FORECASTING THE ENERGY-SAVING BENEFITS OF VARIABLE-SPEED PUMPS

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Abstract: By calculating the energy-saving benefits of a variable-speed pump, the relation between back pressure and energy-saving benefits is analyzed. Based on the energy-saving benefit calculation for four different pump types using a non-dimensional method, it is shown that when the non-dimensional back pressure, $\overline{h}$, and the non-dimensional flow, $\overline{Q}$, are equal, then the corresponding non-dimensional energy-saving benefit Ω of variable-speed control are quite close each to other, thereby advancing a forecasting method of energy-saving benefits of variable-speed control. When the method is used to forecast the energy-saving benefits of pumps whose specific speed is between 15 and 190 and the flow – between 5 and $200\,\mathrm{m^3/h}$ (subject to the condition $\overline{h} \leq 0.9$, $\overline{Q} \geq 0.5$), the error will not exceed 6.55% and thus the method proves to be of reference value for the evaluation of the variable-speed energy-saving benefits of other pumps.

Keywords: pump, back pressure, variable-speed energy-saving, forecasting

1. Introduction

Pipelines can be classified into two types according to the increase in energy resulting from the fluid passing through the pump.

One of them are systems with back pressure, where part of the energy increase resulting from fluid passing through the pump is used to overcome the pipeline's resistance while the remainder is required to boost the fluid's potential energy. Examples of such systems are found in delivery systems for high towers or boilers and in water-filled, pressure-limited systems used in heating systems. The characteristic curve of these systems is $H = h + SQ^2$. In this formula, H is the total energy increase in the pump (m), h is the back pressure, *i.e.* the increased energy resulting from fluid flowing from the system's entrance to its exit (m), Q is the flow ($\mathrm{m^3/h}$), and S is the resistance of the pipeline system.

The other are systems without back pressure, where all of the energy increase resulting from fluid passing through the pump is used to overcome pipeline resistances.

Examples of such systems are chilled water in air-conditioning systems, heating systems and other closed-loop water systems. The characteristic curve of these system is $H = SQ^2$, *viz.* back pressure $h = 0$. Through theoretical analysis and an exemplary calculation, it was demonstrated in [1] that the energy-saving benefits of variable-speed control are greater in systems without back pressure. In systems with back pressure, the energy-saving benefit of variable-speed control is reduced gradually as back pressure increases. With more exemplary calculations found in this paper, the relation between back pressure and the energy-saving benefits of variable-speed control is analyzed and an evaluation method is proposed to forecast these benefits, based on energy-saving benefit calculations for four different pump types.

2. Calculation of pump energy consumption

The output power of a pump is as follows:

$$N_0 = \rho g H \left(\frac{Q}{3600}\right) / 1000 \qquad (\mathrm{KW}), \tag{1}$$

where ρ is the water density, $\rho = 1000\,\mathrm{Kg/m}^3$, $g = 9.807\,\mathrm{m/s}^2$, H is the delivery lift (m), and Q is the flow (m^3/h).

The input power of the pump is:

$$N_p = \frac{N_0}{\eta_p}, \tag{2}$$

where η_p is the pump's efficiency.

The power of the motor is:

$$N_m = \frac{N_p}{\eta_m}, \tag{3}$$

where η_m is the efficiency of the motor.

The input power of a variable-speed drive is:

$$N = \frac{N_m}{\eta_v}, \tag{4}$$

where η_v is the drive's efficiency.

Synthesizing Equations (1), (2), (3) and (4), the energy consumption of the pump is:

$$N = \frac{N_0}{\eta_p \eta_m \eta_v} = \frac{HQ}{367 \eta_p \eta_m \eta_v}. \tag{5}$$

Obviously, in a system without a variable-speed drive $\eta_v = 1$.

3. Calculating the energy-saving benefits of variable-speed control

As shown in Figure 1, the pump's characteristic curve is ①. The system's characteristic curve is ②, namely $H = h + SQ^2$. Point A is the designed working point of the pump, for which the flow is Q_0, when the flow needs to be regulated to Q_1. If variable-speed control is adopted, the working point is point B and the characteristic curve of the pump becomes ③. However, if throttle control is adopted, the working point of the pump is point C. With the same flow objective, we use the difference in the

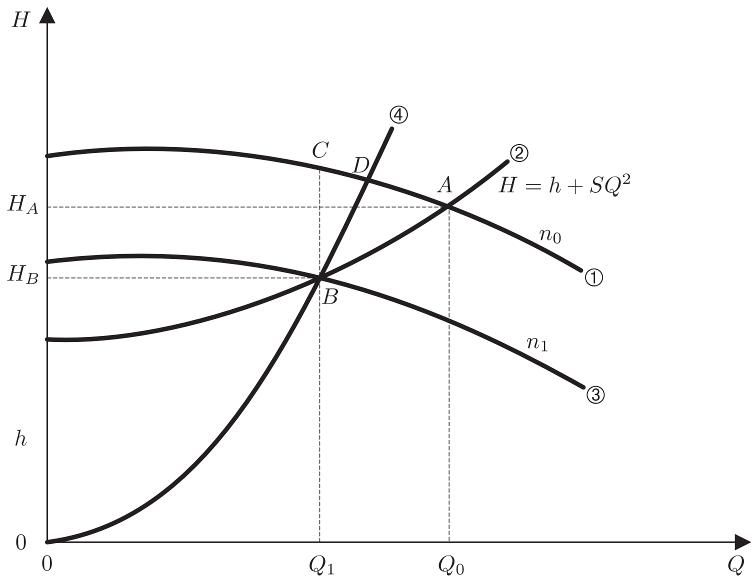


Figure 1. Sketch map of calculating model

power requirements of conditions C and B to reflect the variable-speed energy-saving benefit, $\Delta N = N_C - N_B$, as throttle control is the simplest and easiest to implement.

Subject to the designed working point A and flow goal Q_1 being definite, if the system's back pressure is different, then the variable-speed working point B is different, but the throttle control working point C is independent thereof, so $N_C - N_B$ is different. Selecting a different back pressure, h, by calculating the variable-speed energy-saving benefit, we can find the rules of its variation with back pressure.

4. Exemplary calculations and analysis

The characteristic curve of the ISG150-400 pump is as follows:

$$H = -0.0004861Q^2 + 0.1153Q + 46.39, \tag{6}$$

$$\eta_p = 1.221 \cdot 10^{-8} Q^3 - 2.26 \cdot 10^{-5} Q^2 + 0.007782Q. \tag{7}$$

Its designed working condition are: $H_A = 50\text{m}$, $Q_0 = 200\text{m}^3/\text{h}$, $\eta_A = 75\%$, $n_0 = 1450\text{r/min}$ (see Figure 1). The system's characteristic curve is $H = h + SQ^2$, which passes designed working point A by $H_A = h + SQ_0^2$, so that we can obtain:

$$S = \frac{H_A - h}{Q_0^2} = \frac{1}{4 \cdot 10^4}(50 - h). \tag{8}$$

Thus, the system's characteristic can be expressed as follows:

$$H = h + \frac{1}{4 \cdot 10^4}(50 - h)Q^2. \tag{9}$$

Obviously, if h is different, the pipeline is different. Once back pressure $h$and flow goal Q_1 are given and the variable-speed working condition B and the throttling working condition C are confirmed, $N_C - N_B$ can be calculated. Particular attention must be paid to fixing the efficiency of working point B: creating a similar parabola ④ which passes point B and intersects the curve ① at point D; then the efficiency of working points B and D is equal.

With regard to the efficiency of the motor and the variable-speed drive, we shall adopt the approximate formulas given in [2]:

$$\eta_m = 0.94187 \cdot \left(1 - e^{-9.04k}\right), \tag{10}$$

$$\eta_f = 0.5087 + 1.283k - 1.42k^2 + 0.5834k^3, \tag{11}$$

where k is the speed ratio, or the ratio of the speed of the flow objective to the original speed. More detail of the k calculating method, can be found in [1].

The motor efficiency of the throttling working point C can be calculated with formula (10), subject to $k = 1$. The results so calculated are listed in Table 1.

Table 1. Calculated results of $N_C - N_B$ (KW)

Q_1 (m^3/h) \ h (m)	0.0	10.0	20.0	30.0	40.0	42.0	44.0	46.0	48.0
100.0	21.205	17.816	13.955	9.700	5.134	4.190	3.239	2.280	1.313
120.0	19.579	16.241	12.604	8.715	4.624	3.787	2.944	2.096	1.243
140.0	16.586	13.567	10.391	7.080	3.662	2.968	2.271	1.571	0.868
160.0	12.012	9.641	7.211	4.729	2.205	1.697	1.186	0.675	0.163
180.0	5.710	4.351	2.980	1.601	0.214	–0.065	–0.343	–0.622	–0.901
190.0	1.896	1.176	0.455	–0.269	–0.994	–1.139	–1.284	–1.430	–1.575

It follows from Table 1 that, with the same flow objective, as back pressure increases, the variable-speed energy-saving benefit is reduced. Taking 160m^3/h as an example of the flow objective, when $h = 0$, $\Delta N (= N_C - N_B)$ is 12.012KW, when $h = 20$, ΔN is 7.211KW, when $h = 48$, ΔN is reduced to 0.163KW. The conclusion is in agreement with [1], obtained through academic analysis.

As a whole, the data of Table 1 suggest the following rules. The nearer to the top left corner, or the smaller the back pressure, h, and the flow objective, Q_1, the greater ΔN. On the contrary, the nearer to the top right corner, or the greater h and Q_1 is, the smaller ΔN. ΔN decreases when Q_1 increases, the reason being that, as Q_1 increases, working points B and C are getting closer and closer to each other. ΔN decreases with increasing h for reasons given in [1] corroborated by our calculated results. It was proposed in [1] that, when back pressure increases to reach a certain magnitude, the difference between pump shaft power of working points B and C is reduced (though the shaft power of working point C still exceeds that of B), but when the variable-speed drive is considered, the energy consumption of working point B is slightly greater than that of C.

We can infer from the above analysis that the variable-speed energy-saving benefit is closely related to the system's back pressure and the flow objective, and thus under certain conditions there will practically no notable benefit. Therefore, when deciding issues of variable-speed control of a pumping system, the variable-speed energy-saving benefit must be evaluated and forecast according to the system's characteristic and the flow distribution during an operational cycle.

5. Forecasting the variable-speed energy-saving benefit

Let us propose a method of evaluation to forecast the variable-speed energy-saving benefit conveniently and quickly for various types of pumps and pipeline systems (of varying back pressure, h).

We shall deal with back pressure, flow and the energy-saving benefit by the following non-dimensional method.

In Figure 1, $\overline{h} = \frac{h}{H_A}$ is the non-dimensional back pressure, $\overline{Q} = \frac{Q_1}{Q_0}$ – the non-dimensional flow, $\Omega = \frac{N_C - N_B}{N_C}$ – the non-dimensional energy-saving benefit of variable-speed control.

The type and rating working conditions (RWC) of four different pumps of widely different performance are listed in Table 2. The rating working conditions are taken as the system's designed working conditions, or working points A of Figure 1.

Table 2. RWC and Type of four different pumps

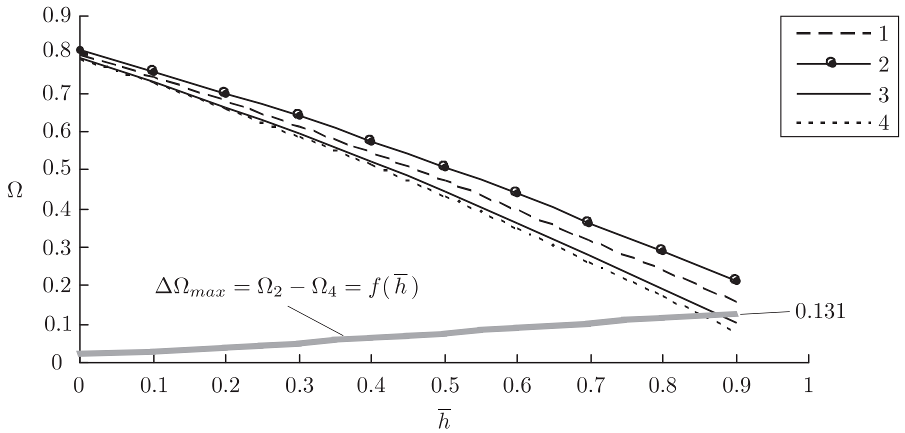
Number	RWC / Type	Flow (m^3/h)	Lift (m)	Efficiency (%)	Speed (r/min)	Specific Speed (Ns)
1	ISG65–100 (I)	50	12.5	73	2900	188
2	ISG80–160 (I)	100	32	76	2900	131
3	ISG150–400	200	50	75	1450	66
4	ISG40–250A	5.5	70	26	2900	17

Figure 2 shows the calculated results of Ω changes with $\overline{h}$ and $\overline{Q}$. We can see that Ω differs for the four pumps, being the greatest for the ISG80–160 (I) pump and the smallest for the ISG40–250A. The Ω difference curve of the two pumps is also shown in Figure 2 as $\Delta\Omega_{\max} = \Omega_2 - \Omega_4 = f(\overline{h})$. As $\overline{h}$ increases, this curve gradually ascends. Comparing Figures 2a, 2b and 2c, we can see that $\Delta\Omega_{\max}$ increases as $\overline{Q}$ decreases. Actually, $\Delta\Omega_{\max}$ with increasing $\overline{h}$, which in turn increases with decreasing $\overline{Q}$. However, when $\overline{h} = 0.9$, $\overline{Q} = 0.5$, $\Delta\Omega_{\max}$ is only 0.131. Therefore, although the performance of the four pumps differs widely, the energy-saving benefit of their variable-speed control is only marginally different. Thus, we take the average Ω of pumps ISG80–160 (I) and ISG40–250A, forming Table 3. When Table 3 is used to evaluate the four pumps' variable-speed energy-saving benefit, with $\overline{h} \le 0.9$, $\overline{Q} \ge 0.5$, the maximum error will not exceed 6.55%. According to our results, provided that ratio Ns is between 15 and 190, flow is between 5 and 200 m^3/h, and $\overline{h} \le 0.9$, $\overline{Q} \ge 0.5$, using Table 3 to calculate the pumps' variable-speed energy-saving benefit, will involve error of no more than 6.55%. We also expect pumps exceeding this range to keep the the variable-speed energy-saving benefit values of Table 3.

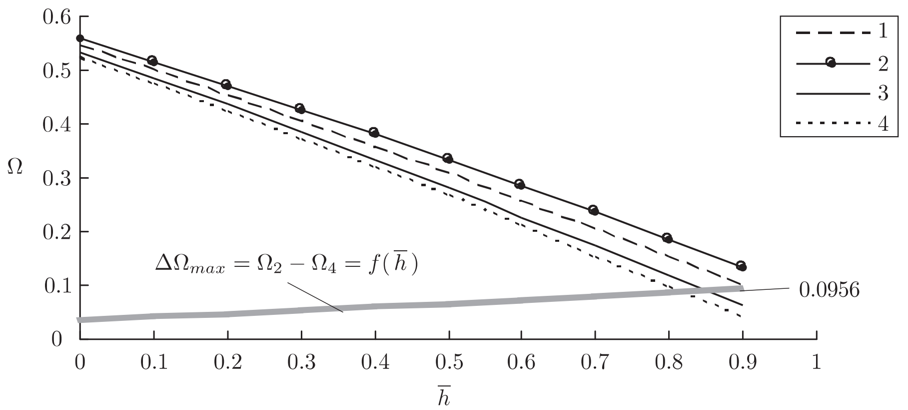
6. Conclusions

1. The variable-speed energy-saving benefit of a pump decreases as the system's back pressure increases, and increases as flow goal decreases (according to the present calculations, when the flow is not less than 50% of the rating flow).

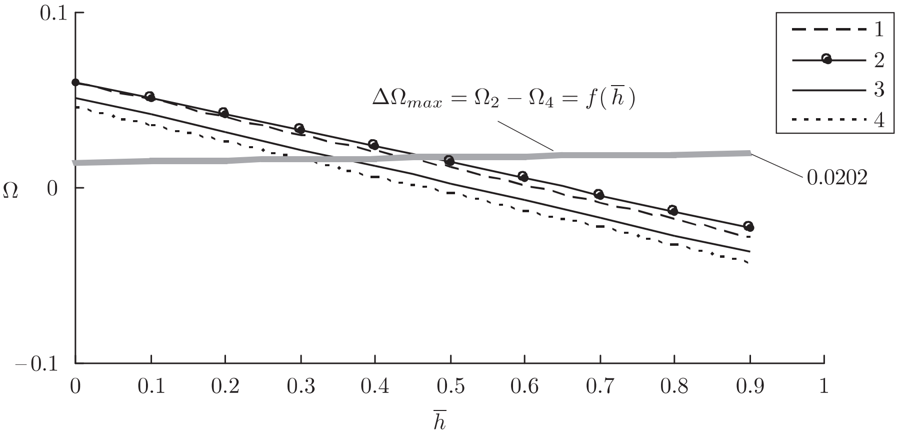
2. The variable-speed energy-saving benefits of different pumps are similar when their non-dimensional back pressures and flows are equal.



(a) $\overline{Q} = 0.5$



(b) $\overline{Q} = 0.7$



(c) $\overline{Q} = 0.95$

Figure 2. Ω changes with $\overline{h}$ of four pumps' variable-speed control

Table 3. Evaluation table of variable-speed energy-saving benefit, Ω

$\overline{Q}$ \ $\overline{h}$	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.5	0.7983	0.7412	0.6794	0.6131	0.5428	0.4689	0.3918	0.3118	0.2292	0.1444
0.6	0.6847	0.6315	0.5753	0.5166	0.4553	0.3919	0.3265	0.2593	0.1905	0.1204
0.7	0.5404	0.4949	0.4478	0.3995	0.3499	0.2991	0.2474	0.1947	0.1412	0.0869
0.8	0.3656	0.3317	0.2973	0.2625	0.2271	0.1914	0.1552	0.1187	0.0819	0.0448
0.9	0.1628	0.1444	0.1260	0.1075	0.0889	0.0702	0.0515	0.0327	0.0138	–0.0052
0.95	0.0527	0.0433	0.0338	0.0244	0.0149	0.0054	–0.0041	–0.0137	–0.0232	–0.0327

3. If $15 \leq \mathrm{Ns} \leq 190$, the flow is between 5 and 200 m^3/h, and $\overline{h} \leq 0.9$, $\overline{Q} \geq 0.5$, then the usage of Table 3 to evaluate a pump's variable-speed energy-saving benefit will involve an error not greater than 6.55% and thus the method proves to be of reference value for the evaluation of the variable-speed energy-saving benefits of other pumps.

References

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