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ABSTRACT

This paper deals with the development of the performance evaluation method for a split air conditioning system using the compressor characteristic curve. In a split air conditioning system, refrigerants expand directly in an evaporator and heat-exchange is done on the indoor unit. It is too difficult to measure the actual capacity and the accurate performance evaluation method is not established.
The performance evaluation method using the compressor curve method based on the characteristic curve of compressor mass flow rate was developed. We examine the application of the method to this system which has different compressor type. The regression equations for refrigerant mass flow rate were obtained based on characteristic curve of the sliding vane compressor. The relative error between air conditioning capacity based on measurement and regression equation is obtained within 9.0%. The applicability of method for another type compressor is confirmed.

1. INTRODUCTION

A split air conditioning system was installed in comparatively small building or used in order to compensate for central air conditioning systems in Japan. However, these systems were adopted rapidly as main air conditioning system in a building larger than 100,000 m^2 because of highly efficiency and easy installation. But, the actual performance evaluation method of split air conditioning systems isn't established.
In Japan, current performance evaluation method of these systems is used Air-enthalpy method which prescribed by JIS (Japan Industrial Standard) and JRA (The Japan Refrigeration and Air Conditioning Industry Association) rules. This method is measured the test unit in the calorie box, so measuring condition is limited. The results of this method quite differ from actual performance. For these reasons, Kato (*et al*, 2009) developed the performance evaluation method measuring the heat flux on the outdoor unit. They measured three-dimensional wind velocity, temperature and relative humidity of airflow on the outdoor unit. However, it is so hard to measure the gas engine-driven heat pump air conditioning systems (GHP) by this method. Because the gas engines waste heat contained a heat flux blowing out from outdoor unit. And a heat flux on outdoor unit is affected by disturbance so it is different to measure the exact value in actual condition. Additionally, Takahashi (*et al*, 2008) evaluated the performance of GHP which had scroll compressor using compressor curve method. This method is based on the characteristic curves of compressor mass flow rate. Cooling and heating capacity were given by multiplying the approximated

mass flow rate by the differential of refrigerant enthalpies pass the indoor unit's heat exchanger. As the result, the relative error between the performance of measurement and approximate was within 10.0% in actual condition. In this study, we measure the split air conditioning system which has another type of compressor, so the effectiveness and the applicability of this method are confirmed.

2. EXPERIMENTAL METHOD AND EQUIPMENTS

2.1 Out line of compressor curve method

The refrigerant mass flow rate is calculated by compressor revolution and saturation temperature of suction and discharge. Cooling and heating capacity are estimated using calculated refrigerant mass flow rate on the Mollier diagram. The discharge rate of compressor is able to be clarified by compressor revolutions, density of suction compressor, volumetric efficiency and compressor displacement. Generally speaking, these values are not disclosed to the public, so we can not estimate refrigerant mass flow rate exactly. However if the super heat of refrigerant and compressor are constant, the suction refrigerant density and pressure ratio between suction and discharge are able to be obtained from saturation temperature on suction and discharge of compressor. So we made the regression equation in order to determine refrigerant mass flow rate. The equations are obtained by refrigerant mass flow rate as response variable as G_{comp} and saturation temperature of suction and discharge as predictive variables as T_s and T_d. Furthermore the parameter of compressor revolution was added to the equations for applied to a wide range of revolutions. The regression equations are shown in (1) and (2).

$$G_{comp} = a_1 \times T_s + a_2 \times T_d + a_3 \tag{1}$$

$$a = b_1 \times N + b_2 \tag{2}$$

2.2 Experimental method

To clarify a regression equation, refrigerant mass flow rate and pressure of suction and discharge were measured in the constant compressor revolution before the actual measurement. Then the measurement points are as follows: (1) refrigerant mass flow rate, (2) gas engine and compressor revolutions, (3) temperature (refrigerant on suction and discharge of compressor, subcooler, outdoor, etc), (4) pressure of the refrigerant, (5) electric power of outdoor unit and indoor units, (6) fuel gas volume flow rate. The test unit was operating actual condition.

2.3 Experimental equipments

The test unit (GHP) was set in the experiment station at Tokyo Gas Co., Ltd. The test unit had four indoor units and one outdoor unit. Test unit has two sliding vane compressors that are connected with gas engine by V-belt. To engage or to disengage the clutch, number of operating compressors is able to be change. It is possible to change the discharge volume in order to comply with the thermal load. So we obtained two types regression equation, one is single drive mode (compressor 1) and another is double drive mode (compressor 1&2). The piping diagram of test unit is shown as Figure 1 and specifications of a test unit are shown in Table 1.

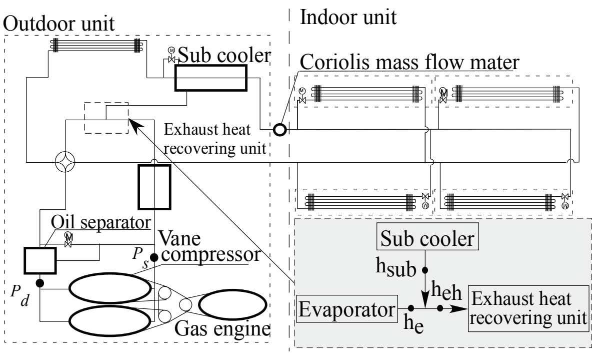

Figure 1: Piping diagram of test unit

Table 1: Specifications of a test GHP unit

Designation		Specifications
Compressor Type		Sliding vane compressor
Compressor Displacement	No.1	172.6 cc/rev
	No.2	98.2 cc/rev
Fuel gas		LNG
Refrigerant		R410A
Rating Capacity	Cooling	56.0 kW
	Heating	63.0 kW

2.4 Experimental conditions

The Measurement was performed through one year. The preliminary experiment was done for creating regression equation model before the actual measurement. For this model, we measured refrigerant mass flow rate and saturation temperature on the suction and discharge of the compressor every 10 seconds. The constant compressor revolutions are shown in Table 2 and Table 3. Outdoor and indoor temperatures are shown in Figure 2(in cooling operation) and Figure 3(in heating operation). In cooling operation, temperature of indoor units which were set on experiment station were over 24℃ constantly. And the temperatures were over 26℃ in heating operation.

Table 2: Gas engine and Compressor revolution in cooling operation

Operating mode	Engine Revolution [rpm]	Compressor Revolution [rpm]
Compressor 1 1 & 2	950	1349
	1200	1704
	1500	2130
	1800	2556
	2100	2982

Table 3: Gas engine and Compressor revolution in heating operation

Operating mode	Engine revolution [rpm]	Compressor revolution [rpm]
Compressor 1 1 & 2	1200	1704
	1500	2130
	1800	2556

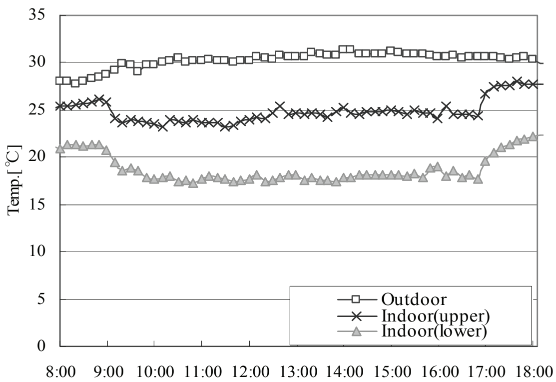

Figure 2: Outdoor and indoor temperature in cooling operation

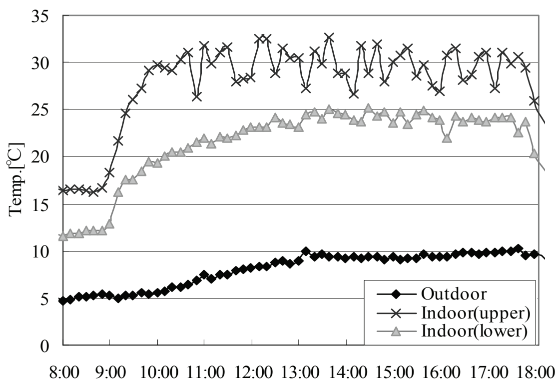

Figure 3: Outdoor and indoor temperature in heating operation

3. EXPERIMENTAL RESULTS AND DISCUSSION

3.1 Refrigerant mass flow rate

The refrigerant mass flow rate is measured with a coriolis mass flow mater. The test unit has the heat exchanger which is called subcooler. Subcooler is separated before indoor heat exchanger and confluent before exhaust heat recovery unit on piping as Figure 1. It is difficult to measure refrigerant mass flow rate passed through sub cooler, so these mass flow rates were calculated by the ratio of enthalpy before and after the confluence point. The heat balance at the confluence point is shown by equation (3). Equation (4) is rewritten from equation (3) for refrigerant mass flow rate which passed through subcooler. And the refrigerant mass flow rate after confluence point is shown by equation (5).On the other hand, because all refrigerant passed through oil separator, this value was dealt with equal to compressor discharge.

$$(G_e + G_{sub}) \times h_{eh} = G_e \times h_e + G_{sub} \times h_{sub} \tag{3}$$

$$G_{sub} = \frac{(h_{eh} - h_e)}{(h_{sub} - h_{eh})} \times G_e \tag{4}$$

$$G_{eh} = G_e + G_{sub} \tag{5}$$

3.2 Comparison of refrigerant mass flow rate between approximation and measurement

The refrigerant mass flow rate by approximation and measurement, and relative error on each seasons are shown in Figure 4 and Figure 5. Relative error between approximation and measurement were so large when the test unit was starting or engaging the clutch. At that time, different control from normal operating was done. A part of discharge refrigerant didn't flow through the indoor units. However, relative error was obtained within 12.0% in normal operating. Furthermore, the error was obtained within 7.0% in continuous operating.

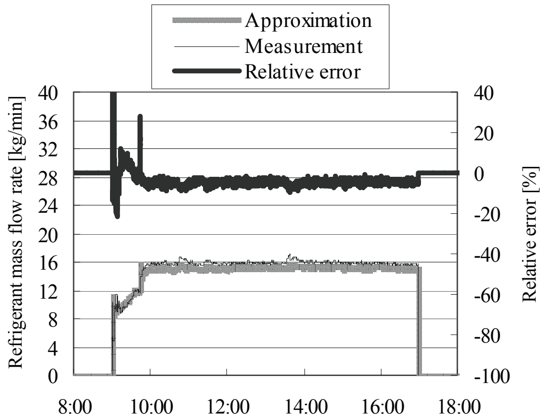

Figure 4: Refrigerant mass flow rate and relative error in cooling operation

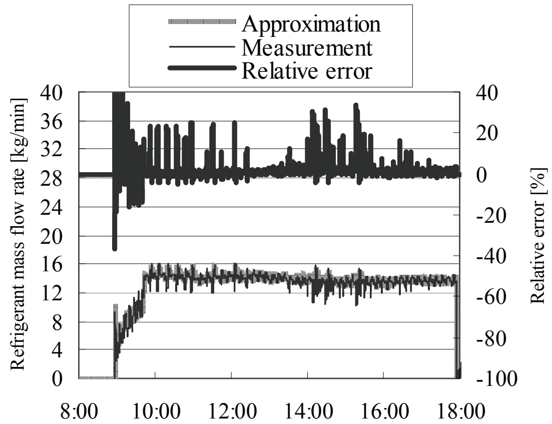

Figure 5: Refrigerant mass flow rate and relative error in heating operation

3.3 Comparison of air condoning capacity between based on approximation and measurement

The cooling and heating capacity based on measurement refrigerant mass flow rate and approximant, relative error between these values are shown in Figure 6. In immediately after starting, the test unit repeated of starting and stopping. The integration of heating capacities was 231.5[kW] based on approximant. As heating capacity calculated based on measurement, the integration was 226.4[kW]. Then the relative error between these values is only 2.3%, so the relative error of refrigerant mass flow rate in immediately after starting is neglected. The relative error between the integrative capacity based on measurement and regression equation on each month is shown in Table 4. The relative errors were controllable within 9.0% in cooling operation and within 4.0% in heating operation. As a result, the effectiveness of this method was clarified.

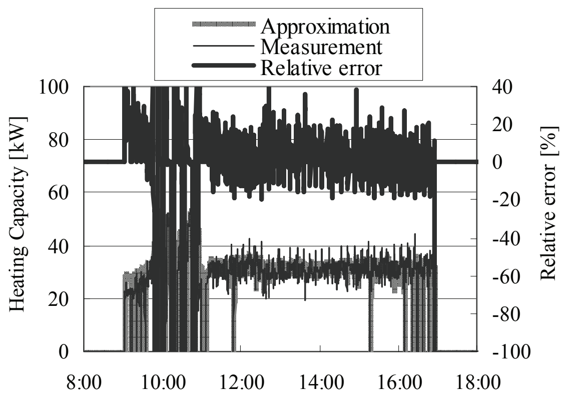

Figure 6: Capacities calculated from measurement and regression model and relative error

Table 4: The relative error of integrative capacity

Month	Relative error of integrative capacity [%]
August	-7.44
September	-1.89
October	8.86
December	0.36
January	3.35
February	0.82
total	-1.82

3.4 The result of analysis using compressor curve method

The various performance of a split air conditioning system is estimated in detail using compressor curve method. COP ratio is defined as COP calculated from compressor curve method divided by rating COP. The load factor is defined as measurement capacity divided by rating capacity. The histograms of load factor are shown in Figure 7 and Figure 8. In cooling operation, the relative frequency of the operating hour is the highest when load factor is 50.0%. In the case of heating operation, the relative frequency of the operating hour is the highest when load factor is 70 to 80.0%. Relationships of COP ratio and load factor on each outdoor temperature were shown in Figure 9 and Figure 10. When the load factor in cooling operation is less than 45.0%, COP ratio is in proportion to the increase of load factor. In the case of heating operation, COP ratio is the peak value when load factor is 55.0%. Then when outdoor temperature was getting lower, COP ratio was increased in the case of the same load factor.

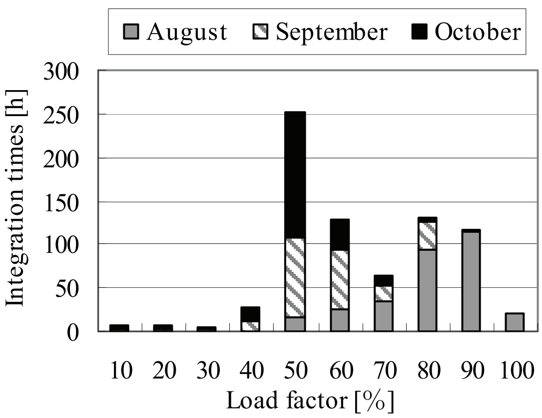

Figure 7: The histogram of load factor in cooling operation

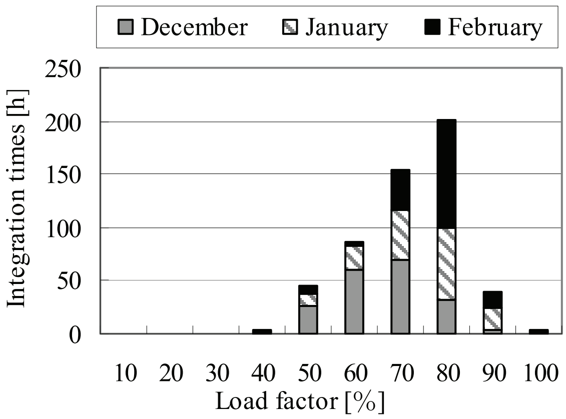

Figure 8: The histogram of load factor in heating operation

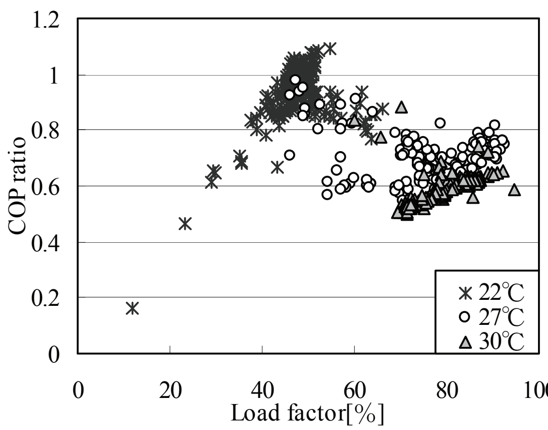

Figure 9: COP ratio and Load factor each outdoor temperature in cooling operation

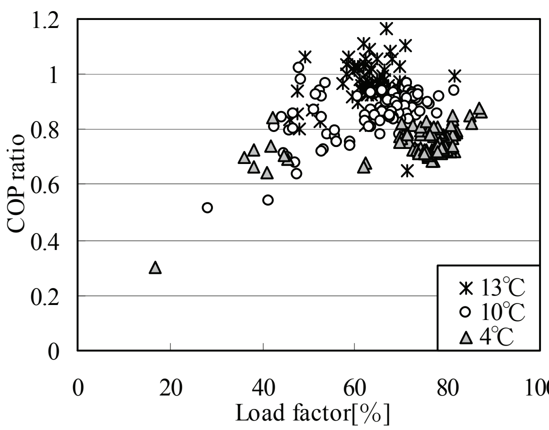

Figure 10: COP ratio and Load factor each outdoor temperature in heating operation

3.5 Application of compressor curve method

According to similarity of fluid machinery, it is possible to estimate refrigerant mass flow rate using ratio with different compressor displacement in case of similar compressor type. However, effectiveness of this method has not

verified in the case of different type compressor. Then the refrigerant mass flow rate of the test unit was obtained using the known scroll compressor's characteristic curve and compared with refrigerant of approximant.
Regression equipments are shown in equation (6) and equation (7).The response variable of regression equation is the value that refrigerant mass flow rate of scroll compressor divided by compressor displacement as G_u. The refrigerant mass flow rate is obtained by the product of this value and this compressor's displacements as $G_{comp,}$ and it is shown in equation (8). Specification of scroll compressor is shown in Table 5. The driving modes of compressor are as follows: (1) compressor 1 drive in cooling operation (2) compressor 2 drive in cooling operation (3) compressor 1 drive in heating operation. The regression equations, multiple correlation coefficient and standard errors are shown in Table 6 and the regression equations, correlation coefficient and standard errors are shown in Table 7.
As the results, comparison between refrigerant mass flow rate of approximate and measurement, the maximum relative error is 9.7% in the case of compressor 1 in cooling operation, -24.8% in the case of compressor 2 in cooling operation and -12.6% in heating operation. It seems that the causes of these errors are volumetric efficiency, difference of suction pressure and different compressor type. Then, correction factors are obtained about two driving mode (compressor 2 drive in cooling operation / compressor 1 in heating operation). As a result, the relative error of compressor 2 drive in cooling operation is able to be obtained within 18.0%, and it is possible to obtain the errors of compressor 1 in cooling operation and heating within 10.0%. Regression curves of refrigerant mass flow rate which were multiplied the correction coefficient s are shown in Figure 11 to Figure 13.
The relative error is obtained within 18.0% using different type compressor's characteristic curve from the test unit. However, it is unnecessary to obtain the regression equation each test units, so this method is easier than the present compressor curve method. Accordingly, it is effective to evaluate the actual performance of a split air conditioning system as simple method.

$$G_u = c_1 \times T_s + c_2 \times T_d + c_3 \tag{6}$$

$$c = d_1 \times N + d_2 \tag{7}$$

$$G_{comp} = G_u \times V \tag{8}$$

Table 5: Specification of scroll compressor

Designation	Specifications
Compressor Type	Scroll compressor
Compressor Displacement	120 cc/rev
Refrigerant	R410A

Table 6: Regression equations of refrigerant mass flow rate

Gas engine Revolution(rpm)	Regression Equation	Multiple Correlation Coefficient	Standard Error (kg/min)
700rpm	$G_u = 1.16\times10^{-3}T_s + 4.62\times10^{-5}T_d + 3.90\times10^{-2}$	0.701	2.86×10^{-3}
1100rpm	$G_u = 1.87\times10^{-3}T_s - 9.48\times10^{-4}T_d + 3.43\times10^{-2}$	0.928	2.86×10^{-3}
1500rpm	$G_u = 2.95\times10^{-3}T_s + 1.15\times10^{-4}T_d + 7.83\times10^{-2}$	0.912	2.15×10^{-3}
2000rpm	$G_u =3.87\times10^{-3}T_s + 3.82\times10^{-4}T_d + 9.65\times10^{-2}$	0.992	1.81×10^{-3}

Table 7: Regression equations of partial regression coefficients

partial regression coefficients	Regression Equation	Correlation Coefficient	Standard Error (kg/min)
T_s term	$a_1 = -3.73\times10^{-4}N + 2.14\times10^{-6}$	0.993	1.18×10^{-4}
T_d term	$a_2 = 3.84\times10^{-8}N + 3.22\times10^{-4}$	0.003	5.01×10^{-4}
constant term	$a_3 = 5.04\times10^{-5}N - 4.74\times10^{-3}$	0.856	1.41×10^{-2}

Table 8: The correction Factor

	Comp.2 in cooling	Comp.1 in heating
Correction Factor	1.05	1.03

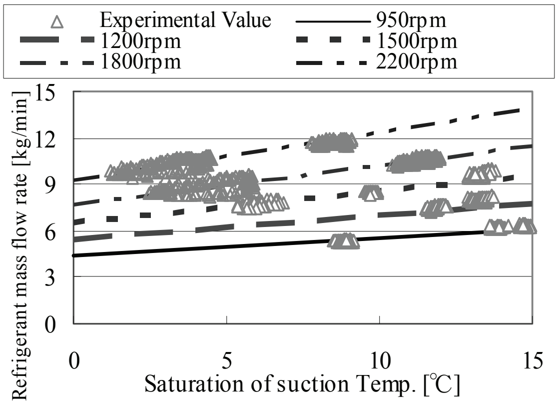

Figure 11: Regression curves of refrigerant mass flow rate in cooling operation (Saturation of discharge temp: 35°C, displacement: 98.1cc)

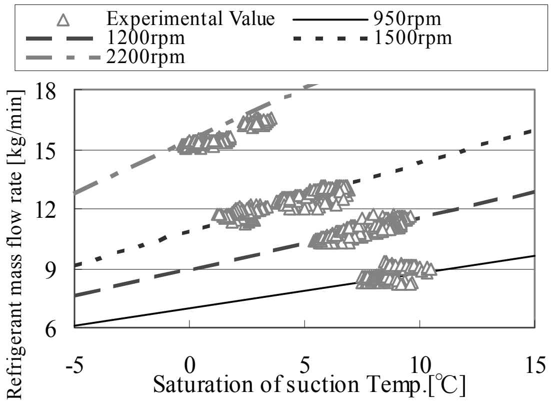

Figure 12: Regression curves of refrigerant mass flow rate in cooling operation (Saturation of discharge temp: 35°C, displacement: 172.6cc)

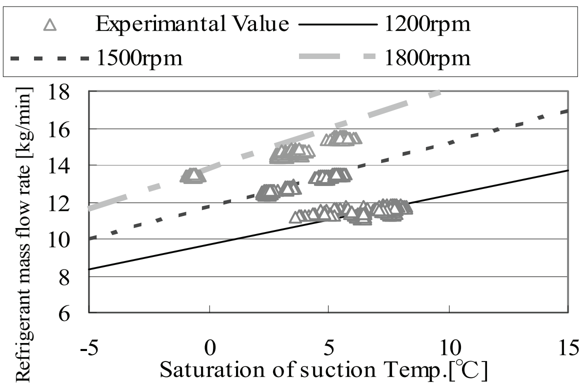

Figure 13: Regression curves of refrigerant mass flow rate in heating operation (Saturation of discharge temp: 43°C, displacement: 172.6cc)

4. CONCLUSIONS

This study deals with extending the applicability of compressor curve method by analyzing the actual performance data of GHP which has sliding vane compressor. The following results were obtained.

- Relative error between refrigerant mass flow rates based on regression equation and measurements were obtained within 12.0% in normal operating. Furthermore, the errors were obtained within 7.0% in continuous operating by constant load. As a result, extending the applicability of compressor curve method to GHP which has sliding vane compressor was confirmed.
- When test unit was starting and clutch on compressor was operating, relative error of approximations and measurements was large because refrigerant's quantity of state is not steady. However, heat exchange is not done in indoor units and the values of capacity are very small. In consequence, these values are not large error.

- The various performances of a split air conditioning system are able to be estimated in detail using compressor curve method.
- The relative error between refrigerant of measurement and approximant is expected within 18.0% using different type compressor's characteristic curve from the test unit. Accordingly, it is effective to evaluate the actual performance of a split air conditioning system as simple and broad method.

For the above mentioned result, the applicability of compressor curve method was confirmed measuring GHP which had sliding vane compressor. This method is not effective only GHP but also Electric-driven Heat Pump system (EHP) units.

NOMENCLATURE

G	refrigerant mass flow rate	(kg/min)	**Subscripts**	
T	temperature	(℃)	comp	compressor
P	pressure	(MPa)	s	suction
h	enthalpy	(kJ/kg)	d	discharge
N	revolution	(rpm)	in	inlet
Q	capacity	(kW)	out	outlet
V	compressor displacement	(cc/rev)	sub	subcooler
			u	per unit displacement

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