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ABOUT VARIABLE SPEED HEATING AND COOLING PUMPS

BY

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Abstract. The present work has the purpose of underlying the advantages of variable speed heating and cooling pumps use for the perspective of general and particular pumping costs and efficiency.

The study approaches comparisons between constant flow pumps and variable flow pumps in different given situations and comparatively analyses the pumping costs.

Key Words: variable speed pumps; heating; cooling; pump head; pumping cost.

1. Introduction

By changing the rotation speed, the pump head may be modified and can therefore be selected to suit the requirements, which can also reduce noise.

Lowering the pump head, when possible also lowers the pumping power costs.

Several applications of variabile speed pumps can be considered.

2. Maintaining a Constant Pump Head

At low loads, pressure drops in pipes and equipment reduce and the differential pressure in the most remote circuits increases (Fig. 1), correspondingly reducing the authority of the control valves which could also become noisy. A constant speed pump increases the pump head (from *A* to *B* - Fig 2), which worsens this problem.

It is therefore beneficial to use a variable speed pump which maintains its pump head. This gives a pump with a "flat" characteristic.

Hydronic balancing remains essential to avoid overflows in some parts, which create underflow elsewhere, and to obtain flow compatibility between the production and the distribution, making the maximum heating or cooling power installed transmittable.

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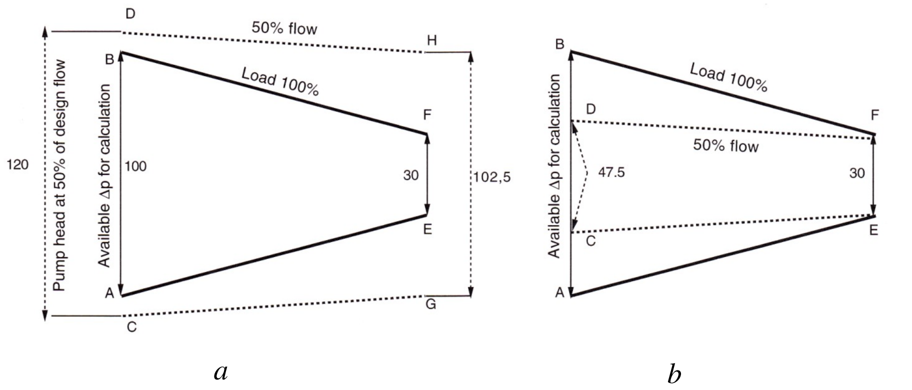

Fig. 1. – Evolution of the differential pressure according to the load: *a* – constant speed pump; *b* – variable speed pump.

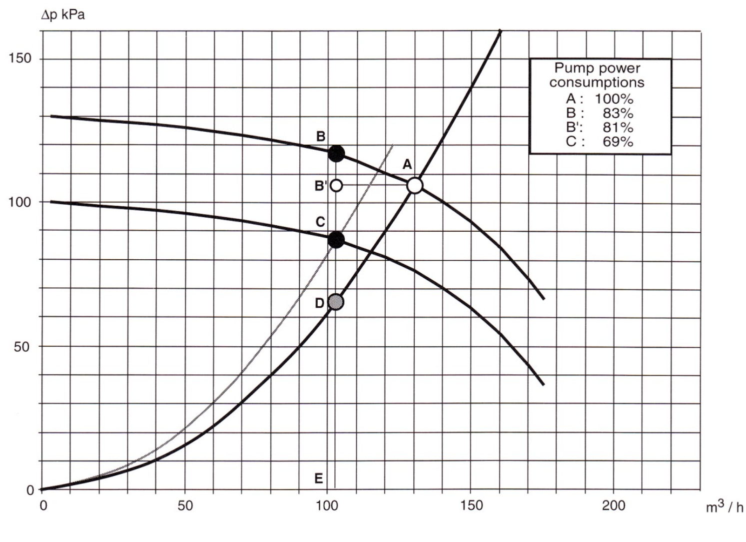

Fig. 2. – The pump characteristic changes with the speed of the motor.

3. Reduction of the Pump Head During Night Time

In heating, the water temperature is usually lowered during the night in order to reduce the room temperature and therefore the energy consumption. During this period, the control valves on terminal units (for example thermostatic valves) open fully as the room temperatures are below their set points. The plant therefore works at maximum flow. The pump power consumption is unusefully high and the circulation noises may become intolerable.

A variable speed pump can generate a lower pump head during the night, simultaneously reducing both pumping costs and noises. The speed of the pump is normally managed between 50 and 100% of nominal value.

The same function can be obtained with two pumps in parallel, the smallest one being used during night time.

4. Reducing the Pump Head at Low Loads

In a heating plant it would seem reasonable to reduce the pump head in spring and autumn. Controlling the water temperature to suit external conditions should take account of this situation. When the plant is controlled by an optimizer with self-adapting functions, the calculation algorithm must take into account the change in the pump head, otherwise the optimizer will search in vain for a suitable heating curve.

The maximum pump head can also be reserved for the highest loads (start up for instance) and a lower pump head is adopted in other conditions.

In variable flow circuits with constant supply water temperature (cooling plants), most 2-way control valves close at low loads. It therefore appears logical in this case to reduce the pump head. The result is a reduction in pumping costs which depends on the product Hq, and both components of this product are reduced simultaneously. For constant speed pumps, the flow alone reduces with the load and the pump head increases.

However, the keyword is ”most”. This is the heart of the problem.

If 50% of the units are stopped while 50% are at full load, an average load drop of 50% is measured. Reducing the pump head under these conditions would make it impossible for units which require full load, to obtain it.

The variation in the average load may be estimated by measuring the total flow, by the change in the differential pressure at the centre of the plant, by detecting return water temperature variations..., however the problem remains. An average is not necessarily representative of all circuits.

In Fig2, the design conditions are represented by the point A which is the cross point between the pump characteristic and the plant characteristic. This plant characteristic is obtained when all the control valves are fully open.

When the control valves start to shut, the water flow decreases and the pump head reaches the value B with a constant speed pump. With a variable speed pump, the head can be reduced theoretically to D, forcing the control valves to reopen fully. However, in this case, it is not possible to adjust the pump speed to increase the head when the flow increases because the flow can not increase anymore as the control valves cannot open beyond 100%. Consequently, the pump head reduction is limited to C, for instance and the pumping costs are reduced in the proportion of the pump heads, EC/EB. This has also to be corrected according to the evolution of the pump and the motor efficiencies which are both decreasing. The reduction of the pump head can be obtained in several ways.

4.1. To Maintain Constant the Differential Pressure Across the Last Terminal Unit

What happens in a plant working with a constant speed pump?

The control valves close to the pump are calculated to take a pressure drop, at design flow, of at least 50% of the available Δp_{AB} (Fig 1). At small loads, the available Δp increases to Δp_{DC} and the control valve authority is reduced by the factor $\Delta p_{AB}/\Delta p_{DC} = 100/120 = 0.83$, which is normally acceptable. Remember that the valve authority = (Δp in control valve fully open and design flow) divided by (Δp in the control valve fully shut) . For good control, this authority, in the worse case, has normally to be higher than 0.25.

For the last terminals, the control valves are calculated to take a pressure drop, at design flow, of at least 50% of the available Δp_{EF} this means 15 kPa in the example chosen. At small loads, the available increases to HG (102.5 kPa) and the control valve authority is reduced by the factor $\Delta p_{EF}/\Delta p_{GH}$, which is sometimes unacceptable. In this example, the valve authority varies from 0.5 to $0.5 \times 30/102.5 = 0.15$ and the proportional band of the controller has normally to be doubled to obtain the same control loop stability.

The situation can be improved by increasing the pump head slightly, this supplement being taken in the control valves. If the pump head is increased by 14 kPa, the valve authority in design condition becomes $(15+14)/(30+14) = 0.66$. At small loads, this authority is reduced to $(15+14)/(102.5+14) = 0.25$, which remains acceptable.

When one circuit has a specially high pressure drop, it would be beneficial to install a secondary pump instead of increasing the pump head of the main pump.

Another approach to the problem is to install a variable speed pump to maintain constant the differential pressure Δp_{EF}. However, at small loads, all Δp are reduced practically to Δp_{EF} and the terminals close to the pump can not reach their full load if necessary.

If these control valves are selected based on the minimum differential pressure, they will be oversized when the plant is working at full load. This could make start up problematic.

Example

Pump power consumption, $\% = k\times$ pump head % $\times$ flow % / $(\eta_{\text{pump}}\eta_{\text{motor}}) =$ $= kHq/(\eta_{\text{pump}}\eta_{\text{motor}})$.

P_M full load = $k \times 100 \times 100/(0.8 \times 0.9) = 100\%$ with $k = 0.0072$.

For a water flow of 50%:

With a constant speed pump: $P_c = k \times 120 \times 50/(0.65 \times 0.86) = 77\%$.

With a variable speed pump: $P_v = k \times 47.5 \times 50/(0.6 \times 0.48) = 59\%$.

The use of a variable speed pump, in these conditions, reduces the pumping costs, calculated in % of maximum value of

$$\frac{P_c P_v}{P_M} 100 = 18\%. \tag{1}$$

4.2. To Maintain Constant the Differential Pressure in the "Middle" of the Plant

The pump head is controlled to maintain constant the differential pressure at the centre of the plant. In comparison with constant speed pump, this is particularly beneficial for central and remote circuits in which the increase in the differential pressure is limited.

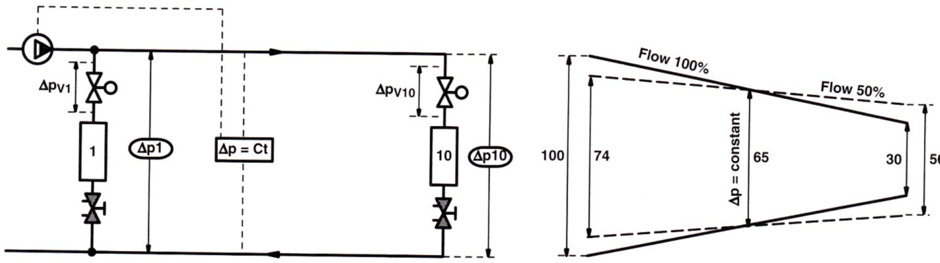

Fig. 3. – A variable speed pump controlled by the differential pressure in the middle of the plant.

At low average loads, the differential pressure is reduced for circuits close to the pump. If balancing has been designed for the maximum differential pressure, these circuits will no longer be able to provide the maximum flow. It may be necessary for some circuits to obtain full load even though the majority of the plant does not. If these circuits are designed based on the minimum differential pressure, they will be oversized at high loads, when the pump is running at full speed, and the resulting overflows may induce underflows in remote circuits If these circuits are designed based on the minimum differential pressure, they will be oversized at high loads, when the pump is running at full speed, and the resulting overflows may induce underflows in remote circuits.

However, when the pressure drops in pipes represent less than 70% of the pump head, this solution is applicable and the plant is balanced in design condition. As the differential pressure variations are reduced by half, in comparison with a constant speed pump, the minimum authority of the control valves is generally improved for the same design pump head.

5. Pumping Costs

It's easy to calculate the pumping power consumption in a balanced plant working with constant water flow. When working with variable flow in the distribution, the pumping costs decrease to reach normally a value of less than 50% of the one calculated hereafter.

5.1. Pumping Power Consumption

The pumping consumption

$$P_c = \frac{9.81Hq}{3600\eta_p\eta_m}, \quad (2)$$

were: P_c, [W], H – pump head, [m WG], q – water flow, [Kg/h] (approximately water flow in 1/h); η_p – pump efficiency = 0.75 for instance (can be less than 0.4 for small circulators); η_m – motor efficiency = 0.85 for instance.

Finally,

$$P_c = \frac{Hq}{367\eta_p\eta_m} \approx \frac{Hq}{234}W. \quad (3)$$

The efficiency for variable speed pumps, including the motor and variable frequency inverter, may drop to 0.67 at 50% load to 0.35 at 15% load, as for example.

5.2. Pumping Costs

The pumping costs

$$C_{\text{pt}} = \frac{P_c t C_w}{1,000}, \quad (4)$$

with t – running time, [h], C_w – price of electricity for one kWh.

5.3. Real Pumping Costs (Constant Flow Distribution)

a. Heating

1• The energy lost by the motor is $(1-\eta_m)P_c t C_w/1,000$.

2• The rest of the power ($\eta_m P_c$) is changed into heat in the water, therefore it not a loss. The cost of this energy is $\eta_m P_c C_w/1,000$.

3• If this energy was provided by a boiler, the cost would be $\eta_m P_c t C_f)/(1,000 \times \times 12\eta_b)$, where C_f – price of one litre of fuel with $C_f/C_w = 1.9$ for example $-12 = 12$ kWh in one litre of fuel; η_b – seasonal efficiency of the boiler = 0.75 for instance.

The real pumping costs $= 1^\bullet + 2^\bullet - 3^\bullet$

$$C_{pr} = C_{pt}\left(1 - \frac{C_f\eta_m}{12C_w\eta_b}\right) \approx 0.82C_{pt} = \frac{HqtC_w}{285,366}. \quad (5)$$

b. Cooling

1• The energy lost by the motor is $(1-\eta_m)P_c t C_w/1,000$.

2• The rest of the power ($\eta_m P_c$) is changed into heat in the water.

3• The cost of this energy is $\eta_m P_c C_w/1,000$.

This heating energy has to be compensated by the chillers which have a performance coefficient, COP (including the motor efficiency) on the evaporator side of 3 for instance. This costs on the chiller side is $\eta_m P_c t C_w/1000$.

The real pumping costs is $1^\bullet + 2^\bullet + 3^\bullet$

$$C_{pr} = C_{pt}\left(1 - \frac{\eta_m}{\mathrm{COP}}\right) \approx 1.3C_{pt} = \frac{HqtC_w}{180,000}. \tag{6}$$

5.4. Pumping Costs in Percent of the Plant Seasonal Consumption

The maximum heat output of the plant $= 1.16q\Delta T_C$, [W].

The mean seasonal load $= 1.16qS_c\Delta T_C$, were S_c – ratio between the mean seasonal load and the maximum designed load = 0.4 for instance.

a. Heating

The heating costs are

$$\frac{1.161qS_c\Delta T_C C_f}{12\times 1,000\cdot\eta_b}. \tag{7}$$

Real pumping costs, in percent, of the plant seasonal consumption $= 100 \times$ (5)/(6) namely

$$C_{pr\%} = \frac{H}{\Delta T_c}\frac{0.235}{S_c\eta_p}\left(\frac{12C_w\eta_b}{C_f\eta_m} - 1\right) \approx 3.58\frac{H}{\Delta T_c}. \tag{8}$$

Example. For $H = 10$ m WG and ATC $= 20$ K, $C_{pr} = 1.8\%$ and the distribution pumping costs represent in this plant, working with constant flow, 1.8% of the fuel costs.

b. Cooling

The cooling costs are

$$\frac{1.161qS_c\Delta T_C C_W}{1,000\mathrm{COP}}. \tag{9}$$

Real pumping costs, in percent, of the plant seasonal consumption $= 100 \times$ (6)/(9)

$$C_{pr\%} = \frac{H}{\Delta T_c}\frac{0.235}{S_c\eta_p}(\mathrm{COP} + \eta_m) \approx 3.58\frac{H}{\Delta T_c}. \tag{10}$$

In cooling, pump heads are generally higher than in heating and ΔT_C are smaller.

Example. For $H = 20$ m WG and ATC $= 5$ K, $C_{pr} = 14.3\%$ and the distribution pumping costs represent in this plant, working with constant flow, 14.3% of the average energy consumption on the chillers side.

6. Distribution Pumping Cost Compared with the Energy Cost, for a Difference of 1 K Between the Room Temperature and the Comfortable Temperature

In heating, a 1 K increase in the room temperature above the required one will cause an increase in consumption which may be estimated by the following relation

$$S,[\%]=\frac{100}{S_c\,(t_{ic}-t_{ec}-ai)}, \tag{11}$$

where: t_{ic} is the design room temperature (20°C); t_{ec} – design outside temperature (-10°C); ai – internal heat gain expressed in degrees of influence on the room temperature (2K); S_c – ratio between the average seasonal heating power and the maximum necessary power (0.4).

With the values used as an example, $S\% = 9\%$, which represents more than four times the pumping consumption in the distribution.

In practice, this relation gives an underestimate value for an apartment building since the increase in the room temperature causes additional exchanges between the apartments. These exchanges are ignored in the above relation.

In cooling

$$S,[\%]=\frac{100}{S_c\,(t_{ec}-t_{ic}+ai)}. \tag{12}$$

For $S_c = 0,4$, $t_{ec} = 35$°C , $t_{ic} = 23$°C , $ai = 4$ K, we get $S = 16\%$, which is equivalent to the pumping cost in the distribution.

In conclusion, any actions intended to reduce pumping consumption must be taken so that they do not adversely affect the operation of terminal unit control loops.

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DESPRE POMPELE CU TURAŢIE VARIABILĂ A SITEMELOR
DE INCĂLZIRE ŞI RĂCIRE

(Rezumat)

Scopul prezentei lucrări este acela de a sublinia avantajele utilizării pompelor cu turaţie variabilă atât pentru instalaţii de încalzire cât şi pentru răcire din perspectiva costurilor generale şi particulare de pompare şi a eficienţei.

Se abordează unele comparaţii între pompele cu turaţie constantă şi cele cu turaţie variabilă în diferite situaţii şi se analizează comparativ costurile pompării.