

# **Some Preliminary Studies on the Contact Stress Distributions in a Compressor Disk Dovetail Type Joints using FEM Approach**

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**ABSTRACT:** This paper presents results of preliminary FEM investigations carried out on 2D equivalent models representing typical blade/disk contact regions in aeroengine fan/compressor designs. The pressure load on the blade represents the centrifugal force due to rotating blade-mass about engine axis at rated RPM. The flank angle was varied from 0° to 45° in steps of 15°. The commercial software ANSYS was used in these investigations. The contact region was modelled using CONTACT12 elements with appropriate settings. The obtained results bring out the effect of flank angle on the stresses around the contact region and corroborate well with earlier published results.

**KEYWORDS:** Compressor Disk, Dovetail Joint, Flank Angle, Contact Stresses, ANSYS FEM Software

## **I. INTRODUCTION**

The stress analysis of single tooth attachments or 'dovetails' used to secure fan and compressor blades to disks in gas turbines has received the attention of several experimental and numerical investigations over the last few decades [1 to 4]. The major emphasis of these investigations however has been on the stress values on the unloaded boundaries of the component with particular attention to the fillet radius at the junction of the dovetail and the radial shank of the blade [1]. Consideration of the failures occurring in practice suggests that the fillet region is not always the site for fatigue cracks and that the region below and adjacent to the contacting flanks of the joint is just as likely to provide a site for crack initiation. This paper presents the details of some preliminary FEM investigations conducted to study the stress distributions in and around the contact region.

The dovetail fixing is usually achieved by means of a uniform axial slot in the compressor disc. It was therefore considered that a two-dimensional model of blade-disc joint would be more appropriate for carrying out the analysis to understand the nature of internal stresses in and around the contact region. The aerodynamic and thermal loads on compressor blades are much smaller in comparison to the radial loads due to rotation. Hence the radial loads accounting for the centrifugal force due to rotating blade-mass about engine axis is only considered in these studies.

## **II. GEOMETRY AND DIMENSIONS OF THE MODEL**

Figure 1 depicts the geometry and dimensions of a typical blade-disc configuration considered in the present investigation which corroborates with the one considered by the first author in his earlier publications [5, 6]. The blade flank has a width of 75 mm and height of 160 mm. The holding frame has appropriate matching dimensions and represents the dovetail groove geometry on the periphery of fan/compressor disc. It has horizontal displacement constraints on its left and right vertical edges. As already mentioned the flank angle was varied from 0° to 45° in steps of 15°. The equivalent pressure traction,  $p$  on the blade is calculated as 48.0 MPa considering the central force/blade.

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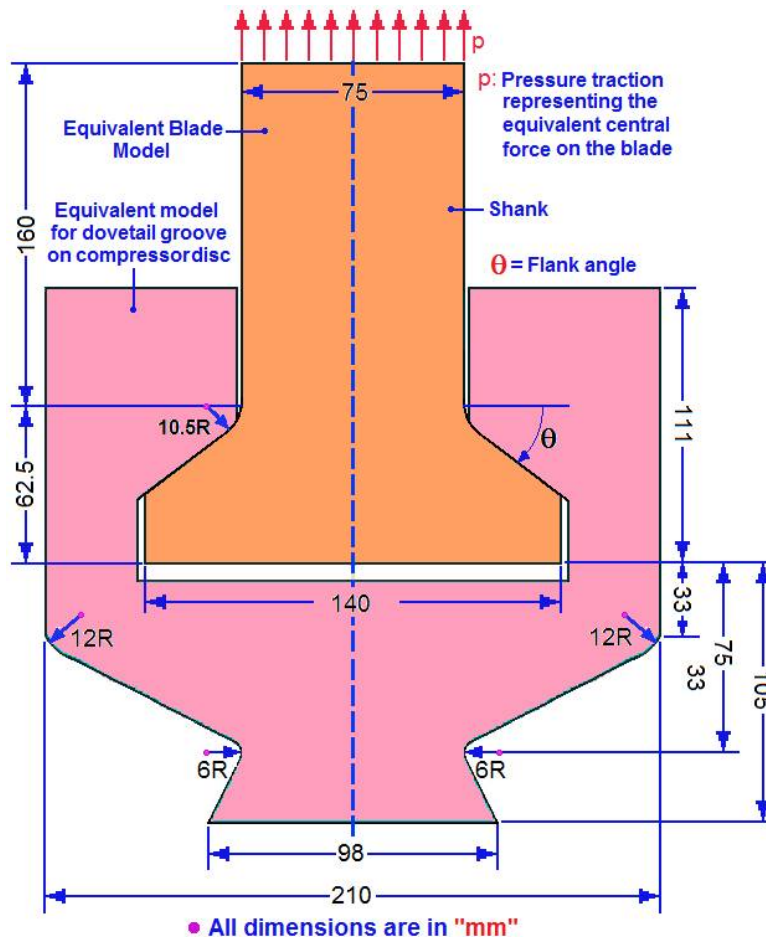


Figure 1: Geometry and dimensions of the equivalent blade/disc model (flank angle = 45°)

### III. FINITE ELEMENT ANALYSIS

#### FEM MODEL

The FEM model was generated using PLANE42 elements. The contact region was modelled using CONTACT12 elements (grid-to-grid-contact) with appropriate settings. The mesh density was arrived at through a systematic convergence study. Figure 2 shows a typical FEM mesh for 45° flank angle model. The FEM mesh consists of a total of 3428 elements including 11 contact elements. The total number of nodes is 3164 with total DOFs as 6328.

#### MATERIAL PROPERTIES

The material used in the design of compressor blade and disc is the titanium alloy Ti64. Table 1 presents the material properties of Ti64 at maximum operating temperature conditions.

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**Table 1: Material properties of Ti64**

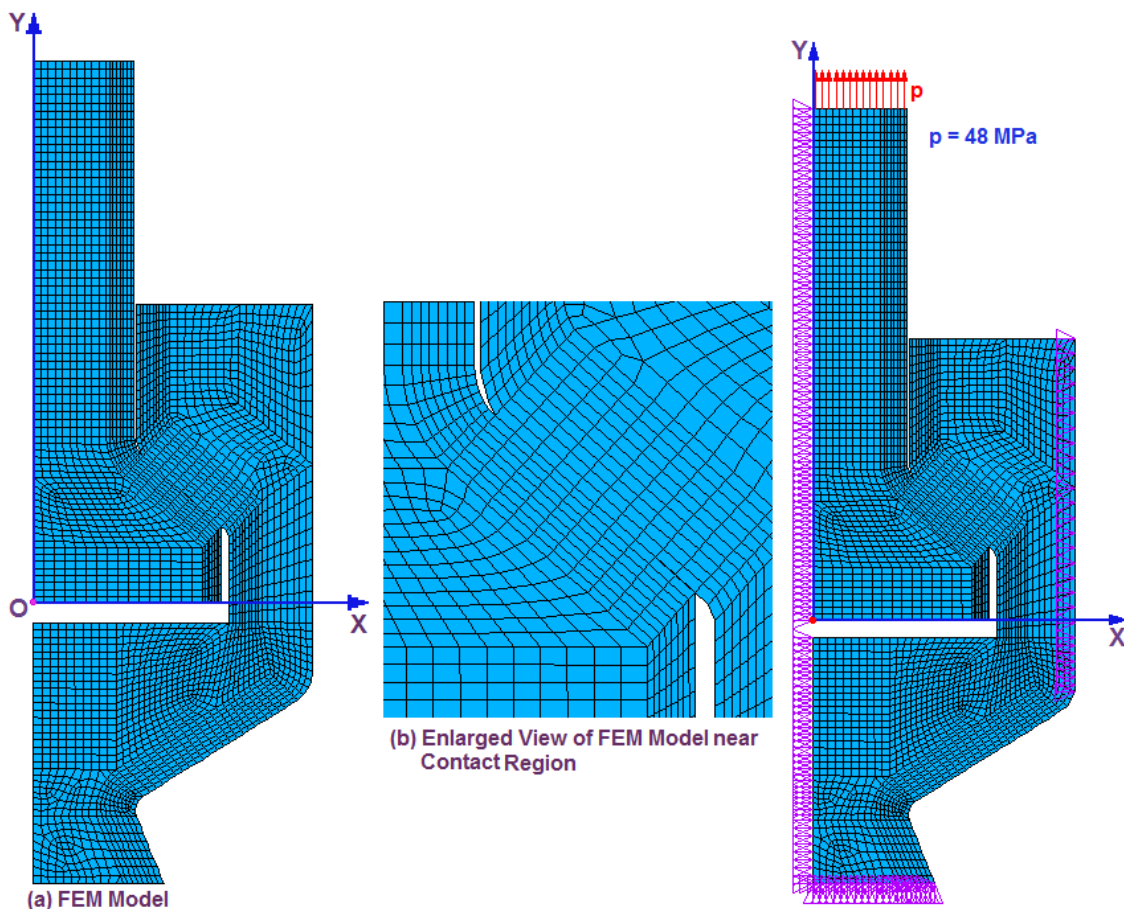
Material (Titanium Alloy)	Elastic Modulus(E) GPa/MPa	Poisson ratio ( $\nu$ )	Density ( $\rho$ ) kg/m <sup>3</sup>	Yield Stress ( $\sigma_y$ ) MPa
(IMI 318) [Ti-6Al-4V]	106/106000	0.33	4420	925

## BOUNDARY CONDITIONS

The problem has a symmetric boundary condition (BC) with respect to y-axis as axis. To take advantage of symmetric BC the right-half portion of the model is only considered for analysis. Figure 3 represents the FEM model with symmetric BC applied. The bottom nodes of the holding frame have a fixed BC ( $U_x = U_y = 0$ ). The nodes on the right vertical edge of the holding frame are suppressed from undergoing displacement in horizontal direction ( $U_x = 0$ ). The applied traction on the blade is  $p = 48.0$  MPa. Along the line of symmetry (y-axis)  $U_x = 0$ .

## FEM ANALYSIS

The problem was analysed using the front end commercial software ANSYS [7] using its static stress analysis capabilities with appropriate settings for the nonlinear features of CONTACT12 element.



**Figure 2: FEM Mesh for the blade/disc model (flank angle = 45°)**

**Figure 3: Loading and Boundary conditions (flank angle = 45°)**

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## IV. RESULTS AND DISCUSSION

Typical results of the investigation are presented in Figures 4 to 9. The resultant displacement (USUM) for 0° flank angle model is shown in Fig. 4. It is seen that the peak value of resultant displacement gradually increases with increase in flank angle. For 45° flank angle model the peak value of USUM is 0.3059 mm.

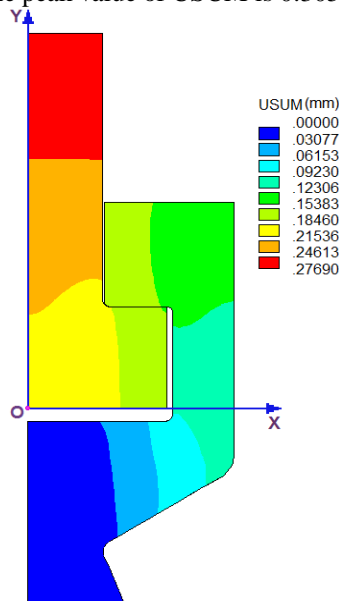


Figure 4: Resultant Displacement (USUM) Distribution (flank angle = 0°)

In Fig. 5 is shown a typical von Mises stress distribution for 15° model with a close-up view near contact region having peak value of stress (228.17 MPa) occurring in the fillet region for the blade model.

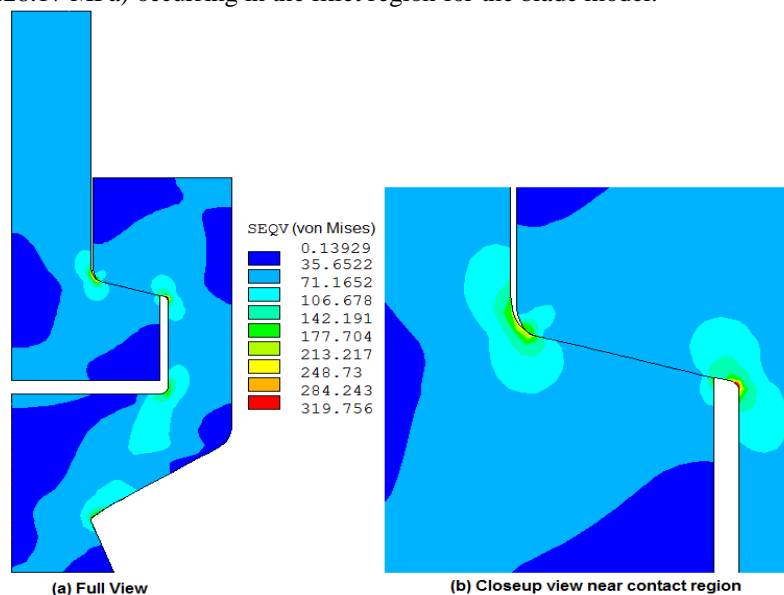
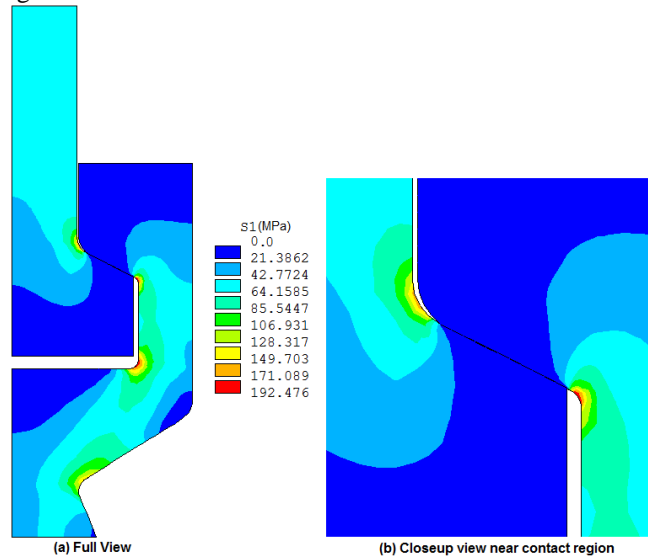


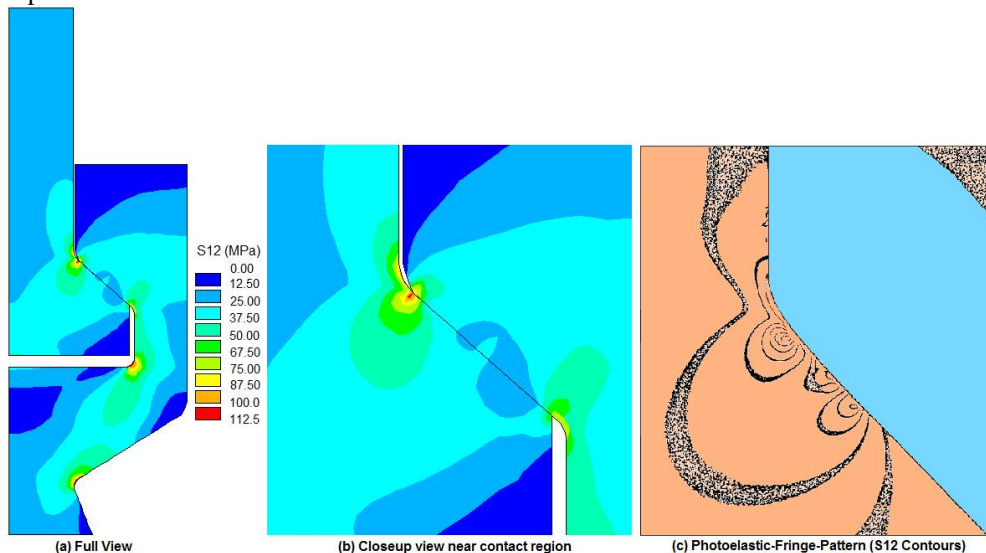
Figure 5: von Mises Stress (SEQV) Distribution (flank angle = 15°)

Figure 6 shows a typical maximum principal stress distribution for 30° flank angle model, the maximum stress (165.24 MPa) occurring in the fillet region.



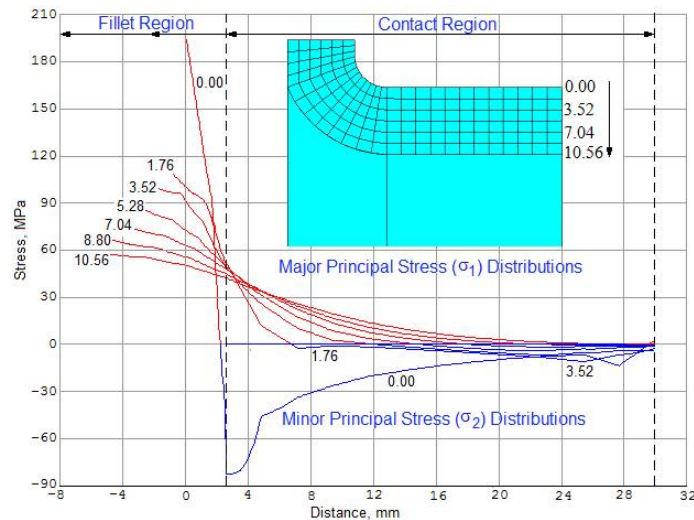
**Figure 6: Maximum Principal Stress (S1) Distribution (flank angle = 30°)**

In Fig. 7 is shown a typical maximum shear stress ( $S12 = \tau_{12}$ ) distribution for 45° flank angle model, the maximum stress (=178.08 MPa) occurring at the edge of the contact region towards fillet side. An experimental (Photoelastic) fringe pattern for 45° flank angle model with slightly different loading condition is shown in Fig. 7(c) which has a close qualitative comparison with FEM maximum shear stress contours.



**Figure 7: Max Shear Stress (S12 =  $\tau_{12}$ ) Distribution (flank angle = 45°)**

A detailed comparison and analysis of stresses around the contact zones and fillet regions was carried out considering all the four cases. It was found that the stresses for the 0° flank angle model were more critical from the point of fatigue loading. Figure 8 represents the major and minor principal stress ( $\sigma_1, \sigma_2$ ) distributions at different depths for the 0° flank angle model. One can note the steep stress gradients at the edge of contact and towards the fillet region.



**Figure 8: Principal stress distributions ( $\sigma_1$ ,  $\sigma_2$ ) at different depths in the contact region (flank angle =  $0^\circ$ )**

#### V. CONCLUSION

1. The results obtained bring out the effect of flank angle on the stresses around the contact region and corroborate well with earlier published photoelastic experimental results [5,6].
2. The steep principal stress gradients existing immediately below the surface at the end of the contact zone in the area adjacent to the fillet could give rise to severe fatigue conditions. Further, any fluctuation in the contact area conditions (due to possible blade vibrations during engine operation) would give rise to a significant change in stress magnitude and even a sign reversal, leading to fatigue of blade material at this location.
3. The load distribution along the flank and fillet regions did not appreciably affect the maximum principal stress distributions below the contact surface.
4. For all four cases studied, the maximum principal stress occurs on the fillet boundary. The stress at this location however is slightly affected by a change of contact conditions and so it is likely that fatigue is not a problem in this locality.
5. The tensile principal stresses tend to increase with depth from the contact surface; however they tend to increase with depth towards the fillet side. The variation of principal stress magnitude (and also direction) was greatest as the flank surface was approached.
6. The maximum principal stress was found to be in the fillet root. However, it is unlikely that fatigue conditions would be critical for the fillet region.

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