Preliminary analysis of turbochargers rotors dynamic behaviour

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Abstract. Turbocharger rotors for the spark and compression ignition engines are resistant steels manufactured in order to support the exhaust gas temperatures exceeding 1200 K. In fact, the mechanical stress is not large as the power consumption of these systems is up to 10 kW, but the operating speeds are high, ranging between 30000 ÷250000 rpm. Therefore, the correct turbochargers functioning involves, even from the design stage, the accurate evaluation of the temperature effects, of the turbine torque due to the engine exhaust gases and of the vibration system behaviour caused by very high operating speeds. In addition, the turbocharger lubrication complicates the model, because the classical hydrodynamic theory cannot be applied to evaluate the floating bush bearings. The paper proposes a FEM study using CATIA environment, both as modeling medium and as tool for the numerical analysis, in order to highlight the turbocharger complex behaviour. An accurate design may prevent some major issues which can occur during its operation.

1. Introduction
Turbocharger is a key component in modern engines in order to increase their power. There are also great benefits by supercharging the engine because it is possible to obtain the same power output considering that the engine size and weight are more and more reduced. Turbocharging combined with this “downsizing” is also an effective way to reduce the engine consumption [1, 2].

There are various methods to increase the engine power, but one of the most used is to increase the airflow into the cylinder. A turbocharger consists of a turbine and a compressor located on a shaft. The turbocharger running is initiated by the engine exhaust gases acting upon the turbine. Therefore, this component is subjected to high temperatures exceeding 1200 K and the working turbine rotation speeds can reach 100000 ÷250000 rpm [3]. On the other shaft end, the compressor who aspires fresh air and compresses it is positioning. The resulting air pressure (commonly 1.5÷2.7 bar) is called boost pressure and the turbocharger systems are measured by its amount. Therefore, the fuel/air combination substantially improves the engine volumetric efficiency with consequences on the engine power, which could be considerable increased and also, the fuel consumption is reduced. The turbine and the compressor are usually supported on floating ring or semi-floating ring bearings and the oil film also contributes to the complex turbocharger system behaviour. The engine technological development also requires the improvement of turbochargers dynamic characteristics, therefore theoretical research becomes more significant and necessary in order to realize an exhaustive covering model. First, they were one-dimensional codes expressing quasi-static approach based on the turbocharger rotors characteristic maps usually provided by the manufacturers [4]. However, the performance data from
maps are recorded on the test bench in a steady operation of the turbocharger system, while in a real situation, the turbine and the compressor operate under unsteady conditions caused by the engine pulsations [5, 6]. As consequence, another serie of codes with varying degree of complexity were resulting in different approaches taking into account of heat transfer, mechanical losses and unsteady behaviour during the engine operation [7].

In the last years, the models are verified through experimental measurement procedures with accelerometers [8] and laser [9, 10]. Contribution to improve the knowledge in this field is related to the lubrication problems studied both experimentally, by measuring the thermodynamic variables in turbocharger test benches [11], and theoretical, by means of computational fluid dynamics CFD models [12, 13].

In the paper, some dynamic aspects of turbocharger operation, considering von Mises stress resulting from the mechanical and temperature effects and vibration data have been explored using a general computer-aided engineering tool (CATIA). The results highlight the FEM capabilities in order to evaluate the turbocharger behaviour, as a preliminary analysis of its potential issues.

2. Turbocharger model considerations

The turbocharger turbine is of steel and the compressor wheel is of aluminum [3], with the appearance from figure 1.

![Figure 1. Turbocharger supported on semi floating ring bearing.](image)

The turbocharger is supported by floating or semi floating bush ring bearing, which is free to rotate. The floating bush ring is commonly used because it is an efficient and cheap component. The design concept for the floating bush bearing was to reduce friction losses. Therefore, the inner clearance is smaller than the outer clearance, so the bushing rotation is a fraction of shaft speed. The ring speed is determined by the friction torque balance between the inner and outer films. The rotation of the outer surface of the floating bush bearing acts as a film damper. The pressure distribution is mainly generated by the rotational speed of the ring, its precessional rate and in addition the thermal effects influence the analysis [14].

There are several reasons for turbocharger failure including the lubricant lack and contamination, the insertion of foreign objects in the system and low power or boost caused by a gas leak or blocked cooler restricting air injection [3]. The most common turbocharger fault is the rotors unbalance followed by system severe vibrations. The phenomena complexity leads to a difficult analysis, mainly caused by its nonlinearities. However, a linearized analysis can be performed if the nonlinear oil film forces are expressed as the linear combination of the stiffness and damping coefficients [15]. Simplified concepts, illustrating a possible algorithm which can be applied to the turbocharger model (figure 2), include the oil films pressures and the reaction forces computation (figure 3) and then the rotors dynamic investigation through the displacements, stress and frequencies evaluation [16].
The film pressure $P$ of both outer and inner oil film, with the viscosity $\eta$, could be estimated by Reynolds equation:

$$
\frac{\partial}{\partial \psi} \left( \frac{h^3}{12\eta} \frac{\partial P}{\partial \psi} \right) + \frac{\partial}{\partial z} \left( \frac{h^3}{12\eta} \frac{\partial P}{\partial z} \right) = \frac{V}{2} \frac{\partial h}{\partial \psi} + \frac{\partial h}{\partial t}.
$$

(1)

For the outer oil film: $h$ is the floating ring – bearing central film thickness, $V$ is the floating ring linear velocity, $\psi$ and $z$ are the circumference and axial coordinate of outer oil film. For the inner oil film: $h$ is the journal – floating ring central film thickness, $V$ is the floating ring – journal relative linear velocity, and $\psi$ and $z$ are the circumference and axial coordinate of inner oil film respectively.

As the effective analytical expression of the whole Reynolds equation is still difficult to obtain, the equation (1) are numerically solved and two pressure oil films and reaction forces are estimated [15].

The vibration analysis allows the observation of the each mode shape and its stress distribution. The energy of deformation occurring in the most sensitive areas and generating significant vibratory movements can be also estimated, thus representing an efficient tool for assessing the dynamic behavior of the turbocharger system.

The calculation of natural vibration modes ignores the external force and, without damping, the equation of motion for the system can be written in matrix form as follows [15]:

$$
[m] \cdot \{\ddot{u}\} + [k] \cdot \{u\} = \{F_{unbalance}\} + \{F_{inner\,film}\} - \{F_{shaft}\}.
$$

(2)

where: $[m]$ is the mass matrix, $[k]$ is the stiffness matrix, $\{u\} = \{x, y\}$ is the vector representing the system displacement, $\{F_{unbalance}\}$ is unbalance centrifugal force, $\{F_{shaft}\}$ is the shaft force due to its weight, $\{F_{inner\,film}\}$ is the hydrodynamic force of the inner oil film.

The ring of floating bearing may have a vibration of following form:

$$
m_{ring} \cdot \ddot{\mathring{u}}_{ring} = \mathring{F}_{outer\,oil} - \mathring{F}_{inner\,oil}.
$$

(3)

where: $m_{ring}$ is the ring mass, $\mathring{u}_{ring}$ represents its displacement and $\mathring{F}_{outer\,oil}$ is the hydrodynamic force of the outer film.

The solutions for the equation (2) represent a synchronous harmonic motion in all its coordinates and are adopted of the following form:

$$
\{u\} = \{a\} \cdot \sin(\omega t + \theta).
$$

(4)

where: $\{a\}$ is the natural vector, $\omega$ is the natural pulsation, $\theta$ is the phase shift.
The results of equation (1) may be introduced into equations (2) and (3) and an integration iterative method can be applied for the specified time interval. [15]

Solving together equations (2), (3) and (4) can provide the theoretical solutions for the natural vibrations. Through the modal analysis, each natural pulsation $\omega_i$ ($i = 1, 2, \ldots, n$) with a corresponding modal vector $\{u_i\}$ defines a state where it is possible a harmonic movement, therefore the mode shape can be underlined.

3. Preliminary results and discussions

It is proposed a FEM study using CATIA environment, both as a modeling medium and as a tool for the numerical analysis. Two methods suggested in figure 4 were used.

![Figure 4](image)

(a) Shaft with turbine, without compressor; (b) Shaft with turbine and compressor.

The shaft with the turbine rotor from figure 4a was conceived of steel, as common body, because the important temperatures values are only reaching the turbine area. For the model from figure 4b, the compressor rotor was also designed of steel, integral with the turbine and the shaft, but with the same equivalent mass as the aluminium rotor (as it is the real compressor made), in order to assure the same vibrational level as the real one.

The simulations are focused on the following analyses:
- the von Mises stress distribution as result of the engine exhaust gas mechanical effect, on the model from figure 4a;
- the thermal effects on stress distribution, using the model from figure 4a (the compressor is cooled by the fresh air);
- the thermal and mechanical stress distribution with overlapping effects;
- the shaft with turbine (figure 4a) and the whole ensemble (figure 4b) behaviour in terms of their natural frequencies and shape modes.

The main dimensions used in the simulations are presented in table 1. There are 12 blades both for the turbine and the compressor.

| Component                    | Dimensions (mm) |
|------------------------------|-----------------|
| Turbine rotor diameter       | 48              |
| Compressor rotor diameter    | 52              |
| Turbine rotor length         | 18.5            |
| Compressor rotor length      | 26              |
| Bearing spindle length       | 31              |
| Bearing spindle diameter     | 8               |
| Compressor mounting length   | 28              |
3.1. Mechanical effects

A static torque resulting from the turbine power (due to the exhaust gas), for various operating speeds, was considered. Figure 5 reports von Mises stress distribution, on the model from figure 4a. The model is subjected to a torque of $10^3$ Nmm corresponding to $10^5$ rpm shaft speed. The simulation provides significant von Mises stress (up to 39 MPa) in the embedded rotor blades.

![Figure 5](image1.png)

**Figure 5.** Von Mises stress distribution for the shaft with turbine, at 1000 Nmm torque.

Obvious, the largest displacements are found at the top of the blades, but they are no more than $7 \cdot 10^{-3}$ mm. The nodal stress from the axial surface of the rotating contact between the floating ring and the turbine body don’t exceed 10 MPa, therefore the operating conditions are ensured. From this point of view, the turbine blades are the elements supporting the maximum stress. It can be noticed that finite elements of octree tetrahedron parabolic with 7 mm size (maximum) were used in the analysis.

![Figure 6](image2.png)

**Figure 6.** Maximum von Mises stress evolution vs. the shaft speed.

![Figure 7](image3.png)

**Figure 7.** Maximum Von Mises stress evolution according to the temperature.
Supposing a constant power engine, the power transmitted to compressor-ventilator ensemble can be also considered constant. Figure 6 depicts the maximum von Mises variation on the turbocharger model from figure 4a, subjected to the FEM analysis at various turbine speeds. It can be remark that there are no dangerous stress for usual operating regimes. The compressor rotor is generally realized of light alloys and is not exposed to the flue gas pressure, as consequence there are no significant mechanical stress developed in its area. For this reason, the compressor isn’t included in the analyses concerning the equivalent stress von Mises characterizing the turbocharger behaviour.

3.2. Thermal effects
For highlighting the flue gas temperatures influence on the turbine and possible on the shaft, it was performed a FEM analysis in the temperature field. As the engine exhaust gas has 1100 ÷ 1500 K temperatures, they are considered in the analyze of mechanical stress induced by this thermal effect.

The analyze was performed on the model from figure 4a, and the compressor rotor was neglected in this regard, as it is cold due to the fresh air flow circulated. The results are presented in figure 7 where the maximum von Mises stress values are calculated for each temperature value. The model subjected to 1400 K temperature is suggested in figure 8. It is emphasized the most thermal exposed zone, by displaying one of its nodal stress value.

The thermal expansion is also the cause of the displacements which occur especially along the shaft.

For the same considered temperature field, as in von Mises stress analyze, the displacements values are of 0.553 ÷ 0.87 mm, representing a normal behaviour for this ensemble.

Briefly, the mechanical stress exclusively due to the turbine absorbed power from the exhaust flue gas was first considered in FEM analysis, and on the other hand, another serie of analysis has focused only to the thermal influence on the stress occurrence on the model from figure 4a.

![Figure 8](image)

**Figure 8.** Von Mises stress nodal value due to the thermal effects, evidenced on the shaft extremity.

3.3. Superposition of mechanical and thermal effects
FEM analysis solves the mechanical and thermal efforts simultaneous applied on the system through the effects superposition. As it can be seen, the thermal stress, determined in the isostatic hypothesis, don’t exceed 10% from the mechanical stress values. For instance, for the same node where the
thermal stress is 2.26 MPa, see figure 8, the addition of the most defavorable mechanical stress produces the increase of von Mises stress to 25 MPa. Ten times increase of the von Mises stress justifies the use of the effects superposition as a correct assumption in evaluation of the turbocharger behaviour. Therefore, its operating conditions are insured, taking into account of mechanical and thermal efforts, if there are allowed enough clearances between the system components.

3.4. Natural frequencies analysis

The turbocharger virtual models from figure 4 were also considered in the estimation of the vibration eigenmodes of the systems. The analysis supposes that the floating bush bearing is linked to the shaft through two points as supports. These two points represent smooth-virtual part links realized between the journal and the floating ring. The circular surface used for the axial support of the turbine represents the third link of the model. In FEM analysis, it were imposed links of User defined Restraint type for the two smooth-virtual parts, eliminating the displacements whose directions are perpendicular to the shaft. The same restriction applied to the compressor impose the axial displacements removing. With the isostatic assumption, the turbocharger natural frequencies can be estimated.

To avoid the use of several materials and to realize a more uniform model, a steel compressor with thin structure (empty inside its body) was used in the analysis of the turbine-shaft-compressor ensemble. Thus, the compressor has the same mass as an aluminium one and the model may be considered identical with the real turbocharger system. The FEM simulations were performed for both models from figure 4.

Figure 9 represents the shape of the compressor in the sixth vibration eigenmode. Obvious, the representing scale leads to a distorted view.

Figure 9. The compressor shape in sixth vibration eigenmode.

Figure 10. Vibration eigenmodes.
In fact, the ten eigenmodes for the two analyzed turbochargers models are expressed in figure 10. The critical speeds with vibrations damage occurrence can be easily calculated. As result, we note the critical speed of 144000 rpm for the compressor-shaft-turbine ensemble (figure 10a) and a critical speed of 135000 rpm for the model with turbine, without compressor (figure 10b). These values correspond to the first vibration mode.

3.5. Floating bush bearing considerations

The lubrication of this bearings type is provided by an engine oil, without pressure, in a hydrodynamic regime. Classical hydrodynamic lubrication approach for the analyzed turbocharger leads to some results, as: the oil thickness is 13 μm; the flow leakage is $12 \cdot 10^{-5}$ l/min; the friction power loss is about 75 W. The results are acceptable and represent a start point in considering this bearing influence on the turbocharger dynamic behaviour.

However, the floating bush bearings have a specific geometry (the width-diameter raport is greater than 3) and the rotation speeds up to $10^5$ rpm. One of the most difficult issues in the analysis of the floating bush is the assumption of the ring speed. The rotation of the outer surface of the floating bush bearing acts as an uncentered squeeze film damper. The analysis is complex because, even with the assumption of ring speed, the system is nonlinear [14].

As consequence, the lubrication study must be consistent and specific for their data, in order to avoid unbalance and large oil thickness affecting the system dynamic stability. The very large obtained values of Sommerfeld number (up to 200) emphasize the need for an exhaustive future theoretical work focused on the turbochargers lubrication.

4. Conclusions

The paper presents some considerations on the dynamic behaviour of turbochargers, using the FEM analyses provided by CATIA soft. The analyses were performed using a commercial turbocharger data. The mechanical stress induced by the engine exhaust gas are active on the turbine blades, but the simulations show that they are not dangerous for the whole system, also including the compressor. On the other hand, the mechanical stress values induced by the high temperature of the flue gas, represent 10% from the mechanical stress due to the turbine operating power. The natural vibrations analysis of the turbocharger leads to ten eigenmodes and it is evidenced the vibration danger if the first mode is reached. The stiffness increasing of the system leads to an increase of the critical frequencies. The floating bush bearing lubrication, assuming the very high ring speeds, has to be detailed studied, using computational fluid dynamics codes. Our analyses show that the suitable design of a turbocharger, using FEM simulations, not necessarily a dedicated soft, may provide a convenient tool for manufactures in order to avoid some major operating issues.

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