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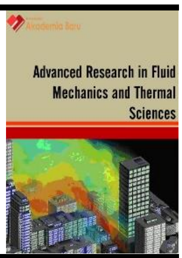

Virtual Flow Meter with Energy Balance Method

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ABSTRACT

Keeping the chiller performance at an optimum level is desirable because chiller is one of the energy-consuming equipment in buildings. Chiller is responsible for 40% to 70% of building energy consumption. Therefore, monitoring chiller performance is essential to keep its performance at a desirable level. Knowing chiller performance needs information about the water flow rate. Unfortunately, not every chiller is equipped with the water flow meter. Fortunately, in the last decade, some researchers have been developing virtual flow meters (VFM) to eliminate the need for flow meter equipment. One of the well-known methods of VFM is the energy balance method. There are two types of energy balance methods. Firstly, using theoretical work input to a compressor in calculating energy balance to estimate the flow rate. However, this method needs free-fault data from manufacturer or field measurement to obtain the theoretical compressor work input from a regression analysis. The second method uses actual compressor work input in energy balance analysis. This method needs trend data generated from the building automation system (BAS) for the energy balance analysis. Our research proposes the use of isentropic compressor work input to estimate the water flow rate. The proposed method may eliminate the need for manufacturer or field measurement data or trend data from BAS. The result shows that the comparison with the measurement values indicates that the use of isentropic work to estimate the water flow rate has good accuracy. However, it is sensitive to a low-temperature difference in evaporator and condenser and refrigerant overcharged fault.

Keywords:
Virtual flow meter; VFM; Virtual sensor; Fault detection and diagnosis; Chiller FDD

1. Introduction

The information about the water flow rate in the chiller system is very important. By knowing the water flow rate, one can monitor the chiller performance such as cooling energy consumption and chiller coefficient of performance (COP). Also, knowing the chilled and cooling water flow is essential

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for diagnostic purposes, for example, identification of reduced condenser and evaporator flow rate for some reasons. Based on our experiences in performing a chiller energy audit in buildings, unfortunately, we found that not every chiller has condenser and evaporator water flow meter installed. It may be because the investment in flow meter equipment is relatively expensive and it also requires routine maintenance and calibration [1]. Therefore, it is difficult for the auditor to assess the chiller performance during the energy audit or chiller assessment. Fortunately, in the last decade, the researchers have been developing virtual flow meters (VFM) to eliminate the need for flow meter equipment [2].

Li reviewed several virtual sensor models for the chiller system to virtually measure refrigerant pressure, refrigerant flow rate, compressor power, and chiller system performance [2]. The application of virtual sensors are cost-effective and they have been applied for other industries [2]. Several methods for chiller virtual flow meter (VFM) have been developed at two-level: pump level and chiller level [3].

At the pump level, Liu developed a VFM model that measured the water flow rate using the pump head (measured from differential pressure transducer), pump speed, and pump curve from the manufacturer data [4-5]. However, the accuracy of differential pressure was low at a lower pump head [4]. Wang proposed models that used pump head and motor power as input [6-7], but the model was sensitive to the fluctuation of the pump speed [6]. Andiroglu used a model that measured motor input power by considering the harmonic energy loss [8].

The second approach is applied to the chiller level. Li predicted the refrigerant mass flow rate using a manufacturer's compressor map [2,9-10]. The manufacturer's compressor map was represented by a 10-coefficient polynomial equation [2,10]. However, the result was only accurate for the normal operating condition [10]. The alternative approaches were proposed that were based on an empirical formula for the volumetric efficiency of the compressor with an error within ±7% and an energy balance approach that considered the compressor energy loss [10-11].

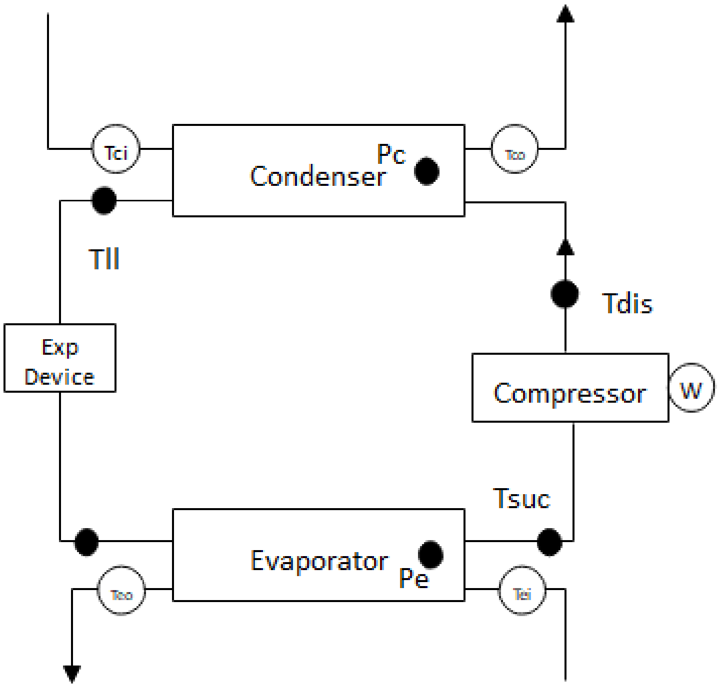
Zhao[1,12-13] and McDonald [3,14-15] proposed an energy balance method to predict the refrigerant, evaporator, and condenser water flow rate. The difference was only in determining the compressor work input. Zhao used the theoretical compressor work input based on a regression model from manufacturer data or field measurement at normal operating conditions [1]. The Zhao approach only worked well when there was no fault in the compressor. However, obtaining manufacturer data or performing field measurements as required by the Zhao method is not always possible. In contrast, McDonald used actual compressor work input for the model, but the model needs chiller trend data from Building Automatic System (BAS) [3, 14-15]. Unfortunately, not every building equipped with a building automatic system (BAS).

This research proposes the possible use of isentropic compressor work input instead of theoretical or actual compressor input to estimate the evaporator and condenser water flow rate. The used method is an energy balance approach. Firstly, using the isentropic compressor work in compressor energy balance analysis to find the refrigerant flow rate. After the refrigerant flow rate is obtained, the water flow rate in the evaporator and condenser can be calculated with energy balance analysis between refrigerant and water circuits. Hopefully, the need for manufacturer data or field measurement and the use of data from BAS may be eliminated.

2. Methodology

The research used experimental data from Comstock in his ASHRAE Research Project 1043-RP [16]. Virtual flow meter (VFM) with an energy balance method requiring ten measurement locations

(Figure 1) and the instrumentation used by Comstock is tabulated in Table 1. Most large chiller systems in Indonesia usually provide all this information for energy balance analysis.



Measurement points:

1.Tci : Condenser inlet temperature
2.Tco: Condenser outlet temperature
3.W : compressor load
4.Tei : Evaporator inlet temperature
5.Teo : Evaporator outlet temperature
6.Tdis : Discharge temperature
7.Tll : Liquid line temperature
8.Tsuc: Suction temperature
9.Pc : Condenser pressure
10.Pe : Evaporator pressure

Fig. 1. Instrumentation locations required for VFM with energy balance method

The first step for energy balance method was to calculate the refrigerant mass flow rate ($\dot{m}_r$) with the use of Eq. (1).

$$W_{isent} = \dot{m}_r(h_{dis_S} - h_{suc}) \quad (1)$$

Table 1
Instrumentation [17]

Measured parameter	Instrument
Flow rate	Vortex flow meter
Compressor power	Watt Transducer
Evaporator water temperature	RTD thermocouple
Condenser water temperature	RTD thermocouple
Evaporator pressure	Pressure transducer
Condenser pressure	Pressure transducer
Suction temperature	Microtech
Discharge temperature	Microtech

where W_{isent} is the isentropic compressor input. The isentropic discharge enthalpy (h_{dis_S}) was calculated at condenser pressure and discharge temperature (T_{dis}) with constant entropy. The suction enthalpy (h_{suc}) was evaluated at evaporator pressure and suction temperature (T_{suc}). To calculate the isentropic compressor work input (W_{isent}), one needs to find the isentropic efficiency of the compressor using Eq. (3) and then calculate the W_{isent} with the use of actual compressor input (W_{ac}) in Eq. (2).

$$W_{isent} = \eta_c W_{ac} \quad (2)$$

$$\eta_c = \frac{Isentropic\ compressor\ input}{Actual\ compressor\ input} = \frac{h_{dis_S} - h_{suc}}{h_{dis} - h_{suc}} \tag{3}$$

After the refrigerant mass flow rate ($\dot{m}_r$) was obtained, the condenser and evaporator water flow rate can be calculated using Eqs. (4) and (5). The liquid line enthalpy (h_{ll}) was calculated at condenser pressure (Pc) and liquid line temperature (T_{ll})

$$\dot{m}_c = \frac{\dot{m}_r (h_{dis} - h_{ll})}{c_p (T_{co} - T_{ci})} \tag{4}$$

$$\dot{m}_e = \frac{\dot{m}_r (h_{suc} - h_{ll})}{c_p (T_{ei} - T_{eo})} \tag{5}$$

The energy balance analysis was applied to seven different scenarios as tabulated in Table 2. Each scenario has 27 data points, therefore, there are 675 data points for the analysis. All data was in a steady state. The Comparison between calculated and measured values was evaluated at the average value for both evaporator and condenser water flow rate ($\dot{m}_e$ and $\dot{m}_c$).

Table 2
The Fault scenario

Fault Type	Fault Level			
Normal				
Reduced condenser flow	10%	20%	30%	40%
Reduced evaporator flow	10%	20%	30%	40%
Refrigerant leak	10%	20%	30%	40%
Refrigerant overcharged	10%	20%	30%	40%
Condenser fouling	12%	20%	30%	45%
Non-condensable gas	1%	2%	3%	5%

According to a study McDonald [14] and McDonald [15], the flow rate estimate was sensitive to water temperature differences in the evaporator and condenser. At lower water temperature difference, the error in flow rate prediction increased [14-15] . Therefore, to see the sensitivity of temperature difference to the estimated value of water flow rate, the data was filtered at four different values of temperature differences (ΔT): First, no filter or all temperature difference included in the calculation. Second, the data with temperature difference larger than 2.5°C ($\Delta T \geq 2.5°C$) was removed from the average flow rate calculation. Next, data with the temperature difference larger than 3.5°C ($\Delta T \geq 3.5°C$) was excluded from the analysis. In the last one, the calculation was performed for water with temperature difference larger than 4.0°C ($\Delta T \geq 4.0°C$).

3. Results

3.1 Sensitivity to Water Temperature Difference (ΔT)

The use of isentropic compressor work in energy balance calculation was very sensitive to water temperature difference in the condenser (ΔT_c) and evaporator (ΔT_e). The simulations were performed at three different ΔT values. Firstly, the calculation included all the data points. Secondly, all data with either ΔTc or ΔTe < 2.5°c were removed from the analysis. In the last calculation, all data with either ΔTc or ΔTe < 3.5°c were taken out from the analysis. To calculate the error or the difference between calculated and measured value, Eq. (6) was used.

$$Error = \frac{calculated\ \ -\ measured}{measured} x100\% \tag{6}$$

The result in error for condenser flow rate (FWC) and evaporator flow rate (FEW) from three different ΔT values were tabulated in Table 3. When the analysis contained all ΔT data (Table 3), the minimum and maximum error in the average water flow rate were within ±17.3% and ±40.2% respectively. The largest error occurs in the condenser fouling scenario. Some fault scenario was sensitive to fault severity level such as increased in condenser and evaporator water flow rate (no 3 to 10, and no 11 to 18 in Table 3). The error in FWC increased as the condenser flow rate decreased and the error in FWE also increased as the evaporator flow rate decreased.

When the data with ΔT < 2.5°C was removed from the analysis, the accuracy of FWC and FWE for all fault scenario improved significantly (Table 3). Mostly, the error was reduced by more than 50%. As a result, the minimum and maximum error for the average water flow rate were reduced to ±0.6% and ±19.4% respectively. The best accuracy belongs to normal and refrigerant leak scenario, both have an error below 10% (Table 3). In addition, the trend in error increased as the fault level increased for the following fault scenario: a reduced evaporator, a reduced condenser flow rate, and a refrigerant leak fault scenario.

The estimated water flow rate improved when the data with ΔT < 3.5°C were excluded from the calculation (See the 6th column in Table 3). The minimum and maximum error for the average water flow rate were reduced to ± 0.0% and ±18.2% respectively. For all scenarios except for refrigerant overcharged, the error was within ±10%. Table 2 shows that the error in the flow rate estimate improved significantly when data with higher ΔT value were used in the calculation.

The result also shows that the method was sensitive to a refrigerant overcharged fault, when the calculation included the refrigerant overcharged, the maximum error for both flow rates were 40.2%, 19.4%, and 18.2% for data with all ΔT, ΔT≥ 2.5°C, and ΔT≥ 3.5°C respectively. But, when the refrigerant overcharged scenario was removed from the analysis, the accuracy improved significantly. The maximum error in the flow rate estimate became 40.2%, 15.3%, and 9.9% for all ΔT, ΔT≥ 2.5°C, and ΔT≥ 3.5°C respectively.

3.2 Water Flow Rate Estimate With ΔT ≥ 4°C

The first scenario simulated a normal operating condition. The data points were taken from the Normal 2 dataset [16]. The use of isentropic work in energy balance analysis can perform well in estimating the average evaporator and condenser water flow rate. The error for both cases was within 1.9% (Table 4).

The second scenario was a reduced condenser water flow rate. For this scenario, the condenser flow rate was reduced by 10% for each fault level. The average evaporator water flow rate can be well-predicted within ±3.2% for all fault severity levels (Table 5). The same was also valid for average condenser water flow rate, the error in estimating the flow rate was within ±1.1% as tabulated in Table 5. For both evaporator and condenser, the error was within ±3.2%.

The next fault type is a reduced evaporator water flow rate (FWE). The evaporator water flow rate is reduced by 10%, 20%,30%, and 40% with the increase of the fault severity level. For the evaporator water flow rate, the model can perform well in estimating the average evaporator water flow rate as shown in Table 6. The average error for the evaporator was within ±3.0%. The same result was also obtained for the condenser water flow rate, the model also works well in predicting the flow rate. It has an average error within ±4.0% as shown in Table 6. For both evaporator and condenser, the error is within ±3.0%.

Table 3
The error in FWC and FWE for different ΔT

No	Scenario	Parameter	All ΔT	ΔT ≥ 2.5 °C	ΔT ≥ 3.5 °C
1	Normal 2	FWC Difference	-19,1%	-7,5%	-2,8%
2	Normal 2	FWE Difference	-17,8%	-6,7%	-1,9%
3	FWC 10%	FWC Difference	-20,7%	-9,0%	-3,9%
4	FWC 20%	FWC Difference	-20,4%	-8,8%	-1,2%
5	FWC 30%	FWC Difference	-24,0%	-11,9%	-1,2%
6	FWC 40%	FWC Difference	-25,9%	-12,3%	-3,0%
7	FWC 10%	FWE Difference	-18,8%	-7,6%	-2,5%
8	FWC 20%	FWE Difference	-18,1%	-7,0%	0,4%
9	FWC 30%	FWE Difference	-20,8%	-8,2%	1,0%
10	FWC 40%	FWE Difference	-22,9%	-10,1%	-0,6%
11	FWE 10%	FWC Difference	-21,3%	-10,1%	-3,0%
12	FWE 20%	FWC Difference	-22,8%	-11,5%	-2,7%
13	FWE 30%	FWC Difference	-23,9%	-13,8%	-4,0%
14	FWE 40%	FWC Difference	-26,2%	-13,8%	-5,3%
15	FWE 10%	FWE Difference	-19,7%	-9,0%	-2,2%
16	FWE 20%	FWE Difference	-20,6%	-10,0%	-1,4%
17	FWE 30%	FWE Difference	-21,6%	-12,3%	-3,0%
18	FWE 40%	FWE Difference	-23,6%	-12,2%	-4,3%
19	RL 10%	FWC Difference	-18,7%	-6,9%	-2,0%
20	RL 20%	FWC Difference	-18,3%	-6,3%	-0,6%
21	RL 30%	FWC Difference	-18,9%	-7,1%	-2,2%
22	RL 40%	FWC Difference	-18,9%	-7,1%	0,4%
23	RL 10%	FWE Difference	-17,6%	-6,4%	-1,4%
24	RL 20%	FWE Difference	-17,3%	-5,7%	0,0%
25	RL 30%	FWE Difference	-17,5%	-5,9%	-0,8%
26	RL 40%	FWE Difference	-21,2%	-8,6%	0,0%
27	RO 10%	FWC Difference	-23,3%	-11,3%	-5,9%
28	RO 20%	FWC Difference	-20,8%	0,6%	10,4%
29	RO 30%	FWC Difference	-25,9%	-16,4%	-13,2%
30	RO 40%	FWC Difference	-28,4%	-19,4%	-18,2%
31	RO 10%	FWE Difference	-22,9%	-11,0%	-5,4%
32	RO 20%	FWE Difference	-20,0%	1,9%	11,9%
33	RO 30%	FWE Difference	-21,9%	-12,1%	-8,6%
34	RO 40%	FWE Difference	-21,3%	-11,8%	-10,5%
35	CF 12	FWC Difference	-37,8%	-13,4%	-9,9%
36	CF 20	FWC Difference	-23,4%	-12,5%	-7,4%
37	CF 30	FWC Difference	-21,9%	-10,7%	-6,2%
38	CF 45	FWC Difference	-25,2%	-14,0%	-8,9%
39	CF 12	FWE Difference	-40,2%	-12,3%	-8,3%
40	CF 20	FWE Difference	-20,2%	-9,5%	-4,2%
41	CF 30	FWE Difference	-18,6%	-7,5%	-2,7%
42	CF 45	FWE Difference	-19,4%	-8,1%	-2,9%
43	NC 1%	FWC Difference	-21,9%	-9,7%	-3,8%
44	NC 2%	FWC Difference	-19,5%	-7,8%	-1,7%
45	NC 3%	FWC Difference	-19,5%	-7,8%	-0,5%
46	NC 5%	FWC Difference	-19,5%	-15,3%	-5,2%
47	NC 1%	FWE Difference	-20,2%	-8,8%	-3,2%
48	NC 2%	FWE Difference	-17,6%	-6,7%	-0,7%
49	NC 3%	FWE Difference	-18,5%	-7,8%	-1,0%
50	NC 5%	FWE Difference	-17,0%	-13,0%	-3,6%

Table 4
The comparison between average flow rate for normal test 2

No	Scenario	FWE (Average)				FWC (Average)			
		Calculated	Measured	Unit	Difference	Calculated	Measured	Unit	Difference
1	Normal	13.84	13.58	kg/s	1.9%	16.99	16.79	kg/s	1.2%

Table 5
The comparison between the average flow rate for reduced condenser flow

No	Scenario	FWE (Average)				FWC (Average)			
		Calculated	Measured	Unit	Difference	Calculated	Measured	Unit	Difference
1	FWC 10%	13.89	13.63	kg/s	1.97%	15.31	15.17	kg/s	0.9%
2	FWC 20%	13.92	13.60	kg/s	2.40%	13.63	13.54	kg/s	0.6%
3	FWC 30%	14.06	13.62	kg/s	3.2%	12.02	11.89	kg/s	1.1%
4	FWC 40%	13.78	13.59	kg/s	1.4%	10.13	10.24	kg/s	-1.1%

Table 6
The comparison between average flow rate for reduced evaporator flow

No	Scenario	FWE (Average)				FWC (Average)			
		Calculated	Measured	Unit	Difference	Calculated	Measured	Unit	Difference
1	FWE 10%	12.29	12.32	kg/s	-0.2%	16.64	16.80	kg/s	-1.0%
2	FWE 20%	10.98	11.11	kg/s	-1.2%	16.46	16.82	kg/s	-2.1%
3	FWE 30%	9.49	9.78	kg/s	-3.0%	16.17	16.84	kg/s	-4.0%
4	FWE 40%	8.57	8.71	kg/s	-1.6%	16.45	16.85	kg/s	-2.4%

To simulate the refrigerant leak, the refrigerant capacity was reduced by 10% for each fault level. The use of isentropic compressor input in energy balance analysis worked well in predicting the average evaporator water flow rate as depicted in Table 7. The error in the average calculated values ranged from 2.3% to 3.6% for all fault severity levels (Table 7). It also worked well in estimating the average condenser water flow rate with an average error ranges from 2.3% to 2.9% as shown in Table 7. For both evaporator and condenser, the error was within ±3.6%.

Table 7
The comparison between average flow rate for reduced refrigerant (LR)

No	Scenario	FWE (Average)				FWC (Average)			
		Calculated	Measured	Unit	Difference	Calculated	Measured	Unit	Difference
1	RL 10%	13.93	13.62	kg/s	2.3%	17.27	16.89	kg/s	2.2%
2	RL 20%	14.05	13.60	kg/s	3.3%	17.33	16.84	kg/s	2.9%
3	RL 30%	14.07	13.58	kg/s	3.6%	17.21	16.79	kg/s	2.5%
4	RL 40%	13.93	13.58	kg/s	2.6%	17.21	16.79	kg/s	2.5%

Opposite to refrigerant leak, the next fault scenario was a refrigerant overcharged. The refrigerant capacity was increased by 10%, 20%, 30%, and 40% to simulate an overcharged condition. The model, as shown in Table 8, can not predict the average evaporator and condenser water flow rate very well for higher fault levels. The mean error for fault level 2 (Evaporator) reaches -13.8% (Table 8). For the evaporator water flow rate, the error can reach -15.5% [1]. One study also experienced the same significant error for a refrigerant overcharged scenario [1]. According to Zhao, when more refrigerant was added to the system, the refrigerant flow increased. The increase in refrigerant flow increased the theoretical compressor input. As a result, the water flow rate became larger. To overcome this issue, one can use one of fault detection and diagnosis (FDD) methods for refrigerant overcharged as proposed by Zhao [12-13] and Comstock [18] to identify this type of fault. For both evaporator and condenser, the error in estimating the flow rate was within ±15.1%.

To simulate condenser fouling, the tubes were plugged at a different level. The condenser consists of 164 tubes. At fault level 1, 12% of tubes were blocked. The next fault level blocked 20% of condenser tubes. At fault level 3, 30% of tubes were plugged. The highest fault level plugged 45% of condenser tubes. The evaporator water flow rate was well-predicted for all fault levels with errors range from -5.3% to 0.8% (Table 9). The model also can estimate the average condenser water flow rate within ±-6.5% (Table 9).

Table 8
The comparison between average flow rate for refrigerant overcharged (RO)

No	Scenario	FWE (Average)				FWC (Average)			
		Calculated	Measured	Unit	Difference	Calculated	Measured	Unit	Difference
1	RO 10%	13.15	13.58	kg/s	-3.2%	16.24	16.79	kg/s	-3.3%
2	RO 20%	15.63	13.58	kg/s	15.1%	19.08	16.79	kg/s	13.6%
3	RO 30%	12.85	13.58	kg/s	-5.4%	15.10	16.79	kg/s	-10.1%
4	RO 40%	12.79	13.58	kg/s	-5.8%	14.47	16.79	kg/s	-13.8%

Table 9
The comparison between average flow rate for condenser fouling (CF)

No	Scenario	FWE (Average)				FWC (Average)			
		Calculated	Measured	Unit	Difference	Calculated	Measured	Unit	Difference
1	CF 12%	12.86	13.58	kg/s	-5.3%	15.73	16.83	kg/s	-6.5%
2	CF 20%	13.26	13.58	kg/s	-2.4%	15.93	16.79	kg/s	-5.1%
3	CF 30%	13.69	13.58	kg/s	0.8%	16.39	16.79	kg/s	-2.4%
4	CF 45%	13.54	13.58	kg/s	-0.3%	15.74	16.79	kg/s	-6.3%

The last fault scenario was performed by adding nitrogen to the system by 1%, 2%, 3%, and 5%. The lab simulation was performed in reverse order [17], and the highest severity fault level was tested first. The average evaporator water flow rates were predicted with an error within ±1.7% (Table 10). In the coolant side, the average condenser water flow rate was estimated within ±2.7% (Table 10). It seems that the model can estimate the water flow rate well-enough for both evaporator and condenser.

Table 10
The comparison between average flow rate for non-condensable gas (NC)

No	Scenario	FWE (Average)				FWC (Average)			
		Calculated	Measured	Unit	Difference	Calculated	Measured	Unit	Difference
1	NC 1%	13.50	13.58	kg/s	-0.6%	16.65	16.79	kg/s	-0.8%
2	NC 2%	13.74	13.58	kg/s	1.2%	16.90	16.79	kg/s	0.7%
3	NC 3%	13.59	13.58	kg/s	0.1%	16.90	16.79	kg/s	0.7%
4	NC 5%	13.35	13.59	kg/s	-1.8%	16.31	16.76	kg/s	-2.7%

4. Conclusions

The virtual flow meter using isentropic compressor work to estimate the evaporator and condenser water flow rate was proposed. The result shows that the method is sensitive to water temperature difference in evaporator and condenser, and the refrigerant overcharged fault. The larger the water temperature difference, the more accurate the water flow rate prediction.

When the calculation of water flow rate included the refrigerant overcharged scenario, the maximum error for both flow rate were 40.2%, 19.4%, 18.2%, and 15.1% for data with all ΔT, ΔT≥ 2.5°C, ΔT≥ 3.5°C, and ΔT≥ 4.0°C respectively. The maximum error reduced about 50% from 40.2% to 19.4% at ΔT≥ 2.5 C. Then, the error slightly improves when data with ΔT≥ 3.5° C and ΔT≥ 4.0° C used in the calculation. The result indicates that the maximum error in flow rate improved at higher water temperature difference. In addition, when the data with the refrigerant overcharged scenario was removed from the analysis, the accuracy of flow rate prediction improves significantly. The maximum error in flow rate estimate became 40.2%, 15.3%, 9.9%, and 6.5% for all ΔT, ΔT≥ 2.5°C, ΔT≥ 3.5°C, and ΔT≥ 4.0°C respectively. Future research should apply the proposed method at real chiller operation and the use of nano refrigerants in the chiller system [19].

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