HVAC SYSTEM LOGIC & PNEUMATIC CONTROL COURSE

HVAC

SYSTEM

LOGIC

&

PNEUMATIC

CONTROL

COURSE

&

HVAC/COVID VIRUS TRANSMISSION

EXCESSIVE FRESH AIR VENTILATION

IMPACT ON GLOBAL WARMING

&

BOURDON TUBE

CHARACTERISTIC ASSESSMENT REGARDING

POTENTIAL PROOF OF $W_{out} > W_{in}$

&

REPORTS REGARDING

THE HYDRAULIC DISPLACEMENT MOTOR

PROVING THAT $W_{out} > W_{in}$ IS ACHIEVABLE

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PREFACE

Every building and control system is different from all others, making it an impossibility to foresee each situation you will face in the field. The training is intended to allow one to understand the operational range relating to each type of component and present control concepts as simply as possible. The performance of each control set up depends on the quality of the design and commissioning. The responsibility of each system rests on the persons working directly with the system.

Normally the end sequence can be achieved by more than one combination of control components. Typically an assessment of the existing control system determines the best approach in retrofitting a system from a cost point, using as much of the existing components as possible.

You must have a clear vision of the final performance level for your system considering safety, health, comfort, energy, etc. You must assure that the system has all information available that would allow good decisions with your performance goal in mind. You must also assure that the control system has proper control over all the functions required to achieve the performance level desired.

Under any set of conditions the control system should position devices identically to what an intelligent person would do, if manually operating the mechanical system.

The most positive assessment is via directly viewing the dampers, valves, etc.

Computer screen shots, via the BAS, if the sensors are calibrated properly, can be a means of identifying illogical control situations. Look at them thoughtfully.

The logic in this text is presented with pneumatic controls; however, it equally applies to DDC, electronic and manual control of mechanical HVAC systems.

A nightschool class, comprised of experienced HVAC Professional Engineers and control service mechanics, retrofitted a school's control system, based on the logic in this course. They reduced the gas consumption by 50% and the electrical consumption by 27%, translating to an annual reduction of 248 tons of carbon dioxide not entering the atmosphere, as well as the other pollutants associated with combustion of fossil fuels.

The project represented about one person-week of work, eliminating the equivalent pollution of approximately thirteen North Americans for one year.

During your career you may easily eliminate an impact equaling thousands of person-years of pollution.

The United Nations scientists stated publicly in 2018 that we have less than twelve years to find real solutions to burning fossil fuels, producing greenhouse gasses.

You can be a contributing factor in producing solutions, addressing climate change.

The following is a suggested path to specific pages in the Training Manual hopefully inspiring the reader to think deeper on why they unconditionally accept established procedures and physical laws.

Why Care:

- Based on the BBC survey on the psychological impact on our youth, the majority of our children often worry about global warming. (page 11.189)
- A recent news report stated that more than 60% of Canadian adults are very concerned with global warming.
- We are needlessly increasing operational costs via wasted energy.
- We are damaging the planet rather than respecting the planet.
- We have the technology, to significantly help in our environment, which is being dismissed.

<u>The APS training manual addressed in categories:</u>

1) HVAC energy performance improvements:

- Pages 6.44 to 6.47 present sample energy reductions for buildings calculated by the owners.
- Some examples of illogical control systems hiding energy waste are presented on pages 5.38 to 5.43A.
- Case studies pages 8.63 to 8.159 present some case studies focusing on logic advanced over much of current industry approaches.

2) COVID/HVAC relationship to escalated global warming:

- Current increased minimum fresh air ventilation is escalating global warming.
- The disabling of Demand Ventilation Control (DVC) is escalating global warming.
- Current scientific data and a response from ASHRAE's Epidemic Task Force (Page 11.191) may imply the current ventilation solutions are not logical in COVID virus transmission reduction, but damaging to the planet via escalated global warming. (Pages 11.187 to 11.209)
- Critical data is missing in this area of research. (Page 11.203)
- The ventilation changes from conservation logic (DVC) is increasing GHG emissions by up to 700% and direct heat injection into the atmosphere by up to 670% at many HVAC fan systems that have been altered to mitigate COVID transmission. (Page 11.190)

3) Hydraulic Displacement Motor:

- Some scientists/engineer positive peer reviews on pages 13.276 to 13.282.
- A patented system producing completely clean mechanical work. Patent office link:

https://worldwide.espacenet.com/inpadoc?submitted=true&DB=EPODOC&CC=US&NR=2002178719&KC=&F=8&OREQ=0&textdoc=TRUE&FT=E

(NOTE: Patent examiners in the PCT, European, Canadian and USA offices passed it.)

- The fundamental science explained on pages 13.233 to 13.237. (Also page 13.254)
- System application drawings and performance graph on pages 13.295 to 13.298.
- The invention challenges the Laws of Thermodynamics; therefore, I suggest a thought process to determine if you accept the Laws of Thermodynamics based on faith or scientific proof on pages 13.300 to 13.304.
- A suggested logic path is on page 13.313.

4) Bourdon tube investigation:

- Some industry experts' opinions are that a Bourdon tubes' volume does not change when flexing, while the remaining experts contacted have no data on the subject.
- Pascal's Principle is an accepted scientific fact.
- If the Bourdon tube does not change when flexing and Pascal's Principle is valid, a demonstration that work output can exceed work input may be developed. ($W_{out} > W_{in}$)
- The experiments presented on pages 12.210 to 12.231 tested the industry experts' opinion that the Bourdon tube's volume does not change.
  The results to date did not confirm that opinion and indicated that the volume does change; however, air bubbles in the Bourdon tube may have simulated change via compression and decompression. We cut open a Bourdon tube and the interior had rough areas that could trap air bubbles.
  The experiments could have proven the industry experts' opinion true, but cannot prove their opinion is untrue. Air bubles in the experimental tube may have simulated a volume change.
  The patent (US 7,467,517, B2), applying Pascal's Principle to Bourdon tubes is linked below:

https://image-ppubs.uspto.gov/dirsearch-public/print/downloadPdf/7467517 (Patent # 7467517)

CLICK HERE ^

The Hydraulic Displacement Motor science has been dismissed based only on faith in the laws of Thermodynamics, with no challenge to the actual system design drawings presented on pages 13.295 to 13.298.
SIEMENS had one of their control experts examine the drawings and he initialed all of the drawings with no changes, which confirmed our applied control logic.

Some scientist/engineers' opinions:

https://static1.squarespace.com/static/5ee6829b4abd4867f862c3ca/t/61e6fb5a7ea4d658250bc320/1642527582218/SCIENTIFIC+AND+ENGINEERING+OBSERVATIONS+AND+OPINIONS.pdf

The industry should be training our service and commissioning persons to be highly sensitive to their work being important in the Climate Change mitigation effort. The men at APS have each eliminated over 20,000 person years of pollution via applied common sense.

Most control service persons are potentially capable of the same if provided with the training and opportunity.

Regards

President
APS

CONTENTS

SECTION ONE

(CONTROL KNOWLEDGE QUIZ)

-Quiz (Answers on pages 10.177 to 10.183) 1.1 to 1.7

PURPOSE OF SECTION—Allow self assessment of reader before course.

CONTROL QUIZ

This quiz should allow you an indication of your basic control understanding before studying the control course. This quiz does not cover every aspect of potential situations you will face. The hope of the course is bringing you to a comfort level with the various control techniques available to you in solving challenges.

Only select True or False if you are positive. Record your number of correct, incorrect and unanswered items. After studying the course repeat the quiz comparing the two results.

The answers are at pages 10.177 to 10.183 in this training manual.

PNEUMATIC THERMOSTATS

-1- In a reheating application you may find a reverse acting thermostat controlling a normally closed valve.

True False

-2- Two pipe relay thermostats usually are capable of passing a greater volume of compressed air than a one pipe thermostat.

True False

-3- Day/night and summer/winter thermostats alter modes via a change in main air pressure.

True False

-4- There are several different pressure combinations used for day/night and summer/winter change-over in the industry.

True False

-5- Relay type thermostats bleed a small amount of compressed air most of the time.

True False

-6- Two thermostats are required to control a room with a VAV box and a heating coil.

True False

-7- If the flapper nozzle (leak port) is plugged on a direct acting thermostat, controlling a normally open heating valve, the room will likely over-heat.

True False

-8- If an operator accidentally bends the bimetal away from the flapper nozzle (leak port) on a reverse acting thermostat controlling a normally closed cooling valve, the room will likely over-heat.

True False

-9- The sensitivity of a thermostat is the relationship of temperature change to output signal (branch signal) change.

True False

-10- The sensitivity can be changed on most thermostats, but recalibration is required after that adjustment.

True False

-11- If the main air and signal connections are reversed the thermostat typically makes an air hissing noise.

True False

-12- If the main air and signal connections are reversed the thermostat will still control, but a little slower to react.

True False

-13- If a thermostat is controlling a normally open heating valve and a normally closed cooling valve with identical spring ranges, the temperature will control, but energy will be wasted.

True False

-14- An air leak on the branch of a relay thermostat can consume more compressed air than fifty extra thermostats working normally on the system.

True False

-15- The maximum volume of compressed air thermostats can pass out their branch lines varies based on different manufacturers.

True False

-16- In an existing system, another manufacturer's thermostat may usually be used as long as the change-over point is adjusted to match the existing system and the thermostat is the same action.

True False

-17- When calibrating a day only thermostat you set the dial at the current room temperature and adjust the branch signal to midrange of the valve or damper being controlled.

True False

-18- The change-over point is not adjustable on summer/winter or day/night thermostats.

True False

-19- In one stage of calibrating a thermostat, you must set the branch pressure for midpoint of the controlled device and use your breath to heat the bi-metal, checking for proper operation.

True False

-20-When you heat the thermostat in question 19, the pressure will go lower with a direct acting thermostat.

True False

VALVES

-1- Normally open valves allow their maximum fluid flow when their diaphragm pressure is below the bottom end of the spring range.

True False

-2- The bench stroking spring range of a valve is the pressure difference between the pressure required to start the stroking of the valve and the pressure when the valve is fully driven.

True False

-3- The stroking spring range of a valve is the same on the bench as it is in service with the pumps running or steam pressure present.

True False

-4- The stored force in the spring of some control valves can be as high as 1800 pounds, setting the potential for injury or death if disassembled incorrectly.

True False

-5- It is best to remove the stem from the packing box to completely clean the stem and packing box before installing the new packing rings.

True False

-6- Two valves the same pipe size will not have the same flow capability if their CV factors are different.

True False

-7- There is no difference between three-way diverting valves and three-way mixing valves.

True False

-8- Normally open valves are usually installed on heat exchangers serving domestic hot water.

True False

-9- The system differential pressure of the water or steam tends to widen the in service stroking spring range compared to the bench stroking spring range of a valve.

True False

-10- A valve installed backward in a system may cause a hammering noise as it attempts to modulate.

True False

-11- A pilot positioner on a valve eliminates the effect of system differential pressure on the valve`s stroking range.

True False

-12- When the hot water supply temperature is hotter than required by the served area with the greatest heating requirement, the seats and discs on the system's valves tend to experience more wear and tear, shorting the valves' life span.

True False

Automatic Dampers

-1- The normal position of a damper is the position it reaches with no air pressure on the actuator.

True False

-2- Most often the fresh air and exhaust air dampers are normally open and the return air damper is normally closed.

True False

-3- There is never danger of injury from an actuator`s spring when replacing a diaphragm.

True False

-4- Exposing an actuator to pressure higher than recommended by the manufacturer could result in damage to the building, personal injury or death.

True False

-5- The return air damper should start closing before the fresh air damper starts opening.

True False

-6- If the return air is closed and the fresh air damper is closed there is no danger of collapsing duct work.

True False

CONTROLLERS

-1- A controller, with its sensing mechanism, senses the controlled condition directly.

True False

-2- Normally a controller has an averaging sensing element when sensing the mixed air.

True False

-3- If a fluid or gas filled sensing element is broken, the controller will believe the temperature suddenly increased.

True False

-4- Sensitivity and throttling range are the same. They relate to the relationship of the output pressure change relative to a specific temperature change. (example: 5#/F°)

True False

-5- Normally the slower the controlled condition potentially changes, the higher the sensitivity (lower the throttling range) (lower the proportional band)on the controller.

True False

-6- A controller acting as a mixed air low limit typically has a set point of 55°F to 60°F.

True False

-7- The sensitivity is not normally adjustable on controllers.

True False

-8- Throttling range is the amount of the sensed variable change required to stroke the controlled device from fully open to fully closed.

True False

-9- A single controller can be used to satisfy only one function in a control circuit.

True False

RECEIVER CONTROLLERS AND TRANSMITTERS

-1- Transmitters do not normally control anything; they just sense and report the condition to receiver controllers and indication gauges.

True False

-2- Receiver controllers may have as many as five connection ports.

True False

-3- The majority of SIEMENS, TAC, Johnson Controls and Honeywell transmitters are two pipe instruments.

True False

-4- The indicating pressure range of transmitters is normally 5 PSIG to 18 PSIG.

True False

-5- Applying 9 PSIG to a 0°F to 100 °F transmission gauge would cause the gauge to indicate 50°F.

True False

-6- A two position receiver controller can gradually modulate a control valve from fully open to fully closed.

True False

-7- Some receiver controllers provide a restricted air supply to its associated transmitter and some others require an external restricted air supply to the associated transmitter.

True False

-8- If a receiver controller that can provide restricted air to its associated transmitter is receiving main air at variable pressures, its internal restrictor must be blocked and a constant main air external restrictor must be added.

True False

-9- If the transmission line is cut between the receiver controller and the (0°F to 100 °F) transmitter, the receiver will believe the temperature is above 100 °F.

True False

-10- A transmitter can only report to one receiver controller.

True False

-11- Room temperature transmitters look like a thermostat with no dial adjustment.

True False

-12- Proportional band, gain and sensitivity all refer to the same function in receiver controllers.

True False

-13- Receiver controllers may be arranged to automatically raise and lower a temperature of one medium based on a variation sensed in another medium.

True False

-14- Receiver controllers with a CPA can have their set point adjusted from remote locations via pressure changes.

True False

RELAYS

-1- Pneumatic relays allow design logic achieving almost any sequence of events required.

True False

-2- A two input high selector averages the input signals.

True False

-3- A volume boosting relay may also be used as a two input low selector.

True False

-4- A snap acting air switching valve may be piloted by a two position controller or a modulating controller.

True False

-5- A modulating receiver controller may pilot a gradual air switching valve.

True False

-6- A reversing relay can effectively change a direct acting signal into a reverse acting signal and offset the signal.

True False

-7- Multi high-low selector indicates the highest input signal on its H output port and the lowest input signal on its L output port.

True False

-8- A minimum positioning relay assures the fresh air dampers are closed at night.

True False

-9- A biasing relay (ratio relay) allows co-ordination of heating and cooling in a logical sequence.

True False

-10- If two thermostats, set for 70°F, are averaged with one room at 80°F and the other at 60°F the average will be 70°F. The comfort in the occupied space will be acceptable.

True False

-11- Pilot positioners (positive positioning relays) are used on valves and damper actuators to position the devices accurately, compensating for varying resistance, as well as allowing alteration of the stroking range and start point of the devices stroke.

True False

-12-Relays are available to add or subtract a determined amount of pressure from another variable pneumatic signal at a constant rate.

True False

-13 A PE switch passes control air when energised.

True False

SAFETY LOOPS

-1- Low limits (sometimes called freeze stats) have an averaging element and shut down the fan if the air is too cold.

True False

-2- High limits (sometimes called fire stats) should be only installed in the supply and mixed air.

True False

-3- Circuits sensing the coil water temperature with an electric thermostat (45°F) and the water electric flow switch wired in series, also have an outdoor temperature electric thermostat (45°F) wired in parallel with the series circuit previously mentioned.

True False

-4- Low and high limit circuits may be fully electric or pneumatic with a pressure switch.

True False

-5- Normally the safety circuit should be wired through the automatic side of the starter switch and not the hand side of the switch.

True False

-6- Static pressure high limit on VAV systems shut down the fan before the pressure reaches the system's static control point.

True False

-7- If a low limit is causing nuisance fan shut downs, it is usually OK to jumper the contacts of the low limit to keep the fan running.

True False

-8- If the high pressure relief valve on the compressor keeps blowing off, you should remove it and plug the hole.

True False

-9- The pressure switch on the compressor can be jumpered, if defective, until you get a new one.

True False

-10- The steam valve on the heat exchanger for domestic hot water should fail closed.

True False

SECTION TWO
(CONTROL ACTION & DEVICES)

PURPOSE OF SECTION

Present applicable components and logic.

Any logic statement regarding event sequencing, while controlling valves, dampers, etc. can be achieved by proper application of the control relays on pages from 2.12 to 2.18.

COMPONENT NORMAL POSITIONING AND ACTIONS

NORMALLY OPEN VALVE

A normally open valve **allows** the flow of fluid through the valve's body when air pressure is removed from the valve actuator's diaphragm.

NORMALLY CLOSED VALVE

A normally closed valve **disallows** flow of fluid through the valve's body when air pressure is removed from the valve actuator's diaphragm.

NORMALLY OPEN PRESSURE SWITCH

A normally open pressure switch **disallows** electrical flow through the pressure switch's contacts when pressure is removed from the pressure switch's diaphragm.

NORMALLY CLOSED PRESSURE SWITCH

A normally closed pressure switch **allows** electrical flow through the pressure switch's contacts when pressure is removed from the pressure switch's diaphragm.

DIRECT ACTING CONTROL

A direct acting control increases its output (branch) signal on an increase in the sensed medium.

REVERSE ACTING CONTROL

A reverse acting control decreases its output (branch) signal on an increase in the sensed medium.

PROPORTIONAL AND TWO POSITION CONTROL

Proportional control allows output branch line signals to be infinitely variable between 0 PSIG to the supply air maximum pressure. Proportional control allows modulation.

Two position control only allows the output branch pressure to be 0 PSIG or equal to the maximum supply air pressure. There is no modulation with two position control.

EXAMPLES

-1- A **direct acting** thermostat controls a **normally open** heating valve. As the thermostat senses an increase in room temperature it increases its output signal driving the normally open valve closed to reduce heat gain in the room.

-2- A direct acting thermostat controls a normally closed cooling valve. As the thermostat senses an increase in the room temperature, it increases its output signal driving the normally closed cooling valve open providing cooling to the room.

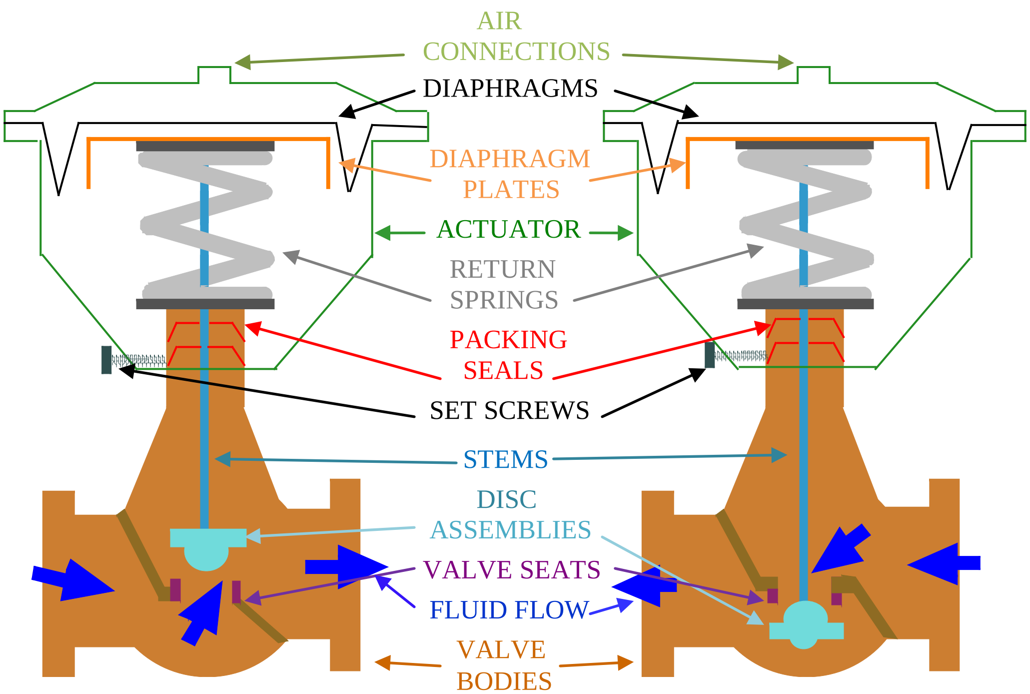
<u>TWO WAY CONTROL VALVES</u>

NORMALLY OPEN VALVE

Compressed air drives the valve closed. The return spring forces the valve open on a decrease in air pressure.

NORMALLY CLOSED VALVE

Compressed air drives the valve open. The return spring forces the valve closed on a decrease in air pressure.



Pneumatic valves stroke through ranges dependent on the range of the return spring. Some valves have fixed start points while other valves may have the start point altered via spring tension adjustment on the actuator.

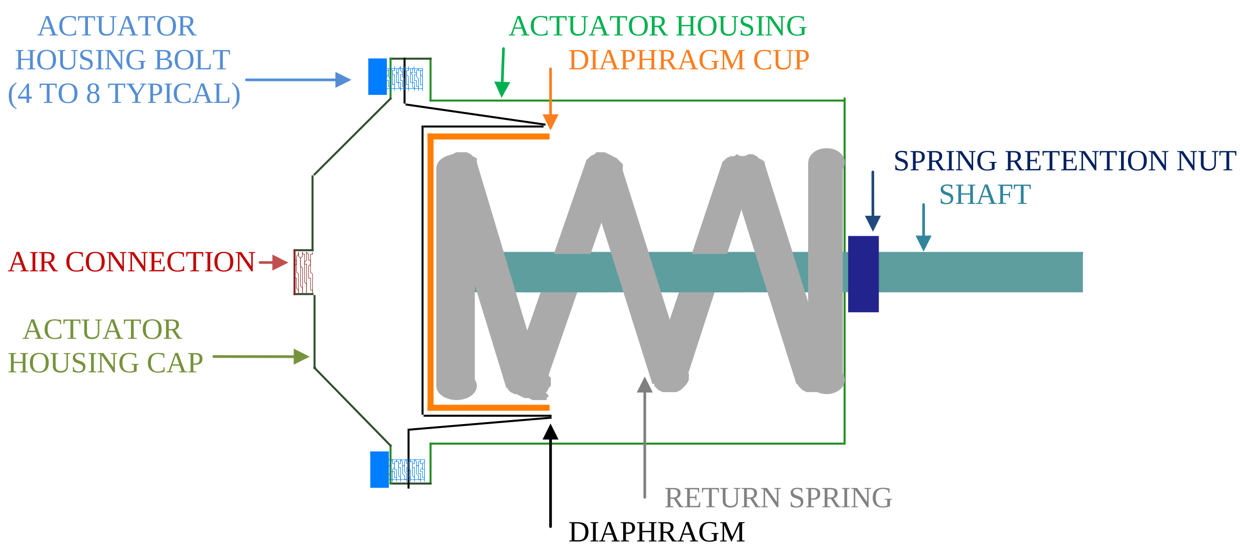
The spring range on the valves specification sheet is a bench spring range. Typically, in a system, a normally open valve with a bench spring range of four PSIG to eight PSIG will start at four PSIG, but close above the eight PSIG bench closing point. The fluid system differential pressure, from the supply to the return, causes a force against the disc assembly, shifting the spring range closing point of eight PSIG, for example, to ten PSIG.

Valves must be installed with the fluid flow as illustrated. If a valve is installed backwards, it will usually cause hammering in the pipes just as the valve is close to the normal closing point.

Honeywell and some Johnson Control valves have a normally open body, with an actuator that drives the valve closed when no pressure is on the diaphragm. These valves have to be considered normally closed in the logic of a system.

Some older Johnson Control valves have the potential to be very dangerous during servicing. Consider the illustration of the normally open valve. If the disc assembly is detached from the stem, the set screws are the only means of holding the spring in the actuator. Some valves have an 8R actuator and a valve range of nine to thirteen PSIG. In this situation, the actuator could be thrown, if the set screws are loosened, with a force around fifteen hundred pounds. It could cause severe injuries or death.

DAMPER ACTUATOR (MOTOR)



Damper actuators position automatic dampers via a control pressure entering the damper actuator through the air connection. The air pressure is distributed evenly over the face of the diaphragm, exerting a total force on the diaphragm cup, which transfers that force to the shaft. The spring resists the force generated by the air pressure, returning the actuator to its normal position when air pressure is removed.

Damper actuators have a spring range (throttling range). The spring range on an actuator's specification sheet is based on the actuator with no load. An actuator with a spring range of 8 PSIG to 13 PSIG will start to move at 8 PSIG and reach the end of its stroke at 13 PSIG.

There are many spring ranges with the most common being 8# to 13#, 5# to 10#, 3# to 7#, 2# to 12#, 3# to 13# and 10# to 15#. Some Barber Colman and some older Powers damper actuators allow shifting of the spring range via an adjusting nut, which has the shaft running through its middle.

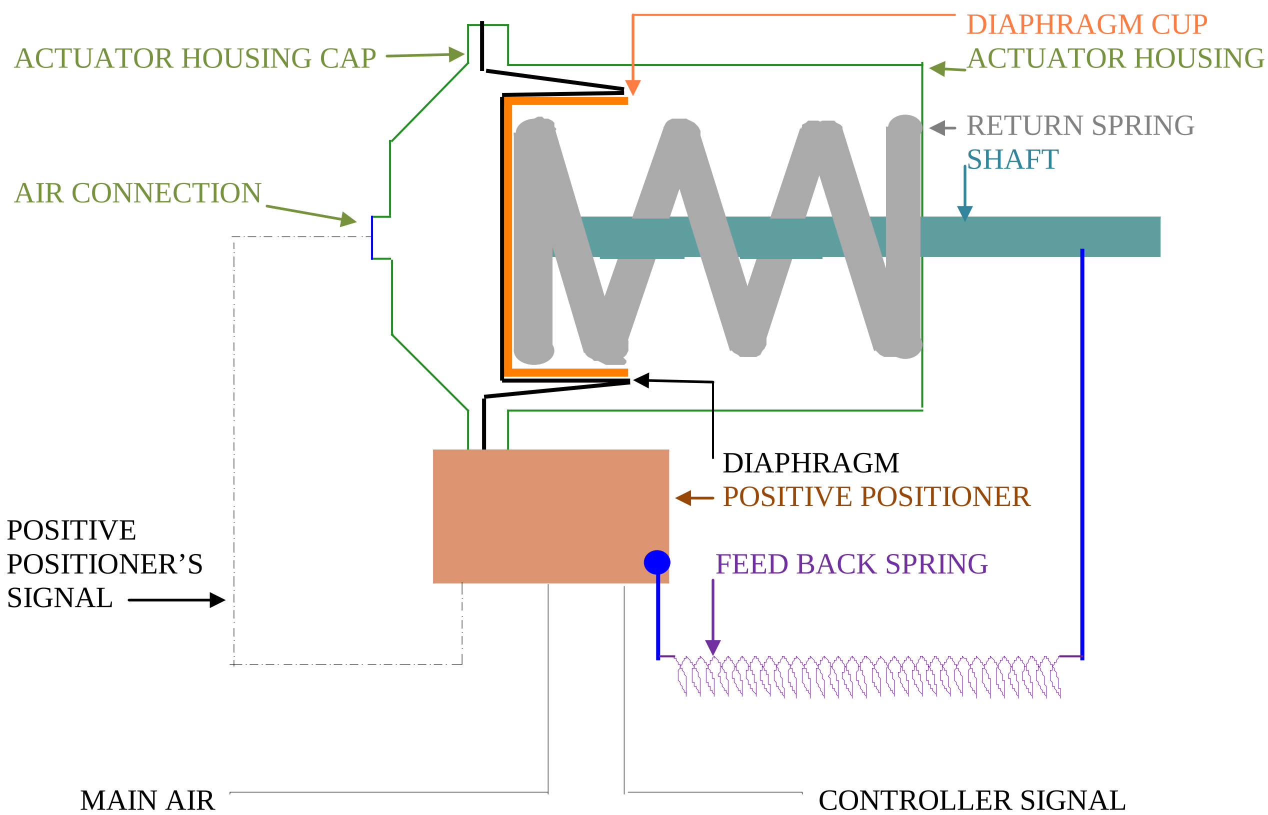
The diaphragms are normally replaceable, but not always.

Springs in pneumatic actuators can be potentially very dangerous; therefore, you should respect the power stored in the spring while disassembling an actuator.

When replacing a diaphragm, remove air pressure from the actuator and then back off the actuator housing nuts just enough to separate the actuator housing cap from the actuator housing. Push on the actuator housing with your hand giving you a sense of the power of the spring. If this is difficult to push, take appropriate steps to protect yourself replacing the actuator housing bolts, one at a time, with longer bolts and nuts for adjustment.

Do not apply air pressure higher than recommended by the manufacturer, as you may:
-1- damage the actuator
-2- injure or kill yourself or others

DAMPER ACTUATOR (MOTOR)
WITH POSITIVE POSITIONING RELAY



Damper actuators with no positive positioning (pilot positioner) relay often position the damper at a different degree of opening for the same pressure when the pressure is increasing to that pressure or decreasing to that pressure. As the damper increases its frictional impact with dirt and age, positioning will be different for the same pressure relative to when the damper was new.

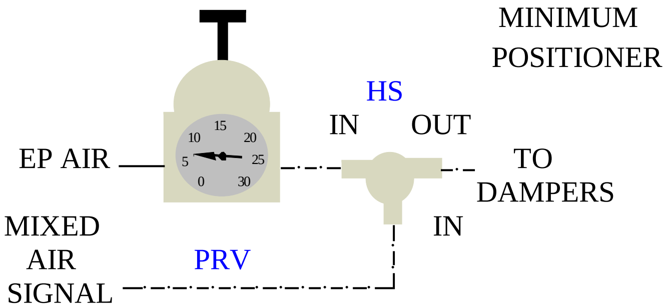
Positive positioners present three benefits:

-1- The throttling range of the controller is adjustable.
On the old Honeywell the six screws under the round cap allowed selection of three, five or ten pound ranges.
The position of the spring connected to the linkage from the shaft allows span adjustment for Johnson Controls, Barber Colman and SIEMENS positive positioners.
The feedback springs can be changed to have three, five and ten pound spans on the new Honeywell, Kreuter and Robertshaw positive positioners.

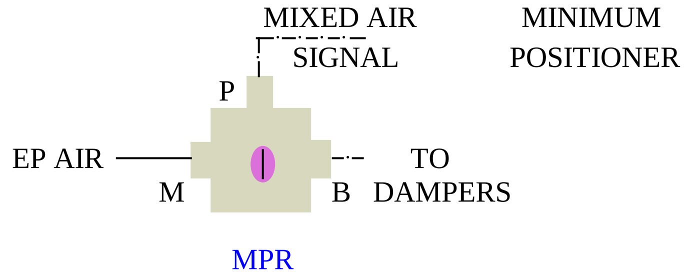
-2- The feedback spring allows the positioner to “know” the percentage of stroke, regarding the actuator’s shaft, at all times. The positive positioner will apply air pressure to the actuator’s diaphragm from main air pressure to zero PSIG maintaining the damper exactly at the percentage of stroke demanded by the controller. The start point for the stroke is adjustable.

-3- Positive positioners boost a low volume signal to a high volume signal.

CONTROL RELAYS

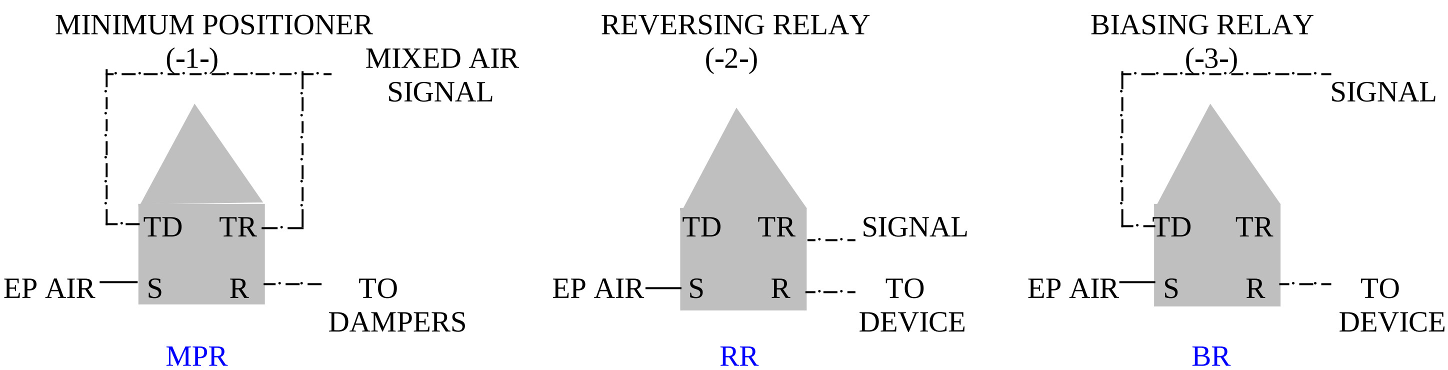


Minimum ventilation is some times achieved via a pressure reducing valve (PRV) and a high selector (HS), as illustrated above. If the controller signal drops below the pressure required keeping the dampers at or above the minimum ventilation rate, the HS passes the PRV signal to the dampers, maintaining minimum ventilation.



Minimum ventilation is sometimes achieved via a minimum positioning relay (MPR). The MPR is set at a pressure which holds the fresh air damper open enough to assure at least minimum ventilation when the controller signal is below that pressure.

SIEMENS MULTIPURPOSE RELAY



SIEMENS multipurpose relay may used as:

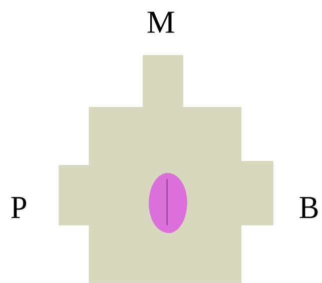
-1- A minimum positioning relay (MPR), as per the top left hand MPR in this illustration.

-2- A reversing relay (RR), which reverses the input signal (TR) and allows biasing of the signal.
Example: If the signal increases by 2 PSIG, the output (R) decreases by 2 PSIG. The output may also be biased relative to the signal.
Example: The relay may be set for 10 PSIG signal provides 10 PSIG output: you may adjust the relay to 10 PSIG provides an out put pressure other than 10 PSIG suiting the sequence of the circuit.

-3- A direct acting biasing relay (BR), which offsets the signal (TD) in the output signal (R), and remains a direct acting signal. This is used to sequence devices, eliminating function over-laps (heat/cool, preheat/reheat, face & bypass damper/valve, etc.)

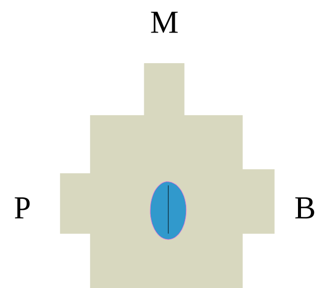
NOTE: The TAC AKR-40605 is also a multi use relay. The AKR-40605 may be used as a minimum or maximum positioning relay, a 1:1 ratio relay (as #3 SIEMENS illustrated)), a two signal low sector or a manual positioner.

REVERSING RELAY



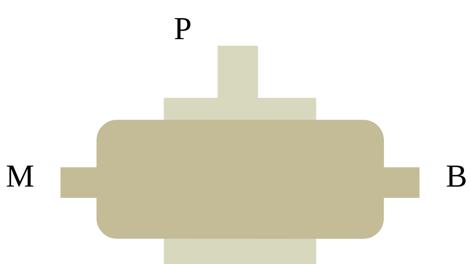
Reversing relays increase their branch signal in response to a decrease in their pilot signal. Most have a 1:1 relationship. Johnson Controls made a reversing relay with a 1:2 relationship. The branch pressure can be biased for sequencing purposes.

BIASING RELAY



Biasing relays increase their branch signal in relationship to the pilot signal. The branch signal may be off set to a different pressure than the pilot signal. The branch may increase at ratios of 1:1, 1:2, 1:3.33 or 1:5.

RATIO RELAY

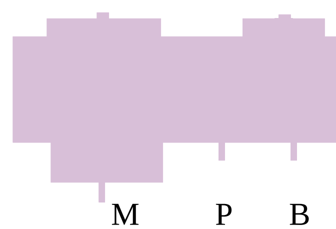


Ratio relays increase their branch signal at a fixed ratio relative to their pilot signal. The ratios are 1:2, 1:4, 1:6.

Reducing ratio relays decrease the branch signal relative to the pilot signals.

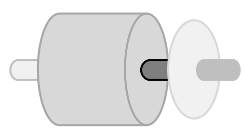
The output signal cannot be biased with these relays.

TURNDOWN RATIO RELAY (KREUTER)



The turndown ratio relay allows the branch signal to be biased relative to the pilot signal. The branch signal may reduced at a selectable ratio via a dial on the relay.

GRADUAL SWITCH



Gradual switches can be one or two pipe. They can be used as a minimum positioner as the PRV on page 2.12. They are very useful in simulating transmission signals while calibrating receiver controllers.

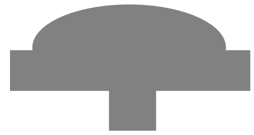
CONTROL RELAYS

LOW SELECTOR

Low selectors receive two input signals and produce an output signal matching the lower input.

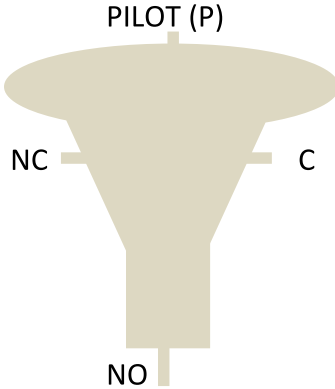
Low selectors often can be used as a volume booster relay, depending on the manufacturer.

HIGH SELECTOR



High selectors receive two input signals and produce an output signal matching the higher of the two inputs.

AIR SWITCHING VALVE



Air switching valves are open from the NO to the C port when the pilot signal is lower than the valve's spring range. The valve is open from the NC to the C port when the pilot pressure is higher than the valve's spring range.

Gradual acting air valves must have a two position pilot signal (0 PSIG or 15-20 PSIG).

Snap acting air valves may have a gradually changing or two position pilot signal.

Some air valves have adjustable switching points.

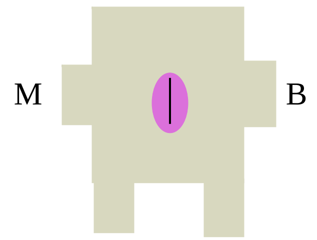
MULTI/HIGH/LOW SELCTOR

Multi-high/low selectors receive more than two input signals. Manufacturers allow a variety of options. The highest signal is duplicated as the high output signal and the lowest signal is duplicated as the low output signal. Both the multi-high and multi-low can be built with check valves and a .005" restrictor.

Flow from each of the selected signals, through a check valve, to a common line and the .005" restrictor bleeding to atmosphere from the common line will produce the highest signal.

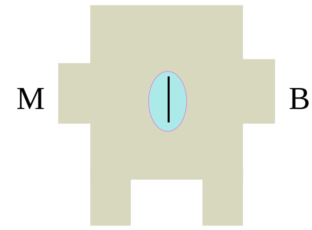
Flow through check valves, to each of the selected signals from a common line which has main air entering the common line through the .005" restrictor produces the lowest signal on the common line.

SUBTRACTION RELAY



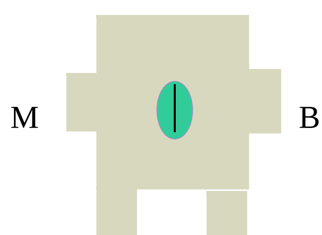
Subtraction relays are available from Kreuter and have two inputs, main air and branch ports. The branch signal is the product of subtracting port 2's signal from port 3's signal (KREUTER). The branch signal can also be biased. They are useful in conservation loops such as enthalpy comparison between outdoor air enthalpy and return air enthalpy.

ADDITION RELAY



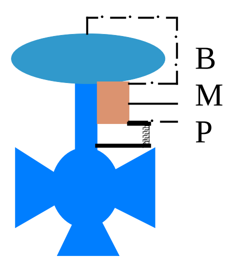
Addition relays are also available from Kreuter and have two input signals, main air and branch signal ports. The branch signal is the sum of the two input signals, but cannot exceed the main air pressure. The branch signal may also be biased. They are useful in conservation loops if you wish to simulate a transmission signal greater than the actual condition.

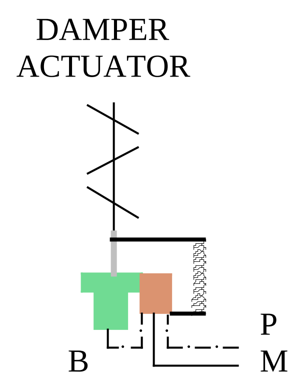
AVERAGING RELAY



Averaging relays have two inputs (some have four), main air and branch ports. The branch port signal is the average of the input signals.
CAUTION: One input signal may indicate an extremely hot room and the other input signal may indicate an extremely cold room, but the average indicates that all is well.

POSITIVE POSITIONER (PILOT POSITIONER)



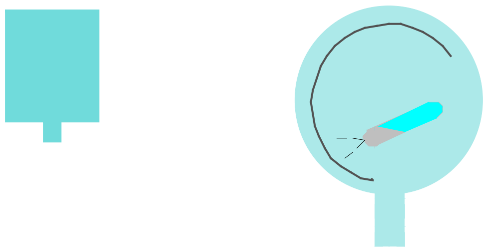


Pilot positioners (positive positioners) sense the stroke of the valve or damper actuator from 0% to 100%. The controller signal does not go directly to the valve or damper actuator. It goes only to the pilot positioned indicating the required percentage of stroke. The pilot positioned uses from full main air pressure to zero PSIG positioning the device as requested.

<u>PNEUMATIC/ELECTRIC COMPONENTS</u>

PRESSURE ELECTRIC (PE)

MERCOID (PE)

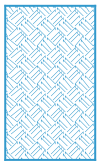


Pressure electric switches and mercoid switches are the means of controlling an electrical signal via a pneumatic signal. They come normally open, normally closed or allow a choice when wiring. Do not exceed the pressure, voltage or amperage recommended by the manufacturer.

ELECTRO/PNEUMATIC VALVE (EP)
SOLENOID AIR VALVE (SAV)

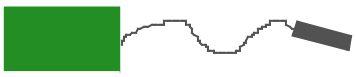
Electro/pneumatic (EP) valves, also referred to as solenoid air valves (SAV) are the means of allowing or disallowing pneumatic signals in a circuit. Be sure the voltage rating is correct and the air pressure is below manufacturer's recommendations.

TRANSDUCERS



One type of transducers are the means of using an analogue pneumatic signal (3#-15#) to generate an analogue electrical signal. (0-5VDC,0-10 VDC or 4-20ma)
Another type is the means of using an analogue electrical signal (0-5VDC, 0-10VDC or 4-20ma) to generate a 3#-15# pneumatic signal. They are available as relay and non-relay transducers.

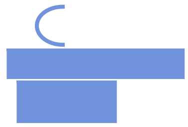
ELECTRIC CONTROLS



Electric thermostats allow or disallow electrical flow based on the controller's set point and the state of the medium the control is sensing.
They are often used in safety circuits.

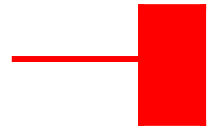
SAFETY CIRCUIT COMPONENTS

DIFFERENTIAL PRESSURE SWITCH



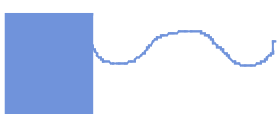
Differential pressure switches open and close their electrical contacts based on pressure variation sensed by the switch. They normally have low and high pressure connections. Often they are used providing status on fans or used as safeties on electric heating coils preventing the coil from heating when the air flow stops.
They are normally electric devices, but pneumatic units exist, which must cycle a pressure electric switch, attaining the same result.
If a diffuser is blocked the switch will sense pressure, but no air flow is allowed by the blockage; therefore, the electric coil may still be damaged.

HIGH LIMIT



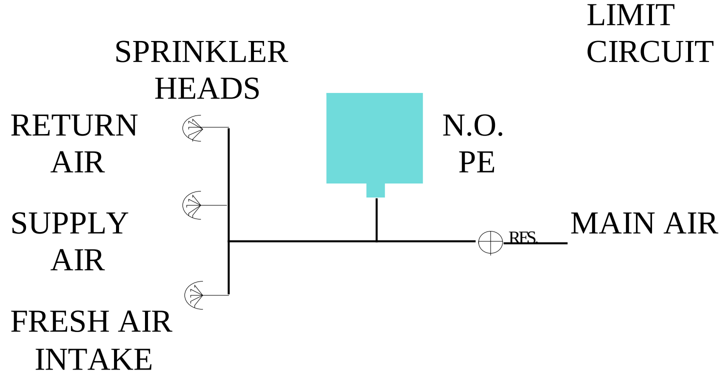
High limits shut down the fan system if they sense a temperature above their set point. The set point should be no more than 50F° above the normal highest temperature in the duct. They usually have to be manually reset after they have shut down the fan system.
They can be electric or pneumatic. If they are pneumatic, they cycle a pressure electric switch attaining the same result as the electric units. Some have adjustable set points, while some have fixed set points.

LOW LIMIT



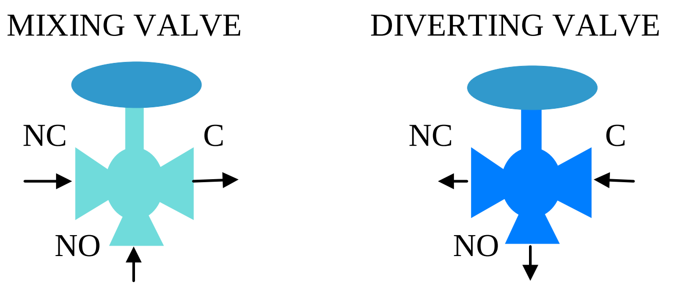
Low limits are sometimes called Freeze Stats. The coil in a fan can still freeze even after the low limit has shut down the fan in some situations. They do not have averaging elements. The element will cause the low limit to trip at its set point (40°F) if any one foot of the element senses a temperature below set point. They can be electric or pneumatic with a PE being controlled by the pneumatic low limit. They can be manual reset or automatic.

HIGH LIMIT CIRCUIT



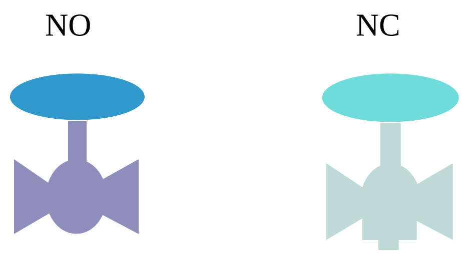
High limit circuits are sometimes installed using sprinkler heads, with the correct temperature rating for the locations and a pressure switch fed by restricted main air. If any sprinkler head melts, the air is bled from the common line down-stream of the restrictor and the pressure switch opens its contacts shutting down the fan system. If the compressed air supply fails, the fan system will shut down, as a high limit situation will be simulated.

THREE-WAY VALVES



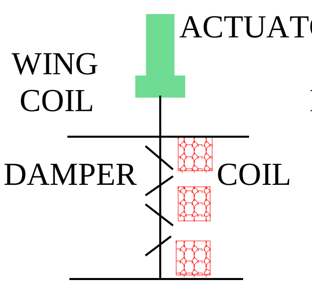
Three-way valves are usually available as mixing or diverting. Mixing valves have two inputs and one output. Diverting valves have one input and two outputs.
Usually there is a flow through the common port of three-way valves; however, rarely one type of three-way valve allows flow from the common port and only through either the normally open port or the normally closed port and sometimes no flow at all through the valve. The direction of flow must be correct.
The spring ranges can vary from valve to valve creating different throttling ranges.
The effective throttling range on valves can be adjusted if a pilot positioner (positive positioning relay) is installed on the valve.

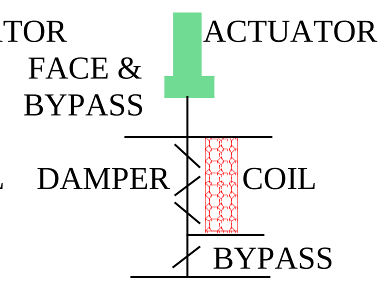
TWO-WAY VALVES



Two-way valves are either normally open (NO) or normally closed (NC).
The spring ranges can vary from valve to valve creating different throttling ranges.
The effective throttling range on valves can be adjusted if a pilot positioner (positive positioning relay) is installed on the valve.
The direction of flow must be correct.

FACE & BYPASS AND WING COIL DAMPER ARRANGEMENTS

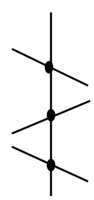




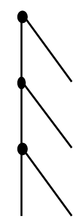
Wing coil and face & bypass damper arrangements are often used when the upstream air temperatures may fall below the freezing point
The control circuit should allow the heating coil full flow before exposing the coil to air flow via the dampers.
The wing coil damper arrangement allows the sections of the heating coil to be completely enclosed when no heat is required.
Most often the heating coil has full flow when the outside air is below freezing.
Proper safety circuits normally are installed on these systems.

AUTOMATIC DAMPERS

OPPOSED BLADE

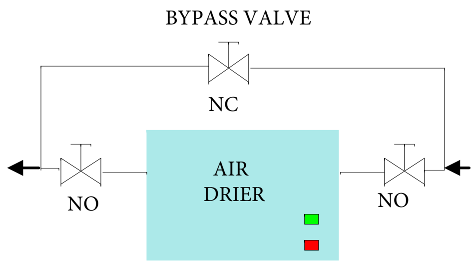


PARALLEL BLADE



Automatic dampers are used to isolate the building systems when fans are off and blend fresh air with building return air when the fans are running.
Opposed blade dampers achieve more linear flow respecting percentage open to percentage of maximum air flow than parallel blade dampers.
Parallel blade dampers, if installed properly, tend to achieve better mixing of fresh air and return air than opposed blade dampers.

REFRIGERATED AIR DRIER

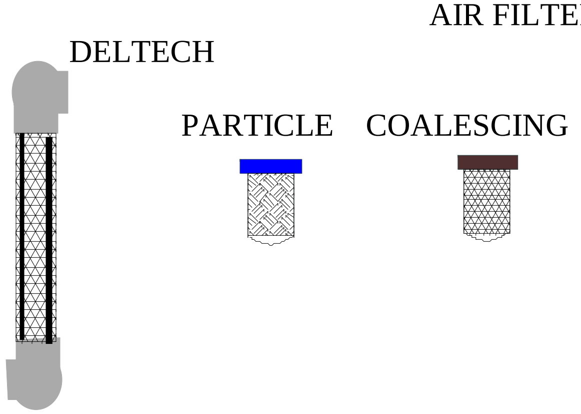


Air driers dry the compressed air by cooling the air to dew point and then continuing cooling to about 38°F, condensing the moisture from the air.
The condensed water collects at an auto drain which relieves the water from the system either via a float mechanism or a timer with a solenoid.

RELIEF VALVES

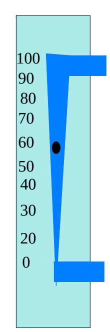
Relief valves are safeties preventing over-pressurization of a tank or air line.
They should not be painted and should be tested regularly as part of normal maintenance.
Relief valves on compressor tanks are set respecting the design operating pressure of the compressor. Often relief valves on control compressors are set at 100 PSIG.
The relief valves installed after the pressure reducing valve on control systems are normally set between 23 PSIG and 30 PSIG.
Sometimes the pressure relief valve is built into the pressure reducing valve.

AIR FILTERS



The orifice that feeds the "brain" of many pneumatic controls is only .005"; therefore, proper filtration is critical.
Particle filters remove particles down to the rating of their elements. (usually 40 micron or 5 micron)
Coalescing filters remove oil and particles. (usually down to .1 micron) Data sheets suggest replacing the element at 10 PSIG pressure drop at 10SCFM airflow.
Deltech filters remove oil and particles down to 1 micron. The element is changed based on colour change in the element from white (pinkish/white) to deep red.

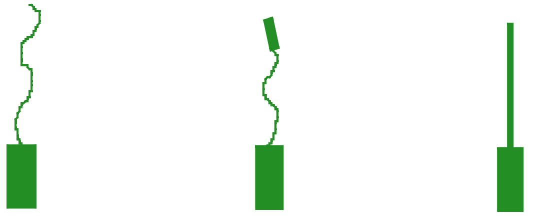
FLOW METERS



Flow meters are installed indicating the quantity of compressed air passing to the control system. (standard cubic feet per hour SCFH) They alert the operator to air leaks in the system, which waste energy and burden the compressor.
They are very useful in trouble shooting compressed air leaks in building systems.

PNEUMATIC TRANSMITTERS

AVERAGING BULB ROD



Pneumatic transmitters sense a medium with their element and only report the condition to a gauge, receiver controller or both: they do not control.
All commercial transmitters have a 3 PSIG to 15 PSIG range. A few are two pipe with relay volume capabilities, but most are low volume, one pipe using a .005”, .007” or a .0075” restrictor.
The reporting characteristics may be temperature, relative humidity, pressure, enthalpy, etc., with multiple ranges in each.
The indication gauges seen on the control panels are actually pressure gauges with a 3 PSIG to 15 PSIG range calibrated to indicate temperature, humidity, etc. relative to their associated transmitter.

RECEIVER CONTROLLERS

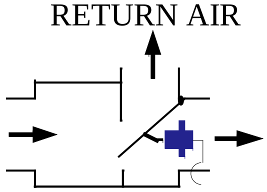
SINGLE INPUT MULTIPLE INPUT

Receiver controllers receive a pneumatic signal or signals allowing the receiver controller to determine what position it should put the valve or damper to attempting to maintain the receiver controller’s set point.
The proportional band determines the throttling range of the controlled valve or damper actuator.
The authority determines the relationship of temperature change on the primary and secondary ports to rebalance the instrument.
The control point adjustment (CPA) allows remote adjustment of the set point regarding the instrument.

DUMP BOX (BYPASS BOX)

CEILING SPACE
RETURN AIR

SUPPLY DUCT

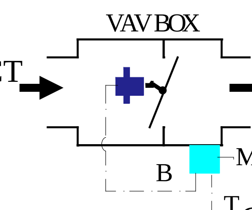


OCCUPIED
SPACE

Dump (bypass) boxes are the mechanical means of regulating the quantity of cooling air allowed into the occupied space based on the thermostat’s control of the damper actuator in the box.
The fan provides a constant volume, as the supply duct always passes the same quantity of air.

VAV BOX (VARIABLE AIR VOLUME)

SUPPLY DUCT



OCCUPIED
SPACE

AIRFLOW
CONTROLLER

VAV boxes are the mechanical means of regulating the quantity of cooling air allowed into the occupied space based on the thermostat’s control of the damper actuator in the box.
The fan requires inlet vanes or variable frequency drive control (VFD) controlling the supply duct pressure as the total quantity of air required by the VAV boxes varies on thermal load changes.
Systems with the airflow controller are called pressure independent and systems where the thermostat goes directly to the damper actuator are called pressure dependent.

NOTES

SECTION THREE
(THERMOSTATS)

PUPOSE OF SECTION:

Present understanding of range and types of thermostats and required calibration equipment.

PREFACE

Proper calibration of thermostats is critical to comfort as well as proper energy control.

Day/night and summer/winter thermostats are switched modes by a change in the main air pressure. There are several combinations of pressures achieving this change-over depending on the manufacturer and the particular building.

Many thermostats are considered defective and replaced when the service person does not have a calibration change-over board illustrated on page 3.22. All that is required most often is an adjustment in the change-over point and recalibration of both operational modes causing the thermostat to perform as new again.

On one occasion a client bought five hundred day/night thermostats to upgrade a whole high school. APS was hired to replace the thermostats with the school board's control mechanic.

Although the sealed boxes, containing the new thermostats, indicate "factory calibration", we never trust that statement. Logically the calibration can only be accurate for one spring range and will be out of calibration for all others even if the factory calibration was done properly and the thermostat was treated gently from that point onward.

On this occasion we tested three hundred thermostats for day calibration, night calibration and change-over point. Not one was accurate on all three considerations. We also found multiple other problems with the three hundred "new" thermostats.

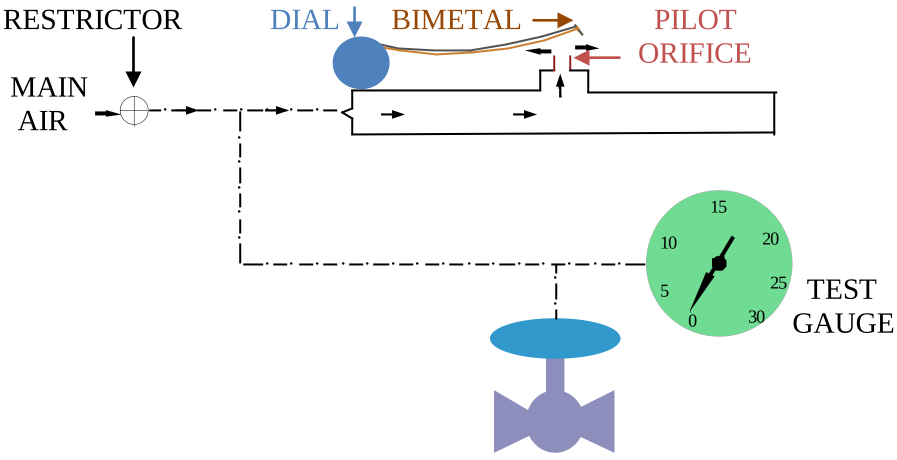
We very much advise to check every thermostat for proper calibration and operation before putting it into service, even though the box states "factory calibrated".

The signal from occupied space thermostats can control their own associated valves, VAV's or damper motors and also be used to co-ordinate the heating and cooling provided by the base building heating and cooling equipment. The thermostats' output signals from 0 PSIG to 20 PSIG create a ramp where functions can be selectively allowed in various segments of the ramp. This allows the system to minimize the over-lap of heating and cooling.

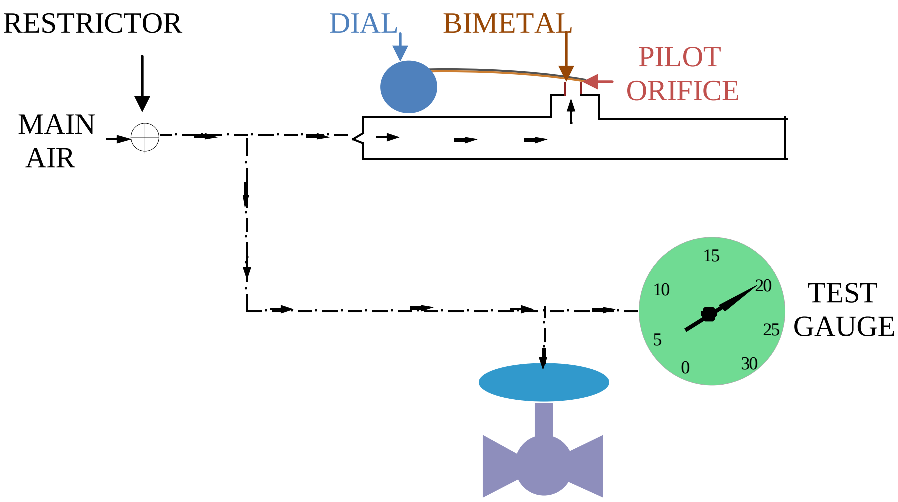
Air leaks waste a lot of energy and may damage your compressor via extended run periods. Page 3.27 presents the impact of a thermostat signal air leak, while still maintaining enough pressure to control the temperature properly. A Johnson Control T4002 thermostat controlling a valve with a 3# to 6# range and a diaphragm leak causing the branch signal to drop from 20 PSIG to 10 PSIG will, while still maitaining temperature control, be consuming as much compressed air as 40 thermostats with no leaks.

Control air compressors are normally sized to handle 25% to 33% run time: not large air leaks.

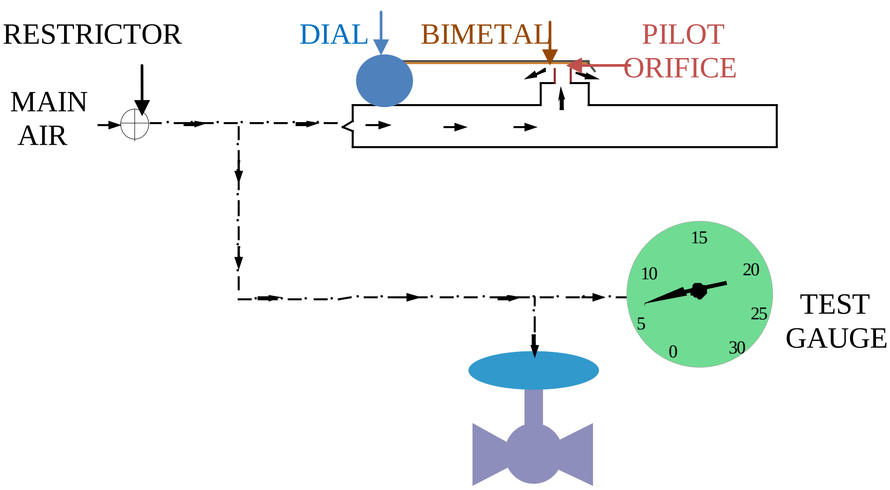
NON-RELAY THERMOSTAT



VALVE FULLY OPEN



VALVE FULLY CLOSED



VALVE MODULATING

Non-relay thermostats are usually one pipe as illustrated; however, they are available as two pipe thermostats. As two pipe, they usually are two one pipe thermostats built into one unit allowing the heating and cooling functions separate control with a dead band.

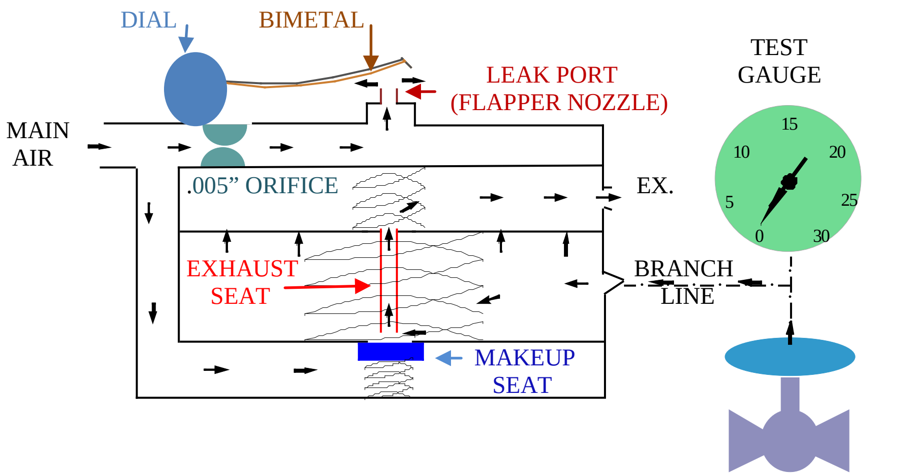
Non-relay thermostats may be direct or reverse acting depending on if the function is heating or cooling and if the valve is normally open or normally closed.

The restrictor is usually either a .005” or a .007” orifice; therefore, this type of instrument builds its branch line signal much more slowly than a relay instrument. A small leak in the branch tubing or the valve diaphragm will cause lack of control much sooner than with relay type thermostats.

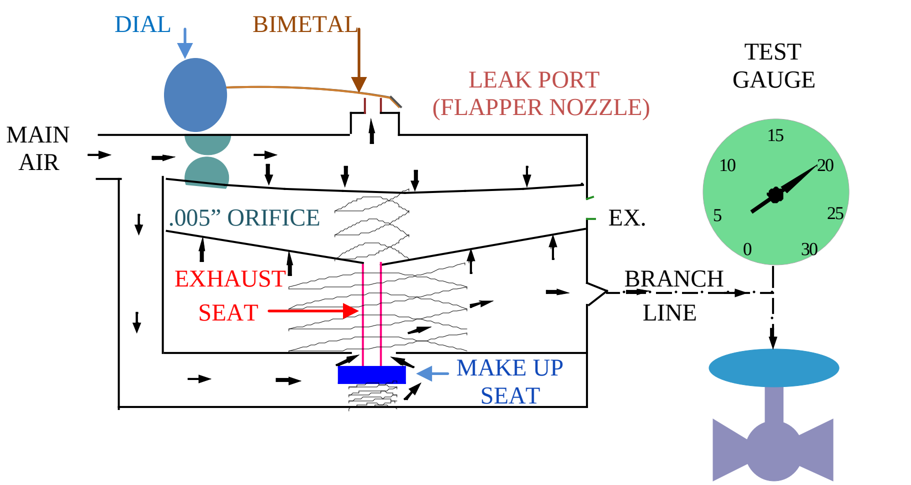
The test gauge illustrated is not normally installed in a building. The service technician calibrating the thermostat plugs his test gauge into the branch line of the thermostat allowing calibration.

When calibrating the thermostat you must know the temperature of the room at the thermostat. Plug your gauge into the branch line. Turn the dial all the way up and all the way down witnessing that the pressure goes from zero PSIG to the main air pressure. Then set the dial at the actual room temperature. Then adjust the branch pressure to mid-range of the device being controlled. Then breathe your warm breath on the thermostat checking that the pressure increases for direct acting thermostats and decreases for reverse acting thermostats. Then set the dial to the desired temperature for the room and put the cover back on the thermostat.

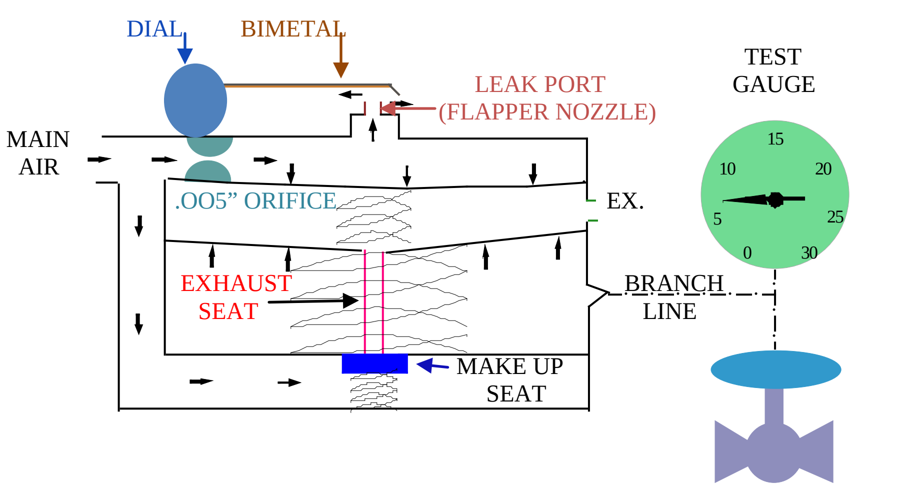
RELAY THERMOSTAT



VALVE FULLY OPEN



VALVE FULLY CLOSED



VALVE MODULATING

Relay thermostats are usually two pipe, but some can be three pipe found in unitventilators, typically in schools. (see unitventilator sheets)

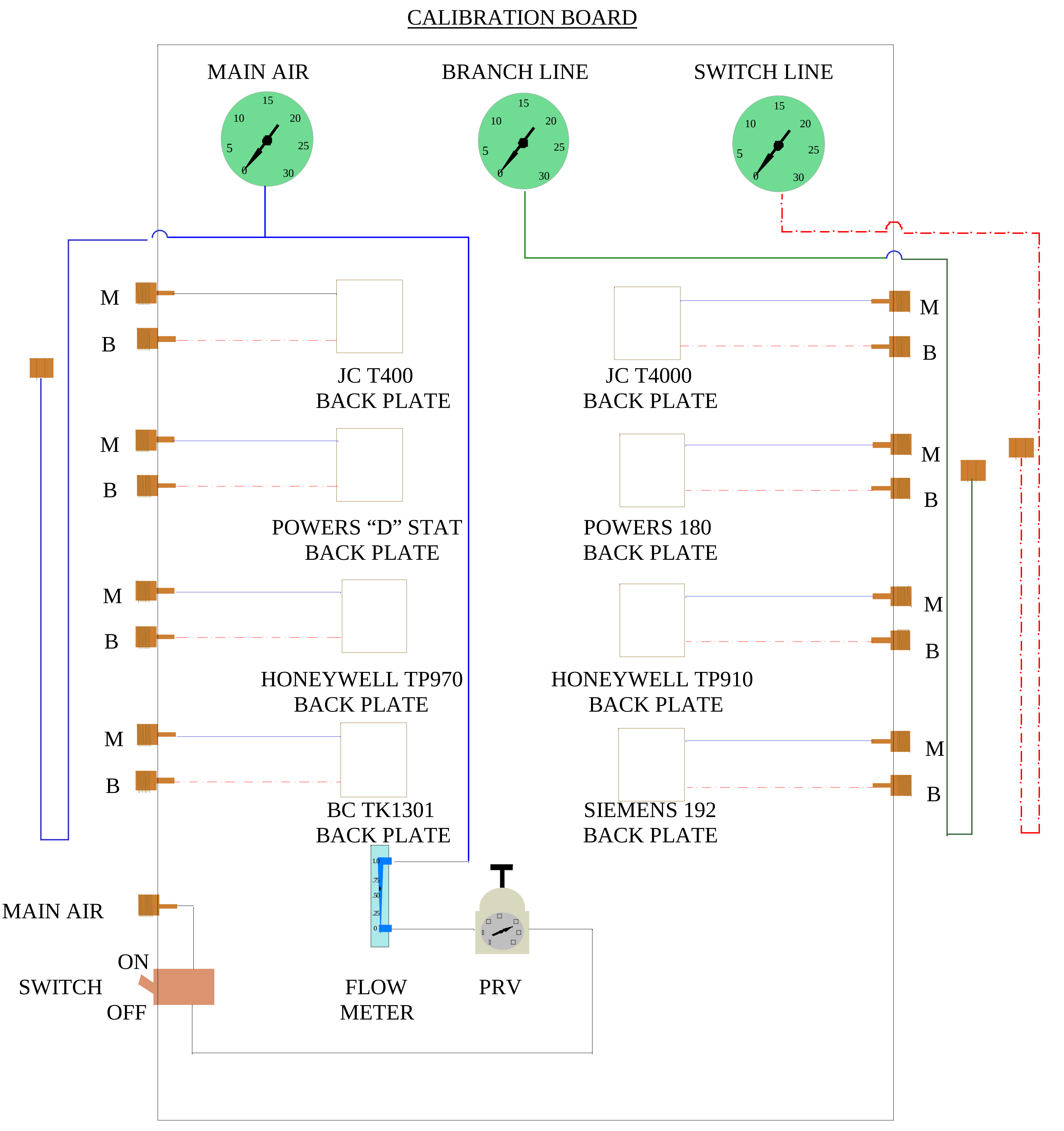
Most relay thermostats are similar to the illustrations on this page.

Both day/night and summer/winter thermostats are more complex. Night and day set points are both direct acting and the summer and winter are usually reverse and direct acting respectively.

Working normally, relay thermostats bleed approximately the same amount of compressed air as non-relay thermostats via the .005” orifice. Through the makeup seat, some relay thermostats pass up to 100 times as much air as non-relay thermostats.

The test gauge is not normally installed in a system. Service technicians plug their test gauges into the thermostat only during calibration.

When calibrating the thermostat you must know the temperature of the room at the thermostat. Plug your gauge into the branch line. Turn the dial all the way up and all the way down witnessing that the pressure goes from zero PSIG to the main air pressure. Then set the dial at the actual room temperature. Then adjust the branch pressure to mid-range of the device being controlled. Then breathe your warm breath on the thermostat checking that the pressure increases for direct acting thermostats and decreases for reverse acting thermostats. Then set the dial to the desired temperature for the room and put the cover back on the thermostat.



Connect the main air (blue line) and the branch line (green line) to the M and B fittings for the type of thermostat you are about to calibrate. Connect the switch line (red line) to the third line of three pipe thermostats for calibration.
The back plates illustrated must be replaced for the type of back plates suiting the thermostats you calibrate.

PNEUMATIC THERMOSTATS

SINGLE BAND, SUMMER/WINTER, DAY/NIGHT

There are **many** different pneumatic thermostat models presently in the field. Pneumatic thermostats come in single, two and three pipe configurations. A collection of the manufacturers' data sheets on all makes and models of pneumatic thermostats would be overpowering; therefore, we are going to take a generic approach to troubleshooting and calibration regarding types of thermostats, developing a common sense approach to the subject. You should be able to trouble shoot and properly calibrate any thermostat even from a manufacturer you never knew existed. Most often specialized tools and gauge adaptors are required.

Pneumatic thermostats come as relay and non-relay units. Page 3.20 contains functional illustrations and calibration instructions for single band one pipe non-relay thermostats. Page 3.21 contains functional illustrations and calibration for single band relay thermostats. The odd thermostats in the industry are the POWERS (SIEMENS) "D" stat. They do not have a pilot orifice. They operate off a double ball arrangement for the makeup and exhaust seats. There are non-relay two pipe thermostats in the industry which are basically two single ban thermostats with one controlling the heating and the other controlling the cooling, in a dead band arrangement.

Thermostats are either direct or reverse acting, (see page one) based on the controlled medium characteristics (hot water/chilled water, hot air/cold air, etc.) and the normal position of the valve, damper, transducer, VAV box, bypass box or pressure switch being controlled. Some thermostats have a direct acting bimetals for winter operation and a reverse acting bimetals for summer operation.

Each time you calibrate a thermostat set the branch pressure at mid range of the controlled device and breathe on the bimetal, witnessing the branch pressure of the thermostat. The signal should be responsive and in the correction direction for the application.

The various combinations, obtaining the desired end result, are extensive. The best failsafe position for the controlled devices must always have first consideration. Usually heating fails on and cooling fails off; however, logical exceptions exist. Sometimes you are required to pick the least unsafe of a few choices. For example: If your choice is risking freezing a coil, or blowing up a building in an explosion-proof environment, you choose the risk of freezing the coil as the safest failsafe position.

The vast majority of pneumatic thermostats are proportional, meaning they vary their branch (output) signal at infinite pressures between zero PSIG (pounds per square inch gauge) and the main air PSIG.

Johnson Controls made a two position, T400 series, thermostat, meaning it can only produce a branch signal of either zero PSIG or the main air PSIG. It cannot modulate between those pressures. In over forty years in the pneumatic service field I've only seen one of these thermostats: they are rare.

The pilot orifice (see illustration on page 3.20 for relay thermostats) is normally non-adjustable at .005". The obsolete, Johnson Control, T400 series, to my knowledge, is the only pneumatic thermostat with an adjustable pilot orifice. The adjustment is a screw under the cap-screw facing directly toward you at the bottom of the large brass plate. A special Johnson Control gauge is required, which has a full range of 35"WG. The T400 factory pilot setting is 5"WG.

This is the pressure read on the gauge with the leak port completely clear to atmosphere.
The T400 pilot orifice may be set without the gauge. Screw a gauge, with adaptor, into the branch line test port at the top of the thermostat. Lift the lid assembly, opening the leak port fully to atmosphere. The branch test gauge should read zero PSIG. Let the lid assembly rotate to the leak port, putting slight pressure on it to seal off the leak port. There should be a slight time hesitation before the branch gauge starts to raise its pressure, climbing steadily to the main air pressure. If this is not observed, remove the pilot orifice cap-screw from the thermostat. Adjust the pilot screw until you can barely hear a slight hissing sound from the air. Plug the cap-screw opening and rotate the lid assembly lifting the lid assembly, clearing the leak port completely to atmosphere. The Branch pressure should read zero PSIG. Let the lid assembly rotate to the leak port, putting slight pressure on it to seal off the leak port.
There should be a slight time hesitation before the branch gauge starts to raise its pressure, climbing steadily to the main air pressure. Repeat adjustment on the pilot screw until you witness a slight hesitation before the branch pressure starts to rise steadily to the main air pressure.
While we are on this obsolete thermostat, lets address the pivot adjustment. There are two pivots for the lid assembly, which varies the air flow from the leak port and two pivots on the bimetal assembly. You can make a spring hook with a paperclip to temporarily remove the small spring from the bimetal assembly. Both assemblies then may freely rotate on their pivots. Adjust the pivots allowing the assemblies to rotate freely with no lateral move at all. (Attempt to wiggle laterally with your fingers.) The leak port is only .0012"; therefore, flopping of either assembly will cause major calibration issues. (This pivot adjustment applies to all Johnson Control instruments with pivots.) Re-install the spring. Calibrate the thermostat as per page six. The calibration screw is at the bottom of the thermostat under the dial.
Contaminants in the main air supply can be devastating to a pneumatic control system. (see report on Compressed Air Care starting at page 9-160) If the pilot orifice becomes plugged, the branch signal drops to zero PSIG. If the leak port (flapper nozzle) becomes plugged, the branch signal is locked at the main air pressure.

Do not assume that factory calibration stated on the shipping box is accurate. I once checked three hundred new day/night thermostats, from a major manufacturer, for day set point, night set point and change-over point on one of our calibration boards. I did not find one (not one) that was correct for all three functions of the three hundred (300) thermostats. Other issues found were dead bimetals, wrong action bimetals, thermostats non-responsive to the dial and thermostats that would not build satisfactorily to maximum branch pressure.
If a thermostat that is factory calibrated correctly for a control point of say six PSIG and your application requires a control point of 13.5 PSIG, it is still out of calibration by three Fahrenheit degrees, based on the common sensitivity of 2.5 PSIG per Fahrenheit degrees.
Factory calibration being correct is based on luck, not good instrumentation.

DAY/NIGHT HEATING THERMOSTATS

Day/night heating thermostats control the room at a lower temperature during unoccupied periods than during occupied periods.

The means of altering from unoccupied set point to occupied set point is a change in main air pressure. There are several different combinations achieving the change-over, from manufacturer to manufacturer and even sometimes within the same manufacturer. Some common arrangements have

15#D (day)/20#N(night), 13#D/18#N,14#D/22#N, 16#D/25#N. The change-over is achieved normally by a time clock or the Building Automation System (BAS). If you replace a thermostat in an existing day/night system you must be sure that the change-over point on the new thermostat is adjusted to be suitable in the existing system.

These thermostats must be calibrated using a calibration change-over board, allowing calibration of the change-over point as well as the day and night set points. (see calibration board page 3.22)

The change-over point is calibrated by causing day side to have a 0# branch signal and the night side to have a full branch signal. Slowly raise and lower the main air pressure from the day main air setting to the night main air setting. The thermostat branch pressure will jump from 0# to full branch at the change-over set point of the thermostat as you raise the pressure. There is a means of adjusting this change-over function on every day/night thermostat. Adjust this point and check by raising and lowering the main air pressure until the change-over point is mid-range of the day main air pressure and the night main air pressure.

Some day/night thermostats have only one dial and others have two dials. They all have an adjustment for calibrating the day mode and an adjustment for calibrating the night mode.
-The two dial models are calibrated for day and night in the same manner at the single band thermostat illustrated on page 3.21. This is similar to calibrating two single band thermostats on one base. After calibration be sure to set the night dial to the desired night setting and the day dial to the desired day setting.
The single dial models are a little more complicated to calibrate. The day bimetal is calibrated in the same manner as the single band thermostat illustrated on page 3.21.
-For the night side you either have to cool the room down to the night set point, which is very impractical or cause the thermostat to “believe” that the room is 10F° cooler than actually exists. When calibrating the night side, turn the thermostat dial up 10F°, causing the thermostat to “believe” the room became 10F° cooler. Adjust the branch pressure to the desired control pressure reflecting the thermostat is satisfied. Breathe on the bimetal checking that the bimetal is correct and active. Turn the thermostat dial back down to the desired day set point for the thermostat.

Some day/night thermostats are three pipe units, typically used on unitventilators (see page7.61), allowing one room day operation while the rest of the building is on night mode. The indexing switch on these thermostats must be manually pushed to re-establish day mode when the system is on nights. The next day, when the main system switches back to day mode, the indexing switch, which was pushed the night before, automatically resets to normal operation, cycling day/night with the rest of the building.

<u>SUMMER/WINTER THERMOSTATS</u>

These thermostats must be calibrated using a calibration change-over board, allowing calibration of the change-over point, the summer set point and winter set point. Calibrate during the season operating on the higher main air pressure, allowing calibration of the three functions at the thermostat locations. (calibration board page 3.22)
Most are non-relay, low volume instruments; therefore, slow to drive the valve (s). (Some are relay, high volume, instruments.) The normally closed pneumatic push button on the induction unit calibration

board allows you to test the branch line and valve diaphragm (s) while you calibrate the induction unit thermostat.

These thermostats are typically found on induction unit systems (see page 7.58) where the valve being controlled receives hot water in the winter mode and chilled water in the summer mode. Opening the valve in the winter gets heat, while opening the valve in the summer gets cooling; therefore, the thermostat has to reverse its logic for each mode of operation.

These thermostats operate similar to the day/night thermostats. They have one direct acting bimetal and one reverse acting bimetal, where the day/night thermostats have two identical bimetals. The means of changing from one bimetal to the other, reversing the thermostats' logic, is also via altering the main air pressure to the thermostats. The summer/winter pressures used vary to the same extent as the day/night systems. The change-over is normally manual and must always match the summer/winter mode of the mechanical system.

The calibration method is the same as the day/night calibration. You can control to a lower temperature for winter than summer by offsetting the control point pressure. If the thermostat's sensitivity causes a branch change of 2.5 PSIG per Fahrenheit degree and you wish to control two Fahrenheit degrees cooler in the winter, you adjust your winter control pressure five pounds higher than the pressure used for summer control pressure when calibrating.

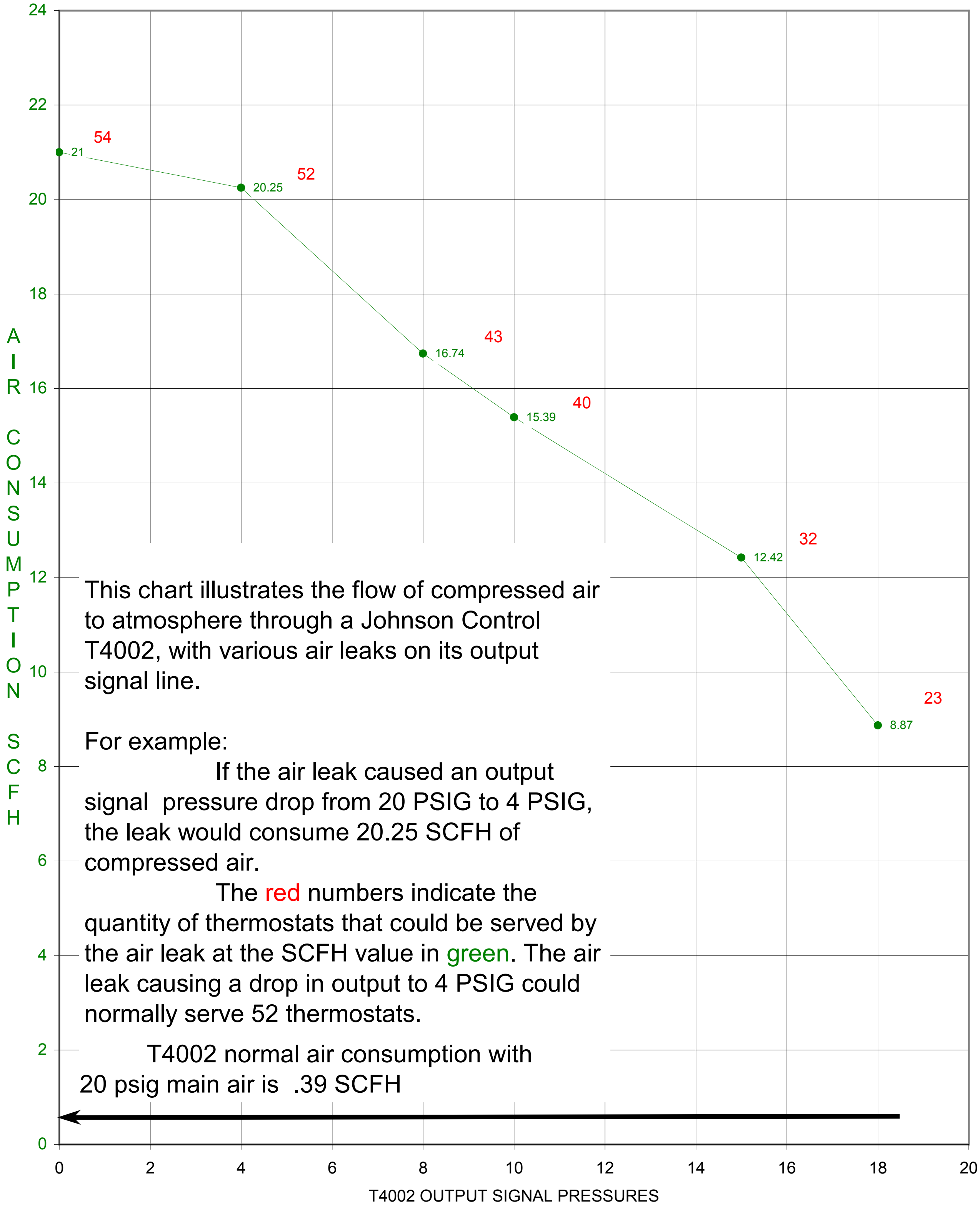
If the system is, for example, a 15 PSIG winter and 20 PSIG summer with a change-over point of 17.5 PSIG and air leaks cause the main air in an area to drop below 17.5 PSIG, the thermostats in that area will lose control, as their logic will be completely backwards in addressing the requirements of the area. I remember one building where we were on a contract. The building had many leaks not associated with our work causing the main air pressure to drop below the change-over point in some areas. We advised the building operator to find and correct the air leaks in order to correct the situation. He chose to run around the building with a magic marker reversing the dials, which read "Cooler/Warmer". Obviously he corrected nothing and acquired no control in those areas. Hopefully, when he fan out of magic markers he considered our suggestion to fix the air leaks.

There are models which look similar to a day/night thermostat. There are models which have thermal capillary element attached directly to them. There are models which have three ports on the thermostat, being the main and branch ports, as well as a sensing port, which is tubed to a remote bimetal controlling the pilot pressure of the thermostat's relay.

THERMOSTATS REGARDING ENERGY CONSERVATION

While manually or automatically controlling a building, providing a healthy environment, comfort and best energy performance, a continuous input of information is required, allowing logical positioning of heating, cooling and ventilation functions.

JOHNSON CONTROLS T4002
TOTAL AIR CONSUMPTION (SCFH) WITH BRANCH LINE LEAKS CAUSING VARIOUS PRESSUR DROPS



This chart illustrates the flow of compressed air to atmosphere through a Johnson Control T4002, with various air leaks on its output signal line.

For example:

If the air leak caused an output signal pressure drop from 20 PSIG to 4 PSIG, the leak would consume 20.25 SCFH of compressed air.

The red numbers indicate the quantity of thermostats that could be served by the air leak at the SCFH value in green. The air leak causing a drop in output to 4 PSIG could normally serve 52 thermostats.

T4002 normal air consumption with 20 psig main air is .39 SCFH

SECTION FOUR
(TRANSMITTERS, CONTROLLERS & RECEIVER CONTROLLERS)

PURPOSE OF SECTION

Present controllers, transmitters and receiver controllers

PNEUMATIC TRANSMITTERS

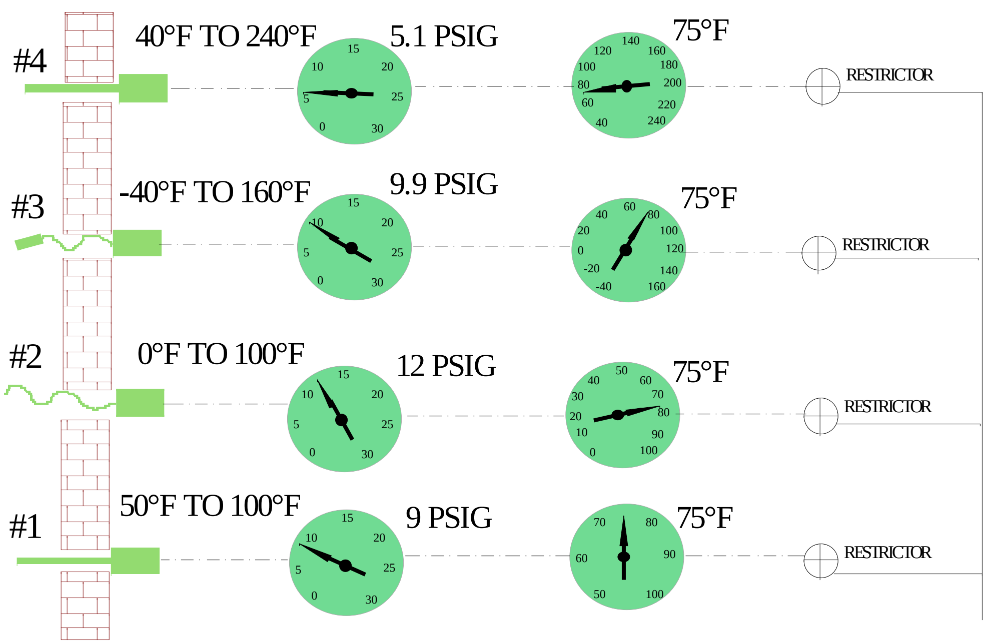
Most commercial pneumatic transmitters are one pipe, as illustrated; however, two pipe transmitters do exist. Note that they all have a 3 PSIG to 15 PSIG range regardless of their temperature, humidity, static pressure, etc. range. They only report the condition; they do not do the controlling.

They are all direct acting; therefore, when the element loses its gas, a low value is simulated on the transmission signal.

Most transmitters use restrictors in the range of .007", but some use restrictors in the range of .005". The graphs on pages 4.29 and 4.30 illustrate the relative air consumption of most of the manufacturers' restrictors.

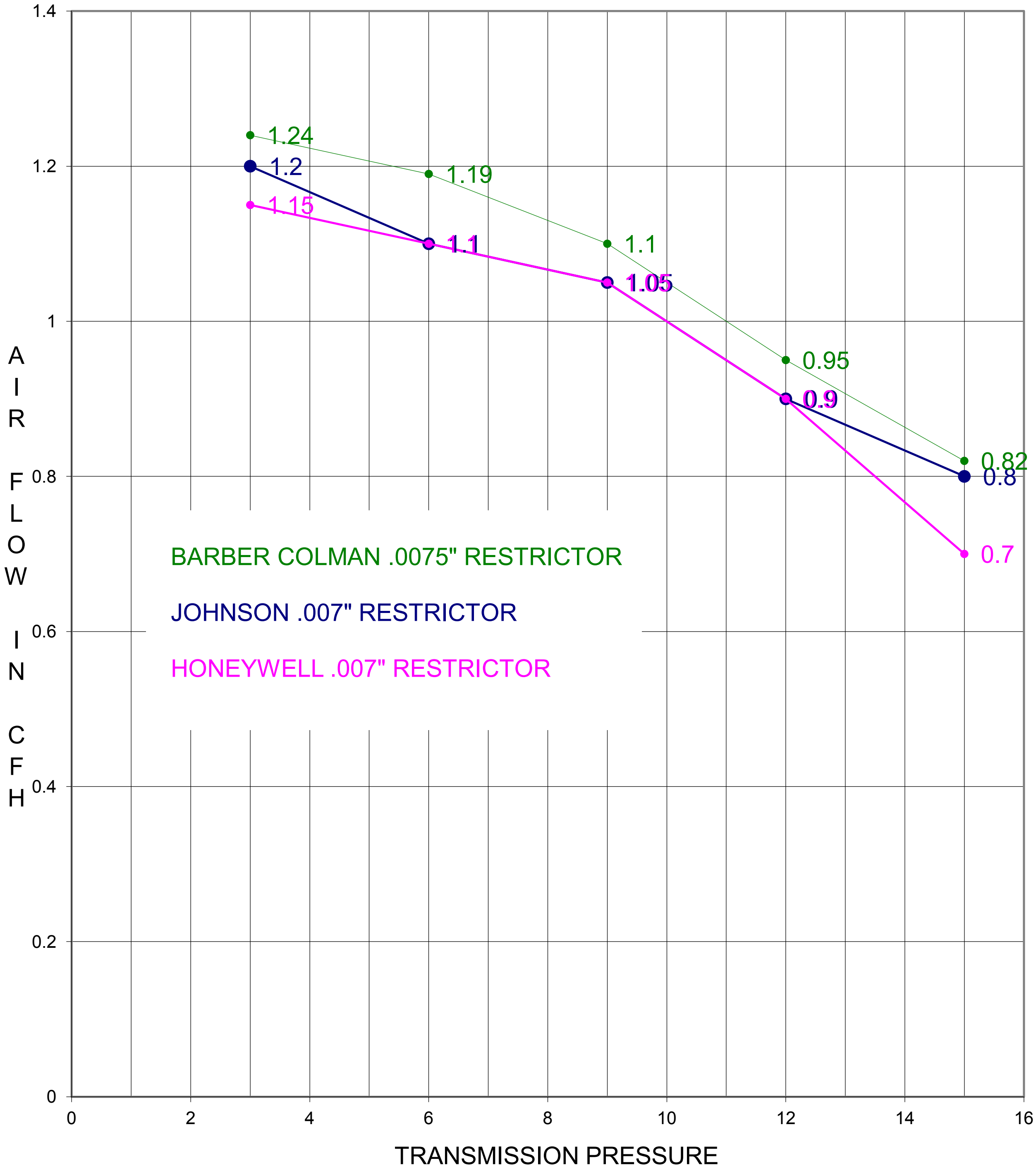
Transmitters' operational characteristics are very useful in energy conservation efforts.

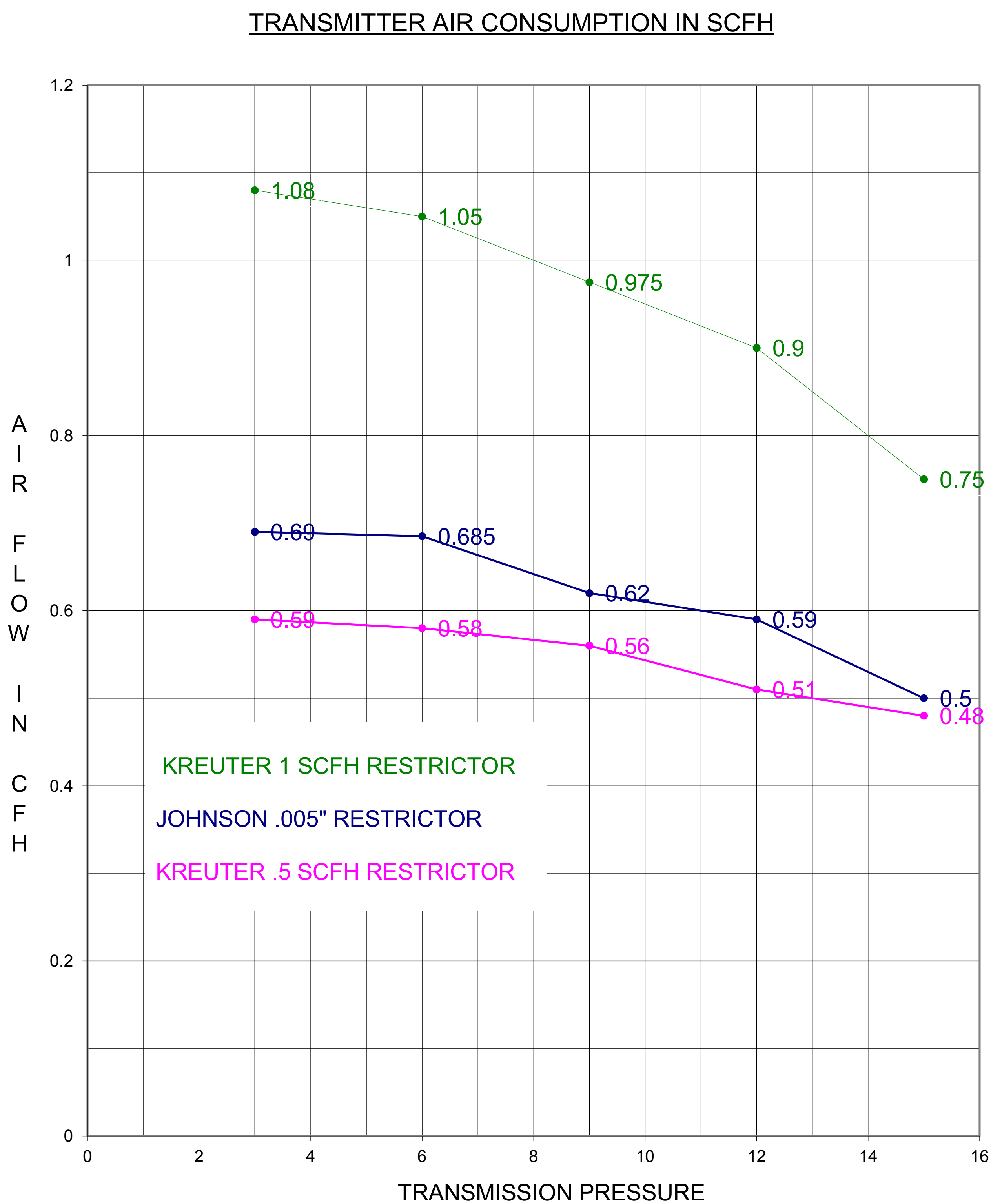
For Example: If the outside temperature is 10°F, the –40°F to 160°F transmitter will be sending a 6 PSIG signal to the control system, resetting the main heat to the building. If the area requiring the most heat over-rides that 6 PSIG signal with a 9 PSIG signal, via a selector, the control system will believe that the outside air temperature is 60°F rather than the true 10°F. This is the means of matching the main heat source temperature to the actual requirement of the coolest area of the building.



	#1	#2	#3	#4
15 PSIG	100°F	100°F	160°F	240°F
12 PSIG	87.5°F	75°F	110°F	190°F
9 PSIG	75°F	50°F	60°F	140°F
6 PSIG	62.5°F	25°F	10°F	90°F
3 PSIG	50°F	0°F	-40°F	40°F

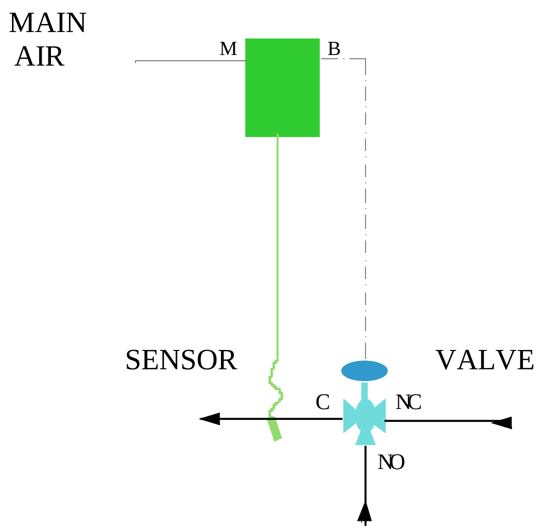
TRANSMITTER AIR CONSUMPTION IN SCFH




TRANSMITTER AIR CONSUMPTION IN SCFH
AIR FLOW IN CFH
TRANSMISSION PRESSURE
1.2
1
0.8
0.6
0.4
0.2
0
0
2
4
6
8
10
12
14
16
1.08
1.05
0.975
0.9
0.75
0.69
0.685
0.62
0.59
0.5
0.59
0.58
0.56
0.51
0.48
KREUTER 1 SCFH RESTRICTOR
JOHNSON .005" RESTRICTOR
KREUTER .5 SCFH RESTRICTOR

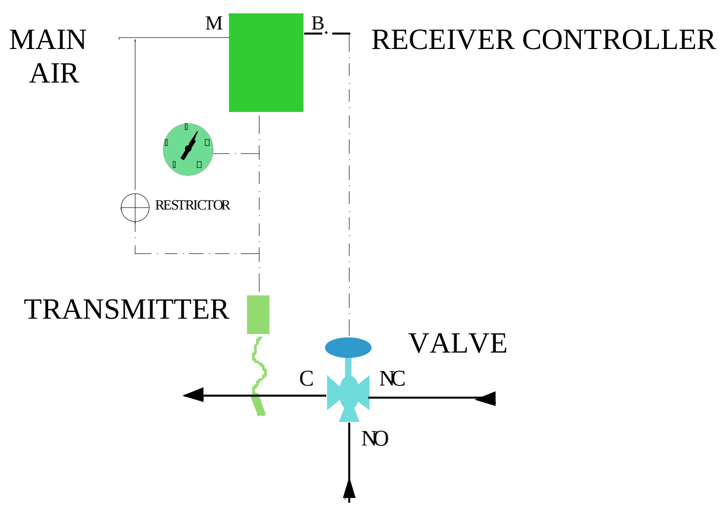
<u>CONTROLLER</u>

Controllers sense the controlled medium directly with their sensing mechanism.
This could be a thermal sensor, humidity sensor, air-flow sensor, etc.
Most controllers have sensitivity adjustment. Controllers can be either direct or reverse acting. Controllers may have proportional or two position control.



<u>RECEIVER CONTROLLER</u>

Receiver controllers do not directly sense the controlled medium. Transmitters directly sense the controlled medium and report the condition of that medium. The receiver controller controls the condition of the medium.
Receiver controllers have sensitivity adjustment. Some have secondary port reset. Some have control port adjustment. Some have integral reset. Receiver controllers can be direct or reverse acting. Receiver controllers may be two position or proportional control.



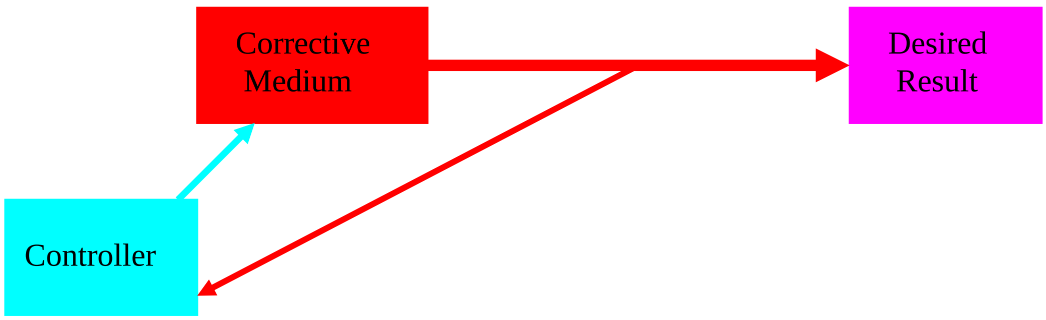
CONTROL LOOPS

OPEN LOOP

-The corrective medium is applied with no feed back indicating the actual condition of the addressed function.

-Example- Setting your furnace to run 50% of every hour with no thermostat sensing the temperature in your house.

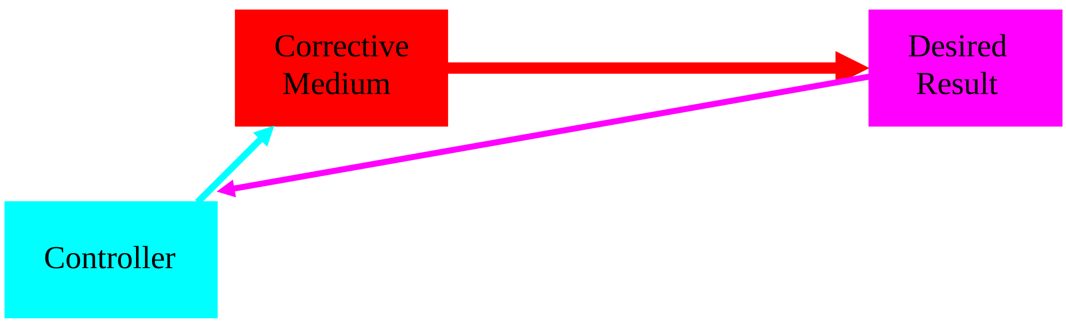
-The accompanying diagram illustrates the controller receiving information regarding the amount of corrective medium being applied, but does not sense the desired result.



CLOSED LOOP

-The corrective medium is applied based on the actual condition of the addressed function.

-The accompanying diagram illustrates the controller controlling the corrective medium based on receiving information regarding the desired result.



CONTROL LOOP COMPONENTS

DESIRED RESULT

-The target condition or set point for the addressed function.
-Temperature in a room, duct, pipe, etc.
-Humidity level in a room or duct.
-Level in a tank.
-Air quality level in occupied space.
-Open a door when some one approaches.
-Etc.

CORRECTIVE MEDIUM

-A replacement supply of what ever is required to maintain the desired condition of your addressed function.
-Hot or cold water, hot or cold air, pressurized air, steam, liquid, refrigerant, etc.

LOAD

-A varying condition that causes the desired result to be lost by consuming the corrective medium.
-Loss or gain of heat, loss or gain of humidity, loss or gain of liquid level in a tank, loss or gain of pressure in a duct, pipe, tank, etc.

CONTROLLER

-The decision-maker that determines when and how much corrective medium to apply.

CONTROLLED DEVICE

-The apparatus used by the controller applying the corrective medium to the addressed function. (Valve, damper, etc.)

CONTROLLER FUNCTIONS

-The controller is the decision-maker in the control loop.
-You are the controller when a manual hand valve requires adjustment.

INPUT SIGNALS

-The input signals are information telling the controller the existing conditions relating to the desired result.
-The controller may receive one input signal, two or more in determining the appropriate corrective action.
-The controller may sense the existing conditions directly with its own sensing element or receive information signals from transmitters.
-When you are manually adjusting a valve, you must obtain information allowing decision to be made regarding the desired position of the valve.

OUTPUT SIGNAL

-The output signal is the action taken by the controller after the controller compares the desired result (set point) to the actual condition of the controlled environment. The difference is the error from set point or sometimes called droop.
-Output signals can be either two position or proportional. Two position will have the controlled device (valve, damper, etc.) either fully open or fully closed. Proportional will gradually modulate the controlled device ranging from fully open to fully closed.
-Some controllers can be set up for several specific steps from fully open to fully closed.
-When you are required to manually adjust a valve, you must be able to physically change the degree of opening of the valve.

PROPORTIONAL BAND, SENSITIVITY, GAIN

-These terms relate to the same function, which is the amount of output signal change relating to a specific amount of input change on the controller.
-The more insensitive a controller, the more error from set point occurs at load extremes. This error from set point is called droop or offset.
-If a controller is hunting or sometimes called cycling, the controller is over-shooting and the sensitivity has to be lessened to settle the system.
-When you are manually operating a valve you must gradually open the valve to the proper position or the system will over-shoot.

INTEGRAL RESET

-Reset is a controller function that corrects the error from set point or droop. Not all controllers have this function.
-When manually operating a hand valve on steam you would position the valve more open in cold weather than in warm weather to attain the same degree of heat in the air or water. This is similar to integral reset in controllers.

AUTHORITY, RESET

-This function allows the set point to vary based on another variable. An example is outdoor/indoor reset on heating hot water. As the outside air becomes colder, the set point for the heating hot water becomes hotter.
When manually operating a water temperature for heating, you would likely target a hotter water temperature in cold weather than in warm weather conditions. This is similar to the authority function in controllers.

SECTION FIVE
(BASIC LOGIC)

PURPOSE OF SECTION

Present opportunities for energy reduction over conventional design.

HEATING INPUT

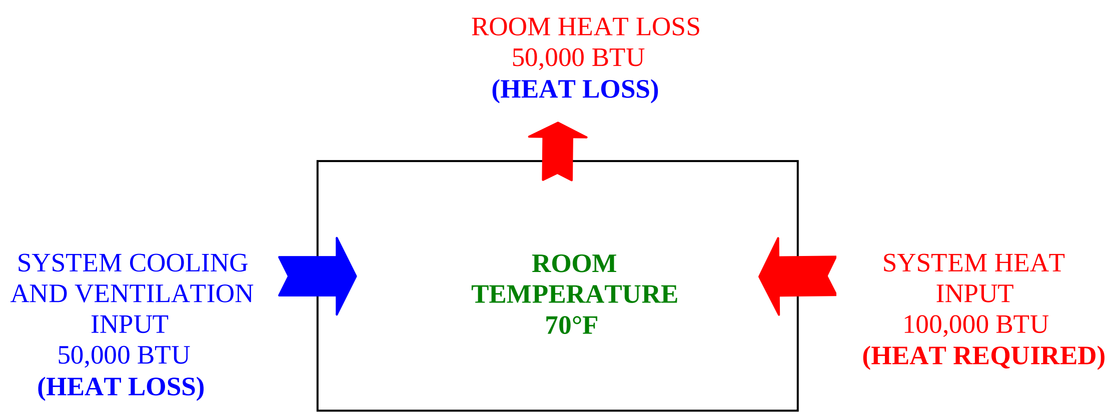
Many control systems determine the main building supply heating temperature based only on the outside air temperature. This similar to allowing your furnace to run for varying number of minutes per hour based on the outside air temperature. This might sound logical, but consider these scenarios.

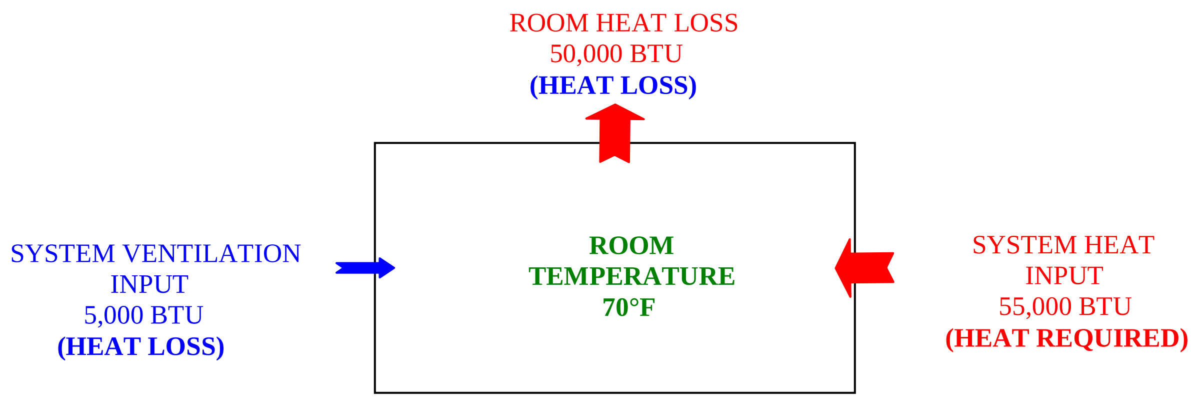
SCENARIO	OSA TEMP.	FURNACE RUN	WIND	OCCUPANCY	SUN CONDITION	ROOM TEMP.
ONE	30°F	15 MIN./HR	STILL	25 (GATHERING)	BRIGHT NOON SUN	?
TWO	30°F	15 MIN./HR	BLIZZARD	YOU ALONE	NIGHT	?

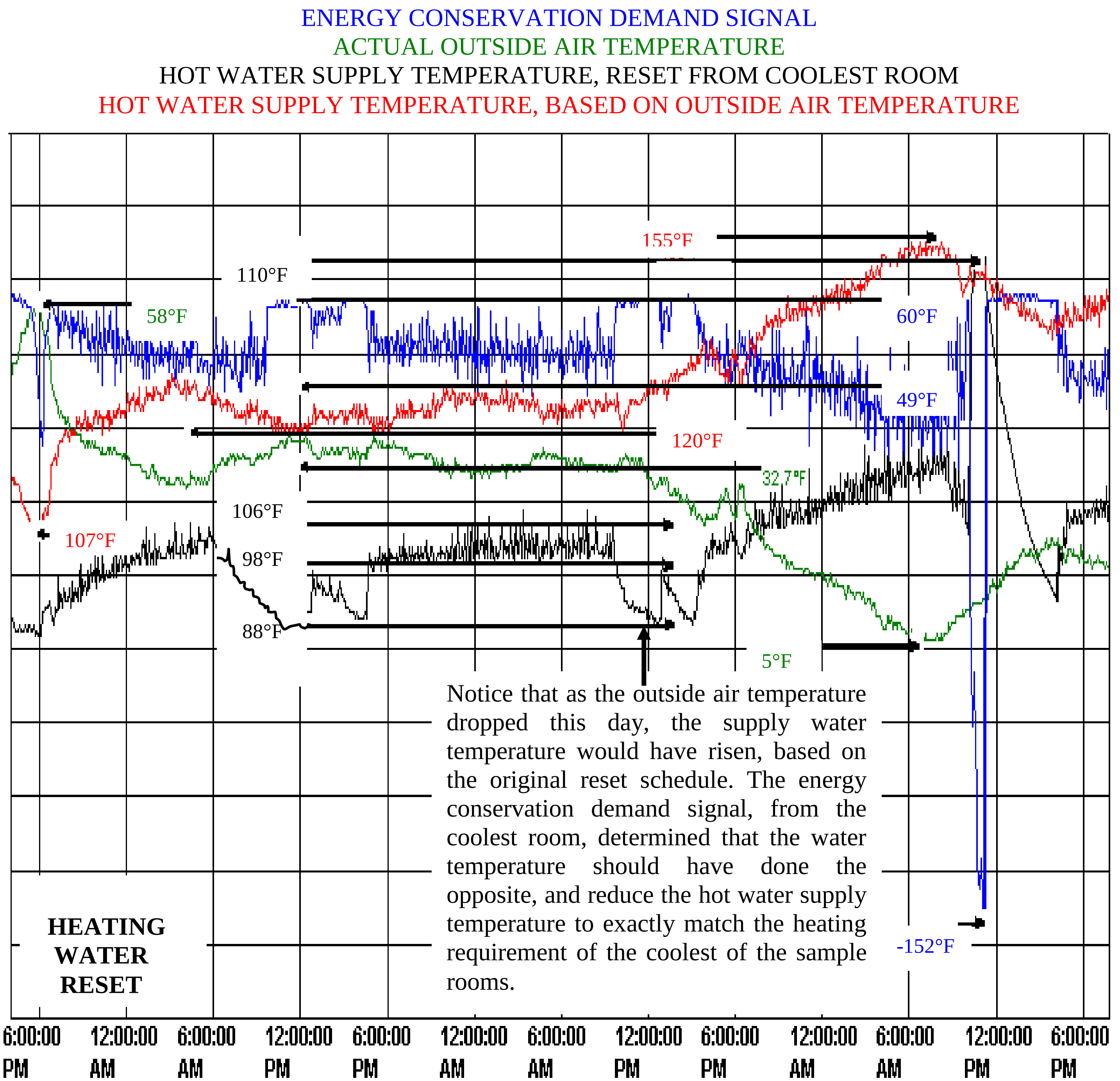
If the fifteen minute run time per hour, based on the 30°F outside air temperature is enough to keep you warm in scenario two's conditions, the gathering will over-heat under the conditions of scenario one with the same run time based only on the outside air temperature.

The graph on page 5.36 illustrates a comparison of actual heating water requirements of the coolest area of the building to heating water determined by only outside air temperature.

CO-ORDINATING THE HEATING AND COOLING
WHICH ARRANGEMENT SEEMS BETTER?







This graph illustrates the heating water temperature difference between resetting only from the outside air temperature and resetting from the coolest room of the area served.

The red line illustrates the water temperature based on the actual outside air temperature, illustrated by the green line. This is the originally designed reset schedule. The red and green lines are symmetrical.

The black line illustrates the water temperature based on the heating requirement of the coolest area served illustrated by the blue line. This is the conservation circuit. The black and blue lines are symmetrical.

The coolest room's signal "tells" the main reset controller that the outside air temperature is a warmer than the actual outside air temperature, causing the reset controller to produce heating water at exactly the correct temperature to keep the coolest room in the comfort zone.

Note that the water temperature drops when the coolest school class becomes occupied in the morning: the water temperature rises at noon when the bodies leave the room: it drops when the bodies re-enter the room for the afternoon and rises again when the people leave for the day.

MIXED AIR

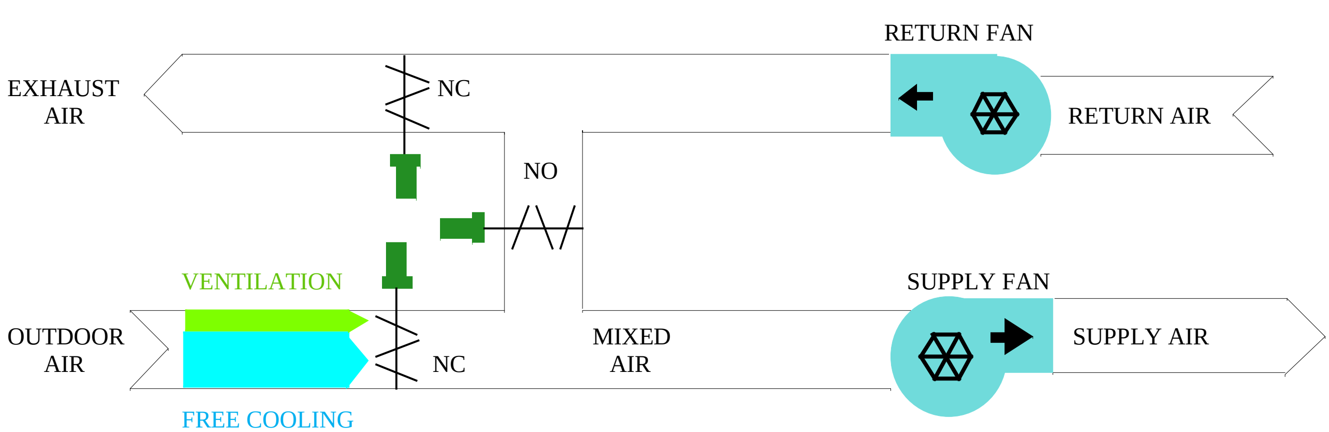
Mixed air in a fan system is the body of air that is a mixture of outside air and return air from the building. Most conventional fan systems attempt to control the mixed air temperature in the range of 55°F. A certain percentage of the fresh air is required for ventilation purposes and exhaust air replacement, while fresh air beyond that percentage is required for cooling purposes only.

Often the mixed air is referred to as "FREE COOLING", but is it always? Most buildings require varying degrees of cooling, even during the winter. The mixed air in the range of 55°F is the design temperature for times when the building requires maximum cooling. What happens when that amount of cooling is forced into the building and the maximum amount of cooling is not required?

The building will either become uncomfortably cold or the building system will have to add heat to compensate for the unwanted cooling.

This is similar to going home and putting a manual switch on your central air conditioner to have refrigeration active all the time. When the load for cooling is great enough to match the capability of the refrigeration unit the building will be comfortable. When the cooling load drops, you will either have to find a way to have the cooling delivered match the actual requirement or turn on your furnace to remain comfortable.

"FREE COOLING" is not "free" if the amount of cooling delivered is not required by any part of the receiving areas. This "FREE COOLING" cost is equal to the heating energy cost to correct for the over-cooling.



HIDDEN HEAT LOSSES VIA THE CONTROL SYSTEM'S LOGIC

CONTENTS

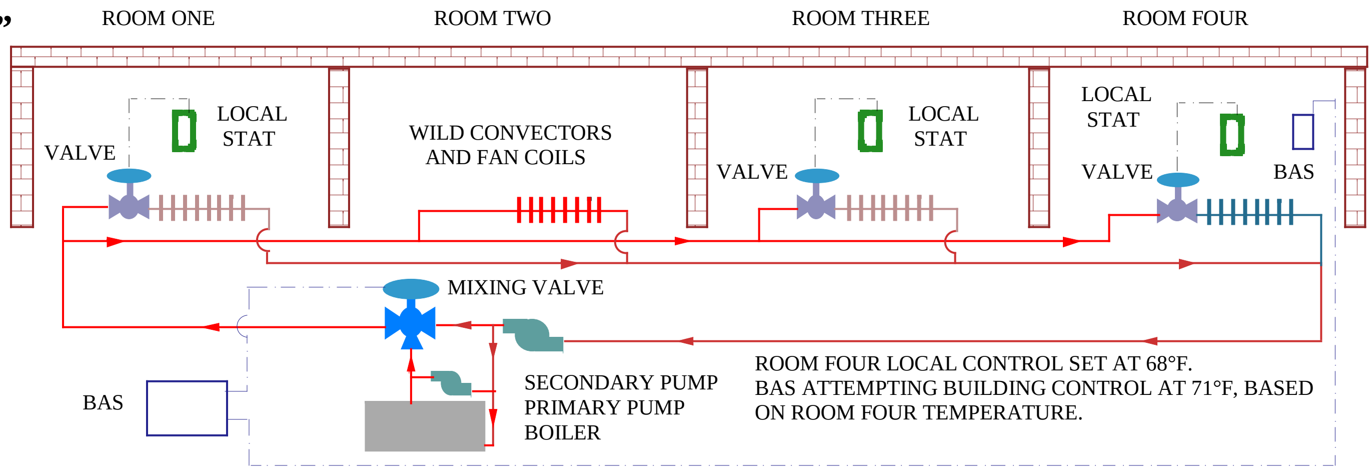
SEVEN EXAMPLES OF SYSTEMS HIDING ENERGY LOSSES

NOTE:

The fact that the space is comfortable is not assurance of proper or logical control.

ENERGY LOSS VIA CO-ORDINATION LACK , REGARDING LOCAL AND BAS CONTROL

“A”



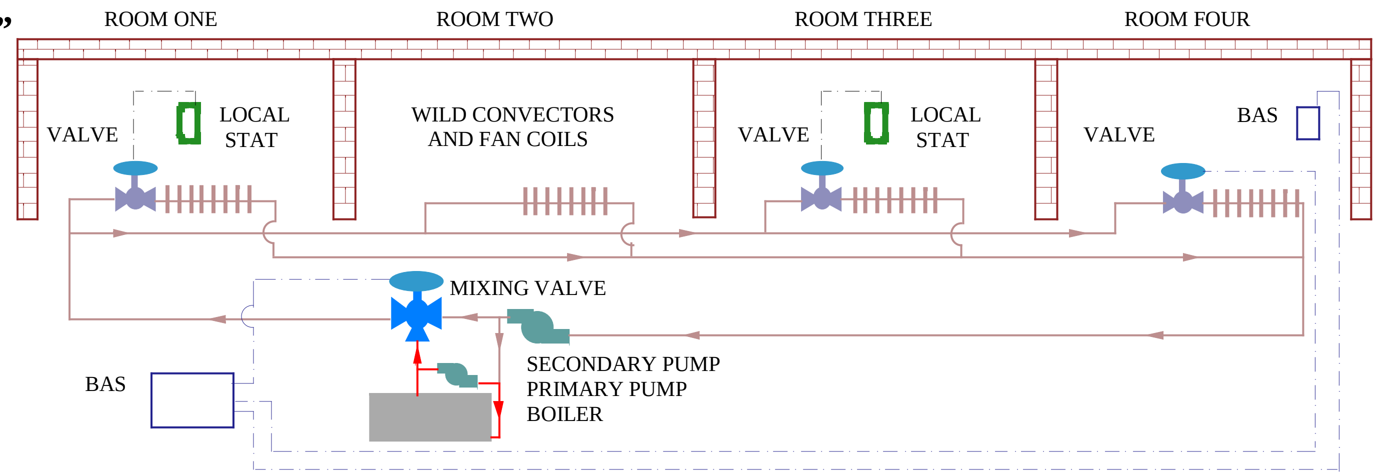
Scenario “A” presents a very common problem where a Building Automation System (BAS) over-lays an existing building control system.

The new BAS scans sample rooms, determining the greatest requirement for heat. Room four (one of several sample rooms) is the coldest room with a temperature of 69.5°F. The local thermostat is set at 68°F; therefore, the valve controlling heat entry to the room is closed. The BAS attempts to control the coldest room to 71°F; therefore, raises the secondary water temperature. The increased water temperature can not correct the cool condition in room four, as the local thermostat disallows flow to the room.

At higher water temperatures, increased heat is lost in the main heating distribution piping and in areas where wild convectors or fan coils exist.

In locations with local thermostat control, other than room four, the valves have an increased risk of damage to their seats and discs, as they are forced to modulate their valves at a more closed position.

“B”



Scenario “B” presents one solution regarding the problem of scenario “A”.

The BAS takes local control of the valves regulating the hot water flow into the sample rooms. The BAS opens the local control valve completely, assuring full water flow into the room before the BAS starts to raise the secondary hot water temperature in response to that room’s temperature set point.

The average cooler water temperature causes less heat loss regarding the main hot water piping, as well as in areas with wild convectors or fan coils.

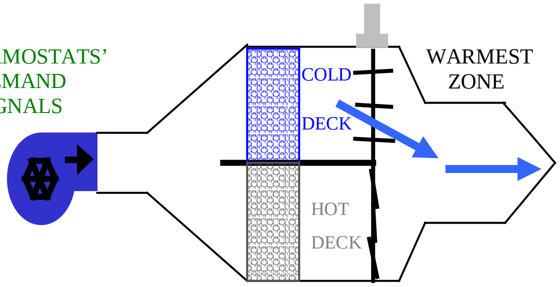
The average cooler water temperature causes valves in other areas to be modulated in a more open position, reducing the risk of disc or seat damage in the valves.

The same logic is attainable leaving local pneumatic thermostats controlling their individual valves. The sample rooms’ signals are tapped and the coolest room’s signal is determined via a multi-low selector. The lowest signal (coolest room signal) is tubed to a transducer, which produces a voltage or ma signal used by the BAS to control the secondary water temperature.

ZONE HEATING-COOLING SEQUENCING IN MULTIZONE LOGIC

18#

THERMOSTATS' DEMAND SIGNALS



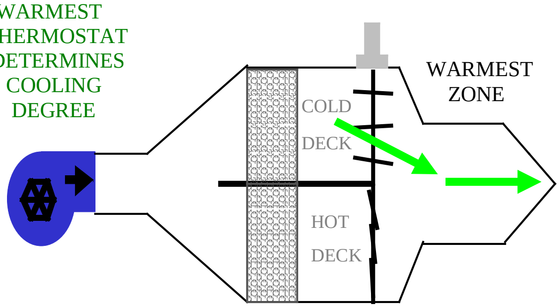
The zone thermostat with the greatest requirement for cooling modulates the free cooling in the winter and the mechanical cooling in the summer, to varying degrees, just enough to keep the served areas within the comfort zone.

The zone demanding cooling does not mix any hot deck air with the air sent to that zone during this mode of operation, allowing the least amount of cooling, in the cold deck, to maintain comfort conditions.

At least minimum ventilation is maintained during all occupied times.

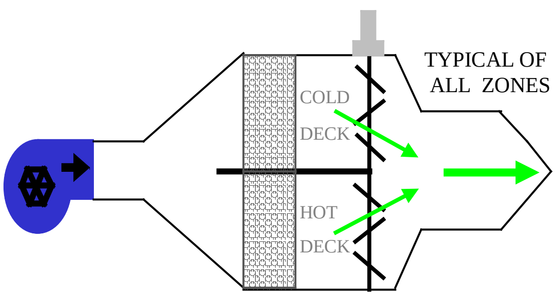
13#

WARMEST THERMOSTAT DETERMINES COOLING DEGREE



The zone thermostat with the greatest requirement for cooling modulates its zone dampers to completely close its hot deck and completely open its cold deck. At this point the cooling is still not active.

At least minimum ventilation is maintained during all occupied times.

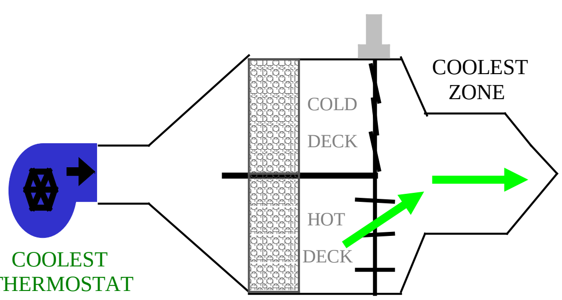


All zone thermostats are modulating their zone hot deck and cold deck dampers within their spring range; therefore, all are within one Fahrenheit degree of set point.

The hot deck is maintained at minimum heat and the cold deck for summer holds the cooling off and for winter the free cooling is off.

At least minimum ventilation is maintained during all occupied times.

8#

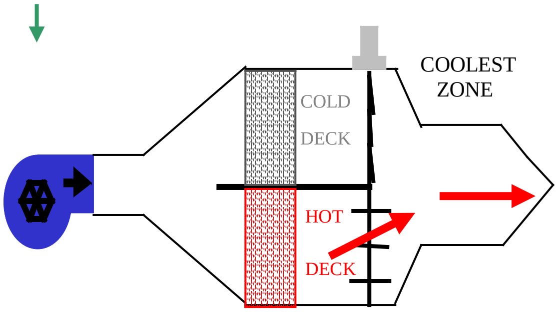


The zone thermostat with the greatest requirement for heating modulates its zone dampers to completely close its cold deck and completely open its hot deck. At this point the heating is still not active.

At least minimum ventilation is maintained during all occupied times.

COOLEST THERMOSTAT DETERMINES HEATING DEGREE

7.5#



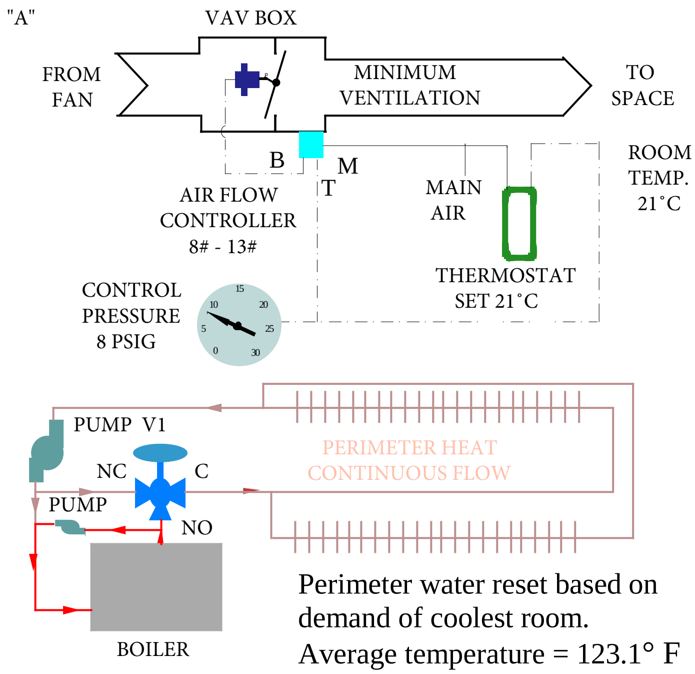
The zone thermostat with the greatest requirement for heating modulates the hot deck heating valve, to varying degrees, just enough to keep the served areas within the comfort zone. The hot deck is also limited by the outside air temperature and sometimes the return air temperature.

The zone demanding heating does not mix any cold deck air with the air sent to that zone during this mode of operation, allowing the least amount of heating, in the hot deck, to maintain comfort conditions.

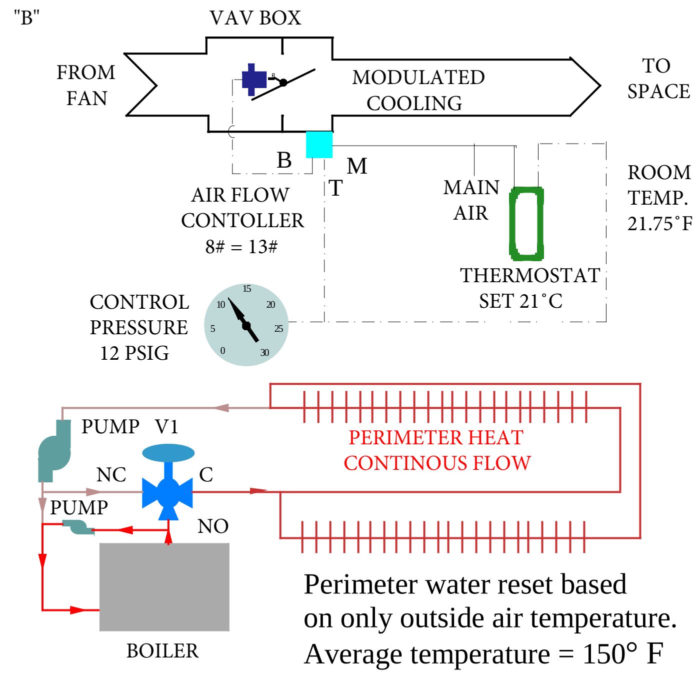
At least minimum ventilation is maintained during all occupied times.

2#

HIDDEN ENERGY LOSS: VAV WITH PERIMETER HOT WATER OSA RESET



Perimeter water reset based on demand of coolest room.
Average temperature = 123.1° F



Perimeter water reset based on only outside air temperature.
Average temperature = 150° F

Scenario “A” illustrates a building circuit resetting the perimeter water temperature based on the heating requirements of the coolest area, after it has modulated its cooling, via the VAV box, back to minimum ventilation. This assures that the coolest room eliminates excessive cooling via the fan system before that room requests heat.

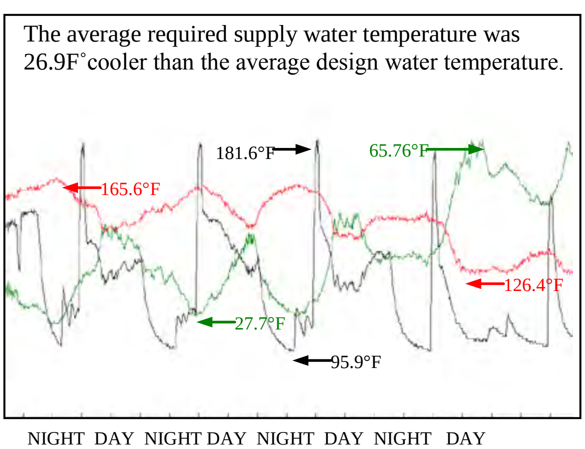
Scenario “B” illustrates a building circuit resetting its perimeter heating water temperature based only on the outside air temperature. As per the two graphs below, reset from only the outside air temperature will provide more heat than required by the coolest room. When unwanted heat is forced into a building, the VAV boxes provide cooling to remove the unwanted heat from the rooms. This is similar to starting your furnace: providing too much heat: and then opening your windows to regulate the room temperature.

Caution must be exercised, regarding impact on the boilers; if night set back is part of your control strategy. Consider the impact of a large volume of cool secondary water entering the primary loop at the changeover from nights to days. Proper control logic gradually increases the secondary loop temperature, keeping the boilers above their minimum allowable firing temperature.

Caution must be exercised in selecting the sample signals, determining greatest heating demand.

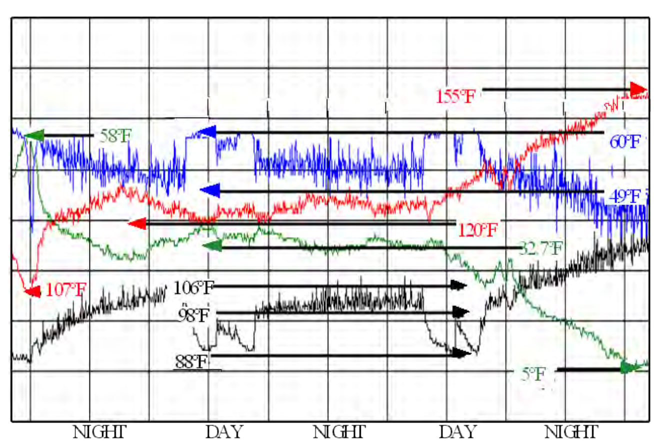
COMMERCIAL OFFICE BUILDING

- Actual outside air temperature.
- Design reset water temperature based on outside air only.
- Actual water temperature required by the coolest room.
- Night set back feature active.

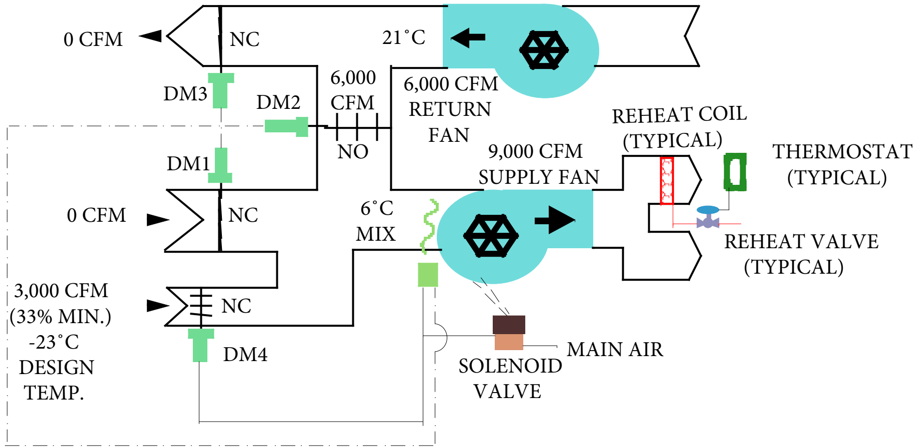


PUBLIC HIGH SCHOOL

- Actual outside air temperature.
- Design water temperature based only on outside air.
- Coolest room heating demand signal.
- Actual water temperature required by coolest room.
- Night set back feature not active.



OUTSIDE AIR PREHEAT WITH MIXED AIR SYSTEMS



Fresh air has two functions, regarding HVAC systems. The first relates to a fixed quantity of outside air, which must always enter the system during occupied periods addressing air quality, exhaust air replacement and building pressurization: the second is free cooling when the outside air is suitable for cooling purposes. Free cooling is normally limited to a minimum fan supply temperature of about 13°C (55°F), preventing draft complaints in the space.

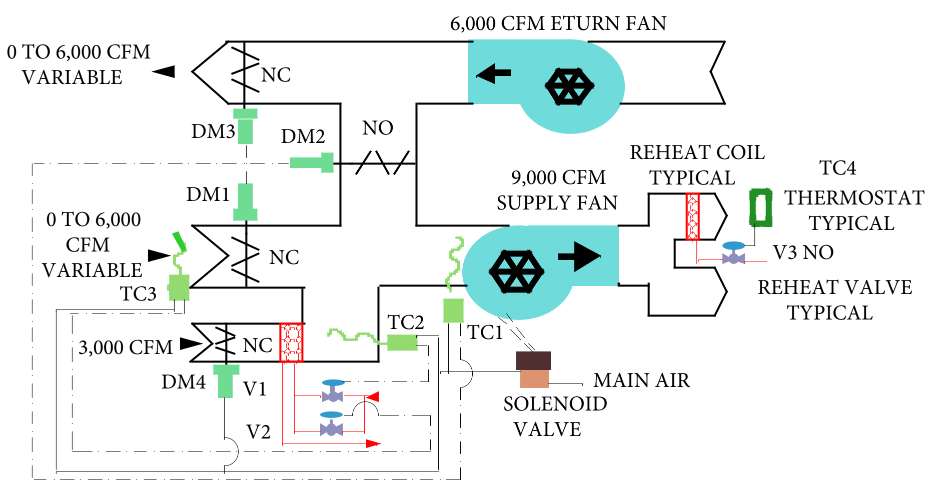
In this example the minimum ventilation requirement is 33.3% of the fan's total volume.

The ratio of mixed air causes the mixed air temperature to drop to the minimum desired temperature of 13°C (55°F) when the outside air temperature drops to 8.6°C (47.5°F). When the outside air temperature drops to the design temperature of –23°C (-10°F) the mixed air temperature will drop to 6°C (43°F).

At design outside air temperature (-10°F in this example), the 9,000 CFM of mixed air requires 116,640 BTU's to bring its temperature from 8.6°C to 13°C. (43°F to 55°F)

(CFM X 1.08 X $\triangle$T = 9,000 CFM X 1.08 X 12F° = 116,640 BTU)

A preheat coil is required preventing the mixed air temperature from dropping below 13°C (55°F).



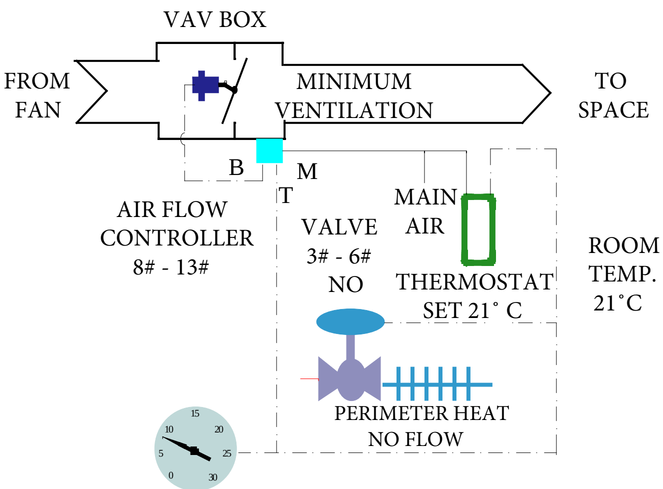
This drawing illustrates a common approach to preheating fresh air. A preheat coil is added with control valves (V1) and (V2). Temperature controller (TC3) opens V2 completely when the fresh air drops to 38°F preventing the preheat coil from freezing. TC2 modulates V1, maintaining the preheat discharge temperature at 13°C (55°F).

The circuit exists in many buildings and does provide preheated outside air; however, the design tends to waste energy. With a return air temperature of 21°C (70°F) and the preheat discharge temperature of 13°C (55°F) the mixed air temperature will be 18°C (65°F) if 33.3% is preheat air. TC1 is commonly set for 13°C (55°F); therefore, automatically brings in cooling via damper (D1) solely to compensate for the excessive heat introduced through the preheat coil. Often V2 opening forces the preheat discharge temperature to rise above 13°C (55°), causing greater energy loss.

We developed a circuit, which tends to correct this matter.

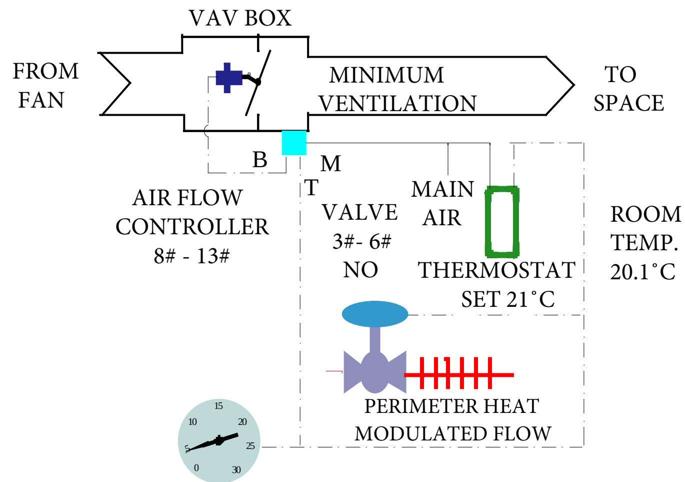
There are a few other means of preheating fresh air. This is the one we have seen most often.

HIDDEN ENERGY LOSS: VAV WITH DEFECTIVE REHEAT HEATING VALVE



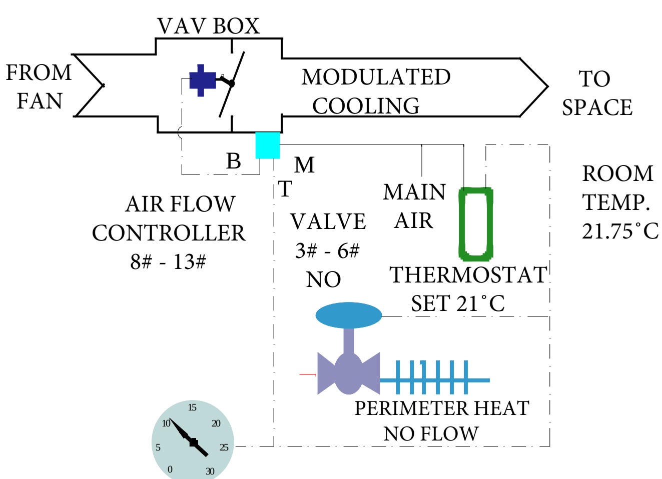
SCENARIO ONE (normal operation)

The thermostat is set at (21°C). The room is 21°C. The thermostat branch signal is at 8 PSIG, forcing the 3# to 6# heating valve closed and the 8# to 13# VAV box to only allow minimum ventilation.
There is no heating/cooling conflict.



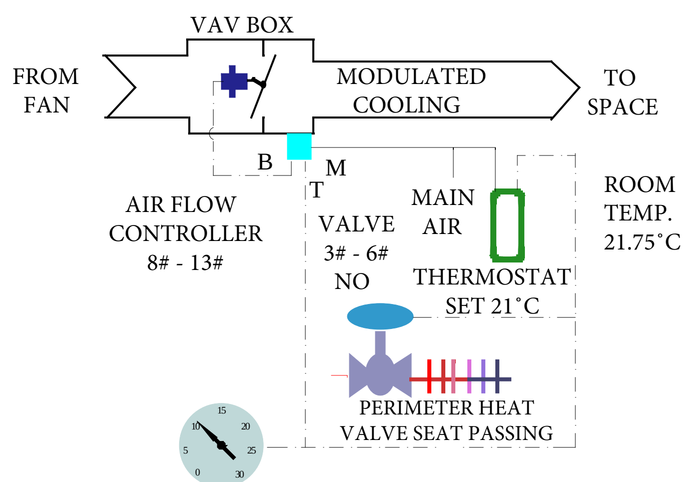
SCENARIO TWO (normal operation)

The thermostat is set at (21°C). The room temperature is 20.1°C. The thermostat branch signal is at 4 PSIG, modulating the 3# to 6# heating valve open and holding the 8# to 13# VAV box to only allow minimum ventilation.
There is no heating/cooling conflict.



SCENARIO THREE (normal operation)

The thermostat is set at (21°C). The room temperature is 21.75°C. The thermostat branch signal is at 11 PSIG, closing the 3# to 6# heating valve and modulating the 8# to 13# VAV box to 60% of maximum cooling airflow addressing normally expected heat gains. (solar, body, lights, etc.)
There is no heating/cooling system conflict.



SCENARIO FOUR (serious problem)

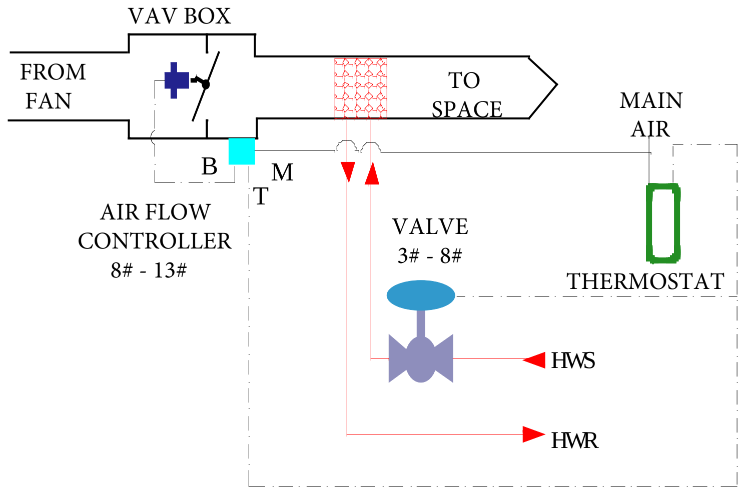
The thermostat is set at (21°C). The room temperature is 21.75°C. The thermostat branch signal is at 11 PSIG, closing the 3# to 6# heating valve and modulating the 8# to 13# VAV box to 60% of maximum cooling airflow. The heating valve's defective seat is allowing unwanted heat gains equal to the solar, body, lights, etc. heat gain in scenario three.

There is heating/cooling system conflict. The cooling is active removing unwanted heat injected via the defective valve seat.

Scenario four is a problem of unwanted heat entering the building via defective control valve seats. This typically occurs when the perimeter heating water is controlled via only outside air reset. The valves are forced to modulate just barely cracked open, causing wear and tear increase on the seats and discs. The problem is not easily detected, as you can see comparing scenarios three and four; the room temperature is identical. The cooling function hides the heating valve's defect. In scenario three no energy is wasted. Scenario four the waste can be severe, similar to turning on your furnace at home and opening windows to dissipate any unwanted heat.

Resetting the heating water temperature based on the area with greatest heating demand, after its control valve is fully open, will tend to save the seats on your control valves and will save energy in reducing heat loss.

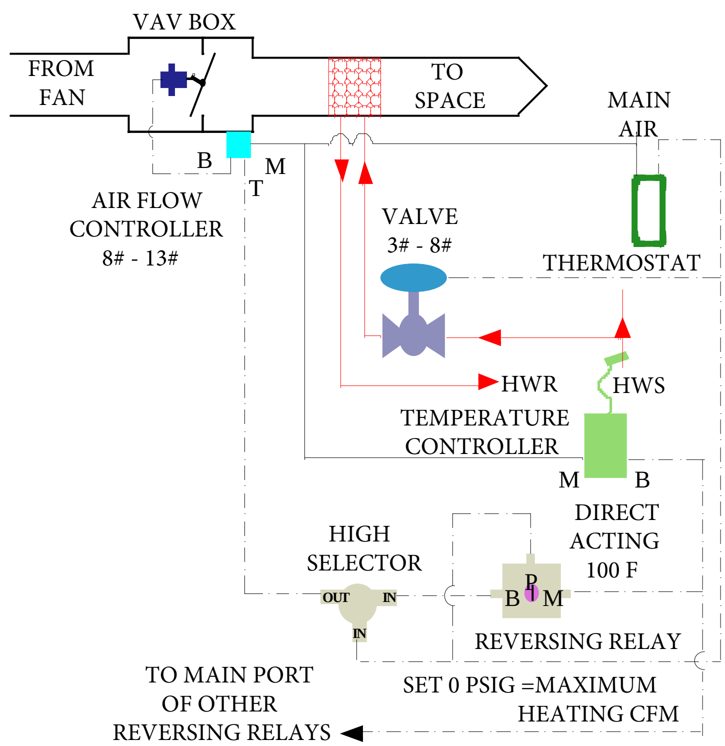
VAV/REHEAT ENERGY PERFORMANCE IMPROVEMENT



AS EXISTING

The airflow controller has minimum, as well as maximum, airflow settings. The maximum setting addresses the maximum airflow requirement at 100% design cooling load. The minimum setting normally addresses the required ventilation rate (example 25%) for the area being served, respecting air quality.

When a reheat coil is associated with a VAV box, the minimum setting relates to the quantity of air required carrying heat to the space (example 50%). This is normally much more than required for air quality maintenance. This arrangement requires that the heating be available all year, as the space will over-cool at the minimum setting of 50%, if no heat is available to warm the air.



SUGGESTED SOLUTION

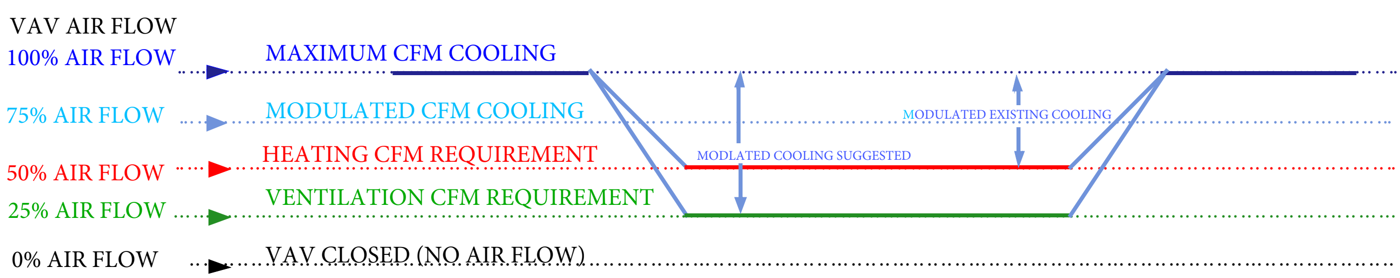
This arrangement allows the VAV box to be set at the ventilation rate (example 25%) as a minimum setting.

As the room cools, the thermostat will modulate the VAV box back to the 25% minimum ventilation position. As the room continues to cool, the thermostat opens the reheat valve. When heat is available and if the room continues to cool, the thermostat, through the reversing relay and high selector, increases the airflow through the VAV box from 25%, variable as needed, up to 50% airflow, delivering the required heat.

If no heat is available, as determined by the supply heating pipe temperature controller, the reversing relay will not receive main air, thus the VAV box is disallowed modulation from the minimum ventilation rate of 25% up to the maximum heating rate of 50% airflow.

This circuit provides the most effective energy use while still attaining the best comfort conditions in the space, both summer and winter, while allowing the boiler to be turned off during the summer.

NOTE:
The Kreuter RCC1502 reversing relay allows the required bias. Predetermine if the unit intended for use can achieve this.

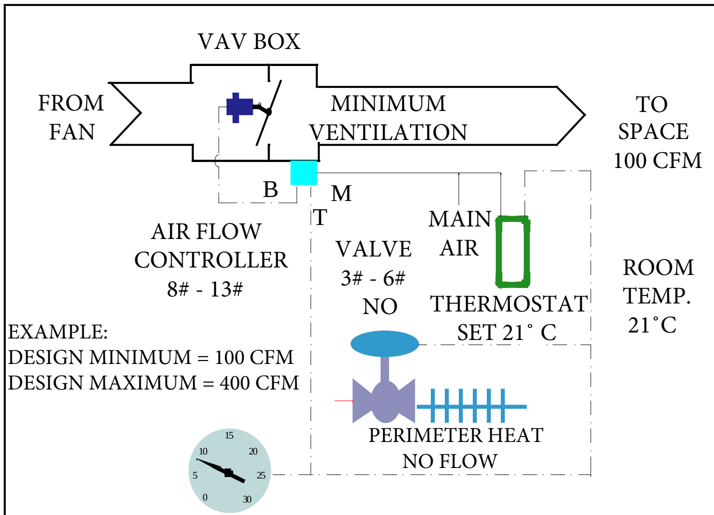


The existing design caused the VAV to over ventilate at a minimum airflow of 50%, when the actual ventilation airflow required was only 25%.

Heat, via the heating valve, had to be added compensating for the over ventilation in addition to the space's normal heat loss.

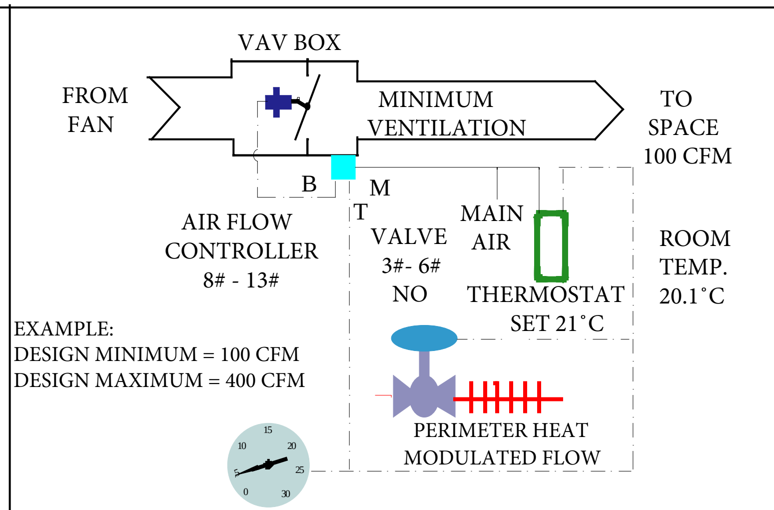
The suggested solution allows the VAV to modulate airflow down to the actual ventilation minimum rate of 25% if cooling is not required. If heating is required, the thermostat opens the heating valve fully: then increases the airflow from the minimum 25% to a maximum of 50%, as required, delivering the desired heat.

HIDDEN ENERGY LOSS: VAV WITH DEFECTIVE AIR FLOW CONROLLER OR DIAPHRAGM



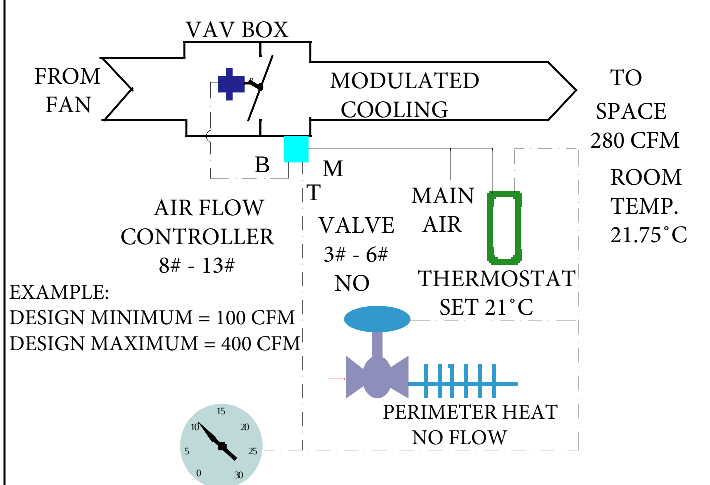
SCENARIO ONE (normal operation)

The thermostat is set at (21°C). The room is 21°C. The thermostat branch signal is at 8 PSIG, forcing the 3# to 6# heating valve closed and the 8# to 13# VAV box to only allow minimum ventilation.
There is no heating/cooling conflict.



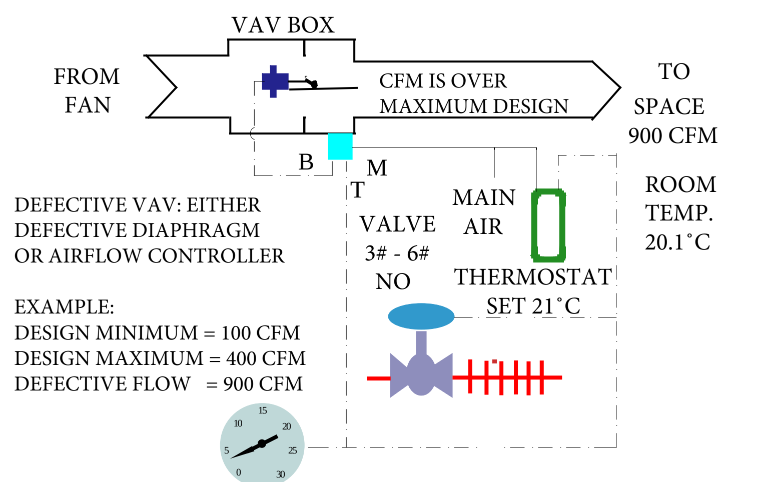
SCENARIO TWO (normal operation)

The thermostat is set at (21°C). The room temperature is 20.1°C. The thermostat branch signal is at 4 PSIG, modulating the 3# to 6# heating valve open and holding the 8# to 13# VAV box to only allow minimum ventilation.
There is no heating/cooling conflict.



SCENARIO THREE (normal operation)

The thermostat is set at (21°C). The room temperature is 21.75°C. The thermostat branch signal is at 11 PSIG, closing the 3# to 6# heating valve and modulating the 8# to 13# VAV box to 60% of maximum cooling airflow addressing normally expected heat gains. (solar, body, lights, etc.)
There is no heating/cooling system conflict.



SCENARIO FOUR (serious problem)

The thermostat is set at (21°C). The room temperature is 20.1.°C. The thermostat branch signal is at 4.0 PSIG, modulating the 3# to 6# heating valve, compensating for the defective VAV which is causing unwanted cooling to enter the occupied space.

There is heating/cooling system conflict. The heating is active, compensating for unwanted over-cooling via the defective VAV box.

Scenario four is a problem of unwanted cooling entering the building via a defective VAV. This typically occurs when the VAV airlow controller or actuator diaphragm is defective. The problem is not easily detected, as you can see comparing scenarios two and four; the room temperature is identical. The heating function hides the VAV's defect.

The heating input in scenario two compensates for perimeter heat loss: the heating input in scenario four compensates for unwanted cooling via the defective VAV box.

In scenario two no energy is wasted. In scenario four the waste can be severe, similar to opening your windows at home and turning on you furnace to compensate for the unwanted cooling..

SECTION SIX
(SOME CLIENT CALCULATED ENERGY REDUCTION IMPACTS)

PURPOSE OF SECTION

Present actual reductions calculated by the building owners.

THE BOARD OF EDUCATION FOR THE BOROUGH OF SCARBOROUGH

CONTROL MODIFICATIONS BY

A P S

SCHOOL	UTILITY	ANNUAL REDUCTION	1979 UTILITY COST	UTILITY SAVINGS **	$	IMPROVEMENT COST	PAYBACK YR.
MILITARY TRAIL JR. PS	GAS	31.1%	9,863	2315	4,073	4,072	1
	ELECTRIC	20.9%	11,217	1758			
JACK MINER PS	GAS	20.4%	14,338	2194	4204	3,287	0.8
	ELECTRIC	17.8%	15,059	2010			
ALEXMUIR JR. PS	GAS	29.5%	7,308	1617	3515	2,699	0.8
	ELECTRIC	24.6%	10,285	1898			
SILVER SPRINGS PS	GAS	37.8%	6,425	1822	3,973	1943	0.5
	ELECTRIC	36.6%	7,835	2151			
WENDELL STATTON SR. PS	GAS	57.6%	10,221	4415			
	OIL	19.5%	16,463	3210	10,114	5300	0.5
	ELECTRIC	22.6%	14,684	2489			
TIMBERBANK PS	GAS	17.1%	8,142	1044	2,784	2110	0.8
	ELECRIC	22.1%	10,499	1740			
WEST HILL C.I.	OIL	8.1%*	46,552	3771			
	ELECTRIC	10.9%*	51,974	4249	24,419	36,370	1.5
	GAS	51.8%*	42,211	16,399			
				TOTALS	53,082	55,781	1.05

- * - PART YEAR ONLY
- ** - GAS AND ELECTRIC @ 75% OIL @ 100%

JRM/sc June 18, 1981

LETTER FROM SCARBOROUGH BOARD ACCOMPANYING CHART ABOVE

Gentlemen:

During early 1979 control improvements were carried out by your firm on a number of our schools. These schools are listed on the attached schedule which indicates the savings that have been achieved.

For clarification purposes, it should be noted that:

a) no allowance has been made for the fact it was 4.7% colder in 1980 than in 1979.

b) The utility costs are 1979 actuals and no allowance has been made for escalation.

c) The majority of the savings are undoubtedly higher as the modifications were not in effect for the entire year.

d) We have assumed only 75% of the actual gas and electricity savings because of the sliding scale rate structures.

e) Approximately $17,000. Is included in the cost of improvements at West Hill Collegiate for other work that was performed at the time aimed primarily at improving poor environmental conditions.

It is almost needless to say that we are very pleased with the results and the manner in which they were carried out.

Yours very truly
J.R. Mazanik

NOTE: Actual Board information copied into Word.

THE DURHAM BOARD OF EDUCATION

CONTROL MODIFICATIONS BY

A P S

	ENERGY USE YEAR PRIOR TO CHANGE		ENERGY USE YEAR FOLLOWING CHANGE		ENERGY SAVINGS		ECONOMIC BENEFIT
SCHOOL	ELECTRIC	FUEL	ELECTRIC	FUEL	ELECTRIC	FUEL	SAVINGS
ADMIN.	575,040	767,690	469,120	597,192	105,920	170,498	$4,641.00
GENERAL VANIER	1,847,200	4,584,768	1,520,720	4,127,945	326,480	456,823	$13,480.00
HENRY ST. HS *	1,049,9162	2,772,563	961,263	2,671,423	88,653	101,138	$3,385.00
O'NEILL CVI	949,025	2,294,845	886,925	2,236,678	62,100	58,167	$2,219.00
DUNBARTON HS	1,686,500	3,376,841	1,586,086	3,001,606	100,414	375,235	$6,962.00
G.L. ROBERTS HS	1,129,996	2,433,187	1,001,192	1,338,622	128,804	896,565	$13,914.00
SOUTHWOOD PARK PS	820,080	NO FUEL	554,640	NO FUEL	265,440	NO FUEL	$6,503.00

TOTAL SAVINGS	$51,104.00
TOTAL COST	$42,340.00
PAYBACK 8 MONTHS, 1 WEEK	

- * - CHANGES WERE IN EFFECT FOR 3 MONTHS.
- ALL CONSUMPTION VALUES ARE IN KILOWATT-HOURS.
- DOLLAR VALUES ARE 1981 FUEL COSTS

DM: mm
1982 01 14

Notes:

- Actual chart of School Board's data copied into Word.
- Administration and Henry St. are multizone systems.
- Southwood Park PS is electric baseboard and electric reheat.
- General Vanier SS, O'Neill CVI and Dunbarton have a variety of HVAC fan systems, but no multizones.
- G.L. Roberts HS was VAV with perimeter heating only controlled via indoor/outdoor reset from one main valve.

ACTUAL GAS CONSUMPTION AMOUNTS FOR THE ETOBICOKE EDUCATION CENTRE ILLUSTRATING THE IMPACT OF EVAPORATIVE COOLING AND LOAD ANALYZING TECHNIQUES

READING DATE	GAS CONSUMPTION (X 100)
AUG. 29/75	1,324.8
SEPT. 30/75	2,069.8
OCT. 30/75	1,965.1
DEC. 01/75	1,776.8
DEC. 30/75	1,930.1
JAN. 30/76	2,806.9
FEB. 27/76	2,308.1
MAR. 30/76	2,624.8
APRIL 29/76	1,749.6
MAY 28/76	1,676.1
JUNE 29/76	1,053.2
JULY 28/76	924.6
JAN. 26/77	9,168.4
FEB. 21/77	1,276.8
MAR. 22/77	1,192.7
APRIL 21/77	613.6
MAY 20/77	522.5
JUNE 21/77	609.2
JULY 21/77	683.7
AUG. 22/77	1,066.6
SEPT. 22/77	1,156.2
OCT. 21/77	1,328.5
NOV. 22/77	1,224.2
DEC. 21/77	815.1
JAN. 20/78	927.5
FEB. 20/78	1,078.8
MAR. 22/78	920.5
APRIL 21/78	771.2
MAY 19/78	526.8

WHEN COMPARING THE GAS CONSUMPTION FOR THESE TWO SETS OF MONTHS, ONE FINDS JUNE AND JULY REQUIRED 13.5% MORE GAS BEFORE THE CHANGES, THAN DECEMBER AND JANUARY REQUIRED AFTER THE CHANGES.

INITIAL CONSERVATION DESIGN EMPLOYING EVAPORATIVE COOLING EFFECT OF THE HUMIDITY WATER SPRAYS. OTHER TECHNIQUES WERE IMPLEMENTED SIMULTANEOUSLY. [JAN. 26/77 – JULY 21/77]

THE WATER SPRAYS WERE REPLACED WITH STEAM HUMIDIFIERS CAUSING THE COOLING EFFECT TO BE LOST. [AUG. 22/77 – NOV. 22/77]

AN ALTERNATE CONSERVATION CONTROL TECHNIQUE WAS DESIGNED AND INSTALLED. LOAD ANALYSING FROM THE WARMEST SAMPLE THERMOSTAT RE-ESTABLISHED THE CONSUMPTION REDUCTION. [DEC. 21/77 – MAY 19/78]

52.6% REDUCTION WITH EVAPORATIVE COOLING TECHNIQUE.

60% REDUCTION WITH ALTERNATE TECHNIQUE.

Wycliffe Property Management Limited

January 8th, 2003

To Whom It May Concern.

Re: Analysts of Pneumatic Services- Energy Savings Initiatives

Since its inception over 40 year ago Wycliffe Property Management Limited has grown to the point where e manage over two million square feet of commercial and industrial property in the greater Toronto area for various owners. As part of our portfolio, for the last eight (8) years we have acted as property managers of shops on Steeles and 404, a 280,000 square foot shopping Centre located in Thornhill, Ontario.

Approximately 6 years ago upon a recommendation received from Carrier Canada we met with Dave Strain, owner of analysts of Pneumatic Services (APS) to evaluate the Shops Centre's H.V.A.C. system with the goal of completely automating the control pneumatics to bring in "free" outside air when temperature fell below 10C.

After upgrading the pneumatic system and installing safety features to prevent freezing of the coils etc., we were able to annually shut down our 347 ton centrifugal chiller for 5 ½ months, relying solely on the "free" air provided, this resulted in energy savings of over 30%.

In another mater A.P.S. found perimeter electric base board heaters were being used indiscriminately by Tenants while the cooling system was functioning. In light of this A.P.S. developed a morning "warmup" process which was initiated by using excess heat from our boilers to supply warm air through V.A.V. boxes in each Tenant space. The electric baseboard heaters were then put under mall control, only activating if temperatures dropped below set point. This additional energy savings iniative helped to bring our consumption cost savings down another 10% from previous usage.

The quality of the air has actually improved with these innovations. Dave Strain's professional expertise has benefited The Shops on Steeles and 404 tremendously over the years and we have found him completely dedicated in his ongoing quest for helping the environment through saving energy.

Yours truly,
WYCLIFFE PROPERTY MANAGEMENT LIMITED

Grant Brunne
Operations Manager

Mark Murphy
Property Manager

Head Office
1485 Whitehorse Road
Downsview, Ontario M3J 2Z2
Telephone (416) 635-2910
Fax (416) 635-7869

NOTE:
Orininal letter retyped for clarity, but exactly the same print.

Branch Office
2900 Steeles Av. East Ste 204
Thornhill, Ontario L3T 4X1
Telephone (905) 881-7422
Fax (905) 881-1224

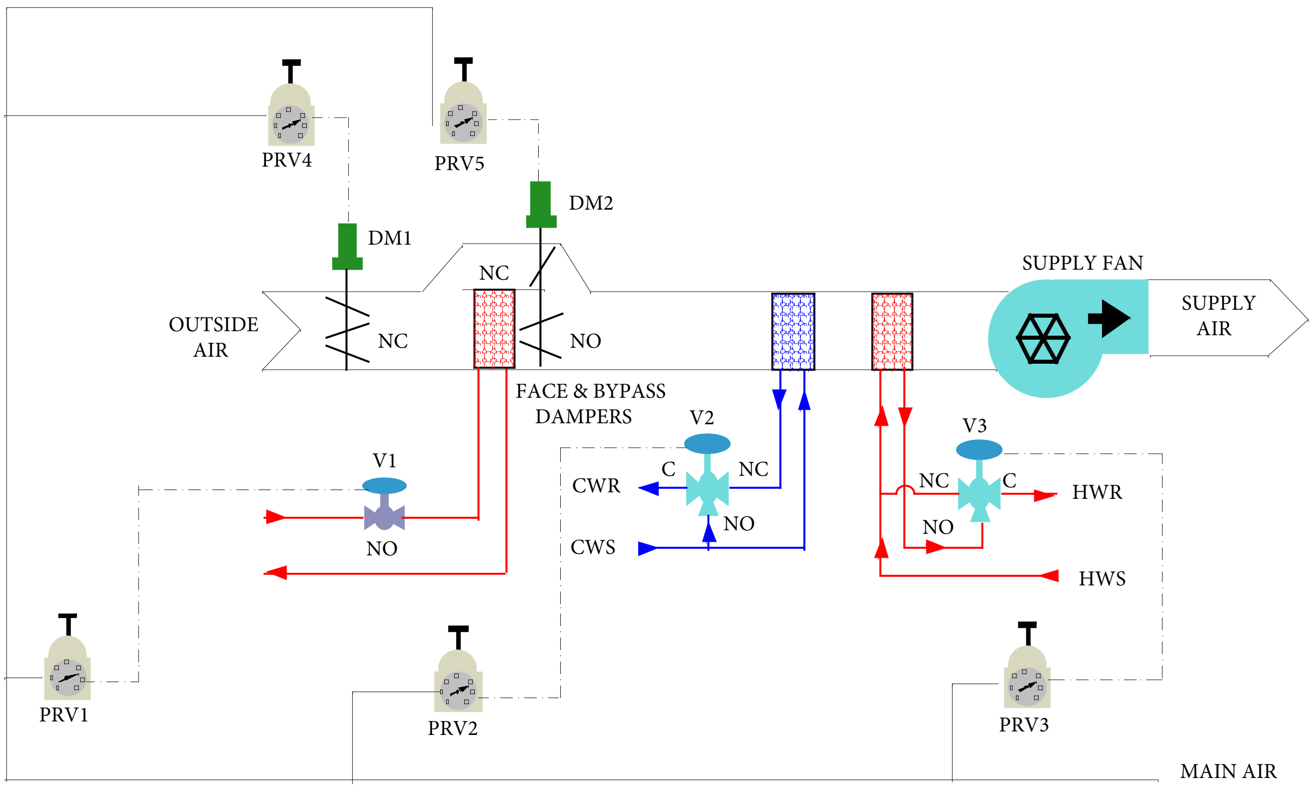
NOTES

SECTION SEVEN
(SOME TYPICAL FAN SYSTEMS)

PURPOSE OF SECTION

Provide a sense of the variety of fans systems faced in the field. There are many more combinations than presented here, but this should establish a good foundation for future challenges.

MANUAL ADJUSTMENT OF A 100% OUTSIDE AIR FAN



What are the considerations for the illustrated system regarding safety of the mechanical components?

What are the considerations for the illustrated system regarding energy use?

What is a typical application of the type of system illustrated?

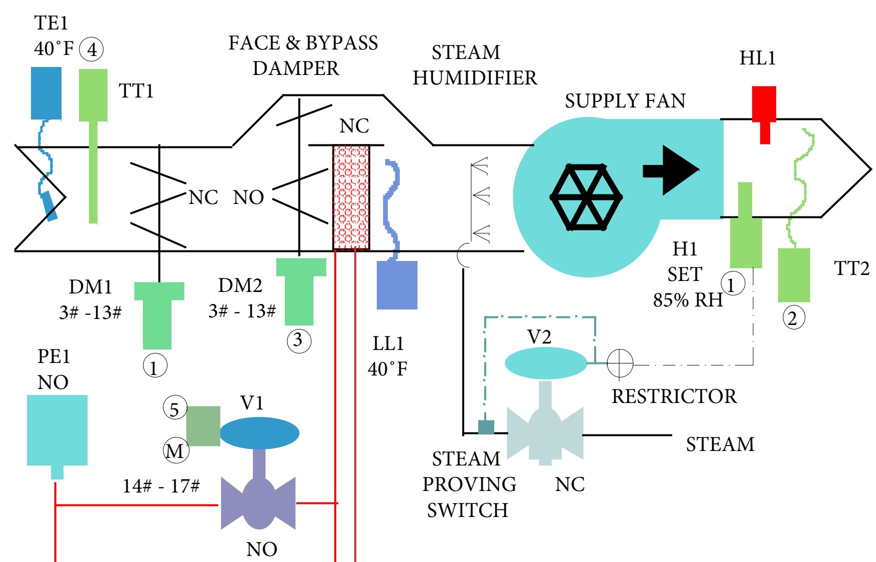
How would you manually operate this fan system to address ventilation, comfort and energy use requirements under these different conditions?

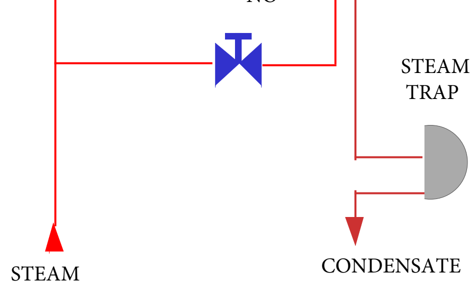
(1) At unoccupied times when the outside air (OSA) temperature is below 35°F?
(2) At occupied times when the OSA temperature is below 35°F?
(3) At unoccupied times when the OSA temperature is above 35°F?
(4) At occupied times when the OSA temperature is above 35°F?

100% OUTSIDE AIR FAN CONTROL EXAMPLE

SEQUENCE OF OPERATION

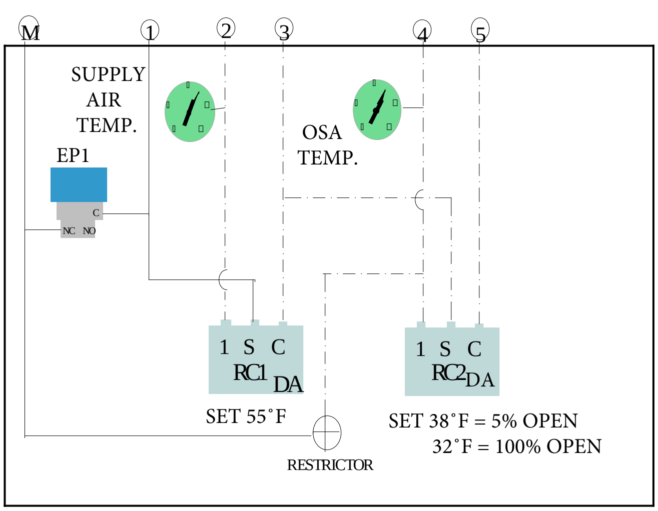
When the fan starts solenoid air valve (EP1) energizes and passes control air to drive damper motor (DM1) fully. EP1 also passes control air to humidity controller (H1). Receiver controller (RC1) senses the discharge air temperature via transmitter (TT2). RC1 sends its signal to modulate face & bypass damper motor (DM2). RC1 also sends its signal to feed receiver controller (RC2). RC2 senses the outside air temperature via transmitter (TT1). RC2 passes its signal to modulate control valve (V1).
H1 sends its signal to modulate humidifier valve (V2).
If low limit (LL1) senses a temperature below its set point on anyone foot of its element the fan will shut down. If pressure switch (PE1) senses a loss of steam pressure in cold weather as determined by controller (TE1) the fan will shut down.
The humidifier must be valved off for summer operation.

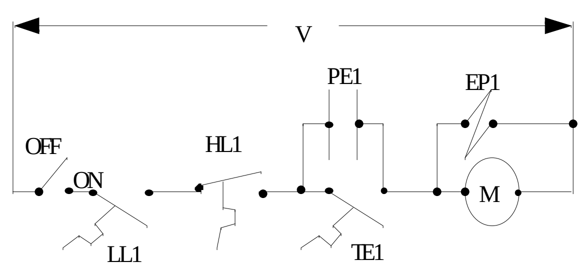




This is a system installed in a hospital, serving patient rooms; therefore, the fan runs continuously.

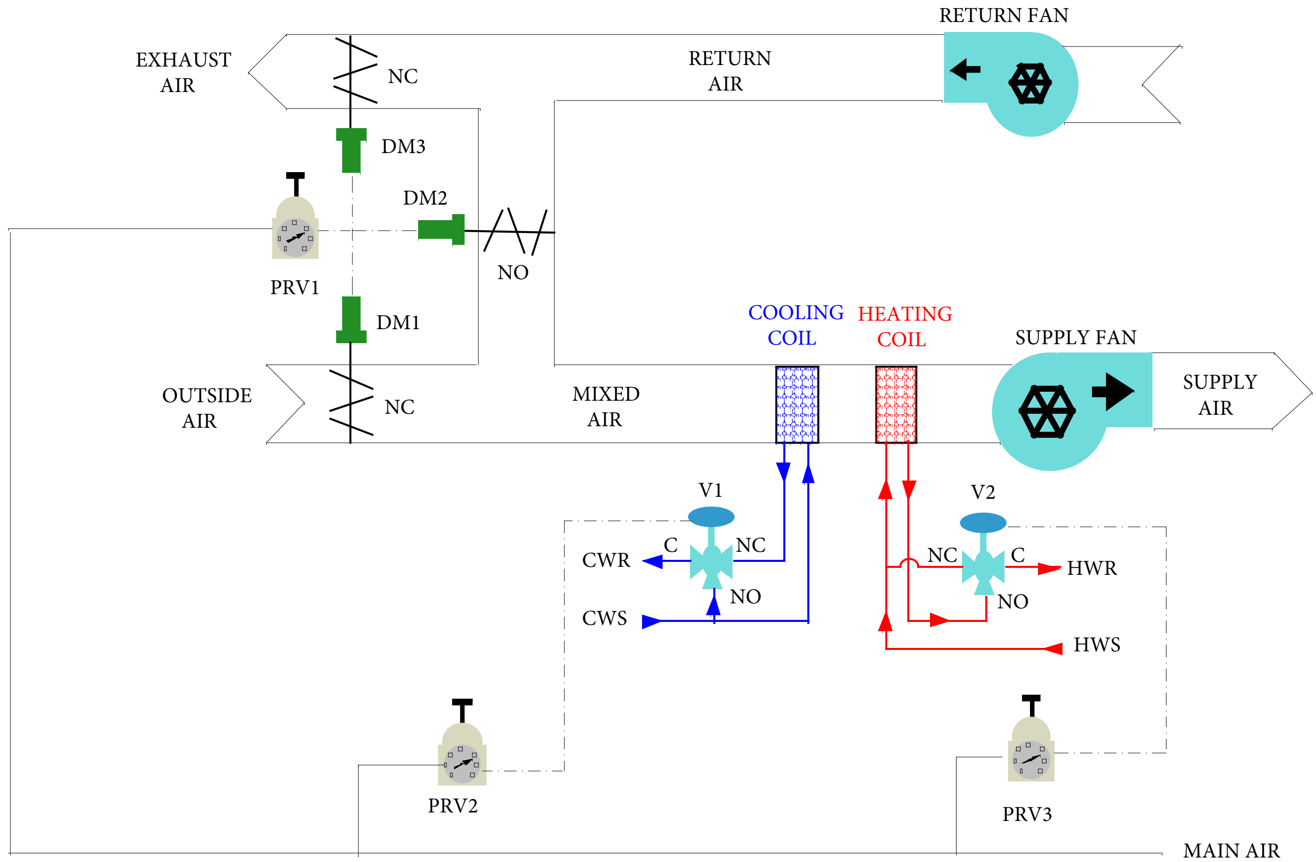
If the fan were to shut down at times, you may consider controlling the valves from a pneumatic low limit with automatic reset between the outside air damper and the preheat coil, preventing over-heating of the duct when the fan is down.





NOTE: RC2'S 100% OPEN MAY HAVE TO BE ALTERED TO SUIT SYSTEM.

<u>MANUAL ADJUSTMENT OF MIXED AIR FAN</u>



What are the considerations for the illustrated system regarding safety of the mechanical components?

What are the considerations for the illustrated system regarding energy use?

What is a typical application of the type of system illustrated?

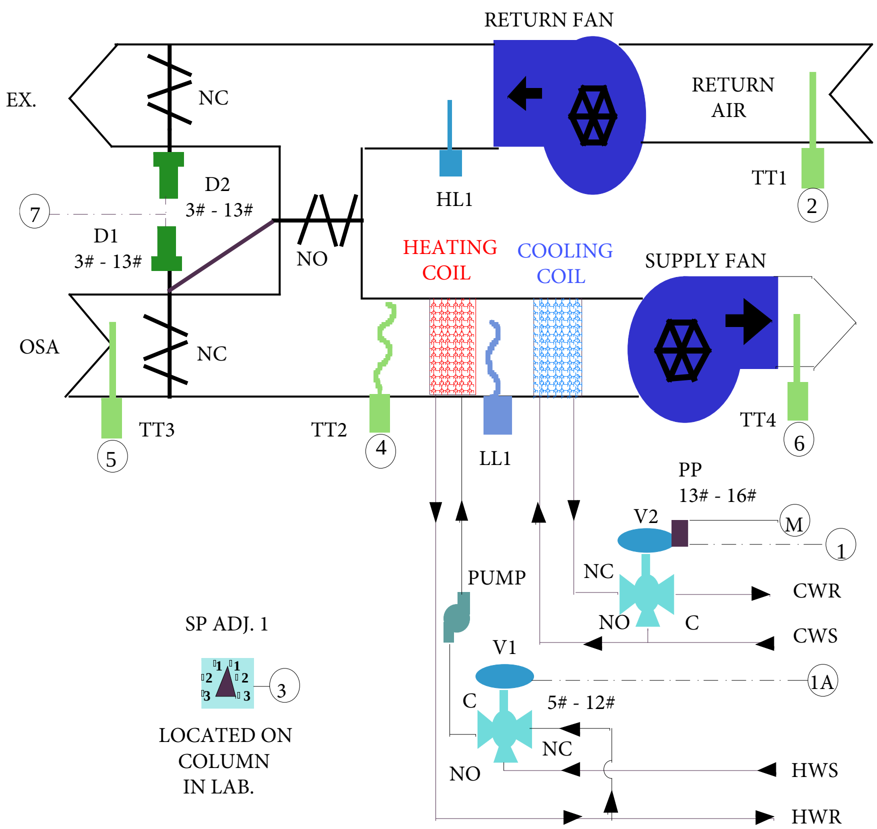
Where and what safeties should be installed?

Where is the mixed air plenum and what are the mixed air's three purposes?

How would you manually operate this fan system to address ventilation, comfort and energy use requirements under these different conditions?

(1) At unoccupied times when the outside air (OSA) temperature is below 35°F? (If the fan is the only source of heating? If adequate perimeter heating also exists?)
(2) At occupied times when the OSA temperature is below 35°F?(3) At unoccupied times when the OSA temperature is above 35°F?
(4) At occupied times when the OSA temperature is above 35°F?

MIXED AIR FAN EXAMPLE FROM SINGLE ZONE CASE STUDY PAGE 8.63



SEQUENCE OF OPERATION

-The fan runs continuously to suit laboratory operation.

-Solenoid air valve (EP1) passes control air to the fan controls when the fan runs.

-Receiver controller (RC1) senses the return air temperature via transmitter (TT1) and receives a set point demand from (SP ADJ 1). RC1 sends its signal to biasing relays (BR1), (BR2), (HS4 on drawing three), as well as to cooling valve (V2). The instrument arrangement causes the heating valve (V1) to close first, then the free cooling dampers (D1) and (D2) to be maximized and then the mechanical cooling valve (V2) to be used on a demand for cooling.

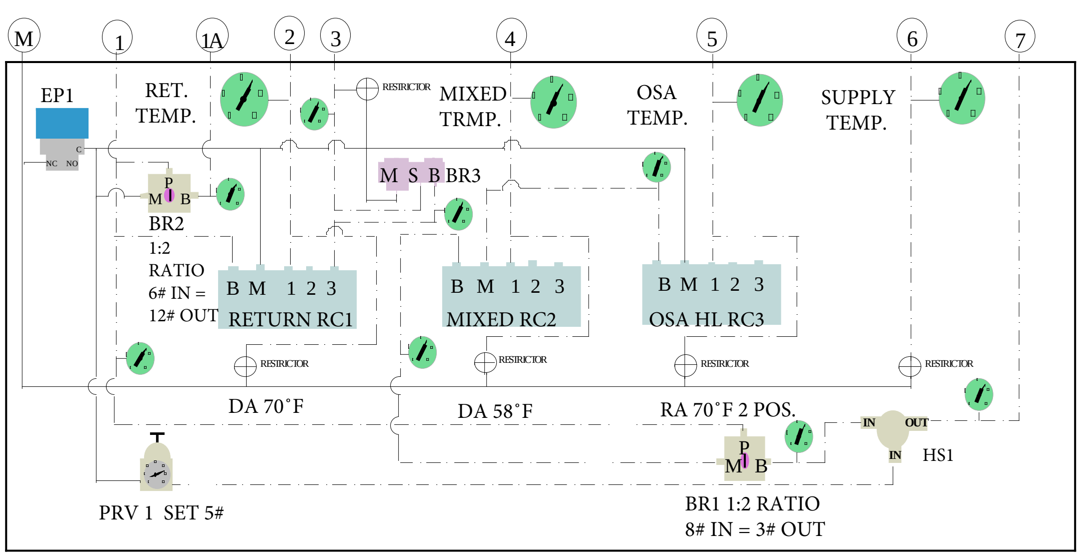
-BR1 sends its signal to high selector (HS1) which also receives a signal from regulator (PRV1) to assure the system of at least minimum ventilation.

-Receiver controller (RC3) senses the outside air temperature via transmitter (TT3). RC3 passes main air to receiver controller (RC2) only when free cooling is available based on the outside air temperature to economize on mechanical cooling. RC2 senses the mixed air temperature via transmitter (TT2). RC2 limits the mixed air temperature to a minimum of 58°F by limiting the main air to BR1.

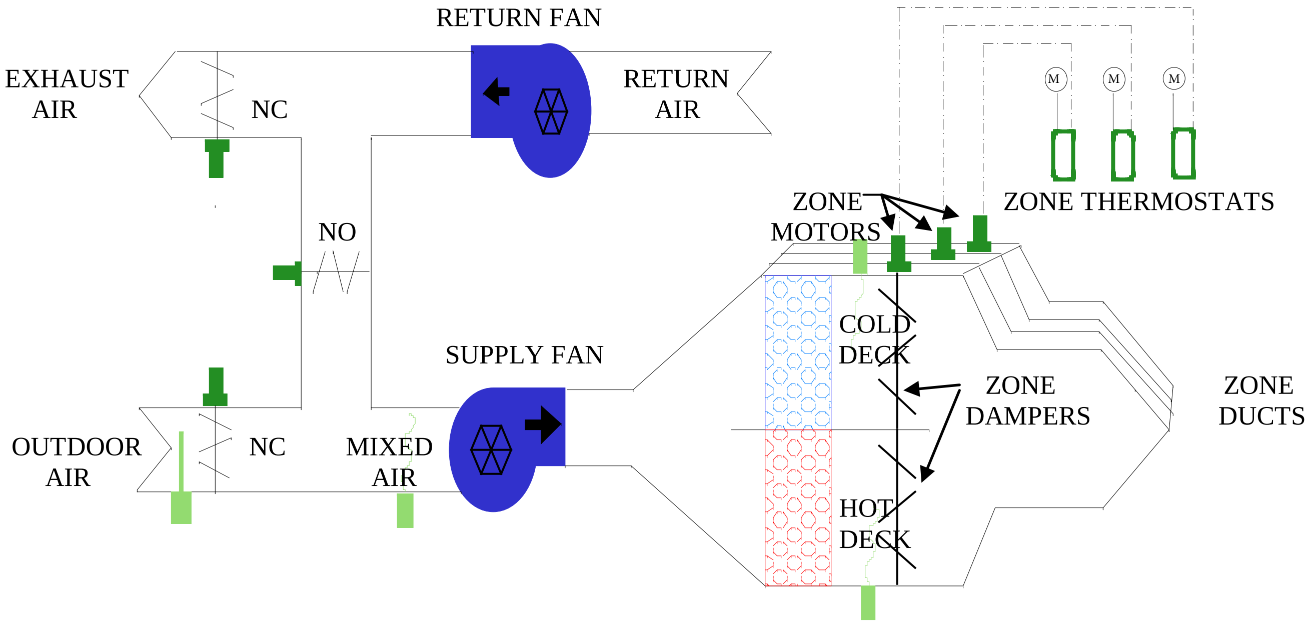
-Turndown ratio relay (BR3) is set to limit the degree of set point demand from the laboratory.

-Static pressure controller (SPC1) senses the differential pressure between the outside air and the lab. SPC! Sends its signal to limiting relay (LM1) which limits the vanes to not close beyond LM1's set point. LM1 sends its signal to modulate vane motor (D3).

-If low limit (LL1) senses a temperature below its set point on any one foot of its element the fan will shut down. If high limit (HL1) senses a temperature above its set point, the fan will shut down. Both HL1 and LL1 require resetting after they have tripped.



THREE ZONE MULTIZONE FAN SYSTEM



MULTIZONE FAN SYSTEM

The physical arrangement of most multizones is illustrated above.

Original design caused a constant mixed air temperature of 55°F. When the outside air exceeded 70°F the mixing dampers returned to their minimum ventilation setting.

The cold deck was controlled to 55°F constantly via the cooling coil for summer operation.

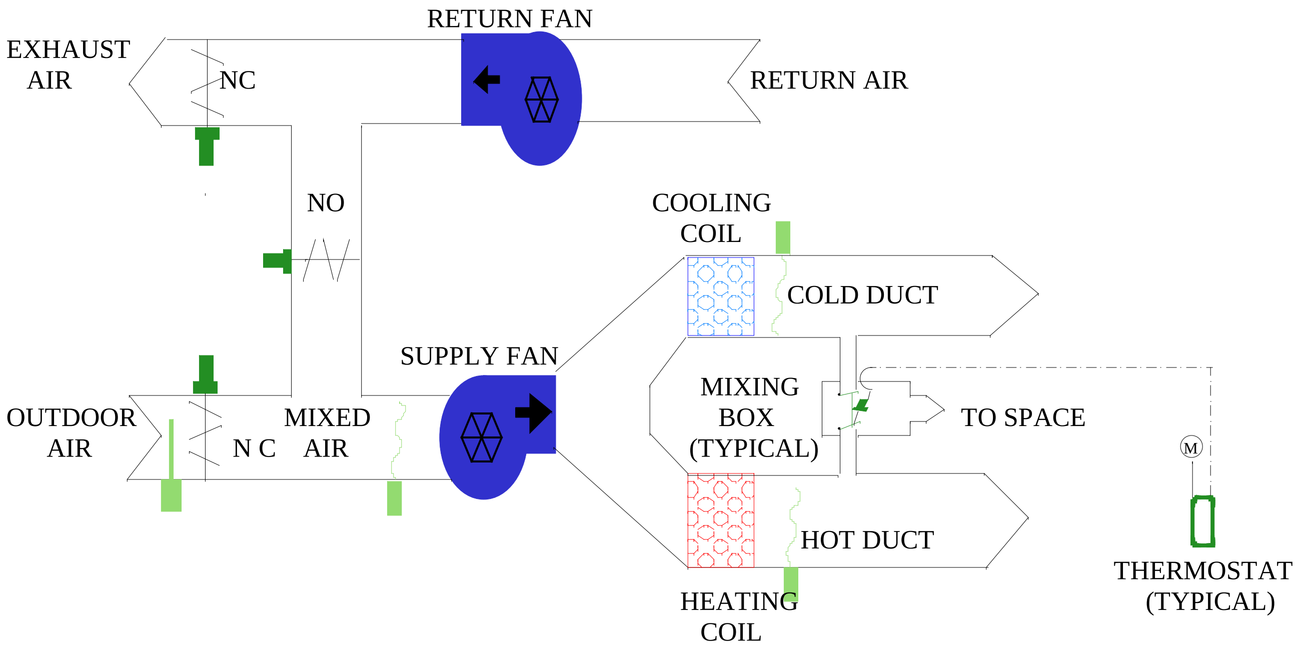
The hot deck temperature was scheduled based on outside air temperature variation. The colder the outside air temperature, the hotter the hot deck.

The thermostats blended air from the cold deck with air from the hot deck to achieve a supply air temperature satisfying the individual zone.

Modern logic knows the maximum demand for cooling and heating from all the zones. If all zone signals are within the range of the zone damper motors, the heating and cooling are kept at a minimum value. If any zone opens its cold deck 100% and closes its hot deck 100%, that zone can gradually cause the cold deck temperature to be lowered. Conversely, if any zone opens its hot deck 100% and closes its cold deck 100%, that zone can gradually raise the hot deck temperature. The demand for cold deck is limited to a minimum of 55°F and the demand for hot deck is limited by an outside air reset schedule similar to original design. The system receives at least minimum ventilation during occupied mode.

The gas sections of Wendell Statton PS and West Hill C.I. in Scarborough and the Administration Office and Henry Street HS in Durham are all multizones. The saving achieved with modern logic are illustrated later in this book.

DUAL DUCT FAN SYSTEM



The physical arrangement of most dual duct systems is illustrated above.

Original design caused a constant mixed air temperature of 55°F. When the outside air exceeded 70°F the mixing dampers returned to their minimum ventilation setting.

The cold duct was controlled to 55°F constantly via the cooling coil for summer operation.

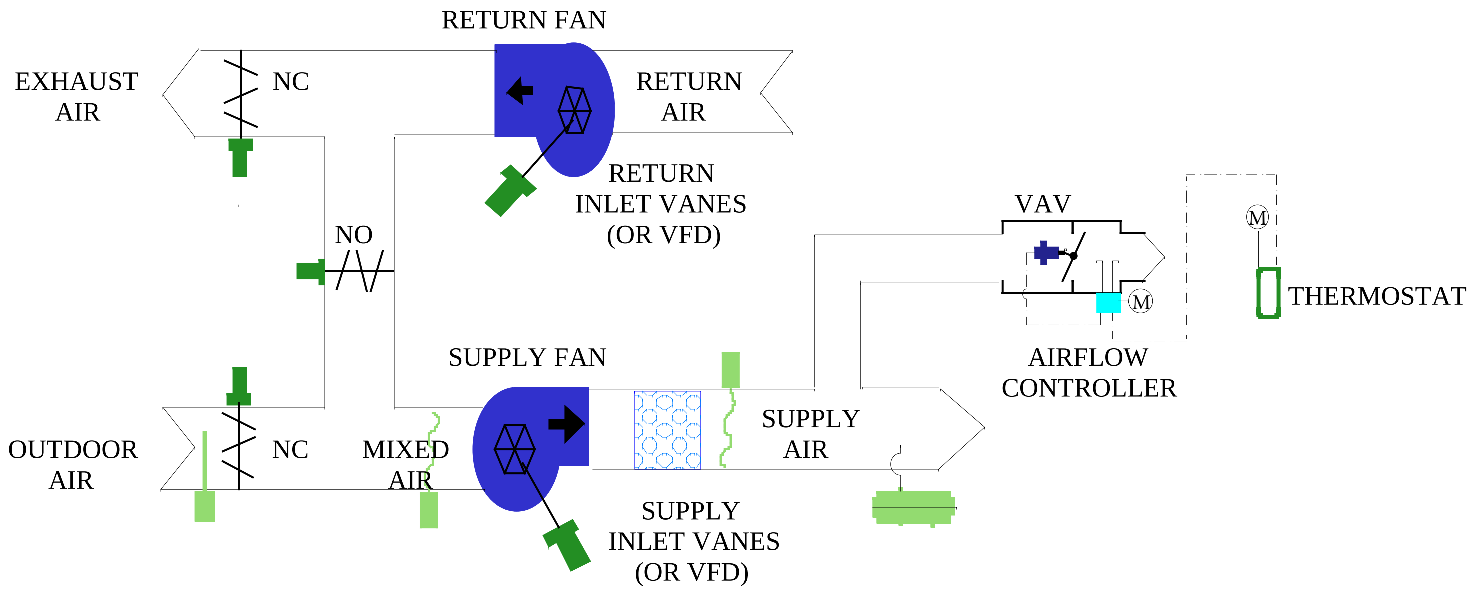
The hot duct temperature was scheduled based on outside air temperature variation. The colder the outside air temperature becomes, the hotter the hot duct becomes.

The thermostats blended air from the cold duct with air from the hot duct in a mixing box, usually in the ceiling of the room with the thermostat, to achieve a supply air temperature satisfying the individual zone.

Modern logic knows the maximum demand for cooling and heating from all the zones. If all zone signals are within the range of the mixing boxes' damper motors, the heating and cooling are kept at a minimum value. If any zone opens its cold duct 100% and closes its hot duct 100%, that zone can gradually cause the cold duct temperature to be lowered. Conversely, if any zone opens its hot duct 100% and closes its cold duct 100%, that zone can gradually raise the hot duct temperature. The demand for cold duct is limited to a minimum of 55°F and the demand for hot duct is limited by an outside air reset schedule similar to original design. The system receives at least minimum ventilation during occupied mode.

The set up and logic for a dual duct system is similar to that for a multizone.

VAV SYSTEM PRESSURE INDEPENDENT



VAV (VARIABLE AIR VOLUME) SYSTEM

This type of system allows each area's thermostat to vary the amount of cold air entering the occupied space, addressing the varying cooling loads.

The VAV box illustrated has an air flow controller that senses the velocity pressure exerted by the airflow. The airflow controller sets the maximum and minimum amount of air allowed through the VAV box. The thermostat is allowed to vary the airflow between these two limits. This arrangement is referred to as pressure independent, as the airflow controller tends to correct for varying upstream static pressure values.

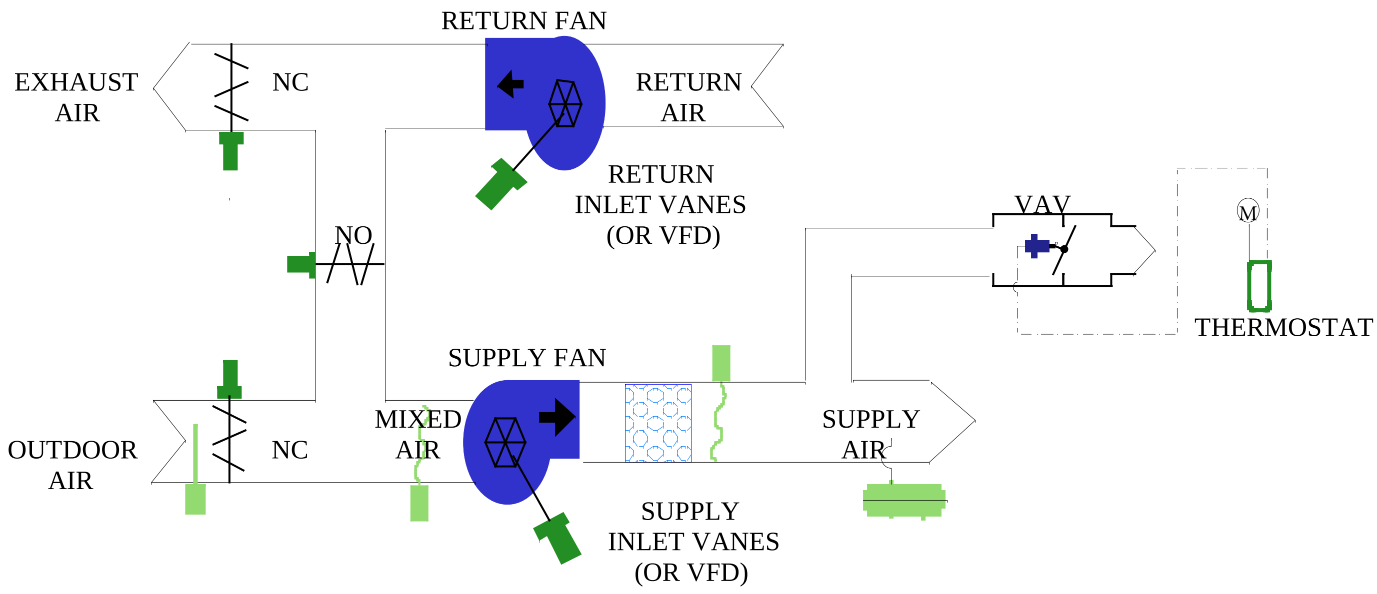
The airflow controllers are proportional with no integral reset; therefore, significant shift in control point is experienced if the static pressure variation is excessive.

The main duct pressure is controlled based on input from a pressure sensor located normally two-thirds of the way down the duct, based on air volume. The controller varies the position of the inlet vanes on the supply and return fans or controls variable frequency drives for these fans, maintaining constant duct static pressure.

Most systems have a diversity factor. This factor is the percentage difference between the total maximum volume of the VAV boxes and the maximum CFM of the supply fan. This is based on the fact that the sun can not shine on all sides of the building at one time; therefore, full cooling will not be required every where, at any one time.

The minimum ventilation requirement of a building is usually a fixed CFM value and the setting is normally a percentage of the supply fan's volume. The fact that the supply fan volume varies, requires that the relative percentage of air for minimum ventilation must vary also. This fact is rarely considered in setting the minimum ventilation logic.

VAV SYSTEM PRESSURE DEPENDENT



VAV (VARIABLE AIR VOLUME) SYSTEM

This type of system allows each area's thermostat to vary the amount of cold air entering the occupied space, addressing the varying cooling loads.

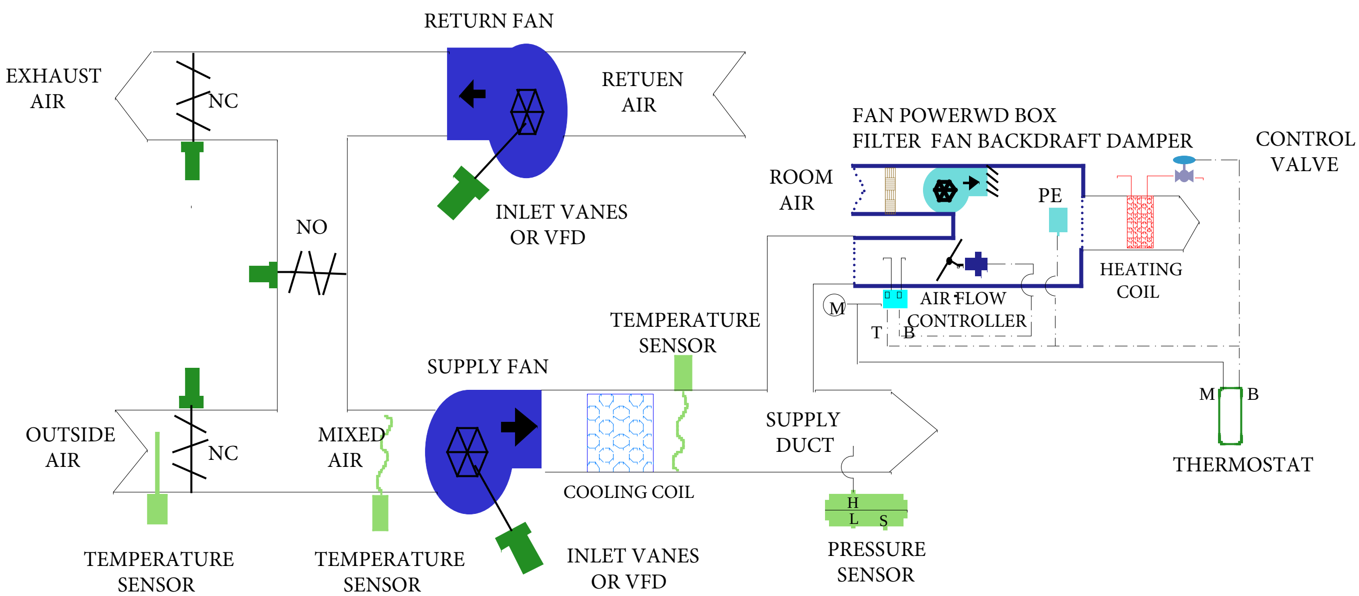
The VAV box illustrated does not have an air flow controller. This arrangement is referred to as pressure dependent. If the static pressure in the main duct varies, the amount of air entering the occupied space will vary even with the VAV box damper fixed at one degree of opening. Precise control of the main duct pressure is critical in these systems.

The main duct pressure is controlled based on input from a pressure sensor located normally two-thirds of the way down the duct, based on air volume. The controller varies the position of the inlet vanes on the supply and return fans or controls variable frequency drives for these fans, maintaining constant duct static pressure.

Most systems have a diversity factor. This factor is the percentage difference between the total maximum volume of the VAV boxes and the maximum CFM of the supply fan. This is based on the fact that the sun can not shine on all sides of the building at one time; therefore, full cooling will not be required every where, at any one time.

The minimum ventilation requirement of a building is usually a fixed CFM value and the setting is normally a percentage of the supply fan's volume. The fact that the supply fan volume varies, requires that the relative percentage of air for minimum ventilation must vary also. This fact is rarely considered in setting the minimum ventilation logic.

<u>PARALLEL FAN POWERED BOX</u>



The fan powered box illustrated has an air flow controller that senses the velocity pressure exerted by the primary airflow. The airflow controller sets the maximum and minimum amount of air allowed through the fan powered box on cooling mode. The thermostat is allowed to vary the primary airflow between these two limits, modulate the heating valve and cycle the fan.

On a drop in temperature, the thermostat reduces the cooling primary air from maximum to minimum airflow. If the temperature drops further the control valve will open to the heating coil and then closes the pressure switch, increasing the airflow through the coil on heating mode. A backdraft damper prevents primary air exiting to the ceiling space.

The primary supply fan may be shut down at night and the primary damper in the fan powered boxes close. The fan powered boxes may be cycled at the night set back temperature.

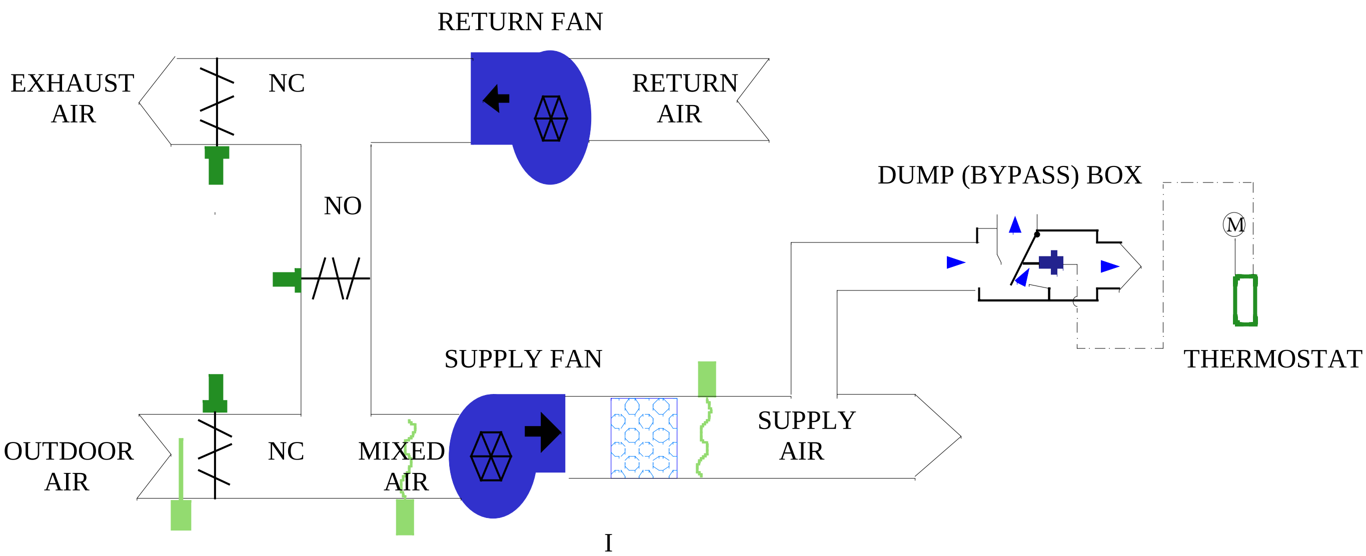
The main duct pressure is controlled based on input from a pressure sensor located normally two-thirds of the way down the duct, based on air volume. The controller varies the position of the inlet vanes on the supply and return fans or controls variable frequency drives for these fans, maintaining the required duct static pressure. On some systems the supply fan's inlet vanes are controlled based on supply duct pressure and the return fan's inlet vanes are controlled from building pressure. More advanced systems control the supply duct pressure just high enough to satisfy the VAV terminal with the greatest cooling requirement.

Most VAV systems have a diversity factor. This factor is the percentage difference between the total maximum volume of the VAV boxes and the maximum CFM of the supply fan. This is based on the fact that the sun cannot shine on all sides of the building at one time; therefore, full cooling should not be required everywhere, at any one time.

When the air volume required exceeds the maximum volume capability of the supply fan, the duct static will drop below the design value. Some factors causing this are supply duct leaks, VAV boxes allowing more than their original design, more VAV boxes added to the system, cooling load exceeding original design or the supply fan providing less volume than its original design.

The minimum ventilation requirement of a building is usually a fixed CFM value and the setting is normally a percentage of the supply fan's volume. The fact that the supply fan volume varies, requires that the relative percentage of air for minimum ventilation must also vary. This fact is rarely considered in setting the minimum ventilation logic.

DUMP (BYPASS) BOX SYSTEM CONSTANT VOLUME



DUMP BOX (VARIABLE AIR VOLUME) SYSTEM

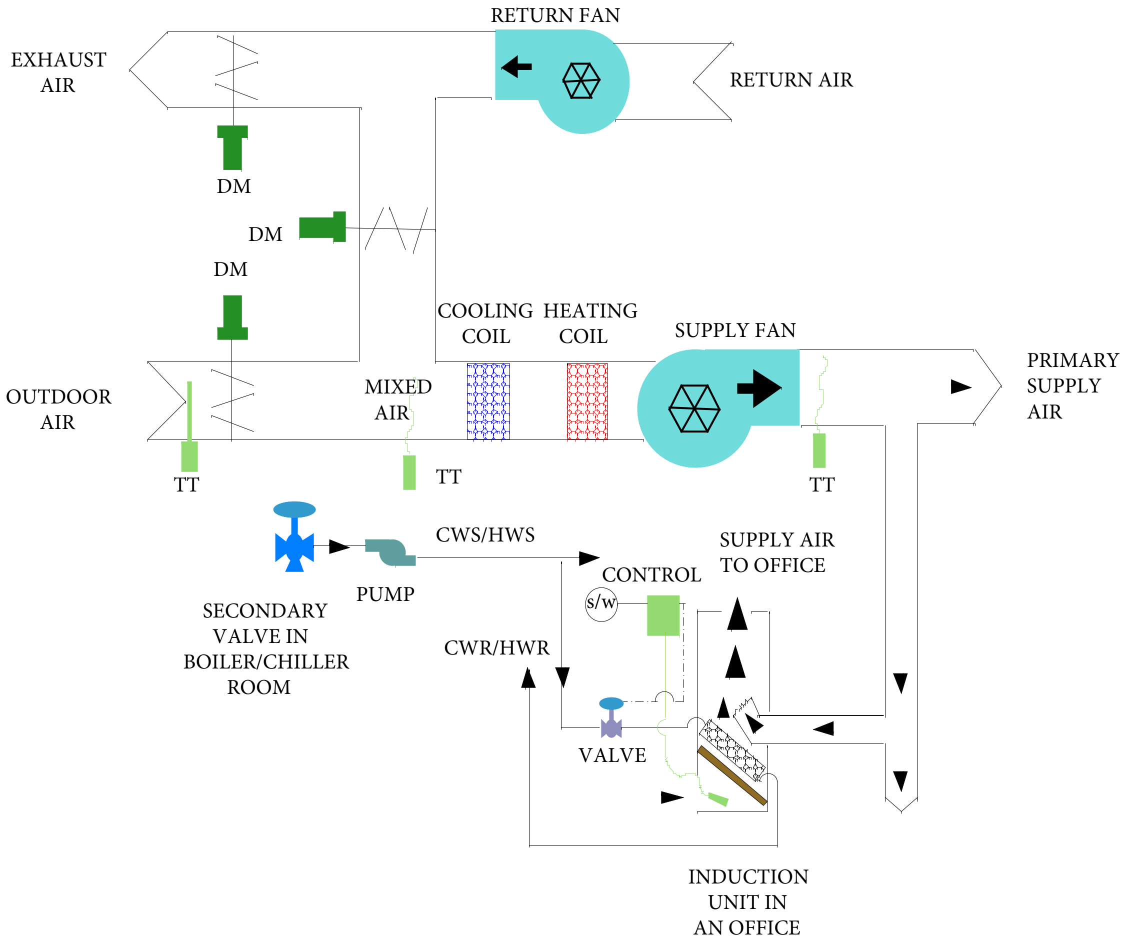
This type of system allows each area's thermostat to vary the amount of cold air entering the occupied space, addressing the varying cooling loads.

The dump box (bypass box) illustrated, always (in theory), consumes the same CFM from the main supply duct. The air is bypassed to the return air duct, forced into the occupied space or split between the two destinations. At a practical level the total CFM is different when the box is positioned to full bypass or full air flow to the occupied space relative to mixing positions between these two points.

The main fans do not require a means of varying their CFM as the system requirement in the main supply and return ducts does not vary.

As with all the fan systems, the air balancing should be performed by a qualified and experienced Air Balancing Technician.

INDUCTION UNIT SYSTEM



Induction unit systems are designed to heat and cool the perimeter area of a building up to about fifteen feet from the windows.

The primary air fan forces air out of the jets in the induction unit, causing an induced flow of return air from the room, through the filter and coil. The room controller senses the return air at this point. The total volume discharging from the grills of the induction unit is the sum of the primary airflow to the unit and the induced airflow.

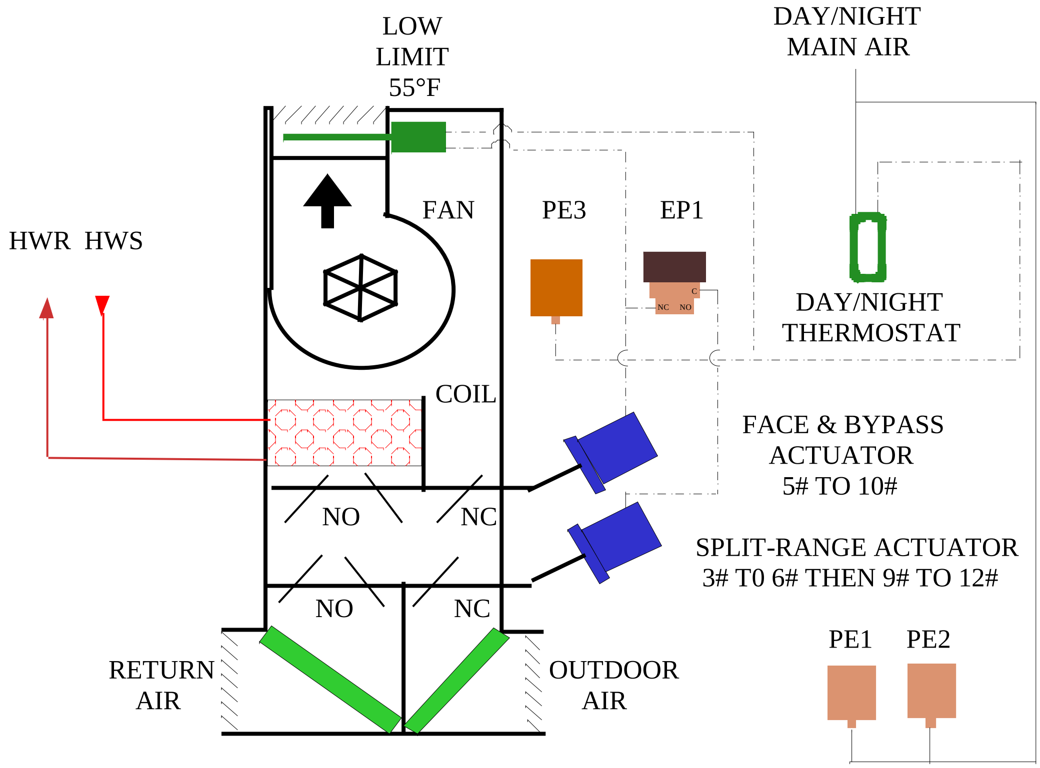
The primary air is cold during winter mode and heated water is provided via the secondary loop to the control valve associated with each induction unit. The controller has the option of heating the return air or allowing it to pass unheated through the coil to be mixed with the cold primary air.

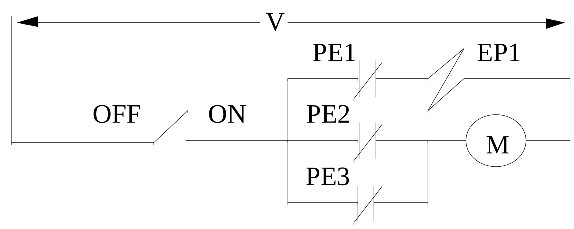
The primary air is heated during summer mode, based on reset from the outside air temperature. The induction unit receives chilled water via the secondary water loop. Usually the water temperature is controlled to minimize condensation at the room level.

Some systems work with chilled secondary water or heated secondary water all year with the heating or cooling source respectively being the primary air. These units do not require reversal of the control action regarding the induction unit controllers.

These systems have the potential for great inefficiency.

UNITVENTILATOR FACE & BYPASS





A unitventilator is a fan system dedicated to one area. They normally run continuously during occupied mode providing ventilation, heating and cooling.

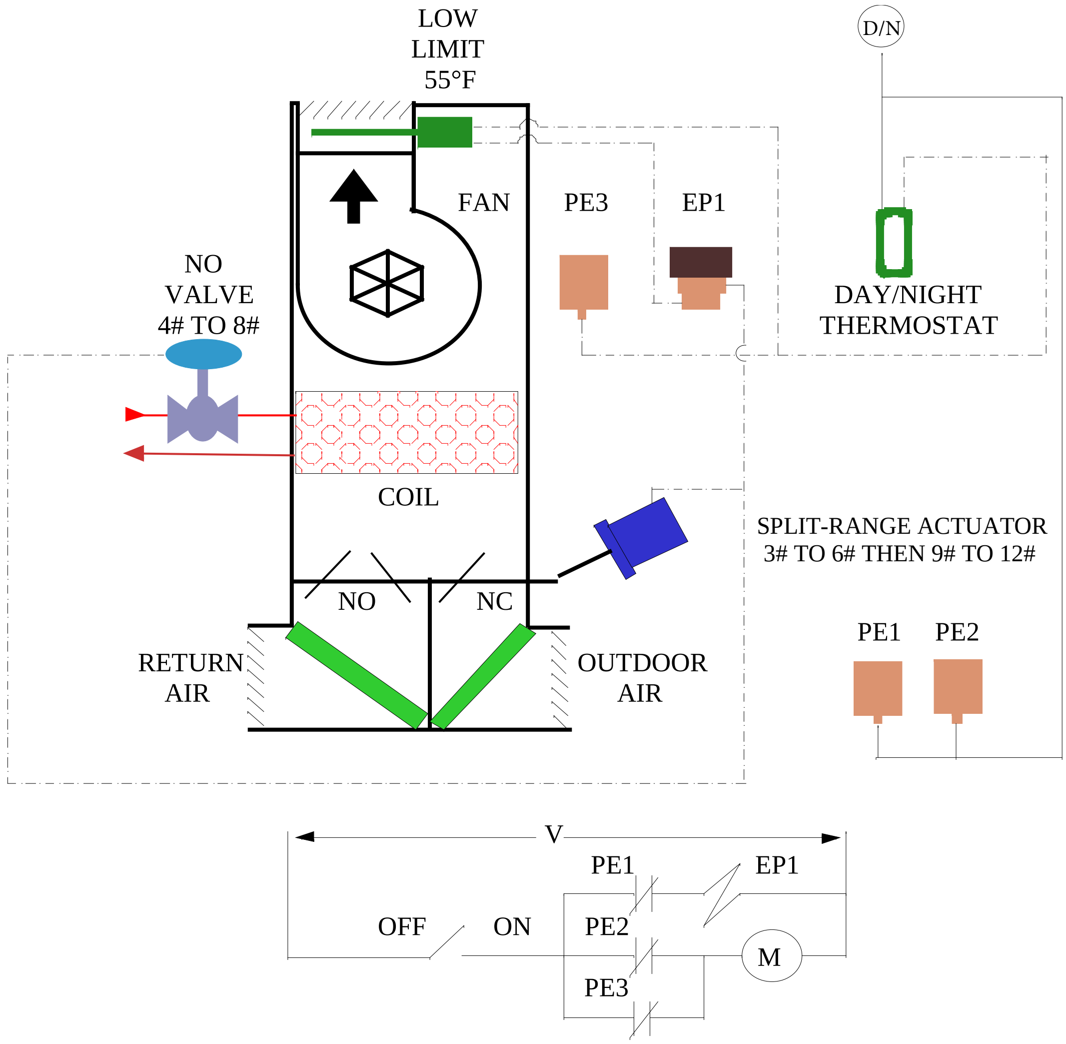
On day mode PE1 and PE2 close their contacts causing the fan to run, allowing a signal through the solenoid valve to modulate the fresh air damper and the heating control valve. The signal from the thermostat goes through the low limit to both the mixing damper and the control valve. As the thermostat signal starts to rise, first the split-range, fresh air motor drives from 3# to 6#, opening to the minimum ventilation requirement; then the heating valve drives from full heating to no heating. If the thermostat requires cooling the fresh air motor will restart to drive from 9# to 12# attaining 100% outside air. The low limit will over-ride the thermostat's demand for cooling if the supply air temperature decreases to the 55°F setting of the low limit.

On night mode PE1 and PE2 open their contacts and the unit shuts down. The thermostat changes to the night set point. When the room cools to the night setting the unit is duty cycled via PE3. The fresh air damper is completely closed and the heating valve is open on night mode.

The thermostat may have an indexing switch, allowing manual day mode operation of the room at times that are normally night mode.

There are many control arrangements used on unitventilators; however, each should address safety, comfort and energy requirements.

UNITVENTILATOR CONTROL VALVE



A unitventilator is a fan system dedicated to one area. They normally run continuously during occupied mode providing ventilation, heating and cooling.

On day mode PE1 and PE2 close their contacts causing the fan to run, allowing a signal through the solenoid valve to modulate the fresh air damper and the heating control valve. The signal from the thermostat goes through the low limit to both the mixing damper and the control valve. As the thermostat signal starts to rise, first the split-range, fresh air motor drives from 3# to 6#, opening to the minimum ventilation requirement; then the heating valve drives from full heating to no heating. If the thermostat requires cooling the fresh air motor will restart to drive from 9# to 12# attaining 100% outside air. The low limit will over-ride the thermostat's demand for cooling if the supply air temperature decreases to the 55°F setting of the low limit.

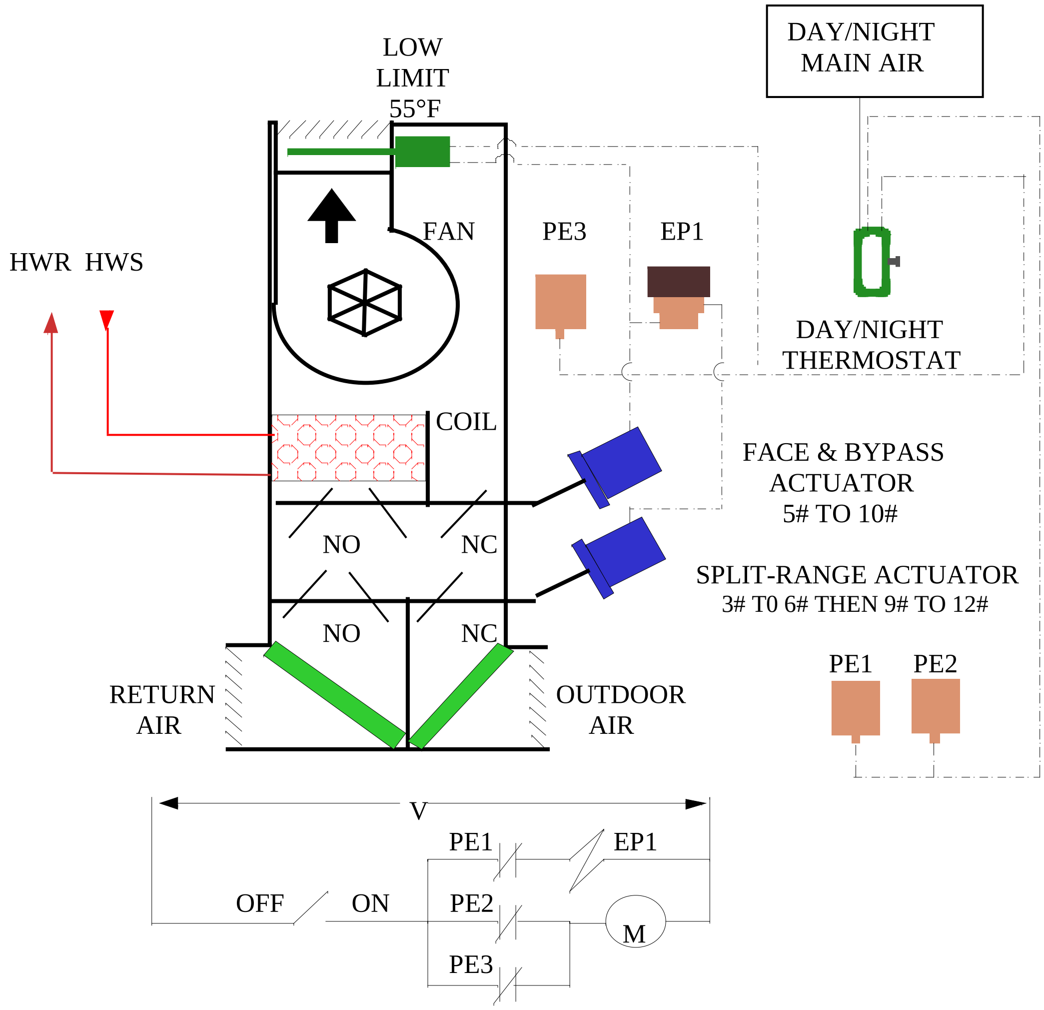
On night mode PE1 and PE2 open their contacts and the unit shuts down. The thermostat changes to the night set point. When the room cools to the night setting the unit is duty cycled via PE3. The fresh air damper is completely closed and the heating valve is open on night mode.

The thermostat may have an indexing switch, allowing manual day mode operation of the room at times that are normally night mode.

There are many control arrangements used on unitventilators; however, each should address safety, comfort and energy requirements.

UNITVENTILATOR FACE & BYPASS

THREE PIPE THERMOSTAT



A unitventilator is a fan system dedicated to one area. They normally run continuously during occupied mode providing ventilation, heating and cooling.

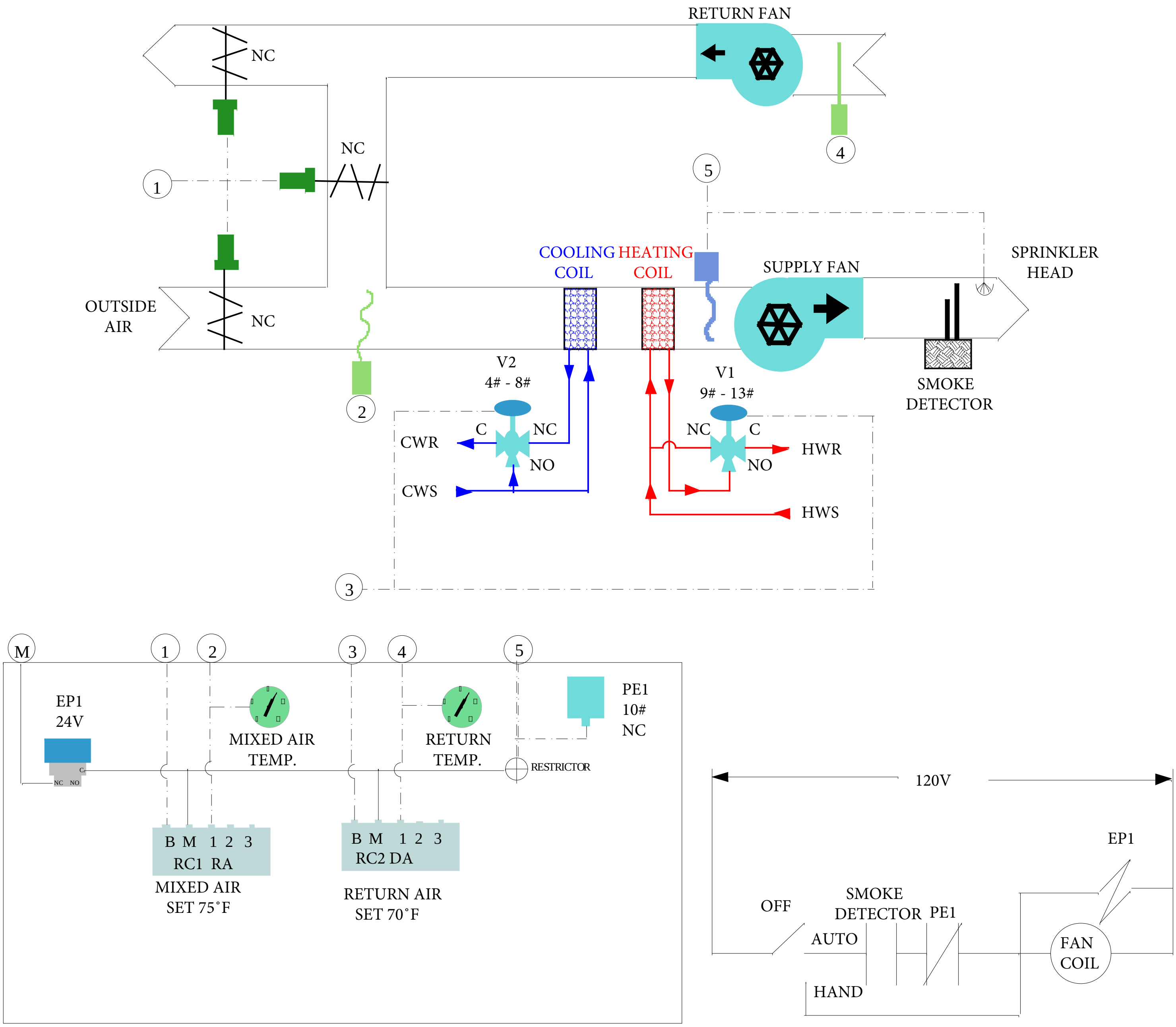
On day mode PE1 and PE2 close their contacts causing the fan to run, allowing a signal through the solenoid valve to modulate the fresh air damper and the heating control valve. The signal from the thermostat goes through the low limit to both the mixing damper and the control valve. As the thermostat signal starts to rise, first the split-range, fresh air motor drives from 3# to 6#, opening to the minimum ventilation requirement; then the heating valve drives from full heating to no heating. If the thermostat requires cooling the fresh air motor will restart to drive from 9# to 12# attaining 100% outside air. The low limit will over-ride the thermostat's demand for cooling if the supply air temperature decreases to the 55°F setting of the low limit.

On night mode PE1 and PE2 open their contacts and the unit shuts down. The thermostat changes to the night set point. When the room cools to the night setting the unit is duty cycled via PE3. The fresh air damper is completely closed and the heating valve is open on night mode.

The illustrated thermostat has an indexing switch, allowing manual day mode operation of the room at times that are normally night mode. The indexing switch automatically resets back to automatic the next morning when the main air pressure switches to the day value.

There are many control arrangements used on unitventilators; however, each should address safety, comfort and energy requirements.

FAN SYSTEM WITH MANY ERRORS



POTENTIAL ISSUES

-1- What are the errors in the mechanical arrangement?
-2- What are the errors with the pneumatic arrangement?
-3- What are the errors with the safety circuit?
-4- What are the errors with the wiring?

Answers are on page 10.184

NOTES

SECTION EIGHT
(SOME CASE STUDIES)

PURPOSE OF SECTION

Provide before and after data from conventional design to logic circuits. Illustrate poor performance of Johnson Controls enthalpy N9000 compared to APS circuit.

SINGLE

ZONE

CASE

STUDY

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http://www.apscontrols.ca

CONTENTS

<u>SINGLE ZONE CASE STUDY</u>

The intent of this case study is to allow an understanding of a significant problem relating to the Heating, Ventilation and Air Conditioning (HVAC) industry. The issue addressed involves simultaneous heating and cooling, due to poor design, installation and commissioning of control systems. This paper's intent is promotion of a thorough understanding relating to the problem, enabling one to address similar situations as they are encountered.

A significant quantity of greenhouse gas (GHG) enters our atmosphere directly relating to poor control arrangements as illustrated in this case study.

This case study is presented in four steps:

(1)- The fan system is considered manual as illustrated in drawing one.
What are you manually controlling?
What are your target conditions?
How would you logically adjust the operations?

(2)- The original design is illustrated in drawing two.
We identify the design problems.

(3)- The design was altered, as per drawing three, by a representative of another large control company.
We identify the design and commissioning problems.
The energy performance relationship is illustrated.
The performance of this arrangement is graphically illustrated from data collected before we altered any controls.

(4)- Drawing four is the same as drawing three except the preferred energy use relationship is illustrated.

(5)- Drawing five illustrates the control arrangement that will achieve the same common sense control you concluded in your manual operation on drawing one.
The performance of this arrangement is graphically illustrated.

(6)- An energy use comparison is drawn via the two performance graphs.
The 49.74% energy reduction enjoyed relates to the specific load conditions.
Based on practical experience, the alterations applied to this system normally produce reductions in the range of 15% to 25% annually.

(7)- Remember, if the control system is not making the same common sense, device positioning decisions as an intelligent person, with identical information input, there is something wrong with the control system!

DRAWING ONE

If charged with the responsibility of manually controlling the fan system illustrated in drawing one, what considerations exist?

(1) What are you required to achieve?

-(A)- A **healthy living environment** for occupants where the supply air is being directed.

-(B)- A **comfortable environment** in the same location.

-(C)- The most **efficient energy performance**.

(2)What components do you have to achieve your desired results? What are the functions of each of the components?

-(A)- A supply air fan and a return air fan.

The function is to force the supply air into and draw the return air from the occupied space.

-(B)- A heating source controlled via a manual heating valve.

The function is to provide heat to the system at varying quantities.

-(C)- A mechanical cooling source controlled via a manual cooling valve.

The function is to provide mechanical cooling at varying quantities.

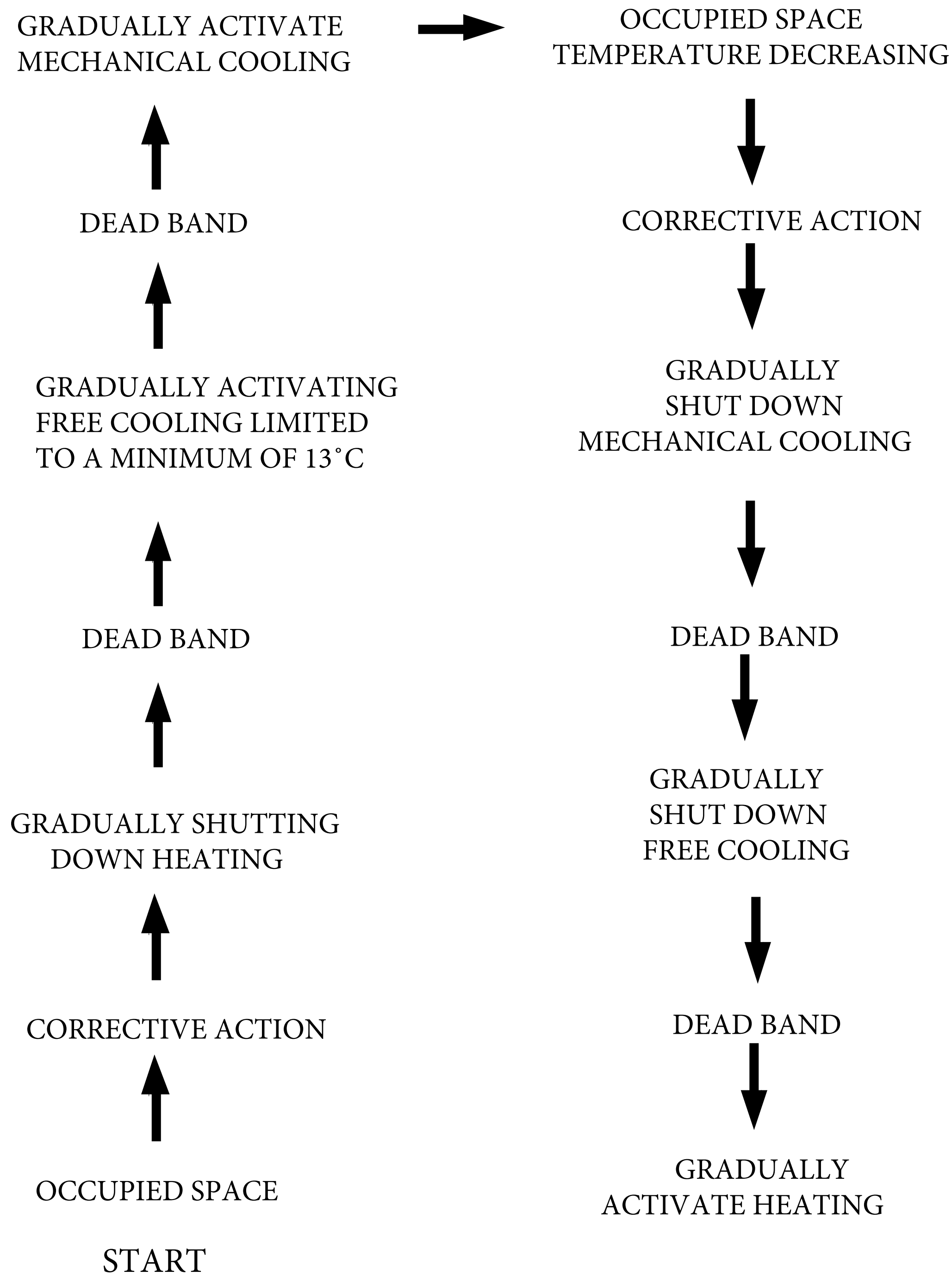
-(D)- Manual dampers enabling your selection of fresh air to varying degrees and rejecting air from the building to varying degrees.

The function is **two fold**.

-1- Provide at least minimum ventilation based on the greater demand of either the building code relating to human occupancy or provide exhaust air replacement relating to building operations.

-2- Provide ventilation in excess of minimum ventilation requirements to achieve cooling when the outside air temperature is lower than the building temperature.

<u>ACTION FLOW SHEET</u>



<u>FIGURE 1</u>

MANUAL OPERATION DRAWING ONE

VENTILATION

The manual dampers are manually adjusted to allow at least minimum ventilation at all times the space is occupied, addressing the greater of occupancy rates or exhaust fan air replacement quantities.

Shut down the fan at unoccupied times if it is not required.

Adjust the manual dampers to full recirculation at unoccupied times if there is no danger of collapsing the ductwork or creating another problem.

TEMPERATURE CONTROL

-1- If the occupied space is thermally satisfied, keep the heating valve closed to the heating coil, keep the cooling valve closed to the cooling coil and leave the manual dampers in the minimum ventilation position.

-2- If the occupied space starts to become cool, hold the manual dampers in the minimum ventilation position, keep the cooling valve closed to the cooling coil and open the heating valve to the heating coil only enough to hold the occupied space within the thermal comfort zone.

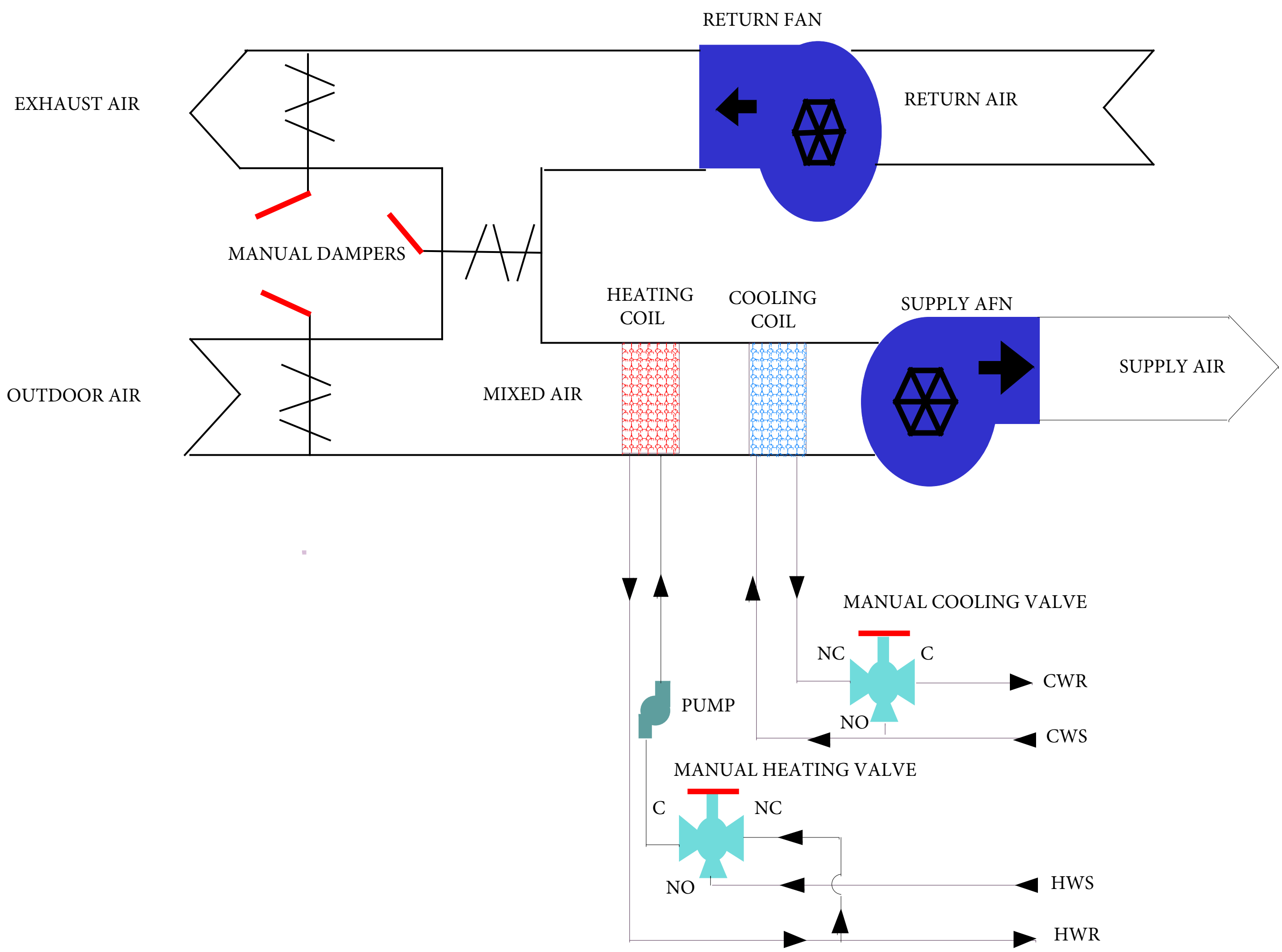
-3- When the occupied space starts to become warm below 13°C outside.

- Gradually reduce the quantity of hot water entering the heating coil. If the heating valve is completely closed off to the heating coil and the occupied space continues to increase in temperature, keep the heating valve closed to the heating coil, adjust the manual dampers to gradually reject the warm return air and bring more cool outside air into the occupied space.
- Do not allow the mixed air to drop below 13°C.
- Do not starve the supply fan or create a high discharge pressure on the return fan.
- If the air temperature after the heating coil drops below 5°C in any area of the coil, shut the fan down to prevent freezing of the coils.

-4- When the occupied space starts to become warm above 13°C outside:

- Follow the procedure outlined in "3". If the outside air can not cool the occupied space, gradually open the cooling valve to the cooling coil just enough to keep the occupied space within the thermal comfort zone.
- When the outside air becomes less efficient to cool than the return air from the occupied space return the manual dampers back to the minimum ventilation position.

DRAWING ONE


RETURN FAN
EXHAUST AIR
RETURN AIR
MANUAL DAMPERS
HEATING COIL
COOLING COIL
SUPPLY AFN
SUPPLY AIR
OUTDOOR AIR
MIXED AIR
MANUAL COOLING VALVE
NC
C
CWR
PUMP
NO
CWS
MANUAL HEATING VALVE
C
NC
NO
HWS
HWR

DRAWING TWO

ORIGINAL DESIGN

The original design logic was typical of many systems designed and installed in the past sixty years.

A return air controller modulated the heating valve (V1) and the chilled water cooling valve (V2) in proper sequence as the means of controlling the occupied space temperature. The heating valve closed completely before the cooling valve allowed any chilled water flow through the cooling coil.

This part presented no problem relating to the system logic.

A mixed air controller modulated the mixing dampers (D1) and (D2) to maintain a constant 58°F upstream of the heating coil. An outside air high limit controller over-rode the mixed air controller, preventing free cooling when the outside air temperature exceeded 70°F. A minimum positioning relay (MPR1) assured at least minimum ventilation when the fan system was active.

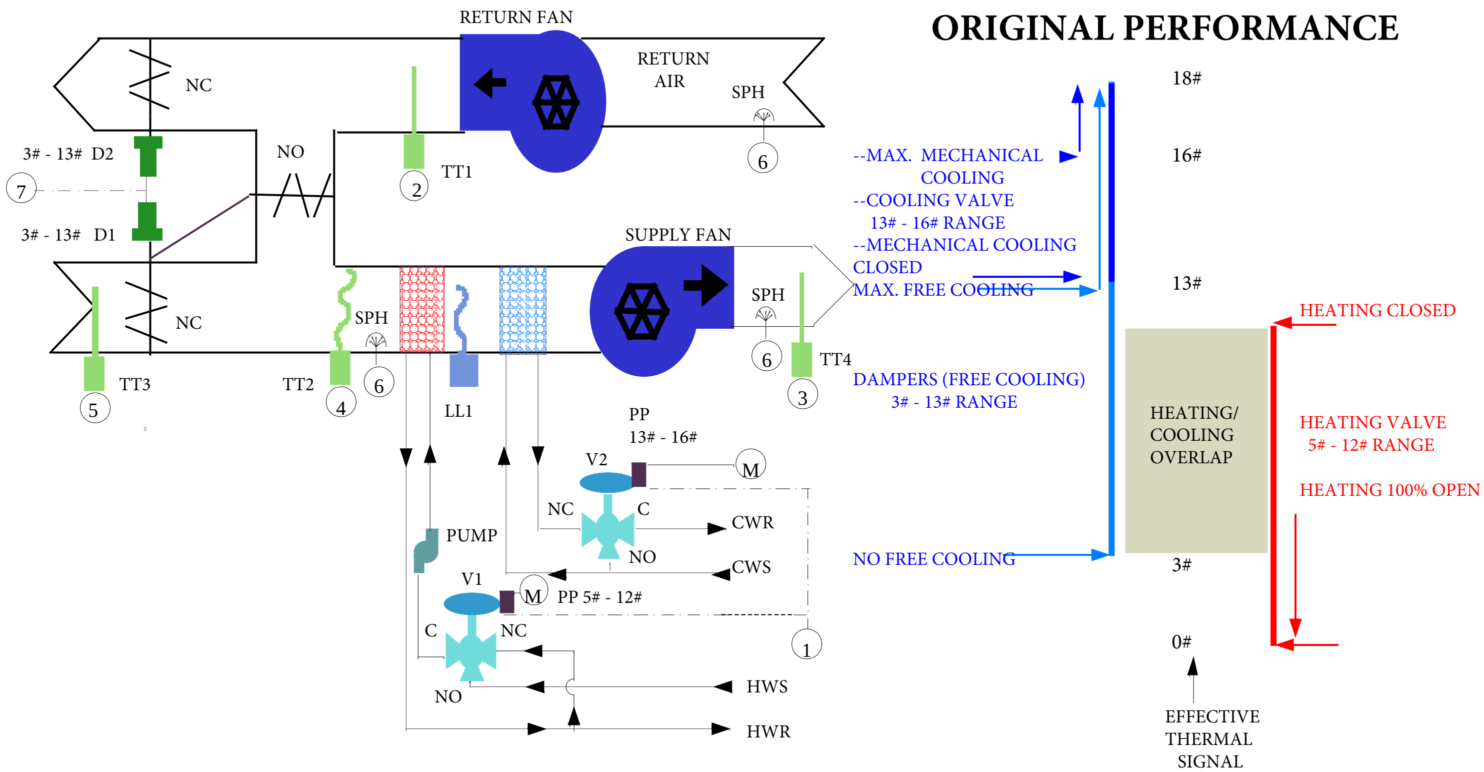
The outside air high limit could be upgraded to an enthalpy function, comparing the total cooling load of the outside air to the total cooling load of the return air. This loop selects the air stream with the lesser load reducing the workload on the cooling coil.

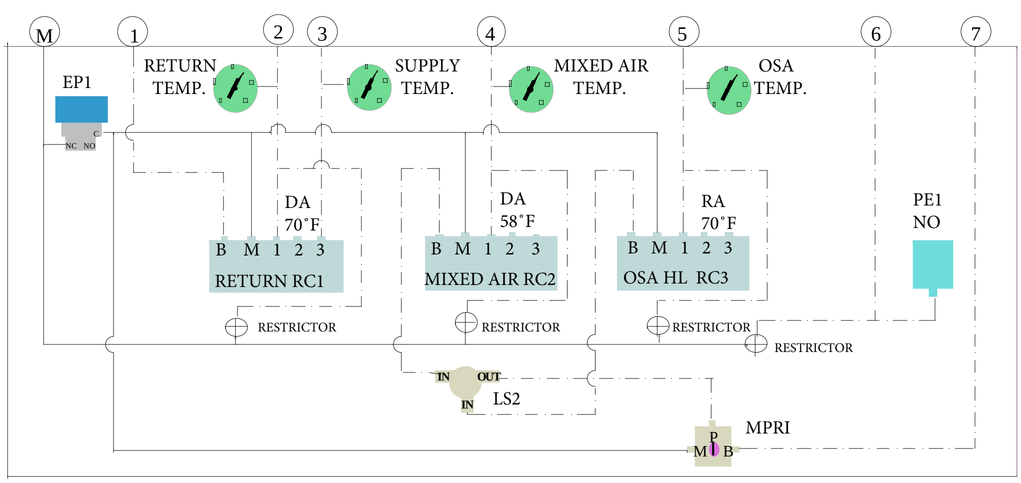
The minimum positioning relay could be upgraded to a carbon-dioxide sensor or VOC (Volatile Organic Compound) sensor better addressing the occupied space contaminant load.

The mixed air design caused the "free cooling" to be active all the time the fan ran. The 58°F mixed air design intended to address the free cooling requirements when the space required its maximum amount of cooling. **At all other times heat had to be active to compensate for the over-cooling of the mixed air.** This is similar to opening the windows in your house and then turning on your furnace to compensate for the cold air coming through the windows. If you just closed the windows, the furnace would be required at a lesser rate.

Drawing two illustrates the overlap of free cooling and heating. The total BTU quantity of unwanted free cooling had to be matched with an equal BTU quantity of heating only to address this problem with the control logic. Additional heat was required to address the normal building heat losses of the occupied space.

DRAWING TWO


RETURN FAN
RETURN AIR
SPH
NC
NO
3# - 13# D2
3# - 13# D1
TT1
SUPPLY FAN
TT2
TT3
TT4
LL1
PUMP
V1
V2
PP 13# - 16#
PP 5# - 12#
CWR
CWS
HWS
HWR
ORIGINAL PERFORMANCE
18#
16#
13#
3#
0#
--MAX. MECHANICAL COOLING
--COOLING VALVE 13# - 16# RANGE
--MECHANICAL COOLING CLOSED
MAX. FREE COOLING
DAMPERS (FREE COOLING) 3# - 13# RANGE
NO FREE COOLING
HEATING/ COOLING OVERLAP
HEATING CLOSED
HEATING VALVE 5# - 12# RANGE
HEATING 100% OPEN
EFFECTIVE THERMAL SIGNAL


EP1
RETURN TEMP.
SUPPLY TEMP.
MIXED AIR TEMP.
OSA TEMP.
DA 70˚F
DA 58˚F
RA 70˚F
B M 1 2 3
RETURN RC1
MIXED AIR RC2
OSA HL RC3
PE1 NO
RESTRICTOR
LS2
IN OUT
MPRI

DRAWING THREE
SYSTEM AS WE FOUND IT

Another contractor altered the control logic, as illustrated in drawing three.

The control contractor attempted to address two problems regarding the original design.

(1) They allowed the return air controller access to the mixed air loop.

The conservation intent was correct, but the application made no difference. The two mistakes occurred when they selected the biasing relay and commissioned that relay. The first error was selecting a biasing relay with a 1:1 ratio that can not squeeze the ten pound damper function between the heating and cooling valves where there is only a one pound span available. The second error observed caused the biasing relay signal to be advanced to the dampers rather than retarding the signal.

In theory the changes caused a greater over-lap of free cooling and heating; however, at a practical level the system performed exactly as the original design.

(2) The second alteration performed was installing a gradual switch in the occupied space, allowing the occupants the option of raising or lowering the return air set point.

This caused problems as the gradual switch had a blind dial with only six dots. The set point impact range was 42F° (7F° per dot) while the occupants thought each dot represented 1F°.

The two intended improvements were both valid points; however, the implementation of both actually had an over-all negative impact on control of the occupied space.

Graph one, at the back of this report, illustrates the system's energy performance as we found it. Graph two, at the back of this report illustrates the system's energy performance after we corrected the over-sights in logic.

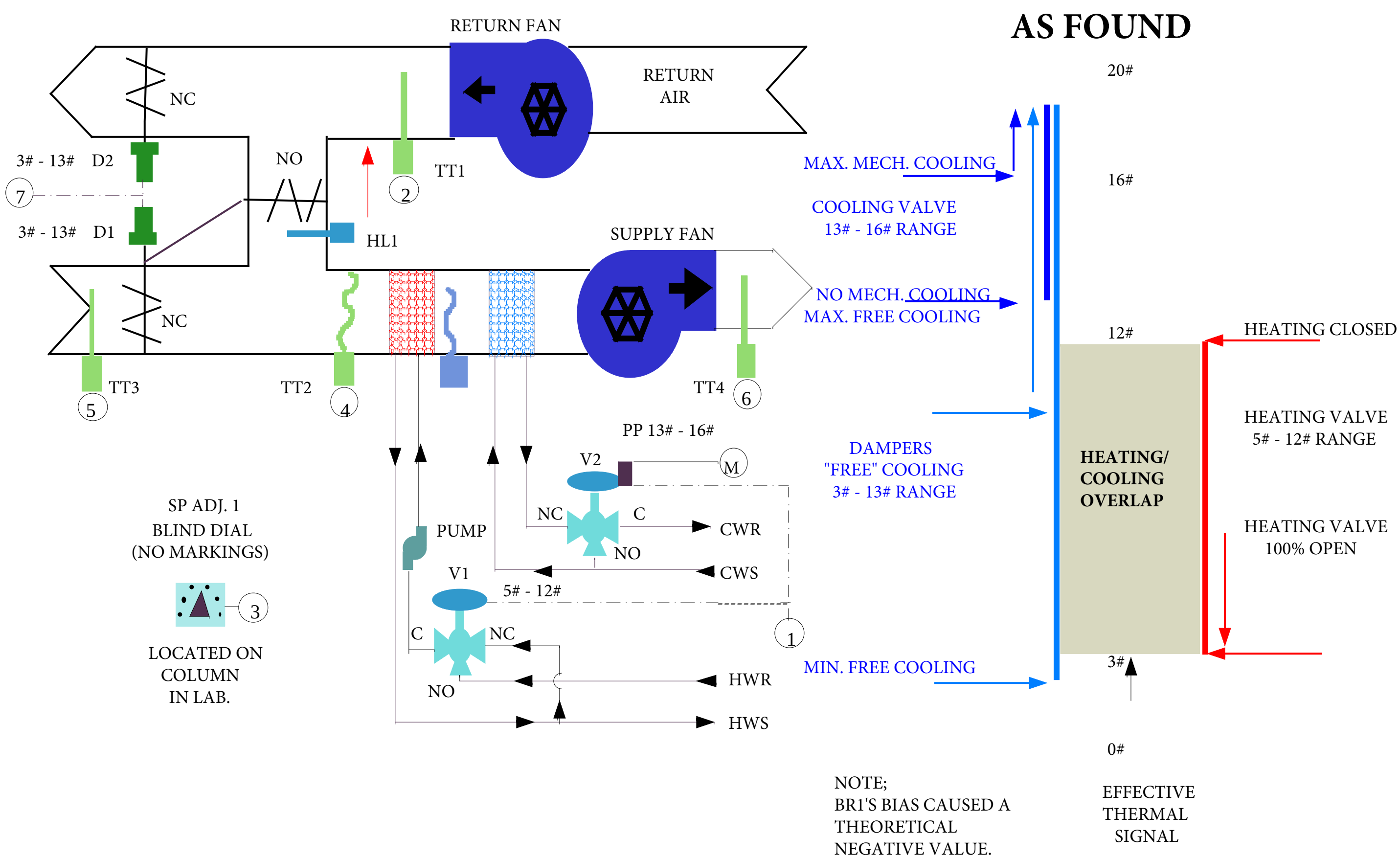
Attempting to equalize the load, we deliberately chose time slots for both graphs that were at night, eliminating solar impact and human activity variation. The times chosen experienced identical average outside air temperatures. Other unknown factors caused the space to require 81.6°F supply air during data collection for the first graph and 70.66°F for the second graph. *During the period of the first graph, the system would have required 49.75% less heat attaining the 81.6°F supply air temperature if it employed the logic present for the second graph.*

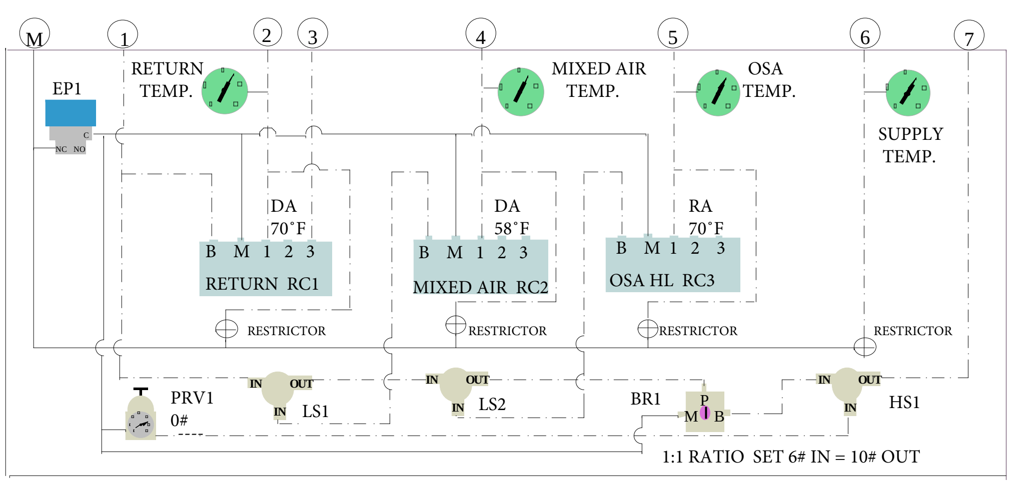
The system, as we found it, brought in an average of 74.86% fresh air, causing the return air controller to open the heating valve an average of 41.4%. In comparison the system as it now stands brought in an average of 27.4% fresh air, causing the return air controller to open the heating valve an average of 3.4%.

The old system over-cooled the mixed air and then activated the heating coil to compensate for the unwanted cooling. The new system logic allows the controls variable use of heat in the return air that has already been purchased.

DRAWING THREE

NOTE: TT1 SHOULD BE MOVED UP STREAM OF RETURN FAN PREVENTING FAN HEAT GAIN.
HL1 SHOULD BE MOVED TO DISCHARGE OF RETURN FAN AVOIDING DEAD AIR LOCATION.





DRAWING FOUR

PREFERRED PERFORMANCE CHART

Drawing four is the same as drawing three with the exception of the performance chart at the upper right corner. The chart on drawing three illustrates the actual performance as we found the system and Drawing Four illustrates the performance chart of the desired sequence of operation.

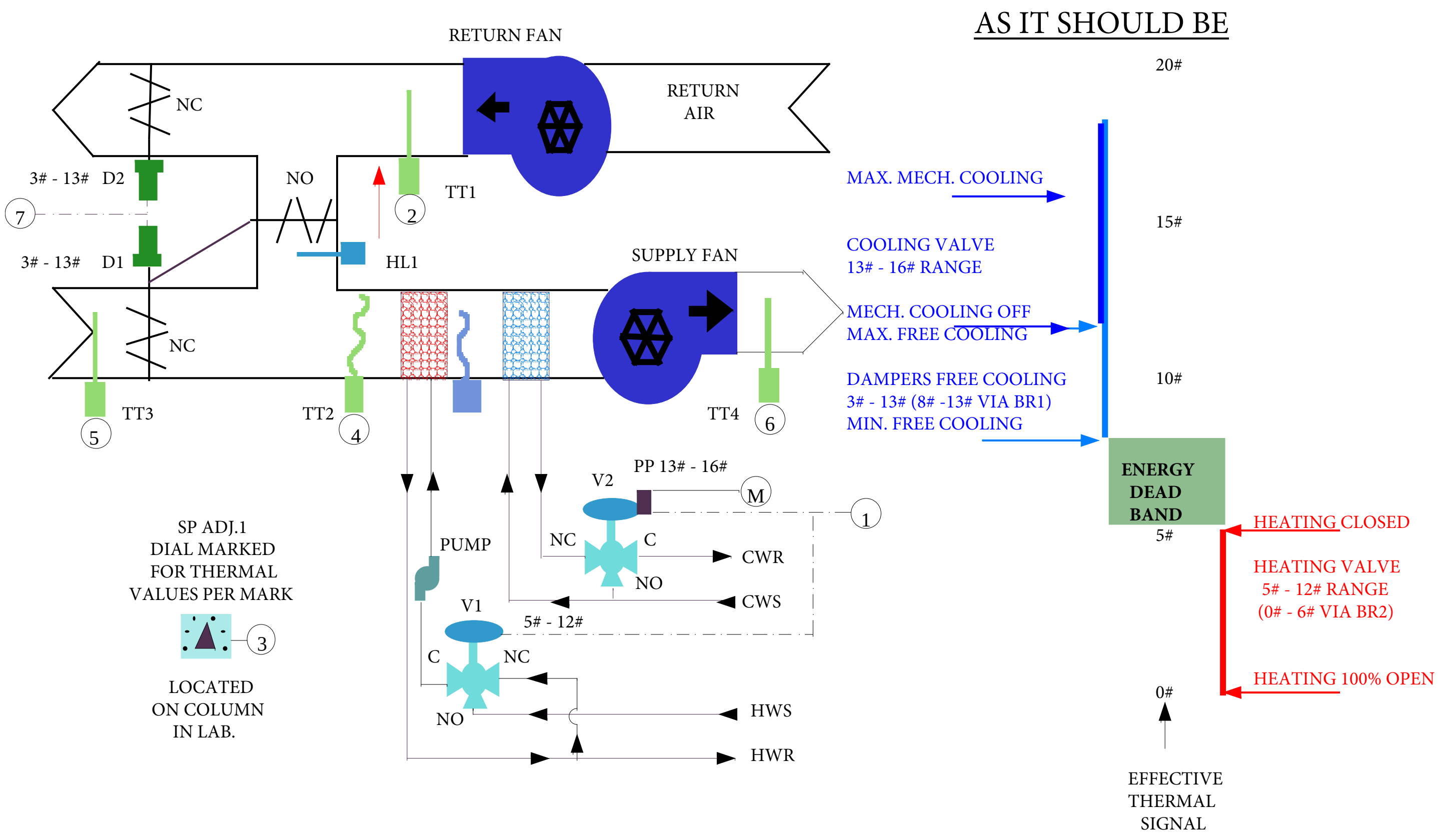
Two other points worthy of note are the locations of the return air temperature transmitter (TT1) and the electric high limit (HL1).

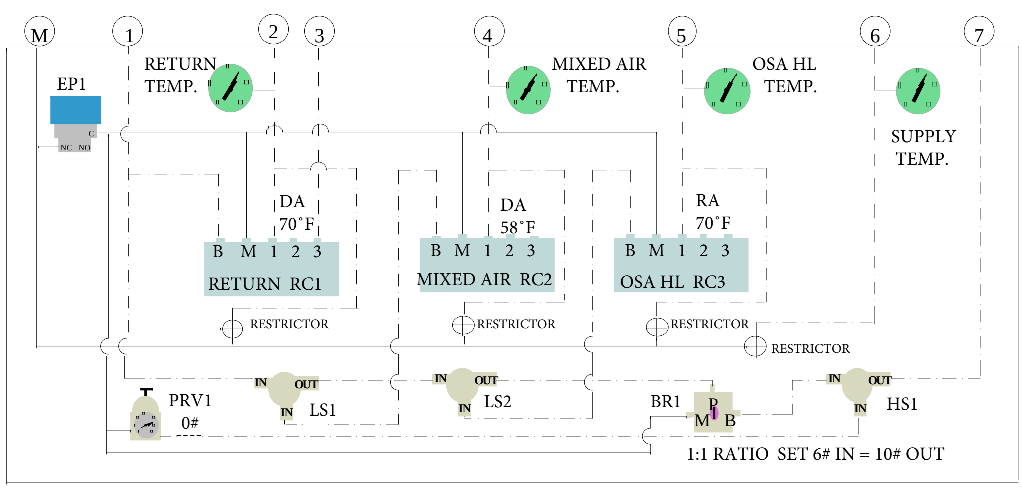
The electric high limit replaced PE1 and its associated sprinkler heads (SPH) in the original design. The original design was well arranged regarding sensing for high temperatures. The location of HL1 could hinder the device from sensing a high temperature if the building were on fire. If the outside air temperature were between 55°F and 70°F, the dampers would drive to full fresh air as the fire warmed the return air. HL1 would be sheltered from sensing the fire while the system fed the fire with oxygen.

TT1 senses the heat gain from the return fan. Up stream of the return fan is a better location for sensing the return air temperature.

DRAWING FOUR

NOTE: TT1 SHOULD BE MOVED UP STREAM OF RETURN FAN PREVENTING FAN HEAT GAIN.
HL1 SHOULD BE MOVED TO DISCHARGE OF RETURN FAN AVOIDING DEAD AIR LOCATION.





DRAWING FIVE
SYSTEM AFTER LOGIC CORRECTIONS

Drawing five illustrates the control arrangement now controlling on site with the sequence of operation.

The performance chart illustrated on drawing four is achieved by this control system.

The high limit (HL1) is moved to sense the return air temperature at all times.

TT1 is moved to sense the true return air temperature.

It is interesting to note that the same number of instruments were required to achieve the logic system in drawing five as the first conservation attempt, as illustrated in drawing three.

GENERAL

The enthalpy circuit was not installed at the time of writing this report.

The minimum ventilation is being controlled as per the original design at the time of writing this report.

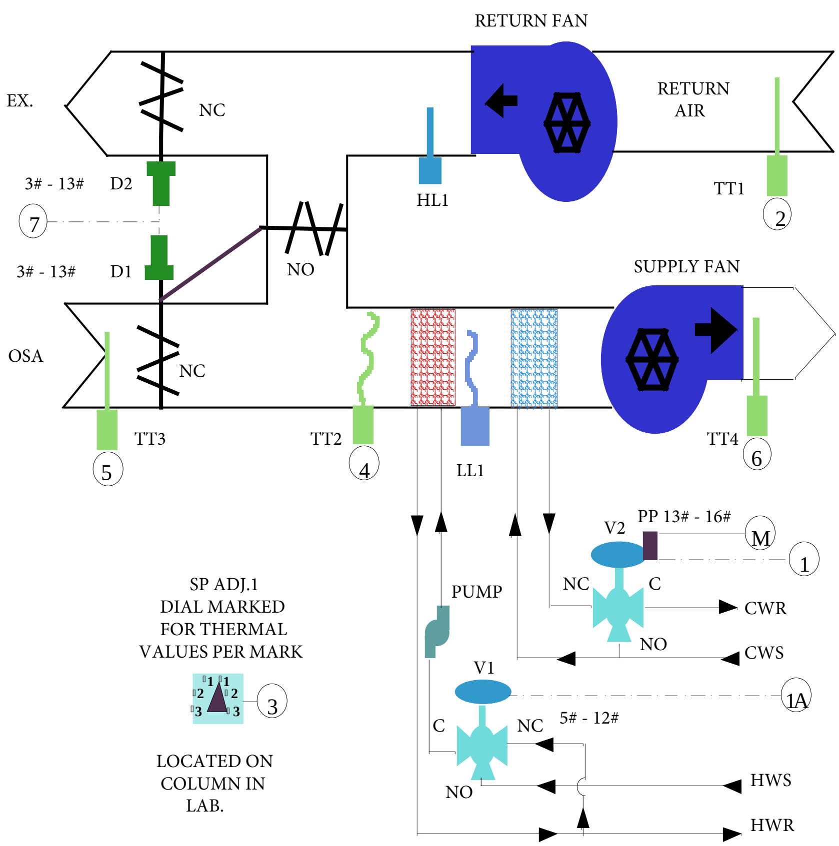
Two potential opportunities relating to energy reduction that required further investigation are:

-1- The fan could possibly be duty cycled, at unoccupied times, to a night set back temperature. This would save energy relating both to the heating and the electrical consumption.

-2- The minimum ventilation rate is 27% on the fan as it is currently running. The return fan has inlet vanes currently set at a fixed point. If the exhaust fans run intermittently, the opportunity may exist to reduce the minimum ventilation rates automatically based on the pressure differential from the laboratory to atmosphere.

This case study is the approach employed for one system in one building. The responsibility rests on the individual designer using any techniques presented in this report. The focus must remain on the safety and well being of the building occupants and the building. The only energy available for savings is the energy consumed by illogical control.

DRAWING FIVE



SEQUENCE OF OPERATION

-The fan runs continuously to suit the laboratory operations.

-Solenoid valve (EP1) passes control air to the fan controls when the fan is running.

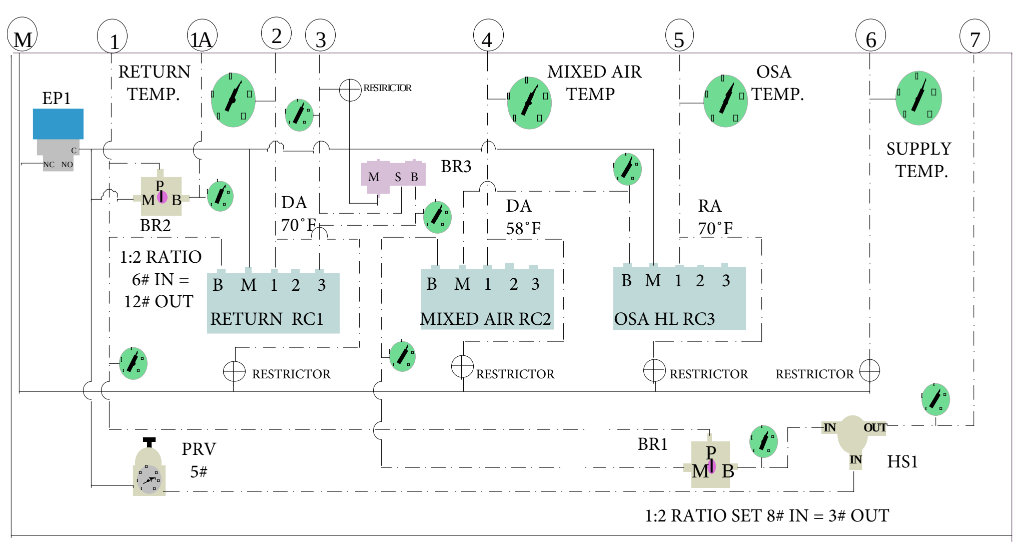
-Receiver controller (RC1) senses the return air temperature via transmitter (TT1) and receives a set point demand from (SP ADJ 1). RC1 sends its signal to biasing relays (BR1) and (BR2) as well as to cooling valve (V2). The instrument arrangement causes the heating valve (V1) to close first, then the free cooling via damper actuators (D1) and (D2) is maximized and then mechanical cooling is employed via V2 on a call for cooling.

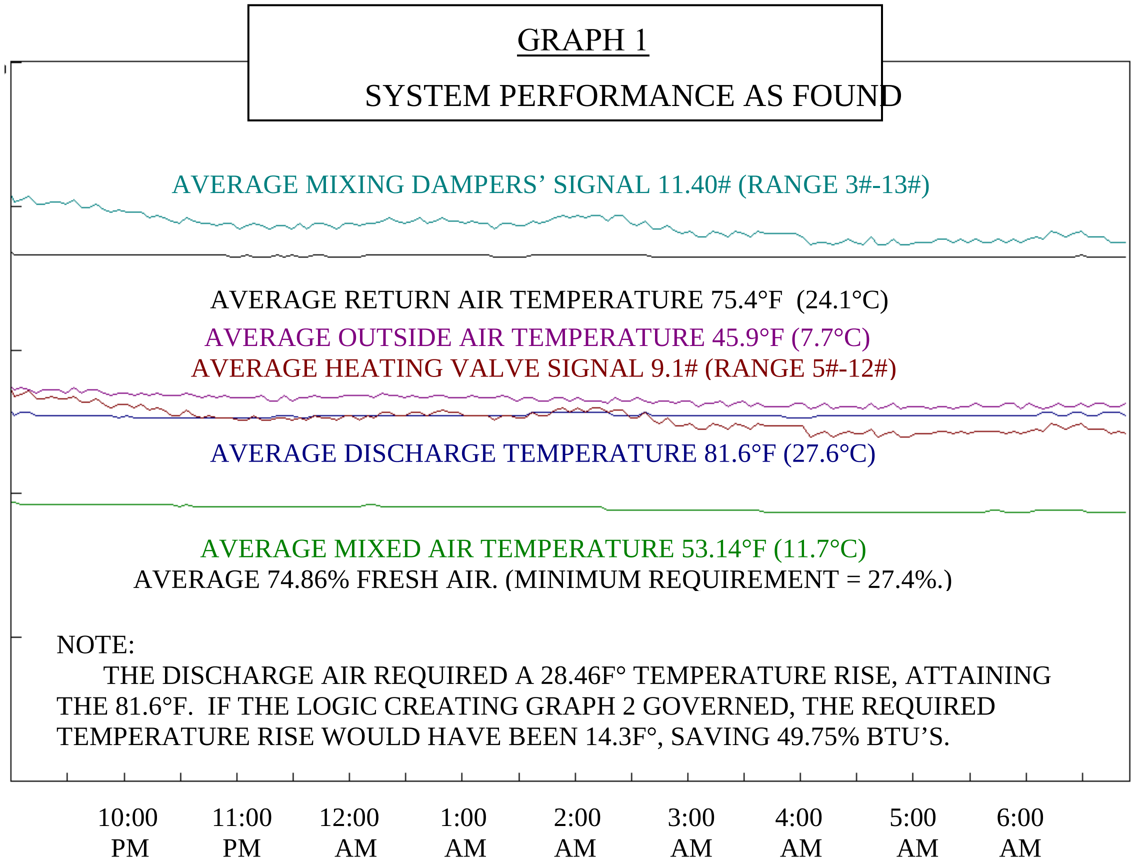
-BR1 sends its signal to high selector (HS1) which also receives a signal from regulator (PRV1) to assure the system of at least minimum ventilation while the fan is running.

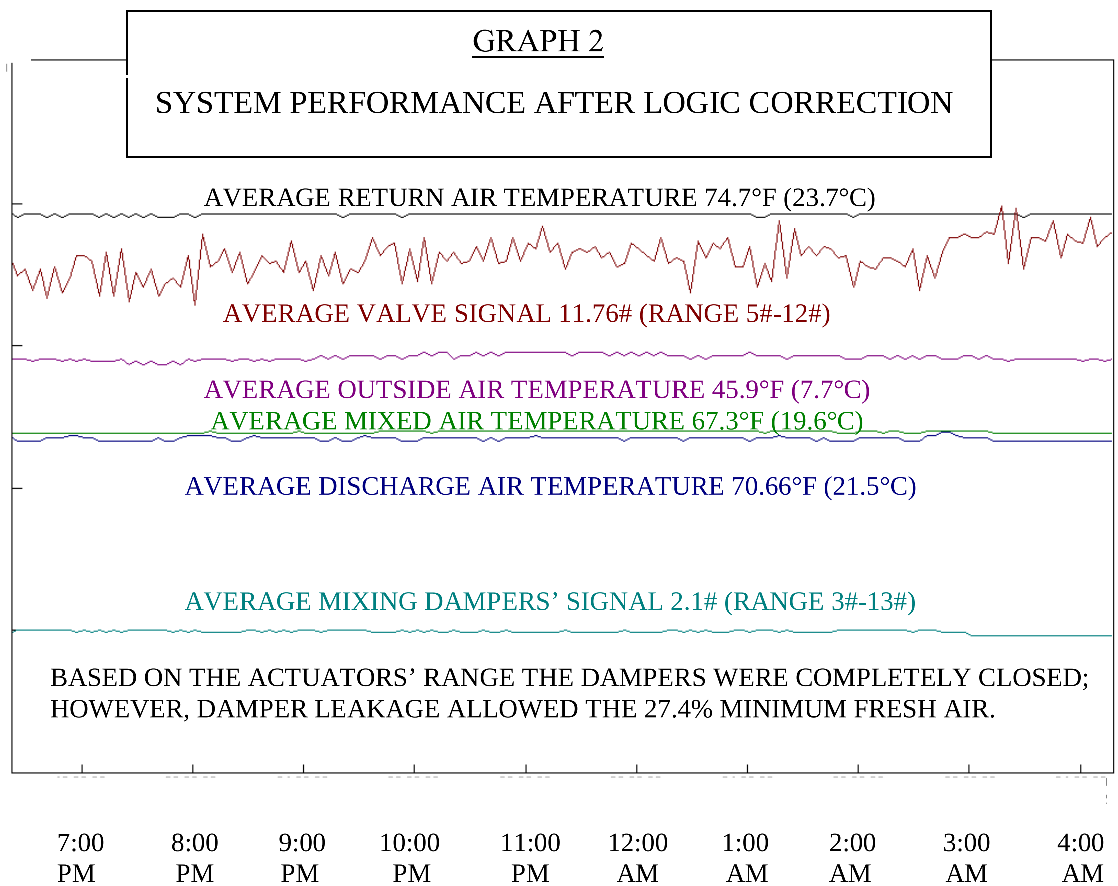
-Receiver controller (RC3) senses the outside air temperature via transmitter (TT3). RC3 passes its signal to receiver controller (RC2) only when free cooling is available, based on the outside air temperature to economize on mechanical cooling. RC2 senses the mixed air temperature via transmitter (TT2). RC2 limits the mixed air to a minimum of 58°F.

-Turn down ratio relay (BR3) is set to limit the range of set point demand from the laboratory.

-If low limit (LL1) senses a temperature below its set point on any one foot of its element the fan will shut down. If high limit (HL1) senses a temperature above its set point the fan will shut down. Both LL1 and HL1 require resetting after being tripped.




GRAPH 1
SYSTEM PERFORMANCE AS FOUND
AVERAGE MIXING DAMPERS' SIGNAL 11.40# (RANGE 3#-13#)
AVERAGE RETURN AIR TEMPERATURE 75.4°F (24.1°C)
AVERAGE OUTSIDE AIR TEMPERATURE 45.9°F (7.7°C)
AVERAGE HEATING VALVE SIGNAL 9.1# (RANGE 5#-12#)
AVERAGE DISCHARGE TEMPERATURE 81.6°F (27.6°C)
AVERAGE MIXED AIR TEMPERATURE 53.14°F (11.7°C)
AVERAGE 74.86% FRESH AIR. (MINIMUM REQUIREMENT = 27.4%.)
NOTE:
THE DISCHARGE AIR REQUIRED A 28.46F° TEMPERATURE RISE, ATTAINING THE 81.6°F. IF THE LOGIC CREATING GRAPH 2 GOVERNED, THE REQUIRED TEMPERATURE RISE WOULD HAVE BEEN 14.3F°, SAVING 49.75% BTU'S.
10:00 PM
11:00 PM
12:00 AM
1:00 AM
2:00 AM
3:00 AM
4:00 AM
5:00 AM
6:00 AM


GRAPH 2
SYSTEM PERFORMANCE AFTER LOGIC CORRECTION
AVERAGE RETURN AIR TEMPERATURE 74.7°F (23.7°C)
AVERAGE VALVE SIGNAL 11.76# (RANGE 5#-12#)
AVERAGE OUTSIDE AIR TEMPERATURE 45.9°F (7.7°C)
AVERAGE MIXED AIR TEMPERATURE 67.3°F (19.6°C)
AVERAGE DISCHARGE AIR TEMPERATURE 70.66°F (21.5°C)
AVERAGE MIXING DAMPERS' SIGNAL 2.1# (RANGE 3#-13#)
BASED ON THE ACTUATORS' RANGE THE DAMPERS WERE COMPLETELY CLOSED; HOWEVER, DAMPER LEAKAGE ALLOWED THE 27.4% MINIMUM FRESH AIR.
7:00 PM
8:00 PM
9:00 PM
10:00 PM
11:00 PM
12:00 AM
1:00 AM
2:00 AM
3:00 AM
4:00 AM

NOTES

MULTIZONE

ZONE CASE

STUDY

APS
PHONE (905) 640-2333
FAX (905) 640-2444
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http://www.apscontrols.ca

MULTIZONE ZONE CASE STUDY

The intent of this case study is to allow an understanding of a significant problem relating to the Heating, Ventilation and Air Conditioning (HVAC) industry. The issue addressed involves simultaneous heating and cooling, due to poor design, installation and commissioning of control systems. This paper's intent is promotion of a thorough understanding relating to the problem, enabling one to address similar situations as they are encountered.

This case study reflects a relatively simple conservation circuit. More complicated circuits may assess the probability of a thermostat demand signal being false by looking at the outside and return air temperatures, as well as the thermostats' demand signals.

A significant quantity of greenhouse gas (GHG) enters our atmosphere directly relating to poor control arrangements as illustrated in this case study.

This case study is presented in these steps:

(1)- The fan system is considered manual as illustrated in drawing one.
What are you manually controlling?
What are your target conditions?
How would you logically adjust the operations?

(2)- The original design is illustrated in drawing two.
We identify the design problems.

(3)- The conservation circuit is illustrated in drawing three.

(4)- Graph #1 illustrates the original design heating and cooling function performance relative to varying outside air temperatures.

(5)- Graph #2 illustrates a comparison of original design cooling levels and actual cooling requirements of the warmest zone on the system.

(6)- Graph #3 illustrates a comparison of the original design heating levels and the actual heating requirements of the coolest zone on the system.

(7)- **Remember, if the control system is not making the same common sense, device poitioning decisions as an intelligent person, with identical information input, there is something wrong with the control system!**

DRAWING ONE

If charged with the responsibility of manually controlling the multizone fan system illustrated in drawing one, what considerations exist?

(1) What are you required to achieve?

-(A)- A **healthy living environment** for occupants where the supply air is being directed.

-(B)- A **comfortable environment** in the same location.

-(C)- The most **efficient energy performance**.

(2)What components do you have to achieve your desired results? What are the functions of each of the components?

-(A)- A supply air fan and a return air fan.

The function is to force the supply air into and draw the return air from the occupied space.

-(B)- A heating source controlled via a manual heating valve.

The function is to provide heat to the system at varying quantities.

-(C)- A mechanical cooling source controlled via a manual cooling valve.

The function is to provide mechanical cooling at varying quantities.

-(D)- Manual zone mixing dampers enabling you to blend the hot deck air and the cold deck air.

The function is to satisfy individual zone thermal requirements.

-(E)- Manual dampers enabling your selection of fresh air to varying degrees and rejecting air from the building to varying degrees.

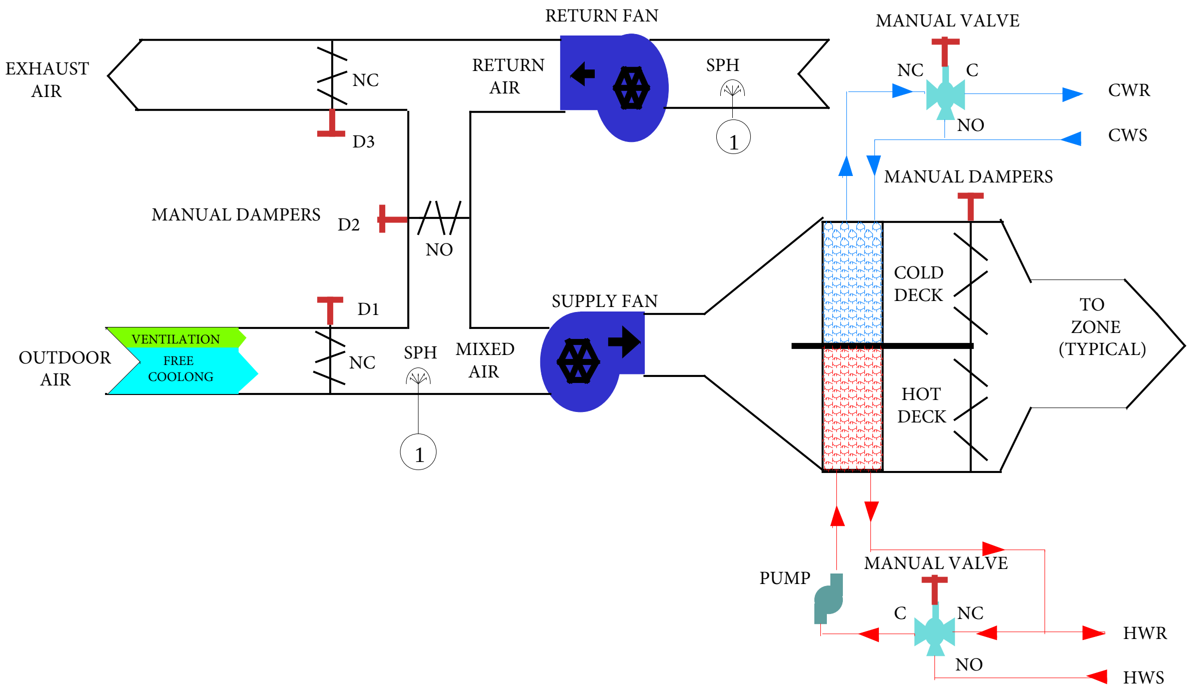
The function is **two fold**.

-1- Provide at least minimum ventilation based on the greater demand of either the building code relating to human occupancy or provide exhaust air replacement relating to building operations.

-2- Provide ventilation in excess of minimum ventilation requirements to achieve cooling when the outside air provides cooling benefit.

DRAWING ONE

MULTIZONE MANUAL CONTROL



MANUAL OPERATION DRAWING ONE

VENTILATION

The manual mixed air dampers are manually adjusted to allow at least minimum ventilation at all times the space is occupied, addressing the greater of occupancy rates or exhaust fan air replacement quantities.

Shut down the fan at unoccupied times if it is not required.

Adjust the manual dampers to full recirculation at unoccupied times if there is no danger of collapsing the ductwork or creating another problem.

Duty cycle the fan at unoccupied times from a set back thermostat if this suits the system.

TEMPERATURE CONTROL

-1- If the all the zones are thermally satisfied, keep the heating valve closed to the hot deck coil, keep the cooling valve closed to the cold deck coil and leave the manual mixed air dampers in the minimum ventilation position.

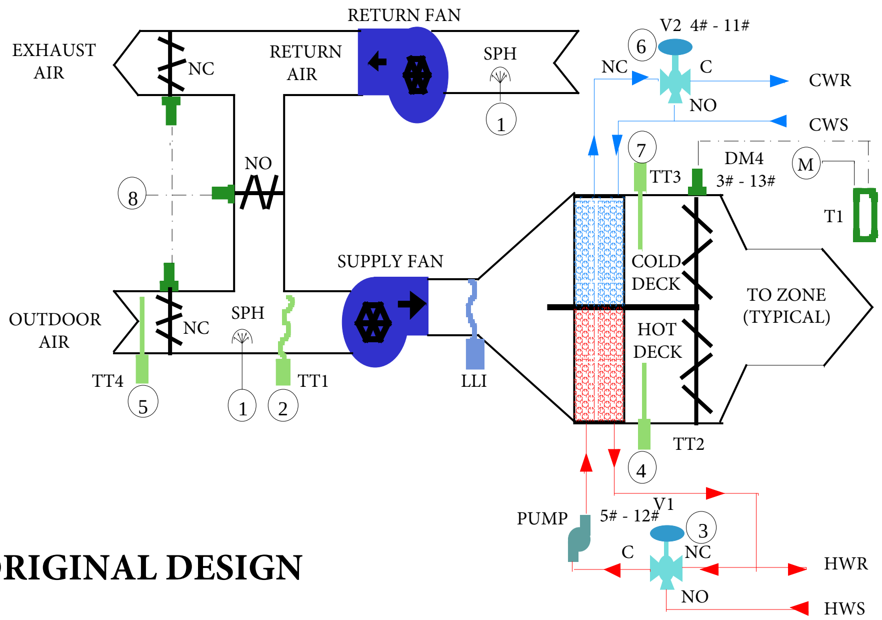
-2- If any zone starts to become cool, adjust that zone's manual zone dampers to full hot deck and no cold deck. Then increase the hot deck temperature **just enough** to keep that zone within the comfort zone.
- Adjust the other zones' dampers to blend hot and cold deck air to satisfy each zone's particular thermal requirement.

-3- If any zone starts to become warm when the outside air is below 55°F, adjust that zone's manual zone dampers to full cold deck and no hot deck.
- Gradually open the fresh air damper and exhaust air damper while closing the return air damper.
- Do not allow the mixed air to drop below 55°F.
- Do not starve the supply fan or create a high discharge pressure on the return fan.
- Adjust the other zones' dampers to blend hot and cold deck to satisfy each zone's particular thermal requirement.
- If the air temperature after the heating coil drops below 38°F in any area of the coil; shut the fan down to prevent freezing of the coils.

-4- When any zone starts to become warm above 55°F outside:
- Follow the procedure outlined in "3". If the outside air can not satisfy the warmest zone, gradually open the cooling valve to the cold deck cooling coil just enough to keep the warmest zone within the thermal comfort zone.
- When the outside air becomes less efficient to cool than the return air from the occupied space, return the manual dampers back to the minimum ventilation position.

NOTES

DRAWING TWO



ORIGINAL DESIGN

SEQUENCE OF OPERATION

The cold deck is always maintained cold. Outside air is used in winter and the cooling valve (V2) is used in summer.

The hot deck is always maintained hot, to varying degrees, based only on the outside temperature.

The thermostats modulate their respective zone damper motors, mixing hot and cold air, attaining the desired supply air temperature to satisfy the zone requirements.

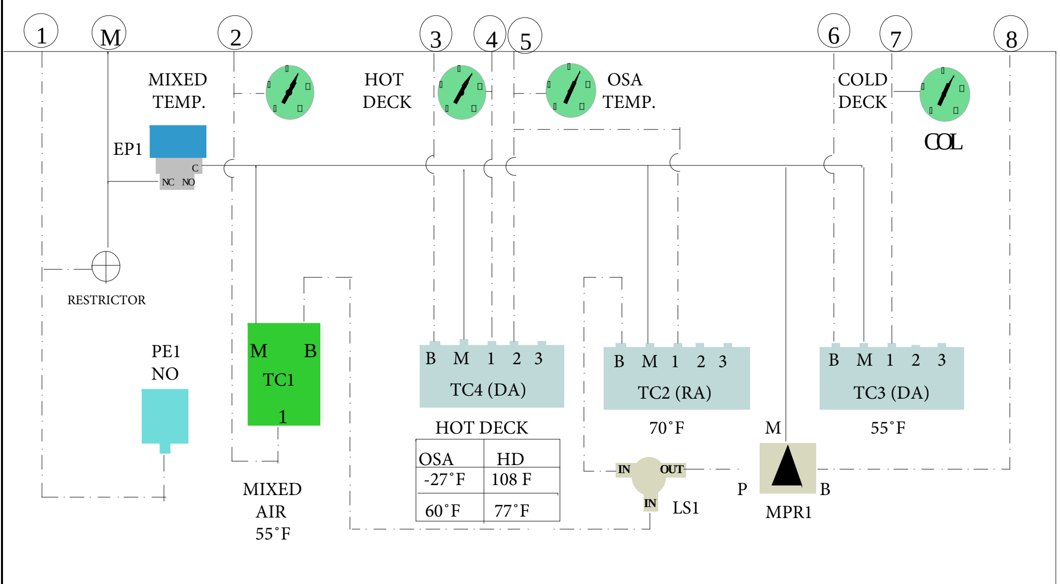
BASIC LOGIC PROBLEMS

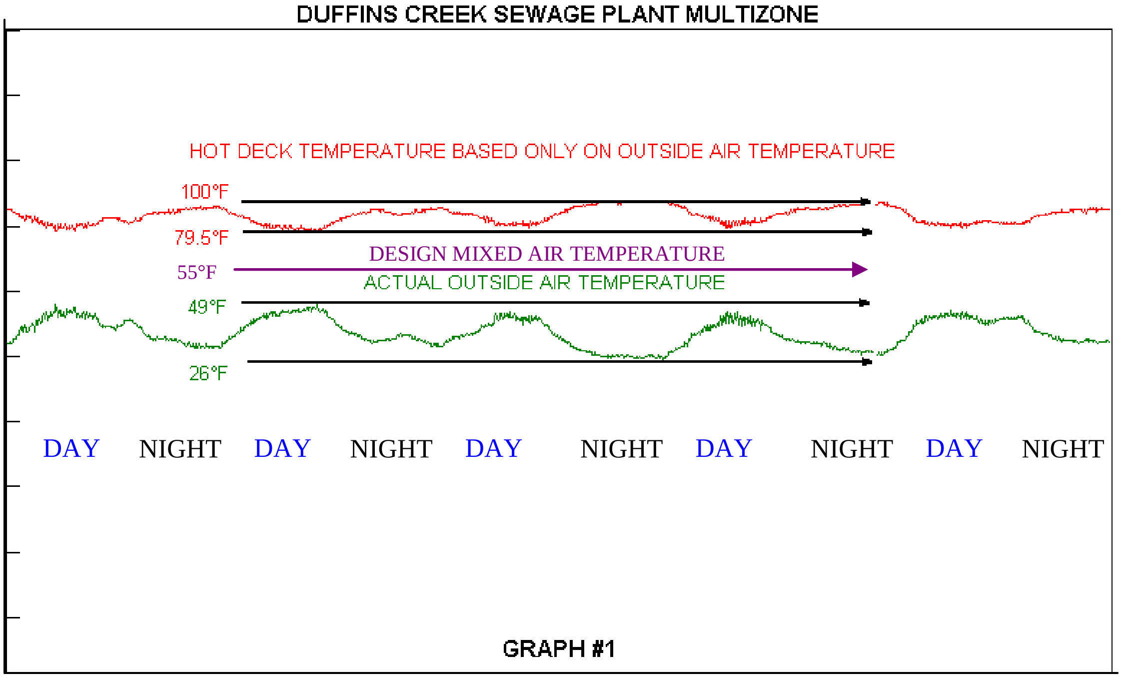
The cold deck temperature will satisfy the maximum cooling requirement the system will experience; therefore, it is too much cooling for all other conditions.

The hot deck temperature will satisfy the maximum heating requirement the system will experience at each specific outdoor temperature; therefore, it is too much heating for all other conditions.

The system produces cooling and heating, at times, when no zone requires either.

The system runs during unoccupied times.





ORIGINAL DESIGN TEMPERATURE GRAPH FOR HOT DECK AND COLD DECK

This graph illustrates the relationship between the temperature of the hot deck air and the outside air temperature. The hot deck becomes warmer as the outside air becomes cooler, at a fixed ratio, set in the hot deck controller.

On drawing two you see that the control loop controlling the hot deck does not communicate with the space heating requirements. The loop simply produces the degree of heating required for the maximum anticipated heating requirement of the occupied space. There is no compensation for load variations in the occupied space.

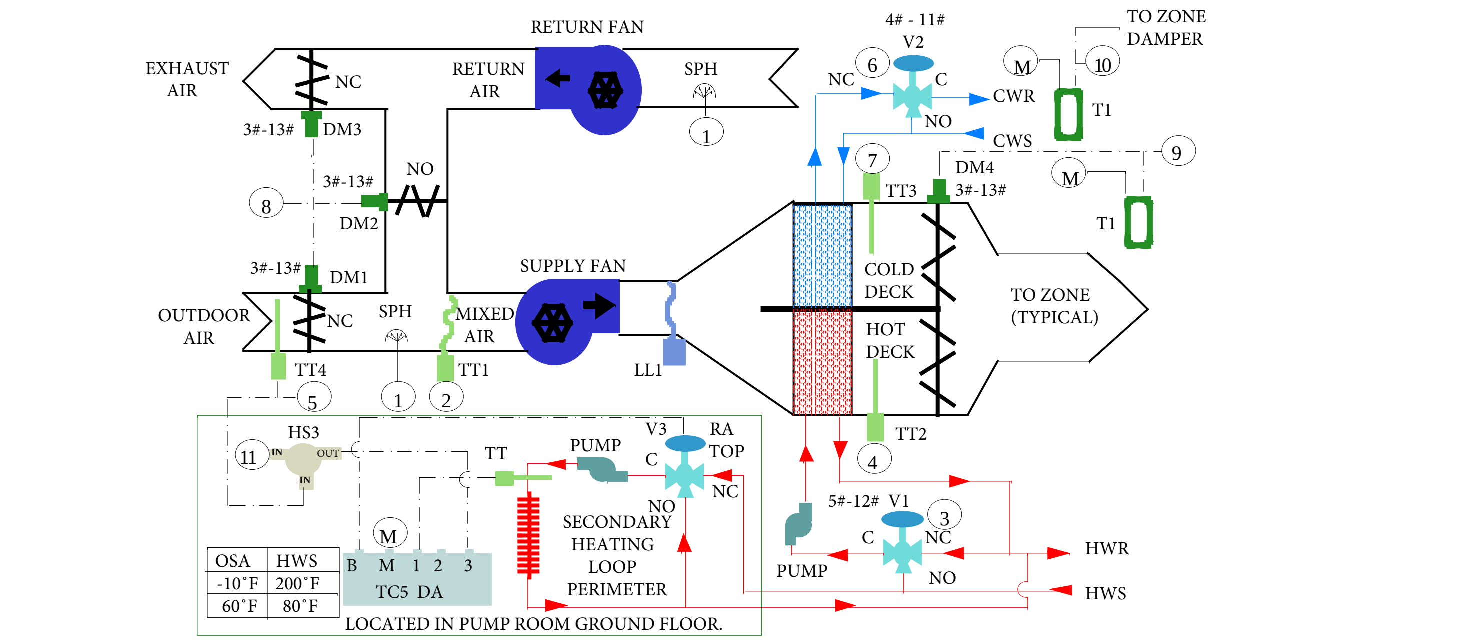
The graph also illustrates the mixed air temperature (cold deck temperature in the winter).

On drawing two you see that control loop controlling the mixed air does not communicate with the space cooling requirements. The loop simply produces the degree of cooling required for the maximum anticipated cooling requirement of the occupied space. There is no compensation for load variations in the occupied space.

This arrangement of ignoring the actual requirements of the zones tends to inject too much heat into the system. The system has to introduce extra cooling to compensate for the portion of the heat that is not actually required to address the heat loss.

At a domestic level, this is similar to running your furnace 60% of the time while the building heat loss only requires the furnace to run 50%. The house would over-heat unless you either reduced the run time to 50% or opened your windows to compensate for the extra heat generated by the excessive heat production. You would keep your windows closed and reduce the furnace run time. Most multizone systems actually follow the other path, bringing in extra cooling to get rid of the excess heat.

DRAWING THREE



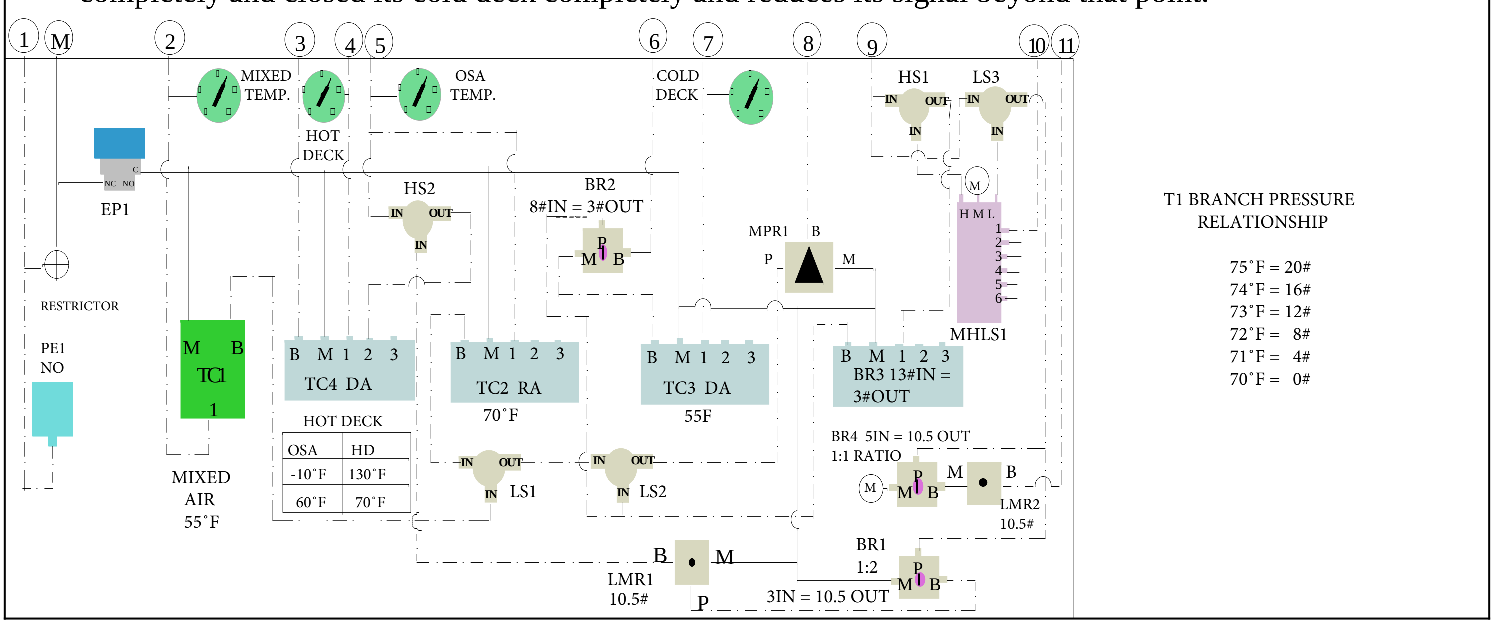
LOGIC OF OPERATION

The controls look at all the thermostat signals determining the greatest demand for heating and cooling via MHLS1, HS1 and LS3. If all thermostats are modulating their respective zone dampers the system "knows" that all zones are within 1F° of set point. Under this condition, the cold deck is controlled to minimum cooling and the hot deck is controlled to minimum heating.

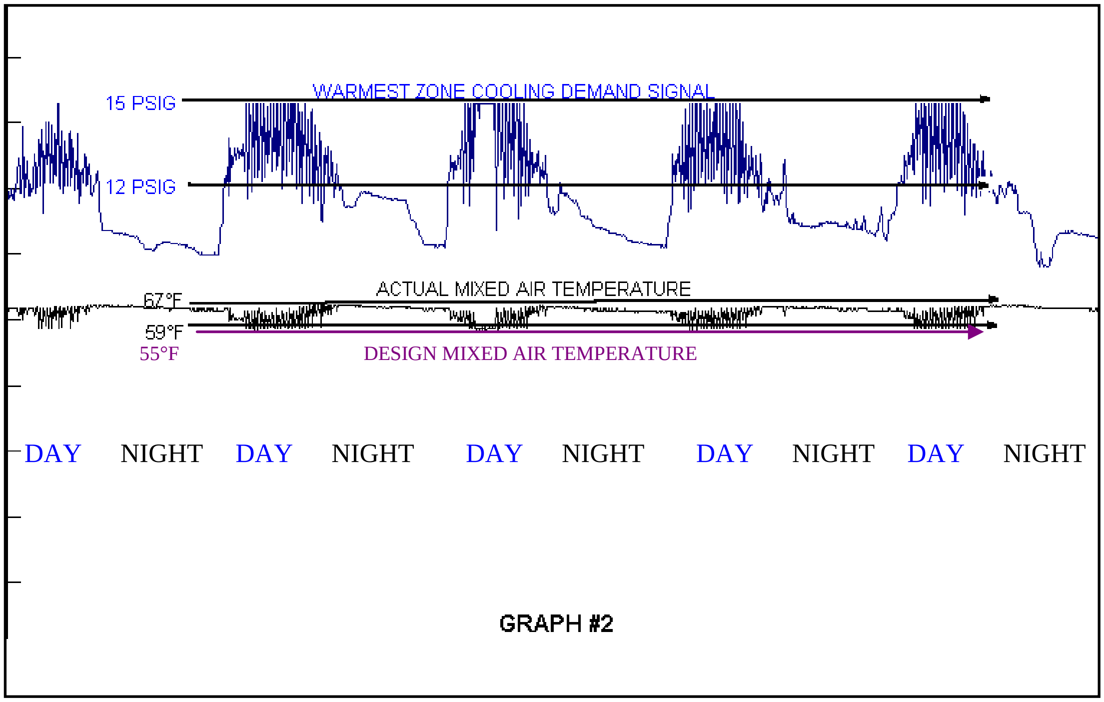
Any thermostat may lower the cold deck temperature limited by TC1 and TC3. The warmest thermostat "tells" the fan controls via the logic relays that at least one thermostat has closed off its hot deck completely and opened its cold deck completely. As the highest signal rises the cold deck will gradually become cooler.

The dampers return their minimum position when the outside air is more difficult to cool than the return air via TC2.

Any thermostat may raise the hot deck temperature, limited by TC4, after it has opened its hot deck completely and closed its cold deck completely and reduces its signal beyond that point.



DUFFINS CREEK SEWAGE PLANT MULTIZONE



GRAPH #2

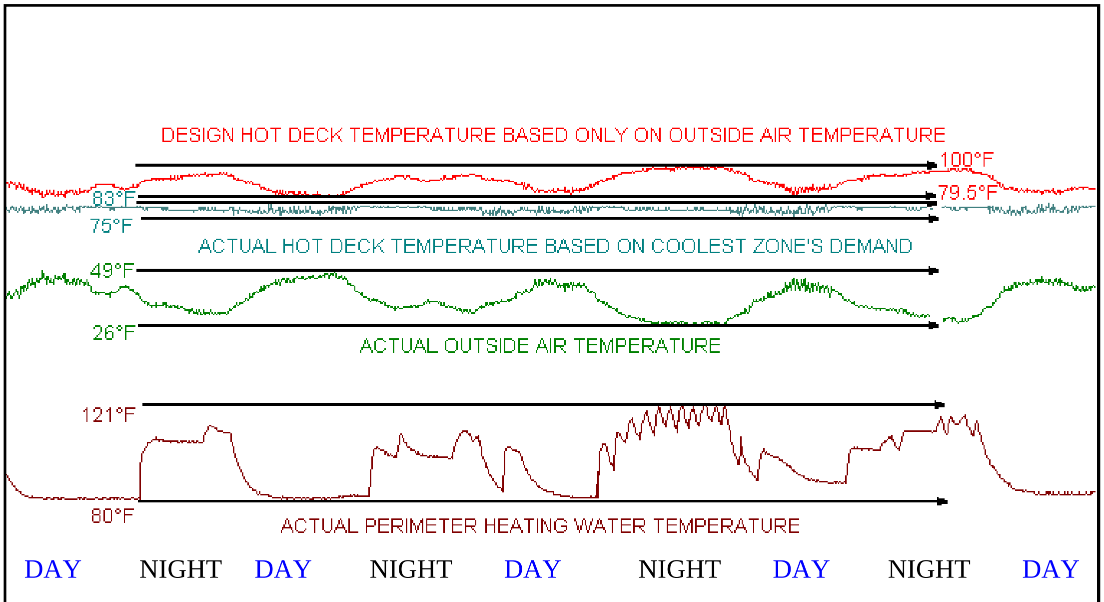
This graph illustrates the relationship of the requirement for cooling by the warmest room to the actual mixed air temperature. (The mixed air temperature is the cold deck for winter operation.) The mixed air temperature based on original design is also illustrated.

On drawing two you see that control loop controlling the mixed air does not communicate with the space cooling requirements. The loop simply produces the degree of cooling required for the maximum anticipated cooling requirement of the occupied space. There is no compensation for load variations in the occupied space. Drawing three shows the communication between the zone with the greatest requirement for cooling and the cooling functions on the fan system.

The graph shows that the warmest zone’s cooling requirements increase during the day when the outside air temperatures are higher and heat gains from lights, bodies, solar gain, etc. are present. When the warmest zone signal is at 12 PSIG or below there is no requirement for cooling; therefore, the mixed air loop provides only enough fresh air to satisfy the ventilation demands, addressing air quality requirements. When the warmest zone’s signal exceeds 12 PSIG the system “knows” that zone has its cold deck damper completely open and its hot deck damper completely closed. With this assurance that the demanding zone will not mix hot deck air with the requested cooling, the cold deck is gradually cooled as the warmest zone’s signal approaches 15 PSIG. The mixed air temperature is limited to a minimum of 55°F.

The average actual mixed air temperature of the total points logged, on the illustrated graph is 65°F. The original design maintained the mixed air at a constant 55°F. Based on a fan volume of 15,000 CFM the original design would have used 162,000 BTU/HR more than the new conservation circuit.

DUFFINS CREEK SEWAGE PLANT MULTIZONE



GRAPH #3

HOT DECK AND PERIMETER HEATING COMPARISON FROM ORIGINAL DESIGN TO CONSERVATION CIRCUIT.

This graph illustrates a comparison of the design hot deck temperature, as per drawing two, to the actual hot deck temperature required, as per drawing three, by the coolest zone demand for heat. The actual temperature required in the perimeter heating, based on the demand from the coolest zone, is also presented.

The average actual hot deck temperature over the five day period was 79.5°F. The average hot deck design temperature was 90.2°F. The conservation circuit trimmed an average 10.7F° off the hot deck design temperature. This reduction in hot deck temperature, which provided only enough heat to satisfy the coolest zone, allowed the mixed air temperature to be raised to an average of 65°F, which provided only enough cooling to satisfy the warmest zone. The mixed air design temperature was 55°F.

The average perimeter heating water temperature was 95.2°F. The average perimeter heating water design temperature was 119°F. The conservation circuit trimmed an average 23.8F° from the design perimeter heating water temperature.

GENERAL

Multizone fan systems often present the most potential benefit relative to the dollars spent on retrofit. Pages 6.44 and 6.45 illustrate savings achieved in the Durham Board of Education and the Scarborough Board of Education.

We normally expect heating reductions ranging from 30% to 50% on multizone retrofits. The range depends on the condition of the system before we touch the controls.

Dual duct systems with mixing boxes at the individual zones require the same logic changes as multizones. Dual duct systems are more expensive to retrofit, as the zone signals have to be run back to the main fan control panel. Multizones have the zone signals at the fan from the original installation.

No two systems are completely identical. This case study illustrates the approach taken for this individual building. When applying this logic to another building, you must consider the safety of the occupants, building systems, etc.

REPORT

ON THE IMPACT OF

EVAPORATIVE COOLING

VIA HUMIDIFIER WATER

SPRAYS REGARDING

ENERGY USE

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CONTENTS

EXERCISING COMMON SENSE

Before a person can improve a system's energy performance, or even calibrate an existing system, it is necessary that he or she acquire a full understanding of the mechanical and control components' relationships. **This means that person completely understands a logical plan that they would employ if they were manually operating the mechanical system.**

THE PLAN

1) Look at the fan system drawing on the next page and consider operating it manually.

2) Relating to this case study you must have access to all relevant values for temperatures and humidity. If you do not know the temperature or the relative humidity values, you can not determine the action required correcting an unacceptable condition.

3) You must be able to gradually open or close the dampers for fresh air, return air and exhaust air.

4) You must be able to gradually open or close the chilled water valve and the heating valve.

5) You must be able to start and stop the humidity spray pump

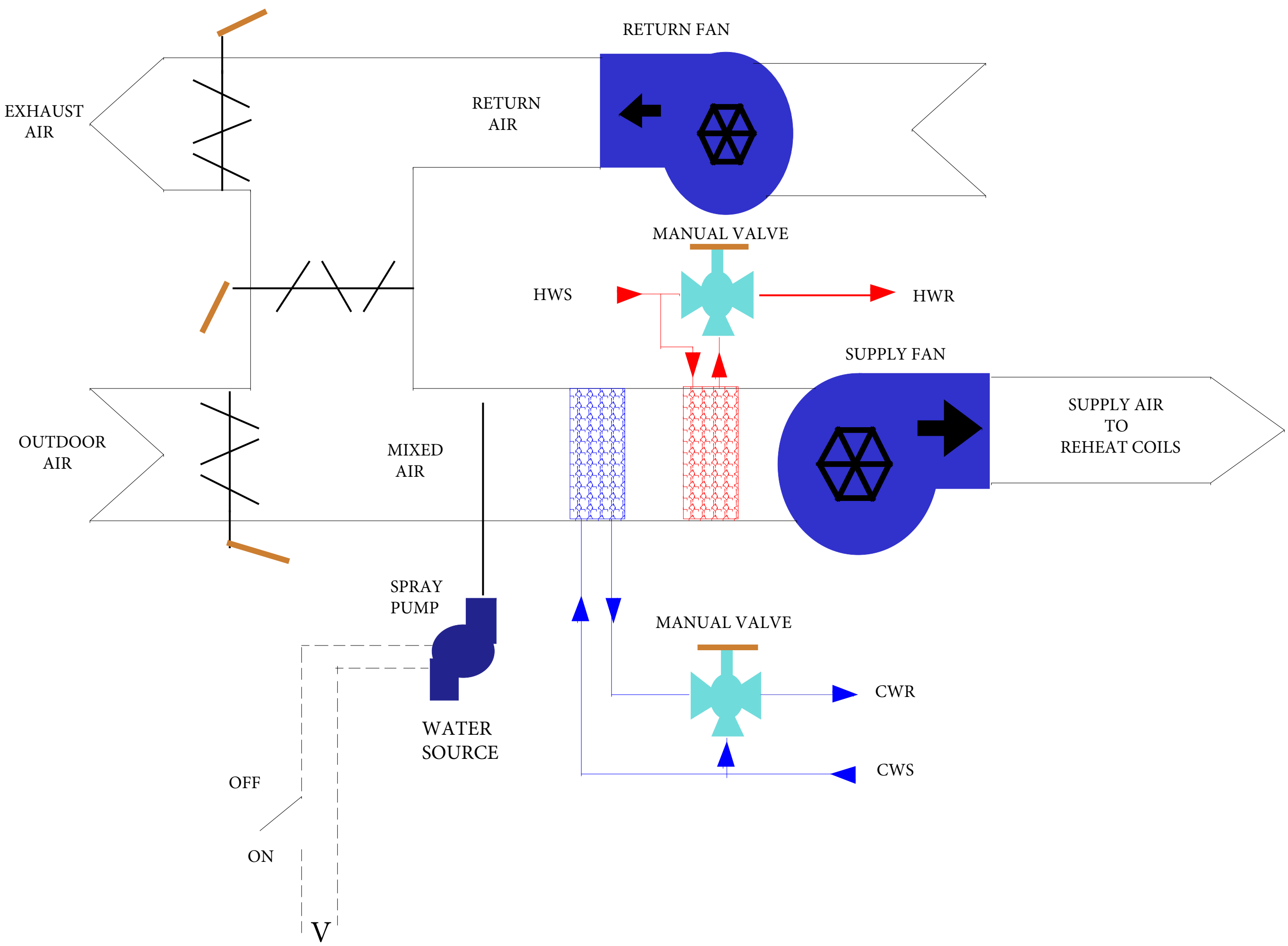
6) You must be able to start and stop the fan if it is not required at unoccupied times.

7) You must understand the impact that each function exerts overall on the system when employed to varying degrees.

When you understand all the interrelationships between the mechanical system and the building served, you are prepared to assess the control system.

Remember if the control system does not make the same mechanical adjustments as an intelligent human, given the same input values, there is something wrong with the control system.

MANUAL OPERATION OF FAN SYSTEM



-1- How would you adjust the manual damers to address minimum ventilation and free cooling requirements? What would be the minimum mixed air temperature?

-2- How would you control the manual heating and cooling valves? What would be the lowest target supply air temperature? Under what conditions would you allow flow into the heatign coil?

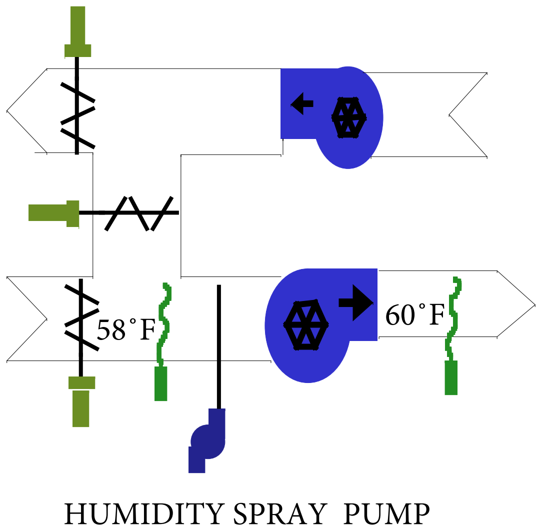
-3- How would you use the humidity spray pump? What would a normal range be for return air relative humidity?

-4-If the mixed air is at target, for example 58˚F, and the supply air temperature was at 60˚F from fan heat gain,what would the supply air temperature do when the humidity sprays are activated?

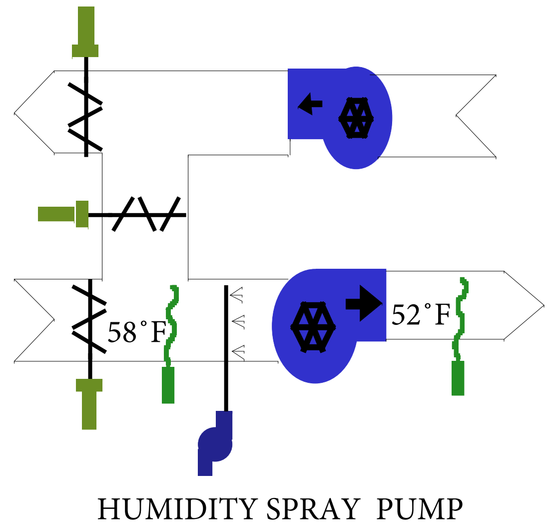
-5- What action would be taken to maintain the supply air temperature at 60˚F?

BEFORE CONTROL CHANGES

HUMIDITY SPRSYS OFF

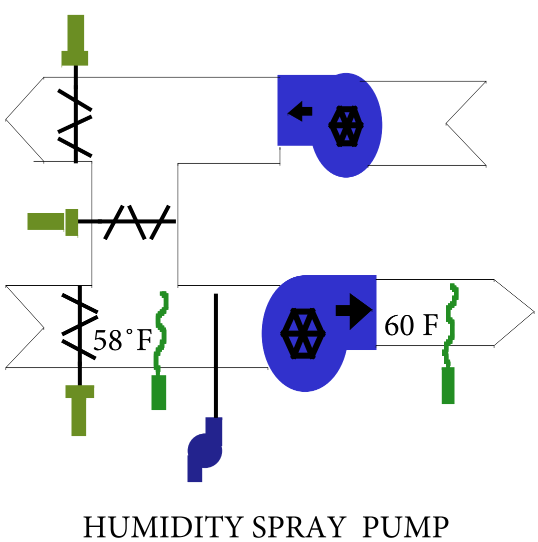


HUMIDITY SPRAYS ON

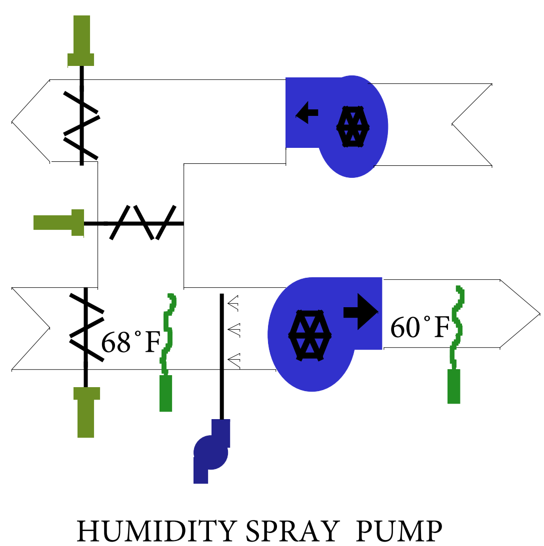


AFTER CONTROL CHANGES

HUMIDITY SPRAYS OFF



HUMIDITY SPRAYS ON



EFFECT OF EVAPORATIVE COOLING CAUSED BY HUMIDITY SPRAYS ON ENERGY USE

Many HVAC systems use water sprays to satisfy the humidity demands of the building based on the return air relative humidity level. Improper control reference points can result in large losses in energy to achieve the desired humidity levels. This report graphically illustrates how the losses occur, the magnitude of the losses and a control solution to the problem.

THE PROBLEM

Many systems control the mixed air to fifty-eight degrees Fahrenheit via averaging sensors in the mixed air plenum. The mixed air is drawn through the fan and forced to the occupied space.

If the humidity water sprays are not active, the normal discharge air temperature from the fan will be in the range of sixty degrees Fahrenheit. (Allowing for some heat gain from the fan.)

If the humidity sprays are active the supply air will drop to near dew point. This will cause the heating equipment to use more heat than while the sprays are not active. The heat, which has already been paid for, is being forced out the exhaust dampers via the mixed air control loop.

This report demonstrates the loss in one such system and clearly illustrates the savings created by causing the control logic to use heat contained in the return air for purposes of evaporating the moisture required for humidification. The system automatically raises the mixed air temperature to compensate for the cooling effect of the humidity sprays. This maintains a relatively constant supply air temperature while maximizing the use of energy already paid for in the building.

GRAPH #1

This graph uses the black line to illustrate the mixed air temperature, which was averaging about fifty-nine degrees Fahrenheit. The mixed air temperature was not affected by the status of humidity sprays; therefore, it remained relatively constant.

The blue line illustrates the discharge air temperature.

When the humidity sprays were not active the discharge air temperature averaged about sixty-three degrees Fahrenheit.

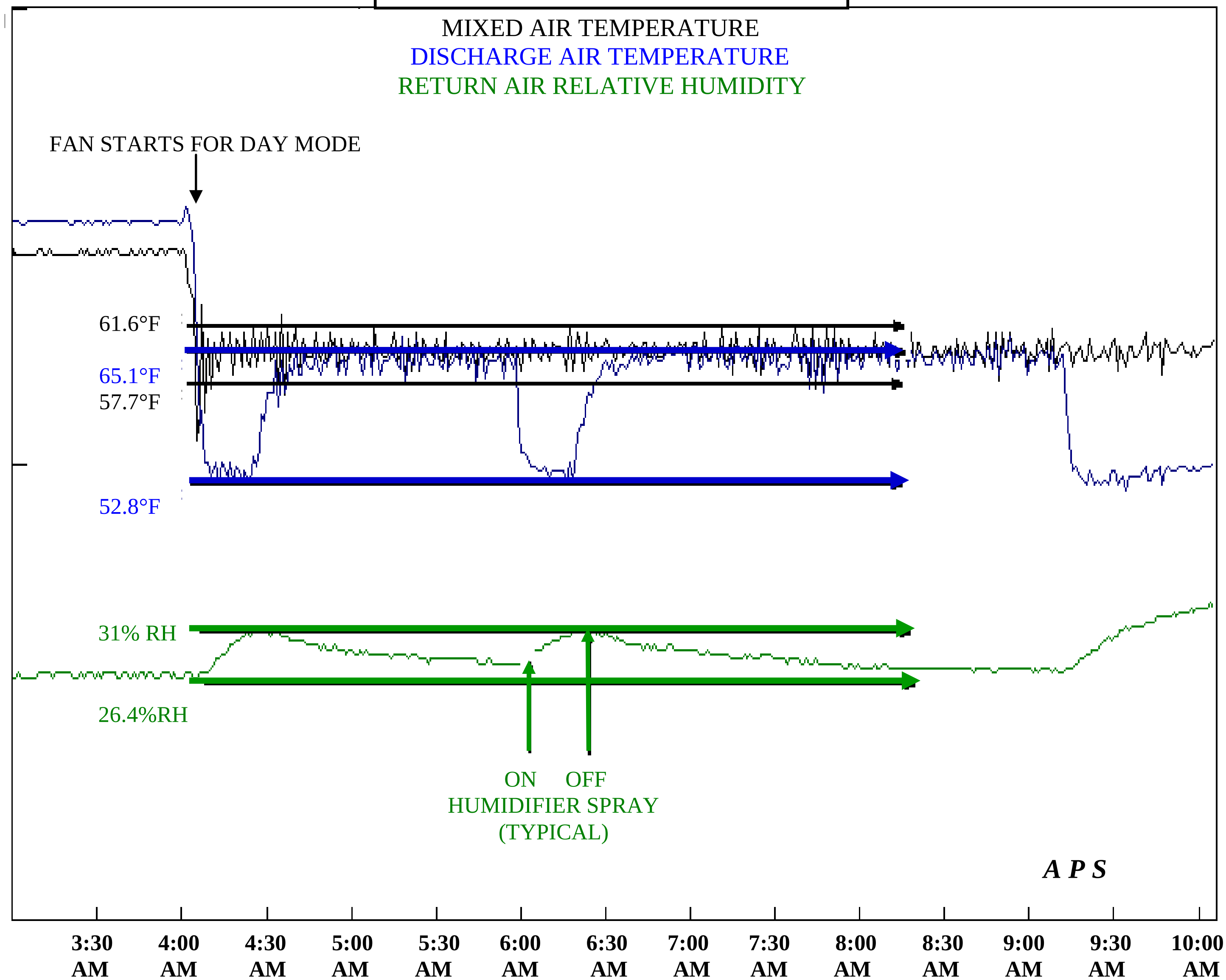
When the humidity sprays were active the discharge air temperature averaged about fifty-three degrees Fahrenheit.

The green line illustrates the return air relative humidity. This line indicates the run times for the humidity spray pump.

The energy loss is demonstrated by the dips in the discharge air temperature (blue line). At these times the heating system has to contend with a heating load approximately ten Fahrenheit degrees cooler than if the building did not have a humidification capability.

A control strategy was developed to allow the discharge air temperature controller to over-ride the mixed air controller to raise the mixed air temperature automatically when the humidity sprays were active. This was intended to keep the discharge air temperature relatively stable and remove the large dips in the discharge air temperature created by the spray activation. This would save energy and reduce the complaints of cold drafts by the occupants. The discharge air controller also retained its control over the fan system's chilled water valve for mechanical cooling.

GRAPH #1


MIXED AIR TEMPERATURE
DISCHARGE AIR TEMPERATURE
RETURN AIR RELATIVE HUMIDITY
FAN STARTS FOR DAY MODE
61.6°F
65.1°F
57.7°F
52.8°F
31% RH
26.4%RH
ON
OFF
HUMIDIFIER SPRAY
(TYPICAL)
A P S
3:30 AM
4:00 AM
4:30 AM
5:00 AM
5:30 AM
6:00 AM
6:30 AM
7:00 AM
7:30 AM
8:00 AM
8:30 AM
9:00 AM
9:30 AM
10:00 AM

<u>GRAPH #2</u>

A pneumatic low selector was added to the circuit to allow the discharge temperature controller to automatically over-ride the mixed air controller; as well as control the cooling valve as per its original intent. The chiller is off during free cooling mode; therefore, the discharge controller will only have an effect on the mixed air dampers during the winter

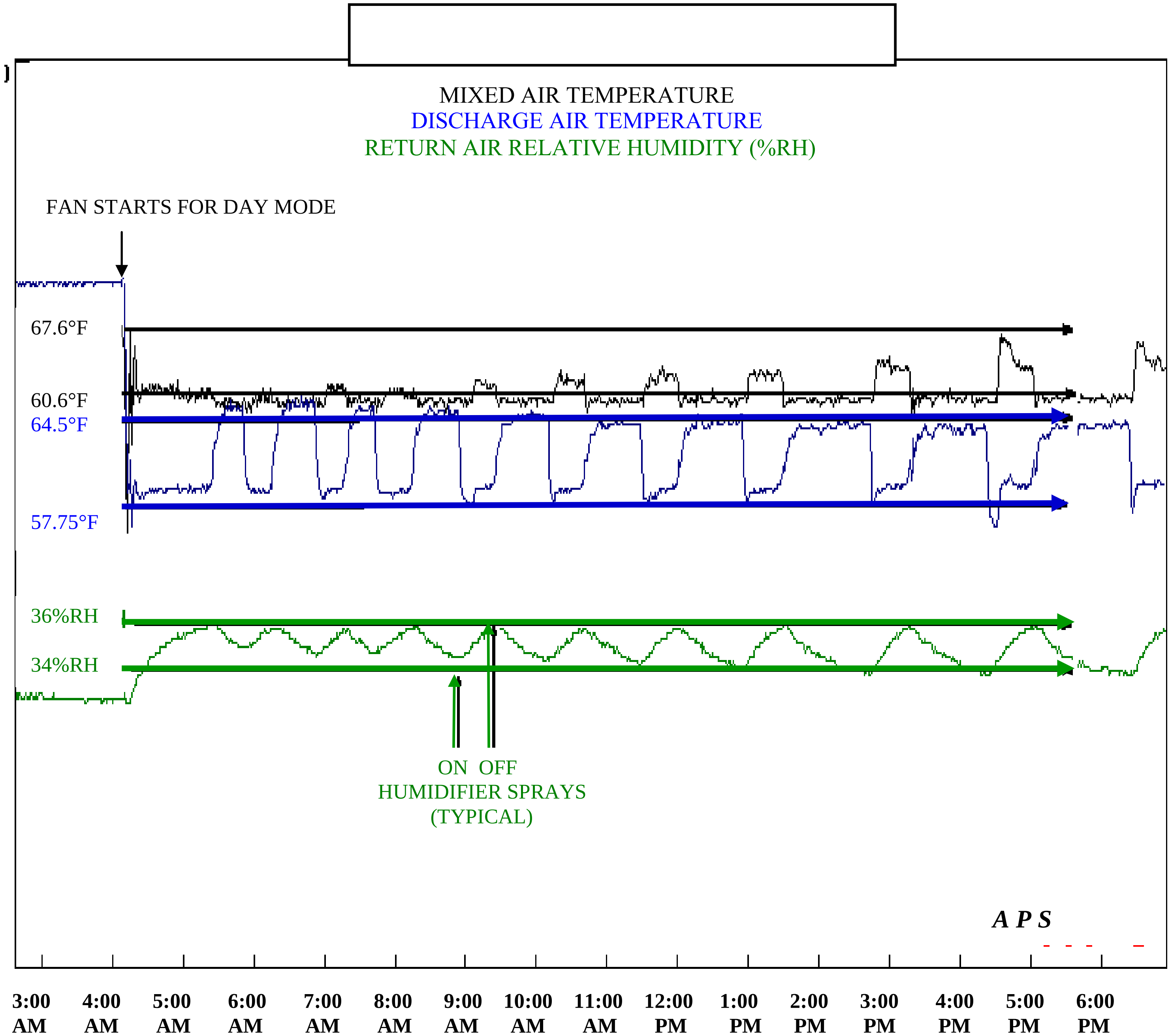
The black line illustrates the mixed air temperature, the blue line illustrates the discharge air temperature and the green line illustrates the return air relative humidity.

The graph demonstrates that the change tended to correct the problem. The mixed air temperature tended to rise when the humidity sprays were active; however, the discharge air temperature still dropped an unacceptable amount when the sprays were active.

We had to increase the sensitivity (lower the proportional band) of the discharge controller to allow that controller to respond with a reduced control point shift. (Lessen the drop in the discharge air temperature when the sprays were active.) The problem was that the existing sensitivity was set up to eliminate the cycling of the loop when controlling the cooling valve as per the original intent.

We altered the circuit via a turn down ratio relay to effectively allow the discharge air controller to have two different sensitivities: one when controlling the cooling valve and the other when it was controlling the mixing dampers. We first used a simulator to establish the exact sensitivity with which the controller was historically running. We increased the sensitivity to the highest level to minimize the drop in discharge air temperature when the sprays were active. We allowed the discharge controller to impact on the mixed air temperature based on evaporative cooling by sending its signal directly to a low selector, which also received a signal from the mixed air controller. The discharge controller also sent a signal to the turn down ratio relay, which allowed the cooling valve loop to operate with the same sensitivity as it had before we entered the job. The turn down ratio relay sent its signal to the chilled water valve.

GRAPH #2



MIXED AIR TEMPERATURE
DISCHARGE AIR TEMPERATURE
RETURN AIR RELATIVE HUMIDITY (%RH)
FAN STARTS FOR DAY MODE
67.6°F
60.6°F
64.5°F
57.75°F
36%RH
34%RH
ON OFF
HUMIDIFIER SPRAYS
(TYPICAL)
A P S
3:00 AM
4:00 AM
5:00 AM
6:00 AM
7:00 AM
8:00 AM
9:00 AM
10:00 AM
11:00 AM
12:00 PM
1:00 PM
2:00 PM
3:00 PM
4:00 PM
5:00 PM
6:00 PM

GRAPH #3

The black line illustrates the mixed air temperature, the blue line illustrates the discharge air temperature and the green line illustrates the return air relative humidity.

The graph three shows almost an inverse situation from graph one. On graph three the mixed air temperature rises significantly while the discharge experiences only a small drop in temperature when the humidity sprays are active.

This graph illustrates the more effective use of energy where the system automatically uses heat contained in the return air to provide the energy required to evaporate the moisture for humidification demands. Prior to the control change the humidity circuit forced the heating equipment in the building to provide the heat to compensate for the evaporization.

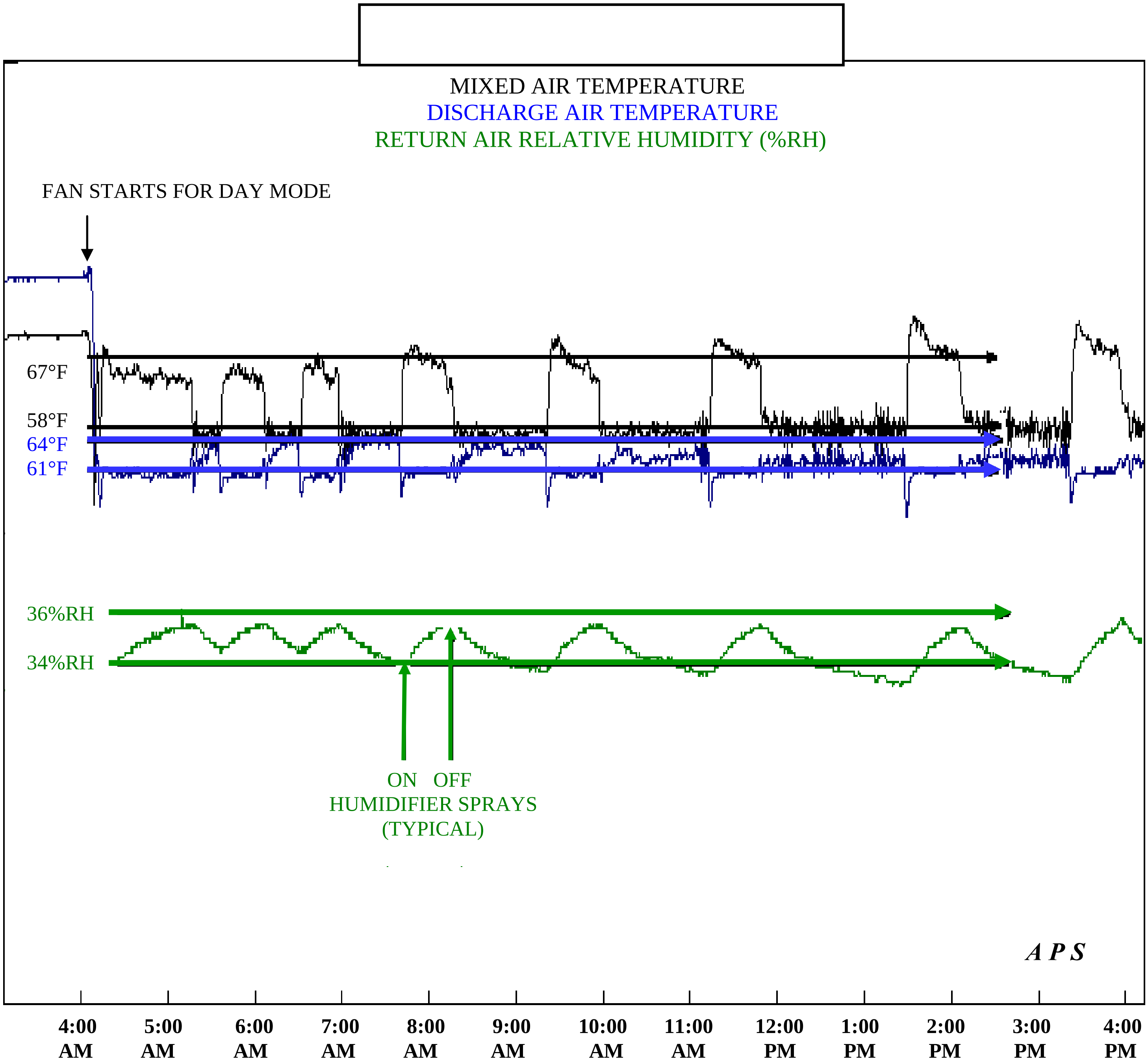
GENERAL

The evaporative cooling has to be compensated for with heat or the space will become too cold. The heat can come from:

1. the mechanical heating system as illustrated in graph one at great expense and pollution generation from extra combustion, or
2. the return air where the heat is already paid for, which saves money and reduces pollution by reducing combustion.

The discharge air temperature could have the drop in temperature eliminated completely by automatically resetting the discharge controllers set point as required, but the range of control experienced with the sprays active and inactive was well within the acceptable limits.

GRAPH #3



MIXED AIR TEMPERATURE
DISCHARGE AIR TEMPERATURE
RETURN AIR RELATIVE HUMIDITY (%RH)
FAN STARTS FOR DAY MODE
67°F
58°F
64°F
61°F
36%RH
34%RH
ON OFF
HUMIDIFIER SPRAYS
(TYPICAL)
A P S
4:00 AM
5:00 AM
6:00 AM
7:00 AM
8:00 AM
9:00 AM
10:00 AM
11:00 AM
12:00 PM
1:00 PM
2:00 PM
3:00 PM
4:00 PM

ACTUAL GAS CONSUMPTION AMOUNTS FOR THE ETOBICOKE EDUCATION CENTRE ILLUSTRATING THE IMPACT OF EVAPORATIVE COOLING AND LOAD ANALYZING TECHNIQUES

READING DATE	GAS CONSUMPTION (X 100)
AUG. 29/75	1,324.8
SEPT. 30/75	2,069.8
OCT. 30/75	1,965.1
DEC. 01/75	1,776.8
DEC. 30/75	1,930.1
JAN. 30/76	2,806.9
FEB. 27/76	2,308.1
MAR. 30/76	2,624.8
APRIL 29/76	1,749.6
MAY 28/76	1,676.1
JUNE 29/76	1,053.2
JULY 28/76	924.6
JAN. 26/77	9,168.4
FEB. 21/77	1,276.8
MAR. 22/77	1,192.7
APRIL 21/77	613.6
MAY 20/77	522.5
JUNE 21/77	609.2
JULY 21/77	683.7
AUG. 22/77	1,066.6
SEPT. 22/77	1,156.2
OCT. 21/77	1,328.5
NOV. 22/77	1,224.2
DEC. 21/77	815.1
JAN. 20/78	927.5
FEB. 20/78	1,078.8
MAR. 22/78	920.5
APRIL 21/78	771.2
MAY 19/78	526.8

WHEN COMPARING THE GAS CONSUMPTION FOR THESE TWO SETS OF MONTHS, ONE FINDS JUNE AND JULY REQUIRED 13.5% MORE GAS Before THE CHANGES, THAN DECEMBER AND JANUARY REQUIRED AFTER THE CHANGES.

INITIAL CONSERVATION DESIGN EMPLOYING EVAPORATIVE COOLING EFFECT OF THE HUMIDITY WATER SPRAYS. OTHER TECHNIQUES WERE IMPLEMENTED SIMULTANEOUSLY. (JAN. 26/77 – JULY 21/77)

THE WATER SPRAYS WERE REPLACED WITH STEAM HUMIDIFIERS CAUSING THE COOLING EFFECT TO BE LOST. (AUG. 22/77 – NOV. 22/77)

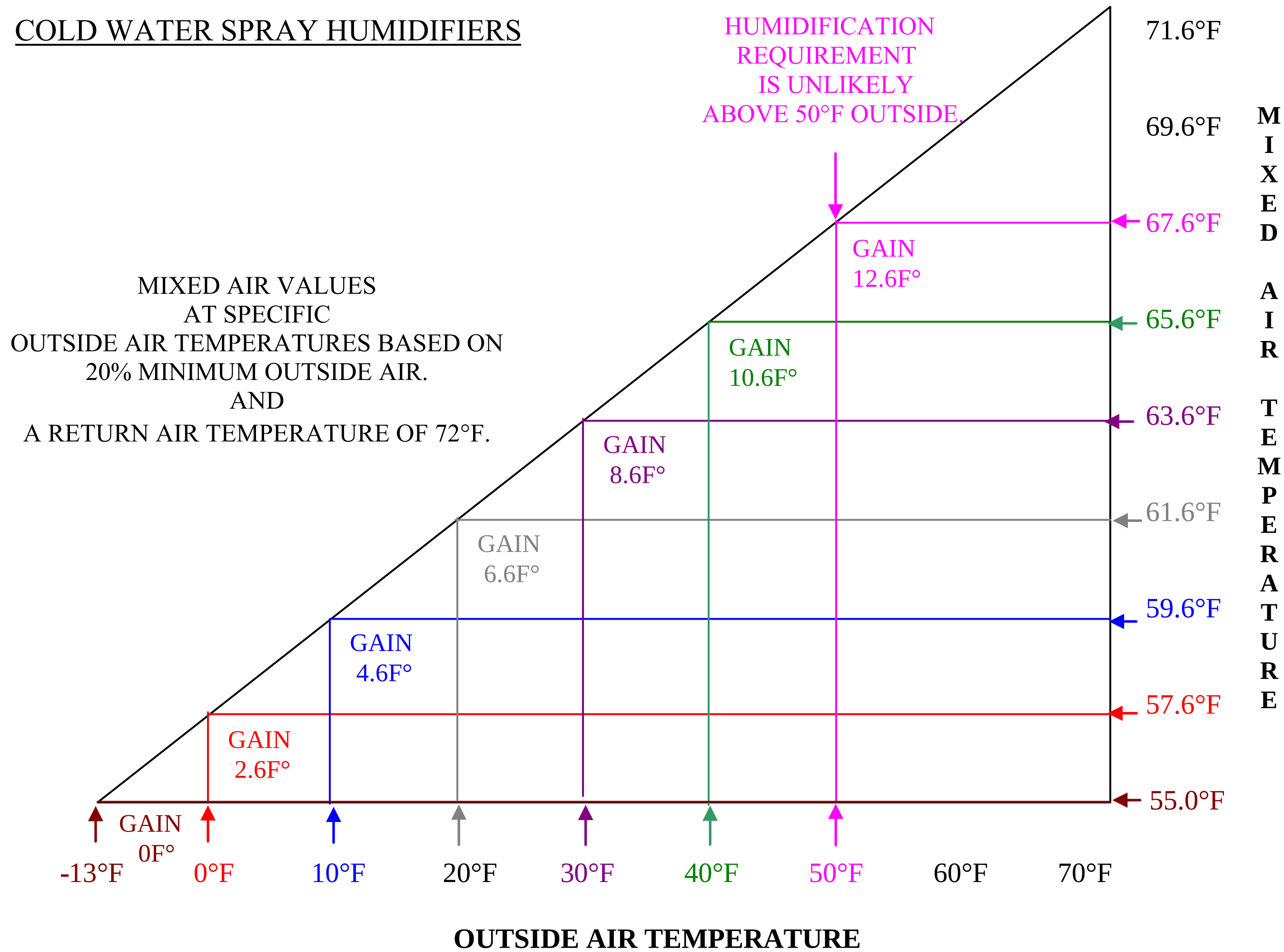
AN ALTERNATE CONSERVATION CONTROL TECHNIQUE WAS DESIGNED AND INSTALLED. LOAD ANALYSING FROM THE WARMEST SAMPLE THERMOSTAT RE-ESTABLISHED THE CONSUMPTION REDUCTION. (DEC. 21/77 – MAY 19/78)

52.6% REDUCTION WITH EVAPORATIVE COOLING TECHNIQUE.

60% REDUCTION WITH ALTERNATE TECHNIQUE.

RETURN AIR THERMAL RECOVERY VIA

COLD WATER SPRAY HUMIDIFIERS



NOTES:

-1- The mixed air set point is normally 55°F. Proportional controllers will droop below 55°F in cold weather conditions increasing the potential heat recovery.

-2- If the outside air is -13°F, the return air is 72°F and the required quantity of minimum fresh air is 20%, the mixed air temperature will be 55°F. At this outside air temperature, or below, there is no savings available, via sequencing logic with the humidity sprays as the system is on minimum ventilation only.

-3-The energy savings available, when the sprays are active, at outside air temperatures above -13°F, gradually increases as the outside air temperature rises.

Example:

At 20°F outside the control logic will automatically raise the mixed air from its normally controlled value of 55°F to 61.6°F. The humidification system will use the energy contained in the 6.6F° gain to evaporate the moisture in the sprays.

-4- The heat recovered, during spray activation, from the return air per 1000 CFM =

(Actual mixed air temperature - 55°F) x 1.08 x 1000 = BTU/HR/1000 CFM

-5- When the fan system is on 100% outside air at 55°F and 100%RH the air will drop to 35%RH when the temperature rises to 72°F, if no other moisture is introduced into the air.

ENTHALPY COMPARATOR LOOP

PNEUMATIC CONTROL

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CONTENTS

DEFINITIONS

DRY BULB TEMPERATURE

A measurement of sensible heat, usually expressed in degrees Fahrenheit or degrees Celsius, measured with a thermometer.

WET BULB TEMPERATURE

The lowest temperature that evaporation of water causes, measured using a thermometer with a wet wick over its bulb.

RELATIVE HUMIDITY

The percentage of moisture contained in air relative to the air saturated at that dry bulb temperature.

ENTHALPY

The total heat content of air, considering the dry bulb temperature and the relative humidity. (Every pound of evaporated water contains 970 BTU of latent heat, as well as its sensible heat.)

PREFACE

Enthalpy comparison in HVAC systems presents a significant opportunity in control logic, where society may reduce electrical consumption. This applies to mechanical cooling systems with outdoor air, return air and exhaust air dampers, where the controls alter the dampers from full outdoor air to minimum ventilation, based on calculations that the outdoor air will consume more cooling energy than the return air of the building.

Air at 70°F and 100% RH contains 34 BTU/lb of dry air and at 70°F and 15% RH contains 19 BTU/lb of dry air. (Environment Canada, last year, reported that Calgary experienced 100% RH as the outdoor high and 15% RH as the outdoor low during the mechanical cooling season.) These conditions present a 78.9% BTU increase from the low to the high regarding air at 70°F. Return air 75°F at 40% RH contains 26 BTU/lb of dry air, which is midpoint considering the high and low, Altering the damper position to minimum ventilation, based only on a temperature of 70°F, which occurs in many systems, would be a bad idea if the outdoor relative humidity is at the lowest recorded value and a good idea if the outdoor relative humidity is at the highest recorded value. The system must know the relative humidity level of both air sources and calculate the actual enthalpy of each to make a proper decision.

This report illustrates our control circuit that calculates the enthalpy of both the return air and outdoor air. The circuit compares the two enthalpy values and selects the lower value as the air stream passing through the mechanical cooling coil. We believe that installation and set up can be accomplished by a pneumatics technician with average ability.

A couple of scenarios where enthalpy logic may not apply are:

-1- Some chillers will create an artificial load, internally matching a lessening load from the fan systems. The chiller's energy consumption will not change at low load levels.

-2- A unit with Freon coils controlled from the space may tend to run the compressors more frequently under certain conditions with enthalpy comparison logic.

For example:

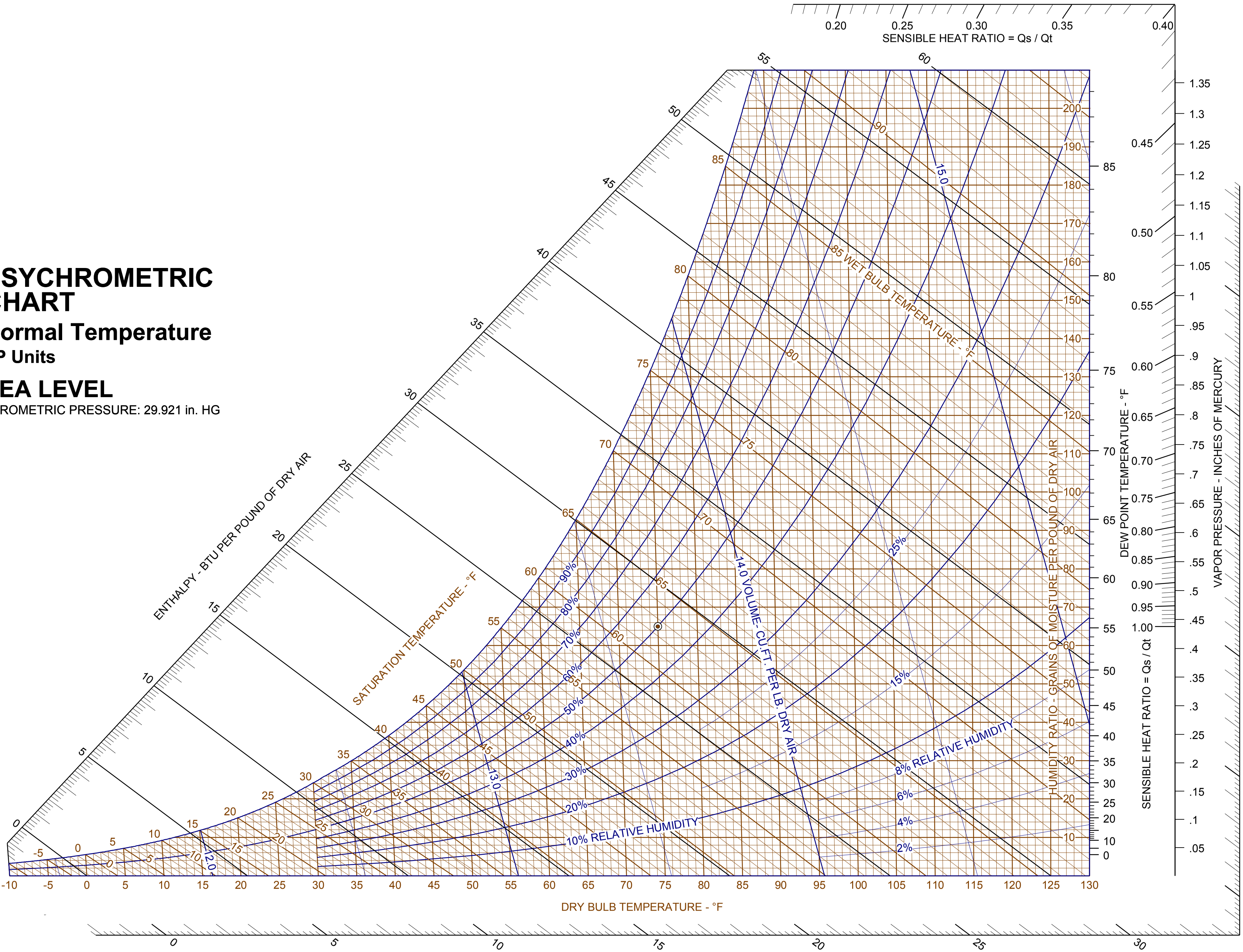
The conflict is that the dampers are positioned based on enthalpy comparison and the mechanical cooling is based only on space dry bulb temperature.

If the outdoor air is 80°F at 15% RH, the enthalpy is 23 BTU/lb of dry air and if the return air is 73°F and 40% RH the enthalpy will be 25 BTU/lb of dry air. The enthalpy comparator will select the outdoor as the air stream passing through the cooling coil.

When the mechanical cooling is active, the outdoor 80°F at 15%RH containing 23 BTU/lb of dry air is the more efficient air stream passing through the cooling coil; however, when the mechanical cooling is not active, the outdoor air will raise the room temperature restarting the cooling in a shorter time than the cooler return air.

Consider allowing enthalpy selection when the mechanical cooling is active and use the air stream with the cooler sensible dry bulb temperature during periods when the mechanical cooling is inactive.

The only other pneumatic logic circuit, currently commercially available, to my knowledge, is the Johnson Control N9000 Logic Center. We tested this device's performance and our data and Johnson Control's specification sheet indicate that the N9000 is not a true enthalpy comparator as per pages nine and ten in this report.



PSYCHROMETRIC CHART

Normal Temperature

I-P Units

SEA LEVEL

BAROMETRIC PRESSURE: 29.921 in. HG

ISSUE

Many systems revert the mixing dampers back to minimum ventilation when the outside air exceeds 70 ^{o}F dry bulb temperature, attempting to reduce the cooling load at the cooling equipment. Environment Canada information records the average lowest Toronto outdoor relative humidity at 32.5% RH for June, July, August and September 2010. They record the average high at 100% RH and the average mean at 73.61% RH for those months during 2010, in Toronto.

Air 70 ^{o}F at 32.5% RH contains 22.32 BTU/lb of dry air while air 70 ^{o}F at 100% RH contains 34.08 BTU/lb of dry air. Air 70 ^{o}F at 100% RH (average high) contains 52.7% more heat energy than air 70 ^{o}F at 32.5% RH (average low). Switching from free cooling to no free cooling, based only on temperature is often an incorrect decision.

The relative humidity, as well as the dry bulb temperature must both be considered for both the return air as well as the outdoor air when determining the lesser cooling load. This combination of humidity and temperature is called enthalpy. The psychrometric chart on page two illustrates this relationship.

EXAMPLE: Pick the point where the temperature and relative humidity of your air stream intersect and follow a line diagonally up to the left, parallel with the other lines, to the enthalpy line that presents values from 0 BTU/lb of dry air to 60 BTU/lb of dry air. The point you cross presents the BTU's/lb of dry air your air stream contains.

GOAL

Use the least amount of mechanical cooling in the HVAC system, while satisfying the requirements of the occupied space.

REQUIREMENTS

-1-Psychrometric chart.

-2- Transmitters for sensing both the return air and the outdoor air relative humidity and dry bulb temperatures.

-3- Two reset receiver controllers producing two pneumatic, varying signals, ranging from 3 PSIG to 15 PSIG, representing a change in enthalpy from 21 BTU/lb of dry air to 39 BTU/lb of dry air for both the outdoor air and return air.

-4- Custom make two overlays for 3# to 15# transmission gauges, ranging from 21 BTU/lb of dry air to 39 BTU/lb of dry air. (A CD/DVD labeling program can be used.)

-5- A relay to subtract the return air enthalpy signal from the outdoor air enthalpy signal.

-6- A snap acting air switching valve or receiver controller acting as the determining point to select the air stream with the lesser BTU content.

-7- Install components as per drawing.

CALIBRATION PROCEDURE

-1- Calibrate all transmission gauges to be accurate at nine PSIG with a certified test gauge or one tested as accurate.

-2- Calibrate the temperature and relative humidity transmitters to be accurate relative to their respective transmission gauges.

-3- Temporarily install gradual switches allowing simulation of varying temperature and relative humidity values for both return and outdoor air.

-4- For each of the two direct acting, reset receiver controllers, tube the temperature signal into the primary port and the relative humidity into the reset port with direct re-adjustment. (This means that an increase on either port signal will cause an increase in the receiver controllers' out put signals.)

-5- -a- Take one receiver controller and set the relative humidity value at 40% RH with your manual gradual switch and leave it at 40% RH, while you address the temperature side of the receiver controller.

-b- Find the proportional band (sensitivity or gain) setting that causes an output signal change from 23.70 BTU/# of dry air to 29.10 BTU/#of dry air on your enthalpy indication gauge as you change the temperature from 70 ^{o}F to 80 ^{o}F.

-c- Set the temperature gauge to 70 ^{o}F and leave the proportional band at the setting determined in "b".

-d- Find the authority (reset) setting that causes an output signal change from 23.70 Btu/lb of dry air to 28.10 BTU/lb of dry air on your enthalpy indication gauge when you change the relative humidity from 40% RH to 60% RH.

-e- Check the enthalpy values at the extremes on the chart and several points in the middle. You can use the psychrometric chart as well.

-6- Remove your manual simulation gradual switches and re-connect the temperature and relative humidity transmitters as required.

We did not expect the system values to be exactly the same as the psychrometric chart, as we are blending a linear signal (temperature) with a non-linear signal (relative humidity) into one signal. As you should see during checking the system values produced are plus or minus one BTU/lb of dry air.

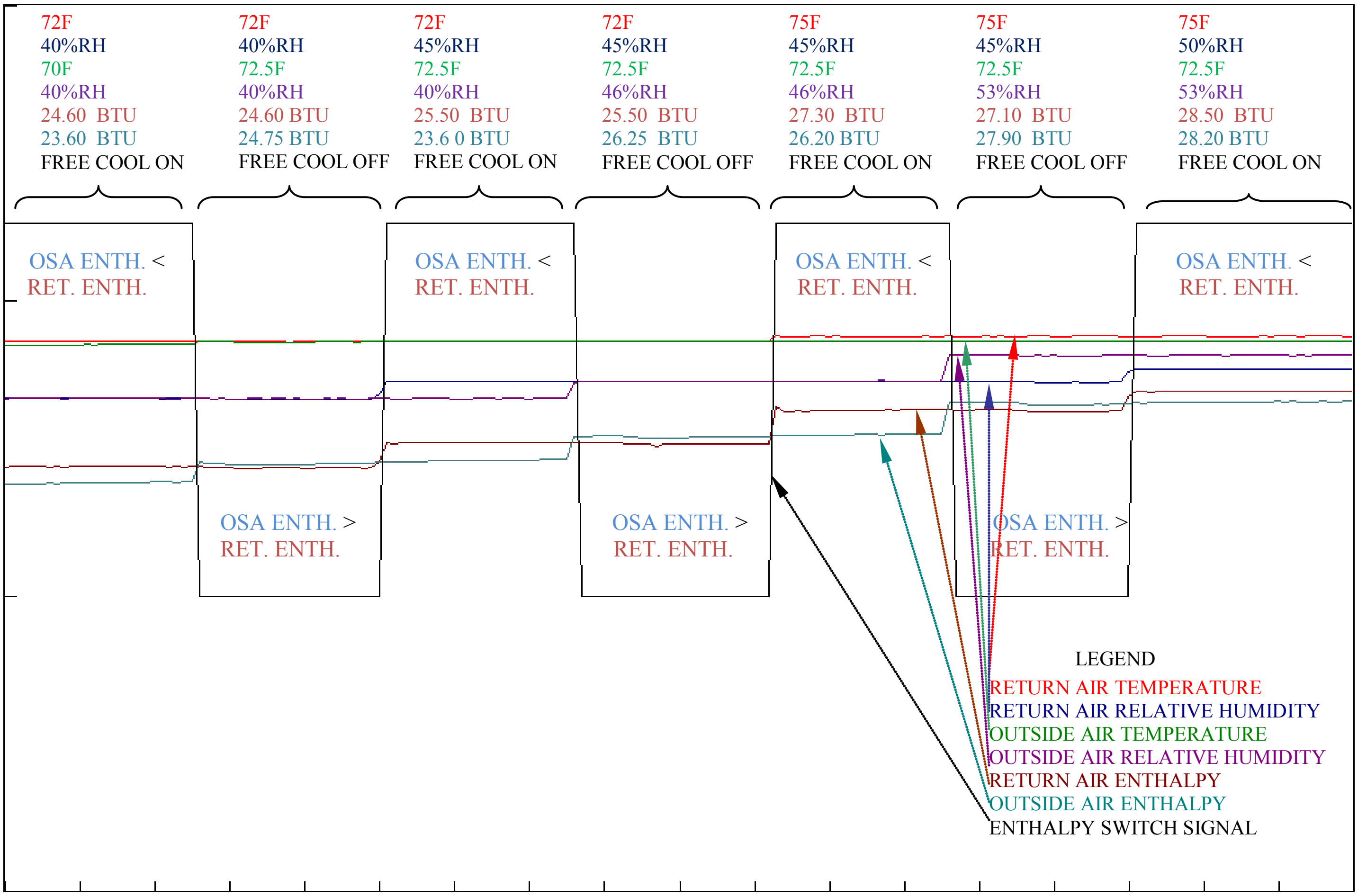
ENTHALPY VALUES

(PSYCHROMETRIC CHART IN BLACK. (BTU/LB DRY AIR)
(VALUES PRODUCED VIA CONTROL LOOP IN BLUE. BTU/LB DRY AIR)

TEMP. °F		BTU	BTU	BTU	BTU	BTU		
80 (10.20#)		28.79 29.10	30.00 29.90	31.23 31.10	32.45 32.30	33.68 33.80		
79 (9.96#)		28.23 28.40	29.41 29.60	30.59 30.70	31.77 32.30	32.96 33.30		
78 (9.72#)		27.68 27.40	28.82 28.80	29.96 30.00	31.10 31.10	32.25 32.30		
77 (9.48#)		27.14 27.30	28.24 28.20	29.34 29.60	30.45 30.60	31.56 32.00		
76 (9.24#)		26.61 26.60	27.67 27.90	28.74 29.10	29.81 30.40	30.88 30.00		
75 (9.00#)		26.09 26.00	27.12 27.10	28.15 28.30	29.18 29.50	30.21 30.00		
74 (8.76#)		25.58 25.50	26.57 27.20	27.56 27.80	28.56 29.20	29.56 30.50		
73 (8.52#)		25.07 25.70	26.03 26.40	26.99 27.70	27.95 28.70	28.92 30.00		
72 (8.28#)		24.58 24.60	25.50 25.80	26.43 27.10	27.36 28.10	28.29 29.20		
71 (8.04#)		24.09 24.20	24.98 25.20	25.87 26.30	26.77 27.30	27.67 28.40		
70 (7.80#)		23.61 23.70	24.47 24.60	25.87 25.90	26.20 27.10	27.06 28.10		
	RH%	40%	45%	50%	55%	60%		

Temperature transmitter is 0°F to 100°F.
Humidity transmitter is Robertshaw 0% RH to 100% RH.
Branch signal is 21 BTU/lb dry air (3 PSIG) to 39 BTU/lb dry air (15 PSIG).
Note: As the dry bulb temperature drops and the relative humidity rises, the BTU values remain within one percent variation. There are more BTU's in the air at 71° F than at 76° F.

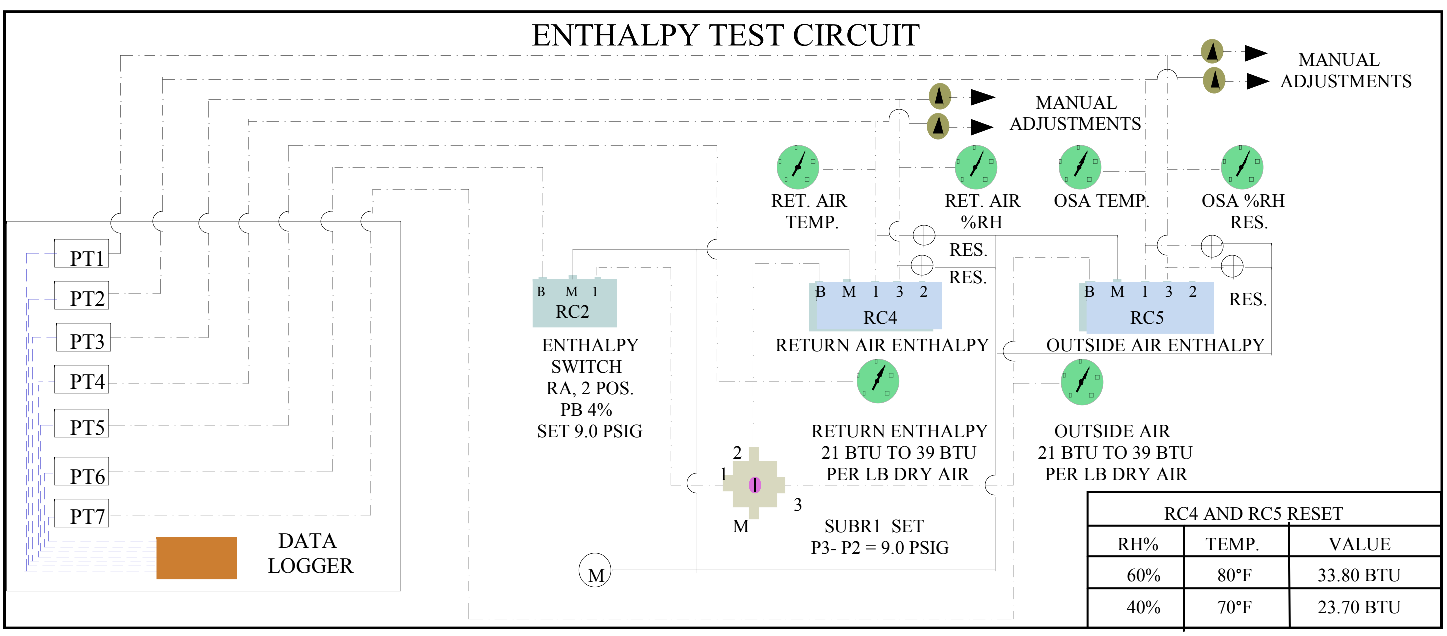
ENTHALPY COMPARATOR PERFORMANCE GRAPH



The graph illustrated above presents the values regarding temperature, relative humidity and enthalpy with the presented test values colour matched to the graph colours.

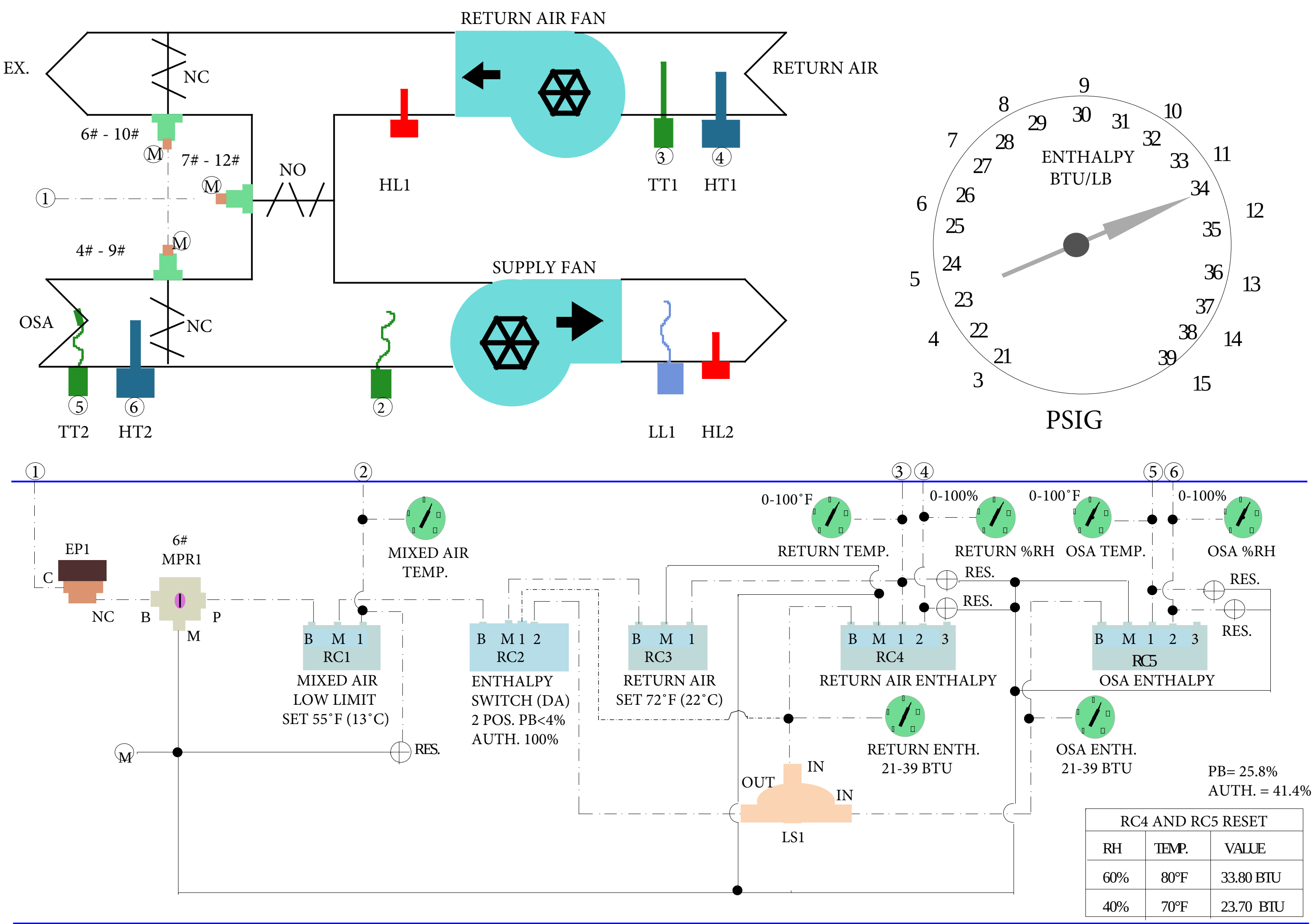
The control circuit illustrated below allowed simulation of various relative humidity and temperature values, causing variations in the enthalpy values regarding both the return air and the outside air. The values were recorded via pressure transducers and collected through a data logger.

The conclusion of the enthalpy comparison, allowing selection of the air stream with the lesser enthalpy value, is represented by the black line.



RC4 AND RC5 RESET		
RH%	TEMP.	VALUE
60%	80°F	33.80 BTU
40%	70°F	23.70 BTU

The Kreuter subtraction relay became obsolete; therefore, this circuit was developed allowing enthalpy comparison.



RC4 AND RC5 RESET		
RH	TEMP.	VALUE
60%	80°F	33.80 BTU
40%	70°F	23.70 BTU

SEQUENCE OF OPERATION

When the fan is off, solenoid valve (EP1) is de-energized. The outside air and exhaust air dampers close and the return air damper opens.

Receiver controller (RC5) senses the outside air humidity and temperature via transmitters (HT2) and (TT2) respectively. RC5 produces a signal representing the outside enthalpy as the humidity and temperature vary according to the reset schedule. RC4 senses the return air humidity and temperature via transmitters (HT1) and (TT1). RC4 produces a signal representing the return air enthalpy as the humidity and temperature varies according to the reset schedule. Both RC4 and RC5 send their signals to low selctor (LS1). RC4 also sends its signal to RC2.

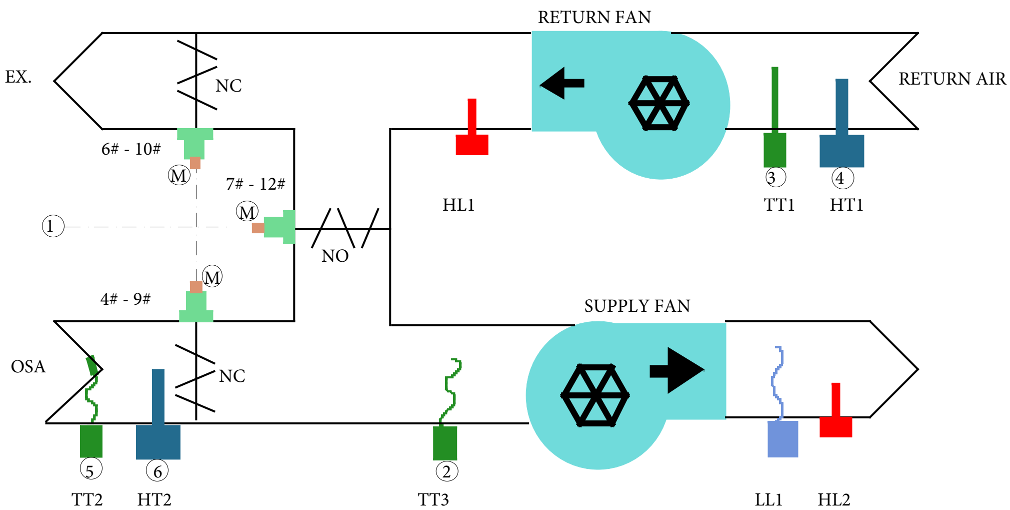
RC2 receives its main air from receiver controller (RC3), which senses the return air temperature via TT1.

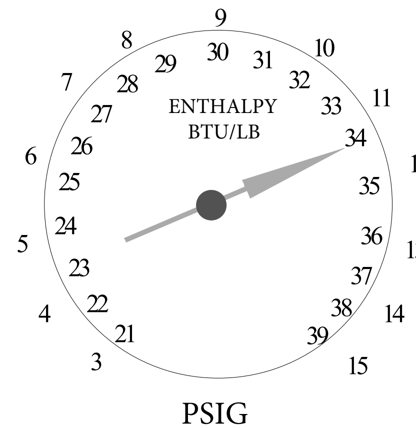
RC2's signal provides the main air for receiver controller (RC1) which senses the mixed air temperature via transmitter (TT3). RC1 sends its signal to minimum positioning relay (MPR1), which sends its signal through EP1 to modulate the mixing dampers.

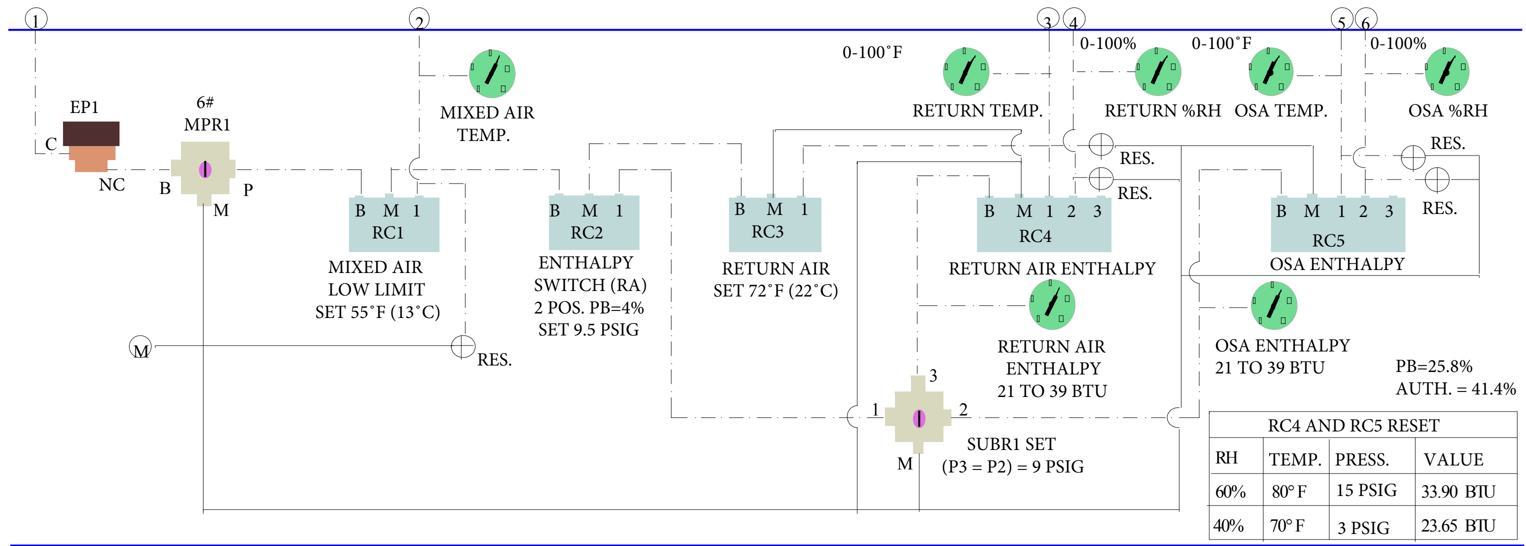
The safeties will shut down the fan when the temperature exceeds their set points.

NOTE: The proportional band and authority may have different values, depending on the controller manufacturer.

<u>Alternate relay to achieve circuit intent</u>.







RC4 AND RC5 RESET			
RH	TEMP.	PRESS.	VALUE
60%	80° F	15 PSIG	33.90 BTU
40%	70° F	3 PSIG	23.65 BTU

SEQUENCE OF OPERATION

When the fan is off, solenoid valve (EP1) is de-energized. The outside air and exhaust air dampers close and the return air damper opens.

Receiver controller (RC5) senses the outside air humidity and temperature via transmitters (HT2) and (TT2) respectively. RC5 produces a signal representing the outside enthalpy as the humidity and temperature vary according to the reset schedule. RC4 senses the return air humidity and temperature via transmitters (HT1) and (TT1). RC4 produces a signal representing the return air enthalpy as the humidity and temperature varies according to the reset schedule. Both RC4 and RC5 send their signals to subtraction relay (SUBR1). SUBR1 subtracts the return air enthalpy signal from the outside air enthalpy signal. SUB1's output signal is 9 PSIG when both input signals are equal. If the outside air enthalpy is greater than the return air enthalpy SUB1's output signal will start to rise by the difference that RC5's signal is greater than RC4's signal indicating that the outside enthalpy is greater than the return air enthalpy.

SUB1's signal goes to receiver controller (RC2), which prevents free cooling when the outside air enthalpy is greater than the return air enthalpy.

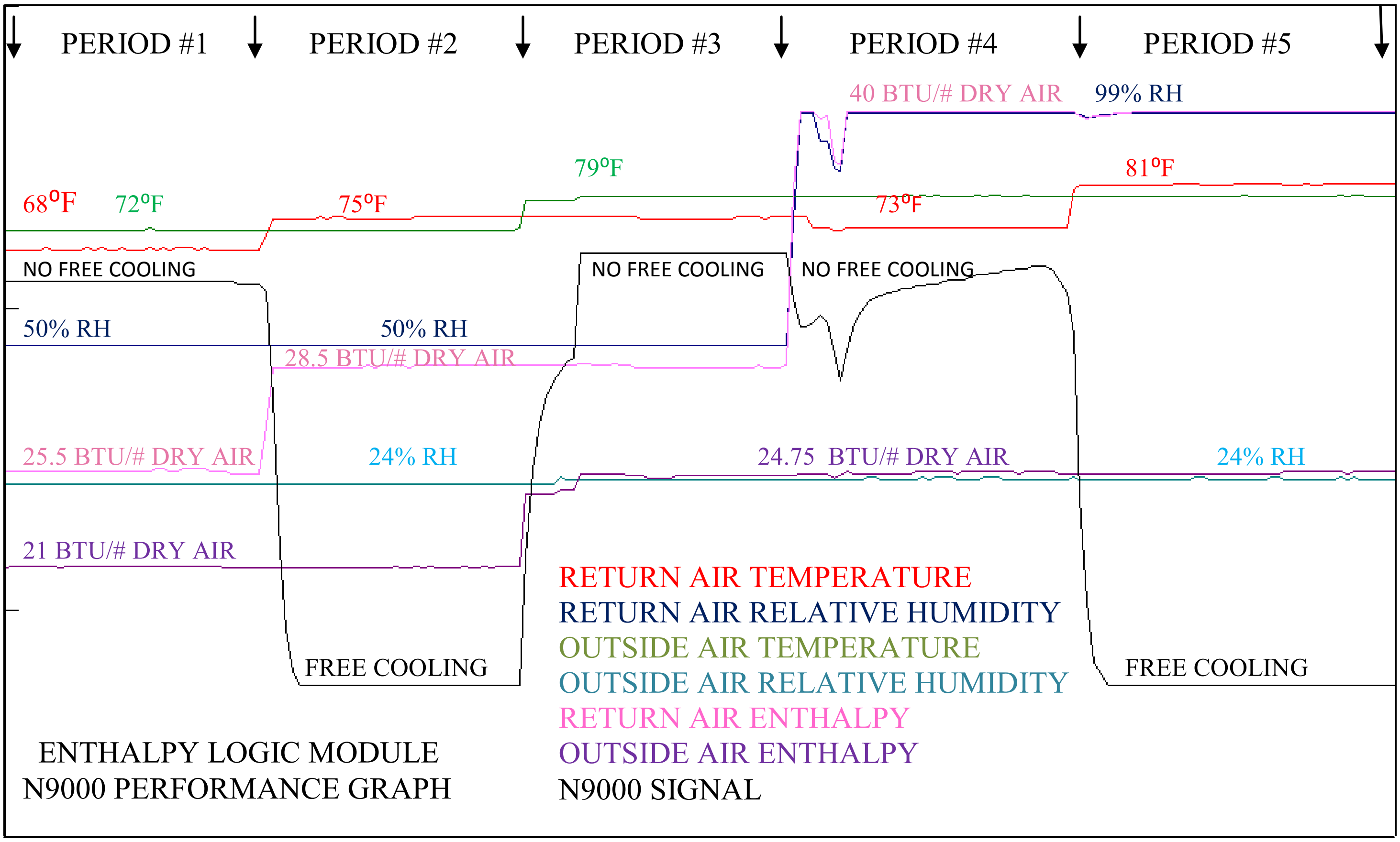
RC2 receives its main air from receiver controller (RC3) which senses the return air temperature via TT1.

RC2's signal provides the main air for receiver controller (RC1) which senses the mixed air temperature via transmitter (TT3). RC1 sends its signal to minimum positioning relay (MPR1), which sends its signal through EP1 to modulate the mixing dampers.

The safeties will shut down the fan when the temperature exceeds their set points.

NOTE: The proportional band and authority may have different values, depending on the controller manufacturer.
The subtraction relay may be obtained from Kreuter Marketing (RCC-1508).
Kreuter Marketing manufactures the receiver controllers used in the testing. (CCC-1002)

JOHNSON CONTROL N9000 ENTHALPY CENTRE



The Johnson Control N-9000 Enthalpy Logic Centre's specification sheet states:

-1- "The N9000 imposes the lowest cooling load on the mechanical cooling equipment."

-2- "When outside air enthalpy is greater than return air enthalpy or when the outside air temperature is greater than the return air temperature, the N9000 provides a maximum output signal of 10 PSIG or greater. This signal returns the outside air damper to its minimum position. Note: If the outside air temperature is greater than the return air temperature, the output signal is at maximum and will not be reduced by a change in the humidity signal.

When outside enthalpy is less than return air enthalpy *and* the outside air temperature is less than the return air temperature, the N9000 provides a 0 PSIG output signal. This signal places the dampers under control of the system controller to obtain free cooling."

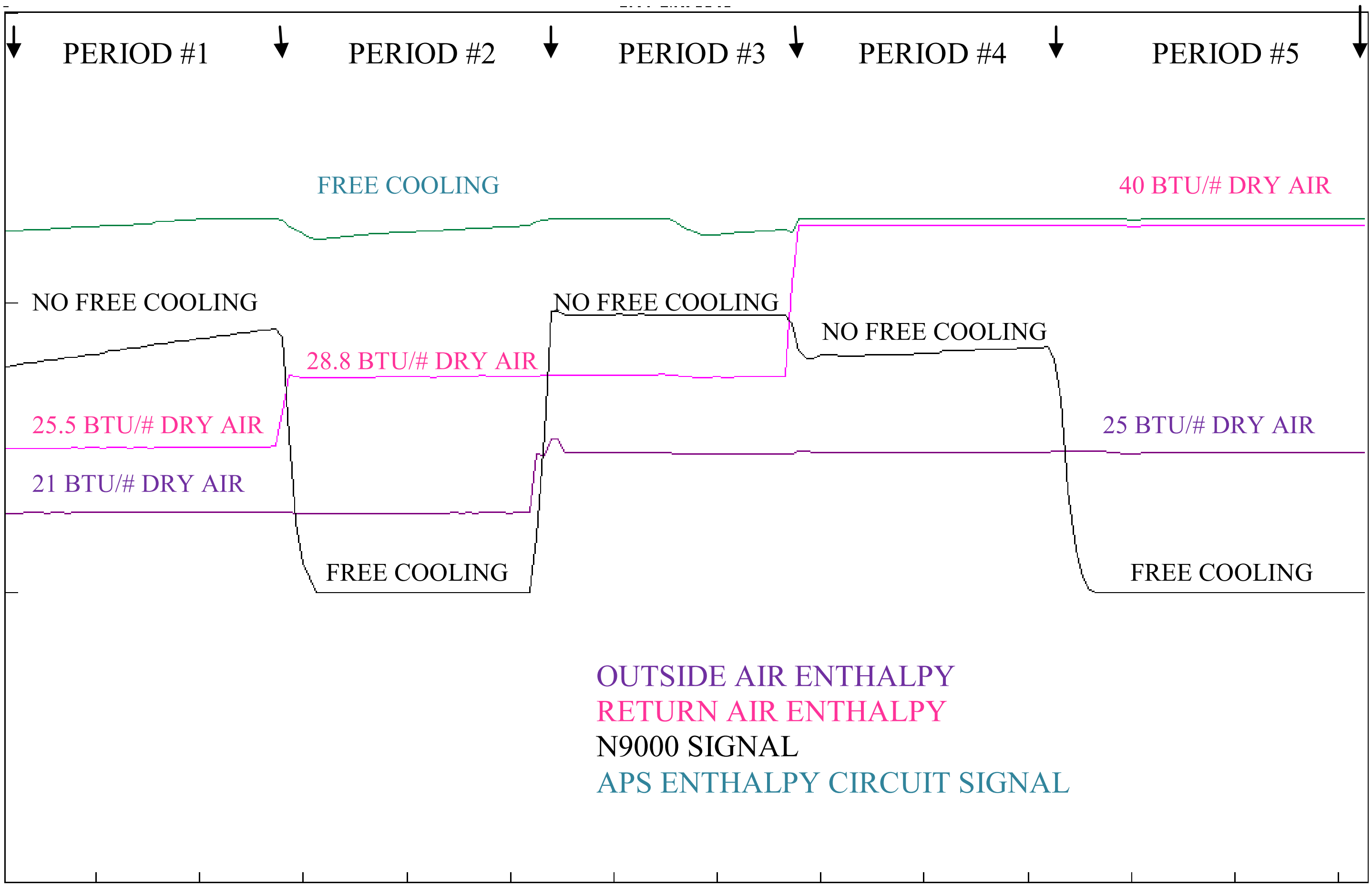
-3- "By enthalpy and sensible heat comparison of both outside and return air conditions, the N9000 provides the most efficient use of free cooling and thus true economizer operation."

These quotes are contradictory, as they state the N9000 will select the air stream with the lesser cooling load. They also state the N9000 will switch based on dry bulb temperature and humidity will have no effect. At times the N9000 positions the dampers solely on a sensible heat comparison and ignores the humidity level, as per quote number two; therefore, ignores the enthalpy levels.

The performance graph above illustrates the N9000's signal under varying humidity, thermal and enthalpy levels. The enthalpy levels were obtained via the enthalpy circuit which is the subject of this paper. During the five periods of testing, the outside air enthalpy was always lower than the return air enthalpy; however, the N9000 determined that free cooling was appropriate for two periods and no free cooling was appropriate for three periods. A true enthalpy controller would have kept the system on fresh during all five periods, as the outdoor enthalpy was lower than the return air during the five periods.

We suggest that existing N9000 installations be altered to the use the circuit presented in this paper, if enthalpy comparison is a valid feature in those buildings.

JOHNSON CONTROL N9000 AND APS ENTHALPY PERFORMANCES



The above graph compares the performance of the Johnson Control N9000 Enthalpy Logic Module and the APS enthalpy comparison circuit.

The return air enthalpy and outdoor enthalpy values were determined via the APS enthalpy comparison circuit illustrated on pages 8-111 and 8-113 in this report. The accuracy relative to the standard psychrometric chart on page 8-107 is presented on page 8-110.

As illustrated above the outdoor enthalpy was lower than the return air enthalpy during the whole test. The APS enthalpy comparison circuit correctly determined that the system should remain on free cooling from outdoor air. The Johnson Control N9000 Enthalpy Logic Module determined that free cooling was appropriate for two periods and no free cooling was appropriate for three periods.

The APS enthalpy circuit compares enthalpy of the return air and the outdoor and these are the only factors determining the switching point. The Johnson Control N9000 Logic Module literature states " When outside air enthalpy is greater than return air enthalpy or when the outside air temperature is greater than the return air temperature, the N9000 provides a maximum output signal of 10 PSIG (70 kPa) or greater. Note: if the outside air temperature is greater than return air temperature, the output signal is at maximum and will not be reduced by a change in the humidity signal."

<u>SUMMARY</u>

Many systems use only outdoor dry bulb temperature when determining the point when outdoor air requires more mechanical cooling than the return air maintaining comfort in the occupied space. At this point the dampers revert back to mainly return air, with outdoor providing only minimum ventilation. The change-over dry bulb temperature is often seventy degrees Fahrenheit. 70°F air at 100% RH contains 78.9% more heat energy than 70°F air at 15% RH. The system must sense relative humidity and dry bulb temperature in both the return air and the outdoor air and calculate the actual enthalpy in determining which air stream is more economical for mechanical cooling.

SIEMENS' enthalpy transmitter, part number 184-0101 and enthalpy logic module, 243-0043 are no longer available. The Barber Colman enthalpy transmitter, part number HKS8065, and enthalpy logic module, part number AK-52101, are no longer available. We did not test the performance of these components through their whole range, but spot checking them over time, during service, we believe that they functioned properly as enthalpy comparators.

Honeywell made an enthalpy comparator, part number HP973A, which is an obsolete item now. We have never seen one in the field.

Johnson Controls Enthalpy Logic Centre, part number N9000, is still commercially available. We do not consider it a true enthalpy comparison circuit, as the dry bulb temperature can cause the unit to switch from free cooling, while a change in the relative humidity will not affect that decision. Enthalpy is a combination of dry bulb temperature and relative humidity; therefore, both must continually have an impact on the decision to choose return air or outdoor air as the primary component of the air passing through the mechanical cooling equipment.

We suggest that each circuit, whether pneumatic or DDC, claiming enthalpy comparison, be checked with simulated temperatures and relative humidity levels. The operator, using the psychrometric chart on page 8-107, can select various enthalpy values for the return air and the outdoor air commanding particular dry bulb temperatures and relative humidity values. The positioning of the mixed air dampers will demonstrate the true logic of the enthalpy system.

NOTES

ENERGY OPPORTUNITY

REGARDING

MECHANICAL COOLING

VIA DX REFRIGERATION

NOTE:

This has not been tested in practical use to date Feburary 17, 2023. Logic to address relative humidity variation in the occupied space may become a requirement.

Application is the full responsibility of those using this consideration.

APS requires no payment for the use of this concept.

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CONTENTS

DEFINITIONS

DRY BULB TEMPERATURE

A measurement of temperature usually expressed in degrees Fahrenheit or degrees Celsius, measured with a thermometer.

WET BULB TEMPERATURE

The lowest temperature that evaporation of water causes, measured using a thermometer with a wet wick over its bulb.

RELATIVE HUMIDITY

The percentage of moisture contained in air relative to the air saturated at that dry bulb temperature.

ENTHALPY

The total heat content of air, considering the dry bulb temperature and the relative humidity. (Every pound of evaporated water contains 970 BTU of latent heat, as well as its sensible heat.)

SENSIBLE HEAT

Heat added or taken from a substance that changes the dry bulb temperature of that substance.

LATENT HEAT

Heat absorbed or discharged from a substance as that substance changes state from solid to liquid, liquid to solid, gas to liquid or liquid to gas. Latent heat loss or gain does not change the dry-bulb temperature of the substance.

VALUES

-1- 7,000 grains of moisture = one pound
-2- 13.35345 cubic feet of air = one pound
-3- 970 BTU's are required to evaporate one pound of water
-4- 970 BTU's are produced in condensing one pound of water vapour
-5- .018 BTU's are required to raise one cubic foot of air one Fahrenheit degree

EXECUTIVE SUMMARY

This paper addresses a means of reducing the total annual running hours of mechanical cooling compressors in DX applications, while maintaining exactly the same dry bulb temperature in the occupied space.

The subject systems have outdoor, exhaust and return air dampers with an enthalpy comparison economizer circuit. During warm weather, typically these systems position the dampers at either 100% outdoor air or primarily return air with minimum outdoor air, addressing ventilation requirements. The circuit selects the air stream with the lesser total heat content (enthalpy) to pass through the refrigeration coil. This is the most efficient means of mechanical cooling, when the refrigeration compressor is running and tends to reduce the run time of the refrigeration compressor. If the particular unit runs properly with potentially low load conditions, this is the correct operation of the DX mechanical cooling, minimizing compressor run time.

With the compressor run time minimized, we now have to consider the duration of the rest time, when the refrigeration compressor is inactive, but the fan is running. The magnitude of both the run time and the rest time determine the annual run time of the refrigeration compressor.

When the controlled space cools to the bottom of the cooling dry bulb differential range, the refrigeration compressor shuts down, the fan continues running, providing the required ventilation, respecting building codes. The dampers conventionally maintain their position based on the air stream with the lesser enthalpy.

The cycling of the refrigeration compressor is based on dry bulb temperature of the occupied space, not enthalpy. The cooler the supply air is to the controlled space, while the refrigeration compressor is inactive, the longer the rest time of the refrigeration compressor.

Consider an outdoor condition of 80°F, with lesser enthalpy than a return air condition of 73°F. Conventional systems, based on enthalpy control of the dampers, maintain an 80°F supply air, while the refrigeration compressor is inactive; however, this new circuit switches to a 73°F supply air, when the refrigeration compressor is inactive. Supply air of 73°F causes a longer rest time, prior restarting the refrigeration compressor, than the 80°F supply air.

When the outdoor air enthalpy is less than the return air enthalpy, the dry bulb temperature of the return air is less than the dry bulb temperature of the outdoor air and the refrigeration compressor is inactive, switch the dampers to return air. This will maximize the rest time of the refrigeration compressor**.**

Combining enthalpy logic when the refrigeration compressor is active with dry bulb logic when the refrigeration compressor is inactive, minimizes the annual run time of the refrigeration compressor, while maintaining identical dry bulb temperature conditions in the occupied space.

ENERGY OPPORTUNITY REGARDING DX COOLING

This paper addresses a significant opportunity regarding HVAC systems with full mixing dampers, limited by a mixed air controller and an economiser controlled via enthalpy comparison of the outdoor air and the return air. The mechanical DX cooling, which cycles the refrigeration compressor via dry bulb temperature, is controlled from either the return air or the occupied space.

A logical control system should run the refrigeration compressors for the least amount of time, while maintaining comfort level in the occupied space. This is achieved by minimizing the refrigeration compressor run time and maximizing the compressor rest time. In order to achieve this situation we must view damper economiser control from a new perspective.

An enthalpy logic circuit calculates the outdoor air enthalpy and the return air enthalpy. The air stream with the lesser enthalpy is selected as the air stream passing through the evaporator coil of the refrigeration cooling system.

For example:

-The thermostat starts the refrigeration compressor when the room temperature rises to 76°F and stops the refrigeration compressor when the room temperature reduces to 72°F.
-Return air 76°F at 40% RH contains 26.61 BTU/lb of dry air.
-Return air 72°F at 51% RH contains 26.61 BTU/lb of dry air. (%RH rises as the room cools, with the same moisture content.)
-Outdoor air 80°F at 15% RH contains 22.76 BTU/lb of dry air.

In this example, enthalpy logic chooses to position the mixing dampers to 100% outdoor air, as this air stream has a lesser enthalpy value. (22.76 BTU < 26.61 BTU) This is the correct choice when the refrigeration compressor is running, as a lesser amount of heat exists in the outdoor air than the return air; therefore, the compressor run time will be minimized.

Now consider the rest time of the refrigeration compressor, as the room temperature rises from 72°F to 76°F. Heat gain factors such as body load, light load, solar load, machine load, etc. are a constant, regardless if the fan's mixing dampers are on 100% outdoor air or on return air with minimum outdoor air and will not alter the rate of temperature gain when the refrigeration compressor is resting or running.

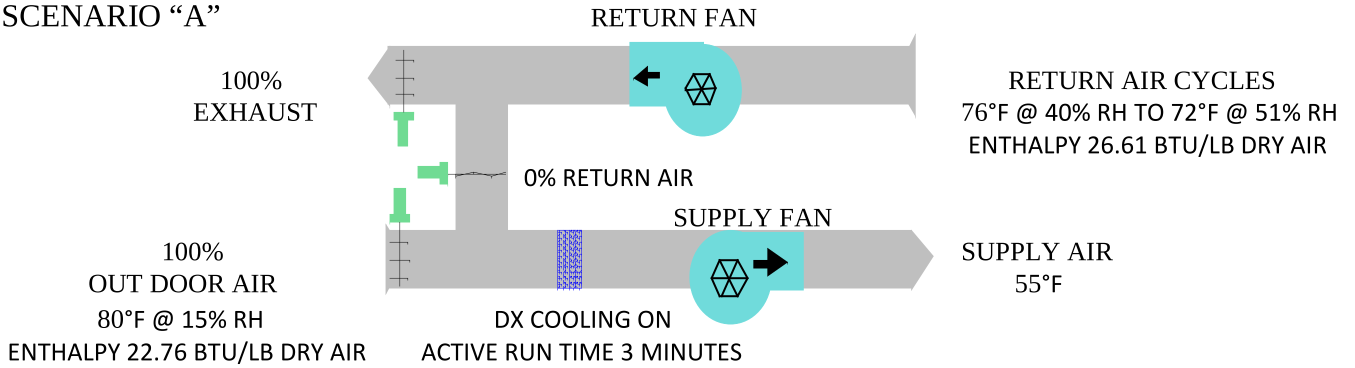
Keep in mind that the refrigeration compressor is controlled via dry bulb temperature. If the system is on 100% outdoor air at 80°F, the time for the room to rise from 72°F to 76°F will be less than if using the return air at 72°F to 76°F, with all other cooling loads being equal.

If the refrigeration compressor is running, the dampers should be positioned to the air stream with the lesser enthalpy, which will minimize the compressor run time.

If the refrigeration compressor is cycled off, when the outdoor enthalpy is less than the return air enthalpy, but the return air dry bulb temperature is less than the outdoor air temperature, select the return air with just minimum outdoor air, which will maximize the compressor rest time.

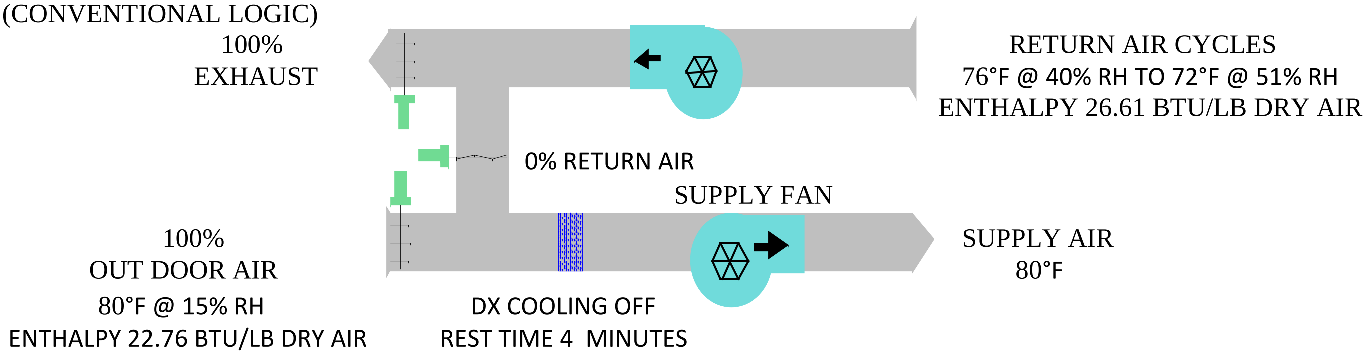
If the return air enthalpy is less than the outdoor air enthalpy, but the outdoor dry bulb temperature is less than the return air dry bulb temperature, do not switch to the cooler outdoor air, when the refrigeration compressor is not running, as you may flush the building with moist air, which will cause a greater cooling load when the refrigeration compressor running.

SCENARIO "A"



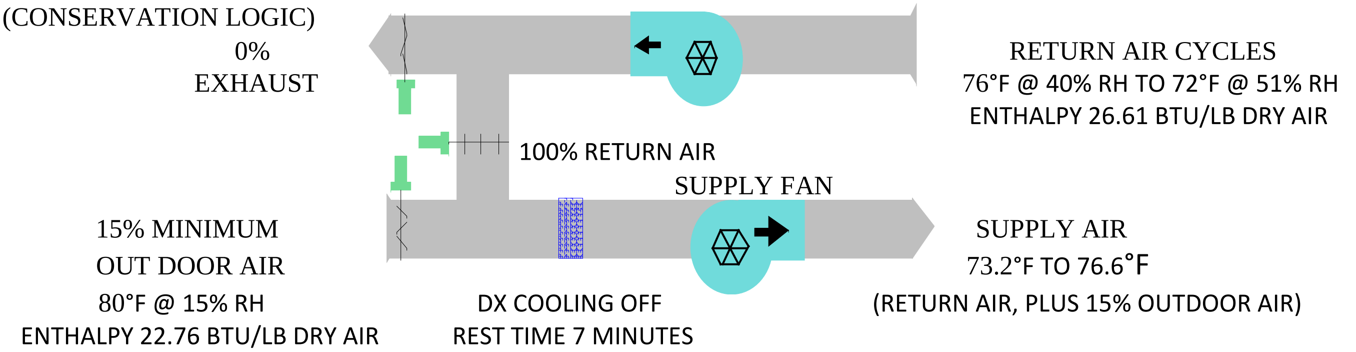
Scenario "A" illustrates a fan system, when the DX cooling compressor is active. Enthalpy comparison determines the heat content of the return air and outdoor air. The system dampers are on full outdoor air, as the total heat content of the outdoor air is approximately 14.5% less than the total heat content of the return air. Selecting the air stream with the lesser heat content causes the refrigeration compressor run time to be minimized.

SCENARIO "B"
(CONVENTIONAL LOGIC)



Scenario "B" illustrates the fan system, when the DX cooling compressor is not active. Enthalpy comparison determines the heat content of the return air and outdoor air. The dampers are on full fresh air, as the outdoor air contains 14.5% less total heat than the return air.

SCENARIO "C"
(CONSERVATION LOGIC)

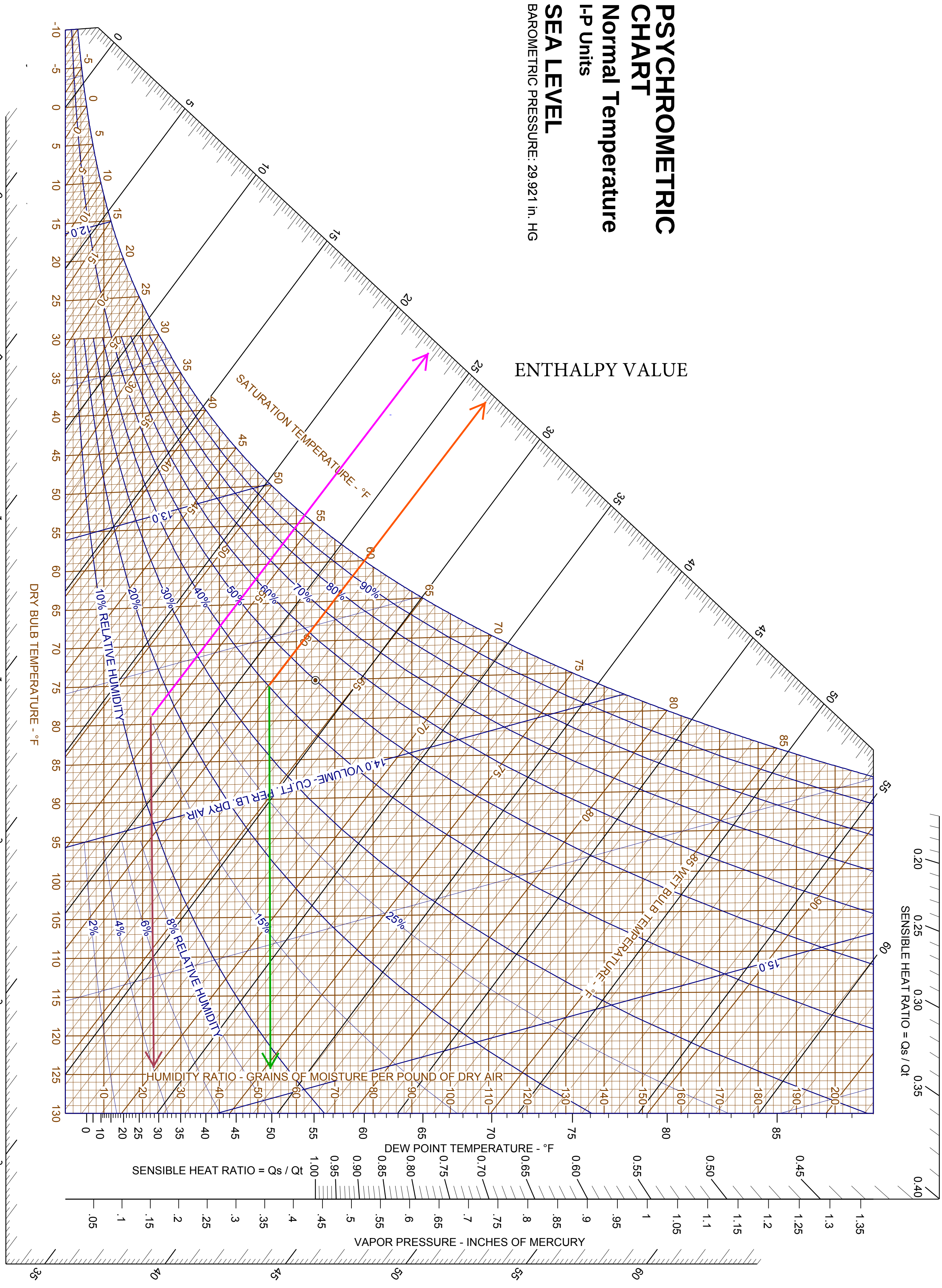


Scenario "C" illustrates the fan system when DX cooling compressor is not active. Cycling the mechanical cooling is based on dry bulb temperature of the space; therefore, the **rest** time of the mechanical cooling will be greater if the supply air is at the lower dry bulb temperature of the return air than the higher outdoor air.

SUMMARY

The mixing dampers should be positioned based on return air and outdoor air enthalpy comparison when the mechanical cooling is active, lessening the run time of the mechanical cooling.

The mixing dampers should revert back to minimum ventilation, when the mechanical cooling is at rest, if the return air temperature is less than the outdoor air temperature, increasing the rest time of the mechanical cooling.


PSYCHROMETRIC CHART
Normal Temperature
I-P Units
SEA LEVEL
BAROMETRIC PRESSURE: 29.921 in. HG
ENTHALPY VALUE
SATURATION TEMPERATURE - °F
DRY BULB TEMPERATURE - °F
10% RELATIVE HUMIDITY
8% RELATIVE HUMIDITY
14.0 VOLUME - CU.FT. PER LB. DRY AIR
85 WET BULB TEMPERATURE - °F
HUMIDITY RATIO - GRAINS OF MOISTURE PER POUND OF DRY AIR
DEW POINT TEMPERATURE - °F
SENSIBLE HEAT RATIO = Qs / Qt
VAPOR PRESSURE - INCHES OF MERCURY
ENTHALPY - BTU PER POUND OF DRY AIR

ENERGY CONSERVATION BENEFIT OF ENTHALPY COMPARISON IN HVAC

Energy conservation benefit regarding enthalpy comparison is significant. If owners, operators and service technicians have a true practical understanding of their importance in maintaining the control logic, they will better fill their respective roles of good management, proper monitoring and precise calibration.

The psychrometric chart on page four high lights the thermal characteristics of the return air and outdoor air illustrated in the sample drawing on page three.

The orange and green lines follow two psychrometric characteristics of the example return air. The orange line overlays the enthalpy line starting at 76° F at 40% RH running up to the enthalpy value of 26.61 BTU per pound of dry air. The green line overlays the moisture content line running to the value of 53.5 grains of moisture per pound of dry air.

The pink and brown lines follow two psychrometric characteristics of the example outdoor air. The pink line overlays the enthalpy line starting at 80° F at 15% RH running up to the enthalpy value of 22.76 BTU per pound of dry air. The brown line overlays the moisture content line running to the value of 22.5 grains of moisture per pound of dry air

Each pound of evaporated water produces 970 BTU's of heat, as it condenses from gas to liquid. This latent heat of evaporization often causes cooler, moist air to consume more cooling energy than warmer dry air. Enthalpy comparison loops tend to minimize the condensing of moisture at the cooling coil.

On page four the saturation temperature figures following the 100% humidity line toward the left side of the chart present the temperatures (dew point) where the air is not capable of holding more moisture than the grains of moisture per pound of dry air scale figure you see if you draw a line straight across the graph to the far right.

The target supply air temperature for cooling is often 55° F.

As the return air cools to 55° F, it becomes saturated at 60°F, containing 78 grains of moisture per pound of dry air. At 55° F the air is only capable of holding 64 grains of moisture per pound of dry air; therefore, 14 grains of moisture must be condensed out of the air as it cools from 60° F to 55° F. The latent heat of evaporization, as it condenses, plus the sensible heat reduction of the air, must be addressed by the cooling system.

As the 80° F outdoor air, at 15% RH, cools to 55°F, the relative humidity level rises to only 95.5% RH; therefore, no condensing takes place, allowing only sensible heat as the cooling load.

Based on 20,000 CFM, when the refrigeration compressor is running, the cooling load, using return air completely, is 7,560 BTU's sensible heat plus 2,906 BTU's latent heat = 10,466 BTU's total.
Based on 20,000 CFM, when the refrigeration compressor is running, the cooling load, using outdoor air completely, is 9,000 BTU's sensible heat and no latent heat = 9,000 BTU's total.
The cooling energy required by DX mechanical systems not comparing enthalpy, under the sample conditions, is 16.3% over a system comparing return air and the outdoor air enthalpy.

Regarding the example figures, when the refrigeration compressor is resting, enthalpy control of the dampers will add an extra 15.7% more sensible heat to the air supply, shortening the refrigeration compressor rest time. The net result will be more running time for the refrigeration compressor with absolutely no temperature difference in the space served.

NOTES

HUMIDEX CONTROL SYSTEM

PREFACE

The requirement arose to maintain the humidex level below thirty, based on the Occupational Health Clinic for Ontario Workers chart on page 8.125 of this report.

A control system was developed controlling the occupied space humidex level below thirty, while using enthalpy comparison control of return air to outdoor air assuring the lesser cooling load on the cooling coils.

As per the graph on page 8.124 both the humidex target and enthalpy targets were met.

CONTENT

PAGE

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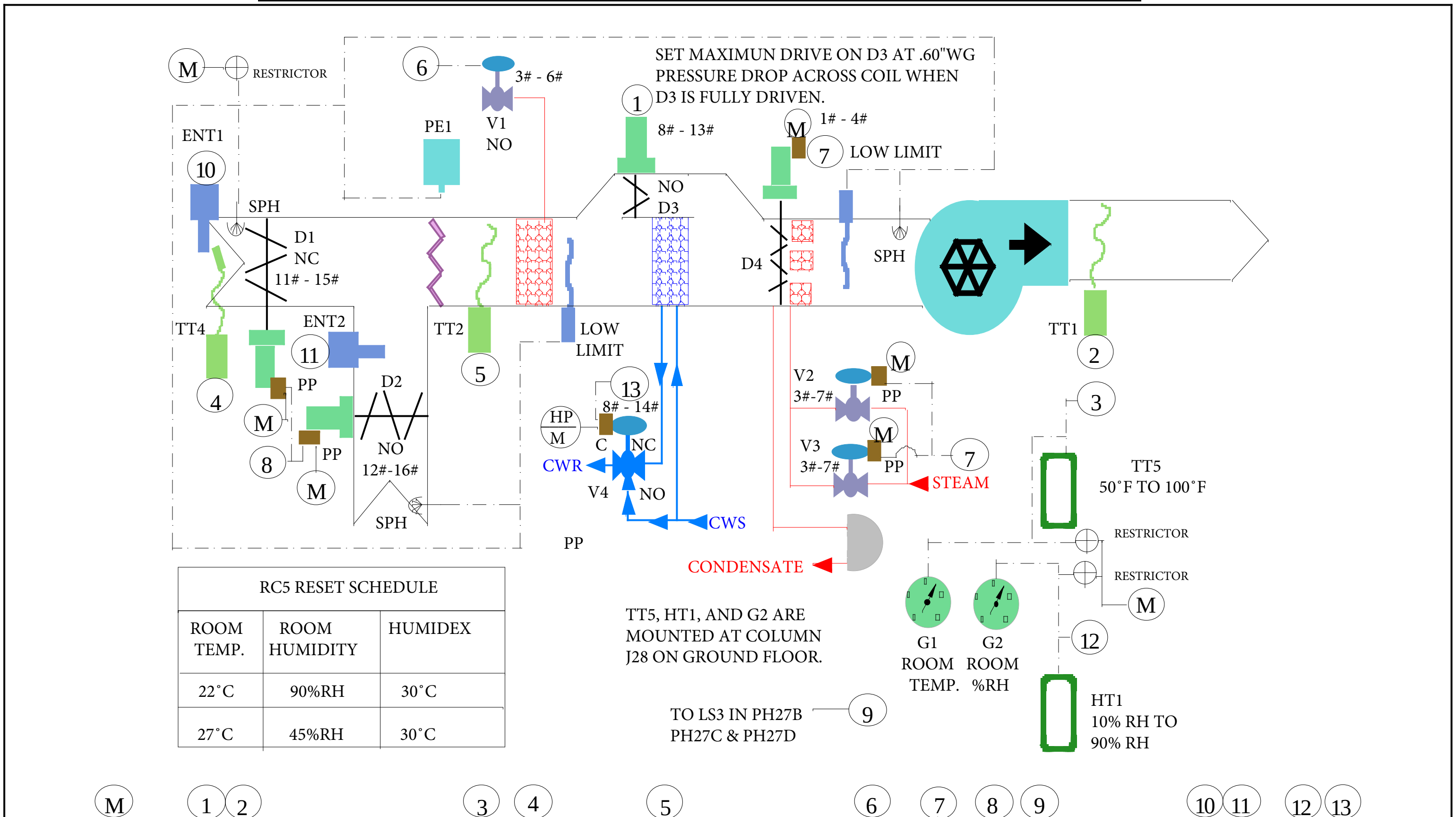
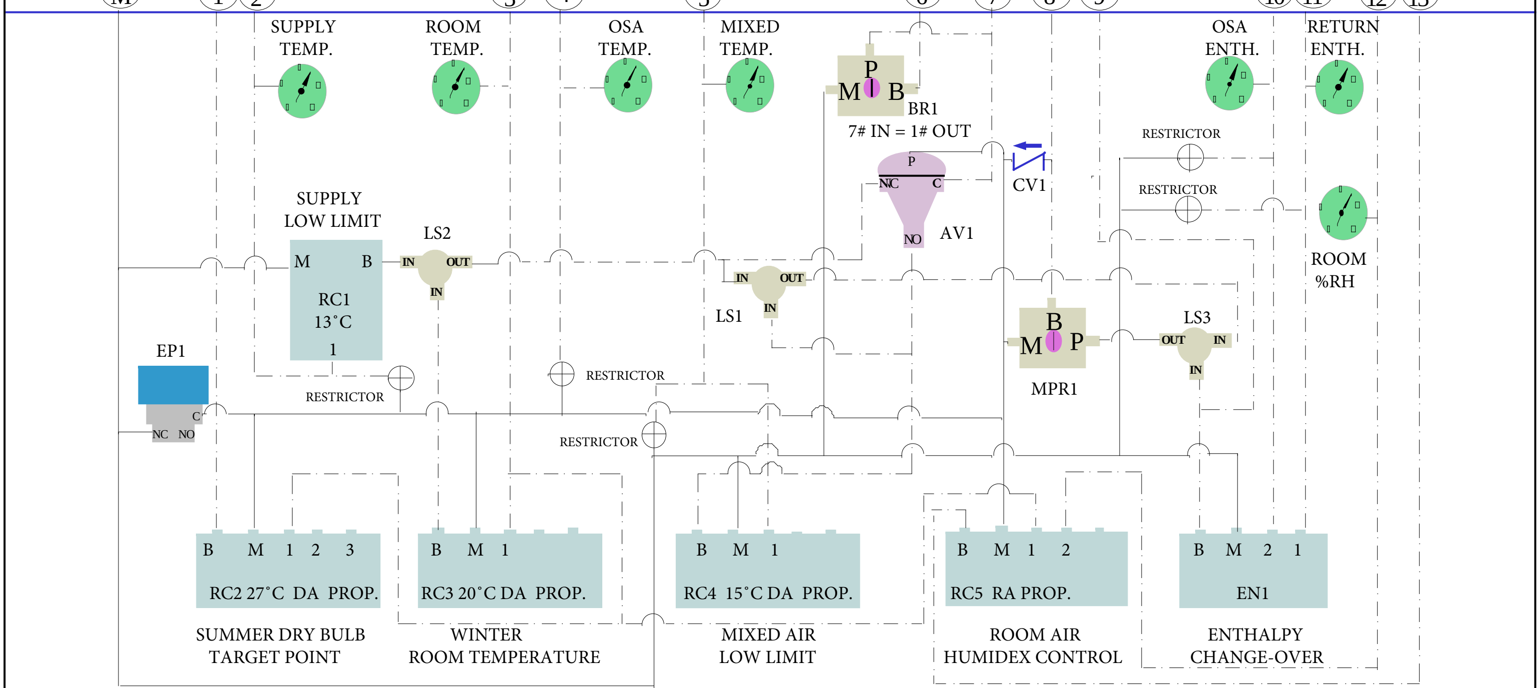
HUMIDEX CONTROL LOGIC WITH ENTHALPY CONTROL LOGIC



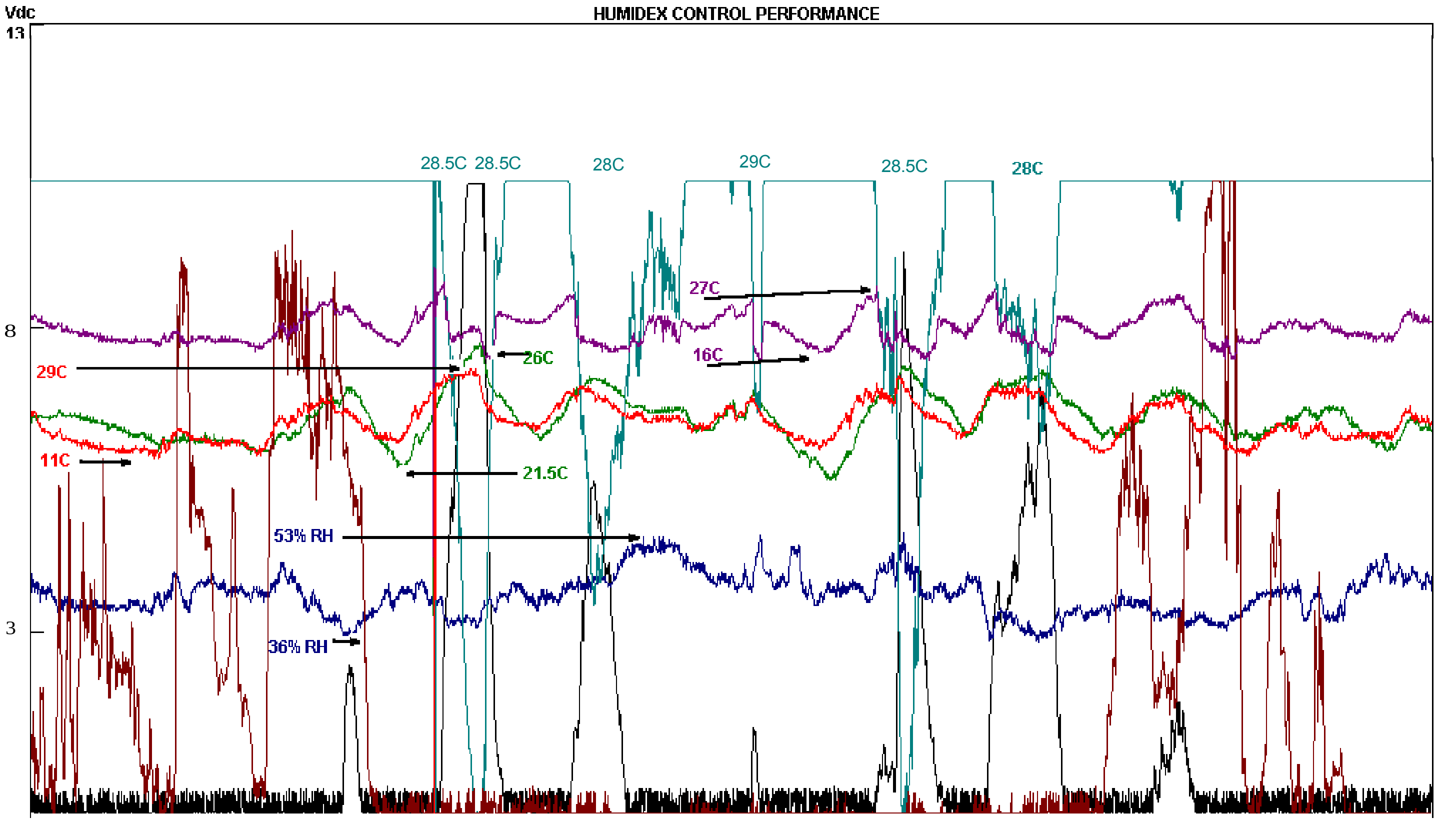
RC5 RESET SCHEDULE		
ROOM TEMP.	ROOM HUMIDITY	HUMIDEX
22˚C	90%RH	30˚C
27˚C	45%RH	30˚C



SEQUENCE OF OPERATION

When the fan is off the outside air damper is closed, the return air damper is open and the steam valves are controlled by the mixed air low limit controller.

When the fan is running the steam valves (V1), (V2), (V3) and (D3) and the mixing dampers (D1), (D2) are modulated in sequence, such that on a drop in room temperature: first the free cooling modulates back to the minimum ventilation position, then the steam valve (V1) opens as required; then V2 and V3 are opened and next D3 exposes the coil to an air flow if required. This loop is limited in the supply air to a minimum of 13°C and limited to a minimum in the mixed air at 15°C. When the outside air, enthalpy is greater than the return air enthalpy, D1 and D2 return to the minimum ventilation position. Enthalpy controller (EN1) from PH27A sends its signal to low selector (LS3) allowing free cooling with the air stream containing the lesser BTU content.



COLOUR LEGEND

RED----Outdoor air temperature
Green---Return air temperature
Purple---Supply air temperature
Black----Branch summer dry bulb bypass dampers
Brown—enthalpy branch signal
BLUE----Humidex branch (cooling valve)

Humidex Heat Stress Response Plan

Temp (in oC)	RH = 100%	95%	90%	85%	80%	75%	70%	65%	60%	55%	50%	45%	40%	35%	30%	25%	20%	15%	10%	Temp (in oC)
49																			50	49
48																			49	48
47																		50	47	47
46																		49	46	46
45	NEVER IGNORE ANYONE'S SYMPTOMS DESPITE YOUR MEASUREMENTS!!!																50	47	45	45
44	Humidex	Action															49	46	43	44
43	45+	stop work														49	47	45	42	43
42	43-44	75% relief													50	48	46	43	41	42
41	40-42	50% relief													48	46	44	42	40	41
40	38-39	25% relief												49	47	45	43	41	39	40
39	34-37	warning & double water											49	47	45	43	41	39	37	39
38	30-33	alert & water										49	47	45	43	42	40	38	36	38
37	25-29	water as needed									49	47	45	44	42	40	38	37	35	37
36									50	49	47	45	44	42	40	39	37	35	34	36
35								50	48	47	45	43	42	40	39	37	36	34	33	35
34							49	48	46	45	43	42	40	39	37	36	34	33	31	34
33					50	48	47	46	44	43	41	40	39	37	36	34	33	32	30	33
32			50	49	48	46	45	44	42	41	40	38	37	36	34	33	32	30	29	32
31	50	49	48	47	45	44	43	42	40	39	38	37	35	34	33	32	30	29	28	31
30	48	47	46	44	43	42	41	40	39	37	36	35	34	33	31	30	29	28	27	30
29	46	45	43	42	41	40	39	38	37	36	35	33	32	31	30	29	28	27	26	29
28	43	42	41	40	39	38	37	36	35	34	33	32	31	30	29	28	27	26	25	28
27	41	40	39	38	37	36	35	34	33	32	31	30	29	28	27	26	25			27
26	39	38	37	36	35	34	33	33	32	31	30	29	28	27	26	25				26
25	37	36	35	34	33	33	32	31	30	29	28	27	26	26	25					25
24	35	34	33	33	32	31	30	29	28	28	27	26	25							24
23	33	32	31	31	30	29	28	28	27	26	25									23
22	31	30	30	29	28	27	27	26	25	25										22
21	29	29	28	27	26	26	25													21

HOT WATER RESET

REPORT CONSIDERING

OCCUPANTS' COMFORT

AS WELL AS

ENERGY CONSUMPTION

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CONTENTS

SECTION ONE 1.0

SECTION TWO 2.0
COMPARISON OF ACTUALLY REQUIRED HOT WATER SUPPLY TEMPERATURE TO OUTDOOR RESET HOT WATER SUPPLY TEMPERATURE.

SECTION THREE 3.0
ILLUSTRATION OF SUPPLY WATER TEMPERATURE IMPACT ON ENERGY CONSUMPTION AND COMFORT LEVELS FOR THREE DIFFERENT SET OF DETERMINING FACTORS REGARDING SUPPLY WATER TEMPERATURE.

SECTION FOUR 4.0

SECTION FIVE 5.0

The data for this case study was collected in a school at two locations. One illustrates the actual temperature of the supply water required to satisfy the coolest zone served and the other illustrates the impact of supply water temperature on energy performance and comfort.

The logic presented in this report does not only apply to school buildings. It applies to any heating system with the capability of varying the primary supply water or air temperature into the occupied areas.

Systems resetting hot water or supply air only from outdoor air temperature are likely wasting heat, increasing the wear and tear on the control valves and creating discomfort, at times, in the some served areas.

The logic applies to all control systems: DDC, electronic, pneumatic and manual.

EXECUTIVE SUMMARY

In many classrooms in Ontario, the heating and ventilation is provided by a fan system called a unitventilator. (See Figure #1 at the back of this report.)

The unitventilator is a small fan system dedicated to only one room.

Figure #1 is a drawing of the type of unitventilator, which is the subject of this report. They are very inefficient and, at times, have difficulty maintaining the comfort conditions in the classrooms.

Over the years, during service, we have witnessed the symptoms of excessive energy use and discomfort in many buildings with unitventilators. This report graphically illustrates the problems with these units and a solution that improves both the energy use as well as the controllability.

The problem, from an energy point, with this type of unitventilator is that the hot water flows continuously through the heating coil. A damper arrangement is used to allow the air to flow through the heating coil, deflect the air to bypass the heating coil or blend the air to various temperatures to select the amount of heat the room requires. Even when all the air is theoretically bypassing the coil, heat escapes into the air stream. This causes the unit to automatically bring in excessive fresh air to compensate for the unwanted heat.

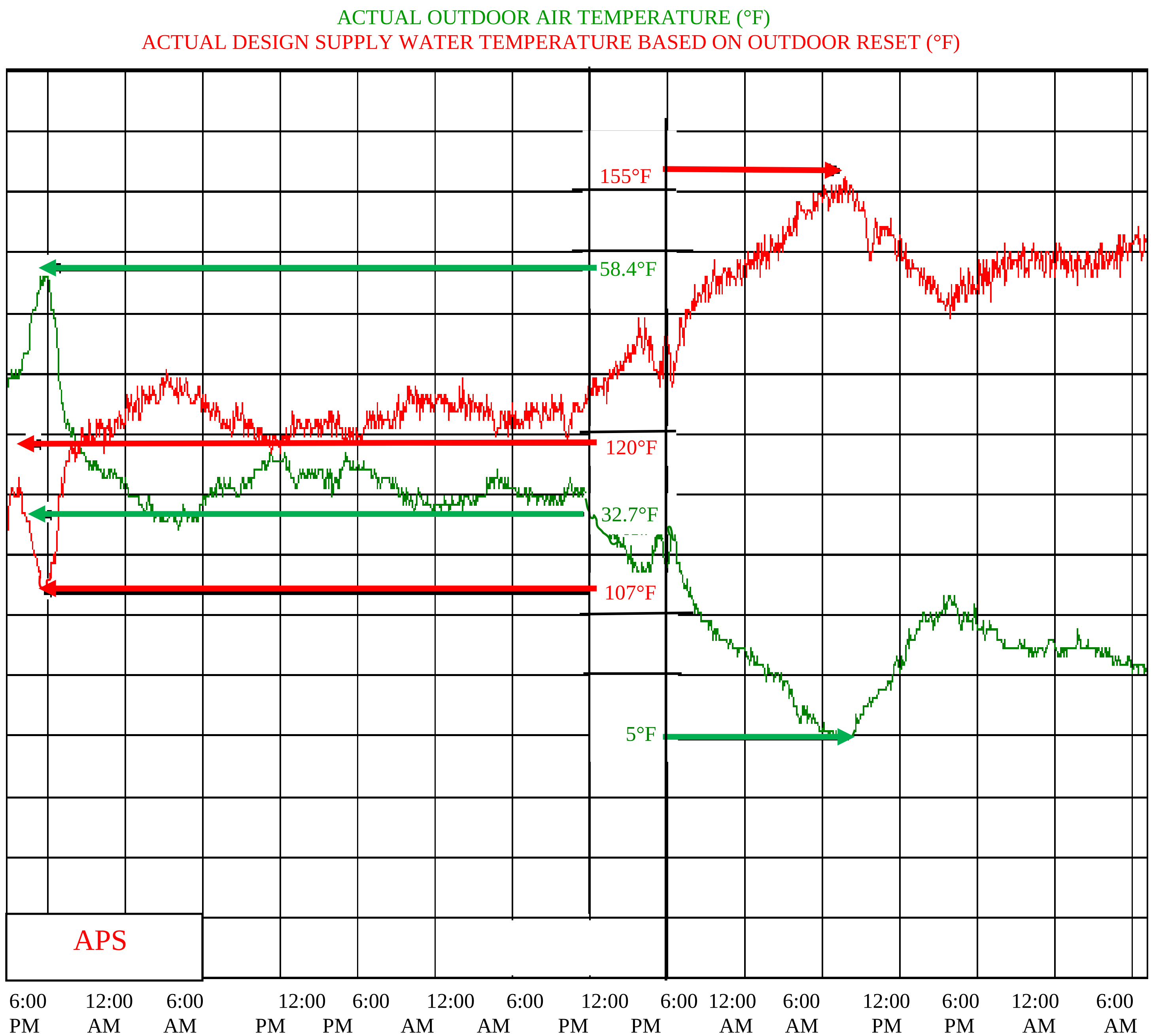
The problem, from a comfort point, is that often the amount of heat escaping is greater than the cooling capability of the unit, even when the unit is on full cooling. The room over-heats while it is actually ventilating with about 500% more cold fresh air than required by code.

Figure #2, at the back of this report, is a drawing of the Energy Conservation Control Circuit that replaced the original heating supply water control circuit. The original circuit was reset based on only the outside air temperature.

By modifying the controls to the Energy Conservation Circuit, the system saved:

- ✓ 81.7% on heating energy, on day operation, compared to a system supplied with 171°F supply water (boiler temperature)
- ✓ 58.4% on heating energy, on day operation, compared to a system supplied with water reset from the outside air temperature
- ✓ the unit used 247% more heat with the 171°F supply water on full cooling than the Energy Conservation Circuit used on full heating
- ✓ comfort levels are superior in the occupied space when using the Energy Conservation Circuit

GRAPH #1


ACTUAL OUTDOOR AIR TEMPERATURE (°F)
ACTUAL DESIGN SUPPLY WATER TEMPERATURE BASED ON OUTDOOR RESET (°F)
155°F
58.4°F
120°F
32.7°F
107°F
5°F
APS
6:00 PM
12:00 AM
6:00 AM
12:00 PM
6:00 PM
12:00 AM
6:00 AM
12:00 PM
6:00 PM
12:00 AM
6:00 AM
12:00 PM
6:00 PM
12:00 AM
6:00 AM

GRAPH #1

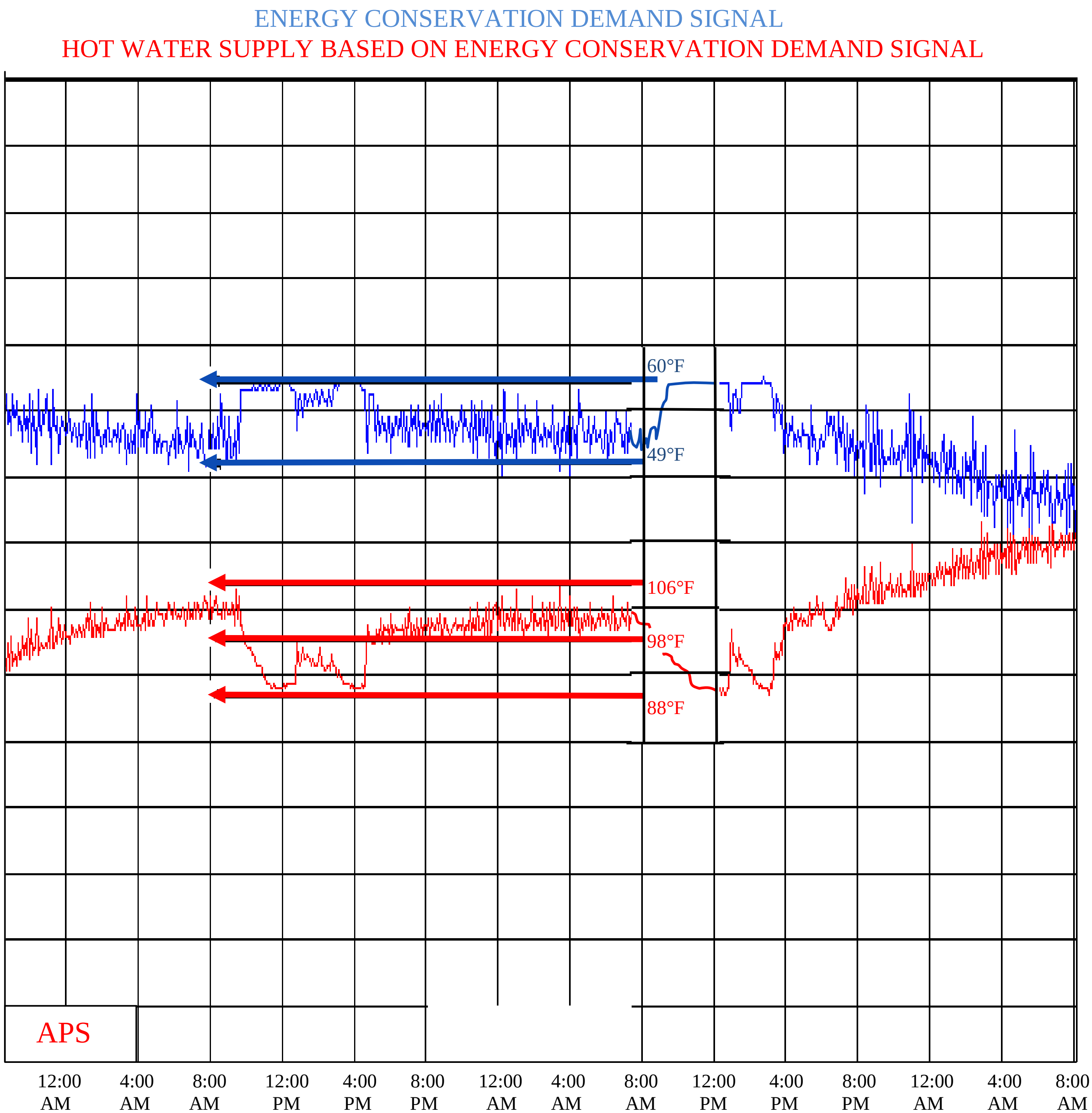
Graph #1 shows the relationship of the supply water temperature with respect to the actual outside air temperature. This graph illustrates the performance of a typical indoor/outdoor reset system circuit. This was the existing design reset schedule for this case study.

Sensing point (F) on FIGURE #2 is the location where the supply water temperature readings were obtained and sensing point (G) on FIGURE #2 is the location where the actual outside air temperature readings were obtained.

The supply water temperature increases as the outside air temperature decreases.

Control from only the outside air temperature mistakenly assumes that the only factor worth considering in hot water reset is the dry bulb outside air temperature.

GRAPH #2


ENERGY CONSERVATION DEMAND SIGNAL
HOT WATER SUPPLY BASED ON ENERGY CONSERVATION DEMAND SIGNAL
60°F
49°F
106°F
98°F
88°F
APS
12:00 AM
4:00 AM
8:00 AM
12:00 PM
4:00 PM
8:00 PM
12:00 AM
4:00 AM
8:00 AM
12:00 PM
4:00 PM
8:00 PM
12:00 AM
4:00 AM
8:00 AM

GRAPH #2

GRAPH #2 shows the relationship of the supply water temperature with respect to the actual amount of heat required by the coldest room on the system. This graph illustrates the performance of an Energy Conservation System Circuit.

Sensing point (F) on FIGURE #2 is the location where the supply water temperature readings were obtained and sensing point (I) on FIGURE #2 is the location where the Energy Conservation Demand Temperature Signal readings were obtained.

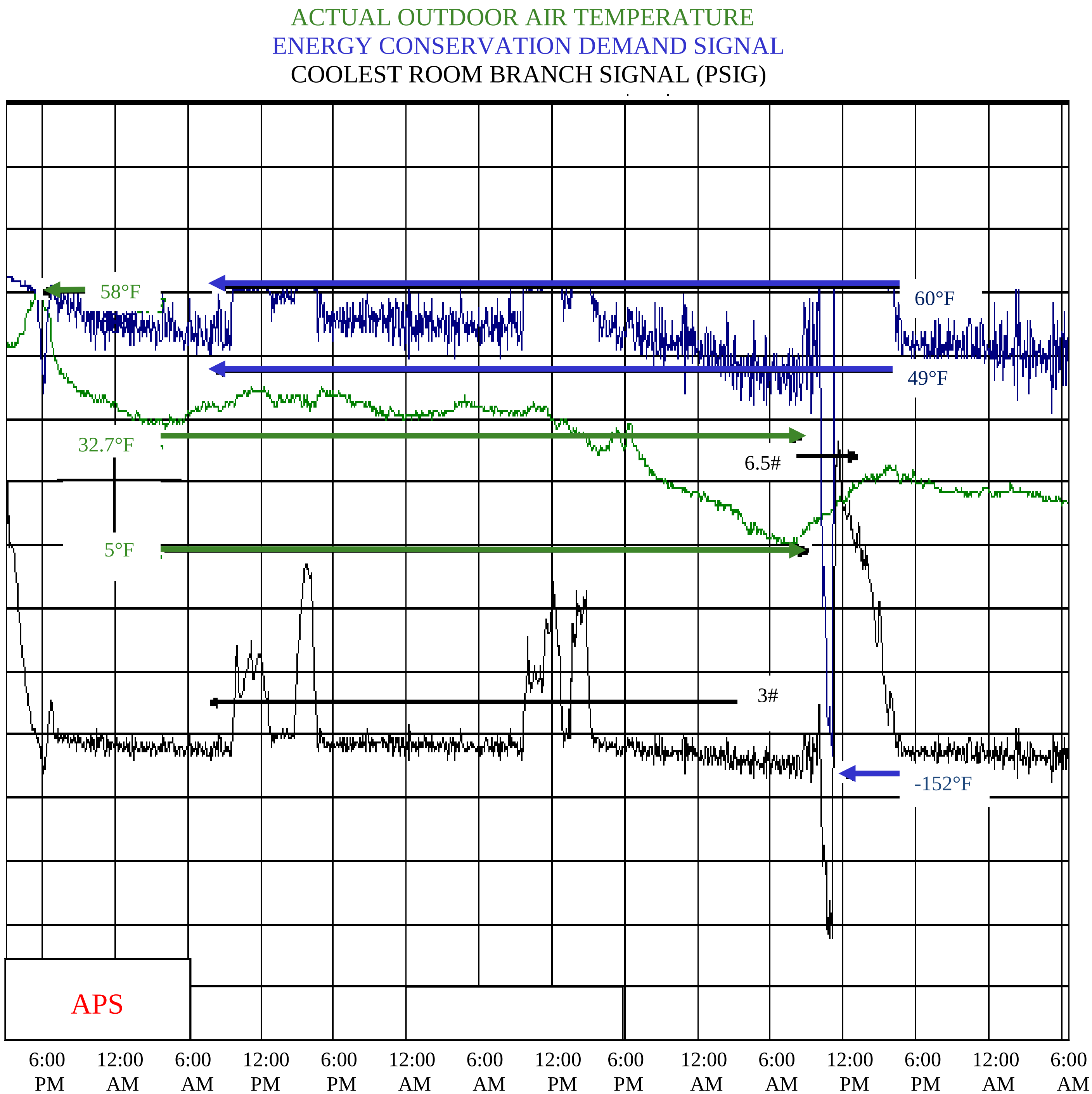
The Energy Conservation Demand Temperature Signal is created by the demand of the coldest room in the system. **The Energy Conservation Demand Temperature Signal "tells" the main heating controller that the outside temperature is warmer than the actual outside condition to allow the system to attain the exact temperature in the supply water to satisfy the building's requirement.**

A multi-selector determines the heating requirement of the coldest room. It selects the lowest signal to indicate the true requirement for heating. Relays are used to offset the coldest room's signal such that it will create an Energy Conservation Demand Temperature Signal of 60°F any time the room has either started to close off its heating device or has closed off its heating device. The only time the coldest room is allowed to ask for more than the minimum level of heat from the main system is after the room thermostat has opened its heating device to maximum flow. This delivers the required BTUs to the room with the minimum degree of temperature in the supply water.

Graph #2 illustrates the supply water temperature dropping when the body load of the students and other internal heat sources are generating enough heat to lessen the load on the heating system. The water temperature rises at night and at lunch when the other heat sources have been removed from the classrooms.

It is worth observing that the supply water temperature graph is symmetrical to both the actual outside air temperature graph as well as to the Energy Conservation Demand Temperature Signal graph on both Graph #1 and Graph #2. The difference is that Graph #1 is using open loop logic which makes the poor assumption that the outside air temperature is the only factor required to determine the heating supply water temperature and Graph #2 is using closed loop logic which assesses the actual heating requirements of the building.

GRAPH #3



ACTUAL OUTDOOR AIR TEMPERATURE
ENERGY CONSERVATION DEMAND SIGNAL
COOLEST ROOM BRANCH SIGNAL (PSIG)
58°F
60°F
49°F
32.7°F
6.5#
5°F
3#
-152°F
APS
6:00 PM
12:00 AM
6:00 AM
12:00 PM
6:00 PM
12:00 AM
6:00 AM
12:00 PM
6:00 PM
12:00 AM
6:00 AM
12:00 PM
6:00 PM
12:00 AM
6:00 AM

GRAPH #3

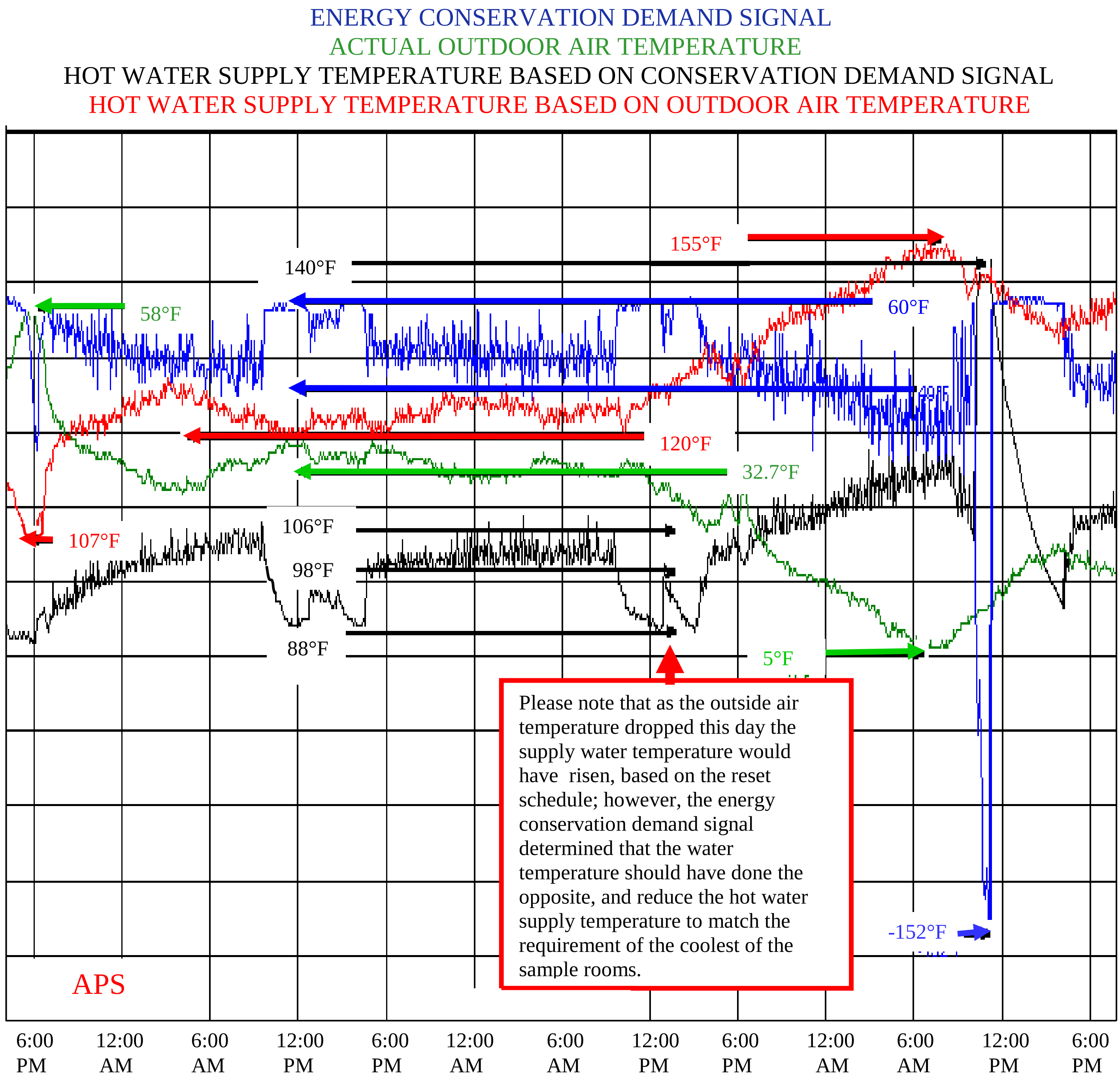
Graph #3 illustrates the method of creating the Energy Conservation Demand Signal.

The "COLDEST ROOM BRANCH SIGNAL" is attained via a multi-selector, which produces the lowest signal of the rooms that are sampled. The coldest room signal is obtained at sensing point (H) on FIGURE #2.

The lowest signal is sent to a biasing relay which offsets the signal so that the pressure at which the heating function in the unitventilator is fully open will cause a branch signal from the relay to simulate the warmest outside air temperature considered in the building's original reset schedule. The relay's signal is limited to that value, as a maximum, which sets the minimum water temperature at which the system will be allowed to operate. The signal from the limiting relay goes to a high selector, which also receives a signal from the actual outside air temperature transmitter that prevents the supply water temperature from exceeding the original reset water schedule.

The actual outside air temperature readings were obtained at sensing point (G) on FIGURE #2 and the Energy Conservation Demand Temperature Signal readings were obtained at sensing point (I) on FIGURE #2.

GRAPH #4


ENERGY CONSERVATION DEMAND SIGNAL
ACTUAL OUTDOOR AIR TEMPERATURE
HOT WATER SUPPLY TEMPERATURE BASED ON CONSERVATION DEMAND SIGNAL
HOT WATER SUPPLY TEMPERATURE BASED ON OUTDOOR AIR TEMPERATURE
155°F
140°F
58°F
60°F
49°F
120°F
32.7°F
107°F
106°F
98°F
88°F
5°F
Please note that as the outside air temperature dropped this day the supply water temperature would have risen, based on the reset schedule; however, the energy conservation demand signal determined that the water temperature should have done the opposite, and reduce the hot water supply temperature to match the requirement of the coolest of the sample rooms.
-152°F
APS
6:00 PM
12:00 AM
6:00 AM
12:00 PM
6:00 PM
12:00 AM
6:00 AM
12:00 PM
6:00 PM
12:00 AM
6:00 AM
12:00 PM
6:00 PM

GRAPH #4

Graph #4 illustrates the relationship of the actual outside air temperature and the Energy Conservation Demand Signal. Graph #4 also illustrates the simultaneous relationship of the reset water temperatures for control from the Energy Conservation Demand Signal as well as control from actual outside air temperature.

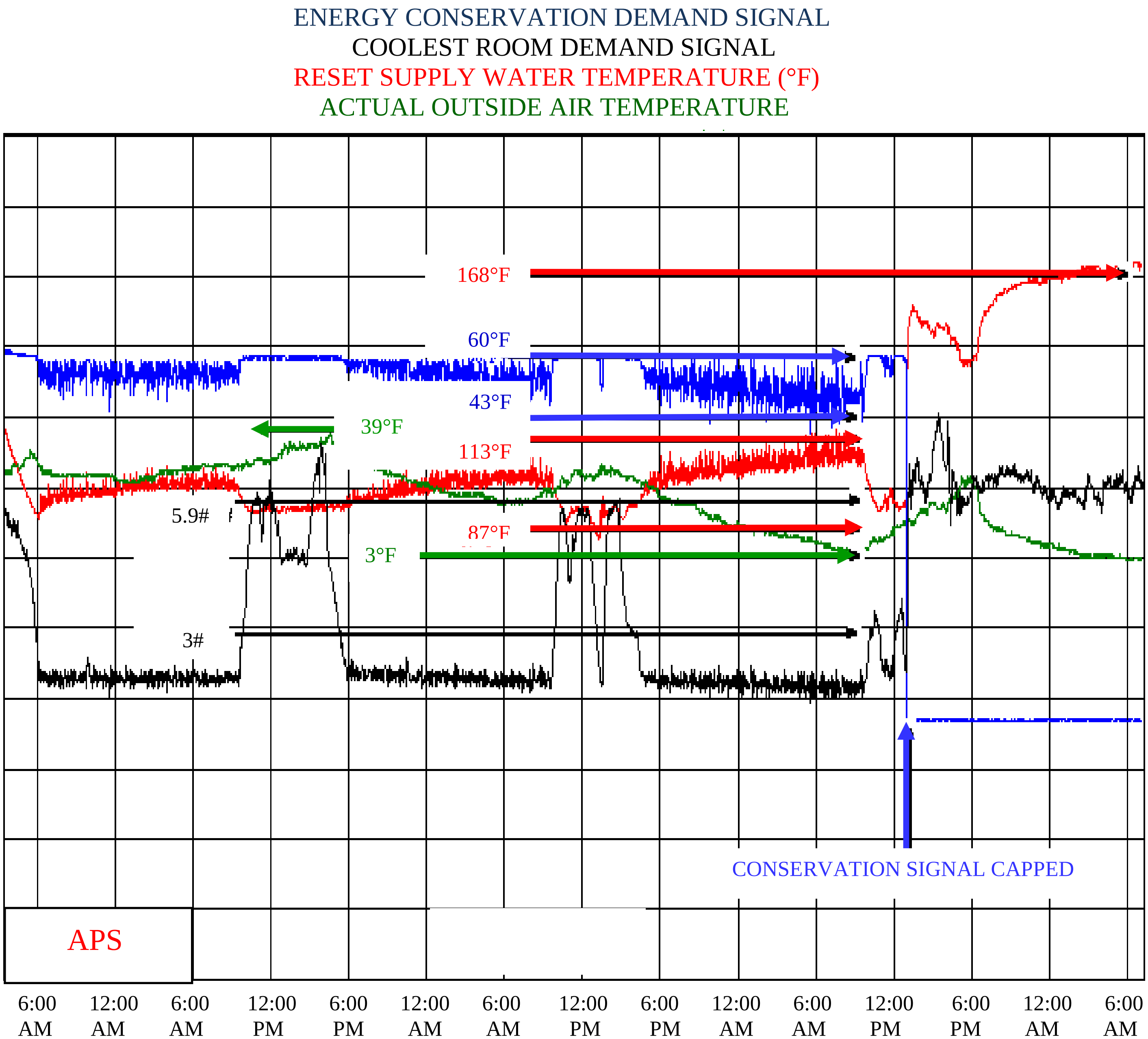
The sensing points for Graph #4 were obtained as shown on FIGURE #2.

The Energy Conservation Demand Signal responds directly to the actual heating requirements of the coolest spot in the building while the outside air temperature sensor is oblivious to the building's needs and only reports the outside air temperature condition. During the afternoon of February 25, 1996 the actual outside air temperature was declining which would have increased the supply water temperature; however, the Energy Conservation Demand Signal was increasing because of the return of the body heat from the students to the classrooms. This caused the supply water temperature to be reduced to reflect the actual heating requirements of the building.

In the morning of February 26, 1996 the Energy Conservation Demand Signal dropped to -152°F. The actual outside air temperature was only 10°F. The system is designed to believe the higher of the actual outside air temperature or the Energy Conservation Demand Signal; therefore, the supply water temperature was limited to the temperature which was the old outside air reset schedule desired.

The reason the Energy Conservation Demand Signal dropped was that an unauthorized person had turned off the circuit breaker for the unitventilator. The room dropped in temperature, which caused the thermostat to demand full heat; however, the heat, which the thermostat was demanding from the building's primary supply system, was prevented from getting to the room because the room's fan was turned off. As you will see later in this report, this caused the other rooms on the same water supply to bring excessive amounts of fresh air into their unitventilators to attempt to compensate for the unwanted BTUs that escape from the coils. The result is a loss of comfort, a waste of energy and an increase in pollution from products of combustion while producing the unnecessary heat.

GRAPH #5



ENERGY CONSERVATION DEMAND SIGNAL
COOLEST ROOM DEMAND SIGNAL
RESET SUPPLY WATER TEMPERATURE (°F)
ACTUAL OUTSIDE AIR TEMPERATURE
168°F
60°F
43°F
39°F
113°F
5.9#
87°F
3°F
3#
CONSERVATION SIGNAL CAPPED
APS
6:00 AM
12:00 AM
6:00 AM
12:00 PM
6:00 PM
12:00 AM
6:00 AM
12:00 PM
6:00 PM
12:00 AM
6:00 AM
12:00 PM
6:00 PM
12:00 AM
6:00 AM

GRAPH #5

Graph #5 illustrates the difference in the supply water temperature, comparing control when the reset system was allowed to sense the Energy Conservation Demand Signal to reset based on only the actual outside air temperature.
The change of reference points was done at noon on March 4, 1996.

The sensing points for Graph #5 were obtained as shown on FIGURE #2.

If you look back at Graph #1 and Graph #2 to compare them to Graph #5 you will see that the relationship of the supply water up to noon on March 4, 1996 is similar to Graph #2 which graphed resetting from the Energy Conservation Demand Signal. After noon on March 4, 1996 on Graph #5 the relationship is similar to Graph #1 which graphed resetting from actual outside air temperature.

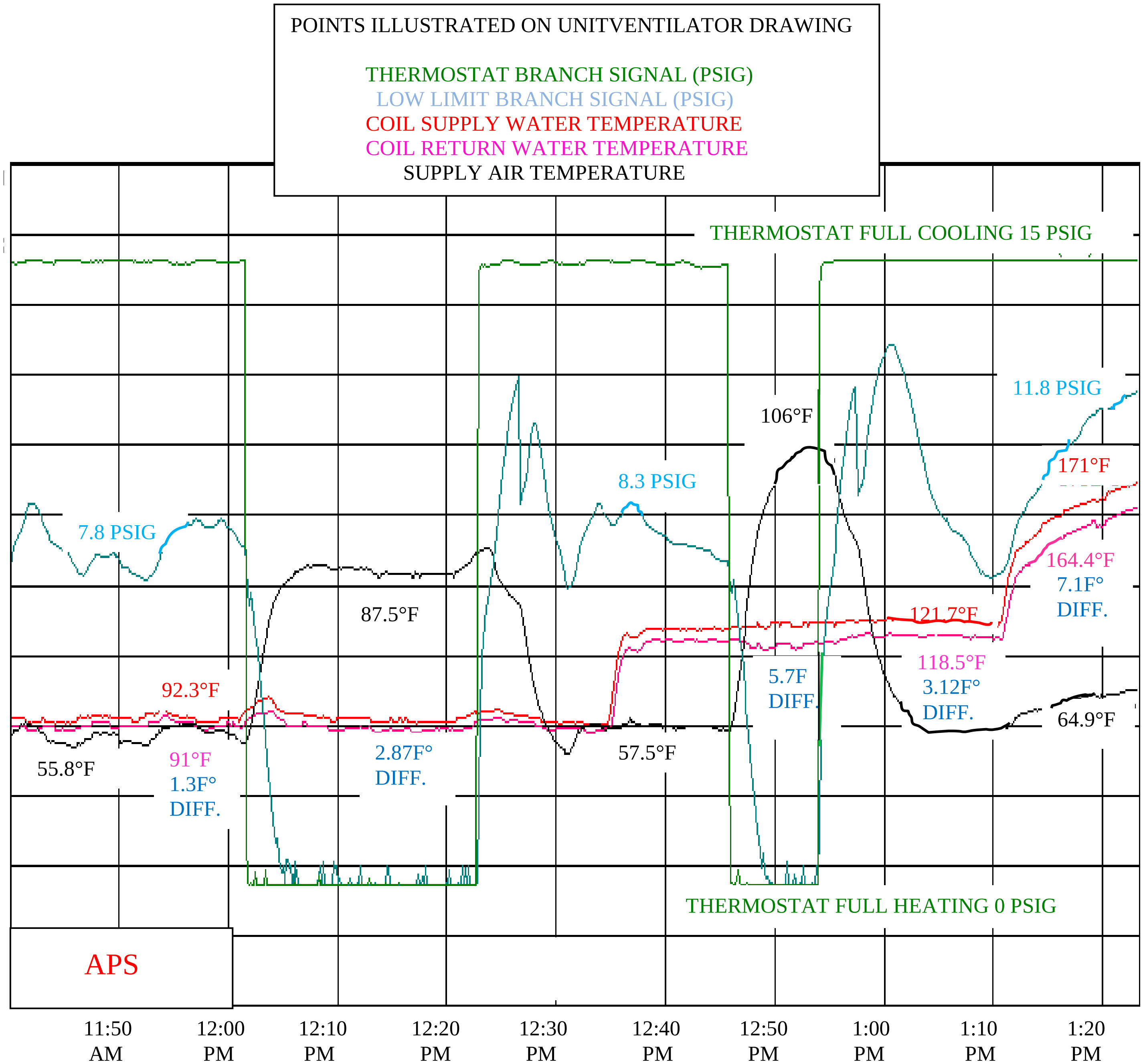
The coldest room demand signal reflected the impact of the body heat of the students up to noon March 4, 1996. The system required cooling while the students were in the classrooms during the day, but the signal dropped at night when the effect of the student body heat was gone to stop the cooling and only use heating.

After noon on March 4, 1996 the coldest room signal increased to bring in cooling to compensate for the heat which was being forced into the system by the main supply water temperature increase. **The unitventilator in the coldest room required more cooling at night, when the students' body heat was not present, than the room did when the classroom was full of students.**

The increased cooling required is the result of heat escaping into the unitventilator.

NOTE: Graphs 1, 2, 3, 4 and 5 were all developed to illustrate the method and need for controlling the supply water temperature based on the Energy Conservation Demand Signal which is created by the coldest room. This signal activates the main heating supply after the coldest room has opened its heating device to receive the requested heat. The information for these charts was obtained from signals in the boiler room at the main steam heat exchangers for the supply water.

GRAPH #6


POINTS ILLUSTRATED ON UNITVENTILATOR DRAWING
THERMOSTAT BRANCH SIGNAL (PSIG)
LOW LIMIT BRANCH SIGNAL (PSIG)
COIL SUPPLY WATER TEMPERATURE
COIL RETURN WATER TEMPERATURE
SUPPLY AIR TEMPERATURE
THERMOSTAT FULL COOLING 15 PSIG
11.8 PSIG
106°F
8.3 PSIG
171°F
7.8 PSIG
164.4°F
7.1F°
DIFF.
87.5°F
121.7°F
118.5°F
5.7F
DIFF.
3.12F°
DIFF.
92.3°F
64.9°F
91°F
1.3F°
DIFF.
2.87F°
DIFF.
57.5°F
55.8°F
THERMOSTAT FULL HEATING 0 PSIG
APS
11:50 AM
12:00 PM
12:10 PM
12:20 PM
12:30 PM
12:40 PM
12:50 PM
1:00 PM
1:10 PM
1:20 PM

GRAPH #6

Graph #6 was prepared to illustrate the increased energy loss in the unitventilator and the loss of control in the room as the supply water temperature rises.

All the readings for the graph were taken in a classroom from sensing points shown on FIGURE #1. The thermostat branch pressure was obtained at (C); the low limit branch signal was obtained at (D); the supply air temperature was obtained at (E); the supply water temperature was obtained at (A) and the return water temperature was obtained at (B).

The supply water temperature was provided at three different modes of operation:

1. The first was based on the Energy Conservation Demand Signal logic with an average supply water temperature of 92.3°F.
2. The second was based on actual outside air temperature reset with an average supply temperature of 121.7°F.
3. And the third was based on boiler supply water at an average supply temperature of 171.5°F.

We ran the thermostat on full cooling with a fifteen pound branch pressure and on full heating with a zero pound branch pressure as shown by the green line.

We observed two characteristics of the unitventilator's response. One was the energy use based on the differential of the supply and return water temperatures and the second was the ability to maintain comfort based on the supply air temperature.

The coil always had the same rate of water flow; therefore, the differential temperature was a direct indication of the relative amount of heat escaping into the room when on cooling under the three different supply water temperatures.

The following chart indicates the supply water temperature, the water differential temperature, the discharge air temperature and the low limit branch pressure when the thermostat was demanding full cooling.

SUPPLY TEMPERATURE	DIFFERENTIAL TEMPERATURE	DISCHARGE AIR TEMPERATURE	LOW LIMIT PRESSURE
92.3°F	1.3F°	55.8°F	7.8 PSIG
121.7°F	3.12F°	57.5°F	8.3 PSIG
171.5°F	7.1F°	64.9°F	11.8 PSIG

The information obtained from the chart gives a clear picture of the supply water temperature's impact on the energy use of a unitventilator as well as the effect on the comfort level in the room.

ENERGY

The building was running on the conservation circuit using the Energy Conservation Demand Signal when we got to the building. The rooms were all satisfied and the water supply temperature was 92.3°F.

On full cooling the unitventilator was only losing 1.3F° of heat from the supply water as it passed through the unit.

When the supply water temperature was raised to the original reset schedule control point of 121.7°F, the unitventilator, which was on full cooling, was losing 3.12F° of heat from the supply water passing through the unit. This is 240% more unwanted heat escaping into the air stream than on the conservation circuit.

When the supply water temperature was raised to 171.5°F, to simulate a typical supply water temperature without a secondary loop, the unitventilator, which was on full cooling, was losing 7.1F° of heat from the supply water. This is 546% more unwanted heat escaping into the air stream than on the conservation circuit.

The low limit branch pressure increased from 7.8 PSIG on the conservation circuit to 8.3 PSIG on the original reset schedule to 11.8 PSIG on the high temperature water supply. The damper motor has a spring range of five to ten pounds. The actual outside air temperature at the time of the readings was 27°F. **While on the high temperature water the unit had to drive to 100% outside air and still could not get the supply air temperature below 64.9°F. The escaped heat caused a 37.9F° temperature gain in the supply air temperature with the unit on full fresh air.**

COMFORT

The low limit branch signal had to gradually increase as the supply water temperature was increased to compensate for the greater amounts of unwanted heat escaping into the unitventilator. **The low limit finally lost control in its attempt to provide cooling and the room over-heated while the thermostat and the low limit were both producing signals to demand full cooling.**

NOTE: When the thermostat was demanding full heat while on the conservation circuit the water differential was 2.8F°. When the thermostat was demanding full cooling on the high temperature water, the water differential was 7.1F°. The system used 247% more heat when on cooling with the high temperature water than it used when on full heating with the conservation circuit.

GENERAL

On any heating system where the level of heat is to be determined by the actual requirement of the space, it is imperative that the control point which sends a signal to the main plant to demand heating levels also controls the local device which regulates the heat into the local area. It is a very common error to have computer points in several classrooms to scan the building for the greatest demand for heat while having local thermostats controlling the flow of heat to the rooms. This can create a situation where the computer is demanding full heat to satisfy a room where the thermostat is set to not allow the heat to enter the room.

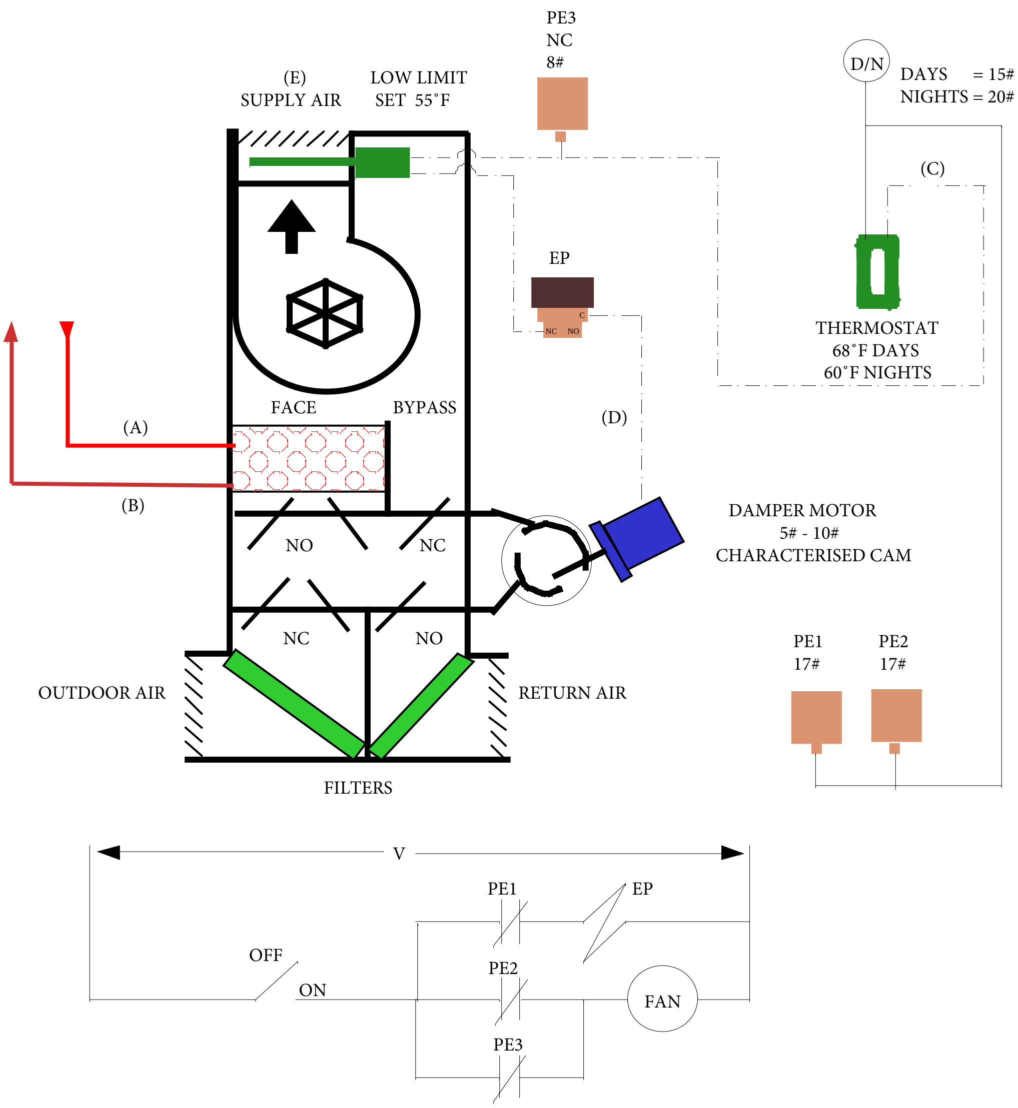
The system can be controlled equally well with either completely computer logic or completely pneumatic logic.

If you choose to use computer logic, each of the room thermostats in the sample rooms has to be replaced with a transducer to control the valve (or the valve can be replaced with an electronic modulating valve) and the software has to be able to determine the room with the greatest requirement for heating. The software then has to coordinate the supply of heat with the position of the local control valve in the coldest room.

If you choose to use pneumatic logic, you will tee into the existing branch lines of the sample room thermostats and often run them to a multi-low selector. A biasing relay will likely be used to establish an Energy Conservation Demand Temperature Signal.

The safety of the building occupants, the mechanical equipment and the building must be the first concern to a designer making changes to save energy. It is the responsibility of the designer to know every potential problem area for each system and address each one in a logical manner. No two buildings are exactly the same; therefore, no two conservation circuits are exactly the same.

FIGURE ONE

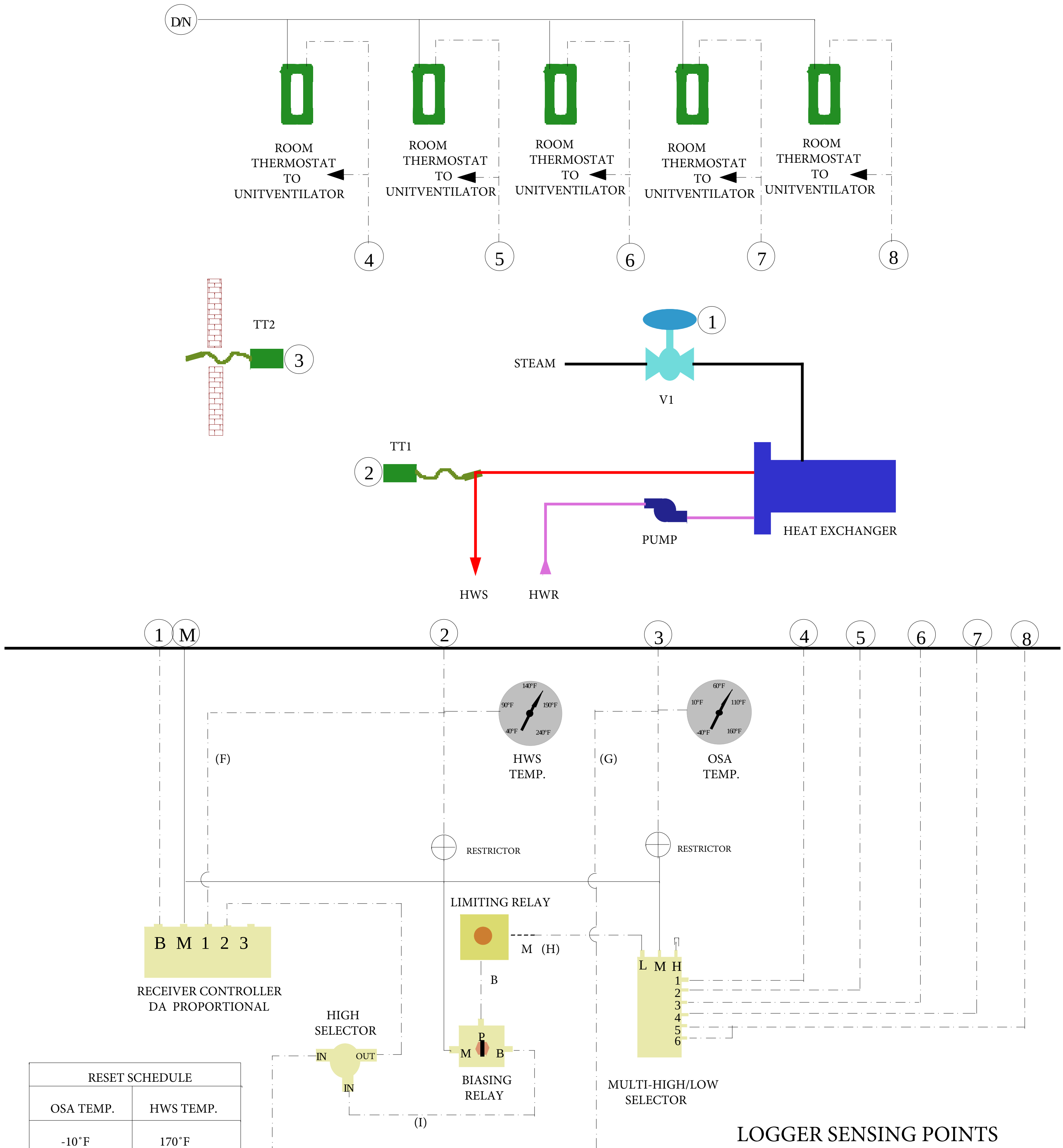


FACE & BYPASS UNITVENTILATOR

LOGGER SENSING POINTS

(A) Hot water supply temperature
(B) Hot water return temperature
(C) Thermostat branchpressure
(D) Low limit branch pressure
(E) Discharge air temperature

FIGURE TWO



RESET SCHEDULE	
OSA TEMP.	HWS TEMP.
-10˚F	170˚F
60˚F	80˚F

LOGGER SENSING POINTS

(F) Supply water temperature
(G) Actual OSA temperature
(H) Coolest room branch signal
(I) Energy demand signal PSIG

VAV PERFORMANCE CASE STUDY

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VAV PERFORMANCE CASE STUDY

THE PROBLEM

The building experienced many complaints of stuffiness and over-heating.

Three different air balancing companies calibrated and checked the VAV boxes and the building continued to have many complaints.

Analysts of Pneumatic Systems Limited (APS) was called to investigate. APS found the static pressure in the supply duct dropped to 0.4”WG in the afternoon, which is very low from the design set point of 1.0” WG. The inlet vanes were fully open, but the system was requiring more CFM (cubic feet of air) than the fan’s capability.

The VAV’s were pressure independent, with airflow controllers (see page 7.54), which should compensate for some drift in the supply air static pressure value, based on the instrument’s data sheet.

The drifting static pressure values in the supply duct would cause the air balancers’ efforts to become out of calibration if the varying static pressure had an impact.

We tested to see if the manufacturers’ data sheets were unconditionally true in the pressure independent claims.

TEST PROCEDURE

-1- We selected a VAV box and installed a o PSIG o 30 PSIG pressure gauge and 0” WG to .5” WG Magnehelic gauge allowing us to read the box’s velocity pressure and airflow controller’s branch pressure.

-2- We checked the damper motor stroking range which was 3# to 13#.

-3- We dropped the supply duct static pressure in steps, recorded the maximum velocity pressure, determined the relative CFM and recorded the branch pressure from the airflow controller as per the chart below.

SYSTEM PRESSURE (“WG)	MAXIMUMVELOCITY PRESSURE (“WG)	CUBIC FEET/MINUTE (CFM)	AIRFLOW CONTROLLER BRANCH PRESSURE (PSIG) (ACTUATOR RANGE 3# to 13#)
1.60	.200	450	6.3
1.50	.190	440	6.0
1.25	.180	430	6.0
1.00	.150	400	5.0
0.75	.125	370	4.0
0.60	.070	290	3.0
0.50	.060	270	0.0
0.20	.020	187	0.0

OBSERVATIONS

-1- The thermostat was demanding maximum cooling at all times through the test period.

-2- The box CFM dropped as the supply static pressure dropped.

-3- At .60" WG the damper motor, with a range of 3# to 13#, was still theoretically under control of the airflow controller, which was producing a branch pressure of 3 PSIG.

-4- From 1.60" WG to 0.60" WG the maximum controlled airflow dropped from 450 CFM to 290 CFM as the controlling branch pressure of the airflow controller dropped from 6.3 PSIG to 3.0 PSIG.

CONCLUSIONS

-1- In this test we witnessed a 35.5% drop in maximum air flow when the supply duct static dropped from 1.60" WG to 0.60" WG.

-2- The drop of 35.5% was when the airflow controller had continual control over the throttling range of the VAV box damper motor.

-3- The airflow controllers are proportional only control and the observations are exactly what a proportional controller should do. Proportional controls require an error from set point for the branch pressure to change. From a maximum air flow of 450CFM to a maximum air flow of 290 CFM the controls were exactly on calibration.

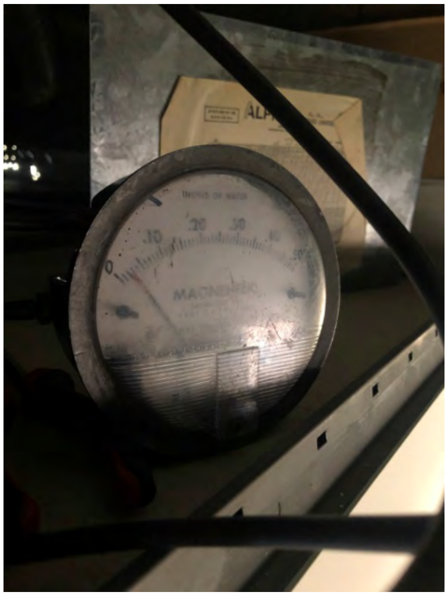
-4- Air balancers should be very sensitive to the supply duct static pressure, assuring it is at normal set point while adjusting VAV boxes.

-5- The term "pressure independent" creates the illusion that the supply duct static can vary without impacting the VAVs' performance. This is a false understanding.

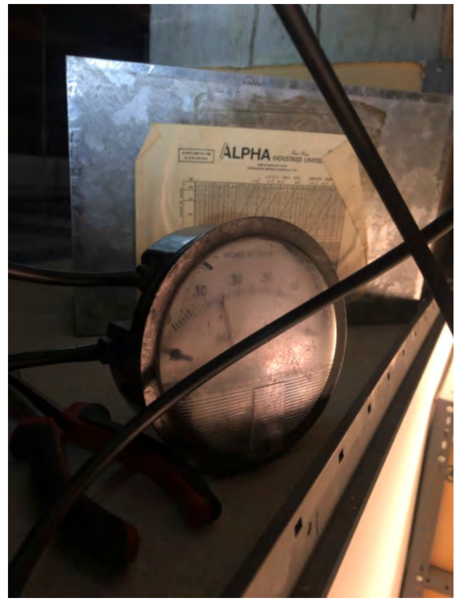
IMPACT OF DEFECTIVE AIRFLOW CONTROLLER OR BLOWN DIAPHRAGM ON A PNEUMATIC VAV BOX

VAV BOX IS 8" WITH AN AIRFLOW CONTROLLER RESET RANGE OF 8 PSIG TO 13 PSIG.

THERMOSTAT SIGNAL BELOW 8 PSIG;
THEREFORE, MINIMUM AIRFLOW SETTING.
VELOCITY PRESSURE .03" WG = 150CFM

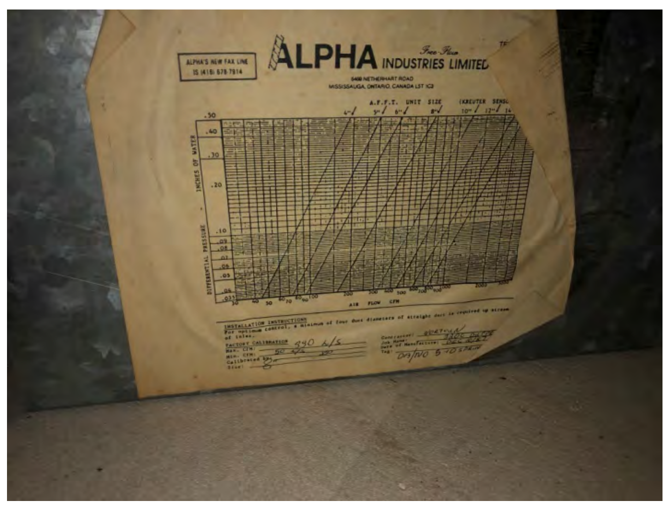


THERMOSTAT SIGNAL ABOVE 13 PSIG;
THEREFORE, MAXIMUM AIRFLOW SETTING.
VELOCITY PRESSURE .14" WG = 340CFM



BLOWN DIAPHRAGM
VELOCITY PRESSURE .23" WG = 450 CFM
(DAMPER WIDE OPEN)

VAV CALIBRATION CHART (8" BOX)



The diversity factor on a VAV system is the percentage the fan is undersized relative to the total of all the VAVs' maximum airflow settings. (Example: If a fan can pass 100,000 CFM and the VAV's total maximum capability is 115,000 CFM, the diversity factor is 15%.) This leaves the fan 15% undersized, but the sun cannot shine on all building faces at once, so maximum airflow should never be required and the duct static pressure setting of (example) 1" WG should be attainable.

The above example of defective airflow controller or blown diaphragm demonstrates that this VAV passes 32% more system air than when working properly at maximum air flow.

Once issues cause the system demand to exceed the example fan's 100,000 CFM the duct static pressure set point of 1" WG cannot be maintained and the duct static could drop to a level we have witnessed of, for example .4" WG.

The airflow controllers' data sheets claim to be pressure independent, but as per our VAV case study this is not true if the duct static variation is large. Calibrating a building at various duct static pressures can result in a very poor accomplishment.

Duct leaks up stream of the VAV's, dirty filters and coils, plus adding more VAV's to the system, without consideration, will also impact the system's ability to maintain proper duct static pressure.

HOUSE

NIGHT TEMPERATURE

REDUCTION

ENERGY

CASE STUDY

A P S

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CONTENTS

APS DOES NOT WORK ON RESIDENTIAL HEATING AND COOLING SYSTEMS. WE HAVE PREPARED THIS CASE STUDY SOLEY TO ADDRESS A COMMON MYTH REGARDING NIGHT SET BACK IN HOUSES.

INCORRECT INFORMATION RELATING TO RESIDENTIAL NIGHT SET BACK, NEEDLESSLY INCREASES OUR TOTAL POLLUTION IMPACT.

USING NIGHT SET BACK AND REDUCING YOUR DAYTIME TEMPERATURE WILL REDUCE YOUR CONTRIBUTION TO THE POLLUTION PROBLEM AND SAVE YOU MONEY.

THE BACK PAGE OF THIS REPORT IS A DRAWING ILLUSTRATING A CHANGE MADE TO A HOUSE IN 1981.

THE CHANGES IMPROVED THE COMFORT IN THE HOUSE AND REDUCED THE ENERGY CONSUMPTION, BY MAXIMIZING THE BENEFIT OF SOLAR GAIN.

PURPOSE OF CASE STUDY

Often people express the opinion that more heat is required regaining daytime temperature, than is saved over night, if night temperature reduction is used. Based on this opinion, they control their houses to daytime temperatures, both day and night.

This case study graphically disproves this opinion illustrating the energy use of one house under three operational scenarios.

(1) The house operating at 21°C (70°F) day and night.
(2) The house operating at 21°C (70°F) day and 14.5°C (58°F) night.
(3) The house operating at 17.2°C (63°F) day and 14.5°C (58°F) night.

This single point of false information **contributes cumulative damage to our children's future** by causing excessive emission of green house gasses into the atmosphere. We wish to address this situation, saving families money, as well as relieving the environment of the additional pollution burden. (Work with your heating experts to assess the best set up for your house.)

DATA COLLECTION METHOD

The component arrangement allowed reading and graphing the boiler's percentage firing time, the outside temperature and the indoor temperature.

The readings occurred every eight seconds and averaged into forty second logging points.

ANALYSIS OF DATA

We consider the relative run time of the boiler, while running under the three different scenarios, the indicator of comparative energy use.

GENERAL

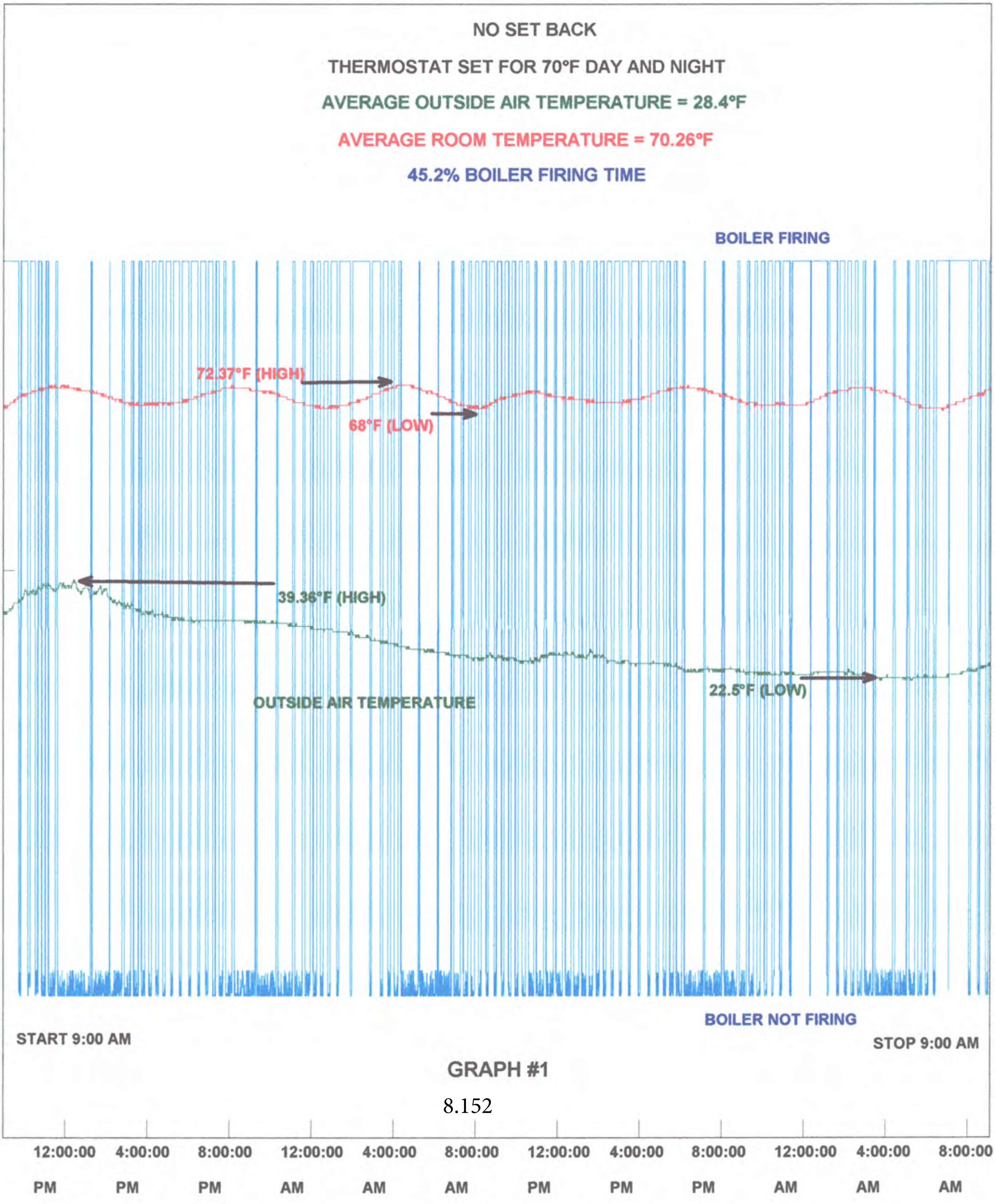
We have focused on improving building energy performance since 1976 in commercial, industrial and institutional environments. The vast majority of buildings required a multiple front approach to reducing energy use with night set back being only one of many techniques simultaneously employed. Although night set back has been a factor in nearly all, there are only two examples allowing isolation of the benefit relating to night set back.

(1) In 1976, we worked in cooperation with the Scarborough Board of Education assessing the relative savings as different techniques were employed. We used an identical building, built to the same specifications as our test building, as a control reference. The energy use reduction relating to only night set back was 15%, while the total energy reduction employing the three tested techniques simultaneously, was 43%.

(2) The oil section of Wendell Statton S., PS in Scarborough only enjoyed the benefit of night set back allowing a 19.5% reduction in energy use. (See chart page 8-158)

The dollar saving in this case study is significant, comparing the house at a constant 70°F to its normal operation; however, this case study is not about saving dollars. It is intended as one technical document, added to our collective effort, in the attempt to preserve the children's future.

HOUSE SET BACK COMPARISON



GRAPH #1

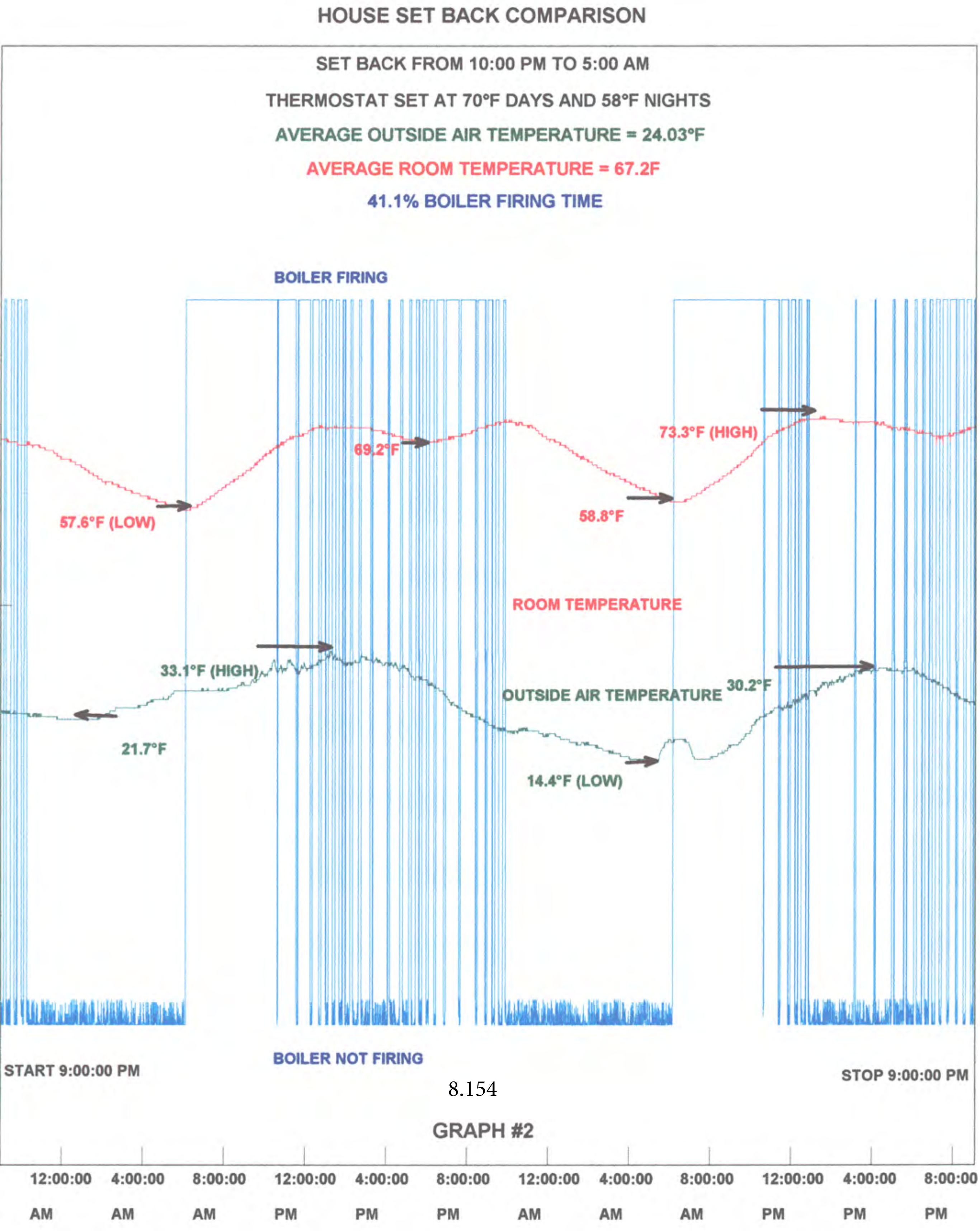
GRAPH #1

Graph #1 illustrates the energy performance of a house that has a hot water boiler with cast iron radiators.

The red line illustrates the room temperature, the green line illustrates the outside air temperature and the blue line illustrates the run time of the burner.

The computer trending program provides the highest, the lowest and the average value for each of the three channels.

The boiler was required to run 45.2% of the time, maintaining the house at an average temperature of 70.26°F, with an average outside air temperature of 28.4°F.


HOUSE SET BACK COMPARISON
SET BACK FROM 10:00 PM TO 5:00 AM
THERMOSTAT SET AT 70°F DAYS AND 58°F NIGHTS
AVERAGE OUTSIDE AIR TEMPERATURE = 24.03°F
AVERAGE ROOM TEMPERATURE = 67.2F
41.1% BOILER FIRING TIME
BOILER FIRING
73.3°F (HIGH)
69.2°F
57.6°F (LOW)
58.8°F
ROOM TEMPERATURE
33.1°F (HIGH)
OUTSIDE AIR TEMPERATURE
30.2°F
21.7°F
14.4°F (LOW)
BOILER NOT FIRING
START 9:00:00 PM
STOP 9:00:00 PM
GRAPH #2
12:00:00 AM
4:00:00 AM
8:00:00 AM
12:00:00 PM
4:00:00 PM
8:00:00 PM
12:00:00 AM
4:00:00 AM
8:00:00 AM
12:00:00 PM
4:00:00 PM
8:00:00 PM

GRAPH #2

Graph #2 illustrates the energy performance of the same house with the thermostat reprogrammed to control at 70°F during the day and 58°F during the night.

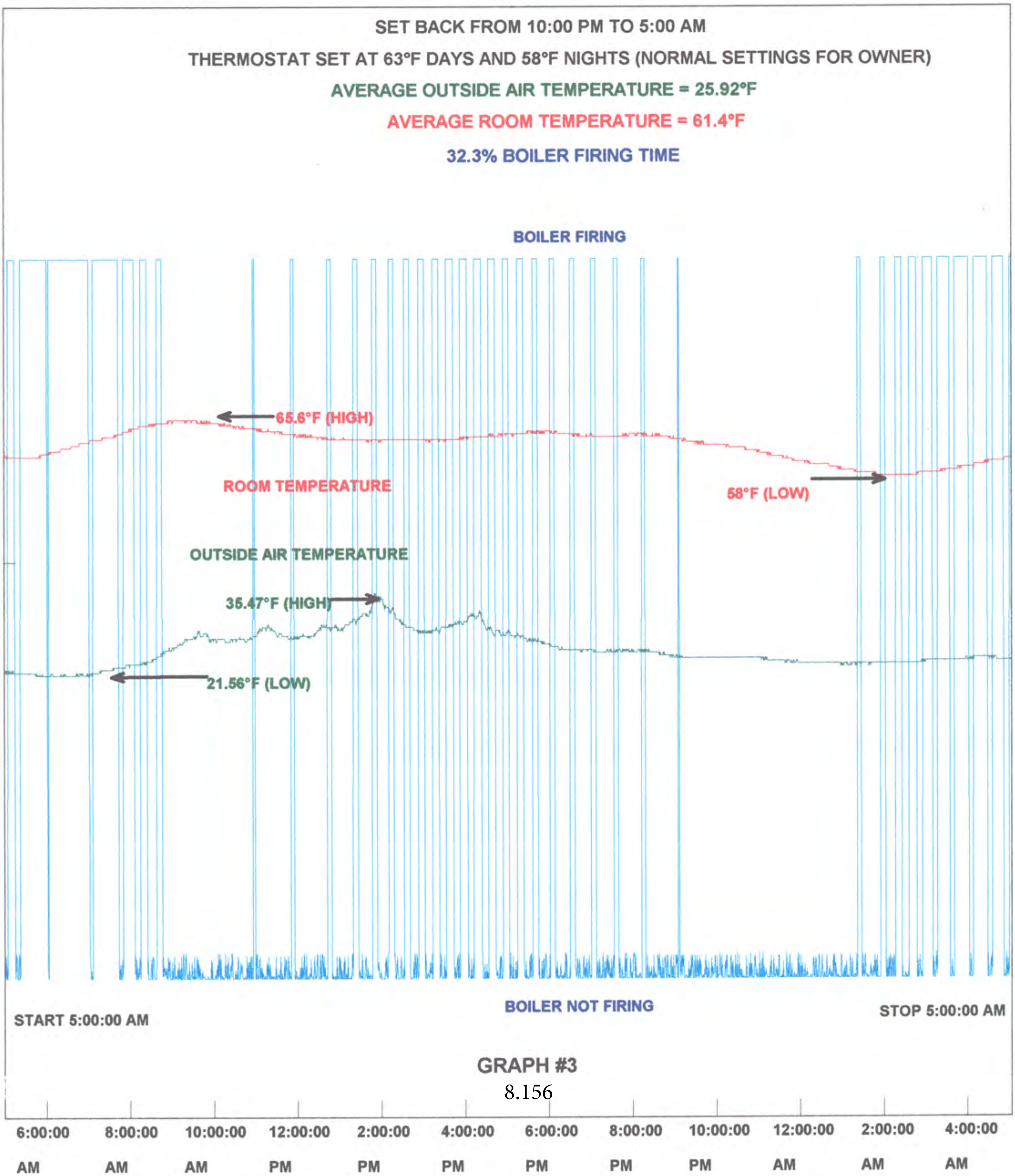
The red line illustrates the room temperature, the green line illustrates the outside air temperature and the blue line illustrates the run time of the burner.

The computer trending program provides the highest, the lowest and the average value for each of the three channels.

The boiler was required to run 41.1% of the time, maintaining the house at 70°F during the day and limiting the house to a minimum of 58°F during the night with an average outside air temperature of 24.03°F.

The boiler run time dropped by 9% using the night set back schedule; despite the fact that the outside air average temperature was 4.37F° colder than the first scenario.

HOUSE SET BACK COMPARISON



GRAPH #3

GRAPH #3

Graph #3 illustrates the energy performance of the same house with the thermostat reprogrammed to control at 63°F during the day and 58°F during the night. This programming reflects the normal values of operation that suit the owner.

The red line illustrates the room temperature, the green line illustrates the outside air temperature and the blue line illustrates the run time of the burner.

The computer trending program provides the highest, the lowest and the average value for each of the three channels.

The boiler was required to run 32.3% of the time, maintaining the house at 63°F during the day and limiting the house to a minimum of 58°F during the night with an average outside air temperature of 25.92°F.

The boiler run time dropped by 28.5% from the first scenario using the reduced day setting with the night set back schedule; despite the fact that the average outside air temperature was 2.48F° colder.

From our environment's viewpoint, the increase in run time from 32.3% in the third scenario, to the 45.2% in the first scenario, represents an increased energy consumption of 39.9%.

THE BOARD OF EDUCATION FOR THE BOROUGH OF SCARBOROUGH

CONTROL MODIFICATIONS BY

A P S

SCHOOL	UTILITY	ANNUAL REDUCTION	1979 UTILITY COST	UTILITY SAVINGS **	$	IMPROVEMENT COST	PAYBACK YR.
MILITARY TRAIL JR. PS	GAS	31.1%	9,863	2315	4,073	4,072	1
	ELECTRIC	20.9%	11,217	1758			
JACK MINER PS	GAS	20.4%	14,338	2194	4204	3,287	0.8
	ELECTRIC	17.8%	15,059	2010			
ALEXMUIR JR. PS	GAS	29.5%	7,308	1617	3515	2,699	0.8
	ELECTRIC	24.6%	10,285	1898			
SILVER SPRINGS PS	GAS	37.8%	6,425	1822	3,973	1943	0.5
	ELECTRIC	36.6%	7,835	2151			
WENDELL STATTON SR. PS	GAS	57.6%	10,221	4415	10,114	5300	0.5
	OIL	19.5%	16,463	3210			
	ELECTRIC	22.6%	14,684	2489			
TIMBERBANK PS	GAS	17.1%	8,142	1044	2,784	2110	0.8
	ELECRIC	22.1%	10,499	1740			
WEST HILL C.I.	OIL	8.1%*	46,552	3771	24,419	36,370	1.5
	ELECTRIC	10.9%*	51,974	4249			
	GAS	51.8%*	42,211	16,399			
				TOTALS	53,082	55,781	1.05

- * - PART YEAR ONLY
- ** - GAS AND ELECTRIC @ 75% OIL @ 100%

JRM/sc June 18, 1981

LETTER FROM SCARBOROUGH BOARD ACCOMPANYING CHART ABOVE

Gentlemen:

During early 1979 control improvements were carried out by your firm on a number of our schools. These schools are listed on the attached schedule which indicates the savings that have been achieved.

For clarification purposes, it should be noted that:

a) no allowance has been made for the fact it was 4.7% colder in 1980 than in 1979.

b) The utility costs are 1979 actuals and no allowance has been made for escalation.

c) The majority of the savings are undoubtedly higher as the modifications were not in effect for the entire year.

d) We have assumed only 75% of the actual gas and electricity savings because of the sliding scale rate structures.

e) Approximately $17,000. Is included in the cost of improvements at West Hill Collegiate for other work that was performed at the time aimed primarily at improving poor environmental conditions.

It is almost needless to say that we are very pleased with the results and the manner in which they were carried out.

Yours very truly
J.R. Mazanik

A system was designed and installed as per figure "A".

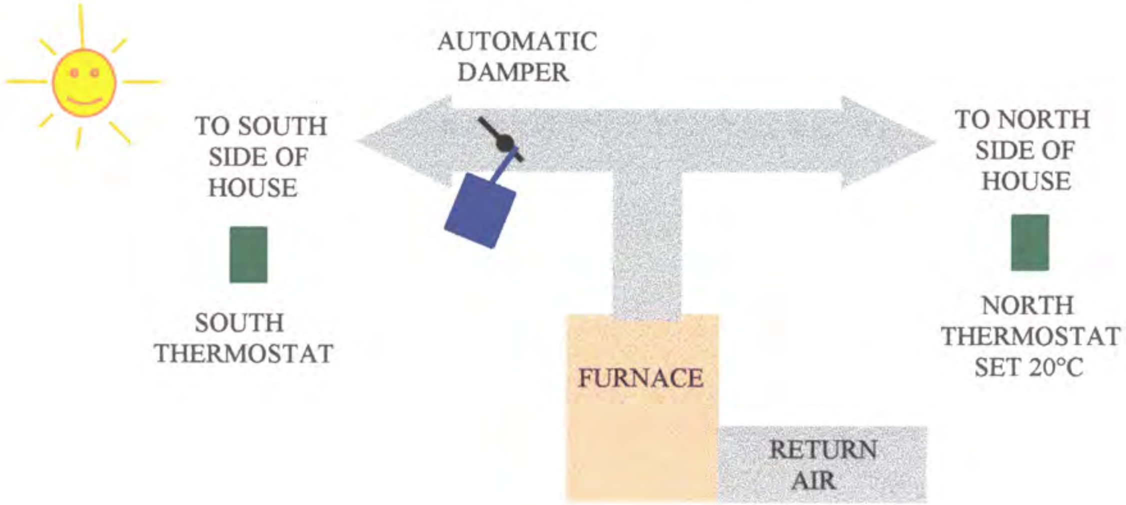


FIGURE "A"

The three modes of operation allowed;

(1) If the south side is below 20°C, the automatic damper is open and the north thermostat operates to heat the whole house with the furnace fan at medium speed.

(2) If the south side is between 20°C and 23°C, the automatic damper closes and the north thermostat operates heating only the north side of the house with, the furnace fan at low speed. (At these times, the solar gain is heating the south side of the house.)

(3) If the south side is over 23°C, the automatic damper opens and the furnace fan runs at high speed. (This mode uses the heat of the south side to warm up the north side and the cooler air from the north side to reduce the over-heating on the south side. If the north thermostat activates the heating during this mode, the automatic damper closes until the gas heating terminates.)

The alterations improved the comfort levels and reduced the energy use of the house.

NOTES

SECTION NINE
(COMPRESSED AIR CARE)

PURPOSE OF SECTION

Provide suggestions relating to filtration and air leaks.

COMPRESSED AIR

SOLUTIONS

FOR YOUR

PNEUMATIC CONTROL

SYSTEMS

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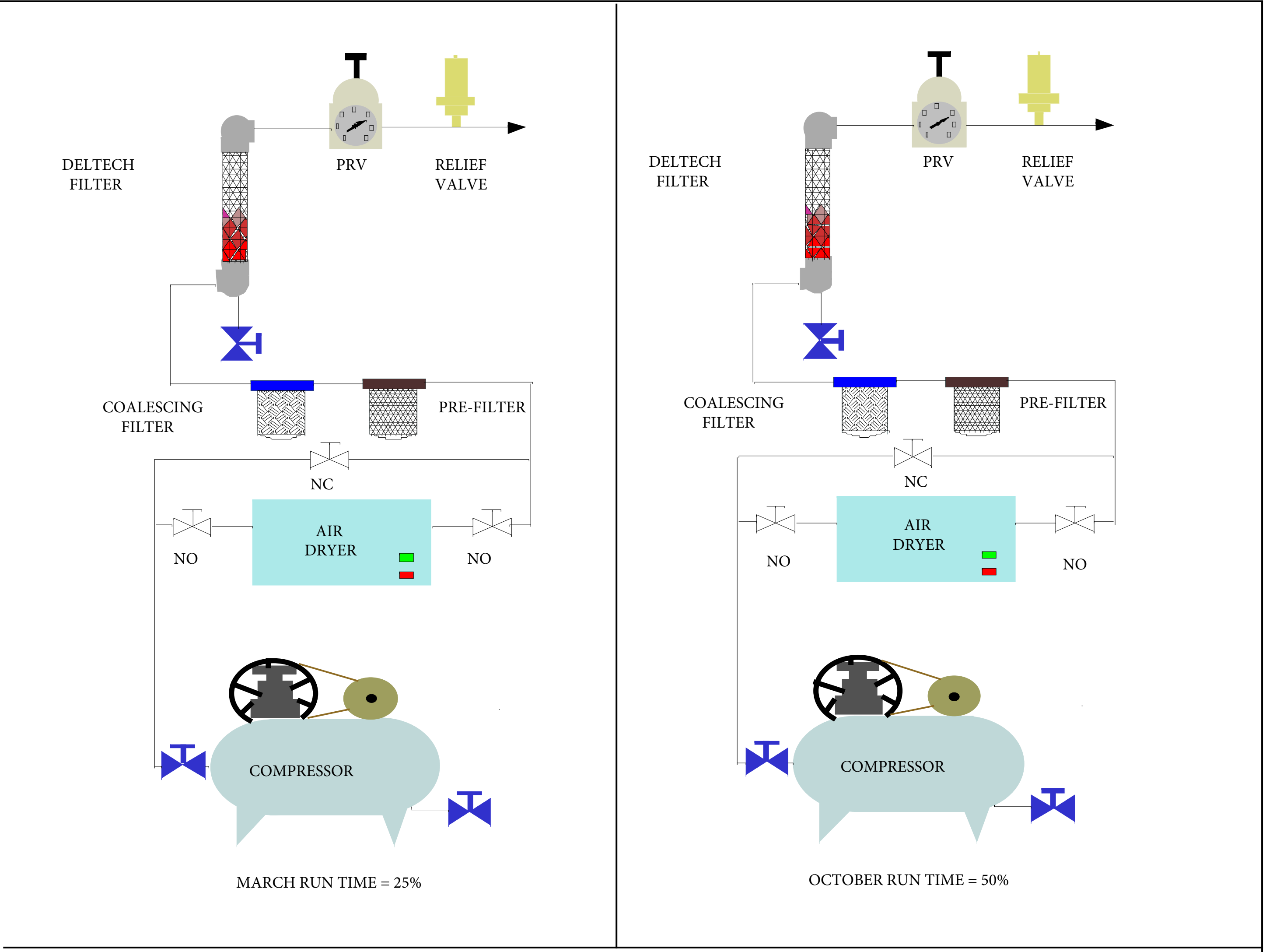
http://www.apscontrols.ca
PH: (905) 640-2333
FX: (905) 640-2444

CONTENTS

AIR COMPRESSOR ISSUES



-1- You time your compressor in March at 25%.

-2- You time your compressor in October at 50%.

-3- Do you spend a lot of time looking for a leak to find the compressor pump is the problem?

-3-Do you replace your compressor pump to find that the old one is fine and there is a leak in the system?

-4- What will you do to determine the proper corrective path?

-5- If you determine that there is a leak in the system do you have a logical path to follow for correction?

FIGURE A

SUPPLY AIR CARE FOR CONTROL SYSTEMS

THE PROBLEM

Most pneumatic control systems have no means of alerting building operators to air leaks. Excessive compressor run time is normally the only indicator available, which may result from deterioration of the compressor or air leaks in the control system. **Both generate the same symptom. Which is causing the current problem?**

Some causes of leaks in your control system are:

(1) Diaphragms rupture in control components.

(2) Vandals break thermostats off walls.

(3) Renovation workers break airlines.

(4) Cable installers break airlines.

Leaks in your control system will:

(1) Cause poor control in areas of low pressure.

(2) Cause changeover controllers to be on the wrong mode for summer/winter or day/night.

(3) Cause your compressor to run excessively.

Excessive run time on the compressor will:

(1) Shorten the life expectancy of your compressor.

(2) Waste electricity.

(3) Over-heat the pump, increasing the probability of passing destructive oil in to the control system.

Having no means to assess the cause of your compressor's excessive run time, you may:

(1) Choose to change the compressor pump finding that the new unit runs excessively as well. The real problem being air leaks in the control system.

(2) Choose to dedicate days of labour looking for air leaks that do not exist. The real problem being a deteriorated compressor pump.

This report contains a simple solution allowing you complete confidence and control, identifying which path leads to correcting your problem.

THE SOLUTION

Do a compressor station analysis complete with a new air flow meter and CAD drawing documenting your system.

The benefits enjoyed are:

(1) A current database is created relating to your compressed air station performance relative to a new unit.

Figure "C", page 9-169, is a sample data sheet from a building with this advantage.

(2) The flow meter provides a simple visual indicator of air leak problems in the system. Operators can identify when air leaks appear in a system immediately allowing correction before associated problems occur.

Figure "B", page 9-165, is a document illustrating the assessment procedure determining if excessive compressor run time is the result of air leaks or compressor deterioration.

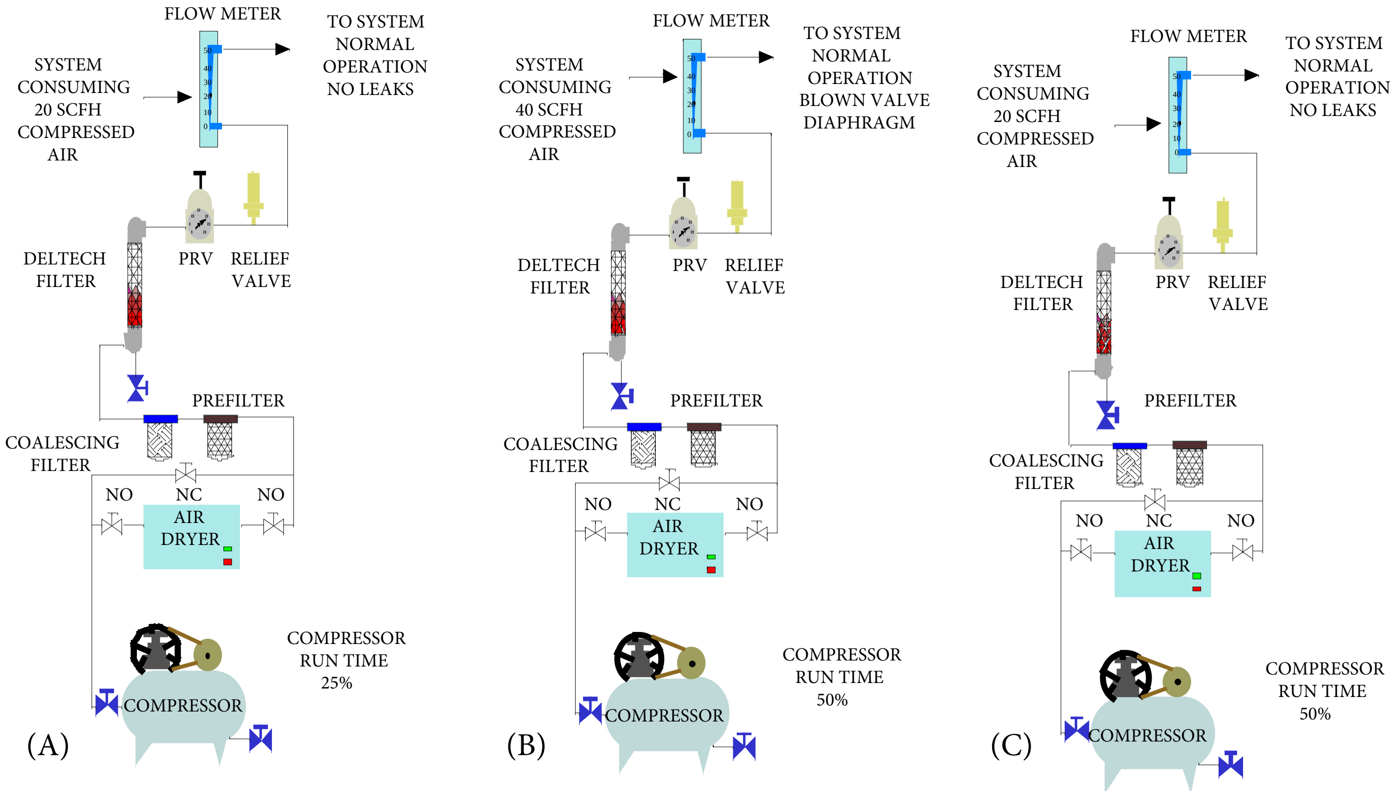
(3) The flow meter provides critical information during the search for air leaks making the procedure logical and as efficient as possible.

The procedure is explained on pages 9-166, 9-167 and 9-168.

Compressor service data sheets, similar to the one on page 9-171, might be used establishing a documented history of the performance of your compressor station.

ASSESSING EXCESSIVE COMPRESSOR RUN TIME

FIGURE B



THE PROBLEM

-In this example the compressor run time is normally 25% as illustrated in (A); however, the run time has increased to 50% as illustrated in (B) and (C). The excessive run time wastes energy, shortens the compressor life expectancy and increases the probability of getting oil into the control system.

THE QUESTION

-Is the excessive run time the symptom of a leak in the control system or a decrease in the compressor performance? Both conditions create the same symptom at the compressor.

THE ANALYSIS

-The flow meter in scenario (A) indicates that the system normally consumes 20 CFH (Cubic Feet Per Hour) of air, at a compressor run time of 25%, which you initially recorded when you installed the meter.
-If (B) illustrates your current situation with the flow meter reading 40 CFH, at a compressor run time of 50%, you know that the problem is a leak in the control system as it normally should only consume 20 CFH.
-If (C) illustrates your current situation with the flow meter reading 20 CFH, at a compressor run time of 50% you know there is a problem with your compressor station as the control system is consuming its normal 20 CFH.

LEAK ANALYSIS VIA A FLOW METER

Looking for an air leak in a large building can be like looking for a needle in a haystack, if a logical approach is not employed.

We suggest using the information provided by a flow meter in the main air supply line of the control system. When using the flow meter information, one can break the building down into three potential areas where the leaks may occur.

(1) The exhaust fans.

(2) The supply fan systems.

(3) The thermostat loops associated with the main air distribution system.

CHECKING FOR LEAKS AT THE EXHAUST FANS

(1) Using walkie-talkies have one person at the flow meter and one person at the central control to stop and start the exhaust fans.

(2) Note the airflow reading at the meter with all the exhaust fans turned off. Turn the exhaust fans on one at a time and note the air flow meter reading after each one has started. The person at the meter should notice an initial surge when an exhaust fan with pneumatic actuator starts, but the meter should settle to the same point of the fan not running, after the actuator has filled.

(3) If the airflow reading is the same with the exhaust fans on or off, there is no leak associated with the exhaust fans; therefore, start with the supply fan air leak investigation.

(4) If the air flow meter jumps up when an exhaust fan starts and remains up, you likely have a leak either on the line from the solenoid to the pneumatic actuator or the diaphragm in the actuator has a leak.

(5) If the airflow meter jumps up when an exhaust fan is turned off, you likely have a leak in the solenoid air valve from the normally closed port to the normally open port and the solenoid should be replaced.

CHECKING FOR AIR LEAKS AT SUPPLY FANS

After your control service mechanic has verified that there are no air leaks in your fan systems, turn the units on and off, individually, while reading the normal air consumption for each unit. Use the chart on page eleven of this report to record the normal air consumption for each unit. This information will provide value during future assessments regarding air leaks.

If you have not noted the normal consumption per fan system, you can estimate the consumption by counting the instruments on the system and allow the manufacturer's controller value relating to estimated air consumption per instrument.

Follow the same manning arrangement as with the exhaust fans. One person should be at the flow meter and the other at central control to operate the fans. Stop and start the fans one at a time while noting the air consumption relative to the expected air consumption for each fan system.

If one fan is using more air than expected, investigate the cause at the fan system.

THERMOSTAT LOOPS AND MAIN AIR DISTRIBUTION

If the exhaust fans and the supply fans prove to have normal consumption, you have narrowed the likely location of the leak(s) to the third and largest component of your pneumatic system, the thermostat loops and main air distribution piping.

The investigation, at this stage, must be approached logically to reduce frustration and costs.

We suggest that you obtain a floor plan of the building to plot the maximum branch pressure readings establishing a pressure picture of the whole piping system. The areas of the lowest pressures are the most likely areas to find significant air leaks.

TRAINING YOUR EAR TO THE DIFFERENT SOUNDS OF A THERMOSTAT MAY ASSIST YOU IN FINDING LEAKS

The normal air usage of a relay thermostat produces a local hissing noise, while air passing through a thermostat relay, to a leaking diaphragm, makes a lower rushing sound.

To train your ear, at a practical level, find a thermostat in a non-critical location.

(1) Turn a direct acting thermostat to full heat and listen to the hissing sound of the pilot air (approximately .5 SCFH of compressed air) which most thermostats are designed to bleed. (Powers "D" stats do not bleed any air except when reducing its branch pressure.)

(2) Turn the setting of the thermostat down and listen to the air rushing through the thermostat relay to fill the diaphragm of the controlled device. The air volume you hear is about 27 SCFH to 70 SCFH, depending on the model. (See T4002 consumption graph, page 3.27.) The sound of the rushing air will taper off to silence when the diaphragm is completely filled at the main supply line pressure.

(3) Now you are going to simulate a blown diaphragm. Turn the thermostat down completely. Go to the controlled device (valve or damper actuator) and disconnect the airline. Go back to the thermostat and listen to the sound it is making. This is the same sound the thermostat makes while filling the diaphragm, except it does not taper off to silence because the air continues to rush to atmosphere via the simulated (or real) blown diaphragm.

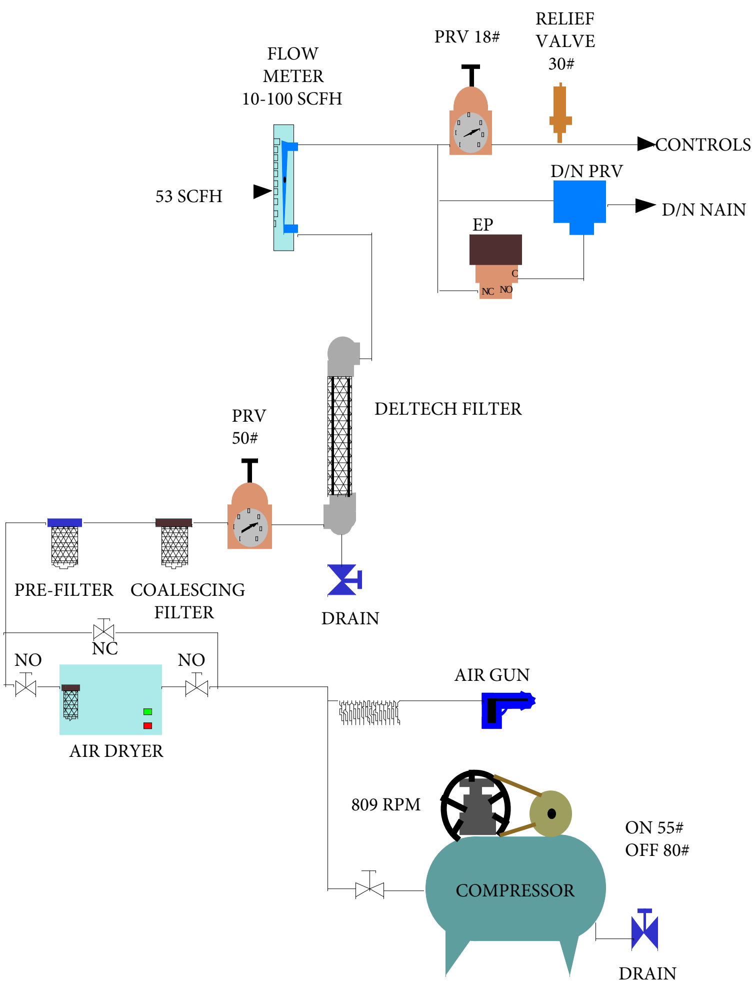
Knowing the difference between the sounds of a thermostat assists greatly in finding and correcting leaks. REMEMBER TO RECONNECT THE BRANCH LINE AT THE VALVE OR DAMPER ACTUATOR AFTER DOING THIS EXERCISE.

CAUTION

Be sure that you do not turn off an exhaust fan or supply fan without checking with the building management. You may cause serious problems if the wrong fan is turned off at the wrong time.

Be sure to check with the building management for the appropriate time to install the flow meter. Advise the building management of the likely impact on parts of the building when control air is turned off during the meter installation. (For example: The heating will likely open fully, fresh air and exhaust air dampers will close, etc.)

FIGURE C



NOTE:
ALWAYS TURN OFF THE COMPRESSOR WHILE SERVICING!

-When the red area on the Deltech filter advances by 1/2" from the last time you had the coalescing element changed, this indicates that oil is passing by the coalescing filter. At this time submit a work order to have the coalescing element changed and mark the top of the red advancing area on the Deltech filter element with a felt pen to guide you for the next 1/2" change to indicate the next required element change.

-When the Deltech filter turns red over about 2/3 of its lower area, submit a work order to have both elements changed.

-Time the compressor's run time once a month. Record this run time as well as the flow meter reading. Send recorded figures to the energy department at end of each year. (Run time/run time + rest time X 100 = % run time)

-Change oil in compressor every six months. FILL TO PROPER LEVEL. EXAMPLES: To check Devilbiss, with dipstick, screw dipstick in all the way. If no dipstick, on Devilbiss, fill to first thread below oil fill port opening. (The correct oil is none detergent 20 that is available from the stock room.)

-Report any unusual sounds, air leaks, large increases in compressor run time or large increases in flow rate at the time of discovery.

-Check compressor belt, with the compressor turned off, for cracking and tension when the oil is being changed. Replace belt at first sign of cracking. The belt should flex by your finger tension about 3/4" at the middle of the belt.

-Use the air gun to clean the air dryer coil. Wear eye safety glasses. Do not use the air gun to clean yourself or your clothing.

MANUFACTURER	DEVILBISS
MODEL	123
HORSE POWER	1
TIME TO FILL TANK (0# T0 80#	5 MIN. & 30 SEC.
TANK SIZE	60 GAL
PERCENTAGE RUN TIME	40%
UNIT CFM @ 80 PSIG	3.1
BELT	41580
INTAKE ELEMENT	M2003 (3" PAPER)

FIGURE D

ANALYSTS OF PNEUMATIC SYSTEMS LIMITED *(APS)*
COMPRESSOR STATION DATA LOG SHEET

BUILDING ______________________________________

YEAR ________ MONTH	FLOW METER READING	% RUN TIME	BELT CHECK	FILTER ELEMENTS' CHECK	OTHER COMMENT	MECH. INITIALS
JAN.						
FEB.						
MARCH						
APRIL						
MAY						
JUNE						
JULY						
AUGUST						
SEPT.						
OCTOBER						
NOV.						
DEC.						

NOTE:

- RUN TIME + REST TIME = TOTAL TIME
- RUN TIME/TOTAL TIME x 100 = % RUN TIME
- IF THE % RUN TIME INCREASES BY MORE THAN 10% FROM ANY PREVIOUS MONTH REPORT TO PLANT OFFICIALS.

SERVICE DATA FOR AIR STATIONS

BUILDING: ___________________ **LOCATION:** ___________________

DATE: ___________________ **TECHNICIAN:** ___________________

COMPRESSOR:

MANUFACTURER: _____________________ **MODEL:** ____________ **RATED AMPS**_______

HORSE POWER: ______________________ **VOLTAGE:** __________ **PHASE:** _______

AMPS AT CUT OUT: ____________________ **FRAME:** __________ **PUMP:** _______

FILTER ELEMENT:_____________________ **OIL TYPE:** __________ **BELT SIZE:** _______

AIR FLOW METER READING: NOW_________________ **AT INSTALLATION**__________________

% RUN TIME_______________ **CHANGE OIL**_________ **CHANGE INTAKE FILTER**____________

BELT________ **SPACE FROM WALL**________ **TANK CHECK**________ **UNLOADER**____________

GENERAL NOTES: ____

AIR DRYER

MODEL_____________ **MOISTURE UNLOADER**_______________ **FREE AIR FLOW**____________

ROOM TEMP. ______ **CONDENSER TEMP.** _______________ **CLEAN COIL** ____________

GENERAL NOTES: ____

SUPPLY AIR FILTERS:

PREFILTER---------------------MAKE: ________________ **ELEMENT:**__________ **CHANGED**_______

COALESCING FILTER------MAKE: ________________ **ELEMENT:**__________ **CHANGED**_______

DELTECH FILTER MODEL_______________________ **ELEMENT:**__________ **CHANGED** _______

GENERAL NOTES:____

A P S

3.4

FAN SYSTEM NORMAL AIR CONSUMPTION

FAN SYSTEM	NORMAL AIR CONSUMPTION (SCFH)

COMPRESSED AIR FILTRATION

THE PROBLEM

Most pneumatic air supply stations, for controls, have a coalescing filter. They are excellent filters; however, the indication to change the filter element is based on a pressure drop through the filter at a specific flow rate. (Often a 10 PSIG pressure drop at a 10 SCFM flow rate.)

It is extremely rare to find a control air supply at exactly 10 SCFM; therefore, the pressure drop is not relevant in most cases. (A 7.5 HP compressor, running 33% of the time, should produce approximately 10 SCFM.) Most operators do not have a reliable means to determine when the element requires changing.

We often witness elements saturated to the point of allowing oil into the control system. In one case, the element was laying in the bottom of the filter bowl.

The operators did not have an indicator to trigger correction of the dangerous situation.

THE SOLUTION

Install a Deltech filter after your existing filter station to act as an "INDICATOR/SAFETY NET"**.** (See the photograph on page 9.174.)

As an "**INDICATOR**", the element of the Deltech filter turns a deep red as it absorbs oil from the control air passing through.. The red starts at the bottom of the element and gradually advances to the top. (See photograph on page 9-175.) The red colour will not start its advance until the upstream coalescing filter starts to fail and pass oil.

When the red advances about one-half of an inch, change the coalescing filter element. Follow this procedure with each new coalescing element, until the Deltech filter is red about two thirds of the element's height. At this time, change both the coalescing and Deltech filter elements.

As a "**SAFETYNET**", the Deltech filter captures oil that passes the coalescing filter, as well as filtering particles down to about one micron in size.

CLEAN, DRY CONTROL AIR IS THE MOST IMPORTANT REQUIREMENT OF A RELIABLE PNEUMATIC CONTROL SYSTEM. YOU MUST GIVE THIS THE ATTENTION IT DESERVES TO AVOID SEVERE PROBLEMS AND ASSOCIATED COSTS. FIGURE "E" ON PAGE 9-176 ILLUSTRATES A SUGGESTED FILTER ARRANGEMENT.

4.2

DELTECH FILTER

4.3

DELTECH FILTER ELEMENTS
CLEAN AND LOADED

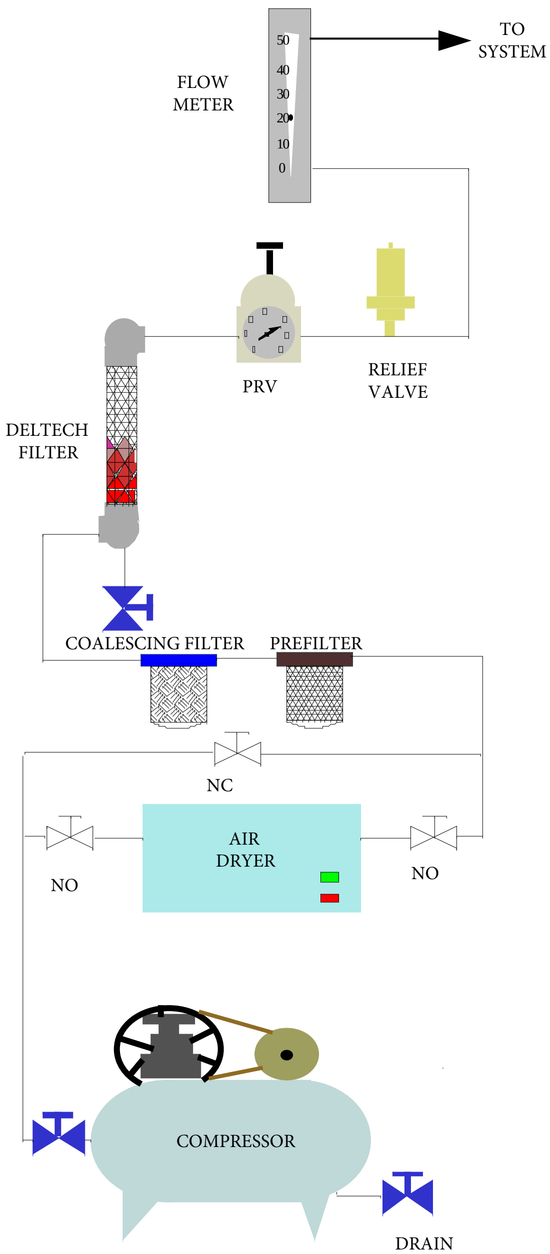
TOP

BOTTOM

NOTE:

New elements range from white to a pinkish white in colour.
Elements turn deep red from the bottom as they load with oil.

FIGURE E



NOTE:

The Deltech filer element changes colour from a pinkish/white to a deep red as it loads with oil. The change to red gradually climbs up the pinkish/white element over time exposure to oil.

The Deltech filter's intent in this arrangement is to act as a safety net / indicator.

1) As a safety net the Deltech captures and contains oil that passes by the pre-filter and coalescing filter if there is a filter failure.
2) As an indicator, the Deltech filter visually alerts the operator that the coalescing filter is passing oil and it is time to change its element. (The industry means of determining when to change elements of most coalescing filters is a specific pressure drop at a specific air flow rate. This is ineffective in most applications we have observed.)
3) We suggest changing the Deltech filter element when it has become red over approximately two-thirds of its element.

NOTES

SECTION TEN
(ANSWERS)

PURPOSE OF SECTION

Provide answers to questions. There are usually more than one method of providing requirements.

CONTROL QUIZ

The underlined answer is correct.

PNEUMATIC THERMOSTATS

-1- In a reheating application you may find a reverse acting thermostat controlling a normally closed valve.

<u>True</u> False

-2- Two pipe relay thermostats usually are capable of passing a greater volume of compressed air than a one pipe thermostat.

<u>True</u> False

-3- Day/night and summer/winter thermostats alter modes via a change in main air pressure.

<u>True</u> False

-4- There are several different pressure combinations used for day/night and summer/winter change-over in the industry.

<u>True</u> False

-5- Relay type thermostats bleed a small amount of compressed air most of the time.

<u>True</u> False

-6- Two thermostats are required to control a room with a VAV box and a heating coil.

True <u>False</u>

-7- If the flapper nozzle (leak port) is plugged on a direct acting thermostat, controlling a normally open heating valve, the room likely will over-heat.

True <u>False</u>

-8- If an operator accidentally bends the bimetal away from the flapper nozzle (leak port) on a direct acting thermostat, controlling a normally closed cooling valve, the room will likely over-heat.

<u>True</u> False

-9- The sensitivity of a thermostat is the relationship of temperature change to output signal (branch signal) change.

<u>True</u> False

-10- The sensitivity can be changed on most thermostats, but recalibration is required after that adjustment.

<u>True</u> False

-11- If the main air and branch signal connections are reversed, the thermostat typically makes an air hissing noise.

<u>True</u> False

-12- If the main air and branch signal connections are reversed the thermostat will still control, but a little slower to react.

True <u>False</u>

-13- If a thermostat is controlling a normally open heating valve and a normally closed cooling valve with identical spring ranges, the temperature will control, but energy will be wasted.

<u>True</u> False

-14- An air leak on the branch of a relay thermostat can consume more compressed air than fifty extra thermostats working normally on the system.

<u>True</u> False

-15- The maximum volume of compressed air thermostats can pass out their branch lines varies based on different manufacturers.

<u>True</u> False

-16- In an existing system, another manufacturer's thermostat may usually be used as long as the change-over point is adjusted to match the existing system and the thermostat is the same action.

<u>True</u> False

-17- When calibrating a day only thermostat you set the dial at the current room temperature and adjust the branch signal to midrange of the valve or damper being controlled.

<u>True</u> False

-18- The change-over point is not adjustable on summer/winter or day/night thermostats.

True <u>False</u>

-19- In one stage of calibrating a thermostat, you must set the branch pressure for mid-point of the controlled device and use your breath to heat the bi-metal, checking for proper operation.

<u>True</u> False

-20-When you heat the thermostat in question 19, the pressure will go lower with a direct acting thermostat.

True <u>False</u>

<u>VALVES</u>

-1- Normally open valves allow their maximum fluid flow when their diaphragm pressure is below the bottom end of the spring range.

<u>True</u> False

-2- The bench stroking spring range of a valve is the pressure difference between the pressure required to start the stroking of the valve and the pressure when the valve is fully driven.

<u>True</u> False

-3- The stroking spring range of a valve is the same on the bench as it is in service with the pumps running or steam pressure present.

True <u>False</u>

-4- The stored force in the spring of some control valves can be as high as 1800 pounds, setting the potential for injury or death if disassembled incorrectly.

<u>True</u> False

-5- It is best to remove the stem from the packing box to completely clean the stem and packing box before installing the new packing rings.

True False

-6- Two valves the same pipe size will not have the same flow capability if their CV factors are different.

True False

-7- There is no difference between three-way diverting valves and three-way mixing valves.

True False

-8- Normally open valves are usually installed on heat exchangers serving domestic hot water.

True False

-9- The system differential pressure of the water or steam tends to widen the in service stroking spring range compared to the bench stroking spring range of a valve.

True False

-10- A valve installed backward in a system may cause a hammering noise as it attempts to modulate.

True False

-11- A pilot positioner on a valve eliminates the effect of system differential pressure on the valve`s stroking range.

True False

-12- When the hot water supply temperature is hotter than required by the served area with the greatest heating requirement, the seats and discs on the system's valves tend to experience more wear and tear, shorting the valves' life span.

True False

AUTOMATIC DAMPERS

-1- The normal position of a damper is the position it reaches with no air pressure on the actuator.

True False

-2- Most often the fresh air and exhaust air dampers are normally open and the return air damper is normally closed.

True False

-3- There is never danger of injury from an actuator`s spring when replacing a diaphragm.

True False

-4- Exposing an actuator to pressure higher than recommended by the manufacturer could result in damage to the building, personal injury or death.

True False

-5- The return air damper should start closing before the fresh air damper starts opening.

True False

-6- If the return air damper is closed and the fresh air damper is closed there is no danger of collapsing duct work.

True False

CONTROLLERS

-1- A controller, with its sensing mechanism, senses the controlled condition directly.

True False

-2- Normally a controller has an averaging sensing element when sensing the mixed air.

True False

-3- If a gas or fluid filled sensing element is broken, the controller will believe the temperature suddenly increased.

True False

-4- Sensitivity and throttling range are the same. They relate to the relationship of the output pressure change relative to a specific temperature change. (example: 5#/F°)

True False

-5- Normally the slower the controlled condition potentially changes, the higher the sensitivity (lower the throttling range) (lower the proportional band) on the controller.

True False

-6- A controller acting as a mixed air low limit typically has a set point of 55°F to 60°F.

True False

-7- The sensitivity is not normally adjustable on controllers.

True False

-8- Throttling range is the amount of the sensed variable change required to stroke the controlled device from fully open to fully closed.

True False

-9- A single controller can be used to satisfy only one function in a control circuit.

True False

RECEIVER CONTROLLERS AND TRANSMITTERS

-1- Transmitters do not normally control anything, they just sense and report the condition to receiver controllers and indication gauges.

True False

-2- Receiver controllers may have as many as five connection ports.

True False

-3- The majority of SIEMENS, TAC, Johnson Controls and Honeywell transmitters are two pipe instruments.

True False

-4- The indicating pressure range of transmitters is normally 5 PSIG to 18 PSIG.

True False

-5- Applying 9 PSIG to a 0°F to 100 °F transmission gauge would cause the gauge to indicate 50°F.

<u>True</u> False

-6- A two position receiver controller can gradually modulate a control valve from fully open to fully closed.

True <u>False</u>

-7- Some receiver controllers provide a restricted air supply to its associated transmitter and some others require an external restricted air supply to the associated transmitter.

<u>True</u> False

-8- If a receiver controller that can provide restricted air to its associated transmitter is receiving main air at variable pressures, its internal restrictor must be blocked and a constant main air external restrictor must be added.

<u>True</u> False

-9- If the transmission line is cut between the receiver controller and the (0°F to 100 °F) transmitter, the receiver will believe the temperature is above 100 °F.

True <u>False</u>

-10- A transmitter can only report to one receiver controller.

True <u>False</u>

-11- Room temperature transmitters look like a thermostat with no dial adjustment.

<u>True</u> False

-12- Proportional band, gain and sensitivity all refer to the same function in receiver controllers.

<u>True</u> False

-13- Receiver controllers may be arranged to automatically raise and lower a temperature of one medium based on a variation sensed in another medium.

<u>True</u> False

-14- Receiver controllers with a CPA can have their set point adjusted from remote locations via pressure changes.

<u>True</u> False

<u>RELAYS</u>

-1- Pneumatic relays allow design logic achieving almost any sequence of events required.

<u>True</u> False

-2- A two input high selector averages the input signals.

True <u>False</u>

-3- A volume boosting relay may also be used as a two input low selector.

True False

-4- A snap acting air switching valve may be piloted by a two position controller or a modulating controller.

True False

-5- A modulating receiver controller may pilot a gradual air switching valve.

True False

-6- A reversing relay can effectively change a direct acting signal into a reverse acting signal and offset the signal.

True False

-7- Multi high-low selector indicates the highest input signal on its H output port and the lowest input signal on its L output port.

True False

-8- A minimum positioning relay assures the fresh air dampers are closed at night.

True False

-9- A biasing relay (ratio relay) allows co-ordination of heating and cooling in a logical sequence.

True False

-10- If two thermostats, set for 70°F, are averaged with one room at 80°F and the other at 60°F the average will be 70°F. The comfort in the occupied space will be acceptable.

True False

-11- Pilot positioners (positive positioning relays) are used on valves and damper actuators to position the devices accurately, compensating for varying resistance, as well as allowing alteration of the stroking range and start point of the devices stroke.

True False

-12-Relays are available to add or subtract a determined amount of pressure from another variable pneumatic signal at a constant rate.

True False

-13- A PE switch passes control air when it is energised.

True False

SAFETY LOOPS

-1- Low limits (sometimes called freeze stats) have an averaging element and shut down the fan if the air is too cold.

True False

-2- High limits (sometimes called fire stats) should be only installed in the supply and mixed air.

True False

-3- Circuits sensing the coil water temperature with an electric thermostat (45°F) and the water electric flow switch wired in series, also have an outdoor temperature electric thermostat (45°F) wired in parallel with the series circuit previously mentioned.

True False

-4- Low and high limit circuits may be fully electric or pneumatic with a pressure switch.

True False

-5- Normally the safety circuit should be wired through the automatic side of the starter switch and not the hand side of the switch.

True False

-6- Static pressure high limit on VAV systems shut down the fan before the pressure reaches the system's static control point.

True False

-7- If a low limit is causing nuisance fan shut downs, it is usually OK to jumper the contacts of the low limit to keep the fan running.

True False

-8- If the high pressure relief valve on the compressor keeps blowing off, you should remove it and plug the hole.

True False

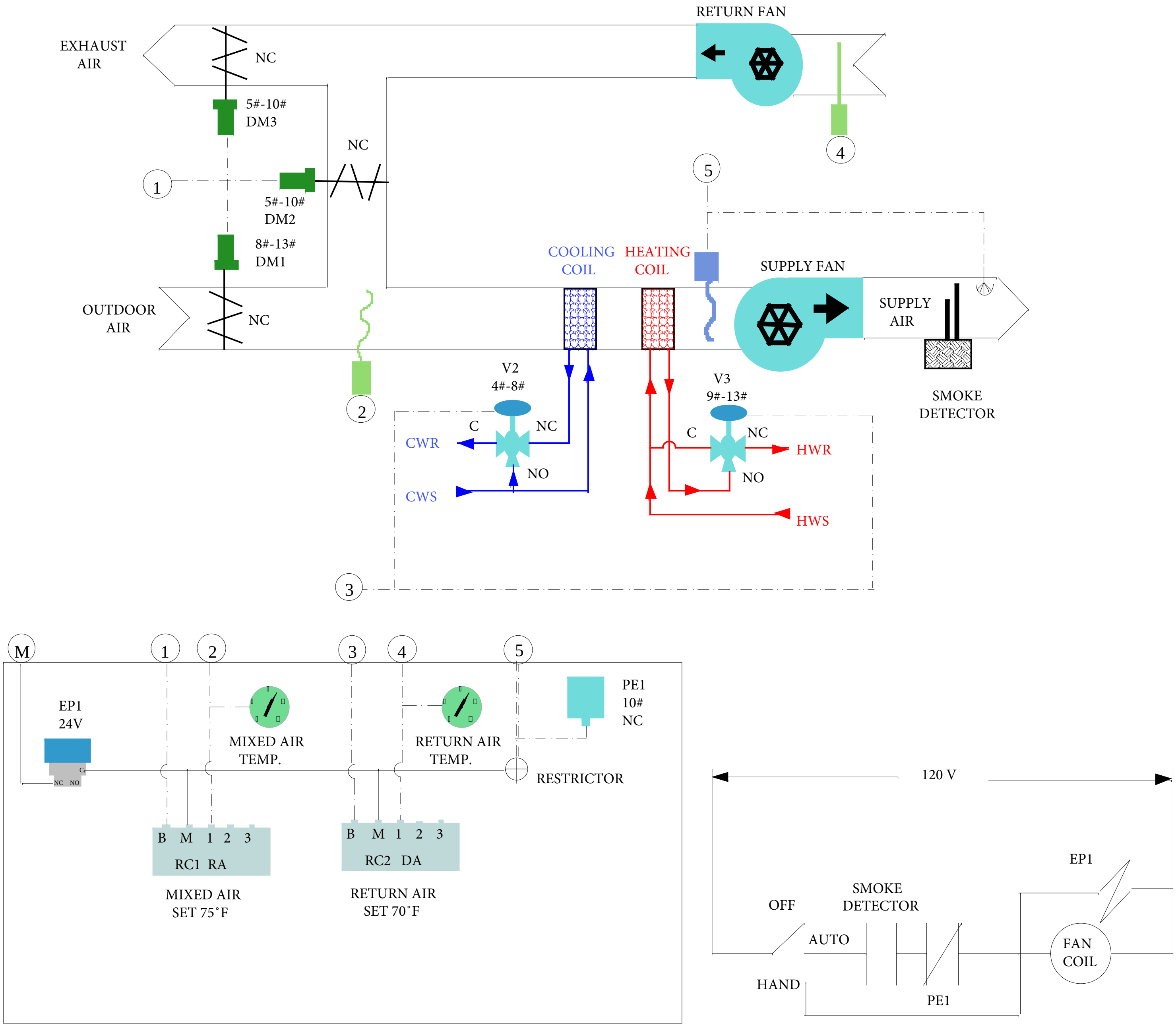
-9- The pressure switch on the compressor can be jumpered, if defective, until you get a new one.

True False

-10- The steam valve on the heat exchanger for domestic hot water should fail closed.

True False

ANSWERS TO FAN SYSTEM WITH MANY ERRORS



PROBLEM AREAS

-1- The return damper should be NO not NC.

-2- The spring ranges on the mixing dampers are incorrect. Exhaust and return should be 8#-13# and fresh air should be 5# to 10#.

-3- Another smoke detector and another sprinkler head (high limit) should be installed in the return air.

-4- PE1 should be NO, not NC.

-5- The hand position on the starter bypasses the safeties. It should not.

-6- The solenoid valve voltage is incorrect. It should be 120V.

-7- There is no high limit from outside air on the dampers for cooling season. There should be one.

-8- The mixed air controller is RA, set at 75 °F. It should be DA, set at 55°F.

-9- The valve ranges on V2 and V3 are reversed.

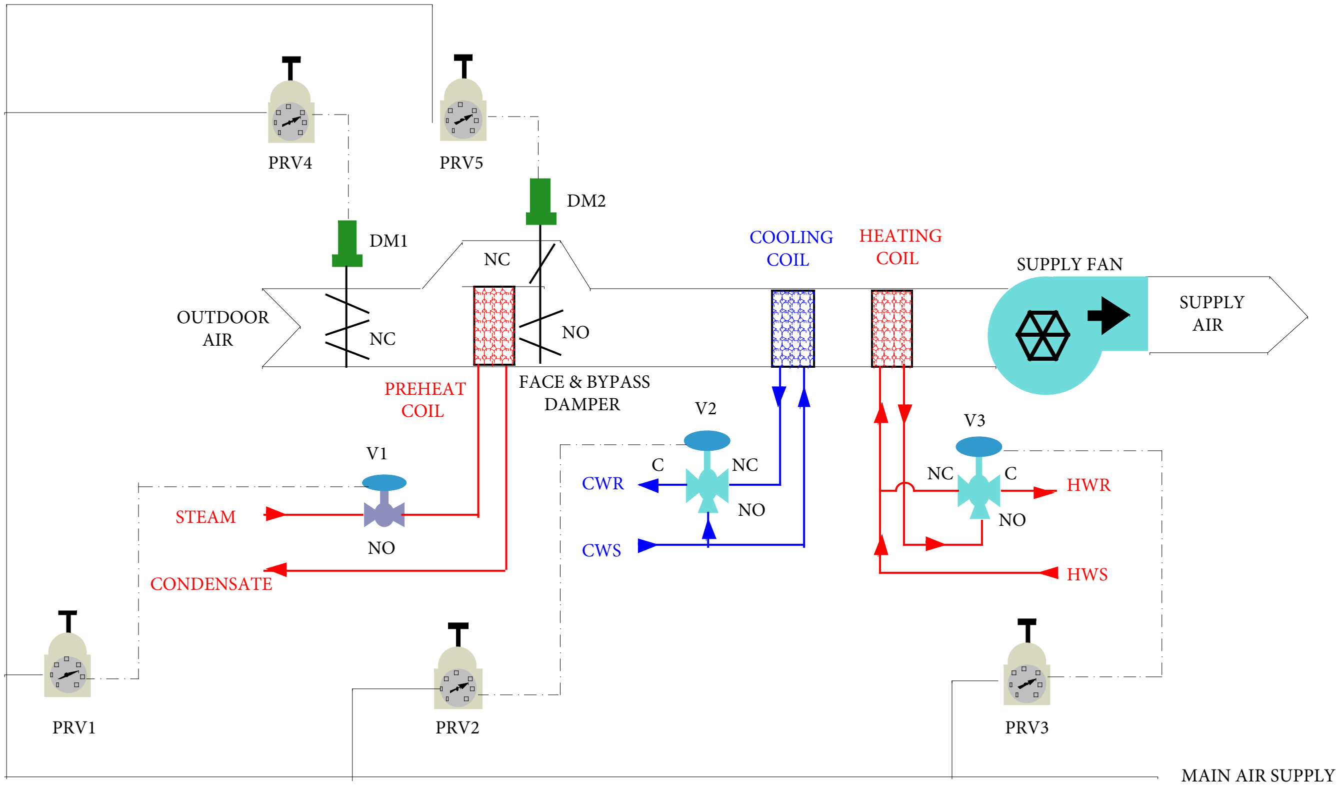
-10- The porting on V3 is incorrect. C and NC should be reversed.

-11- There is no minimum positioner. One should be added.

-12- There is no sequencing between free cooling and heating. It should be added.

-13- The cooling coil must be winterized or the low limit moved upstream of the heating coil.

-14- The safety circuit is fed with EP air not main air.



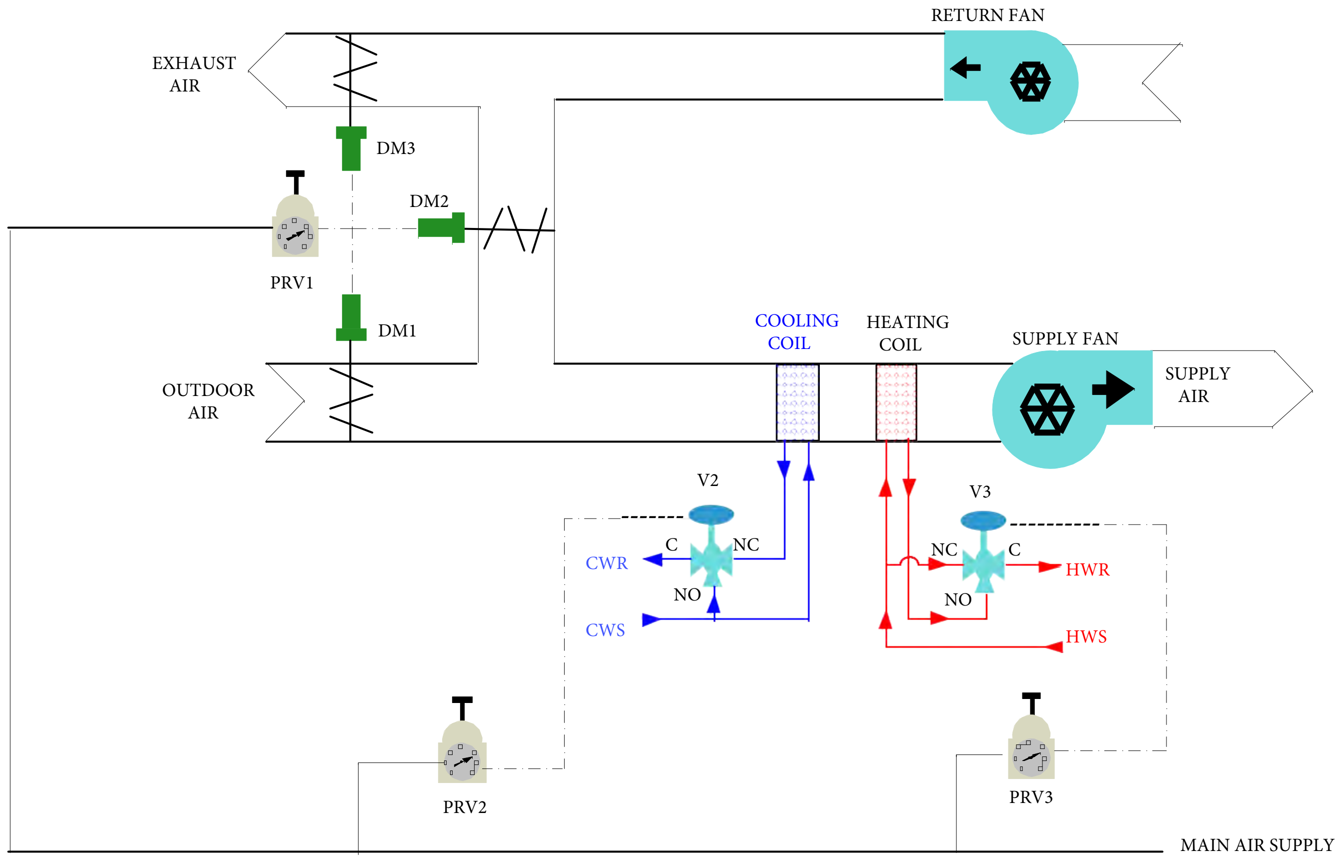
What are the considerations for the illustrated system regarding safety of the mechanical components? There is no low limit or high limit shown. Install low limits after preheat and cooling coils. The compressed air supply pressure must be under the ratings of the actuators as there are not relief valves shown for the PRV's. The upstream temperature of the preheat must be known. Add 30# relief valves after each PRV.

What are the considerations for the illustrated system regarding energy use?
The actuator spring ranges should be known. Pilot positioners or ratio relays will be required to sequence the four functions. Time actual time it is required to run must be known.

What is a typical application of the type of system illustrated?
Hospital OR's, school classroom supply, cafeteria/kitchen supply, for area with large exhaust requirements.

How would you manually operate this fan system to address ventilation, comfort and energy use requirements under these different conditions?

(1) At unoccupied times when the outside air (OSA) temperature is below 35F? Shut down if not required, with OSA damper closed and steam valve controlled from pneumatic low limit with automatic reset at 55°F.
(2) At occupied times when the OSA temperature is below 35°F? Maximize heating with preheat coil and adjusting face & bypass dampers. Use reheat coil only after preheat arrangement is unable to attain desired supply emperature.
(3) At unoccupied times when the OSA temperature is above 35°F? Same as number one.
(4) At occupied times when the OSA temperature is above 35°F? Same as number 2, but shut down heating completely before gradually opening the cooling coil valve.



What are the considerations for the illustrated system regarding safety of the mechanical components? There are no low limits or high limits, the compressed supply air pressure must be below the actuators' maximum allowable pressure rating, The mixed air temperature must be known and not allowed below 55°F. Added 30# relief valves after each PRV.

What are the considerations for the illustrated system regarding energy use?
The actuators' spring ranges must be known. The temperatures of the OSA, return air, mixed air and supply air must be known. The mixed air must be held at the minimum ventilation rate until the heating is completely shut down and cooling is required. Mechanical cooling is gradually used only after free cooling cannot achieve the desired supply air temperature goal. Mixed air to go back to minimum when it is easier to cool return air than OSA.

What is a typical application of the type of system illustrated? Office, classroom, gym, auditorium store.

Where and what safeties should be installed? High limits (HL) and low limit (LL) and smoke detectors (SD).

Where is the mixed air plenum and what are the mixed air's three purposes? Ventilation, pressurization and cooling.

How would you manually operate this fan system to address ventilation, comfort and energy use requirements under these different conditions?

(1) At unoccupied times when the outside air (OSA) temperature is below 35°F? (If the fan is the only source of heating? If adequate perimeter heating also exists?)
Duty cycle with dampers in full recirculation if possible via a night stat setting if only source of heat. Shut down if not required controlling heat from mixed air at 55°F, with OSA and exhaust dampers closed.

(2)At occupied times when the OSA temperature is below 35°F? The mixed air must be held at the minimum ventilation rate until the heating is completely shut down and cooling is required.

(3)At unoccupied times when the OSA temperature is above 35°F? Duty cycle with dampers in full recirculation if possible via a night stat setting if only source of heat. Shut down if not required controlling heat from mixed air at 55°F, with OSA and exhaust dampers closed.

(4)At occupied times when the OSA temperature is above 35°F? The mixed air must be held at the minimum ventilation rate until the heating is completely shut down and cooling is required. Revert back to minimum ventilation when the return air is easier to cool than the OSA.

TWO OF THE MOST SIGNIFICANT

GLOBAL SOCIAL THREATS

ARE LINKED DANGEROUSLY

VIA

HVAC VENTILATION ALTERATIONS

COVID VIRUS/GLOBAL WARMING

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CONTENTS

PREFACE

Two of the most significant global social threats, Climate Change and COVID virus transmission, are dangerously linked via COVID HVAC ventilation solutions.

Authorities have suggested HVAC systems be altered to full fresh air and/or abort Demand Controlled Ventilation (DCV), reverting to design constant supply air temperatures. This applies to both thermal and CO_2 load variation responses. (DCV varies the amount of fresh air above minimum code requirement based on conditions in the building.)

These HVAC ventilation changes will have two severe environmental repercussions:

1. They will increase the HVAC systems' atmospheric heating input directly, via increased associated HVAC exhaust, ranging from 417% to 670% over conservation circuit performance.

2. They will also increase GHG production ranging from 333% to 700% to produce the required heating of the increased outdoor air.

We have searched for data regarding live virus content in the HVAC supply air as it enters the occupied space with the HVAC dampers at maximum and minimum outdoor air conditions, with no such data being found.

Authorities claim the live virus tends to be killed during attempted collection caused by turbulence. If they are correct, the turbulence of going through the spinning blades of the return and supply fans will also kill the live virus.

The net result allows no live virus leaving the HVAC system diffusers before or after the suggested HVAC ventilation changes; however, the changes escalate global warming impact of millions of fan systems.

Based on available current data the COVID HVAC ventilation solutions are extremely destructive regarding Climate Change, with likelihood of no benefit addressing COVID virus transmission.

This paper suggests a testing method assessing the live virus content in the HVAC system establishing a non-turbulent supply air environment at full air recirculation and no recirculation scenarios. Complete data is required. With over forty years of redesigning HVAC conservation control logic, APS will volunteer its time assisting a credible laboratory and donate $10,000.00 toward the performance of the required testing.

WHY CARE?

BBC NEWS:
September 14, 2021

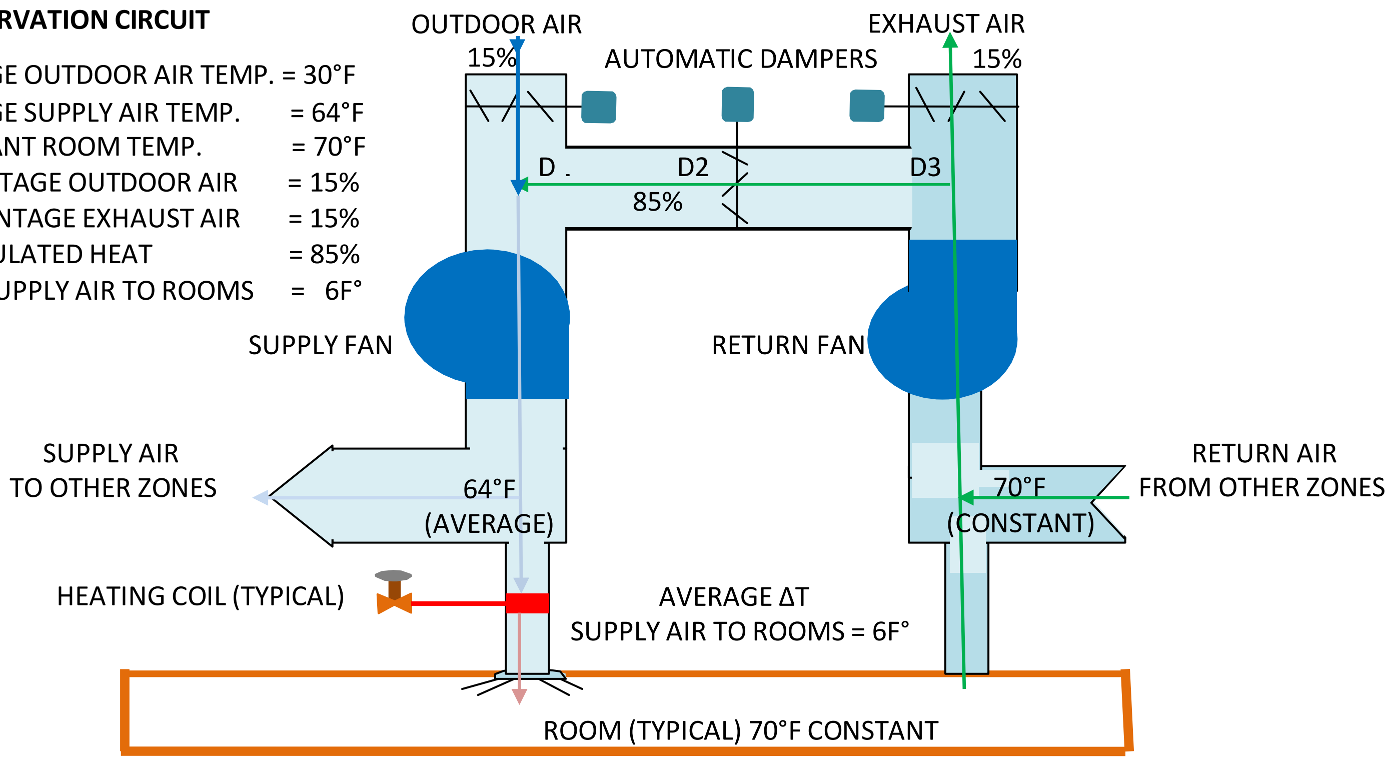
A new global survey illustrates the depth of anxiety many young people are feeling about climate change.

- Nearly 60% of young people approached said they felt very worried or extremely worried.
- More than 45% of those questioned said feelings about the climate affected their daily lives.
- Three-quarters of them said they thought the future was frightening. Over half (56%) say they think humanity is doomed.
- Two-thirds reported feeling sad, afraid and anxious. Many felt fear, anger, despair, grief and shame - as well as hope.
- One 16-year-old said: "It's different for young people - for us, the destruction of the planet is personal."

EXPLANATION OF HOW SUGGESTED HVAC VENTILATION INCREASE IMPACTS GLOBAL WARMING

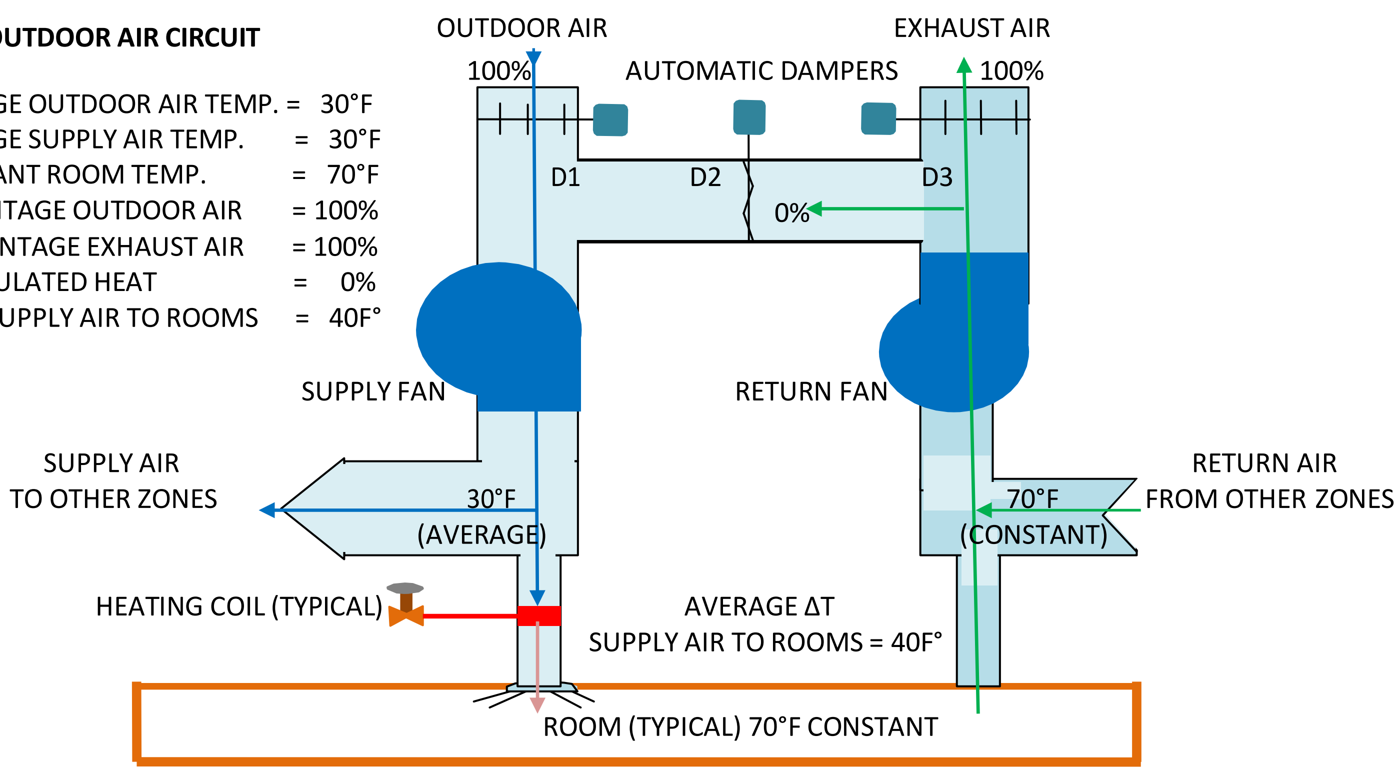
CONSERVATION CIRCUIT

AVERAGE OUTDOOR AIR TEMP. = 30°F
AVERAGE SUPPLY AIR TEMP. = 64°F
CONSTANT ROOM TEMP. = 70°F
PERCENTAGE OUTDOOR AIR = 15%
PERECENTAGE EXHAUST AIR = 15%
RECIRCULATED HEAT = 85%
ΔT OF SUPPLY AIR TO ROOMS = 6F°



100% OUTDOOR AIR CIRCUIT

AVERAGE OUTDOOR AIR TEMP. = 30°F
AVERAGE SUPPLY AIR TEMP. = 30°F
CONSTANT ROOM TEMP. = 70°F
PERCENTAGE OUTDOOR AIR = 100%
PERECENTAGE EXHAUST AIR = 100%
RECIRCULATED HEAT = 0%
ΔT OF SUPPLY AIR TO ROOMS = 40F°



NOTE: INCREASED CLIMATE CHANGE DAMAGE AT TWO DIFFERENT POINTS:

The direct heat increase, via exhaust air, into the atmosphere from altering to the 100% outdoor air circuit is 670%.

The ncreased GHG emissions, caused by increasing the ΔT of the supply air to the rooms is 700%.

This is worst case scenario, as most systems' safeties disallow this alteration, but various percentages are achievable. Disabling all Demand Control Ventilation (DCV), as per ASHRAE suggestions, reverts to minimum supply temperature set point causing at a 417% increase in exhaust heat and at a 333% increase in GHG emissions to atmosphere. This is best case scenario. ASHRAE altered from suggesting 100% outdoor air, but many governing authorities do not know this fact. ASHRAE's position of disabling DCV remains, based on available papers. **If the live virus cannot travel through the turbulence of the HVAC fans and return alive to the occupied spaces, the alterations in the HVAC equipment presents no benefit, as the supply air should have the same live virus content before and after the alterations; however, the alterations are extremely damaging regarding global warming.**

We could not find data on live COVID virus quantities at various locations in HVAC systems. We did find reports indicating the live virus could not travel from occupied spaces, through the HVAC system and back to the occupied space. This communication is one from an ASHRAE member on their Epidemic TASK Force. We need to know where the virus survives in HVAC systems before setting plans. The accompanying papers may assist in that requirement.

Subject: RE: COVID-19 transmission consideration

Hello,

Please see the following response from a member of the ASHRAE Epidemic Task Force:

Hello David.

Thank you for your question to the ASHRAE Epidemic Task Force (ETF) and for forwarding the interesting attachments.

The science surrounding the possibility and likelihood of airborne transport and transmission of COVID-19 is rapidly changing. As time goes on, more and more respected scientists are convinced that the airborne pathway for transmission is likely. For instance, in early July, 239 well-respected scientists (including some serving on the ASHRAE ETF) from 32 countries petitioned the World Health Organization (WHO) to rethink the importance of airborne transmission of the SARS-CoV-2 virus. Their letter can be found here. It prompted the WHO to change their position and recognize the airborne pathway in the ongoing pandemic. As you mentioned, there is not a lot of hard evidence showing that airborne viral particles can travel long distances and through HVAC systems. However, this lack of evidence is not proof that airborne transmission cannot happen. Sampling and culturing viable virus (any virus) from the air is extremely difficult to do. The relatively turbulent act of sampling tends to kill the virus, making true viability very difficult to ascertain. Given those limitations, it is likely viable SARS-CoV-2 viral particles are traveling further in the air than we can concretely prove. HIGHLIGHTING ADDED.

After reviewing all of the literature we have available (and doing our best to keep up with new reports), the conclusions of our group of experts is that the transmission through the air is likely enough that we believe countermeasures are warranted. That is why we put together what we hope are practical suggestions for ways for owners/managers/occupants to proceed. We also tell people to use the suggestions with caution. As you suggest, difficulty maintaining the RH level is one reason to consider limiting the amount of incoming outdoor air.

We also wholeheartedly agree with your assertion that more research is needed to be sure our recommendations are the best possible. However, in the throes of a global pandemic, we firmly believe we need to be intelligently proactive and not wait for absolute proof before acting. There is a significant amount of research going on currently, and there will likely be much more in the future. Eventually, we may learn more about how effective our guidance truly was during the pandemic. In the meantime, we are choosing to err on the side of caution.

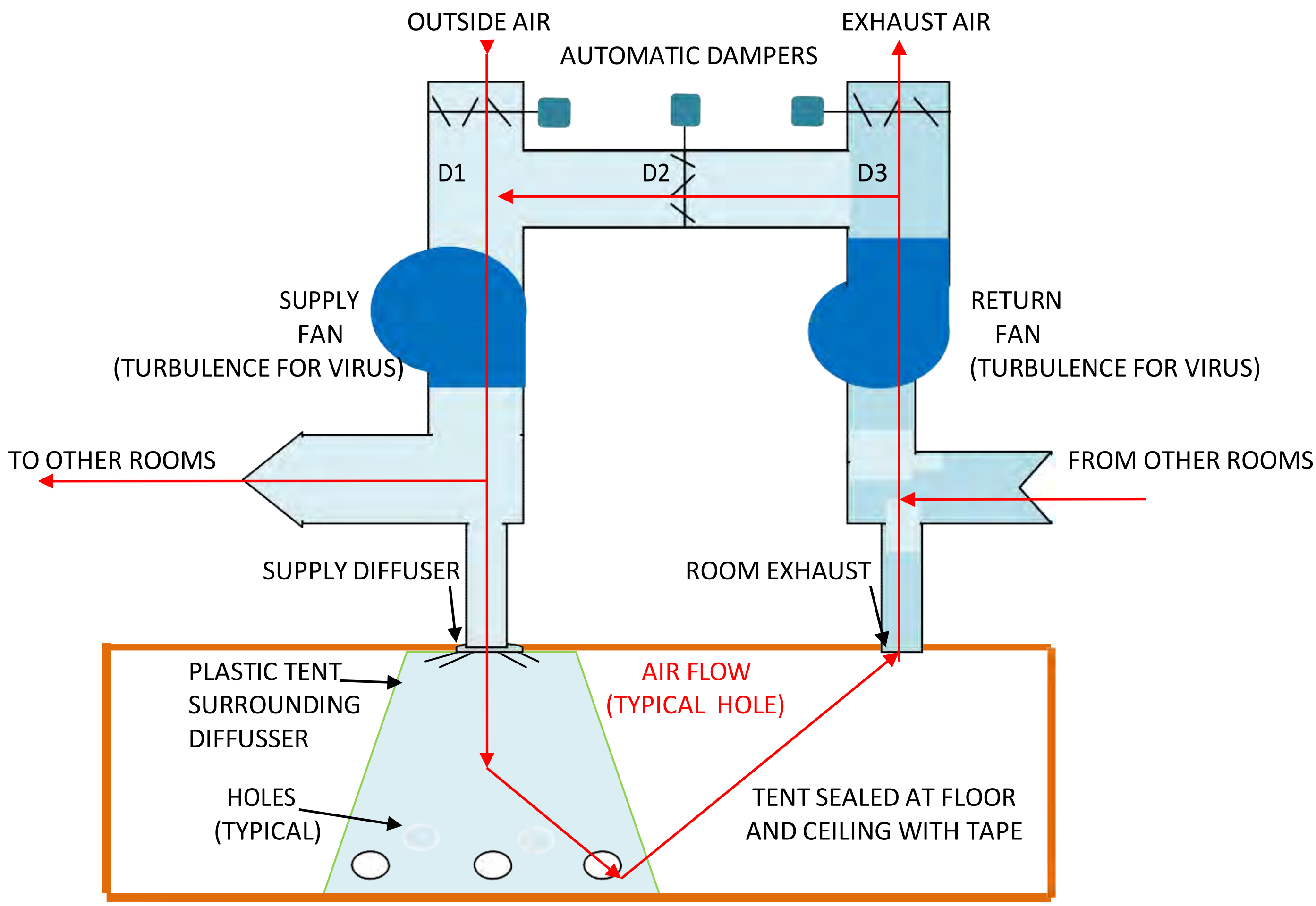
Regards,
Steve Hammerling
ASHRAE Manager of Technical Services

Answers to technical inquiries are provided as a service to the public. While every effort is made to provide accurate and reliable information, they are advisory, are provided for informational purposes only, and may represent only one person's view. They are not intended and should not be relied upon as official statements of ASHRAE.

SUGGESTED TEST COMPLETING REQUIRED DATA

COVID VIRUS TEST IN NON-TURBULENT HVAC FAN SUPPLY AIR

CONSIDERING ESCALATED GLOBAL WARMING AND POSSIBLE VIRUS TRANSMISSION IMPACTS



THE PROBLEM------ Increasing outdoor air ventilation is escalating global warming in two manners:

-1- The increased heating required increases GHG emissions caused by the HVAC system up to about 700%

-2- The increased exhaust air increases atmospheric heat input via the HVAC system up to about 670%.

The only information we have found indicates that no testing has been done in all the appropriate locations on typical HVAC systems for live COVID virus, supporting the maximum ventilation position of much of governing society. Opinions expressed by authorities indicate that the virus cannot live through turbulent environments. The reports found indicate the live virus cannot travel through the typical HVAC system and return to the occupied space as a threat.

REQUIRED TESTING---The illustration above presents a means of creating a relatively non-turbulent supply air environment to determine if live virus can survive the turbulence of the HVAC fans and return to the occupied areas.

TEST METHOD -1- Build a tent as illustrated and run system until the air in the tent matches supply air quality.

-2- Alter the HVAC automatic dampers to have D1 and D3 closed and D2 open (full recirculation) if fan CFM's allow.

-3- Test the tent air and the room air for the live virus.

-4- Introduce the live virus into the room outside of the tent.

-5- Run for enough time to have tent refilled completely with return air that had the live virus introduced.

-6- Test again for the live virus in the tent and the room. Record the quantity of live virus at both locations.

-7- Alter the HVAC automatic dampers to have D1 and D3 open and D2 closed.(full outdoor air) if allowable. Repeat testing.

CONSIDERATION---If the tent live virus is absent in both the full outdoor air and full recirculation scenarios there is no benefit in altered ventilation: however, the full outdoor air position will have significant damage implications regarding Climate Change. If there is benefit in extra ventilation, it should be weighed against the negative impact, regarding Climate Change.

<u>SUMMARY</u>

1. All in the industry have been on a learning curve regarding COVID and HVAC; therefore negative comments are not helpful and everyone with a positive solution or input should speak up.

2. The ASHRAE-suggested ventilation changes initially had HVAC systems altered to maximum outdoor air where possible, which was a reasonable first position reaction.

3. ASHRAE altered their position in some steps as new information became available, which is also a logical path.

4. The current ASHRAE suggestion of disabling DCV will prevent conservation circuits from recirculating a lot of heat in buildings, which will dramatically increase HVAC systems' direct heat atmospheric input and also increase GHG emissions.

5. The current understanding of experts is that the virus is killed by the turbulent act of collecting samples. Accepting this, the turbulence of going through the return and supply fan blades should kill the virus.

6. Knowing whether there is live virus coming through the diffusers feeding the rooms is critical. If the diffuser air has no live virus before the HVAC ventilation changes and no live virus after the ventilation changes, the changes at the fan systems are of no benefit regarding COVID protection; however, the changes are very damaging regarding Climate Change. The suggested testing method on page 11.192 of this manual will allow a non-turbulent environment in the HVAC supply air, providing the required information relating to the HVAC ventilation alterations.

7. If the HVAC alterations provide benefit, it should be weighed against the negative impact on Climate Change created by these alterations.

Please send questions or comments to:

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HVAC CONSIDERATIONS

ADDRESSING CURRENT SUGGESTIONS

REGARDING

COVID-19 VIRUS AIRBORNE TRANSMISSION

AND SUBSEQUENT NEGATIVE IMPACT ON

EFFORTS TO AVERT

CLIMATE CHANGE

NOTE: A CASE STUDY WAS SENT TO US SINCE WRITING THIS PAPER WHICH READS VIRUS AT A FAN SYSTEM. MUCH OF THE DATA POINTS WE NORMALLY REQUIRE TO ASSESS SYSTEM PERFORMANCE WAS MISSING FROM THE SLIDES FOR THAT STUDY.

FURTHER INVESTIGATION IS REQUIRED TO COMPARE THAT STUDY TO THE SEEMINGLY CONTRADICTORY STUDIES FROM THE ALBERTA HEALTH SERVICES REPORT.

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PREFACE

1) The seriousness of the current COVID-19 pandemic is understood by most people and we are united in accepting that we must address the threat.

2) Most people also understand the seriousness of Climate Change and we are united in accepting that we must address that threat.

3) The HVAC suggestions via ASHRAE and REHVA alter the fan systems from mixed air to maximum fresh air, which can increase the systems' negative impact regarding Climate Change via increasing Green House Gas (GHG) emissions into the atmosphere by 700% in some cases.

4) If the suggested improvement to MERV13+ filters will capture the total airborne COVID-19 virus, we may provide protection from the virus and not increase Climate Change.

5) The assumption that the virus can travel from an infected person, through the occupied space and then through the route to the fan system in a remote fan room and back to the occupied space seems questionable when social distancing rules maintain the virus cannot travel more than two metres.

6) A study is required that identifies the virus content in the HVAC air at the defined locations of the drawings on page three of this report. We have to know where the virus actually exists before logical solutions can be established.

7) ASHRAE and Dr. Stephanie Taylor present the fact that relative humidity levels between 40% RH and 60% RH are beneficial to human health and detrimental to viruses. REHVA does not support this claim. This relationship of humidity level is very important in COVID-19 considerations in that introducing larger amounts of cold outdoor air will reduce humidity levels in the occupied space, which will benefit viruses and negatively impact human health, if Dr. Taylor and ASHRAE are correct on this matter.

8) Persons assessing the HVAC situation regarding COVID-19 virus airborne characteristics must not lump all HVAC airflow into one category.

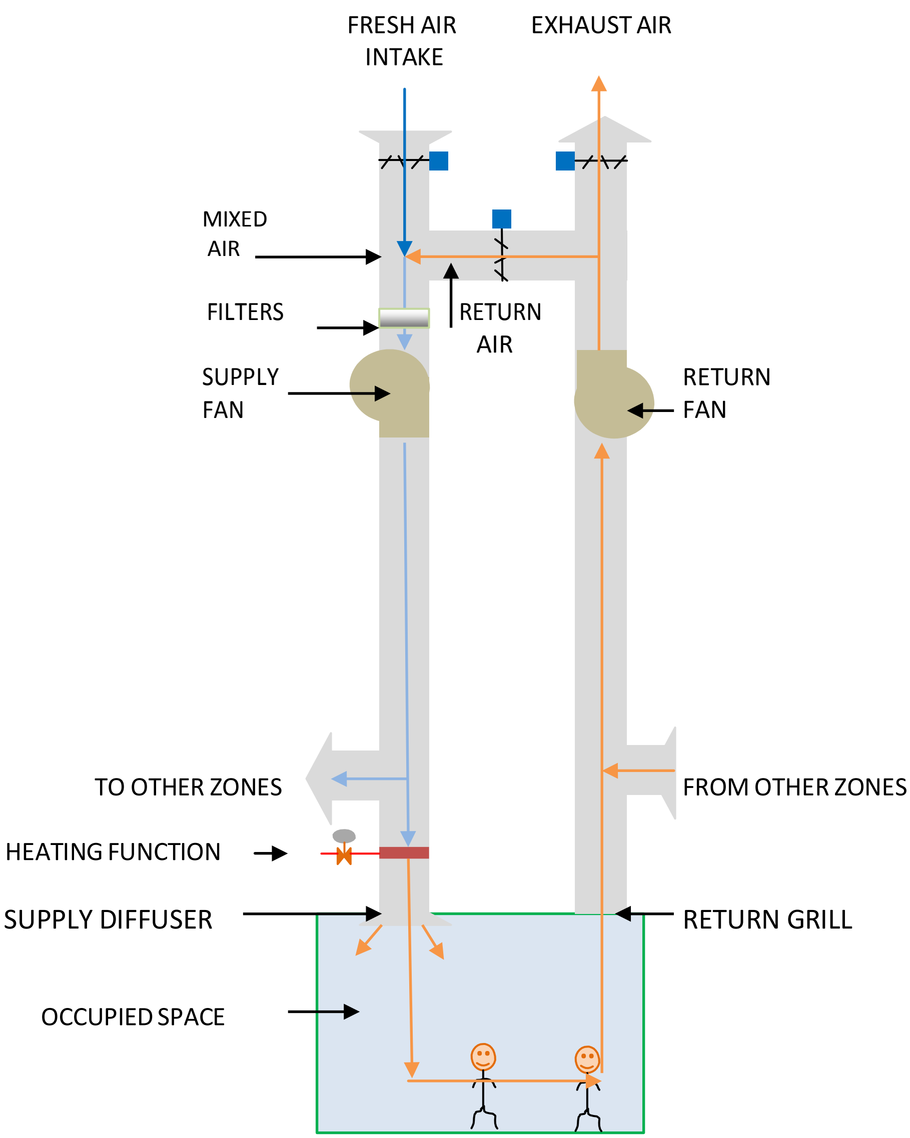
   They must separate the airflows into:

   a) Airflow from the supply fan to the point of entering the occupied space.

   b) Airflow through the occupied space.

   c) If the virus transmission is in the occupied space, while the supply fan's air is virus-free, no changes such as more fresh air or better filtration will improve the situation.

SIMPLE HVAC MIXED AIR SYSTEM



I'm interpreting Demand Control Ventilation (DCV) as variation of fresh air quantity from 0% to 100% based on minimum ventilation code, exhaust air replacement, occupied space CO_2 control or free cooling based on warmest zone requirements. ASHRAE may have a different definition.

On October 5, 2020 ASHRAE posted a new sheet regarding "GUIDANCE FOR RE-OPENING BUILDINGS" with some significant changes from the previous HVAC re-opening advice of July 17, 2020 and August 20, 2020.

We have two very dangerous social challenges simultaneously occurring globally:

1) **The COVID-19 virus threat**. The HVAC actions suggested via the major Engineering societies in Europe (REHVA) and North America (ASHRAE) will produce added Green House Gas emissions.
2) **Climate change.** Action required is to reduce Green house Gas emissions.

We should not needlessly feed one of these monsters while attempting to address the other.

Current scientific studies show that the virus cannot travel from infected persons in the occupied space to the remote HVAC fan system and back to the supply air diffusers in the occupied space. Scientific studies are required to quantify the virus content at HVAC locations identified at the locations A, B, C, D, E and F on page 11-203 of this report. Social distancing rules imply the virus cannot travel more than two metres to infect another person. The net result of these facts is the supply air entering the occupied space cannot contain virus whether it contains fully fresh air or fully recirculated air. With no added virus protection benefit, the system altered to full fresh air may require over 700% more heat (page 11-202) to keep the occupied space warm. This will produce 700% more GHG emissions, which will tend to escalate Climate Change.

Often statements for better ventilation are voiced. The ventilation air from the supply fans must be considered from two different positions:

1) The air from the fan to the entrance point of the occupied space.
2) The air as it passes through the occupied space.

If the issue is with virus being passed from an infected person to another within the occupied space, the problem will still exist even if the supply air to that space is completely virus-free fresh air. Current scientific studies and the virus travel distance defined by social distancing indicate that the supply air to the space must be virus-free. Altering fresh air rate and filters at the HVAC fan is pointless when the supply air is virus-free before and after the HVAC alterations. If the problem is only in the occupied space, it must be solved in the occupied space. We should avoid the addition of pointless GHG emissions to the atmosphere.

The shift in operational changes via ASHRAE in the new positions posted October 5, 2020 must be conveyed very clearly to officials in school boards, government and teacher unions on ASHRAE's altered opinion regarding maximum fresh air in ventilation. The understanding should be conveyed that the supply air to the rooms would be virus-free before or after the original guidelines and the GHG emissions addition can be avoided if the systems are allowed to control under energy conservation logic that existed prior to the COVID-19 issue. Facility staff will not likely alter the public and teacher unions' opinion for maximum fresh air without very clear and public support of ASHRAE indicating the new position will not negatively impact the effort to correct the COVID-19 fight, but will prevent further damage to the planet via unnecessary GHG emissions.

The comparisons of earlier ASHRAE guidance to the October 5, 2020 ASHRAE guidance are:

1) August 20, 2020 "Ventilation: A good supply of outside air, in accordance with ASHRAE Standard 62.1-2019, to dilute indoor contaminants is a first line of defense against aerosol transmission of SARS-CoV-2. Pre- and post-occupancy purge cycles are recommended to flush the building with clean air."

   October 5, 2020: "Pre- or Post-Occupancy Flush with Outdoor Air: Focus on removing bio-burden pre-or post-occupancy of the building. Flush building for a time required to achieve three air changes of outdoor air (or equivalent, including effect of outdoor air, particulate filtration, and air cleaners)."
   Red highlighting added.

2) July 22, 2020: "During the Pandemic, disable any Demand Control Ventilation (DCV) and introduce the maximum possible OA flow 24/7 until further notice (including DOAS)."

   October 5, 2020" "Energy Savings: During Evaluation and Inspection, determine optimized control strategies that can be implemented per ASHRAE Guideline 36-2018, High-Performance Sequences of Operation for HVAC Systems."

   October 5, 2020: "Modes of Operation for the Building: Operate in Occupied Mode when people are present in the building, including times when the building is occupied by a small fraction of its allowable capacity."

3) October 5, 2020: "Ventilation and Filtration: Confirm systems provide required minimum amounts of outdoor air for ventilation and that the filters are MERV 13 or better filters for recirculated air. Combine the effects of outdoor air, filtration, and air cleaners to exceed combined requirements of minimum ventilation and MERV-13 filters.

The environmental benefits in the apparent shift are:

1) From two purge cycles a day to one is a 50% reduction in additional pollution generated.

2) The shift from disabling the Demand Control Ventilation to maintaining energy conservation logic is a potential 700% improvement in pollution reduction on some fan systems.

3) The shift from running fans 24/7 on maximum fresh air to operate in Occupied Mode when people are present in the building is the greatest reduction in additional pollution generation.

The initial opinion of the public may over-look or refuse to accept these three alterations. This would be environmental devastation regarding GHG emissions if we produce unnecessary GHG emissions. I believe a path must be found to scientifically quantify the virus content in the supply air of mixed air systems in 100% recirculation, with virus present in the occupied space. (Worst case scenario) The two case studies in the Alberta Health Report and the social distancing rule of two metres suggest the virus cannot travel through the fan system and back to the occupied space, but real data in virus content of the supply air is required to confirm this.

Demand Ventilation Control (DVC) based on temperatures only, allows the code requirement of minimum fresh air (example 20% fresh air) until the warmest zone has closed its heating device and requires cooling. The remaining 80% of the mixed air volume is increased in fresh air beyond the minimum just enough to satisfy the cooling requirement of the warmest zone. The variable cooling demand is limited to a minimum temperature of 55˚F (13˚C).

DVC may also vary the minimum fresh air quantity based on CO_2 levels in the occupied space.

The requirement to alter back to (DVC) is urgent now in that:

1) Many systems with DVC have been altered from variable mixed air temperatures to provide constant mixed air temperature at 55˚F (13˚C).

2) The day time outdoor air temperatures during spring and fall will cause the fans to be on about 100% outdoor air most of the time.

3) This causes a 15F˚ (8.3C˚) ΔT heating demand to warm the room to 70˚F (21˚C).

4) If the DVC is re-instated the mixed air temperature would vary automatically adjusting to 67˚F (19.4˚C) with outdoor air temperature at 55˚F (13˚C) and 20% minimum fresh air if the warmest zone required no cooling.

5) DVC control on minimum ventilation would require a 3F˚ (1.7C˚) ΔT to rise to the 70˚F (21˚C) room temperature rather than the altered circuit's 15F˚ (8.3C˚) ΔT. This is a 500% increase in heating required with the altered circuit.

6) During many spring and fall days the school's boiler with DVC can be run for a couple of hours in the morning to take the chill off the building and then shut down for the remaining day. The altered circuit with a 55˚F (13˚C) mixed would require the boiler to run all day compensating for the unnecessary cooling.

A study identifying virus concentrations at various locations in the HVAC system is required immediately. If that study contradicts existing studies identifying maximum virus travel and the social distancing rule of two metres, then both positions have to be thoroughly analyzed.

If new studies indicate that the virus can travel from the occupied space, through the HVAC fan system and back to the occupied space, the provable fact of escalating Climate Change by increasing HVAC fresh air intake should be considered in the final plans.

ASHRAE's support to convey the information in their October 5, 2020 announcement to the public is imperative now as the maximum benefit of DVC occurs in the spring and fall of the year.

CONTENTS

The Guardian

This article is more than 8 months old

We have 12 years to limit climate change catastrophe, warns UN

Urgent changes needed to cut risk of extreme heat drought, floods and poverty, says IPCC

Overwhelmed by climate change? Here's what you can do.

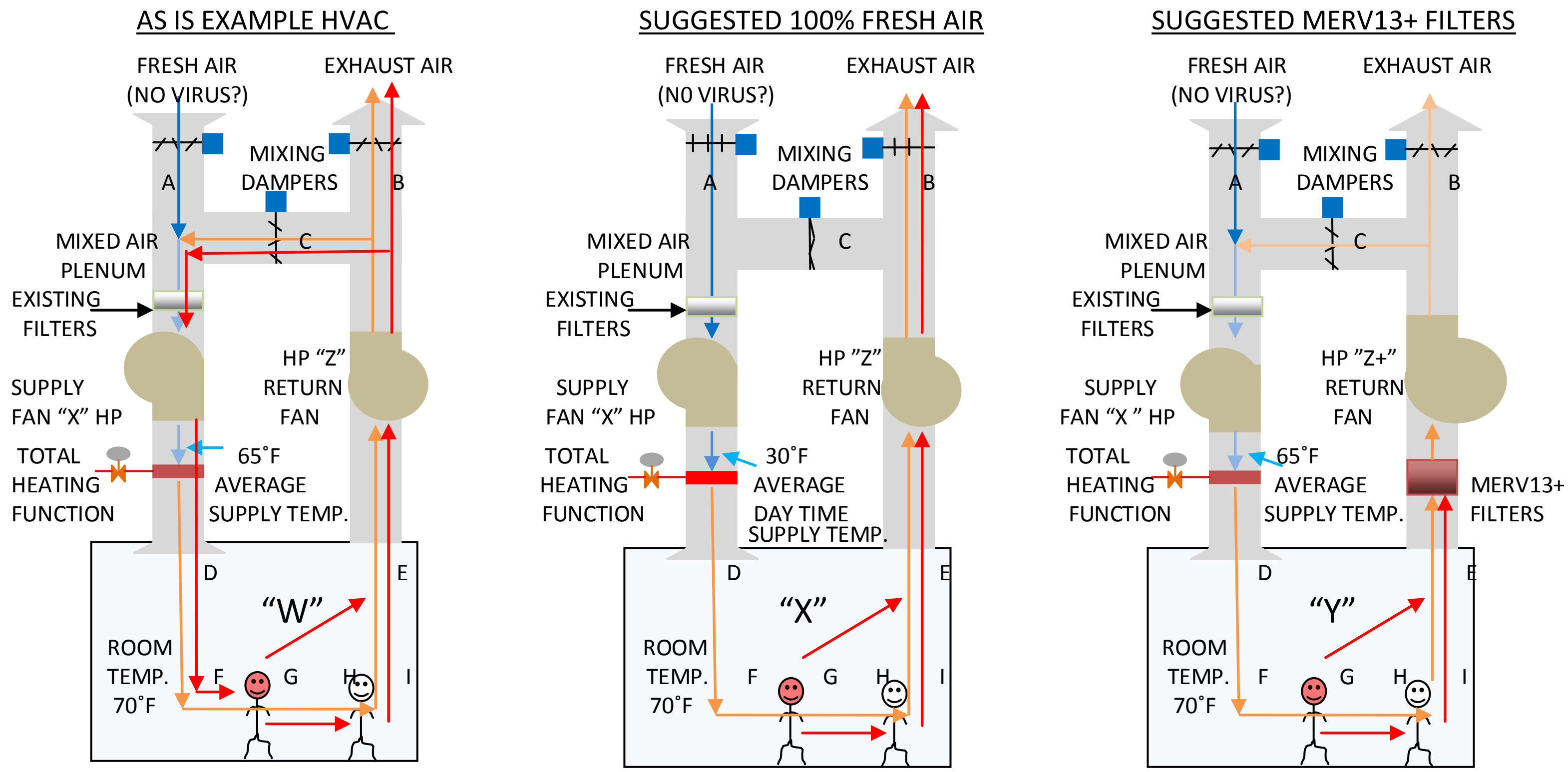
Jonathan: watts *Global environment editor*

Mon 8 Oct 2018 07.23 BST

The world's leading climate scientists have warned there is only a dozen years for global warming to be kept to a maximum of 1.SC, beyond which even half a degree will significantly worsen the risks of drought, floods, extreme heat and poverty for hundreds of millions of people.

The authors of the landmark report by the UN Intergovernmental Panel on Climate Change (IPCC) released on Monday say urgent and unprecedented changes are needed to reach the target, which they say is affordable and feasible although it lies at the most ambitious end of the Paris agreement pledge to keep temperatures between l.SC and 2C.

100% VENTILATION IMPACT ON CLIMATE CHANGE VIA INCREASED GREEN HOUSE GAS (GHG) EMISSIONS



NOTE: red arrows are virus travel range. All other arrows are HVAC airflow paths.

1) If the virus can be recirculated back to the occupied space as per example "W", positive correction is required; however, the impact on Climate Change, via increased Green House Gas (GHG), should also be considered.
   In example "W", the ΔT from the supply air temperature to the occupied space temperature is 5F˚ based on conservation logic's average mixed air demand temperature of 65˚F and an occupied space temperature of 70˚F.

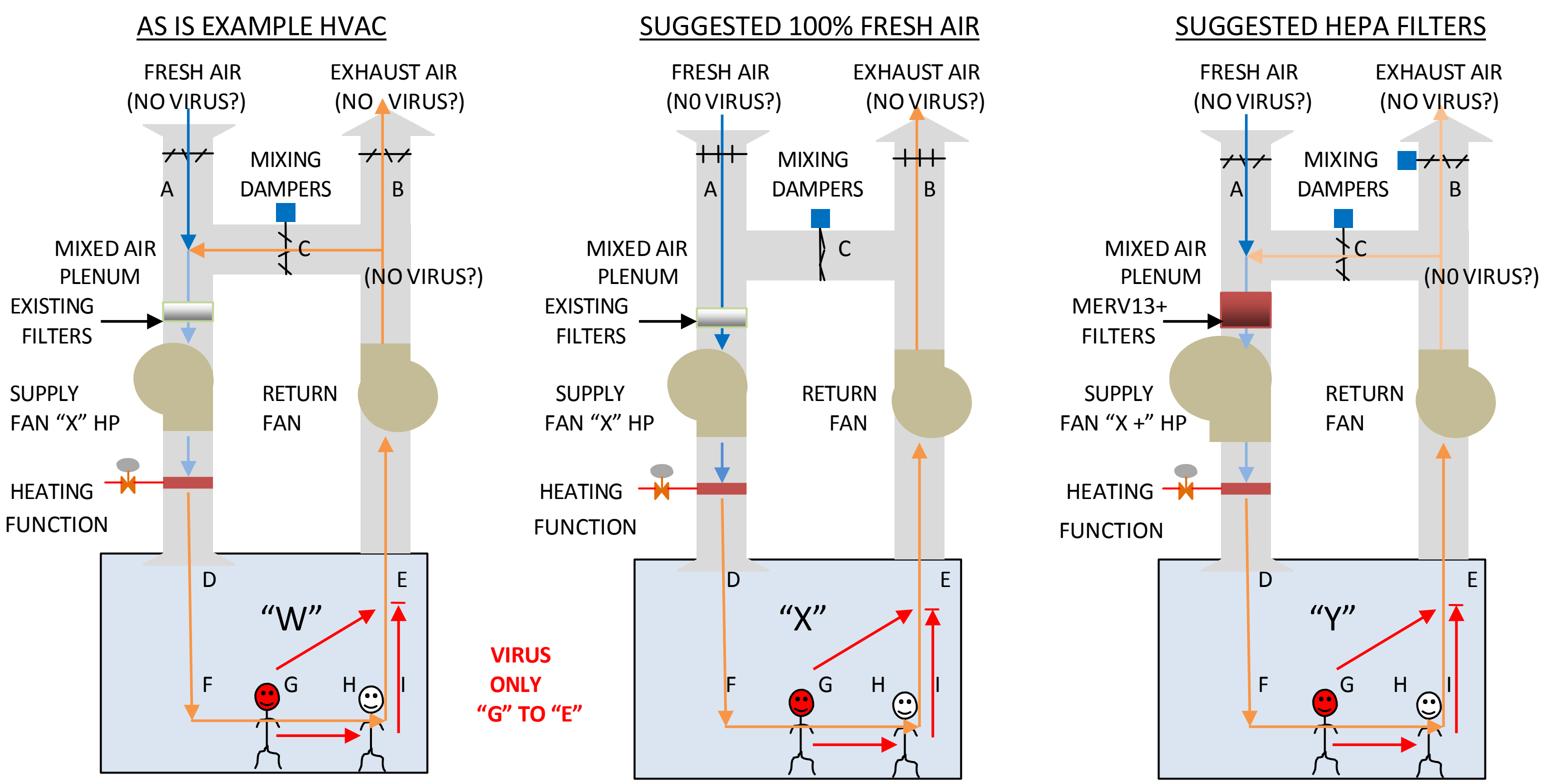
2) The supply air "D" may become virus free by altering the HVAC system to 100% fresh air as per example "X"; however, this action will cause a significant increase in GHG emissions.
   In example "X", the ΔT from the supply air temperature to the occupied space temperature is 40F˚ based on an average day time winter fresh air temperature of 30˚F and occupied space at 70˚F.

3) Altering from "W" to "X" illustrate a 700% increase in heating energy required at the HVAC system to maintain the occupied space at 70˚F. This will devastate operational budgets and increase the rate of Climate Change via increased GHG emissions. The design of most mixed air systems will not allow this operational change.

4) If the MERV13+ filters can completely capture the airborne viruses, example "Y" might be considered.
   The benefits are:
   a) The current energy conservation logic of "W" may be maintained; therefore, the 700% increase in energy use can be avoided, with no extra GHG emissions.
   b) If the virus is airborne over large distances, the exhaust in "Y" will be filtered to be virus free, removing the possibility of virus returning into the building via exhaust air short circuiting, via wind, into the supply air intakes.

   The down sides are:
   a) The return fan will consume more energy to overcome the pressure drop of the MERV13+ filters.
   b) If the supply air "D" is currently virus free, because the virus cannot travel from the occupied space to the mixing plenum; therefore, the supply air at "D" has absolutely no change in virus content from before to after the changes, the public may develop a false sense of safety and back off on other safety measures.
   c) This may actually cause COVID-19 infections to increase.

PRO'S AND CON'S OF CONSIDERATIONS FOR ALTERATION TO 100% FRESH AIR AND MERV13+ FILTERS



NOTE: red arrows are virus travel range. All other arrows are HVAC airflow paths.

According to case studies in the "COVID-19 Scientific Advisory Group Rapid Evidence Report, prepared for the Alberta Health Services, the virus can only travel up to 6m with wind speeds of 4km/h to 15km/h. (Alberta Health Services, Dbouk and Drikakis 2020 Page 11.207) and another case study states a maximum of 4m travel in patient rooms. (Alberta Health Services, Guo et al 2020 Page 11.206). Common knowledge accepts that the virus can travel no more than 2m, respecting social distancing. It seems unlikely the virus can travel from the occupied space to the mixed air plenum of the fan system.

As the supply air enters the room at "D", If it carries no virus, there is no virus contamination threat from that point to the person at "H".
As the air from "D" passes over the virus carrier person at "G", the air may pick up virus and carry it to the person at "H".
If "G" to "E" is more than 6m, current studies indicate that the air from "E" to "C" will contain no virus. Better filtration or more fresh air will not alter this outcome.

Altering from scenario "W" to "X" will provide no additional virus protection if the virus content is the same at "A" as it is at "B".
With no added virus protection, the system will consume much more energy, produce much more GHG emissions, have increased risk of freezing a coil, experience more low limit shut downs and will likely not be able to heat the space in severe winter weather.

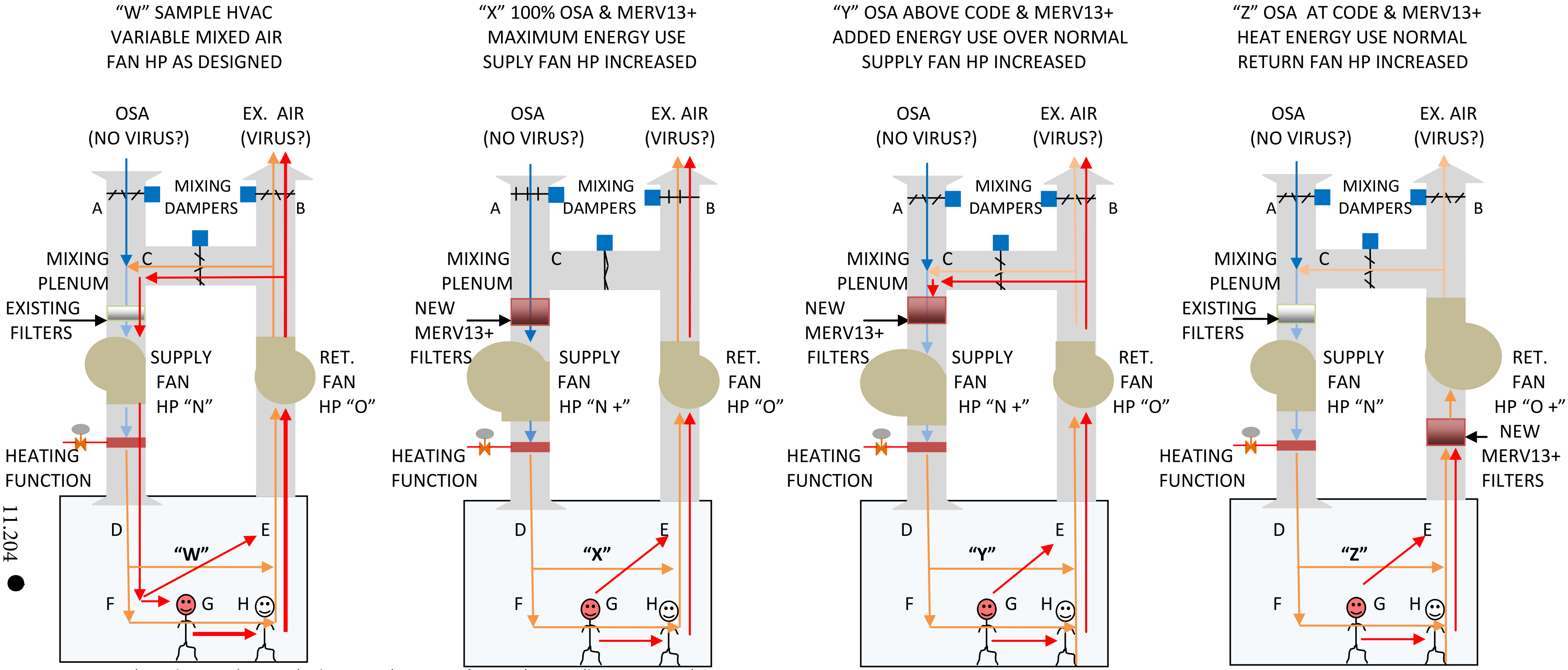
If MERV13+ filters are installed with scenario "X" and the virus content in the fresh air is considered to be virus-free, the virus content in the air before and after the MERV13+ filters will be equal, with no virus. The extra pressure drop of the MERV13+ filters over the existing filters will cause the fan's air volume to be decreased.

If MERV13+ filters are installed as per scenario "Y", the air entering the filters will likely be virus-free, as the fresh air at "A" and the return air "C" are likely both virus-free. The fan horse power will require increasing to produce the same airflow as scenario "W". With no added virus protection, the system will consume more energy and produce more GHG emissions.

We require a study identifying the virus presence at locations A, B, C, D, E and F. Current studies appear to assess the virus content from G to H. If the virus cannot travel from G to C, the alterations increasing the percentage of fresh air in the supply air at "D" and the change to MERV13+ filters will achieve nothing in addressing the virus transmission issue. Both changes will cost a lot of money that could be applied to other relevant solutions.

An additional concern is that excessive fresh air supply in cold weather tends to lower the relative humidity in the occupied space, benefitting the virus and is detrimental to the human occupants.

CONSIDERATION FOR MERV13+ FILTER INSTALLATION, IF THE VIRUS CAN TRAVEL FROM THE OCCUPIED SPACE TO THE HVAC MIXING PLENUM. CONSIDERING THE TIMING ISSUE, IT MAY BE WISE TO INSTALL MERV 13+ FILTERS, BUT STILL RESPECT CLIMATE CHANGE ENERGY CONCERNS.



11.204

NOTES: OSA-- (Outdoor Air), EX.-- (Exhaust air), RET..—(return), HP—(horse power)

RED arrows are suspected virus travel routes. Darker **blue** arrows are OSA. Lighter **blue** arrows are mixed air. **Orange** arrows are heated supply air.

1) The live virus transmission issue must be solved with minimum negative impact on Climate Change via added GHG, considering extra energy requirements.
2) Virus airborne travel from G to H is likely based on case studies; however, I'm unaware of case studies proving virus travel from E to C. If A and C contain no live virus, D will be virus-free, whether on 100% OSA or full recirculation. In this case the problem cannot be solved with alterations at the fan system, but in the occupied space.
3) If the live virus can be recirculated as per example "W" and if the upgraded MERV13+ filters can capture the live virus, Merv13+ filter installation should be considered.
4) Example "X" has MERV13+ filters, clean OSA and exhausts 100% at B. This will be extremely expensive, increasing Climate Change and most systems cannot do this in cold weather.
5) Example "Y" increases the OSA above current minimum ventilation code requirements and with no load analyzing logic has the same negative impact as "X", but not quite as severe.
6) Example "Y" could maintain current code levels and conservation logic if the live virus can travel to C and the MERV13+ filters can remove the live virus from the air.
7) Example "Z" MERV13+ filters capture the live virus, preventing it from entering the mixing plenum, allowing HVAC conservation design to continue, supporting Climate Change reduction.
8) The MERV13+ filters have a greater pressure drop than existing filters; therefore, will require more fan power (more electricity) to maintain equal airflow as existing arrangements.
9) Increasing the OSA intake in cold weather will tend to reduce the indoor relative humidity, which benefits the virus and is detrimental to humans.
10) If the live virus currently cannot travel from E to C, then D has always been virus-free. If the public believes that better filters and more OSA significantly solved the problem, when D is actually the same, the false sense of security may cause more careless activities in other protective measures. This could make our situation worse.
11) The MERV13+ filters, if required, may require new filter racks and spacing may be an issue in "Y", possibly making "Z" a more practical location for new MERV13+ filters.

COVID-19 Scientific Advisory Group
Rapid Evidence Report

Key Research Question:

Has there been documented transmission of SARS-CoV-2 virus (or similar viruses) through Heating, Ventilation, and Air Conditioning (HVAC) systems in hospitals or non-hospital settings?

Context

- Respiratory viruses are believed to transmit over multiple routes and the relative significance between aerosol and droplet transmission may vary among pathogens.
- The Public Health Agency of Canada (2020) and the World Health Organization (2020) consider the route of human-to-human transmission of SARS-CoV-2 to be predominantly via respiratory droplets or contact during close and unprotected contact, with a recommendation to use N95 respirators only in the context of aerosol-generating medical procedures.
- A reported outbreak of COVID-19 in an air-conditioned restaurant in Guangzhou, China that involved three family clusters was attributed to a longer range of droplet transmission, with the authors suggesting that was related to airflow generated by air conditioned ventilation but uncertainty over means of transmission remained.
- University of Alberta researchers have received research funding from the Canadian Institutes for Health Research to examine how SARS-CoV-2 may be transmitted through airborne fine particles, and how its movement is controlled by current HVAC designs in non-healthcare settings.
- There is concern about the possibility of promoting transmission of SARS-CoV-2 through HVAC systems inside and outside hospital settings.
- Evaluating the relative contribution of airborne versus droplet and contact transmission of SARS-CoV-2 is beyond the scope of this review. However, this review is presented under the working assumption that SARS-CoV-2 is primarily transmitted through respiratory droplets or contact and potentially short-range aerosols.
- Standards exist for the construction and maintenance of HVAC systems in healthcare and other settings, therefore recommendations for HVAC system design are beyond the scope of this review.

Key Messages from the Evidence Summary

- There is no clear evidence to date of transmission of SARS-CoV-2 associated with HVAC systems in hospitals or health care facilities, although there is a mechanistic possibility of this occurring. Studies that have identified the presence of viral RNA in procedure generated aerosols have not demonstrated viable virus that would be capable of infecting susceptible hosts, however viral culture may be relatively insensitive.
- There is epidemiologic evidence that HVAC conditions may have contributed to transmission of SARS-COV-2 in community settings including a restaurant, call centre and airplane, though in these events spread through close contact was not ruled out, and longer distance localized droplet spread related to airflow (given proximity to the index cases) is more likely than classic airborne transmission.
- The need to assess HVAC systems in the control of SARS-CoV-2 and other viruses is highlighted by various interim guidelines (Saran et al., 2020). Notably, rooms with higher air exchanges tend to have less viral RNA detected in the air, based on the literature identified.
- Given the complexity in the transmission modalities of SARS-CoV-2 and other similar viruses, lack of data on viable virus in air samples, and the wide variety of HVAC systems, studies have not been able to consider and evaluate all HVAC configurations and their potential to affect transmission of infection.

Alberta Health Services

June 5, 2020

and isolation rooms in relation to spread of infectious diseases via the airborne route. Many of the epidemiological studies did not include adequate airflow studies (Y. Li et al., 2007).

Luongo et al. (2016) systematic review assessed epidemiologic studies published after 2000 and investigating the association of at least one HVAC-related parameter with an infectious disease-related outcome in buildings. The authors indicate that the data implies that HVAC system factors in buildings have a role in airborne pathogen transmission, but more robust, interventional studies are needed (Luongo et al., 2016).

SARS-CoV-2 & HVAC Systems in Healthcare Settings

Liu et al. (2020) investigated the aerodynamic nature of SARS-CoV-2 by measuring viral RNA in aerosols in different areas of two Wuhan hospitals during the COVID-19 outbreak in February and March 2020 (Liu et al., 2020). They collected thirty-five aerosol samples of three different types (total suspended particles, size-segregated, and deposition aerosol) in Patient Areas (PAA) and Medical Staff Areas (MSA) of Renmin Hospital of Wuhan University (Renmin) and Wuchang Fangcang Field Hospital (Fangcang), and Public Areas (PUA) in Wuhan, China during the outbreak. The ICU, CCU and general patient rooms inside Renmin, patient hall inside Fangcang had undetectable or low airborne SARS-CoV-2 concentration but deposition samples inside ICU and air sample in Fangcang patient mobile toilet room tested positive. The toilet room was a temporary single toilet room of approximately 1m^2 in area without ventilation and had the highest viral load detected (19 copies/m^2). The airborne SARS-CoV-2 in Fangcang MSA had bimodal distribution with higher concentrations than those in Renmin during the outbreak but were negative after number of patients were reduced and rigorous sanitization was implemented. Public areas had undetectable airborne SARS-CoV-2 concentration but obviously increased with accumulating crowd flow. The authors interpreted this to suggest overall low risks in the well ventilated or open public venues. The authors also concluded that room ventilation, open space, proper use and disinfection of toilets can effectively limit aerosol transmission of SARS-CoV-2. For example, the negative pressure ventilation and high air exchange rate inside ICU, CCU and ward room of Renmin Hospital were effective in minimizing airborne SARS-CoV-2. The authors further concluded that transmission within crowds via airborne transmission is possible. The virus aerosol deposition on protective apparel or floor surface and their subsequent resuspension is a potential transmission pathway and effective sanitization is critical in minimizing aerosol transmission of SARS-CoV-2 (Liu et al., 2020).

Guo et al. (2020) tested surface and air (including air outlets) samples for SARS-CoV-2 using real-time PCR from an ICU and a general COVID-19 ward at Huoshenshan Hospital in Wuhan, China (Guo et al., 2020). Thirty-five percent (14/40) of the samples collected from the ICU and 12.5% (2/16) of the general ward samples were positive. Air outlet swab samples also yielded positive test results, with positive rates of 66.7% (8/12) of ICUs and 8.3% (1/12) for general wards. Rates of positivity differed by air sampling site with 44.4% (8/18) samples in patients' rooms, 35.7% (5/14) near air outlets and 12.5% (1/8) in the doctors' office area. The authors indicate that virus-laden aerosols were mainly concentrated near and downstream from the patients, with a maximum transmission distance of 4m. The air sampling sites in the general ward were distributed in different regions around the patient, under the air inlet, and in the patient corridor. Only air samples around the patient were positive. One of their conclusions was that SARS-CoV-2 was widely distributed in the air and on surfaces but did not associate this with HVAC systems (Guo et al., 2020). Both this and the Liu et al. (2020) study noted above are limited by the lack of viable virus testing. It is unclear whether environmental contamination with viral RNA contributes to clinical infection.

Ong et al. (2020) collected surface environmental samples at 26 sites from three airborne infection isolation rooms (12 air exchanges per hour) with anterooms and bathrooms in the dedicated SARS-CoV-2 outbreak center in Singapore between January 24 and February 4, 2020. Note: viral culture was not done to demonstrate viability. There was extensive environmental contamination by one SARS-CoV-2 patient with mild upper respiratory tract involvement. Toilet bowl and sink samples were positive, suggesting that viral shedding in stool[5] could be a potential route of transmission. Post-cleaning samples were negative, suggesting that current decontamination measures are sufficient. Air samples were negative despite the extent of environmental contamination. Two of the three swabs taken from the air exhaust outlets tested positive, suggesting that small virus-laden droplets may be displaced by airflows and deposited on equipment such as vents. The authors conclude the environment is a potential medium of transmission and supports the need for strict adherence to environmental and hand hygiene.

Buonanno et al. (2020) estimate the SARS-CoV-2 viral load emitted by a contagious subject on the basis of the viral load in the mouth, the type of respiratory activity (e.g. breathing, speaking), respiratory physiological parameters (e.g. inhalation rate), and activity level (e.g. resting, standing, light exercise). The authors conclude that the results obtained from the simulations highlight that a key role is played by proper ventilation in containment of the virus in indoor environments (Buonanno, Stabile, & Morawska, 2020).

Dbouk and Drikakis (2020) used computational multiphase fluid dynamics and heat transfer to investigate the transport, dispersion, and evaporation of saliva particles arising from a human cough (Dbouk & Drikakis, 2020). An ejection process of saliva droplets in air was applied to mimic the real event of a human cough. Their model took into account relative humidity, turbulent dispersion forces, droplet phase-change, evaporation, and breakup in addition to the droplet-droplet and droplet-air interactions. The authors further investigated the effect of wind speed on social distancing. For a mild human cough in air at 20 °C and 50% relative humidity, human saliva-disease-carrier droplets may travel up to unexpected considerable distances depending on the wind speed. When the wind speed was approximately zero, the saliva droplets did not travel 2 m, which is within the social distancing recommendations. However, at wind speeds varying from 4 km/h to 15 km/h, the saliva droplets can travel up to 6 m with a decrease in the concentration and liquid droplet size in the wind direction. The findings imply that considering the environmental conditions, the 2 m social distance may not be sufficient. Further research is required to quantify the influence of parameters such as the environment's relative humidity and temperature among others. The authors further highlight that further research is required to assess the probability of viral transmission and that a holistic approach to address these questions is needed. This would require closer interactions between individuals in medicine, biology, engineering fluid physics and social sciences.

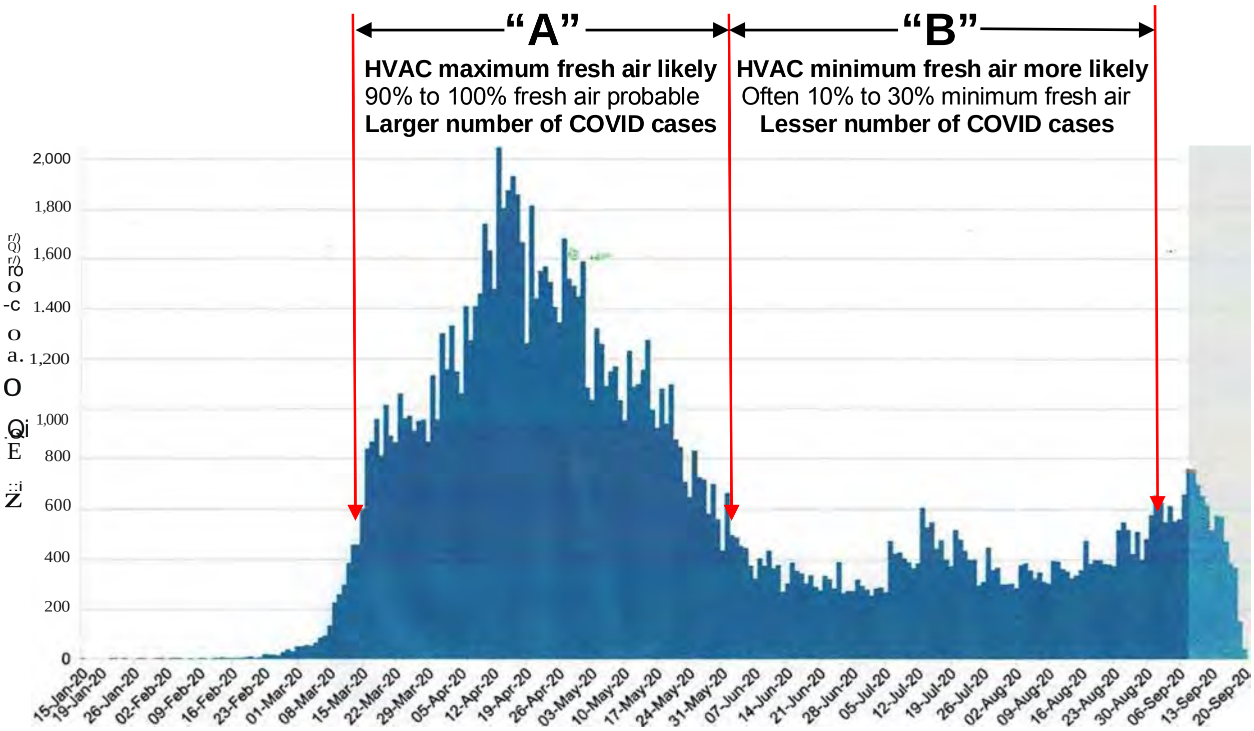
Morawska and Cao (2020) stated that based on the trend in the increase of infections, and understanding the basic science of viral infection spread, they strongly believe that the SARS-Cov-2 virus is likely to be spreading through the air and that it will take several months for this to be confirmed by science. The authors recommend all possible precautions against airborne transmission of SARS-CoV-2 virus in indoor scenarios be taken and that these precautions include increased ventilation rate, using natural ventilation, avoiding air recirculation, avoiding staying in another person's direct air flow, and minimizing the number of people sharing the same environment based on Qian & Zeng (2018). Morawska & Cao (2020) also recommend personal protective equipment, in particular masks and respirators should be recommended, to be used in public places where density of people is high and ventilation potentially inadequate, as they can protect against infection (by infected individuals) and infecting others (L. Morawska & Cao, 2020).

Other Viruses & HVAC Systems in Healthcare Settings

Li et al. (2004) presented a detailed air distribution study of a hospital ward during a major nosocomial outbreak of SARS-CoV in Hong Kong in March 2003 (Y. Li, Huang, Yu, Wong, & Qian, 2004). Retrospective on-site inspections and measurements of the ventilation design and air distribution system, three months after the outbreak, showed that the flow rates in the supply diffusers and exhaust grilles were not balanced. Measurements performed using bio-aerosol generator (with diameters between 0.1 and 10um) placed in one of the beds next to the index patient's bed (since it was occupied). At a height of 1.1m, concentration decreased as the virus laden bio-aerosols moved away from the index patient's cubicle. The concentrations at the doorstep of patient's toilet and store/clean room were relatively high – so risk if other patients in distant cubicles visited the toilets. Extraction fans in the store/cleaning room and in the patient's toilet seems to have also contributed to spread of bio-aerosols from index patient's cubicle to corridor and nurses station. Using their simulations and measurements, the predicted bio-aerosol concentration agreed with the spatial infection pattern which the authors indicated it suggested a probable airborne transmission route, in addition to the commonly accepted large droplet and close personal contact transmission (Y. Li et al., 2004).

Yu et al. (2005) conducted a retrospective cohort study of the SARS-CoV outbreak, noted above, on a hospital ward at the Prince of Wales Hospital, China (Yu, Wong, Chiu, Lee, & Li, 2005). Information on roles of healthcare workers and the ventilation system (location and size of supply diffusers, exhaust grills, supply air temperature, and the air-flow rate through each supply diffuser, exhaust grille and exhaust fan) were collected. Dispersion of hypothetical virus-laden aerosols, originated from the index case patient's bed, through the entire ward was

Normal fresh air ventilation trend in HVAC mixed air systems
Consideration of normal fresh air intake to COVID cases relationship



Canada's COVID infection graph. Date of illness onset

APRIL AVERAGE CANADA LOW/HIGH TEMPERATURES
Vancouver, BC: 44 / **56F**, (7 / 13C)
Edmonton, AB: 31 / **52F**, (-0 / 11C)
Winnipeg, MB: 32 / **51F**, (0 / 11C)
Ottawa, ON: 34 / **52F**, (1 / 11C)
Toronto, ON: 39 / **54F**, (4 / 9C)
Montréal, QC: 34 / **49F**, (1 / 9C)
Halifax, NS: 33 / **47F**, (0 / 8C)
St. John's, NF: 30 / **41F**, (-1 / 5C)
(Temperature values via tripsavvy)

MAY AVERAGE CANADA LOW/HIGH TEMPERATURES
Vancouver, BC: 49 / **63F**, (9 / 17C)
Edmonton, AB: 42 / **64F**, (6 / 18C)
Winnipeg, MB: 44 / **64F,** (7 / 18C)
Ottawa, ON: 47 / **67F**, (8 / 19C)
Toronto, ON: 50 / **66F**, (10 / 19C)
Montréal, QC: 46 / **63F**, (8 / 17C)
Halifax, NS: 42 / **58F**, (6 / 14C)
St. John's, NF: 36 / **51F**, (2 / 11C)

JUNE AVERAGE CANADA LOW/HIGH TEMPERATURES
Vancouver, BC: 54 / **67F**, (12 / 19C)
Edmonton, AB: 50 / **70F,** (10 / 21C)
Winnipeg, MB: 56 **/ 74F,** (13 / 23C)
Ottawa, ON: 56 / **75F**, (13 / 24C)
Toronto, ON: 59 / **76F**, (15 / 24C)
Montréal, QC: 58 / **75F**, (14 / 29C)
Halifax, NS: 51 / **67F**, (11 / 19C)
St. John's, NF: 43 / **60F**, (6 / 16C)

1) HVAC mixed air fans are more likely to maximize fresh air close to 100% in the seasonal range **"A"** because the outdoor air day time average temperature is close to the supply fans' normal low limit set point of 55˚F to 60˚F.
2) HVAC mixed air fans are more likely to minimize fresh air to design minimum fresh air quantities in the seasonal range **"B",** as the fresh air "free cooling" is often prevented, as determined by enthalpy comparison or dry bulb fresh air high limits via economiser logic. (Fresh air and return air enthalpies compared or fresh air over about 70˚F reverts fresh air quantity to minimum reducing mechanical cooling energy requirements.)
3) The current position that more fresh air will assist in the reduction of COVID-19 infections seems to be questionable given the infection rate and normal HVAC seasonal fresh air intake as graphed. Other factors seem dominant.
4) Most case studies focus on virus concentrations in the occupied space, but do not quantify virus concentrations in the airflow to and from the HVAC fans. Data is required showing virus concentrations in the occupied space, fresh air, return air as it enters the mixing plenum, in the exhaust air and the supply fans' discharge. This data is necessary in developing the most effective and most energy efficient plan.
5) It can be argued via some current case studies that there will be no gain in virus transmission reduction by flushing with 100% fresh air over mixed air, but 100% fresh air will increase operating costs and GHG emissions.

SUMMARY

1) I suggest that in every organization an experienced HVAC engineer should oversee each total COVID-19 protection plan regarding HVAC alterations. Organizations employ people with varying types of expertise, however having a governing expert to provide oversight during this pandemic will ensure integration of knowledge, identification of gaps and overall success.

2) The effectiveness of the MERV13+ filters in capturing the COVID-19 virus should be verified. If 100% effective, return air filtration may allow conservation logic to remain, respecting the Climate Change threat.

3) A study assessing the virus content at HVAC locations presented on page 11.203 is required via a credible governing body. Understanding the locations of the virus is critical to establishing a safe plan.

4) The air from the supply fan must be viewed differently at two separate locations in the HVAC system. The airflow should first be considered from the supply fan to where it enters the occupied space and secondly as a different airflow as it passes over people in the occupied space. Completely virus-free air entering the occupied space can possibly pick up and transfer viruses between occupants in the space. Alterations to the supply air at the fan will be pointless if the air is already clean and the virus transfer is only within the occupied space.

5) If the virus cannot be in the return air at the entrance to the mixed air plenum by either return air filtration or the virus being incapable of travelling that far, allowing recirculation of air as per existing conservation logic should be considered respecting the Climate Change threat.

6) The impact of humidity levels between 40% RH and 60% RH on viruses and human health should be clarified. ASHRAE and REHVA have differing opinions. ASHRAE indicates between those RH values, viruses are less dangerous and humans' health is improved. If ASHRAE is correct, flushing with maximum quantities of cold fresh air is beneficial to the virus and detrimental to humans, because it will reduce relative humidity levels in the occupied space.

7) Information is changing on this issue as research is being reported. Everyone is putting their best effort into solving this common threat. Working as a positive team has value.

8) Questions, constructive criticism on any points or added information will be appreciated.

9) Acceptance and application of any concepts suggested in this report is the total responsibility of those designing, installing and commissioning the individual projects.

BOURDON TUBE EXPERIMENTS

JULY 7, 2021

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Fax: (905) 640-2444

PREFACE

This investigation is intended as a stepping stone to methods significantly addressing the Climate Change issue. We wish to partner with the R & D division of a major manufacturer, regularly using Bourdon tubes, based on three scientific facts they currently accept.

A supporting letter from Dr. Gorber, past President of SENES Consultants (energy scientists), regarding an associated patented invention (link to patent offices page 12.214), concludes with:

"I believe that Mr. Strain's invention will advance the scientific community's understanding of thermodynamics relating to pressurized fluids and energy to a new level. If fully developed the invention has the potential to reduce energy and as a result a reduction in the use of fossil fuels, thus assisting in the battle against climate change."

Practical application must be finalized. To survive together, we must work together.

The three accepted facts are:

1) Pascal's Principal (page 12.216) is valid respecting fluid pressure in a contained volume.
2) The volume of a Bourdon tube does not change on being pressurized. (Industry experts contacted accept this fact or state they have no data on the matter.)
3) The tip of a Bourdon tube exerts a force at its tip as it travels through a distance.

The three facts combine into a concept challenging physical laws. When the potential working capability of a Bourdon tube with a lesser range exerts its force on the non-compressible surface of a second Bourdon tube, with a higher range, the work output of the second bourdon tube is greater than the work output of the first Bourdon tube. (illustrated page 12.225)

The concluding fact is Work input < Work output. ($W_{input} < W_{output}$)

I urge you to consider the basis for accepting the physical laws established in the 1800's. If the claim is made that anything is unachievable, those making such claims must have all knowledge regarding the subject; past, present and future.

There are two recently patented inventions directed at challenging these laws. (Patent office links page 12.214). I'm quite confident the men from the 1800's did not time travel and refute inventions that challenge their claims. Multiple engineers/scientists/professors have assessed the inventions, which contest the opinion from the 1800's, but their current challenges contained nothing more than statements of blind faith in the "Laws" or a no contest position. None refuted the mechanical design, control circuitry or logic in the reports given to them.

Quoting Einstein: " *A foolish faith in authority is the worst enemy of truth.*"

If the scientific community does not adjust to thinking creatively and support their opinions with actual science, humanity is headed for disaster.

CONTENTS

NOTE:

Initial testing of appartaus presented on pages 12.222 and 12.223, as well as the appartaus presented on page 12.225, contrary to industry expert opinion, indicate that the volume of a Bourdon tube changes as it flexes. Further experimentation is required as of May 31, 2022 to better understand the causes of the intial test results. The volume change of the intruding plunger relative to the Bourdon tube's flexing volume change must be determined.

BOURDON TUBE

WORK IN < WORK OUT ASSESSMENT

RESEARCH APPROACH

-1- Assess if the volume of a Bourdon tube changes on pressurization.

-2- Investigated current engineering and scientific knowledge on the subject.

-3- Investigated other attempts of identifying Bourdon tube characteristics.

-4- Formulated experiments to determine Bourdon tube characteristics.

THE BOURDON TUBE EXPERIMENT

-1-The need was to provide an alternate and simple demonstration, supporting the diamond-shaped actuator patent, suggesting that the Laws of Thermodynamics have over-looked important fluid relationships, regarding energy/mechanical work.

Successful experiments would also confirm Pascal's Principle.

The intent was to determine if a Bourdon tube with a lesser pressure range than a second bourdon tube will produce enough work at its tip to pressurize the second Bourdon tube, producing work at the tip of the second bourdon tube greater than the work of the first Bourdon tube.

-2 We contacted many gauge manufacturers to determine if the volume of a Bourdon tube changed when being pressurized and depressurized. A few thought the volume did not change: most had no data on the matter.

-3- Cynthia Conway in her 1995 Engineering Masters thesis investigated the question regarding Bourdon tube characteristics. She was required to assume the volume changed as she found no conclusive data on the matter after extensive investigation. The concluding result is that it appears unknown if the volume changes in a Bourdon tube when pressurizing and de-pressurizing. View Cynthia Conway's full Thesis at https://core.ac.uk/download/pdf/36724053.pdf

-4- If the volume does not change or changes less than the submersed volume of a plunger forced into the Bourdon tube's fluid, a demonstration may be presented proving that Work input < Work output ($W_{in} < W_{out}$), which contradicts the Laws of Thermodynamics.

The adaptation compensating for small volume changes is addressed in the patent, "PATENT USING BOURDON TUBE CHARACTERISTICS".

Preparation for the experiments include:

-a- obtaining materials required for testing.
-b- obtaining information from the NASA site addressing Pascal's Principle.
-c- preparing an illustration of Pascal's Principle as it relates to Bourdon tubes.
-d- design a method of assessing Pascal's Principle.
-e- design a method of measuring the varying force at the tip of a Bourdon tube.
-f- design a method of measuring the varying displacment at the tip of a Bourdon tube.
-g- design a method of testing if a Bourdon tube changes volume on pressurization.
-h- design a method of demonstrating Work input < Work output

PATENT using Bourdon tube characteristics

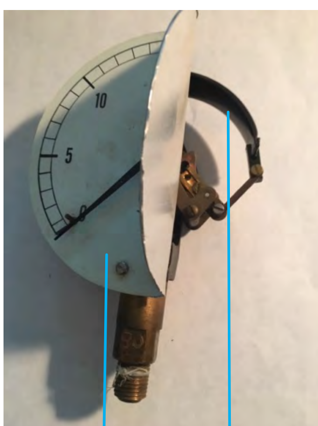
https://image-ppubs.uspto.gov/dirsearch-public/print/downloadPdf/7467517

Copy and paste link into your browser. (Patent# 7467517)

Link for the first patented invention.

https://worldwide.espacenet.com/inpadoc?submitted=true&DB=EPODOC&CC=US&NR=2002178719&KC=&F=8&OREQ=0&textdoc=TRUE&FT=E

SPECIALIZED COMPONENTS FOR EXPERIMENTS



gauge face   Bourdon tube

Bourdon   whisker   pneumatic
pneumatic tube   sensor   sensor
push button

Regarding the experiment presented on page 12.222;

Testing determined that the push buttons required too much force to use them in the experiment.

Testing determined that the pneumatic proximity sensors were not satisfactory for the experiments.

Testing determined that the pneumatic whisker sensors were the best option for the experiments.

PASCAL

Pascal (19 June 1623 – 19 August 1662) was a significant scientist advancing the knowledge of mankind in the fluidics field and more.

PASCAL'S PRINCIPLE

"*A change in pressure applied to an enclosed fluid is transmitted undiminished to all portions of the fluid and to the walls of its container.*"

Page 12.216 was obtained from the NASA WEB site which presents the basic understanding of Pascal's Principle.

Page 12.217 presents the relationship of Pascal's Principle in a practical application to Bourdon tubes in gauges.

Understanding Pascal's Principle at a very practical level is imperative inorder to grasp this fluidic advancement.

Pascal's Principle and Hydraulics

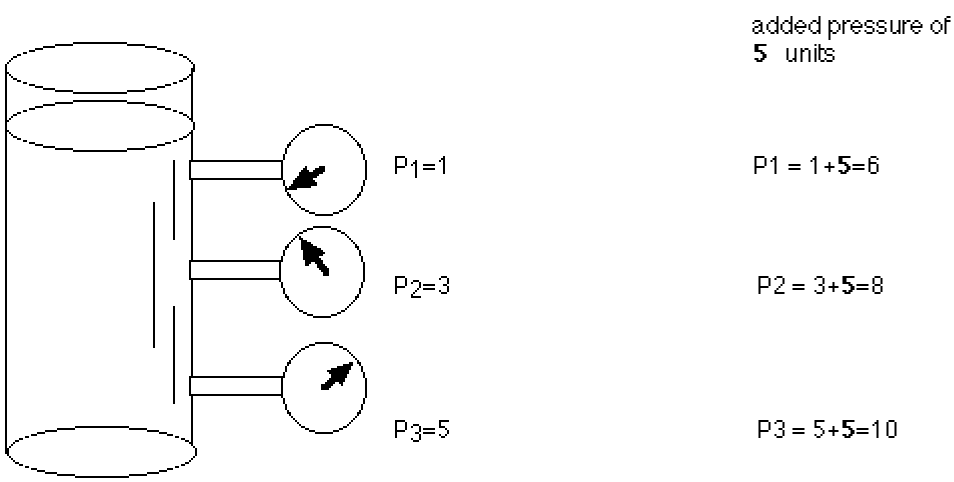
SUBJECT: Physics
TOPIC: Hydraulics
DESCRIPTION: A set of mathematics problems dealing with hydraulics.
CONTRIBUTED BY: Carol Hodanbosi
EDITED BY: Jonathan G. Fairman - August 1996

Hydraulic systems use a incompressible fluid, such as oil or water, to transmit forces from one location to another within the fluid. Most aircraft use hydraulics in the braking systems and landing gear. Pneumatic systems use compressible fluid, such as air, in their operation. Some aircraft utilize pneumatic systems for their brakes, landing gear and movement of flaps.

Pascal's law states that when there is an increase in pressure at any point in a confined fluid, there is an equal increase at every other point in the container.

A container, as shown below, contains a fluid. There is an increase in pressure as the length of the column of liquid increases, due to the increased mass of the fluid above.

For example, in the figure below, P3 would be the highest value of the three pressure readings, because it has the highest level of fluid above it.



If the above container had an increase in overall pressure, that same added pressure would affect each of the gauges (and the liquid throughout) the same. For example P1, P2, P3 were originally 1, 3, 5 units of pressure, and 5 units of pressure were added to the system, the new readings would be 6, 8, and 10.

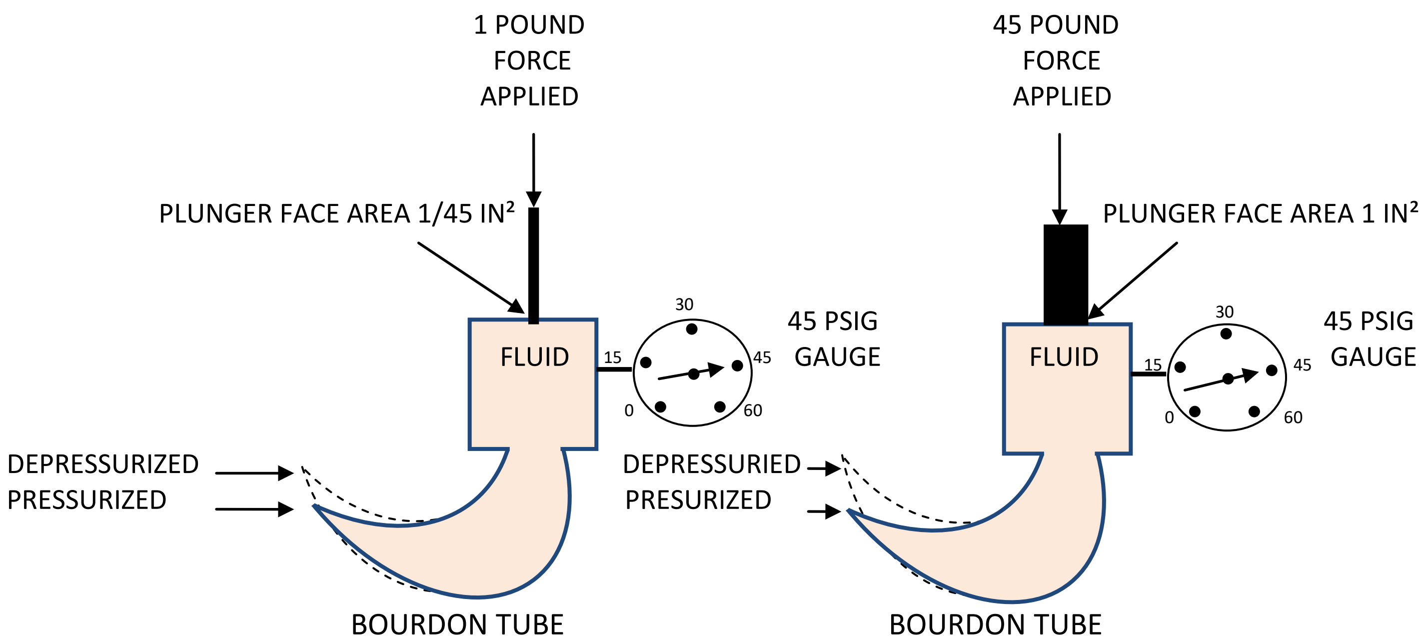
PASCAL’S PRINCIPLE AS IT APPLIES TO BOURDON TUBE PERFORMANCE

PASCAL’S PRINCIPLE

"*A change in pressure applied to an enclosed fluid is transmitted undiminished to all portions of the fluid and to the walls of its container.*"

IF NO VOLUME CHANGES ON PRESSURIZATION

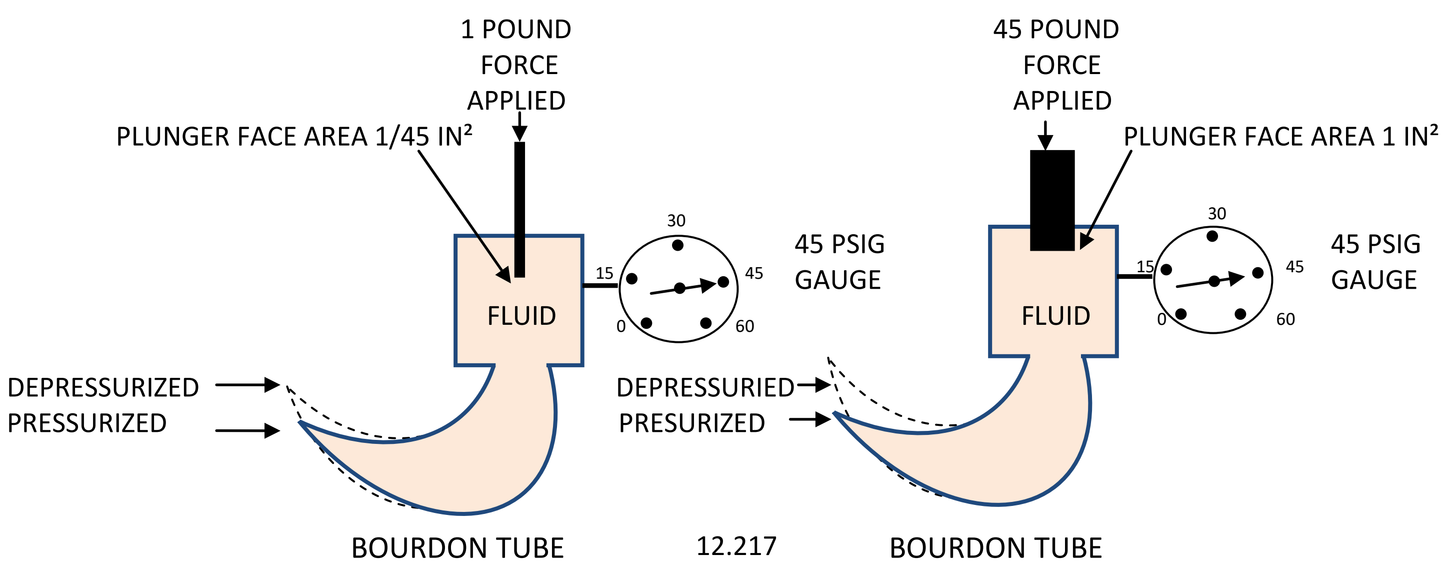
Assuming the volume of a Bourdon tube does not change on pressurization, as per opinion of industry experts and Pascal’s Principle is valid, the illustrated Bourdon tubes can be pressurized to 45 PSIG by various combinations of force and fluid surface area where the force is applied. Two examples are illustrated.



IF VOLUME CHANGES ON PRESSURIZATION

Assuming the volume of a Bourdon tube does change on pressurization, contrary to opinion of industry experts and Pascal’s Principle is valid, the illustrated Bourdon tubes can be pressurized to 45 PSIG by various combinations of force and fluid surface area where the force is applied. Two examples are illustrated.

This is valid if the displaced fluid volume, via the plunger, is greater than the volume change in the fluid when altering from depressurized to pressurized.



BOURDON TUBE FORCE VARIATION AND DISPLACEMENT VARIATION

1) Page 12.219 presents the experiment determining the Bourdon tube tip force variation at different internal pressures.

2) Page 12.220 presents the experiment determining the Bourdon tube tip displacement at various internal pressures.

FORCE VARIATION AT DIFFERENT PRESSURES IN A BOURDON TUBE

"A"

FORCE = 9 OZ.

PRESSURE = 20 PSIG

BOURDON TUBE

FORCE = 1 POUND = 16 OZ.

"B"

PRESSURE = 45 PSIG

"C"

FORCE = 1 POUND & 15 OZ. = 31

OZ. PRESSURE = 60 PSIG

NOTE:

The 31 OZ. force of "C" can be attained by increasing the fluid supply pressure, as illustrated, to 60 PSIG, or if Pascal's Law is true, by isolating the Bourdon tube from the pressure source and applying the one pound (16 OZ.) force of "B" to 1/60th IN^2 of the incompressible fluid surface of "C", causing a 60 PSIG the pressure reading.

If **W**ork = **F**orce X **D**istance (**W=FD**) and ("**D**" FOR "B") = ("**D**" FOR "C"); then **W** "B" = 16 units and **W** "C" = 31 units; however **W** "B" can generate **W** "C", as explained; therefore, "B" can attain 193.75% (31 OZ/16 OZ) more **W** if its force is applied to the fluid surface of the incompressible fluid surface of "C". Conclusion: Lesser $\mathbf{W}_{input}$ can produce greater $\mathbf{W}_{\mathbf{output}}$.

DISPLACEMENT AND FORCE EXERTED OF A BOURDON TUBE AT VARIOUS PRESSURES

21 PSIG

1.6 mm DRIVE

(FORCE = 9 OZ)

START

PRESSURE GAUGE

DIGITAL CALIPER

BOURDON TUBE

45 PSIG

3.23 mm DRIVE

(FORCE = 16 OZ)

59 PSIG

4.56 mm DRIVE

(FORCE = 31 OZ)

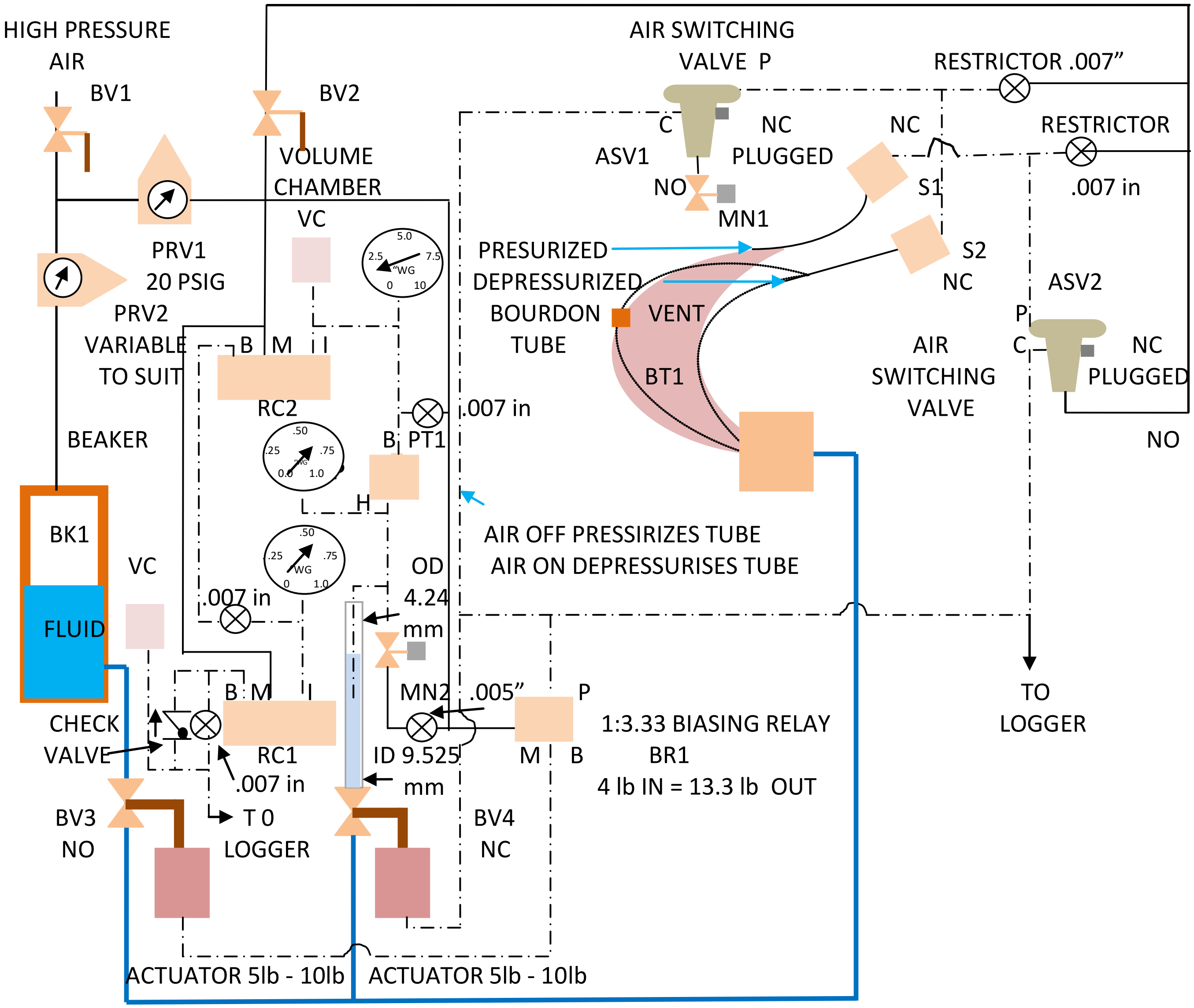
VOLUME RELATIONSHIP EXPERIMENT

The presentation on page 12.222 illustrates the experiment to assess if the volume changes on pressurization of a Bourdon tube.

The photo on page 12.223 presents the physical arrangement of the test.

More pneumatic components and a different valve arrangement were required to cause the system to reciprocate as require for the test.

Venting the air from the system has presented a serious challenge.



-1- No air pressure; therefore, Bourdon tube (BT1) is relaxed, hitting sensor (S2).

-2- Open ball valve (BV1) pressurizing beaker (B1). Ball valve (BV3) allows fluid pressure to Bourdon tube (BT1), flexing BT1 to hit sensor (S1). BT1, via S1, causes air switching valve (ASV2) to position NO to C.

-3- Open ball valve (BV2), allowing pressure to build via S1 to ASV1's pilot, closing ASV1's NO port.

-4-BT1 opens S2, causing pressure from PRV1 to go through ASV2 to switch fluid BV3 and BV4 and to the logger. BV3 closes BT1 to beaker (BK1) and opens BT1 the 9.525 mm ID clear tube VIA BV4, depressurizing BT1. Biasing relay BR1 coordinates BV3 and BV4 so only one can be open at a time and there is two pound dead span where both are closed during switching.

-5- BT1 relaxes and returns to its depressurized position.

-6- If a volume increase is experienced by BT1 on pressurization, fluid will flow to the 9.525 mm ID tube on depressurization.

-7- On depressurization BT1 will return to hit S1 and close S2. ASV2's NO port will close. S1 will open opening ASV1's NO port to C.

-8- Microneedle valve (MN1) will determine the rate of cycles as BV3 and BV4 change their porting on switching.

-9- This creates a reciprocating cycle. Note the "WG level change of the 9.525 ID tube after a few cycles.

-10- If it appears that no level change is experienced, continue experiment. (If level increases; calculate % change.)

-11- Adjust MN1 to establish a fixed number of cycles per minute.

-11- The change in fluid in the 9.525 mm ID tube is sensed by pressure transmitter (PT1) via the 4.24 OD tube with air slowly bubbled into it via MN2. Minimum bubbles maintained while keeping the 4.24 OD tube the waterless.

-12- Program trend reader to read cycles over prolonged period of time and calculate level change for 1,000 cycles.

-13- The open area to atmosphere of fluid in the 9.525 ID tube with 4.24 mm tube inserted is .57107 cm².

-14- Testing indicates that one drop of water alters the level in the 9.525 mm ID tube by .02" WG.

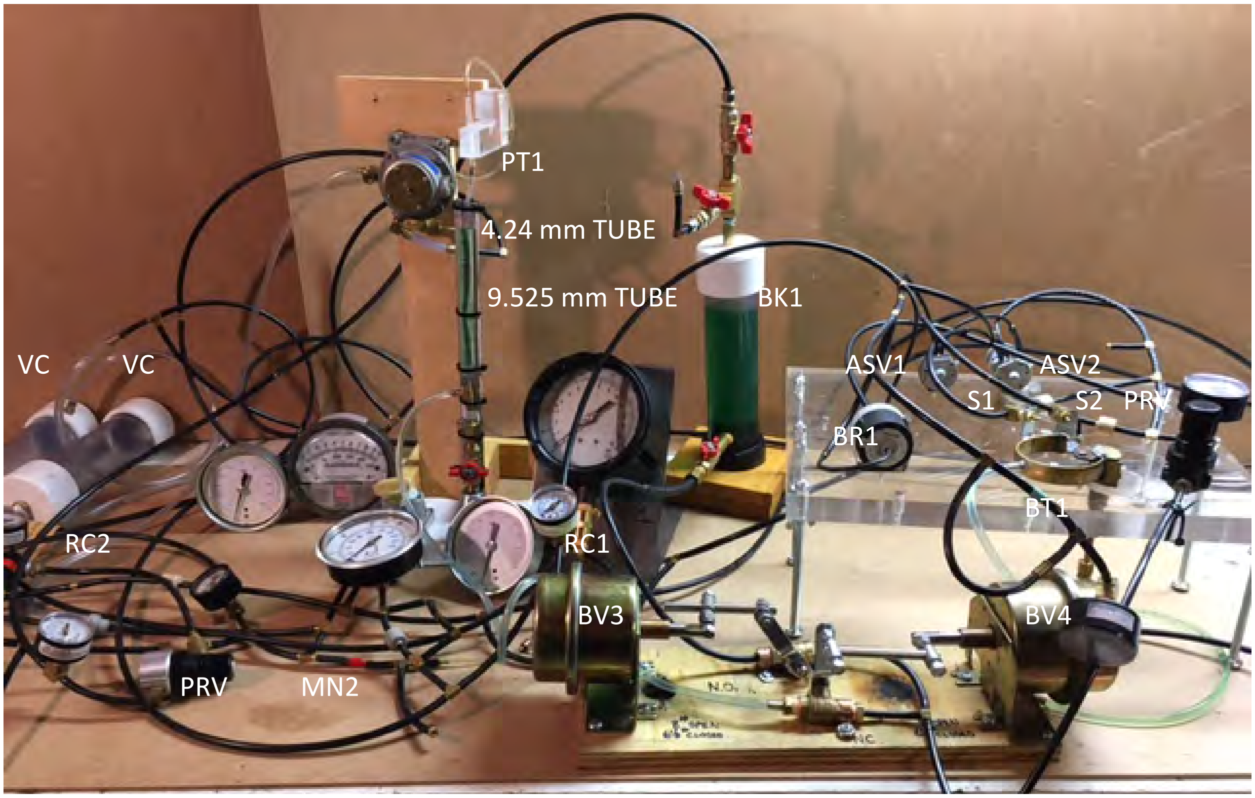
-15- .02" WG change equals a volume change of about .029 cm³ (.029 gram), producing a 3 PSIG output change from RC1, which has an input range of 1.0" WG.

-16- On the Trend reader each PSIG represents 1.49 volts which can be subdivided into thousands of a volt.

-17- The test equipment is able to record volume changes of about .000001 cm³ (.000001 gram of pure water).

PHYSICAL TEST MODEL OF CIRCUIT PRESENTED ON PAGE 12.222

The test equipment is able to record volume changes of about .000001 cm^3



WORK INPUT INCREASED TO WORK OUTPUT

-Page 12.225 presents the experiment regarding increase of work output over work input.

-Page 12.226 is a page from USA patent 7467517 commenting on a volume increase compensation method.

If the volume does increase on pressurization, the entering shaft, exerting the force on the second tube can mitigate the volume change impact if the volume of the shaft displacement is greater than the volume change of the bourdon.

POTENTIAL WORK INCREASE APPLYING PASCAL'S PRINCIPLE USING BOURDON TUBES

Using the experimental data presented on pages seven and eight, the relationship illustrated below of two Bourdon tubes presents the case of $Work_{output} > Work_{input}$. (.3492 in-lb output work potential generated by .127 in-lb input work potential)

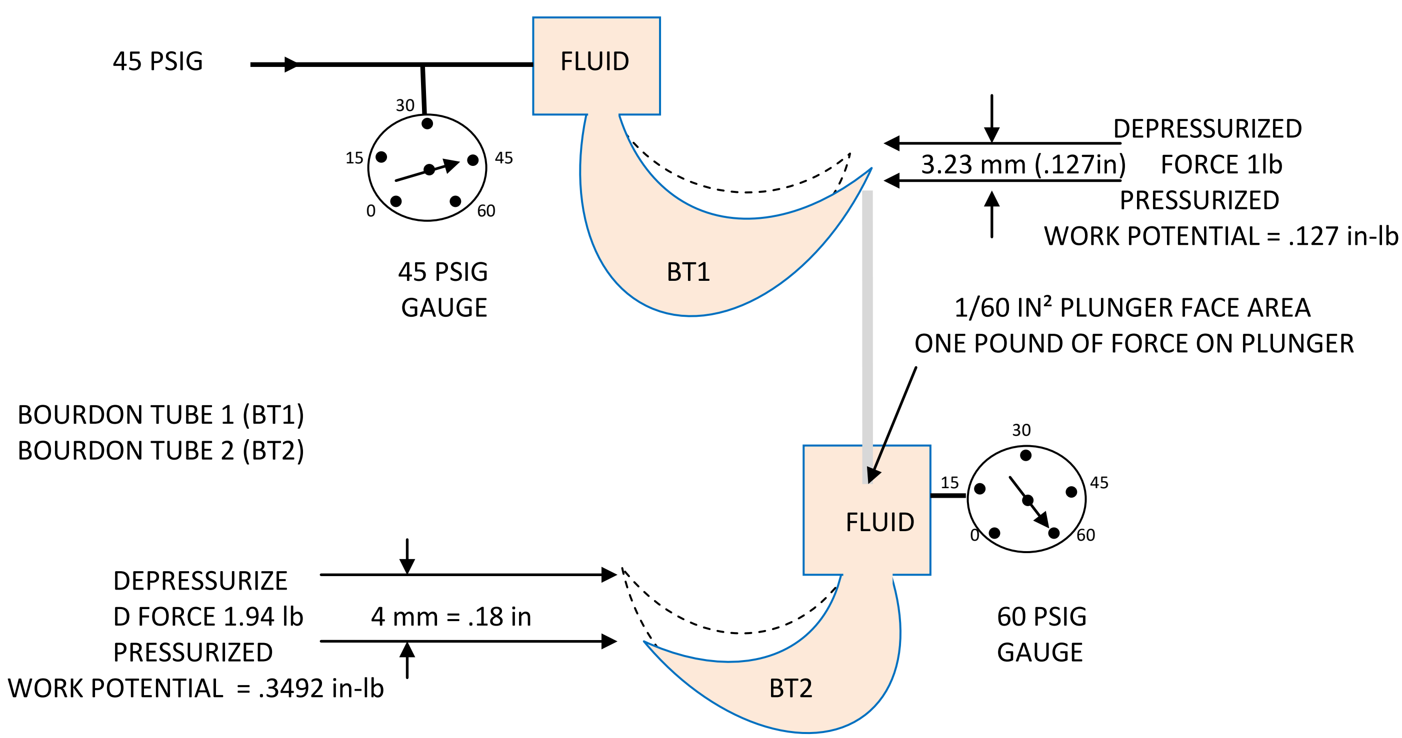
This conclusion is based on:

1) Pascal's Principle being valid and either
2) The volume does not change in Bourdon tubes when being pressurized or
3) The volume displaced by the plunger into Bourdon tube (BT2) is greater than the volume change in BT2 caused by pressurization.

NOTE: Of several R & D departments contacted of manufacturers using bourdon tubes, most stated they had no data on if the volume in a Bourdon tube changes on pressurization. Two manufacturers stated that they believe the volume does not change.

Extensive research concludes that data on Bourdon tube volume change is likely unavailable in the industry.

A volume change compensating feature in a current patent may compensate for volume change while maintaining the fluidic advancement presented at the top of this page.



CONCLUSION: .3492 in-lb > .127 in-lb; therefore $W_{output} > W_{input}$

In one aspect or embodiment, the invention uses a solenoid to apply a force to a fluid in communication with a near constant volume fluidic linkage. A voltage applied to the solenoid pressurizes the fluid causing the constant volume fluidic linkage to displace. In this way, a transducer is created that converts voltage into a displacement. To create a motor, the displacement is used to stop the application of the voltage to the solenoid allowing the constant volume fluidic linkage to return to an initial state. The return to the initial state then triggers the re-application of the voltage. In this way, the voltage source is used to create a reciprocating motion. Because the displacement is produced through a constant volume fluidic linkage, the solenoid travels only through a minimal, if any, stroke. Because very little, if any, change in volume is required, the fluidic piston may be replaced with another linkage, such as one with a diaphragm wall that does not require moving seals. Alternately, the size of any moving seals may be minimized. Among other advantages, the transducer or linkage may avoid one or more of the inefficiency, mechanical complexity and wear related problems associated with long-stroke solenoids or conventional stroking fluid pistons. Although a voltage source is used as the input energy source, analogous transducers or motors may use fluid pressure sources, such as compressed air or a liquid pressure, as the input energy either to replace the solenoid as the drive for the plunger or to pressurize the inside of the constant volume fluid linkage directly. Liquids, particularly minimally compressive liquids, are preferred for use in all fluid filled parts of the transducer or motor to reduce volume changes due to compression of the fluid. A volume compensating circuit or device may be used to reduce or eliminate the effects of any change in the volume contained in the transducer or motor or the effects of compression of the contained fluid.

This summary is intended to introduce the reader to the invention but not to define or limit the invention. Other aspects of the invention may reside in other combinations or sub-combinations of elements or steps described above or in other parts of this patent.

BRIEF DESCRIPTION OF THE DRAWINGS

One or more embodiments of the invention will now be described with reference to the following figure(s).

FIG. 1 is a schematic representation of a motor having a transducer with a constant volume fluidic linkage.

DETAILED DESCRIPTION OF THE INVENTION

FIG. 1 shows a motor 10 having a transducer 12 which converts electrical energy or pressure into a reciprocating energy. The transducer 12 has a solenoid 14 that applies a force to a plunger 16 when a voltage is applied to the solenoid 14. The plunger 16 may or may not move in response to the applied force. The plunger 16 bears on a fluid 18 in a container 20. For example, the plunger 16 may protrude into the container 20 through a seal 22 which may be an O-ring, U-cup or other type of seal. The seal 22 permits movement of the plunger 16 while maintaining a seal between the plunger 16 and container 20. Thus, if the contained volume in the motor 10 increases by a small amount or if the fluid compresses by a small amount when the fluid is pressurized, movement of the plunger 16 into the container 20 can compensate for these effects by decreasing the contained volume of the motor 10. Alternately, the plunger 16 may bear on the outside of a diaphragm wall of the container 20 or the plunger 16 and container 20 may be a conventional fluidic piston.

The container 20 is connected through fitting 24 to the base 26 of a Bourdon tube 28. As discussed above, other sorts of (near) constant volume fluid linkages may be used in place of the Bourdon tube 28. The base 26 allows the Bourdon tube 28 to be mounted to a structure and also provides a path for fluid communication between the fitting 24 and first end 30 of the Bourdon tube 28. A second end 32 of the Bourdon tube 28 has a hook 34 for attaching the motor 10 to a driven device, for example a crankshaft. A vent valve 36 allows air to be bled from the Bourdon tube 28 so that it is filled entirely with the fluid 18. The fluid 18 is preferably a minimally compressive liquid such as water, mercury or glycerine.

When the fluid 18 is at an initial pressure, the second end 32 of the Bourdon tube 28 rests in a first position 40. When a voltage, resulting in a current, is applied to the solenoid 14, a force is applied to the plunger 16 which pressurizes the fluid 18. This causes the second end 32 of the Bourdon tube 28 to move to a second position 42. When the voltage is removed, the second end 32 of the Bourdon tube 28 returns to the first position 40. An electrical circuit 50 connects a

OPINIONS OF INDUSTRY EXPERTS

Pages 12.228, 12.229 and 12.230 are from an Engineering Master's Thesis prepared by Cynthia Conway. After extensive research no data could be found on if the volume of a Bourdon tube changed on pressurization.

We also did extensive research attempting to determine if the volume of a Bourdon tube changed on pressurization. The last attempt questioned the R & D divisions of gauge manufacturers who work with bourdon tubes. Of the many approached, four stated they have no data on the matter and two stated that the volume does not change.

The net result left us concluding that it is an unknown fact.

If it does change, the volume compensation characteristic will still allow the proof of $W_{in} < W_{out}$, if the volume change is less than the volume displacement of the entering shaft forced by the first Bourdon tube illustrated on page 12.217.

NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA

THESIS

ANALYTICAL ANALYSIS OF TIP TRAVEL IN A BOURDON TUBE

by

Cynthia D. Conway

December, 1995

Thesis Advisor: R. Mukherjee

Approved for public release; distribution is unlimited.

Approved for public release; distribution is unlimited.

ANALYTICAL ANALYSIS OF TIP TRAVEL IN A BOURDON TUBE

Cynthia D. Conway
Lieutenant, United States Navy
B.S., Clemson University, 1990

Submitted in partial fulfillment
of the requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

from the

NAVAL POSTGRADUATE SCHOOL
December 1995

Author: [redacted]
Cynthia D. Conway

Approved by: [redacted]
Ranjan Mukherjee, Thesis Advisor

[redacted]
Matthew D. Kelleher, Chairman
Department of Mechanical Engineering

12.229

C. PROBLEM FORMULATION

1. Geometric Relationships

Assuming that the volume of the Bourdon changes as deformation occurs, the change in the semi-major axis *da* is expressed as a function of the change in the semi-minor axis *db* by the equation

$$da = \frac{a \cdot b \cdot (E - K)}{E \cdot a^2 - K \cdot b^2} \cdot db \qquad 2.8$$

The derivation of equation 2.8 is presented in Appendix A. For a unit length, the volume of the Bourdon is defined by the equation

$$V = \pi \cdot a \cdot b \qquad 2.9$$

The change in volume then becomes

$$dV = \pi (a \cdot db + b \cdot da) \qquad 2.10$$

or

$$dV = \pi a \left[\frac{\left(a^2 + b^2\right) \cdot E - 2Kb^2}{Ea^2 - Kb^2} \right] \cdot db \qquad 2.11$$

by making the substitution for *da* in equation 2.10.

Bourdon tip travel is a function of the sectional deformation that occurs as a result of pressurization. For an applied internal pressure, a curved circular tube will not

SUMMARY

If Pascal's Principle is valid and the industry experts' opinion that no volume change occurs during pressurization of a Bourdon tube is true, proof that $W_{in} < W_{out}$ can be achieved.

We believe that, although indicating the Laws of Thermodynamics require reassessment, the actual extra work potential available is likely minor; however, open doors to significant gains.

The purpose of this exercise is to attempt breaking down the psychological barrier causing uncontested acceptance of the Laws of Thermodynamics.

I once met a man who claimed to have had the position in both the Federal and Provincial governments assessing such inventions that challenged the status quo. He stated "For the life of me, I could not see why some of those inventions would not work, but we had to reject them anyway because the government could not be seen as supporting such inventions."

We believe this was dishonest and unscientific on the Government's part and retarding to the advancement of real science.

We believe it will require only one invention breaking through the unsupported faith in the Laws of Thermodynamics to open the doors to many rejected inventions that will help mankind in addressing the Climate Change threat we all face.

The patented invention linked in this report (page 12.214) to the Diamond-shaped fluid powered linkage, system and engine has been assessed by many scientist/engineers. All have agreed with the fact that the new style actuator is more efficient than a convention piston. The basic principle of more efficiency was patented in 1874 by Mr. Reilley. This invention has the potential to provide pollution-free work addressing the Greenhouse Gas issue associated with Climate Change.

Progress has been blocked by scientist/engineers/professors who have not provided any supporting explanation of their rejection of the detailed drawings and physical demonstrations, other than quoting the Laws of Thermodynamics.

Please send questions or comments.

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FUNDAMENTAL SCIENTIFIC DISCOVERY

REGARDING

THE DIAMOND-SHAPED ACTUATOR

ADVANTAGE OVER

CONVENTIONAL PISTONS

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HYDRAULIC DISPLACEMENT MOTOR FUNDAMENTAL SCIENCE

The foundational point of the following discovery establishes that a diamond-shaped actuator can produce 100% of it's fluid requirements, as illustrated in FIGURE "D", forcing the fluid from a standard piston, while using only 85% of its work potential. The remaining 15% work potential can be captured as free work. **No other energy source is required.** This paper develops an understanding of this fact.

FIGURE "A"

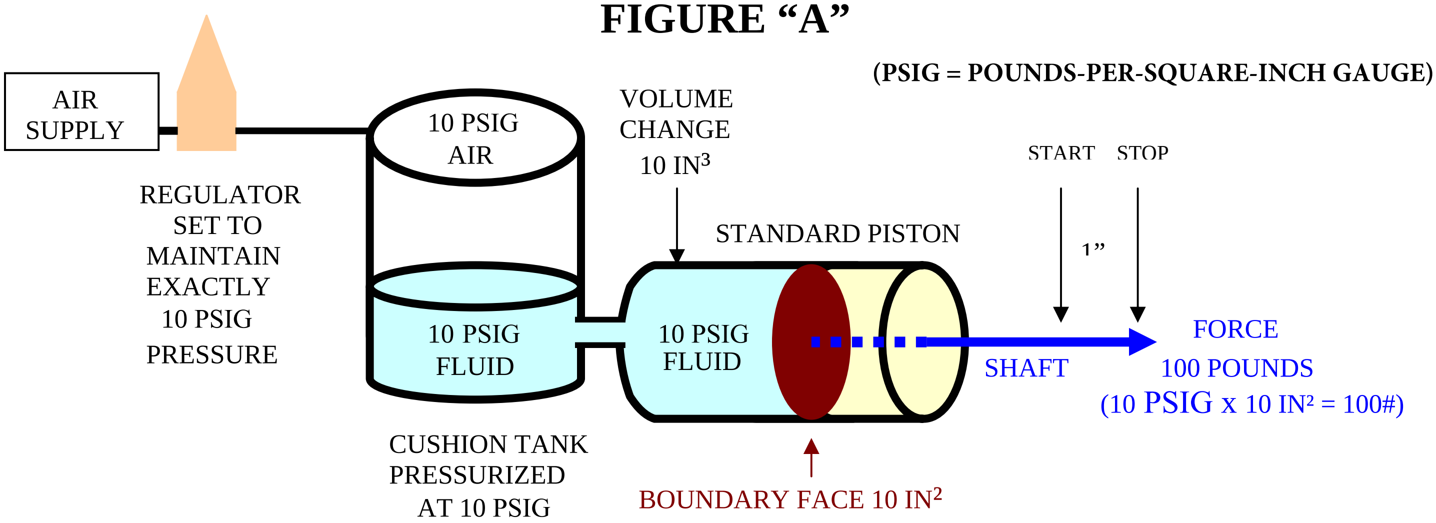


FIGURE "A" illustrates a standard piston with a constant fluid source at 10 PSIG pressure, applied to the piston's 10 in² boundary face, resulting in 100 pounds of force applied to the piston's shaft. The travel distance is one inch, consuming 10 in³ of fluid.

FIGURE "B"

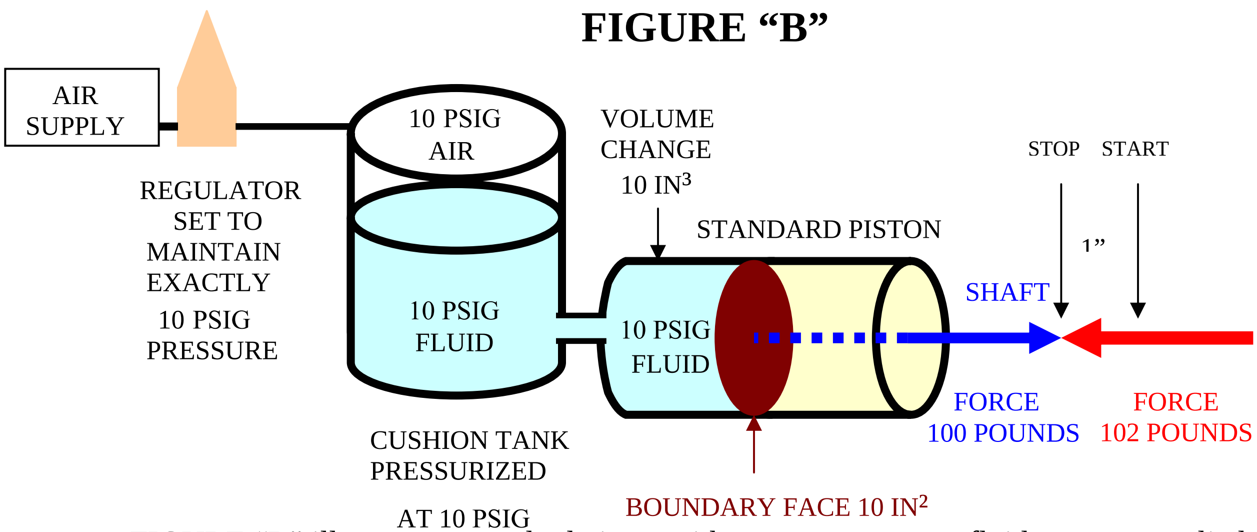


FIGURE "B" illustrates a standard piston with a constant 10 PSIG fluid pressure applied to the piston's 10 in² boundary face, resulting in 100 pounds of force applied to the piston's shaft. A 102-pound force is applied to the shaft, over-powering the 100 pounds of force generated by the pressurized fluid. The shaft travels one inch forcing 10 in³ of pressurized fluid from the standard piston back into the cushion tank at 10 PSIG pressure.

FIGURE "B" illustrates a standard piston employed as a pump producing 10 in³ of fluid at 10 PSIG pressure with an energy source producing 102 pounds of force through a one-inch travel.

FIGURE "C"

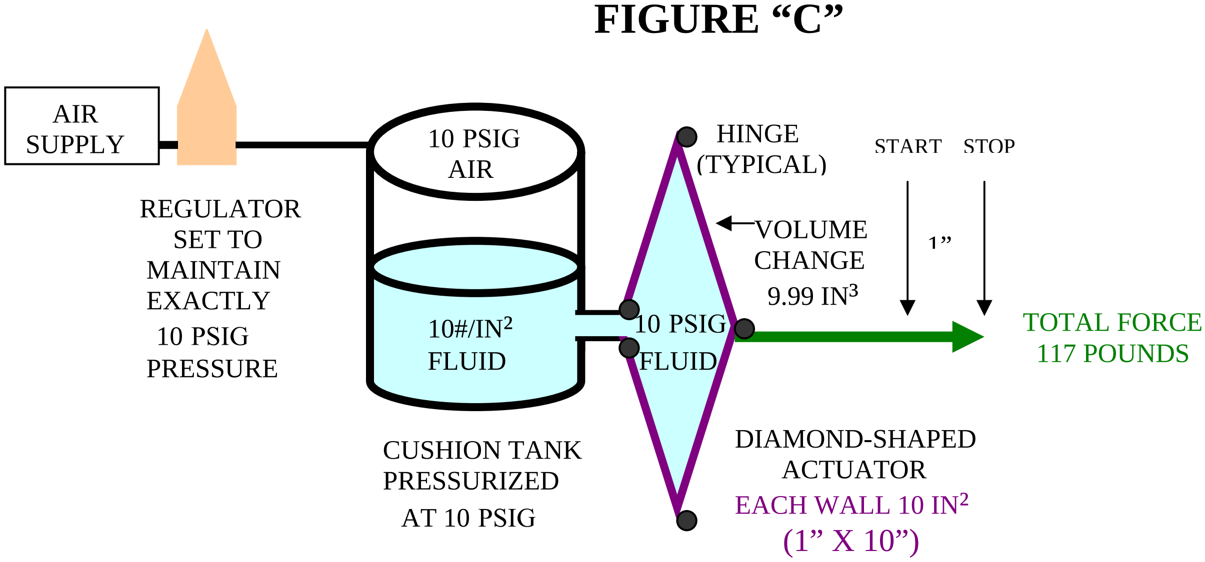


FIGURE "C" illustrates a diamond-shaped actuator with a constant 10 PSIG fluid pressure applied to the actuator's four 10 in², hinged moving walls, resulting in 117 pounds of force at the actuator's forward tip. The travel is one-inch consuming 9.99 in³ of fluid.

NOTES TO CONSIDER:

-1- The diamond-shaped actuator in FIGURE "C" produces 17% more force, with .01% less fluid than the standard piston in FIGURE "A".

-2- The fluid in the standard piston in FIGURE "B" does not care what provides the 102 pounds of counter force on the piston's shaft. If the 102 pounds is applied through one inch, over-powering the 100 pounds of force generated by the fluid, the fluid exits the piston. The 10 in³ of fluid at a pressure of 10 PSIG is pumped from the piston and can be used as a fluid source in another device.

-3- The diamond-shaped actuator does not care where its 117 pounds of force through one inch is applied.

-4- The diamond-shaped actuator may apply 102 of it's 117 pounds of force to pump 10 in³ of fluid from the standard piston.

-5- The diamond-shaped actuator can use 9.99 in³ of the 10 in³ it pumped from the standard piston to provide its own fluid requirements.

-6- After completely satisfying its requirement of pressurized fluid, the diamond-shaped actuator has 15 pounds of force remaining through the one inch travel.

-7- FIGURE "D" illustrates the required arrangement.

FIGURE "D"

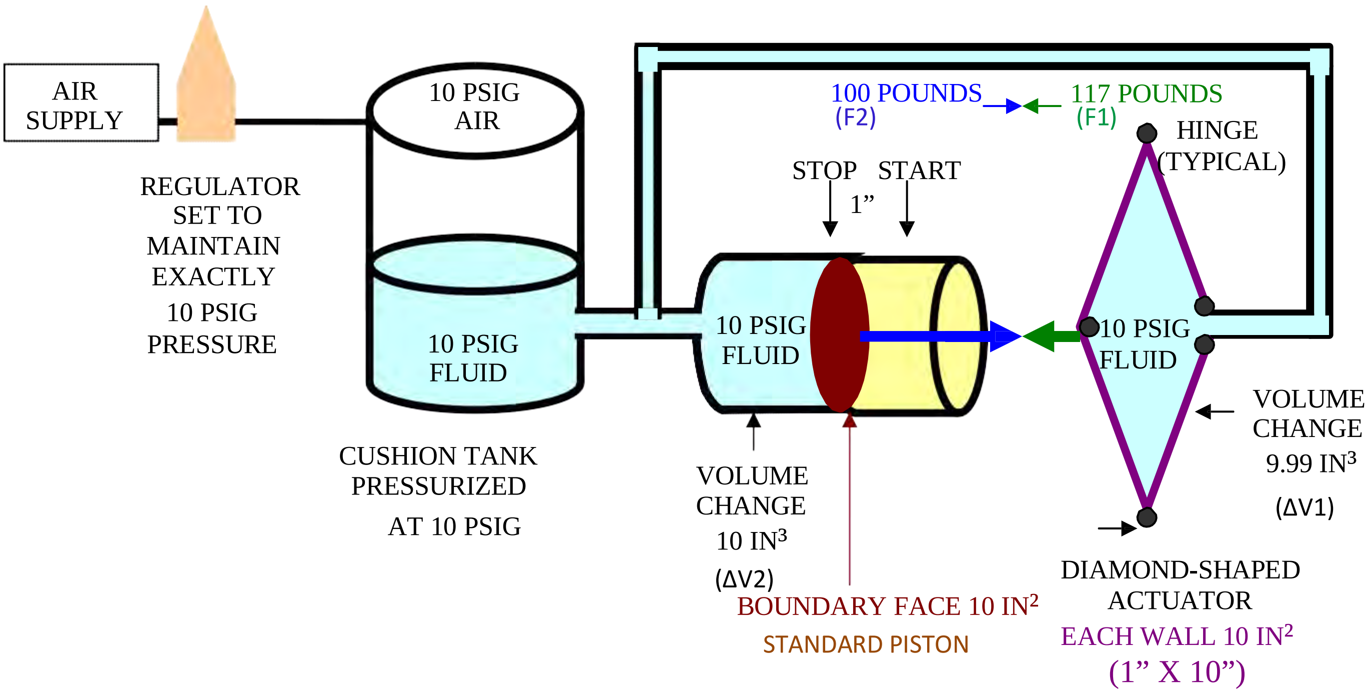


FIGURE 'D" illustrates a standard piston and a Diamond-Shaped actuator piped to a common pressurized fluid source.

The standard piston attains a total force of 100 pounds (F2) while the Diamond-Shaped actuator attains a force of 117 pounds (F1) when they both experience fluid pressure at 10 PSIG through one inch of travel.

The Diamond-Shaped actuator forces 10 in³ of fluid at a pressure of 10 PSIG from the standard piston.

The Diamond-Shaped actuator requires 9.99 in³ of the 10 in³ it forced from the standard piston.

The Diamond-Shaped actuator applies 102 pounds of its 117 pounds of force to the piston's shaft, while simultaneously attaining its required 9.99 in³ of fluid from the piston.

There is an extra 15 pounds of force through the one-inch travel in each stroke that can be extracted for external work.

More detail and a control circuit required for a running machine is presented at http://www.apscontrols.ca, in the patent section.

Work input < Work output

1) Work = Pressure X Volume change (W=PΔV) is an accepted formula in physics.
2) The pressures in the piston and the Diamond-Shaped actuator are equal at 10 PSIG.
3) The work output is the volume change (10 in³) of the piston (ΔV2).
4) The work input is the volume change (9.99 in³) of the Diamond-Shaped actuator (ΔV1).
5) The volume change (ΔV1) in the Diamond-Shaped actuator is less than the volume change (ΔV2) in the piston and the pressures are equal; therefore, applying the formula W = PΔV, the work input is less than the work output. ($W_{in} < W_{out}$)
6) The .01% volume differential is enough to prove $W_{in} < W_{out}$, which is a .01% efficiency advantage of the Diamond-Shaped actuator over the piston.; however, actual testing by several scientists/engineers proves > 17% efficiency advantage of the Diamond-Shaped actuator over the piston proving $W_{in} < W_{out}$ at a greater magnitude, which proves the formula W=PΔV does not apply to the Diamond-Shaped actuator. The formula W=PΔV applies to the piston.

FREE WORK VIA
SURPLUS FORCE DIFFERENTIAL IN HYDRAULICS

The subject invention provides a means to produce energy without producing any pollution. Man has publicly known the basic scientific components since 1874, via US patent 147,519, crediting Mr. Terence F. Reilley. The system is hydraulic.

Man benefits from hydraulic actuators in every day life, by applying pressurized fluid to one side of a hydraulic actuator, developing a force through the stroking range of the actuator's shaft, thus attaining work.
The actuator shaft is returned to the start point by removing the pressurized fluid from the one side of the actuator, plus one of two approaches:

(1) Pressurized fluid is applied to the actuator causing force in the opposite direction forcing the shaft back to the start point.

OR,

(2) A return spring is loaded while the shaft is driving and unloads, forcing the shaft back to its start point.

All hydraulic systems are designed based on a force (F1) applied in the direction of the desired work action, counter force (F2) in the opposite direction and frictional force (F3). The net force (F4) of any hydraulic actuator is F1 - F2 - F3 = F4.

Lets consider the relationship between the volume of pressurized fluid and the hydraulic actuator. Imagine a volume of fluid forced into an actuator producing one hundred pounds of force, through the actuator's stroking range. How much force would you have to apply to the shaft to push it back to the start point, if the fluid's supply pressure remained constant? Physical laws maintain that one hundred pounds of force (ignoring friction at this point) would only achieve equilibrium; however, one hundred and one pounds of constant force would push the shaft back to the start point.

What just happened? You displaced a volume of pressurized fluid identical to that required by the hydraulic actuator to produce its hundred pounds of force with your one hundred and one pounds of force**. With only 1% more force than the hydraulic actuator develops you can produce that actuator's required fluid supply.**

Are we getting to a point where a system could provide its own source of pressurized fluid if one actuator pushed fluid from another?

What do you think we need?

(1) Imagine that we have a hydraulic actuator that is 20% more efficient than conventional hydraulic actuators. This means the new actuator produces one hundred and twenty pounds of force through the same stroking range of the shaft, rather than the one hundred pounds attained by the conventional actuator in the first consideration.

(2) Imagine that the new, more efficient hydraulic actuator could relieve you of the burden suffered in pushing with one hundred and one pounds of force to displace the volume of pressurized fluid in the previous example.

(3) Imagine that the more efficient actuator actually required less fluid than the conventional actuator attaining identical stroke.

If we had these three points what can be accomplished?

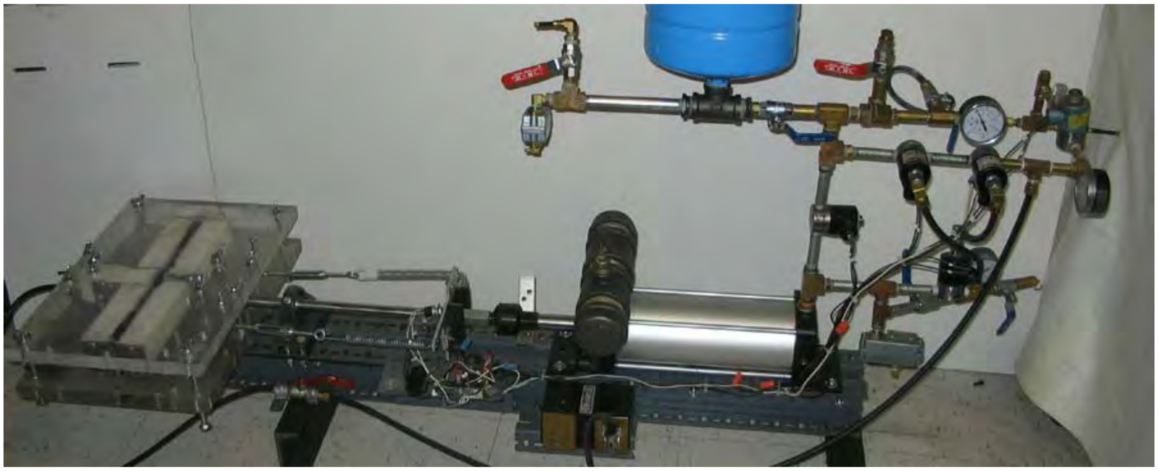
If we had all three points we could develop a machine that provides its own source of pressurized fluid while stroking through its shaft's range. Part of the 20% surplus force, created by the 20% greater efficiency, would be required over coming friction, causing motion and resetting the machine for the next reciprocating cycle. **Here's the kicker! The remaining surplus force, through the stroking range, can be extracted from the system producing work external to the system.**

Well…, Reilley in 1874 (US patent #147,519), Powers in 1886 (US patent #345,446), Sleeper in 1901 (US patent # 696,768) and Strain in 2004 (US patent #6,782,800 and European patent #1240435) were all granted patents defining actuators achieving exactly these three points.

NET RESULT = WORK OUT > WORK IN

DIAMOND-SHAPED FLUID POWERED LINKAGE, SYSTEM AND ENGINE

TEST MODEL PROVING OPERATION OF THE CONTROL STRATEGY.



USA PATENT # 6,782,800

EUROPEAN PATENT # 1240435

AUSTRIAN PATENT # AT E 280 331 T1

GERMAN PATENT # DE 600 15 181 T2 2005.11.24

UK, IRISH, SWISS, BELGIAN, ITALIAN AND FRENCH

PATENT # 1240435

CANADIAN PATENT # CA 2 424 712

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CONTENTS

TEST MODEL
PROVING FUNDAMENTAL SCIENCE

EXECUTIVE SUMMARY

The major puzzle pieces, regarding a Global Warming solution, have been registered patents in the US for over a century. The missing component, allowing the creation of a completely clean energy source, was a control logic strategy. A Canadian developed the required control logic from 1971 to 1999, obtaining patents in several countries. We now have at least one tool to assist in the battle against the Global Warming threat.

Three century-old patents, registered in the US Patent Office, presented the design of hydraulic actuators that are more efficient than actuators currently used in modern society. These inventors understood that their actuators achieve more work than conventional actuators when allowed identical fluid inputs.

The three patents are:

- US Patent #147,519 in the year 1874.
- US Patent #345,446 in the year 1886.
- US Patent #696,768 in the year 1901.

A new development, conceived in 1971, was first filed as a patent application in 1999. This application made the step of fitting the century old knowledge into a practical solution regarding a clean energy source. The inventor was awarded patents in the USA, UK, Irish, Belgian, German, Austrian, French, Italian, Swiss and Canadian patent offices.

The current patents pit the work potential of the conventional actuator against the work potential of the Diamond-Shaped Actuator. The Diamond-Shaped Actuator won, as it produces more force through identical travel. As the diamond-shaped actuator pushed the conventional actuator, the volume of fluid required by the Diamond-Shaped Actuator was forced from the conventional actuator. After the Diamond-Shaped actuator produced its own source of fluid from the conventional actuator, there was surplus work potential in the Diamond-Shaped actuator's stroke.

The fundamental science was known about 150 years ago. The new developments provided a mechanical/control system, creating a machine that runs itself, while allowing extraction of the work differential.

This invention is intended to add to the total effort addressing the Global Warming Threat.

Unfortunately, during the 1800's, early inventors were very close; however, did not see this functional application. If they had, the world would likely be much cleaner now.

REPORT GUIDE

-1- What is the topic of this report?
A solution to global warming. Page 13.244 gives some history and an overview.
-2- Has this science been reviewed by scientists?
Many scientists and Engineers have assessed the invention finding nothing incorrect with the claims. It has granted patents in Europe and North America.

-3- What is a conventional actuator?
Page 13.245 and 13.246 illustrate a conventional actuator (hydraulic piston).

-4- What is a diamond-shaped actuator?
Pages 13.247 and 13.248 illustrate a diamond-shaped actuator.

-5- What were the patented actuators from 1874, 1886 and 1901? Page 13.249 illustrates the concept of these actuators.

-6- What is meant by volume displacement regarding actuators? Page 13.250 illustrates this concept.

-7-Which type of actuator has greater volume change through identical travel?
Page 13.251 illustrates this point.

-8- Which type of actuator produces more power through identical travel? Page 13.252 illustrates this point.

-9- How does the diamond-shaped actuator provide its own source of pressurized fluid?
Page 13.253 illustrates this point.

-10- How does surplus power develop when using a diamond-shaped actuator with a conventional actuator, but not with two conventional actuators?
Page 13.254 illustrates this point.

-11- How did these century old facts allow development of a machine to generate energy?
Pages 13.255, 13.256 and 13.257 illustrate the machine's flow patterns and circuitry.

-12- Can the machine's speed and power output be controlled?
Page 13.258 is a graph illustrating the relationship of system static pressure to power output and speed.

DIAMOND-SHAPED FLUID POWERED LINKAGE, SYSTEM AND ENGINE

The Diamond-shaped Fluid Powered Linkage, System and Engine patent application was first filed in 1999 in the USA as a provisional patent application. The patent process took it through the Patent Co-operation Treaty (PCT**),** via the European Patent Office, where it gained general international approval. The next steps gained patent approval in the USA under patent number 6,782,800 and approval in Europe under the European patent number 1240435. Patents were granted in Ireland, the UK, Belgium, France, Switzerland and Italy using the European patent number 1240435 as reference in those countries. The patent was granted in Austria under patent number AT E 280 331 T1 and in Germany under patent number DE 600 15 181 T2 2005.11.24. The Canadian Patent Office assigned number CA 2 424 712 to the Canadian patent.

The operation of the Diamond-Shaped Fluid Powered Linkage, System and Engine challenges one of the most accepted concepts in the human scientific knowledge base. "Energy can not be created or destroyed." The fathers of the "laws", declaring the physical results of this invention impossible, did not have the benefit of our current materials and precise test equipment that allowed a more in depth understanding, regarding their claims.

The holder of these patents does not believe that energy is truly understood by our collective human knowledge. The definitions witnessed define energy by what it can do, but do not state what it is.

Declaring anything "impossible", with the knowledge we have accumulated to date, clearly states that for all eternity humanity will not surpass our current level of practical knowledge regarding the matter at hand. That is a lot of confidence, if not arrogance. We have to truly keep open minds if we hope to move forward effectively.

I request that you think your way through this assessment, rather than react based on your belief system. Every point is scientifically supportable, challenge it with logical thinking.

The primary concepts regarding the invention are:

-1- If two unequal forces oppose one another, the stronger force will over-power the weaker force until equilibrium is reached.
-2- If a hydraulic actuator is full of fluid causing its shaft to be extended, the fluid can be forced back out of the actuator by applying force to the shaft greater than the counter force applied by the fluid that fills the actuator.
-3- Conventional cylindrical actuators, currently employed in hydraulics, are less efficient than other patented actuators. The alternate actuators were patented by Terence F. Reilley in 1874 under US patent number 147,519, Titus Powers in 1886 under US patent number 345,446, Frank H. Sleeper in 1901 under US patent number 696,768 and myself, David Strain, in 1999 under the patent numbers outlined in the first paragraph on this page.

This paper walks you through the century old fundamental science and illustrates a control circuit regarding one potential Global Warming Solution.

CONVENTIONAL ACTUATOR

The shaft is extended on the actuator in this picture. The two fluid ports are visible on the two black ends of the actuator.

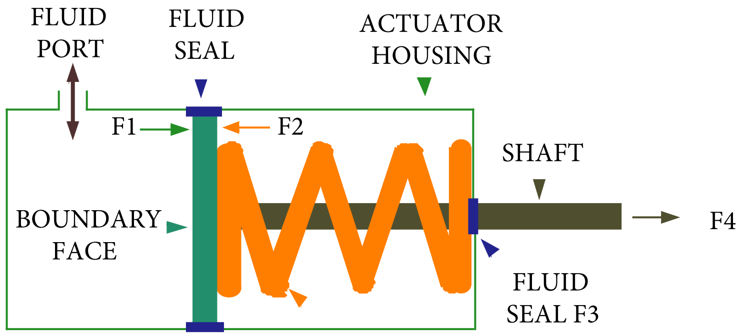
CONVENTIONAL ACTUATOR
DISASSEMBLED

The shaft and boundary face assembly (1) is at the front of the photograph.
The black seals (2) around the boundary faceplate that separate the two sides of the actuator are visible.
The black end caps (3) and two of the four assembly rods (4) are present.
The body of the actuator (5) is the hollow tube where the boundary faceplate slides back and forth.

ACTUATOR WITH SPRING RETURN

Fluid actuators produce work via the movement of their shafts caused by fluid exerting pressure on the actuator's boundary face.

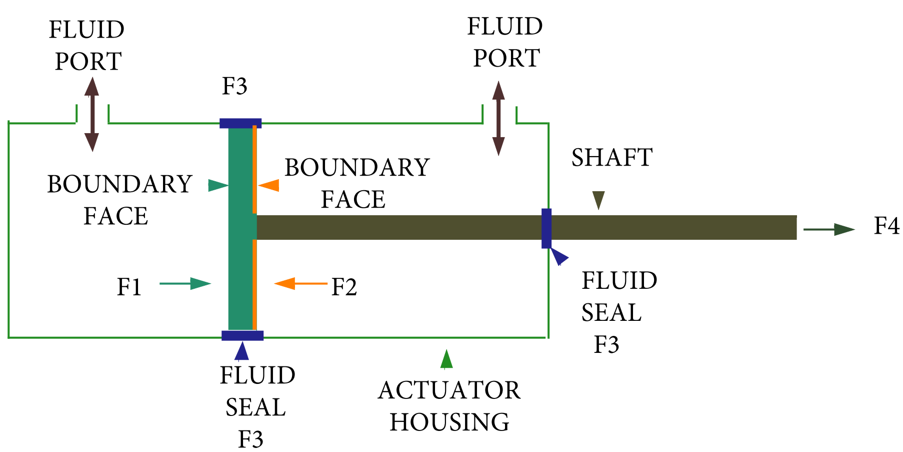
The potential work available at any point in the travel is the pressure of the fluid multiplied by the area of the boundary face, minus the counter force of the return spring, minus the frictional resistance of the actuator.



FORCE ONE (F1) = FLUID PRESSURE X THE BOUNDARY FACE AREA
FORCE TWO (F2) = SPRING PRESSURE RETURNING TO REST POSITION
FORCE THREE (F3) = FRICTIONAL RESISTANCE OF PISTON
NET FORCE (F4) = F1 - F2 - F3

ACTUATOR WITH NO SPRING RETURN

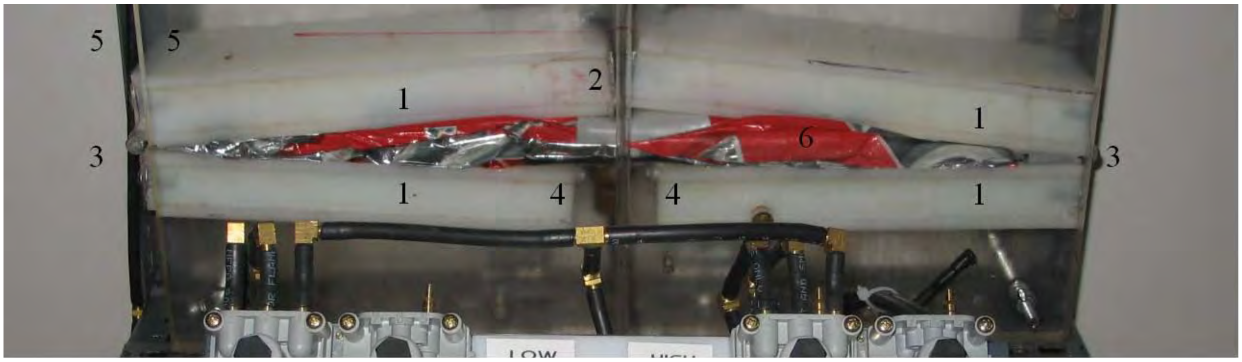
Double drive actuators do not have spring return. The work potential is based on the pressures of the fluid applied to the two opposing boundary faces generating forces (F1) and (F2), as well as the frictional resistance of the actuator (F3).



FORCE (F1) = FLUID PRESSURE X THE AREA OF THE BOUNDARY FACE
FORCE (F2) = FLIUD PRESSURE X THE AREA OF THE BOUNDARY FACE
FORCE (F3) = FRICTIONAL RESISTANCE OF THE ACTUATOR
NET FORCE (F4) = F1 - F2 - F3

DIAMOND-SHAPED ACTUATOR IN
NON-DRIVEN POSITION

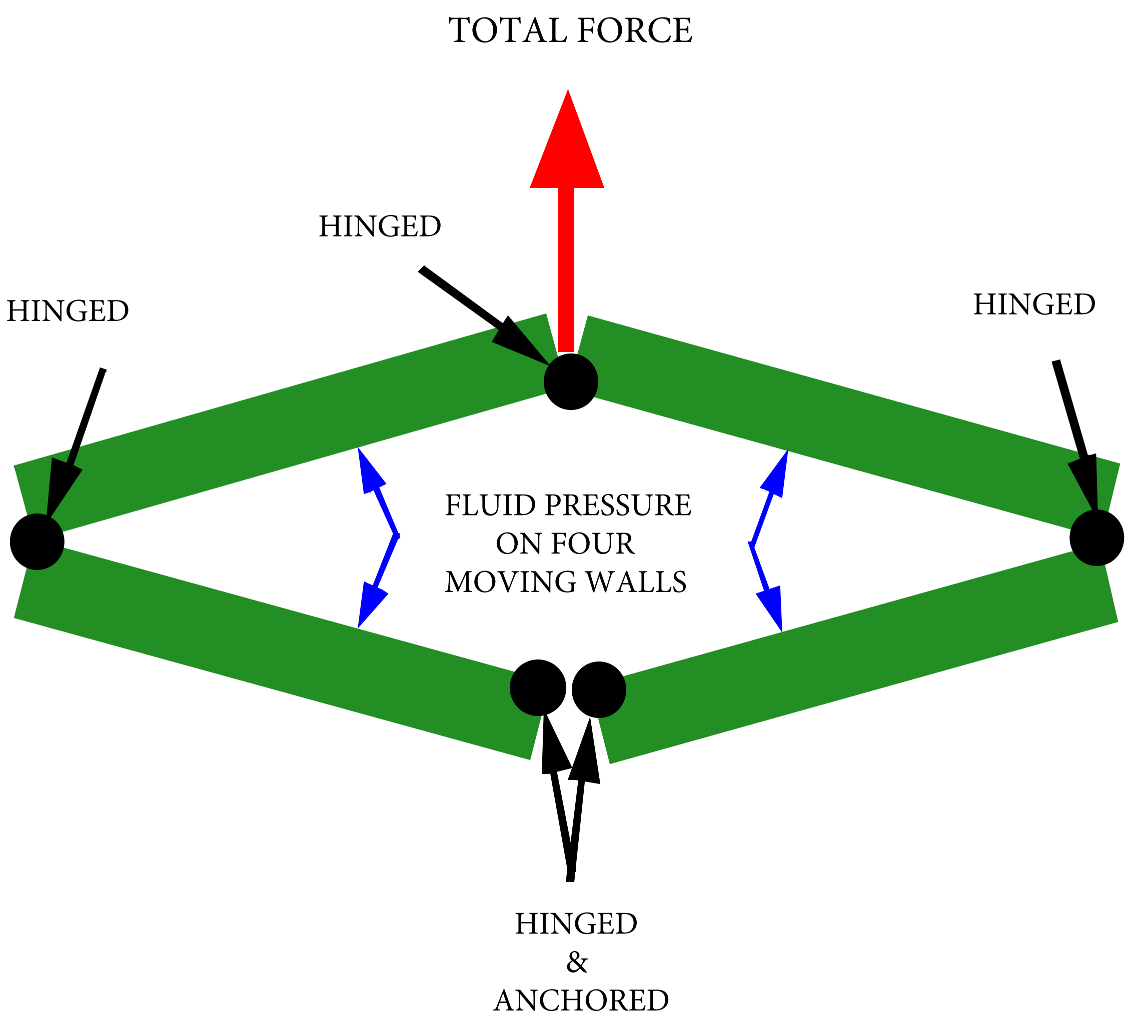
This photo shows our test model of the diamond-shaped actuator.
The four white walls (1) are hinged at the upper connection (2) and the two outside connections (3).
The lower white walls are hinged separately and anchored at the lower mid point (4).
The walls are enclosed by two large pieces of plexiglass (5) allowing the four walls to move inside.
The coloured diaphragm (6) contains the fluid, preventing leaks.



DIAMOND-SHAPED ACTUATOR IN
DRIVEN POSITION

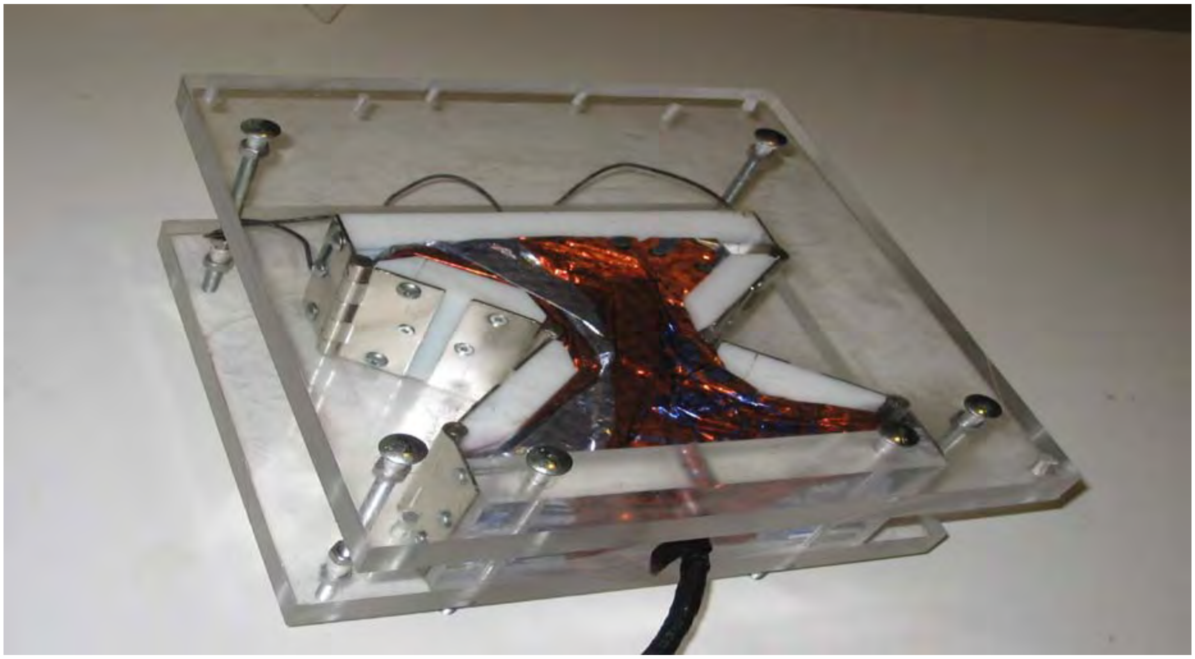
This photo shows the diamond-shaped actuator driven.
The force created by the pressure applied to the four moving walls is totaled at the upper hinge.

DIAMOND-SHAPED ACTUATOR DESIGN CREDITED IN 2002 & 2004
USA PATENT AND PATENT CO-OPERATION TREATY



ACTUATOR CONCEPTS FROM 1873 AND 1886

Mr. Terence F. Reilley patented the actuator concept presented in the photograph in 1873. Mr. Titus Powers's patent of 1886 employed the same basic concept. Unaware of the earlier inventions, I developed the same actuator between 1971 and 1999, but was informed in 2002, by the US patent office, that I was only late by 127 years, as they referenced Mr. Reilley's and Mr. Power's patents.



ACTUATOR CONCEPT FROM 1901

Mr. Frank H. Sleeper patented the actuator concept in the photograph below in 1901. I developed the actuator in the photograph below between 1971 and 1999. I used linear motion bearings and rails where Mr. Sleeper used a different mechanism allowing for the lengthening of the walls on the forward drive. In 2002 the US patent office made me first aware of Mr. Sleeper's patent. I was a little more up to date with this one, only ninety-seven years too late.

VOLUME DISPLACEMENT

Start with a conventional actuator as in "FIGURE 1".

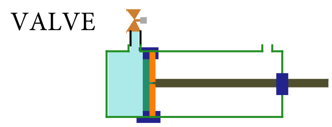


FIGURE 1

Connect a pressurized fluid source to the valve attached to the fluid port of the actuator in "FIGURE 2". Open the valve to fill the actuator and close the valve when a 20 in^3 volume change occurs. The shaft drives out exerting a force dependent on the pressure of the fluid.

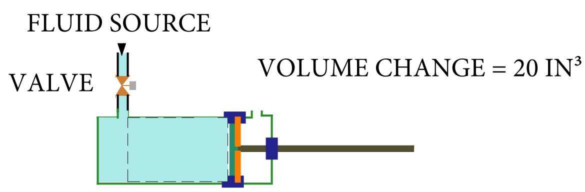


FIGURE 2

Disconnect the fluid supply and connect a balloon to the valve attached to the fluid port of the actuator as in "FIGURE 3". Re-open the valve.

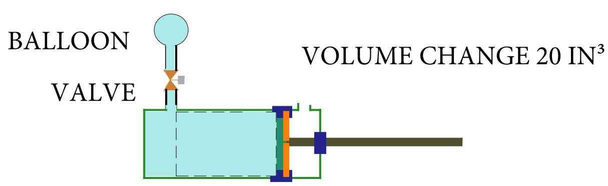


FIGURE 3

Push on the shaft, forcing it back to its position before you first added the fluid supply as in "FIGURE 4".

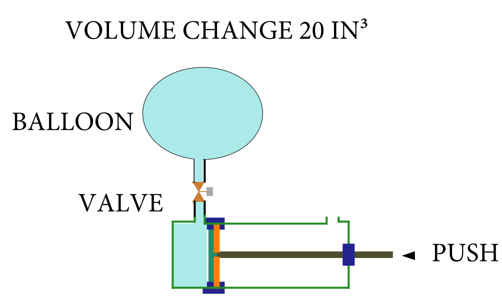


FIGURE 4

Identical volume change occurs, whether filling an actuator or emptying an actuator, for equal linear travel.

VOLUME CHANGE COMPARISON

A conventional actuator and a diamond-shaped actuator are arranged as per "FIGURE 5". The conventional actuator's boundary face area is the same as the area of each of the four equal sides of the diamond-shaped actuator. The valve is closed.

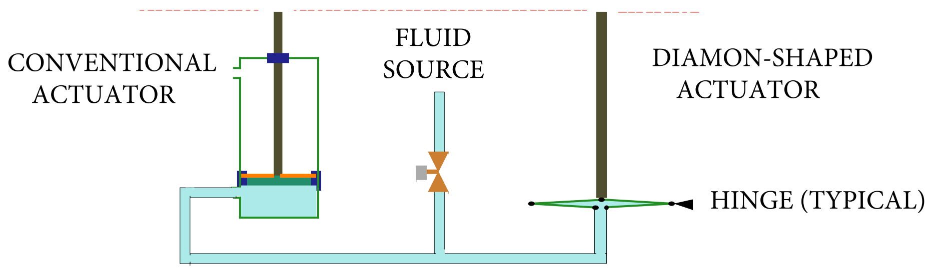


FIGURE 5

The valve is opened causing both the actuators to drive equal distances as per "FIGURE 6".
The relationship is linear regarding the volume change relative to travel for the conventional actuator.
The relationship is non-linear regarding the volume change relative to travel for the diamond-shaped actuator.
For any change in linear travel the diamond-shaped actuator requires less fluid than the conventional actuator achieving identical travels.

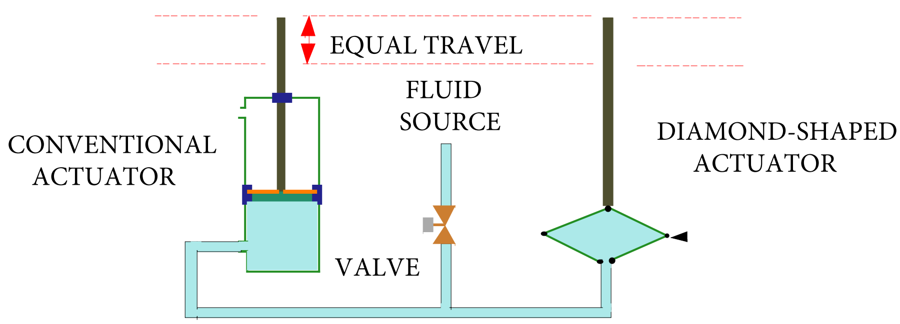


FIGURE 6

POWER COMPARISON

A conventional actuator and a diamond-shaped actuator are arranged as per "FIGURE 7". The conventional actuator's boundary face area is the same as the area of each of the four equal sides of the diamond-shaped actuator. The valve is closed.

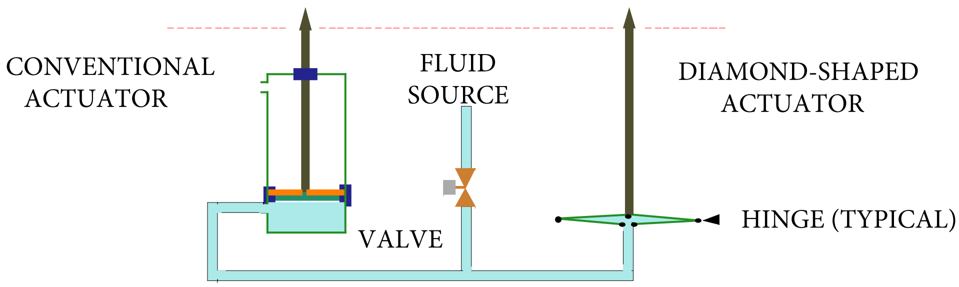


FIGURE 7

Open the valve, driving both actuators with the common source of pressurized fluid as per "FIGURE 8".
The driving force on the conventional actuator is the result of the fluid pressure on the one moving boundary face times the boundary face area.
The driving force at the tip of the diamond-shaped actuator is the effect of the combined force on the four moving walls of the actuator.

We determined the equilibrium point for our diamond-shaped model. This is the point the actuator is able to suspend a weight with neither rising nor lowering. The best our model has demonstrated in actual testing is a 26% advantage over a conventional actuator.

NOTE: The diamond-shaped actuator requires less fluid achieving the same degree of lift with 26% more load.

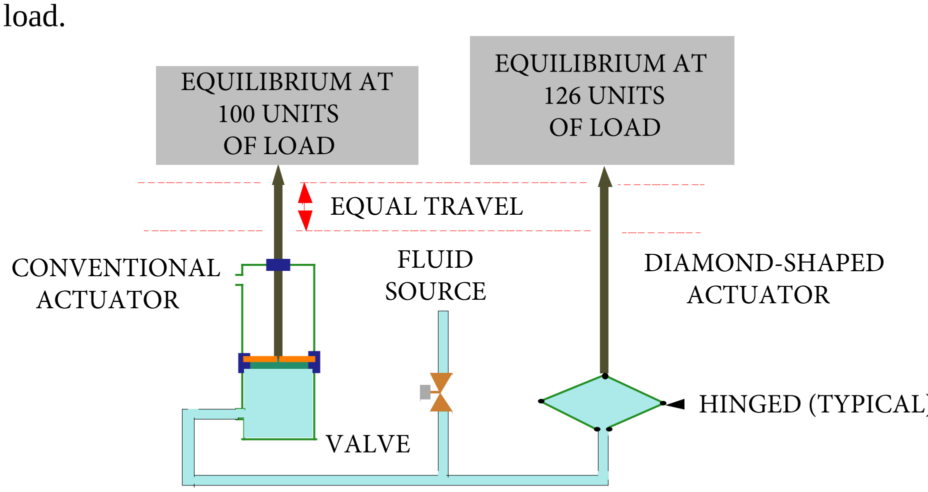


FIGURE 8

POWER/VOLUME RELATIONSHIP

A conventional actuator and a diamond-shaped actuator are arranged as per "FIGURE 9".
The conventional actuator's boundary face area is the same as the area of each of the four equal sides of the diamond-shaped actuator. The valve is closed.

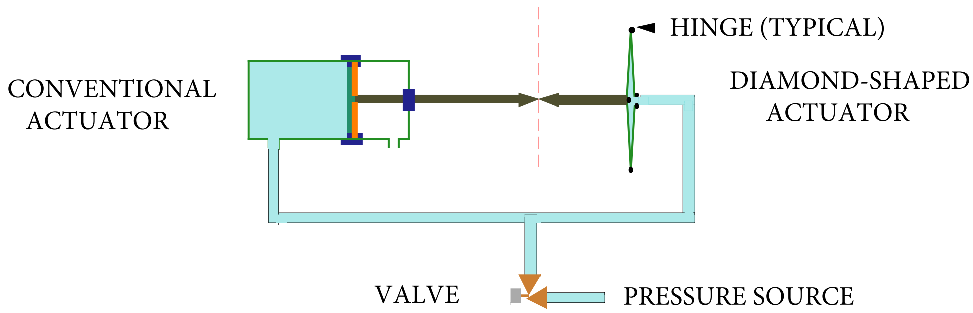


FIGURE 9

Open the valve, exposing both actuators to the common pressure source, as per "FIGURE 10".
The driving force of the conventional actuator is the result of the fluid pressure on the one moving boundary face times the boundary face area.
The driving force at the tip of the diamond-shaped actuator is the effect of the combined force on the four moving walls of the actuator.
The diamond-shaped actuator develops more force than the conventional actuator; therefore, drives forward, forcing fluid from the conventional actuator.
The volume of fluid forced from the conventional actuator is slightly more than the diamond-shaped actuator requires achieving this action.
The diamond-shaped actuator displaces a greater volume of fluid from the conventional actuator than the volume of fluid it requires for the same length of stroke, causing a slight flow of fluid toward the pressure source.
The diamond-shaped actuator sustains a surplus of work potential after it has produced its own source of pressurized fluid for the action.

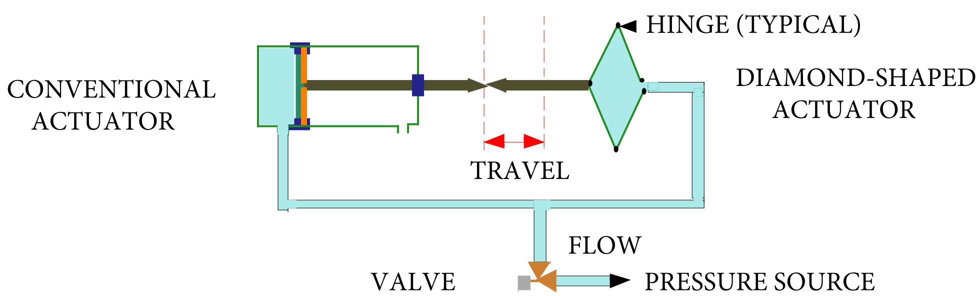
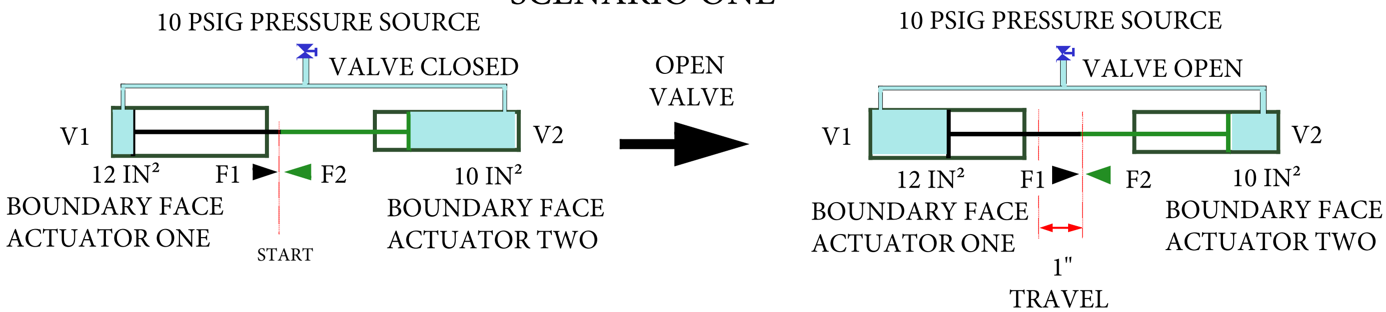


FIGURE 10

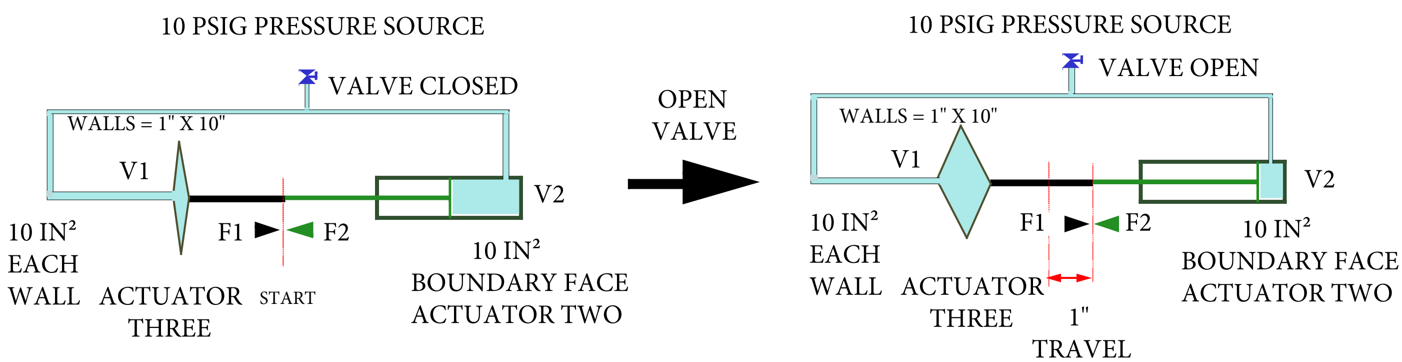
ACTUATOR COMPARISON

SCENARIO ONE



VOLUME ONE (V1) CHANGE = 12 IN^3
VOLUME TWO (V2) CHANGE = 10 IN^3
FORCE ONE (F1) = 120 POUNDS
FORCE TWO (F2) = 100 POUNDS
TRAVEL IS EQUAL FOR BOTH ACTUATORS.

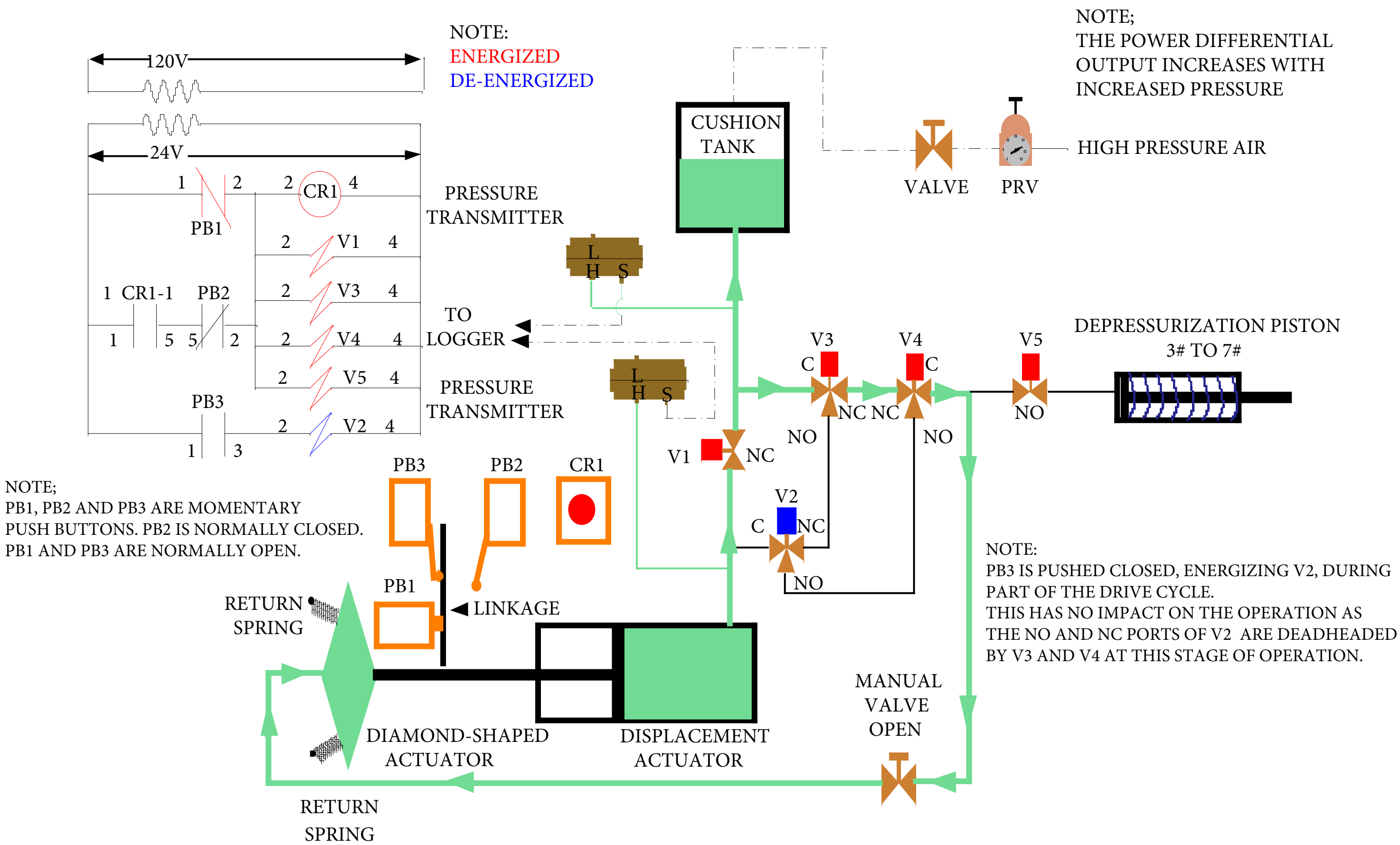
SCENARIO TWO



VOLUME ONE (V1) CHANGE = 9.99IN^3
VOLUME TWO (V2) CHANGE = 10 IN^3
FORCE ONE (F1) = 120 POUNDS
FORCE TWO (F2) = 100 POUNDS
TRAVEL IS EQUAL FOR BOTH ACTUATORS.

COMPARISON

-1- Actuators one and three displace 10 in^3 of fluid from actuator two during these actions.
-2- Actuators one and three each exert a 120 pounds of force against the 100-pound counter force of actuator two.
-3- Actuator one requires **20% more fluid** than the volume displaced from actuator two during this action.
-4- Actuator three requires **.01% less fluid** than the volume displaced from actuator two during this action.
-5- Actuator one can not satisfy its fluid volume requirement with the displaced fluid of actuator two.
-6- Actuator three can satisfy its total fluid volume requirement with the displaced fluid of actuator two.
-7- Part of actuator three's 20% surplus power must overcome the system's power requirements and frictional losses while running itself.
-8- The remaining power can generate external mechanical work.



DRIVING STATE

The driving state is started when the linkage forces push button (PB1) closed to energize V1, V3, V4, V5 and CR1. The contact of CR1 closes to lock power on these devices when PB1 re-opens as the drive movement starts.

When V1, V3, V4, and V5 are energized the flow pattern is as illustrated on this drawing.

The diamond shaped actuator, the displacement cylinder and the cushion tank experience the same pressure.

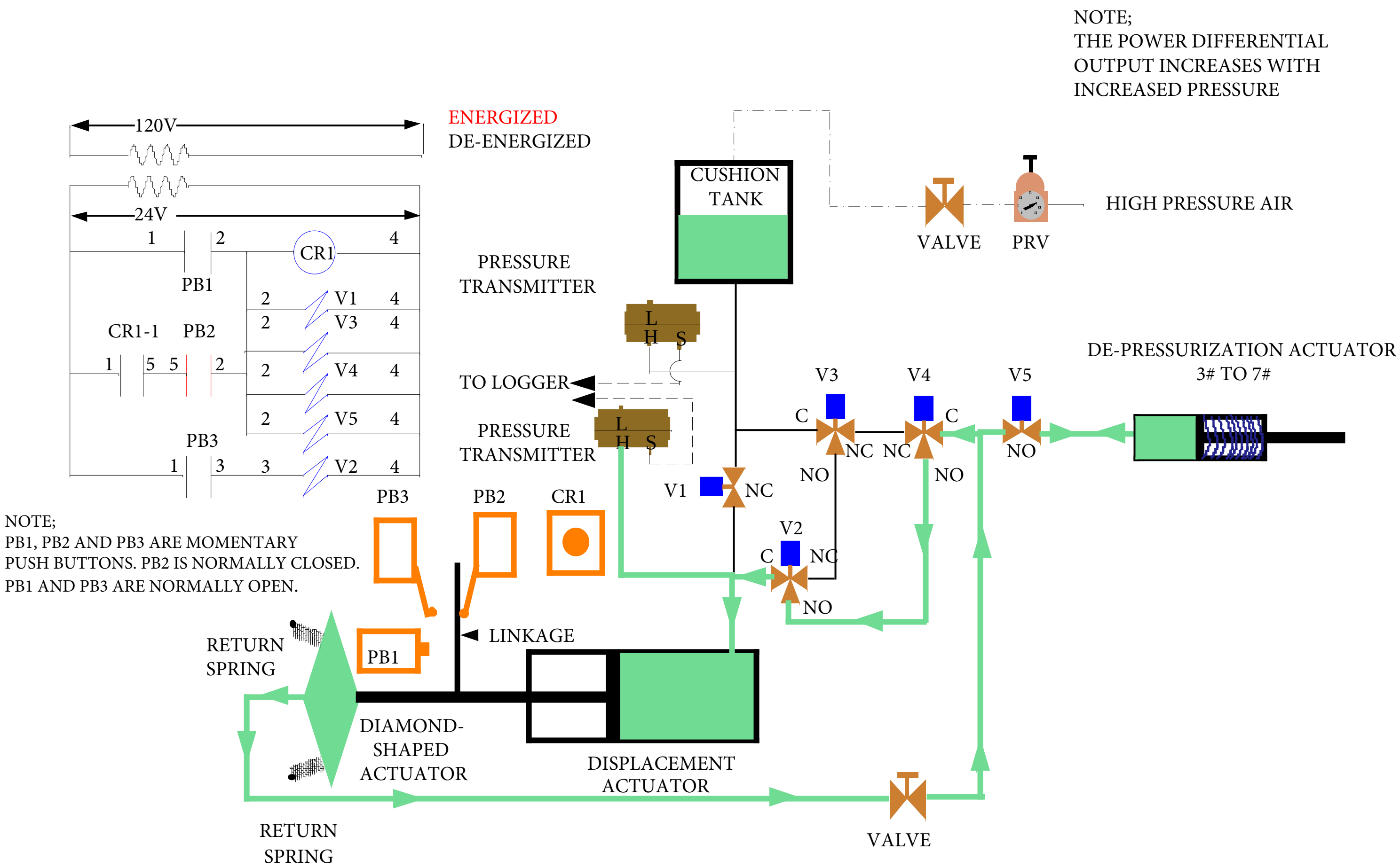
The diamond shaped actuator develops about 10% to 15% greater total force at its tip than the counter force developed in the displacement cylinder.

Each of the four faces of the diamond shaped actuator have the same area as the displacement cylinder's boundary face. This causes more fluid to be driven out of the displacement cylinder than is required to drive the diamond shaped actuator for the same linear travel.

About 99% of the fluid driven out of the displacement cylinder flows to the diamond shaped actuator and about 1% of the fluid flows into the cushion tank.

The differential in force between the diamond shaped actuator and the displacement cylinder may be used to drive any mechanical device external to this machine, such as a generator or pump.

When the linkage forces push button (PB2) open V1, V3, V4, V5 and CR1 are de-energized which causes the driving state to stop.



FIRST RECHARGE CYCLE

The linkage pushes PB2 open, which causes V1, V3, V4, V5 and CR1 to de-energize and the flow pattern illustrated on this drawing is established.

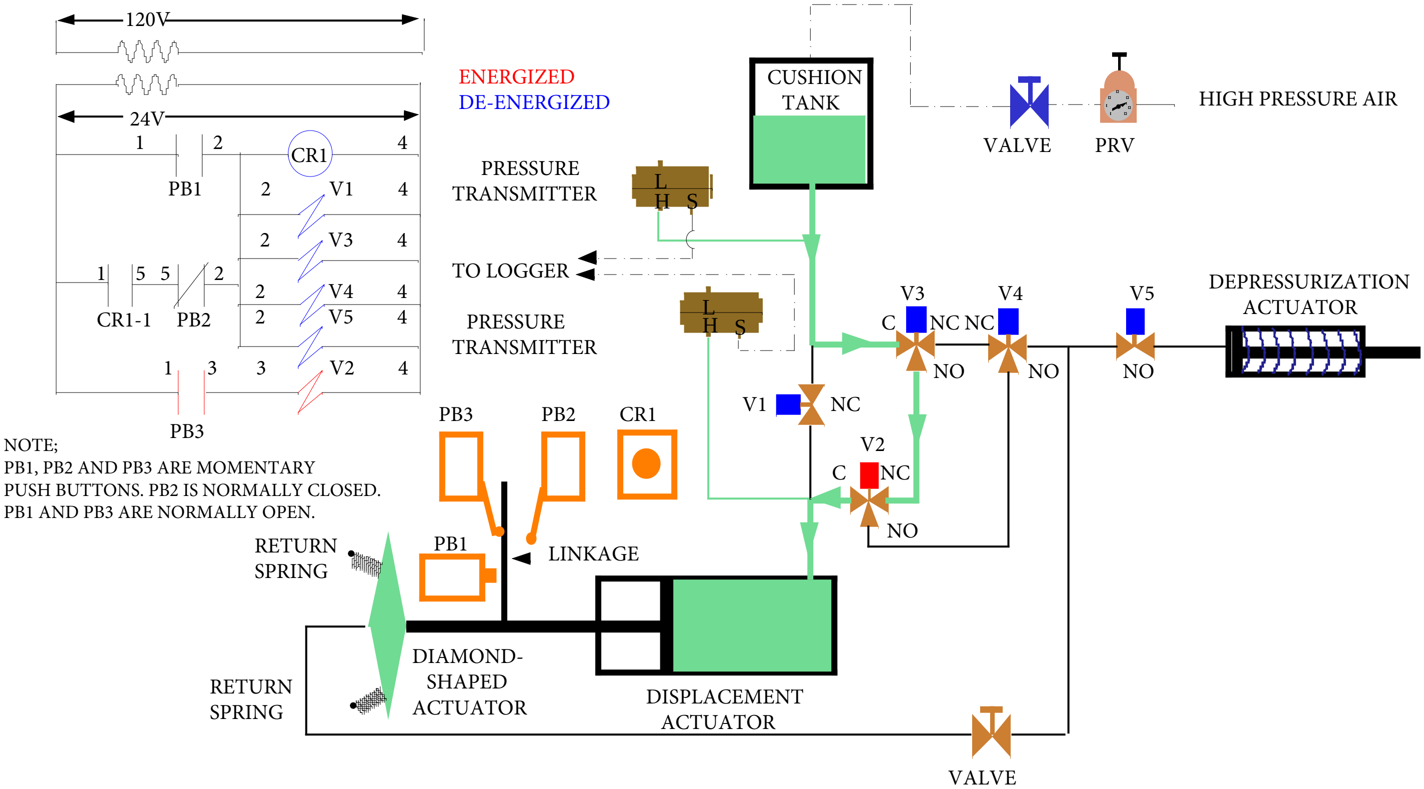
The pressurized cushion tank is isolated.

The diamond shaped actuator, the displacement cylinder and the depressurization cylinder all experience a common pressure.

The depressurization cylinder strokes, which de-pressurizes all three components. This removes the diamond shaped actuator's power advantage over the return springs. The return springs retract to force the fluid in the diamond shaped actuator back into the displacement cylinder.

When the common pressure drops below the spring range of the depressurization cylinder, the fluid is forced into the displacement cylinder from the depressurization cylinder.

When the 99% of the fluid that came out of the displacement cylinder has been returned to the displacement cylinder, the linkage forces push button (PB3) to close which energizes V2.



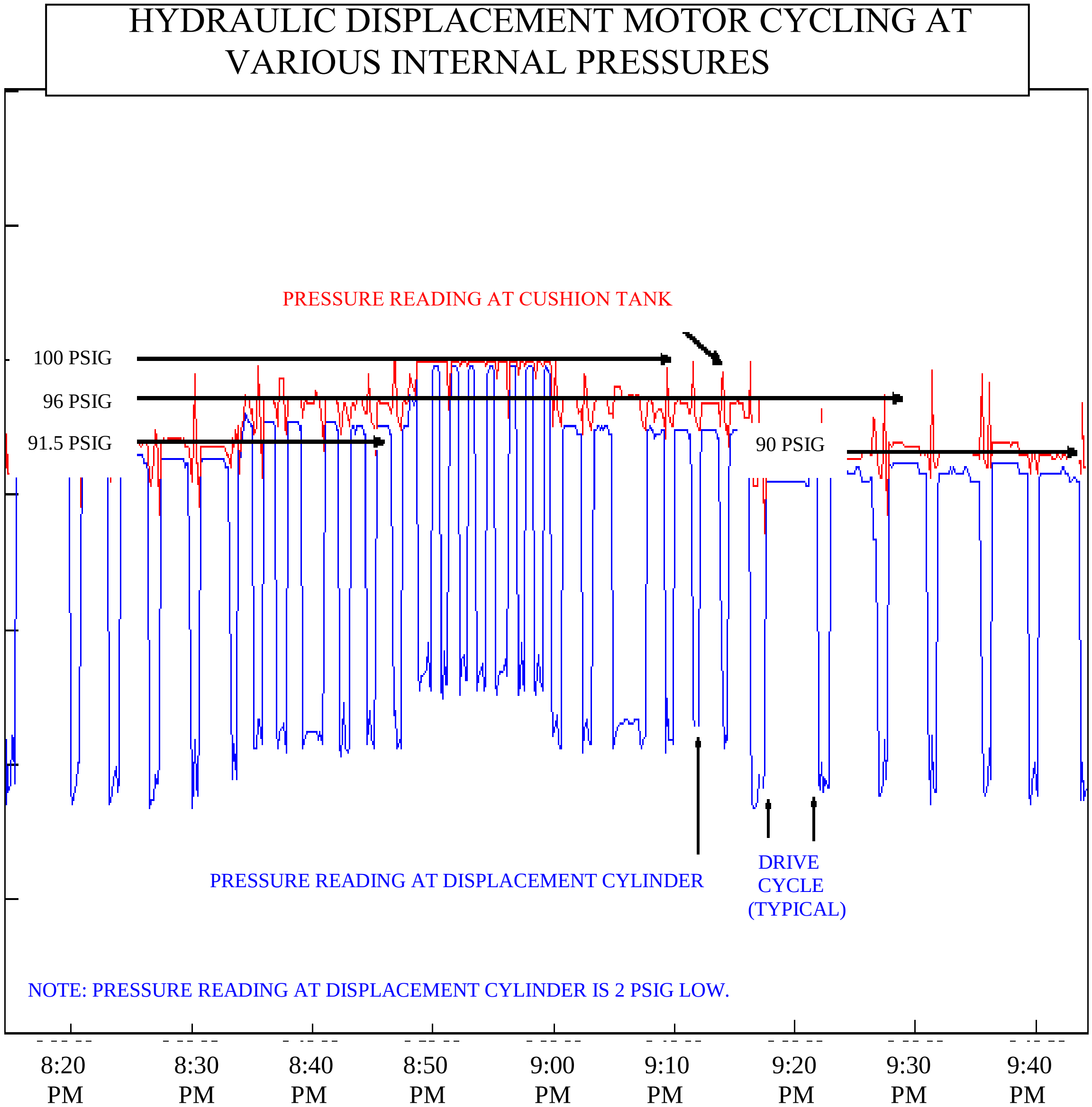
SECOND RECHARGE CYCLE

The linkage pushes PB3 closed to energize V2. This establishes the flow pattern illustrated on this drawing.

The 1% of the fluid that was forced from the displacement cylinder into the cushion tank is forced back into the displacement cylinder from the cushion tank.

When the displacement cylinder is completely refilled, the linkage of the displacement cylinder pushes PB1 closed and the cycle repeats.

NOTE: The linkage is not mechanically connected to the diamond shaped actuator; therefore, there will be a temporary gap between them during the second recharging stage.



Data was collected at transmitter points illustrated on the previous drawings on pages 13.255, 13.256 and 13.257.
Data extracted from motor as per exact layout of drawing 12 in patent application document.

TEST RESULTS

REGARDING

EFFICIENCY DIFFERENTIAL

COMPARING

CYLINDRICAL ACTUATORS

TO

DIAMOND-SHAPED ACTUATORS

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REPORT PURPOSE

This report's focus is the fundamental science regarding an invention patented in Europe, Canada and the USA.

The concept under consideration is when a diamond-shaped actuator and a conventional cylindrical shaped actuator oppose one another, with a common fluid source, there are two results observed:

(1) The diamond-shaped actuator displaces a volume of fluid from the cylindrical actuator. The displaced volume is greater than the volume added to the diamond-shaped actuator.

(2) The diamond-shaped actuator maintains **>**11% work potential, in each stroke, after attaining its own source of fluid from the cylindrical actuator.

TABLE OF CONTENTS

Note that the differential in the model illustrated on page four is required to over-come the inertia of two cylindrical actuators. The >15% advantage witnessed by scientist/engineers was based on the diamond-shaped actuator causing motion; therefore, had over-come its inertia and the differential is required to over-come the inertia of only one cylindrical actuator.

SURPLUS WORK DEMONSTRATION MODEL

TWO TYPES OF ACTUATORS

The diamond-shaped actuator shown in FIGURE 2 has been tested by several scientist/engineers, all observing an efficiency advantage >15% over the cylindrical actuator shown in FIGURE 1. (Pages six to twelve contain scientific observations.)

The advantage is gained via the pressure applied on the four moving walls of the diamond-shaped actuator compared to the single moving boundary face of the cylindrical actuator. The diamond-shaped actuator uses slightly less fluid, at the same pressure, as the comparable cylindrical actuator, accomplishing >15% more work.

FIGURE 1
CYLINDRICAL ACTUATOR

FIGURE 2
DIAMOND-SHAPED ACTUATOR

POWER RELATIONSHIP OF OPPOSING ACTUATORS

FIGURE 3
TWO IDENTICAL AND OPPOSING CYLINDRICAL ACTUATORS

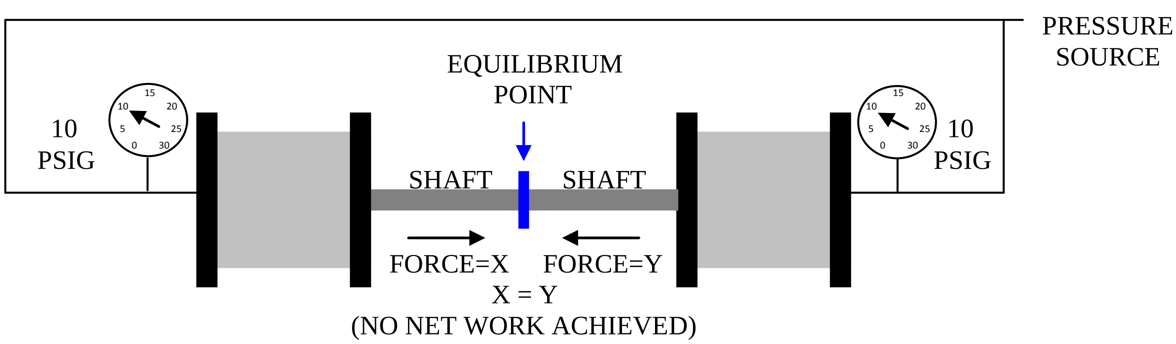


FIGURE 3 illustrates two opposing identical cylindrical actuators.

If both actuators receive fluid at the same pressure, the opposing forces on the two shafts are identical; therefore, they are in a state of equilibrium.

Three different approaches, causing one actuator to exert 15% more force, on its shaft, than an opposing actuator are:

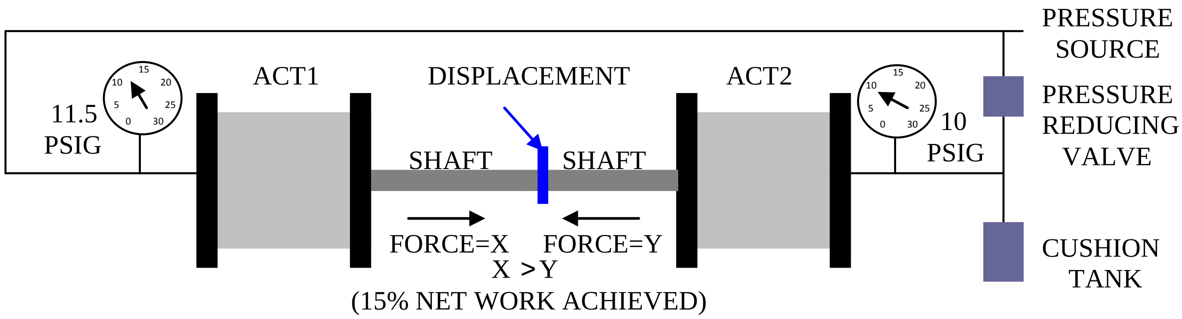
(1) Provide fluid at a 15% greater pressure than the opposing actuator receives. (FIGURE 4)

(2) Replace the actuator with a 15% larger actuator, thus consuming 15% more fluid.

(3) Increase the efficiency of the actuator, over the other, by 15%. This is achieved when replacing one with a diamond-shaped actuator, that is shown in FIGURE 2 and illustrated in FIGURE 5.

FLUID VOLUME RELATIONSHIP OF CYLINDRICAL OPPOSING ACTUATORS
TWO IDENTICAL OPPOSING ACTUATORS

FIGURE 4



If actuator (ACT1) receives fluid at a higher pressure than the other (ACT2), ACT1 develops more force, on its shaft, forcing fluid from ACT2.
As ACT1 and ACT2 are identical actuators, travelling identical distances, the amount of fluid forced from ACT2 is identical in volume to the fluid required by ACT2.

FLUID VOLUME RELATIONSHIP OF CYLINDRICAL AND
DIAMOND-SHAPED OPPOSING ACTUATORS

FIGURE 5

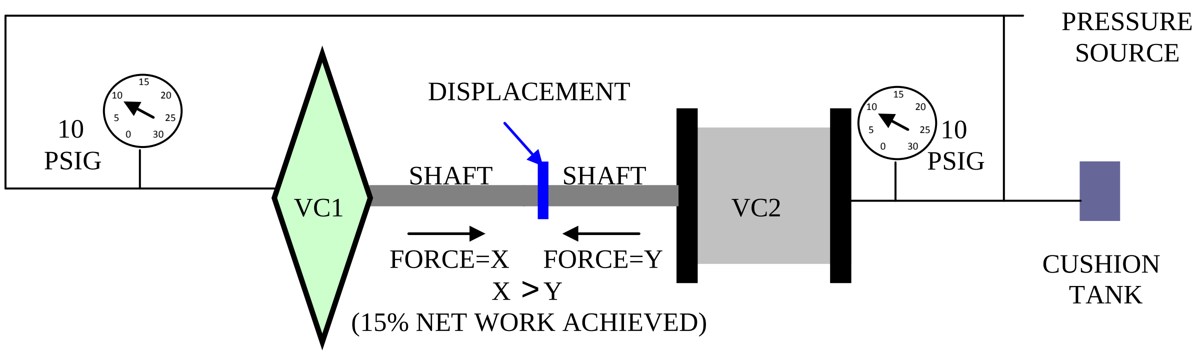


Figure 5 illustrates a diamond-shaped actuator and a cylindrical actuator receiving pressurized fluid at the same pressure with the boundary face of the cylindrical actuator equal in area to each of the four moving walls of the diamond-shaped actuator. Volume change one (VC1) < Volume change two (VC2).

The diamond-shaped actuator is >15% more efficient than the cylindrical actuator, as proven by scientist/engineers; therefore, develops >15% more force than the opposing cylindrical actuator.

If the diamond-shaped actuator's >15% force advantage forces fluid from the cylindrical actuator, the cylindrical actuator's displaced volume is always greater than the fluid volume added to the diamond-shaped actuator.

To this point we have:

(1) The cylindrical actuator and the diamond-shaped actuator, as in FIGURE 5, pressurized from a common source; therefore, at the same pressure.

(2) The diamond-shaped actuator has been tested by scientist/engineers to exert >15% more force than the cylindrical actuator when the boundary face of the cylindrical actuator is equal in area to each of the four moving walls on the diamond-shaped actuator.

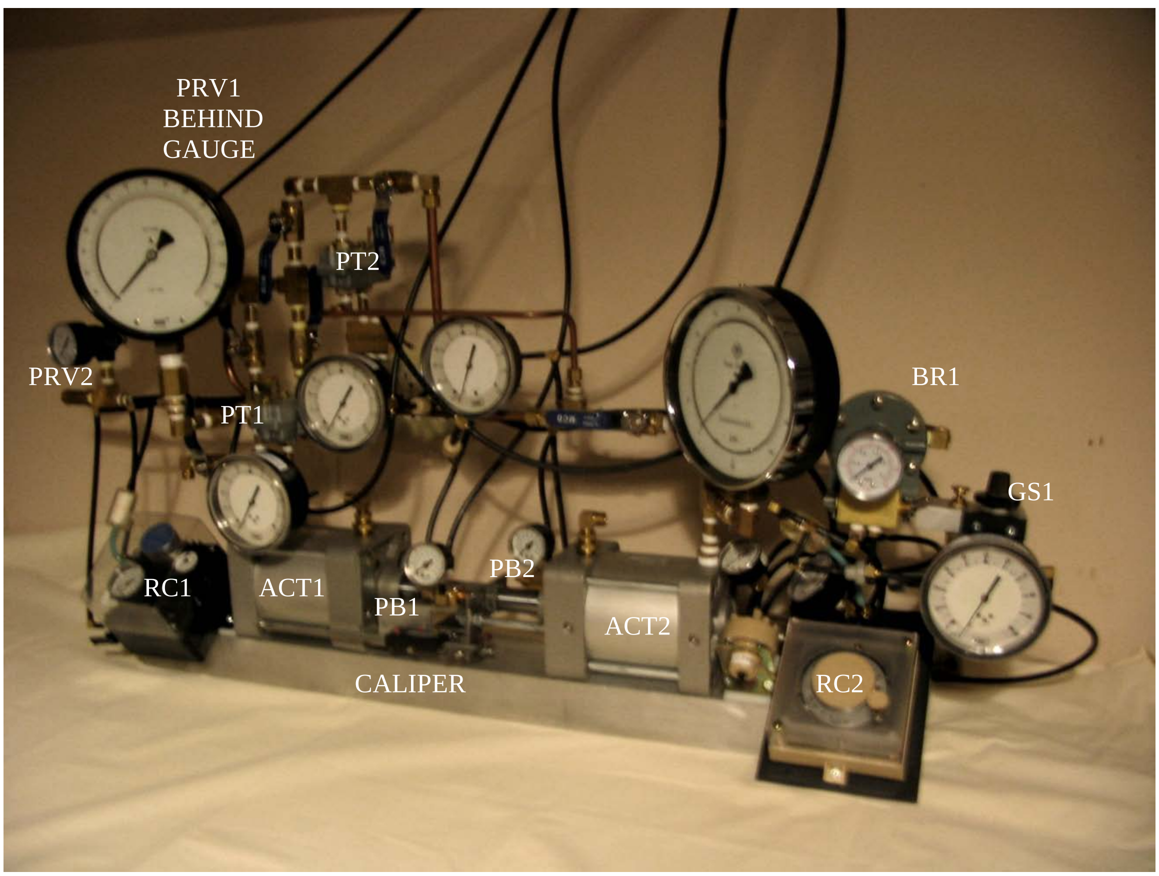
(3) If the diamond-shaped actuator's >15% greater force displaces fluid from the cylindrical actuator, the displaced fluid's volume is more than the volume added to the diamond-shaped actuator.

The question now is: “What is the counter force requirement to displace fluid from a pressurized cylindrical actuator?”

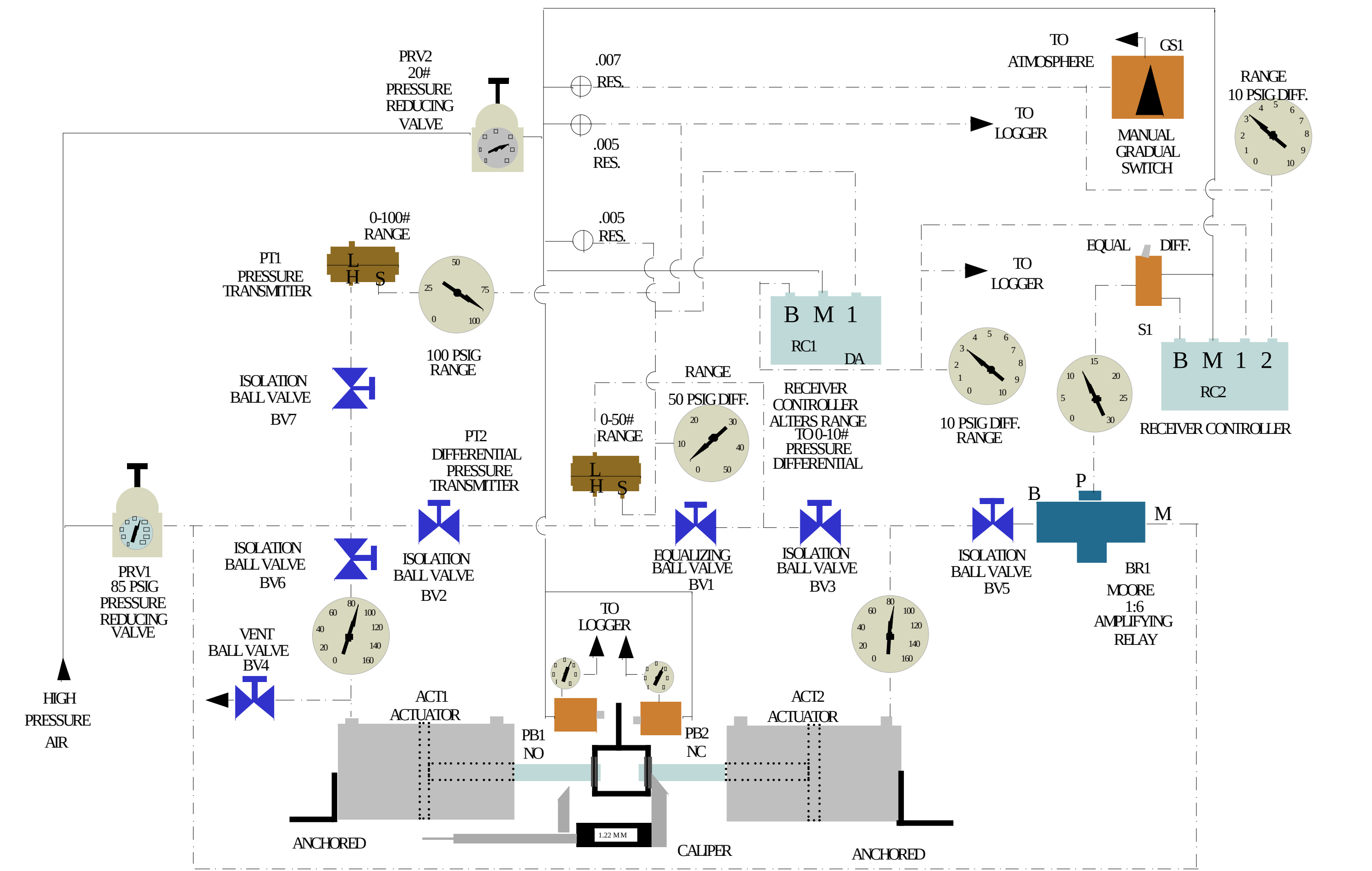
The physical model, presented in the photograph on this page, demonstrates that <4% greater counter force is required, forcing fluid from a cylindrical actuator. The diamond-shaped actuator develops >15% greater counter force based on investigations via scientist/engineers.

When the relationship illustrated in FIGURE 5 is employed, with the appropriate control system, the diamond-shaped actuator can produce its own source of pressurized fluid with >11% power potential remaining in each stroke.

PHYSICAL MODEL USED IN DETERMINING THE FORCE DIFFERENTIAL REQUIRED, DISPLACING FLUID FROM AN ACTUATOR, WHEN OPPOSED BY A MORE POWERFUL ACTUATOR.



The control drawing relating to the system in the above photograph is on page four.



NOTES

-Actuators ACT1 and ACT2 are identical.

-The apparatus's purpose is:

(1) Precisely determine the minimum differential pressure required, causing ACT1 to over power ACT2, driving the linkage from push button (PB1) to push button (PB2).

(2) Illustrate the varying rate of achieving the travel from PB1 to PB2 relative to the differential pressure changes for 3 PSIG, 4 PSIG, 5 PSIG and 6 PSIG.

-Pressure transmitter (PT1) measures the main pressure in ACT1 and provides data.

-Differential pressure transmitter (PT2) measures the differential pressure between ACT1 and ACT2 and provides data.

-Receiver controller (RC1) allows a magnified range, displaying only a ten PSIG differential, for more accurate readings in the data collection.

-Receiver controller (RC2) controls the differential pressure to match the desired differential pressure, between ACT1 and ACT2, as set manually via gradual switch (GS1)

TEST PROCEDURE

(1) Manually set PRV1 to 85 PSIG, as the base test pressure, and PRV2 to 20 PSIG serving the control circuit.

(2) Manually set GS1 to demand 0 PSIG differential and manually open BV1.

(3) Manually close BV6 and slowly bleed air to atmosphere via BV4 until PB2 removes its signal from logger.

(4) Manually close BV4 and open BV6; then close BV1; then zero caliper.

(5) Manually set RC2's set point, via GS1, to 3 PSIG differential. ACT1 pressure = 85 PSIG and ACT2 pressure = 82 PSIG at this point.

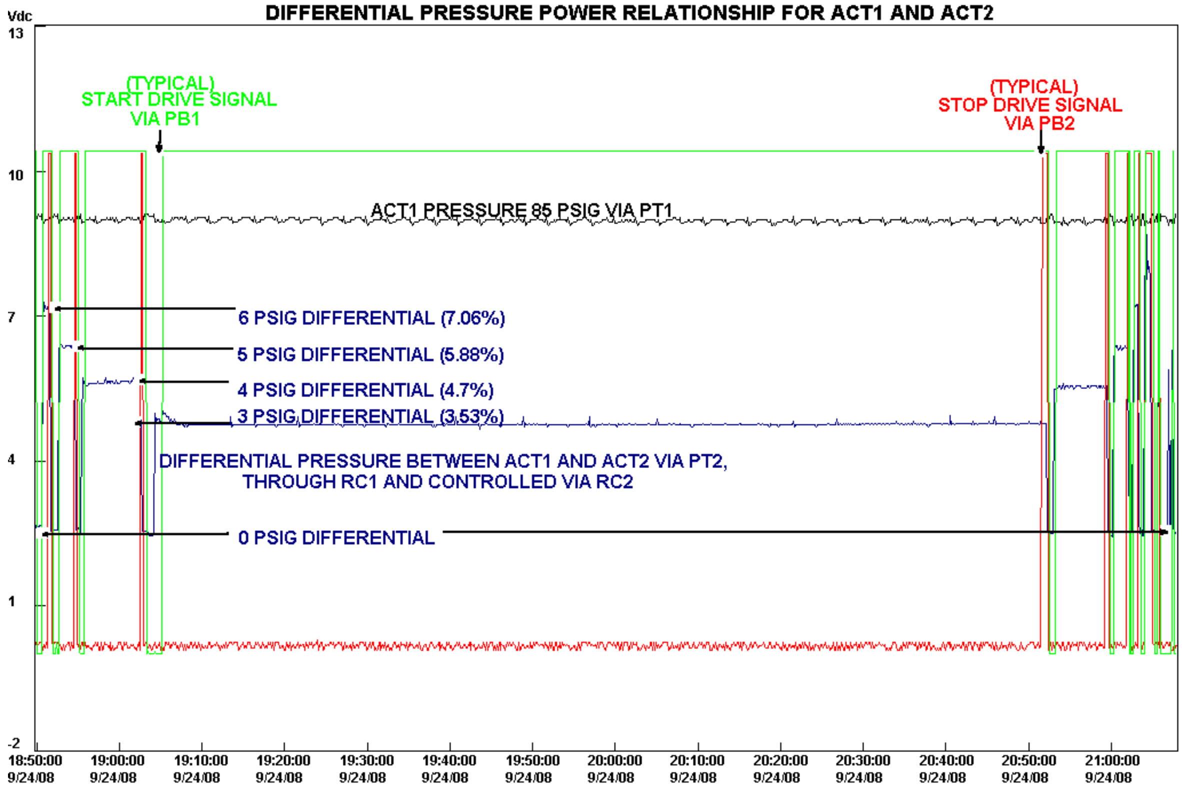
(6) As the linkage moves toward ACT2, PB1 is released, sending its signal to the logger, which alters the graph green line (page 5) from minimum to maximum. When the linkage pushes PB2, it sends its signal to the logger altering the graph red line from minimum to maximum.

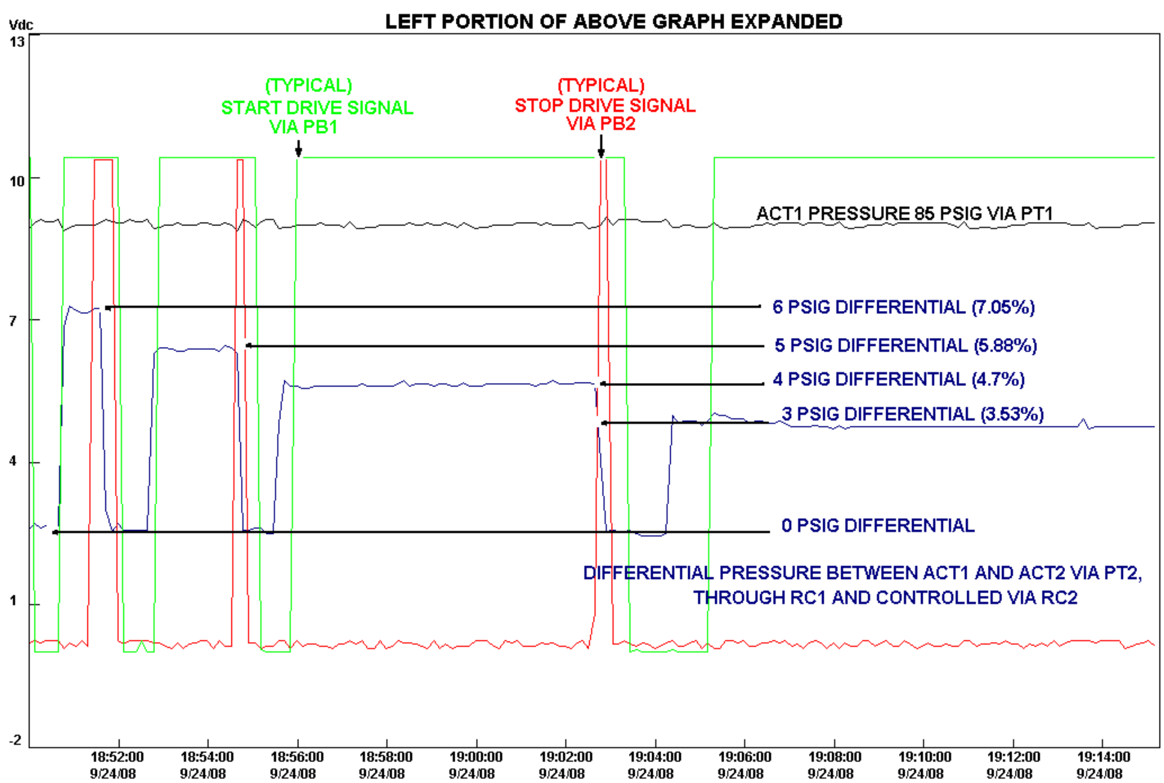
(7) Repeat steps two to five for 4 PSIG, 5 PSIG and 6 PSIG differential.

The two graphs below illustrate that only 3.53% more counter force is required on a cylindrical actuator's shaft to force fluid from that pressurized cylindrical actuator.

The scientist/engineers' testing demonstrates that the diamond-shaped actuator's efficiency advantage over cylindrical actuators is >15%; therefore, produces >15% counter force compared to the opposing cylindrical actuator illustrated in FIGURE 5.

The volume of fluid required by the diamond-shaped actuator is less than the volume of fluid it forces from the cylindrical actuator; therefore, the diamond-shaped actuator can generate its own source of pressurized fluid, while having >11% work potential remaining in each stroke.





ASSESSMENT REGARDING THE FLUID EQUATION

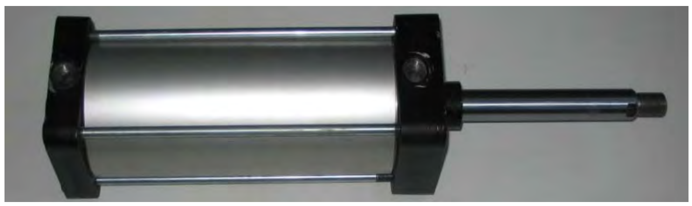
W = PV (WORK = PRESSURE x VOLUME)

AS IT RELATES

TO THE WORKING CAPABILITY OF PRESSURISED FLUIDS

APPLIED IN

THE CONVENTIONAL PISTON ACTUATOR



AND

THE DIAMOND-SHAPED ACTUATOR

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EXECUTIVE SUMMARY

The fluid equation WORK = PRESSURE x VOLUME (W = PV) applies to the linear fluid performance regarding conventional piston actuators.

In these cases work input = work output (WI = WO).

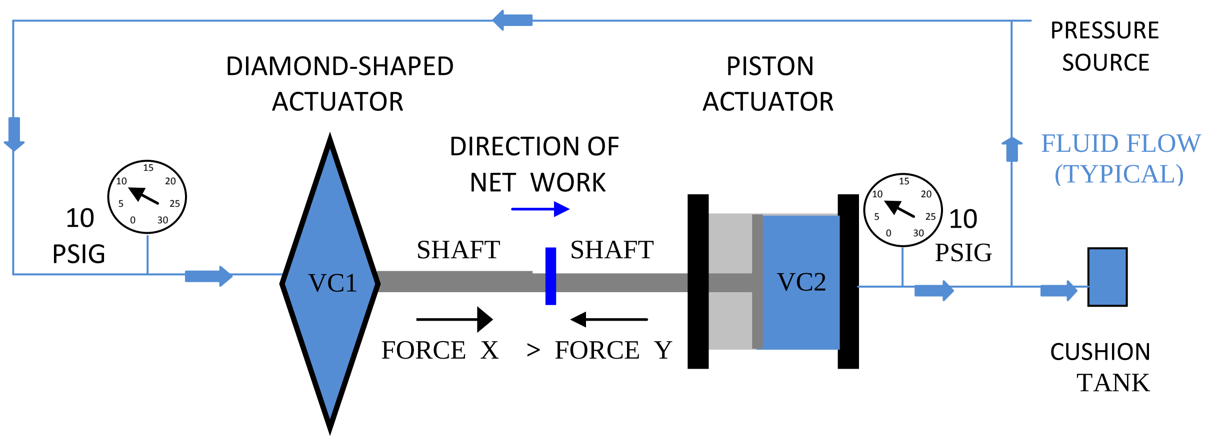
The fluid equation WORK = PRESSURE x VOLUME (W = PV) does **not** apply to the non-linear fluid performance regarding diamond-shaped actuators.

In the initial stages of the diamond-shaped actuators' stroke, work input is less than work output (WI < WO).

Many engineer/scientists have tested the physical model concluding this fact.

Diamond-shaped actuators are more efficient than conventional piston actuators in the early stages of the diamond-shaped actuators' stroke.

PATENTED APPLICATION CONCEPT



The diamond-shaped actuator expends 85% of its work potential generating its own source of fluid, pumping fluid from the piston actuator.

The remaining 15% of the diamond-shaped actuator's work potential may be extracted, addressing work requirements other than producing its own fluid source.

15% net work achieved = ((force "X" – force "Y") x travel)

VC1 (volume change one) < VC2 (volume change two)

REPORT'S PURPOSE

Present information illustrating:

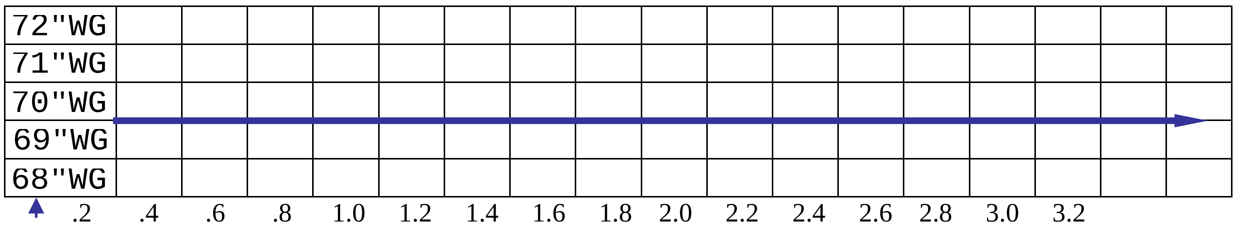
-1- The equation W = PV applies to the linear fluid performance in conventional piston actuators, but does **not** apply to fluid performance in diamond-shaped actuators.

-2- The diamond-shaped actuator performance is non-linear as it progresses through its stroke regarding the work/pressure/volume relationship.

-3- Diamond-shaped actuators are more efficient than conventional piston actuators in the early stages of the diamond-shaped actuators' stroke.

CONTENTS

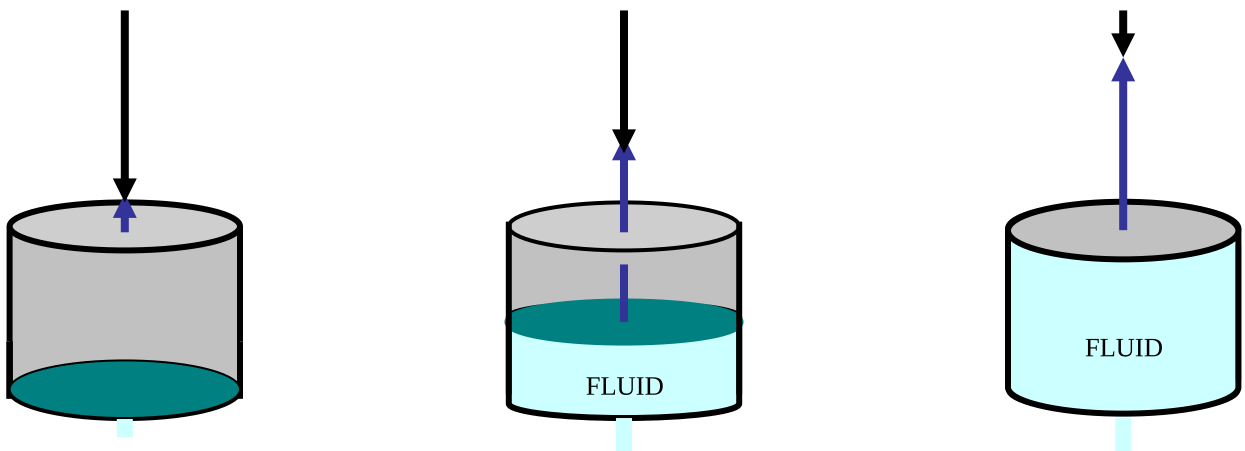
EQUILIBRIUM PRESSURE REGARDING
CONVENTIONAL PISTON ACTUATORS WITH BOUNDARY FACES OF 23.953 IN²
AND A CONSTANT LOAD OF 59.75 POUNDS



69.12"WG (2.494 PSI) PRESSURE
APPLIED TO PISTON'S 23.953 IN²
BOUNDARY FACE AT EQUILIBRIIUM

INCHES OF LIFT AT EQUILIBRIUM

LOAD 59.75#	LOAD 59.75#	LOAD 59.75#
PISTON FORCE < 59.75#	PISTON FORCE = 59.75#	PISTON FORCE > 59.75#
FULLY RETRACTED	EQUILIBRIUM	FULLY DRIVEN



FLUID PRESSURE	FLUID PRESSURE	FLUID PRESSURE
< 69.12 " WG (2.494 PSI)	= 69.12"WG (2.494 PSI)	> 69.12"WG (2.494 PSI)
LOAD FORCES PISTON TO	**ANY POINT** IN PISTON'S	FLUID LIFTS LOAD TO
PISTON'S MINIMUM STROKE	STROKING RANGE	PISTON'S MAXIMUM STROKE

The above graph illustrates that a piston with a constant load of 59.75 pounds and a boundary face of 23.953 IN² requires a pressure of 69.12" WG (2.494 PSI), maintaining equilibrium at all points in its stroke.

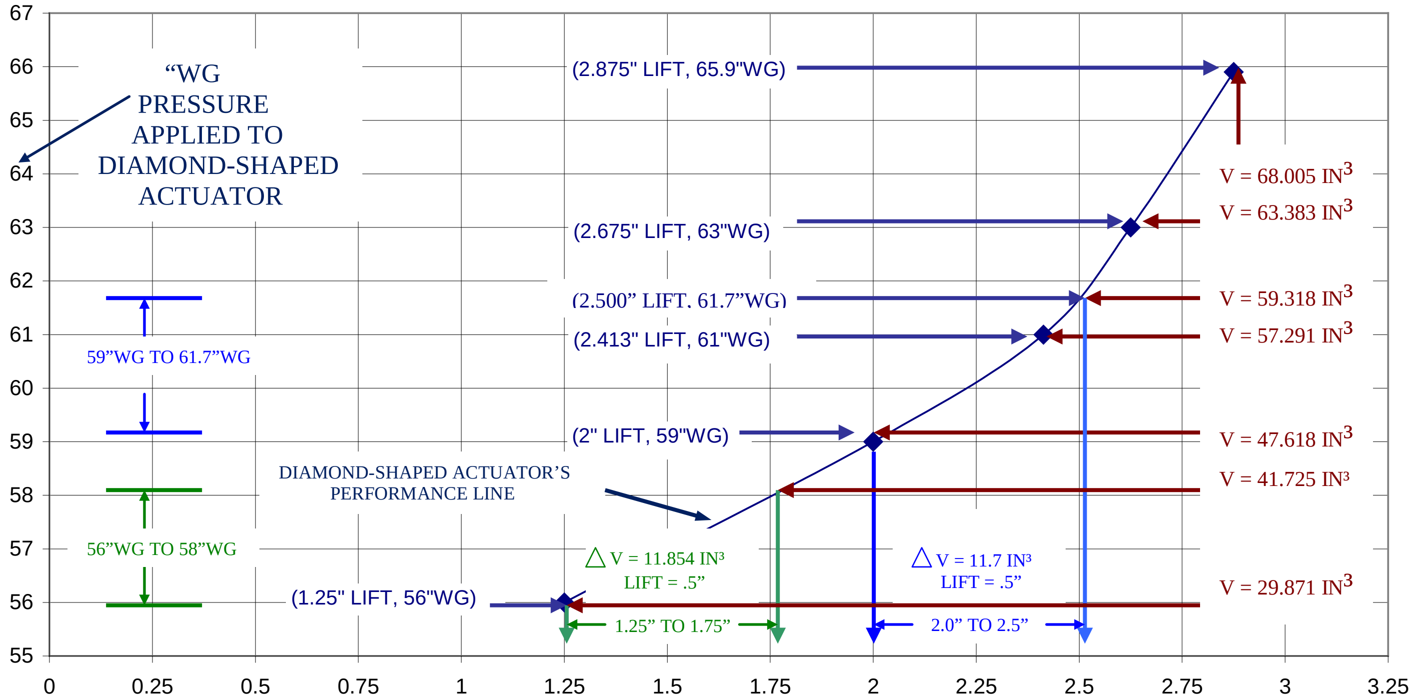
If another piston actuator receives an identical volume of fluid as the above example, with a boundary face twice as large (2 x 23.953 IN² = 47.906 IN²), its force is doubled; however, the stroke is one-half. Travelling one-half the distance of the above piston actuator, while producing double the force, generates exactly the same amount of work as the illustrated piston

The fluid equation W = PV applies to conventional piston actuators regarding work, pressure and volume. The relationship is linear.

The work input required for each .5" stroke for the illustrated piston is:

$$\begin{aligned} W &= PV \\ &= 2.494 \text{ PSI} \times (.5" \times 23.953 \text{ IN}^2) \\ &= 2.494 \text{ PSI} \times 11.9765 \text{ IN}^3 \\ &= 29.87 \text{ in/lb} \end{aligned}$$

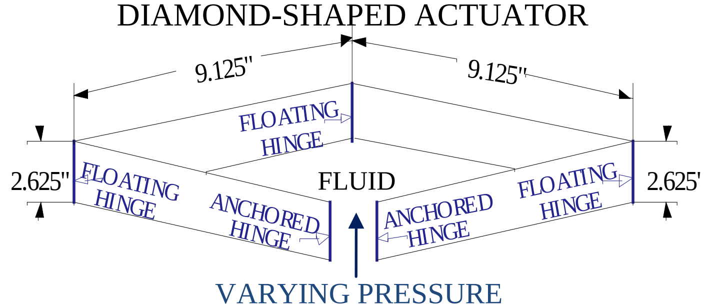
DIAMOND-SHAPED ACTUATOR EQUILIBRIUM POINTS AT VARIOUS PRESSURES WITH A CONSTANT 59.75# LOAD



INCHES OF LIFT AT EQUILIBRIUM WITH A CONSTANT 59.75# LOAD

The above graph illustrates the equilibrium performance of our test model with a constant 59.75 pound load.

DIAMOND-SHAPED ACTUATOR



The table below presents data regarding two segments of travel: each being one half inch.

SEGMENT	TRAVEL	VOLUME CHANGE	LOAD	FLUID PRESSURE		
				LOW	HIGH	AVERAGE
1.25"-1.75"	.5"	11.854 IN³	59.75 lb	56" WG	58.0" WG	57.00" WG
2.00"-2.50"	.5"	11.700 IN³	59.75 lb	59" WG	61.7" WG	60.35" WG

The output is equal regarding both segments of travel, each lifting 59.75 lb through .5".

(W=FD) W = 59.75 lb x .5 in
= 29.87 in/lb

The input work for one segment from 1.25" to 1.75" is:

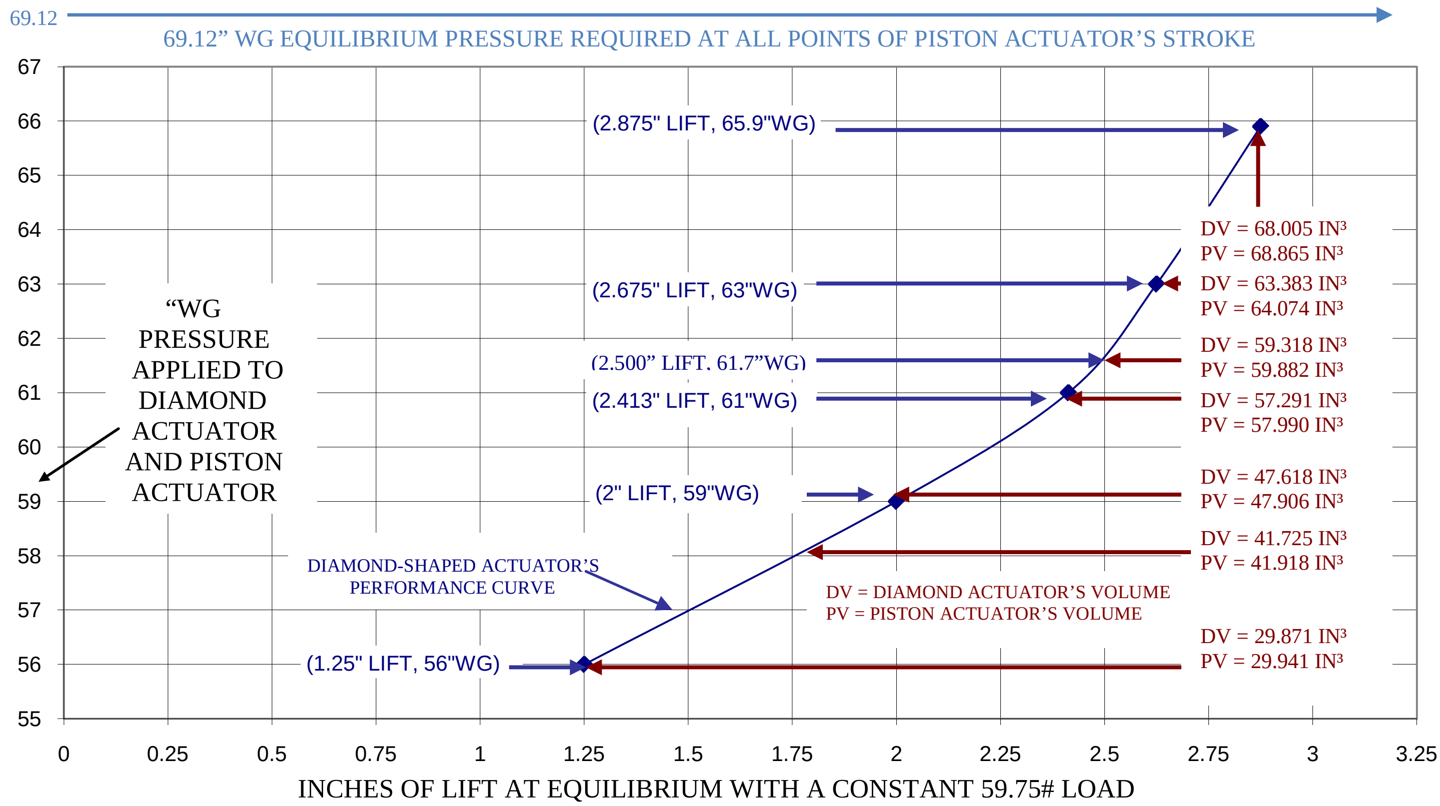
(W=PV) 57" WG (2.057 PSI) x 11.854 IN³ = 24.38 in/lb

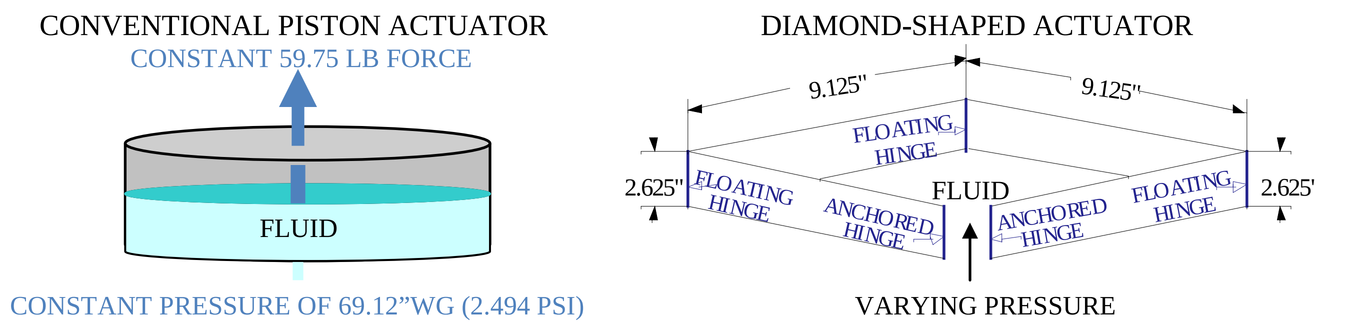
The work input for the segment from 2.0" to 2.5" is:

(W=PV) 60.35" WG (2.178 PSI) x 11.7 IN³ = 25.48 in/lb

NOTE: The work output for both segments is equal (29.87 in/lb), maintaining equilibrium, but the work inputs are different (24.38 in/lb and 25.48 in/lb).

EQUILIBRIUM COMPARISON REGARDING
THE DIAMOND-SHAPED ACTUATOR AND THE CONVENTIONAL PISTON ACTUATOR





The table below presents the efficiency comparison regarding equilibrium with an equal load of 59.75 pounds on both the piston actuator and the diamond-shaped actuator.

TRAVEL	PISTON ACTUATOR EQUILIBRIUM PRESSURE	DIAMOND EQUILIBRIUM PRESSURE	DIAMOND'S ADVANTAGE
1.250"	69.12" WG (2.4944 PSI)	56.0" WG	18.98%
1.750"	"	58.0" WG	16.08%
2.000"	"	59.0" WG	14.64%
2.413"	"	61.0" WG	11.75%
2.500"	"	61.7" WG	10.73%
2.675"	"	63.0" WG	8.85%
2.875"	"	65.9" WG	4.66%

NOTE:

The piston actuator requires the same pressure of 69.12" WG (2.494 PSI) at all points in its travel, maintaining the equilibrium force of 59.75 pounds to match the 59.75 pound load.

The diamond actuator requires varying pressures during its illustrated travel maintaining an equilibrium force of 59.75 pounds matching the 59.75 pound load.

SUMMARY

-1- The equation W = PV defines fluid performance regarding conventional piston actuators. The work input for every .5" of the example conventional piston actuator's travel requires 29.87in/lb of work. The input work matches the output work of 29.87 in/lb. The relationship is linear.

-2- The equation W = PV does not define fluid performance regarding diamond-shaped actuators. The two example .5" travel segments of our model produced identical work output of 29.87 in/lb, but their work input was different, being 24.38 in/lb and 25.48 in/lb.
Both the volume and pressure were different for each travel segment.
In this case W = W; however, PV for the first segment of travel ≠ PV for the second segment of travel.

WORK SUMMARY OF EXAMPLE ACTUATORS

ACTUATOR STYLE	TRAVEL RANGE	WORK RELATIONSHIP		
CONVENTIONAL PISTON	ANY .5" OF DRIVE	WORK IN (29.87 in/lb)	=	WORK OUT (29.87 in/lb)
		≠		=
DIAMOND-SHAPED	.5" (1.25" TO 1.75")	WORK IN (24.38 in/lb)	≠	WORK OUT (29.87 in/lb)
		≠		=
DIAMOND-SHAPED	.5" (2.00" TO 2.5")	WORK IN (25.48 in/lb)	≠	WORK OUT (29.87 in/lb)

-3- The diamond-shaped actuator achieves more work relative to a conventional piston actuator with identical fluid input, regarding both pressure and volume. This occurs only during the first portion of the diamond-shaped actuator's travel.

-4- The diamond-shaped actuator is more efficient than the conventional piston actuator, at varying magnitudes, depending on the angle variation of the diamond-shaped actuator's walls.

-5- This is the main corner stone regarding the patented invention "DIAMOND-SHAPED FLUID POWERED LINKAGE, SYSTEM AND ENGINE".

The efficiency differential produces a work differential, allowing the diamond-shaped actuator to attain its complete source of fluid via pumping the fluid from the conventional piston actuator. The diamond-shaped actuator expends approximately 85% of its work potential generating the total fluid requirement of that action. The diamond-shaped actuator experiences a surplus work potential of approximately 15% after generating its own total fluid requirement relating to each stroke.

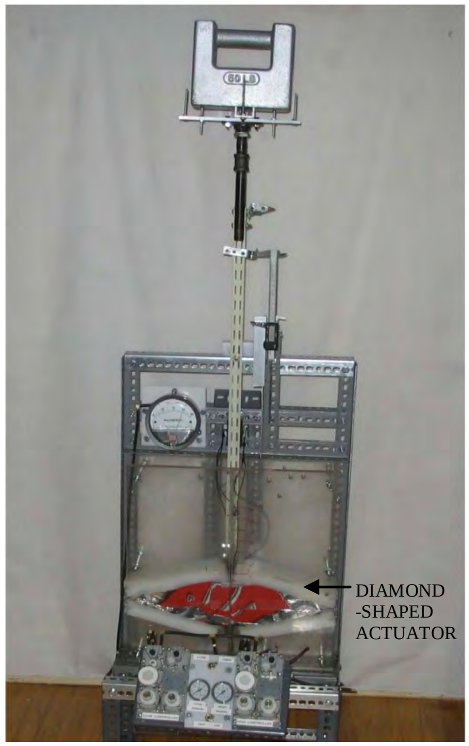
A NEWLY IDENTIFIED HYDRAULIC **MECHANICAL ADVANTAGE**

EXTENDS OUR UNDERSTANDING OF THE FLUIDIC EQUATION

W = **P**Δ**V** (**W**ORK = **P**RESSURE x **V**OLUME CHANGE),

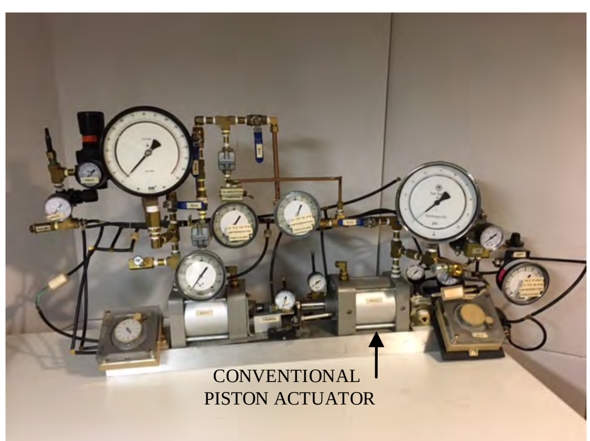
ALLOWING A MACHINE'S OPERATION TO ESTABLISH $W_{out} > W_{in}$

IMAGE 1



TEST MODEL PROVING THE **15%+ MECHANICAL ADVANTAGE** OF DIAMOND-SHAPED ACTUATORS OVER CONVENTIONAL PISTON ACTUATORS

IMAGE 2



TEST MODEL PROVING **< 4% MECHANICAL ADVANTAGE** IS REQUIRED FOR AN ACTUATOR TO PUMP PRESSURIZED FLUID FROM A CONVENTIONAL PISTON ACTUATOR.

CONTENTS

PREFACE

- The focus of this report is to present an explanation of a newly identified **mechanical advantage** in hydraulic applications. With this **mechanical advantage** applied, we are able to assist in facing one of our planets greatest threats; Climate Change.

  Quote from Donald Gorber Ph. D., P. Eng.

  " I believe that Mr. Strain's invention will advance the scientific community's understanding of thermodynamics relating to pressurized fluids and energy to a new level. If fully developed the invention has the potential to reduce energy and as a result a reduction in the use of fossil fuels, thus assisting in the battle against climate change."

  Three peer review letters are presented on pages 13.276 to 13.282 from Scientists/Engineers who tested the actual models, concluding that the Diamond-Shaped Actuator has >15% more **mechanical advantage** over conventional hydraulic piston actuators. Apparatus used is presented in photo (IMAGE 1).

- The current scientific understanding of the working capability of pressurized liquids accepts the hydraulic formula **W**ork = **P**ressure X **V**olume change (**W**=**P**Δ**V**) in designing conventional hydraulic piston systems.

- The newly identified fluid-driven **mechanical advantage**, presented in an altered hydraulic actuator design, requires an extended assessment of the formula (**W**=**P**Δ**V**).

- This report illustrates a method where a lesser volume of fluid at a lower pressure generates a larger volume of fluid at a higher pressure.

- Applying the formula W = PΔV to both the larger volume of fluid ($Work_{out}$), at a higher pressure, and the lesser volume of fluid ($Work_{in}$), at a lower pressure, concludes with $Work_{out} > Work_{in}$.

- The formula W=PΔV does not apply to the diamond-Shaped Actuator.

- The designed control circuit and photo (image 3) are presented of the actual apparatus that proves < 4% **mechanical advantage** of the Diamond-Shaped Actuator enables it to produce its full volume of hydraulic fluid by pumping the fluid from a conventional piston actuator.

- The remaining **mechanical advantage** is +15% - 4% = +11%, which produces $W_{out} > W_{in}$, concluding with +11% surplus work potential in each cycle of a reciprocating machine.

- Page 13.254 illustrates a practical comparison of the Diamond-Shaped Actuator's **mechanical advantage** over a conventional piston's performance.

- Pages 13.255 to 13.258 present a reciprocating machine design in its three modes of operation and a performance graph.

- More reports, You Tube videos and links to the patent files can be viewed via our website site at: http://www.apscontrols.ca

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90001

9 March 2007

TO WHOM IT MAY CONCERN

Mr. David Strain requested that I provide a letter of support relating to his invention currently patented in the USA and Europe. (The Canadian patent is pending.) He also requested that I state my credentials allowing the reader some assessment of my opinion.

I am Donald M. Gorber, Ph.D., P.Eng., current and founding President of SENES Consultants Limited established in Ontario in 1980. I hold a doctorate degree in Chemical Engineering and have more than thirty-five years experience in the energy and environmental field.

Mr. Strain made a presentation to SENES to discuss his invention. This presentation involved myself and our senior energy scientist/engineer, Dr Mehran Monabbati and provided us with a clear understanding of the principles relating to the invention.

The fundamental basis of the invention is the efficiency differential when comparing a conventional hydraulic actuator to the new diamond-shaped actuator. The efficiency advantage of the new actuator was clearly demonstrated during his presentation. Dr. Monabbati, who holds a doctorate degree in Chemical Engineering, tested the actual model, at both Mr. Strain's location and at SENES, reviewed certifications for the test equipment, and was able to confirm Mr. Strain's claims.

The tests indicated an efficiency advantage of approximately 17% over conventional actuators.

The work done through the stroke of the diamond-shaped actuator can push back a conventional cylindrical actuator. The displacement volume of conventional actuator is slightly greater than that volume of fluid required by the diamond-shaped actuator to accomplish the work. This indicates that the diamond-shaped actuator requires less volume of hydraulic fluid to accomplish the work compared to that of the conventional actuator (at the same pressure).

It should be mentioned that in an old 1874 USA patent (No. 147,519), Mr. Terrance Reilley demonstrated the same efficiency advantage. However, specific knowledge and recent technological advancement in mechanical equipment and instrumentation were required to achieve the results of Mr. Strain's invention.

I believe that Mr. Strain's invention will advance the scientific community's understanding of thermodynamics relating to pressurized fluids and energy to a new level. If fully developed the invention has the potential to reduce energy and as a result a reduction in the use of fossil fuels, thus assisting in the battle against climate change.

Yours very truly,

SENES Consultants Limited

Donald M. Gorber, Ph.D., P.Eng.
President

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April 26, 2007

To Whom It May Concern:

My name is Rajendra K. Singh, a doctoral graduate of Kansas State University.
I am actively engaged as a consulting scientist at R& D division of Intellimeter Canada Inc.
I have 30+ years of experience with multi-national companies in Canada in the area of engineering, product development, and ITU standards.

I am writing this letter of support, regarding an invention developed by
Mr. David Strain, who has patents granted in Europe, the USA and Canada.
Mr. Strain has a document from Mr. Ed Komadowski, SIEMENS Building Technologies, which confirms the performance of his control circuitry.

Mr. Strain brought a model to our facility and demonstrated the efficiency advantage of the diamond-shaped actuator. The load was measured with certified scales. The objective was to develop a perpetual machine.

During the demonstration, the internal pressure was raised to 65" WG lifting the load and then dropped to 54" WG causing the load to fall. During the fall, the pressure was switched to 60" WG. The diamond-shaped actuator stopped the fall of the fifty-nine and three-quarter pound load and lifted it upward with 60" WG pressure. A conventional cylindrical actuator holding the same elevation with identical fluid volume requires a pressure of 69.013" WG, just holding equilibrium.
This demonstrated an efficiency advantage greater than 15% favoring the diamond-shaped actuator over a conventional actuator. The force/travel relationship is non-linear; therefore, readings vary depending on the percentage of stroke achieved when readings are taken.

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At the time of testing there was an air leak in the diaphragm of the diamond-shaped actuator. Correcting the leak could have improved the performance of the diamond-shaped actuator. Because of the high cost of manufacturing a proper diaphragm, the experiment could not be continued.

I wish him success.

Rajendra K Singh, Ph.D.
R&D
Intellimeter Canada Inc
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Mr Dave Strain,
35 O'Brien Avenue
Stouffville, Ontario,
L4A 1G6

Dear Mr Strain:

Re: DIAMOND-SHAPED FLUID POWER LINKAGE

On November 18, 2001 I visited your laboratory to observe the operation of your DIAMOND-SHAPED FLUID POWER LINKAGE, perform the collection of data with my TOUR & ANDERSSON calibrated electronic manometer, and discuss some questions you presented.

(8) What force is required to move the linkage **without** the walls of the diamond shaped piston?

- Four lb. and ten oz. was recorded on the certified digital scale to overcome the inertia.

(9) What force is required to move the linkage **and** the walls of the diamond shaped piston?

- Nine lb. and twelve oz. was recorded on the certified digital scale to overcome the inertia. The linkage and walls dropped when the force was reduced to 6 lb. on the digital scale.

(10) What is the total load for the face of the fluid in the diamond shaped piston model to lift considering the walls, linkage, and the 50 lb. weight?

Fifty lb. plus nine lb. and twelve oz. = 59.75 lb.

(1) What is the lowest pressure in the diamond shaped actuator that causes an upward lifting motion?

- I observed the threshold of upper motion to occur at 60" w.g. on the water column.

(7) What is the travel to the equilibrium point at 60" w.g. from completely collapsed?

- The short diameter of the rhombus measured about 1". see note for question 6.

(2) Does the electronic certified equipment agree with the 0-5# certified gauge?
(3) Does the electronic certified equipment agree with the water column?

- I used a certified electronic manometer connected in parallel with the 0-5# certified gauge, the water column, and a 0-60" w.g. magnehelic to collect the following data:

- The 0-5# gauge was within ±1.3% and the 0-60" magnehelic, and the water column were both within ±0.7% of the electronic meter. This data shows quite exceptional agreement when considering the decimals of the analogue instruments are interpolated by eye.

	Electronic Meter	0-5# certified gauge	0-60" w.g. magnehelic	water column
	4.823 ft.w.g.	2.09#	58.9 " w.g.	58.33"
	4.753 ft w.g.	2.04#	58.7 " w.g.	57.35"
	4.684 ft. w.g.	2.005#	58.2" w.g.	56.75"
Threshold of motion	4.961 ft.w.g.	2.12#	59.95" w.g.	60.0"
Increase to 61"	5.030 ft.w.g.	2.19#	60.5" w.g.	61.0"

Note: The reading at the electronic meter increased slightly when the air supply pressure was held constant and the valve to the water column was closed, then returned to the previous reading when the valve was opened. We repeated this observation three times with consistent results. We did not investigate the correlation of individual meters or gauges with respect to two or more connected in parallel. There is reason to believe that each device reads a force and thus consumes some small portion of the energy applied. While there is merit in applying three or four devices to establish calibration, the data used for calculations should be collected from a single gauge to negate the effect of the energy consumed by the other devices.

(6) What is the volume of the diamond shaped actuator at the equilibrium point for 60" w.g? Do not allow for the volume of the diaphragm, but qualify that it reduced the fluid volume by some amount.

- The calculated gross area contained by four 9.0" long sides of a rhombus with the short diameter of 1.0" is 8.9861in^2. Thus the volume contained by the rhombus with a depth of 2.625" is 23.589in^3. The diaphragm within the rhombus that contains the force exerted on the walls of the rhombus occupies a considerable portion of the volume. The interior of the clear sides of the test rig are smeared with grease to address the friction of the moving sides of the rhombus. Precise measurement of the length of the interior walls of the rhombus, the short diameter, or the percentage of space occupied by the diaphragm is difficult because of the grease.

Although the force of about 60" w.g. is exerted equally and perpendicular over the entire inside surface of the diaphragm, the force does not appear to be transferred equally over the entire interior surface of the rhombus because of the wrinkles in the diaphragm and spaces.

(5) What is the travel to the equilibrium point at 62" w.g. from completely collapsed?

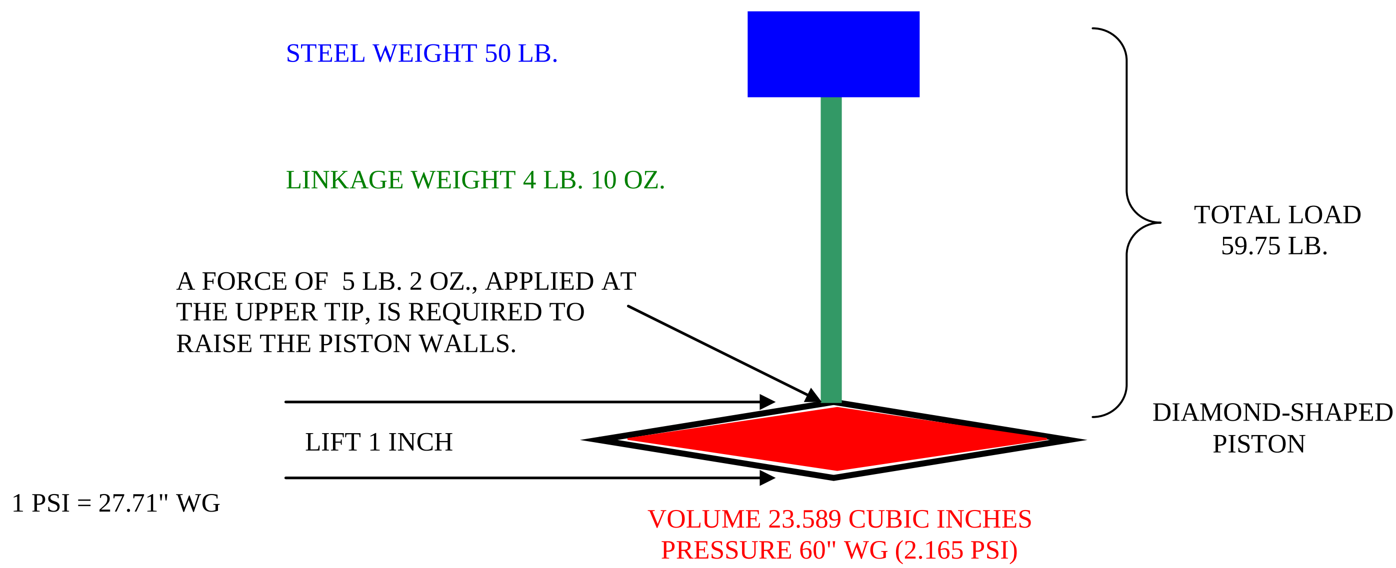
- The short diameter of the rhombus increased by 0.0858" from about 1" at 60" w.g.

(4) What is the volume of the diamond shaped actuator at the equilibrium point for 62" w.g? Do not allow for the volume of the diaphragm, but qualify that it reduced the fluid volume by some amount.

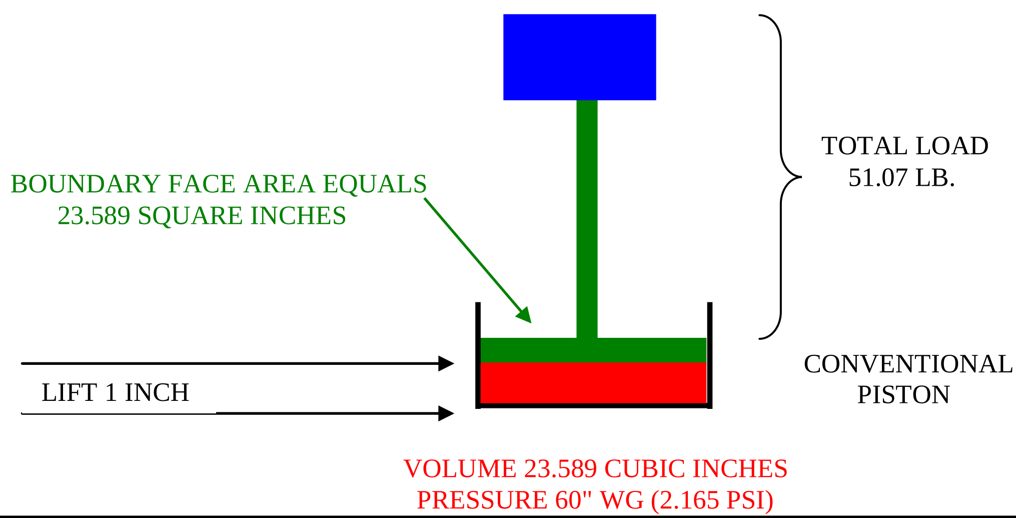
- When the air pressure exerted upon the diaphragm was increased from 60" w.g. to 62" w.g. the short diameter increased by 2.18mm (0.0858in).(Lifted the weight 0.0858). Using the previous data as a base, the new volume increases to 25.605in^3 a difference of 2.016in^3

MECHANICAL ADVANTAGE OF THE NEW DIAMOND-SHAPED PISTON RELATIVE TO THE CONVENTIONAL PISTON

This drawing is based on Robert J. Blanchard P.Eng.'s recorded observations, while using certified test equipment.
Note that 23.589 cubic inches of fluid at 60" WG (2.165 PSI) pressure, is at the point of equilibrium, with a 59.75 pound load, at one inch of travel..

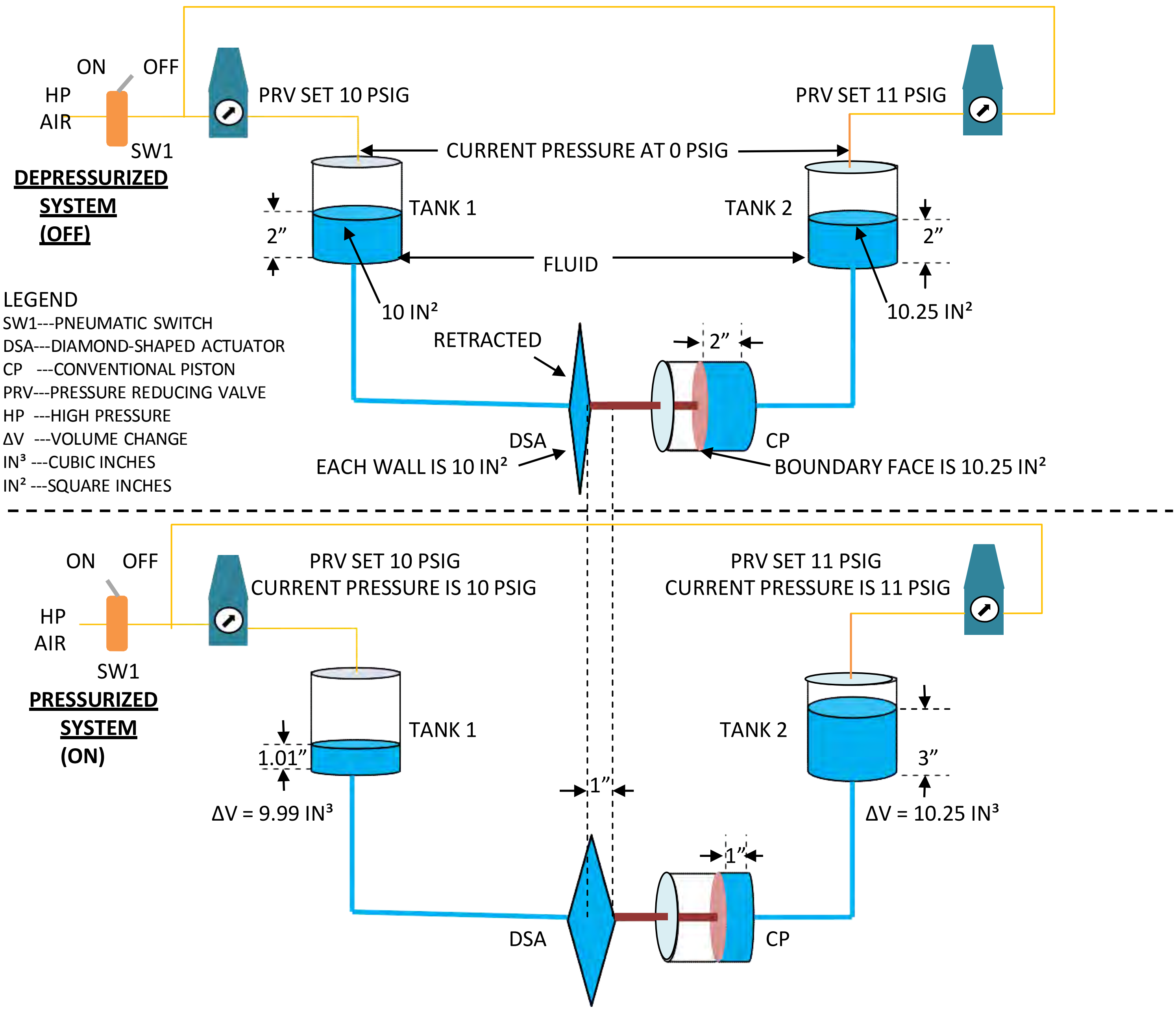


This drawing is based on completely frictionless conventional piston.
Note that 23.589 cubic inches of fluid at 60" WG (2.165 PSI) pressure is at the point of equilibrium with a 51.07 pound load, at one inch of travel.



Note that the diamond-shaped piston lifts 16.9% more load than the conventional piston through one inch of travel with the same volume of fluid, at the same pressure.

$W(P\Delta V)_{out} > W(P\Delta V)_{IN}$



TEST PROCEDURE

Prepare apparatus as per the upper illustration, DEPRESSURIZED SYSTEM (OFF), which disallows pressurization on both the Diamond Shaped Actuator (DSA) and the Conventional Piston (CP).

Turn switch (SW1) on to PRESSURIZED SYSTEM mode, allowing both PRV’s to attain their pressure settings. (10 PSIG and 11 PSIG.) Observe the volume changes in TANKS 1 and 2.

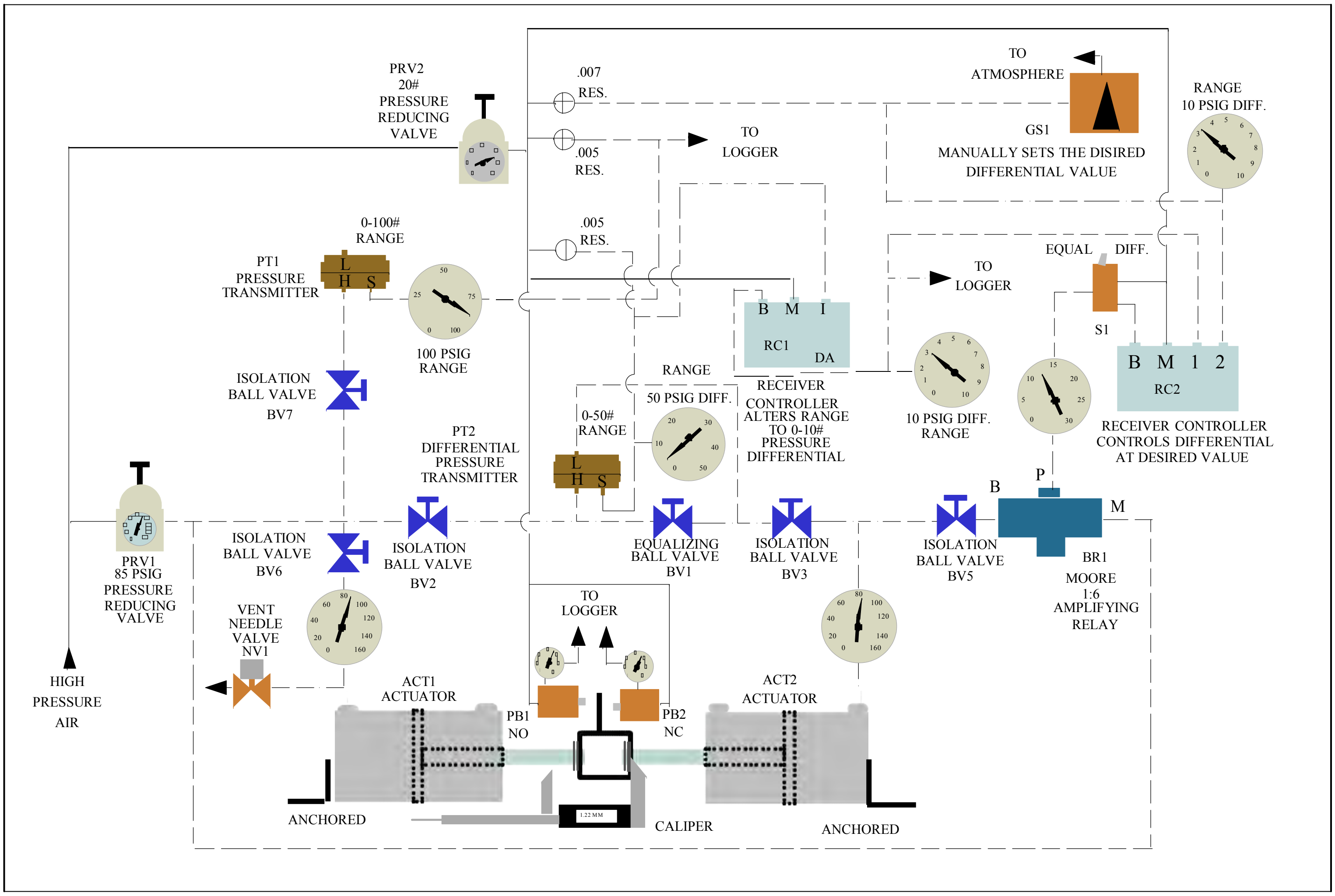
The minimum mechanical advantage of a DSA over a CP, reported by testing scientists is 15%. Please see the following scientists’ letters. The DSA, with 10 IN^2 walls and 10 PSIG pressure exerts a minimum force of 115 lb. The CP, with a 10.25 IN^2 boundary face and 11 PSIG pressure exerts a force of 112.75 lb. The DSA overpowers the CP (115 lb > 112.75 lb), forcing fluid from the CP.
The volume change (ΔV) in the DSA is 9.99 IN^3; therefore, ΔV in TANK 1 is 9.99 IN^3.
The volume change in CP is 10.25 IN^3; therefore, TANK 2 ΔV is 10.25 IN^3

Using the fluidic formula Work = Pressure x Volume Change ($W = P\Delta V$)
TANK 1’s work input potential................W = $P\Delta V$ = 10.0 PSIG X 9.99 IN^3 = 99.90 in-lb
TANK 2’S work output potential.............W = $P\Delta V$ = 11.0 PSIG X 10.25 IN^3 = 112.75 in-lb

CONCLUSION: Lesser work input can produce greater work output ($W_{out} > W_{in}$).

TEST APPARATUS PROVING THAT IF ACTUATOR ONE (ACT1) HAS MORE THAN **4%** MECHANICAL ADVANTAGE OVER ACTUATOR TWO (ACT2), ACT1 CAN PUMP ITS TOTAL FLUID REQUIREMENT FROM ACT2. SCIENTISTS TESTING THE MODELS CONCLUDE THAT THE DIAMOND-SHAPED ACTUATOR (IN ACT1'S PLACE) HAS A MECHANICAL ADVANTAGE OVER CONVENTIONAL PISTONS EXCEEDING **15%**

CONTROL CIRCUIT



PHYSICAL MODEL OF TEST APPARATUS

IMAGE 3

SUMMARY

- The formula W = PΔV applies the conventional hydraulic piston.

- The formula W $\neq$ PΔV applies to the Diamond-Shaped Actuator.

- Scientists who tested the actual physical models confirm the mechanical advantage of the Diamond-Shaped actuator at +15% over the conventional piston.

- The mechanical advantage of < 4% produces a work advantage in favour of the Diamond-Shaped Actuator.

- The Diamond-shaped Actuator generates its total fluid requirement by pumping the fluid from a conventional piston with > 11% mechanical advantage remaining. (+15% - 4% ≥ 11%)

- A reciprocating machine can be produced which produces enough work to run itself and deliver surplus work externally, suiting other requirements.

- The machine will only stop if commanded to do so by blocking fluid flow, depressurization or mechanical failure.

- A control circuit is presented on pages 13.255 to 13.257 and page 13.254 illustrates the mechanical advantage of the Diamond-Shaped Actuator.

- Please send questions or comments.

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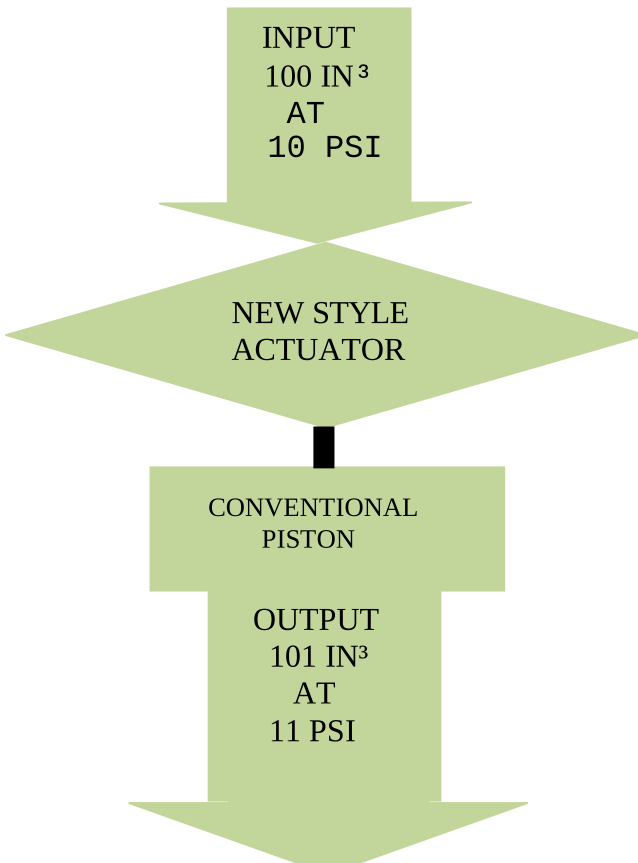
A NEW VIEW OF THERMODYNAMICS RELATING TO THE WORKING CAPABILITY OF PRESSURIZED FLUIDS



TYPICAL CYCLE

PREFACE

This report contains an extended viewpoint regarding the Laws of Thermodynamics, concerning the working potential of pressurized fluids.

The newly developed Hydraulic Motor is based on concepts contained within our patent application. The USA patent was issued on August 31, 2004, patent number 6,782,800. The European patent has been granted under number 1240435. The patents have been granted in Switzerland, Belgium, France, Italy, the UK and Ireland under patent number 1240435. The patent has been granted in Germany under patent number DE 600 15 181 T2 2005.11.24. The patent has been granted in Austria under patent number AT E 280 331 T1. The whole principle is based on squaring off a conventional piston, hinging the walls and rotating it by 45°.

CONTENTS

This report contains two different approaches to explain the new discovery relative to the currently accepted Laws of Thermodynamics.

Prepared by:
Dave Strain
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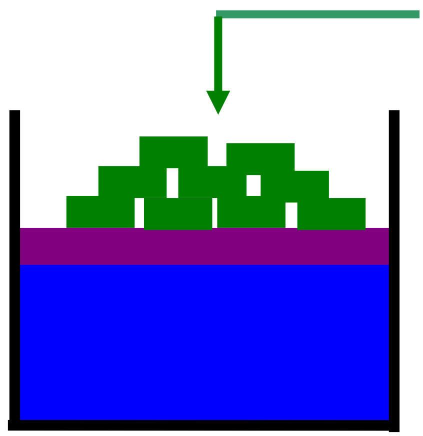
analystsofpneumatic@bellnet.ca

QUASIEQUILIBRIUM EXPANSION OR COMPRESSION PROCESS

The figure 2.4, below, is copied from section 2.2.3 in the Second Edition of "FUNDAMENTALS OF ENGINEERING THERMODYNAMICS" written by MICHAEL J. MORAN of Ohio State University and HOWARD N. SHAPIRO of Iowa State University and Technology.

A thorough understanding of this segment in the Laws of Thermodynamics is required for one to understand the development of the hydraulic displacement motor.

FIGURE 2.4



The Laws of Thermodynamics conclude that the upward force generated by applying pressurized fluid over the piston's moving boundary face is equal to the downward force exerted by the lifted mass when the piston is in a state of equilibrium. I completely agree with this conclusion for a conventional piston as illustrated.

This paper uses pressures and areas identical to our model to establish an understanding of why our new type of piston attains a power increase over a conventional piston.

When the face area of the conventional piston is 23.95 IN^2 and the pressure is 2.51 PSI, the upward force is 60 pounds. If the piston is at equilibrium, the downward force of the mass is 60 pounds.

The piston illustrated in Figure 2.4 has linear characteristics regarding its travel to volume relationship. For 2.5 inches of travel, with a face area of 23.95 IN^2, 59.875 IN^3 of volume change is required to lift the 60 pounds of mass.

If you rotate a squared off piston 45° and hinge its walls, as per our model, you will create a new type of piston that expands in a diamond shape. It will have four moving walls and exert its total force at the tip. It will only require 59.318 IN^3 of fluid to lift the same 2.5 inches. This is .9% less fluid to travel the same distance. The relationship of travel to volume is *not* linear for this new piston.

The experiments with our model have demonstrated that the diamond shaped piston requires 2.24 PSI to lift the 60 pounds of mass to an elevation of 2.5 inches. This is 10.7% less pressure to do the same amount of work as the piston in the illustration 2.4 in "FUNDAMENTALS OF ENGINEERING THERMODYNAMICS"

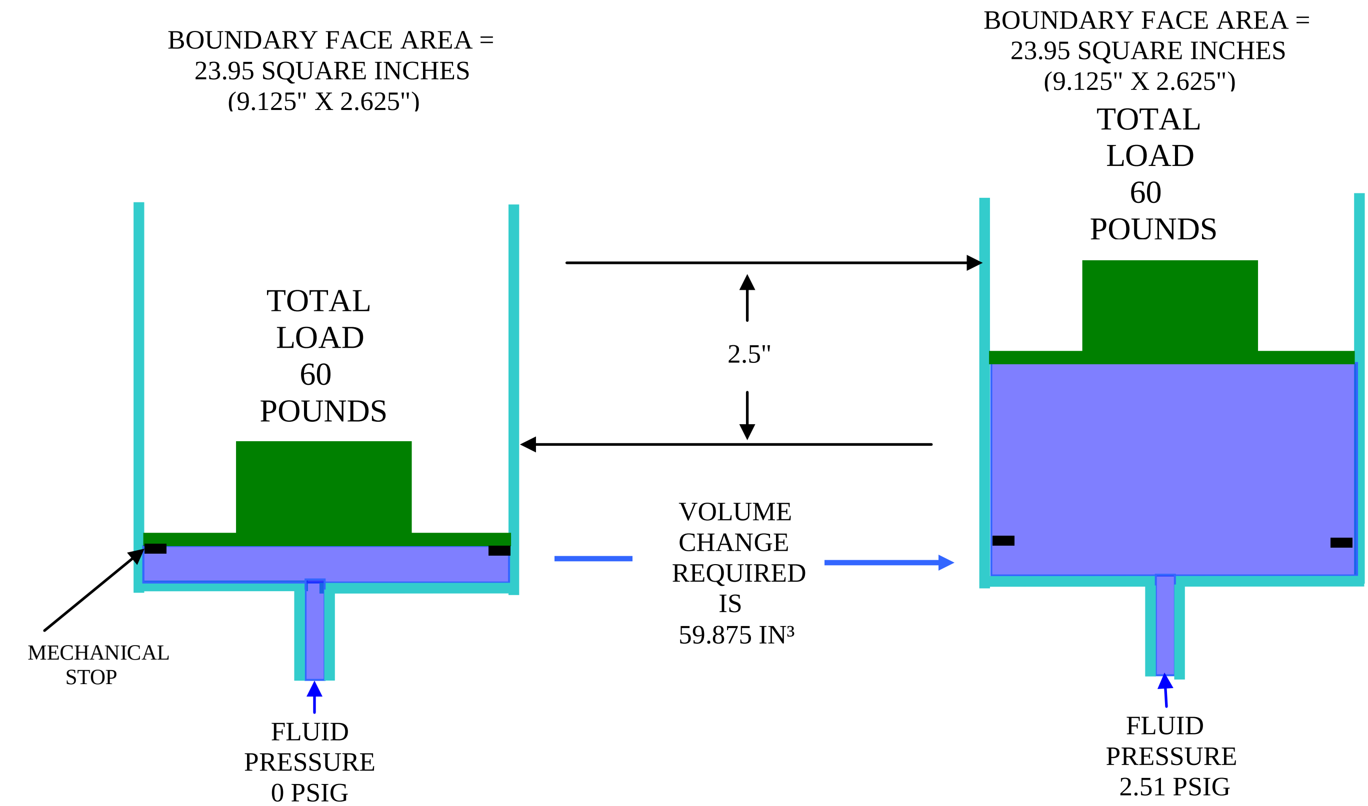
In comparison to the illustration 2.4 in "FUNDAMENTALS OF ENGINEERING THERMODYNAMICS", the diamond shaped piston can achieve the same work with **.9% less fluid volume** and with **10.7% less pressure.**

This allows the diamond shaped piston to displace fluid out of a conventional piston, at greater volume and at greater pressure, than the volume of fluid required to initially drive the diamond shaped piston.

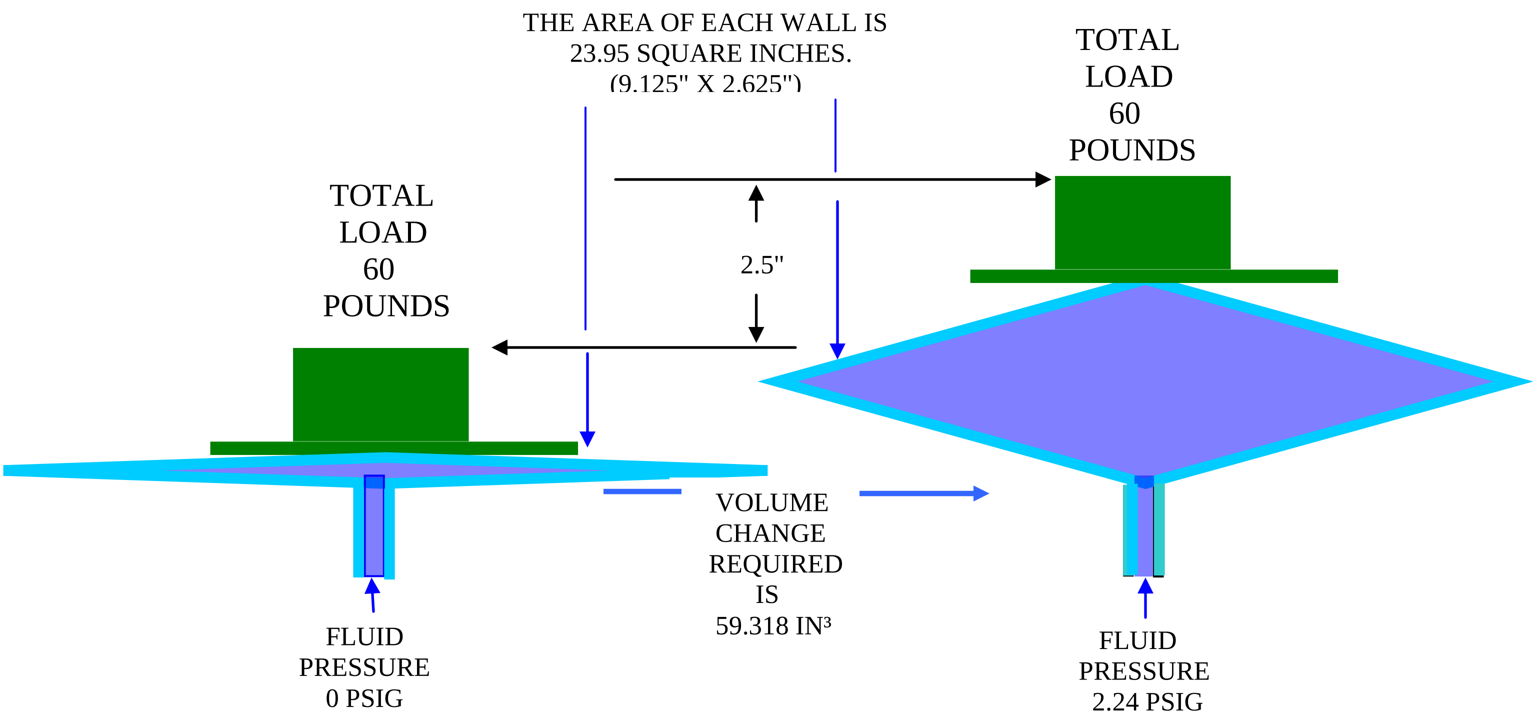
A model, that actually functions, has been built using this relationship to demonstrate a reciprocating motor that powers itself and generates surplus mechanical work.

AN EXTENDED VIEW OF THE LAWS OF THERMODYNAMICS RELATING TO THE QUASIEQUILIBRIUM EXPANSION OR COMPRESSION PROCESSES

NORMALLY ACCEPTED RELATIONSHIP OF PRESSURIZED FLUID AND WORK.

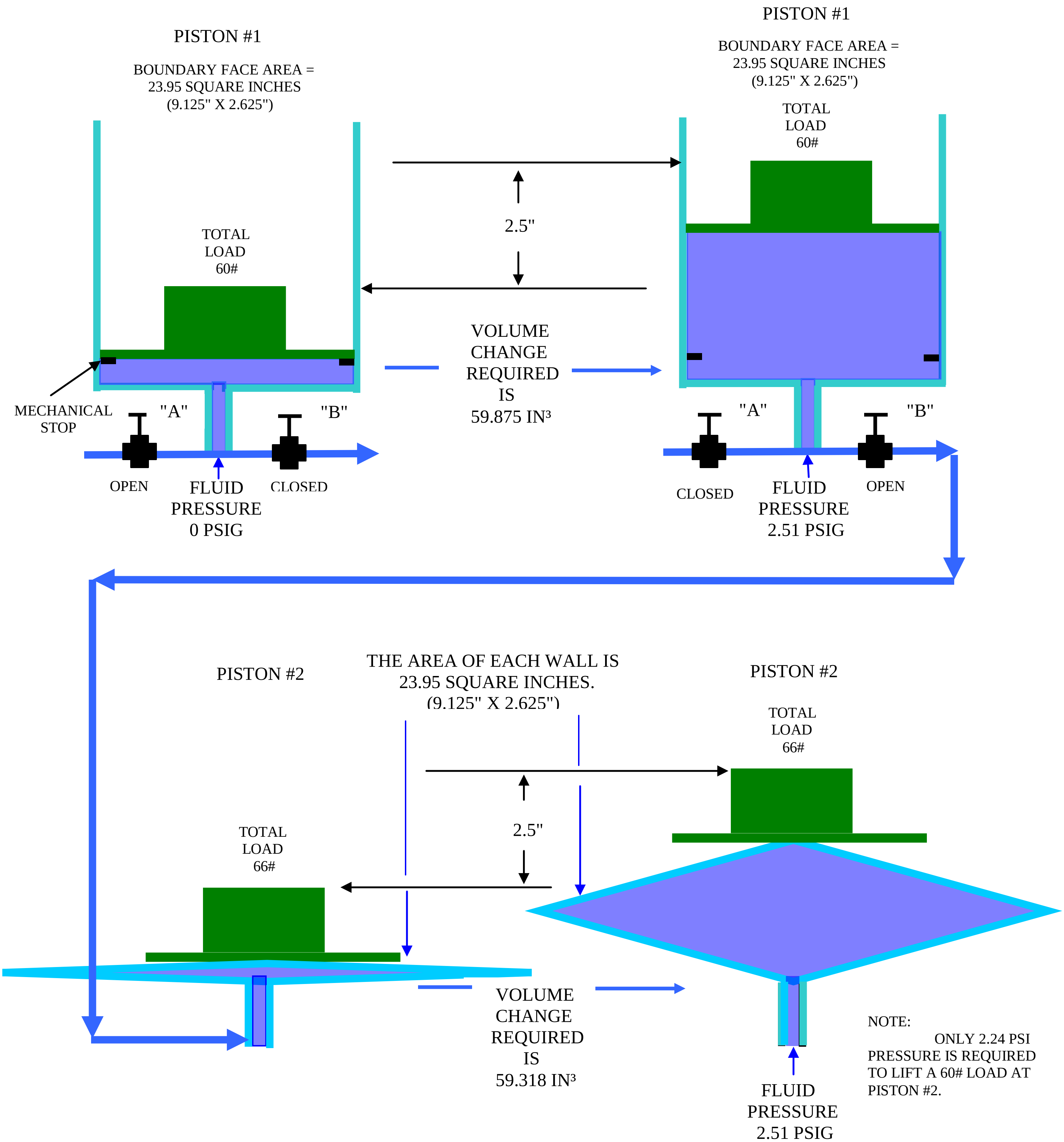


A NEW VIEW POINT
INCREASED WORK CAPABILITY OF FLUID BY ROTATING PISTON 45°.
(FIGURES USED REFLECT ACTUAL EXPERIMENTAL RESULTS)



EXAMPLE OF HOW POWER GAIN IS ACHIEVED.

-AT PISTON #1, CLOSE VALVE "B" AND OPEN VALVE "A" TO FILL PISTON #1 FROM A FLUID SOURCE UNTIL THE 60# LOAD HAS LIFTED 2.5".
-CONNECT PISTON #1 TO PISTON #2 AS SHOWN. CLOSE VALVE "A"; THEN OPEN VALVE "B".
-THE 60# LOAD ON PISTON #1 WILL FORCE THE FLUID FROM PISTON #1 INTO PISTON #2, LIFTING THE 66# LOAD OF PISTON #2, 2.5", WHILE USING ONLY 99% OF THE FLUID AVAILABLE IN PISTON #1.
-THE FORCE DEVELOPMENT AVAILABLE AT THE TIP OF PISTON #2, VIA THE FLUID FLOWING FROM PISTON #1 INTO PISTON #2, IS MORE THAN 10% GREATER THAN REQUIRED TO LIFT THE 60# LOAD ON PISTON #1.
--THE 66# LOAD LIFTED BY PISTON #2 CAN BE RE-APPLIED TO PISTON #1 TO DISPLACE A GREATER VOLUME AT A GREATER PRESSURE ON THE NEXT CYCLE.



HYDRAULIC MOTOR EXPLANATION

GENERALLY ACCEPTED CONDITIONS

The accepted Laws of Thermodynamics describe the working capability of pressurized fluid. This is addressed in the text, "Fundamentals of Engineering Thermodynamics", written by Shapiro and Moran; section 2.2.3, "Work in Quasiequilibrium Expansion or Compression Processes".

The Laws of Thermodynamics establish that a piston will move in the direction of lesser force until the forces on both sides of the piston are equal. If the force is reduced on one side, the piston movement will continue until equilibrium has been regained.

Shapiro and Moran used a conventional piston, with incremental masses as a load to illustrate this fact. (FIGURE 1) (Page 13.293)

A piston with a ten square inch boundary face and a fluid pressure of 10 PSIG would reach equilibrium if the mass it was lifting weighed one hundred pounds.

The potential work in the lifted mass has a linear relationship to the work required to put the pressurized fluid into the piston "B" and they are equal at all times equilibrium has been reached.

The same fact can be demonstrated by linking the drive rods of two identical pistons. (FIGURE 2) (Page 13.293) If both pistons have the same pressure applied to them the force of each piston would be equal and there would be no movement. If either piston's pressure were reduced, the piston with the higher pressure would attain a force advantage and start to push back the other piston. If the pressures become equal at some point, the movement will stop because the forces on the pistons will have become equal; therefore, the system will have regained equalibrium.

At mid-point of the pistons' stroke, the amount of work to provide the pressurized fluid to both pistons is equal and the work relationship potential is linear.

EXAMPLE:

Consider both pistons to have a rectangular drive face of one inch by ten inches and a maximum stroke of ten inches.

When 10 PSIG pressure is applied to the pistons, they would both exert a force of one hundred pounds.

Equalibrium, at the middle of their strokes, would require the same volume of fluid applied to each piston.

Consider the pistons arranged as in FIGURE 3 (Page 13.294). Piston "A" is filled with fluid and connected to a cushion tank "C" that is able to accept piston "A's" volume at a constant pressure of 10 PSIG.

Piston "B" receives fluid from another source.

When piston "B" receives fluid pressurized to about 10.01 PSIG it will start to displace piston "A's" fluid into cushion tank "C".

In this case the volume required driving piston "B" is the same as the volume displaced in piston "A". The pressure of the fluid in piston "B" is very slightly higher than the pressure of the fluid displaced from piston "A".

The work potential relationship of the two fluid volumes is linear.

NEW CONSIDERATIONS

The accepted facts from the Laws of Thermodynamics in the previous part of this paper are the basic building blocks to understand the operation of the hydraulic motor. There is one significant alteration to the arrangement of piston "B". It is rotated forty-five degrees, hinged at its walls and anchored at two points to make a different piston, "B2", as in FIGURE 4 (Page 13.294).

This arrangement allows piston "B2" to expand in a diamond shaped form with piston "B2's" external force, the total of four moving walls**,** culminating at the tip connected to piston "A".

The face area of each of piston "B2's" walls is equal to the boundary face of piston "A". This relationship assures that when piston "B2" expands to displace the fluid in piston "A", the amount of fluid required causing movement in piston "B2" will always be less than the volume of fluid displaced from piston "A".

The volume displacement relationship of piston "A" and piston "B2" is NOT linear.

This relationship allows a lesser volume of fluid to displace a greater volume of fluid for the same linear travel.

The larger volume of displaced fluid in piston "A" has to attain a pressure equal to or greater than the pressure of the fluid in piston "B2" in order for the hydraulic motor to function. This is achieved, as the force at the tip of piston "B2" is greater than the force applied to any one of piston "B2's" four equal walls. The boundary area of piston "A" is equal to each wall of piston "B2".

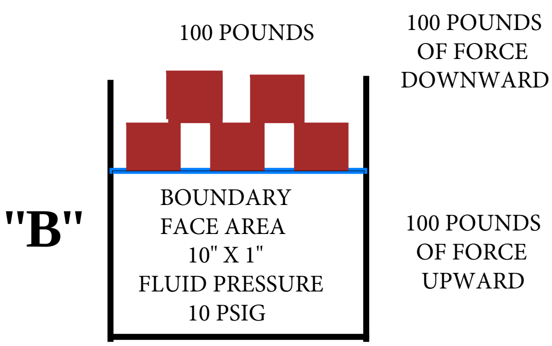
Our experiments indicate that there is a force advantage from zero degree expansion to about twenty-two degree expansion. The power gain starts in the range of fifteen to twenty percent and decreases to zero percent at about twenty-two degree rotation.

The net result of these considerations is that a smaller volume of fluid at a lesser pressure can displace a larger volume of fluid at a higher pressure.

A circuit has been developed, using this fact, to build a machine that runs itself with no energy input other than the initial start. The machine can be opposed to extract a percentage of the surplus power generated in each cycle while the machine continues to run.

FIGURE 1

STATE OF
EQUILIBRIUM



ILLISTRATION IS SIMILAR
TO FIGURE 2.4 ON PAGE 34 IN
"FUNDAMENTALS OF ENGINEERING
THERMODYNAMICS"
WRITTEN BY MORAN & SHAPIRO.

FIGURE 2

STATE OF
EQUILIBRIUM

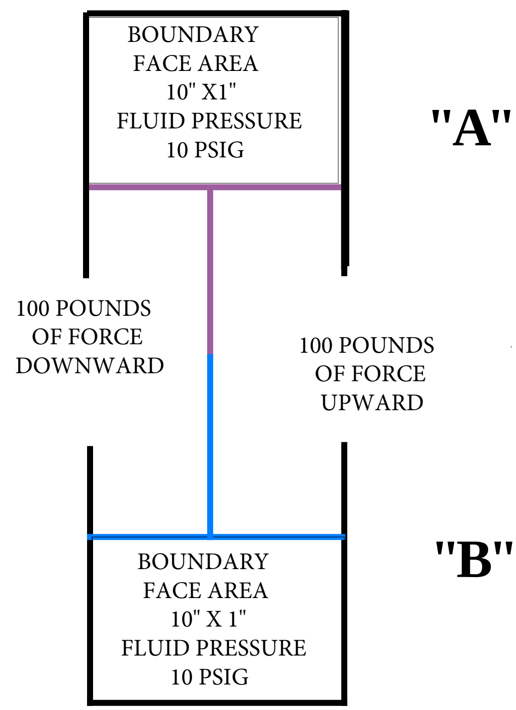


FIGURE 3

FLUID BEING FORCED INTO PISTON "B"
DISPLACES N EQUAL AMOUNT FROM PISTON "A"
INTO CUSH TANK "C"
NOTE: THE PRESSURE IN PISTON "B" IS HIGHER
THAN THE PRESSURE IN PISTON "A"

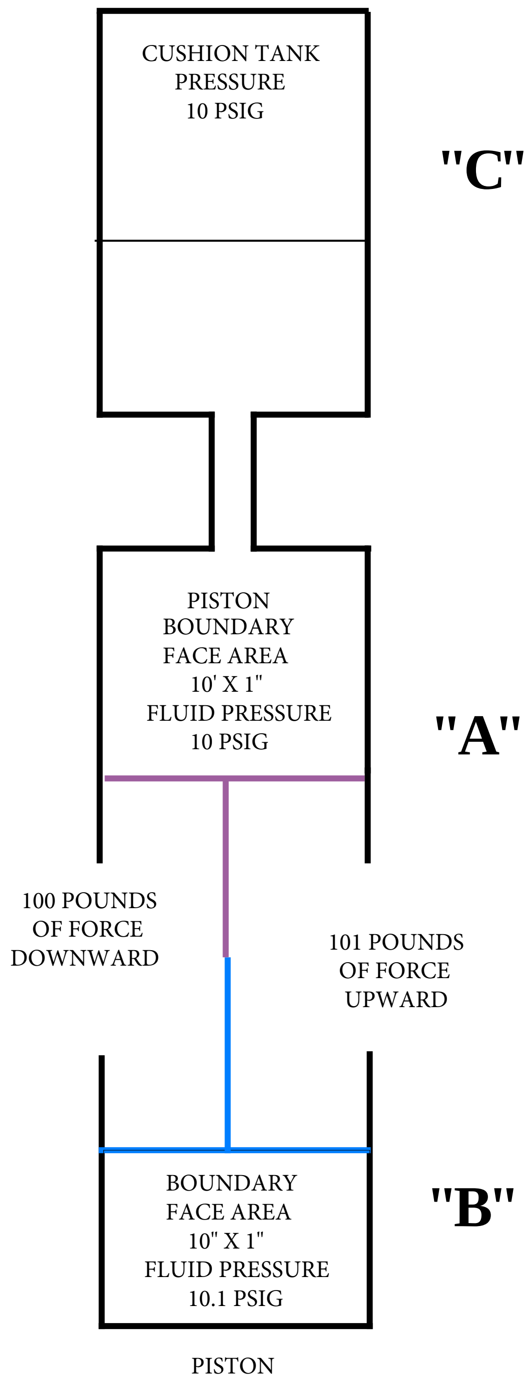
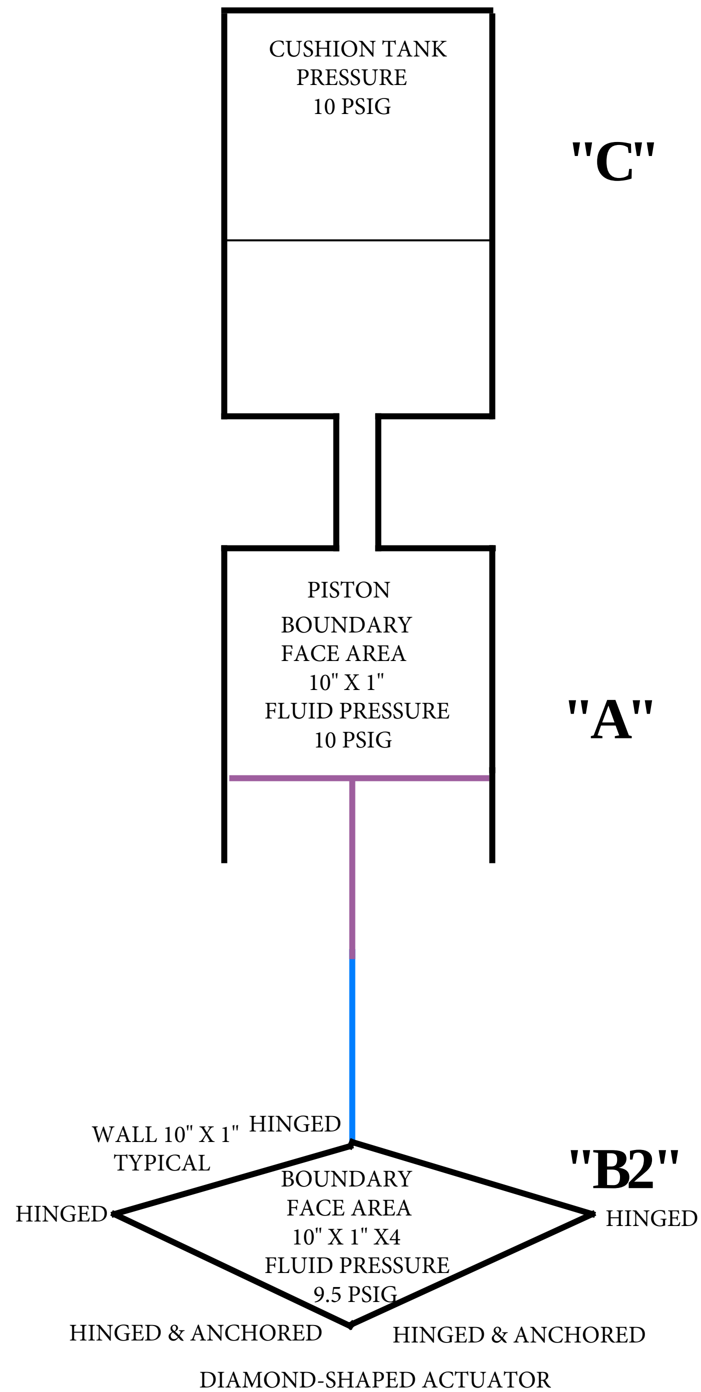
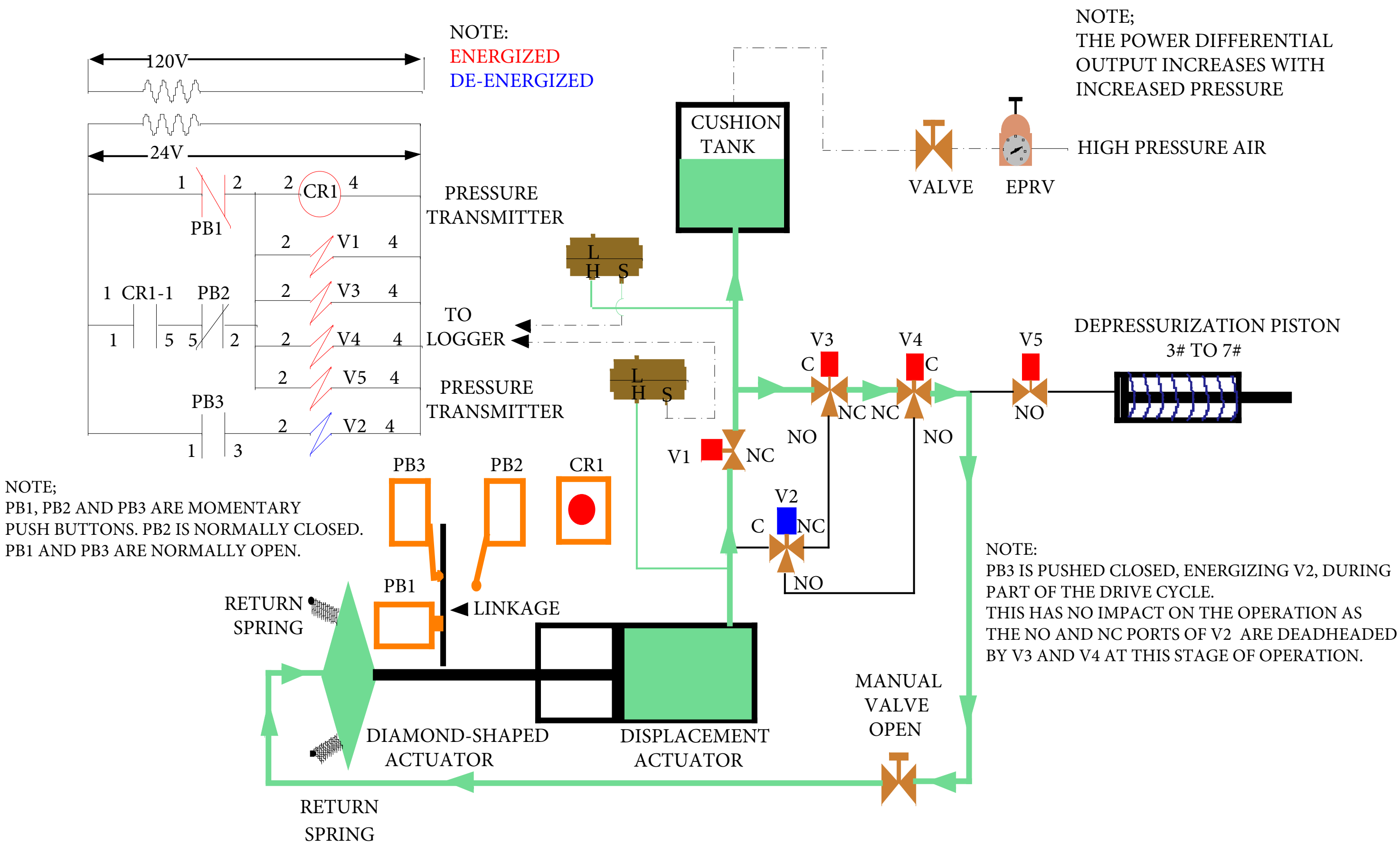


FIGURE 4

FLUID BEING FORCED INTO AVTUATOR "B2"
DISPLACES A GREATER AMOUNT FROM PISTON "A"
INTO CUSHION TANK "C"
NOTE: THE PRESSURE IN ACTUATORE "B2" IS LESS
THAN THE PRESSURE IN PISTON "A"





DRIVING STATE

The driving state is started when the linkage forces push button (PB1) closed to energize V1, V3, V4, V5 and CR1. The contact of CR1 closes to lock power on these devices when PB1 re-opens as the drive movement starts.

When V1, V3, V4, and V5 are energized the flow pattern is as illustrated on this drawing.

The diamond shaped actuator, the displacement cylinder and the cushion tank experience the same pressure.

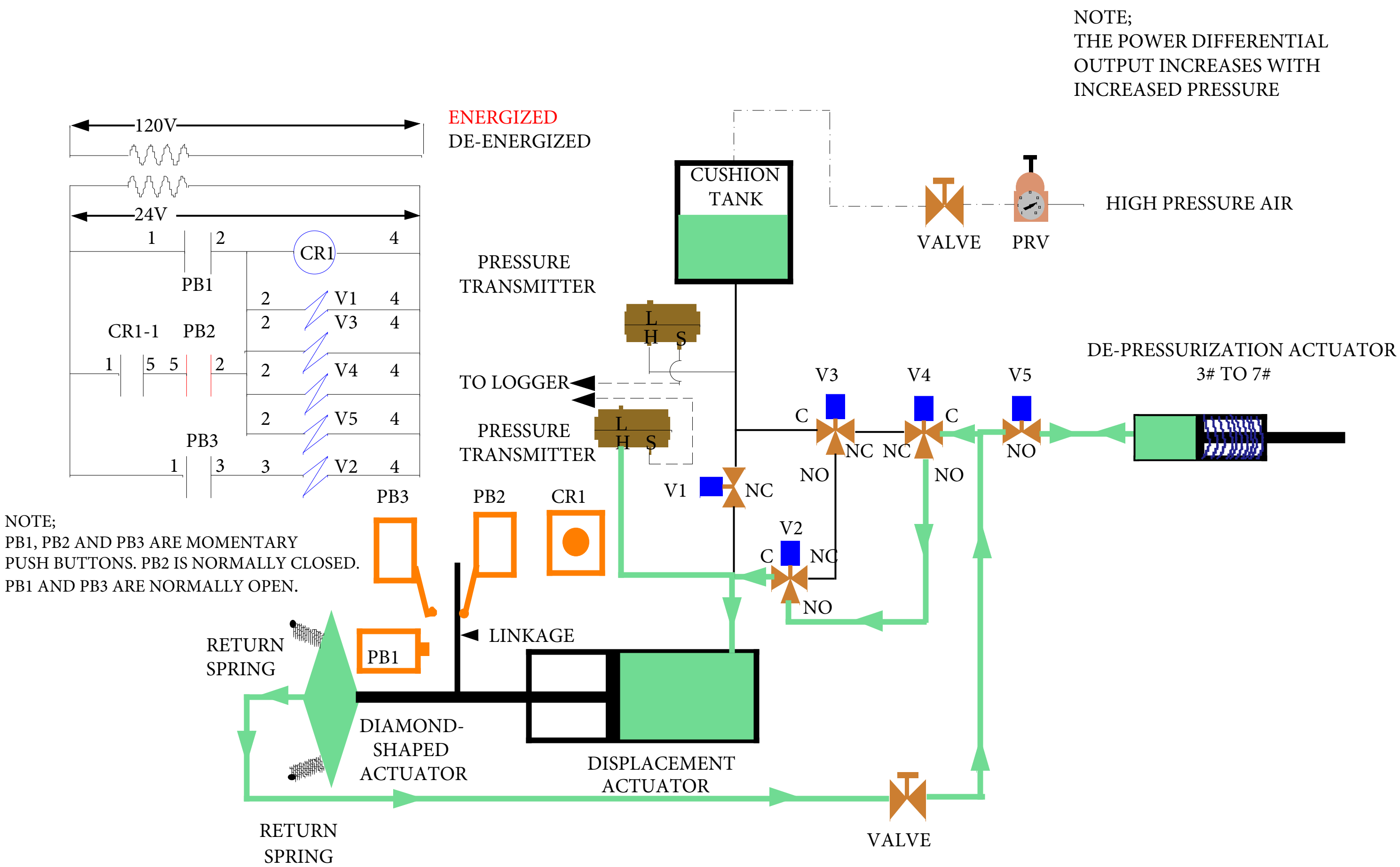
The diamond shaped actuator develops about 10% to 15% greater total force at its tip than the counter force developed in the displacement cylinder.

Each of the four faces of the diamond shaped actuator have the same area as the displacement cylinder's boundary face. This causes more fluid to be driven out of the displacement cylinder than is required to drive the diamond shaped actuator for the same linear travel.

About 99% of the fluid driven out of the displacement cylinder flows to the diamond shaped actuator and about 1% of the fluid flows into the cushion tank.

The differential in force between the diamond shaped actuator and the displacement cylinder may be used to drive any mechanical device external to this machine, such as a generator or pump.

When the linkage forces push button (PB2) open V1, V3, V4, V5 and CR1 are de-energized which causes the driving state to stop.



FIRST RECHARGE CYCLE

The linkage pushes PB2 open, which causes V1, V3, V4, V5 and CR1 to de-energize and the flow pattern illustrated on this drawing is established.

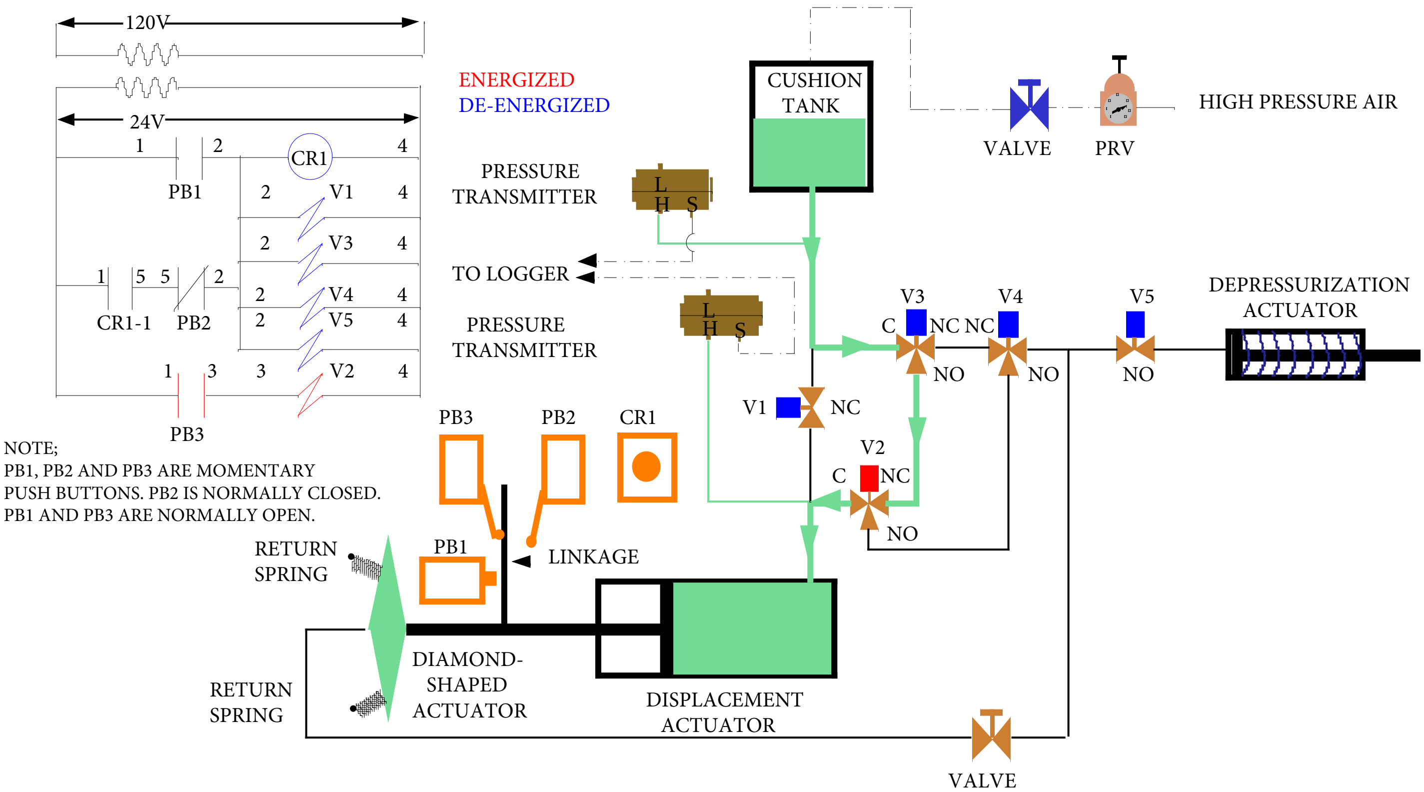
The pressurized cushion tank is isolated.

The diamond shaped actuator, the displacement cylinder and the depressurization cylinder all experience a common pressure.

The depressurization cylinder strokes, which de-pressurizes all three components. This removes the diamond shaped actuator's power advantage over the return springs. The return springs retract to force the fluid in the diamond shaped actuator back into the displacement cylinder.

When the common pressure drops below the spring range of the depressurization cylinder, the fluid is forced into the displacement cylinder from the depressurization cylinder.

When the 99% of the fluid that came out of the displacement cylinder has been returned to the displacement cylinder, the linkage forces push button (PB3) to close which energizes V2.



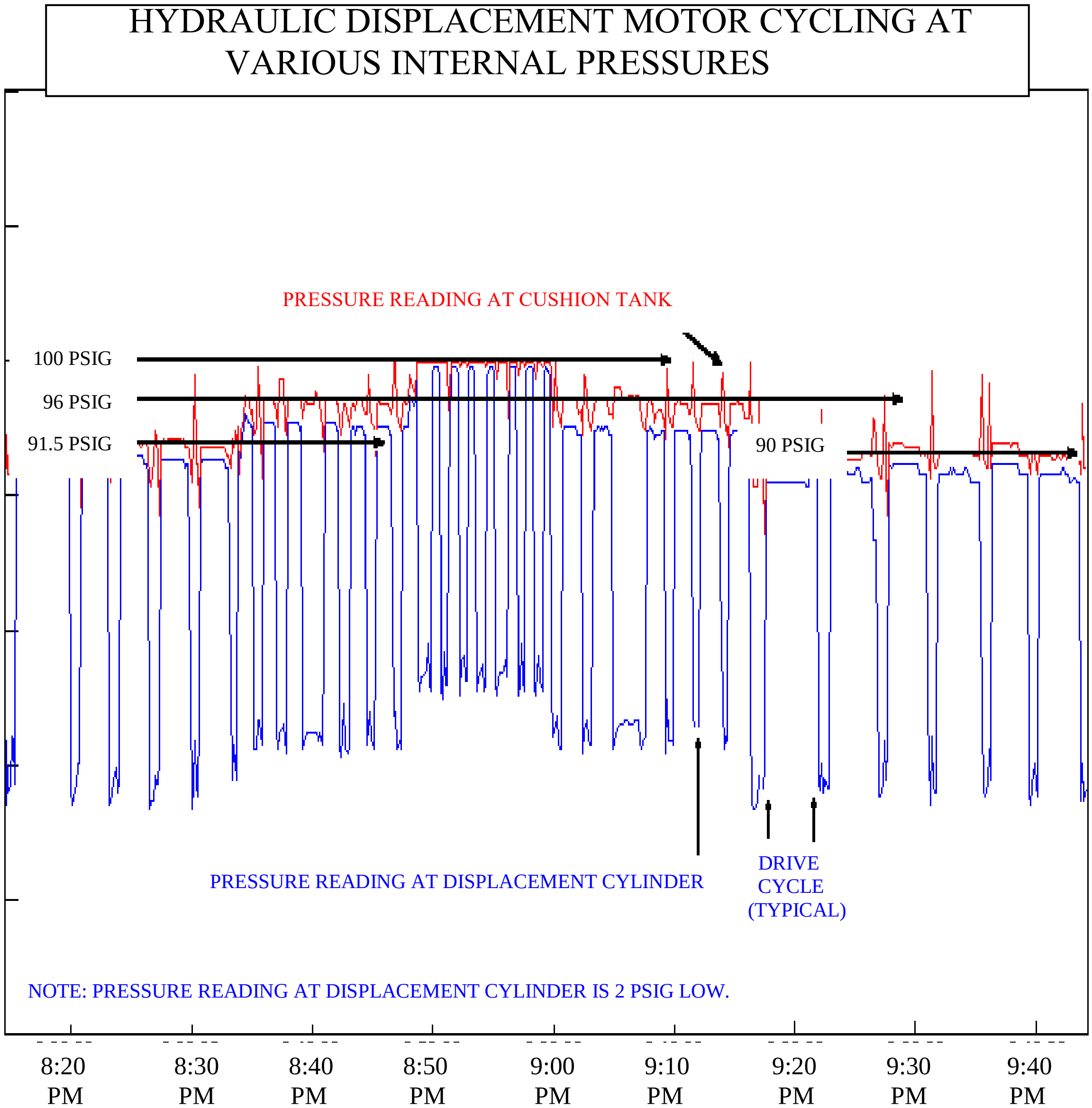
SECOND RECHARGE CYCLE

The linkage pushes PB3 closed to energize V2. This establishes the flow pattern illustrated on this drawing.

The 1% of the fluid that was forced from the displacement cylinder into the cushion tank is forced back into the displacement cylinder from the cushion tank.

When the displacement cylinder is completely refilled, the linkage of the displacement cylinder pushes PB1 closed and the cycle repeats.

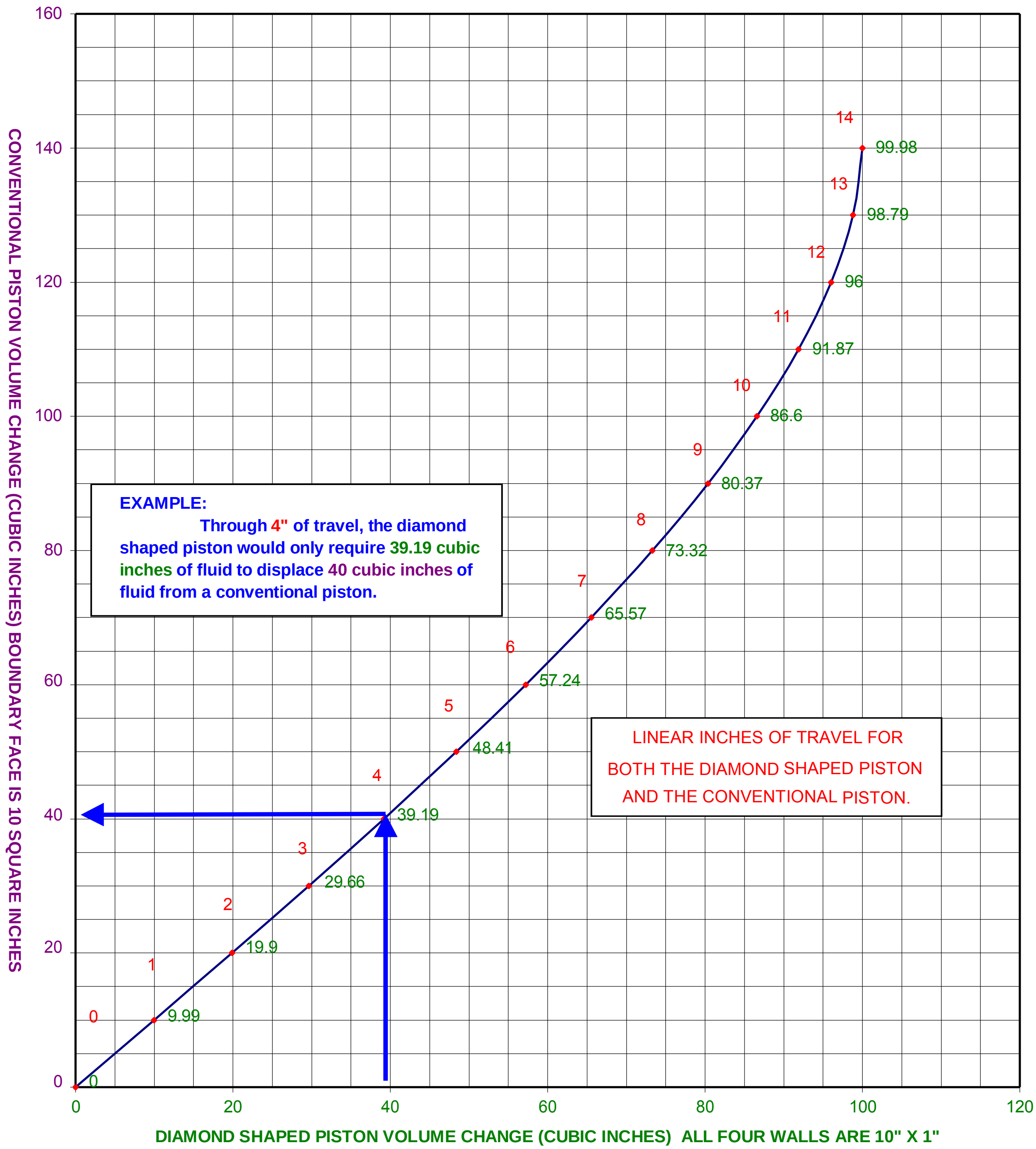
NOTE: The linkage is not mechanically connected to the diamond shaped actuator; therefore, there will be a temporary gap between them during the second recharging stage.



Data was collected at transmitter points illustrated on the previous drawings on pages 13.295, 13.296, and 13.297.

Data extracted from motor as per exact layout of drawing 12 in patent application document.

VOLUME RELATIONSHIP OF A DIAMOND SHAPED PISTON RELATIVE TO A CONVENTIONAL PISTON FOR THE SAME LINEAR TRAVEL



The Laws of Thermodynamics have been accepted without question in the scientific community for about 200 years. But everything else changes, everything else evolves.

We've seen other scientific laws change over time as we learned new information.

What if the Laws of Thermodynamics are outdated?

And what if, by not questioning these old laws, we are preventing innovation that can positively address Climate Change?

Technological advancements that would assist in the battle against climate change are being dismissed by scientists/engineers based solely on the scientific community's unconditional acceptance of the Laws of Thermodynamics.

We have a patented solution to reduce pollution: why no scientific interest?

Design drawings and functional models were made available to many scientists/engineers. Here are the results:

- Most declined the invitation to test the physical models at all
- Most provided no feedback on the actual design, but still rejected the invention based on the existing Laws of Thermodynamics
  - Appendix "A" (page 13.303): The responses from engineers/scientists with respect to the invention; it's all about the laws – nothing about the actual design or models

Unconditional faith in text book opinions is blocking application of real Climate Change solutions.

But some in the scientific community who decided to look at our designs and drawings provided positive feedback…

Some scientists/engineers have tested the models with approval

- *"I believe that Mr. Strain's invention will advance the scientific community's understanding of thermodynamics relating to pressurized fluids and energy to a new level. If fully developed the invention has the potential to reduce energy and as a result a reduction in the use of fossil fuels, thus assisting in the battle against climate change."* ~ Donald M. Gorber PH.D, P. Eng.

- A control systems expert, representing SIEMENS, one of the world leaders in automatic controls, peer reviewed the control logic and initialled the drawings with no changes.

Again, what if accepting the Laws of Thermodynamics without question blocks innovation that can reduce Climate Change?

Here are some questions we need to consider (and ask!) when thinking about our ongoing acceptance of the 200-year-old Laws of Thermodynamics

		YES	NO
1)	Is it true that in the pursuit of absolute truth, some past scientific laws and beliefs that were held as true have been proven to actually be wrong and untrue?	◻	◻
2)	Is it logically possible that some scientific laws and believes we hold now could be wrong?	◻	◻
3)	Is it scientific to update old/wrongs laws and beliefs, to new proven designs and understandings?	◻	◻
4)	Experts often quote the Laws of Thermodynamics when contesting innovation designed to combat climate change. Should they be allowed to simply cite the laws and block progress, without a modern and thorough explanation?	◻	◻
5)	Experts will often recite that energy cannot be created or destroyed. Many of the same experts maintain that in the "Big Bang" time, space, matter and energy came into existence from nothing. Is the contradiction apparent to you?	◻	◻
6)	Many scientists today say that we don't know what energy actually is—which is fundamental to fully understanding the Laws of Thermodynamics. Could the developers of the Laws of Thermodynamics in the nineteenth century understand what energy is in its entirety, yet more technically advanced modern-day scientists do not?	◻	◻
7)	Imagine a hydraulic product exists (Product A) and through innovation, a more efficient version is patented (Product A+). As it turns out, Product A+ can derive its total hydraulic fluid requirement, pumping it from Product A, with work potential remaining in product A+ ($W_{in} < W_{out}$). This outcome contests the Laws of Thermodynamics. Can the Laws of Thermodynamics remain valid with this consideration?	◻	◻

Answered yes to 4, 6, 7 and no to 1, 2, 3, 5? Please send your explanations, we'd love to hear your thoughts.

Answered no to 4, 6, 7 and yes to 1, 2, 3, 5? Then pass this on to scientific experts and ask for urgent support to developing good solutions to address climate change.

Take action AND demand answers and accountability

When people participate in coordinated rallies in cities across the globe, it visually shows collective desire for change, but after a day, everyone disappears from the public eye. To effect real and lasting change, we need to be consistent and persistent in our pursuit of answers and accountability. We need to knock on every door and ask "why" over and over and over again until we hear a real answer.

So, (1) if it's true that the Laws of Thermodynamics are outdated, and (2) that fact is preventing innovation from being used to fight climate change, then the younger generation must demand answers from the governing bodies and professors for their unconditional acceptance of the Laws of Thermodynamics. Innovations must be assessed on whether they produce the desired outcomes or not.

Do not let 200-year-old laws stop innovation. **We must address climate change now.**

THE NEXT PRACTICAL ACTION

Concerned youth should search for an Engineer, Scientist or Professor who believes they can publicly contest the drawings illustrated on pages 13.295 to 13.297, that describes work output greater than work input ($W_{out} > W_{in}$), contesting conventional knowledge. Quoting the Laws of Thermodynamics is an unacceptable faith statement, not a technical argument.

If no technical contesting position is presented in detail, concerned youth should collectively demand this advancement be brought to practical application. The other inventions rejected based only on faith in the Laws of Thermodynamics should be reassessed respecting technical merit, not unsupported faith in the Laws of Thermodynamics.

Relying on the established laws with faith only is a very serious error.

Link to patent offices for subject invention:

https://worldwide.espacenet.com/inpadoc?submitted=true&DB=EPODOC&CC=US&NR=2002178719&KC=&F=8&OREQ=0&textdoc=TRUE&FT=E

Link to You Tube videos presenting the subject invention:

https://www.youtube.com/channel/UCVqNrHjb2nj-wo-h7Shzigw

APS respects the advancement of science through practical methods with the goal of global environmental betterment. So, we ask all scientists to test our designs and drawings based on technical merits and not simply restating the Laws of Thermodynamics, but engage the drawing and designs and come to conclusions based on logical assessments, as the evolvement of science starts with the questioning of science itself. The questioning serves to either strengthen the pre-existing scientific understanding or belief, or it serves to advance science as a whole by changing the pre-existing scientific understanding or belief. So we ask everybody who reads this to be a part of science and the scientific process.

Please send any questions or comments.

Dave Strain
President
APS
analystsofpneumatic@bellnet.ca
Office—(905) 640-2333

Appendix A

Some actual quotes from engineers'/scientists' responses with respect to the invention, based only on their faith in the Laws of Thermodynamics:

A. "You claim that the diamond-shaped actuator produces a greater displacement of fluid at a lower pressure than for a conventional piston-cylinder configuration. From this you conclude that there is an excess of energy output compared to the energy input. Although I confess to only a rudimentary understanding of your principle of operation, I cannot accept the conclusion that this device would produce more energy output than is put in." ...Same letter... "Finally, I must admit that I cannot fully understand your arguments. Regretfully, though, from what I already know I doubt that I could be persuaded that this device is a perpetual motion machine as you claim." (University Engineering Professor and Engineering text author)

B. "Quite frankly, I reject outright any proposal that claims to overcome the basic laws of physics." (University Professor and environmental organization leader.)

C. "As you know, your results contradict the First Law of Thermodynamics. Despite numerous attempts over the centuries to demonstrate otherwise, this fundamental Law has never been violated. In fact, the theory and all available evidence support this Law so strongly that we hold it as true. All modern science and technologies are developed and work based on the truth of this Law. Examples of experiments, demonstrations, and machines developed and/or built that purportedly invalidate this Law, have, without exception, all been shown to be false." (University Professor and director of applied technology at that University.)

D. **"Discussion and conclusion** [*TO BE COMPLETED*]

Figure 7 compares the estimates of equation 5 with the data of Ref. 2. As compared with the calculated results including the effects of both actuator and fluid weight, the measured pressures are significantly lower, being about 26% low at z/L = 0.25, decreasing to 19% low at z/L = 0.42.Since the possible effects of such sources of discrepancy as solid and fluid friction might be expected to increase the pC required to lift the weight—at least in the process of raising the it—this appears to support Mr Strain's claim."

"To my knowledge, in two centuries of scientific development there has never been a case in which the laws of classical physics when used in their proper sphere of applicability have been violated. I have 50 years of direct personal experience using these laws, leading me to believe that I can no more question them than I can make two plus two equal five, and thus improve my financial affairs."(University Engineering Professor and renowned scientist.)

NOTE: The differential test model proves that less than 4% efficiency differential is required for the diamond-shaped actuator to produce its full fluid requirement by pumping the fluid from a conventional piston. This fact leaves 22% remaining work potential in each operation, allowing a reciprocating machine to run itself and produce external mechanical work with no external energy input.

E. "I chose to write to you on my personal, rather than business stationary. I figure that you get enough of the formal business stuff. Most importantly, you should understand that in building your piston system, you conformed to the laws of Newtonian Mechanics. Those laws helped you build something better than was before, but at the same time, restricted you to the valdity of CPE. More simply, you can't have your cake and eat it too. Take it all, or take none of it. Them's the rules." (Scientist used by the NRC to assess "out of the box" inventions)

F. "I have understood novelty of the diamond-shaped actuator and its potential advantages. Also, I do not question validity of experimental data that you and Mr. Blanchard collected. Quite contrary, I trust that the experiments and all measurements were conducted professionally. Our views differ with regard to the system that uses the diamond-shaped actuator to produce useful mechanical work without spending any external energy. Even in this case I do not challenge validity of your experimental data (that is, I trust that the system worked at some point). I only think that the data is incomplete -- some part of the process went unnoticed. Thus, I trust that the system can work, but you/we do not have a proper explanation for how it works." (NRC scientist)

Quotes A to F are typical of comments based on unconditional acceptance in the Laws of Thermodynamics.

Not one has provided any scientific or technical logic to support their position specifically addressing the invention's design drawings they possessed.

SIEMENS, via one of their controls experts, peer reviewed the detailed control drawings and initialled the drawings with no changes, confirming the control circuits. Pages 13.305 to 13.310 present the SIEMENS assessment. Pages ten to fifteen of the link below present the same, plus other scientists/ engineers assessments in the report.

Patent examiners from the PCT (Patent Co-operation Treaty), European, USA and Canadian patent offices examined the design data and granted the patents.

Rejection should be based on technical assessment of the actual invention, with supporting arguments.

https://static1.squarespace.com/static/5ee6829b4abd4867f862c3ca/t/61e6fb5a7ea4d658250bc320/1642527582218/SCIENTIFIC+AND+ENGINEERING+OBSERVATIONS+AND+OPINIONS.pdf

Analysts of Pneumatic Systems Limited (APS)
35 O'BRIEN AVENUE - STOUFFVILLE, ONTARIO L4A 1G6
TEL: (905) 640-2333 FAX: (905) 640-2444

Siemens,
2 Kenview Blvd.,
Brampton, Ontario,
L6T 5E4
Att.- Mr. John Osmond
February 10, 2004

Dear Sir;

As per our discussion, we have invented a more efficient hydraulic actuator than the conventional cylindrical piston. The new actuator achieves **more** work than a conventional piston with equal fluid pressure, at a lesser fluid volume.

The efficiency differential gained acceptance by the scientific communities who have assessed the invention; however, understanding the main application of the new actuator requires one to possess a reasonable knowledge of control circuitry. We have built models proving the claims; however, a written expert collaborating opinion is required regarding the function of the illustrated control and fluid circuitry.

We request that Siemens evaluate the three drawings illustrating:

(1) the driving cycle,
(2) the first recharge cycle
(3) and the second recharge cycle.

Two facts are required to be assumed true:

(1) The diamond-shaped actuator develops approximately 10% to 15% more force than a conventional piston with an identical fluid source. Investigating scientists have accepted this fact.
(2) The force differential is more than adequate to over come all frictional losses in the running machine.

The accompanying graph illustrates the internal pressure changes of the model running over two hours at various fixed static pressures. The exact model that produced this graph is illustrated on the last page.

The basic question presented is: "If the differential in efficiency exists, will the mechanical system fluid patterns and control circuit perform as per the illustrations on the three accompanying CAD drawings?"

A meeting with the person performing the analysis may simplify their task.

Thank you for your consideration.

Sincerely,

Dave Strain

SIEMENS

May 06, 2004

Analysts of Pneumatic Systems Limited (APS)
Attention: Mr. Dave Strain
35 O'Brien Avenue
Stouffville, Ontario
L4A 1G6

RE: Diamond-shaped actuator

Your letter dated February 10,2004 requested a review of the control drawings by a controls expert. With 35 years industry experience, I feel I am qualified to do that evaluation. The evaluation is based on the diagrams supplied and the discussions in our office, March 31,2004 with Len Shaddock (Service Manager) and Mitch Proude (National Operations Manager SBT). It should be noted that we did not witness a live demonstration of the circuits depicted on the control drawings. No conclusions are made regarding the fluid flow. For this exercise it will be assumed the flows are as described and the switches are activated by these flows. For this to work, a 100 PSI air source and a 24volt AC power supply are required.

Drawing (Driving state)

The cycle is initiated by manually opening the isolation valve between the PRV and the cushion tank. When switch PB1 is closed,V1, V3, V4, V5 and CR1 are energized. CR1 latches through Normally Closed switch PB2. The valves switch positions as described.
It is not mentioned, but given the location of PB3, it is closed at this time, by the linkage and valve V2 will be energized for part of the "Driving State". PB3 closes before PB1 as the linkage travels to the left. When PB2 opens, V1, V3, V4, V5 and CR1 are de-energized. The valves return to their normal positions. (Start of "First Recharge Cycle")

Drawing (First Recharge Cycle)

Switch PB2 opens causing V1, V3, V4, V5 and CR1 to de-energize. The valves will return to their normal positions. When PB3 closes, valve V2 will be energized. (start of "Second Recharge Cycle")

Drawing (Second Recharge Cycle)

When PB3 closes, valve V2 will be energized

Drawing (Fig. 12)

This is supposed to be a depiction of a complete working model. This drawing shows an additional external air source and associated circuitry that is not mentioned on the "Driving State', "First Recharge Cycle' or the "Second Recharge Cycle" drawings..

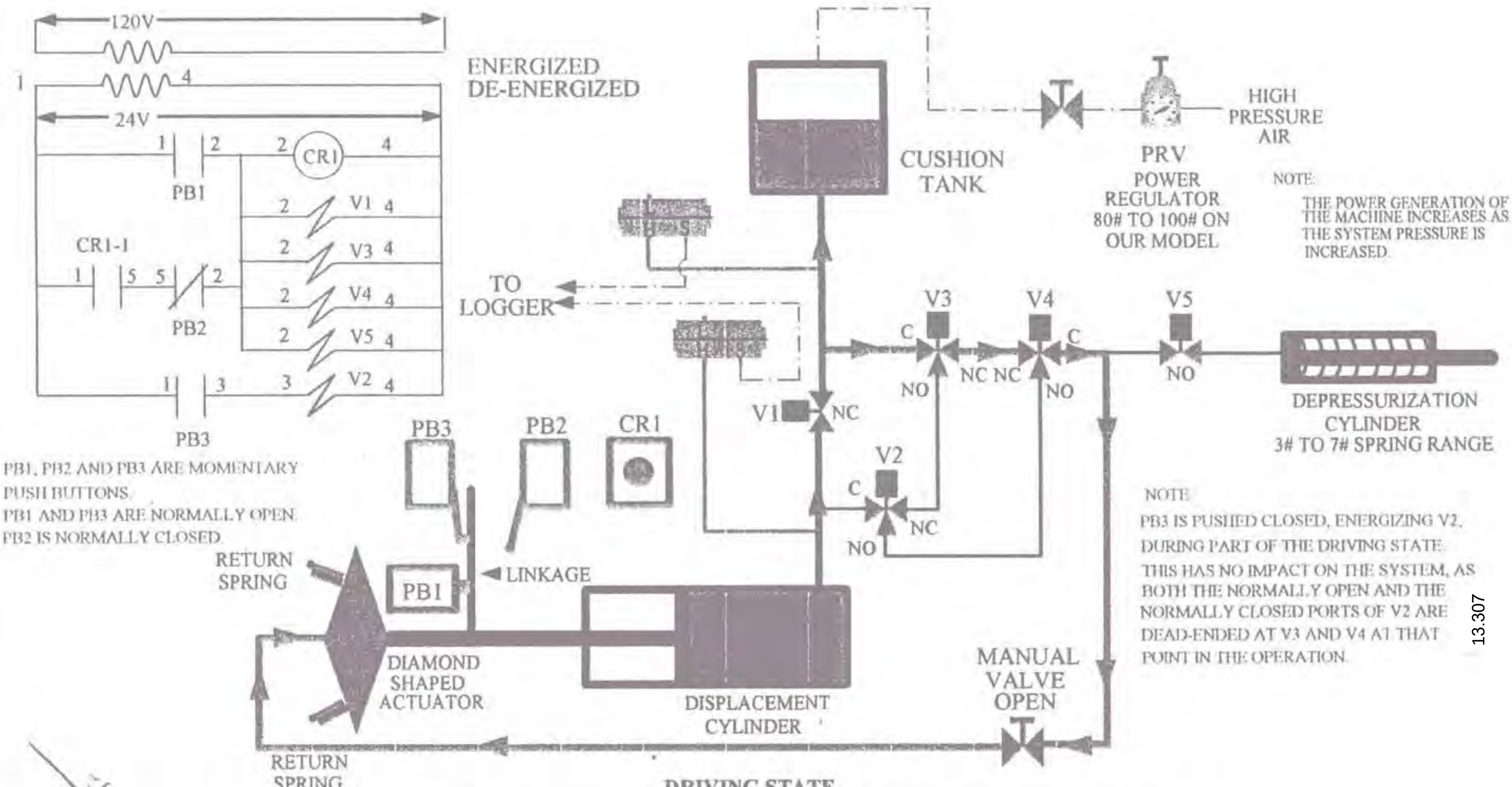
Please call if you have any questions.

Ed Komadowski
Service Operations Supervisor
cc: Len Shaddock, Mitch Proude

Siemens Building Technologies, Ltd.

2 Kenview Blvd.
Brampton, Ontario L6T 5E4
Canada

Tel: (905) 799-9937
Fax: (905) 799-9277
Toll Free: (800) 268-6831



MAY 06, 2004

DRIVING STATE

The driving state is started when the linkage forces push button (PB1) closed to energize V1, V3, V4, V5 and CR1. The contact of CR1 closes to lock power on these devices when PB1 re-opens as the drive movement starts.

When V1, V3, V4, and V5 are energized the flow pattern is as illustrated on this drawing.

The diamond shaped actuator, the displacement cylinder and the cushion tank experience the same pressure.

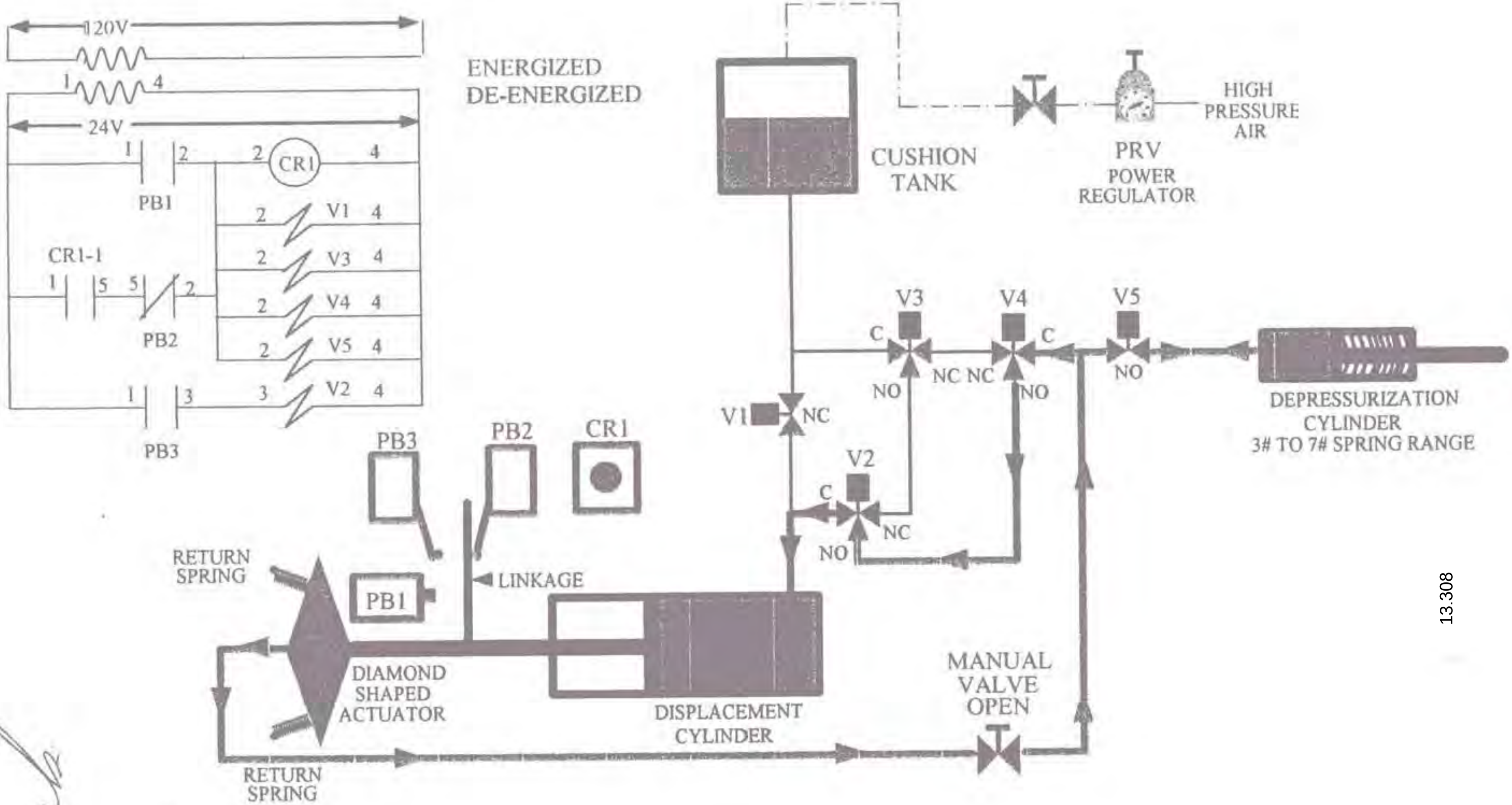
The diamond shaped actuator develops about 10% to 15% greater total force at its tip than the counter force developed in the displacement cylinder.

Each of the four faces of the diamond shaped actuator have the same area as the displacement cylinder's boundary face. This causes more fluid to be driven out of the displacement cylinder than is required to drive the diamond shaped actuator for the same linear travel.

About 99% of the fluid driven out of the displacement cylinder flows to the diamond shaped actuator and about 1% of the fluid flows into the cushion tank.

The differential in force between the diamond shaped actuator and the displacement cylinder may be used to drive a generator external to this machine.

When the linkage forces push button (PB2) open V1, V3, V4, V5 and CR1 are de-energized which causes the driving state to stop.



FIRST RECHARGE CYCLE

The linkage pushes PB2 open, which causes V1, V3, V4, V5 and CR1 to de-energize and the flow pattern illustrated on this drawing is established.

The pressurized cushion tank is isolated.

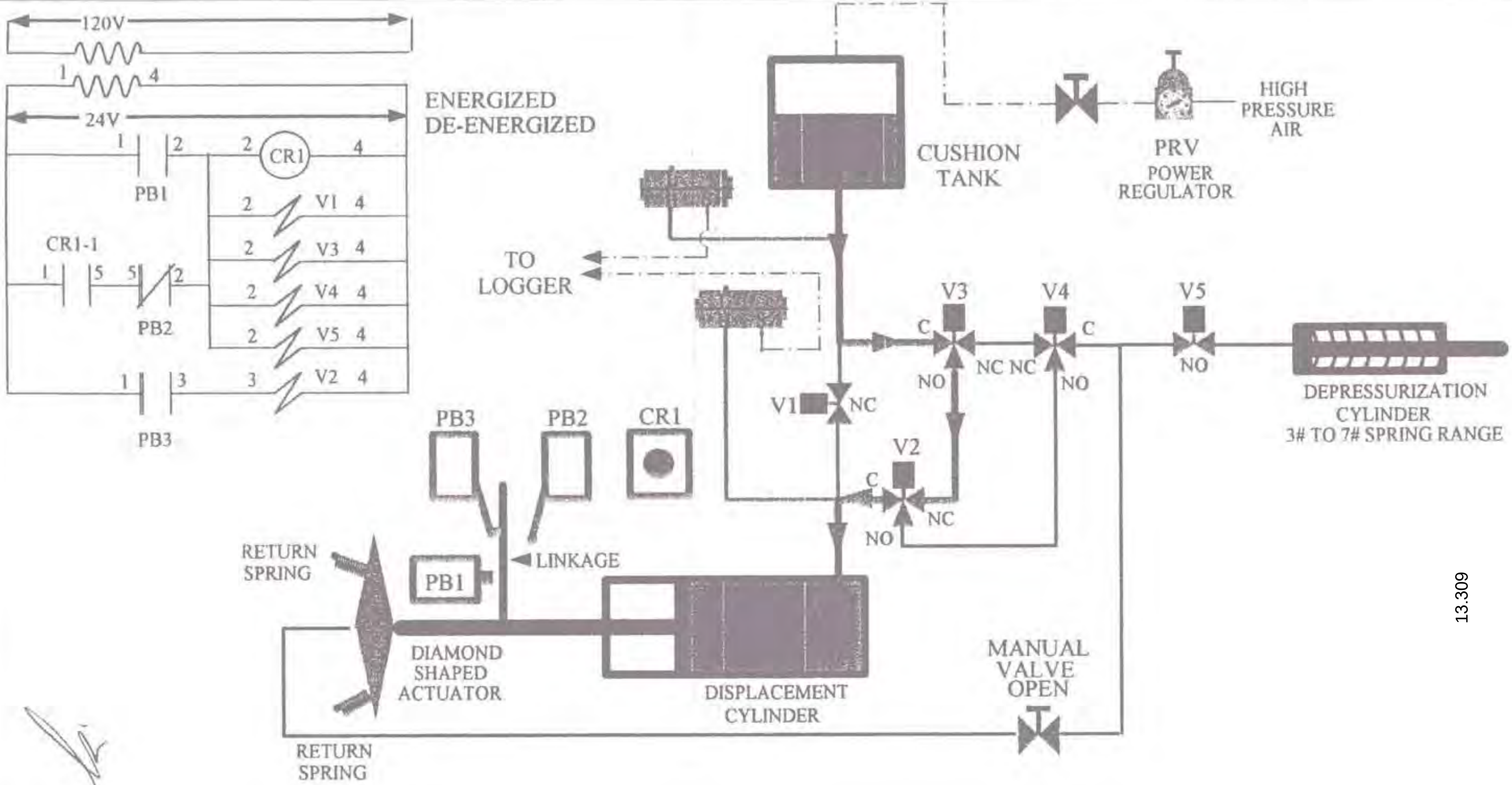
The diamond shaped actuator, the displacement cylinder and the depressurization cylinder all experience a common pressure.

The depressurization cylinder strokes, which de-pressurizes all three components. This removes the diamond shaped actuator's power advantage over the return springs. The return springs retract to force the fluid in the diamond shaped actuator back into the displacement cylinder.

When the common pressure drops below the spring range of the depressurization cylinder, the fluid is forced into the displacement cylinder from the depressurization cylinder.

When the 99% of the fluid that came out of the displacement cylinder has been returned to the displacement cylinder, the linkage forces push button (PB3) to close which energizes V2.

MAY 06, 2004



SECOND RECHARGE CYCLE

The linkage pushes PB3 closed to energize V2. This establishes the flow pattern illustrated on this drawing.

The 1% of the fluid that was forced from the displacement cylinder into the cushion tank is forced back into the displacement cylinder from the cushion tank.

When the displacement cylinder is completely refilled, the linkage of the displacement cylinder pushes PB1 closed and the cycle repeats.

NOTE: The linkage is not mechanically connected to the diamond shaped actuator; therefore, there will be a temporary gap between them during the second recharging stage.

MAY 06, 2004
Edmond Kmadourk

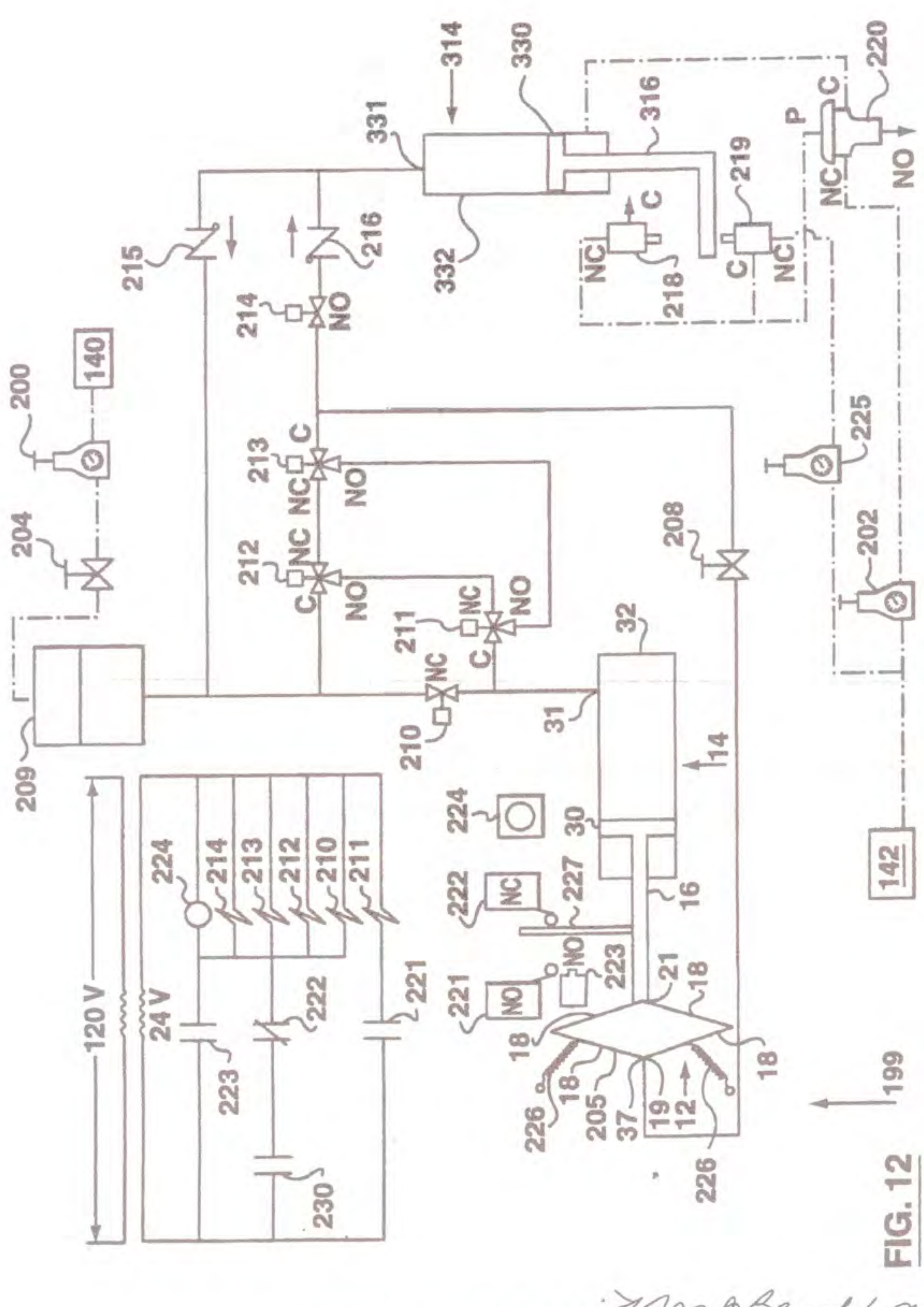


FIG. 12

SUBSTITUTE SHEET (RULE 26)

MAY 06, 2004

Hydraulic Displacement Motor

Accepting this patented invention requires a person to think logically, not just recite physical laws. Please challenge any or all points with logic, not unsupported faith in textbook laws. (analystsofpneumatic@bellnet.ca)

Patent office link:

https://worldwide.espacenet.com/inpadoc?submitted=true&DB=EPODOC&CC=US&NR=2002178719&KC=&F=8&OREQ=0&textdoc=TRUE&FT=E

Some scientists/engineer positive peer reviews on pages 13.276 to 13.282.

> *"I believe that Mr. Strain's invention will advance the scientific community's understanding of thermodynamics relating to pressurized fluids and energy to a new level. If fully developed the invention has the potential to reduce energy and as a result a reduction in the use of fossil fuels, thus assisting in the battle against climate change." Donald M. Gorber, Ph.D., P.Eng., President, SENES*

(NOTE: Patent examiners in the PCT, European, Canadian and USA offices passed it.)

- The fundamental science explained on pages 13.233 to 13.237. (Also page 13.254)
- System application drawings and performance graph on pages 13.255 to 13.258.
- The invention challenges the Laws of Thermodynamics; therefore, I suggest a thought process to determine if you accept the Laws of Thermodynamics based on faith or scientific proof on pages 13.300 to 13.304.

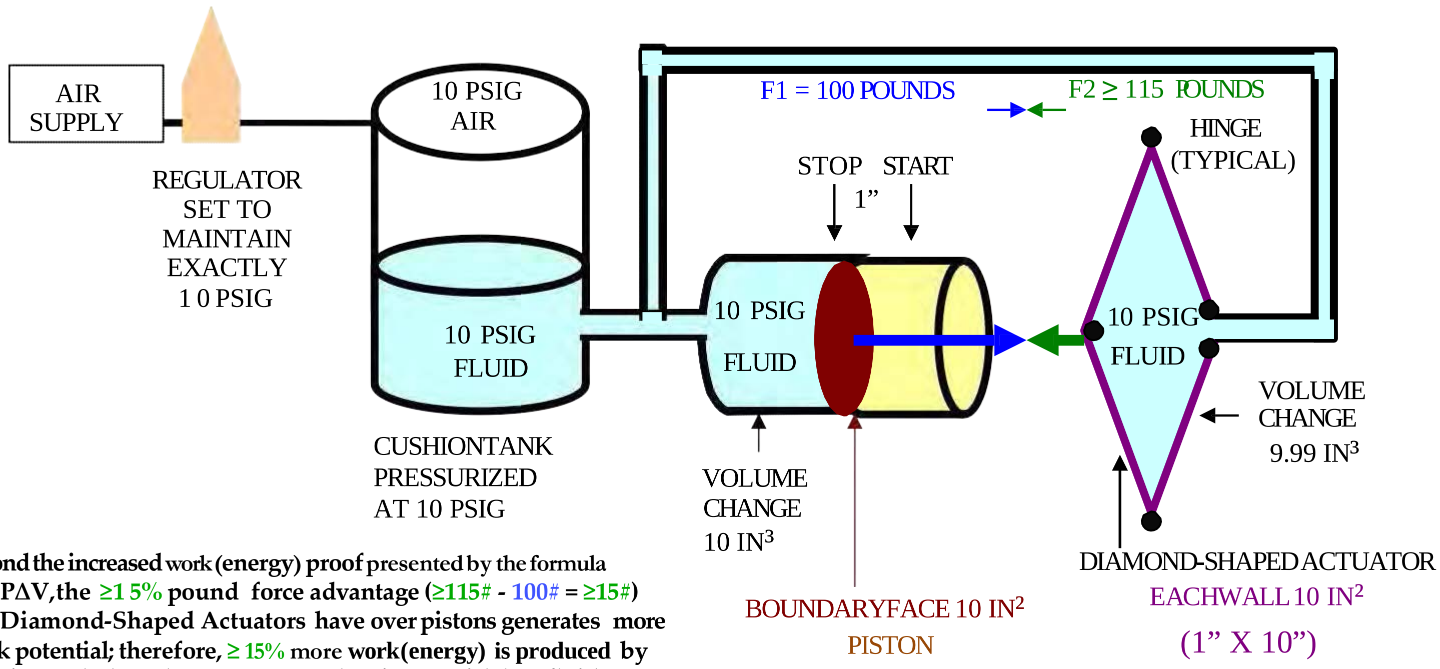
A work increase assessment using formulae $W = P\Delta V$ and $W=FD$ is presented on page 13.312.

A suggested logic path for concerned persons is presented on page 13.313.

A suggested action plan is presented on page 13.314.

THE DIAMOND-SHAPED ACTUATOR HAS BEEN VERIFIED BY SCIENTISTS
(PAGES 13.276 - 13.282)
TO BE ≥15% MORE EFFICIENT THAN CONVENTIONAL PISTONS

WORK OUTPUT > WORK INPUT ($W_{OUT} > W_{IN}$) USING THE FLUIDIC FORMULA W= PΔV



Beyond the increased work (energy) proof presented by the formula W= PΔV, the ≥1 5% pound force advantage (≥115# - 100# = ≥15#) that Diamond-Shaped Actuators have over pistons generates more work potential; therefore, ≥ 15% more work(energy) is produced by the Diamond-Shaped Actuators over the pistons with less fluid volume change at equal pressures.

INTENT

1) The formula W = PΔV (Work = Pressure times volume change) is used to determine the work potential of a volume of fluid.

2) In the above presentation the work input is based on the pressure and volume change of the Diamond-Shaped Actuator.

3) In the above presentation the work output is based on the pressure and the volume change of the piston.

4) The work input = 10 PSIG x 9.99 in^3 = 99.9 in-lb

5) The work output = 10 PSIG x 10.0in^3 = 100 in-lb

6) 100 in-lb > 99.9 in-lb

7) $\therefore W_{out} > W_{in}$ via the fluidic formula W = PΔV

NOTE: The differential of $W_{out} > W_{in}$ is much more than indicated by application of the fluidic formula W = PΔV.

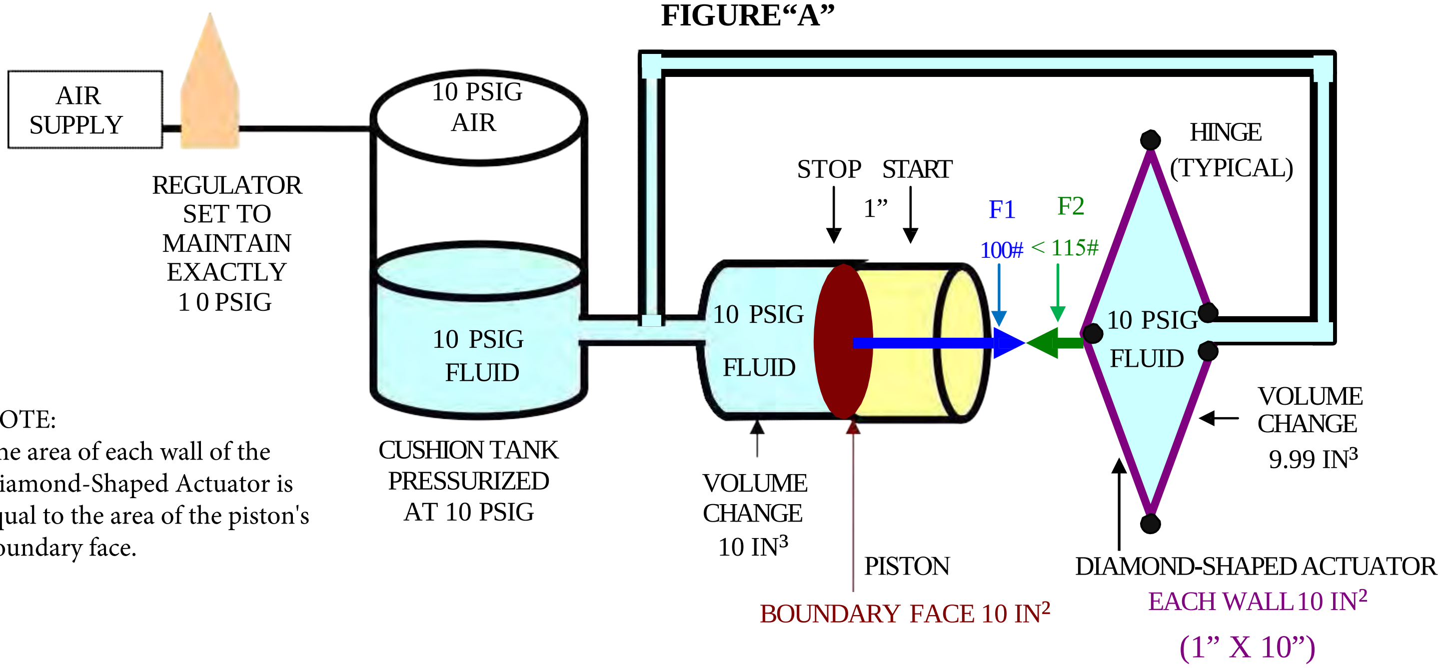
1) The geometrics of the piston boundary face and Diamond-Shaped Actuator walls creates a work increase > 15%, via a mechanical advantage. (Peer review presented in pages 13.276 to 13.282.)

2) Less than 5% advantage is required for the Diamond-Shaped Actuator to pump its total fluid requirement from the piston. (Demonstrated by the physical model on page 13.284)

3) This arrangement leaves >10% surplus work in each forward stroke.

ONE CLIMATE CHANGE SOLUTION REQUIRING CONCERNED CITIZEN ACTION

FIGURE"A"



NOTE:
The area of each wall of the Diamond-Shaped Actuator is equal to the area of the piston's boundary face.

IMAGE 1 (MODELS ARE AVAILABLE FOR TESTING) IMAGE 2

PATENTED CLIMATE MITIGATION INVENTION LOGIC

1) When two forces are opposed, the greater force overpowers the weaker force as per FIGURE"A". (> 115# > 100#)
2) The Diamond-shaped actuator is > 15% more efficient than a conventional piston as tested by Dr. Gorber. Dr. Singh, Dr. Ostojic, Dr. Sullivan, Robert Blanchard P.Eng., using the test model presented in IMAGE 1.
3) As per FIGURE "A", the Diamond-Shaped Actuator's force "F2", opposing the piston's lesser force "F1", pumps fluid from the piston in a volume fully satisfying the fluid requirement of the Diamond--Shaped actuator. (10 in³ > 9.99 in³)
4) IMAGE 2 presents the test model proving < 5% greater opposing force is required to pump fluid from a piston.
5) As the Diamond-shaped actuator produces > 15% more force through the identical stroking range, the surplus work potential is > 15% - < 5% ≥ 10% in each stroke.
6) CONCLUSION: Work input is less than Work output. ($W_{in} < W_{out}$)
7) A reciprocating machine's patents and design is presented, that produces completely clean mechanical work with no external energy input in the WEB site http://www.apscontrols.ca (Contact information is on the WEB site.)
8) This logic has been arbitrarily dismissed by scientists from multiple agencies with no supporting logic addressing the provided design drawings that explain the machine.
9) Persons concerned with seeking climate change solutions should request logical and scientific explanations from persons dismissing this solution.
10) Challenging the Laws of Thermodynamics should be accomplished via organized groups, not individuals, as response from those not questioning the laws is sometimes very harsh once the responder realizes they have no scientific basis or logic to support their belief, other than quoting the laws.

NO ACTION: NO RESULTS!!

CONCERNED PERSONS' ASSISTANCE REQUIRED IN ADDRESSING HVAC NEGATIVE IMPACT ON CLIMATE CHANGE

Why Care:

- Based on the BBC survey (14/09/21) on the psychological impact of global warming on our youth, the majority often worry about the issue. (page 11.189)
- A recent news report stated that more than 60% of Canadian adults are very concerned with global warming.
- We are damaging the planet rather than respecting the planet.
- HVAC design is available, which is being dismissed, that could significantly improve the HVAC industry's effort in addressing climate change.
- The advancements in this manual have been, ignored and dismissed by government and private scientist/engineers with no logic addressing the detailed designs provided.
- The logic advancements should be applied, where appropriate, by well trained field personnel in cooperation with well trained designers. Old logic is still being applied.

1) HVAC energy performance improvements:

- Pages 6.44 to 6.47 present sample energy reductions for buildings calculated by the owners.
- Some examples of illogical control systems hiding energy waste are presented on pages 5.38 to 5.43A.
- Case studies pages 8.63 to 8.159 present some case studies focusing on logic advanced over much of current industry approaches.

2) COVID/HVAC ventilation climate change negative impact:

- Available scientific data implies that **live** COVID virus cannot be carried through the HVAC fans from an infected room to other rooms served by the particular HVAC system as **live** COVID virus.
- More research is required to determine if **live** virus from an occupied room can travel through the HVAC fans and be distributed to the other occupied rooms as **live** virus.
- Altering HVAC systems from Demand Ventilation Control (DVC), which is applied conservation logic, to 100% fresh air on fans that are designed to reuse heat already in the building increases the fan's GHG emissions by up to 700%. The same fan also increases direct heat injection into the atmosphere by up to 670%.
- Section eleven from pages 11.187 to 11.209 presents the logic supporting these claims.

POSSIBLE ORGANIZATIONAL ACTION:

- Organize in student bodies, unions, environmental groups, etc. to understand the negative global warming impact of inefficient HVAC logic and the opportunities that are being dismissed.
- As a united body, approach universities, governmental officials (Minister of Energy, Minister of Environment, MPPs and MPs) and media environmental writers asking why the negative global warming approaches are being applied and advances ignored in HVAC systems.
- Identify and call out organizations presenting themselves as environmentally responsible, seeking social credit in addressing the requirements of environmental improvement, while providing little to nothing. They are gaining unjustified credit off our youth's future.
- Understand the issue and share your knowledge that significant improvement is available with others; however, **real change will likely be driven via the passion and energy of the YOUNG.**

STUDENT PROTESTS, REGARDING GLOBAL WARMING, WITH LOGICAL REQUESTS, SHOULD BE MORE EFECTIVE!!!

<u>ORIGINAL SOURCE OF KNOWLEDGE</u>
<u>DEVELOPING THE MOTOR TECHNOLOGY</u>

When I was twenty-two years old, Dr. David Suzuki was predicting environmental disaster within twenty-five years, if mankind did not learn to respect the natural environment. Being a new father, I became extremely concerned with the threat to the planet. Literally, I prayed to our Father, via Jesus, for the knowledge to prevent the environmental disaster. Although, I had no mechanical experience or education, God provided a mechanical design addressing the request. In 1972 I sent a registered letter to myself with the conceptual design of the invention and the intended use of the financial benefits (Pages 14.319 and 14.320). Hoping to donate the invention to Pollution Probe, I found no interest there. The next twenty some-odd years were spent on developing the control logic ability to support the invention. The registered letter was opened in 1998 to see what it contained. The enclosed drawing clearly demonstrated the technical intent and the financial benefits were to help the natural environment and children.

Most of the next thirty years from 1971 were spent attempting to build a model that would prove the concept. The guilt of not bringing the invention on line soon enough caused me to approach several engineers to seek assistance. Each time they became bogged down with the "impossible" and did not respond positively.

Around 1995 concern increased with where I should go with the motor. I did not know if it was going to do more damage than good; therefore, I prayed for direction. Praying that if what I had asked was not best, then prevent it from becoming a reality, but if it was the correct request; then let it happen. The King James Bible was used to seek guidance from God.

I opened the King James Bible randomly, with a verse number in mind, to see what message it would contain. It said "Go forth and speak of your good works on behalf of the Lord." Initially I was awed by the verse. The argument was presented that it could have been a coincidence to turn to that page and that verse. A few weeks later a couple of Mormon missionaries came to our door in Pickering and wished to lend me the Book of Mormon for a week. This had never happened before. I felt that this was a sign to repeat the same exercise as with the King James Bible; therefore, randomly thinking of a verse number I opened to a random page in the Book of Mormon. The verse said "Go forth and speak of your good works on behalf of the Lord." A contesting opinion was presented that the Book of Mormon was identical to the King James Bible and I subconsciously remembered the verse number and the exact page to randomly open the Book of Mormon, repeating the exact exercise as done with the King James Bible. The two missionaries came back the next week. Returning the Book of Mormon, I thanked them.

The occurrences gave me much to ponder. A few weeks later, two different Mormon missionaries came to a summer home my wife and I were restoring and said they wanted to give me a Book of Mormon. I felt that this was an indication to read the Book of Mormon. That book is still in our office. It is a completely different lay out from the King James Bible. The rationalization was completely incorrect that I subconsciously remembered the verse number and page number of the King James Bible, leading to the same wording in the Book of Mormon, as the books are very different. These were the only two visits from Mormon missionaries in my life.

Through God allowing me to improve sequential logic in many types of Engineered building systems, our work has generated cost avoidance, in energy consumption, of roughly one hundred million dollars. The new sequential logic developments God allowed in energy conservation, is the same type of sequential logic required in developing the control circuitry relating to the first patented invention. Along with the multiple unpatented HVAC inventions, He has led me to develop another invention patented under USA patent number US 7,467,517, B2.

The cost of lawyers, my redirected time from the business, hired help, machining and parts has been quite expensive, driving the business close to bankruptcy on a few occasions.

On one of those occasions I was sitting at my desk looking at a letter from our patent lawyers advising that the next bill from the European legal associates would be in the $28,000.00 range. The overdraft on the account could not stand such a bill at that point and the company was at a very low point financially. With almost no reasonable place to go, I prayed to Jesus stating that I knew He did not like divorces, but the cheque to the lawyers' had to be written from the line of credit on our house and my wife, Susan, would not likely tolerate that. Susan was working on the books in the other room at the time and as soon as I committed to paying the bill, via our house line of credit, Susan called out "Why did you not enter this money into the account?" I went to see what she was talking about. I had gone to the bank on four different occasions in the previous months and deposited money without recording it in our books. The amount was about $28,000.00.

At another point, one month after my wife, Susan, died from cancer, the business was in the worst financial condition it had seen in over thirty-six years of business. The overdraft was as deep as it could go and the accounts receivable were at an all time low. I went to the super mail box, hoping for some early paid cheque. On opening the box nothing was there. I was done. Turning to go back to my van I smiled and looked up saying "Well Jesus, you're the accountant now." I had not walked two steps when the cell phone rang. Fred, one of the men who works with us, called informing me that a client's accounting department had given him a cheque to take to the office. It was for about $47,000.00. This completely relieved the financial pressure of the moment. Fred had been with us about twenty years at that point and that was the first time that any client, had given him a cheque to take to the office.

Respecting a statement from Jesus, John 15:5 (King James Bible), "*I am the vine, ye are the branches: He that abideth in me, and I in him, the same bringeth forth much fruit: for without me ye can do nothing*.", I know clearly that my personal abilities could produce nothing of what has been achieved through my hands if God did not provide the required wisdom, protection and guidance.

SOME OTHER FAITH BUILDERS IN MY LIFE

-1- At about eleven years old I sat beside a beaver meadow with the barrel of a shotgun to the roof of my mouth. As my finger touched the trigger a man's voice shouted "Do not shoot, there is a job you have to do." I jumped up and circled the tree I was leaning against to see who was talking to me, but there was no person.

-2- We took our middle daughter to have her baby picture taken at Sears. The photographer took the baby and put her on the table. He turned his back on her and started walking back to his camera. The baby fell off the table and was falling head first to the floor.

From behind the camera position, I ran past the photographer to catch her. My mind moved into slow motion and in terror I realized I was not going to get to her. I cried out in my mind "Jesus, please get me to my baby girl."

Instantly I was diving with the clear understanding that my right hand had to grasp both of her legs together and my left hand had to go between her head and the floor to cushion her head and compress her body, so I would not break her neck with the sudden stop.

She fell about forty inches. Jesus caused me to accelerated approximately three times the acceleration rate of the earth's gravitational pull with a zero reaction time.

My tested reaction time is the human average of 0.25 seconds.

She would have been at the floor before I normally would have reacted.

Jesus saved our daughter.

-3- In 1974 I was setting up the control system for an office.

It was on a mezzanine and the refrigeration compressors were on another mezzanine with a walk way between of about fifteen feet.

I commissioned the pneumatic circuits in the panel and then allowed the signal to activate the refrigeration compressors. When the signals were altered to de-activate the refrigeration compressors, the coolant valves closed, but the compressors did not stop as they should. The machine started roaring like I had never heard before.

I quickly rechecked the pneumatic circuit, determining that it was an electrical issue.

Convinced it was going to blow up I ran across the mezzanine to the stairs (about thirty feet) when I saw people walking toward the danger area. I did a 180˚ turn and started running back toward the panel and the machine I was sure was about to explode, crying out "Jesus, I do not know anything about this." (Being the electrical circuitry.)

I got to the panel and it was if I was witnessing some other person's hands rip through the bundles of tubing and wiring in the cable tray. One wire came out of the mess and He said cut that wire. I did and the machine fell silent. I went back to the stairs and sat at the bottom stair with my hands shaking. A security guard came to me and said "What the "bleep" is the matter with you?"

Without Jesus the occurrence could have ended in disaster.

-4- As my wife was dying from cancer, she initially experienced a severe issue with constipation. Once I became aware of her problem, I prayed for God to allow her free bowel movements. Instantly after praying she had a smooth and easy bowel movement. This occurred every time she indicated that she was worried regarding this issue.

-5- Although I did not feel ready, God compelled me to approach intellectual property lawyers in 1999 to start the patent process. The claims were considered impossible by the engineer/lawyers in that office and they applied much effort to disprove the claims. Their unsuccessful attempts allowed the patent process to proceed. At the meeting they agreed to proceed a critical logic illustration had not been developed. It was required in two weeks. I did not have the required knowledge at that time, but ordered them to proceed knowing Jesus would provide the required design on time. The design illustration which was beyond my ability was provided via Jesus within the two week requirement.

These are only a few of the Divine interventions in my experience. I believe we all have extraordinary experiences, but many people assign credit to themselves rather than the true source of knowledge.

SOME THOUGHTS TO PONDER

The eternal threat to unbelieving adults and children may be much more frightening than the environmental threats of today. Many people reject Jesus on the basis that they are scientific. That has been an issue for apparently a long time, as per 1 Timothy 6:20 in the New Testament, *"O Timothy, keep that which is committed to thy trust, avoiding profane and vain babblings, and oppositions of science falsely so called:"*

Atheists may alter their position if they analyzed the current, real science available today regarding the statistical improbability of life forming from non-life, the fine tuning factors of the universe allowing life to exist, the staggering complexity of the systems and interrelationship of God's systems in nature, the biological functions within each cell, the programming of DNA, etc. Systems by my hands are relatively very simple.

Of the hundreds of unique systems illustrated through my hands, not one was produced by putting a blank piece of paper and pencils into a box and shaking it for 13.5 billion years to have a functional design pop out by chance. Intelligence infinitely beyond mine was required. I was just the tool.

Consider Isaiah 1:18 *"Come now, and let us reason together saith the Lord."*

Unsupported faith in the atheist religion is not scientific. Think deeper than that. Christianity is evidence based faith, considering real science and the historic record via eye witnesses.

If one concludes, considering the known science, that there must be a source of intelligence far beyond any level humankind can even imagine, then they might:

-1- look at the predictions of the Prophets comparing those predictions to recorded history available to all of us.

-2- study the records of eye witness reports regarding Jesus.(New Testament)

-3- consider the testimonies of persons who are willing to tell their experiences with God, such as this paper.

-4- ask God to touch their life with the truth.

A COMMON SENSE POSITION FROM ONE OF HISTORY'S GREATEST SCIENTISTS

Pascal's wager:

Practical argument for belief in God formulated by Blaise Pascal. In his *Pensées* (1657–58), Pascal posed the following argument to show that belief in the Christian religion is rational:

"If the Christian God does not exist, the agnostic loses little by believing in Him and gains correspondingly little by not believing. If the Christian God does exist, the agnostic gains eternal life by believing in Him and loses an infinite good by not believing."

The Editors of Encyclopedia Britannica

YOUR CHILDREN'S WELL-BEING RISK ASSESSMENT APPLYING PASCAL'S LOGIC

On earth most parents do everything they can to protect their children's well-being.

According to Pascal's logic there are two pathways available for guiding your children into eternity:
----Path #1 with either neutral impact or total misery.
----Path #2 with either neutral impact or total well-being.

I believe that current science, logic and recorded history should lead you to Jesus, Who is path #2.

Do not make the wrong choice based on false science.

PATENT OFFICES' LINKS FOR TWO PATENTED INVENTIONS:

https://worldwide.espacenet.com/inpadoc?submitted=true&DB=EPODOC&CC=US&NR=2002178719&KC=&F=8&OREQ=0&textdoc=TRUE&FT=E

https://image-ppubs.uspto.gov/dirsearch-public/print/downloadPdf/7467517

Dave Strain
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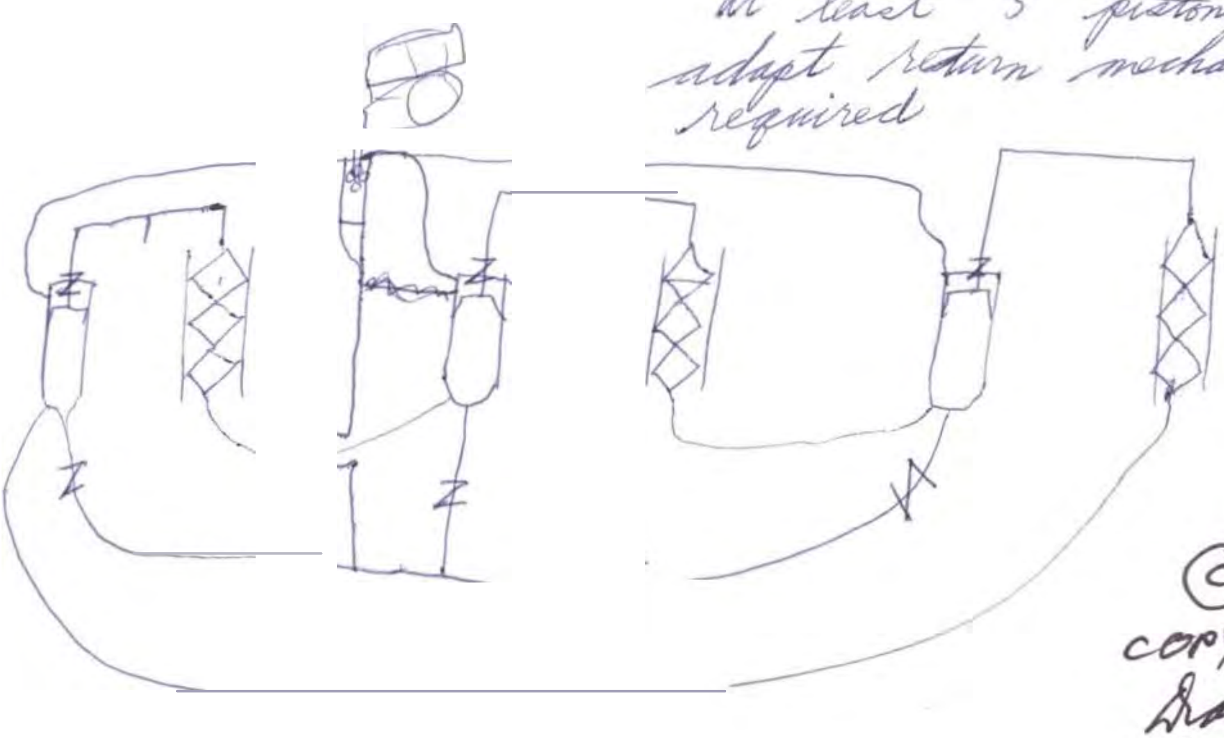
CONCEPT PRESENTATION AND LETTER OF INTENT REGARDING THE FIRST PATENTED INVENTION GIVEN TO DAVE STRAIN (OCTOBER 2, 1972)

Sept 30/72

I'm going to polution probe with a means to perpetual motion. The patent is to benefit man kind, not any individual. Money to my nick, will be assigned to cleaning up the environment and helping children.

Dave Strain

at least 5 pistons to adapt return mechanism. required



REGISTERED LETTER ENVELOPE CONTAINING LETTER PRESENTED ON PAGE 14.319