Thermomechanical Analysis of the Turbo-Compressor Sliding Bearing Mount Units

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Abstract

On SKIF k-1000 supercomputer the temperature fields in turbo-compressor sliding bearings were defined. Dynamic temperature loads were estimated by calculating of heat conduction in the rotor shaft from 600 C heated turbine to the compressor wheel. Conduction and radiation losses were taken into account. The role of friction in heating the contacting parts of the bearing mount assembly was estimated. The calculation of heat conduction through the rotor shaft were made for different exploitation regimes namely for the starting and working regimes and for the lubrication absence conditions. Different constructive shaft variations were used that made it possible to smooth the temperature peaks in thickening ring of the shaft. The temperature fields in bearing mount assembly details were calculated and the thermal stresses were estimated.

The role of lubrication was estimated in two ways. In preliminary calculations its influence was estimated by taken into account only its convection and radiation properties as the properties of the environment in the bearing unit. Then the whole process of the oil- bearing unit parts interaction was modeled. The problem was solved in ALE formulation. The gap between rotor shaft and sliding bearing was filled with lubrication. The areas of the ALE-mesh that corresponded to the inflow of the oil was prescribed its initial cool temperature. Then the incoming oil was taken with rotating rotor shaft, the later conducting the heat from the turbine wheel.

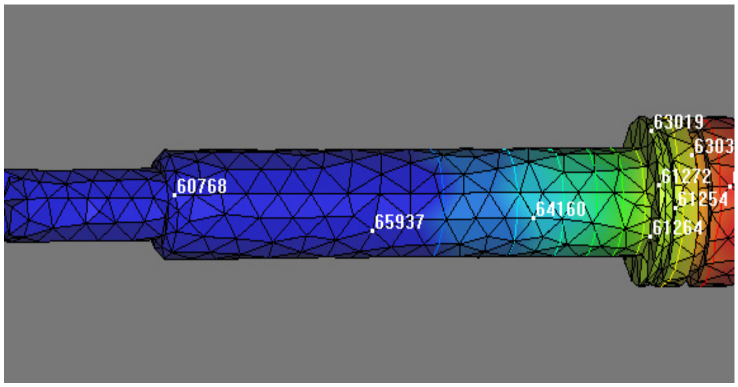
Introduction

While elaborating a new constructive model of turbo-compressor. its units are subjected to manifold tests. To simplify the experiments procedure and to shorten the necessary time the numerical simulations are to be carried out on some crucial aspects of turbo-compressor behavior. In this paper the results of the rotor thermomechanical properties study are represented.

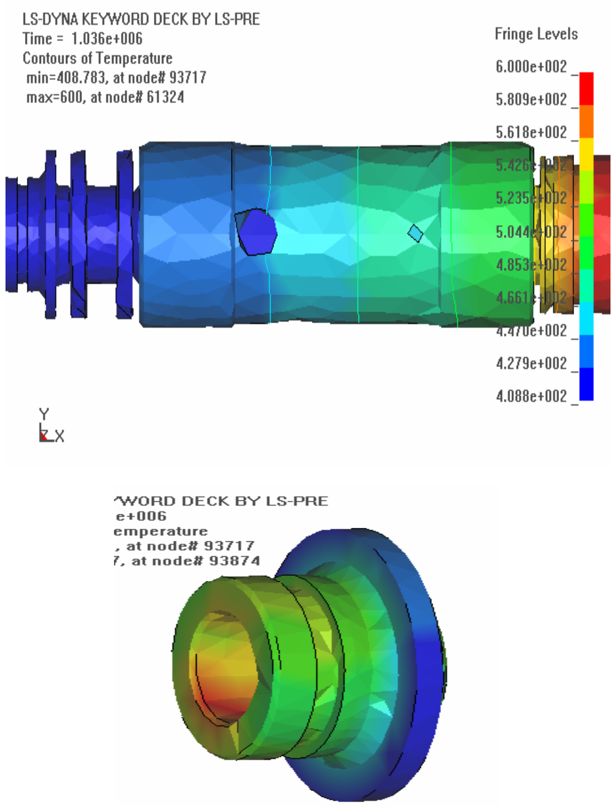
Several different thermomechanical phenomena make influence on the strength of the rotor shaft. Thermal stresses are induced by the dynamic temperature field, friction in sliding bearings may lead to the bearing surface melting, the behavior of the oil in the bearing mount affect the temperature fields in the thickening ring of the rotor shaft.

First the simplest problem of the heat conduction through the rotor shaft from 600^{o}C heated turbine wheel was solved. As the thickening ring of the shaft should not be heated up to the oil coking temperature the different exploitation regimes were modeled to find out if the temperature in the thickening ring was less then critical. We let the turbine have the constant temperature equal to the heated gas temperature (600^{o}C) and supposed that the compressor wheel is also heated up to the constant temperature of 100^{o}C. After a while a quasi-static temperature field distribution through the rotor shaft was obtained. The convection losses were calculated through assigning the oil and air heat transfer coefficient for the cylindrical rotor shaft parts, the flux was supposed to be laminar. For the working regime the temperature of the environment was set to 20^{o}C for air and to 100^{o}C for the oil. The radiation losses were also taken into account. As can be seen from the figure 2c the temperature of the point B from the thickening ring of the rotor shaft (figure 1a) has its static value of about 480^{o}C that is 20^{o}C less then critical. For the starting regime the temperature of the oil was supposed to be 20^{o}C and the obtained temperature values of the thickening ring were about 15-20oC less (figure 2b).

The temperature of the thickening ring was also calculated for the lubrication absence condition (Figure 2a) that was modeled by using only an air as an environment. It is seen that point B of the thickening ring reached 580^{o}C level that is 80^{o}C higher then critical. From figures 2a, 2b, 2c one can see also that the time needed for the temperature reaching its maximal level in the area of the thickening ring is approximately 50c for all exploitation regimes that was approved during the experiments. The units of mc, kN, mm, kg were used in these calculations.

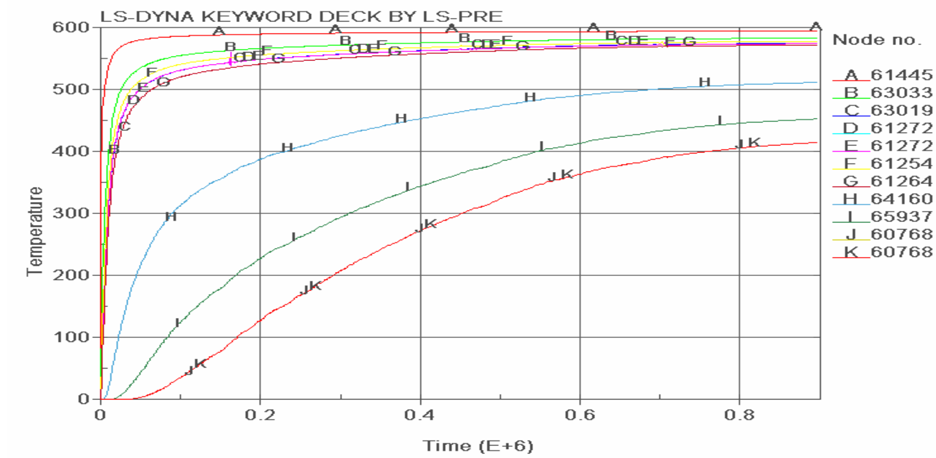


a)



b)

Figure 1 - The temperature fields in bearing mount assembly. a) - temperature fields in rotor shaft, b) - temperature fields in parts of bearing unit.



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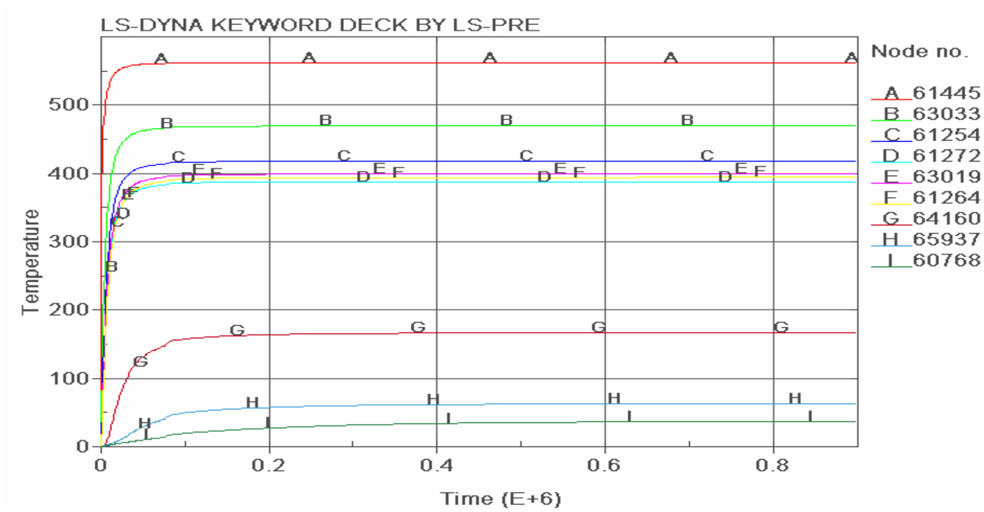

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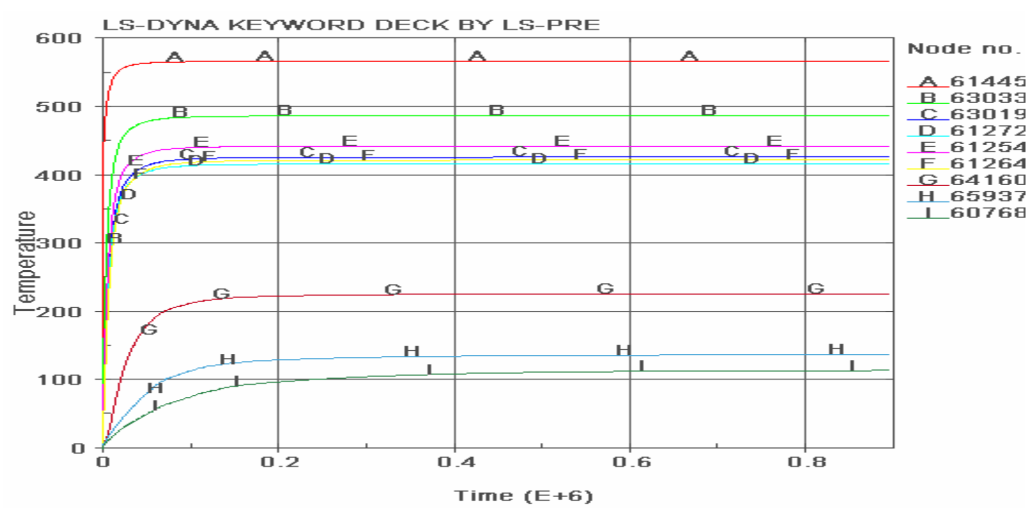
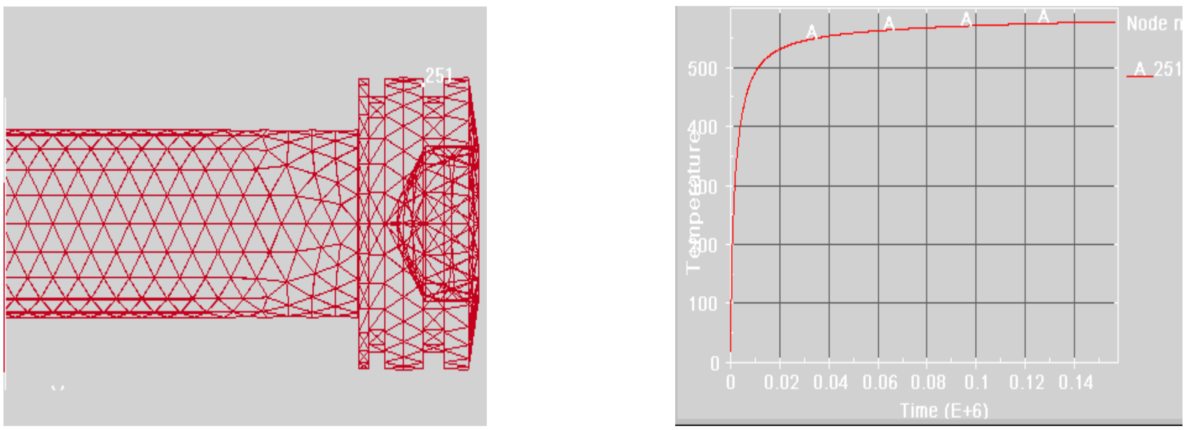


c)

Figure 2 - Heat conduction dynamics in the rotor shaft from 600 C heated turbine to the compressor wheel for the lubrication absence conditions (a), for the starting (b) and working (c) regimes.

As the lubrication absence regime is "bad" for the considered rotor shaft model and the presence of other bearing mount units (figure 1b) has little influence on the quasi-static temperature field distribution in rotor shaft the important question is how to modify the geometrical model of the shaft so that to lessen temperature peaks in thickening ring. The way proposed by the constructors was to increase the thermal resistance of the critical part of the shaft by putting a hole in this area (figure 3a). As can be seen in figure 3b the temperature of the point B for lubrication absence condition is 570°C that is about 15°C less than for the first shaft model but still 65°C higher than oil coking level. The dimensions and form of the hole were varied but the anticipated result was not achieved. Therefore the proposal of changing the rotor shaft design was declined.



a) b)

Figure 3 - Heat conduction dynamics from 600 C heated turbine to the compressor wheel for the lubrication absence conditions in the rotor shaft with the hole

The second problem was to estimate the thermal stresses of the rotor and wheels. The thermal stresses of the wheels were found also in "quasi-static" regime, the temperature of the wheels being considered constant (600°C for the turbine wheel and 100°C for the compressor wheel). Even with such simplification the not trivial stresses distribution on the wheels were seen (figure 4). The complicated geometry leads to the constraining effects of some areas of the wheel on the other. Thus the most thermally "loaded" parts are the ends of the blades and the areas on the wheel disk peripheral. The later is accounted for by the stiffener role of the blades roots. The level of the thermal stresses of the wheels (~10MPa) is about 5% of the inertial dynamic stresses (~250MPa) and can to some extend be ignored in the strength analysis. The rotor shaft thermal stresses are induced mostly by thermal gradients and therefore their values can also be neglected in the rotor shaft strength analysis.

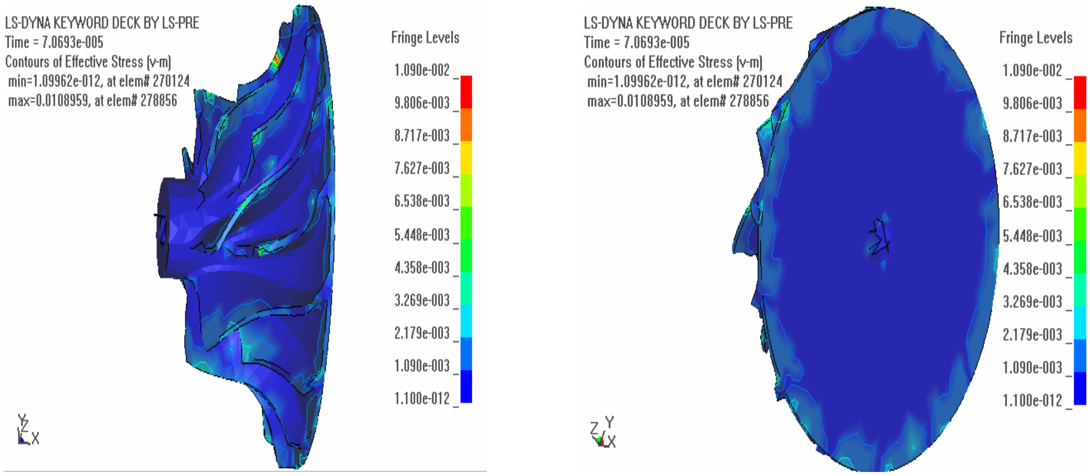


Figure 4 – Thermal stresses distribution on the compressor wheel

Much greater effect on the rotor shaft stresses values is made by friction between rotor shaft and bearing mount units. The calculated value of the temperature increase in the bearing mount due to the friction during shaft rotation was about 90-100^{o}C (Figure 5), the gap filled with oil being 0.01-0.02mm. The numerical experiments were carried out for the rotor shaft angular velocity range 0.8-2 kHz, the oil characteristics were 0.8kg/mm^2 density, 1600MPa bulk modulus, 3.5 mPa*c dynamic viscosity. Friction influence was calculated with the rotor shaft being thermo-mechanically loaded by quasi-static thermal field induced by the constantly heated wheels and an ALE-formulation for the fluid-structure interaction being used (Figure 6).

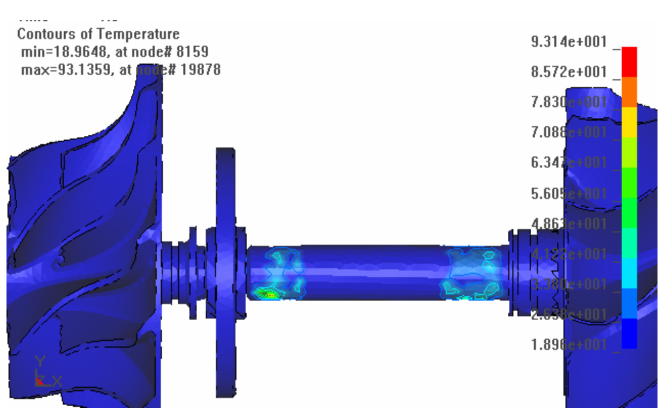


Figure 5. Heating the contacting parts of the bearing mount assembly caused by friction. No temperature loads from turbine wheel and no lubrication are taken into account.

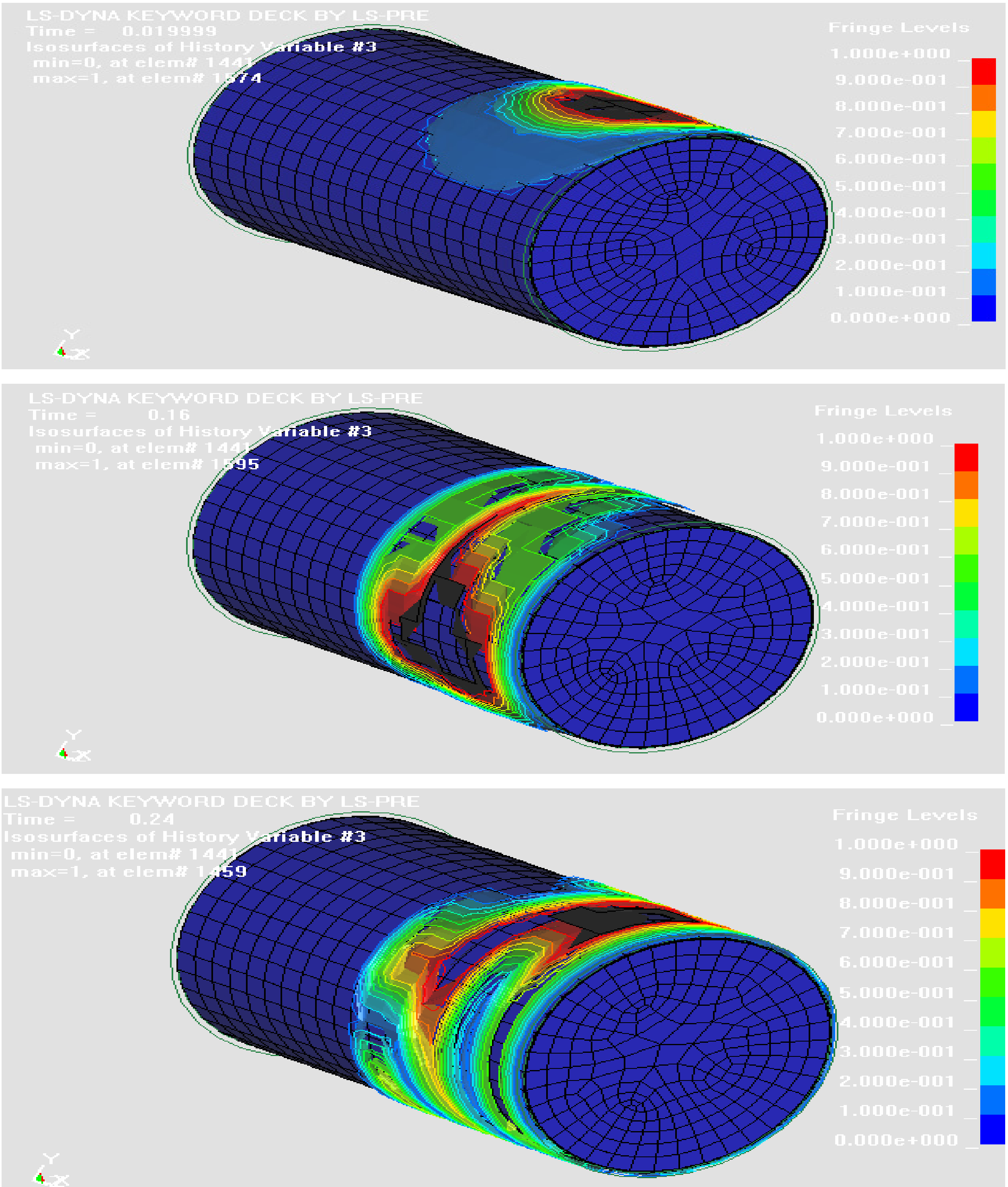


Figure 6 – Fluid-structure interaction between oil spot and rotating cylindrical shaft part

We tried also to model the phenomenon of the sliding bearing melting. It is know that in the absence of lubrication the surface of the babbit bearing melts. We calculated the time needed in this case for the tin babbit material (7.2kg/mm^2, σ_T=600MPa, T_{melt}=350^{o}C) and it occurred to be about 2-3min for the different angular rotor shaft velocities. Mechanical equivalent of heat was set to 1 and static and dynamic coefficients of frictions were varied between 0.1-0.2.

References

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