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ASPECTS REGARDING SIMULATION AND TESTING TURBOCHARGER SHAFTS

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Abstract: *Due to down-sizing phenomena, in designing and development of internal combustion engines the accent is put on turbocharging in order to maintain, or to raise the engine performances with less fuel consumption.*

Keywords: Turbocharger; vibration; shaft; modal analysis; natural frequencies; mode shapes

From the most common methods of charging engines we mention the turbocharging using a turbocharger. Recent researches in the field of turbochargers show that specific models of turbochargers are capable, in some functioning conditions, of reaching rotational speeds over 250000 rot/min. Turbocharging always raised, from the very beginning, major problems related to noise and vibration traced to the turbocharger rotor. Due to high rotational speeds it is absolutely necessary to maintain a rigorous design and also to test the dynamical behavior of the turbocharger rotor. Classical solutions of turbochargers use hydrodynamic bearings, but modern turbochargers try to use hybrid rolling bearings with ceramic rolling elements. Also due to the friction and dilatation properties of the ceramic materials, the classical rolling bearings with steel rolling elements were replaced with bearings with ceramic rolling elements and steel raceways, in domains where it is necessary to maintain functioning at high speeds. In order to mount the rolling hybrid bearings on turbochargers shafts it is necessary to accomplish some modifications of the shaft. Two identical rotors one with hydrodynamic bearings, and one with rolling hybrid bearings, with same geometrical characteristics regarding the compressor and turbine, will not behave in the same way due to different shaft solutions.

This paper is focused on showing the vibration behaviour of different types of turbocharges shafts in order to establish which solution is probably more fitted to reach a stable dynamic behaviour under vibrations (1). In order to present the aspects mentioned above it were made some simulations using software solutions and laboratory tests.

MAIN SECTION

In this study it will be presented some results obtained by simulating and testing of two turbocharger rotors that equip engines for automobiles.

For 3D modeling it were taken into consideration two types of turbocharger rotors: one that is a classical model equipped with hydrodynamic bearings figure (1), and the other rotor is modified in order to be equipped with hybrid rolling bearings figure (2).

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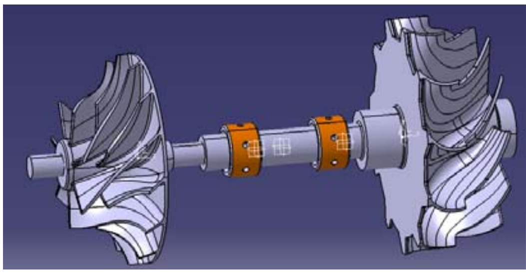
Figure 1. Classical turbocharger rotor **Figure 2.** Modified turbocharger rotor

The first step in modeling the two rotors mentioned above, was to identify the main geometrical parameters of the rotors. In this stage it were accomplished several precise measurements using specialised equipment.

The second step was to decide which software solution program will be right one in order to model the two turbocharger rotors showed in Figure (1) and (2). Taken into consideration criterion of precision and easiness in modeling it were used the following software programs:

- For modelling the shaft type elements it was used CATIA V5;
- For modelling the blade type elements it was used ANSYS.

The models obtained after 3D modelling are presented in figures (3) and (4) as follows:



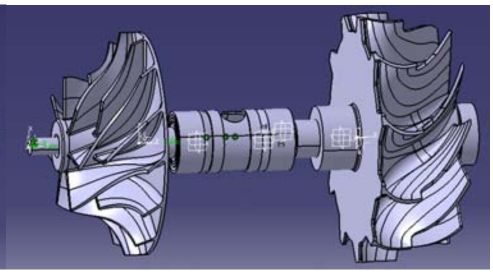


Figure 3. 3D model of classic rotor **Figure 4.** 3D model of modified rotor

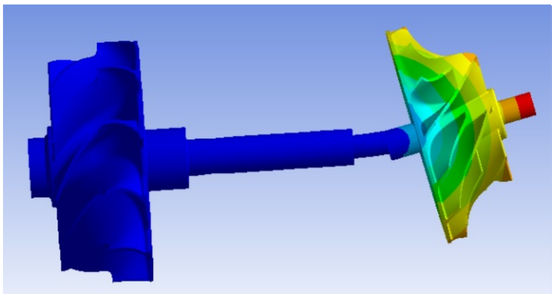
For understanding the phenomena that appears at the level of turbochargers in functioning conditions it were been made several simulations using the software solutions programs mentioned above.

In order to identify the vibrations that need to be avoided during functioning, it has been accomplished a modal analysis of the two turbocharger rotors. The modal analysis revealed the values of vibration mode shapes of the rotors, values, which were furthermore used in evaluate the modal analysis values obtained on the test rig, using the impact hammer method. The values of the natural frequencies obtained using software simulations for the classic rotor are presented in table no.1 and some modes shapes in figures (5) and (6).

Table1

Tabular Data

	Mode	Frequency [Hz]
1	1.	900.98
2	2.	905.32
3	3.	921.04
4	4.	1338.5
5	5.	1343.5
6	6.	2462.3
7	7.	2501.4
8	8.	2957.6
9	9.	3188.4
10	10.	3190.1
11	11.	3191.9
12	12.	3199.2
13	13.	3200.7
14	14.	3201.7
15	15.	3205.2



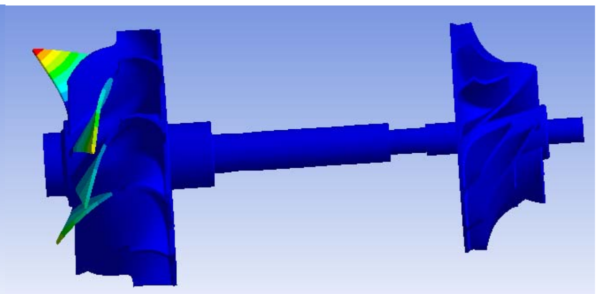


Figure 5. Mode shape for the classic rotor **Figure 6.** Mode shape for the classic rotor

The values of natural frequencies for the modified rotor presented in figure (2) are presented in table no.2 as follows:

Table 2

Tabular Data

	Mode	Frequency [Hz]
1	1.	825.56
2	2.	844.34
3	3.	1170.3
4	4.	1176.6
5	5.	1434.6
6	6.	1476.3
7	7.	3178.3
8	8.	3182.1
9	9.	3183.8
10	10.	3185.
11	11.	3186.7
12	12.	3190.2
13	13.	3190.6
14	14.	3196.7
15	15.	3219.5

It is mentioned that in the software analysis accomplished in this paper work it were taken into consideration only the natural frequencies corresponding to shaft type elements, due to the fact that in functioning conditions the damage of turbochargers appear due to bearing malfunctioning, or shaft bending determined by forces and high rotational speed functioning (3). The modal analysis done using software solution programs was accomplished in such conditions like the ones used in laboratory tests, referring to environment temperature and rotors material.

Even if the materials of which the rotors are made of and also the testing conditions were maintained the same in the software simulations and also laboratory testing it will be observed that a small difference appear between the values of the natural frequencies obtained using the two methods.

In order to validate the modeling data obtained by software simulations it has been accomplished a modal analysis of the rotors using laboratory equipment. In this study the method used in laboratory modal analysis tests, was the impact hammer method (5), accomplished with Pulse 12 Platform supplied by Bruel&Kjaer. For the laboratory tests it were used two accelerometers which have a small mass compared to the tested structure, in order not to influence the values obtained.

The first step was to establish the method used to sustain the tested shafts. It was used a metallic frame strong enough to sustain the structure, underneath it has been placed a rubber plate in order to isolate the structure from the environment, from which to the structure could induced unwanted vibrations, which can change the measurement results. The two shafts were sustained by two wires which have a high rigidity. The equipment settings are shown in figure (7). In this figure we can observe the way the links had been made at the level of channels on the measurement platform:

- Channel no.1 links to the impact hammer;
- Channel no.2 links to accelerometer no.1;

- Channel no.3 links to accelerometer no.2.

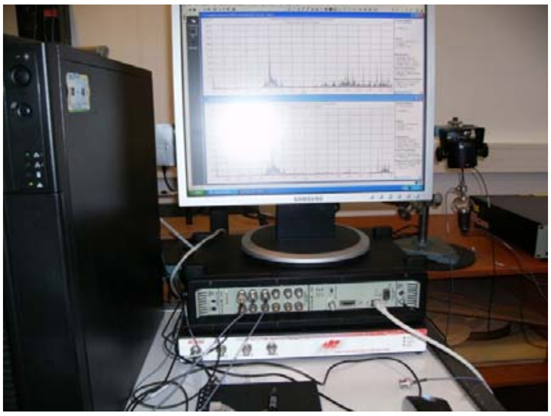


Figure 7. Measurement platform

In figures (8), (9) we can observe the signal gathered from the two accelerometers for the unmodified turbocharger rotor.

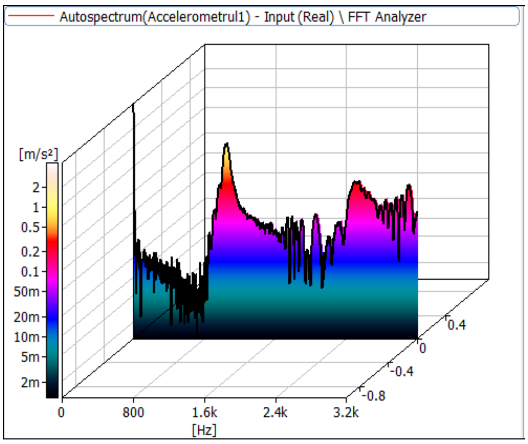


Figure 8. Signal at accelerometer no.1

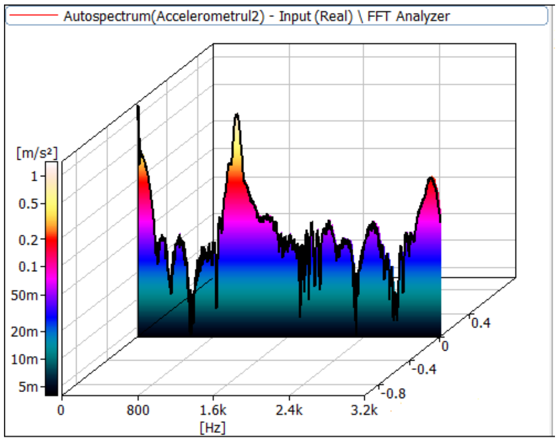


Figure 9. Signal at accelerometer no.2

For accomplishing the first order transfer function (H1) for the classical turbocharger rotor the signals gathered from the force generated by the impact hammer and the two accelerometers needed to be compound and the results are highlighted in figures (10) and (11).

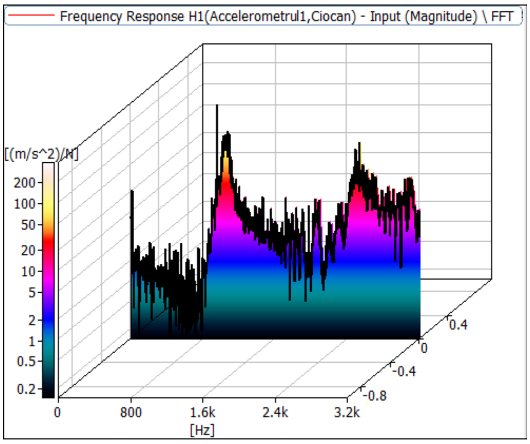


Figure 10. H1 (TF) for accelerometer no.1

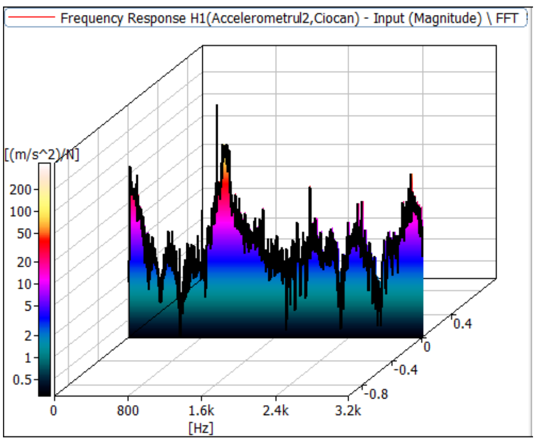
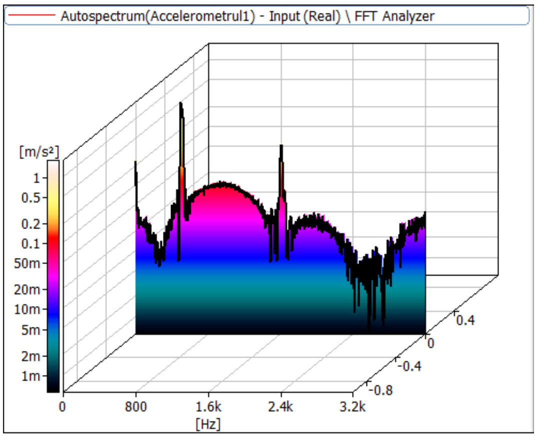


Figure 11. H1 (TF) for accelerometer no.2

The same procedure and the same signals were observed for the modified rotor in order to establish a comparison scale between the performance regarding the mode shapes of the studied turbocharger rotors.

In figures (12), (13) we can observe the signal gathered from the two accelerometers for the modified turbocharger rotor.



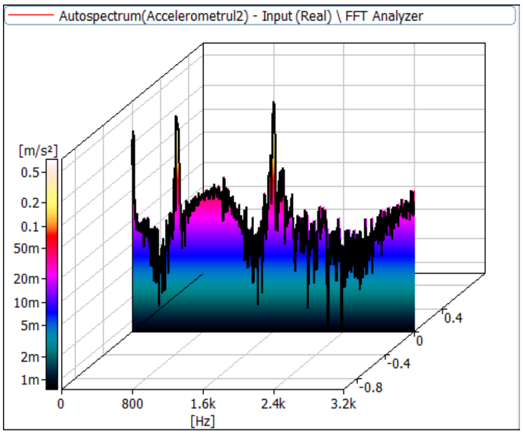
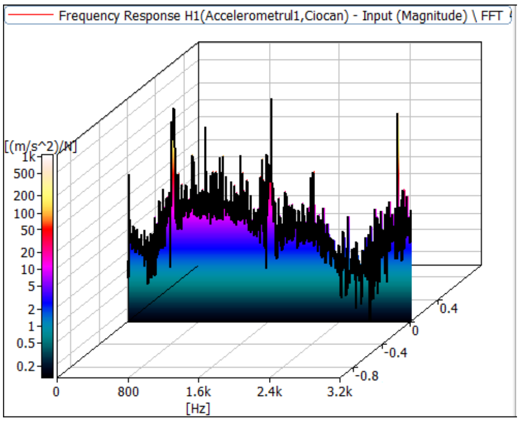


Figure 12. Signal at accelerometer no.1 **Figure 13.** Signal at accelerometer no.2

The first order transfer function (H1) for the modified turbocharger, are presented in figures (14) and (15).



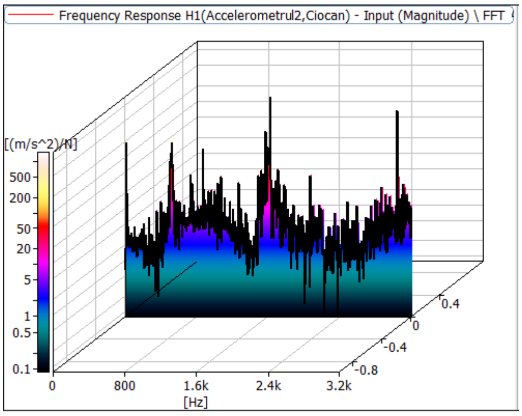


Figure 14. H1 (TF) for accelerometer no.1 **Figure 15.** H1 (TF) for accelerometer no.2

The accelerometers placement and impact point of hammer, are showed in figure (16).

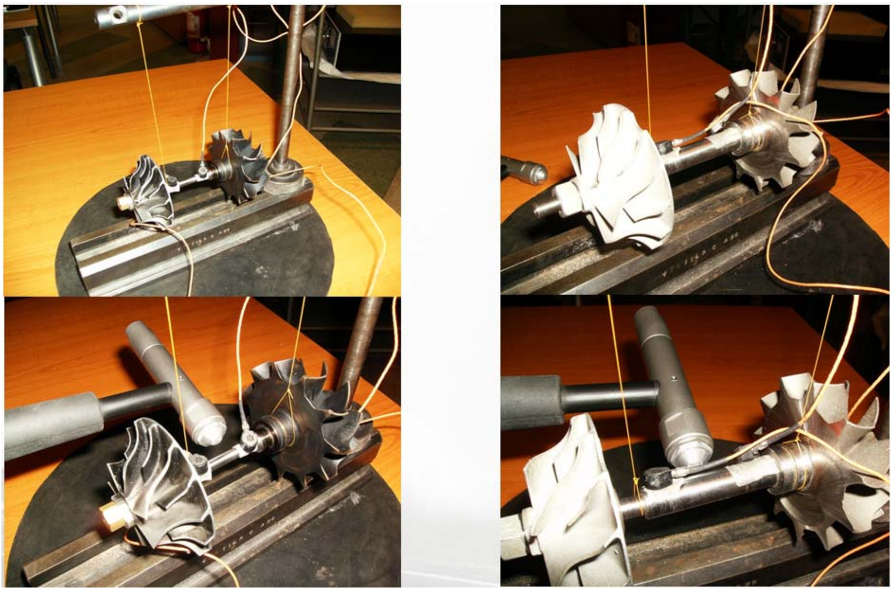
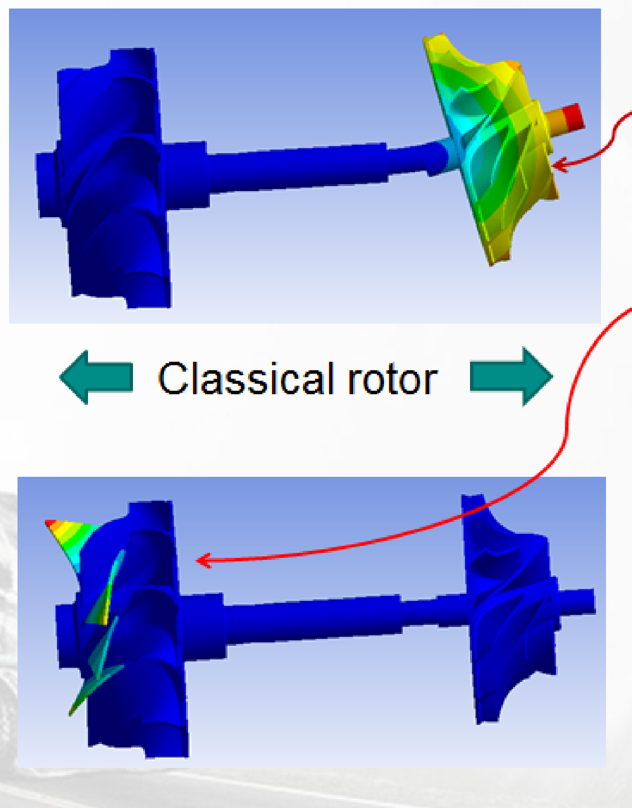


Figure 16. Accelerometer placement and hammer impact point

In the followings it will be presented a comparison between the values of natural frequencies obtained by software simulations and laboratory tests for each turbocharger rotors, the table in the left shows the values obtained by laboratory tests and the right one values obtained by software simulations.

Simulations that have accomplished using software solutions, took into account the exact geometry of the two considered rotors and also the material library that have been used was, in such manner defined in order to maintain the same material properties as the real turbocharger rotors. Studying the specific literature regarding testing turbocharger rotors, it was observed that generally speaking turbocharger rotors have 9 principal mode shapes. Other mode shapes of the shaft can be studied as variations or derived from the 9 principal mode shapes. The simulations accomplished took into consideration only the mode shapes generated at the level of the shaft because the blades mode shapes, do not occur in functioning conditions. The vibrations (bending) of turbocharger shafts can, involve a malfunctioning of the turbocharger, concluding with a total damaging of the mechanical system.

Frequency [Hz]	Damping Ratio [%]
900	0.196
912	0.184
924	0.153
944	0.148
960	0.192
988	0.157
1008	0.154
1024	0.324
1036	0.186
1072	0.191
1088	0.149
1120	0.229
1204	0.112
1036	0.117
1072	0.098
1088	0.098
1120	0.109
1204	0.094
1232	0.121
1412	0.08
1484	0.103
1596	0.135
1760	0.067
1836	0.074
1928	0.062
2004	0.055
2144	0.052
2240	0.053
2304	0.046
2348	0.041
2392	0.05
2484	0.04



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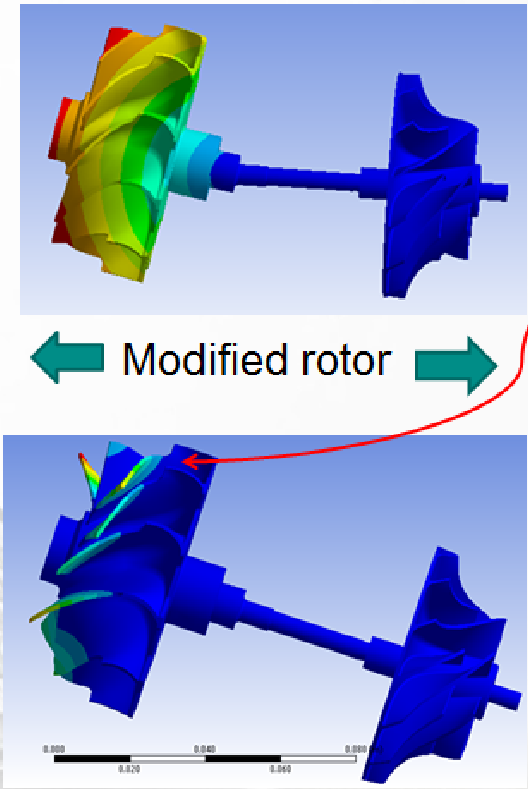
14	14.	3201.7
15	15.	3205.2
16	16.	3207.
17	17.	3330.
18	18.	3336.4
19	19.	4100.6
20	20.	7665.
21	21.	8797.4
22	22.	8962.6
23	23.	8991.
24	24.	9113.1
25	25.	9125.9
26	26.	9134.8
27	27.	9180.2
28	28.	9184.7
29	29.	9239.7
30	30.	9256.4

Frequency [Hz]	Damping Ratio [%]
512	2.19
584	0.985
792	0.807
928	0.684
1248	1.41
1312	0.296
1376	0.44
1424	0.244
1448	0.281
1464	0.365
1512	0.241
1544	0.221
1560	0.278
1624	0.264
1696	0.303
1776	0.486
1824	0.77
1888	0.566
2096	0.238
2216	0.296
2536	0.109
2744	0.102
3024	0.171
3080	0.209
3144	0.221
3192	0.134



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13	13.	3190.6
14	14.	3196.7
15	15.	3219.5
16	16.	3224.2
17	17.	3300.1
18	18.	7591.7
19	19.	7661.
20	20.	8020.5
21	21.	8066.2
22	22.	8805.6
23	23.	8889.9
24	24.	8936.9
25	25.	9006.7
26	26.	9162.5
27	27.	9168.8
28	28.	9222.4
29	29.	9228.3
30	30.	9232.3

CONCLUSIONS

Concluding we can observe that the differences between measurements and simulation are not so relevant, referring to shaft type elements. The blades have an unpredicted mode shape, but these blade mode shapes are not so relevant during functioning conditions of turbochargers. To study the dynamic behaviour of turbocharger rotors is needed to simulate the parameters of rotational speed (4), balancing, waste gases temperature, eccentricity, parameters that require expensive equipment. Regarding the vibration prediction of turbocharger rotors we can observe from practice that the unwanted vibration occur during forced acceleration and deceleration of the rotor shaft. During the acceleration process the rotor can have stabilized motion. This stabilized motion can be obtained at higher rotational speed were the gyroscopic effect takes place. These rotational speeds, in some cases do not need to reach higher levels, the stabilized dynamical motion can be also obtained at slower rotational speeds. If the rotor dynamical motion can't be predicted and if is impossible to obtain a stabilized rotor motion, engineers use in practice additional damping devices in order to stabilize the shaft motion. Additional damping devices can be of the form of squeeze film dampers (SFD) or can use different forms of mechanical systems.

We can say that a full prediction of turbocharger rotors vibrations can't be obtained because, the influence of different parameters related to: speed; balancing; temperature play an important role in developing or not dynamical motion stability.

The paper work accomplished above is focused on developing some information related to vibrational mode shape of turbochargers, rotors involving the influence of rotor geometry over this mode shapes and also on developing a comparison data set between software simulations and laboratory testing.

REFERENCES

(1) Chiriacescu, S. T., “Vibrații în construcția de mașini”, Editura Univesității Transilvania, Brașov, 1982.
(2) Harris & Crede, “Schock and Vibration ” , Editura Tehnică, București, 1985.

(3) Radeş, M., “Maşini rotative”, Editura Universității Politehnice Bucureşti, Bucureşti, 2005.
(4) Rosca, I.C., “Vibratii mecanice”, Editura Infomarket, Brasov, 2002.
(5) Modal testing: theory, Practice and application second edition D.J. Ewins, Wiley,2001

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