HIGH SPEED BALANCE PROCEDURE

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ABSTRACT

The paper describes elastic balancing procedures and experimental results for turbomachine rotors. The commonly known criteria of static and dynamic balancing no longer apply in this case.

The rotating elements are balanced at full speed in their contract bearings. Dynamic bearing forces are measured with respect to direction and magnitude. Balancing weights are attached to the rotor in several planes and the thermally caused deflections, nonsynchronous vibrations and natural frequencies of bearing supports have to be accounted for. The rotor sensitivity can be determined by attachment of test unbalances. Dynamic bearing forces and shaft vibrations are measured and the shock excitations and log decrement calculations yield the stability limits.

INTRODUCTION

Balancing turbomachinery rotors at low speed is a commonly used practice. It is, however, very often the case that such balanced rotors do not have the required running smoothness after startup. To achieve low vibration amplitudes and dynamic forces during operation, it is better to balance the rotors at full operating speed after overspeed testing (high speed or elastic balancing).

A bunker with thick concrete walls, as illustrated in Figure 1, is required. This bunker must be sealed off, air-tight and evacuated to a vacuum level of approximately 5 mbar. There are two main reasons for this. One is to keep the energy requirement at a low level; the other is to avoid non-allowable heating resulting from ventilation losses. It is necessary to run the rotor in its contract bearings in order to warrant a similar vibrational behaviour as occurs in the machine. Also, in order to detect existing residual unbalances and required corrective balancing weights, the indication of dynamic bearing forces as to amplitude and phase in the rotating system is required. Measurement of shaft vibration must also be possible.

The operation, monitoring and measurement are effected by means of a closed circuit television transmission to a separate control room (Figure 2). The installation, as shown, is

Figure 1. Vacuum Bunker for Balancing and Overspeed Testing.

Figure 2. Control Room.
1. Polar diagram for bearing forces.
2. Absolute value for bearing forces.
3. Oscillograph for shaft vibrations.
4. Shaft vibration indicator.
5. Recorders for speed, shaft vibrations, bearing housing vibrations.
6. Speed control.
7. Control and monitoring board for oil and air.
8. Monitor with test object.
9. Control for strobscopic lighting of bunker.
10. Window to testbed.
suitable for balancing rotors up to 1.80 m in diameter, 6 m in length and 15000 kg in weight, with speeds up to 25000 rpm.

WHY IS HIGH SPEED BALANCING NECESSARY?

When balancing at low speed, the rotor behaves as a rigid body. By applying corrective unbalance weights, Uc, at two planes selected at random (I and II in Figure 3a), specified balance qualities can be achieved. The axial location of the residual unbalance, Uc, remains unknown and is not neutralized at the point of occurrence. If the rotor later deforms elastically at high speed (Figure 3b), then the corrective weights could be useless. The attempt is made to eliminate this disadvantage by incremental balancing during the assembly of the rotor.

![Figure 3](image)

Figure 3. Low Speed Balanced Rotor. 
a. Rigid  b: Elastical

Each part is shrunk separately onto the shaft. Preferably after each shrinkage operation, but at least after each second one, the shaft is balanced at low speed. Corrective weights may only be applied to the last parts which are mounted on the shaft. These then can only be located on that part of the shaft to coincide with the residual unbalances.

This procedure provides a reasonable level of balance, but not 100% assurance for smooth running of the completely assembled rotor at full speed. It cannot be ruled out that some parts shrunk onto the shaft will be displaced as a result of the centrifugal force. There is then a change in the balance condition as opposed to the condition at the start. If shrinkage is inadequate at certain points, some parts will loosen or internal friction will result. A loosening of parts could result in unbalances varying with time and could thus lead to vibrations of a higher amplitude. Internal friction can cause instabilities, meaning nonsynchronous vibrations which also have higher amplitudes. It is not possible to carry out incremental balancing on rotors which are manufactured in one piece, and such rotors are quite numerous.

It is advisable to check and correct the rotor under those conditions at which it is later to be operated. This is the full operational speed.

BALANCING LIMITS

The correct approach to the balancing procedure is to clarify beforehand the criteria of assessment and to determine the tolerances which are to be applied. Limits for residual unbalances, U, for low speed balancing are specified in international standards [1, 2, 3]. The test object has mass m, thus the eccentricity of the center of gravity (c.o.g.) is U/m. The speed of the c.o.g. is limited to a value of G. For turbo rotors, G is between 1 and 2.5 mm/s.

If \( \omega \) is the angular velocity at full speed, the following limit applies for low speed balancing:

\[
\omega \frac{U}{m} < G
\]

Based on model laws, it can be shown that the dynamic stresses in the rotor and the bearings are the same for all machine sizes if this limit is adhered to. Defining \( n \) as the operating speed in rpm, m in kg and U in g-mm, the following is derived from equation (1) (see Figure 4):

\[
U < \frac{9550 \times G \times m}{n}
\]

![Figure 4](image)

Figure 4. Allowable Unbalance for Low Speed Balancing as per ISO.

If the rotor behaves elastically, individual unbalances distributed along the axial length lead to different deformations which depend on the speed. It is no longer sufficient to limit the magnitude of the unbalance according to the equations (1) or (2). Neither is it possible to compensate the effect of existing natural unbalance in two levels selected at random.

The following values can be used for measurement criteria and tolerances when balancing rotors at high speeds:
Equivalent Unbalance [4]

Residual unbalances are distributed at random over the axial length of the rotor. Deflections result at the rotor bearings at a certain speed, \( n \), and take the form of vibrations or forces. These are measured in a coordinate system rotating with speed and represented by the vector \( \bar{A} \) in amplitude and phase (Figure 5a). Then, a test unbalance, \( U_T \), is attached, and its effect depends on the flexural form of the rotor. Thus, the axial location of \( U_T \) is to be adjusted to the speed in question.

The size of \( U_T \) is selected as per equation (2) with, for example, \( C = 1 \). With the same speed as in the case of the first measurement, a deflection results which is represented by the vector \( \bar{B} \) (Figure 5b). The difference \( \bar{B} - \bar{A} \) is the effect of the unbalance \( U_T \) on the deflection (Figure 5c). Then, as will be shown later, corrective unbalance weights, \( U_C \), are to be placed in such a way that the deflection \( \bar{C} \) remains smaller than \( \bar{B} - \bar{A} \) (Figure 5d). By applying this procedure, only pure unbalances are compared to each other; thus having the advantage of being independent of the boundary conditions resulting from the bearings and the measuring device.

The disadvantage of this procedure is that vibrations and forces occurring during the later operation at the site are not directly limited. We do not apply this procedure due to its obvious disadvantages. We have gained sufficient experience with the method described in the following sections.

Shaft Vibration Amplitudes

The objective of the balancing procedure is to obtain low amplitudes of vibration over the entire length of the shaft. Limits for vibration amplitudes at the bearings can be found in the standards [5, 6] and are also shown in Figure 6. In addition to the size of the unbalances, the vibrations are influenced by the bearing type and by the bearing supports as well. We use the contract bearings for high speed balancing so that this influence is precluded.

We discovered, nevertheless, that the magnitude of the vibration amplitudes measured in the vacuum bunker does not always coincide with the measurements recorded on the testbed or in the plant. We do not use these vibrations when assessing balance quality. They are, however, subjected to regular checking in order to detect any nonsynchronous vibrations.

Amplitude of Bearing Housing Vibrations [4]

If one observes the pure vibration amplitudes of the bearing housings, one discovers that these are considerably influenced by the bearing support stiffness. In order to compensate for this we apply the measured support spring deflections as follows.

Dynamic Bearing Forces

First the vibration amplitudes of the bearing pedestals are measured. Multiplied with the coefficients of the test springs the bearing housing is supported on, the dynamic bearing forces \( \bar{F} \) result. These are exerted from the rotor to the bearings.

With \( B \) as the bearing width and \( D \) as the bearing diameter, the dynamic bearing compression is \( \bar{F} = \bar{F}(B-D) \). The applicable limit for use is,

\[
\bar{F} \leq 0.3 \text{ MPa}
\]

This value approximately corresponds to the dynamic force for each bearing,

\[
\bar{F} \leq 0.1 \times m \times g \times \frac{n}{6000} \text{ N}
\]

where \( m \) is the rotor mass in kg, \( n \) the maximum operating speed in rpm and \( g = 9.81 \text{ m/s}^2 \), the gravitation constant.

The smaller limit, as per equations (3) and (4), is applied as the final limit for the high speed balancing procedure. This is the assessment of measurement variables at full operating speed. One obtains lower dynamic values if the bearing forces that have been measured during the low speed balancing procedure, are extrapolated to the operational speed. Looking back on the last ten years of experience in this field, we discovered that the required vibration values as shown in Figure 6 were always met during plant test runs and startups, provided of course, that the bearing forces in the bunker were limited as per equations (3) and (4).

One disadvantage of the selected limitation is the occurrence of a natural frequency resulting from the measuring device.
BALANCING AND OVERSPEED TESTING

First of all, every rotor is balanced at low speed. If possible, incremental balancing is carried out as stated earlier. This is done to simplify the elastic balancing procedure. After low speed balancing, the completely assembled rotor is subjected to examination.

Overspeed Testing

The rotor is accelerated to overspeed level. This speed is 15% above the maximum operating speed. In this case, 1.8 MPa is the limiting value of the dynamic bearing compression, and this is six times the value of the final balancing procedure. Figure 7 shows, as an example for a compressor rotor, the behavior of the dynamic bearing compression versus the speed before and after the overspeed testing. A worsening of the balanced condition can be clearly seen here. This behavior is not always the case. Generally, unbalance changes only slightly or not at all during the overspeed testing procedure.

If, after overspeed testing, the normal dynamic bearing forces are not exceeded, the rotor is delivered. In all other cases, modal balancing is carried out in order to meet the limits for the dynamic bearing forces.

General Remarks on Balancing Procedures [7]

Elimination of the effects of the unbalance in any of the axial planes of the rotor is done as follows:

- In Figure 8a the natural unbalance is represented by the bearing force $\bar{X}$.
- The bearing force $\bar{Y}$ represents the natural force plus a test unbalance $\bar{U}_T$. The resulting vector $\bar{Y} - \bar{X}$ is the effect of the test unbalance.
- In order to compensate for the residual unbalance, a corrective unbalance $\bar{V}_T$ is to be attached. Its size is $U_T \frac{X}{\bar{Y} - \bar{X}}$, its direction is to be turned by the angle $\alpha$ as opposed to that of $\bar{U}_T$ (see Figure 8b).

Balancing in the First Mode

This balancing is to be applied for speeds where the natural form of the rotor is predominantly determined by the first natural frequency. The test and/or corrective unbalance weights shall be applied near the shaft center. This correction, however, change the balancing quality achieved during the low speed balancing procedure. If this must be avoided, three unbalance weights shall be applied (Figure 8c) instead of the unbalance only in the shaft center. In addition to $U$ in the shaft center, two unbalances are attached near the bearings for test as well as for corrective unbalances. Their sizes are $U/2$ each and their phase angle is at 180° to $U$ [8].
Balancing in Higher Modes

This balancing procedure makes sense for speeds where the mode of the rotor is determined by higher natural frequencies. In this case, test and corrective weights are to be attached in at least two planes.

Two additional planes are necessary if unbalance at low speed balancing is to be limited in addition to limiting bearing forces at full speed. In this case, a number of test runs are necessary. Calculation of corrective weights based on the results of the runs can only be carried out by sophisticated formulas. There are programs for pocket calculators for this.

ACCOMPANYING PHENOMENA

Up to now, the principle of balancing was discussed. There are, however, other accompanying phenomena which are discussed as follows:

Natural Frequency of the Measuring Device

The bearing housing is spring supported in order to measure the bearing forces. Natural frequencies occur here which are independent of those of the rotor. The level of natural frequency depends on the bearing test stands used as well as on the weight of the rotor. Figures 9 and 10 compare measuring results for the same rotor, in the bunker and on the testbed. The first natural frequency of the rotor is 7500 rpm; the second is above the operating speed of 16500 rpm. The amplitude increase at 12000 rpm is attributable to the bearing stands of the bunker. It does not exist if the rotor is running in the compressor.

Generally, we try to balance in such a way that, despite the natural frequency mentioned above, the limits of the bearing forces can be met. The bearing forces, when the machine is in full operation, are therefore usually smaller than those recorded in the bunker.

Figure 8. Compensation of Unbalance Effects.

Figure 9. Polar Diagram of Bearing Forces in the Bunker.

Figure 10. Bearing Forces in the Bunker and Shaft Vibrations During the Test Run.
Instabilities

The vibrations of the bearing pedestals and the resulting dynamic bearing forces are measured in a coordinate system rotating with speed. Nonsynchronous vibrations are not recorded. These can be caused by the oil film in the radial bearings or through internal friction resulting from inadequately shrunk parts. For this reason, we monitor the shaft vibrations near the bearings and sometimes also at the shaft center in addition to the dynamic bearing forces. Instabilities are indicated by high amplitudes with nonsynchronous frequencies.

As an example, Figure 11 shows the vibration records of an instability. Synchronous vibrations with a low amplitude occur up to 10000 rpm; at 10400 rpm there was a considerable increase in the amplitudes. The vibrational frequency was less than half of the rotational frequency. After a reduction of speed to 9500 rpm the instability had disappeared.

Thermal Deflections

If turbomachinery rotors are not turned during cool-down, different temperatures can result across the shaft diameters. This leads to deflections and thus, subsequently, to incorrect balancing results. One degree Celsius temperature difference over the diameter on the entire axial length leads to a deflection in the range of 0.1 mm. This is about 300 times more than the tolerance on balancing allows.

It is therefore necessary to be sure of the steady state conditions of the test values for every test and check run. Otherwise, the rotor is to be turned until such time as steady state conditions have been achieved. Figure 12 shows the gradual reduction of shaft bending caused by thermal conditions, over time t, for various diameters [9].

![Figure 12. Vibration Decay of Thermally Caused Shaft Deflection with Respect to Time for Various Diameters. h₀ = initial value at point of commencement zero.](image)

EXPERIMENTAL TESTING

In addition to tests directly contract connected, we also carry out experimental testing in our vacuum bunker.

Unbalance Vibrations

These measurements serve the purpose of studying the sensitivity of the rotors, particularly within the natural frequencies. Figure 13 shows shaft vibrations measured on a

![Figure 11. Vibration Recordings.](image)
Figure 13. Vibration Measured at the Compressor Shaft with an Unbalance of G = 7.5.

Figure 14. Test of System Damping.

compressor rotor and dynamic bearing compression versus the
speed. A test unbalance was attached to the rotor as per
equation (2) with G = 7.5.

Decay Function

Above a certain speed, rotors in journal bearings can
become unstable, meaning, nonsynchronous vibrations can
occur. This speed limit depends on the damping μ of the
system which is made up of the rotor and the bearings. μ is
determined from the decay rate of the vibrations after a shock
excitation of the rotor. The quotient μ/V is the log decremen δ
and V is the natural frequency. μ or δ = 0 characterizes the
speed above which there are instabilities. Figure 14 shows
relevant test results.

SUMMARY

High speed balancing of turbomachinery rotors provides a
better assurance for later operation than low speed balancing.
The dynamic bearing forces have proven themselves as mea-
surement criteria of high speed balancing. Additional natural
frequencies of the measuring device can lead to incorrect
results. These influence the balancing operation only as
regards higher level accuracy. Shaft vibrations at the bearings
are to be additionally monitored in order to detect nonsyn-
chronous vibrations. Modal balancing can also be necessary
in order to maintain specified dynamic bearing forces. There
must be no differences in temperature along the rotor before
each test or check run.

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