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CALCULATION OF OPTIMAL VALUE OF TAPER FOR THE DRIVE PIN OF THE SCROLL COMPRESSOR CRANKSHAFT

by

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ABSTRACT

The following paper calculates the optimal value of taper for a crankshaft drive pin of the scroll compressor such that under several load conditions encountering an edge-loaded situation is minimized. Load distributions and forces are calculated using finite element analysis software and the results are used with some mathematical analysis to arrive at an optimal value of taper.

INTRODUCTION

Unlike piston-type compressors which by comparison to scroll compressors are well-understood, there are many investigative analyses that need to be performed to design scroll compressors. In particular, finite element analysis is one option that can be used with much success. In the present instance, finite element is used to model a nonlinear drive pin contact problem. Below the problem is stated, the finite element model is outlined and results are discussed. Finally the taper calculation is performed and a conclusion reached.

PROBLEM

In the present case with reference to Figure 1 under maximum load conditions, there is visible wear on the sleeve bearing and drive pin. It is postulated that the situation can be relieved by tapering the pin a certain amount, so that in the deflected state the resulting deformation would be more favorable and avoid the edge-loaded situation already described. The purpose of the present study is to more fully understand the loading distribution phenomena and then to calculate an optimal value of taper which will eradicate the edge loading and wear problems.

FINITE ELEMENT MODEL

The problem may be viewed simplistically as in Figure 2, that is as two cantilevers opposing one another, in which the lower one has a linear taper. Figure 3 shows the model in more detail. The complete crank shaft has been modeled and the hub is modeled as a cantilever with a rolling support in the Y direction. Gap elements exist between the hub and drive pin which are modeled with beam-column finite elements [1]. The moment of inertia of the crank shaft and drive pin are complicated by the fact that there is an oil feed hole, linearly displaced from the center line in order to facilitate centrifugal pumping of the oil. This passageway must be accounted for, especially in the pin. The force exerted on the pin is applied as shown in Figure 3. This represents the maximum force required to compress the refrigerant gas during a cycle.

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RESULTS OF THE FINITE ELEMENT MODEL

Using Ansys finite element software, the model has been run for five values of taper, (.005, .001, .00125, .002) where the dimensions are in inches. For each value of taper there are five load conditions, (300, 700, 1000, 1300 and 1700) where the dimensions are in pounds. In particular 1000 and 1700 pounds represent important maximum load conditions, during which survivability is an issue. Having run these load cases for all values of taper, the results are displayed in Figures 4 through 8.

Observing Figure 4, it is seen that for .0005 inches of taper, edge loading always occurs. The situation would be worse for no taper and agrees with observation of worn parts. At the other end of the spectrum for .002 inches of taper it is observed that edge loading occurs not now in the inner edge of the pin but always at its exterior end, as illustrated in Figure 7, especially for lower loads of 300 and 700 pounds. An observation of Figures 5 and 6 shows loadings which are between these two extremes. Figure 8 also shows the moment at the pin/shaft interface for various loadings and tapers. It is not possible to choose any arbitrarily large value of taper to eradicate the edge loading problem, because the resulting high moments will cause the shaft to become deformed and that will effect bearing performance. As would be expected the bending moment increases with increasing taper. Also large moments increase the likelihood of a fatigue crack at the fillet where the pin joins the shaft.

Therefore from observation of the data, and an understanding of the effect of taper on both edge loading and moment, the problem then is to calculate an optimal value of taper. This is done in the next section.

CALCULATION OF THE OPTIMAL VALUE OF TAPER

It is necessary to quantify the types of loading in terms of their desirability. Clearly it is necessary to avoid an edge loaded situation, where the loading is weighted to one side or the other as indicated in Figures 4 and 7. A convenient measure or indication of an edge loaded situation is calculated by using standard deviation. Such a value is always positive. Also low values of standard deviation indicate progressively less one-sided pressures. A standard deviation of zero could only be achieved in fact, if the pressure was absolutely uniform across the pin. One of our objectives then is to search for small values of the standard deviation of the gap forces obtained from the finite element analysis.

Similarly it is decided to use the moment at the pin/shaft interface as another criteria for the reasons discussed above. The loads of 1000 lbs. and 1700 lbs. are chosen for the calculation as they are the predominant maximum load conditions and are larger than the force of 300 lbs. which the pin would have to withstand under normal running conditions.
Therefore for each value of taper there are four values to obtain, the standard deviations of 1000 lbs. and 1700 lbs. and the moments at the pin/shaft interface for each of these loads. As there are five values of taper, then four vectors are formed:

\[
\sigma_1 = \begin{bmatrix}
150.374 \\
56.5934 \\
43.9429 \\
56.0521 \\
88.6288 \\
\end{bmatrix}
\]

\[
\sigma_2 = \begin{bmatrix}
292.983 \\
191.661 \\
143.571 \\
100.906 \\
78.319 \\
\end{bmatrix}
\]

\[
M_1 = \begin{bmatrix}
268 \\
254 \\
546 \\
634 \\
788 \\
\end{bmatrix}
\]

\[
M_2 = \begin{bmatrix}
390 \\
574 \\
666 \\
760 \\
944 \\
\end{bmatrix}
\]

\(\sigma_1\) and \(\sigma_2\) are the standard deviations of the gap forces for 1000 lbs. and 1700 lbs. respectively. \(M_1\) and \(M_2\) are the moment for 1000 lbs. and 1700 lbs. respectively. It is necessary to normalize these vectors, to give each the same importance.

\[
\sigma_1 = \frac{\sigma_1}{|\sigma_1|}
\]

\[
\sigma_2 = \frac{\sigma_2}{|\sigma_2|}
\]

\[
M_1 = \frac{M_1}{|M_1|}
\]

\[
M_2 = \frac{M_2}{|M_2|}
\]

The first row in each of these vectors represents a value of the quantities concerned for a certain value of taper. Looking at the second row, there is another combination of the four variables for a new value of taper. These four variables represent a four dimensional space. We could theoretically find infinitely many points that would be joined by a smooth curve. As we vary the value of taper we traverse this four space along the curve. For a certain value of taper we will be closer to the origin than at any other point on an interpolated curve connecting these points. This will represent small or minimal values of the quantities concerned, which is what we are trying to achieve. A useful way to calculate this minimal point is to measure the distance from the origin to these points and to then plot this as a function of taper. With this new graph it is possible to find the minimum.
Calculate a new vector representing the distance from the origin to each of the five points held in the vectors above.

\[ \delta_i = \sqrt{a1^2 + a2^2 + b1^2 + b2^2}, \quad i = 1 \text{ to } 5 \]  

(9)

At this point we have five points in the X-Y plane and a quartic can be fit through them. It is required to determine the coefficients \( a, b, c, d, \) and \( e \) from equation 10.

\[ f(x) = a x^4 + b x^3 + c x^2 + d x + e \]  

(10)

Taking each point of the \( \delta \) vector in equation 9, with its corresponding value of taper we substitute into equation 10, and solve the resulting set of five equations in five unknowns. This is illustrated in equation 11.

\[
\begin{bmatrix}
a \\
b \\
c \\
d \\
e
\end{bmatrix} =
\begin{bmatrix}
.005^4 & .0005^3 & .0005^2 & .0005 & 1 \\
.001^4 & .001^3 & .001^2 & .001 & 1 \\
.00125^4 & .00125^3 & .00125^2 & .00125 & 1 \\
.0015^4 & .0015^3 & .0015^2 & .0015 & 1 \\
.002^4 & .002^3 & .002^2 & .002 & 1
\end{bmatrix} \cdot \delta \]  

(11)

Figure 9 shows a plot of equation 10, with the values calculated in equation 11. Also shown is the derivative which intersects the Y axis at .0011 inches. This is the value of taper which is chosen to minimize the variables discussed above.

CONCLUSION

A finite element model has been developed which shows agreement with the observed wear on the drive pin and journal bearing of the orbiting scroll hub. By inspection of the graphs of the gap forces between the pin and the hub a greater understanding of the role and magnitude of taper is obtained. The graphs suggest that some value of taper should be employed between .0005 and .002 inches. The optimal value of taper for this geometry is calculated to be .0011 inches. The calculated value of taper has been found to be similar to the experimentally determined taper.

Further studies of taper design could include oil film effects and a more accurate hub deflection by including the flexibility of the scroll base plate.

There is a practical limit to the amount of taper that can be utilized. Taper helps to compensate for the drive pin and scroll hub flexibilities and large forces involved, however there is a practical limit to how much taper can rectify the situation. For too large a value of force or too flexible a drive pin, it would be necessary to use a very large value of taper. The result of this would mean that during normal running at lower loads higher bending moments would occur at the pin fillet which might cause the origination of fatigue cracks. It is important to take the appropriate steps using finite element analysis and experimental methods to investigate the fatigue strength in this area.

REFERENCES

FIGURE 1  EXPLODED DRAWING OF RUNNING GEAR ASSEMBLY

FIGURE 2  SIMPLISTIC REPRESENTATION OF PROBLEM
FIGURE 3 CLOSE-UP OF DRIVE-PIN

FIGURE 4 FORCES ON PIN FOR .0005 INCHES OF TAPER
FIGURE 5 FORCES ON PIN FOR .001 INCHES OF TAPER

FIGURE 6 FORCES ON PIN FOR .0015 INCHES OF TAPER
FIGURE 7 FORCES ON PIN FOR .002 INCHES OF TAPER

FIGURE 8 MOMENT AT PIN/SHAFT INTERFACE
FIGURE 9  GRAPH OF EQUATION 10 AND ITS DERIVATIVE