2008

Transmission Error in Screw Compressor Rotors

Jack Sauls
Trane

Transmission Error in Screw Compressor Rotors

Jack SAULS
Phone: 608-787-2517; FAX: 608-787-2669
Email: jsauls@trane.com

Trane
Compressor Center of Excellence
3600 Pammel Creek Road; La Crosse, WI 54601 USA

ABSTRACT

Vibration stemming from interaction of the rotors is one of several factors in overall screw compressor sound and vibration. This paper will look at the definition, origin and effects of rotor transmission error. The work is limited to a study of static transmission error where dynamic forces and material deformation are not considered in the evaluations. Calculation methods are reviewed and examples of design studies using the calculations are presented. The analyses lead to a modified rotor design aimed at reducing the levels of transmission error. Results of testing to demonstrate the effects of this modification are presented.

1. INTRODUCTION

Alignment of the band of contact between male and female rotors in rotary screw compressors is an important factor in the running quality of the rotor pair. Errors in relative lead between rotors give rise to non-uniform motion of the driven rotor, leading to vibration and noise. The state of the lead is measured by the rotor pair's mesh transmission error (Holmes, 2006). The mesh transmission error (MTE) is the deviation of the actual angular motion of the female rotor compared to its theoretical motion based on the gear ratio determined by the number of rotor lobes.

Deviation from the theoretical rotation requires acceleration and deceleration of the female rotor. This motion might be in the form of a more or less continuous variation in the female rotor angular velocity leading to dynamic contact forces which are transmitted through the bearings to the housings, eventually contributing to compressor vibration and noise. The deviations in rotor geometry and alignment can also lead to female rotor motions characterized by periods of constant angular velocity (although not at the theoretical rate) followed by loss of contact between the rotors with a rapid adjustment to a new contact state. The impacts occurring in this case are heard as rotor rattle.

The variations in relative motion can be caused by dynamic forces on the rotors. They can also come from deviations in rotor geometry – profile form, lobe spacing and lead – and errors in shaft alignment. Except for errors in profile form, geometric and alignment errors effectively result in deviation of the relative lead between the rotors at the contact band. This observation is used in Section 2 to provide a simplified description of how transmission error arises in screw rotor pairs.

Reducing the errors in geometry and alignment would reduce transmission error, of course. However, there are a large number of critical dimensions that assemble to create the actual rotor-to-rotor contact characteristics. With the relative lead being a primary measure of transmission error, it should be possible to reduce MTE by changing the lead to which the rotors are actually produced relative the theoretical value normally used as the production target. However, this would suffice only if the errors to be compensated for were consistent. The most practical approach would seem to be a rotor design that was less sensitive to the effects of the errors on MTE. Reduction in MTE from crowning - a special form of rotor lead modification - is reviewed in Section 3.

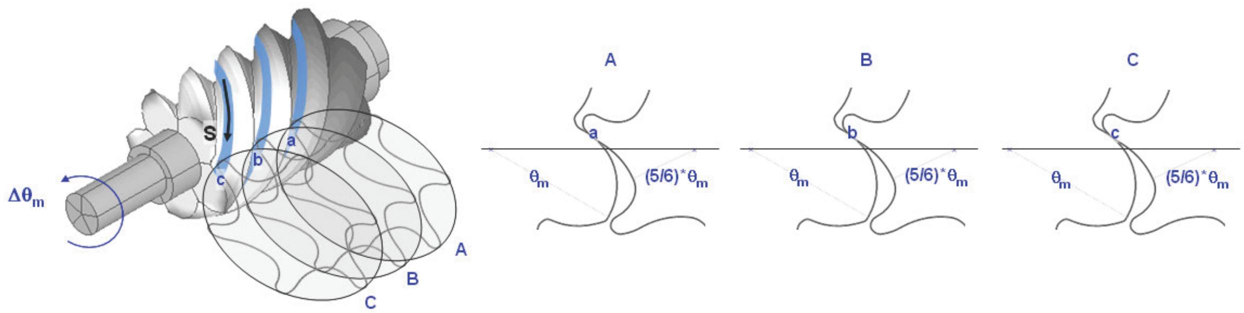
Experience with running crowned rotors in actual compressors is reviewed briefly in Section 4. The report ends with some concluding remarks in Section 5.

2. MESH TRANSMISSION ERROR

As noted above, we can look at MTE as the result of a difference in relative lead of the male and female rotors compared to the theoretical values that are different by design in the ratio of the rotor pitch diameters or lobe counts. To begin the illustration, we first consider the case of perfectly made, perfectly aligned rotors which will rotate without generating transmission error. Examples in this section are based on a rotor set designed for use in R-134a water chillers. The male rotor has 5 lobes, the female has 6. In this case, an angular rotation of the male rotor of $\Delta\theta$ will result in a rotation of the female rotor of (5/6) x $\Delta\theta$.

Figure 1 shows a male rotor paired with a female rotor represented by three sections, labeled **A**, **B** and **C**. The rotors are designed to contact only within a narrow band near the pitch line, indicated by the shaded area on the male rotor lobes in the figure. Contact areas at sections **A**, **B** and **C** are labeled **a**, **b**, **c**, respectively.

The sections shown are exactly one lobe pitch apart. That means that the angular orientation of the lobes in each section and the state of the meshing with the male rotor are, in the case of no errors in form or alignment, exactly the same as in the other sections. We can follow the meshing of one male and one female lobe in this way: consider the male to be at an angular orientation such that the contact with the female lobe is at point **a** in the middle of the rotor, as shown in the figure. After 72° of male rotor rotation (one lobe pitch for the 5 lobe rotor of the example) the contact point will have moved along the flank in the **S** direction to point **b**, which is also rotated to the orientation shown in section **B** in the figure. Another 72° of male rotor rotation moves the contact in the **S** direction to point **c**.



(a) Meshing Rotors

(b) Angular Orientations

Figure 1 Rotor Contact Characteristics

At each section in Figure 1(a), the male rotor lobe is oriented at the angle θ_m and the female at $\theta_f = (5/6)^* \theta_m$. We can quantify the error of rotation of the female rotor as $e = \theta_f - (5/6)^* \theta_m$. In this case, a plot of e vs. θ_m would be simply a horizontal line at e = 0.

Next we consider the case where there is an error in the male rotor lead. This can be represented as shown in Figure 2. Here we see only the male rotor with the contact band highlighted on one lobe. The variation in shading along the band represents the material state – plus metal at the end marked **a**, nominal material at **b** and minus metal at **c**.

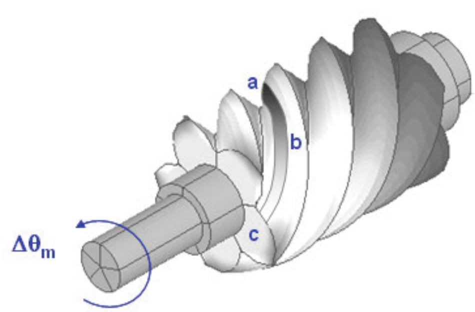
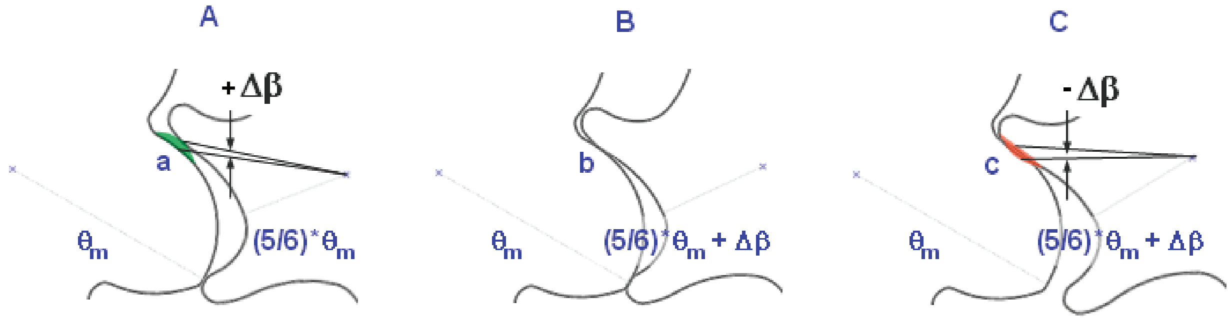


Figure 2 Male Rotor with Lead Error

Figure 3 shows the male and female rotors engaged at the **a**, **b** and **c** points. In 3(a), we see that the excess material at **a** means that the female rotor is rotated an increment Δβ more than it would have been had there been no lead error on the male rotor. Figure 3(b) shows point **b**, the location of the contact after 72° rotation of the male. Here, the male rotor profile is at its nominal material state. The female has rotated an amount (5/6)*72° in response to the male rotor rotation PLUS an additional amount, Δβ, to stay in contact with the flank that now does not have the excess material. Continuing on to point **c**, Figure 3(c), where the male rotor has again rotated 72°, we see the female once again has rotated the theoretical (5/6)*72° plus the error increment, Δβ.



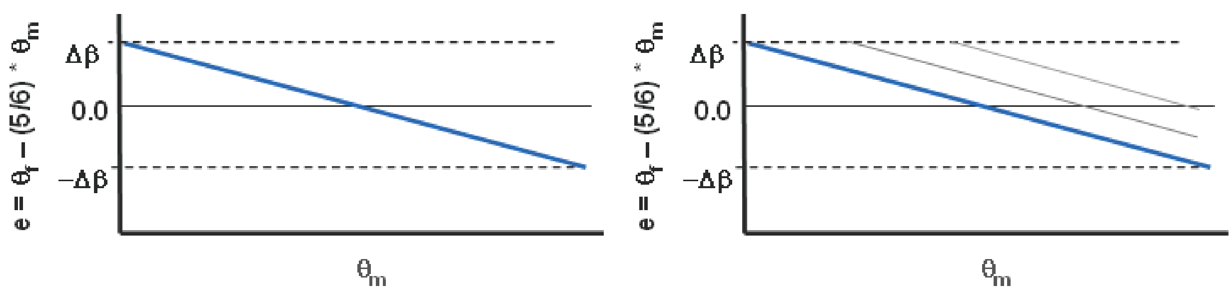
(a) **(b)** **(c)**

Figure 3 Rotor Contact Characteristics with Male Rotor Geometric Deviation

The error, e, in the female rotor rotation caused by the male rotor geom.+etric deviation is

$$e = \theta_f - (5/6)*\theta_m = (5/6)*\theta_m + \theta_m*2*\Delta\beta / 72 - (5/6)*\theta_m = \theta_m*2*\Delta\beta / 72 \quad (1)$$

If we plot the error term against θ_m, we get a straight line as shown by the bold line in Figure 4a. In this example, the female rotor is rotating faster than its theoretical speed, the additional rotational velocity being needed to keep up with the changing material state. Had we postulated an error going from minus metal to plus metal, the line in Figure 3 would have had a positive slope, indicating that the female rotor would run more slowly than its theoretical speed. In either case, the female rotor angular velocity would be constant so, as we have considered the situation so far, there are no accelerations or decelerations and hence no additional dynamic forces.



(a)
Single Lobe Meshing Characteristic

(b)
Meshing Characteristics for Multiple Lobes

Figure 4 Rotor Meshing Characteristics with Rotor Geometry Deviation

So far we have considered only the action of a single pair of lobes. In an actual rotor pair, there are multiple lobes engaged at the same time and assuming, for our simple example, that the profile errors are the same on each lobe, then the rotor pair will be characterized by a series of meshing error characteristics, each offset from the other by one lobe pitch. This is shown in Figure 4(b) where the characteristics for two other lobe pairs are added to the chart. While in the case of the single lobe of Figure 4 the female rotor error would run from Δβ to -Δβ, the actual rotor would not follow this course, rather, its rotational characteristic would be as shown in Figure 5.

The actual rotational characteristic is controlled by the extremes in variation of the individual lobe characteristics. If we follow the process illustrated in Figure 5 from left to right, we see that the error tracks downward (female rotor

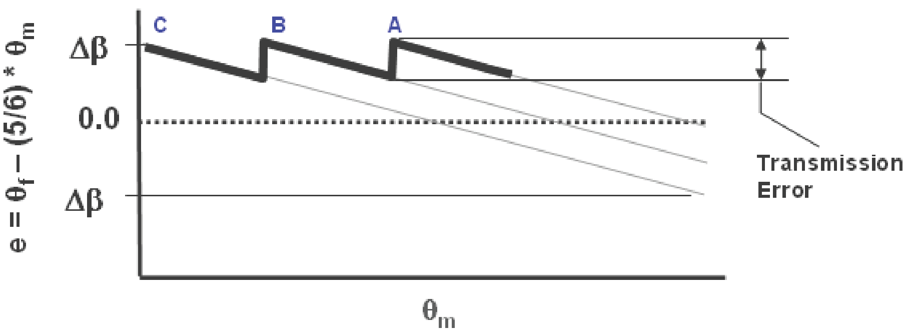


Figure 5 Female Rotor Rotational Characteristics and MTE Definition

rotating at a higher than theoretical rate) over the first 72° of male rotor rotation. At this point there will be another meshing lobe pair, denoted by characteristic **B** in the figure, that will be in the same orientation as was lobe pair **C**. Pair **B** is in the orientation where the female rotor must rotate an increment Δβ to account for the excess material. Hence, it is this meshing condition that controls where the female rotor must be, so the rotor adjusts. The process repeats and the overall female rotor rotation characteristic takes the saw tooth form shown by the heavy line in Figure 5.

The mesh transmission error is defined as the difference in the extremes of the error (e) characteristic. In this report the MTE will be expressed as the angular rotation difference in degrees although it is often given as the linear motion of the rotor measured at the female rotor pitch diameter.

Use of a 3D rotor meshing analysis program for the computation of meshing characteristics is described in Sauls, et al, 2007. Errors of rotor form caused by manufacturing variation and deformation due to operating pressure and thermal loads can be defined for the analysis. Errors in alignment due to the same effects can also be prescribed. With the geometry thus defined, the program finds the angular orientation of the female rotor required for there to be at least one point of contact with the male. The calculation is carried out for a number of male rotor orientations, providing the information necessary to build up the meshing error characteristics.

Figure 6 shows the results of this analysis for a rotor pair with misaligned shafts. The calculations were carried out at 72 male rotor angular orientations evenly spaced 1˚ apart. The computations with actual rotors using the 3D analysis confirm the meshing characteristics deduced from the observations in this section.

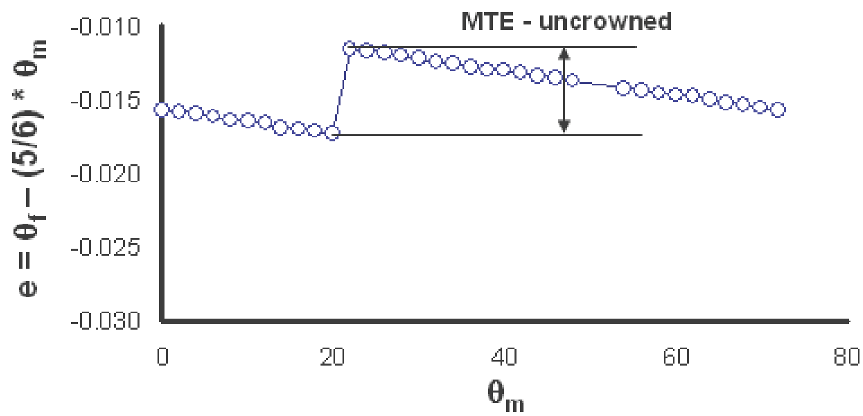


Figure 6 Computed Mesh Characteristics for Rotors with Misaligned Shafts

3. ROTOR CROWNING

Crowning of screw rotors for control of mesh transmission error has been discussed by Holmes, 2006 and Sauls, 2007. The crown is achieved by removing material along the male contact band a both ends of the rotor. The modification is applied with a parabolic distribution, the transition from maximum material removal to nominal occurring over 30% of the rotor length at each end, leaving the middle 40% of the rotor in its nominal form.

Figure 7 shows the meshing characteristics for crowned rotors after the approach used in Figure 5. The crowning results in a change from the straight line characteristics for the single lobe rotation error of Figure 5 to the curved lines of Figure 7.

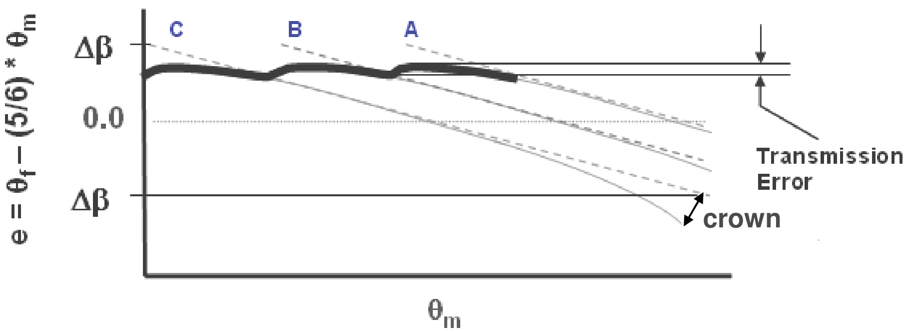


Figure 7 Female Rotor Rotational Characteristics and MTE for Crowned Rotors

Following the description of the meshing process for uncrowned rotors of Section 2, we can develop the meshing characteristics for the case of crowned rotors. The result is the heavy line in Figure 7. Here, we see that there is a continuous variation of the error term over most of the process. The discontinuity in contact still appears at the transition of control of the motion from one lobe to the next, but the magnitude of the MTE is reduced significantly.

Figure 8 shows the calculations for a rotor pair with the same misalignment as in Figure 6. Rotors are the same as in the previous example except for a crowning of the male rotor.

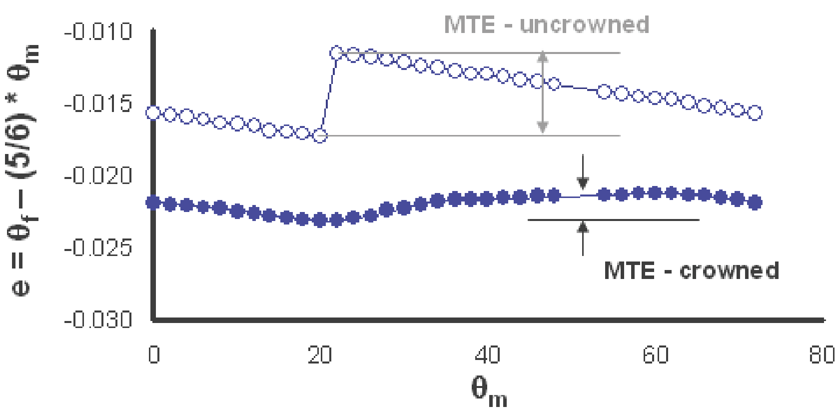


Figure 8 Computed Mesh Characteristics for Rotors with Misaligned Shafts and Crowned Male Rotor

The 3D computations for actual rotors with misalignment demonstrate the expected effect for the crowning. For this example, MTE was reduced by 65%. For the case of no crowning, the MTE requires no acceleration of the female rotor except at the discontinuity. Here there must be a rapid adjustment and there will almost certainly be rotor-to-rotor impact. These factors are not evident in the pure geometric analysis which shows static MTE only. While not evident on the chart, there is a discontinuity at the point of minimum e for the crowned rotor characteristic. This situation may or may not cause loss of rotor-to-rotor contact with resulting impacts. However, even if it does, the impact velocities are likely to be considerably lower than in the case of the uncrowned rotors.

The effect crowning has on the MTE is dependent on the magnitude of the rotor geometric and alignment errors. For perfectly made rotors with no misalignment, the MTE is zero. If the rotors were to be crowned, then MTE would increase. The rotor design issue is then to select the right crown for the expected application deviations.
Figure 9 shows the effect of crowning using the same rotor pair whose mesh characteristics are illustrated in Figures 6 and 8. The crowning selected for the example in Figure 8 is defined as the 100% level and several values of crowning between 0 and this value were analyzed.

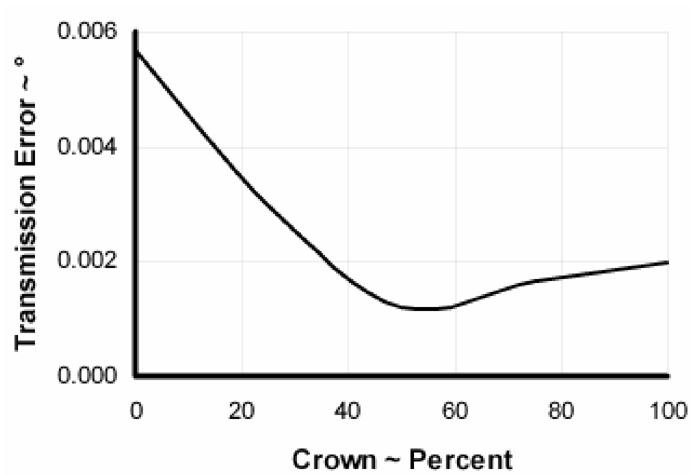


Figure 9 Crowning Optimization

There are some important observations from this analysis. First, there is an optimum crown for a given error. Here the optimum is about 50% of the level first investigated. This results in an MTE reduction of 79% relative to the uncrowned case compared to a 65% reduction for the original crowning. The other feature of this MTE vs. crowning characteristic is the difference in the effect of crowning below the optimum compared to crowning above the optimum. For over-crowning, MTE increases but at a much lower rate than if the rotors were under-crowned. Knowledge of this characteristic is important in minimizing average MTE in production quantities of compressors where errors vary.

4. EXPERIENCE WITH CROWNED ROTORS

An experimental investigation into the effects of rotor crowning on compressor noise was initiated in the early 1990's. Several rotor pair with differing degrees of crowning were produced and tested in the same compressor housings. The crown was applied only to the male rotor and tests were run with each of the male rotors paired with the same female rotor. Results are summarized in Figure 10.

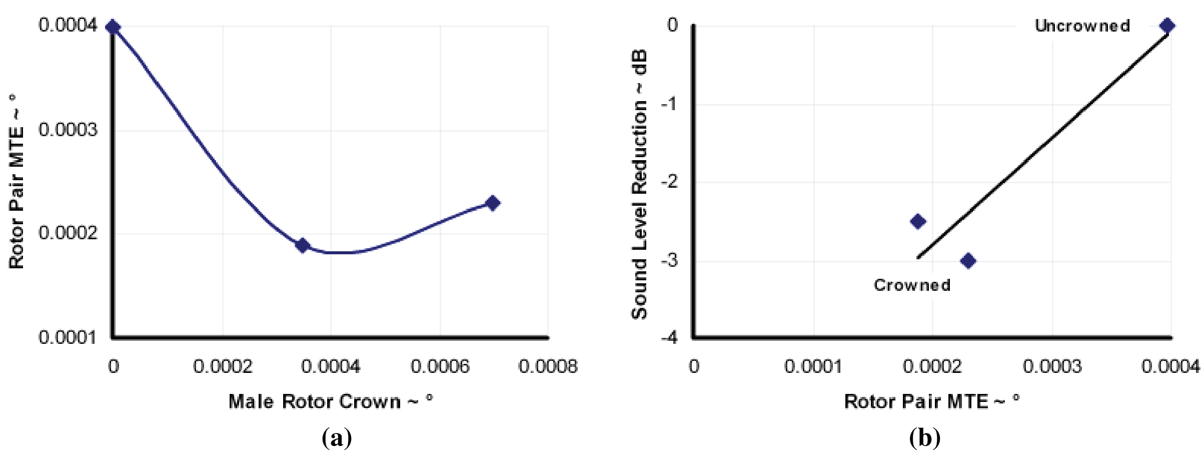


Figure 10 Effect of Crowning on MTE and Compressor Noise

Figure 10(a) shows the relationship between static MTE and crowning. The MTE was determined in a special test machine. The shafts were aligned and all of the static MTE was due to errors in the rotor geometry. The data from these three rotor pairs do conform to the characteristic shown in Figure 9. Figure 10(b) shows the measured compressor sound levels plotted against the static MTE. The effect of reducing the MTE is to improve noise by about 3 dB.

Finally, the tests provided information on the effect of the crowning on compressor efficiency. Using the uncrowned efficiency as the baseline, we saw that a crown of 0.00035° improved efficiency by 2.1% while the rotors crowned at 0.00070° resulted in a loss of efficiency of about 1%. The reason for this seems to be that the crowning compensates for thermal effects that often result in rotors running tight at the discharge end. Providing some relief by removing material from the male rotor seems to help up to a point. Increasing crowning beyond this optimum results in increased mesh clearance and higher leakage losses.

5. CONCLUDING REMARKS

Rotor mesh transmission error has been shown experimentally to affect overall compressor noise levels. Results of analyses and rotor pair measurements explain how geometrical deviation of the rotors and parts that control their alignment give rise to non-uniform rotation of the female rotor ~ the mesh transmission error. As a result, it is possible to design rotors for quieter operation using relatively simple tools as demonstrated in this report.

For the actual design process, it is necessary to consider the effects of rotor thermal and pressure load induced deformations (Sauls, 2007) and to explore the range of manufacturing variability. The design principles for minimizing MTE and the tools used are the same, but the process is more complex due to the extra inputs.

The analytical studies reported here show that there is an optimum crown for a given rotor geometry and alignment. Calculated MTE characteristics show that the crowning more than the optimum has less of a detrimental effect than does under-crowning. A result of this is that over-crowning can be used to reduce variation in MTE from compressor to compressor in production, although on the average, the MTE of the design will be slightly higher. Finally, tests showed that going too far in the over-crowning will result in a loss of efficiency. It is, as with so many things in compressor design, a balancing act.

NOMENCLATURE

A, B, C	Rotor lobe identifiers	-
a, b, c	Point identifiers	-
e	Meshing error term	°
MTE	Mesh transmission error	°
S	Arc length on mesh line	m
β	Geometric error factor	°
θ	Rotor angular orientation	°

Subscripts

f	Female rotor
m	Male rotor

REFERENCES

Holmes, C., Noise Reduction in Screw Compressors by the Control of Rotor Transmission Error; Proceedings of the 2006 International Engineering Conference at Purdue, Purdue University; July, 2006.

Sauls, J., G. Powell, B. Weathers; Thermal Deformation Effects on Screw Compressor Rotor Design; Proceedings of the International Conference on Compressors and their Systems; IMechE; September, 2007, p. 159-168.

ACKNOWLEDGEMENTS

I would like to thank Trane for the opportunity to present the work reported here. My thanks also to Dr. Chris Holmes and Dr. Bob Munro for their ideas and insights into the issues of mesh transmission error for screw compressor rotors. Their guidance was invaluable in development of the design processes described here and their friendship has been much appreciated over these many years.