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DEVELOPMENT OF HIGH EFFICIENCY ROTARY COMPRESSOR FOR DOMESTIC REFRIGERATOR USING HFC-134a

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ABSTRACT

In response to the trend for phasing out CFCs, we have developed a high efficiency HFC-134a rotary compressor, whose total adiabatic efficiency is 9% higher than that of the present CFC-12 one. This paper describes the causes for the usual poor performance of rotary compressors using HFC-134a and explains how we improved the performance.

INTRODUCTION

CFCs will be phased out in the mid 1990s. Since CFC-12 is the usual refrigerant for domestic refrigerators, alternative refrigerants urgently need to be developed. The most promising alternative seems to be HFC-134a. However, it has been reported that when HFC-134a is substituted for CFC-12, the performance of refrigerators and compressors decreases [1,2]. Our initial test of rotary compressors using HFC-134a also showed a large drop in performance. Therefore, we investigated the causes of the decrease in performance and developed ways to improve performance. As a result, we have developed a high efficiency rotary compressor using HFC-134a.

HFC-134a REFRIGERANT PROPERTIES

The refrigerant properties of CFC-12 and HFC-134a are compared in Table 1 for the same temperature [3]. Since HFC-134a has a lower suction pressure and a higher discharge pressure than CFC-12, the pressure ratio of HFC-134a is 24% higher than that of CFC-12. The theoretical coefficient of performance (COP) on the Mollier chart of HFC-134a is similar to that of CFC-12 and this indicates no drop in efficiency using HFC-134a. On the other hand, the refrigerating capacity of HFC-134a is 8% smaller than that of CFC-12 and compressors using HFC-134a need a larger displacement to maintain the same capacity.

INITIAL TEST

Since HFC-134a is not compatible with the mineral oil or alkylbenzen oil used with CFC-12, it needs a new oil such as polyalkylene glycol (PAG) oil or ester oil. For the initial test, we used HFC-134a and PAG oil for a compressor calorimeter test using the rotary and reciprocating compressors designed for CFC-12. For CFC-12, high viscosity grade (VG) oil is usually used for rotary compressors, whose chamber is filled with high pressure gas, and low VG oil is used for reciprocating compressors, whose chamber is filled with low pressure gas. In this test, the same selection was applied.

Three rotary compressors and two reciprocating compressors with different displacements were tested. The results (see Fig. 1) show that the COP of the reciprocating compressors using HFC-134a is almost the same as for CFC-12, but is 10 - 15% lower for rotary compressors.

Therefore, we investigated the causes of the decrease in performance of rotary compressors using HFC-134a [4].

EFFECT OF REFRIGERATION OIL ON PERFORMANCE

Different oil types

The performances of rotary compressors using different types of oil are shown in Fig. 2. PAG, ester 1, and ester 2 oil with a high VG were compared for a compressor with 5.8 cm^3/rev displacement. To examine the effect of different viscosity grade on performance, ester 1 oil with a low VG was also tested. With PAG oil, volumetric efficiency η_v and total adiabatic efficiency η_{tad} decreased by 11% and 15% respectively. The compressor using ester 1 oil with a high VG had a 3% higher η_v and a 5% higher η_{tad} than a compressor using PAG oil. The ester 1 with a low VG showed a similar η_v to the high VG ester 1, but a 12% higher η_{tad} than the PAG oil. The COP for low VG ester 1 oil also showed a 12% increase. The ester 2 showed a 1% higher η_v and a 3% higher η_{tad} than the PAG.

The cause of the difference in performance of compressors using different types of oil may be the difference in lubricity of each oil, but the difference in oil viscosity is more likely to have great effect on performance from the results for ester 1 oil with a high VG and a low VG.

Different oil viscosity

The viscosity of the HFC-134a / oil mixture is the key factor that influences the compressor performance. The viscosity of the mixture was measured inside the compressor chamber during operation by attaching a viscosity sensor to the compressor (see Fig. 3). In this sensor, the viscosity is determined from the velocity of an object which moves in a fluid. Ester 2 oils having four different VG were used. As shown in Fig. 4, there is an optimum viscosity that maximizes COP. Refrigerating capacity has a maximum. On the other hand, energy consumption decreases, as the viscosity decreases in the measured range. The optimum viscosity for COP is rather smaller than the viscosity of the present CFC-12 / alkylbenzen oil mixture. There are two reasons for the existence of the optimum viscosity for COP. First, with regard to mechanical loss, there is an optimum viscosity that minimizes the friction loss at bearings. Second, the compression chamber of a rotary compressor is sealed with leak-in oil through clearance and the quantity of leak-in oil depends on the viscosity of refrigerant / oil mixture, so there is an optimum quantity of leak-in oil that is adequate for sealing.

Fig. 5 shows COP versus viscosity of HFC-134a / oil mixtures, including the results of other types of oil described before. The results for all types of oil are almost on a line and the effect of different types of oil on performance is small.

This optimization of oil viscosity reduced the drop in COP of the compressor using HFC-134a from -15% to -3%.

EFFECT OF RE-EXPANSION LOSS ON PERFORMANCE

Another factor causing the drop in COP is the greater re-expansion loss of gas in clearance volume V_{cl}. Replacing CFC-12 with HFC-134a increases the compression ratio Π $(=P_d/P_s)$ by 24% and reduces the adiabatic exponent κ from 1.14 to 1.12. This results in an increase in the re-expansion loss ΔL_{cl}.

$$\Delta L_{cl} = P_s \cdot V_{cl} \left[\Pi^{\frac{1}{\kappa}} \frac{\kappa}{\kappa - 1} \left(\Pi^{\frac{\kappa-1}{\kappa}} - 1 \right) - \Pi + 1 \right] \quad \cdots\cdots\cdots\cdots \quad (1)$$

In the theoretical pressure-volume diagram in Fig. 6, the shaded part represents the re-expansion loss. The re-expansion loss for HFC-134a is 34% higher than for CFC-12 at the same clearance volume ratio. Thus the decrease in η_{tad} due to the difference in re-expansion loss is estimated to be about 2% for the measured compressor with the clearance volume ratio of 1.5%.

To avoid a drop in COP or η_{tad}, we must reduce the clearance volume such as discharge port volume. This will also increase η_v because of the increase in theoretical volumetric efficiency.

HIGHER EFFICIENCY COMPRESSOR

We investigated ways to reduce mechanical and heating losses.

Reduction of mechanical loss

We reduced the diameter of the crankshaft to reduce the friction loss at the bearings. The bending induced by this was reduced by adding a supplementary bearing to the end of the motor rotor. Calculations show that the load on the supplementary bearing is much smaller than on the other two bearings, so the increase in friction loss due to the supplementary bearing is negligible.

The roughness of a vane slot was halved to reduce the friction loss at the sides of a vane.

These two reductions of mechanical loss led to an estimated 3% increase in η_{tad}.

Reduction of heating loss

Since the flow rate of refrigerant in a domestic refrigerator is much smaller than that in an air-conditioner, the refrigerant gas in a refrigerator compressor is affected more by heating. First, the suction pipe was insulated. After the gas in the suction pipe enters the chamber of a compressor, the low temperature gas is heated through the suction pipe wall by the high temperature discharge gas inside the chamber. Thus the specific volume of the suction gas becomes large and η_v decreases. Consequently, η_{tad} also decreases. Therefore, to reduce the heating loss of suction gas, we used a double suction pipe with an insulating layer between the two pipes, instead of a single pipe. Experiments showed that the double suction pipe increases η_v by 2% and η_{tad} by 0.7%. The increase of η_{tad} is rather small, compared with that of η_v.

Fig. 7 shows the change in gas temperature inside the compressor. For the double pipe, the gas temperature at the cylinder inlet decreases by 12°C, compared to that for the single pipe, but the gas temperature at the cylinder outlet only decreases by 2°C. The gas temperatures inside the chamber and at the discharge pipe are almost the same. This result suggests that the reason the double suction pipe only produces a small increase in η_{tad}, compared to the large increase in η_v, may be as follows. With the double pipe, the difference in temperature between the suction gas and the cylinder wall is larger and the suction gas is more subject to heating. This increases indicated work and reduces indicated efficiency. Therefore, to obtain a large increase in η_{tad}, we should lower the temperature of the cylinder itself and reduce the quantity of heat transferred to the suction gas.

As Fig. 8 shows, in the present compressor, the high temperature gas discharged from a cylinder goes through the silencer next to the cylinder and is led to the motor side. As a result, the cylinder is heated. During suction, the low temperature suction gas is heated and expanded, thus η_v decreases. At the early stage of the compression process, as previously described, the gas receives heat from the cylinder wall and indicated work becomes large and indicated efficiency decreases. Therefore, as shown in Fig. 8, we chose a direct discharge passage which decreases the quantity of heat transferred to the cylinder. The high temperature discharge gas is led directly to the outside of the chamber. Then it is cooled by passing it through the radiating pipe and is returned to the inside of the chamber. With the change in length of the radiating pipes, we examined the performance versus the quantity of radiating heat (see Fig. 9). The quantity of radiating heat is represented as the percentage of the consumption energy. As shown in this figure, as the quantity of radiating heat increases, the temperature of the cylinder wall and motor wire decreases and η_v increases significantly. η_{tad} also increases significantly but it has a saturation point. Since in the study of refrigeration oil, HFC-134a / oil mixture viscosity increased as the temperature decreased, η_{tad} is likely to be influenced by the increasing viscosity of the mixture. This method increases η_v by 4% and η_{tad} by 5% for a temperature decrease of 15°C.

PERFORMANCE OF PROTOTYPE COMPRESSOR USING HFC-134a

Structure

We constructed a prototype compressor incorporating all of the improvements described above; namely, optimized oil viscosity, reduced clearance volume at discharge port, reduced crankshaft diameter, reduced vane slot roughness, insulated suction pipe, and added direct discharge passage. The structure of this prototype is shown in Fig. 10.

Performance

The performance of the prototype compressor is shown in Fig. 11, compared with the present compressor using CFC-12 and the initial test using HFC-134a. η_{tad} of the prototype compressor is 9% higher than that of the present compressor using CFC-12. This is a 24% increase over the initial test and is close to the sum of performance increases expected from each improvement. Although it is difficult to distinguish individual contributions, they are probably; 43% by optimized oil viscosity, 15% by reduced clearance volume, that is, reduced re-expansion loss, 27% by reduced heating loss, and 15% by reduced mechanical loss.

The η_v of the prototype compressor is 5% higher than that of the present one. This results from the optimized oil viscosity, reduced clearance volume, and reduced heating loss.

CONCLUSION

When HFC-134a is substituted for CFC-12 in a rotary compressor, the performance decreases. The main causes of this performance decrease are the higher viscosity of the HFC-134a / oil mixture, and the greater re-expansion loss of gas in clearance volume.

The performance can be improved by optimizing the oil viscosity by directly measuring the viscosity of the HFC-134a / oil mixture inside the compressor chamber with a viscosity sensor, and by reducing heating loss during compression with the direct discharge passage which decreases the temperature of the cylinder.

Using these methods, we have developed a high efficiency HFC-134a rotary compressor, whose η_{tad} is 9% higher than that of the present CFC-12 one.

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Table 1 Refrigerant properties of CFC-12 and HFC-134a

	CFC-12	HFC-134a
Suction pressure (MPa)	0.984	1.043
Discharge pressure (MPa)	0.111	0.095
Pressure ratio (−)	8.86	11.01
COP (−)	2.92	2.91
Relative Capacity (−)	1.0	0.92

Condition : −27.6 ℃ evaporating temp.
32 ℃ compressor suction temp.
41 ℃ condensing temp.
32 ℃ expansion temp.

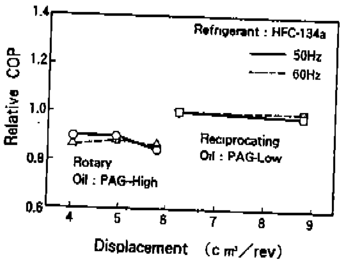

Fig. 1 Performance of compressor using HFC-134a (relative to CFC-12)

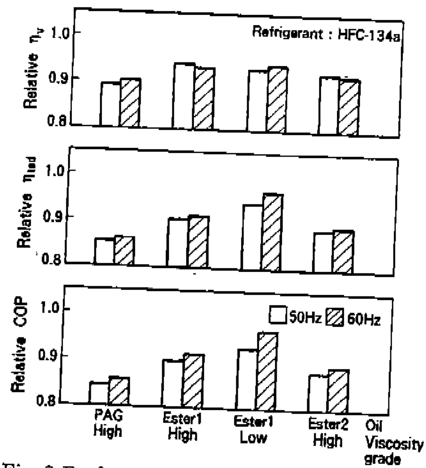

Fig. 2 Performance for different types of oil (relative to CFC-12)

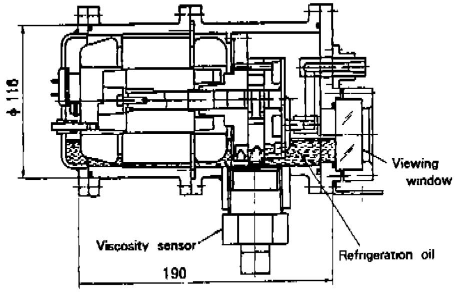

Fig. 3 Structure of compressor for measuring viscosity of HFC-134a / oil mixture

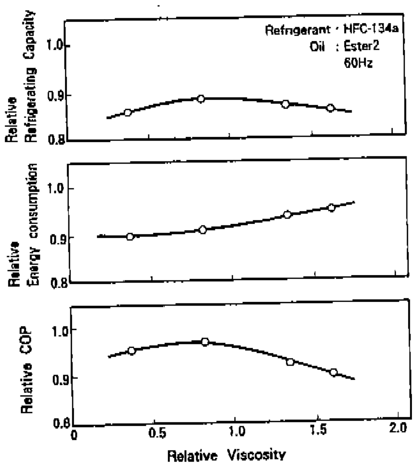

Fig. 4 Performance versus viscosity of HFC-134a / oil mixture (relative to CFC-12 / alkylbenzen oil mixture)

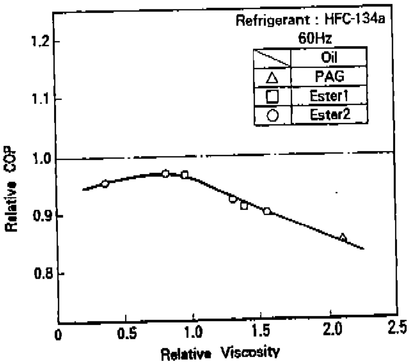

Fig. 5 COP versus viscosity of HFC-134a / oil mixture (relative to CFC-12 / alkylbenzen oil mixture)

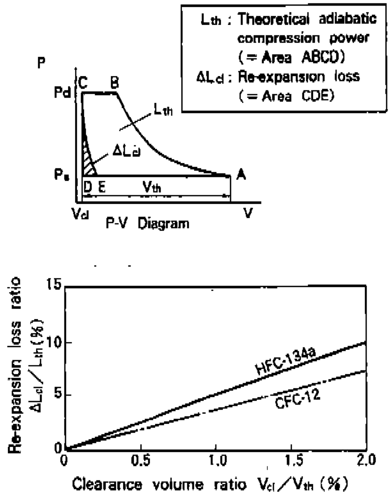

Fig. 6 Comparison between CFC-12 and HFC-134a re-expansion loss

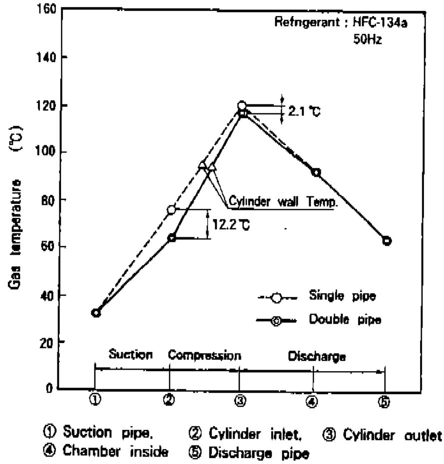

Fig. 7 Change in gas temperature inside compressor

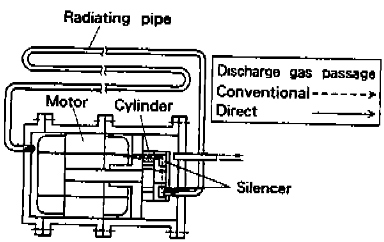

Fig. 8 Passage of discharge gas

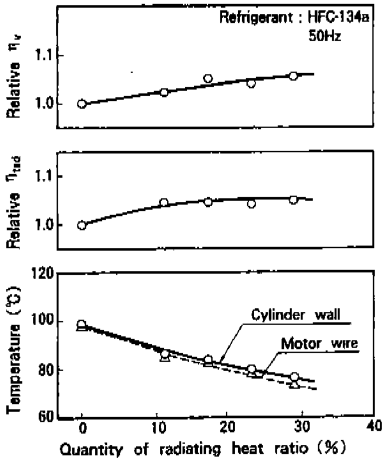

Fig. 9 Performance and temperature versus quantity of radiating heat

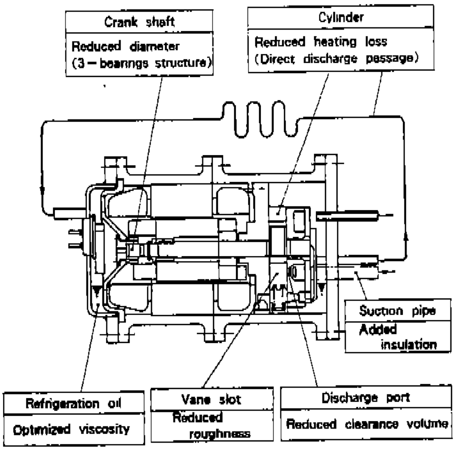

Fig. 10 Structure of prototype compressor

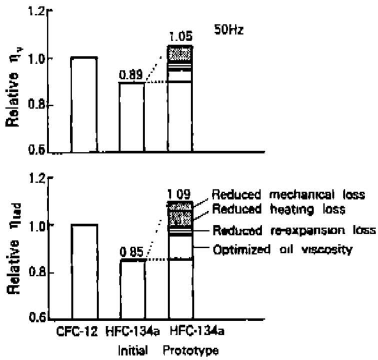

Fig. 11 Performance of prototype compressor