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Optimization of Scroll Compressor Performance with Manufacturing Capability and Reliability Constraints

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ABSTRACT

This paper reviews an approach to the problem of selecting scroll geometry for optimum performance under conditions of defined manufacturing capability and reliability constraints. The focus is on definition of the meshing of fixed and orbiting scrolls. Flank gap and flank contact force characteristics are determined from Monte Carlo analyses of compressor assemblies. Monte Carlo analyses using a thermodynamic simulation are then carried out to compute the combined effect of leakage and flank friction on overall compressor performance and performance variation. The performance characteristics and reliability constraints are then used to define the optimum configuration for the specific manufacturing capability set used in the analyses.

1. INTRODUCTION

Scroll flank meshing characteristics – the minimum clearance between flanks of the fixed and orbiting scrolls or the force applied when flanks are in contact – affect three critical user-sensible compressor characteristics: performance, reliability and noise. Involute design is the process of selecting proper dimensions for all features in the stack-up of parts that create the mesh. This must be carried out with consideration of manufacturing process capabilities and load induced changes in shape and position of the parts. Design objectives are to maximize performance, minimize noise and minimize cost with the constraint of meeting reliability targets. The process also leads to identification of the most critical factors of the design and the levels to which they must be controlled to achieve the design results.

Cost and reliability benefit from application of the simplest design. The study reported here is applied to the design of a fixed-throw compressor, minimizing the number of parts in the running gear. Performance is evaluated at the ARI rating condition of 45°F (7.2°C) saturated suction temperature, 130°F (54.4°C) saturated discharge temperature with 20°F (11.1°C) suction superheat.

Performance is determined using a comprehensive thermodynamic simulation of the scroll compressor. The model accounts for losses in all components of the compressor, including losses arising from component interactions, e.g. heat transfer effects. Reliability of the involute set is assessed based on a limiting flank contact force. When assemblies analyzed at load are found to have an average contact force above this level, they are counted as failed. Issues related to noise are not included in the study reported here.

Assessments are carried out for 500 running gear assemblies using the Monte Carlo analysis technique. Independent variables in this study are the crankshaft throw (the offset of the orbiting scroll relative to the fixed scroll) and manufacturing process capabilities for each feature affecting the running gear alignment. One study is carried out with process capabilities fixed while varying the throw. A second study varies the manufacturing process capability with the throw selected to maximize performance while satisfying the reliability constraint.

The two specialized assembly models used in the analysis are introduced in Section 2. Results of the assembly model Monte Carlos, which are inputs to the performance analysis, are reviewed in this section. Section 3 is a review of the performance analyses and shows the results of the studies mentioned in the previous paragraph. Concluding remarks are offered in Section 4.

2. ASSEMBLY MODELING

The overall process for flank mesh analysis relates the effect of the most basic design decisions, feature dimensions and tolerances, to the important customer-sensible characteristics of the compressor. Moving from one end of this chain to the other requires combination of the features into subassemblies, subsystems and finally the full compressor assembly. This process is illustrated in Figure 1.

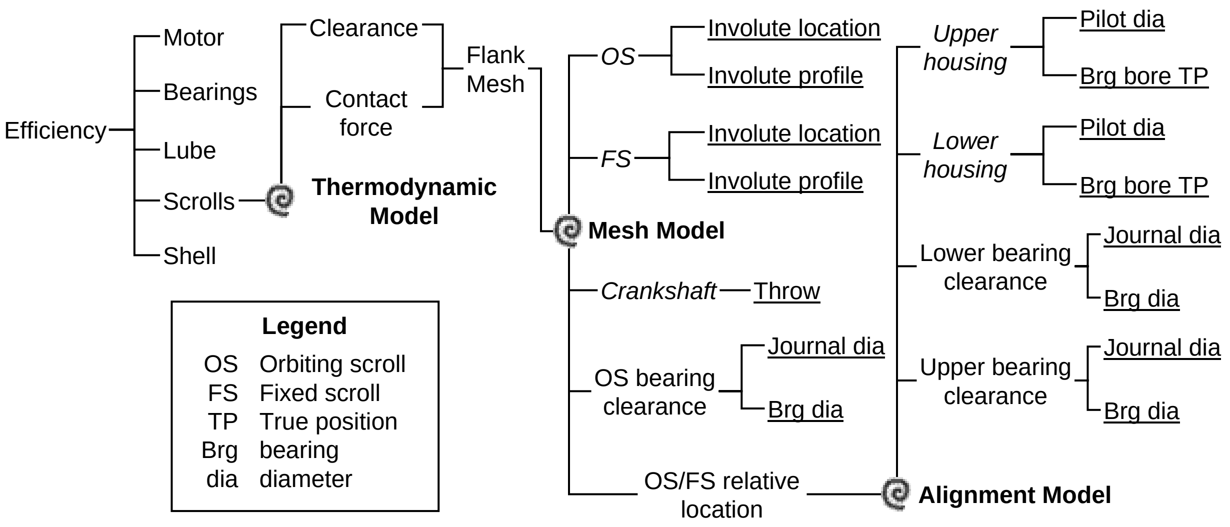

Figure 1. Chain from basic features to operating efficiency

In the figure, the right hand side shows the more basic information, individual features (which are underlined) and parts (in *italics*). Moving to the left, these basic elements appear in characteristics of assemblies, such as bearing clearance and further as factors affecting the measured output of the running compressor, in this case the compressor efficiency. Moving from features to assemblies and from characteristics of the assembly to performance requires analysis by the “transfer functions”, noted in the figure with **bold text** and the 🌀 symbol.

The analytical assembly of the involutes for the purpose of analyzing the meshing characteristics is carried out in two steps using the Alignment and Mesh Model transfer functions. These models are discussed in Sections 2.1 and 2.2. In these sections, the models themselves are only briefly described, as the primary purpose of this report is to illustrate the use of the analyses, the type of information and the way that this information generated is used to define and control the design. Output from the Mesh Model is used as an input to the Thermodynamic Model, which is discussed in Section 3.

2.1 Alignment Model

Shaft alignment is an issue for bearing design and the positioning of the orbiting scroll relative to the fixed. The latter factor is of interest for the meshing analysis. The Alignment Model is an analysis of the assembly based solely on geometric considerations. Nine separate inputs defining size and position of key elements in the assembly are used in the analysis. Mean and variation characteristics of these features are input. The model computes distributions of bearing clearance and location of the drive pin are output which are then used as input for the mesh analysis.

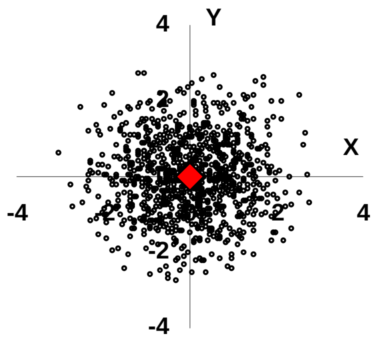

Figure 2. Crank pin location

Typical solution results are shown in Figure 2. In this case, X and Y are the Cartesian coordinates of the location of the drive pin’s (the journal of the orbiting scroll bearing) orbit center that positions the orbiting scroll relative to the fixed scroll. The nominal value is zero and deviations in X and Y are shown normalized to the standard deviation of the location variations. In this example, all of the features in the stack-up of this position result are assumed to have variations that conform to a normal distribution with mean values equal to the nominals and standard deviations consistent with a process capability (Cpk) of 1. The resultant distributions of the X and Y locations of the crank pin are also normal.

The variation in location is due to misalignment of the main bearings supporting the shaft caused by variations in dimensions that determine the relative positions of the lower bearing to the upper bearing. The significance of shaft alignment and the importance of manufacturing processes that control variation are pointed out by Ginies and Ancel (2003).

Key assumptions in the Alignment Model are that the shaft is straight and its centerline passes through the geometric centers of the lower and upper bearings. Bearing clearances are computed in the Alignment Model, but do not affect the alignment output. The effect of bearing clearance is dealt with in the Mesh Model, discussed in Section 2.2.

2.2 Mesh Model

Flank mesh characteristics of clearance and contact force are determined using the Mesh Model. This program accepts geometric and operational factors as input and computes the mesh under conditions of operating loads and speeds. This model computes the two elements of flank mesh which are subsequently used to evaluate compressor performance: clearance and contact force.

The effect of operating loads is included in the calculations. The Mesh Model includes a sophisticated film thickness analysis for the journal bearings used in the compressor. Bearing clearances are taken from the Alignment Model and used to compute minimum film thickness. The magnitude and direction of the minimum film thickness locates the shaft in the bearing so that the actual position during operation determines the final location of the drive pin. Similar analysis is carried out for orbiting scroll bearing. The final position of the orbiting scroll relative to the fixed is then based on the position of the pin (the journal of the orbiting scroll bearing), which is determined by the geometry of the upper and lower shaft bearings and the minimum film thicknesses of each, and the magnitude and location of the minimum film thickness of the orbiting scroll bearing.

A thermal analysis model has been developed for the scroll compressor. The model uses information generated by the Thermodynamic Model to define temperature and heat transfer boundary conditions for application to finite element (FE) models of the fixed and orbiting scrolls. Deformations calculated in the FE analyses are then transferred to the Mesh Model. The operating-load-induced deformations play a large role in the mesh results. The target is a near-zero clearance. As can be seen in Figure 3, this must be accomplished with parts whose dimensions change by 0.0014 inches (35.5 micrometers [μm]) and where the variation of this change within the part itself is of the same magnitude.

The part deformation, which does not vary, is added to the manufacturing variation for each trial assembly in the Monte Carlo analysis. This requires specification of process capabilities for variation in the shape of the involute independent of its location on the part and, separately, the location of the involute relative to datums that position the fixed and orbiting scrolls relative to each other.

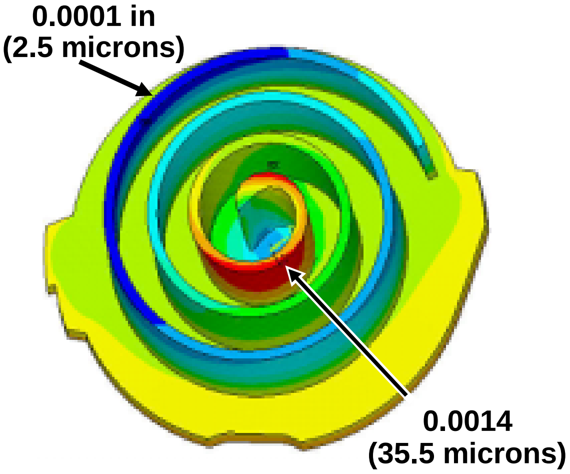

Figure 3. Computed distortion of the orbiting scroll due to thermal loads

The Mesh Model evaluates conditions at the minimum-distance points between the fixed and orbiting scrolls at 30 steps over one complete revolution of the crankshaft. Perfect involutes positioned at their theoretical orbit radius (crank throw = involute design orbit radius) with no deformation or rotation of one relative to the other will exhibit a mesh characteristic of zero clearance and zero contact force at all crank angles. If the crank throw is set to a value below the orbit radius, the mesh characteristic will be one of uniform clearance equal to the difference in the throw and the orbit radius. Increasing the throw to a value above the orbit radius complicates things a bit, but the end result is a mesh characteristic of uniformly zero clearance with contact force between the flanks.

When manufacturing variation and thermal and pressure loads are introduced, the meshing characteristics become more complex. These perturbations result in a variation in the meshing as a minimum clearance point progresses from the outer end of the involute to the inner. Due to the non-uniformity of the parts, there may be clearance at some points while other points are in contact, resulting in a net contact force.

The first of the two studies reported here is one in which manufacturing capabilities for all features used in the model are defined by a Cpk = 1. The crankshaft offset or throw is varied in four steps from a relatively small value, where one would expect to see assemblies with high flank clearance, to relatively large value with lower flank clearances. This exercise will result in a relationship between the clearance and throw, the smaller clearances being better for performance as internal leakage is reduced.

Results of this analysis are shown in Figure 4. Here, the distributions of flank clearance for the four levels of throw are compared. The chart legend shows the mean and standard deviation values for flank clearance.

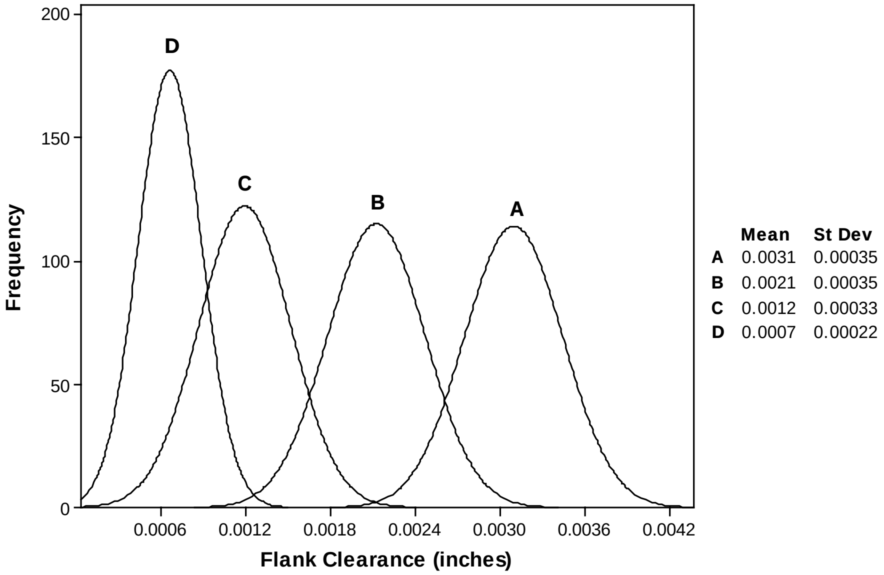

Figure 4. Clearance characteristics computed from the Mesh Model

The data in this figure shows the expected results. As the throw is increased (moving from case A to case D), the mean clearance is reduced. The variation in clearance is essentially the same for cases A, B and C. For case D, the mean is close enough to zero that the variation is reduced, it not being possible for clearances to be less than zero. From this data alone, it would seem that case D is the best; in fact, a further increase in throw with a corresponding reduction in the mean and standard deviation of the clearance distribution might be considered.

To this point, however, we have not considered the other half of the meshing characteristic – contact force. As noted above, the asymmetries in an involute set caused by manufacturing variation and operating load effects result in a situation where, over one revolution of a single assembly, the average clearance and the average contact force can both have positive values. We would expect that the number of assemblies with positive contact force and the level of this force to increase as the mean clearance is reduced. Case D in Figure 4 should have higher counts and higher levels in force than case A. This characteristic is illustrated in Figure 5. In this presentation of the data, the relative contact force is plotted against the clearance. A limiting value of contact force representing flank damage is used to non-dimensionalize the forces presented in this study

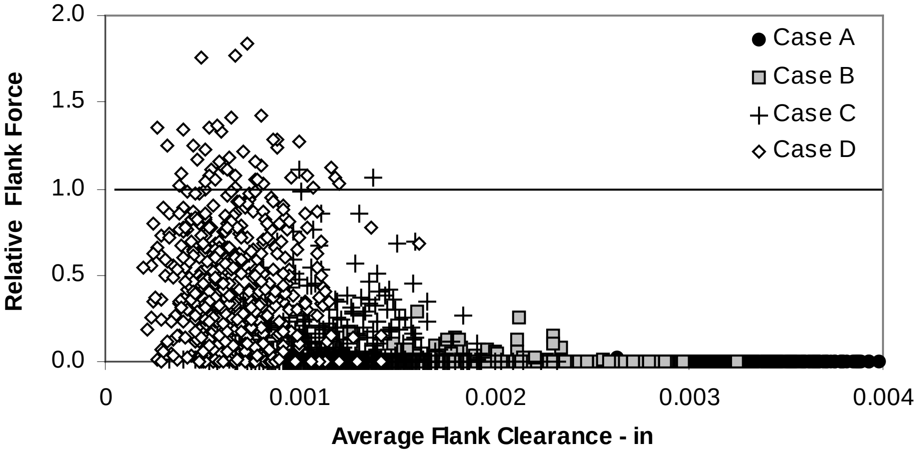

Figure 5. Relationship between flank clearance and contact force

Case A resulted in 500 assemblies having the largest mean value of the average clearance. The results in Figure 5 show that there are no occurrences of contact force greater than zero within this set of assemblies (the solid symbols which are distributed along the x axis). As the crankshaft throw is increased for case B, the mean clearance is reduced and we now see assemblies with contact force – 40 of the 500 total. Even at clearances greater than 0.002 inches (51 μm) there are assemblies with average contact force of up to 25% of the presumed limit value. The next increase in throw generates 273 assemblies, over 50%, with the combination average clearance / average force characteristic. When we get to the extreme in case D, 473 out of the 500 assemblies generate contact force. Also, as can be seen in Figure 5, even this extreme condition results in no occurrences of zero average clearance.

The contact force impacts the analysis in two ways. First, there is the friction resulting in increased power and reduced efficiency. As we explore increasing the crank throw, this effect counters the improvement in efficiency that comes from lower clearance and reduced internal leakage losses. In addition, there is a reliability effect. We can see from the data in the figure that cases C and D result in a number of assemblies that fail the reliability test , having contact forces above the limit (relative forces > 1) representing a failure.

The study continues with an analysis of the performance characteristics of the compressors for which the mesh analyses have been completed. Results of this part of the investigation are reported in Section 3.

3. PERFORMANCE MODELING

A comprehensive thermodynamic model is available for calculation of the performance in the face of the variety of flank clearance / contact force combinations generated by the assembly analyses. The program includes models of the motor, bearings and flowpath in addition to the model for compression in the involute set. A version of this program was created for Monte Carlos analyses. A spreadsheet tool is used to create a table of program inputs, one line for each assembly. This file is read by the thermodynamic model which executes the analyses and prints results for statistical analysis of the performance characteristics.

The first study uses the assembly analyses reviewed in Section 2 where manufacturing process capability for all features is assumed to be represented by Cpk=1 and the crank throw is varied. Figure 6 shows performance characteristics for case A. In this and all other analyses in this report, the only factors that vary in the Monte Carlos are the computed mesh characteristics: flank clearance and contact force. The dashed line in the figure

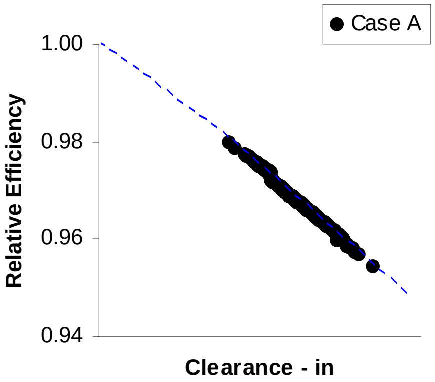

Figure 6. Relative Efficiency / Case A

is the computed characteristic of efficiency vs. flank clearance with no flank contact force effects. The computed actual value of efficiency at zero clearance with zero contact force from that analysis is used to normalize all of the performance results. Figure 6 shows that the 500 trial assemblies of case A follow the clearance characteristic. This is as expected; recall that the mesh analysis for this case (Section 2.2) produced no assemblies with contact force. The mean clearance for the case A assemblies is 0.0031 in (79 μm) and the mean relative contact force is 0. The mean and standard deviation of the relative efficiency characteristic are 0.968 and 0.004, respectively. The standard deviation in this case reflects simply the slope of the efficiency vs. clearance characteristic.

Figure 7 shows the results of the performance analysis of the assemblies of case B. The case A results remain in the figure, but with the small, light symbols. Again, the dominant factor in the efficiency variation is the simple effect of leakage due to the clearance variation. However, we now see some points deviating from this trend, appearing at slightly lower levels of efficiency. These are the cases in which the assemblies exhibited a combination of positive average clearance and positive average flank contact force. Mean values of the average clearance and relative contact force are 0.0021 in. (54 μm) and 0.2%. Mean and standard deviation for the efficiency are 0.978 and 0.004.

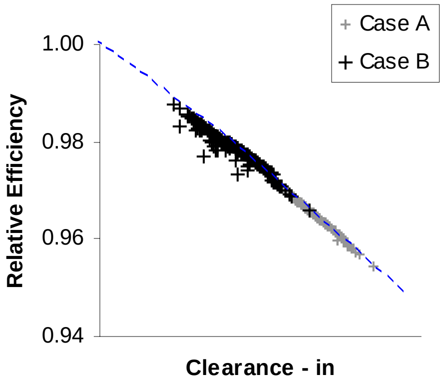

Figure 7. Relative Efficiency / Case B

Performance characteristics for the assemblies of cases C and D are shown in Figure 8. Results from cases A and B remain in the figure, again de-emphasized with the small, light symbols. These analyses show the increasing influence of the contact force on the efficiency distributions. The increasing number of points that fall off the efficiency vs. clearance characteristic shows the real benefit of reducing flank clearance is less than the idealized case of clearance variation without consideration of the contact force.

As documented in Section 2.2, cases C and D have 273 and 473 assemblies, respectively, that had both positive average clearance and positive average contact force. For case C, the mean clearance and contact force are 0.0012 in. (30 μm) and 17.5%. The values for case D are 0.0007 in. (17 μm) and 46%. Mean and standard deviation for the relative efficiency of these last two cases are (0.985, 0.0049) and (0.981, 0.0104), respectively.

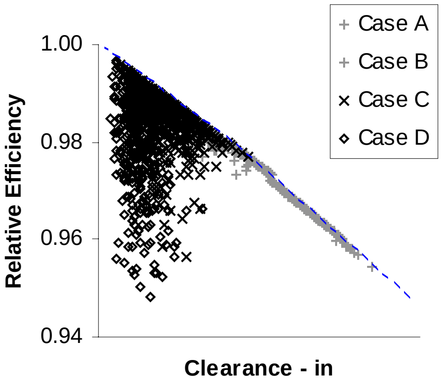

Figure 8. Relative Efficiency / Cases C and D

The effect of the contact force factor is quite visible here. Comparing the mean levels of efficiency for the four cases shows that there is a maximum at case C (0.985) with further reduction in clearance resulting in a lower efficiency (case D, 0.981). And, even though the variation in clearance (and hence leakage loss) is much lower for case D, we now see a very high standard deviation of efficiency in this case.

Reliability is a primary consideration in all design decisions. The mesh analyses showed that cases C and D contained 2 and 42 assemblies, respectively, where the contact force exceeds the failure limit. We would reject case D in any event since there is a higher efficiency option available. Case C is also problematic since 2 of 500 cases fail the contact force limit criteria.

Results of this study lead to the question of how we can move to lower clearances and higher performance. One obvious possibility is to reduce variation in the assemblies so that mean clearances can be pushed to lower levels while controlling the contact force effect on both performance and reliability. Both the design details and the manufacturing process capabilities are candidates for review. The capability effect is illustrated in the other study reported here.

Three cases are run to illustrate the capability effect. For each case, Cpk for all factors in the assembly is set at the same level; these levels are Cpk = 0.6, 1.0 and 2.0. For each capability level, the crankshaft throw is varied in order to determine the value of throw at which we see the condition of 1 of the 500 cases having an average contact force above the limit. Throw is then reduced slightly to create the last case of no violations of the reliability criteria over the 500 sample assemblies.

Results of the performance calculations for the three levels of process capability are shown in Figure 9. Results of the first study (Cpk = 1, crankshaft throw varies) are shown with the + symbols. The open circle symbols are the results of the Cpk effect calculations.

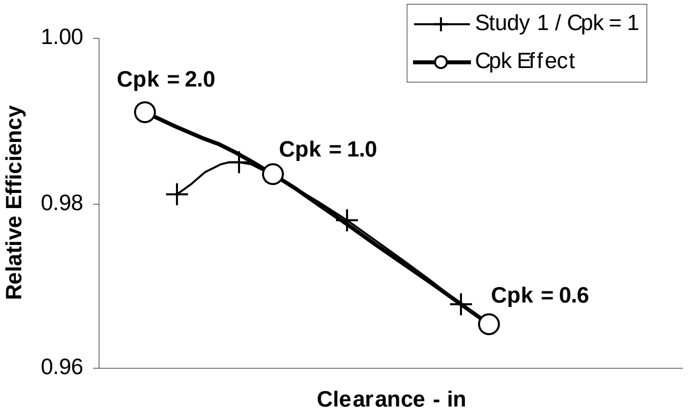

Figure 9. Process capability effect

As seen in the figure, improving capability has a powerful effect on performance. Respecting the reliability limit of contact force means a performance level of 0.965 at Cpk=0.6. Improving to a Cpk level of 1.0 raises performance by 1.8% to 0.984.

4. CONCLUSIONS

Many design decisions call for selection of factors that affect more than one customer-sensible characteristic of the compressor. In some cases, preferred values for control of one result are the opposite of what is desirable for another. This is the case for the involute mesh characteristics that were the focus of the studies reported here.

Assembly analyses that include effects of manufacturing and operational factors are the key to understanding the important interrelationships. Performance analysis allows the impact of the factors to include the efficiency effect and is also the source of the loads and temperatures that feed into the operational effects in the assembly calculations.

Conclusions from this design study are:

- The scroll mesh characteristics of clearance and contact force need to be balanced to realize the highest performance while meeting reliability constraints. A conservative selection of the mesh characteristic in favor of either performance or reliability will compromise the other.
- Real scroll assemblies will have assembly and operational load induced non-uniformity that allow the involute set to exhibit both running clearance and contact force characteristics in a single assembly.
- Assembly and performance analyses are the proper way to assess the need for manufacturing process capability and for guidance to the features whose control provides the greatest leverage at the level of customer-sensible compressor characteristics.
- Understanding and control of these features is fundamental to achieving the simplest design meeting performance and reliability goals.

NOMENCLATURE

Cpk	process capability index	(-)	Brg	bearing
FS	Fixed scroll	(-)	dia	diameter
OS	Orbiting scroll	(-)	in	inch
TP	True position	(in / mm)	μm	micrometer
X, Y	Cartesian coordinates	(in / mm)		

REFERENCES

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