Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

1988

# Deformation Analysis of Scroll Members in Hermetic Scroll Compressors for Air Conditioners

Kazutaka Suefuji
*Hitachi*

Masao Shiibayshi
*Hitachi*

Rumi Minakata
*Hitachi*

Kenji Tojo
*Hitachi*

# DEFORMATION ANALYSIS OF SCROLL MEMBERS IN HERMETIC SCROLL COMPRESSORS FOR AIR CONDITIONERS

Kazutaka Suefuji*, Masao Shiibayshi*, Rumi Minakata*
Kenji Tojo**
Mechanical Engineering Research Laboratory, Hitachi Ltd.
* 390, Shimizu-shi, Shizuoka-ken, Japan (Zip code 424)
Phone: 0543-34-1111
** Tsuchiura, Ibaraki, Japan

Naoshi Uchikawa, Takao Mizuno
Shimizu Works, HITACHI Ltd.
Shimizu, Shizuoka, Japan

## ABSTRACT

Deformation of fixed and orbiting scroll members of hermetic scroll compressors is analyzed to find better scroll wrap shapes for the compressor performance. In the scroll compressor, axial and radial seal gaps between the fixed and orbiting scroll members exert a great deal of influence on performance. Under actual operating conditons, the seal gaps undergo changes (subsequently described) from the initial setting gap (clearance) due to deformation of the fixed and orbiting scroll members. In this paper, deformation of the fixed and orbiting scroll members under actual operating conditions is simulated. This is accomplished using the finite element method based on measurement data concerning pressures and temperatures. As a result, it is found that compression performance is sensitive to axial deformation and better scroll wrap shape can be predicted. The predicted scroll wrap shape is tested in experiments and 4% inprovement in adiabatic efficiency is obtained compared with conventional scroll compressors. The deformation analysis here is comfirmed to be very useful for design purposes.

## INTRODUCTION

The popularization of air-cooled and heat-pump air conditioners requires improvement in the energy efficiency ratio and high compressor efficiency over a wide range of operating conditions must be improved. Low noise and vibration level at all rotational speeds is another important consideration for compressors. Thus, rotational speed control is coming into wide use. With this in mind, a new type of compressor with scroll wraps was developed and put into commercial production in 1983[1]. This compressor has a highly efficient simple mechanism--- the so-called "Self-adjusting back pressure mechanism"[2]. This mechanism controls thrust force to support the orbiting scroll in the axial direction by back pressure provided from an intermediate process compression chamber. Developed compressors are highly efficient, but even higher efficiency is expected with the help of computer simulation analysis.

To date, the following technology and simulation code has been developed. (1) Optimize the proper range of back pressure [2][3]. (2) Simulate the scroll compressor performance[4]. These factors have contributed to the widespread applications of scroll compressors[5]. The other important area for analysis is deformation analysis of the scroll members taking into account actual operating conditions to minimize gas leakage and friction loss between the fixed and orbiting scroll. Deformation analysis results of fixed and orbiting scroll members by the finite element method, is reported in this paper. In addition, a method for improving efficiency was derived from this analysis. The effect of this inproved method was experimentaly confirmed.

## CONFIGURATION OF SCROLL COMPRESSOR

The assembled compression component is shown in Fig. 1. The main elements in a scroll compressor are a fixed scroll and an orbiting scroll, both of which have identical involute scroll wraps. These scroll members are assembled at a relative angle of 180 degree. So that they make contact with at several points and form a series of sealed compression chambers. The relative angle of the two scrolls is maintained by means of an anti-rotation coupling mechanism, such as an Oldham ring, that is located behind the orbiting scroll. The orbiting scroll is driven by a simple crank mechanism and orbits around the center of the fixed scroll. This scroll assembly is housed in the hermetic casing which is filled up with discharge gas. As the crankshaft rotates, gas is drawn to the periphery and trapped in a pair of compression chambers. Then it is compressed by volume reduction while moving toward the center of the scroll wrap. Next, the compressed gas is released through the discharge port at the center of the fixed scroll. A back pressure chamber is provided behind the orbiting scroll. This chamber is pressurized automatically at a level between the suction and discharge pressure by means of gas introduced through small apertures (back pressure ports). The pneumatic force applied to the back of the orbiting scroll supports it in the axial direction with low frictional loss and wear.

The major dimensions of the scroll compressor provided as the subject of the analysis are shown in table 1. This is a compressor for heat-pump air conditioners, and R22 refrigerant is used. The capacity is 3.0 kW at 60 Hz of electric source frequency. The fixed scroll and orbiting scroll are assembled with an axial clearance of 5 microns between the tip of the wrap and the end plate. The standard value of relative radial clearance between the flanks of the wrap is 30 microns.

## PRESSURE AND TEMPERATURE FIELD IN SCROLL MEMBERS

The gas compression process is shown in Fig. 2. The pressure of the peripheral side suction chamber is suction pressure, while that of the inner end compression chamber, open to the discharge port, is discharge pressure. Between them, a few symmetric pairs of compression chambers are formed. The pressure is identical in each pair of compression chambers. Process (1) shows the state after the suction process has just been completed. Then seal points between the two wraps shift along the wrap flanks toward the center as seen (2) to (4), and the compression chambers are gradually reduced in volume. The gas is trapped in the compression chamber at the periphery of the scrolls and compressed as the compression chamber moves toward the center.

In Fig.2 (1), for example, wraps AB and CD receive radial force due to the pressure difference as indicated by the arrows, while wraps BC and DE receive no radial force because there is no pressure difference between the inner and outer sides of the wrap.

The axial pressure distribution is shown in Fig. 3. The pressure acting on the compression chamber side of both end plates changes in steps between suction pressure and discharge pressure along the radial direction. The outer surface of the fixed scroll is in an atmosphere of discharge gas in the casing. The intermediate back pressure, need to support the orbiting scroll and to make contact with the fixed scroll, acts on the rear portion of the orbiting scroll. The supplied oil pressure affects the area in the housing of the bearing. Oil pressure is nearly as strong as discharge pressure. The total force of outer pressure on the fixed scroll or orbiting scroll is higher in both cases than the total force of inner pressure. In consequence, both scroll end plates are forced to bend inward. Inner pressure changes during one revolution of the crankshaft according to pressure change in the compression chamber. Minimum total force of inner pressure occurs just at the completion of the suction process as shown in process (1) in Fig. 2. An this point, the differential force between the outer and inner sides and inner side of each scroll end plate maximizes. Consequently, the deformation of each endplate also maximizes. The analysis subsequently mentioned is performed under this condition.

Temperature distribution is obtained by measurement at many points in the fixed scroll member while the compressor is operating under steady states of suction pressure 0.57 MPa and discharge pressure 1.88 MPa. The isothermal lines on one radial section are shown in Fig. 4. Gas temperature is low in the peripheral compression chamber, but higher in the inner compression chamber according to the degree of compression. On the other hand, the surface of the fixed scroll is exposed to the high temperature discharge gas. Furthermore, the member receives the heat from supplied oil and mechanical friction. Temperature distribution is a consequence of these heat balances.

## ANALYSIS AND APPLICATIONS

### F.E.M. MODEL

In order to analize deformation of the fixed and orbiting scroll in the pressure and temperature field, the finite element method analysis code ADINA (developed at M.I.T.) was used. A three dimensional solid element with eight nodes was used as the finite element. The configurations of analysis models are shown in Fig. 5. The fixed scroll is supported in the Z direction at all nodes on the peripheral line on the plane of the wrap tips. It is also supported in the X direction at two nodes on the Y axis and in the Y direction at two nodes on the X axis. The orbiting scroll is supported in the Z direction at all nodes on the periphery line on the end plate. It is also supported in the X and Y directions as with the fixed scroll. Regarding temperature conditions, temperature distribution in the orbiting scroll was assumed to be the same as that in the fixed scroll.

### RADIAL DEFORMATION OF SCROLL WRAP

The deformation of fixed scroll wrap tips and orbiting scroll wrap tips, at suction pressure 0.57 MPa and discharge pressure 1.88 MPa conditions, is shown in Fig.6. It is seen that the wrap deforms as it inclines in a definite radial direction as a result of gas force. Furthermore, the wrap deforms as it expands according to temperature increases. The maximum displacement is approximately 60 microns for the fixed scroll and approximately 20 microns for the orbiting scroll. It is considered that the smaller deformation of the orbiting scroll wrap is caused by stiffness of the bearing housing below the end plate. In accordance with the deformation direction, the wrap inclines to the perpendicular direction to the line of seal points at each scroll wrap. Maximum displacement along the line of seal points is approximately 20 microns. In consequence, the seal effect is not considered to be significantly damaged by the gap change according to this degree of radial deformation[4].

### AXIAL DEFORMATION OF FIXED SCROLL AND ORBITING SCROLL MEMBERS

Analyzed axial deformation of the wrap tip and end plate of the independent fixed scroll when affected by pressure and temperature is shown in Fig. 7. Both the wrap tip and end plate are displaced towards the compression chambers by about 30 microns at the center as a result of pressure. Due to rising temperature, the end plate is moved in - direction about 20 microns, but the wrap tip does not move significantly. Therefore, it is clarified that pressure acts to diminish the axial gap and rising temperature acts to enlarge the axial gap on the end plate.

Analyzed axial deformation of the wrap tip and the end plate of the independent orbiting scroll is shown in Fig. 8. Both the wrap tip and end plate displace toward the compression chambers about 25 microns at the center as a result of pressure. Due to rising temperature, the end plate is moved in + direction about 25 microns and the wrap tip is moved in + direction about 45 microns. Therefore, it is clarified that pressure and temperature both act to diminish the axial gap.

### AXIAL DEFORMATION OF ASSEMBLED SCROLL MEMBERS

The deformation is limited when the fixed scroll member and the orbiting scroll member are actually assembled. The deformation is limited by the other scroll member in the case, for example, when the wrap tip of the orbiting scroll contacts the end plate of the fixed scroll. Accordingly, contact force occurs at the contact points and the deformation of each scroll member is balanced at the point where the external forces are balanced. To simulate this phenomenon, axial displacement by gas pressure and temperature and displacement by unit axial pressure acting on a certain area in the central region were combined.

Analyzed axial deformation under the standard operating condition for all 5 micron initial clearances is shown in Fig. 9. Scroll wraps contact at several points in the central region and axial gap becomes 10 to 15 microns in the peripheral region. Accordingly, gas leakage increases compared with the assumption of a 5 micron constant axial gap. Further, friction loss also increases if the friction coefficient at the wrap contact area is greater than that at the peripheral contact area on the end plate. It has been found that effective improvements are obtained by decreasing the peripheral gap and contact force. One of the improvement was confirmed by this analysis and experiments as mentioned in the next section.

## APPLICATION FOR DESIGN OF SUITABLE SCROLL SHAPE

To decrease peripheral axial gap and contact force, initial axial clearance must be changed. For this purpose, the axial clearance in the central region is enlarged while that of the periphery remains at the same value as in the conventional case. Concretely, as Fig. 10 shows, the central region of the end plate, within 10 radians in the involute angle has a slightly stepped down surface (by 10 to 20 microns) compared with the end plate in the periphery. This causes the 15 to 25 micron axial clearance in the central region when axial clearance in the periphery is 5 microns.

The analized axial gap resulting from changing the degree of the step is shown in Fig. 11. In comparison with the conventional flat end plate, the 10 micron stepped down surface has a slightly larger gap through 6 to 10 radians of the involute angle, but it has a smaller gap below 3 radians and above 10 radians. The 20 micron stepped down surface has a much larger gap through 5 to 10 radians however it has the smallest gap above 10 radians. It is considered that synthetic gas leakage is minimized in the 10 micron stepped down surface.

These results were comfirmed by experiments as shown in Fig. 12. It is seen that the 10 micron stepped down surface within 10 radians of the involute angle has the highest volumetric and adiabatic efficiency.

Subsequently, the axial gap resulting from changing the degree of stepped down surface area is analyzed. Here, there are four cases as follows;

Axial clearance is

(1) 5 microns for the entire scroll wrap
(2) 15 microns within 5 radians and 5 microns in the periphery
(3) 15 microns within 10 radians and 5 microns in the periphery
(4) 15 microns for the entire scroll wrap

The results are shown in Fig. 13. In comparison with case (1), case (2) has a slightly smaller gap above 5 radians, however, it has significantly larger gap below 5 radians. The main reason is that there are axial contact points remaining outside of the stepped down surface area and the gap in the central region is enlarged due to the stepped down surface. Although case (3) has a slightly larger gap through 6 to 10 radians, it has the smallest gap below 5 radians and above 10 radians. Case (4) has a very large gap above 10 radians. In these cases, case (3) has the best by for seal effect.

These results were confirmed by experiments as shown in Fig. 14. It is seen that the 15 micron axial clearance within 10 radians with 5 microns in the periphery is 2% higher in volumetric efficiency and 4% higher in adiabatic efficiency than conventional compressors.

Thus the deformation analysis of actual operating scroll compressors contributes to the design of suitable scroll wrap shapes which result in high efficiency to scroll compressors.

## CONCLUSIONS

Deformation of fixed and orbiting scroll members of a hermetic scroll compressor was analized under actual operating conditions. Based on this analysis, an improved scroll wrap shape has been realized and 4% improvement in adiabatic efficiency has been obtained.

## REFERENCES

(1) K. Tojo, et al., "A Scroll Compressor for Air Conditioners", Proc. of the 1984 Purdue Compressor Technology Conference,

(2) M. Ikegawa, et al., "Scroll Compressor with Self-Ajusting Back Pressure Mechanism", ASHRAE Transactions, Vol.90, Pt.2, No.2846, 1984.

(3) N. Arai, et al., "Scroll and Screw Compressors: The Latest Compressor Technology for Air Conditioning and Refrigeration", Hitachi Review Vol.34, No.3, 1985.

(4) K. Tojo, et al., "Computer Modeling of Scroll Compressor with Self-Ajusting Back-Pressure Mechanism", Proc. of 1986 Compressor Engineering Conference at Purdue.

(5) K. Tojo, et al., "Scroll Compressors for Air Conditioning and Refregeration", IIR XVIIth International Congress of Refrigeration, 1987 (Submitted)


Table 1. Major dimensions of the tested compressor

| | | |
|---|---|---|
| Displacement volume | | 64 cm /rev |
| Built in volume ratio | | 2.7 |
| Orbiting radius | | 4.5 mm |
| Wrap thickness | | 3.6 mm |
| Wrap height | | 31 mm |
| Fixed scroll | diameter | 169 mm |
| | end plate thickness | 12 mm |
| Orbiting scroll | diameter | 131 mm |
| | end plate thickness | 10 mm |

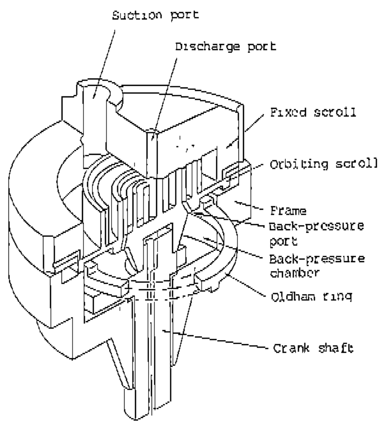


Fig. 1 Basic configuration of scroll compressor

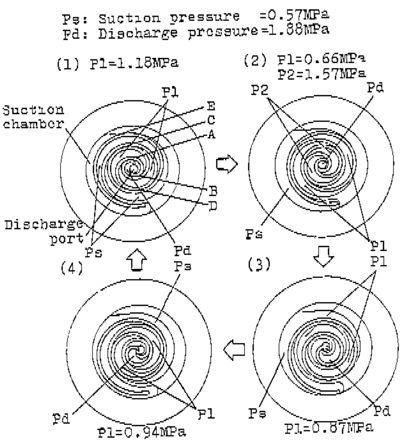


Fig. 2 Radial pressure distribution

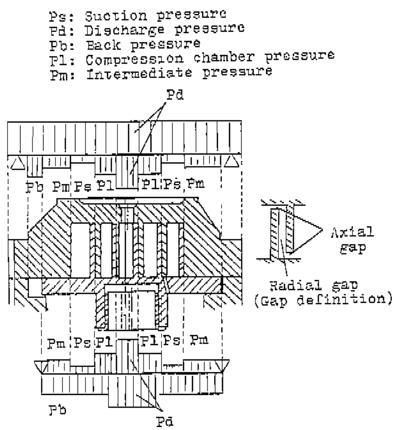


Fig. 3 Axial pressure distribution

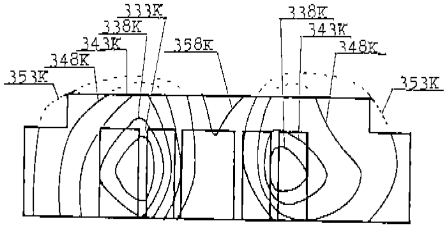


Fig. 4 Temperature distribution in fixed scroll

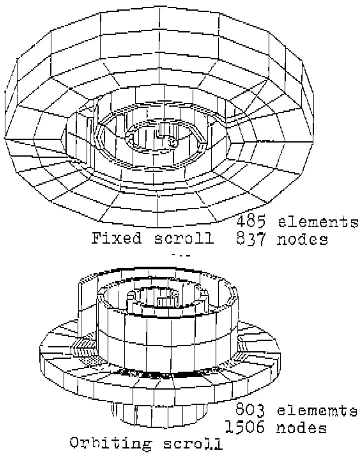


Fig. 5 F.E.M. models

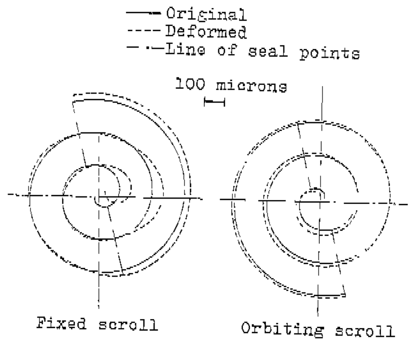


Fig. 6 Radial deformation at scroll wrap tip

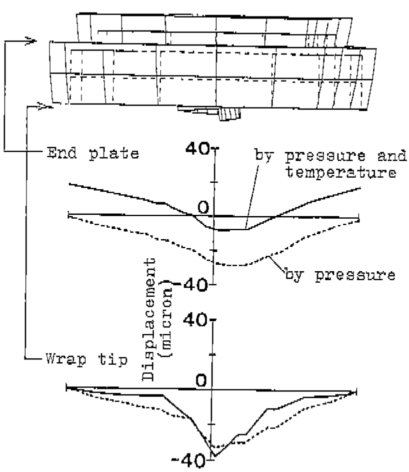


Fig. 7 Axial deformation of fixed scroll

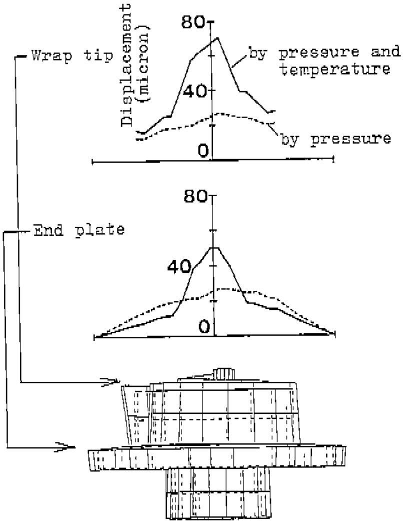


Fig. 8 Axial deformation of orbiting scroll

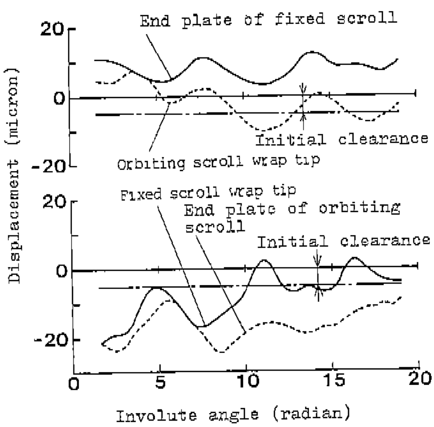


Fig. 9 Axial deformations of assembled scrolls

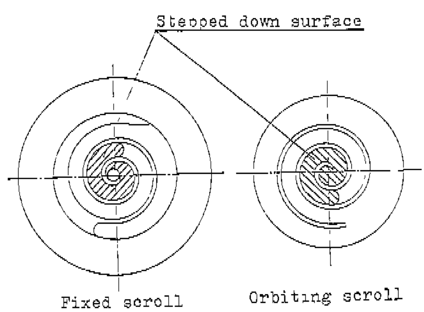


Fig. 10 Wrap shape for improving efficiency

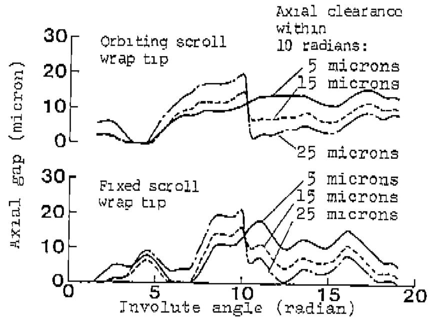


Fig. 11 Axitial gap with degree of stepped down clearance

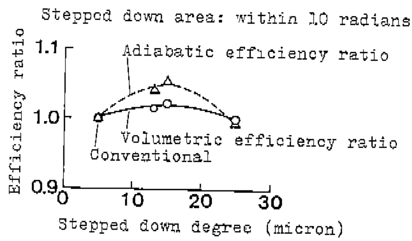


Fig. 12 Effect of stepped down degree change

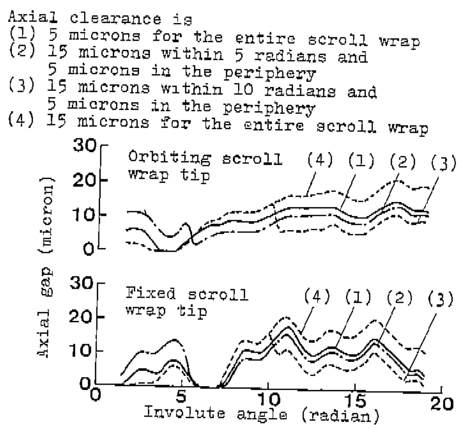


Fig. 13 Axial gap with degree of stepped down surface area

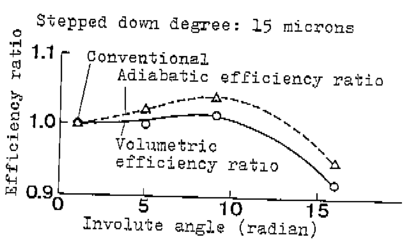


Fig. 14 Effect of stepped down area change