BALANCING OF INTEGRAL GEAR-DRIVEN CENTRIFUGAL COMPRESSORS

by

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ABSTRACT

Due to the complex dynamic behavior of gear-driven centrifugal turbocompressors, selection of the proper balancing and assembly procedures is essential to obtain a smoothly running rotor. Rigid and flexible balancing techniques as applied to a typical rotor system are explained and evaluated, and possible improvements are discussed.

INTRODUCTION

The gear-driven centrifugal compressor was invented some 40 years ago and has become the dominating method for air separation. Quite a few manufacturers supply this kind of machine. A very important part in both manufacturing and maintenance is proper balancing.

TYPICAL DESIGN

A spur gear drives one or more pinion shafts with impellers on one or both ends (Figure 1), thus forming systems with one to six stages. The design principle provides for different speeds of the stages or at least pairs of stages (Figure 2). While the main gear wheel runs at lower speed (normally synchronous speed, such as 3000 cpm or 3600 cpm), the pinion shafts can reach speeds up to 60000 cpm. Therefore, a discussion of balancing problems should focus on the high speed components.

ASSEMBLY PROBLEMS

The impellers always are separate components, i.e., they are mounted on to the pinion shaft. Most manufacturers use a proprietary method of fit and fastening (Figure 3) to attain the following objectives:

- accurate location of the impeller throughout the entire working range (speed, load, temperature).
- sufficient reproducibility of the impeller location, in case the impeller has to be removed from its shaft for reasons of final assembly or inspection.

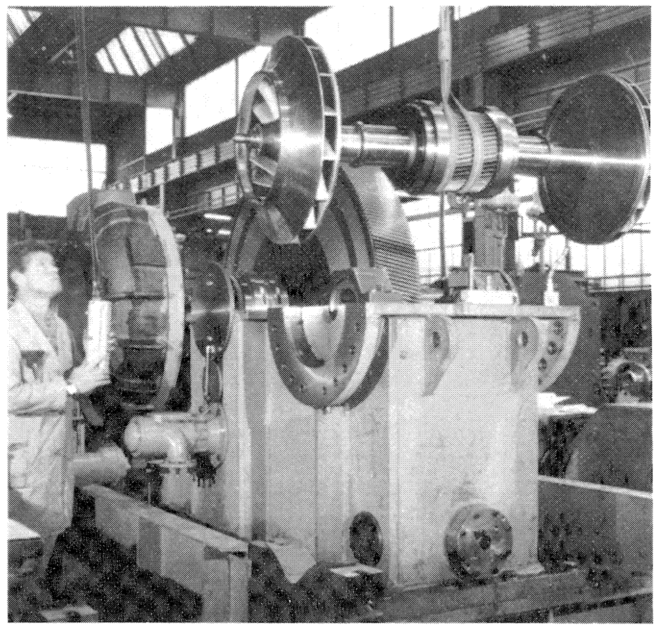

Figure 1. Assembly of a Multistage Gear-Driven Centrifugal Compressor.

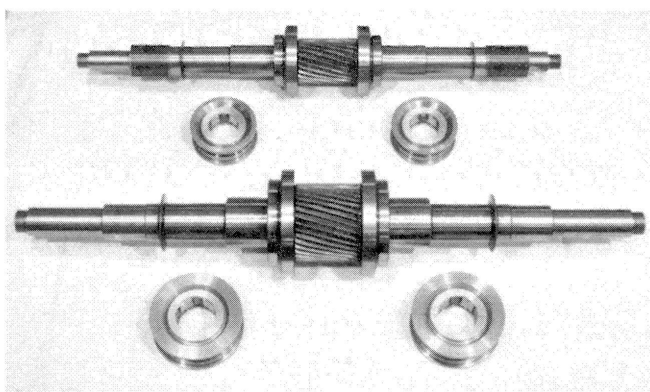

Figure 2. Two Shafts of a Four-Stage Compressor.

Changes in residual unbalance due to dis- and reassembly should not exceed more than 20 percent to 50 percent of the balance tolerance.

Suggestions on how to handle larger changes are listed in Table 1.

BALANCING

The International Standard ISO 5406 [1] lists five classes of rotors (Table 2). To understand the typical behavior of these rotors and the balancing procedures to be applied, the definition of "rigid rotor" must be recalled:

Table 1. Suggestions for Proper Handling in Case of Concentricity and Reproducibility Problems.

<table>
<tr><td colspan="2" rowspan="2"></td><td colspan="2">Concentricity of impeller location</td><td rowspan="2" colspan="2"></td></tr>
<tr><td>good</td><td>insufficient</td></tr>
<tr><td rowspan="4">reproducibility of fit</td><td rowspan="2">good</td><td rowspan="3">• standard balancing procedure</td><td rowspan="2">very unlikely condition</td><td>no</td><td rowspan="4">impeller removed for final assembly</td></tr>
<tr><td>yes</td></tr>
<tr><td rowspan="2">insufficient</td><td>• if impeller location has stabilized after a high speed or over-speed run, balancing at low speed is possible.</td><td>no</td></tr>
<tr><td>• field balancing required</td><td>• a high speed or over-speed run is required after final assembly, and then field balancing.</td><td>yes</td></tr>
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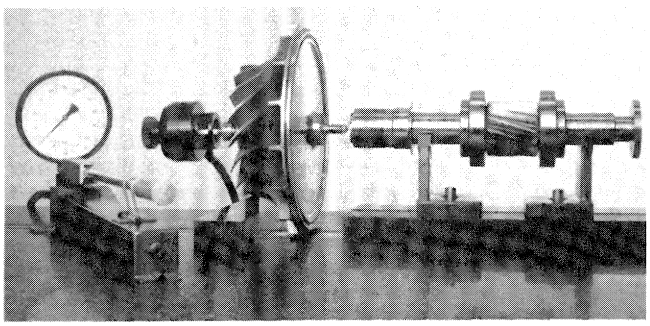
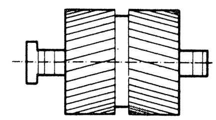
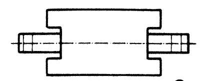
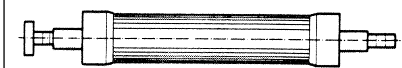
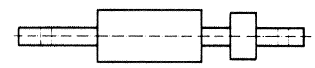
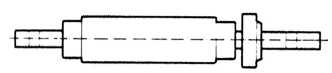


Figure 3. Mounting an Impeller to the Pinion Shaft With a Hirth-Serration, Using a Hydraulic Preloading Device.

Table 2. Rotor Classes According to ISO 5406.

<table>
<tr><td></td><td>Class of Rotor</td><td>Example</td></tr>
<tr><td>1</td><td>Rigid Rotors</td><td> Gear Wheel</td></tr>
<tr><td>2</td><td>Quasi-Flexible Rotors</td><td> Computer memory drum</td></tr>
<tr><td>3</td><td>Flexible Rotors</td><td> Generator Rotor</td></tr>
<tr><td>4</td><td>Special Flexible Rotors</td><td> Rotor with Centrifugal Switch</td></tr>
<tr><td>5</td><td>Single Speed Flexible Rotors</td><td> High Speed Motor</td></tr>
</table>

> A rotor is considered rigid when its unbalance can be corrected in any two (arbitrarily selected) planes and, after that correction, its residual unbalance does not significantly change (relative to the shaft axis) at any speed up to maximum service speed and when running under conditions which closely approximate those of the final supporting system.

A discussion of pinion shafts with impellers has to consider the first four rotor classes:

Class 1—Rigid Rotor. This rotor can be balanced at low speed (compared to service speed) in at least two correction planes.

Class 2—Quasi-Rigid Rotor. This rotor is a flexible rotor (with shaft elasticity); however, by using special procedures, it is possible to balance the rotor at low speed, often by correction in more than two planes.

Class 3—Flexible Rotor. A flexible rotor requires high speed balancing, usually by correction in more than two planes.

Class 4—Special Flexible Rotor. A rotor of Class 1, Class 2, or Class 3 with components, which are flexible or flexibly attached—generally requires high speed balancing.

Balancing As Class 1 Rotors

Although compressor rotors are typically designed currently to run above the first critical speed, there are still some rigid rotors in service, such as the following:

Pinion shaft weight	400 kg
Impeller weight	65 kg
Total weight	465 kg
Service speed	8,000 cpm
First flexural critical speed	12,000 cpm

ISO 1940 [2] recommends balancing turbomachinery compressors to quality grade G 2.5; however, some manufacturers go to G 1 or even lower. This practice may date back to the Bureau of Ships Noise Reduction Standard MIL 167 and to API 617 [3], which limit the residual unbalance (U_{per}) to 4 W/N (with W in lb, N in cpm, and U_{per} in oz-in). Converted into an ISO 1940 quality grade, this limit would be equivalent to G 0.7.

Based on the service speed of 12000 cpm, G 2.5 allows a center of gravity (CG) displacement of less than 0.0001 in, which is likely to cause assembly problems (see ASSEMBLY PROBLEMS). Certainly, a grade equivalent to G 0.7 cannot be maintained through a dis- and reassembly cycle, and therefore, will require rebalancing.

Standard Procedure

The assembled rotor is balanced at low speed using correction planes II and IV (Figure 4) to residual unbalances of $U_{per}II$ = 500 g·mm and $U_{per}IV$ = 900 g·mm.

If for any reason (e.g., easier handling) the components are prebalanced individually (to five to ten times the tolerance), planes I and II are used for the impeller and planes III and IV for the pinion shaft. Assembly balancing remains as described above.

Component Replacement

It may be of interest to be able to replace the impeller (in case of wear or damage) without the necessity to rebalance the

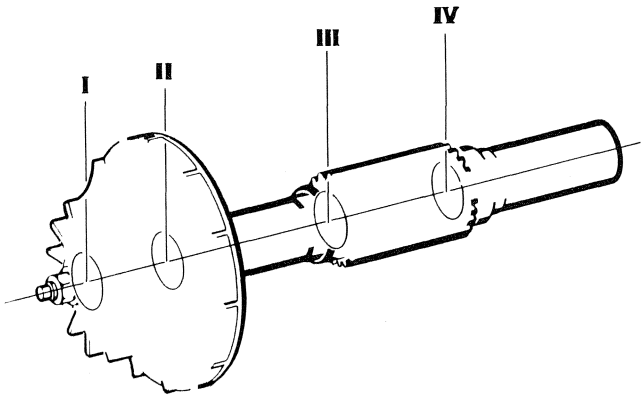


Figure 4. Rotor With One Impeller and Four Correction Planes.

assembly. This may be possible with the following procedure: The impeller is balanced (to tolerance) in planes I and II (U_{per} = 100 g·mm/plane), using a separate mandrel. By applying a 180° indexing/balancing method, all errors from the mandrel are eliminated. The pinion shaft is also balanced to tolerance, using planes III and IV (U_{per} = 600 g·mm/plane).

If the assembly errors (radial displacement and inclination of impeller axis versus pinion axis) exceed the assembly balance tolerance, two alternatives remain: a) Reduce the residual unbalance of the pinion shaft, e.g., to 100 g·mm/plane, thereby allowing for larger assembly errors. b) If the assembly errors are still too large, but are consistently reproducible, use an impeller or 'dummy" rotor that has been balanced on a separate mandrel using the 180° index/balancing method, mount it on the pinion shaft, and then balance the pinion shaft by correcting in planes III and IV. Again, go through an index/balancing procedure between the impeller (or dummy rotor) and the pinion shaft, making corrections only in planes III and IV. Now, all unbalance caused by the geometric errors of the shaft's impeller mounting surface are corrected in the pinion shaft.

The benefit of going through a), and b), will be that any properly balanced replacement impeller will not throw the assembly out of tolerance.

Balancing As Class 2 Rotors

A typical example for this class has two impellers (Figure 5) and the following specifications:

Pinion shaft weight	24 kg
First impeller weight	4 kg
Second impeller weight	3 kg
Total weight	31 kg
Service speed	28,000 cpm
1. flexural critical speed	15,200 cpm
2. flexural critical	16,900 cpm
3. flexural critical speed	53,200 cpm

ISO 5406 [1] subdivides Class 2 rotors into a group of rotors for which the axial distribution of unbalance is known (because of typical rotor designs), and a second group for which the axial distribution of unbalance is not known (Table 3). Even though none of the Class 2 rotor examples look like gear-driven compressors, Class 2 c, nevertheless, applies because a rotor assembly would have more than two transverse planes of unbalance, i.e., two in the pinion and two in each impeller.

Standard Procedure (for quality grade G 2.5)

ISO 5406 recommends balancing the shaft first at low speed. Corrections should be applied in planes III and IV. After mounting of the first impeller the assembly is rebalanced by applying corrections in planes I and II. Then the second impeller is mounted and the assembly is rebalanced in planes V and VI (Figure 5). The intention is to correct all unbalances near the transverse planes where they occur, thereby minimizing internal bending forces, and shaft deflection at the critical speeds.

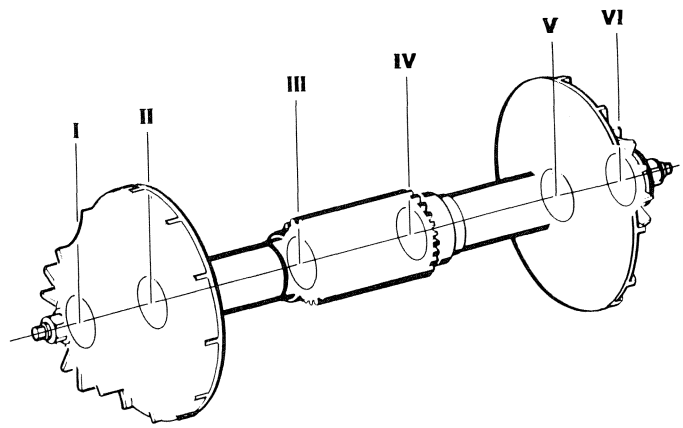


Figure 5. Rotor With Two Impellers and Six Correction Planes.

Table 3. Class 2 Rotors.

Class of Rotor	Example
Rotors Where The Axial Distribution Of Unbalance Is Known	
2A	Shaft with Grinding Wheel
2B	Shaft with Grinding Wheel & Pulley
2C	Jet Engine Compressor Rotor
2D	Printing Press Roller
2E	Computer Memory Drum
Rotors Where Axial Distribution Of Unbalance Is Not Known	
2F	5-Stage Centrifugal Pump
2G	Multi-Stage Pump Impeller
2H	I.P. Steam Turbine Rotor

Reduced Tolerances

ISO 1940 does not provide recommendations on how to apply tolerances to outboard correction planes, but it is obvious that static (actually quasi-static) residual unbalances in these planes will create larger bearing forces than the same residual unbalances, if they occurred between the bearings. Therefore, rotors with significant outboard components are often balanced to quality grade G 0.4 or even better. For these grades ISO 1940

[2] specifies that the rotor be balanced "in its own housing and bearings under service conditions," but this requirement is ignored. Sometimes a final balancing step, using correction planes I and VI, is added. These measures do not really address the problems caused by dis- and reassembly or internal moments. In fact, it appears that such extremely low tolerances are mostly derived from the minimum unbalance that available balancing machines can achieve, and not from rotor considerations.

Unbalance Effects in Different Correction Planes

Each leg of the residual couple unbalance in planes III and IV will increase only slightly when translated into the correction planes I and II of the impeller, namely by the ratio a/b shown in Figure 6.

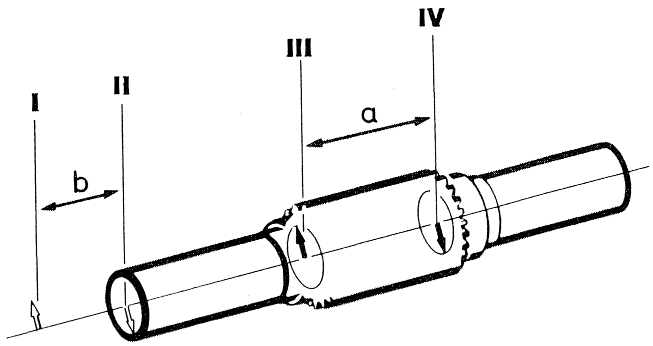


Figure 6. Translation of Couple Unbalance in Planes III and VI to Planes I and II.

However, residual static unbalance of 1 U_{per} each in planes III and IV will translate into approximately 6 U_{per} in plane I and 8 U_{per} in plane II, assuming the dimensions of b and c shown in Figure 7.

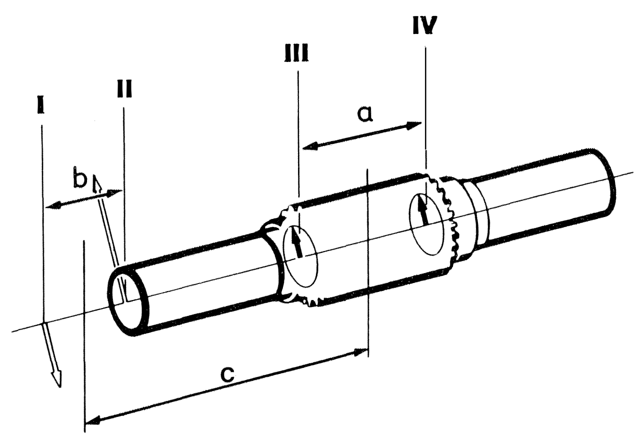


Figure 7. Translation of Static Unbalance in Planes III and IV to Planes I and II.

As can be seen from the preceeding example, small inboard balance errors may translate into significantly larger values, if corrected in outboard planes, particularly if these outboard planes are spaced closely together.

Assume that the previously discussed pinion shaft has been balanced to its recommended tolerance. This tolerance increases six to eight times, if translated into planes I and II. If the first impeller is mounted and the assembly unbalance corrected in planes I and II, the correction will have included the (translated and therefore amplified) effect from the pinion shaft residual unbalance, thus putting an internal bending moment into the shaft. This obviously is contrary to the objectives of a Class 2 c balancing procedure.

The same problem may affect the next assembly step (Figure 8). A residual couple unbalance in planes I and II is only slightly affected in magnitude if translated to planes V and VI, because distances b and d are nearly equal. However, residual static unbalance will appear with approximately 15 U_{per} in plane V and approximately 13 U_{per} in plane VI, assuming d and e approximately, as shown in Figure 8. If the assembly unbalance indicated in planes V and VI is then corrected in these planes down to a level where the indication is 1 U_{per} or less per plane, an unbalance of 14 U_{per} to 16 U_{per} has been put into these impeller planes.

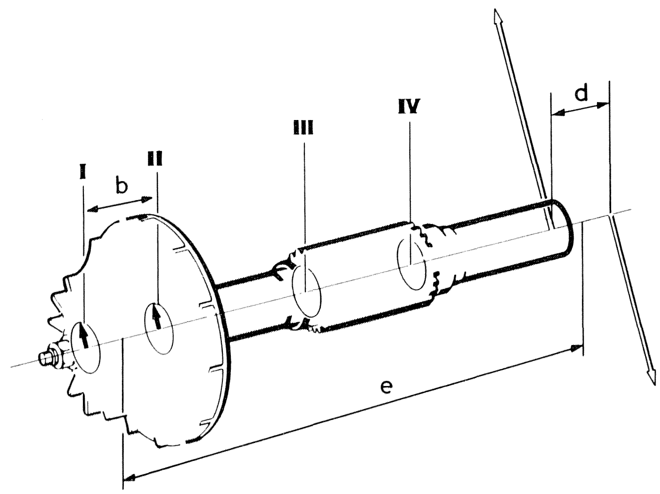


Figure 8. Translation of Static Unbalance in Planes I and II to Planes V and VI.

The examples illustrated would seem to be the worst possible cases. They show what could happen if standard (inboard) balancing procedures are applied to rotors with outboard correction planes.

If the unbalance readings for plane V and VI could be corrected to "zero," the balancing machine would show "in tolerance," no matter which combination of correction planes were chosen. However, the individual assembly components, if checked by themselves, would be mostly out of tolerance (Figure 9).

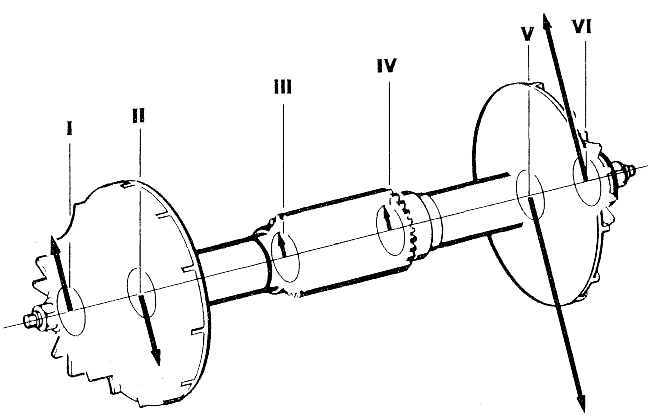


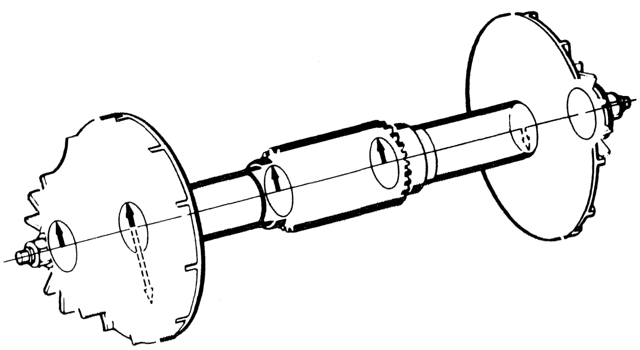
Figure 9. Residual Unbalances in Planes I to VI After the Standard Procedure.

Special Procedures

There are two different procedures that can be used to improve the situation:

Procedure One. After balancing the pinion shaft to its tolerance in planes III and IV, readings for planes I and II are taken *before* mounting the impeller. This serves to establish new "zero points" in these planes. A tolerance circle is drawn around these zero points. Then the impeller is mounted and the unbalances are corrected in planes I and II so that the residual unbalances fall within the tolerance circles.

Next, the readings for planes V and VI are taken, tolerance circles extablished, impeller 2 mounted and balanced in planes V and VI to these tolerance circles. This rotor should now be in far better condition to pass through the critical speeds than if balanced according to the standard procedure, since balance tolerances were maintained in all correction planes (Figure 10).



residual unbalances

final unbalance corrections

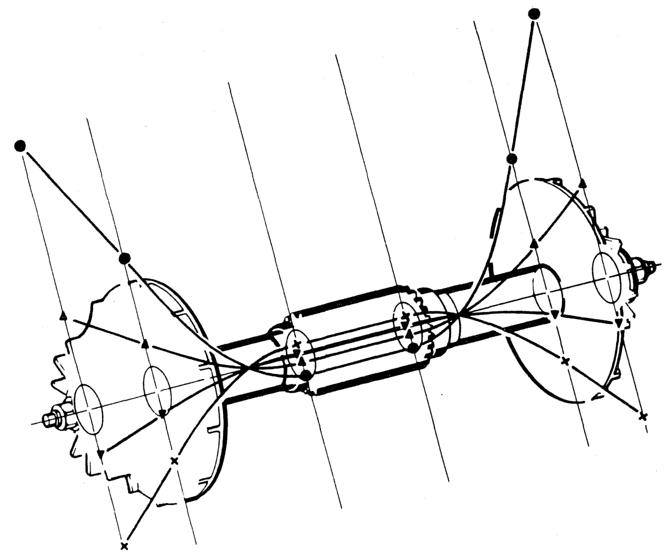
Figure 10. Residual Unbalances in Planes I to VI After the Special Procedure and Final Correction in Planes II and V.

But, there is still a problem. The residual unbalances in the six planes are independent from each other. If these residuals mainly form couple unbalances, there is a good chance that they don't all add up, but are also subtracted from each other to a certain extent. Related to the bearing planes, they might well stay within the desired tolerances. If they mainly form static (quasi-static) residuals, they could add up to 3 U_{per} as a maximum. Naturally, there is always the possibility to stay within the overall tolerances by reducing the tolerances for all three steps to one-third of the standard U_{per} values. Another possibility would be a final (fourth) balancing step, correcting in planes I, II, V, and VI to standard tolerance values. A correction in planes II and V to zero is shown in Figure 10. This would lead to a well balanced rotor at low speed.

To determine the rotor behavior at the critical speeds, it is necessary to consider the typical bending modes of the system (Figures 11 and 12), assuming two flexural critical speeds are of interest.

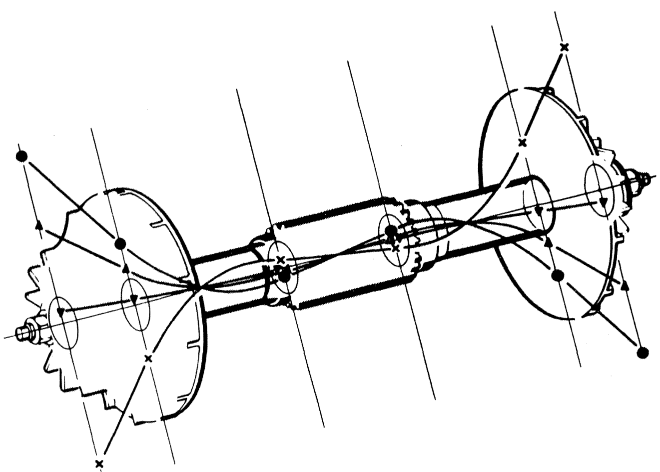
Based on these modes and the residuals left with the different balancing procedures, the following statements on the tendency of the rotor to bend can be made:

- The standard procedure typically leads to a high degree of bending.
- The special procedure without a fourth balancing step tends to improve the situation.
- The special procedure *with* a fourth balancing step reduces the bending considerably if planes III and IV are balanced to reduced tolerances in the beginning.
- Static residuals mainly excite the first bending mode.
- Couple residuals mainly excite the second bending mode.
- A good compromise would be to distribute the corrections equally to planes I and II as well as to V and VI.



x——x first bending mode
●——● bending acc. to figure 9
▲——▲ bending acc. to figure 10, without fourth step
▼——▼ bending acc. to figure 10, with fourth step

Figure 11. Typical Bending Mode (First Critical Speed) and Degree of Bending After Different Balancing Procedures.



x——x second bending mode
●——● bending acc. to figure 9
▲——▲ bending acc. to figure 10, without fourth step
▼——▼ bending acc. to figure 10, with fourth step

Figure 12. Typical Bending Mode (Second Critical Speed) and Degree of Bending After Different Balancing Procedures.

Procedure Two. Since the static residuals always have the larger influence when using the standard procedure, separate tolerances for static unbalance could be set, e.g., one-third of the full value in planes III and IV, one-fifth in planes I and II. This helps to avoid large residual couple unbalances. Naturally, the two procedures can be combined.

Practical Applications

To balance properly, the balancing machine must achieve accurate plane separation and calibration, even for outboard correction planes and very small unbalances.

The method described under *Procedure One* is mainly restricted to hard-bearing machines (with permanent calibration), because plane separation is set without influence from rotor mass. This means the accuracy of plane separation and amount/angle indication can be checked with the impeller mounted, and will still hold true when the pinion shaft is run by itself for the purpose of establishing the new zero points. Balancing to these new zeros points can be facilitated by using a compensator network to provide undistorted readings for the necessary correction. *Procedure two* can be performed with both hard-bearing and soft-bearing balancing machines, if plane separation can be achieved for static/couple (more accurately: quasistatic/couple) unbalances.

Conclusion

Procedure one allows for "standard" tolerances with a similar, or even better, result for high speeds than the very tight tolerances practiced. *Procedure two* requires tight static tolerances, but at least the residual couple unbalances can be of the "standard" size. This conforms with the observation that balancing machines provide better accuracy for indications of static unbalance than for couple unbalance in narrow, outboard planes.

Balancing As Class 3 Rotors

High speed balancing usually requires speeds up to the service speed. For bladed rotors, a vacuum chamber is necessary to reduce the drive power and thermal problems (Figure 13).

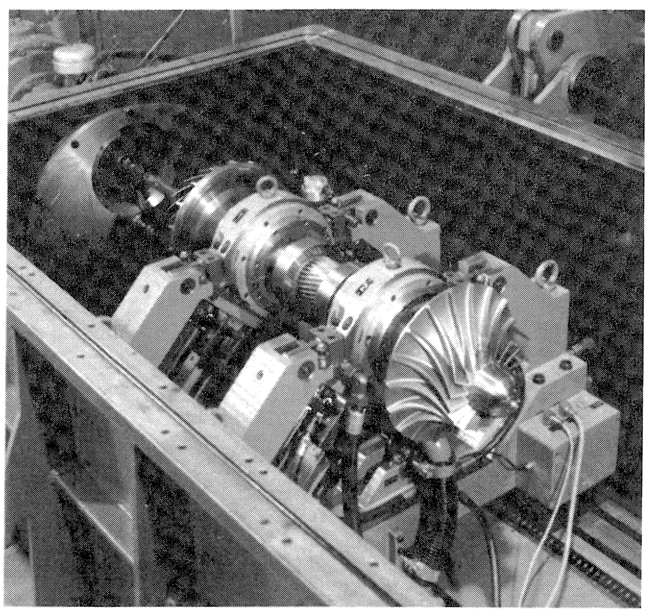


Figure 13. Pinion/Impeller Assembly in Pedestals of a High Speed Balancing Machine Inside a Vacuum Chamber.

Balancing Methods

ISO 5406 [1] lists and recommends three methods to balance Class 3 rotors. In all cases, trial unbalances are applied to the correction planes. By observing the changes in vibration at certain speeds, corrections are calculated to improve the initial vibration level.

Modal Balancing

By using basically one correction for the first critical speed, two for the second and so forth, all amplifications of vibration near resonances are reduced considerably. Since no low speed balancing is performed, the vibration level throughout the speed range might not be low enough.

This is interpreted as the influence of modes outside the speed range. Normally, two additional correction planes are necessary to improve the vibration to tolerances without disturbing the levels at the critical speeds. This method therefore sometimes is called the n + 2 method (n being the number of critical speeds and 2 the additional correction planes).

Combined Rigid Rotor and Modal Balancing

Here the rotor is balanced first in two planes at low speed, i.e., where it still behaves as a rigid rotor.

After that the basic vibration level can be expected to be quite good, although some improvement is still necessary near critical speeds.

High speed corrections should only influence the modes and not disturb the rigid rotor balance. This means for the first critical speed at least three correction planes are needed, for the second four, and so forth, but the correction planes for the rigid rotor balance can be used here, too. This method is sometimes called the comprehensive modal balancing technique.

Influence Coefficient Method Balancing

This method uses a computer system for data acquisition and processing, thus simplifying the work for the operator (Figure 14).

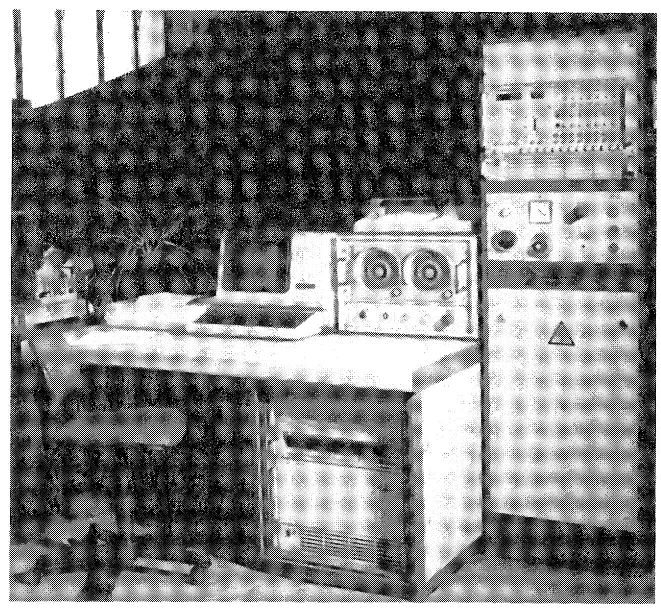


Figure 14. Computer Aided Balancing of Flexible Rotors.

After many years of trying to apply this method without regard to the flexible rotor mode shapes and other criteria, it is now clear that its success depends on a good choice of correction planes, trial weights (or sets of these) and relevant speeds.

Choice of Balancing Method

The initial unbalance of the compressor rotors (mainly the impellers) is so large that the centrifugal force will be a multiple of the static bearing load long before the first critical speed is reached. This means that rigid rotor balancing is necessary and leads to method two. If this prebalancing would be done for the individual components, then method one could also be used in a high speed installation.

A computer system (method three) could be applied in any case, but the necessary procedure depends (as mentioned above) on the initial unbalances.

Mode Shapes

Due to the comparably stiff pinion, the flexible shafts and overhung impellers, the first and second critical speeds are very close together. It may be that sometimes they can't be separately observed at all. Which bending mode occurs first—bearing signals having the same or opposite phase—depends on design values of the system. The example presented shows the in-phase vibration as the first bending mode.

The mode shapes for the first and second critical speeds are shown in Figures 11 and 12. The ordinates of the curves depict how sensitive the different planes are; i.e., for the first mode plane I is approximately twice as sensitive as plane II.

To compare the amount of rotor deflection due to different unbalance distributions, the simplified approach under *Special Procedures* and in Figures 11 and 12 is used regarding the vector directions, splitting vectors into 90° components if necessary.

Correction Planes

For the first mode, six correction planes are more than sufficient. Since the rotor is nearly symmetric, it is recommended to use planes III and IV as one plane, thus distributing the corrections. One plane at each end is necessary and sufficient. Planes I and VI are extremely sensitive to correction (see mode curves) but it can be assumed that the larger initial unbalances occcur in planes II and V, and these are of greater importance.

For the second mode, each rotor end should have two correction planes (I, II and V, VI). This means all six planes are used.

Tolerances

ISO 5343 [4] recommends two different methods:

Vibration level. With the help of a set of conversion factors, the permissible vibration severity in service bearings is reduced to a once per revolution value in the high speed balancing machine.

Permissible residual unbalances. These are derived from ISO 1940 [2] in terms of U_{per} for a rigid rotor. Different limits are set for low speed balancing and the "equivalent modal unbalances" for each mode.

Limits applicable to the centrifugal turbocompressors under discussion are:

- 100 percent U_{per} for low speed balancing
- 30 percent to 70 percent U_{per} as equivalent first modal unbalance (depending on degree of damping)
- 30 percent to 70 percent U_{per} as equivalent second modal unbalance
- 200 percent to 500 percent U_{per} as equivalent third modal unbalance (depending on the distance of the service speed to the third critical speed).

Reasons for High Speed Balancing

High speed balancing is the best way to find out about the high speed behavior of a flexible rotor (Figure 15).

Even Class 2 rotors should be balanced at high speed, if that turns out to be the more economical solution. The choice between low and high speed balancing must consider possible rejections if only low speed balancing is carried out. If a Class 1 or 2 rotor also behaves as a Class 4 rotor, the unbalances which will excite the bending at the critical speeds change with the speed. This means they cannot be corrected sufficiently at low speed.

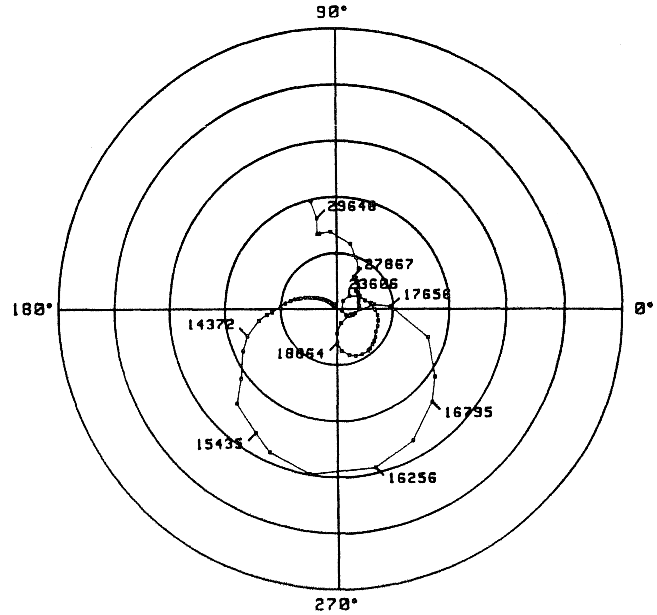
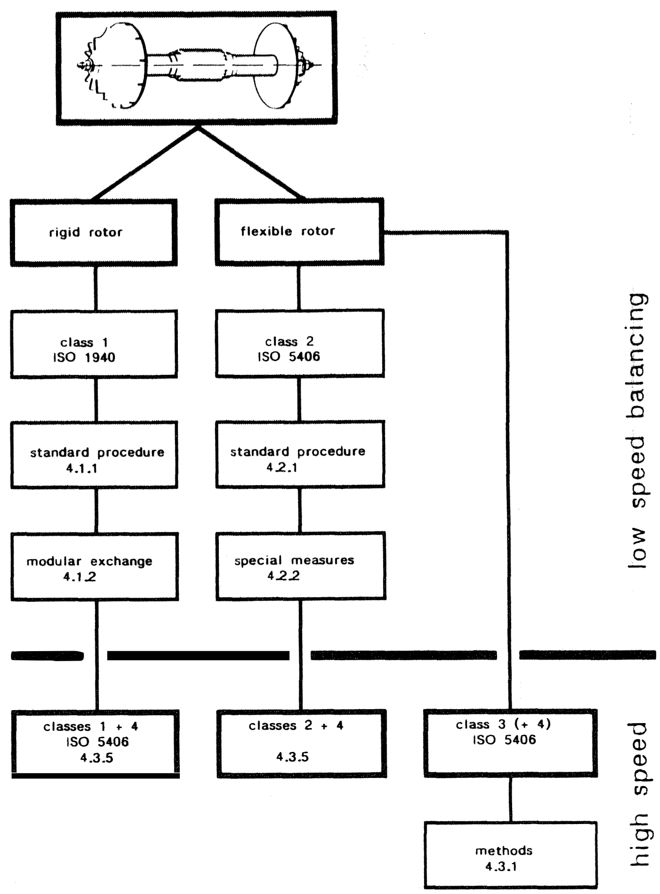


Figure 15. Bearing Signal (Nyquist Plot) of a Rotor Balanced as Class 3 (Full Scale: 0.5 mm/s).

CONCLUSION

The different requirements and methods to balance gear driven centrifugal compressors are summarized on Table 4.

Table 4. Summary of Different Balancing Tasks and Approaches.



None of the various procedures can be considered as *the* industrial standard. Depending on the vibrational behavior of a rotor system, its design characteristics and intended use, several balancing methods are practiced.

This discussion serves to illuminate the theoretical background and the reasons why a particular procedure leads to certain results.

Thus, the reader should be able to evaluate and possibly improve the procedure used heretofore.

Optimizing the balancing procedure can provide significant savings in manufacturing and maintenancce, while at the same time improving operating efficiency.

REFERENCES

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