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HIGH EFFICIENCY COMPACT ROTARY COMPRESSOR USING THE NEW REFRIGERANT HFC-134a FOR AUTOMOTIVE AIR CONDITIONERS

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ABSTRACT

In accordance with worldwide consensus, the use of chlorofluorocarbons, which play an important role in damaging the ozone layer, should be completely abolished by the end of 1995. This provided the impetus for developing the three vane rotary compressor. Formerly CFC-12 was used as the refrigerant in automotive air conditioners, until HFC-134a was substituted for CFC-12. In response to the change to HFC-134a, we developed the vane rotary compressor, which is smaller in size than, but superior in effectiveness to, the previous rotary compressor. We eliminated the problems caused by changing the refrigerant to HFC-134a by examining the materials for seals, choosing lubricating oil specifically for this compressor, and lightening lubricating conditions in the compressor, as well as reconsidering the materials for the rubbing parts. Moreover, by measuring the fluctuating pressure in the compression process and by performing a quantitative analysis of the compression losses, we reduced the gas leakage, which was a fundamental component of the loss. We redeveloped the seal ring, and reduced gas leakage by optimizing the oil supplying passages. These developments made it possible to improve the compression efficiency by 15% over that of the previous rotary compressors. In this report we explain why the three vane rotary compressor is so efficient, concentrating on the aforementioned developments.

1. INTRODUCTION

A worldwide ban on ozone-depleting CFC-12, the most common automotive air conditioner refrigerant, is set to go into effect by the end of 1995. Hence, the development of mechanisms which work efficiently using the new refrigerant, HFC-134a, is an urgent task. We have been developing sliding-vane type rotary compressors for automotive air conditioners that have the advantages of low noise and low vibration. By incorporating a range of advanced technologies into these compressors, we have developed a compact, highly efficient compressor which uses the refrigerant HFC-134a. The ozone-depleting coefficient of HFC-134a is zero because it does not have any chlorine atoms that cause ozone-layer depletion. However, it has the following disadvantages as to physical characteristics:

(1) Strong polarity and no compatibility with conventional mineral oils;
(2) Small molecular diameter and high permeability with regard to organic materials;
(3) Inability to improve the lubricity of oil used in the refrigeration system. Due to its zero chlorine content, it provides no extreme pressure reaction caused by chloride film under boundary lubrication conditions;
(4) The condensing pressure is 0.2 - 0.3 MPa higher than that of CFC-12 over the normal operating temperature range.

In view of these disadvantages, the following problems concerning the compressor must be solved before HFC-134a can be used effectively as a refrigerant in automotive air conditioners:

(1) Select optimal oil: The oil for the refrigeration system, mixed with the refrigerant, circulates in the works as refrigerant. Therefore, it is necessary to choose the best oil which has a high compatibility with HFC-134a and a high lubricity as the compressor lubricant;
(2) Select optimal material for seals: Because many O-rings are used in the sealing portions of the compressor, it is necessary to choose an appropriate type of rubber which is resistant to permeation by the refrigerant;
(3) Improve lubricity: It is necessary to increase the wear resistance of sliding materials and to reduce the loads at sliding faces;
(4) Increase efficiency: It is necessary to boost the efficiency of the compressor above that of the conventional-type compressor, because the power required for the operation of the former is higher due to the higher compression ratio.

In this paper, methods for solving these problems, basic technologies and also the structure, characteristics, and operating performance of a high efficiency three vane rotary compressor are described.

2. STRUCTURE OF A COMPACT HIGH EFFICIENCY ROTARY COMPRESSOR

The structure of a rotary compressor and its specifications are shown in Fig. 1 and Table 1, respectively. Its mass is 5.7 kg and its dimensions are 195 mm long, 108 mm wide and 140 mm high. Its cooling performance is 5.1 kW (1800 rpm, Pd 1.67 MPa, Ps 0.20 MPa, S.H 10, S.C 5) and its rotation speed varies between 700 and 9,000 rpm.

The rotary compressor has a three vane rotary-type compression mechanism, as shown in the right-hand side view in Fig. 1. The rotor is positioned eccentrically to the center of the cylinder and supported by the needle bearings in the front and rear housings. A small clearance is maintained between the rotor and cylinder. Three slidable vanes are inserted into the slit portion of the rotor. The maximum volume formed by two vanes is 38.3 cm^3 and the total displacement volume per cycle is 115 cm^3, because this three vane configuration gives three discharges per cycle.

Table 1 Specifications of developed compressor

Type			Sliding vane type rotary
Displacement		cm^3	115
Rotation speed		r/min	700 - 9000
Performance *	Freezing	kW	5.1
	Electric power required	kW	2.5
Dimensions	Width	mm	108
	Length	mm	195
	Height	mm	140
Refrigerant used			HFC-134a
Refrigerant oil			PAG B

* Nc	1800 r/min
Pd	1.67 MPa
Ps	0.20 MPa
SH	10 °C
SC	5 °C

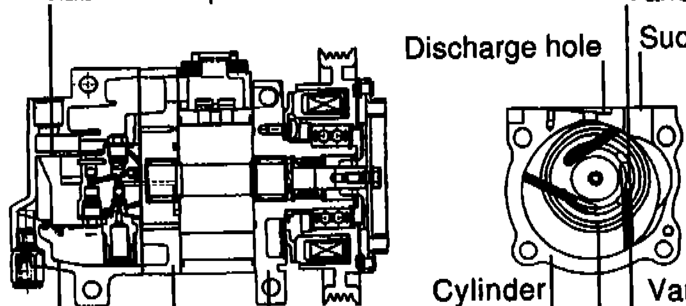

Fig. 1 Structure of compressor

During operation of the compressor, lubricating oil stored in the high pressure case is supplied to the back portion of each vane and pushes each vane to the cylinder inner surface. This high pressure applied to the back portion of the vane is called "vane back pressure." The pressure on the tip side of the vane becomes higher than that in the high pressure case because of over-compression in the compression chamber during the compression process. Then the vanes jump from the wall of the cylinder. To prevent this jumping, the groove for supplying oil and the space behind the vane are separated from each other at a rotation angle of 48° in front of the top dead center (the minimum clearance between the rotor and cylinder). Thus the space behind the vane is closed and thereby the mixture of lubricating oil and refrigerant stored in the closed space behind the vane is compressed by the vane displacement itself. The resulting high pressure prevents the vane from jumping. This high back pressure section is called the "confined section." The inside of the compressor is sealed from the outside by using the shaft seal and the O-rings in the contact portion between the cylinder and side housing and in the engagement portion between the cylinder and thermal protector.

3. SELECTION OF REFRIGERATION OIL

The major characteristics required for a lubricating oil in a compressor for automotive air conditioners are high compatibility with refrigerant, high chemical stability, moderate viscosity, high lubricity, low hygroscopicity and complete resistance to hydrolysis. Oils with high compatibility with HFC-134a include the fluoride oil, the polyall ester oil (POE), the polyalkylene glycol oil (PAG), and the polycarbonate oil [1]. As shown in Table 1, PAG B was selected as the refrigeration oil, due to its resistance to hydrolysis, low cost, high worldwide availability, and high two-phase-separation temperature on the higher temperature side.

4. SELECTION OF RUBBER MATERIAL

In the rotary compressor, the shaft seal is used for sealing between the inside and outside of the compressor, and O-rings made of fluoride rubber are used in the contact portion between the cylinder and side housing and in the engagement portion between the cylinder and thermal protector. The change to HFC-134a makes it imperative that the rubber material used for t his O-ring is completely resistant to both the refrigerant and the oil.

Generally, when a refrigerant foams, causing rubber to swell, the rubber cracks [2]. After rubber materials were caused to swell by oil and refrigerant, the pressure was quickly reduced and heat was applied until a constant temperature was reached. Following this, the temperature was measured where cracks were generated in the rubber materials. Table 3 shows the results. These results indicate that the polymer where the connection density of HNBR was increased is effective and the resistance to PAG and HFC-134a foaming is better than the resistance to mineral oil and CFC-12 foamingof conventional fluoride rubber.

Table 2 Selection of refrigeration oil

		Paraffin mineral oil	PAG A	PAG B	PAG C	POE A
Hue	ASTM	L2.0	L0.5	L0.5	L0.5	L0.5
Dynamic viscocity mm^2/s	40 °C 100 °C	190.3 17.2	54.6 10.6	51.6 9.9	102 14.4	71.5 10.0
Viscocity index		96	187	181	146	124
Total acid value	mgK0H/g	0.01	0.01	0.01	0.01	0.01
Two-phase-separation temperature	High temp. °C Low temp. °C	> C.T -25	80 < -70	70 < -70	75 < -70	> 80 < -70
Chemical stability	175 °C x 14 days	L2.0 CKC-12	L0.5	L0.5	L0.5	L0.5
Water saturation quantity %	30 °C 60%	0.005	1.1	1.1	0.8	0.2
Lubricity *1	mg	8	27	25	7	10
Resistant to hydrolysis *2		No change	No change	No change	No change	Change

Table 3 Cracking temperature of rubber materials

Material / Heating Temperature	Fluoride rubber /CFC-12	HNBR A /HFC-134a	HNBR B /HFC-134a
130°C	No crack	No crack	No crack
150	No crack	Cracks	No crack
170	No crack	Cracks	No crack

*1 FALEX 100°C, 1.11kN x 1h (blowing of refrigerant)

*2 Water: 1000 ppm, 175°C x 14 days

5 . IMPROVEMENT OF LUBRICITY

The main sliding portions inside the compressor are the shaft and the needle bearing pair, the rotor and the side housing pair, the vane and the side housing pair, the vane and the cylinder pair, and the rotor and the vane pair. In particular, the friction between the rotor shaft and the needle bearing, and the vane and the side housing, tends to generate a adhesive wearing, because the sliding pairs are of the same material: the former, Fe/Fe, the latter, Al/Al. The actual machine test results showed that the wear of the shaft under Condition A in Table 4 was greater than that under the conditions in combination with CFC-12 and mineral oil. The results also showed that under Condition B, the wearing of the side housing and the vane increased. Generally, when an increase of the face pressure P shifts the fluid lubrication condition to the boundary lubrication condition, a chlorine film on the sliding surface is formed by the reaction between the chlorine in the refrigerant and the metal surface substance when CFC-12 is the refrigerant. The chlorine works as an extreme pressure additive. In the case of HFC-134a, the aforementioned increase in wear is thought to occur because the chlorine film is not formed. To solve these problems under Conditions A and

B, the following studies were carried out.

5.1 Improvement of Wear Resistance in the Bearing Section

The shaft and the needle bearing of the compressor are in rolling friction and are permanently supplied with lubricating oil because they are positioned in the oiling path. During operation of the compressor, the fluid lubrication condition is formed under low discharge pressure and low discharge temperature, but it changes to the boundary lubrication condition in the range over the discharging pressure, Pd = 2.45 MPa, and the discharge temperature, Td=155 °C. As the endurance test results in Fig. 2 show, therefore, the wear on the shaft of a conventional compressor becomes extremely great.

Table 4 Durability test conditions for actual machine

Conditions		A	B
Rotation speed	r/min	4500	700 - 8500
Discharging pressure	MPa	2.74	2.16
Suction pressure	MPa	0.10	0.08 - 0.12
Overheat temperature of suction gas	°C	10 -20	10
Interrupted time ON/OFF	s	No interruption	5/2
Testing time	h	500	300

The frictional force F on the boundary lubrication surface can be described as follows [3]:

$$F = A \{ a\ Sm + (1-a)\ Se \}, \qquad (1)$$

where the lubrication surface area is represented by A, the ratio of the broken area of the oil film by "a ", the shearing strength of the metal contact by Sm and the shearing strength of the lubricant film by Se. The smaller "a " and Se become, the smaller the boundary friction becomes. Therefore, difficulty of breakage and ease of shearing are desirable for the boundary lubricating film. Generally, phosphoric esters, fatty acids, chlorinated paraffins, and organic metals are effective additives as boundary lubricating agents, which form a boundary film with the above characteristics on the metal surface. Because of its chemical stability with HFC-134a and PAG, a phosphoric ester was studied. The Falex test, which involved blowing of HFC-134a, was adopted as an evaluating method. Fig. 3 shows the results.

Adding phosphoric ester A to around 1% reduces the degree of wear considerably. Further addition does not reduce wear much further. In addition, there is the danger of increasing the total acid value at hydrolysis. Therefore, approximately 1% phosphoric ester A was selected as the best alternative.

By performing a special heat treatment on the shaft surface of the compressor, the decrease in hardness of the shaft caused by the temperature rising during operation was reduced and the fatigue strength of the shaft surface was increased. As a result, the wear on the shaft of the newly-developed compressor was decreased to within several mm, as shown in Fig. 2, and its life was extended.

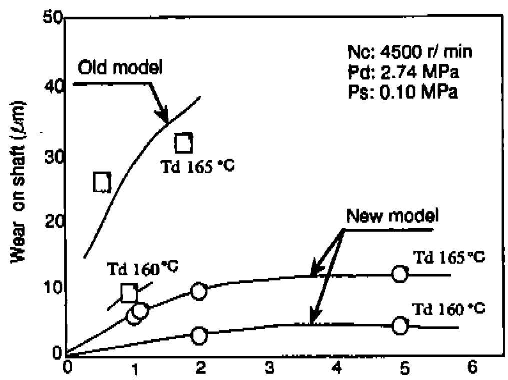

Fig.2 Wear on shaft

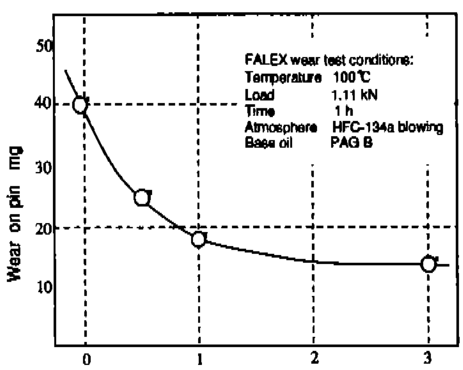

Fig.3 Percent by weight of phosphoric ester A added

5.2 Improvement of Wear Resistance in the Side Housing and Vane

Under Condition B of the actual test model endurance test, the degree of wear on the sliding section in the vane and the side housing increased. To determine how this wear could be reduced, the following tests were performed.

The conventional method for increasing the wear resistance is to coat resin on the sliding section between the side face of the vane (Al) and the housing (Al) so as to avoid direct metal contact, decreasing the values of "a " and Se. The high pressure wearing test machine shown in Fig. 4 was used to evaluate this wear resistance in a refrigerant atmosphere [4]. In this machine, a test piece was made to slide under a combination of refrigerant and refrigeration oil and the extreme pressure reaction of the refrigerant was also evaluated. After coating resin over the sliding section of the vane of the test piece, the face pressure was measured with the torque made to rise sharply by increasing the compression load under the conditions shown in Table 5.

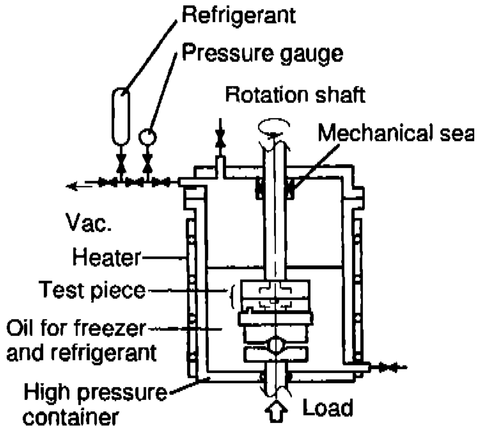

Fig.4 High pressure wearing test machine

Table 5 Conditions of high pressure wearing test

Refrigerant/Oil for freezer	80/20 wt%
Atmosphere temperature	80 - 90 °C
Rotation speed	5000 r/min
Load	Seizing is determined by increasing the load another 49N every three minutes which causes torque deviations of the disc.
Test piece material	Vane: Various types fo resin coating on the sliding surface of high silicon content aluminum alloy Disc: ADC12

Table 6 shows the results, which indicate that even when the conventional coating material is used, the sliding section seizes under smaller loads in an HFC-134a atmosphere than in CFC-12. The seizing loads for combinations of several kinds of coating materials and refrigeration oils were evaluated. The value of the seizing load with the conventional CFC-12 and coating A was set as the target before the tests were performed. We conclude that adding graphite to the coating material reduces the frictional coefficient. Changing the binder from polyamide-imide (PAI) to denatured PAI increases the binding strength and makes the seizing load using a combination of HFC-134a and PAG increase by 30% over that using a combination of CFC-12 and mineral oil.

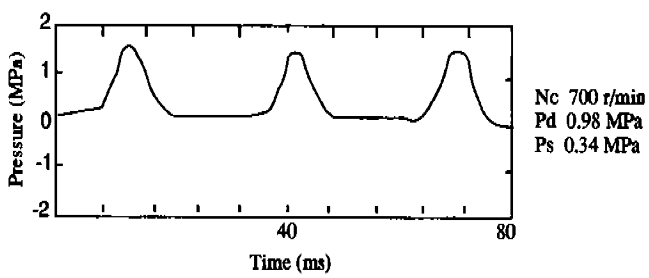

Fig.5 Pressure variation in the confined section

Table 6 Results of seizing load resistance test

Coating	Thickness of film	Seizing load kN		
		Paraffin mineral oil / CFC-12	PAG B / HFC-134a	PAG C / HFC-134a
A	20	1.77	1.28	0.93
B	20	0.88	0.69	-
C	20	1.81	0.64	1.06
D	20	2.21	1.91	0.88
D	30	-	2.35	-

The following hypothesis was made concerning the origin of the face pressure between the vane and the housing in the actual test model, because wearing had occurred in the rear housing. The vane back pressure during the rotation of the vane affects the force on the rear housing of the vane because a round-shaped groove was made and a seal ring was inserted to reduce leakage in the front housing. As described in Section 2, to prevent the jumping from the cylinder wall that occurs before the top dead point, the vane back pressure was increased by using the high pressure generated when the lubricating oil is confined in the vane back pressure chamber and pressure is transmitted through this incompressible fluid. Fig. 5 shows the pressure variation in the vane back pressure chamber under normal operation, indicating that in the "confined section," the pressure in the vane back pressure chamber is 1.7 times greater than the discharging pressure. This change in back pressure further increases on start of high-speed rotation. It is assumed that this occurs because, when the compressor stops at low rotation speed, the vane back pressure chamber and oiling path are filled with refrigeration oil; when the compressor starts up at high rotation speed, the refrigeration oil inside the vane back pressure chamber is compressed by rapid immersion of the vane in the confined section, and this generated pressure causes an excessive face pressure on the side face of the vane. To reduce this pressure, a pressure reduction mechanism was installed on the downstream side of the oiling path so that lubricating oil and refrigerant mixing fluid can be supplied into the vane back pressure chamber. This pressure reduction mechanism has reduced the vane back pressure at startup and the face pressures on the vane side face and side housing face by 20%.

The aforementioned improvement in load resistance performance of the coating material and reduction of the vane back pressure have reduced the wear on t he actual test model of the side housing to approximately 2 to 6 mm and have helped improve reliability.

6. IMPROVEMENT IN EFFICIENCY

The saturated pressure of HFC-134a is lower below 17 °C and higher above 17 °C, compared with that of conventional CFC-12. The operational efficiency of the compressor drops because in the refrigeration cycle using HFC-134a as the refrigerant, the evaporating pressure is lower and the condensing pressure is higher. To solve this problem, we took the following measures to improve efficiency.

6.1 Factor Analysis of Loss

A P-V diagram of the compression process was made when a conventional rotary compressor was operated at 1,000 rpm, 1,800 rpm, and 2,500 rpm with Pd = 1.76 MPa and Pa = 0.20 MPa, and the ratio of losses at each speed to the total power loss was calculated. Fig. 6 shows the results. The ordinate shows each ratio to the total power loss. These results suggest that the mechanical loss including the clutch transmission loss, the sliding loss of the bearing, and the frictional loss inside the compressor comprise a relatively small percentage, namely, 13 - 20% of the total loss .

On the other hand, the compression loss comprises a larger portion, namely, 80 - 87%. The lower the rotation speed, the larger the compression loss. Further analysis of this compression loss shows that at 1,000 rpm, for example, the leakage and heat acceptance loss during the suction process comprise the largest part, namely, approximately 47%, and the leakage and heat acceptance loss during the compression process comprises the second largest part, approximately 33%. Suction losses, such as pressure losses during the suction process, comprise 1.2%, and the discharge loss that over-compression effects during the discharge loss, comprising 4.8%, are both relatively small. Thus, the leakage and heat acceptance loss comprises 80% of the total power loss. Fig. 7 shows the causes of this leakage and heat acceptance loss. Reducing the leakage around the vane by decreasing the clearance around the vane is problematic because the decrease has a negative effect on the endurance resistance due to the difference of the thermal expansions between the vane made of Alalloy and the surrounding metal. Furthermore, it is not easy to decrease the diameter of the over-compression hole because it generates an excessive pressure during liquid compression. We have taken the following fundamental measures to reduce the leakage on the side face of the rotor.

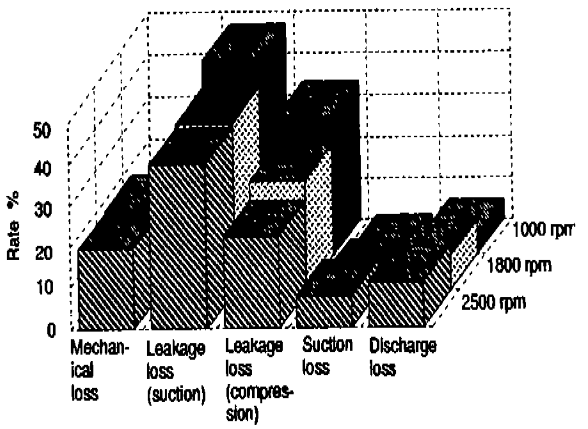

Fig.6 Analysis of power loss

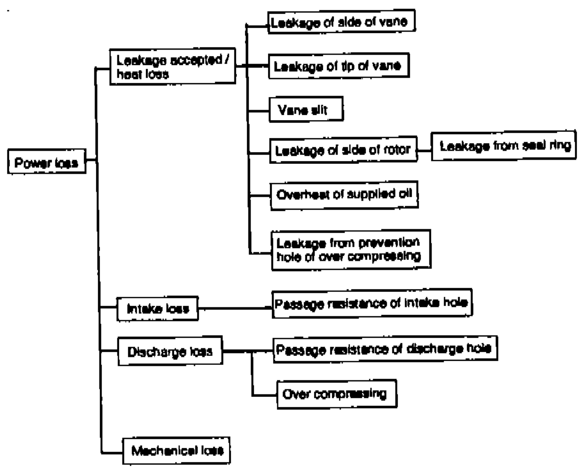

Fig.7 Factor analysis of power loss

6.2 Improvement of the Shape of the Seal Ring

The oil supplied from the high pressure case goes along the oiling path from the groove for supplying oil in the rear housing, to the vane back pressure chamber. On the path between the vane back pressure chamber and the compressing chamber, the oil leaks from the clearance between the rotor and the housing. In a conventional seal ring, the leak pressure is used to float and move the seal ring, and it pushes the seal ring onto the side of the rotor, creating an effective seal. In the newly-developed seal ring, a lip has been added to the rotor side of the ring (see Fig. 8) to allow a slight but continuous contact with the rotor. In addition, as with conventional designs, during operation of the compressor, the leak pressure is supplied to the inside of the seal ring and pushes the ring onto the rotor. This helps improve seal performance.

Improved seal performance contributes to the smaller leakage loss. On the other hand, mechanical losses such as the frictional loss between the seal ring and the rotor increase. The mechanical loss is 10 to 13% of the total loss, which is fairly small. However, it is necessary to minimize this loss. The diameter of the newly-developed seal ring was decreased to 91% of that of a conventional seal ring to make the contact area with the rotor smaller and to reduce the friction loss.

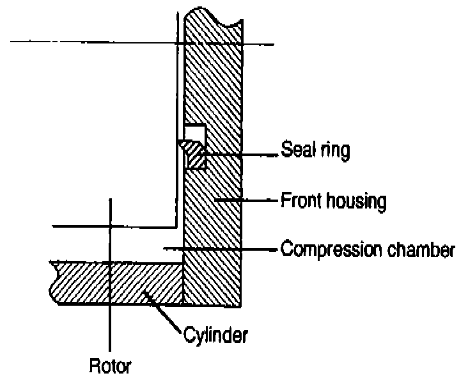

Fig.8 Section drawing of seal ring

6.3 Improvement of the Pressure Balance between the Front and Rear Sides of the Rotor

As mentioned above, improved sealing performance of the front housing helps decrease leakage loss. However, if the quantity of the supplied oil is the same, the pressure inside the seal ring set in the front side will be higher. Therefore, it is necessary to optimize the shape of the groove in order to improve the pressure balance between the front and rear sides of the rotor, since the central pressure of the front side of the rotor is high. Improvement of the sealing performance makes the vane back pressure high, too, which increases the friction loss of the vane tip and the mechanical loss. As for the quantity of the lubricating oil supplied, the greater the supply of oil, the higher the vane back pressure and the greater the mechanical loss. Conversely, the lower the supply of oil, the less mechanical loss. This is accompanied by reduced seal performance and decreased volume efficiency, however.

Thus, there exists a point that maximizes the operational efficiency in proportion to the quantity of lubricating oil supplied. In the newly-developed model, to maximize the operation efficiency, the shape of the groove for the lubricating oil and the supply of lubricating oil have been optimized so that the pressures between the front and rear side of the rotor are almost the same.

6.4 Improvement of Separation Performance of Lubricating Oil

The lubricating oil in the compressor circulates in the refrigeration cycle. Reducing the ratio of the circulated quantity of oil to the refrigerant (the oil circulation ratio) helps increase the efficiency of the heat exchanger and reduce the pressure loss in the suction pipes. This maximizes the cooling performance of the air conditioner system. To reduce the outflow of lubricating oil from the compressor and to reduce the oil circulation ratio, the oil separator in the high pressure case was improved. As shown in Fig. 9, in the conventional model, the oil mist discharged from the discharge hole in the cylinder collides with the inside wall of the high pressure case and is separated from the fluid with which it is mixed. However, the lubricating oil is easily discharged because the discharge pipe is close by. A U-shaped rib was installed on the upper section of the oil mist collision area of the newly-developed model to make the separated lubricating oil pass along the rib to the discharge pipe and prevent the oil from discharging easily. In addition, a number of vertical ribs were installed on the inside wall of the high pressure case to make the remaining oil mist, that was not separated on the inside wall, collide many times with the inside wall and separate from the mixing fluid while circulating inside the high pressure case. Fig. 10 shows the effects of adopting the aforementioned oil separator, and the results suggest that the quantity of outflow of the lubricating oil was reduced and the oil circulation ratio was also reduced in spite of the quantity of enclosed lubricating oil in the refrigeration cycle being the same as that of a conventional-type compressor. Good cooling performance can be obtained, although the action of the compressor unit was the same as that of a conventional one. Moreover, high reliability of the compressor is made possible because the lubricating oil prevented from flowing out is left inside the compressor.

7 . CHARACERISTICS OF THE COMPRESSOR

Fig. 11 shows a comparison between the characteristics of the newly-developed high efficiency compressor and that of a conventional compressor with equivalent displacement. In the area of 1,800 rpm, the volume efficiency is increased by 11%, and the power force required is reduced by 4%, which contributes to a 15% improvement in the performance coefficient. The lower the rotation speed, the greater the degree of improvement. This result was made possible by reducing the leakage loss. The maximum allowable rotation speed of the new compressor is 9,000 rpm. Thus a greater number for the pulley ratio (ratio of the rotation speed of the engine to that of the compressor) can be selected and the better cooling performance can be obtained. In addition, in the low speed area, the compressor is used at a relatively high rotation speed, which can further reduce leakage and improve efficiency.

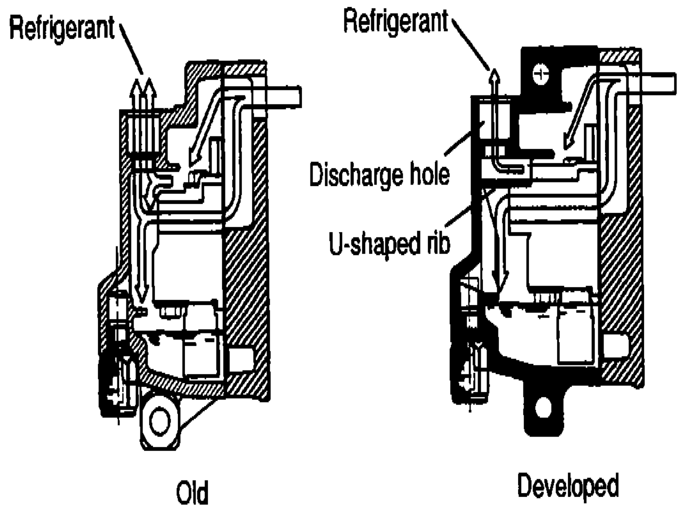

Fig.9 Section drawing of the oil separator

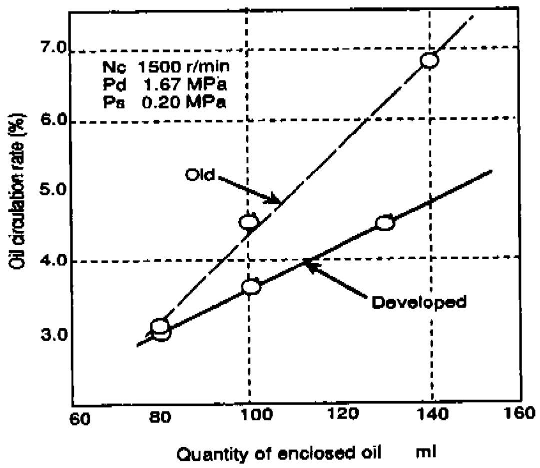

Fig.10 Quantity of enclosed oil vs. Oil circulation rate

8. SUMMARY

Responding to the worldwide demand for protection of the ozone layer, we have developed a compact high efficiency rotary compressor using the refrigerant HFC-134a in automotive air conditioners and put it to practical use. Its operation efficiency is 15% higher than that of a conventional compressor, and this has been achieved by developing new materials, studying the lubrication path, and developing new basic technologies, such as the seal ring for reducing leakage. We intend to continue to develop a range of technical innovations for new compressors which we hope will contribute to environmental protection.

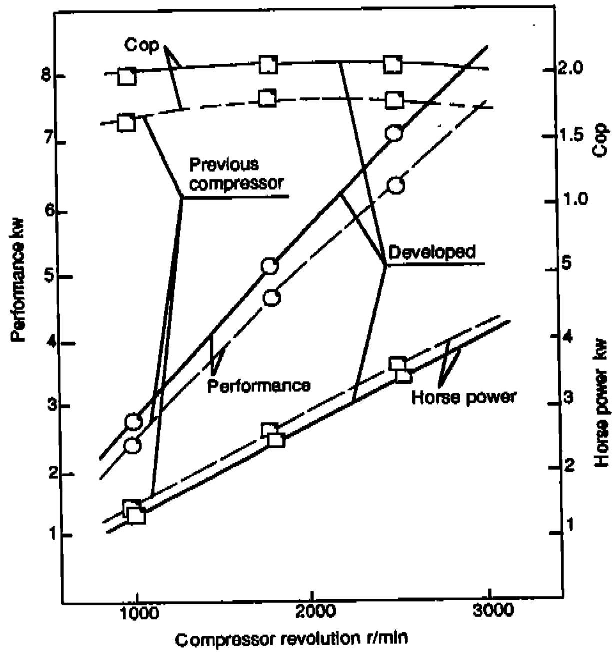

Fig. 11 Comparison of performance characteristics

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