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Evaluation of the Centralized HVAC Chilled Water Pumps of Suranaree University of Technology Hospital Main Building

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Abstract. Pumps are the main issue of the chilled water distribution in a centralized system. In purpose of evaluating the chiller pump system of the centralized HVAC for Suranaree University of Technology Hospital (SUTH) main building, a model comprised evaporator, pump and pipe network was proposed. An analysis of energy rate losses due to pressure losses was applied in a model composed of vertical and horizontal distribution pipes. Pressure loss in pipes was assumed to follow Darcy-Weisbach equation. The resistance of the systems were also calculated. The results show that distribution of losses in vertical shaft and stories were unequal. The pressure loss and energy rate loss were function of flow-rate. But the resistance was an inverse function of the flow-rate. The difference distribution emphasized the need of reconfiguring chilled water pump for accommodating flexibility usage.

1. Introduction

Centralized HVAC is a common method for large building as it has high cooling load. The centralized HVAC works more efficiently than split system when the system needs high heat rejection at the same time. A centralized system works with less moving parts, higher heat working fluid capacity and higher COP [1]. Less moving parts mean less maintenance. Higher heat working fluid capacity can handle more heat rejection load. The COP tells energy consumption over heat rejection. Therefore, the centralized HVAC system is preferable for large system. However, the high capacity system has vulnerability of load variation range.

Variation of load in a HVAC system can degrade the efficiency of the system. Out of designed working point, the efficiency of the system mostly decreases. Because the centralized HVAC depends on pumps for transferring heat among the components, increasing load above the designed load forces the pumps to increase the flow rate of working fluid. It implies higher flow rate and less efficient motor. When the load is less than it designed, the pump can work better, but the motor might work at less efficient condition.

Variable speed motor is proposed for variable load system. Variable speed motor can work over long range speed with high efficiency [2]. Therefore, load variation does not affect the performance of motor. When the load increases, the motor can adjust the speed to provide higher pressure and vice versa. To set at high performance, control system of the motor is important. Unfortunately, the



variable speed motor is still has problem of the high range variety and the price could be very high for a high capacity motor and its controller.

Constructal law emphasizes a design to follow the nature design in order to get high efficiency. Combination of few larges and many smalls is considered the best design as the nature shows in many nature flows [3]. It has been studied in many engineering area such as cooling system [4]–[6], heat storage [7], and energy harnessing [8]. These studies focused on design base and not for retrofitting the established system.

The objective of the study is to evaluate the chilled water pumps of HVAC system in a multi-story building, the main building of Suranaree University of Technology Hospital (SUTH). Evaluation of centralized HVAC system pump can be considered as the compromise between high variety of load and large system [1]. Moreover, it is also a new field of constructal law area to be more practical in established system. In order to evaluate the HVAC system in large building with high load variety in term of constructal study, a case of SUTH main building is proposed.

The structure of the article is mainly composed of introduction, methodology, result and discussion, and conclusion. At the introduction, the state of the problem is presented to show the research gap of the proposed topic. The methodology part consists of description of the HVAC system of SUTH main building and its performance records. In this part, the model of the system will be presented for evaluating the performance. Result will have some data that show analysis of the performance at the current condition. Conclusion will close the paper with some main trend found.

2. Methodology

2.1. Centralized HVAC of SUTH main building

Rattanjapevat is the operational main building of Suranaree University of Technology Hospital. It has 12 floors which has been functioned as the administration building, OPD clinics, wards, operation rooms, emergency, pharmacy, laboratories and some support systems. The clinics areas are distributed in 1st, 2nd, and 3rd floors. The administration office is centered at 4th floor. The operation rooms are located in 5th floor. The 6th floor is dedicated for delivery, while the 7th floor is the place for intensive care units. Hemodialysis machines and its recovery rooms are located in 8th floors. The rest floors; they are 9th – 11th, are wards for women, men, and VIP, respectively. The ground floor is the place for some support systems, they are engineering, CCDS, pharmacy stocks, securities, and morgue.

Total HVAC system capacity of the building is 750 TR which consists of 3 chillers with 250 TR each. The system is also supported by 3 x 30 kW CDPs, 3 x 37 kW CHPs and 3 x 3.8 million BTU/hr. The coil units capacity of the building is 7.2 million BTU/hr. The chilled water flow-rate capacity of the building is 83.9 l/s. The chilled water is designed to work at 7-13.5°C. The chilled water output pressure is set to be 65 psi. The condenser water pumps have to manage 40 psi pressure head loss compensation.

The chilled water is distributed to the building by 3 x 37 kW pumps operated alternately. Every time, only two of the three pumps work. These pumps are connected to vertical main shaft of 10" pipe. The shaft is connected to distribution pipes in every floor. Different sizes of pipes are applied for the distribution pipes. Some floors have equal size of the supply and return pipes. Some of them have different sizes for the reason of price. The size of the pipes for the HVAC and the chilled water flow-rate of the system are provided in table 1. The designation of the stories follows the building condition. The 0 and 12 are used for the basement and rooftop, respectively. This chilled water flow-rates also show the cooling capacity in respective floors.

The chilled water is assumed to be distributed in proportional velocity at steady state condition. The valves are set to provide proportional water flowing at every floor according the capacity of the cooling system. The relationship of the chilled water flow-rate and pressure difference at the evaporator is determined by equation (1) as provided by the provider company [9]. The pressure difference is in psi and the flow-rate is in gallon per minute. The model of the chiller system is

provided at figure 1. As the input of the pump is set to be 65 psi, the pressure loss of the pipes network follows equation (2).

$$Q_{chw} = -0.0226\Delta p_{ev}^2 + 8.9826\Delta p_{ev} + 136.65 \tag{1}$$

$$\Delta p_n = -3 \times 10^{-10} Q^4 + 4 \times 10^{-7} Q^3 + 1 \times 10^{-4} Q^2 + 0.0309Q + 0.6721 \tag{2}$$

Table 1. Story, chilled water flow-rate, and pipe sizes.

Story		Water (l/min)	Inhabitant (day equivalent)	pipe size S (inch)	pipe size R (inch)
Name	Designation				
Ground	0	77.84	18	3	3
1	1	236.10	55	4	4
2	2	203.43	23	4	4
3	3	137.43	23	2	3
4	4	110.60	7	2	3
5	5	120.34	13	2	3
6	6	123.26	5	2	3
7	7	100.65	14	2	3
8	8	86.66	17	2	3
9	9	22.10	25	2	3
10	10	22.10	29	2	3
11	11	74.36	17	2	3
R	12	15.00	1	2	3

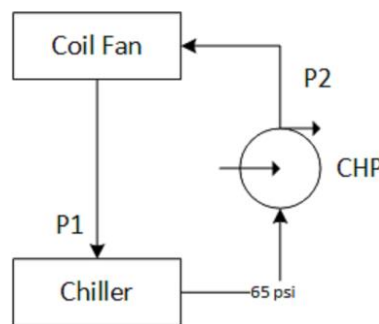


Figure 1. Model of the chilled water loop of the HVAC.

2.2. The evaluation of the chilled water pump configuration

There are two questions of the configuration in chilled water distribution problem. The first question is distributing cooling water as load needed. The second question is minimum energy to flow the chilled water to the load which has horizontal and vertical part. The first question can be considered as sufficiency problem of the chilled water. Assuming load as proportional to number of people and its density is proportional to the area, it can inferred that the scale of the load is L². L is length scale. At the same time the chilled water flow-rate expresses the cooling capacity of the network in a story. Higher water flow-rate means higher cooling capacity and vice versa.

To evaluate the chilled water pump, distribution of chilled water is mapped. To simplify the system, a model as shown in figure 2 is proposed. The model consists of evaporator, pump, and pipe networks which their flow rates and pressure difference. The pipes network is composed of shafts (main pipes), and story distribution pipes. The story pipe line is described as a single equal diameter

line connecting inlet main shaft and outlet main shaft. It is assumed to be a single segment which has pressure difference between its inlet and outlet. Every main shaft segment connecting two adjacent stories is assumed to be a single segment. In the model, the segment has its own pressure difference and flow rate.

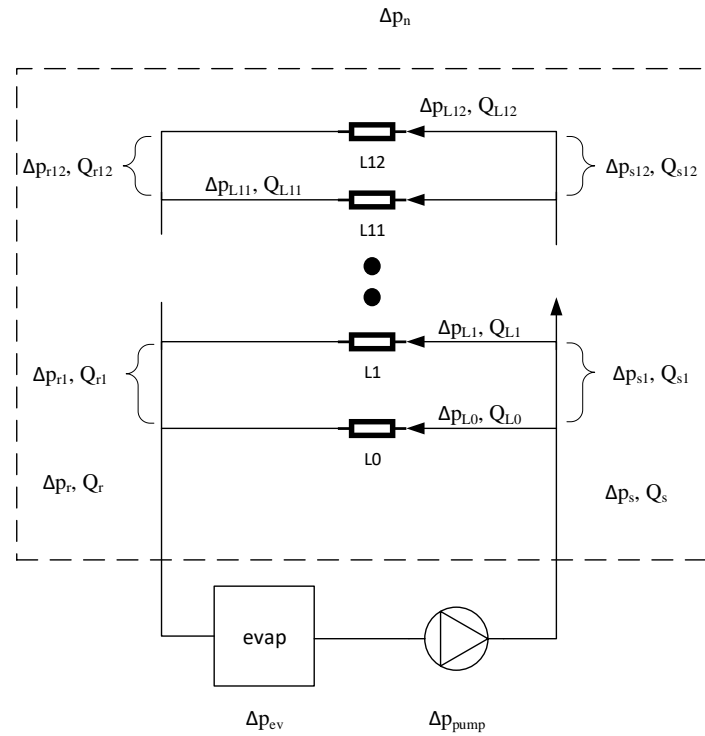


Figure 2. The model of chilled water loop and its pressure and flow-rate.

The pressure of the loop can be shown in equation (3) with η as the total efficiency of pump and motor. As the figure 2 shows, the Δp_n is the pressure of the network and the Δp_{ev} is the pressure difference of evaporator inlet-outlet.

$$\Delta p_n + \Delta p_{ev} = \eta \Delta p_{pump} \tag{3}$$

The pressure of network is composed of pressure of main pipes (Δp_s and Δp_r) and pressure of a story load (Δp_L), that generally can be formed as equation (4).

$$\Delta p_n = \Delta p_s + \Delta p_r + \Delta p_L \tag{4}$$

The total load pressure is equal to load pressure at ground Δp_{L0} and accumulation of other load pressure added by pressure loss at shaft between the ground and the 1st floor. Equations (5) and (6) show the relationship of the pressures.

$$\Delta p_L = \Delta p_{s1} + \Delta p_{r1} + \Delta p_{n1} \tag{5}$$

$$\Delta p_L = \Delta p_{L0} \tag{6}$$

At the branches, equations (7) and (8) take place.

$$Q_{Li} + Q_{S(i+1)} = Q_{Si} \tag{7}$$

$$\Delta p_{Li} = \Delta p_{si} + \Delta p_{ri} + \Delta p_{ni} \tag{8}$$

It can be inferred, that equation (4) can be formed as equation (9).

$$\Delta p_n = \sum_i \Delta p_{si} + \sum_i \Delta p_{ri} + \Delta p_{L12} \tag{9}$$

The flow-rate can be also shown as equation (10).

$$Q_s = \sum_i Q_{Li} \quad (10)$$

According the Darcy Weisbach equation for the pipe, $\Delta p = f_D \rho 8LQ^2 / \pi^2 D^5$, therefore it can be formed equations (11) and (12) that resembles the electricity equation with the current as volume flow-rate rather than mass flow-rate as Bejan did [10] for the reason of incompressibility of the fluid.

$$\dot{E} = \frac{f_D \rho 8LQ^3}{\pi^2 D^5} \quad (11)$$

$$R = \frac{f_D \rho 8LQ}{\pi^2 D^5} \quad (12)$$

The pressure head difference of the adjacent stories can be inferred as equation (13). The first term is the height effect and the second is length effect.

$$\Delta p_{si} = \rho g \Delta h_i + \frac{f_D \rho 8 \Delta h_i Q_i^2}{\pi^2 D^5} \quad (13)$$

The equation (13) emphasizes that the pressure difference of the shaft is proportional to the acceleration of the fluid flow along the vertical shaft. It also happens in horizontal pipes. While at the vertical shaft the potential part is in opposite direction between supply and return, therefore this part collapses in calculation.

Derivation of the pumping power against the pressure different can be shown as in equation (14).

$$\frac{d\dot{E}}{d\Delta p} = 3C_1 \Delta p^{\frac{1}{2}} \quad (14)$$

Equation (14) implies that power of pump is a function of pressure difference by power of 3/2. The derivation of power against flow-rate gives equation (15). The derivation of flow-rate against pressure difference is shown in equation (16).

$$\frac{d\dot{E}}{dQ} = C_2 Q^2 = 3\Delta p \quad (15)$$

$$\frac{dQ}{dp} = C_1 \Delta p^{\frac{1}{2}} \quad (16)$$

For vertical pipe in multistory building, the flow-rate of the vertical main pipe between two adjacent stories can be assumed as iterative function as equation (17).

$$Q_{si} = \sum_{j=i}^n Q_j \quad (17)$$

Equation (13) implies that the pressure difference of vertical pipe is proportional to the height difference and the acceleration. Therefore, it can be formed into equation (18).

$$\frac{d\dot{E}}{dQ} = C_2 Q^2 + C_4 \quad (18)$$

The first term comes from the pressure different of the kinetic and the second from the potential.

3. Result and Discussion

At current operational condition the power of pumping is the function of the total flow-rate. The total flow-rate is determined from equation (1). The total flow-rate is quadratic function of the pressure difference at evaporator. Therefore a pressure difference at evaporator determines power of the pump for distributing the chilled water. Applying single primary pump only for the system, the power of the pump for distributing the chilled water according the flow-rate is shown in figure 3. The function of the pump can determined from equations (1), (2) and (3).

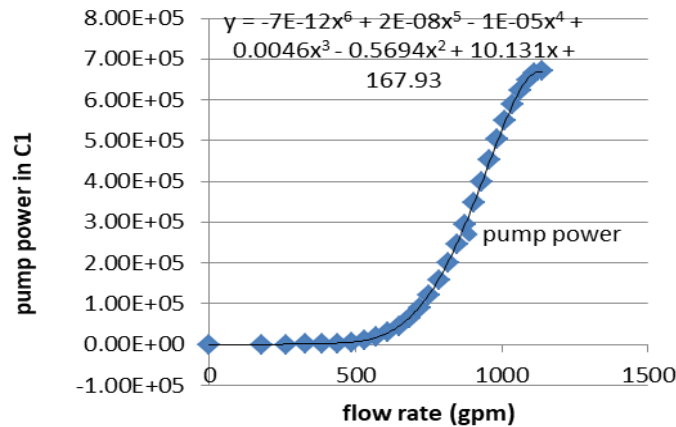


Figure 3. Pump power as function of chilled water flow-rate. The power is scaled into C1.

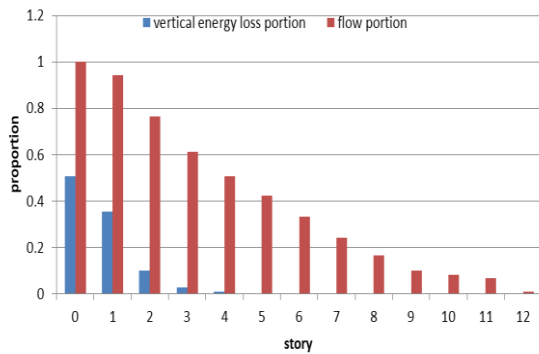


Figure 4. The distribution of energy loss in vertical shaft.

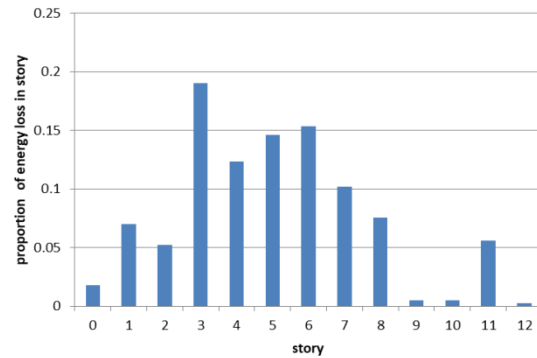


Figure 5. The distribution of energy loss in story.

The energy loss at vertical shaft and at the stories can be traced from equations (13) and (15). Assuming the proportions of the flow rate are equal to the maximum capacity, the energy lost at vertical shaft can be shown in figure 4. Transferring chilled water from the evaporator to the ground story chilled water network is the place where most of the vertical energy loss happened, while transferring chilled water to the roof-top is the least energy consumer.

The energy loss at vertical shaft depends on flow-rate and the height. The distribution of the energy loss according the story shows that vertical effect is far less than energy loss at the story. The pressure loss at vertical shaft for the ground story is ranging from 0.0% - 0.25% of the total pressure loss. Flow rate is more dominant in determining energy loss in distribution of the chilled water than the height.

The maximum delta pressure of the story took place at the ground floor, but the minimum delta pressure happened at the rooftop. Range of the delta pressure between the inlet and outlet at the story is 565.1 – 568.1 kPa at 50 liter/second flow rate of the chilled water. The resistance of the flow at the equal chilled water flow-rate shows different trend. The range of the resistance is 64.39 kPa s/m³ at the 1st story until 1012.68 kPa s/m³ at the rooftop. It is clear that the chilled water flow-rate at the 1st story is the highest and the lowest took place at roof-top. The complete pressure difference among inlet and outlet of the stories, flow-rates of the chilled water at the stories, and the respective resistance are shown in table 2.

The pressure loss at stories is function of the chilled water flow-rate. Figure 6 shows the relation of the pressure loss and the flow-rate as a linear function. Increasing the chilled water flow-rate increases

the pressure loss at the stories linearly. The normalization of the system shows that the increasing takes place proportionally according to the flow-rate.

Horizontal energy rate loss does not depend on the story position. But it depends on the flow-rates and diameter of the pipes as shown in table 1 and table 2. The relation of the energy rate loss and the flow-rate of chilled water is also presented in figure 7. The resistance of the story is also a function of flow-rate as shown in figure 8. Considering relation of equation (13), it shows that the coefficient is a function of flow-rate which is confirming figure 6. This condition emphasizes necessity of considering the constructal configuration at the story as mentioned by Xia [11].

Table 2. The pressure, flow-rate, and resistance of the stories at 50 liter/second flow-rate.

Story	Δp_{Li} (Pa sec/m ³)	Q_{Li} (liter/sec)	R_{Li} (Pa sec/m ³)
0	5.66 10 ⁵	2.90 10 ⁰	1.95 10 ⁵
1	5.66 10 ⁵	8.78 10 ⁰	6.44 10 ⁴
2	5.65 10 ⁵	7.57 10 ⁰	7.47 10 ⁴
3	5.65 10 ⁵	5.11 10 ⁰	1.11 10 ⁵
4	5.65 10 ⁵	4.11 10 ⁰	1.37 10 ⁵
5	5.65 10 ⁵	4.48 10 ⁰	1.26 10 ⁵
6	5.65 10 ⁵	4.59 10 ⁰	1.23 10 ⁵
7	5.65 10 ⁵	3.74 10 ⁰	1.51 10 ⁵
8	5.65 10 ⁵	3.22 10 ⁰	1.75 10 ⁵
9	5.65 10 ⁵	8.22 10 ⁻⁰	6.87 10 ⁵
10	5.65 10 ⁵	8.22 10 ⁻⁰	6.87 10 ⁵
11	5.65 10 ⁵	2.77 10 ⁰	2.04 10 ⁵
12	5.65 10 ⁵	5.58 10 ⁻⁰	1.01 10 ⁶

Unequal distribution of the vertical and story energy loss shows the importance of reconfiguring the pumps of the chilled water in order to minimize the effect of loss. Controlling single primary pump as the current operational condition assumed equal distribution of the vertical and story losses. Increasing power of the pump will increase the water flow-rate proportionally according to the valves setting. At this condition, losses at stories are assumed in accordance with the position. Adjustment of valves should be conducted to allow the change of flow rates. Optimum condition is hardly achieved as the reason of the distribution. In this case, flexibility is important due to interrelation as shown in figure 6, figure 7, and figure 8. Secondary pumps in stories are suggested [2].

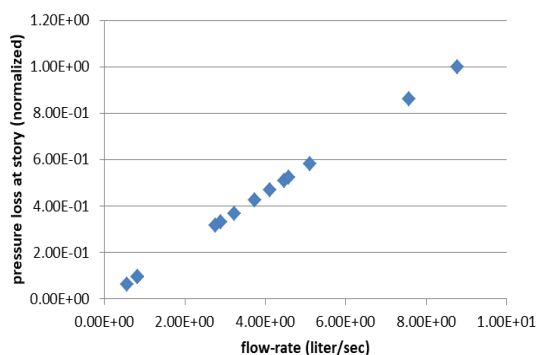


Figure 6. Pressure loss at stories as function of flow-rate.

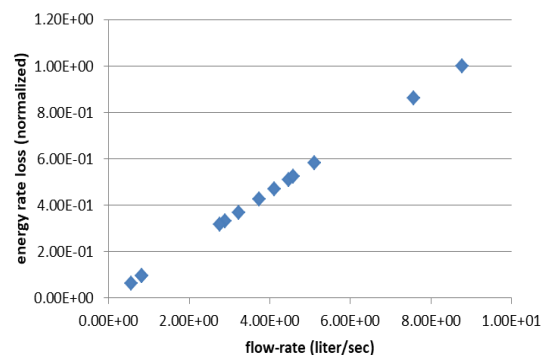


Figure 7. The energy rate loss as function of flow-rate.

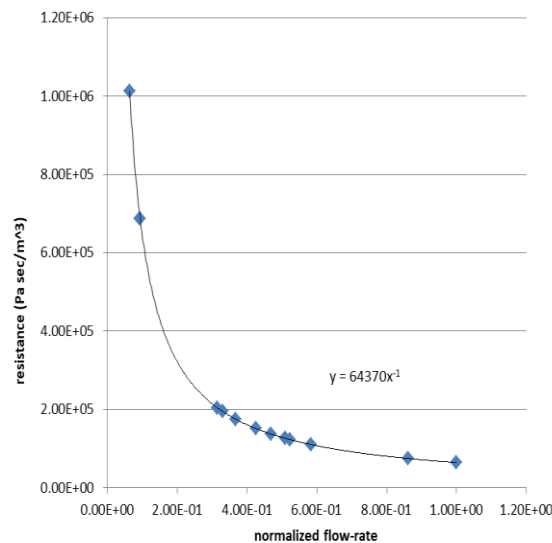


Figure 8. Resistance of the stories as function of normalized flow-rate.

4. Conclusion

The evaluation of the centralized HVAC system at SUTH main building was conducted in order to consider retrofitting the pump configuration. The work shows that distribution of losses at vertical shaft and stories are unequal. The energy losses and pressure losses at the stories are linear function of flow-rate in normalized way. The resistances of stories are inverse function of normalized flow-rate. Simple configuration using single primary pump for the system has drawback of losses at some stories which do not need it. It is suggested applying secondary pumps in every story.

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References

- [1] Perez-Lombard L, Ortiz J and Maestre I R 2011 *Appl. Energy* **88** 5020–31
- [2] Tirmizi S A, Gandhidasan P and Zubair S M 2012 *Appl. Energy* **100** 238–48
- [3] Bejan A and Lorente S 2011 *Phys. Life Rev* **8** 209–40
- [4] Ghodoossi L and Eğrican N 2004 *Energy Convers. Manag.* **45** 811–828
- [5] Adewumi O O, Bello-Ochende T and Meyer J P 2013 *Int. J. Heat Mass Transf* **66** 315–23
- [6] Kalbasi R and Salimpour M R 2015 *Appl. Therm. Eng.* **84** 339–49
- [7] Miguel A F 2008 *Energy Build.* **40** 1020–30
- [8] Lorente S, Koonsrisuk A and Bejan A 2010 *Int. J. Green Energy* **7** 577–92
- [9] TRANE 2012 *Installation, Operation and Maintenance Series R[®] Rotary Liquid Chillers Water-Cooled and Compressor-Chillers* (Ingersoll Rand) p 215
- [10] Bejan A and Lorente S 2008 *Design with Constructal Theory* (New Jersey: John Wiley and Son) p 517
- [11] Xia L, Lorente S and Bejan A 2011 *Int. J. Energy Res.* **35** 806–12