Estimation of Vapor Quality at Compressor Suction of Air Conditioner

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Abstract: To evaluate performance and reliability of an air conditioner, the states of its refrigerant in the refrigeration cycle need to be understood. Since the isentropic exponent of the next-generation refrigerant R32 is larger than that of conventional refrigerant R410A, the compressor discharge temperature of R32 is higher than that of R410A. When a wet refrigerant is entered into a compressor suction line to decrease the discharge temperature, it is generally difficult to estimate the vapor quality at the suction. This paper describes a method that estimates a vapor quality at the compressor suction by applying energy balances on the compressor with the values of suction pressure, discharge pressure, discharge temperature, etc. In the air conditioner test, the vapor qualities at the compressor suction estimated with this method agreed well with those estimated with the conventional method by measuring liquid and vapor flow-rates. Moreover, this paper derived a relational expression for estimating a compressor suction state from a discharge state and discusses the characteristics of the coefficients in the expression calculated from the test results.

Key words: Air conditioner, compressor, wet refrigerant, vapor quality, R32.

1. Introduction

HFCs (hydrofluorocarbons), which have been used as refrigerants, have high GWPs (global warming potentials). They were thus designated as emission control substances under the Kyoto Protocol in 1997. In 2014, the EU (European Union) adopted revised F-gas regulation, which includes CO₂ equivalent phase-down of HFCs to be placed on the EU market [1]. In Japan, revised fluorocarbon law was enacted in 2013 and will come into force in 2015 to promote the use of lower GWP fluorocarbons [2]. Under these circumstances, room air conditioners using the next-generation refrigerant R32 (GWP = 675), whose GWP is one third that of a conventional refrigerant R410A (GWP = 2,088), have been commercialized in Japan [3]. Since the isentropic exponent of R32 is

larger than that of R410A, the compressor discharge temperature of R32 is higher than that of R410A [4]. The conventional way to decrease the discharge temperature is to enter the wet (two-phase vapor-liquid) refrigerant into the compressor suction line [5]. In the compressor test setup that connects a bypass line from the condenser outlet to the compressor suction line, a vapor quality, which represents the state of wet refrigerant, at the compressor suction can be set with flow-rates and enthalpies of vapor from the evaporator outlet and liquid from the bypass line [6, 7]. However, it is generally difficult to estimate a vapor quality at the compressor suction during the practical operation of an air conditioner.

This paper aims at estimation of a vapor quality at the compressor suction by applying energy balances on the compressor with the values of suction pressure, discharge pressure, discharge temperature, etc. Moreover, this paper derives a relational expression for estimating a compressor suction state from a discharge state and discusses the characteristics of the

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coefficients in the expression calculated from the test results.

2. Estimation of Vapor Quality at Compressor Suction

2.1 Estimation of Suction Quality by Applying Energy Balances on Compressor

Vapor quality x is defined as the ratio of the total mass of two-phase vapor-liquid refrigerant m and the mass of saturated vapor m'' :

$$x = \frac{m''}{m} \quad (1)$$

Quality x can also be expressed as the ratio of the total mass flow-rate of two-phase vapor-liquid refrigerant G_r and the mass flow-rate of saturated vapor G_r'' :

$$x = \frac{G_r''}{G_r} \quad (2)$$

Moreover, quality x is given in terms of the enthalpy of two-phase vapor-liquid refrigerant h , the enthalpy of saturated liquid h' , and the enthalpy of saturated vapor h'' :

$$x = \frac{h - h'}{h'' - h'} \quad (3)$$

Since the mass m'' and the mass flow-rate G_r'' of saturated vapor in Eqs. (1) and (2) cannot be measured directly, a method to estimate the quality at the compressor suction will be provided by using the relationship between a quality and enthalpies given in Eq. (3).

A suction quality x_s is expressed by Eq. (3), using index s :

$$x_s = \frac{h_s - h_s'}{h_s'' - h_s'} \quad (4)$$

With Eq. (4), even when the suction state is superheated vapor, the state can be represented as a number (> 1.0).

Fig. 1 shows the P - h diagram for representing the relationships between enthalpies at the compressor

suction and discharge. The method to estimate a suction quality x_s by applying energy balances on the compressor is described below. This method uses the values of a suction pressure P_s , a discharge pressure P_d , a discharge temperature T_d , a refrigerant mass flow-rate G_r , a compressor power consumption W_c , and a compressor ambient temperature T_a during the practical operation of the air conditioner. A suction quality x_s will be identified in such a way that a discharge enthalpy h_{d2} estimated from an assumed suction quality x_s by using energy balances is the same as the enthalpy h_d calculated from the discharge pressure P_d and temperature T_d .

A suction enthalpy h_s is represented as follows from Eq. (4), assuming the value of the quality x_s :

$$h_s = h_s' + x_s(h_s'' - h_s') \quad (5)$$

A compressor efficiency η_c is given in terms of the discharge enthalpy h_{dth} calculated under the assumption of an isentropic process, the measured refrigerant mass flow-rate G_r , and the measured compressor power consumption W_c :

$$\eta_c = G_r(h_{dth} - h_s)/W_c \quad (6)$$

A discharge enthalpy $h_{d'}$ calculated under the assumption of no heat radiation from the compressor is given in terms of compressor efficiency η_c :

$$h_{d'} = h_s + W_c / G_r = h_s + (h_{dth} - h_s) / \eta_c \quad (7)$$

The discharge enthalpy h_{d2} estimated from the assumed value of suction quality x_s is represented as follows, letting Q_a be the practical heat radiation from the compressor:

$$h_{d2} = h_{d'} - Q_a / G_r \quad (8)$$

The heat radiation from the compressor Q_a is modeled as follows, using KA value (K : overall heat transfer coefficient, A : heat transfer area) of heat radiation from the compressor KA_c and the ambient temperature T_a :

$$Q_a = KA_c(T_d - T_a) \quad (9)$$

Fig. 1 Relationships between enthalpies at compressor suction and discharge.

The suction quality x_s can be identified and estimated in such a way that the enthalpy h_{d2} in Eq. (8) is the same as the enthalpy h_d calculated from the discharge pressure P_d and temperature T_d .

2.2 Estimation of Suction Quality by Measuring Liquid and Vapor Flow-Rates

This method is the conventional one used in the compressor test setup. In this study, the air conditioner test setup is constructed so that liquid and vapor flow-rates can be measured. The suction quality can be estimated by reproducing the refrigeration cycle in such a way that the suction pressure P_s , discharge pressure P_d , and discharge temperature T_d are the same as those during the practical operation.

Fig. 2 shows the configurations of refrigeration cycle and P - h diagrams in the practical operation test and reproduction test. In the refrigeration cycle of the reproduction test, part of the liquid refrigerant from the condenser outlet is bypassed, then depressurized, and finally returned to the compressor suction line, joining the superheated vapor refrigerant through the evaporator. The bypassed liquid flow-rate is controlled with a pressure reducing valve in such a way that pressure P_s , pressure P_d , and temperature T_d are the same as those of the refrigeration cycle of the practical operation test. Let h_{sv} be the superheated

vapor enthalpy, h_{bl} the bypassed liquid enthalpy, G_r the total refrigerant mass flow-rate, and G_{bl} the bypassed refrigerant mass flow-rate. Then the suction enthalpy h_s is given by:

$$h_s = \frac{h_{sv}(G_r - G_{bl}) + h_{bl}G_{bl}}{G_r} \quad (10)$$

The suction quality x_s is obtained with Eq. (4).

2.3 Expression for Estimating Compressor Suction State

A relational expression for estimating a compressor suction state from a discharge state is discussed below. The state point s_{th} is newly defined as the state of compressor suction from which the refrigerant changes to the practical state point of the compressor discharge d under the assumption of an isentropic process. Fig. 3 shows relationships between enthalpies at the compressor suction and discharge in this case. The suction enthalpy h_s is represented as follows, using the discharge enthalpy h_d , the isentropic compression work based at the practical state point of the compressor discharge ($h_d - h_{sth}$), and the estimating coefficient for the suction state η_c' .

$$h_s = h_d - (h_d - h_{sth})/\eta_c' \quad (11)$$

Eq. (11) can be transformed with respect to η_c' as follows:

Fig. 2 Configurations of refrigeration cycle and P - h diagrams.

s : Practical state point of compressor suction;
 d : Practical state point of compressor discharge;
 d_{th} : State point of compressor discharge to which refrigerant changes from compressor suction state point s under assumption of isentropic process;
 d' : State point of compressor discharge to which refrigerant changes from compressor suction state point s under assumption of no heat radiation from compressor;
 s_{th} : State point of compressor suction from which refrigerant changes to compressor discharge state point d under assumption of isentropic process.

Fig. 3 Relationships between enthalpies at compressor suction and discharge with s_{th} added.

$$\eta_c' = \frac{h_d - h_{s_{th}}}{h_d - h_s}$$

$$= \frac{h_d - h_{s_{th}}}{h_{d_{th}} - h_s} \frac{h_{d'} - h_s}{h_d - h_s} \frac{h_{d_{th}} - h_s}{h_{d'} - h_s} \quad (12)$$

where, $h_{d'}$ is the discharge enthalpy under the assumption of no heat radiation from the compressor

and $h_{d_{th}}$ is the discharge enthalpy under the assumption of the isentropic process from the suction enthalpy h_s , as described before. The first ratio in Eq. (12) is (the isentropic compression work with regard to the practical state point of the compressor discharge)/(the isentropic compression work with regard to the practical state point of the compressor suction), the second ratio is (the enthalpy difference

between the suction and the discharge under the assumption of no heat radiation from the compressor)/(the enthalpy difference between the suction and the practical discharge), and the third ratio is the compressor efficiency η_c . Let k_w be the first ratio, which is related to the isentropic compression work, k_r be the second ratio, which is related to the radiation heat from the compressor, and k_p be the product of these ratios. Then the estimating coefficient for the suction state η_c' is represented as follows:

$$\eta_c' = k_w k_r \eta_c = k_p \eta_c \quad (13)$$

Let $h_d = h_{d2}$ in Eq. (8), and then k_r is transformed as follows using Eq. (8):

$$k_r = \frac{h_{d'} - h_s}{h_d - h_s} = \frac{(h_d - h_s) + Q_a / G_r}{h_d - h_s} \quad (14)$$

Values of k_w and k_r are both larger than 1.

To evaluate the validity of the method to estimate the suction quality by applying energy balances on the compressor, this method will be compared below with the conventional method by measuring the liquid and vapor flow-rates in a room air conditioner test. The characteristics of the coefficients in the relational expression for estimating the compressor suction state calculated from the test results will also be discussed.

3. Experimental Setup and Test Procedure

A room air conditioner used for the test is one for R410A with a rated cooling and heating capacities of 7.1 kW and 8.5 kW, respectively. Fig. 4 shows the system diagram of the experimental setup. The compressor is an inverter-driven scroll compressor. R410A was replaced with R32, and refrigeration oil developed for R32 was used. The air conditioner was modified for the experimental setup as to bypass the line before an expansion valve and connect it with the compressor suction line through a pressure reducing valve. Pressures and temperatures of the refrigerant were measured with strain gauge pressure transducers and sheath type thermocouples, respectively. A total

mass flow-rate and a bypassed flow-rate of the refrigerant are measured with Coriolis flowmeters. The power consumption of the compressor was measured with a power meter.

In the experiment, first the practical operation test was conducted by closing a bypass valve. The reproduction test was then conducted by opening the bypass valve and controlling the bypass flow-rate with the pressure reducing valve in such a way that the suction pressure P_s , discharge pressure P_d and temperature T_d were the same as those of the practical operation test. At this time, to obtain the enthalpy of the refrigerant through an evaporator and a 4-way valve, the expansion valve was controlled in such a way that the refrigerant was superheated at the superheated temperature of 8 °C. In this case, the refrigerant state in the evaporator was different from that in the practical operation test, and this resulted in a difference between the suction pressure P_s and that in the practical operation. The air temperature conditions were then adjusted to match the P_s .

Table 1 shows test conditions. Outdoor air temperatures for heating operation were -15 °C and -20 °C, under which discharge temperatures of the compressor become very high. Discharge temperatures were varied by adjusting the expansion valve at each compressor rotational speed.

The suction qualities were calculated with measured values in the reproduction test with the methods by applying energy balances on the compressor and by measuring flow-rates. REFPROP (reference fluid properties) [8] was used for the thermodynamic properties of R32. The average value of heat radiation coefficients from the compressor KA_c calculated from Eqs. (5)-(9) under the heating condition at the outdoor air temperature of 7 °C, where the discharge temperature is not very high and the suction state can be set to be superheated, was used as KA_c in the method by applying energy balances. A compressor ambient temperature was measured with a thermocouple installed near the compressor inside the outdoor unit.

Fig. 4 System diagram of experimental setup.

Table 1 Test conditions.

Operation		Heating		
Indoor air temperature	Dry bulb (°C)	20		
	Wet bulb (°C)	15		
Outdoor air temperature	Dry bulb (°C)	-15, -20		
	Wet bulb (°C)	-		
Compressor rotational speed (min ⁻¹)	3,250	5,350	7,150	
Indoor fan rotational speed	Low	Medium	High	
Outdoor fan rotational speed	Medium	Medium	High	

4. Experimental Results and Discussions

4.1 Comparisons of Estimated Vapor Qualities at Compressor Suction

Fig. 5 compares suction qualities x_s estimated with methods by applying energy balances on the compressor and by measuring flow-rates. Suction qualities estimated with the energy balancing method agree well with those estimated with the flow-rate measuring method within $\pm 0.5\%$ in the quality range of 0.86–1.01 at each air temperature and each compressor rotational speed. The energy balancing method can estimate the suction quality as precisely as the flow-rate measuring method without adding the liquid bypass line to the original refrigeration cycle and is therefore effective.

4.2 Characteristics of Coefficients in Relational Expression for Estimating Compressor Suction State

This subsection will discuss the characteristics of

the coefficients in the relational expression for estimating the compressor suction state calculated from the test results. Fig. 6 shows the relationships among suction quality, estimating coefficient, and its related ratios. The discharge temperature T_d decreases as suction quality x_s decreases. T_d increases as compressor rotational speed increases, because the pressure ratio of the compressor also increases. The temperature barely rises between the compressor rotational speeds of 5,350 min⁻¹ and 7,150 min⁻¹, because the indoor and outdoor fan rotational speeds are increased. The compressor efficiency η_c decreases as rotational speed increases and suction quality x_s decreases. η_c also decreases as outdoor air temperature decreases. One reason for η_c decreasing as x_s decreases is that the increase in the amount of liquid refrigerant in the compression chamber as x_s decreases leads to the increase in the amount of the refrigerant that dissolves in the refrigeration oil and the decrease in the viscosity of the oil. This results in the decrease in the sealing performance in the compressor chamber.

The ratio $k_w (= (h_d - h_{sth}) / (h_{dth} - h_s))$ increases as compressor rotational speed increases and suction quality x_s decreases. This is because the decrease in the compressor efficiency η_c as compressor rotational speed increases and x_s decreases leads to the increase in the difference between h_{dth} and h_d . k_w also increases

Fig. 5 Comparisons of suction qualities estimated with methods by applying energy balances and measuring flow-rates.

Fig. 6 Relationships between suction qualities and estimating coefficients

as outdoor air temperature decreases, which leads to the decrease in η_c .

The characteristics of the ratio k_r ($= (h_d - h_s)/(h_d - h_s)$) will be explained with reference to Eqs. (9) and (14). The refrigerant flow-rate G_r increases as compressor rotational speed increases and

$1/G_r$ in Eq. (14) then decreases. Therefore, k_r decreases. The decrease in T_d as x_s decreases leads to the decrease in the heat radiation from the compressor Q_a , resulting in the decrease in k_r . As outdoor air temperature decreases, the increase in the difference between T_d and the ambient temperature T_a leads to

the increase in Q_a and G_r decreases. Therefore, k_r increases.

The range of product k_p becomes small, because the tendencies of k_w and k_r are opposite with regard to the compressor rotational speed and x_s . Especially with changing x_s , k_p barely changes and is nearly constant. As compressor rotational speed increases, k_p increases. k_p also increases as outdoor air temperature decreases. η_c' that is the product of η_c and k_p decreases as x_s decreases. The range of η_c' is smaller than that of η_c with regard to the compressor rotational speed, because η_c decreases and k_p increases as compressor rotational speed increases.

5. Conclusions

From this study, it can be concluded that:

- Suction qualities estimated with the method by applying energy balances on the compressor agree well with those estimated with the conventional method by measuring liquid and vapor flow-rates;
- The energy balancing method can estimate the suction quality without adding the liquid bypass line to the original refrigeration cycle and is therefore effective;
- The suction enthalpy is represented by using the discharge enthalpy, the isentropic compression work based at the practical state point of the compressor discharge, the compressor efficiency, the ratio related with isentropic compression work, and the ratio related with the heat radiation from the compressor;
- The product of the ratios is nearly constant as suction quality changes.

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