GEAR VIBRATIONS

by

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1 Introduction

The speed and power of modern turbo-machines is being constantly increased. This, together with the necessity for high reliability, is making the design and operation of these machines increasingly demanding, not least with regard to their vibrations. With this paper it shall be attempted, therefore, to advance the understanding of the corresponding vibrational behaviour, especially of turbo-gears. The content will concentrate therefore on the lateral vibration of the rotors, which generally demands the most attention. The monitoring of the structure-borne noise, to detect a deterioration of the tooth mesh, will also be examined in some greater detail. The explanations are supported by relevant practical examples.

2 Lateral Vibration of Gear Rotors

The lateral vibration of each rotor shall be looked at independently of the rest of the string, by limiting ourselves to flexibly coupled rotors and by applying the so called "half-coupling" modelling. Fig. 1 shows the Lateral Undamped Critical Speed Map of such a rotor as it is required by API. The model of the rotor and its mode shapes were added to the map to help the visualization of the vibration.

In comparison to other rotating machines, gears support with their bearings not only the weight and dynamic forces of their rotors but also the very large forces transmitted by their tooth contacts. Stiffness and damping of bearing oil films, on the other hand, are strongly dependent on the bearing loading, which makes the representation of the resonance frequencies as a function of bearing stiffness and percentage power in Fig. 1, particularly suitable for gears. As a further distinction, high speed gear rotors are designed in flexion very rigidly and, especially pinions, have very large bearings in comparison to their mass. This is not only essential to take up the very large tooth contact forces, but, as shall be shown, has also decisive advantages with regard to their lateral vibration. This specific vibrational behaviour of gear rotors, and the effect of the most important design parameters, shall briefly be demonstrated here by means of the rotor in Fig. 1. The discussion can be limited to the first two resonance frequencies, since, also in practice, the higher ones seldom need to be considered.

The pronounced vibrational motion of the coupling at the first resonance frequency in Fig. 1 indicates a strong dependence of this frequency on the coupling overhang, i.e. a shortening of that overhang or a lowering of the mass of the coupling would effectively raise the first lateral resonance frequency. Particularly effective, and practical, is a shifting of the coupling joint closer to the bearing, as it is accomplished, for example, with our MAAG ZUD couplings. Shortening the overhand by increasing the bearing span has an equivalent effect, at least for lower bearing stiffnesses. At higher bearing stiffnesses, the measure may become less effective beacause of the resulting reduction of the flexural stiffness between the bearings. A general stiffening of the rotor by increasing its diameters will naturally be limited by the diameter of the toothing and by bearing related parameters, primarily the speed; with all its negative effects on the power loss, temperature, etc.

In the complex design optimization of turbo-gears, the selection of the proper coupling is obviously very important and requires, therefore, a competent and prompt cooperation of all parties involved, especially in regard to vibrations. Thus, not only the torsional vibration of the string needs to be considered but, just as much the lateral vibration of the coupled rotors and, not least, also of the coupling itself must be considered. MAAG has a major advantage by being able to provide an integral solution, i.e. to supply gear and coupling, both optimally matched to each other.

Couplings also deserve a closer look because of their pronounced potential to excite lateral vibrations in the coupled rotors, especially by excessive unbalances. Membrane couplings are particularly prone to such excessive unbalances, due to the need to balance them as a complete assembly, which is apparently rather difficult especially in regard to reproducibility.

Based on the mode shapes in Fig. 1, we had concluded, for the first resonance frequency, on a high sensitivity to the coupling overhang. The same logic lends itself to the visualization of an increased sensitivity of the rotor to exitations from the coupling. Obviously, the further away from a mode an excitation acts, the more effective it is in exciting the particular vibration. An equivalent logic applies to the damping, as shall be shown next.

With some experience, Fig. 1 can even be used to estimate the effectiveness of the damping. The mode shapes at lower bearing stiffnesses show hardly any vibrational flexion of the rotor at all, i.e. the vibrational motion within the bearings is optimally large, such that the excellent damping properties of the large bearings are fully exploited. This damping, in fact, is so effective that, at lighter loads, the rotor in Fig. 1 is supercritically damped. During final shop testing, our gears are frequently running at such conditions. Resonance frequencies, i.e. so called "criticals", are then not even descernable, when one passes through them, since traces of amplitude and phase angle remain too shallow.

With an increase in bearing stiffness beyond the range discussed above, the vibration of the rotor is increasingly composed of flexion, i.e. the participation of the bearings in the overall vibrational motion decreases, which can be concluded also from the levelling off of the first two resonance frequency curves in Fig. 1. An increase in the loading of a bearing, on the other hand, not only increases its stiffness but also its damping coefficients. A gear rotor, therefore, can experience highly effective damping, even in spite of this decreasing portion of its vibrational motion within its bearings. The further that portion of the motion within the bearings decreases, the larger the overall vibrational response of a rotor becomes for a given bearing load and excitation. Taking the rotor in Fig. 1 as an example again, its response would show a pronounced peaking, near the coupling, if one would pass at full torque through its first lateral critical at 15 000 rpm. However, even that relatively small motion within the bearing would still be far better than no motion at all, as would roughly be the case for rotor at bearing stiffnesses of 10^{10} N/m and beyond. At these high stiffnesses, practically only the weak damping of the rotor material would be available, which would demand particularly carefully directed measures to limit the excitations, should one have to operate at this condition.

With the constant increase in power and speed, operation near or at lateral criticals becomes increasingly unavoidable even at full loads. The correspondingly increasing demand for accurate balancing is only to be met by trim balancing the strings in the field, which has been decisively simplified by recent advances in instrumentation.

Although the individual rotors can be balanced to a sufficient accuracy in the shop, the excessive unbalance is usually introduced with the mounting of the couplings in the field. Quite frequently, the coupled machines are then blamed first, since the excessive vibrations are signalled through their probes. Gears appear to be particularly vulnerable to such rash conclusions; all the more when operated at lighter loads. The relatively small mass of their rotors and the lower bearing coefficients at lighter loads make them especially sensitive to such excitations from the coupling.

This sensitivity is particularly pronounced for rotors that are pushed upward by the tooth contact force, and when this uplifting force is about equal to the weight of the rotor. This occurs at torques that typically amount to only a few percent of nominal torque. The loads on the bearings are then lowest and, hence, so are the resulting bearing coefficients. Further, in this condition, slight changes in torque cause the bearing force vector to change its direction rather strongly, such that the journal may move quite rapidly through a larger portion of the bearing clearance. The heightened sensitivity coupled with this transient motion may cause the vibration monitoring system to trigger, although this physically unavoidable vibrational motion is entirely harmless. Vibration monitoring equipment should be designed, therefore, with provisions to ignore such transient events which may occure not only during run-up but also during run-down.

This peculiar torsional-lateral interaction of gears already manifests itself, however, at far lower vibrational torques, expecially when lightly loaded. As a matter of fact, the presence of a torsional vibration problem in a string would often not even be realized were it not made visible or audible by lateral vibrations of the gear rotors. Tracing the source of such vibrations can then be correspondingly difficult, particularly since the suspicion is often limited to that sole source of the vibrational signal, i.e. to the gear. Difficulties in relating the frequency of the lateral vibration with the rotational frequencies of the gear, including harmonics and subharmonics, should serve as an indication for possible torsional vibration problems, especially if the lateral frequency happens to coincide with a torsional resonance frequency of the string.

3 Structure-borne and Air-borne Noise

The structure-borne noise of turbo-gears is excited mainly by the meshing of the teeth. Its frequency spectrum is accordingly dominated by the meshing frequencies and their harmonics. A deterioration of the tooth flanks, for example due to wear or electric erosion, will result in an increase of the excitation. The structure-borne noise is transmitted through the bearing oil films into the casing and foundation of the gear and eventually dissipated as primarily heat. Usually only a small fraction of the original vibration energy is radiated as air-borne noise.

In the optimization of the toothing, the vibration excited by the tooth mesh is given prime consideration, especially with regard to the geometry of the toothing, its flank corrections and, naturally, its accuracy. The propagation of the structure-borne noise within the gear and its foundation is affected mainly by the geometric parameters and the materials employed. The proper geometric parameters are also important to minimize the air-borne noise radiated from these structures, especially in avoiding resonating outside surfaces.

Soundproof enclosures may have to be added in order to meet particular stringent specifications. The decision to soundproof an individual machine should only be based on a thorough analysis that encompasses all machines and structures within a given acoustically significant range. Quite often, such costly enclosures are rendered worthless by other machines with similar noise levels and, possibly, an even more aggravating pitch, or by piping, frames or related structures that conduct the noise outside of the enclosure more effectively than anticipated. Effective subsequent extensions of the sound proofing are usually far costlier than the original enclosures, that is, if they can be implemented at all.

4 Monitoring Vibration

On turbo-gears two distinct sources of vibration can be identified: the rotor-bearing system, with lateral and torsional vibrations described earlier; and the tooth mesh. The lateral vibration of the rotors is monitored primarily by noncontacting displacement probes, which measure the gap to the shaft by means of eddy currents. Two such probes, oriented 90°

to each other, are required to provide a complete picture of the motion of a given portion of a shaft. The vibration excited by the meshing teeth is monitored with piezoelectric accelerometers, by measuring the structure-borne noise of the gear casing.

The non-contacting displacement probe provides, electonically processed, a voltage proportional to its distance from the shaft surface. The shaft vibration is proportional, therefore, to the dynamic component of this voltage. Also the static or quasi-static component of this signal is valuable. During the start-up of a gear, the signals from a pair of probes, for example, trace the entire upward motion of a given portion of a rotor, as it is lifted by the tooth contact force. Also, at steady state operation, a slowly changing shaft position may signal an impending bearing failure, possibly before any indication is shown by a temperature probe. Thus, not only the vibration amplitude but also the shaft position should regularly be logged. The resulting trend should never be corrected, however, without fully understanding its cause. Blindly balancing a high vibration out of existance, may forfeit, for example, the last chance to catch a progressing fatigue crack from developing into a sudden failure.

MAAG recommends the following settings, in mils, as vibration limits:

- Alarm at twice the API-limits required for final shop testing, given by $\sqrt{12\ 000\ /n}$, with n being the rotational speed in rpm.
- Shut-down one mil above alarm.

These settings are approximate values which, based on experience, can be modified for specific string and operating conditions.

The monitoring of the rotor vibration with displacement probes has matured in recent years into a most reliable and useful technique, including the instrumentation employed in analysing, logging and displaying the data. This positive evaluation can not be extended, unfortunately, to the monitoring of the tooth mesh, at least not without major restrictions, as shall be outlined next.

The mechanism of excitation of the structure-borne noise in the meshing teeth is highly complex and effected by a large number of design and performance parameters. This complexity is reflected, for example, in the frequency spectrums of gear noise, especially by their characteristic sidebands, which are distributed, often in bewildering patterns, around the meshing frequency and its harmonics. These sidebands result from amplitude and frequency modulations of the tooth mesh and its harmonics. A well known modulation mechanism is the inaccuracy in the toothing, especially accumulated pitch errors, which explains the dominance of the rotational frequencies and their harmonics in the frequency separations of the sidebands. Consequently, many other mechanisms can also cause the modulations, such as thermal flexion of the rotor, misalignment, torsional vibration, or general rotordynamic phenomena, excited, for example, by unbalances. Thus, many effects not directly related to the toothing, or even the gear, can influence the siedebands strongly and, hence, also the structure-borne noise.

The structure-borne noise is further effected by its complex propagation within the gear. The noise, on its way from the source to the accelerometer, can travel a multitude of paths, within the rotors to the bearings and from there within the casing. The noise, with its complex frequency and phase relationships, is modified thereby through superpositioning, reflections, resonances and various other mechanisms. The propagation is affected by the type and size of the gear and by a large number of its more detailed geometric parameters. The parameters of the foundation and the mounting of the gear also play a role, as well as the coupling configuration of the shafts. Even the operating conditions, especially load, or the lateral and torsional vibrations can effect the propagation.

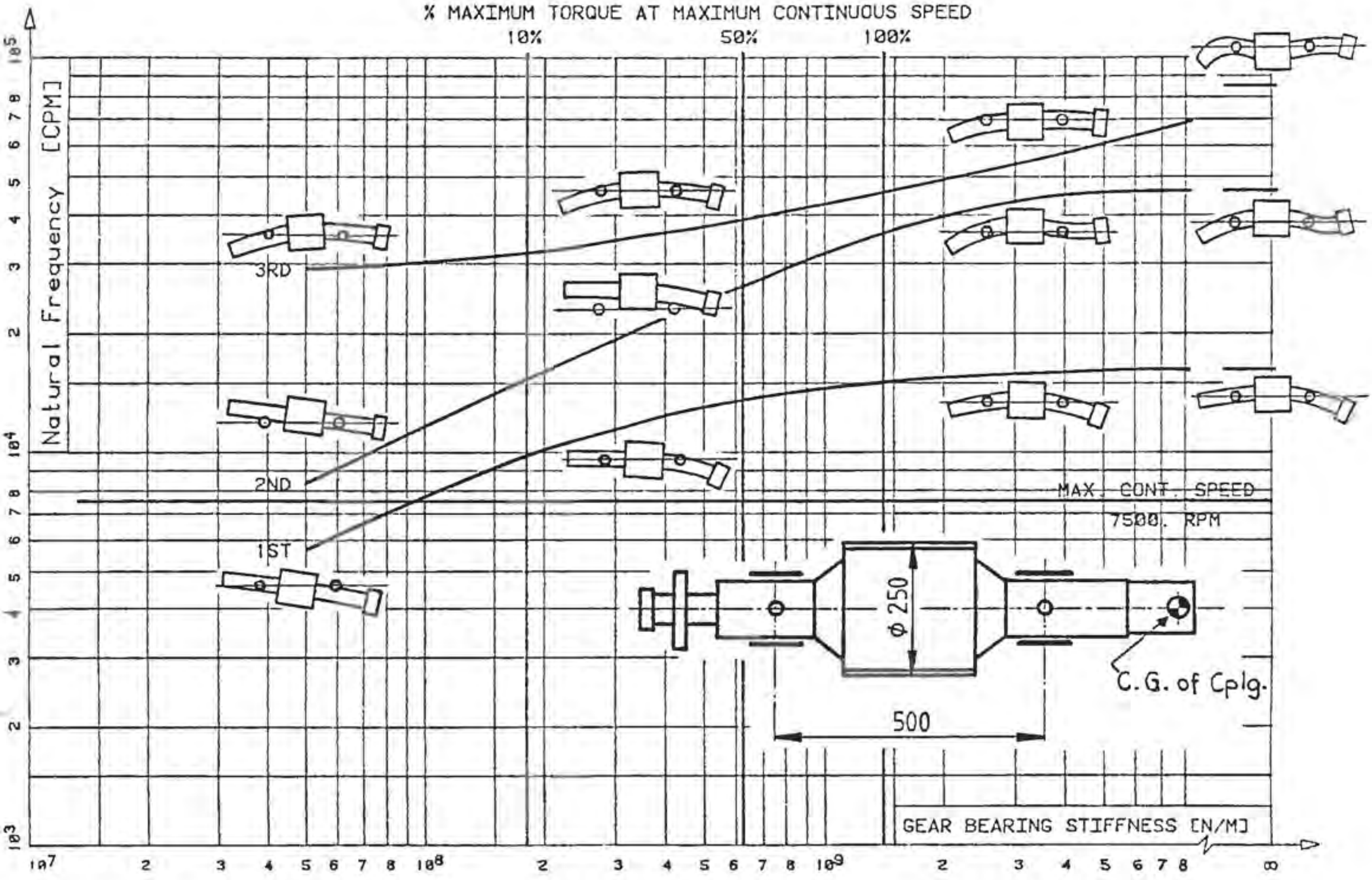
Measurements on gears support this reasoning on the excitation and propagation of the structure-borne noise. Gear noise shows generally rather strong fluctuations, especially in analyses of its frequency spectrum. Changes in position and orientation of the probes have usually also a strong effect. The influence of many parameters is not well understood yet. Some of these parameters often even yield results that appear to be in contradiction with logic.

In view of these difficulties, reliable predictions of the structure-borne noise are practically impossible to make. Too many parameters, partially not even confined to the gear, are interrelated in too complex a fashion to permit the derivation of a sensible expression. This complexity becomes expecially evident if contrasted with the lateral vibration, where just one obvious parameter, the speed, suffices to express a simple yet useful limit. The difficulties in predicting the levels of the structure-borne noise are also reflected by recent activities of various standards organisations. For example an ISO draft standard that contained such recommended limits was reissued with all those limits withdrawn.

The best way to employ the structure-borne noise, in order to catch a deterioration of the transmission, is to monitor its trend. By monitoring just the overall level or its trend, a relevant change in noise may be masked for extended periods by its relatively large scattering or by compensating, and possibly unrelated, effects. Experience actually shows that relevant changes in noise usually are perceived first by our ear; with its fantastic sound analysis capability. A monitoring of the frequency spectrum not necessarily constitutes an improvement because of the even more pronounced scattering and the even larger difficulties in defining the more sophisticated acceptance criteria. Also, monitoring of the velocity, by integrating the signal from the accelerometer, can hardly be advantageous. On the contrary, the resulting emphasis of vibrations at lower frequencies will mask the mesh related signals even further, without relevantly contributing to the monitoring of the vibrations at lower frequencies, since these are already effectively covered by the non-contacting displacement probes.

All these reservations on the monitoring of the structure-borne noise of gears could give rise to the impression that the method is of little value for gears. Such a conclusion was not intended. The analysis of the structure-borne noise is actually especially effective for gears. To develop its full potential, with a high degree of reliability, it belongs, however, in the hands of specialists. Such specialists should have a good knowledge of gears and extensive experience in measuring and analysing structure-borne noise, especially of gears. Some of the most important aspects of this measuring technique shall, in conclusion, be briefly discussed here.

Piezoelectric accelerometers which are used on turbo-gears, are usually rather small in size, in order that they have their resonance frequencies well above the tooth meshing frequencies. The resulting small size of the accelerometers is, however, at the expense of their sensitivity. A resonance frequency that is three times higher than the frequency of the vibration to be measured will impair that reading, causing it to be read about 10% too high. With a closer proximity to the frequency, this error increases strongly, so that low pass filters have to be used. A proper mounting of the accelerometer is decisive for a reliable measurement. An insufficiently rigid mounting can lead to completly erroneous results. The readings are then usually too high, because of a lowering of the resonance frequency closer to the frequencies of the vibrations which are to be monitored. Accelerometers also have a transverse sensitivity with a corresponding resonance frequency well below the one in the direction of measurement. In the presence of strong transverse vibrations, this transverse sensitivity can also lead to erroneous readings, especially if additionally amplified by an inadequate mounting. The machine surface on which the accelerometer is mounted must be rigid, i.e. free of any resonances that could effect the readings. This surface should also be in rigid connection to the bearing structure. Alarming reports from the field of high readings of structure-borne noise can frequently be traced to any of the above described resonance-related problems, i.e. to an improper selection and/or mounting of the accelerometer. Improper use of the instrumentation and connecting cables, as well as incorrect calibration procedures, usually add to these problems.



% MAXIMUM TORQUE AT MAXIMUM CONTINUOUS SPEED
10%
50%
100%
Natural Frequency [CPM]
3RD
2ND
1ST
MAX. CONT. SPEED
7500. RPM
ϕ 250
500
C.G. of Cplg.
GEAR BEARING STIFFNESS [N/M]