Dynamic stability of high speed turboshaft engine with reducer

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Abstract. The dynamic behavior of machines which rotate at extreme high speeds is crucial for stable and long operation. In design process some dynamics calculation must be taken into account to avoid potential operation near resonance areas. This paper considers dynamic analysis of the high speed reducer, designed together with the turbo-shaft engine. Resonances can occur when frequency of rotation and teeth mesh frequency are close to gear pair in mesh natural frequency, to the shafts natural frequencies and to the other parts natural frequencies. Problems can be avoided by translating resonant areas to higher or lower level of frequency by changing some design parameters, mass, stiffness and by variation of the stiffness of bearing supports. The paper presents the approaches to natural frequencies identification for the example of high speed reducer design. Some examples of design solutions, especially for elastic bearing supports and their experimental testing are also presented.

1. Introduction

High speed machines like turbo-jet and turbo-shaft engines operate with high frequencies of rotation which pass through some resonant areas. Operation in resonant areas can at least reduce operating life of the parts but it usually leads to failure of the entire system. Therefore, it is crucial to avoid these areas in permanent operation, 15 to 20 % before and after the resonant frequency value. Operating regimes should be below them. In high speed machines, it can be complicated to achieve this, so their operation can be achieved at frequencies above resonant values which must be achieved by fast run across during start up process. The calculations and experimental testing for predicting these areas are a very important and unavoidable phase in design process of such structures.

The case study in this paper is a high speed reducer for turbo-shaft engine that operates at extreme speeds of revolution. Gas generator provides power of 230 kW by generating the gas (combustion products) at high temperature and pressure. This operating fluid drives the free turbine which is installed after the gas generator, Figure 1. The free turbine is on the input shaft of the reducer and it provides mechanical power on the output of system. The reducer has a task to transform the torque and the speed from 40,000 rpm to 6,000 rpm and it is designed for helicopter application.It consists of three identical auxiliary branches which divide the power from the input gear and deliver the power to single output gear. Basically, it has only two different gear pairs. This type is used in order to achieve the least possible dimensions and weight.

There are three different types of resonant behaviours that may occur inside high speed reducers, and all must be avoided for safe and long operation. The first type is resonant frequencies of gears in mesh [1], the second type is the shaft's modal behaviour [2] and the third type is natural vibrations of machine parts of the system, especially gearbox housing behaviour. Natural frequency of gear pairs in mesh is depended on the equivalent mass of rotating assemblies (mass of parts with their moment of inertia) and stiffness of the teeth [3]. Shaft mods are directly proportional to the mass and stiffness of the shaft and supports and their joins. Natural frequencies of the rest of the parts can be easily altered by increasing their stiffness or mass. Gearbox casings are the most exposed part of the structure on multiple excitations, but in most cases they are the most massive part and their natural frequencies are high enough. But in the case of high speed reducer for aviation application their natural frequencies can be found near resonance areas and it is necessary to perform the dynamic analysis [4].

Dynamic behaviour is the result of the relation between operating – excitation frequencies and natural frequencies. Excitation frequencies of gear transmission units are rotation frequencies and teeth mesh frequencies. These frequencies vary in linear dependence of the input shaft speed, whereas natural frequencies are in relation to design parameters and they do not vary. The relation between excitation (disturbance) frequencies define Campbell's diagram which provides the possibility to identify or avoid local resonances.

The aim of this article is to identify possible natural resonant frequencies in order to provide the possibility for turbo-shaft motor with reducer to operate in supercritical range and minimised resonance effect when the motor starts, when the speed increases and when the speed decreases.

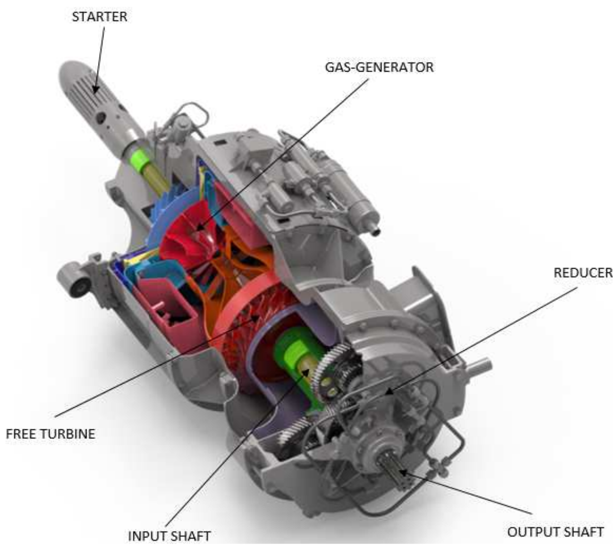

Figure 1. Turbo-shaft engine with reducer.

2. Gear mesh resonance frequencies

The high speed reducer, integrated with the turboshaft engine, transforms 40,000 rpm of free turbine into 6,000 rpm of output shaft. This transmission unit consists of input (free turbine) shaft with input central gear, output shaft with output central gear and of three auxiliary branches of double gears, which divide the power into approximately three parts. The auxiliary branches return all power, reduced for some percent of loses, to the output gear. There are practically two different gear pairs in the mesh and they are presented in Figure 2. The first pair reduces the speed from 40,000 rpm to 16,000 rpm while the other one reduces the output value further to 6,000 rpm. The three parallel branches provide the smallest possible weight of the reducer. The gears are small in their diameter and mass, together with the other rotating masses. The first pair has a normal plane standard teeth module of 2mm, and the number of teeth z_1=24 and z_2=60. The teeth mesh frequency $nz/60$ is very high at 16,000 Hz. The mass of the input rotor (input shaft with free turbine, first gear and other parts that

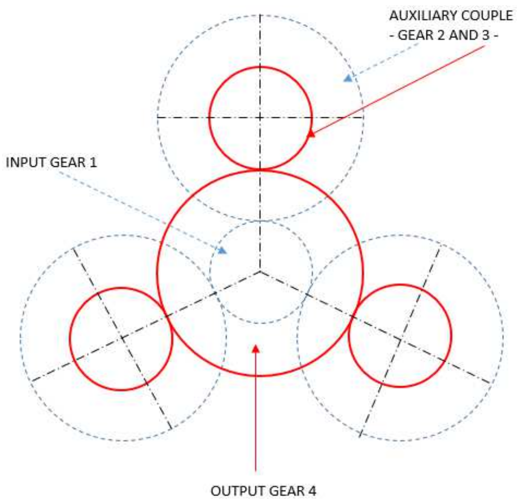

Figure 2. Gear layout in the high speed reducer.

rotate with it) is designed to be 3.8 kg. The mass of the rotating parts on the auxiliary branch (gears 2 and 3 with other parts that rotate around the spin) is 0.9 kg. By calculating the stiffness of the teeth in the mesh together with reduced equivalent masses, according to DIN3990, natural (resonant) frequency of this gear pair is 3,333 Hz with the similar approach as in papers [5, 6, 7]. This calculation shows that the first pair operates in supercritical mesh frequency range with almost 5 times larger mesh frequency regarding the natural frequency. That makes a stable operation by running across the resonant area during the start and operation.

The second pair is unusual because it has lower module value than the first. The main reason for this comes from a need to avoid meshing of teeth close to the resonate area. With lowering the value of normal plane standard teeth module there is space to increase the number of teeth for the same relation of diameters of gears. With a value of 1.5mm for the standard module, the numbers of teeth z_3=30 and z_4=81 are achieved. With these teeth parameters a frequency of 8,000 Hz of meshing of second gear pair is achieved. The natural frequency of the second pair is 4,768 Hz for output rotating mass of 2.2 kg. With these parameters the second pair is designed to operate in supercritical mesh frequency range with the relation of frequencies of 1.67.

3. Modal behavior of the shafts

Shafts in the gear transmission units and in the other structures have natural frequencies (resonant vibrations) and frequencies, especially when the shafts are with higher support distance. The mode shapes are flexural and torsional. Modal shape identification can be obtained by analytic, numerical or experimental approaches. In the presented design solution, the input shaft is with really long distance of supports. This shaft carries the free turbine at one end and the input (first) gear at the other end it is designed to operate at 40,000 rpm. The identification of the critical speeds is carried out by the numerical and experimental approach. During the run-up process, the shaft has to pass through the critical speed, which depends to geometry, rotor mass, connection and position, stiffness of the bearings and bearing supports etc.

High speed of rotation produces a gyroscope effect of rotating masses which makes a difference between the static and dynamic stiffness of supports. With the increase of the rotation frequency (speed), stiffness can increase (FW) or decrease (BW) in comparison to the static one. In order to identify these dependencies, for the turbine shaft (gearbox input shaft), a numerical calculation of critical speeds is carried out. The stiffness of the support near the free turbine is varied across the range from 5000…180,000N/mm and obtained Critical speed map (Figure 3). The first mode has a sloping section in the left half of the plot. This indicates that the critical speed can be shifted up or down as the stiffness in the bearing supports are varied. In the right section of the plot, the shaft

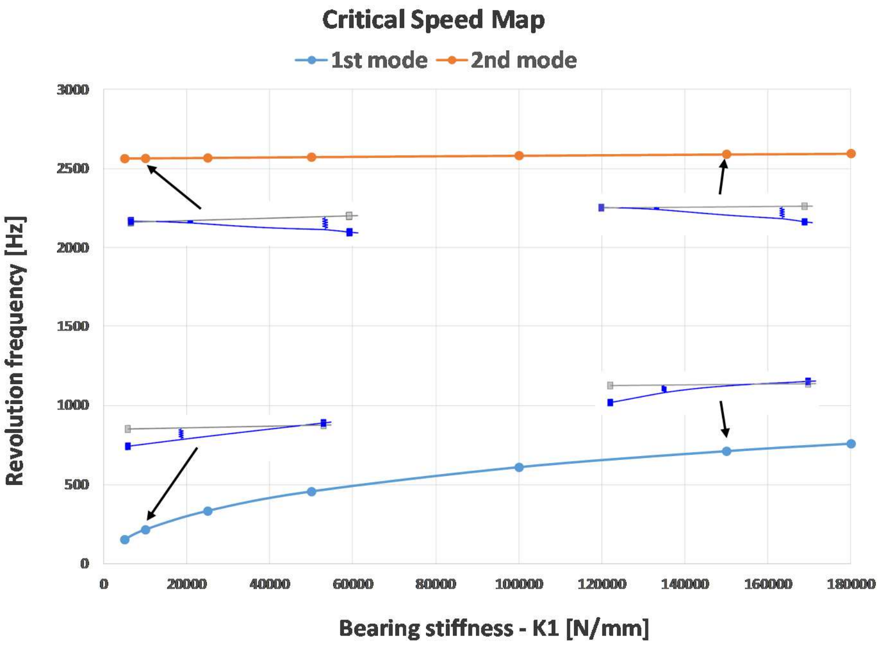

Figure 3. Critical speed map of input shaft.

stiffness dominates, and varying the bearing stiffness does very little to change the critical speed. The value in the flat section resembles an infinitely rigid support and provides an upper limit to the critical speed. The critical speed map provides a convenient format for determining how the speed dependent bearing stiffness affects critical speeds. It is also useful in deciding whether the intended operating speed range is compatible with the locations of the lateral critical speeds.

In general, it is desirable from a rotor-dynamics standpoint to have the supports more flexible than the rotor. In that case, the motion at the bearings allows the damping to be more effective. The shaft appears flexible but it does not bend. On the contrary, the rotor on stiff supports can pose an unacceptable design, which is difficult to balance and is prone to instability. The accomplished compromise is to have one of the stiffness K1 very stiff (180,000 N/mm) so that the shaft at the right end is fixed (bearing support at gear side), while the stiffness K2 is fairly soft (8,000 N/mm). Here the first critical speed is at 193 Hz. This is comparable to the original setup where both bearings were stiff, for example K1 is 180,000N/mm and K2 equals 150,000 N/mm. The critical speed was calculated at 710 Hz which was very close to the operating range of 666.67 Hz (40,000 rpm). By providing a soft bearing support at the free turbine side, the value of the first critical speed dropped from 710 Hz to 193 Hz, while the second critical speed was still far above the operating range. This is the reason why it is necessary in design solution to insert elastic support near the free turbine.

The effect of the elastic support at critical speed of this shaft, also obtained by numerical simulation, is shown in Figure 4. This is the Campbell diagram with natural and shaft rotation frequencies. Natural frequencies increase with the increase of the speed of rotation. When the gyroscopic effect is in the same direction with rotation (FW), dynamic stiffness and natural frequency increase in front of the speed increase. The opposite direction of the gyroscopic effect reduces dynamic stiffness and natural frequency (BW), but this is not a realistic situation.

In order to reduce support stiffness, three types of elastic bearing supports are developed and experimentally tested. The first one is designed with local weakening of the bearing housing in the area before the bearings, so called squirrel-cage support [8], Figure 5. The main drawbacks of this design solution are that it is very vulnerable to alternating loads with high frequencies and that very low stiffness provides a high displacement.

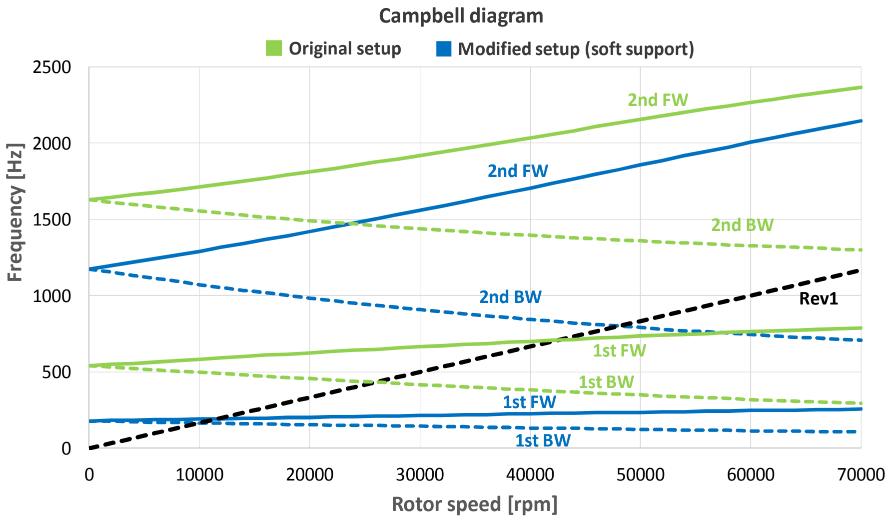

Figure 4. Campbell diagram of two setups superimposed.

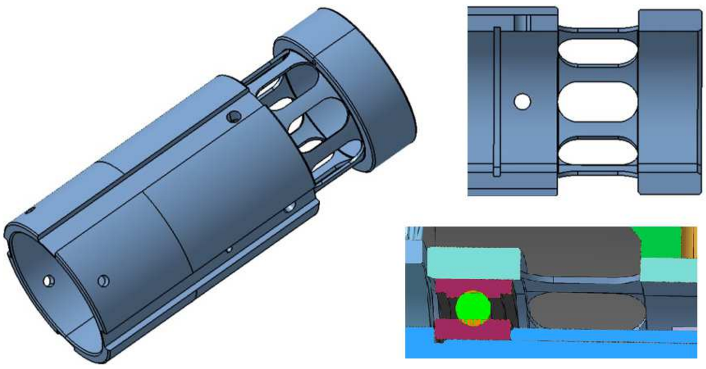

Figure 5. Elastic bearing support – squirrel cage type.

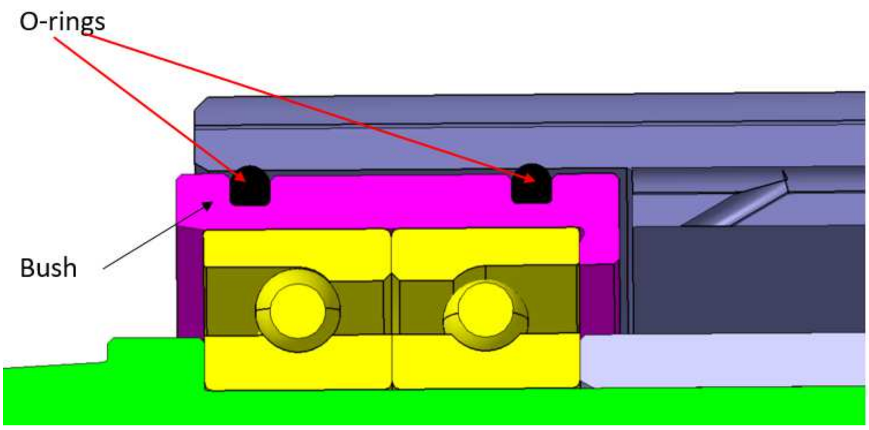

Figure 6. Elastic bearing support – bush with O-ring type.

The second design solution implies a bush, in which bearings are placed, with two O-ring above it, inside the bearing housing. The bearing bush with dumping O-rings is pressed inside the bearing housing while O-rings represent the dumping parts in the system. This type of elastic support is presented in Figure 6. Rubber rings have low durability when the high frequency loads are applied to the shaft so their operating life is questionable.

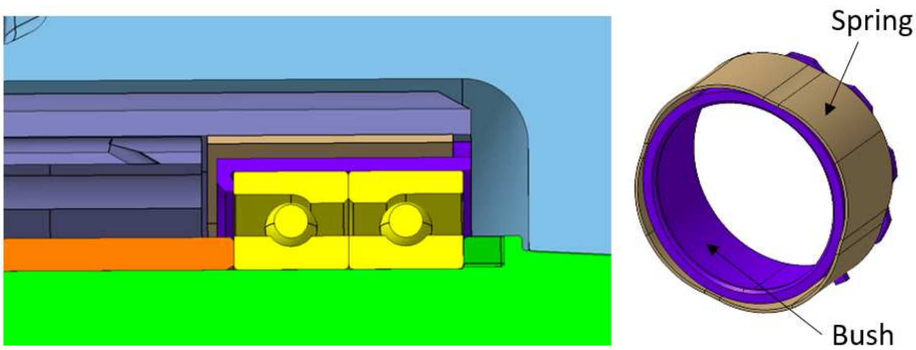

Figure 7. Elastic bearing support – bush with spring type.

The third design solution, Figure 7, is similar to the previous with the difference that it uses a spring steel part instead of O-ring. They are limited in use for very high elasticity because steel is much more rigid that rubber.

4. Modal behavior of housing and other parts

Every part has its own natural vibrations which can be awaken by some excitation. Housings are parts whose oscillation can be awaken by an influence of multiple sources of excitation and this makes them sensitive in resonant areas [1, 9]. In low speed reducers, the housings are the stiffest part and usually the modal analysis of the structure can be neglected. But, in the case of high speed reducers, a high speed of rotation, together with teeth mesh disturbances, can easily excite some of the housing natural frequency (resonance) because it has a high frequency which corresponds to non-rigid structures. Therefore, a modal analysis of high speed reducer for turbo-shaft engine is also carried out and discussed in this paper.

The housing for the reducer consists of three main parts: input shaft bearing housing, dividing plate and main housing. The housing assembly is presented in Figure 6 together with the belonging parts. All parts are joined with several screws. The bearing housing for input shaft is made from stainless steel and it is the part that holds the input shaft with the first gear and free turbine (the green part in Figure 1 and Figure 8). The dividing plate is the carrying part which is placed near the exhausts of the turbo-shaft engine and it is exposed to elevated temperature. For that reason it is made from a stainless steel material (the brown part in Figure 8). The main housing has a function to close and hold all parts in one entirety. In order to make the reducer as light as possible, the main housing part is made from quality aluminium alloy (the grey part in Figure 8).

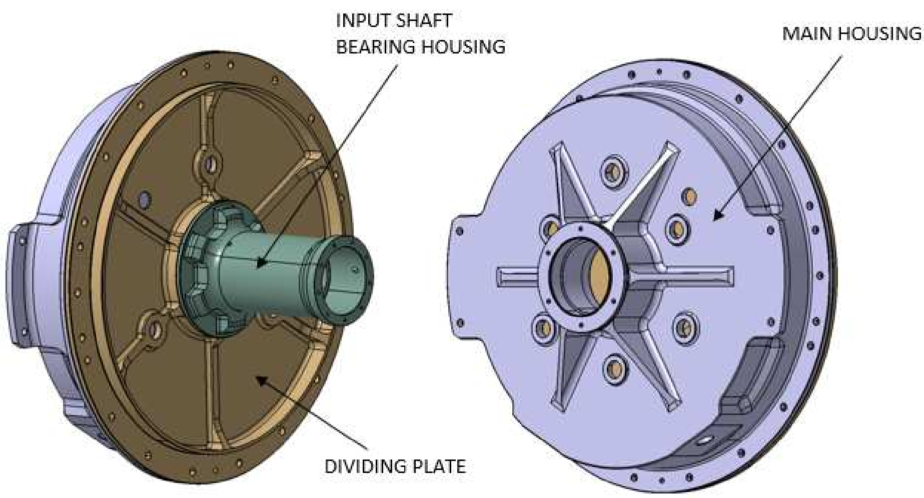

Figure 8. Housing assembly for high speed reducer.

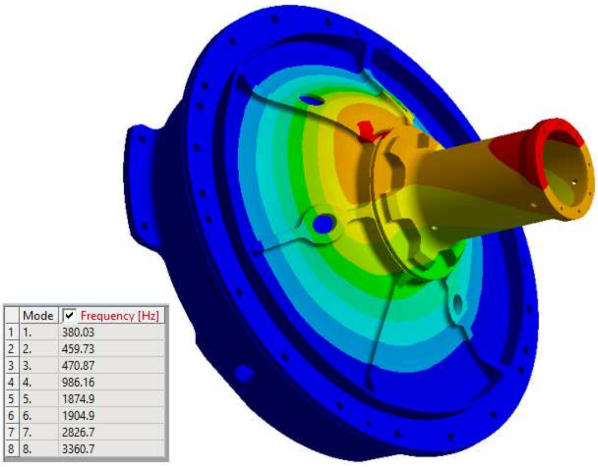

Figure 9. Modal analysis of housing and the shape of first and lowest natural frequency.

The housing's natural vibrations can be excited by main two sources, the frequency of rotating mass and frequency of teeth mesh. The frequencies of the teeth mesh for two gear pairs are calculated in the previous chapter and they are 16,000 and 8,000 Hz. These frequencies are at very high values and they are much higher than housing's natural frequencies (Figure 9). The input shaft with the free turbine rotates at 40,000 rpm which creates an excitation frequency of 666.7 Hz. The auxiliary branches rotate at 16,000 rpm and they create an excitation frequency of 266.7 Hz, including the output shaft with 6,000 rpm with a frequency of 100 Hz. These frequencies are more critical and must be taken into account during design process in order to avoid them for operating regimes of the reducer.

The modal frequency analysis of the housing is carried out by applying the numerical finite element method [10]. The results are presented in the table in Figure 9. The shape of the mode for the first frequency is also presented in Figure 9. It represents the bending of the dividing plate due to the presence of the input shaft bearing housing which is joined on the console principle with screws from the inside. The second and third shape and frequency are more or less similar to the first with the difference that they occur in the other planes.

5. Conclusion

By dynamic analysis and by design parameters variation, an innovative high speed reducer for turbo-shaft engine is designed. The obtained structure can operate steadily with 40,000 rpm input speed of the free turbine and to avoid excitation of any possible natural frequency. The specific results are the next.

- The calculations of teeth mesh resonant frequencies are carried out. The possible resonances in relation to gear speed of rotation and teeth number is avoided by gear parameters variation (module and teeth number). Selected gear parameters provide possibility for steady gears operation in supercritical mesh frequency range.
- The input shaft together with the free turbine and with extreme high speed of rotation produces variation of supports dynamic stiffness in relation to speed of rotation, and also variation of critical shaft speed. The problem is solved by this phenomenon identification using the numerical approach and by specific design solutions of bearing supports which can provide steady operation.
- For housing of high speed reducer a numerical modal analysis is carried out. By design parameters variation teeth mesh frequencies are shifted in the frequency range much higher than calculated natural frequencies. Frequencies of rotation are between the calculated natural frequencies of the housing. It also provides steady operation with extremely high speed of rotation.

Acknowledgments
The authors wish to acknowledge and thank the company EDePro – Engine Design and Production – Belgrade, for taking the project of turbo-shaft engines development, for support and for encouragement in the project realization. This article is also a contribution to the Ministry of Education, Science and Technology Development of Serbia funded projects TR 035006 and TR 35044.

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