Consider Variable Pitch Fans

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Presented is a detailed analysis of the control and energy saving advantages of variable pitch fans. Basic mechanical and operating principles are also covered.

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Variable pitch fans provide the precise amount of airflow to control process temperature and save substantial amounts of energy at the same time ... automatically. This article discusses their ability to control, to conserve energy, and their basic operating characteristics and mechanical aspects. Economic comparisons with other type air flow control systems for axial fans are made.

Auto-variable®, controllable pitch, or variable pitch are words usually used to describe axial fans with blades that change pitch in operation, so that the precise amount of airflow is furnished to meet the requirements for a particular heat exchanger or cooling tower.

In the years of low cost energy, the impetus to use variable pitch fans has been for precise temperature control, generally to within four percent or less of the set point. However there is new interest in variable pitch fans for the energy saving potential as well.

This article will discuss characteristics of variable pitch fans under 30 feet (9.1 in) in diameter and less than 150 horsepower (112 Kw). The most typical sizes would be 10 to 14 feet (3.04-4.3m) in diameter consuming up to 40 horsepower (30 Kw). Shown in Fig. 1 through 3 are several different types of variable pitch fans. Fig. 1 shows the typical type found in air coolers. Fig. 2 shows an inverted actuator type popular in Europe but not in general use in the USA. Its advantage is that critical components are accessible for easy adjustment or maintenance from the walkway.

Fig. 3 shows the electrically operated hub for a 28 ft. (8.5m) diameter fan that will be used in a large air cooled steam condenser. The thrust bearing above the gear box is lubricated by the gear box oil pump.

Operating and Control Characteristics

Variable pitch fan operators used in air coolers can be described in general terms as very similar to diaphragm operated, spring return valves. Instead of moving a valve stem up and down the operator controls fan blade pitch. A force diagram is shown in Fig. 4.

As the blade moves the air, the aerodynamic moment (CW) tries to feather the blade. The hub spring creates an opposing moment (CCW) to make the blade do work. This is a fail-safe mechanism so that if the air pressure on the diaphragm fails, the fan operates as a fixed pitch fan providing design air flow. Fans can be assembled to move to maximum or minimum air flow on loss of air signal. The initial spring preload has to be sufficient to keep the blade from feathering.

To reduce the air flow, or even *reverse* the air flow direction, air pressure is exerted on the diaphragm to oppose the hub spring and decrease blade pitch.

When the blade pitch is about minus 10º, no work is done and essentially "zero" flow is attained. Minimum air velocity obtainable is approximately 50-100 fpm (.25-.50m/s). If the hub has its pitch stops adjusted for reverse flow, the air is directed downward and can be as much as about 60 percent of the upward flow at the same horsepower. The decrease in flow capability is because of poor efficiency in the reverse pitch mode.

A typical variable pitch hub requires a 3-15 psi (21103 kPa) control signal and operates the blades from some maximum pitch down to "zero" air flow. It typically fails to maximum flow if the control signal is interrupted but can be made to fail to minimum or negative flow. Most hubs are capable of 45º total pitch travel and perform as shown in Fig. 5.

Fig. 6 shows the actual results of two full scale tests: One of a 7 ft. (2.1 m) diameter fan with 10 horsepower installed and another of a 24 ft. (7.3 m) diameter fan with 125 hp (93 Kw) installed. The common shape of the curves can easily be seen. Note that the equivalent control signal for the electric operator is 4-20 ma compared to 3-15 psi (21-103 kPa) for pneumatically controlled fans.

Typical components. A typical fan hub mechanism is shown in Fig. 7. The basic components of a hub are:

- Hub spring
- Diaphragm
- Piston
- Blade shafts with eccentric actuator
- Rotary air joint
- Valve positioner

The blade shafts or axles hold the fan blade and have an eccentric bearing on the inboard end. These eccentrics engage the groove in the piston. As the piston moves up or down a twisting motion is imparted to the blades, changing pitch. The rotary air joint is the static/dynamic interface between the rotating fan and its control air system.

Methods of control. The valve positioner is a "closed loop" feedback device which receives the control signal, (usually 3-15 psi) and supply pressure up to 100 psi (689 kPa), receives feedback as to the blade position and adjusts the diaphragm pressure to satisfy the control signal. Valve positioners are used when air flow is critical, as for condensers.

Control signals can be "split-ranged" to actuate the fan control using only 3-9 psi (21-62 kPa), or from 9-15 psi (62-103 kPa) signal pressure.

The valve positioner receives the control air signal and outputs a higher modulated air pressure to the diaphragm which moves the fan blades to their proper position to satisfy the air flow requirements as called for by the temperature controller. For a 3 to 15 psi (21 - 103 kPa) control signal the output pressure of the valve positioner might be 7 to 23 psi (48-159 kPa) to make the fan blades move to their required pitch angle.

The advantage of the valve positioner is precise air flow control due to the feedback of blade pitch position and the ability to output high pressure on the diaphragm to quickly attain that position.

If air flow control is not critical, an "open loop" system can be used. The simplest would operate the fan using the 3-15 psi (21-103 kPa) signal alone. It is an "open loop" system in that an air signal is imposed on the diaphragm and the proper pitch angle is *assumed* to be attained. This will be no problem for coolers or units that are not sensitive to air flow and where 15 psi (103 kPa) diaphragm pressure is sufficient to move the blades through their required pitch travel. Operation with only 3-15 psi is generally limited to small fans.

An alternate device called a bias relay can operate most fans in lieu of a valve positioner. It is very simple, less expensive and requires no maintenance. It is mounted in the "cold" air outside the fan ring.

A bias relay operates by receiving the control signal, adding or subtracting a constant pressure and multiplying the sum by a fixed gain. It outputs a modulated higher pressure to simulate the output of the valve positioner. The bias relay has to be set for a particular fan application to provide *diaphragm* pressure proportional to the *instrument* pressure to make the blades move through their desired pitch travel (Fig. 8).

Bias relays can provide the proper starting point so the blades begin decreasing pitch at 3 psi (21 kPa), but with a fixed multiplier or gain (usually 2 or 3), they usually cannot provide the output pressures to assure an exact 12 psi (83 kPa) span.

Hysteresis. Another point of interest is shown in Fig. 8. Note that there is a difference in the pitch angle versus diaphragm pressure for increasing and decreasing pressure. This is because of hysteresis in the hub operating mechanism caused by friction. Hysteresis is practically nil when a positioner is used and even if it is present and there is a slight discrepancy in air flow to control the process temperature, the T.I.C. (Temperature Indicating Controller) will merely output a signal correction to meet the desired control point.

As an example, consider a fan with 20º blade travel. This fan may require a diaphragm pressure of 7 to 23 psi (48-159 kPa) to get 20º movement from the hub. A valve positioner can achieve this exactly through its feedback mechanism. A bias relay with a bias of + 0.5 psi (3.5 kPa) and gain of 2 can achieve the 7 psi (48 kPa) diaphragm pressure with 3 psi input signal, but at 15 psi signal its output is 31 psi (214 kPa). The 23 psi (159 kPa) diaphragm pressure is attained at a signal pressure of only 11 psi (75 kPa) giving a *span* of 3-11 psi instead of 3-15 psi. Fig. 9 shows this difficulty. The difficulty of obtaining an exact 12 psi (83 kPa) instrument span is caused by the lack of choices available for the fixed multiplier (gain). The solution lies in using a variable ratio bias relay which is under development.

Process Control

There are several methods used to control process temperature in an air cooler:

- Fluid bypass (manual)
- On-off fan operation (manual)
- Two-speed fans (manual or automatic)
- Louvers or shutters (automatic)
- Variable pitch fans (automatic)
- Variable speed fans (automatic)

Of the methods of control, the oldest is probably the bypass method. This is merely bypassing the throughput if the air cooler is overcooling.

"On-off" fan control is simple and is often used if there are a large number of fans in an identical service. This non-modulating control method, however, can cause problems in air cooled condensers such as water hammer or tube to header leakage with differential expansion of bundles in parallel. Differential tube expansion can cause tube buckling. Power savings are at the mercy of the operator unless the bank is computer controlled. Cooling tower fans are a good example of "on-off" fan control. Air flow control is in incremental steps.

Two-Speed fans are a further refinement giving 0, 50 or 100 percent air flows with 1800/900 rpm (T/M) motors or 0, 67 or 100 percent with 1800/1200 rpm (T/M) motors. Naturally, larger numbers of motors give smaller incremental steps of air flow control.

Automatic louvers are the first step to modulated air flow. The only problem is that fan horsepower is wasted as the flow is throttled by the louver. At complete shut-off the fan is stalled and horsepower actually increases.

Variable speed fans for fully modulated air flow are available in two types: hydraulic and electric drive. Either type conserves energy and offers good air flow control. The hydraulic drive system consists of a motor/variable volume pump/reservoir unit connected to a slow speed, high torque motor driving a fan shaft directly. These drives have been used for many years although not widely. Advantages are variable fan speed and elimination of a reduction belt or gear drive. The basic disadvantage is inferior system drive efficiency. Optimum efficiencies are in the range of: motor 0.97, pump 0.92 and hydraulic motor 0.92. The optimum drive system efficiency thus would be: efficiency = (.97) (.92) (.92) = 0.82, not counting hydraulic line losses. This must be compared to a typical motor/v-belt drive efficiency of 0.95 or 0.97 with gear or timing belt drive. As with all systems efficiency will vary from operating to operating point, generally decreasing.

The latest developments in electrical variable speed controls for fans are a.c. adjustable frequency drives (AFD). There are three basic types: VVI (Variable Voltage Inverter), PWM (Pulse Width Modulation) and CS (Current Source). These drives utilize a standard induction motor and require no starting gear. Automatic control is obtained by a process control device to interface the 4-20 m.a. output of the temperature controller with the AFD.

Provision must be made to operate the fan in the reverse direction to counter the natural draft effect in an air cooler and attain essentially zero flow. Advantages of AFD's are reduced noise and vibration during slow speed operation. Disadvantages are high cost per horsepower for the control of a small number of fans in the same service and total loss of service if multiple fans are controlled by one AFD which fails.

The variable pitch fan can provide from 0 to 100 percent upward or about 0-60 percent *negative* flow at the same horsepower. Another way to put it would be: “From plus 100 percent to minus 60 percent modulation.” Negative air flow is useful in winterized type air coolers that seal off freezing outside air and recirculate warm air inside the plenum. The variable pitch fan goes into reverse, pumping air downward and an adjacent fixed pitch fan circulates it upward.

An actual example of the ability of the variable pitch fan to control is shown in Figs. 10 and 11. These are photographs of actual strip charts from a southwest chemical plant for two days of operation. The unit was a liquid condenser with two variable pitch fans using an "anticipatory" or "feed-forward" type control system.

A change in the inlet air temperature was sent to a pneumatic computing relay. The condensate temperature was sensed by the temperature controller which outputs the control signal for the fan pitch. In this case the control signal changes required by the change in inlet air temperature and condensate temperature were averaged to "anticipate" the proper signal to control the variable pitch fans.

In these strip charts, the heavy line represents condensate level and the light line represents condensate temperature. The notable points from these charts are (a) good control during various air temperatures from 40ºF (4ºC) to 70ºF (2 1ºC) (b) the upset and recovery of control in condensate temperature due to windy, gusty rain shower and (c) overall temperature control was within plus or minus VF of a 175ºF (79ºC) set point.

Although the control system was more elaborate than usual, the ability of the variable pitch fan to accurately control a process or condensate temperature has been proven many times over.

Power Consumption and Cost Comparison

When considering power consumed by any axial fan operating at fixed pitch, we must consider not only the fan performance curve but the system resistance characteristics as well.

Fan performance curves are obtained in a wind tunnel with a means of varying the resistance or static pressure head against which the fan must work, and measuring the resulting airflow and horsepower.

System losses are more difficult to determine, and judgement is necessary in many cases to evaluate the losses due to poor inlet conditions, excessive tip clearance, unusual structural conditions, etc. If a cooling service is very critical, it may be wise to model the unit and establish the system losses in a wind tunnel. What you must determine is the sum of the static pressure resistances versus air flow of the *system.* Since the air delivery characteristics of the *fan* have been accurately established (or should have been) the operating point for any air flow requirement can be established. This point is where the output of the fan in terms of pressure and flow exactly meets the requirements of the system. It is the equilibrium point of operation.

Let's examine a typical case of an air cooler with a fixed pitch fan and a means of throttling air flow, such as a shutter, at its design point and several other operating points of' reduced air flow.

For example, consider a 14 ft. diameter fan operating at 14º pitch at its design point 1 (Fig. 12). The system resistance line, shown as a dashed line, is the locus of points obtained by a summation of the static and velocity pressure losses vs. flow through the bundle. The pitch angle line for 14º represents the delivery characteristics to the fan. Assume density remains constant and the decreased air flow is a result of decreased duty.

The louver throttles air flow to control the outlet temperature. From Fig. 12 the points shown as 1, 2', 3' and 4' are the total pressure output and horsepower consumed by the fixed pitch fan. If we were controlling the airflow with a variable pitch fan instead of a throttling device, the total pressure output would exactly match the system requirements as shown in points 2, 3 and 4 and the horsepower requirements would be significantly reduced. The savings in horsepower can easily be seen. In this case the variable pitch fan saved 26, 51 and 73 percent of the horsepower required by the fixed pitch fan at points, 2', 3' and 4'.

Energy comparison. A typical case study to evaluate cost differentials between fixed pitch (single or dual speed) or variable pitch fans would be prepared as follows:

Step 1. Thermal studies would be made to determine the total airflow required as a function of ambient temperature for the particular item of interest. Plot flow versus temperature.

Step 2. Obtain climatological data for the area where the air cooler will be located and derive a table of "degree-hours" for incremental temperature ranges. This tabulates the number of hours per year that each temperature range occurs.

Step 3. For a fixed pitch fan, flow is a direct function of speed. Make a plot of fan output versus ambient temperature for each scheme to be studied. This yields hp-hours per temperature range.

Once the cfm is defined for each range, the fan horsepower for any point (i) can be approximated by the following relation:

$$Hpi = \left(\frac{cfmi}{cfm\ des}\right)^{2.8} \left(\frac{pi}{pdes}\right) hp_{des}$$

Where hpi = Horsepower at $cfmi$
pi = Density at point i lb/ft^3
$cfmi$ = ft^3/min flow at point i
hp_{des} = Design horsepower
$pdes$ = Design density

or:

$$kWe_i = \left(\frac{m^3/si}{m^3/s_{des}}\right)^{2.8} \left(\frac{kg/m^3 i}{kg/m^3_{des}}\right) kWdes$$

Step 4. Using hp-hours for each temperature range, tabulate the total for the year. The power required would be *kWe* = *hp* (.746). Arrive at an energy cost per year for each scheme.

This is a very simple approach but a useful tool.

As an example consider the following: a 20 by 36 ft. (6 by 11 m), 4 row, forced draft air cooled heat exchanger, with 2-14 ft. (4.3 m) fans each having 40hp (30kW) motors. The unit is a propane condenser, item 126C to be located in West Virginia, close to Charleston. Power cost is \$0.035/kW-hr.

Study the direct costs of power and initial investment for a one-year period for three schemes:

1. Two fixed pitch fans.

2. Same, but with one 1800/900 rpm motor.

3. One fixed, one variable pitch fan (single speed).

Design Conditions (per fan):
Flow .... 224,613 acfm (106 m/s)
Power .... 29.0 hp (21.6 kW)
Temperature 45ºF (7ºC)
Elevation ....350 Ft. (107m)

See Fig. 13 for a plot of Air Delivery vs. Ambient Temperature.

Table 1 gives the power study results. The results show that the variable pitch fan would pay for itself within two years. In this particular application the two-speed motor in case 2 made only a small difference in energy cost.

Table 1 – Power Study Results

	Est. total kw-hr/yr	Energy cost/yr $.035/kw-hr	First costs
Case 1 2 fixed pitch	162,670	$5,693	Base
Case 2 2 fixed pitch 1 ½ sp manual)	142,080	$4,973	+$1,000
Case 3 1 fixed pitch 1 variable pitch	70,887	$2,481	+$3,715(a)

(a) includes temp. controller, variable pitch cost adder plus est. $1,500 installation.

Table 2 – No Load Power Consumption

Rated horsepower	Measured no-load power factor	Average no-load horsepower	Per cent rated horsepower
10	.305	1.55	15.5
15	.285	2.22	14.8
20	.355	2.30	11.5
25	.210	3.41	13.6
30	.265	3.95	13.2
40	.330	4.24	10.6
125	.350	14.2	11.4

Overall average no-load power was 12.9% of rated horsepower
Average no-load power factor was 0.300
All values obtained by operating fans at zero-flow condition of (-) 10 pitch
All motors, 440v, “mac”

In an additional case consider a variable frequency inverter drive: Typical cost for a 40 hp (30 kW) inverter would be about $10,000 per fan. Two 40 hp fans could be saved by using an 80 hp inverter at slightly less cost per horsepower. The power savings should not be significantly lower than the fixed/variable pitch combination. Considering two inverters at $20,000, plus $1,000 for temperature controls and $1,800 installation, the first cost would be about $22,800. This must be compared to $3,715 additional first costs for the variable pitch fan.

Minimum power consumption. One of the most misunderstood but important characteristics of the variable pitch fan is the ability to "unload" or feather for minimum power consumption *automatically* when the ambient temperature decreases.

The measurement of this no-load power condition cannot be accurately determined by the ratio method using voltage and amperage. Using this method would lead one to believe the minimum power consumption of a variable pitch fan in its feathered position would still be 30 to 50 percent of full load. This is not true because three-phase power consumed by the motor is calculated by the relation:

Hp_{in}, $= (I)\ (E)$ (1.732) (PF)/746
Where I is amperage, *E is* voltage, PF is power factor. Without measuring power factor, accurate measurements cannot be made. Note, this would be *total* power consumed at "no load" fan operation. If we knew motor efficiency at no load we could calculate "hp_{out},". Since motor manufacture's data for loads less than half are not published, we chose to consider "total power in" for a series of tests shown below. The "no-load" power occurs at the

"zero-flow" position of the fan pitch, usually at about minus ten degrees. This feathered mode occurs quite frequently during daytime-nightime and summer-winter ambient temperature fluctuations and has importance in any power evaluation.

For these tests a power analyzer was used. The minimum power was obtained by two methods and averaged:

Method (a) Read directly in kilowats, *hp* = *k*W/0.746
Method (b) Calculated using volts, amps and power factor.

The results are shown in Table 2. The motors in each case were "MAC" (Mill and Chemical) type, NEMA B, 440 volt of varying horsepower typically used in air cooled heat exchangers. For most tests a 14 ft. (4.3M) diameter fan was used. Note that the average no load power was only 12.9 percent of rated horsepower.

Maintenance experience. The most frequently heard complaints about variable pitch fans concern reliability. In the past some of these complaints were valid, but with improved components, reliability is no longer the "weak link."

The basic enemy of the variable pitch hub is heat--whether in induced or forced draft fans. In the past typical lubricants were hydrocarbon and the elastomers were bunan or neoprene. These are not good enough for exposure to the continuous dry air in an air cooler. *Synthetic lubricants* and *silicone elastomers* are necessary for the high temperatures of Saudia Arabia or the low temperatures of the North Slope.

The rotary joint's weakness is water inclusion which causes bearing failure. After two year's testing of a new type rotary joint with three separate ways to exclude water; synthetic grease, synthetic seals, and carbon/ceramic air seal, field results have been excellent. Designs utilizing dual air passages have generally been much more prone to failure than the single passage types.

Actuator bearings are typically round cam followers acting on hardened washers. This is fine for about a year until the line contact of the circle with the washer turns into a groove at the most used pitch position. A small change of signal would not change pitch. Some of today's fans are square, hardened blocks which offer much lower contact stress since they utilize area contact rather than line contact. This former problem has disappeared, especially using oil as a lubricant.

The point is that the environment in which the AV fan operates is very hostile. It must have premium component materials to be maintenance free.

Acknowledgment

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About the Author

Mr. R.C. "Bob" Monroe had over 30 years of experience in Fan Engineering and was the Manager of Research and Development for two years with Hudson Products Corporation, before his retirement on August 1, 1997. Prior to joining Hudson, Mr. Monroe was engaged in oilfield tool design for 17 years. He earned a B.S.M.E. degree from the University of Houston, and was a past President of the Cooling Tower Institute, and also served many years on their Board of Directors. Additionally, he was a member of ASME, and the Texas Society of Professional Engineers.

Figure 1 Typical Variable Pitch Fan

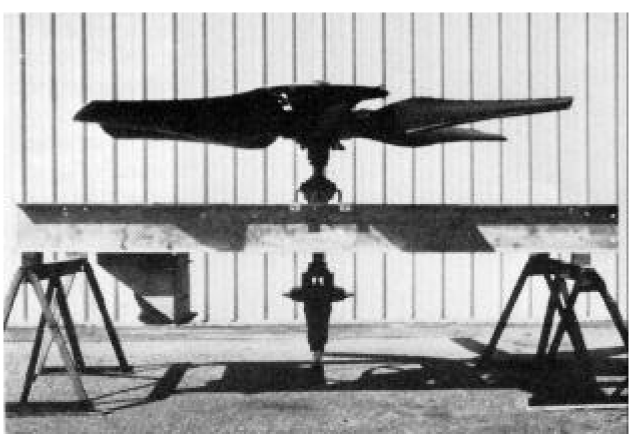

Figure 2 Inverted Actuator Variable Pitch Fan

Figure 3 Electrically Operated Variable Pitch Fan

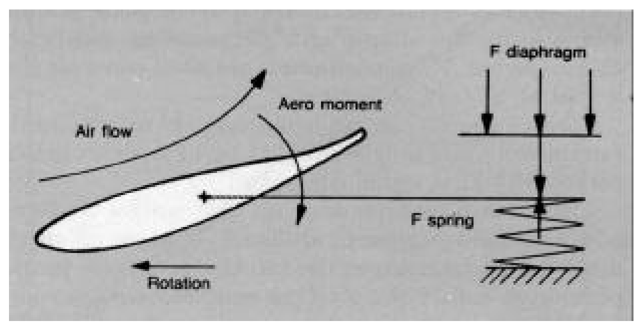

Figure 4 Force Diagram

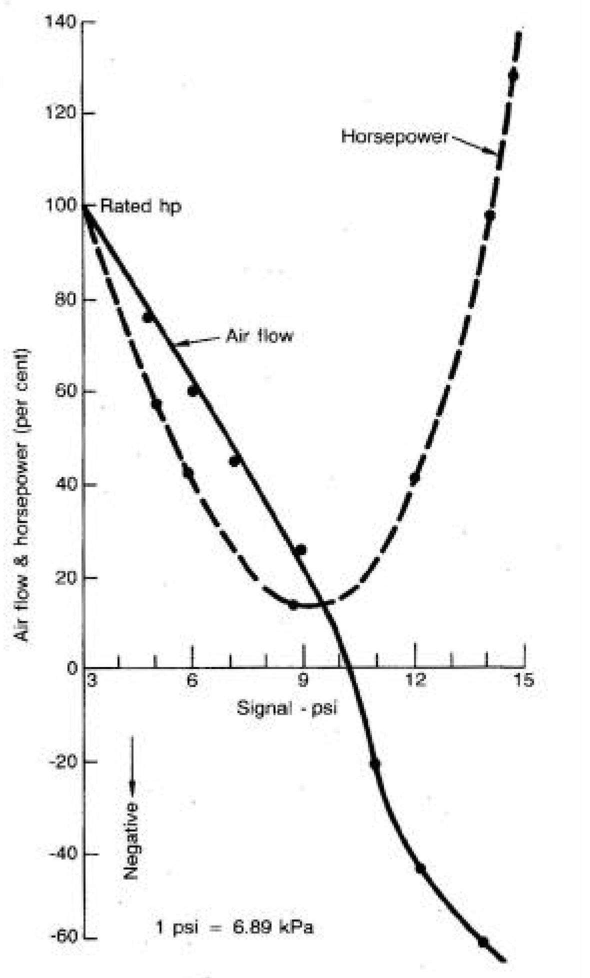

Figure 5 Air Flow and Horsepower Vs. Signal

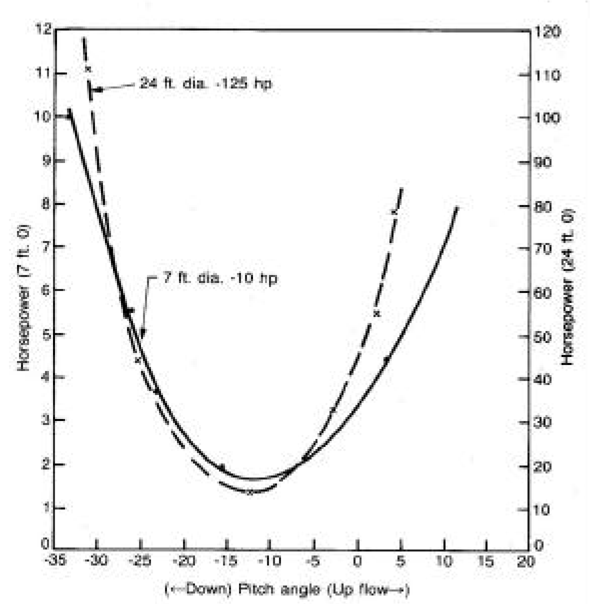

Figure 6 Horsepower Vs. Pitch Angle

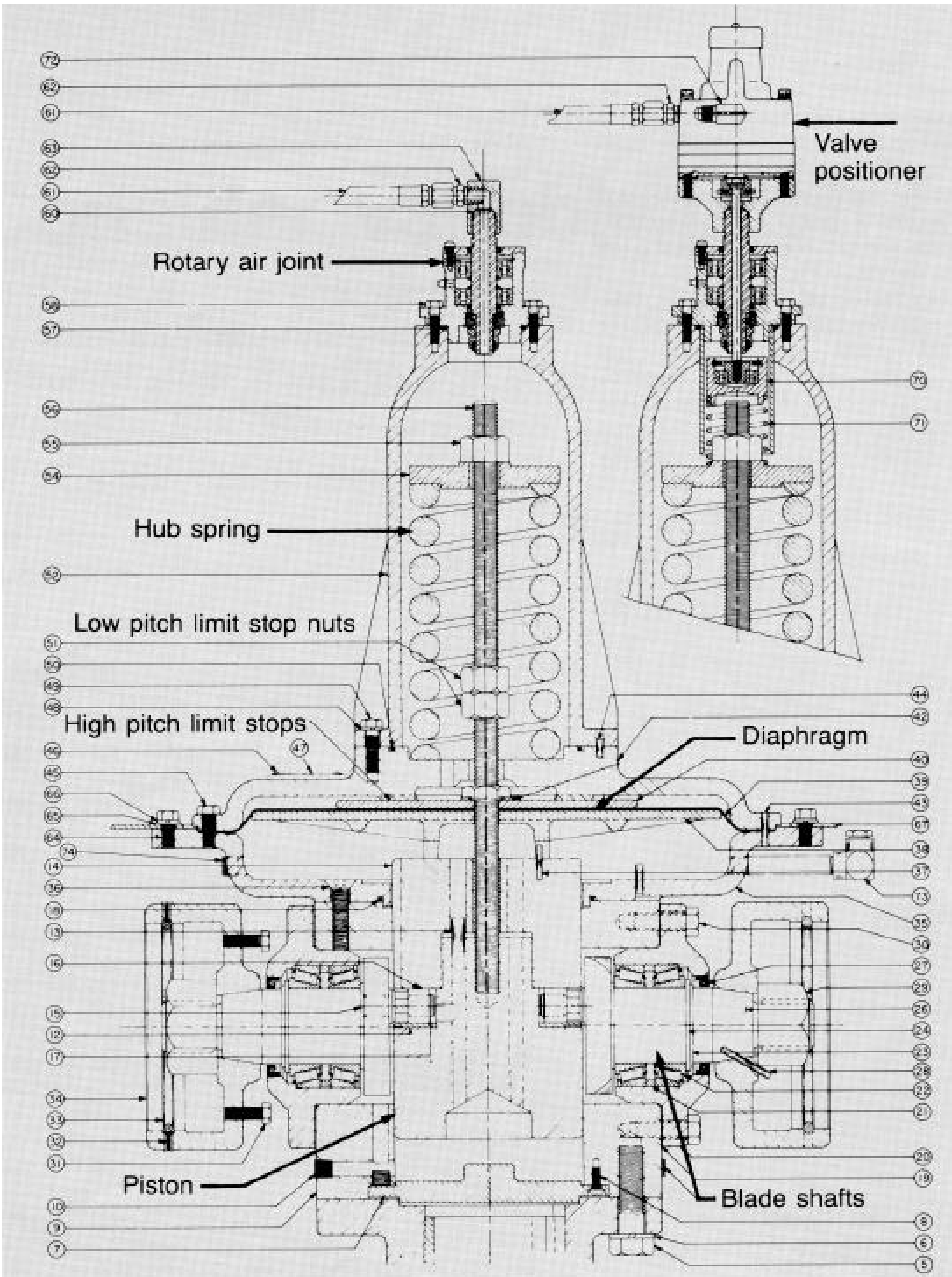

Figure 7 Variable Pitch Hub Components

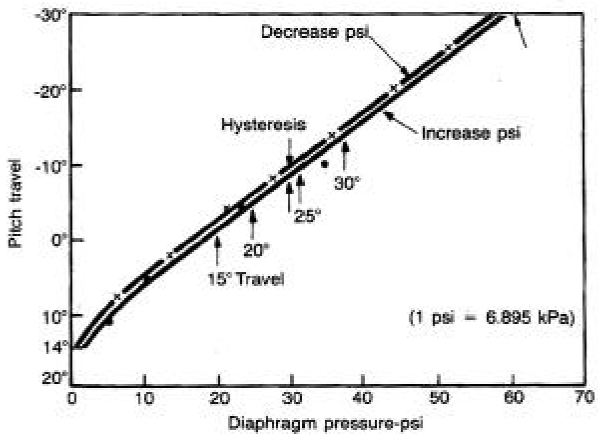

Figure 8 Operating Pressure Test

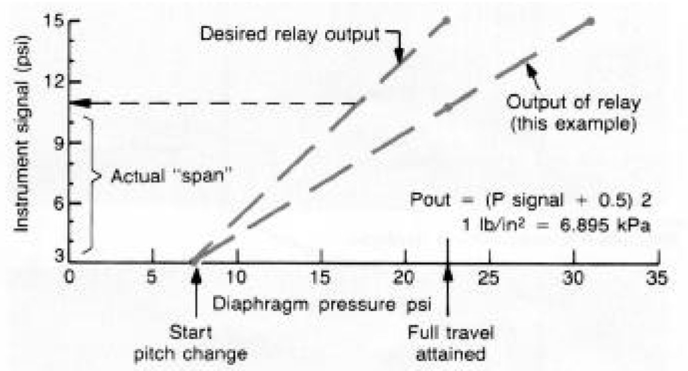

Figure 9 Bias Relay Output

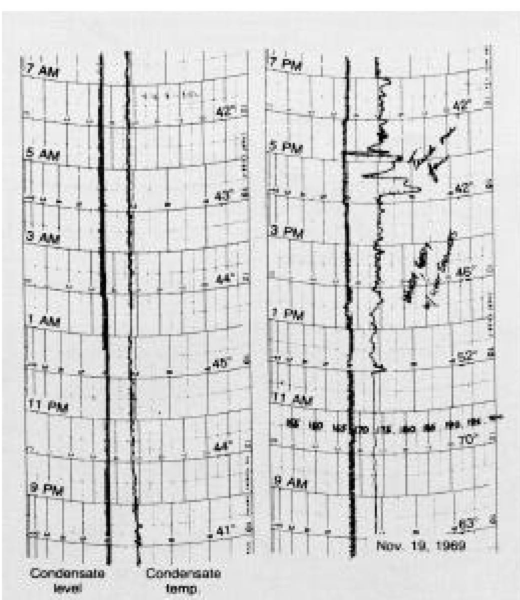

Figure 10 Process Control Example

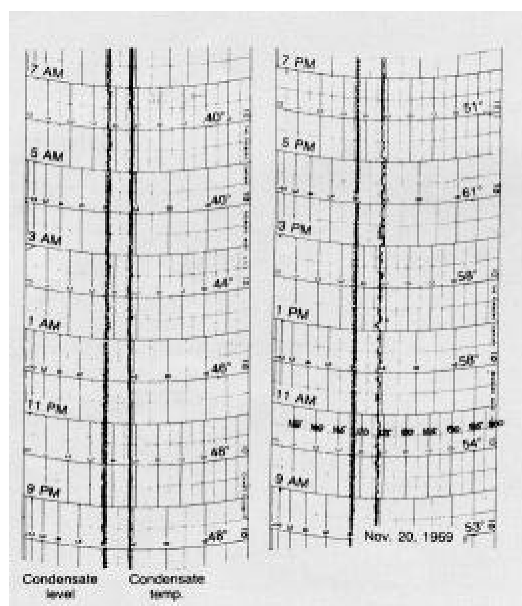

Figure 11 Process Control Example

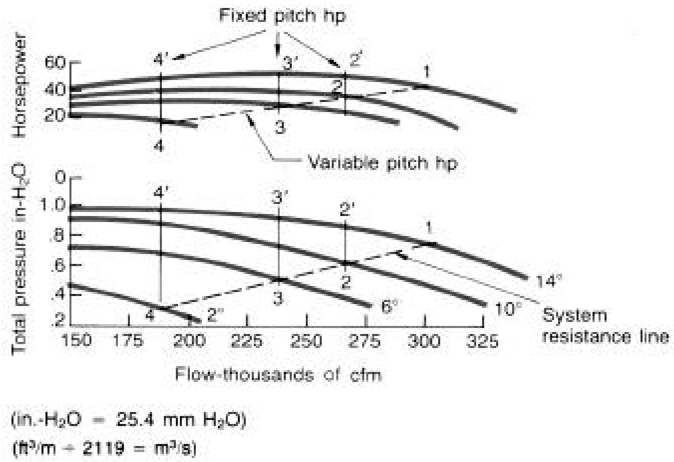

Figure 12 Throttle flow through fan

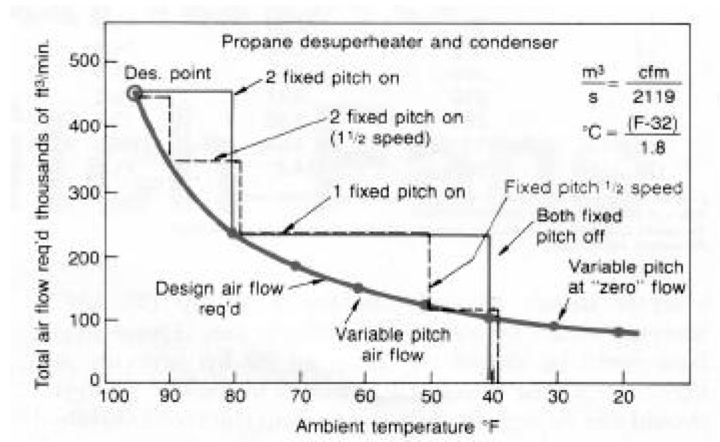

Figure 13 Air Delivery Vs. Temperature