

Research Article

Analysis of Steam Turbine Coupling Finite Element Method

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Abstract

The manufacturing cost of a component is direction proportional to the difference between the lower and upper specification limits and the functional aspects of the assembly of two components are also affected by the magnitude of fit between those components. In the present work detail, a study of interference fit between coupling and shaft in a steam turbine is carried out. Under normal operating conditions, the coupling hub is expected to transmit 75 MW at the rated speed of 3000 RPM. The interference fit between the motor and turbine to transmit the torque is analyzed by estimating the contact pressure between the coupling and shaft using 2D FEM analysis. The analysis were performed for: (i) Stand-still condition, (ii) Operating at the rated speed with short circuit conditions, equivalent to 4.4 times the nominal power transmission and (iii) Operating at rated torque but higher (121% of the rated) speed conditions. The analysis for each cases is performed by having two different type of meshing (i) uniform meshing and (ii) fine meshing at the contact zone and (i) contact pressure, (ii) Tangential stress distribution and (iii) Von-mises stress is estimated. The validation of FEM method was performed by comparing the theoretical and estimated reduction in contact pressure. The maximum value of power developed for the present case is estimated.

Keywords: FEM, Interference Fit, Contact pressure

1. Introduction

In earlier time the majority of the components like bearings (Hirani, 2009, Hirani *et al*, 2000, Hirani *et al*, 1999, Hirani *et al*, 1998, Muzakkir *et al*, 2011, Hirani, 2005, Hirani *et al*, 2001, Muzakkir *et al*, 2013, Hirani 2004, Muzakkir *et al*, 2015, Hirani, Verma, 2009, Hirani, Suh, 2005, Hirani *et al*, 2001, Rao *et al*, 2000, Hirani *et al*, 2000, Hirani *et al*, 2002, Burla *et al*, 2004, Hirani, Samanta, 2007, Lijesh, Hirani, 2015, Lijesh, Hirani, 2014, Shankar *et al*, 2006, Lijesh, Hirani, 2015, Muzakkir *et al*, 2014, Lijesh, Hirani, 2015), brakes (Sarkar and Hirani, 2015, Sarkar and Hirani, 2015, Sarkar and Hirani, 2013, Sarkar and Hirani, 2013, Sarkar and Hirani, 2013, Sukhwani *et al*, 2009, Sukhwan and Hirani, 2008, Sukhwani *et al*, 2008, Sukhwani *et al*, 2008, Hirani and Manjunath, 2007, Sukhwani *et al*, 2007), gears (Shah and Hirani, 2014, Hirani, 2009), seal (Hirani and Goilkar, 2011, Goilkar and Hirani, 2010, Hirani and Goilkar, 2009, Goilkar and Hirani, 2009, Goilkar and Hirani, 2009), coupling etc. are fitted to the shaft considering some fit (i.e. interference, clearance, transition, etc.). The dimensions of these mating components cannot be manufactured to the exact specifications and often defined with lower specification and upper specification limits. The difference between lower and upper specification limits is inversely proportional to the manufacturing cost, which means manufacturing

dimension of a component/shaft to the exact specification (zero deviation from the mean value) demands very high manufacturing cost. In interesting to note that the functional aspects of the assembly of two components are also affected by the magnitude of fit between those components. In the present work detail, a study of interference fit between coupling and shaft in a steam turbine (as shown in figure 1) is performed.

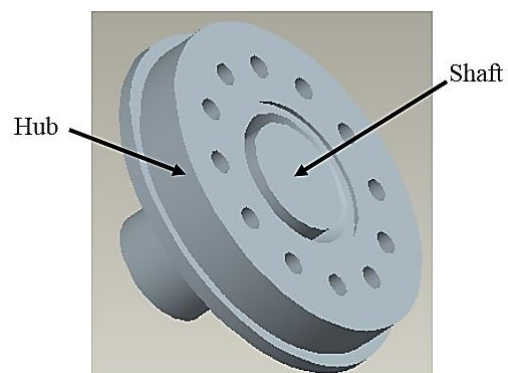


Fig. 1 shaft and coupling hub

Under normal operating conditions, the coupling hub is expected to transmit 75 MW at the rated speed of 3000 RPM. The interference fit is used to transmit the torque between the motor and turbine. If the fit between the sleeve and coupling hub is lesser than a certain limit, then slip between shaft-hub occurs. If interference fit is

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too much then excessive stresses develop resulting failure of the coupling. In the present work, contact pressure and stresses between the shaft and the coupling hub for the following three cases have been evaluated and the results are presented.

- Stand-still condition.
- Operating at the rated speed with short circuit conditions, equivalent to 4.4 times the nominal power transmission.
- Operating at rated torque but higher (121% of the rated) speed.

For estimating contact stress and pressure in the coupling for the above mentioned three conditions, 2D FEM analysis is performed using ANSYS software. The importance of meshing in contact is addressed. The comparison of contact stress and pressure values for the considered three case is performed and results are presented.

2. Finite Element Analysis

The 2D finite element analysis was performed for the 3D part shown in figure 1 and the dimension of the coupling hub considered for analysis is shown in figure 2. In present work contact analysis was performed for to estimate the contact pressure, tangential stress distribution case and Von Mises stress distribution of different three cases using ANSYS. In the present case the coefficient of friction is considered as 0.12 and contact surface offset as 0.4mm. The material proper considered for the coupling hub is: Young’s Modulus = $2.1 \times 10^5 \text{ N/mm}^2$, Poisson’s ratio= 0.29, Density= $7.85 \times 10^{-6} \text{ Kg/ m}^3$. The element type used for the contact analysis is plane 82.

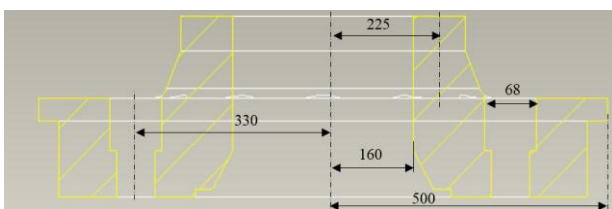
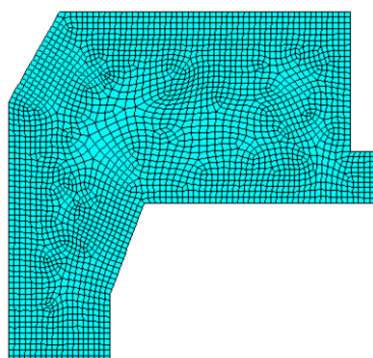
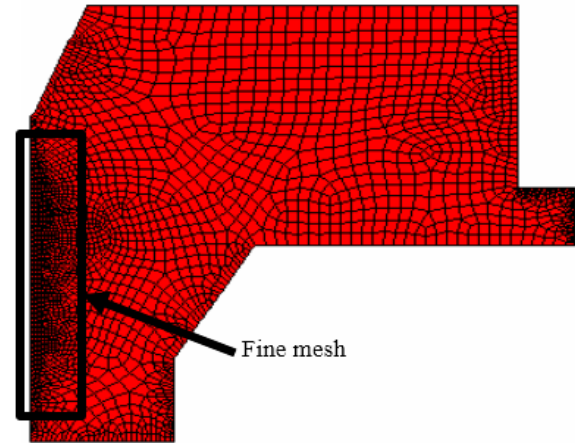


Fig.2 Schematic figure of coupling



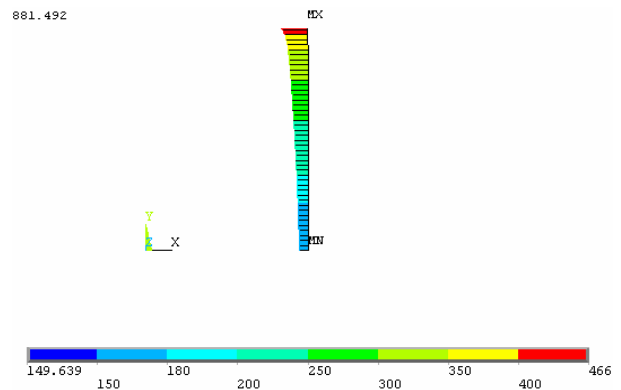
(a) Full model



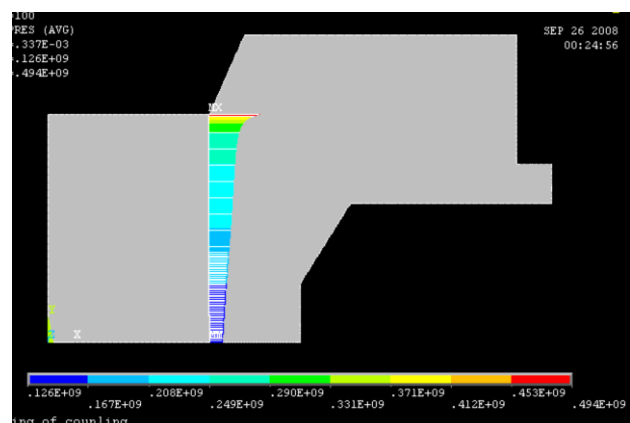
(b) Quarter model

Fig. 2 Assembly of shaft-coupling in ANSYS

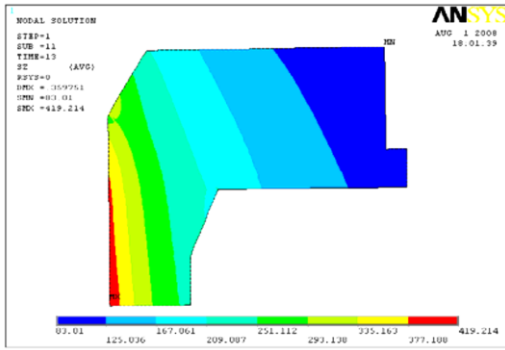
Two types of meshes were generated in ANSYS, having same number of as shown in Figure 2. In (i) first case (figure 2(a)), uniform meshing is done and (ii) second case (figure 2(b)) fine meshing is provided in the contact zone. The number of elements and nodes in the first case is 2528 and 12404 and in the second case is 2556 and 8061. The result obtained after performing analysis in the three different cases is shown in figure 3, 4 and 5 respectively.



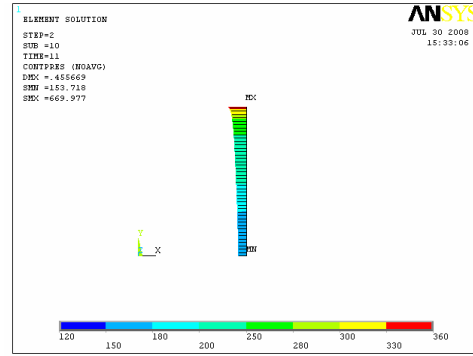
(a) Contact Pressure case (1)



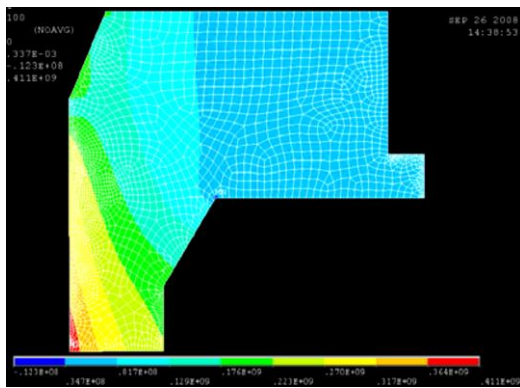
(b) Contact Pressure case (2)



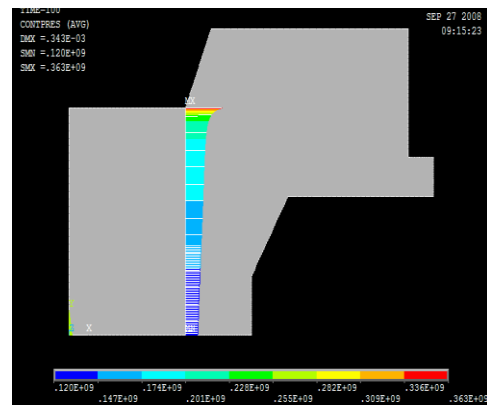
(c) Tangential stress distribution case (1)



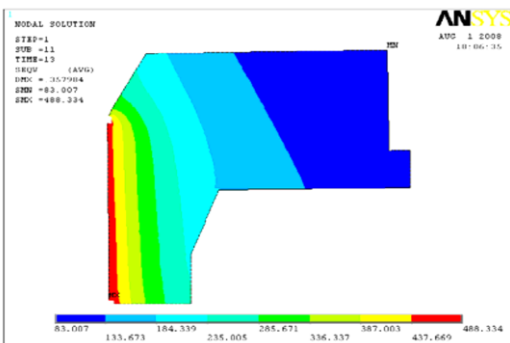
(a) Contact Pressure case (1)



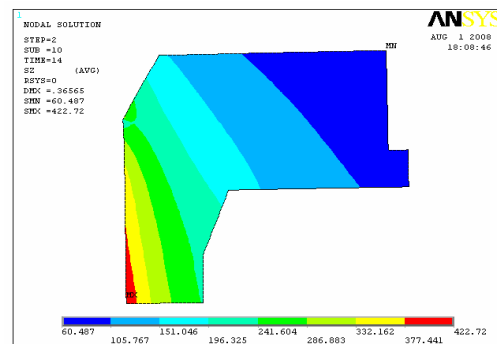
(d) Tangential stress distribution case (2)



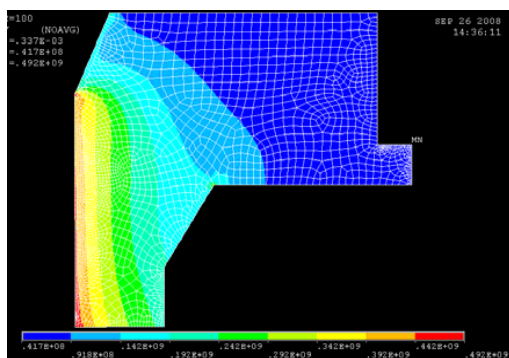
(b) Contact Pressure case (2)



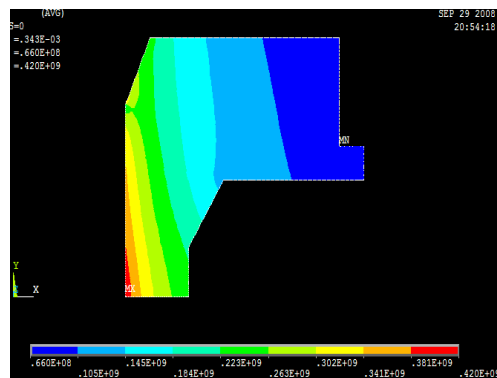
(e) Von Mises stress distribution case (1)



(c) Tangential stress distribution case (1)

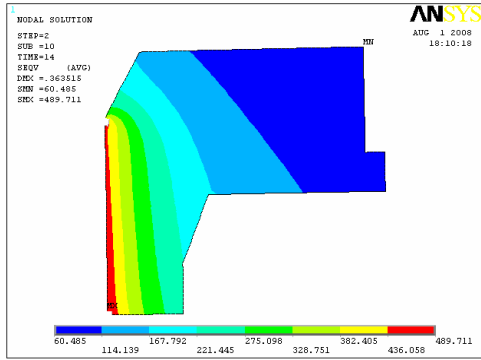


(e) Von Mises stress distribution case (2)

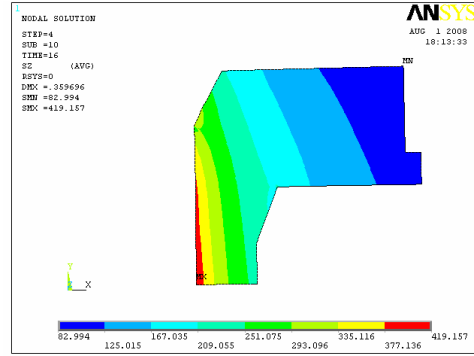


(d) Tangential stress distribution case (2)

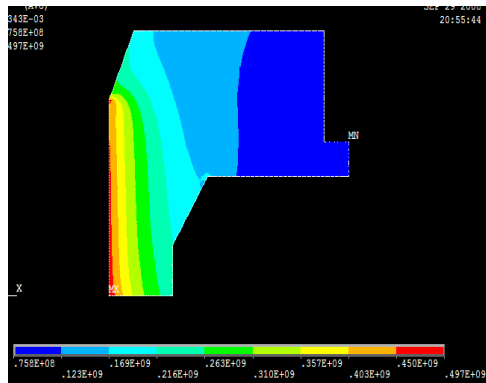
Fig.3 Results obtained for Stand-still condition



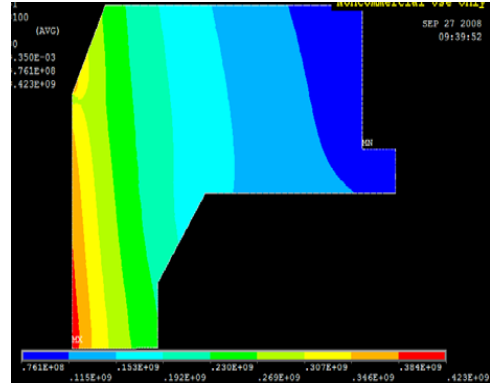
(e) Von Mises stress distribution case (1)



(c) Tangential stress distribution case (1)

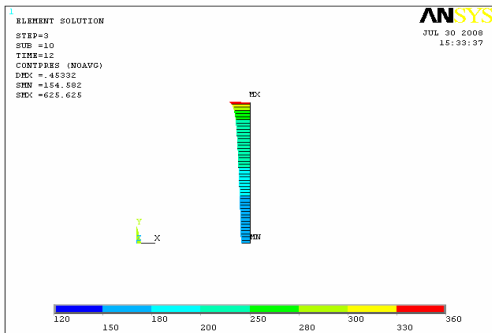


(f) Von Mises stress distribution case (2)

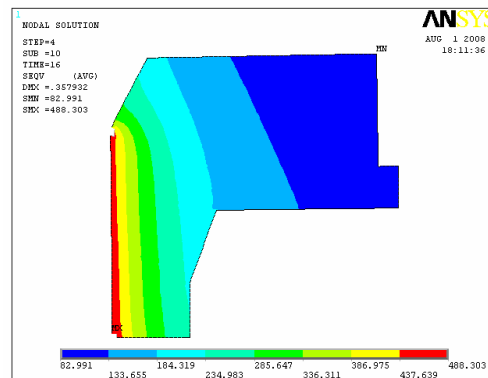


(d) Tangential stress distribution case (2)

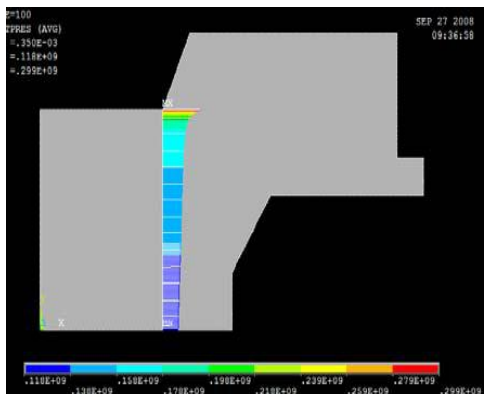
Fig.4 Results obtained at rated speed of 3000 rpm



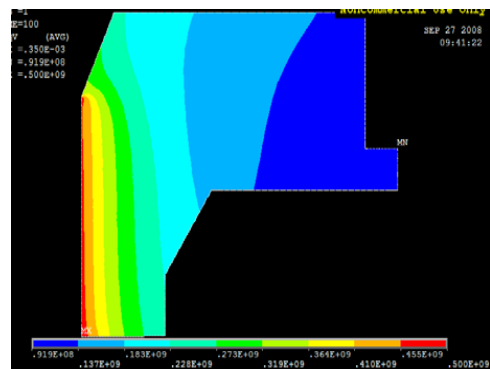
(a) Contact Pressure case (1)



(e) Von Mises stress distribution case (1)



(b) Contact Pressure case (2)



(f) Von Mises stress distribution case (2)

Fig.5 Results obtained at centrifuging speed i.e.121% of rated speed

From figure (3) it can be concluded that the values estimated for contact pressure and circumferential and Von Mises stresses in case (2) is more than the case (1). The maximum contact pressure obtained by case (1) is 466MPa while case (2) is 492MPa. The maximum tangential stresses are 419MPa and 411MPa, respectively. Similarly Von Mises stress maximums are 488MPa and 492MPa, respectively. For all stress distributions, the location of the maximum stresses are identical. Hence by providing the more number of nodes increase the values of all parameters. The length of contact between shaft and the coupling hub is 0.23m. From figure 4, the estimated maximum contact pressures are 360MPa and 363MPa, respectively. The maximum tangential stresses are 422Gpa and 420MPa, respectively and the maximum Von Mises stresses are 489MPa and 497MPa, respectively.

The peak contact pressure for the (i) Stand-still condition is 494MPa, (ii) for rated speed of 3000 rpm it is 363MPa and (iii) at centrifuging speed it is 299MPa. So reduction in contact pressures at 3000 rpm is 131 MPa and 3630 rpm (121% of rated speed) is 195 MPa which is a ratio of 1.488. The theoretical estimated reduction with increase in speed by 121%, is $(1.21)^2$, that is, by a factor of around 1.46. The values indicate the results obtained from FEM is correct.

The length of contact between shaft and the coupling hub is 0.23m. The torque transferred is estimated by using equation (1) and the estimated value is 0.757MNm. The RPM is 3000, which is equal to 314.159 rad/s. Thus, the maximum power that can be transmitted is 237.57 MW. The maximum power that can be transmitted is only 238MW, that is, just 3.17 times the rated power.

$$T = 2\pi R^2 \mu \int_0^{0.23} p(y) dy \quad (1)$$

Conclusion

To analysis the interference between the coupling hub and shaft of an steam turbine, 2D Finite Element Analysis was performed in the ANSYS software. The analysis was performed for: (i) Stand-still condition, (ii) Operating at the rated speed with short circuit conditions, equivalent to 4.4 times the nominal power transmission and (iii) Operating at rated torque but higher (121% of the rated) speed. The analysis for each cases is performed by having two different type of meshing (i) uniform meshing and (ii) fine meshing at the contact zone. From the analysis result it was concluded that the fine meshing at the contact zone provide realistic results and uniform meshing will result in more number of nodes and computational time also. The estimated reduction in contact pressure using 2D FEM and analytical approaching having fine meshes at the contact zone were matching and providing the validation of the FEM work. The estimated maximum power that can be transmitted was only 238MW, which is 3.17 times the rated power.

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