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SCROLL COMPRESSOR WITH NO TIPPING MOMENT

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ABSTRACT

A scroll compressor concept is presented which has a very unique method of eliminating the inherent overturning or tipping moment associated with conventional orbiting scrolls. This design is a high-side compressor with axial compliance. An innovative gas discharge path from the involute effectively reduces the height of the assembly. A special offset is placed in the location of the involute, which enables an inherently large start angle to be reduced. A simple, yet useful computer model is used to optimize the overall design. Experimental test results are included which match the calculated result. Finally, a summary of the advantages and disadvantages with the concept are discussed.

NOMENCLATURE

\[ F_{ag} \] - Axial gas force
\[ F_{br} \] - Orbiting-Scroll baseplate thrust force
\[ F_{ac} \] - Axial compliance force from intermediate gas pressure
\[ F_{ob} \] - Orbiting-Scroll bearing force
\[ F_{pin} \] - Fixed-Scroll pin force
\[ F_{res} \] - Resultant (tangential + radial) gas force
\[ F_{tg} \] - Tip thrust force
\[ R_{bt} \] - Orbiting-Scroll baseplate thrust radius
\[ R_{tt} \] - Tip thrust radius
\[ Z_{t} \] - Tipping moment-arm

INTRODUCTION

From the beginning of scroll technology [1], there have been several major engineering challenges for scrolls to become a viable product for HVAC. First and foremost was the requirement of extremely accurate machining processes since most of today’s scroll compressors rely on metal-to-metal sealing of the compression pockets. Second, a method to accurately and reliably align and join the bearing housings and stator to a sheet metal container. However, the strategy of containing the inherent gas forces developed by the scroll set during compression has inspired the most innovative contributions in the past two decades. Therefore, the subject of this paper is a very unique scroll concept that totally eliminates the tipping moments that are due to these fundamental gas forces. This discussion is not intended to cover the design in great theoretical detail, but contribute to the spirit of advancing scroll technology.

BACKGROUND

The gas pressure forces developed within the scroll pair are well known to be axial, tangential, and radial. These orthogonal force-vectors lie in the x, y, and z planes in all scroll concepts. Each force varies in magnitude as the crank angle cycles through the 360 degrees of revolution. The generally large axial force is produced by the collection of pocket pressures acting on the respective floor areas of each scroll, pushing them apart and generating a large orbiting thrust against a member. The small radial force acts parallel to the lines of flank contact points, and the generally large tangential force acts normal to these lines. The radial force (along with centrifugal and drive-
angle forces, or selective interference) control the magnitude of the sealing force on the flank walls that form each pocket. The radial force is small compared to the other two, thus presenting a less significant problem for the design. The tangential force is produced by the collection of pocket pressures acting on the flank walls of each scroll, pushing each one horizontal in opposite directions. The resultant force \( F_{res} \) is a vector combination of tangential and radial, and controlling it efficiently is a major challenge in all scroll compressor designs.

First of all, tangential-gas force generates most of the radial-bearing loads. Since the force is produced by pressure on the flank walls, a shorter flank will reduce the magnitude. However, there are other significant factors in the design that optimize with taller flanks. For a given displacement, a taller flank requires a smaller pitch, less wrap length, and therefore less tip-leakage length per increment of capacity. Since tip-to-floor leakage is by far the key to scroll efficiency, a design with tall flanks can be superior in volumetric efficiency. Therefore, designs that can effectively tolerate high tangential forces have an inherent advantage.

Secondly, the major challenge due to tangential gas-force is not the radial-bearing force. The big difficulty is controlling the overturning, or tipping, moment created by the force couple of \( F_{res} \) and \( F_{ob} \). Figure-1 shows a force diagram of a typical scroll design (axially compliant orbiting-scroll) used in the residential and light-commercial air-conditioning industry. In this design, the fixed scroll is held rigid to the frame and a gas restoring force \( F_{ac} \) is applied under the center of the orbiting scroll [2]. Since the fixed scroll is held rigid, the only tipping moment is on the orbiting scroll \( F_{res}Z_1 \). It can be seen that the axial-compliance force required to insure stability of the scroll set must be large enough to: one oppose axial gas-force \( F_{ag} \), and two produce a counter moment with the outermost tip radius \( R_{ob} \). This criterion must be met to prevent the scroll set from axially separating at the peak of generated pressure forces.

In an axially compliant fixed-scroll design [2] (shown in Figure-2) the axially-free support for the fixed scroll can vertically located such that there is no net moment on the fixed scroll. Here again, the only tipping moment produced is on the orbiting scroll. But, the counter-tipping moment-arm \( (R_{bt} + R_{bt}) \) is greater, thus insuring stability with a smaller tip-thrust force \( F_{th} \). \( R_{bt} \) is due to the rigid thrust-bearing under the orbiting scroll.

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**Figure 1 - Axially Compliant Orbiting**

**Figure 2 - Axially Compliant Fixed**
DESCRIPTION OF NEW CONCEPT

The arrangement of the subject scroll compressor is shown in Figure-3. It has similarities to Lepsi's "Involute Pump" [3]. Significant differences with this arrangement compared to ones discussed earlier are; one, the orientation of the fixed and orbiting scroll set is reversed. Two, the main bearing is uniquely combined with the fixed-scroll (Figure-4) such that the drive shaft passes through the scroll. And three, the orbiting scroll (Figure-5) houses the eccentric-shaft drive-pin and bushing in the center section of the involute wrap. This third feature is the most significant feature of this concept. With $F_{fe}$ and $F_{ob}$ coplanar, the tipping moment, typical of the other designs, is eliminated.

The axial compliance force ($F_{ac}$) required to keep the scroll set together is generated by gas pressure on top of the orbiting scroll. The area on which this pressure acts is defined by a single face-seal located between the upper shell assembly and the orbiting scroll. Like most gas-loaded scroll designs, this one utilizes an optimized combination of discharge and intermediate gas pressure determined by the precise location of a vent hole drilled through the floor of the orbiting scroll.

Also notable, the concept is a high-side machine with the motor and major volume of the shell open to the discharge gas pressure. Since two bearings are located in the discharge section of the scroll set, high side is the most practical layout. Oil management and lubrication would not be practical if the design were low side. Another interesting feature of this compressor is the discharge technique. Suction gas enters the compressor into the chamber above the fixed-scroll/crankcase and is pulled into the scroll pockets. As the pockets finally reach the point of discharge, the gas leaves the compression pockets via an annulus surrounding the shaft. The gas then exits through a plurality of holes extending radially downward and outward toward the stator windings. The discharge gas flow provides cooling for the motor, after which any entrained oil is separated from the gas before it exits the compressor.

![Figure 3 - Compressor Layout](image-url)
DESIGN DETAILS

The design objective was to produce 61,000 Btu/hr at ARI (45/130/20/15°F) utilizing a motor diameter of 169-mm (6.656-in). The fact that the shaft passes through each scroll involute created a unique challenge in the choices of generating radius, start angle, end angle, and flank height. In addition to the capacity requirement, the design volume-reduction ratio proved to be quite challenging due to the large start angle required by the bearing bore located in the involute center.

The smallest possible start angle was required to keep the overall package diameter within specifications. In order to maintain a volume-reduction ratio of at least 2.05, a special technique was discovered to further condense the package. In both the orbiting and fixed scrolls, the involute was shifted off the center of the part. The start angle was reduced 93° by off-centering the involutes. Each iteration involved changing the offset distance and offset phase-angle along with other key parameters of involute geometry.

It was obvious that machining the long involute wraps would be a significant negative in this design concept. In the mathematical design-model used, the end mill diameter is considered one of the secondary parameters. Such things as resulting slot width, from pitch and wall thickness, and minimum radii in the discharge area were considered. The long involute length inherent to this design could be significantly reduced with hybrid-wrap technology [4]. The resulting shorter lengths would minimize machining length and internal leakage.

Table-1 contains a comparison of the forces for the three arrangements discussed in this paper. A force diagram for this concept is shown in Figure-6. The significance of the arrangement is now obvious - the moment arm \( (Z_1) \) now has a value of zero. Note even with the higher gas forces, the elimination of the tipping moment for this compressor provided the ability to reduce overall frictional losses.

<table>
<thead>
<tr>
<th>Force Description</th>
<th>Symbol</th>
<th>Units</th>
<th>Compliant Orbiting</th>
<th>Compliant Fixed</th>
<th>Zero Tipping Moment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tipping Moment Arm</td>
<td>( Z_1 )</td>
<td>in</td>
<td>1.74</td>
<td>1.74</td>
<td>0</td>
</tr>
<tr>
<td>Force, Resultant Gas</td>
<td>( F_{\text{res}} )</td>
<td>lb(_f)</td>
<td>643</td>
<td>643</td>
<td>873</td>
</tr>
<tr>
<td>Tipping Moment</td>
<td></td>
<td>( \text{in-lb}_f )</td>
<td>1119</td>
<td>1119</td>
<td>0</td>
</tr>
<tr>
<td>Axial Gas</td>
<td>( F_{\text{ag}} )</td>
<td>lb(_f)</td>
<td>655</td>
<td>655</td>
<td>1238</td>
</tr>
<tr>
<td>Axial Compliance</td>
<td>( F_{\text{ac}} )</td>
<td>lb(_f)</td>
<td>1426</td>
<td>961</td>
<td>1478</td>
</tr>
<tr>
<td>Tip Thrust</td>
<td>( F_{\text{t}} )</td>
<td>lb(_f)</td>
<td>771</td>
<td>306</td>
<td>240</td>
</tr>
<tr>
<td>Thrust Bearing</td>
<td>( F_{\text{th}} )</td>
<td>lb(_f)</td>
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<td>961</td>
<td>222</td>
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<tr>
<td>Friction Power</td>
<td></td>
<td>W</td>
<td>461</td>
<td>512</td>
<td>345</td>
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</table>
EXPERIMENTAL RESULTS

Much of the basic hardware for the scroll concept was available from other experimental designs. The scroll set, Oldham coupling, crankshaft, and seal member were fabricated. A fixed-throw eccentric-journal was calculated and ground to provide an optimum flank clearance. Additionally, a special radial compliant slider block was also designed for evaluation. Although, the limited space for the slider-block mechanism made the eccentric-shaft pin hazardously small. Regardless, both designs were built and tested. During the span of development with this concept, (24) laboratory tests were conducted. These tests involved iterations of the scroll involute geometry, oil separation methods, crankshaft, intermediate pressure values and seal sizes, and discharge porting changes. The final test results of this effort are shown in Table-2. Below is a summary of test conclusions:

- The internal leakage across the flank tips was believed to be responsible for the excessive power versus calculated. The long lengths of the involutes require closer height-tolerance between the scroll pair as compared to shorter length designs.
- The seal size could be reduced, thereby reducing the friction power.
- Controlling external oil-circulation rate of the compressor was challenging.
- Further performance improvements would be possible by optimizing the crank eccentric distance (orbit radius).

<table>
<thead>
<tr>
<th>Condition</th>
<th>Capacity (Btu/hr)</th>
<th>Power (W)</th>
<th>EER (Btu/W-hr)</th>
<th>Dis. Temp (°F)</th>
<th>Oil Circ. (%)</th>
<th>Sound Power (dBA)</th>
</tr>
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<tbody>
<tr>
<td>45/130/20/15</td>
<td>61257</td>
<td>5566</td>
<td>11.01</td>
<td>204.2</td>
<td>1.1</td>
<td>69.3</td>
</tr>
<tr>
<td>45/100/20/15</td>
<td>71880</td>
<td>3762</td>
<td>19.11</td>
<td>159.5</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
EVALUATION OF THE DESIGN CONCEPT

There are a number of potential advantages with this design, as well as significant challenges. Below is a summary of the strong points:

- Material cost reduction of at least 8% due to fewer parts.
- Overall compressor height reduction of at least 13%.
- Very favorable sound level is due to the small orbit radius, fixed-throw shaft, internal oil- and high side configuration.
- The fixed eccentric versus radial compliance is more practical with this design. The bearing bores and involutes can be machined at the same time, producing excellent position control with minimal effort.
- Controlled oil injection can be used to minimize internal gas leakage in the scroll set.

The significant disadvantages are:

- Approximately 90% longer involute machining length with the existing design.
- Higher bearing loads.

CONCLUSIONS

The work discussed in this paper presented a unique scroll-compressor concept. While it is not yet positioned for production development, it does offer a very creative solution to the old problem of containing large tipping moments. Elimination of tipping moments provided a means to design for much lower frictional power losses. The entire development work covered less than six months, and was discontinued due to more immediate priorities. Because the involute geometry incorporated more than three revolutions, the next logical step would be to utilize hybrid wraps. Hybrid wraps could significantly reduce the machining wrap-length while improving overall compressor efficiency.

REFERENCES