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Design of Shrink Fit for Low Temperature Rotating Turbine Components

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***Abstract* - Key aspect to designing any turbine component is to ensure its mechanical integrity. Steam turbine rotor mechanical integrity has traditionally been and still is a concern for Original Equipment Manufacturers (OEMs) across the globe due to their catastrophic nature of failure [1]. Any joint in rotating components always present a great risk for integrity of rotating machinery. One such joint is shrink-fit in rotating components. Many steam turbine rotors of earlier designs especially impulse or low reaction bladed rotors had shrink fit discs fitted onto them. Though shrink fit joints are less employed these days, they are still relevant.**

This paper discusses the shrink fit design philosophy for rotating components operating at room temperature. Design philosophy includes identification of loads, safety factor, evaluation of stresses and interference etc. Turbine bladed discs, gear wheel etc. are not regular cylindrical shapes, hence employability of classical formulations have limitations. Additional analysis using Finite Element Method is required to substantiate the design. The methodology is validated in design of one rotor having such shrunk on components.

Keywords: Shrink fit, interference, turbine, rotation, ANSYS

I. INTRODUCTION

Shrink fit is a semi-permanent joint between the shaft and hub. Outer element of the shrink fit joint is hereby referred to as “Hub” and inner element of the shrink fit joint is referred to as “Shaft”, please refer to *Fig. 1*.

Shrink fit is achieved by assembling a cylindrical hub onto a shaft (shaft outer diameter is higher than the inner diameter of hub) either by pressing it in place (or) using differential thermal expansion techniques [2]. An interfacial pressure is generated due to the local elastic deformation due to interference. The state of stress induced between interfacing cylinders is similar to that of an externally (or) internally pressurized pressure vessel and is governed by Lame’s equation.

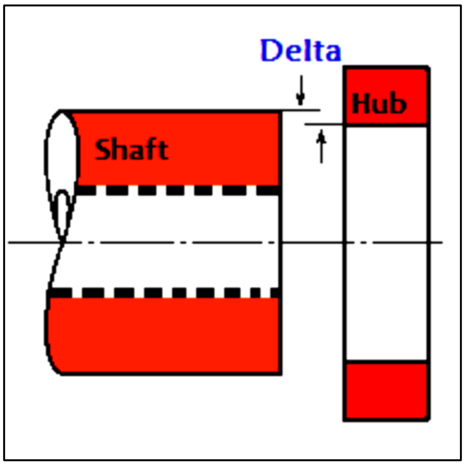

Fig. 1, Schematic of Interference Fit Joint

The case of shrink fit in rotating discs with large diameters is slightly different. These discs with large diameter are subjected to high centrifugal force, resulting in reduction in interference between shaft and hub diameters at the interface due to centrifugal action. Stresses induced due to rotation are also a cause of concern along with interfacial stresses due to interference. Hence, additional care from the designer is required for deciding the interference tolerance, please refer to *Fig. 2*.

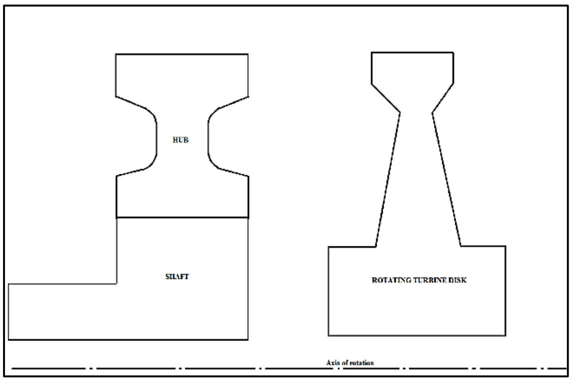

Fig. 2, Rotating turbine wheel and hub of varying geometric shapes

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II. DESIGN CONSIDERATIONS FOR SHRINK FIT JOINT

Designing a shrink fit joint for rotating discs primarily involves following aspects:

- Torque transmission requirement and differential growth between shaft and hub due to centrifugal load. This determines the "*Minimum Interference Condition*"
- Stress in the joint at assembly condition as well as operating condition due to centrifugal loads would decide the "*Maximum Interference Condition*"
- Shaft/Hub material being used

A designer would then integrate these requirements and define an interference tolerance. Shaft and hub materials can differ or be the same depending upon requirement. This needs to be taken care appropriately in the classical formulations.

Differential growth due to thermal loads are not significant in low temperature turbine rotors. However, these are relevant for high temperature turbine rotors. But, they affect the design of shrink-fit joint only in case material of shaft and hub are different.

Deformations induced due to centrifugal loading is function of square of rotational speed, hence joint can loosen up in case of inadequate interference resulting in adverse consequences.

III. DESIGN METHODOLOGY

As discussed earlier, the joint has two limiting (or) bounding conditions:

- "*Minimum Interference Condition*": a joint should not open up under the operating conditions due to lack of interference
- "*Maximum Interference Condition*": Material of Hub/Shaft should be able to withstand the interference stress during assembly. During operation, the joint should be able to withstand stress due to rotation and pressure stresses due to remaining interference

A. Minimum Interference Condition

This condition ensures that there is no slippage in the joint. A loose joint may result in slippage of hub onto shaft and loose its intended purpose. Minimum interference condition would overcome differential radial growth between shaft and hub due to centrifugal loading and slip due to torsional loading during operation. Following methodology is to be followed to address this concern: -

1) Interference to overcome torque requirement:

Friction force (F_T) generated in the joint to overcome operating condition torque, is given as;

$$T = R \times F_T \qquad (1)$$

Where,

T = Torque to be transmitted by the joint during operation
R = Nominal radius at interface
F_T = Force

Interference required to overcome this torsional loading [3] would be given by,

$$\delta_T = P_T * \left[\frac{R}{E_S} * (1-\vartheta_s) + \frac{R}{E_H} * \left(\frac{R_o^2+R^2}{R_o^2-R^2} + \vartheta_H \right) \right] \qquad (2)$$

Where,

δ_T = Minimum interference required for overcoming torsional load
P_T = Interference pressure corresponding to F_T
E_s = Young's Modulus of Shaft
R = Nominal radius at interface
ϑ_S = Poisson ratio of shaft
E_H = Young's Modulus of Hub
R_0 = Outer radius of Hub
ϑ_H = Poisson ratio of Hub

2) Radial growth due to rotation:

For a hollow disc at any radii, radial growth due to centrifugal loads [4] is given by the relation below:

$$\Delta = \left[\left(\frac{\rho\omega^2 r}{E} * \frac{(3+\vartheta)*(1-\vartheta)}{8} \right) * \left(r_o^2 + r_i^2 + \frac{(1+v) r_o^2 r_i^2}{(1-\vartheta) r^2} - \frac{(1+\vartheta)}{(3+\vartheta)} r^2 \right) \right] \qquad (3)$$

Δ = Radial growth of rotating disc (shaft/hub)
ρ = Density of rotating disc (shaft/hub)
ω = Angular Velocity
r = At any radius where growth needs to be estimated
r_0 = Outer Radius of rotating disc (shaft/hub)
r_i = Inner Radius of rotating disc (shaft/hub)
ϑ = Poisson ratio of rotating disc (shaft/hub)
E = Young's modulus of rotating disc (shaft/hub)

Using the above formula, the radial growth of inner element - shaft "Δ_S" and outer element - Hub "Δ_H" are calculated at maximum expected over speed condition at the nominal radius.

The classical formulation for radial growth of shaft (or) hub is based on rotating disc formulation. It is important to note that the radial growth of solid and hollow shafts would be different. However, it is observed that the difference is practically negligible and hence, these formulations can be safely used.

3) Minimum Interference:

"Minimum interference condition" is estimated using Δ_s, δ_T and Δ_H are determined as follows:

$$\delta_{minimum} = [(\Delta_H - \Delta_S) + \delta_T] \qquad (4)$$

The above interference must be achieved at the assembly condition to avoid loosening of joint in operation under combined action of centrifugal and torsional loads. An appropriate safety factor may be employed to increase the value, if needed.

After determination of minimum interference as per above, designer should include manufacturing tolerance upon the nominal shaft/hub diameter. The maximum interference should be evaluated with the help of specified tolerances.

B. "Maximum Interference Condition"

Maximum interference is important from strength point of view and hence, may affect selection of material. The Hub is highly stressed component of the joint compared to shaft. It can fail due to:

a) Stress beyond limits during assembly due to interference pressure
b) Stress beyond limits during operation due to remaining interfacial pressure and rotational loads

Maximum principal stresses have been considered for mechanical strength analysis in this paper. However, it depends upon the designer to choose the appropriate failure criterion.

If the above stresses are beyond the material strength limits, then material selection needs to be relooked or the shrink fit parameters need to be redesigned if possible to bring stress within allowable limits.

1) Stress due to interfacial pressure in hub during assembly/operation:

Hub is a hollow thick cylinder subjected to internal pressure. Hoop stress at the hub inner diameter due to assembly condition/operating condition interfacial pressure is calculated using the equation below [3]:

$$\text{Hoop stress at hub inner diameter} = P * \frac{r_i^2}{r_0^2 - r_i^2} * \left(1 + \frac{r_0^2}{r_i^2}\right) \tag{5}$$

Where,

P = Interfacial pressure at the time of assembly to achieve $\delta_{minimum}$ or remaining interfacial pressure at operating condition
r_0 = Outer radius of Hub
r_i = Inner radius of Hub

2) Stress due to rotation in hub during operation

Hoop stress due to rotation [4] at hub inner radius is given by

$$\frac{\rho_H \omega^2}{4} * \left[(3+\vartheta_H) r_0^2 + (1-\vartheta_H) r_i^2\right] \tag{6}$$

Where,
ρ_H = Density of Hub material
ϑ_H = Poisson ratio of Hub
ω = Angular velocity of Hub
r_0 = Outer Radius of Hub
r_i = Inner Radius of Hub

Stresses at assembly and operating condition should be within material limits with appropriate factor of safety.

IV. FINITE ELEMENT ANALYSIS AND ITS IMPORTANCE

As discussed before, classical formulations have their own set of limitations. Hence, finite element analysis of shrink fit assembly would give designer additional information about the joint behavior and its integrity. Here, we discuss finite element analysis using ANSYS™ software to further validate the integrity of shrink fit design. Outcome of the finite element analysis depends on:

- Quality of mesh (including refinement)
- Interface definition using contact settings [5]. (*Selection of contact and target component (shaft or hub) is designer's prerogative. However, in present work, shaft is used as contact and hub is taken as target surface*)

Surface to Surface contact pair definition was used to define the interference between shrink fit components, please refer to *Fig. 3*

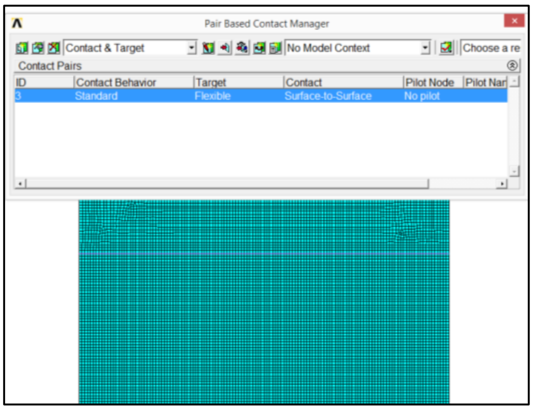

Fig. 3, Surface to Surface Contact Definition

The amount of physical interference between the hub and the shaft is defined through contact surface offset as in *Fig. 4* below:

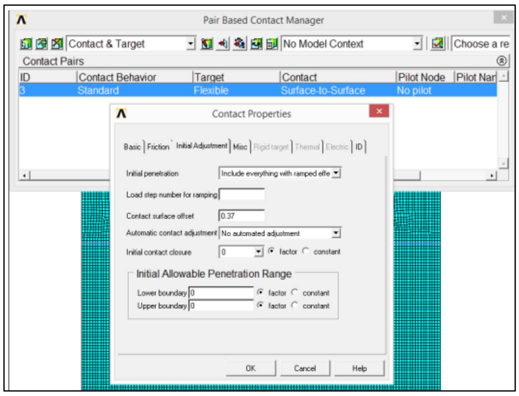

Fig. 4, Interference definition

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After defining the contact assembly, the analysis is performed in two stages. In the first stage, the interference due to minimum interference condition is simulated. In the second stage, centrifugal loading in the form of global angular velocity is applied along with appropriate boundary condition.

Based on requirement different over speed conditions are simulated to check for exact speed at which the joint disintegrates i.e. loosen up. Parameters that are used to check for joint integrity include "*contact status*" and "*contact pressure*"

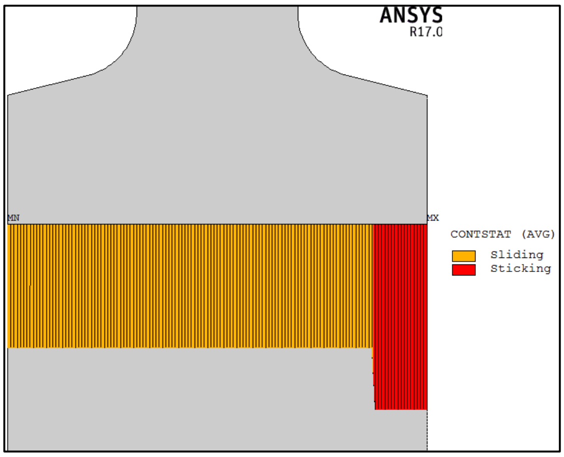

Fig. 5, Contact Status of the shrink fit joint

Contact status from the finite element software is just a qualitative indicator of the condition of the joint. However the contact pressure is a quantifiable term, please refer *Fig. 5*

The contact pressure should match with Interference pressure (please refer equation 2) during assembly condition and the remaining interference pressure at operating condition within acceptable margin of error *Fig. 6*.

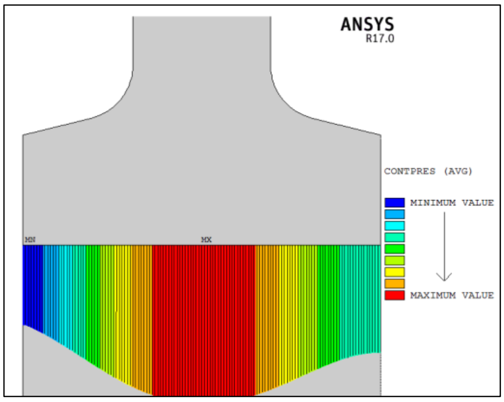

Fig. 6, Contact Pressure at running condition

End effects (or) localized stress raisers can also be observed from the finite element model unlike the classical formulation in relevant cases.

V. APPLICATION OF DESIGN PROCESS IN A RECENT PROJECT

For a recent project, a shrink fit joint had to be designed for a low pressure steam turbine rotor in which a starting motor gear had to be shrink fitted. Through the above established process this recent project was handled.

Fig. 7, Gear shrunk on over a turbine coupling

Post design, the rotor was also over speed tested successfully at factory works, please refer *Fig. 7*

VI. CONCLUSION:

Methodology for design of shrink fit joints for rotating components has been described. Calculations using classical formulations and validation using finite element software (ANSYS^{TM}) has been shared. A case of successful implementation of the procedure to design a shrink fit joint for a low pressure steam turbine rotor is also presented.

VII. REFERENCES:

[1] Nitta A, Kobayashi H , "Burst of Steam Turbine Rotor in Fossil Power plant"; http://www.sozogaku.com/fkd/en/cfen/CB1031027.html
[2] Jack A. Collins, Henry Busby, George Staab; "Mechanical design of machine elements and machines – A failure prevention perspective"; Wiley Publication; 2010
[3] J Edward Pope; "Rules of Thumb for Mechanical Engineers"; Gulf Publishing Company; 1997
[4] EJ Hearn; "Mechanics of Materials 2"; Butterworth Heinemann publication; 2001
[5] ANSYS^{TM} 17.0; Help manual