Modal Properties of Turbo-Compressor Elements

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Abstract

In this paper some modal and vibration properties of turbocompressor elements are studied. Natural frequencies of turbokompressor rotor shaft with turbine and compressor wheels and bearing unit elements are calculated. Their spectrum dependence on the material properties of the elements and on the constructive variants of the bearing unit components are studied. Shape modes of the system are obtained .

The vibrations of the system at the maximal angular velocity are estimated. The oscillations of the turbine and compressor blades are found for that angular velocity . The natural frequencies and shape modes of the separate blades of the turbine and compressor are found. The blades response to short pressure pulses are studied. The conditions of resonance are investigated.

Newly designed object modal properties analyzing

In this paper we represent some results received during the modal analysis carried out over two constructive variants of the turbo machine elements. One of the important task usually stated by the technologists group is the newly designed object modal properties investigation. In such investigation the preliminary calculations on the basic model are usually needed and the following comparison of the results for the both design solutions is necessary. Here we are trying to represent these steps of the newly designed object properties analyzing.

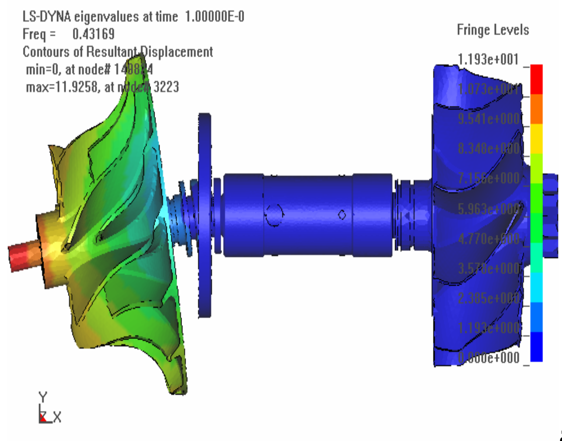
First only rotor and wheels system was analyzed for the basic model. Eigenmodes and eigenforms were found. As it was our first experience in LS-DYNA modal analysis it was found expedient to vary slightly the material properties of the model and check the resulting shifts in the eigenmodes. The density, the Young modulus and the Poisson ratio were varied and the tendencies in the frequencies altering were checked. (Table 1).

Table 1 – The eigenmodes dependence on the material properties for the rotor-wheels system in kilohertz

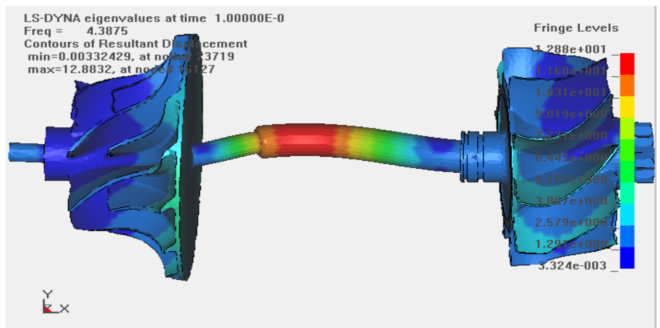
Eigenmode number	ρ=7e-6 kg/mm^3 E=200 kN/mm^2 ν=0.3	ρ=7e-6 kg/mm^3 E=200 kN/mm^2 ν=0.33	ρ=7.7e-6 kg/mm^3 E=200 kN/mm^2 ν=0.3
1	0.5146	0.54633	0.5164
2	0.55575	0.55971	0.5298
3	0.58891	0.59337	0.5615
4	1.5288	1.5422	1.4577
5	1.677	1.6856	1.599
6	1.6994	1.7152	1.6203
7	3.0935	3.1702	2.9495
8	3.5283	3.5597	3.3641
9	3.565	3.5941	3.3991
10	4.0762	4.1288	3.8865
11	4.1879	4.2197	3.993
12	8.9986	9.1294	8.5798
13	9.073	9.1461	8.6507
14	9.1656	9.2321	8.7391
15	9.2028	9.2713	8.7745
16	9.2468	9.3156	8.8165
17	9.3077	9.4353	8.8745
18	9.3366	9.4638	8.9021
19	9.4461	9.5005	9.0065
20	9.9306	10.018	9.4685
21	10.009	10.101	9.5429
22	10.131	10.207	9.6599

It can be seen from the table that the natural results were obtained. The values of the eigen frequencies shift into the higher frequencies range if the strength of the sample increase with the growth of the Young modulus and one gets the lower frequencies range if the density grows. The introduction of the oil lubrication caused the frequencies range to shift in the lower frequencies area playing the damping role. Then the series of preliminary numerical experiments on the old model were made adding the units of the bearing mount into consideration. Such constructive parameters as the rotor shaft length, the sliding bearing length were varied and different variants of the sliding bearing fixture were considered.

The preliminary numerical tests on the basic model having proved the correctness of the calculation model, the same experiments were remade for the new variant of the rotor (Figure 1). The eigenmodes and forms were found for the given values of the material parameters for wheels, rotor shaft and sliding bearing.



a)



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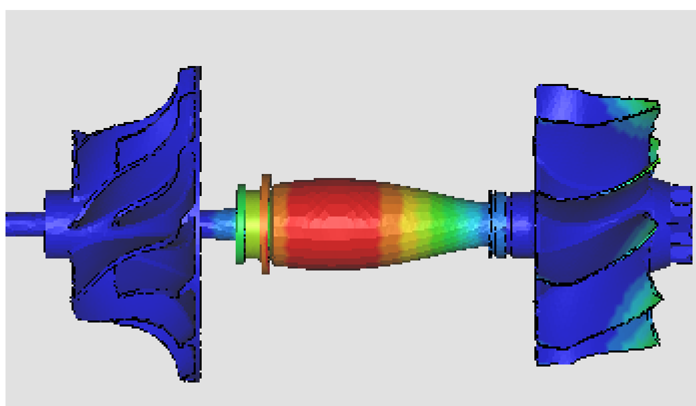


c)

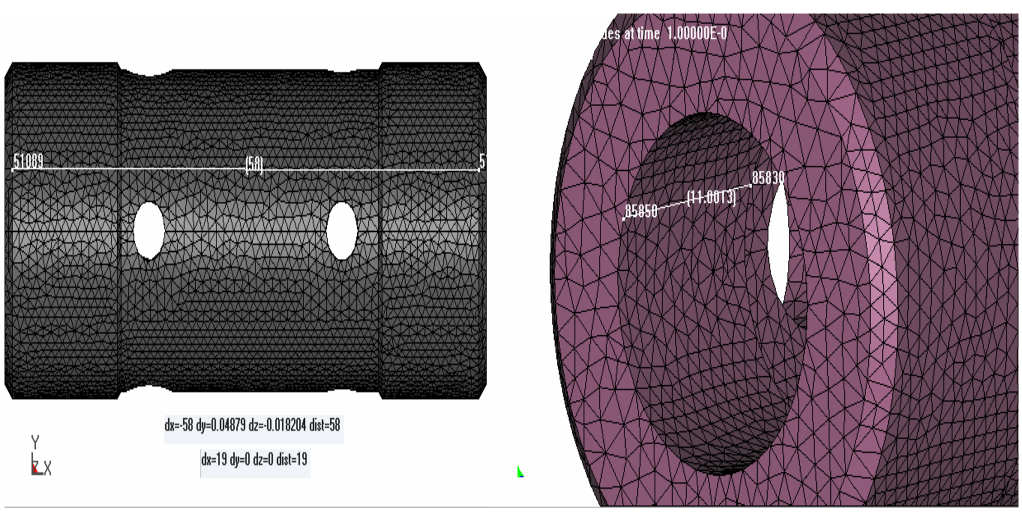
Figure 1 – Types of the lowest rotor eigenform a) - shift eigenform b) - torsion eigenform c) - rotor shaft stability loss eigenform

Only first 25 modes were first taken into account during the eigenmode analysis because for the both models only several first modes may be found in the exploitation area of the turbocompressor. First 15 of them represent the different rotor oscilation eigenforms. That is shift, torsion, rotor shaft stretching and stability loss modes (Figure 1). The answer was found to the question how the eigenmodes are distributed in the exploitation area of rotor frequencies. For the basic model of rotor they fill the mentioned interval evenly (Table 1) while the modified model has the range from 0.8 to 2kGz free from eigenmodes (Table 2). That means the modified model has particularly good modal characteristics. The later experiments on location confirmed the absence of the eigenmodes in this interval of angular velocities while at the level of 50000 rotations per minute angular velocity the rotor began vibrating immensely.

The influence of the sliding bearing length and the sliding bearing support length on the eigenmodes values was analyzed (Figure 2, Table 2).

Table 2 – Eigen modes of the modified turbocompressor rotor-bearing system for different sliding bearing constructive variations in kilohertz

For the 60 mm bearing length and 12 mm support surface length	For the 59 mm bearing length and 12 mm support surface length	For the 58 mm bearing length and 12 mm support surface length	For the 58 mm bearing length and 11 mm support surface length
0.37473	0.37458	0.38007	0.37997
0.37513	0.37489	0.38035	0.3804
0.45407	0.45208	0.45432	0.45164
0.9579	0.74008	0.7259	0.71528
1.4567	0.74492	0.72806	0.71876
1.4613	0.89472	0.92953	0.86938
2.0516	0.91904	2.0873	2.0869
2.0612	2.051	2.0964	2.0946
3.6732	2.059	3.31965	3.1832
7.3779	4.1194	4.4079	4.4138



a) b)

Figure 2 – Constructive parameters of the bearing a) – length b) length of the support

As can be seen from the table 2 the increasing of the sliding bearing length from 58 to 59 mm shorten the free frequency range by 70kHz while the further increasing in bearing length leads to the appearing parasitic eigen frequencies in the initially free interval (1.45 kHz for the 60 mm length bearing). Also as can be seen from the last two columns of the table 2 the increase in the sliding bearing support length also causes the slight shortage of the free frequency interval. Thus the way of operating over the free frequency range of rotor-bearing system by means of sliding bearing constructive parameters altering was found.

While rotating in the free from eigenmodes frequency interval the rotor undergoes chaotic vibrations due to the presence of the gap between rotor shaft and the bearing. They were measured for the higher angular velocities (figure. 3). It was found that their values were not surpassing the 0.1 mm.

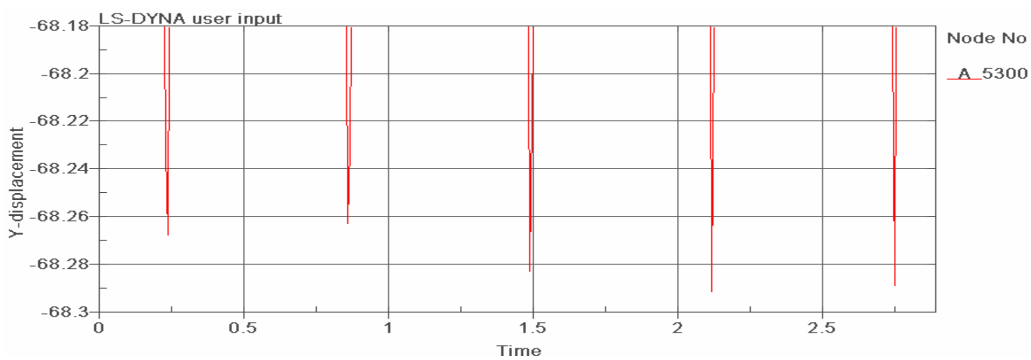


Figure 3 - Transverse oscillations of the turbine points for balanced rotor.

The area of the higher eigen frequenciesis is of importance but another kind. The rotational angular velocities do not reach the level of the eigenmodes that correspond to the eigenform of the wheels blades. The mechanism of the system exitation that may lead to such occilations appearing lies in the fluid-structure interaction between gases and wheel blades. Here we can not yet represent the results of such an analysis as it is still far from finishing. But some preliminary work in that direction was made. In figure 4 two eigen forms of the turbine blade are shown (corresponding eigenmodes are given in the table 3).

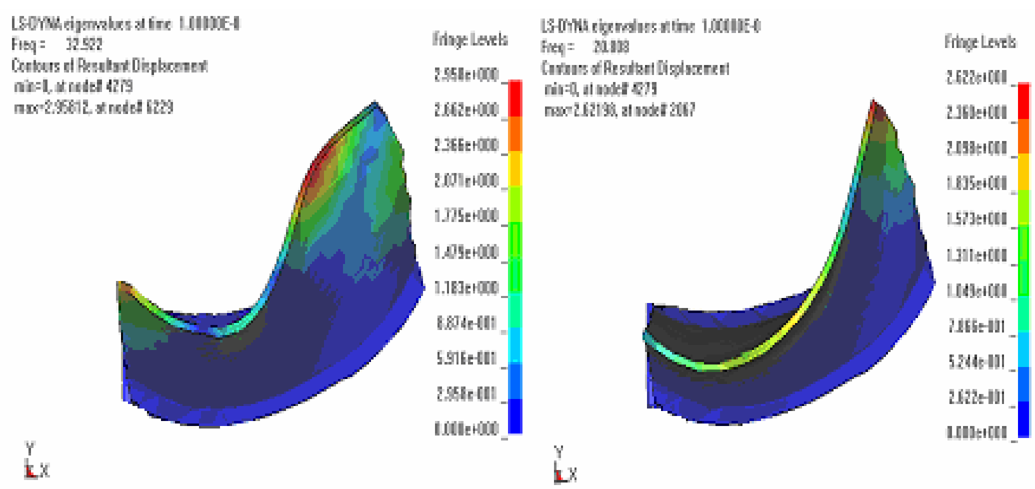
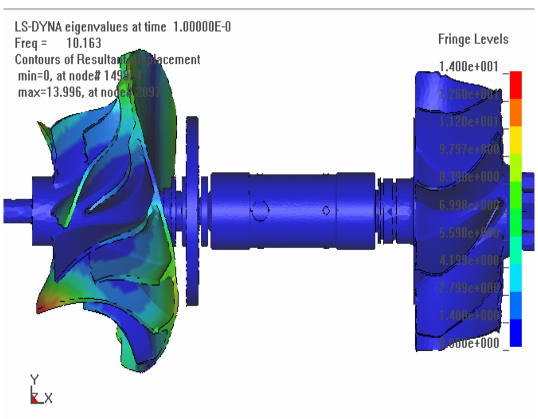


Figure 4 - Shape modes of the turbine blades

Table 3- Turbine blade eigenmodes in kilohertz

Eigenmode Number	Frequency
1	12.356
2	20.808
3	25.599
4	32.922

The values in table 3 differ from values calculated during the whole rotor modal analysis. They are more then 1.5 kHz higher. Anoter surprising fact was that during the whole rotor system modal analysys these 1.5 kHz lower values were obtained for the eigenforms of compressor wheel stability loss and for the wheels blades eigenforms but only for the *groups* of several blades (figure 5)



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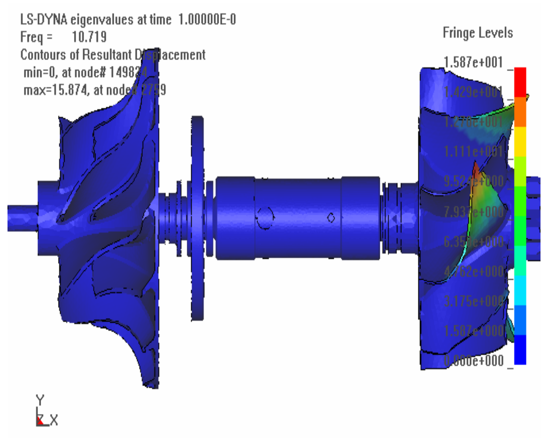
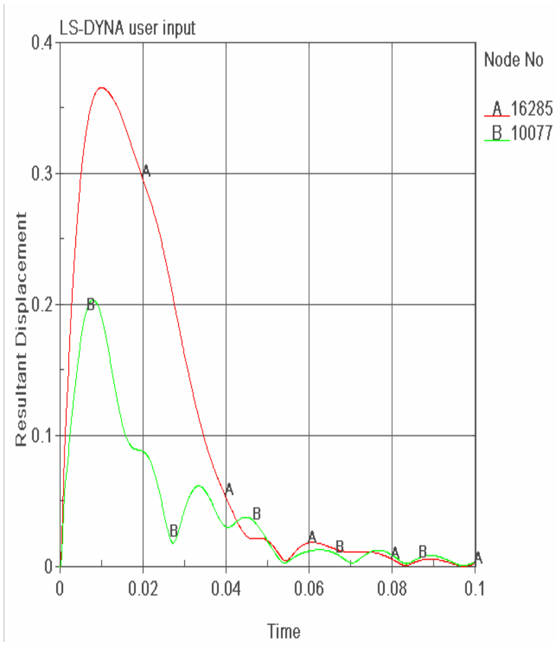
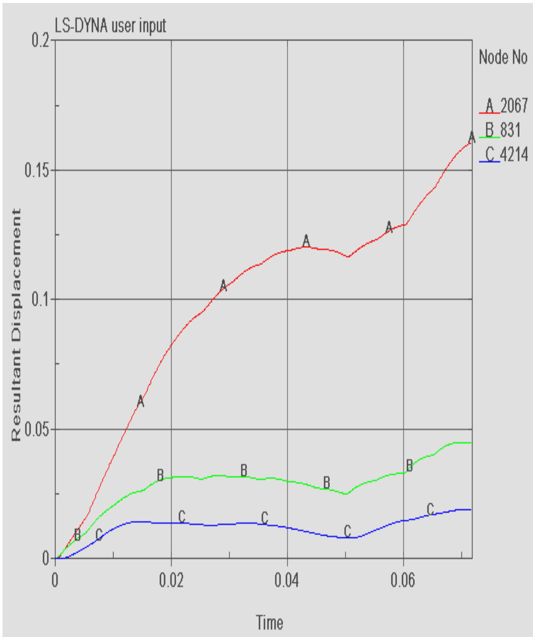


b)

Figure 5 – high frequency eigenform a) - Compressor wheel stability loss eigenform b) - the turbine wheel blades eigenform

We tried to model the simplest blade resonance mechanism when the blade is exited by pulses of any nature. In figure 6a the fading single 0.1 мкс pulse can be seen. If the pulse is repeated with the high frequency it may cause the increase in the blade amplitude (figure 6b).





a) b)

Figure 6 - a) The blades response to short pressure pulse, b). The blades response to the loading conditions that caused resonance.

References

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