Prediction Using Equivalent Ratio in Estimating of the Performance of Heat Pump Air Conditioner

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PREDICTION USING EQUIVALENT RATIO IN ESTIMATING OF THE PERFORMANCE OF HEAT PUMP AIR CONDITIONER

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1. INTRODUCTION

Capacity controlled heat pump air conditioner which is equipped with variable rotating speed compressor and fans has been developed to satisfy comfort requirement in response to the variations of air condition for consumer use in Japan.

More recently, for satisfying more efficient energy savings and comfort requirement, increasing interest has been focused on optimization control of the parameters such as refrigerant flow rate and air flow rates in both indoor and outdoor heat exchangers to achieve the maximum value of COP in heat pump operating at a given air condition.

With regard to system design optimization, most of the focus in the previous works done on the evaluation of the performance analysis has been turned to the case of non-capacity controlled heat pump model /1/.

However, very little information has been given about the system optimization of the capacity controlled heat pump air conditioner. The objective of the present investigation is to develop an optimization method available for predicting the optimum performance of a heat pump air conditioner. In particular, the evaluation of the performance in such a heat pump air conditioner was performed with reasonable model combined compression displacement volume with air flow rates in both indoor and outdoor heat exchangers.

2. PERFORMANCE ANALYSIS OF HEAT PUMP

Capacity controlled heat pump air conditioner system in corporating these components is schematically shown in Fig. 1. In this figure, in order to develop an optimization method for maintaining the optimum performance of heat pump system by controlling air flow rates and compression displacement volume, the performance of each component was required and also the optimization method proposed in the present investigation was discussed.

2-1. General Characteristics of Each Component

As illustrated in Fig. 1, the heat pump air conditioner model was mainly organized into six components :
(1) Compressor, (2) Evaporator, (3) Condenser, (4) Refrigerant, (5) Expansion Device and (6) Fans

R-22 was used as the working refrigerant. The assumptions for performance simulation was given as follows :
1) Refrigerant superheat of 8°C at compressor inlet was held at constant.
2) Refrigerant subcool of 5°C at expansion valve inlet was held at constant.
3) Heat losses to the surroundings were neglected.

Based on the above mentioned assumptions, capacity, input power and COP of each component is expressed as follows:

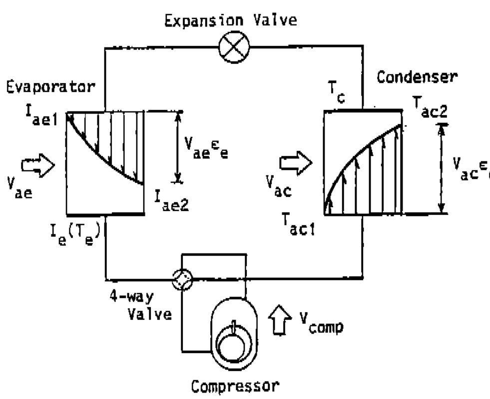

Fig. 1 - Schematic Diagram of Heat Pump Model

Compressor;

$$Qe.comp = Vcomp\ \eta v\ \gamma r\ \Delta Ie \quad (1)$$
$$Ncomp = Vcomp\ \eta v\ \gamma r\ \Delta Iad \quad (2)$$
$$Qc.comp = Qe.comp+Ncomp \quad (3)$$
$$COPe = Qe.comp/Ncomp \quad (4), \qquad COPc = Qc.comp/Ncomp \quad (5)$$

where Qe is a evaporation capacity , Ncomp is input power added to compressor.

Evaporator; $$Qe.ex = Vae\ \gamma ae\ \varepsilon e(Iae_1 - Ie) \quad (6)$$

where εe is a enthalpy efficiency $\varepsilon e = (Iae - Iae)/(Iae - Ie)$ (7)
and Ie is a saturated air enthalpy corresponding to Te.

Condenser; $$Qc.ex = Vac\ \gamma ac\ Cp(Tc - Tac_1) \quad (8)$$

where εc is a temperature efficiency $\varepsilon c = (Tac_2 - Tac_1)/(Tc - Tac_1)$ (9)
Te and Tc are balanced temperatures in a evaporator and a condenser, respectively.

2-2. Evaluation of System Performance Using Non-dimensional Equivalent Ratios of Se and Sc

In order to evaluate the system performance of the heat pump operating at different air conditions, the performance parameters of non-dimensional equivalent ratios, Se and Sc were proposed in the present investigation.

As to the Se and Sc, these derivation were performed based on the heat balance equations. Heat balance equations of heat pump cycle based on the assumptions mentioned earlier were expressed as:

$$Qe.comp = Qe.ex \quad (10), \qquad Qc.comp = Qc.ex \quad (11)$$

From the eqs. (10) and (11), the two non-dimensional groups Vae εe/Vcomp, Vac εc/Vcomp were derived as influence factors on Te, Tc. Namely, the Se and the Sc were expressed as

$$Se \equiv \frac{Vae\ \varepsilon e}{Vcomp} \quad (12), \qquad Sc \equiv \frac{Vac\ \varepsilon c}{Vcomp} \quad (13)$$

Hence, the Se and the Sc were defined as the non-dimensional equivalent ratios. Furthermore, substituting eq. (12) and eq. (13) into eq. (10) and eq. (11), Se and Sc is expressed as:

$$Se = \gamma r\ \eta v\ \Delta Ie/\gamma ae\ (Iae_1 - Ie) \quad (14)$$

$$Sc = \gamma r\ \eta v\ \Delta Iad/\gamma ac\ (Tc - Tac_1) \quad (15)$$

As seen in the eqs. (12) and (13), it can be understood that the evaluation of performance in heat pump air conditioner can be performed using Se and Sc, since those parameters represent explicitly the variation of air flow rates in heat exchangers and compression displacement volume.

Therefore, it can be said that the Se and the Sc derived are the available parameters as a means of the performance evaluation of heat pump air conditioner in case where outdoor ambient temperatures change. Accordingly, the improvement in COP can be performed by controlling the relation between the Se and the Sc.

Referring to Se and Sc, the following expressions are used in different modes,

Heating mode : Se = So , Sc = Si (16), Cooling mode : Se = Si , Sc = So (17)

2-3. Computational Procedure

The computational procedures for evaluating the performance of heat pump air conditioner is briefly described below.

The Se and Sc was calculated first using eqs. (14) and (15) for the given air conditions and the compressor efficiency. In order to examine the variation of Ie and Tc with variables such as air flow rates and compression displacement volume, using eqs. (14) and (15), the following eqs. were written approximately as follows:

$$Ie = (a_0 \cdots\cdots a_{20})\ (1\ x \cdots x^5\ y \cdots y^5\ xy \cdots xy^4) \quad (18)$$
$$Tc = (b_0 \cdots\cdots b_{20})\ (1\ x \cdots x^5\ y \cdots y^5\ xy \cdots xy^4) \quad (19)$$

where, x and y are $\log_{10}Se$ and $\log_{10}Sc$ respectively. In eqs.(18) and (19), the coefficients referred to "a" and "b" are determined with the Ie , the Tc, the Se, and Sc.

Accordingly, the COP and capacity in heat pump air conditioner were obtained from eqs. (18) and (19).

2-4. Optimization for Improvement in COP

To maximize the COP , available combination of various control parameters such as indoor air flow rate, outdoor air flow rate and compression displacement volume is discussed here in case of heating mode.

Heating capacity, Q and the COP in heat pump system are expressed as:

$$Qsys = q_c Vcomp \quad (20), \qquad COPsys = Qsys/(Ncomp+Nfi+Nfo) \quad (21)$$

where qc is a condensation capacity per a compression displacement volume, the Nfi and the Nfo are input powers of indoor fan and outdoor fan, respectively.

On the other hand, heating load Q_1, using a assumption to be proportional to a temperature difference between indoor ambient temperature and outdoor one, is determined and is given as :

$$Q_1 = q_1(Tai-Tao) \quad (22)$$

where, a constraint condition to optimize the COP can be expressed using eq. (20) and (22) as follows:

$$Qsys = Q_1 \quad (23)$$

3. RESULTS AND DISCUSSION

3-1. Validity of Performance Prediction

A comparison of calculated results with experimental data as to the capacity and the COP is shown in Table.I. In Table.I, compressor efficiencies used in predicting the performance were obtained from a capacity and input power measured with a compressor testing unit. Moreover, heat exchanger efficiencies εe, εc were given by a heat exchanger performance program. The measured values of pressure drops were used in Table.I. A experimental data of capacity were obtained from both measured refrigerant flow rate with a anuver flow meter and refrigerant enthalpy difference. Experimental data of COP were obtained from capacity and input power measured with a wattmeter.

As shown in Table.I, it is found that discrepancy between experimental data and predicted results are within about ±3% in capacity and COP. From such findings, it can be said that the present method is fairly effective to predict the performance.

TABLE. I Comparison between predicted results and experimental data

		HEATING MODE			COOLING MODE		
		No.1	No.2	No.3	No.4	No.5	No.6
Tail	°C	21	21	24	27	32	21
RHail	%	50	50	40	50	30	50
Vai	m³/h	16.8	10.6	16.8	15.1	15.1	15.1
Si		99.6	78.3	99.6	71	71	71
Taol	°C	7	7	21	35	43	21
RHaol	%	83	83	50	27	22	50
Vao	m³/h	40.5	40.5	40.5	43.5	43.5	43.5
So		238	238	238	276	276	276
Te °C	Experimental	-2.2	-1.5	7.1	2.6	6.5	-2.2
	Predicted	-2.4	-2.1	6.4	1.6	7.0	-3.9
Tc °C	Experimental	48.5	55.5	59.2	48.5	56.7	33.8
	Predicted	49.6	55.2	58.2	48.0	56.9	33.3
Q w	Experimental	8465	8335	10337	7162	7272	7101
	Predicted	8362	8209	10199	6944	7373	6670
COP	Experimental	3.55	3.03	3.37	2.95	2.50	4.02
	Predicted	3.45	3.02	3.40	2.90	2.52	3.82

3-2. Effect of S Assignment Ratio

The effect of the S assignment ratio on the heat pump performance is discussed. Fig.2 shows the results examined the effect of S assignment ratio, F , on the performance in heating mode, under the constraint condition that Stotal (=Si+So) is kept constant, where the F corresponds to Si/(Si+So), qht is a heating capacity per Vcomp and equals to Qc/Vcomp. COPht equals to COPc. An inspection of this figure reveals that the maximum values of qht and COPht appear at the given values of F. For instance, the heating capacity and the COP attain a maximum at F=0.3, and F=0.7, respectively.

Furthermore, it is remarkable that the Stotal is independent on the F corresponding to maximum values of the COP and the capacity.

Fig.3 shows the results in case of the cooling mode. In the cooling mode, this figure indicates that the maximum values of the qcl and the COPcl attain at F=0.5 and F=0.3, respectively.

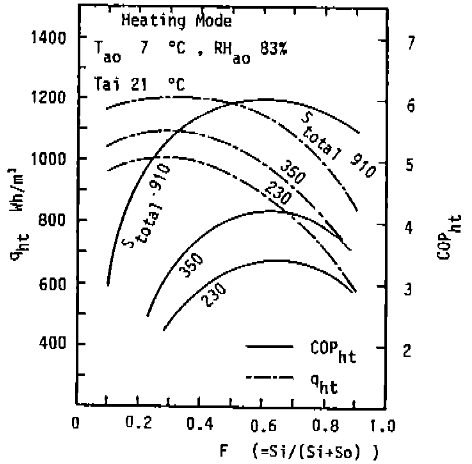

Fig. 2 - Effect of S assignment in heating mode

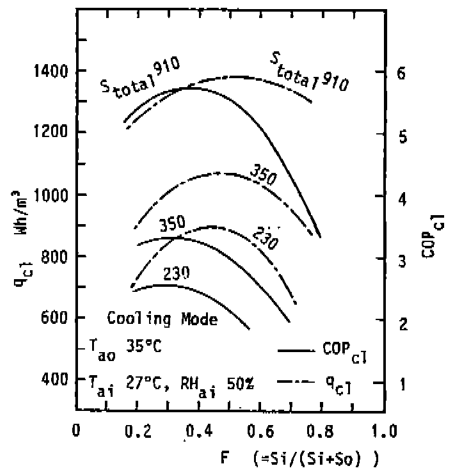

Fig. 3 - Effect of S assignment in cooling mode

Concerning the maximum q and COP, Senshu, et al./2/ examined the effect of assignment in surface areas of both indoor and outdoor heat exchangers. They showed that the assignment ratio of the indoor heat exchanger area in the total heat exchanger area corresponding to maximum COPht was 0.3. However, although their results obtained with only the heat exchanger surface area are available for designing air conditioner, these are inadequate for controlling the capacity in the heat pump.

Furthermore, it is showed that the F of the heat pump listed in Table.I is good agreement with the calculated results to maximize qht and COPcl in heating and cooling modes, respectively.

3-3. Effect of S on heat pump performance

Fig. 4 shows the variations of Q/Qbase and COP/COPbase with S/Sbase. In this figure, the subscript "base" represents the values for actual heat pump in Sec.3-1. The solid line and the dashed line in Fig. 4 correspond to improvement effects due to the Si and So, respectively.

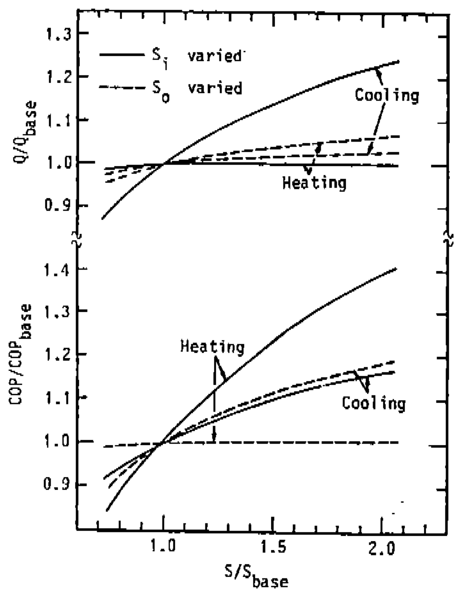

Fig. 4 - Comparison between Si and So in improvement effects

3-4. Optimization in Capacity Controlled Heat Pump

Examination of the control parameters such as Vai, Vao and Vcomp of the capacity controlled heat pump is made of in this section in order to maximize COP in heating mode. The results obtained are shown in Figs. 5 and 6. The specifications of a base model, and some assumptions used in calculations are shown in Table.II.

TABLE. II Specifications of base model and assumptions used in optimization

	Specification	Assumption
Outdoor Heat Exchanger	$\varepsilon o= 0.75$, $Nfo= 39w$, at $Vao=1.2*10^3$ m^3/h	$\varepsilon=1-EXP(-NTU)$ $NTU \propto Va^{-0.65}$ $Nfan \propto Va^{2.5}$
Indoor Heat Exchanger	$\varepsilon i= 0.9$, $Nfi= 64w$, at $Vai=0.45*10^3$ m^3/h	
Compressor	$\eta v= 0.9$, $\eta tad= 0.65$	$\eta v, \eta tad$ =const.

Fig.5 shows the variation of COPsys with the various combinations of Vai, Vao, and Vcomp at the outdoor ambient temperature of 1.5°C. Heating capacity obtained from a combination of Vai, Vao and Vcomp in Fig.5 is kept a constant value which equals to heating load given by eq. (22). The maximum COPsys obtained is 3.6 and also this means the improvement of 11 %, compared with the COP of base model mentioned in Table.II.

Fig.6 shows the variations of Vai, Vao, Vcomp and COPsys with outdoor ambient temperature. The solid line corresponds to the value optimized, and also the chained line corresponds to the value for the base model. This figure demonstrates that the effect of the optimization in COPsys is significant in the lower outdoor ambient temperature region. This is due to the reasons that with decreasing Tao, since the reduction of the S results from the increase of the Vcomp, the effect of the increase in air flow rate appears significantly.

From the above mentioned discussion, with regard to the optimization in the performance of the heat capacity controlled heat pump under the various air condition operations, the proposed optimization method is expected to be a superior technique, compared with the previous optimization which predict the performance by treating individually Vae, Vac and Vcomp.

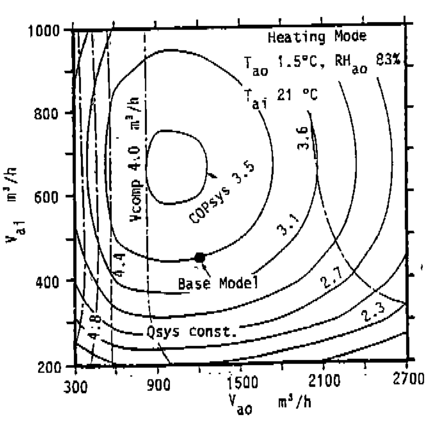

Fig. 5 - Sensitivity of COP to air flow rates at constant capacity

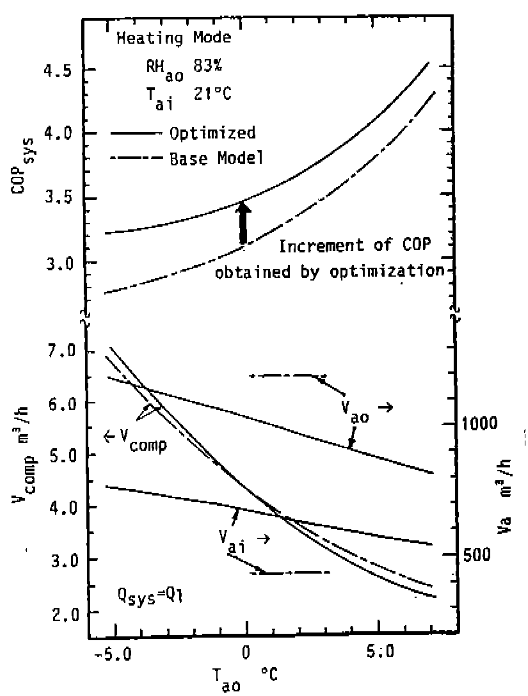

Fig. 6 - Optimization effects on COP at various outdoor temperatures

4. CONCLUSIONS

To predict the optimization of performance of heat pump air conditioner, the performance prediction method using the non-dimensional equivalent ratios Se and Sc is proposed and discussed.

Conclusions in the present investigation are as follows:

(1). Calculated data by the proposed method was good agreement with experimental results.

(2). The optimization of some control parameters, those are compression displacement volume and air flow rates through both indoor and outdoor heat exchangers, was examined in order to achieve a maximum COP of the capacity controlled heat pump in heating mode. Then, it is showed that the optimization of these values cause 7-16% increment of COP in heating mode compared with a base model.

5. REFERENCES

1. C.K.RICE et al. : Design optimization of conventional heat pumps, ASHRAE Trans. Vol.87 (1981), Part 1, pp.1037-1055.
2. T.SENSHU et al. : Review of refrigeration cycle based on evaporators and expansion devices, Refrigeration, Vol.56 (1981), No.649, pp40-48.

6. NOMENCLATURE

S	non-dimensional equivalent ratio (=Va ε/Vcomp)	(-)
F	assignment ratio of S (=Si/(Si+So))	(-)
Q	capacity	(W)
N	electric input power	(W)
q	(=Q/Vcomp)	(Wh/m^3)
n	(=N/Vcomp)	(Wh/m^3)
COP	coefficient of performance	(-)
Vcomp	compression displacement volume	(m^3/h)
ηv	volumetric efficiency of compressor	(-)
ηtad	total efficiency of compressor	(-)
Va	air flow rate through heat exchanger	(m^3/h)
ε	efficiency of heat exchanger	
Ql	heating load	(W)
ql	(=Ql/(Tai-Tao))	(W/°C)
T	temperature	(°C)
I	enthalpy	(Wh/kg)
RH	relative humidity of air	(%)
γa	specific weight of air	(kg/m^3)
Cp	specific heat of air	(kcal/kg°C)
γr	specific weight of refrigerant at compressor suction inlet	(kg/m^3)

Subscripts.

e	evaporator
c	condenser
r	refrigerant
a	air
ad	adiabatic
i	indoor
o	outdoor
sys	heat pump system
comp	compressor
ex	heat exchanger
ht	heating
cl	cooling
l	heating load

RESUME

Au Japon, des pompes à chaleur à controle volumétrique équipées de compresseurs et de ventilateurs à vitesse variable ont été construits pour satisfaire les necessites du confort domestique.

Afin d'économiser un maximum d'énergie tout en maintenant le confort au même niveau, il a été nécessaire de chercher le meilleur rendement possible des pompes à chaleur. Cet article presente une démonstration théorique visant à maîtriser divers paramètres, tels que le débit du réfrigérant et celui de l'air à la fois dans l' échangeur de chaleur extérieur et l' échangeur intérieur. Les resultats de ces calculs ont ensuite été comparés oux données expérimentales.

L'optimisation des deux paramètres de contrôle suivants, le volume d'air déplacé par le compresseur et le débit d'air à travers les deux échangeurs de chaleur a donné comme résultats une augmentation du C.O.P. de 7 à 16% pour la pompe à chaleur fonctionnant en mode chauffage. Les résultats théoriques et expèrimentaux arrivent aux mêmes conclusions.

SUMMARY

Capacity controlled heat pumps equipped with variable speed rotating compressors and fans have been developed to satisfy comfort requirements for consumer use in Japan.

In order to save more energy and at the same time maintain comfort, there is a need to optimize the heat pump operation. This paper discusses a theoretical model to optimization control of the parameter such as refrigerant flow rate, air flow rate in both indoor and outdoor heat exchangers. The output of the model is compared to experimental results.

The optimization of the following control parameters, compressor displacement volume, and air flow rate through both heat exchangers gave 7-16% increase in the COP for the heat pump operating in the heating mode. Theoretical and experimental results were in good agreement.