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ABSTRACT

We have completed development of the new scroll compressors for R410A by upgrading our existing air-conditioners' scroll compressors.
We have incorporated our various technologies into developing these R410A new scroll compressors and have succeeded in several major achievements, including:
1) Optimization of the orbiting scroll back pressure force, which reduces the thrust bearing loss;
2) Optimization of the scroll wrap dimension, which achieves high COP;
3) Improvement of shell strength, which endures 15.2MPa high side pressure.

1. INTRODUCTION

More and more of R410A is being used in air-conditioners lately to improve performance, given that R410A can decrease refrigerant pressure loss by reducing the volume flow rate.
This market trend has given rise to an urgent need to develop a high-efficiency compressor for R410A. The high-efficiency scroll compressors for R22/R407C, which have a unique high/low pressure side structure, are being manufactured annually. Using this R407C compressor as the base allowed us to develop a high-efficiency R410A compressor over a shorter development period at a lower investment.
There were, however, three technical problems to resolve:
1) Thrust bearing loss needed to be reduced to achieve high efficiency.
2) Scroll wrap dimension needed to be optimized to resist the higher pressure.
3) Shell strength needed to be increased to resist the higher pressure.
We have confirmed that the following three technologies provide as effective measures to resolve these issues:
1) Asymmetrical scroll wraps
2) Non-uniform thickness scroll wraps
3) Modified shell shape
In the last section, we show that the R410A high-efficiency compressor which uses these technologies.

2. Structure of compressors

The development model was based on a scroll compressor with a high/low pressure side shell structure, as shown in Figure 1. This structure has the following four characteristics.
1) The scroll compressor has a high/low pressure side shell structure, with the motor located in the high-pressure side and the compressed element is in the low-pressure side.
2) It has an asymmetrical scroll wraps and non-uniform thickness scroll wraps.
3) Is has the axial and radial compliance mechanism.
4) All of discharge gas is let flow from the orbiting scroll into the gas passage of the crank shaft and it is let flow through the effective oil separator to lower side of a motor.

The next section describes the new technologies for R410A that have been developed.

3. Developments

3.1 Reduction in thrust bearing loss

This compressor is structured so that the orbiting scroll is pressed against the fixed scroll by the high-pressure gas supplied to the back of the orbiting scroll (as shown in Figure 2), so that leakage in the scroll is reduced.
The thrust bearing loss is determined by sliding speed and pressure force: therefore, both need to be decreased.

· Reduction in the sliding speed

It is widely known that a smaller orbiting radius will effectively decrease the sliding speed. However, there are several restrictions involved in making the orbiting radius smaller, as shown in Figure 3, including the limited scroll strength because of higher scroll wraps and limited discharge passage area because of the smaller scroll.
Figure 3 shows the relationship between the orbiting radius and the compressor loss.
The smaller the orbiting radius, the smaller the thrust bearing loss. On the other hand, however, the smaller discharge passage increases pressure loss, resulting in minimum value.
The scroll strength limitation also had to be considered to minimize the orbiting radius. Applying the specifications for R407C compressor without modification resulted in locating point "A" indicated in Figure 3. This meant that the orbiting radius could be made smaller. From these observations, it was determined that locating the orbiting radius for R410A compressor at point "B" would significantly reduce the sliding speed. As a result, the loss was decreased 3% against that for R407C compressor. This relationship can also be applied to other compressors to optimize the orbiting radius.

· Reduction in the orbiting scroll back pressure force

When the orbiting scroll back pressure force is too large, friction loss increases. On the other hand, when the orbiting scroll back pressure force is too small, leakage loss increases. The orbiting scroll back pressure force for this compressor was optimized using the following method.
Figure 4 shows the relationship between the orbiting scroll back pressure force and the compressor loss, the coefficient of tipping.
It was found from Figure 4 that when the orbiting scroll back pressure force in the axial direction is reduced, compressor loss can be decreased because of the reduction in thrust bearing loss. When a certain the coefficient of tipping is exceeded, however, the scrolls start separating, and an abrupt increase in loss is seen.
To improve this situation, an asymmetrical scroll wrap was also applied to this compressor to lower the overall the coefficient of tipping, achieving a 1% reduction in compressor loss compared to a symmetrical scroll wrap.

3.2 Optimization of scroll wrap dimension

Because the pressure is higher with the use of R410A, the orbiting radius cannot be made smaller using the scroll wrap dimension designed for R407C compressor because of the limitation of the scroll strength. To solve this problem, non-uniform thickness scroll wraps, which increase its thickness gradually from out side to inside, was further improved.
This modification increased the rate of the change in thickness so that the scroll inside, where scroll strength would be insufficient because of the increased pressure, was thicker than for the R407C compressor. The improving scrolls, as shown in the analysis results in Figures 5 and 6, ensured sufficient safety factor against the higher pressure of R410A.
Although the discharge port area had to be decreased because of this modification, the port area equivalent to that for R407C compressor could be maintained by adapting the modified scroll shape, as shown in Figure 7. Now, both sufficient scroll strength and the discharge port area were secured.

3.3 Shell strength improvement

The technical problem occurred when we tried to increase the shell strength to resist the higher pressure with the use of R410A, a crack in the joint that connects the hermetic terminal and the shell.
Figure 8 is a detail of the terminal section. A flat area was created to fit the terminal to the shell, then projection welding was done to attach and seal the terminal in that area.
Next, the dynamic model shown in Figure 9 was invented to locate the cause of this problem.
Figure 9 shows that when a larger flat area height is set, moment B increases in the opposite direction to offset moment A, thus decreasing the stress.

We also conducted a strength analysis, as shown in Figure 10, and pressure tests using the actual model by gradually changing the flat area height in order to verify our assumption.

The results are shown in Figure 11. As can be seen in this figure, as the flat area height is increased, the shell strength increases, however, the strength begins to drop when it reaches a certain height. This is because the shell strength around the terminal declined due to the thinner shell thickness.
From our study of the above, we have found that the shell strength can be improved by increasing the terminal flat area height, and that there is an optimum point for the flat area height.

3.4 The compressor for R410A

We developed the compressor with compression efficiency equivalent to that of the base model. The specifications of this compressor are shown in Table 1.

4. CONCLUSIONS

We can conclude that it is beneficial to use the following designs for Scroll Compressors for R410A in order to improve on the scroll compressors for R407C.
•The optimum orbiting radius that minimizes the thrust sliding speed was able to select by fully understanding the relationship between the orbiting radius and the compressor loss (the thrust bearing loss and the pressure loss of discharge).
•Use of the asymmetrical scroll wrap was effective in reducing the orbiting scroll back pressure force in the axial direction.
•Optimizing the rate of thickness change of non-uniform thickness scroll wrap proved effective in order to maintain scroll strength against the higher pressure with R410A. Any resulting reduction in the discharge port area could be avoided by using a modified scroll wraps.
•Increasing the flat area height was effective in increasing the strength of the shell around hermetic terminal.

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[3] S. Hagiwara, et al., “Development of Scroll Compressor of Improved High-Pressure-Housing.” Proceedings of the 1998 Compressor Engineering Conference, Purdue University.

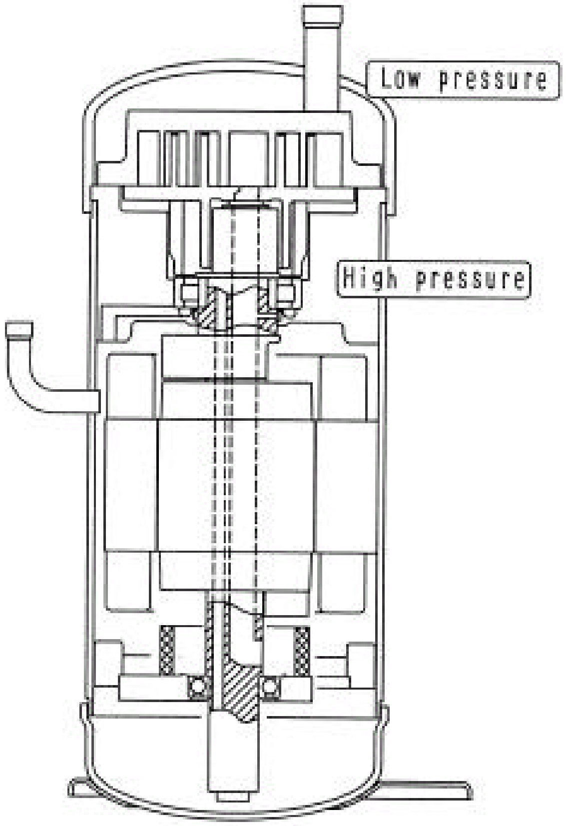

Figure 1 Compressor Cross Section

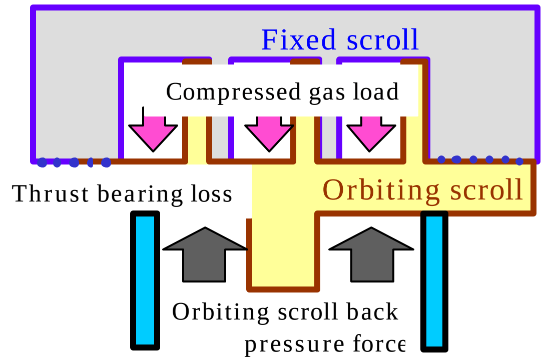

Figure 2 Schematic diagram

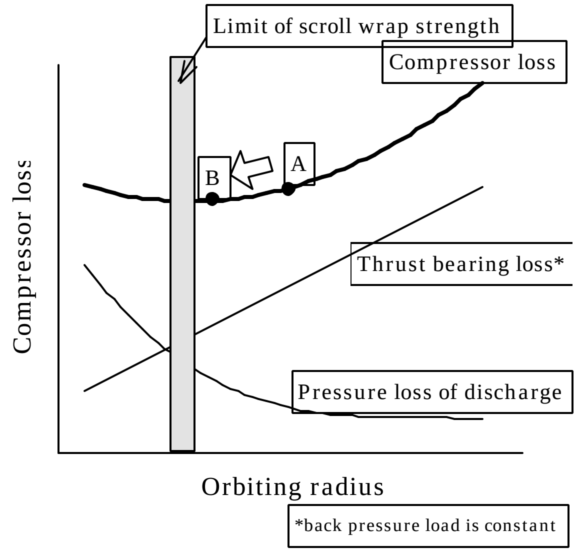

Figure 3 Relationship between the orbiting radius and the compressor loss

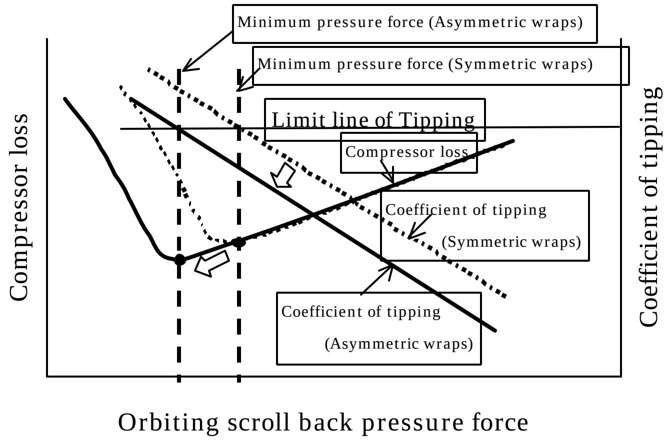

Figure 4 Relationship between the orbiting scroll back pressure force and the compressor loss

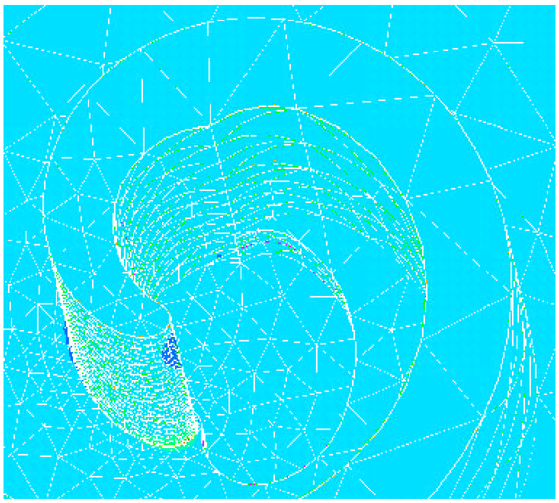

Figure 5 Analysis result of high compression ratio

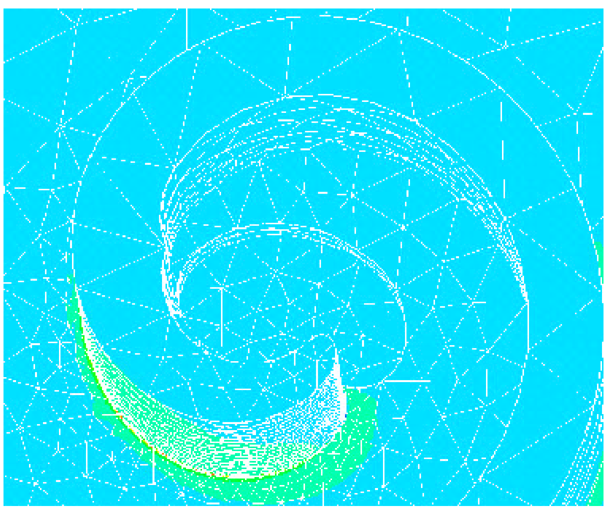

Figure 6 Analysis result of low compression ratio

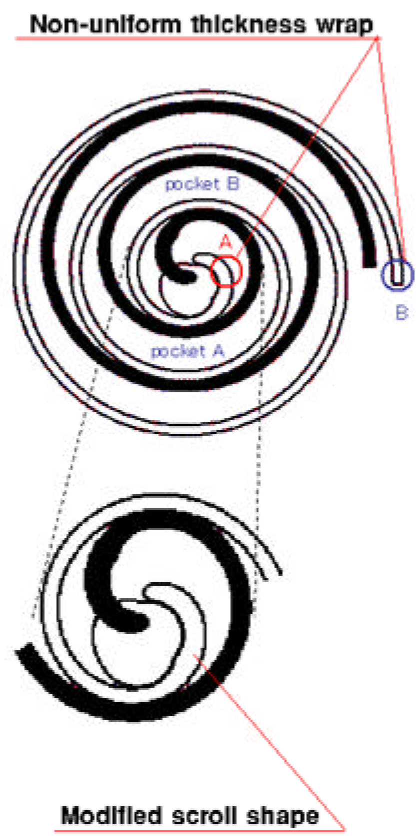

Figure 7 Description of modified scroll shape

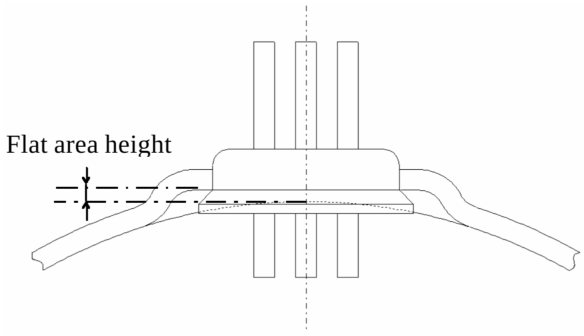

Figure 8 Detail of the terminal section

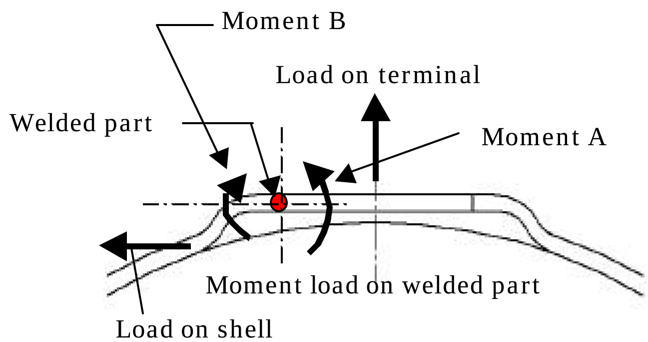

Figure 9 Dynamic model on the welded part

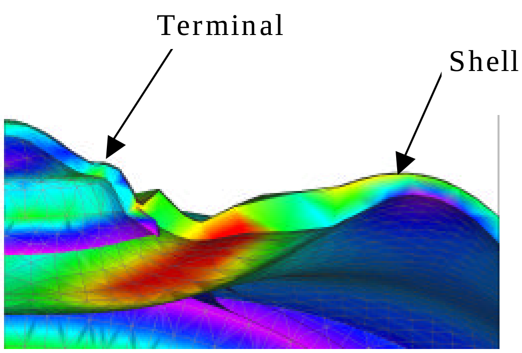

Figure 10 Analysis result of the terminal flat area

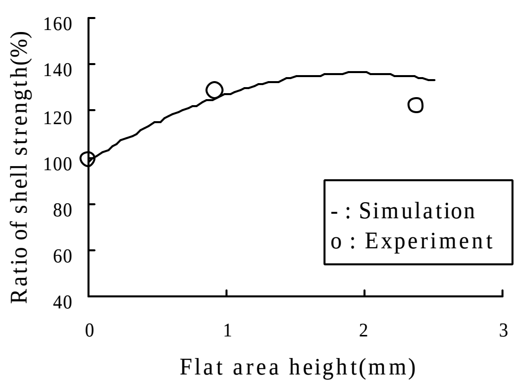

Figure 11 Relationship between the flat area height and the shell strength

Table 1: Specifications of development compressor

Capacity (Revolution speed)	COP	Size	Weight	Power
11200W@60r/s (20~108r/s)	4.11@60r/s	Diameter: 168mm Height: 420mm	33.4kg	DC Inverter

COP condition: Tc/Te/Sh/Sc=42/3/5/8(Celsius)