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Theoretical Prediction of Optimum Chilled Water Distribution Configuration in Air Conditioning Terminal Unit

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Abstract

The distribution of chilled water flow rate in terminal unit is a major factor used to evaluate the performance of central air conditioning unit. In this work, a theoretical chilled water distribution in the terminal units has been studied to predict the optimum heat performance of terminal unit. The central Air-conditioning unit model consists of cooling/ heating coil (three units), chilled water source (chiller), three-way and two-way valve with bypass, piping network, and pump. The term of optimization in terminal unit ingredient has two categories, the first is the uniform of the water flow rate representing in statically permanents standard deviation (minimum value) and the second category is the maximum heat transfer rate from all terminal units. The hydraulic and energy equations governing the performance of unit solved with the aid of FORTRAN code with considering the following parameters: total water flow rate, chilled water supply temperature, and variable valve opening. It was found that the optimum solution of three-way valve case at 8°C water supply temperature, 0.12 kg/s total water flow rate and valve opening order (valve 1: 100%, valve 2: 100% and valve 3: 75%) with total heat rate (987.92 Watt) and standard deviation (1.181E-3). Also, for the two-way valve case the results showed that the optimum condition at 8°C water supply temperature, 0.12 kg/s total water flow rate and valve opening order (valve 1: 75%, valve 2: 75% and valve 3: 50%) with total heat rate and standard deviation (717Watt) and (5.69E-4) respectively.

***Keywords**: Optimization, 3-way valve, 2-way valve, Terminal unit, Standard deviation, heat rate.*

1. Introduction

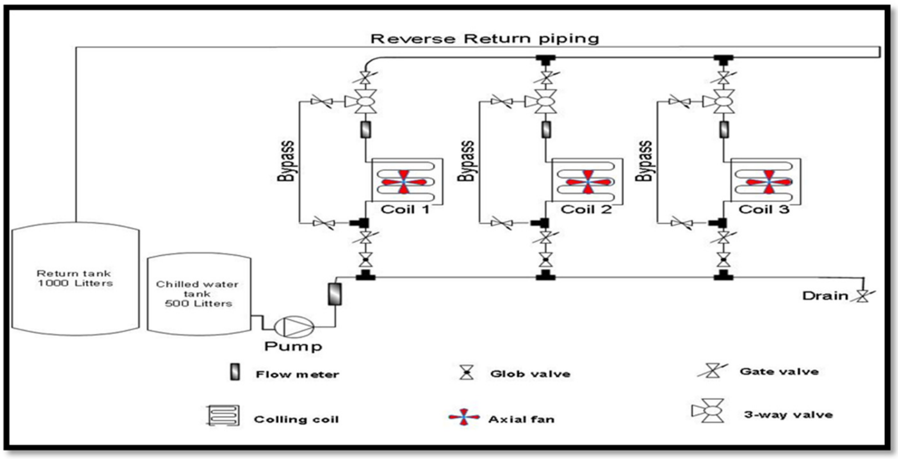
Cooling is essential for all types of buildings in Iraq; Most of the air- central conditioning (A/C) systems used in commercial and institutional buildings are of large capacity. These systems use chillers for cooling production and a network of chilled water piping for cooling distribution to the individual Air-Handling Units (AHUs) or Fan Coil Units (FCUs). The more cooling capacity of cooling system will be accomplished per liter of chilled water distribution if the chilled-water temperature differentials were 8.3°C or greater [1] and if the chilled water temperature difference between supply and return water to the building's side becomes smaller than the designed value. The energy for heat distribution system and the total energy consumption in district heating and cooling system plant increases in considerable degrees [2] but the high chilled water temperature, low supply air temperature, and high outside air intake ratio may results in low cooling chilled water return temperature and the bypass bridge is not necessary and can be removed in the consumer chilled water system with the 2-way valve configuration. The removal not only reduces first cost but also operation cost [3].

The chilled water distribution system stability is the basic engineering knowledge for air conditioning engineer; Proper sizing of control valves, layouts, and self-balancing, flow balancing and control the speed of the pump is an important fundamental factor of chilled water distribution system [4]. Jing jing Liu.et.al (2012)[5] simulated the cooling coil ΔT profiles with different geometric configurations and various water side and air side conditions using an effectiveness-NTU model and found that the existing water side correlations generated significantly different ΔT profiles, with variant geometric configurations, the coil ΔT at full load may be higher, equal, or lower than that at part load, the complexity of cooling coil ΔT characteristics indicates that it is hard to make conclusions about the coil ΔT change with cooling loads. Jin-ping Liu.et.al (2012)[6] built a hydraulic calculation model of pipe network topology, bypass loop hydraulic calculation model and found that when chilled water system is operating in bypass circuit hydraulic adjustable area, pressure difference bypass valve opening and pump operating frequency show an approximately linear relation, which is under little affection of pipe network flow ratio.

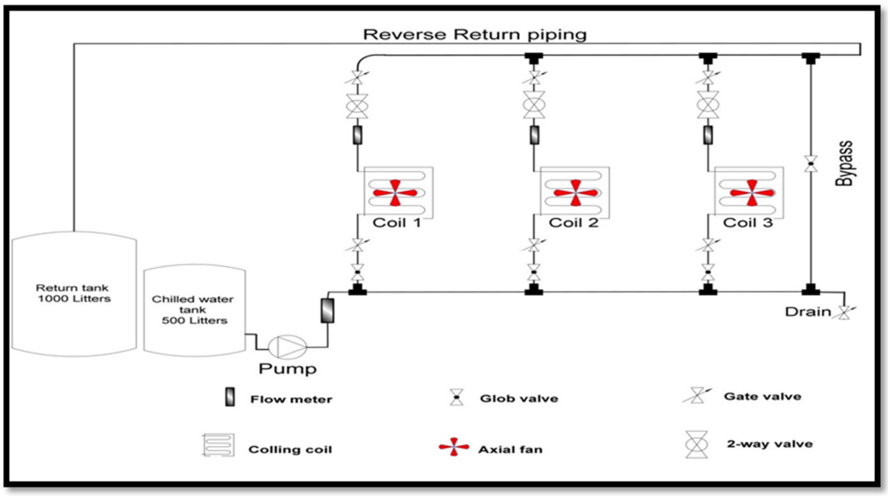
The aim of this work is to determine theoretically the optimum water flow rate distribution in terminal unit to achieve maximum heat transfer between the water and air sides corresponding to the main parameter effect (flow rate, inlet temperature, and valve opening).

2. Governing Equations

In order to investigate the water distribution network in HVAC system, the flow distribution network, pressure drop, heat transfer rate as well as optimization equations were solved using iteration method numerical technique (terminal units), the smallest HVAC system that could be found practically content three circuits model , figure (1a & 1b) shows the schematic diagram of this system.



(a) The system with 3-way valve



(b) The system with 2-way valve.

Fig. 1. a and b: Schematic diagram of the system.

A) Hydraulic model

To calculate the head losses in the pipe:

$h_f = h_{f-major} + h_{f-minor}$ ... (1)

The frictional head loss $h_{f-major}$ is given by Darcy-Weisbach equation, such as:

$$h_{f-major} = f \frac{\ell}{d} \frac{V^2}{2g} \quad \dots (2)$$

where Friction factor for laminar flow is:

$$f = \frac{64}{R_e} \quad \dots (3)$$

and for turbulent flow friction factor is defined by Zigrang-Sylvester formula.:

$$\frac{1}{\sqrt{f}} = -4.0 \log_{10}\left[\frac{\varepsilon/d}{3.7} - \frac{5.02}{R_e}\log_{10}\left(\frac{\varepsilon/d}{3.7} + \frac{13}{R_e}\right)\right] \quad \dots (4)$$

where the relative pipe roughness is $\frac{\varepsilon}{d}$, ... (5)

(ε) is the nickered roughness factor and (d) is tube diameter.

Table (1) illustrates the internal roughness of many types of pipes. [7]

Table 1,
Pipe internal Roughness

Material	(Centimeters)ε
Cast iron	0.026
Galvanized iron	0.015
Asphalted cast iron	0.012
Commercial steel or wrought iron	0.046
PVC	0.00015

The minor head loss $h_{f-minor}$ is given as;

$$h_{f-minor} = K_L \frac{V^2}{2g} \quad \dots (6)$$

where K_L is (local) loss coefficient, although K_L is dimensionless, it is not correlated in the literature with the Reynolds number and roughness ratio but rather simply with the raw size of the pipe.

Fittings such as elbows, tees, valves, and reducers represent a significant component of the pressure loss in most pipe systems, the calculation of pressure losses through pipe fittings and some minor equipment using the K-value method, The resistance coefficient value (K) are given in table (2).[8]

Table 2,
Resistance coefficient (K)

Fitting	Nominal Pipe Size 1/2″	3/4″	1″
	K Value		
globe valve (Full open)	9.2	8.5	7.8
gate valve (Full open)	0.22	0.2	0.18
Tee straight Through	0.54	0.5	0.46
Tee Through branch	1.62	1.5	1.38
Elbow 90°	0.81	0.75	0.69

The pressure drop can be calculated by :

$\Delta P = \rho\, g\, h_f$ ... (7)

The pressure drop through a 3-way & 2-way valve

$$\Delta P_{value} = \left[\frac{Flow}{K_{vs}}\right]^2 * 1296000 \quad \dots (8)$$

Table (3) illustrates the flow coefficient (K_{vs}) of valves with different opening [9].

Table 3,
Flow coefficient (K_{vs})

Valve opening (in)	(mm)	Flow coefficient	Valve opening - Percent of Total Travel 10	20	30	40	50	60	70	80	90	100
3/4	20	Kvs	2.4	4.375	6.625	9.075	11.675	14.25	16.825	18.925	20.525	21.55

B) Heat transfer Model

In this study, the (ε-NTU) method has been used for cross-flow heat exchanger with both fluids unmixed.

NTU depends on both the heat exchanger design (UA) and the operating conditions (C_{min}).

With the dimensionless number of transfer units (NTU) that is used for heat exchanger analysis and was defined as:

$$NTU = \frac{U*A}{C_{min}} \quad \dots (9)$$

Where:

$$U = \frac{1}{R_{th}} \quad \dots (10)$$

$$R_{th} = R_{total} = R_i + R_{wall} + R_o \quad \dots (11)$$

where:

R_{th}: Overall heat resistant
R_i : Inner resistance
R_{wall} : Wall resistance
R_o : Outer resistance
The overall heat resistant R_{th} consists of three parts: convection heat resistant of the chilled water, conduction heat resistant of the metal wall and convection heat resistant of the air.
The overall heat resistance of chilled-water coil was then written as: [10]

$$R_{th} = \frac{1}{h_i\ A_i} + \frac{ln(d_o/d_i)}{2\pi k_w l} + \frac{1}{h_o\ A_o} \quad \ldots (12)$$

Where:

$$h_i = \frac{Nu*K_w}{di} \quad \ldots (13)$$

$$h_o = \frac{Nu*K_a}{d_h} \quad \ldots (14)$$

Effectiveness ε, is defined as the ratio of the actual heat transfer rate for a heat exchanger to the maximum possible heat transfer rate.

$$\varepsilon = \frac{q}{q_{max}} \qquad 0 \le \varepsilon \le 1 \quad \ldots (15)$$

$$\varepsilon = 1 - exp\left[\frac{e^{-NTU^{0.78}\,C}\,_{-1}}{NTU^{-0.22}\,C}\right] \quad \ldots (16)$$

where:

$$C\text{: Capacity ratio} \ = \frac{C_{min}}{C_{max}} \quad \ldots (17)$$

Maximum possible heat rate:

$$q = C_{min}\,(T_{h,i} - T_{c,i}) \quad \ldots (18)$$

The Nusselt Number for air side is calculated as. [10]

$$Nu = 0.117*Re^{0.65}*Pr^{0.33} \quad \ldots (19)$$

and for chilled water side it is calculated by:
For laminar:

$$Nu = 1.86\left(\frac{Re\ \Pr d_i}{\ell}\right)^{0.333} \quad \ldots (20)$$

For turbulent:

$$Nu = 0.023*Re^{0.8}*Pr^{0.4} \quad \ldots (21)$$

where

$$Pr = \frac{CP*\mu}{K} \quad \ldots (22)$$

C) Flow variation and standard deviation

The following equation evaluates the standard deviation:

$$SD = \sqrt{\frac{1}{N}\sum_{i=1}^{N}(flow - \overline{flow}\,)^2} \quad \ldots (23)$$

where,
$(flow_1, flow_2 \ldots . flow_N)$ are the values of the flow calculated and $(\overline{flow}\ )$ is the average value of these flow, while the denominator N stands for the size of the sample(in the present work = 3).

3. Solution Code

Using FORTRAN code with iteration method technique to solve the governing equations from equation (1) to equation (23) according to the flow chart see figure (2).

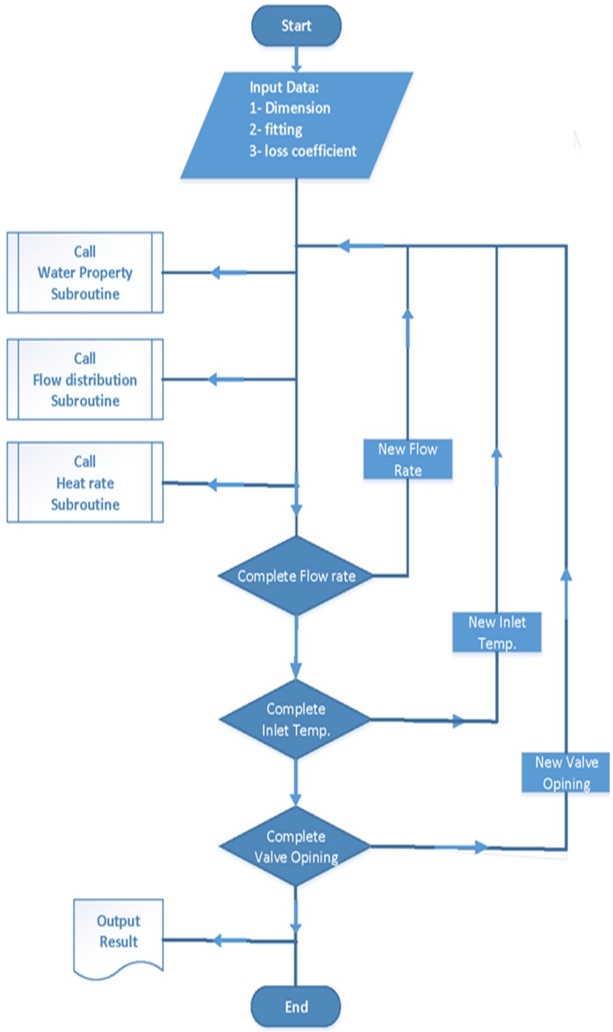


Fig. 2. Flow chart of the water distribution network.

4. Optimization Method

The ultimate goal of the optimization was to maximize the value of the total heat rate (summation of three coils) and minimize the variation of flow in the three circuits representing by the standard deviation (SD). To do that we suggested two factors (F_1) & (F_2) and the target of optimization is to find the maximum value of $(F_1 * F_2)$ where:

$$F_1 = 1 - \left(\frac{M-Q}{R_1}\right) \quad \ldots (24)$$

$$F_2 = 1 - \left(\frac{S-Z}{R_2}\right) \quad \ldots (25)$$

$$Optimization\ factor = F_1 * F_2 \quad \ldots (26)$$

5. Results and Discussion
5.1 Total Heat Rate

Figure (3) shows that the total heat rate of terminal units at different total water flow rates (0.06, 0.08, 0.1 and 0.12 kg/s) and various water supply temperatures were (8, 10, 12, 14 and 16 °C) at 100% valves opening: (valve 1=100%, valve 2=100% and valve 3=100%), Note that the higher total heat rate occurred at minimum water supply temperature (8°C) and at maximum total water flow rate (0.12 kg/s), due to the highest difference in temperature between the water and the air when the temperature of the air is constant, While the increase of the heat rate is slight because the flow type in most cases is laminar and the value of heat transfer coefficient (h_i) is almost constant. The same behavior was found in figure (4) and (5), however changing the valve opening from (100-75%) or to 50% for coil No.3 only gave less water flow rate inside this coil, while the total heat rate remained approximately constant. The figures (6) to (9) explain the changes in other 3-way valve opening of coil. These figures indicate that the total water flow rate and water supply temperature has a significant impact on the total heat rate at any valve opening for any coil. Figure (10) illustrates the effect of valve opening on the overall heat rate at water supply temperature (8°C) and different total water flow rates (0.06, 0.08, 0.1 and 0.12 kg/s). It can be depict that the lower total heat rate occurred at case (valve 1=100%, valve 2=100%, and valve 3=50%), because coil No.3 is always has the higher water flow rate in comparison with other coils in the cycle, which makes a significant effect on the total heat rate. But Figure (11) to (13) shows that the total heat rate of terminal units with the same input parameter in 3-way valve when utilize the 2-way valve. Note in all figures that the higher total heat rate occurred at minimum water supply temperature (8°C) and maximum total water flow rate (0.12 kg/s). Figure (14) illustrates the effect of valve opening on the overall heat rate at water supply temperature (8°C) and multiple total water flow rates. It noted that the lower overall heat rate occurred at case (valve 1=100%, valve 2=100%, and valve 3=50%) for the same reason in 3-way valve.

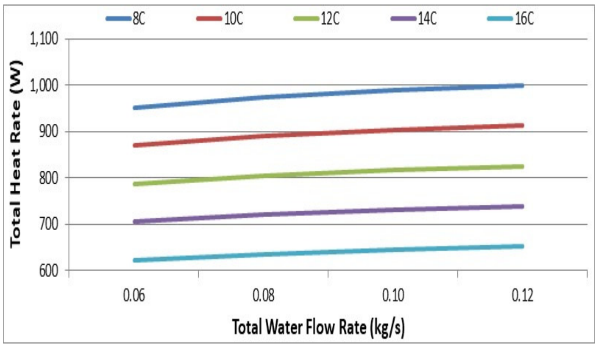


Fig. 3. The total heat rate of terminal units at 8, 10, 12, 14 and 16 °C water supply temperature and valve opening: Valve 1=100%, Valve 2=100% and Valve 3=100%.

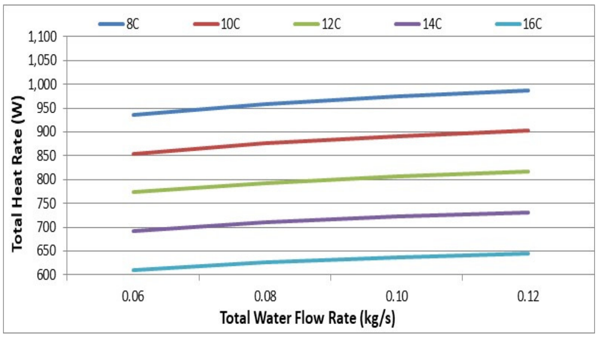


Fig. 4. The total heat rate of terminal units at 8, 10, 12, 14 and 16 °C water supply temperature and valve opening: Valve 1=100%, Valve 2=100% and Valve 3=75%.

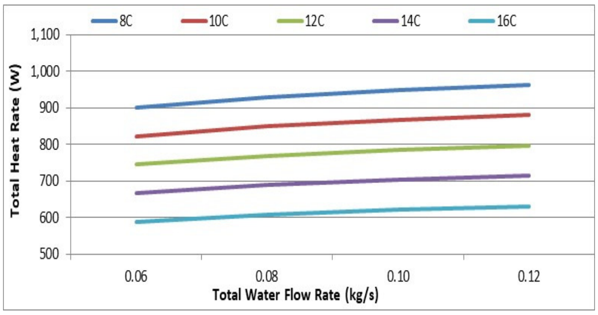


Fig. 5. The total heat rate of terminal units at 8, 10, 12, 14 and 16 °C water supply temperature and valve opening: Valve 1=100%, Valve 2=100% and Valve 3=50%.

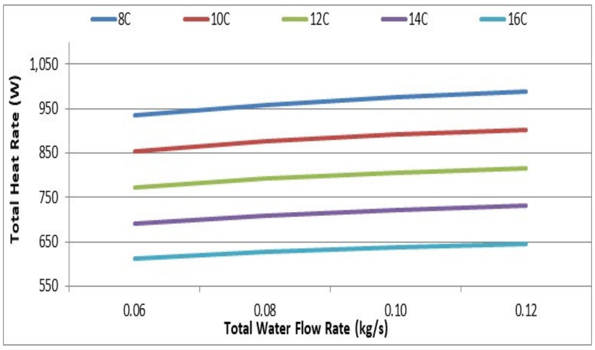


Fig. 6. The total heat rate of terminal units at 8, 10, 12, 14 and 16 °C water supply temperature and valve opening: Valve 1=100%, Valve 2=75 % and Valve 3=100.

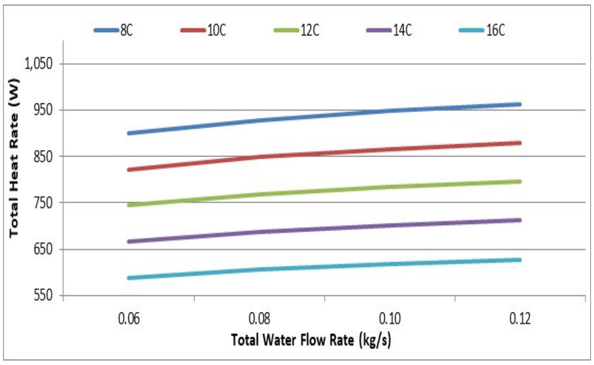


Fig. 7. The total heat rate of terminal units at 8, 10, 12, 14 and 16 °C water supply temperature and valve opening: Valve 1=100%, Valve 2=50 % and Valve 3=100%.

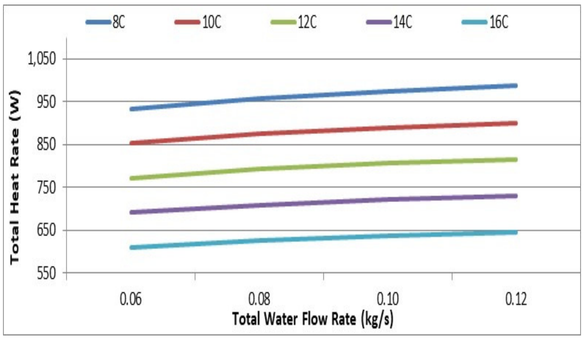


Fig. 8. The total heat rate of terminal units at 8, 10, 12, 14 and 16 °C water supply temperature and valve opening: Valve 1=75%, Valve 2=100 % and Valve 3=100%.

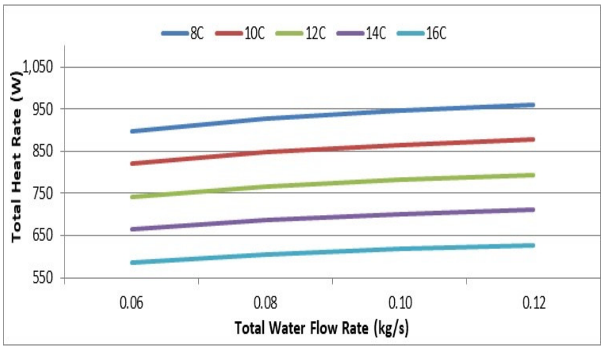


Fig. 9. The total heat rate of terminal units at 8, 10, 12, 14 and 16 °C water supply temperature and valve opening: Valve 1=50%, Valve 2=100 % and Valve 3=100%.

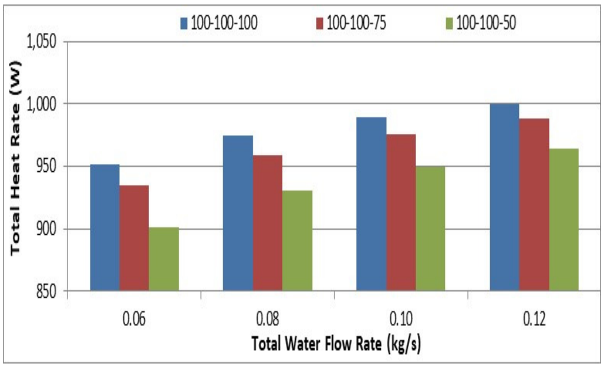


Fig. 10. The total heat rate of terminal units at 8°C water supply temperature and valve opening: Valve 1=100%, Valve 2=100 % and Valve 3=100-75-50%.

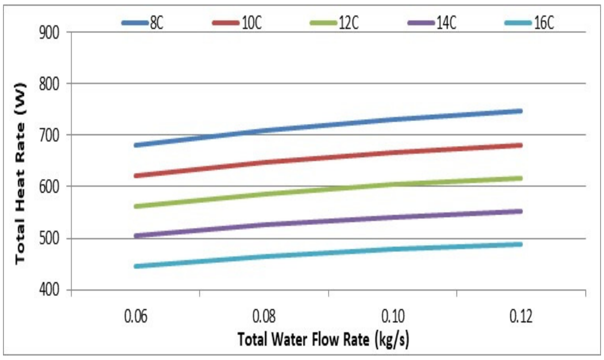


Fig. 11. The total heat rate of terminal units at 8, 10, 12, 14 and 16 °C water supply temperature and valve opening: Valve 1=100%, Valve 2=100% and Valve 3=100%.

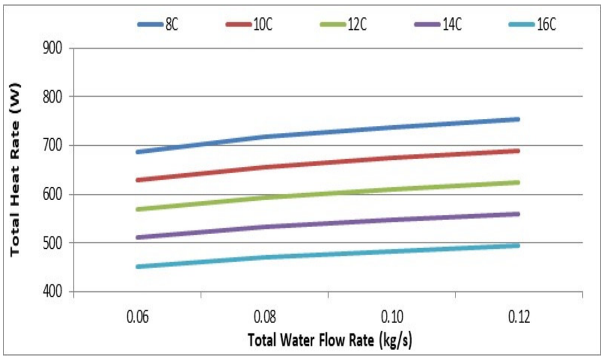


Fig. 12. The total heat rate of terminal units at 8, 10, 12, 14 and 16 °C water supply temperature and valve opening: Valve 1=100%, Valve 2=100% and Valve 3=75%.

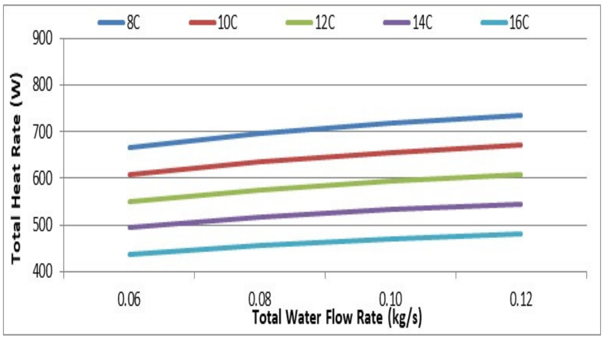


Fig. 13. The total heat rate of terminal units at 8, 10, 12, 14 and 16 °C water supply temperature and valve opening: Valve 1=100%, Valve 2=100% and Valve 3=50%.

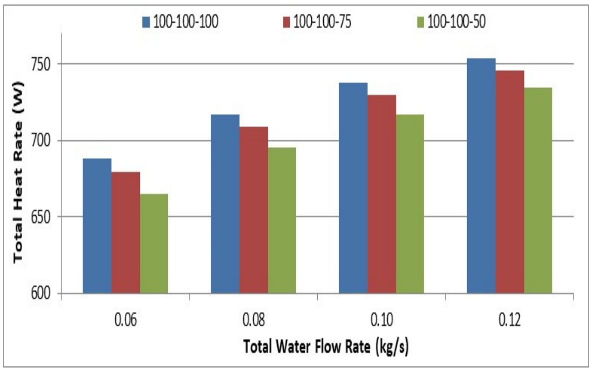


Fig. 14. The total heat rate of terminal units at 8°C water supply temperature and valve opening: Valve 1=100%, Valve 2=100 % and Valve 3=100-75-50%.

5.2 Standard Deviation

Figure (15) illustrates the effect of the total water flow rate on the flow variation representing in standard deviation between the terminal units at (8°C) inlet water supply and constant valve opening of coil No.1, coil No.2 and variable valve opening of coil No.3 (100, 75, 50%) It can be seen that the best standard deviation (minimum) occurred at case (100% - 100% - 75%). That is mean the amount of flow variation between coils is very low, because coil No.3 has the highest flow rate of the others, therefore decreasing the valve opening of coil No.3 to 75%, which decreases the water flow rate inside the coil and gives closer deviation with the other valves. However decreasing the valve opening of coil No.3 more to 50%. The flow rate in coil No.3 will be very low in comparison with the others, which leads to a great increase in standard deviation. Also this figure proves that the standard deviation increases with the increase the total water flow rate, but the water supply temperature does not affect the standard deviation. Also in 2-way valve figure (16) illustrates the effect of the total water flow rate on the flow standard deviation between the terminal units at (8°C) inlet water supply and constant valve opening of coil No.1, coil No.2 and variable valve opening of coil No.3 (100, 75, 50%) . It can be seen like in 3-way valve that the best standard deviation (minimum) at case (100% - 100% - 75%). The reduction in valve opening reduces the standard deviation of flow because the decrease of valve opening means reducing the flow in the coil system, i.e. reducing the variation between the coils (reduce standard deviation) of flow.

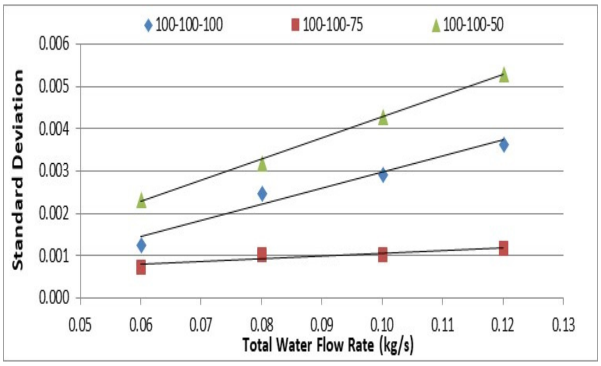


Fig. 15. The flow standard deviation between terminal units at 8 °C water supply temperature and valve opening: Valve 1=100%, Valve 2=100% and Valve 3=100-75-50%.

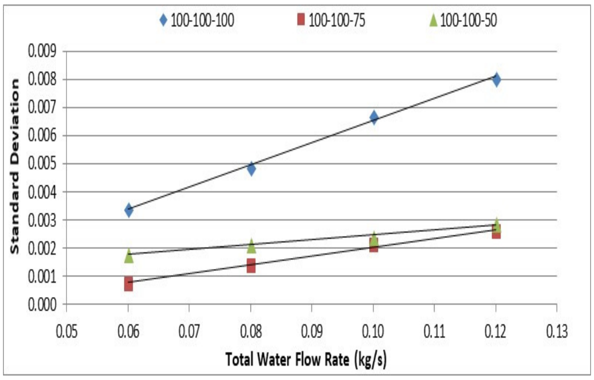


Fig. 16. The flow standard deviation between terminal units at 8 °C water supply temperature and valve opening: Valve 1=100%, Valve 2=100% and Valve 3=100-75-50%.

5.2 Predict the Optimum Case

The optimum conditions are found to give the maximum total heat rate and minimum standard deviation of predetermined values, the optimum values of these circumstances at valve opening (Valve 1: 100%, Valve 2: 100% and Valve 3: 75%), the total water flow rate 0.12 kg/s (7.2 l/min) and the water supply temperature 8°C with a maximum total heat rate of (987.92 watt) and minimum standard deviation (0.001181), the optimization factor could be found by using equations (16), (17) and (18). From figure (17) it can be seen that the maximum optimum factor (0.919894) occurred at 8°C water supply temperature and (0.12 kg/s) total water flow rate. This case is the optimum solution of three-way valve cases. The same procedure can be used to find the optimum condition in two-way valve in A/C system configuration, the optimum values of these conditions at valve opening (Valve 1: 75%, Valve 2: 75% and Valve 3: 50%), the total water flow rate 0.12 kg/s (7.2 l/min) and the water supply temperature 8°C with a maximum total heat rate of (717 watt) and minimum standard deviation (0.000569), Figure (18) shows that the optimization factor at each flow rate and each water supply temperature.

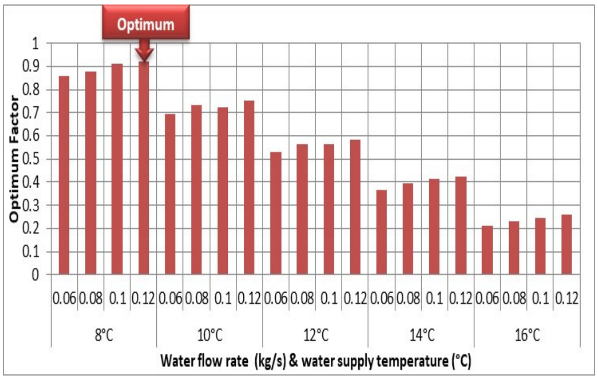


Fig. 17. Optimization factor of three-way valve case.

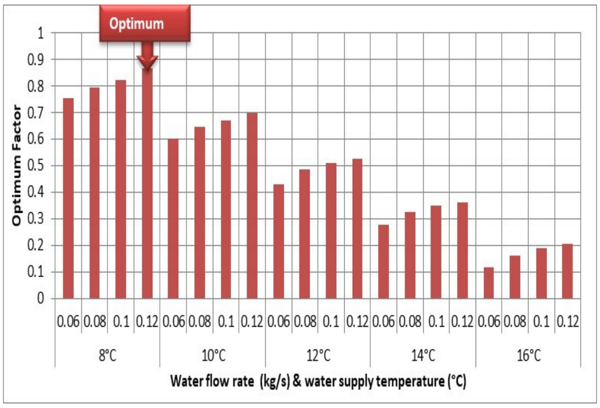


Fig. 18. Optimization factor of two-way valve case.

6. Conclusions

1. The total heat rate in A/C terminal unit is minimum at the lowest valve opening, highest water supply temperature and lowest total water flow rate, whereas the total heat rate is maximum at the maximum valve opening, highest total water flow rate and lowest water supply temperature.
2. The last valve (No.3 in current model) highly effect on the distribution of flow rate at each terminal unit. Therefore, the optimum value of maximum total heat rate and minimum standard deviation occurred when valve order (valve 1= 100%, valve 2=100% and valve3=75%) in three way valve case. Also, in a 2-way valve the optimum value of maximum total heat rate and minimum standard deviation occurred when valve opening (valve 1= 75%, valve 2=75%, and valve 3=50%).

3. The supply temperature has the greater impact on the total heat rate than valve opening and total water flow rate.
4. The valve opening and total water flow rate has the significant influence on the standard deviation, while the water supply temperature does not effect on it in 3-way and 2-way valve.
5. The diameter of the bypass pipe in 2-way valve system should be much lower than the diameters of the system.

Nomenclature

A/C	air- central conditioning
SD	standard deviation
f	Friction factor
ℓ	Length of the pipe with constant diameter in (m).
d	Internal diameter of the pipe in (m)
V	Average velocity of flow in (m/s)
g	Gravity constant
v	Kinematic viscosity of the fluid (m^2/s)
ΔP	Pressure drop in (N/m^2)
h_i	Heat transfer coefficient of water side (W/m^2.K)
C_{min}	Minimum Capacity rate (W/ K)
C_{max}	Maximum Capacity rate (W/ K)
q_{max}	Maximum possible heat transfer rate (W)
T_{hi}	Inlet temperature of air (K)
T_{ci}	Inlet temperature of water (K)
Cp	Specific heat (J/kg.K)
μ	Dynamic viscosity (N.S/m^2)
K	Thermal conductivity (W/m.K)
U	Overall heat transfer coefficient (W/m^2.K)
Ɛ	Relative pipe roughness
h_o	Heat transfer coefficient of air side (W/m^2.K)
ΔT	Temperature difference
F1	Total heat rate factor
F2	Standard deviation factor
M	maximum value of total heat rate
Q	Actual value of total heat rate
R_1	The range between maximum and minimum value of total heat rate
S	Actual value of standard deviation
Z	minimum value of standard deviation
R_2	The range between maximum and minimum value of standard deviation
F	Optimum factor

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التخمين النظري للتوزيع الامثل للماء المثلج في الوحدات الطرفية في منظومة التكييف المركزية

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الخلاصة

يعد توزيع معدل تدفق المياه المثلجة في الوحدات الطرفية عاملا رئيسا يستخدم لتقويم أداء وحدة تكييف الهواء المركزية. في هذا العمل، تم اجراء دراسة نظرية لتوزيع المياه المثلجة في الوحدات الطرفية للتنبؤ بأفضل أداء حراري للوحدة الطرفية. يتكون نموذج وحدة تكييف الهواء المركزية من الملف التبريد / التدفئة (ثلاث وحدات)، مصدر المياه المثلجة (المبرد)، صمام ثلاثي الاتجاه وصمام ثنائي الاتجاه مع مجرى جانبي، وشبكة الأنابيب، ومضخة. إن مصطلح التحسين في الوحدة الطرفية يشمل جزئين الأول هو معدل تدفق المياه المنتظم الذي يتمثل في الانحراف المعياري الثابت (القيمة الدنيا) والفئة الثانية هي أعظم معدل نقل للحرارة من جميع الوحدات الطرفية. المعادلات الهيدروليكية ومعادلات الطاقة التي تحكم أداء الوحدة تم حلها بمساعدة برنامج فورتران مع الاخذ بنظر الاعتبار العوامل المتغيرة الاتية: إجمالي معدل تدفق المياه، ودرجة حرارة إمدادات المياه المثلجة، وفتحات صمام متغيرة. وقد وجد أن الحل الأمثل لحالة الصمام الثلاثي الاتجاه عند درجة حرارة امداد بالمياه تبلغ ٨ درجة مئوية، و٠,١٢ كجم / ثانية إجمالي معدل تدفق المياه وعند فتحة صمام (صمام ١: ١٠٠٪، صمام ٢: ١٠٠٪ وصمام ٣: ٧٥ ٪) مع معدل حرارة كلي (٩٨٧,٩٢ واط) وانحراف معياري (1.181E-3). أيضا بالنسبة لحالة الصمام ثنائي الاتجاه أظهرت النتائج أن الحالة المثلى عند ٨ درجة مئوية درجة حرارة إمدادات المياه، و ٠,١٢ كغم / ثانية مجموع معدل تدفق المياه وعند فتحة صمام (صمام ١: ٧٥٪، صمام ٢: ٧٥٪ وصمام ٣ :٥٠%) مع معدل حرارة كلي وانحراف معياري (٧١٧ واط)، (5.69E-4) على التوالي.