NOISE REDUCTION IN SCREW COMPRESSORS BY THE CONTROL OF ROTOR TRANSMISSION ERROR

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ABSTRACT

Noise reduction in compressors has been achieved by applying helix compensation to rotors during manufacture. An analysis of the effect of generalised misalignments and distortions is modelled and used to predict the transmission errors and accelerations. Evidence of noise reduction supplied by Holroyd's research partners is presented. **Keywords**: screw compressor, noise, rotors, transmission error

1. INTRODUCTION

It has long been known that manufacturing deviations in screw compressor rotors and their housings can lead to undesirable noise phenomena, and that the more extreme manifestations, such as as rotor rattle where rotor contact oscillates rapidly from one flank to the other, can severely shorten the life of the rotors.

There is considerable evidence from the field of helical gears that noise frequencies have a close correlation with transmission errors. The general theory was first published by Harris [1], with subsequent developements by Harris, Gregory & Munro [2]. Munro [3] has written an excellent review of the theory and measurement of transmission error. A text book with a broad coverage of the subject, which includes the effect of helix crowning on transmission error, has recently been produced by Smith [4].

There are relatively few publications on the influence of rotors on compressor noise. Recent work by Yoshimura [5] has focussed on the influence of gas forces and profile shapes, e.g. involutes. In the present study however, it is assumed that motion is transmitted through the helical action rather than by the properties of the profile shape. However, the general theory is equally valid for both profile and helical effects.

Transmission error occurs in the driven component of a pair when its instantaneous angular position differs from the theoretical angular position determined by the gear ratio. By convention, transmission error is positive at any instant when the driven component is ahead of its theoretical position. The error is expressed in microns at the pitch circle radius. The displacements and accelerations can be transformed by Fourier analysis, and this spectrum has been found to be especially relevant to noise in gearboxes.

The Holroyd pair measuring system has been described by Holmes and Munro [6]. The machine measures both transmission error and rotor clearances. A results plot from the machine is shown in Fig. 1. The lower line is the forward transmission error, the upper faint line is the reverse transmission error, and the upper heavy line is the backlash (backlash is obtained by subtracting the forward from the reverse transmission errors). In the transmission error line, the longer waves represent the effect of run-out, the shorter waves represent tooth-to-tooth effects caused by divide and lead errors, and the shortest waves relate to errors in profile or helix form.

The forward transmission error results are converted into a spectrum using Fourier analysis, as shown in Fig. 2. This reveals the basic frequencies existing in the rotors. The amplitude of the displacements is plotted against the

frequency expressed as cycles per revolution of the driving rotor. In this case, the fundamental frequency occurs at 5 cycles per rev., because the driving rotor has 5 lobes. Harmonics can be seen at 10, 15, etc. Since the numbers of lobes on the male and female are usually different, it is possible to identify which rotor relates to which feature on the spectrum. For example, the largest frequency line in the graph is at one cycle per rev. of the 5 lobe male (driving) rotor, and the line immediately to its left is for the 7 lobe female which rotates at 5/7 the rate of the male. As with the transmission graph, the lower frequencies represent the effect of run-out, the medium frequencies represent tooth-to-tooth effects caused by divide and lead errors, and the high frequencies relate to errors in profile and helix form. Angular accelerations can be displayed in a similar manner.



Fig. 1 ARAC Report: Transmission Error & Backlash



Fig. 2 ARAC Report: Fourier Spectrum of Forward Transmission Error

2. TRANSMISSION ERROR PREDICTION

1.1 Illustrative example with linear mis-match only

To understand transmission error, assume a pair of rotors is operating with a helical mis-match of 10 microns over its body length, so that contact between the lobes occurs at the discharge end. Now imagine a single pair of lobes, with all others removed. This pair rotates through a full engagement, from initial contact at the suction end to the point where the final contact occurs at the discharge end. The positional error of the idling (or driven) rotor is taken to be zero when the lobes first make contact at the suction end, and as the pair rotates, the idling rotor will advance steadily ahead of its theoretically correct angle as determined by the gear ratio. This effect is plotted in Fig. 3.

The horizontal axis shows the rotation of the driving rotor expressed as axial movement of the contact point on the drive band. In this paper, the deviation of the idling rotor is expressed in microns normal to the surface at the pitch line. Although not the usual way of expressing transmission error, it makes it easier to combine the deviations from several sources in the assembly, as explained later. The final result can easily be converted into the transverse plane if required, e.g. for comparison with actual noise measurements.

Fig. 4 shows the two adjacent pairs of lobes. The lagging pair (lower line) will not affect the net transmission error over this rotation angle, and the leading pair will dominate for most of the rotation.

The combined effect of all the lobe combinations is shown in Fig. 5. The transmission error, indicated by the heavy enveloping line, has a saw-tooth shape in which control passes suddenly from one lobe combination to the next. In this case, it might be expected that the rotors would lose contact momentarily at the end of each engagement. The direction of the 'saw teeth' could equally be the opposite of that shown, in which case sudden forward accelerations will occur.

1.2 Effect of including other deviations

It may easily be imagined that transmission error will result from many sources in addition to simple mis-alignment. The magnitudes and directions of deviations in any particular situation will depend on design, manufacture, and operating conditions. Many scenarios are possible, and the final judgement must be made by the compressor designer based on his close knowledge of his product.

In this study, the following deviations were included in the simulation: lead mismatch, lead non-linearity, housing bore locations, bearing deflections, and bending under gas forces. Other influences excluded because they were considered either negligible or not applicable in this case were: divide (pitch) errors, profile at drive band, and rogue contacts (e.g. at root). Lobe bending under variations in the gas torque was also excluded, and this could perhaps form the basis of a future study.

The resulting transmission error pattern is shown in Fig. 6.

1.3 Relief (or 'crowning') Strategy

Using a capable machine tool relief can be applied to the helix in a variety of ways. Fig. 7 shows an unmodified helix (heavy line), with reliefs at each end RA1 and RA2. These can take any value and can differ. Relief can be specified as linear (as shown) or parabolic. The lengths RL1 and RL2 can also take any values up to half the body length. In the case of the deviations chosen, the type, amount, and length of relief was optimized to minimize the displacements for the expected mis-match. Parabolic relief was selected. The combined effect of the assumed deviations and the relief is shown in Fig. 8. The net transmission error has been reduced to about 2.5 microns.

3. NOISE PREDICTION

The Fourier spectra of displacements without helix relief is shown in Fig. 9. The effect of including relief is shown in Fig. 10. The reductions in the acceleration spectrum can be seen by comparing Fig. 11 with Fig. 12. If desired, the velocity spectrum can also be displayed, as could the 3rd derivative, ('jerk'). The relation between noise and the various spectra is not yet fully understood, although it is observed that the acceleration spectrum bears a closer relation to actual noise than the dispacelement spectrum. Note that the base line Cycles Per Rev. can be converted to frequency in Hz by multiplying by the rotational speed of the driving rotor in RPM and dividing by 60.







Fig. 5 All 7 lobe pair combinations



Fig. 4 Lobe pair 1 with following and leading pairs



Fig. 6 All deviations combined



Fig. 7 Helix relief



Fig. 8 Smoothing effect of helix relief



Fig. 9 Displacement spectrum without helix relief



Fig. 11 Acceleration spectrum without helix relief



Fig. 10 Displacement spectrum with helix relief



Fig. 12 Acceleration spectrum withhelix relief



Fig. 13 Compressor and instrumentation - 1



Fig. 14 Compressor and instrumentation - 2

4. NOISE MEASUREMENTS

Noise measurements have been carried out at two refrigeration companies following rotor design collaboration with Holroyd. Both report significant reductions. York International Corporation used sound equipment as shown in Fig. 13 and Fig. 14. The noise results are shown in Fig. 15, indicating a reduction of 6 dB at the higher compression ratio. It might be expected from the measurement arrangement that mechanical noise from the compressor is only a part of the overall noise measured, the remainder coming from motor, gas pulsations etc. It is suspected that the noise reduction in the compressor itself is considerably higher than the 6 dB measured.



Fig. 15 Noise tests of standard rotors and relieved rotors (same housing)

5. CONCLUSIONS

The dramatic reduction in both displacements and accelerations which are predicted indicates that a powerful method of reducing rotor-induced noise exists by the application of helix relief, and this is borne out by experiment. Further work is desirable to understand in more detail the correlation between the various spectra and the measured noise. However, the general relationship is clear, and Paul Nemit, York's principal compressor engineer, made the statement: "The difference in noise levels between any two rotor sets can be assessed by the ARAC Fourier results. By comparing the magnitude of the acceleration, we can view the effects of various degrees of crowning". The implication is that rotor-related noise can be both predicted, and prevented, during rotor manufacture.

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