1996

Comparison of Compressor Efficiency Between Rotary and Scroll Type with Alternative Refrigerants for R22

T. Kato
Mitsubishi Electric Corporation

Y. Shirafuji
Mitsubishi Electric Corporation

S. Kawaguchi
Mitsubishi Electric Corporation

COMPARISON OF COMPRESSOR EFFICIENCY BETWEEN ROTARY AND SCROLL TYPE WITH ALTERNATIVE REFRIGERANTS FOR R22.

Taro KATO Yoshinori SHIRAFUJI Susumu KAWAGUCHI

Shizuoka Works, Mitsubishi Electric Corporation
3-18-1, Oshika, Shizuoka City 422, JAPAN

ABSTRACT

This paper compares the compressor efficiency between rotary and scroll type, premising that each type has the highest efficiency mechanism that can be designed at present, in the operation with R22 and its alternative refrigerants, R407C and R410A. The efficiency of compressors optimized for each refrigerants is theoretically figured out, and it is also experimentally confirmed. As the results, the upper limit of cooling capacity range, in which efficiency of rotary type is higher than that of scroll type, is about 8,000 Btu/h with R22 and R407C. In the case with R410A, rotary type is superior to scroll type in the range approximately up to 24,000 Btu/h. This study shows that the range in which rotary type operates more efficiently than scroll type will expand up to small capacity range of unitary air conditioners in the case with R410A.

1. INTRODUCTION

Refrigerants of binary or ternary mixtures of HFC32, HFC125, and HFC134a are considered to be more prosecutable for R22 alternatives than another composites of HFCs. R407C (HFC32 / 125 / 134a = 23 / 25 / 52 wt%) has an advantage that it needs no large change of the compressor dimension for R22 use. R410A (HFC32 / 125 = 50 / 50 wt%), which needs modification to the compressor dimension because it perform the same cooling capacity as R22 with smaller stroke volume, also has an advantage that the expected system COP is higher than R407C in spite of its low theoretical COP. Because R410A performs nearly same as azeotropic, and pressure drop inside the piping is smaller than R407C because velocity of gas flow is smaller than R407C. Table.1 compares the theoretical conditions between R22, R407C and R410A. Fig.1 shows the temperature definition for non-azeotropic refrigerants in this paper.

On the other hand, rolling-piston-rotary (following "rotary" for short) compressor and scroll compressor are broadly used for air conditioners. Rotary type, which has a simple compression mechanism and needs a low cost to produce, is adopted mainly on the air-conditioners with smaller capacity than medium range of unitary use, up to 69,000 Btu/h class. Scroll type, which operates with small vibration and low overshooting loss, is used mainly for unitary air-conditioners, and currently used for room air-conditioners of 9,000 Btu/h class. It is important to compare the performance of different mechanism compressors to supply most appropriate type according to cooling capacity range after R22 is replaced with alternatives. By considering current compressor losses and its alternation by changing refrigerant, we studied technique about dimensional optimization of compressor, and we estimated efficiency of optimized compressor in the case that R22 is replaced with R407C or R410A.

Table.1 Theoretical Condition
CT/ET = 54.4/7.2(C), SC/SH = 8.3/27.8(deg)
NIST Refprop Ver4.0

	R22	R407C	R410A
Discharge pressure Pd (kPa.ABS)	2152	2318	3351
Suction pressure Ps (kPa.ABS)	625	632	996
Suction gas temp. Ts(°C)	35.0	37.3	35.0
Exp entering temp. Texp(°C)	46.1	43.8	46.0
Discharging temp. Td(°C)	103.7	96.3	101.6
Theoretical Cooling Capacity*	1.00	1.03	1.43
Theoretical cooling COP*	1.00	1.00	0.93

* :Relative to R22 base

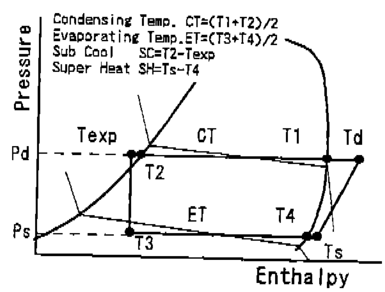


Fig.1 Temperature definition

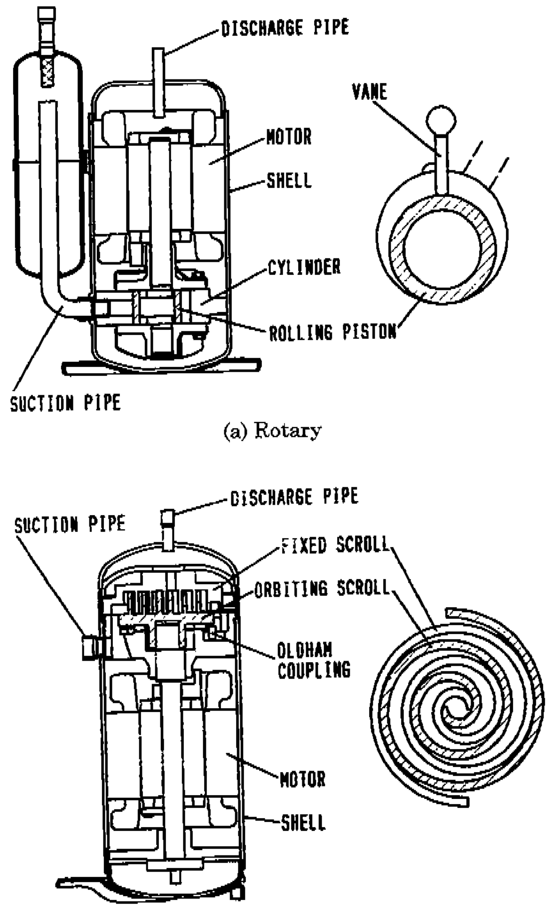
2. COMPRESSOR MECHANISMS

Fig.2 shows the structure of the rotary and scroll compressor which this paper refers to. Rotary compressor is high side pressure shell type. Scroll compressor is low side pressure shell type, fixed scroll and orbiting scroll are forced to contact with each other by the radial and axial compliance mechanisms. Each of compressors is driven by variable frequency 2-pole induction motor. Stroke volume, Vst , of each compressor is given by following equation,

Scroll $$Vst = (2N-1)\pi p(p-2t)H \quad (1)$$

Rotary $$Vst = \frac{\pi}{4}\left(D^2 - d^2\right)h \quad (2)$$

where t is blade thickness, p is pitch of blades, $(N+1/4)$ is number of scroll turns, H is blade height, D is cylinder inner diameter, d is piston outer diameter, h is cylinder height.



(a) Rotary

(b) Scroll

Fig.2 Cross Sectional View

3. PREVIOUS LOSS ANALYSIS

3.1 Definition Of Efficiencies

Table. 2 shows the relation of these losses. Each of efficiency is defined in this paper as follows.

Motor efficiency $$\eta_m = L_m / L_c \quad (3)$$

Mechanical efficiency $$\eta_{me} = L_i / L_m \quad (4)$$

Indicated efficiency $$\eta_c = L_{ad} / L_i \quad (5)$$

Compressor efficiency $$\eta_{comp} = \eta_m \cdot \eta_{me} \cdot \eta_c \quad (6)$$

Where L_m is motor output, L_i is indicated work, L_c is consumption power, L_{ad} is adiabatic work defined as follows.

Adiabatic work $$L_{ad} = \eta_v Gr_{th} \Delta h_{comp} \quad (7)$$

Volumetric efficiency $$\eta_v = Gr/Gr_{th} \quad (8)$$

Where Gr_{th} is theoretical refrigerant flow, Δh_{comp} is theoretical increment of enthalpy, Gr is real refrigerant flow.

Table.2 Classification of Loss Factor

- Consumption power
  - Motor loss
  - Motor output
    - Mechanical loss**
    - Indicated work
      - Indicated loss*
      - Adiabatic work

Indicated loss*

- Rotary
  - Heat loss
  - Over/Undershooting loss
  - Leak loss
  - Reexpansion loss
- Scroll
  - Heat loss
  - Over/Undershooting loss
  - Leak loss
  - Built-in ratio loss

Mechanical loss**

- Rotary
  - Journal loss
  - Thrust loss
  - Vane tip
  - Vane side
- Scroll
  - Journal loss
  - Thrust loss
  - Blade top
  - Blade side
  - Oldum coupling

Although mechanical efficiency and indicated efficiency are depend on the type of mechanism and refrigerant property, motor efficiency is essentially independent of them, therefore this paper refers only to mechanical efficiency and indicated efficiency in the following sections. Table.2 also shows the difference of indicated loss and mechanical loss contents between rotary type and scroll type.

3.2 Loss Analysis

We analyzed the efficiency of compressor with current refrigerant before predicting the compressor efficiency with alternate refrigerant. Loss analysis in this paper is performed according to following process[1].

(1) Experimentally determine the adiabatic work and consumption power.

(2) Obtain a $P-V$ diagram by measuring pressure inside of cylinder to determine indicated work.

(3) Determine overshooting and undershooting loss on the $P-V$ diagram. In the case of scroll compressor, compression loss caused by the fixed built-in volume ratio is also measured.

(4) Calculate leak loss numerically by assuming that the leak flow is one-dimensional flow of a compressible fluid as following equation[2].

$$w = \varphi \cdot A \cdot \sqrt{\frac{2\kappa}{\kappa - 1} \cdot p_2 \cdot \rho_2 \cdot \left(F^{\frac{2}{\kappa}} - F^{\frac{\kappa+1}{\kappa}} \right)} \tag{9}$$

$$\text{when } \; p_1 / p_2 \geq Fc \qquad F = p_1 / p_2$$

$$p_1 / p_2 \leq Fc \qquad F = Fc = \left(2/(\kappa + 1) \right)^{\frac{\kappa}{\kappa - 1}}$$

Where w is the mass flow ratio of leakage, φ is coefficient of flow, A is the cross sectional area of clearance which is geometrically determined at each crank angle, κ is adiabatic exponent, p_1 is the lower pressure, p_2 is the higher pressure, ρ_2 is density, Fc is the critical compression ratio.

(5) Theoretically estimate reexpansion loss at rotary considering back flow ratio[3].

(6) A remainder of adiabatic work is heat loss. It is confirmed by considering η_v and experimentally measured suction port gas temperature.

(7) Experimentally determine the motor loss. Then total amount of mechanical loss is figured out.

(8) Further analysis to determine the contents ratio of each mechanical losses is performed. In the case of rotary compressor, frictional loss at vane tip and vane side are obtained by numerical solution of angular speed of rolling piston[4]. Journal losses are obtained by solving basic equation for journal bearing of finite length under fluctuating load. Thrust loss is calculated considering weight of crankshaft and rotor, axial component of motor torque as load.

(9) In the case of scroll, frictional losses at blade top, blade side, oldham coupling and thrust bearing are determined by considering the coefficient of friction experimentally obtained in advance as a function of load and sliding velocity. Journal bearing loss is calculated by solving the equation of journal bearing of finite length under static load.

4. SIMULATION

We estimated the increment and decrement of each losses when refrigerant is changed from R22 to alternatives, by considering the refrigerant property differences and compressor dimension changes, according to the results of loss analysis at R22. The following shows the model and postulates for this simulation.

4.1 Heat Loss

Heat loss is defined as the decrement of cooling capacity caused by the increment of suction gas specific volume, which is given as a product of heat flow, specific heat at constant pressure of suction gas Cp_s and a ratio of change of specific volume product for temperature $\partial v_s / \partial T_s$. Heat flow is assumed to be proportional to the product of temperature difference and suction chamber surface area approximation, heat loss L_{pre} is given as follows,

$$\text{Heat loss} \quad L_{pre} \propto \frac{1}{v_s} \cdot \frac{\partial v_s}{\partial T} \cdot Cp_s \cdot (Vst)^{2/3} \cdot \left(T_d^{th} - T_s \right) \tag{10}$$

Where T_d^{th} is theoretical discharge gas temperature.

4.2 Reexpansion Loss

This simulation assumes that reexpansion is proportional to the weight ratio of reexpansion gas for suction gas.

4.3 Overshooting Loss And Undershooting Loss

Overshooting loss L_{dis} may be described as $L_{dis} \propto \Delta P \cdot \Delta V \cdot f$, where ΔP is the average overshooting pressure, ΔV is the volumetric difference of compressing chamber during discharging process, f is frequency of crank shaft rotation. ΔP is calculated by assuming incompressible fluid during discharging process, considering gas density and gas velocity which is determined by port area, time of discharge or suction process and ΔV. Suction undershooting loss is also calculated by the same way considering pipe length and inner diameter. Total of overshooting loss and undershooting loss is described as "over/undershooting" for short in this paper.

4.4 Compression Loss Caused By The Fixed Built-in Volume Ratio

Scroll compressor produces loss when compressing ratio is not equal to the built-in compression ratio. This paper assumes that the loss of this type keeps constant ratio to adiabatic work regardless of refrigerant type.

4.5 Leak Loss

Mass flow rate of each of leakage is regarded to be proportional to w which is given by equation (9) at $p_1 = P_s$, $p_2 = P_d$ with adequate φ and A calculated by considering the difference of compressor dimension.

4.6 Mechanical Loss

This paper regards each of mechanical losses is able to be described as following simple equation $L = \mu \cdot F \cdot V$ where L is mechanical loss at the considered point, μ is coefficient of friction, F is reaction force and V is sliding velocity. Generally, μ is regarded as constant when load capacity of each bearings are adequately designed. However, μ at thrust bearing of scroll compressor must be considered as a function of F and V which is experimentally obtained in advance, because it is far from complete fluid lubrication especially in low frequency operation of small capacity class. F is proportion to product of pressure difference and area under the pressure which depends on the major dimension of the compressor. V also depends on the compressor dimension and operating frequency.

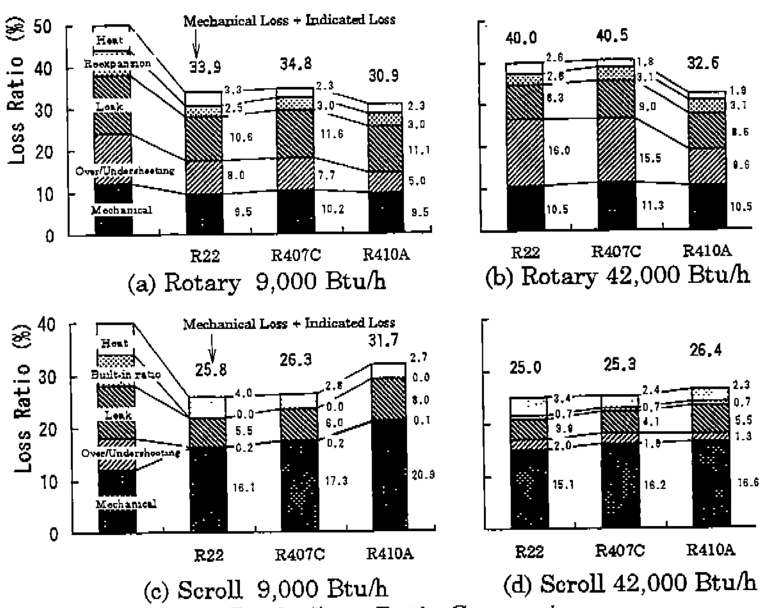


Fig.3 Loss Ratio Comparison

Loss Ratio is relative to Adiabatic work as 100%.
CT/ET = 54.4/7.2(C), SC/SH = 8.3/27.8(deg), 60Hz.

5. SIMULATION RESULTS

Fig.3 compares the ratio of mechanical losses and indicated losses between rotary type and scroll type with R22, R407C and R410A in driving with rating condition. The ratio of R22 is obtained by analysis explained in 3.2, and the others are obtained by the simulation explained in 4. Each compressor has optimized dimension to minimize losses. Each of rotary compressor also has a optimized suction pipe dimension to obtain the maximum effect of super charging[5].

5.1 R407C Loss Alternation

Total loss ratio of mechanical loss and indicated loss at R407C is nearly equal to R22. In detail, mechanical loss, leak loss and reexpansion loss of R407C are expected to increase in comparison with R22 mainly because of the increment of pressure difference between discharge gas and suction gas. Heat loss of R407C decrease, because R407C's theoretical difference of temperature between suction gas and discharge gas is smaller than R22's. Loss ratio increment and decrement are almost canceled out. Although loss ratio composition of rotary is different from that of scroll, there is no remarkable difference of the alternation of total loss ratio from R22 to R407C between rotary and scroll, because alternation of each loss ratio is not large enough.

5.2 R410A Loss Alternation

On the other hand, loss ratio alternation from R22 to R410A is estimated to be quite different between rotary and scroll. In the case of R410A, mechanical loss, leak loss and reexpansion loss tend to increase because of increment of pressure difference and stroke volume decrement. Heat loss tends to decrease mainly because Cp_s is larger than R22, and $\partial v_s/\partial T_s$ is smaller than R22 at equation (10). Over/undershooting loss decreases because effect of velocity decrement caused by stroke volume decrement is more dominant than density increment.

By comparing Fig 3(a) and (b), it is recognized that the larger over/undershooting loss ratio at R22, the larger decrement of the loss at R410A. Over/undershooting loss ratio of rotary compressor is larger than that of scroll, therefore indicated loss of rotary compressor is expected to decrease larger extent than that of scroll compressor.

5.3 Dimensional Optimization At R410A

Furthermore, increment of leak loss and mechanical loss ratio from R22 to R410A is larger extent in scroll compressor than in rotary compressor. This tendency originates in a difference of effect of dimensional optimization between scroll type and rotary type, as shown in fig 4. In the case of rotary, by decreasing h to obtain adequate Vst for R410A, journal load decreases in proportion to $d \times H$ decrement. Rotary type has no thrust load caused by the gas pressure, therefore the smaller journal loss, the smaller total mechanical loss, unless vane side loss turns to dominant. In the case of scroll, H decrement causes not only journal loss decrement but also increment of thrust loss, which is the most dominant loss for scroll compressor especially under 18,000 Btu/h class. Since too large H/p is not allowed to avoid

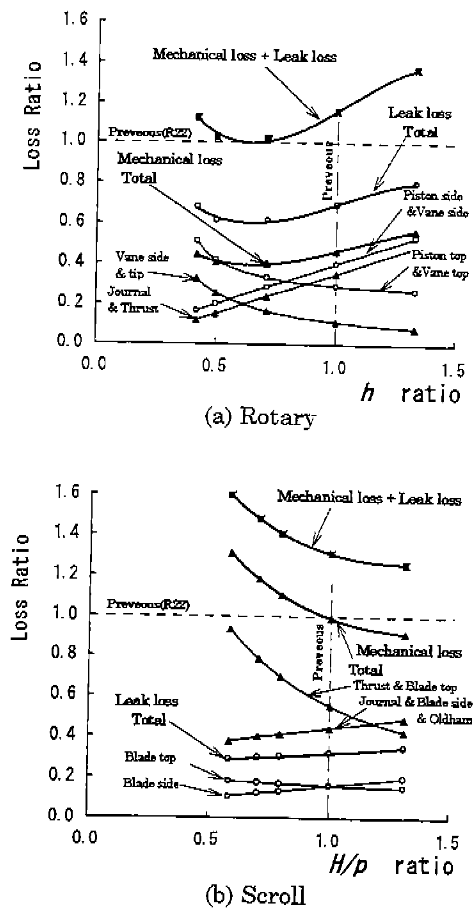


Fig.4 Dimensional Optimization

Loss Ratio is relative to Previous R22's "Mechanical loss + Leak loss."
CT/ET = 54.4/7.2(C), SC/SH = 8.3/27.8(deg), 9,000 Btu/h class, 60Hz.

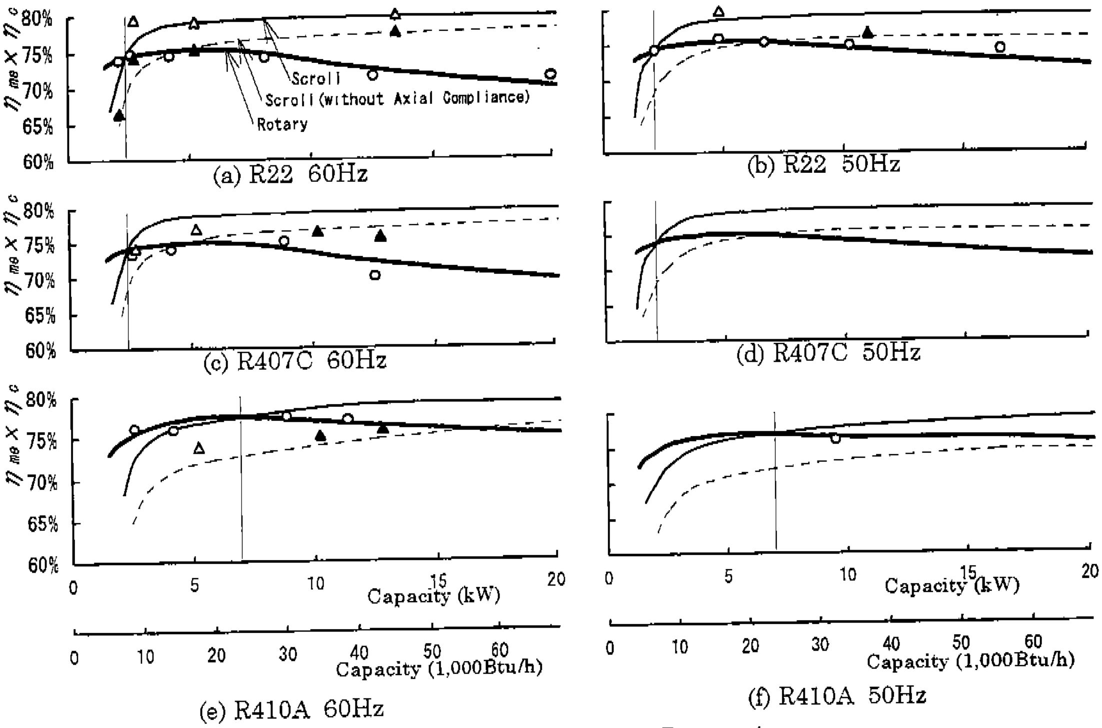


Fig.5 Predicted Efficiency Comparison

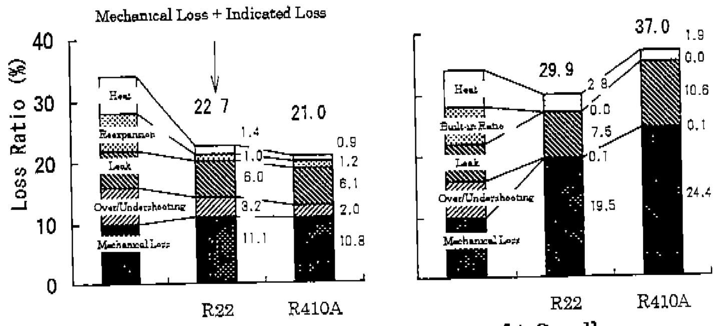
Lines of (a)(b) are approximation of measured value, else are prediction.
Plotted points are all measured results. CT/ET = 54.4/7.2(C), SC/SH = 8.3/27.8(deg)

orbiting scroll overturns, total amount of leak loss and mechanical loss of scroll compressor necessarily larger at R410A than at R22 as shown in Fig.4(b).

6. TEST RESULT

Fig 5 shows the product of mechanical efficiency and indicated efficiency as a function of compressor's cooling capacity. Tested results which have been provided through examination until the present are also plotted on the graph. The predicted curves at R407C and R410A agree with actual tested value qualitatively. It is recognized that efficiency of rotary compressor is actually improved at R410A especially in large nominal output range, as above prediction. Authors predict that rotary type will be potentially superior to scroll type in the range approximately up to 24,000 Btu/h when refrigerant is replaced with R410A, in contrast that authors also predict that the efficiency superiority of scroll type will not change in the case of R407C.

This tendency is predicted to appear more clearly in the case that compressor operates in variable frequency on air-conditioning system. Fig.6 shows the loss ratio of compressors for 9,000 Btu/h class air-conditioning system under the condition which is close to actual use of air-conditioners at medium to low output condition of heating, such as outdoor temperature is higher than 5°C. Total driving time with medium to low condition of heating is nearly 50 % of total air-conditioning time through the year in temperate region in Japan[6], therefore E.E.R. at this condition is dominant to decide seasonal E.E.R. As shown in Fig.6, rotary compressor is superior to scroll compressor in the efficiency at this condition with R22 currently. At rotary compressor, over/undershooting loss is smaller in this condition than in rating condition because of low operating frequency. Leak loss also decreases because of low pressure difference. On the other hand, mechanical loss of scroll compressor increases because of partial boundary lubrication at thrust bearing caused by the low sliding velocity, and leak loss also increases because radial force to keep blades contact is reduced by low pressure difference. Predominance of rotary compressor at this condition is estimated to spread in the case with R410A, mainly because of thrust bearing loss increment at scroll compressor.



(a) Rotary (b) Scroll

Fig.6 Loss Ratio Comparison

Medium-low Output Condition on variable frequency air-conditioner of 9,000 Btu/h class, heating.

CT/ET = 31.2/5.4(C), SC/SH = 5.0/3.1(deg), 30Hz

7. CONCLUSION

We showed an outline about technique to predict the efficiency of compressor with alternative refrigerant. The objective of this study brought the following result.

(1) This paper shows the possibility of prediction about the compressor efficiency with alternative refrigerant by considering the change of refrigerant property and a dimension of compressor, based on the loss composition of existing models.

(2) Scroll type is estimated to be more effective than rotary type over 8,000 Btu/h with R407C same as R22 potentially.

(3) In the case with R410A, rotary type is superior to scroll type in the range approximately up to 24,000 Btu/h which is small capacity range of unitary air- conditioners in the case with R410A.

Based upon this study, and considering other factors such as compactness, silence, etc., we intend to advance further toward perfection of compressor for R22 alternatives, and contribute to global environment protection.

8. REFERENCE

[1] Sakaino, K., et al., "Some Approaches towards a High Efficient Rotary Compressor," Proc. Purdue Compressor Tech. Conf., 1984, pp.315-322.

[2] Inaba, T., et al., "A Scroll Compressor with Sealing Means and Low Pressure Side Shell," Proc. Purdue Compressor Tech. Conf., 1988, pp.887-900.

[3] Yanagisawa, T., Shimizu, T., Proc. JSME No.814-9, 1982-11, pp.31-38.

[4] Yanagisawa, T., et al., "Motion Analysis of Rolling Piston in Rotary Compressor," Proc. Purdue Compressor Tech. Conf., 1982, pp.185-192.

[5] Kakuda, M., et al., "Investigation of Pressure Pulsation in Suction Pipe on Rotary Compressor," Proc. Purdue Compressor Tech. Conf., 1988, pp.591-598.

[6] JIS-C9612