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An Analytical Approach to Optimize the Performance of the Bypass for Two-Way Control Valves in Chilled Water Central Air-Conditioning Systems

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Abstract
Primary-secondary pumping system is widely utilized in the Heating, Ventilating, and Air Conditioning industry. Such system has an energy saving potential over the constant speed primary pumping system. However, the system uses a surplus amount of energy that can be saved, the low delta T syndrome and low pumping efficiency are the main energy wasting reasons. The implementation of a proper design and novel control strategy can eliminate these two problems, knowing that the design of the bypass (Decoupler Bridge) is still a topic of debate. Thus, the focus of this paper will be enhancing the design and control of the bypass, developing an energy efficient control for the secondary loop including the pumps and the two way control valves, and setting an energy efficient sequencing of operation for the primary and secondary pumps. Calculating the bypass pipe diameter and its control strategy are often estimated according to the designer experience, rather than a code of practice, the goal for this paper is setting a guide for the bypass design.

Keywords – Primary-Secondary; Bypass; Two way control valve; Decoupler bridge; Air Conditioning

1. Introduction

The primary-secondary pumping system is one of the most famous configurations that are being used, for its proven energy savings and its familiarity to the engineers and the designers. The pumps regulate the water volume flow rate moving through the piping network, by changing the rotational speed of the pump, therefore reducing a huge amount of energy. In high rise buildings and large sized networks the pressure drop across the pipping network is relatively high, therefore a high head pump should be selected in order to overcome the pressure loss in the system, which makes it hard for the chiller to withstand this hydraulic head, so the pumping terminals are divided into two; primary and secondary pumps, where the primary pumps are constant speed pumps to maintain a constant volume flow rate through the chillers, and variable speed high head pumps for the secondary pumps. In order to decouple the two pumping sets, a bypass is installed, it is also known as the decoupler bridge.

This configuration has proven its energy saving potential and sustainability compared to the constant speed pumping system, but this comes with a more problematic control [1]. The presence of secondary pumps with variable speed drives, two way control valves, and the bypass makes the control strategy and the set points much more complex resulting in performance and efficiency related complications, identified as “low delta T syndrome”. This happens when chillers operate on an inadvisable low temperature difference at part load conditions, because of the cold flow bypassing the loads mixed with the return water before entering the chillers [2]. Many modifications were suggested to reduce the effect of this problem [3], but the bypass design and control have always been a debate.

When designing the primary-secondary pumping system, there are two points of view according to designers; one favors the reduction of the pressure drop across the bypass as much as possible, designing the diameter of the bypass to be equal to the diameter of the main pipe. This achieves a nearly zero pressure drop across it. While the other design strategy suggests that the pressure drop across the bypass is beneficial for the system, accordingly the pressure drop is increased by installing valves on the bypass, usually a check valve or a flow regulating control valve is used [4].

For studying the problem, a model was constructed for a primary-secondary system, applying various controls strategies and bypass designs, in order to optimize the performance of the bypass.

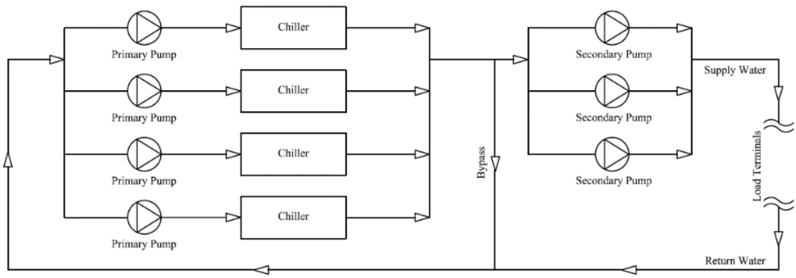

Fig. 1 A schematic diagram for the primary-secondary configuration

2. Model Development

An analytical approach is used to find the most efficient design, where the model simulates the water distribution loop, pumps, chillers and control valves. Pressure drop in the system is divided into three categories; speed dependent, friction loss in pipes, and controlling pressure drops. Pumps are modelled by curve fitting to real pump performance curves. Control valves are modelled with their non-linear flow characteristics.

2.1 Pipes

The pressure drop in circular pipes is represented by using the Darcy-Weisbach equation,

$$h = f\ \frac{L}{D}\ \frac{v^2}{2g} \tag{1}$$

Where h is the head loss in m; L is the pipe length in m; D is the pipe diameter in m; v is the average fluid speed inside the pipe in m/s; g is the gravitational acceleration m/s^2, and f is the friction factor determined according to moody chart. The absolute roughness for the pipes is based on schedule 40 steel pipes.

2.2 Speed dependent friction losses

Pressure drop in the components like fittings, valves, strainers etc., varies directly with the square of the average water flow speed, representing it as constant resistance coefficient, K values as shown in (2),

$$h = K\ \frac{v^2}{2g} \tag{2}$$

2.3 Pump

The head curve for the pump is represented by curve fitting. An actual centrifugal pump performance curve is used and can be plotted as a polynomial equation [5],

$$h_p = c_1 + c_2 Q + c_3 Q^2 + c_4 Q^3 \tag{3}$$

Where h_p is the pump head in m; Q is the volume flow rate in m^3/s and c_1, c_2, c_3, c_4 are constants. A variable speed drive model is developed for the various ways of control. Affinity laws for a pump of the same impeller are used for modeling the performance curves at different rotational speeds,

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \tag{4}$$

$$\frac{H_1}{H_2} = \frac{N_1^2}{N_2^2} \tag{5}$$

Where Q_1 and Q_2 are the volume flow rate for the pump; H_1 and H_2 are the hydraulic heads for the pump at the corresponding rotational speeds N_1 and N_2. To calculate the efficiency of the pump, a set of empirical equations for centrifugal pumps are used, where the efficiency η depends on the dimensionless volume flow rate number ζ [6],

$$\eta = b_1\, \zeta - b_2\, \zeta^2 \tag{6}$$

$$\zeta = b_3\ \frac{Q}{N} \tag{7}$$

Where b_1, b_2, b_3 are constants depending on the dimensions of the impeller. For the primary pumps the efficiency was curve fitted directly from the efficiency curve of the pump, as the rotational speed is constant.

2.4 Control valves

Globe two way valves are used to model the flow regulating control valves in this system, where its *K* values with respect to opening are calculated using the equations developed by Crane [7,8,9,10],

$$K_{cv} = K_{cvf} + \frac{0.5(1-\beta^2) + (1-\beta^2)}{\beta^3} \tag{8}$$

$$K_{cvf} = 340\, f_t \tag{9}$$

$$f_t = \frac{0.25}{log^2\left[\frac{RR}{3.7}\right]} \tag{10}$$

Where K_{cv} is the representative resistance coefficient *K* for the valve at any given opening; K_{cvf} is the representative resistance coefficient *K* at the fully opened position; β is the percentage opening; f_T is friction factor for the pipe; *RR* is the relative roughness of the pipe.

Each valve opening at full and part load condition, depends on almost all other system components. The valve adjusts its opening with an objective to regulate the flow in each heat exchanger to meet the cooling load requirements in each zone, where a thermostat is in charge of controlling the valve opening. This was modeled by setting the volume flow rates for each two way control valve corresponding to its cooling load.

3. Testing Conditions

The model is used to simulate the whole system, where the cooling loads vary from full load to ten percent, more than one system with different volume flow rates are used to test a wide range, starting from 360 up to 1800 cubic meter per hour of chilled water. A single individual primary pump for each chiller is used, with three secondary pumps as shown in Fig. (1), while the number of load terminals varies with volume flow rate for each system. In order to estimate the pumping energy saved for different designs, the average pumping power is introduced as the algebraic average of all part load pumping powers. The average pumping power gives an overall indication for the pumping energy. Actual energy savings should be based on real load plans for each system. Simulations are made to study the effect of the diameter of the bypass on the pumping efficiency, while applying different control strategies.

4. Results and Findings

4.1 Primary pumps sequencing

Primary pumps are one of the main components that needs to be considered when designing a bypass, where the number of pumps in operation alongside their control scheme can directly affect the volume flow rate in the bypass. If an inconvenient sequence of operation is used, and the system needs more volume flow rate than what the primary pumps are suppling, the volume flow rate direction inside the bypass will be reversed. This case must be avoided, in order to have a functioning cooling system.

The performance of the system can be evaluated from the flow inside the bypass, where it's better to keep the flow inside as minimum as possible while avoiding reversed flow for having the best pumping efficiency. The graph below shows the effect of using new set points for sequencing in order to make it near zero flow inside the bypass, by setting a novel control on the primary pumps rather than the flow, temperature, or pressure dependent control [11,12]. The novel control adds the volume flow rate inside the bypass to the control parameters. This configuration can reduce the primary pumps energy to around 4% as an average power consumption.

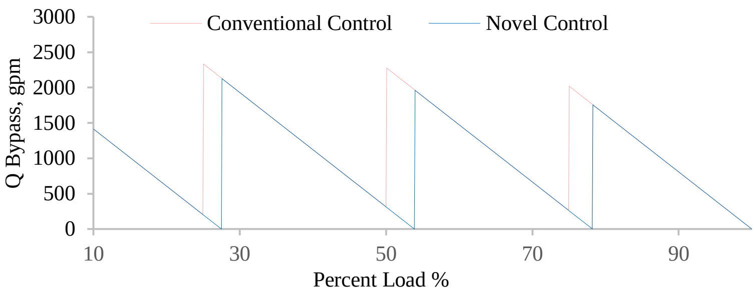

Fig. 2 The effect on the volume flow rate in the bypass with changing the primary pumps operation set points

In the novel control scheme, the unused primary pump shuts down earlier than in the conventional control scheme, thus suppling only the amount of volume flow rate needed by the system. The number of primary pumps operating can be observed from Fig. (2). For example, the conventional control operates four pumps from full load till 75% load, where around 200 gallons per minute are flowing in the bypass just after shutting down the first pump. However, the novel control scheme shuts down one pump earlier at 78% load to reduce the flow in the bypass.

4.2 Secondary pump sequencing

A given point of operation can be obtained by any number of secondary pumps running. This happens due to the variable speed drive, where the performance curves change according to the speed of the impeller in order to obtain the desired point of operation. As a result, the flow rate inside the bypass is ignorant to the secondary pumps sequencing of operation.

The factor that can be taken into consideration is the power consumption and it depends on the number of pumps and their efficiency with respect to their speeds.

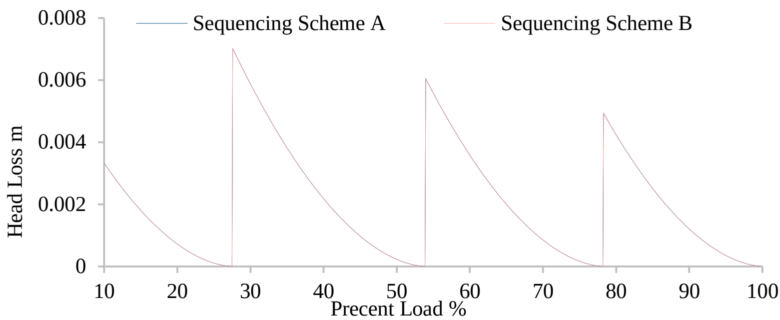

Fig. 3 The pressure drop inside the bypass for different secondary pumps sequencing strategies

Fig. (3) shows the pressure difference across the bypass with different number of pumps in operation. Analytically the same pressure difference and volume flow rate were obtained in both sequencing strategies.

The most efficient sequencing scheme can be obtained by comparing the power consumption for the set of pumps, with different number of pumps running in order to have the same operation point. And then determine the set points for secondary pumps sequencing.

4.3 Secondary pumps control

Secondary pumps control strategies, can be summarized into three types; constant pressure control at the discharge of the pump, differential pressure sensor at the furthest point in the system, and critical valve rest [13].

The constant pressure control is the simplest, however the highest energy consuming strategy. As the variable speed drive of the pump acts to maintain a constant pressure at the sensor, and the excess pressure is demolished by the control valves by being partially opened.

The differential pressure at the furthest point in the system is more energy efficient, but it is not be the best option for highly diverse loads. Because only one control valve controls the pressure of the system [14].

Using critical valve reset method has proven its energy saving potential over the other control strategies. In addition it has an outstanding performance over diverse loads, where the building automation system polls each valve lift, changing the pressure of the pump to make at least one valve nearly full opened, while the other valves are partially opened [15].

Testing the three control schemes using the proposed model, it was found that the control strategy of the secondary pump doesn’t affect the flow in the bypass. The secondary pump and the control valves act to expel the effect of each other. This happens only under the condition when the pressure drop inside the bypass is relatively small, and its effect can be neglected. This high pressure drop will be present at smaller diameters which will be discussed below.

4.4 Bypass dimensions

The diameters of the pipes are usually chosen according to their volume flow rate, with a conventionally recommended maximum velocity range varying from 0.5 to 3 meters per second. Accordingly knowing the maximum flow rate in the bypass will directly determine its diameter. Regarding the primary pumps set points mentioned in this paper, the maximum flow in the bypass is at the point where only two primary pumps are operating and the water volume flow rate needed for the system is just above the operating set value of single pump operation. At this condition the friction is the lowest compared to the multi-pump operation, due to the relatively low volume flow rate flowing in the large pipes. The volume flow rate in the primary circuit is nearly double that of the secondary circuit, forcing almost half of the primary volume flow rate to pass through the bypass. These flow rates can be easily calculated based on the primary system friction loss. To maintain the recommended maximum speed range, the bypass diameter needs to be equal to the diameter of the pipe connected to a single primary pump.

However, the more the friction in the bypass the less the primary pump power, since the pump head increases resulting in less total volume flow rate, due to the reduction in bypass volume flow rate. Therefore the diameter should be even smaller than the pipe of the primary pump, but there is a limit for reducing the diameter, which is the erosive-corrosive velocity determined from the equation below [16],

$$v_c = \frac{c}{\rho^{0.5}} \tag{11}$$

Where v_c is the maximum velocity to avoid the erosion of the pipe in ft/s, c is an empirical constant that varies from 100 to 200 for solid free flow, ρ is the density of the fluid in lbs/ft^3, from (11) the maximum velocity can reach 3.8 to 7.7 m/s.

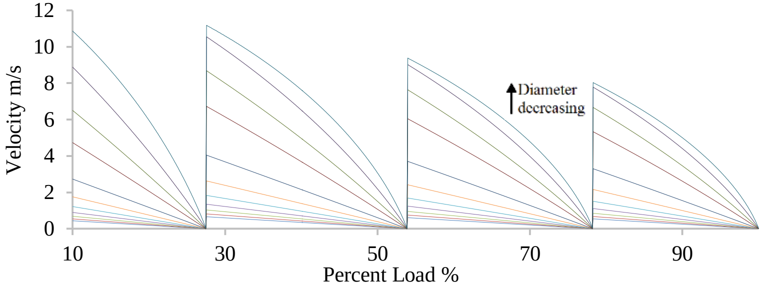

Fig. 4 The effect on the bypass average flow velocity in variable pipe diameters

Fig. (4) illustrates the effect of the reduction of the bypass diameter with respect to flow velocity. The maximum velocity is at a bypass diameter between 10 to 15 % of the main pipe diameter. The reduction of its diameter can save up to 5% power for the primary pump. In addition it will reduce the secondary pumps power consumption, where the high pressure at the secondary pump suction will result in reducing the secondary pumps head, saving around 8% of the secondary pump power on the constant pressure control strategy. The energy saving potential is higher for the secondary pump as the power is proportional to the rotational speed cubed.

Fig. (5) shows the average pumping power for both primary and secondary pumps versus diameter of the bypass as a percentage from the diameter of the main pipe. It is noticeable that there is nearly no effect on the pumping power from 100 to 50 % of the bypass diameter, then a steep drop in the power in the range below 40%, due to the significantly high values of the friction loss in the bypass compared to the rest of the system. This changes the pumps operation points, thus reducing the volume flow rate in the primary circuit.

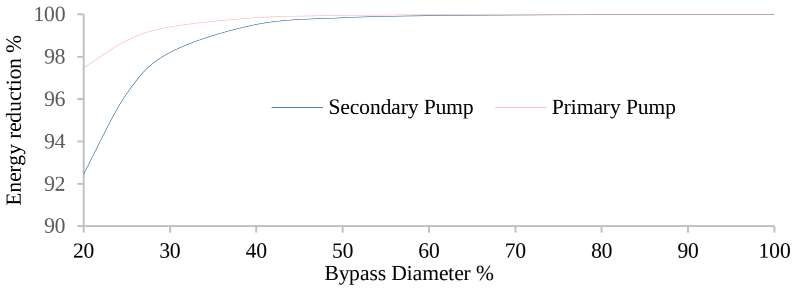

Fig. 5 The effect of changing the bypass diameter on the pumping power

Even a more efficient design can be developed, by reducing the diameter of the bypass to less than 10%, even better energy reduction up to 8% for the primary pumps and up to 23% for the secondary pumps. However this violates the constant flow in the primary pumps, which is desired in the primary-secondary systems. As the diameter of the bypass is reduced below 10% the volume flow rate in the bypass is severely reduced, due to the huge fiction loss inside the pipe, therefore the flow speed is decreased as well. At these small diameters the restriction is blocking the flow, forcing the system to perform as if there is no bypass.

4.5 Valves on the bypass

For a better control over the bypass volume flow rate, valves are installed on the pipe [17,18]. For example, if the flow rate direction is reversed in the bypass, and the return water bypasses the chillers by going directly to the secondary pump, this will increase the pumping energy and will reduce the chiller supply water temperature. As mentioned in the section 4.1, this happens when the heat load is higher than the cooling capacity. A mechanical solution to this problem is introduced, where a check valve is installed to insure that the return water passes through the chillers before going back to the secondary circuit. Although installing a check valve has proven its usefulness in many cases, the same result can be obtained with a control solution. If a proper control strategy is implemented, involving the sequencing of the primary pumps, this problem can be eliminated without the need of any valves. Consequently the check valve improves the efficiency in normal flow direction conditions, due to its significant pressure drop, which is equivalent to reducing the diameter of the bypass.

Another scheme goes for a full control over the bypass by installing a control valve, in which controlling the volume flow rate through the bypass will control the volume flow rate through the system, but it will result in a complex control strategy in order to prove its energy saving potential over the design approach of reducing the diameter of the bypass, where the volume flow rate is controlled by the pumps without the need of any valves on the bypass.

5. Conclusion

In conclusion, the design of the bypass in the primary-secondary chilled water systems has a significant effect on the pumping energy. The bypass is capable of reducing the primary pumps average power consumption up to 6%, and for the secondary pumps up to 8%. The secondary pumping energy is usually more than the primary pumping energy, due to the higher head of the secondary pumps.

These energy savings are achieved by the reduction of the bypass diameter to 20% of the main pipe and setting an operating set points for the primary pumps. Further energy savings for the secondary pumps can be achieved by using the critical valve reset method with the suggested design for the bypass. Then the pumps will operate to supply the minimum required amount of energy, by optimizing the energy devoured in the two way control valves.

Regarding to pumping energy, minimizing the friction loss across the bypass was not the best option, similarly the using of valves on the bypass is not necessary.

The pumping energy increases with the increase of the volume flow rate inside the bypass. Moreover, the flow is affected by the primary and secondary pumps control strategies as well as the bypass diameter. Therefore, when designing the bypass, it is always preferable to always maintain the minimum volume flow rate at all the time restricted by preserving a constant volume flow rate through the chiller.

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