PECULIAR CASES OF PINION VIBRATION IN PARALELL SHAFT DOUBLE HELICAL GEAR UNITS – IDENTIFICATION AND MITIGATION

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Abstract – Speed increasing parallel shaft double helical gear units are commonly used in centrifugal compressors and high energy pump applications. Such turbo-gears are API 613 compliant (Titled as -Special Purpose Gear Units for Petroleum, Chemical, and Gas Industry Services) and have horizontal offset transmission line, with journal bearings on the high speed shaft and low speed shaft. When a full load full speed complete unit test is carried out, pinion dynamic behavior is found to be different than what observed during full speed no load condition as mandated in API 613 . During this test, gear box high speed pinion may show up with high vibration or combination of both during certain load and speed combination in spite of proper balancing and alignment. Since full load full speed test is kept as an option in API 613 , hence these type of vibration may not be detected during a no load test in gear box manufacturer's premises. These types of vibration patterns are not explicitly addressed in API 613 .

The first objective of this paper is to cover analytical, design and diagnostic aspects which can be helpful to mitigate above issues before it is towed out from manufacturer's premises.

The second objective of the paper is to suggest installation /innovation / design modification on high speed pinion as a pre-emptive approach which can save time to mitigate if encountered during any certain speed –load combination.

Keywords – Sub -synchronous , relief , super-synchronous , tuning , nodes, FEM

Introduction-

Normal Vibration phenomena of a gear box includes 1x vibration, hunting tooth error , impacting and large side bands across the gear mesh frequencies . These can be easily identified by various techniques and mitigation action can be chalked out . Sometimes , baffling vibration phenomena such as sub- synchronous / super- synchronous vibration occurs which are not well cataloged in vibration analysis text books and international standards such as API 613 and ISO 6336.

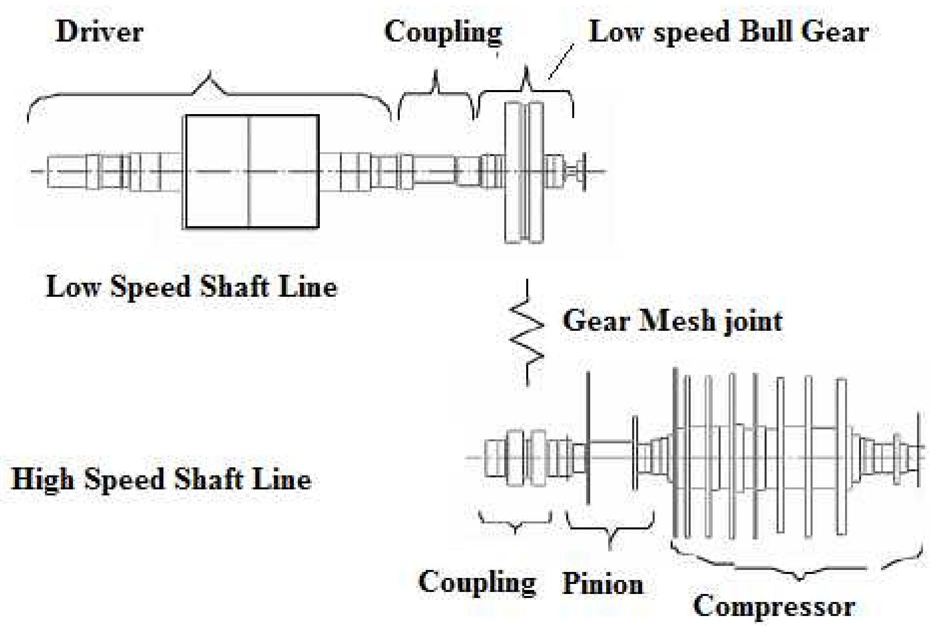

Fig -1 - Torsional Model of a turbo compressor train

Mandates by API 613 5th edition for design and testing -

As per API 613 latest edition, the design of rotor-bearing systems, consideration is given to all potential sources of periodic forcing phenomena (excitation)
The design aspect also covers resonances of structural support systems that may affect the rotor vibration amplitude that occurs within the specified operating speed range or the specified separation margins[1]

As mandated in API 613, three lateral critical speed modes are generally of concern. The frequencies at which these modes occur vary as a function of the transmitted load, primarily due to the resulting stiffness change of the bearing-oil film due to varying stiffness.
API addresses that sometimes it is not possible to operate the gear rotor below the first critical speed. When a gear rotor must operate above the first critical speed, the rotor should be designed with the first critical speed about 60% of operating speed. First critical speeds below 60%of running speed are prone to create bearing instability or oil whip and if running much above 60% are too close to running speed.
2. Gear wheels that operate between 40% – 50% of pinion speed (2:1to 2.5:1 gear ratio) can excite instability in the pinion bearings. Therefore, if gear ratios between 2:1 and 2.5:1 are required special consideration should be given to performing a pinion stability analysis. Instability phenomena at bearings can be caused by excitations having frequency around the half of the shaft speed. This is the reason why the API requires the stability verification when the gear ratio is around 2 to 2.5.
Gear Box are tested as mechanical running tests at OEM (Original Equipment Manufacturer)works as per API 613 and following measurements are taken -
a. Inlet oil temperature and pressure.

b. Oil flow.
c. Outlet oil (drain) temperature.
d. Shaft vibration, frequency, and amplitude, both filtered (synchronous) and unfiltered
e. Bearing pad temperatures.
f. Casing vibration

During API 613 compliant mechanical run test (MRT) of gear box, following criteria are explicitly followed –

1. Steady state shaft and casing vibration - (cl.2.6.6.5 & Table 9)
2. If the amplitude of any discrete, nonsynchronous vibration exceeds 20% of the pinion shaft (or 40% for shafts running below 2000 rpm) of the allowable vibration as defined in 2.6.6.5, the purchaser and the vendor shall mutually agree on requirements for any additional testing and on the gear unit's suitability for shipment.(cl. 4.3.2.2.9)
3. Unbalance response test, if specified –(cl.2.6.3.1)

Dynamics of a pinion in parallel shaft double helical gear unit

The dynamics of high speed pinion of such turbo-gear is significantly different than a centrifugal compressor / turbine / pump rotor. The bearing load on a single casing multistage compressor / pump is approximately half of rotor weight while for a pinion due to torque force developed in a meshing gear, can be several times of its own weight .Due to this , bearings of a parallel shaft industrial gear are heavily loaded at particular direction compared to bearings of compressors. The load direction is based on meshing direction of two gear elements. When these gearboxes are tested with no load (5-10% of actual load during MRT) in OEM works, they are to meet acceptance criteria of casing and rotor vibration as mandated by API. During this test, error profile, pitch line error, hunting tooth issues, gear mesh frequencies and its sidebands are identified with a spectral range of 1500 Hz.
The stiffness and damping characteristics of the bearings are functions of the radial load applied to them.
Depending on the tooth geometry the radial gear load will typically be at a 20 degree angle from the vertical, known as the pressure angle. With a centrally located bull gear, a pinion may be 'upward driven' or 'downward driven'. The gear load and the weight of the rotor shall be oriented approximately 20 degrees apart on a downward driven pinion. On the other hand, upward driven pinions have the gear load and weight vectors oriented approximately 160 (180 minus 20) degrees apart. [7] This creates a larger variation in the resultant load angle in off design conditions i.e. during no load MRT. However , during full load full speed test the dynamic behavior normally simulates actual site running condition.

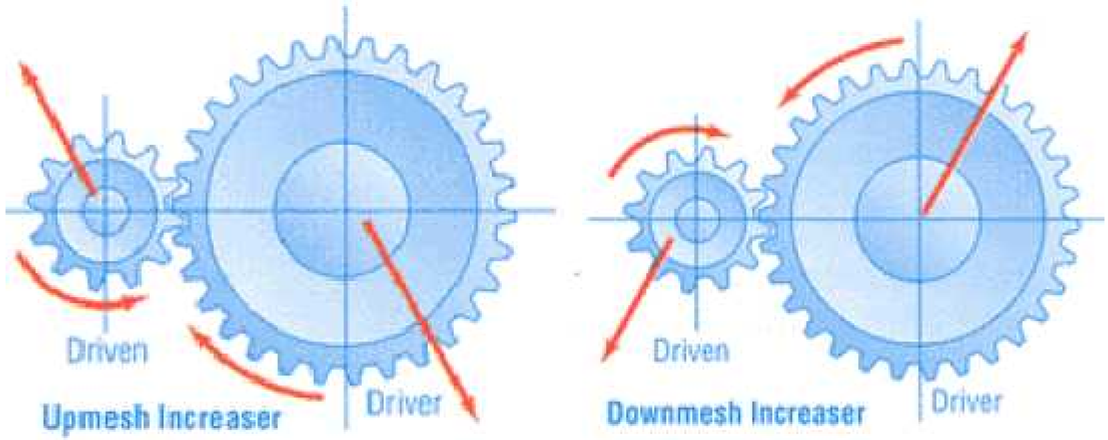

Fig 2 Gear mesh force vectors

Type of Bearings used in high speed pinion of turbo gears -
Bearings play important part in damping the vibrations and sometimes a source of vibration too . Hence, this part gives a brief account of type of bearings used in gear box .

- Cylindrical 2-lobe
- Offset half
- Double offset
- Pressure dam
- Lemon bore
- Tilting pad journal

Axially split journal bearings (Fixed Geometry) are used for the standard design in the gear shafts . If higher speeds are involved (shaft circumferential velocity exceeding 90 m/sec), radial tilting pad bearings are normally used .
For a constant direction of rotation with high load operation (70% -100% load) ,lemon bore bearings are used as a normal practice**.**
If the gear box is to operate at lower speed during driver barring and subsequent change in load direction, pressure dam bearings is a good choice.
The concern for oil film excitation of the rotor lowest natural frequency is eliminated by the use of the tilting pad journal bearings. Tri- pad tilting pad journal bearing has higher load operation capability(P, v) i.e. lower Tmax. (Pad temperature) .Now a days, to reduce power loss, evacuated direct lubrication type journal bearings is used.
For high speed pinion bearings, directed lubrication is used to reduce pad temperatures in high speed, heavily loaded bearings. These bearing designs utilize several design techniques to reduce pad temperatures, including the use of:
• Copper alloy pad materials
• Offset pivots
• Directed lubrication (running evacuated)
• “Spray Bar Blockers” (to reduce hot oil carryover)[5]
• By-pass cooling where extra oil is introduced behind the pad to increase heat transfer.
Evacuated bearing design equipped with spray bar blocker with bypass cooling) are normally used for high speed high load application

Since the loaded pads require the least amount of oil, due to their thin oil films, they typically run with full films while the unloaded top pads run with various degrees of starvation. So , for a pinion ,during various load and speed combination , location of partially unloaded pads may change. On prima facie , SFD (Squeeze film damper) bearings can be thought of a better choice but they cannot be used in such gear units as at high load the damper can settle and rotor may not operate in desired path.

Sub-synchronous vibration in pinion

Sometimes , the tilting pad bearings demonstrate chaotic behavior with a comparatively raised noise floor. It is often found the sub-synchronous vibration issue are at very low frequency and spread all over up to 0.5X region of waterfall spectrum. This type of problems takes long time to identify and mitigate.

This type of sub-synchronous vibration (SSV) has low amplitude at random frequency . The concern from OEM original equipment manufacturers and end user is that this vibration, although small, fails to meet overall API vibration limits. See fig 3 below -

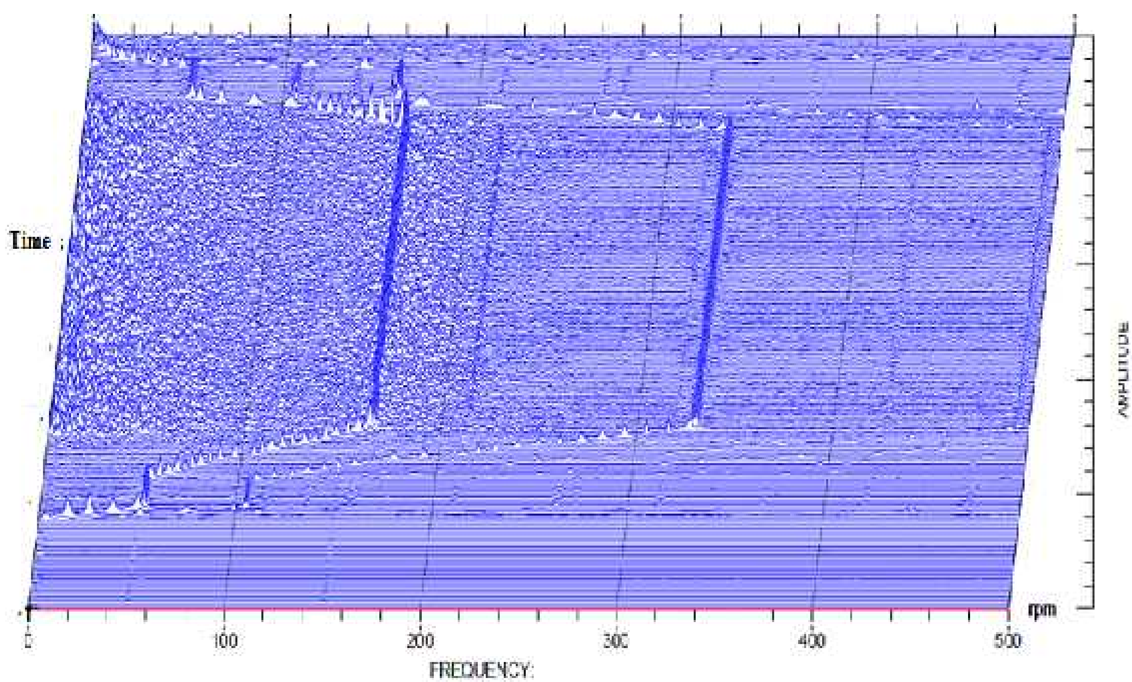

Fig 3 – Low frequency Sub-synchronous vibration

Pad Flutter is phenomena occurring to unloaded pads stationed opposite to load vector and not finding an equilibrium position. This means when the oil supply is not adequate on a relatively unloaded pad as stated earlier , the pad tends to swing in nonlinear mode creating a chaotic behavior and later stage they start spragging (infrequent hitting the journal).[4]

TPJB (Tilting pad journal bearing) particularly a three pad journal bearing, because of their geometrical and nonlinear fluid film characteristics, show high chances of the onset of chaos but are still selected when journal load and rotor peripheral velocity is too high to be accommodated by regular 4,5 pad design. If this is a case , a detailed analysis to be carried out during design phase itself and subsequent remedial action should be ready . During a loaded condition, based on direction of gear mesh as described above, unloaded top or bottom pads and require more oil to satisfy full film requirement. The shortfall of oil may lead to pad flutter.

To further confirm and identify the starved pad all pad temperatures may be recorded and compared with other pads during internal test carried out by gear box manufacturer .

The most effective means of controlling pad fluttering has been explored to be relieving the leading edge of each pad. The depth of the relief must be sufficient to ensure that a convergent wedge is formed at the leading edge of the pad as it approaches the journal.

Fig 4 - Relief created in pad to avoid sub synchronous vibration

Standard tilt pad bearings achieve low cross-coupling through rocking or sliding motion. Flexure Pivot tilt pad radial bearings achieve the same low cross-coupling and high stability through flexure and rotation of the center post while eliminating any pivot wear and high contact stress. The rotational stiffness increases the pad natural frequency and eliminates pad flutter and spragging which are encountered on the unloaded pads in standard type tilt pad bearing. Although this design is used in integrally geared compressor rotor but it is yet to be used on a parallel speed increaser double helical gear box pinion for its high speed and high load capability.

Pinion vibration during start up

A high vibration spike on pinion is observed sometimes during the start up which gets registered in spectrum but does not trigger alarm / trip of machine. The reason is due to

high inertia of bull gear which literally lifts the pinion during meshing . This is considered as benign vibration till it is within 70% of pad and journal clearance.
A common phenomena in speed increasers during API 613 no load test in pinion vibration recordings that exhibit higher vibration levels than the allowable limits set in API 613, clause2.6.6.5.
These vibration values of the pinion vary within a certain range. The above mentioned behavior is due to a slight instability of one of the rotors which can occur when the tangential force (driven torque) is equal to the pinion rotor shaft weight. Once there is load on the gear unit (< approx. 10%),there will be a defined force, which will move the pinion to its normal defined operating position(upward and out). At this point the instability will immediately disappear and the amplitude of the vibration will drop below the maximum allowable values. If this phenomena occurs during the test , it is advised to conduct at least a part load full speed test to assure stability of pinion during its operating speed regime .
In this case the energy associated with the start-up is low enough whereby no harm can occur.

Another phenomena commonly experienced in these type of gear units higher than normal vibration on hot starts. In this case higher vibration levels may be experienced on both rotors after a brief shutdown (hot start). When the gearbox is shut down lube oil continues to spray into the local area of the standstill mesh . The cooling effect of the oil brings the localized temperature down causing a relative thermal contraction of the steel in a localized area relative to the rest of the rotors gear teeth. This cooled area temporarily generates a thermally induced relative transient transmission error .This behavior is experienced for a short while and then gradually diminishes until the rotors are thermally stable to uniform load. In such cases a due diligence is to be taken to employ vibration trip multiplier during startup.

Pinion Super synchronous vibration

In last few years oil and gas industry has encountered. gearbox pinion high radial vibrations at a particular frequency that was well away from the gear mesh frequency, but which was excited by the pinion’s harmonics at 7x , 8x ,9× , 10× or 11× running speed.
ISO 6336-1:2006 has identified super synchronous vibration due to tooth mesh frequency only . When excitation frequencies (such as tooth meshing frequency and its harmonics) coincide or nearly coincide with a natural frequency of vibration of the gearing system, the resonant forced vibration may cause high dynamic tooth loading. When the magnitude of internal dynamic load at a speed involving resonance becomes large, operation near this speed should be avoided

As the super-synchronous vibration on pinion is identified, investigations are systematically carried out by comparing the uncoupled and coupled natural frequencies and their mode shapes with varying gear mesh stiffness, taking into account rotating speeds, and by comparing the strain energies of the lateral and torsional vibration modes. The results show that some modes may generate coupled lateral and torsional mode

characteristics when the gear mesh stiffness increases over a certain value or a dominant mode may change from torsional to a lateral or vice versa .
The parameter excitation is normally generated the pitch error during the gear teeth manufacturing within API accepted manufacturing tolerance. So , 7x, 8x or 9x and so on could be a harmonic multiple inside the pitch error deviation around the circumference. The harmonics can be found by an FFT-analysis of the pitch error

Sample Case study -

With variable speed drive turbo compressor train, turbo gear pinion showed high super-synchronous (7x) vibration at NDE side in certain speed load range which was inside the compressor operating map. It was found that pinion 4th lateral critical speed was excited at 7 times of its operating speed. The shaft circumferential velocity was below just below 100 m/sec and was equipped with tilting pad bearing. It was taken into consideration that train 10th torsional critical is coinciding with pinion 4th critical and mode was amplified.. Changing of coupling and its overhung to shift the 10th torsional critical did not improve the 7x component of vibration.

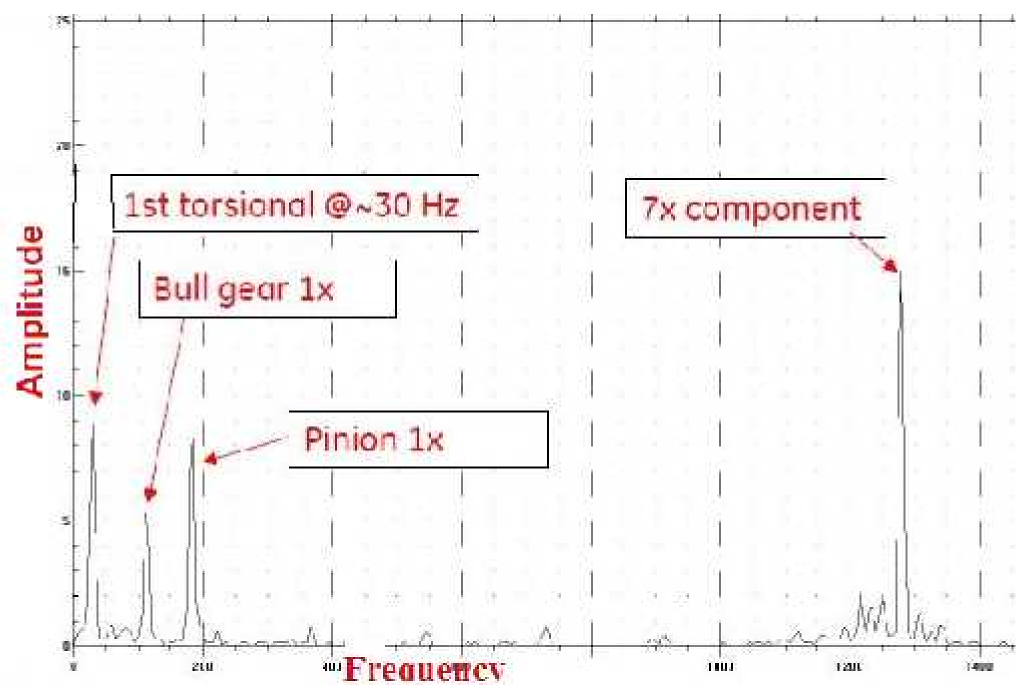

Fig 5 – Pinion super- synchronous vibration

Later on, the bearings were changed to fixed profile (lemon bore) bearings with much significant improvement, with lowered the 7X component but not below API criteria. With a revised rotor dynamic analysis of data gathered at test bed, in was observed that horizontal stiffness of pinion NDE bearing at that particular speed load range was very different than vertical. It was also seen in 4th damped eigenvalue mode shape plot that NDE bearing was at its nodal point .To shift the pinion critical speed and damp 4th mode after analysis of 4th damped eigenvalue plot, bearing damping was increased and as the 4 the mode was very near to nodal point, bearing span was changed by @12 mm.

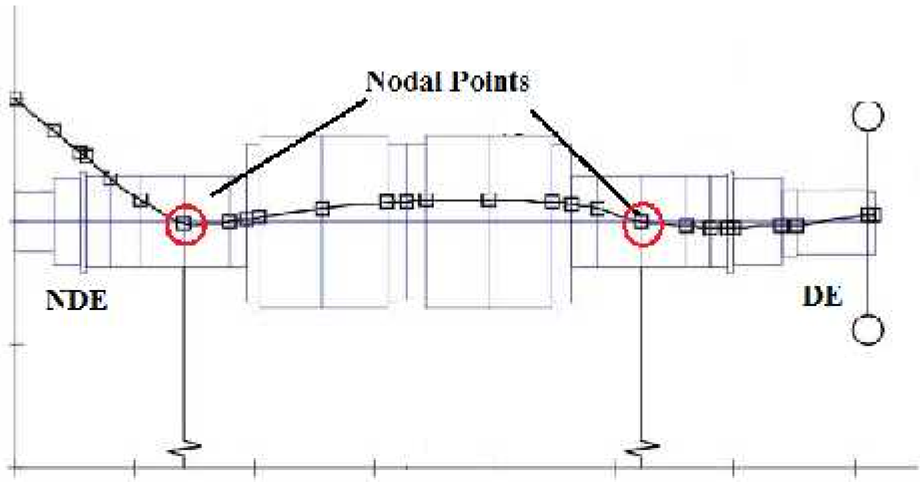

Fig 6 - Nodal points of a mode shape

Pinion vibration came down to meet API criteria to its full operating range.
In other cases, pinion was tuned where it was not possible to change the bearing type due shaft peripheral velocity and journal load (high PV) or location of nodes in mode shape [7]
After analysis of plots, pinion was over tuned by adding mass at NDE side (care was taken the mass does not exceed more that 7% of pinion weight)
In a case the pinion was under tuned by removing the mass from NDE side.

Role of torsional –lateral coupling in pinion vibration –
It is known that two dynamic systems influence each other, if connected. In the case of rotating machinery, lateral and torsional vibrations are usually analyzed separately. Such analysis is allowed in the case when the coupling effect is weak. When the coupling is strong, coupled vibrations is to be analyzed . It is to be noted that in the case of lightly damped structures, there are ranges of unstable rotor speeds. The unstable speed ranges are at frequencies for which the natural frequencies of both dynamical systems intersect with each other on the Campbell diagram.

At certain speed and load, sharp increase in vibration of pinion may be noticed but as the load and speed are increased, the vibration may be disappeared. In such case this can be concluded that a torsional Eigenmode excited by low loaded pinion bearings. When a torsional and lateral coupled system is modeled by a FE based software, it is found that at certain speed both modes exist and coupled at gear mesh point. By undertaking a stability analysis of various load and speed combination as shown below, bearing geometry and type can be changed to stabilize the pinions in its full operating regime

Transmission error (TE) is considered to be an important excitation mechanism for gear vibration during certain load/ interaction with torsional Eigen mode of train.
The causes of transmission error are -

- Cumulative pitch-line error
- Contact deformations (hertzian) in the gear mesh

- Gear teeth bending deflections
- Gear blank deflections
- Shaft deflections
- Bearing and gearbox casing flexibility

Normally practiced evaluation method.

Since the couplings with a very low lateral stiffness are used, it is understood that the rotational frequency from another component such as compressor rotor / pinion in the train will not trigger a lateral-resonant speed. Regarding the possibility of torsional and lateral critical speed affecting each other , they are drawn in different color lines denoting torsional and lateral natural frequencies are drawn in horizontal axis and possible excitation frequencies are drawn .Then they are checked for separation margins to one another

Little known resonant condition and frequencies -

Gear blank resonance- The gear blanks of high-speed, lightweight gearing may have natural frequencies within the operating speed range. If the gear blank is excited by a frequency which is close to one of its natural frequencies, the resonant deflections may cause high dynamic tooth loads.

Ghost Frequency -In addition to the tooth mesh frequency, there are sometimes “ghost” or “phantom” frequencies appear in spectrum, as a result of cyclic errors in the master worm wheel drive in the grinding machine used to finish cut the gears. These errors are imprinted onto the gear teeth as a helix undulation on each tooth and they are sometimes an important source of transmission error.
The ghost frequency usually shows up as the product of the number of teeth on the cutting mechanism and the running speed of the rotor and is caused from the imperfections of the gear hobbing tool that cut the gear at the time of manufacturing. Every gearbox may have a fault frequency set up for the ghost frequency, not to alert us of problems but to show us what that unknown frequency is.

Mitigation strategies

It is understood that all issues encountered during operation , cannot be pre- identified in a mechanical run no load complete unit test. Owing to various reasons such as -project completion schedule , work loads on OEM shop and test bed availability a full load full speed string test cannot be executed. In this case OEM’s proactive approach to mitigate such type of vibratory phenomena plays a key role to successfully commission and hand over the train to end user. Below paragraphs endeavor to draw a road map for such actions .

Before discussing the mitigation strategy , it is imperative to mention that base plate of gear box should be free from any twist in all planes . If there is any shaft driven main lube oil pump , it should be supported adequately ensuring no stress on casing . Return oil drain pipe (which is often a big diameter pipe) shall be fitted without any residual stress on gear box casing and casing shall be properly dowelled after ensuring that there is no soft foot present in installation. In recent years , this issue has baffled condition monitoring and rotor-dynamic engineers to solve gear box vibration problem.

Sub synchronous vibration -Most commonly used design tools do not consider the pad inertia as a standard feature. The inherent stability and self-alignment ability are the most important advantages tilting pad journal bearings have. An essential requirement for rotor-bearing system analysis is the calculation of the oil film force. Reynolds Equation is considered to be acceptable for evaluation of oil film force based on database. The database is based on number of pads , orientation , pad angular extension and length to diameter ratio.
The swing of bearing pads must be taken into account for detailed dynamic analysis of rotor bearing system.
One of proactive mitigation strategy may be to increase oil flow than critical oil flow to load all pads and keep them in dynamic equilibrium.

Super synchronous vibration-To detect the traces of peaks at high harmonics , the Fast Fourier Transform spectrum should be accordingly set up . Normally vibration peaks up to 1500 Hz are measured and identified in no load test.
At lower load at 70% speed, as the meshing stiffness is lower, pinion can excite at higher critical mode. Hence , a detailed asynchronous response analysis to be done addressing forcing functions at 7x,8x,9x and 10 x operating speeds to determine the pinion and bearing system sensitivity to fourth and higher lateral mode as shown in fig 7a . The forcing function shall address the cumulative pitch error just below API 613 acceptance limit .Root locus plot of pinion for various loads and speeds should be drawn to see the stability of the system.

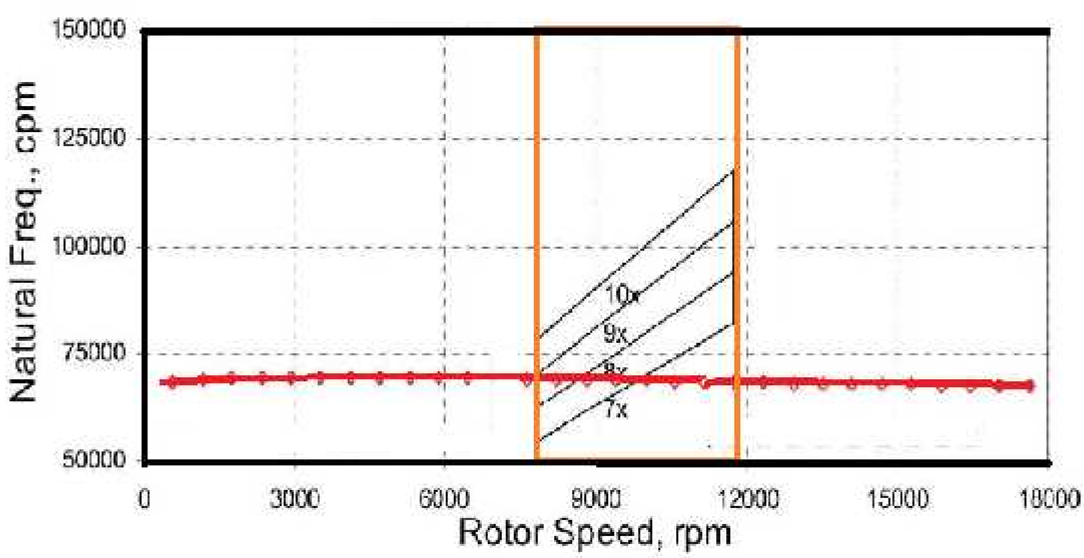

Fig 7 - Identification of corresponding harmonics and 4th lateral frequency

Root Locus plot -

Load force vectors imposed on the pinion journal bearings vary due to load variations and speed changes for a centrifugal compressor driven by variable speed drive motor or two shaft gas / steam turbine. In most of the cases the pinion load in on upward direction due to up mesh. The horizontal load vector is due to separating forces generated at mesh. In mechanical run test, load vectors of such magnitude are not experienced. The resulting bearing load vector during start up is downward and as the speed and torque is increased, it tends to go upward. So it is imperative to draw a root locus plot for shop condition and various load – speed condition.

Prediction of possible torsional coupling with a realistic rotor-dynamic analysis should be carried out by using Finite Element model of bull gear and pinion.. Bearing pad radial movements should be modeled in the analysis.FFT(Fast Fourier Transform)-analysis of the pitch error can be fruitful during no load test; a log based spectrum plot can be generated. The gear box sensitivity analysis can be done based on the pitch error with variables of meshing stiffness and torsional -lateral coupling.[6]

The complete train system is to be analyzed in order to define the optimum compromise to avoid the excitation mechanism (in speed harmonics were known), or to increase the system modal damping. Gear Box OEM along with bearing designer should keep the preliminary design of alternative geometry bearing ready if found that pinion nth lateral critical speed is excited at higher harmonics of pinion and mode can be damped.
In addition to above OEM should ensure to have wider space in bearing housing during design and fabrication so that bearing can be shifted to avoid nodal location in Eigen modes if required . Oil supply passages in gear housing must be designed to meet this modification (if required to be carried out at site). [6]

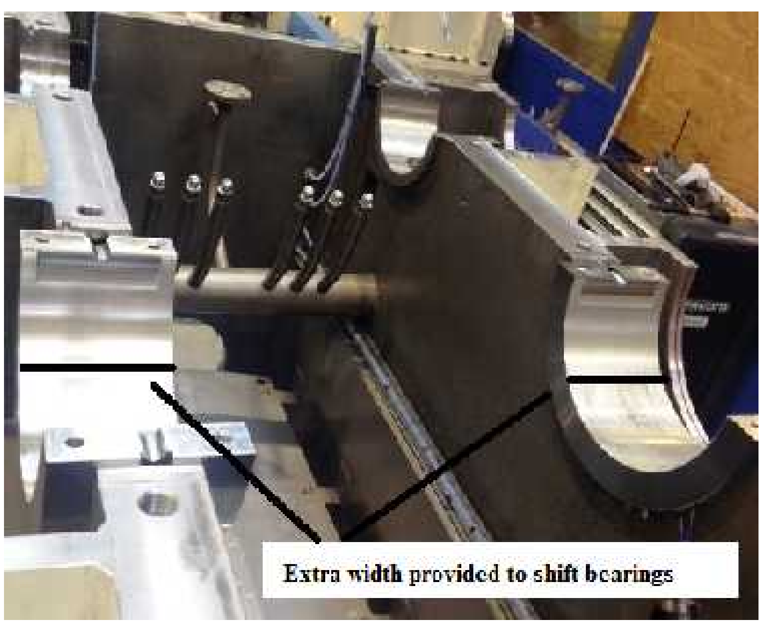

Fig 8 - Gear Box bearing housing

Identification of defect frequency if any during mechanical run test (ghost frequency) can be done and can be stamped in installation and maintenance manual .
Forecasting of tuning of pinion -Addition / removal of 7-10 % of pinion's weight at the non-drive end.

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Proactive design approach to solve the issue at site with minimum time -
Sensitivity analysis of effect of coupling overhung on pinion Eigen frequencies and modes should be carried out as a part of OEM internal test .Before and after installing the bull gear and pinion on the casing a bump test may be carried out and same can be verified with historical data of modal parameters.
If FLFS (Full load full speed) test of complete unit is not an exercised option then proactive approach shall be very beneficial -
1. Tuning of pinion to shift the Eigenmodes is a proven solution . But this type of activity is carried out when after a prolonged and detailed study reveals the excitation of higher mode .Hence as a proactive approach OEM can design the pinion in such way that it is easy to remove weight or add weight to tune at site . The pinion should be drilled with a proper centering device up to location of NDE bearing location . It shall be then closed with a stepped plug before assembly . If the requirement of tuning the pinion is needed , this pug can be removed to shift the natural frequency of pinion.
2.Additionally , a set of two segmented ring / disc can be machined and kept as spare which can be installed during diagnostic tests if issue of super-synchronous vibration arises . The rings can be mounted as interference fit using coupling hub mounting device or induction heater used to mount antifriction bearing . The rings / discs can be joined at NDE end with fatigue proof bolts . The total mass forcasted to tune the pinion can be easily splits into two rings or discs to fine tune it. While designing the rings for mounting , it is important that these can be easily removed by hydraulic or mechanical puller device if required .

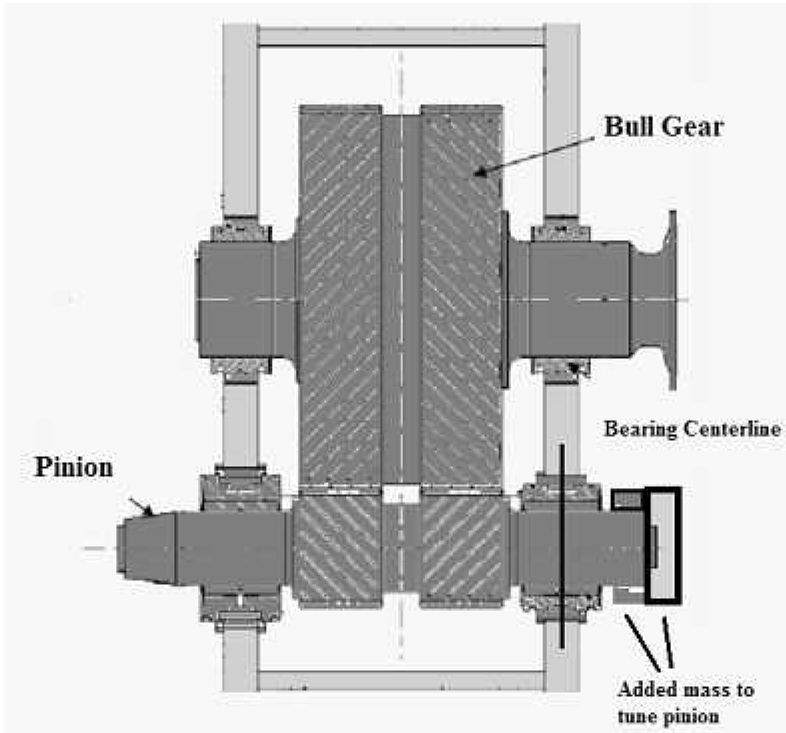

Fig -9 - Tuning of pinion by adding mass

After installing the added mass on NDE side of pinion , in-situ balancing shall be carried out .A balancing machine shall be available at site which can be hooked with machinery diagnostic system .
For both cases a loss of interference study shall be done using a FE analysis .

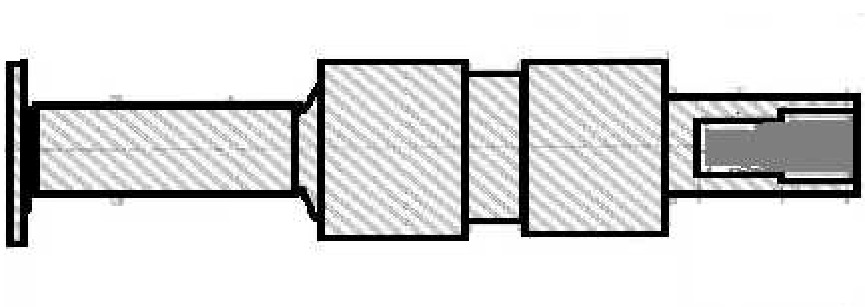

Fig10- Tuning of pinion by removing mass (temporary plug inserted*)*

Technical support at site -
1. Availability of personnel from OEM with a master vibration analyst (Cat III or IV).
2. Vibration analysis tool and software
3. On line availability of rotor-dynamics experts from OEM and active participation of End user to come to a quick decision .

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Conclusion

From instances described above , it is evident that no load mechanical run or no load complete unit test may not accurately predict the probability of sub synchronous and super synchronous vibration . Of course the nonlinearity associated with tooth contact can give rise to instability producing super and sub harmonic responses, and even chaos, but it may not then correspond to actual operating conditions where the load is sufficient to maintain tooth contact.

Instead of focusing just on the fourth lateral vibration mode(as has been the common phenomena as reported in various technical papers), the alignment of any pinion torsional modes with lightly damped lateral ones should be avoided. [6]It is important to synthesize the bearing support dynamics , pad movement , pure lateral and pure torsional mode to create a reliable solution. Coupling modifications may become a less costly, and less time-intensive, and may be explored if the torsional mode high participation is confirmed. Plausible mitigation action suggested above should be deeply reviewed by OEM in terms of risks and rewards (saving time in not conducting a full load string test , analyzing the issue , finding ways to tune the pinion / damping the vibration at pinion lateral mode)

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