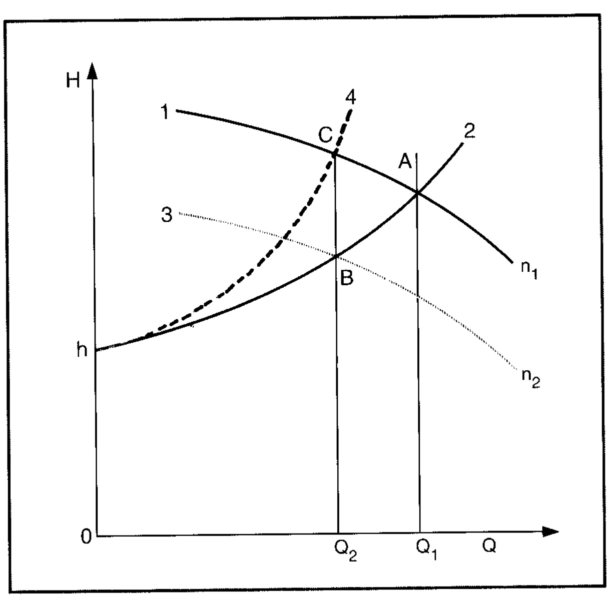
The influences of back pressure on variable-speed control

Variable-speed controls are widely employed in pumps and fans because of the obvious energy-saving benefits. However, the energy-saving benefits gained by using variable-speed controls are linked with individual system characteristics. In this contribution from China, Fu Yongzheng, Wu Keqi and Cai Yaqiao argue that, as a result, variable speed controls offer different energy-saving benefits in different systems.

A pipeline system can be classified into one of two types, according to the increase in energy resulting from the fluid passing through a pump or fan:

(1) **Systems with back pressure.** Part of the energy increase resulting from fluid passing through the pump or fan is used to overcome the resistance of the pipeline, while the remainder is required to boost the potential energy of the fluid, for example in delivery systems for high towers or boilers and in water-filled and pressure-limited systems used in heating systems. The characteristic curve of this system is $H=h+SQ^2$. In this formula, H is the total energy increase resulting from the fluid passing through the pump or fan, Q is the flow, S the resistance of the pipeline system, and h the back pressure, that is, the increased energy resulting from fluid flowing from the entrance to the exit of the system (i.e. the static lift).

Figure 1. The pump-pipeline model on which the analysis is based.



(2) **Systems without back pressure.** Here all of the energy increase resulting from fluid passing through the pump or fan is used to overcome pipeline resistances, such as in ventilation systems, heating systems, air-conditioning or water-cooling systems and closed-loop systems with other liquids. The characteristic curve of these systems is $H=SQ^2$; put another way, the back pressure is zero: h=0.

The energy-saving benefits of variable-speed controlled pumps and fans have a close relation with the system: whether it has back pressure or not and how large that back pressure is. This article studies the relationship between the above two system types and analyses why and how back pressure influences the energy-saving benefits of variable-speed controlled pumps and fans.

Analysis model

For convenience, the following analysis considers a pump system, but the conclusion would also apply in the case of a fan.

In Figure 1, the characteristic pump curve is line 1 and the corresponding speed (in rpm) is n_1. The characteristic curve of the pipeline is line 2, namely, $H=h+SQ^2$. The designed working point of the pump is point A, for which the flow is Q_1 and the delivery lift is H_A. Now, the flow needs to be regulated to Q_2. If throttle control is adopted, the working point of the pump becomes point C and the characteristic curve of the pipeline becomes line 4. However, if variable-speed control is adopted, the working point of pump becomes point B and the corresponding pump speed is n_2.

Obviously, a different back pressure, h, would result in different characteristic pipeline curves, and the position of variable-speed condition B would also be changed. Namely, with variation of h, the position of B is changed. Assuming the same type of pump work in the different pipeline systems and the same design working point, the flow is the same after changing the velocity. For this condition, this paper studies the relationship between the energy-saving benefits of variable-speed control and the back pressure h by analysing the way the parameters of the variable-speed condition B change with h.

How variable-speed condition parameters change with back pressure

Effect of back pressure on lift

After changing the velocity to n_2, the pump's delivered lift is:

$$H = h + SQ_2^2 \tag{1}$$

From the coordinates of point A, S can be derived as follows:

$$S = \frac{H_A - h}{Q_1^2} \tag{2}$$

Inserting Equation 2 into Equation 1 we get:

$$H_B = f(h) = h + \frac{Q_2^2}{Q_1^2}(H_B - h) = \frac{Q_2^2}{Q_1^2} H_A + \left(1 - \frac{Q_2^2}{Q_1^2}\right) h \tag{3}$$

For $(Q_2/Q_1)^2 < 1$, f(h) is monotonously increasing function; namely, lift H_B increases as h increases.

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Change in efficiency with back pressure

As Figure 2 shows, the parabola that passes through point B can be derived according to Q_2 and H_B, which are coordinates of point B.

$$H = \frac{H_B}{Q_2^2} Q^2 \tag{4}$$

Point D is the point that this parabola (curve 5 in Figure 2) intersects the characteristic curve 1 of the pump; if B and D are similar conditions, then the efficiencies of these two conditions are equal. Point D' is the point on the efficiency curve (6) corresponding to D, so the efficiency of condition B is the efficiency coordinate of D'. When h=0, line 7 is the characteristic curve of the pipeline; this curve is also a similar parabola to 5. For this curve, points B_0 for variable-speed control and A are similar conditions. So, as the variable-speed control condition B moves upwards (h increases gradually) from B_0 (h=0), its similar condition D (for which the pump's speed is n_1) moves from A to the left, and the corresponding point D' on the efficiency curve also moves from A' in the same way.

Thus, with the change of h, there are two possible scenarios for the change of efficiency of condition B. The first is that the efficiency of condition B reduces monotonously with the increase of h, when point A', corresponding to design condition A, is at the highest point on the efficiency curve or to its left. The second is that the efficiency of condition B increases slightly at first and subsequently drops gradually with the increase of h, which is the case when A' is positioned to the right of the highest point of the efficiency curve, on the decreasing section of the curve. In general, A should be at the high efficiency section – it should not be far from the highest point even if it isn't at this point. Thus, the efficiency of condition B in general decreases with the increase of h.

Effect of back pressure on power

The power requirement of condition B is:

$$N_B = \frac{Q_2 H_B \gamma}{\eta_B} \tag{5}$$

From the preceding analyses, it can be seen that H_B increases with the increase of h while η_B decreases. Thus, N_B increases with the increase of h.

Back pressure changes energy-saving benefits

Variable-speed control provides an energy saving if there is a decrease in energy consumption compared with other methods that could be used to obtain the same control goal in a given situation. Variable-speed control is mainly compared with throttle control because that is the most simple control method and its application is the most widespread.

The difference in the power requirements of conditions C and B is:

$$\Delta N = N_C - N_B = \gamma Q_2 \left(\frac{H_C}{\eta_C} - \frac{H_B}{\eta_B} \right) \tag{6}$$

Because N_B increases when h increases and N_C is not dependent on h, ΔN reduces with the increase of h; that is, the energy-saving benefits of variable-speed control are reduced as h increases. From this, the relationship between the energy-saving benefits of variable-speed control and the back pressure of the pipeline can be seen easily. The increase of back pressure h results in an increase in the delivered lift and a decrease of efficiency for the variable-speed control condition, thus causing the power increase. Thus, the disparity between the power for the variable-speed control condition and that of the throttle control condition reduces as h increases, thus also reducing the energy-saving benefits of variable-speed control.

The energy-saving benefits of variable-speed control can't be calculated and estimated directly from ΔN: the efficiency of variable-speed installations must also be taken into account. Assuming the efficiency of variable-speed installations is η_m and comparing the variable-speed control with throttle control, the economy in electrical power is:

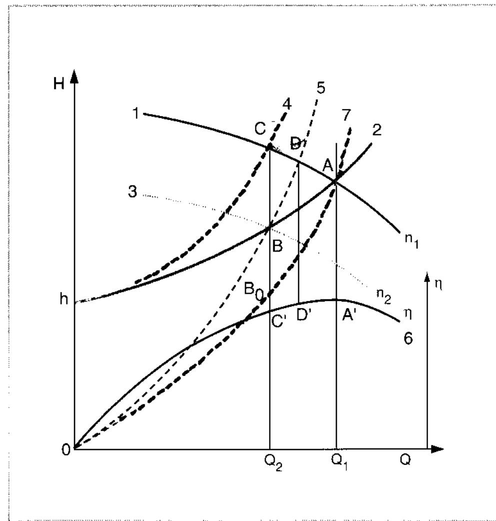


Figure 2. Diagram showing how the efficiency of the variable-speed condition varies with the change in h.

$$N_C - (N_B/\eta_m) \tag{7}$$

Obviously, $N_C - (N_B/\eta_m) < \Delta N = N_C - N_B$, and if h increases to a sufficient extent, the situation may arise that $N_B/\eta_m \geq N_C$. At that stage, instead of economizing energy, the energy consumption is more than before the adoption of variable-speed control.

Example calculation

Let us consider three different pipeline systems:

I: $H = S_0 Q^2$ (h=0)

II: $H = H_1 + S_1 Q^2$ ($h = H_1$)

III: $H = H_2 + S_2 Q^2$ ($h = H_2$)

We assume that in each case the pump work is of the same type and that the design condition is the same, namely point A as shown in Figure 3, with the corresponding rpm of n_1 and flow rate Q_1. If the flow rate is now modulated to Q_2 and variable-speed control is adopted, the conditions of the three systems become respectively D, E and F. If throttle control is adopted condition G applies to all three.

The example considers a centrifugal water pump of type ISG150-400G produced by a factory in Shanghai. According to the property curve given in the manufacturer's literature, the parameters and power of points D, E,

Table 1. Calculated results for example systems I, II and III

	Q_2=0.045 m^3/s ($0.75Q_1$)				Q_2=0.03 m^3/s ($0.5Q_1$)			
	D	E	F	G	D	E	F	G
H (m)	27.0	35.8	46.7	52.2	12.0	27.0	45.8	54.0
η (%)	74.0	73.5	72.0	71.0	74.0	67.0	61.0	60.0
N (kW)	16.1	21.5	28.6	32.4	4.8	11.8	22.1	26.5
n (rpm)	1088	1208	1377	1450	725	1061	1338	1450

F and G can be worked out after choosing Q_1, Q_2, H_1 and H_2.

(1) Taking Q_1=0.06 m^3/s, gives H_A=48 m, η_A=74%.

(2) If H_1=20 m and H_2=45 m, and because the characteristic curves of the three pipeline's systems all pass through point A, it can be worked out that:

$S_0=13.33\times10^3 m/(m^3/s)^2$

$S_1=7.78\times10^3 m/(m^3/s)^2$

$S_2=0.833\times10^3 m/(m^3/s)^2$

Then, the pipeline's characteristic curves for the three systems are respectively:

$$H=13.33\times10^3Q^2 \quad (8)$$

$$H=20+7.78\times10^3Q^2 \quad (9)$$

$$H=45+0.833\times10^3Q^2 \quad (10)$$

(3) After confirming Q_2, the parameters of points D, E, F and G can be worked out or confirmed by examining the property curve. Because condition D and A are similar, the lift and speed of condition D can be calculated according to the equivalence rule, and efficiency of condition D is equal to that of condition A. The delivered lift and efficiency of condition G can be obtained from the property curve. The lift of conditions E and F can be worked out by Equations 8 and 9. Taking condition E as an example, there is a method to confirm the efficiency and speed of conditions E and F, which can be explained as follows:

a. According to flow rate Q_2 and delivery lift H_E of condition E, it can be worked out that the equation of the parabola passing through point E is $H=(H_E/Q_2^2)Q^2$, namely, 0–E–P, shown in Figure 3.

b. The parabola intersects the curve that corresponds to the original rpm n_1 at point P, and points P and E are similar conditions.

c. The speed for condition E is $n_3=(Q_E/Q_P)n_1$.

d. The efficiency η_P of point P can be obtained from the efficiency curve corresponding to the original speed n_1, and because points E and P are similar conditions, their efficiencies are equal, namely, $\eta_E=\eta_P$.

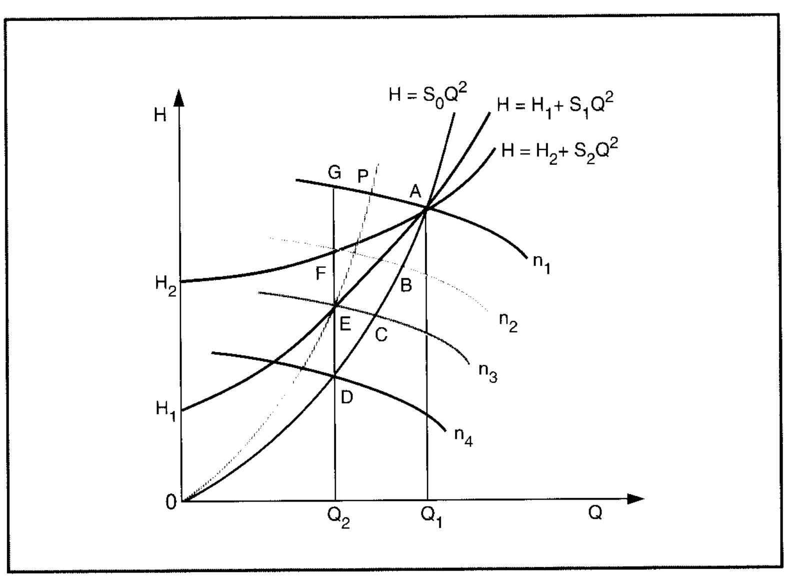
Table 1 lists the parameters for conditions D, E, F and G for two situations where $Q_2=0.75Q_1$ and $Q_2=0.5Q_1$.

The power of the variable-speed control conditions is not difficult to calculate according to the data in this table. If the power required by throttle control condition G is taken as 100%, the power of variable-speed control conditions D, E and F are 49.7%, 66.4%, 88.3% respectively for $Q_2=0.75Q_1$, and 17.1%, 44.5%, 83.4% respectively for $Q_2=0.5Q_1$.

So, the energy-saving benefit of the system without back pressure (system I, condition D) is the best. The power required by variable-speed control gradually tends to that of throttle control with the increase of back pressure, and the energy-saving benefit of variable-speed control decreases gradually. These findings accord with the conclusion drawn from the theoretical analyses above.

Taking the efficiency of variable-speed installations into account (the efficiency range for all kinds of variable-speed installations[1] is 0.8–0.96), the practical energy consumption of variable-speed control will be close to that of throttle control and will even exceed it when the back pressure reaches a sufficient level. For example, for pipeline system III in this example (that is, $H=45+0.833\times10^3 Q^2$) with the flow rate modulated from 0.06 m^3/s to 0.045 m^3/s, the axial power of the throttle control condition is 32.4 kW and that of the variable-speed control condition is 28.6 kW (condition F). Obviously, if the efficiency of the variable-speed installation is taken into account, the practical energy consumption of the variable-speed control condition will be rather close to that of the throttle

Figure 3. Pump and pipeline curves for the example calculation.



control condition. At this point, the variable-speed control method will lose its significance of economizing energy.

Conclusion

With the increase of h, the vertical lift of the variable-speed control condition increases and the efficiency decreases; thus the axial power increases and the disparity between the power of the variable-speed control condition and that of the throttle control condition decreases. Therefore, the energy-saving benefit of variable-speed control is greatest in the system without back pressure.

In a system with back pressure, the energy-saving benefit of variable-speed control decreases gradually with the increase of back pressure. When back pressure increases to a certain level, the power drawn by the variable-speed control condition is very close to that for throttle control. Once the efficiency of the variable-speed installation is taken into account, the actual energy consumptions of the two conditions are rather close. Thus, the variable-speed control method will lose the advantage of energy economy, and the situation may arise that the energy consumption of variable-speed control is more than that of throttle control. Because adopting variable-speed control increases the investment required in a system, the potential benefits must be analysed and calculated carefully according to the actual situation in a system with back pressure. Then, the correct conclusion can be drawn about whether it saves energy and is economical or not. ■

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