1992

Dynamic Axial Compliance to Reduce Friction Between Scroll Elements

J. J. Nieter
United Technologies Research Center

A. J. Marchese
United Technologies Research Center

R. B. L. DeBlois
United Technologies Research Center

Nieter, J. J.; Marchese, A. J.; and DeBlois, R. B. L., "Dynamic Axial Compliance to Reduce Friction Between Scroll Elements" (1992). *International Compressor Engineering Conference.* Paper 905.
https://docs.lib.purdue.edu/icec/905

DYNAMIC AXIAL COMPLIANCE TO REDUCE FRICTION BETWEEN SCROLL ELEMENTS

J. J. Nieter
Research Engineer

A. J. Marchese
Assistant Research Engineer

R. L. DeBlois
Senior Principal Engineer

United Technologies Research Center
411 Silver Lane, Mail Stop 129-19
East Hartford, CT 06108

ABSTRACT

One technique for minimizing tip leakage and providing axial compliance in scroll compressors is to pressurize the back side of an axially compliant orbiting scroll. This technique utilizes a pressurized back chamber which generates sufficient upward force to not only overcome the downward compression force, but to counteract the orbiting scroll overturning moment as well. Since such an approach must be designed for worst-case conditions, excessive friction forces may result over a portion of the compressor operating regime if designed to provide static (or relatively constant) axial compliance force at any given operating condition. However, analysis of the axial forces and overturning moment generated during the compression process show that both vary substantially during each orbit cycle. Thus, if the axial compliance force is made to vary in a similar manner, a lower average force, and therefore lower friction loss per revolution can be achieved. This paper describes an axial compliance mechanism that achieves this characteristic by utilizing two axial compliance chambers that are supplied with pressurized gas taken from the scroll compression chambers. It is shown that the sub-revolution dynamic response of the axial compliance mechanism can be optimized by proper sizing and positioning of the supply ports, delivery passages, and compliance chambers. Experimental test results have verified the predicted sub-revolution response from such an axial compliance mechanism.

NOMENCLATURE

F	Force
M	Moment
r	Radial axis
t	Tangential axis
z	Vertical axis

Subscripts

bc	Back chamber force
cs	Centrifugal force
o	Overturning moment
pa	Axial pressure force
pr	Radial pressure force
pt	Tangential pressure force
s	Seal force at wrap flanks
sbr	Radial scroll hub bearing force
sbt	Tangential scroll hub bearing force
T	Net thrust force

Superscripts

′	Indicates component in plane of view

INTRODUCTION

It is widely recognized that in order to achieve high efficiency with a scroll compressor, some form of compliance mechanism is required. This is because scroll compressor performance is very sensitive to leakage [1-3]. Two primary forms of leakage can occur in scroll displacement machines [3-6]: flank leakage through the radial clearances between the flanks of the mating scroll wraps, and tip leakage through the axial clearances between the ends of the scroll wraps and the mating scroll base plate. Due mainly to the fact that the effective total leakage path width for tip leakage is several times larger than for flank leakage, tip leakage is the more severe of the two [2-4,6]. Attempts to minimize this tip leakage by close-tolerance manufacturing techniques alone are not adequate due to varying wrap deformation over the range of compressor operation.

Axial compliance is an approach utilized in many scroll compressors to minimize tip leakage, usually in one of the following forms: tip seals[4,6], back pressurization of the orbiting scroll[1,3,5], or putting axial compliance in the "fixed" scroll[5]. Although tip seals are effective for sealing, they complicate machining requirements, are expensive, and create additional friction. Allowing the "fixed" scroll to move axially appears to be technically sound, however, it may involve a relatively complex mechanism.

The method of applying a compensating pressure force under the orbiting scroll is shown schematically in Fig. 1. In operation, an annular cavity is pressurized to create sufficient upward force to not only overcome the downward compression force, but to counteract the orbiting scroll overturning moment also. Thus, a net upward force maintains the orbiting scroll in close proximity to the fixed scroll to minimize tip leakage.

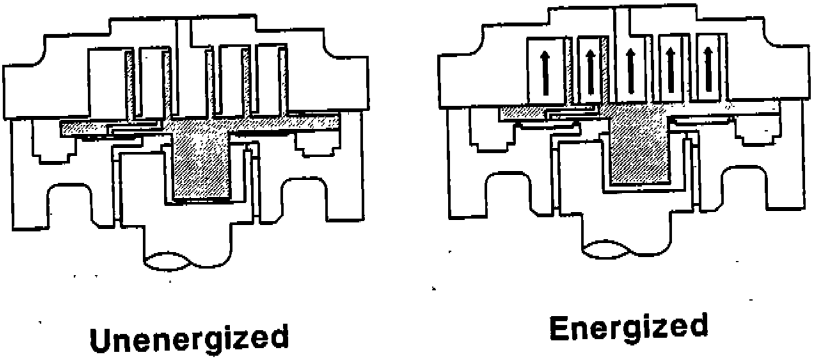

Figure 1. Axial Compliance Mechanism: Pressurized Back Chamber

Typically, the method of back pressuring the orbiting scroll to achieve axial compliance, stability, and minimize friction loss is achieved using a single back pressure chamber with no attempt made to tailor the net upward force on a sub-revolution basis[1,3,5]. Such a system must be designed for worst-case conditions and, as a consequence, excessive thrust force (and therefore friction) is experienced over a wide portion of the compressor operating range. The novel back pressuring method described in this paper has all the positive benefits of standard static back chamber designs, and in addition, minimizes friction losses by taking advantage of sub-revolution dynamics.

ANALYSIS

The major forces acting on an orbiting scroll are shown in Fig. 2 and include those generated from the orbiter motion (pressure and inertia forces) as well as those required to support the orbiter (reaction forces). Force and moment balances on the orbiting scroll and Oldham coupling result in many simultaneous equations which must be solved in order to obtain these reaction forces, as well as other unknowns, and are well documented in the literature[2,7-11]. As a consequence of the simultaneous forces acting on the orbiting scroll, there results a moment, referred to as the overturning moment[7], which may cause the orbiter to tip if not properly supported.

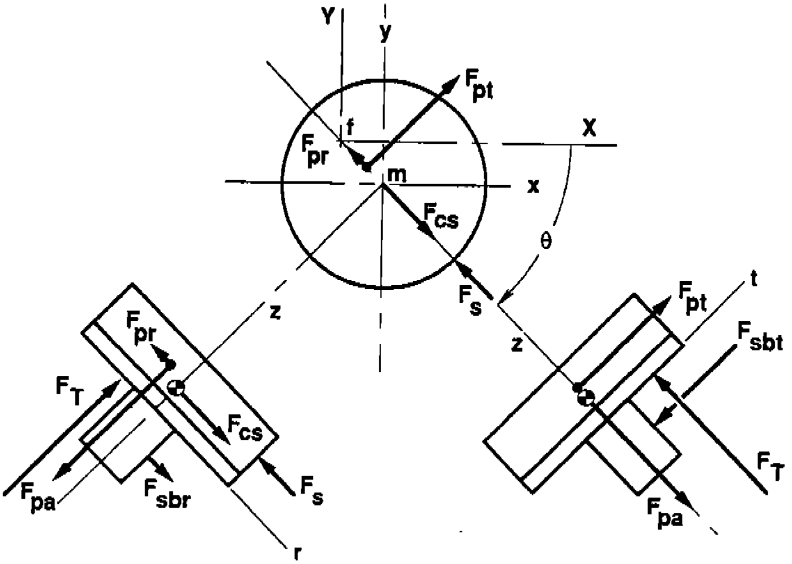

Figure 2. Scroll Orbiter Forces

The magnitude and radial location of the net axial thrust force F_T reacted at the thrust surface should provide adequate support to overcome this moment. The net axial thrust force shown in Fig. 2 is representative of that which would occur if no back pressure chamber were utilized below the orbiting scroll. This configuration is shown in Fig. 3 where the component of overturning moment in this plane is also shown. The t-z plane is used in Fig. 3 to show overturning moment because the resultant overturning moment is due primarily to the forces F_{pt} and F_{sbt}, as shown in the t-z plane in Fig. 2.

In the absence of the back pressure approach, as in Fig. 3, the orbiting scroll is forced down against the crankcase where a very high thrust load F_T and subsequent friction loss occurs. However, when the back pressure approach is used, the thrust surface occurs between the top of the orbiter base plate and the lower surface of the fixed scroll, as shown in Fig. 4. Additionally, a friction surface occurs between the seals and either the bottom of the orbiter base plate, or the crankcase, or both, depending on design. In practice, the friction between the seals and orbiter, or crankcase, is small

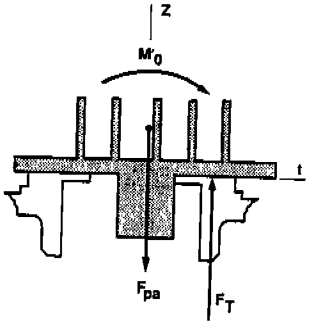

Figure 3. Scroll Orbiter Thrust Force

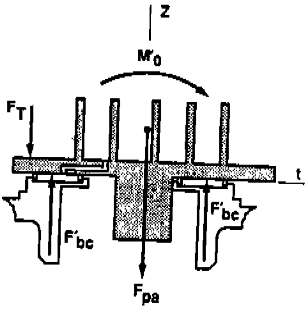

Figure 4. Scroll Orbiter Thrust Force

for proper seal design and material selection. Consequently, the back chamber approach results in reduced thrust load and friction loss well below what occurs without it, while minimizing tip leakage. Further, by minimizing the magnitude of the upward force due to back chamber pressure via proper selection of the supply port as well as positioning and sizing of the back chamber, the net thrust reaction force and friction at the thrust surface can be reduced even more.

A comprehensive scroll compressor simulation was used to analyze the back chamber pressure dynamics[10] and resulting dynamic forces reacted on the orbiting scroll[2,7-11]. This simulation enables one to predict the overturning moment, net thrust force, and other reaction forces acting on the orbiter as a function of orbit angle. An example of the instanteous variation in overturning moment on a sub-revolution basis is shown in Figs. 5a and 5b for an under-pressure and over-pressure condition respectively. As discussed above, current back chamber designs entail obtaining a net axial thrust force F_T sufficient to support the orbiter for the maximum overturning moment during each shaft revolution for the worst-case operating condition. The thrust force results from back pressuring the orbiter using gas as supplied from the high pressure compression pockets. An example of the gas pressure available in the compression pockets to do this is shown in Figs. 6a and 6b for an underpressure and overpressure condition respectively. Also shown in Figs. 6a and 6b are examples of the range of pressure during one orbit cycle (360 deg.) that can supply gas to the back chamber. It is apparent from Figs. 6a and 6b

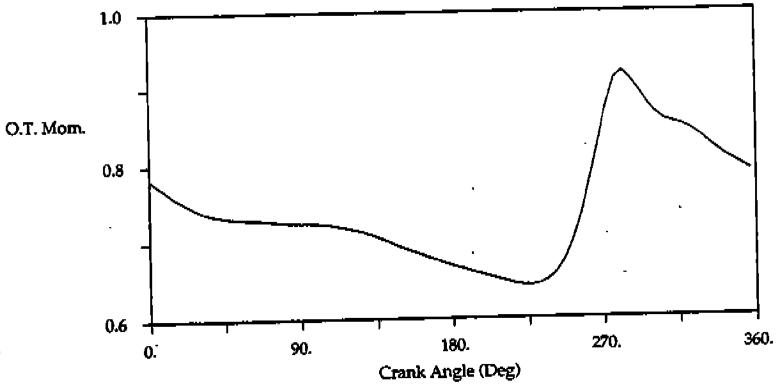

Figure 5a. Instantaneous Overturning Moment: Under-pressure Condition

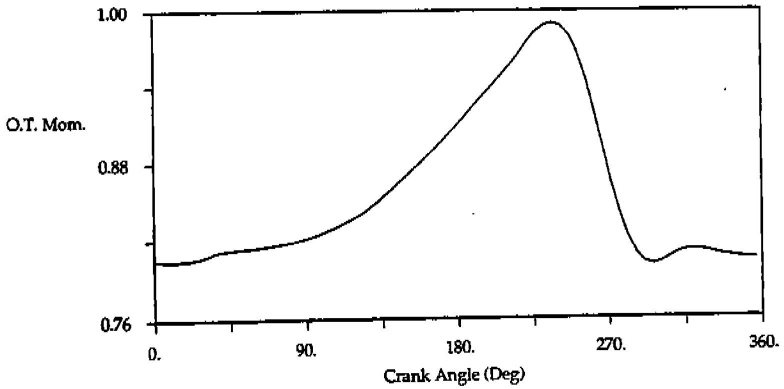

Figure 5b. Instantaneous Overturning Moment: Over-pressure Condition

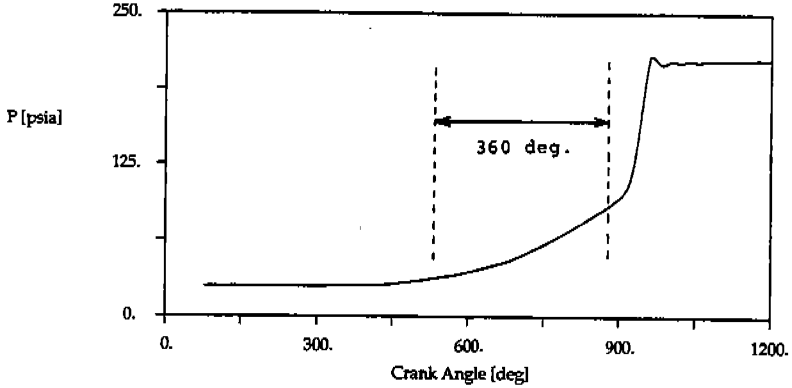

Figure 6a. Scroll Compression Process: Under-pressure Condition

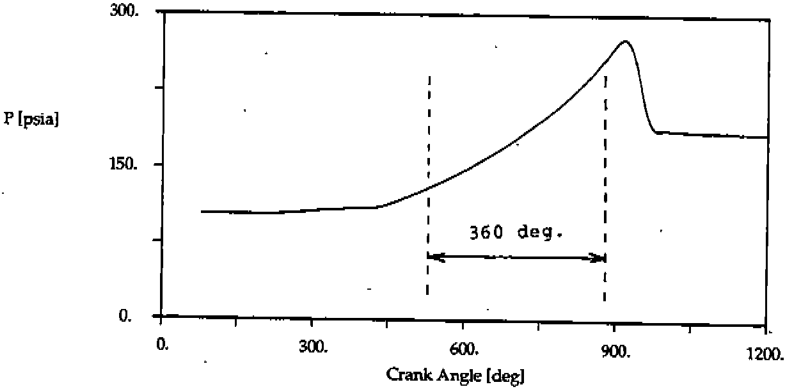

Figure 6b. Scroll Compression Process: Over-pressure Condition

that there is a dynamic character to the supply pressure during each cycle similar to the overturning moment. Thus, if this gas pressure is supplied to the back chamber in an appropriate manner, it is possible to obtain an axial pressure force on the back side of the orbiter which varies during each shaft revolution similar to the overturning moment and, thereby, effectively balancing the moment. Doing this results in a much lower average thrust reaction force per cycle than current static chamber designs and, consequently, lower friction loss.

To accomplish this dynamic thrust force characteristic, an axial compliance device can utilize multiple pressure chambers located at the back side of the orbiting scroll and whose supply pressure ports provide various degrees of damping[10]. By proper sizing and positioning of the supply ports, delivery lines, and back chambers, the pressure force acting on the back side of the orbiting scroll varies on a sub-revolution basis, increasing during portions of the cycle where added force is needed to counteract the increasing overturning moment.

A device with two annular back chambers and two supply pressure ports is shown schematically in Fig. 7. Each supply port supplies high pressure gas to a corresponding back chamber. The first back chamber, referred to as the static back chamber, has a relatively small port diameter and large chamber

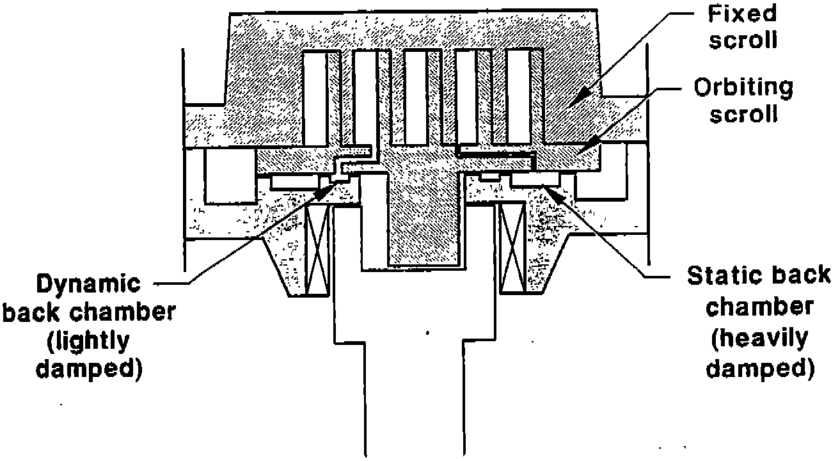

Figure 7. Dynamic Axial Compliance Approach

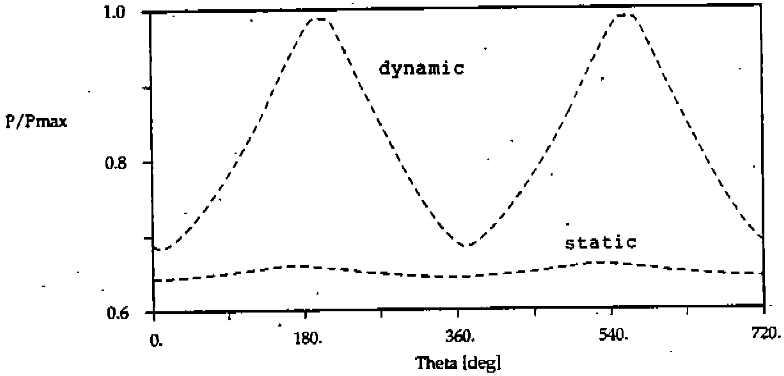

Figure 8a. Simulated Axial Compliance Pressures

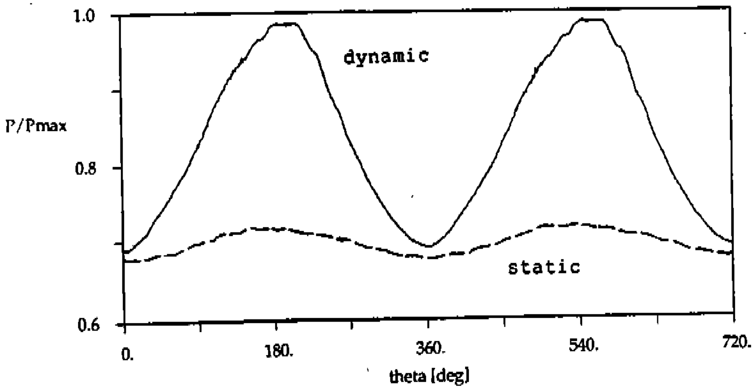

Figure 8b. Measured Axial Compliance Pressures

volume. The port diameter and chamber volume are sized to create sufficient damping such that pressure is nearly constant over the cycle as shown by the lower curve predicted in Fig. 8a. The second back chamber in this example, referred to as the dynamic back chamber, has a relatively large diameter supply port and small chamber volume. The chamber volume and port diameter in this case are sized to create very little damping so that the pressure in this chamber follows the compression process. The upper curve in Fig. 8a shows the predicted sub-revolution pressure variation in the dynamic back chamber.

Areas in the static and dynamic back chambers are sized to produce the desired total pressure force on the back side of the orbiter due to the shell, static back chamber, and dynamic back chamber pressures. As shown in Figs. 9a and 9b this resulting axial pressure force is a close approximation to the minimum force required on the back side of the orbiting scroll to counteract the overturning moment. Moreover, the resulting axial pressure force approximates the minimum required compliance force for under-pressure and over-pressure conditions. Any back-side pressure force which is greater than the minimum required force results in excess thrust load on the orbiting scroll and increased friction. An example of the reduction in net axial thrust force possible with this dynamic approach is shown in Fig. 10 where the net thrust force during one orbit cycle is shown for this approach as well as for the static pressure chamber approach.

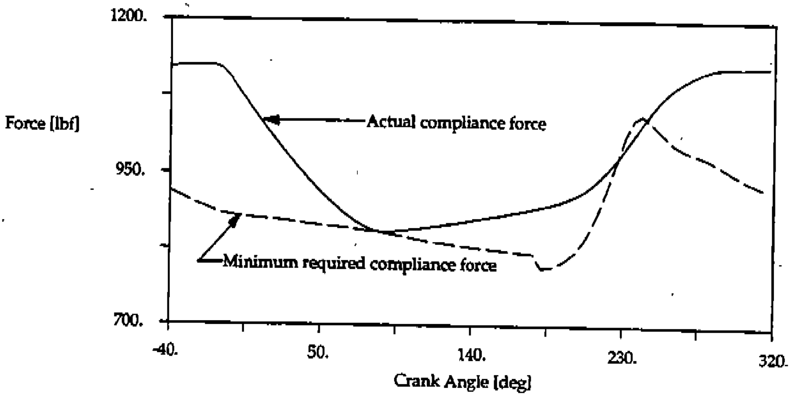

Figure 9a. Axial Compliance Force: Under-pressure Condition

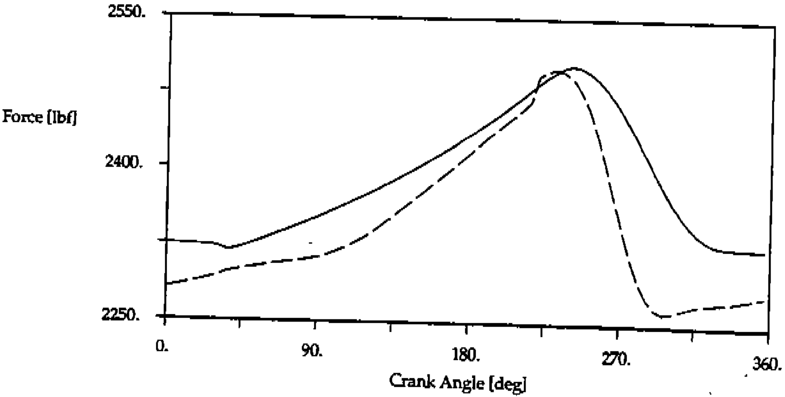

Figure 9b. Axial Compliance Force: Over-pressure Condition

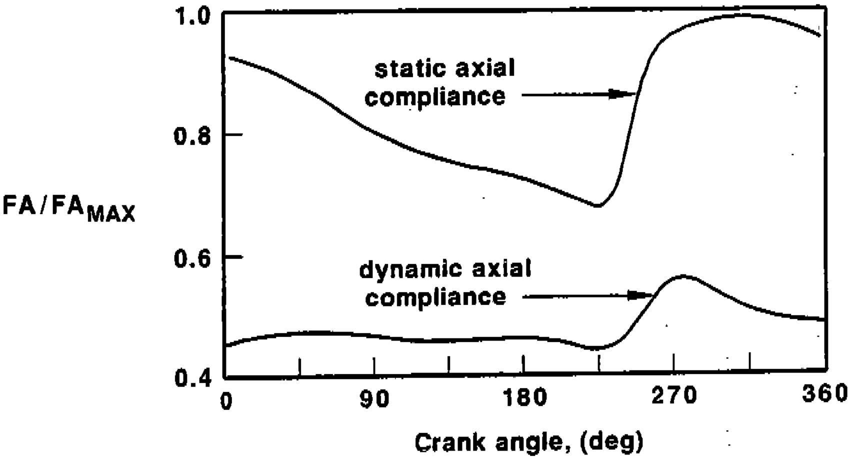

Figure 10. Net Axial Thrust Force on Orbiting Scroll

Since the dynamic pressures are achieved via mass flow between the compression chamber and compliance back chamber, the presence of the dynamic compliance chamber can potentially affect the scroll compression process adversely . Figure 11 compares simulated P-V diagrams for scroll compressors equipped with a standard static axial compliance chamber and a dynamic axial compliance chamber, respectively. This figure indicates that the dynamic axial compliance chamber can potentially result in increased compression power suggesting a trade-off between decreased friction and increased compression power. Fortunately, the dynamic pressures required can be achieved with little increase in compression power by minimizing the volume of the dynamic chamber.

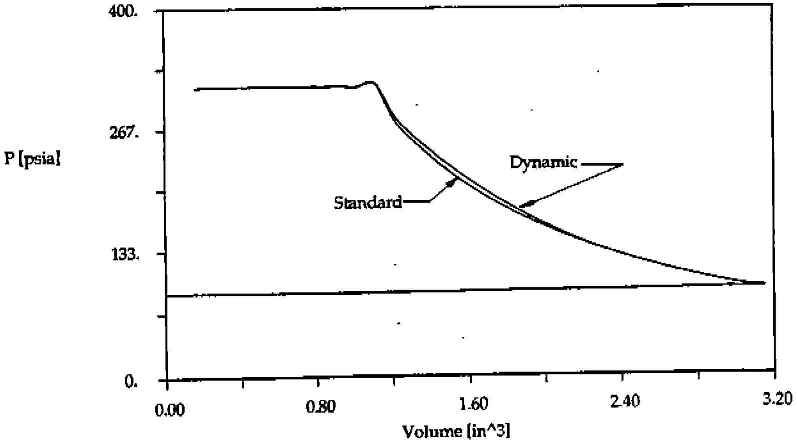

Figure 11. Simulated P-V Diagrams for Dynamic and Standard Axial Compliance

EXPERIMENTAL VERIFICATION

The comprehensive scroll compressor simulation described above is a useful design tool for many of the scroll compressor subsystems. It has been used extensively, for instance, in design of the "standard" static axial compliance systems. In these systems, the static and dynamic compliance chamber pressures predicted by the simulation program have been verified experimentally to within approximately five percent[10].

The dynamic axial compliance approach described in this paper, however, requires the dynamic chamber to undergo severe peak-to-peak pressure pulsations, on the order of 40% of the average. In order to verify that such pressure pulsations were achievable and accurately predicted, a dynamic axial compliance system was designed, installed, instrumented and tested in a laboratory scroll compressor. The laboratory test compressor chosen for these tests was a nominal 3 ton, low-side scroll compressor. The compressor was fabricated with a crankcase which had a removable upper surface that could be interchanged without requiring disassembly of the compressor drive train. This enabled the testing of various compliance chamber and seal configurations.

The compressor was instrumented with thermocouples to measure motor, bearing, and oil temperatures as well as suction gas superheat. Initially, a "standard" axial compliance system was installed into the laboratory test compressor. The compressor was attached to a desuperheater test stand equipped with suction and discharge pressure transducers and thermocouples which enabled accurate determination of operating condition. In this configuration, the compressor was tested at various operating conditions within the standard scroll compressor envelope while compressor power, mass flow rate, and temperatures were acquired using the laboratory data system described in Reference 12.

Having accurately measured the compressor performance and temperature characteristics with the standard axial compliance, the dynamic axial compliance design was installed. Both the static and dynamic chamber were instrumented with high-response pressure transducers. The compressor was then tested in this configuration at the same conditions as the baseline configuration. Figure 8b shows that the dynamic pressures measured experimentally were in good agreement with those predicted by the simulation program (Fig. 8a).

CONCLUSIONS

The resulting axial compliance forces generated by this approach closely approximate the minimum required necessary to counteract the overturning moment acting on the orbiting scroll. Moreover, this approach is equally effective for both over-pressure and under-pressure conditions. The result of this dynamic design is a scroll compressor with greatly reduced net axial thrust loads on the orbiting scroll compared with that in current static designs. In the example case discussed, approximately 45% reduction in the net axial thrust force can be achieved at nominal operating conditions. Any increase in indicated power due to the presence of the dynamic chamber can be minimized by minimizing the volume of the dynamic chamber.

REFERENCES

1. Tojo, K., et al., "A SCROLL COMPRESSOR FOR AIR CONDITIONERS", Proc. of the 1984 Intern. Compr. Eng. Conf. (Purdue), July 1984, pp. 496-503.

2. Hayano, M., et al., "PERFORMANCE ANALYSIS OF SCROLL COMPRESSOR FOR AIR CONDITIONERS", Proc. of the 1986 Intern. Compr. Eng. Conf. (Purdue), Aug. 1986, pp. 856-871.

3. Tojo, K., et al., "COMPUTER MODELING OF SCROLL COMPRESSOR WITH SELF ADJUSTING BACK-PRESSURE MECHANISM", Proc. of the 1986 Intern. Compr. Eng. Conf. (Purdue), Aug. 1986, pp. 872-886.

4. Inaba, T., et al., "A SCROLL COMPRESSOR WITH SEALING MEANS AND LOW PRESSURE SIDE SHELL", Proc. of the 1986 Intern. Compr. Eng. Conf. (Purdue), Aug. 1986, pp. 887-900.

5. Bush, J.W. and Elson, J.P., "SCROLL COMPRESSOR DESIGN CRITERIA FOR RESIDENTIAL AIR CONDITIONING AND HEAT PUMP APPLICATIONS PART I: MECHANICS", Proc. of the 1988 Intern. Compr. Eng. Conf. (Purdue), July 1988, pp. 83-92.

6. Yamamura, M., et al., "COMPACT TYPE SCROLL COMPRESSOR FOR AIR CONDITIONERS", Proc. of the 1990 Intern. Compr. Eng. Conf. (Purdue), July 1990, pp. 140-146.

7. Morishita, E., et al., "SCROLL COMPRESSOR ANALYTICAL MODEL", Proc. of the 1984 Intern. compr. Eng. Conf. (Purdue), July 1984, pp. 487-495.

8. Ishii, N., et al., "A STUDY ON DYNAMIC BEHAVIOR OF A SCROLL COMPRESSOR", Proc. of the 1986 Intern. Compr. Eng. Conf. (Purdue), Aug. 1986, pp. 901-916.

9. Hayano, M., et al., "AN ANALYSIS OF LOSSES IN SCROLL COMPRESSOR", Proc. of the 1988 Intern. Compr. Eng. Conf. (Purdue), July 1988, pp. 189-197.

10. Nieter, J.J. and Barito, T., "DYNAMICS OF COMPLIANCE MECHANISMS IN SCROLL COMPRESSORS PART I: AXIAL COMPLIANCE", Proc. of the 1990 Intern. Compr. Eng. Conf. (Purdue), July 1990, pp. 308-316.

11. Nieter, J.J., "DYNAMICS OF COMPLIANCE MECHANISMS IN SCROLL COMPRESSORS PART II: "RADIAL COMPLIANCE", Proc. of the 1990 Intern. Compr. Eng. Conf. (Purdue), July 1990, pp. 317-326.

12. Marchese, A. J., "DYNAMICS OF AN ORBITING SCROLL WITH AXIAL COMPLIANCE - PART 2: EXPERIMENTAL TECHNIQUES", Proc. of the 1992 Intern. Compr. Eng. Conf. (Purdue), July 1992.