Contact pressure distribution in joints formed by V-band clamps

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Abstract. V-band clamps offer an efficient clamping solution in diverse applications including process equipment, exhaust systems and air handling. This paper studies the distribution of interface contact pressure between the V-band and flange when the coupling is established. The determination of the contact area and pressure distribution in a joint is essential information, as it determines the integrity of the coupling. A three dimensional finite element model has been developed for this purpose. Contrary to the previous assumption in developing axisymmetric models, the 3D results showed that the contact pressure is non-uniform around the circumference of V-band with maximum contact pressure near the T-bolt area. This is in agreement with the theory in the literature. The presence and magnitude of friction has a noticeable influence on the form of the interface pressure distribution curve. It is also shown that the diameter of the band interacts with the effect of friction.

Introduction

V-band clamps are used in connecting circular flanges in a wide range of applications mostly in the automotive and aerospace industries. Weight and volume reduction and also ease of assembly are the major factors which make the V-band clamp a better choice over traditional bolted joints. In addition, the orientation of the connected components can be easily and infinitely adjusted according to design and application requirements. For these reasons they have been popularly adopted in the field of turbochargers [1] to clamp the major housings together.

The V-band coupling working mechanism is shown in figure 1. The clamp band has a flat bottomed, V-shaped cross section. As torque is applied to the T-bolt nut, a radial force is created in the V-band (inward direction). Due to the angle of the V-section, the radial load created by band tension is then transmitted as an axial load on the matting flanges.

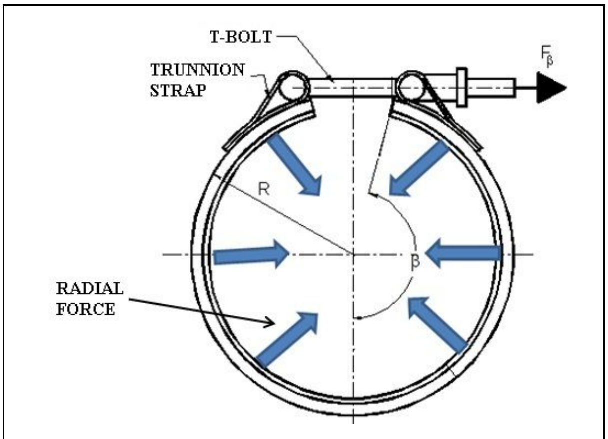

Figure 1. Working mechanism of V-band

The behaviour of such clamps has been studied and theoretical and computational models are proposed in the literature to address this type of coupling. Marmon clamps used in the aerospace industry are similar but comprise relatively stiff, discrete V-section blocks held in place around the flange pair with a flat band. Detail analysis of this application is reported in [2]. This includes axial stiffness, damping characteristic and also investigating the effect of preload, wedge angle, friction coefficient and number of V-segment on the behaviour of the joint. Another application where rigid

V-insert clamps are used on relatively flexible flanges are automobile exhaust pipes. Testing the sealing performance of such joint under different working conditions is reported in [3].

Flexible V-band clamps were first studied in [4] based on empirical data with the results of a series of pressure tests extrapolated out to other load cases. In [5,6] models were developed to study stresses in these clamps during assembly and working conditions. The results from this work suggests a variation in working stresses due to uncontrolled operating parameters and manufacturing tolerances associated with this type of band. Also, in [7] a theoretical model was developed to study contact pressure distribution around the V-band circumference.

The failure mechanism of this type of V-bands under axial loading was investigated in [8] and [9] with the aid of an axisymmetric finite element model. The effect of on the axial stiffness of such a joint of different structural parameters such as friction, band size and wedge angle together with operational parameter such as T-bolt load were studied.

Although the previously proposed models in the literature provide valuable information on the behaviour V-band couplings, the complexity of such systems suggest that axisymmetric FEA models and classical models may be over simplifying the problem. Within the current study a 3D model of V-band coupling is used allowing previously neglected effects to be investigated.

Finite element model

The 3D finite element model was developed within the Abaqus software package (Dassault systems). As shown in figure 2 the model consisted of half the V-band cross section and one flange. The V-band was constrained by a symmetry boundary condition at the plane of symmetry, to prevent it from moving in the axial direction. The V-band was modelled as a solid body, using linear elements with reduced integration as suggested in [10] and as recommended in [11] for similar cases of contact analyses. In the applications being studied here, the flange has a much more substantial geometry than the band and was therefore treated as a rigid body. This approximation was also made in [5] when developing a theoretical model for flat and V-section band clamps. The use of an analytical rigid body that did not require meshing reduced computing time, and the complexity of the contact analysis, enabling it to converge faster. Displacement in the axial direction and rotation about the circumferential direction was constrained at the rigid body reference point (RP) to keep the flange in position. The V-band is made of AISI 304 stainless steel with Young's Modulus of 227 GPa and Poisson's Ratio of 0.29.

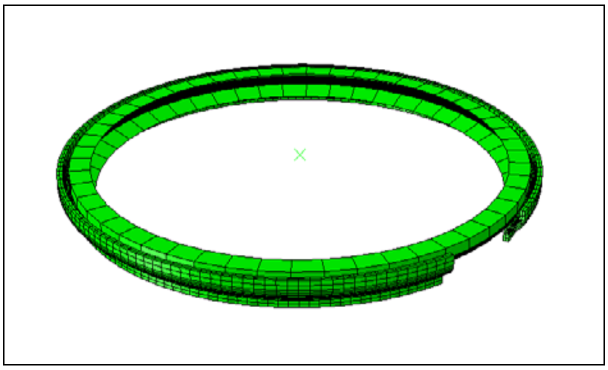

Figure 2 3D finite element model

In the present study the contact pressure distribution is determined following the application of T-bolt load which results in tightening of the V-band. This process is simulated in two different ways for 2D axisymmetric and 3D finite element models. In 2D axisymmetric models, as discussed in [9], tightening is enforced by generating an artificial thermal strain in the V-band to shrink it onto a flange. For the 3D model, the tightening process was modelled by the direct application of a circumferential force applied as a traction on the cross section surface at both ends of the V-band.

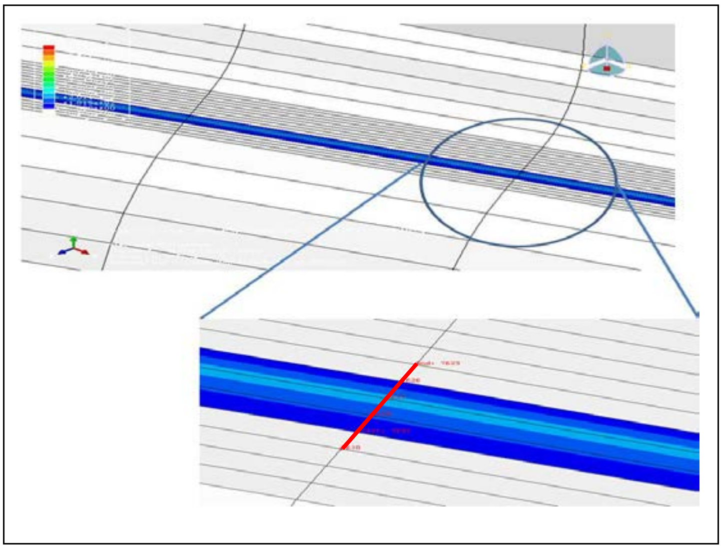

Figure 3. Contact area (in colour) on V-band

The aim of this study was to quantitatively evaluate the contact between the V-band and flange. In the theory [6] this contact is assumed to be on a circumferential line acting normal to the band surface. In the FE model, this same contact is simulated more precisely as a contact pressure on a surface. This is shown in figure 3 where the high density of the FE mesh used to capture the pressure distribution can also be seen. In order to obtain data from the FE model which was comparable to the theory, it was necessary to integrate the contact pressure along a line at each angular position (cross section) sampled around the circumference of the band. An example of such an integration line is highlighted in the detailed view of the contact area shown in Fig. 3.

The load per unit length was calculated as described above at different angular positions (0 to 150 degrees) where 0 degrees is the position furthest from the T-bolt and 150 degree is closest to the T-bolt. The FE analysis and associated post processing was repeated for coefficients of friction between 0 and 0.3, band radii of 57mm, 157mm and 257mm and T-bolt loads from 0 to 15kN.

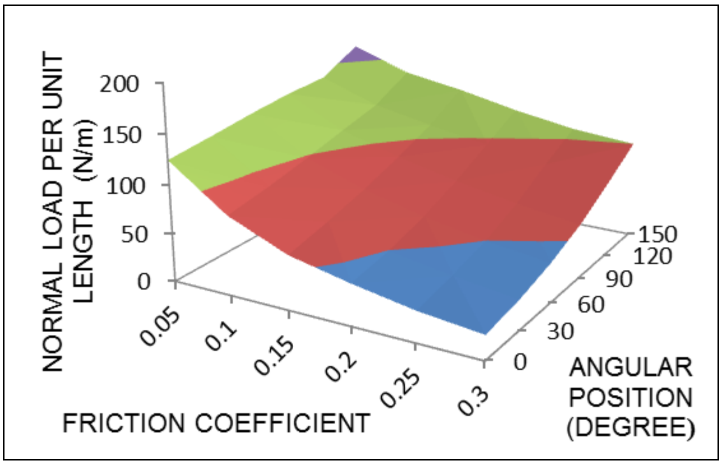

Figure 4. Contact pressure distribution (band radius 57mm, T-bolt load 6kN)

Fig. 4 shows the contact pressure distribution around the band for V-band radius of 57mm and T-bolt load of 6kN with variable friction coefficient.

The results in Fig. 4 show a non uniform distribution of contact pressure over the circumference of V-band with the maximum values near the T-bolt. This non uniform pattern is more distinct when the coefficient of friction is increased with the contact pressure dropping markedly away from T-bolt. It can also be seen that with an increase in coefficient of friction for a fixed T-bolt load the contact pressure all around the band is reduced. Hence, the overall clamping force in the coupling system will be reduced. Repeating this study with higher T-bolt loads showed a similar pattern but with higher contact pressures.

Comparison of finite and theoretical models

Shoghi et al [7] presented a simple equation to determine the contact force per unit length, q, at a particular angular position around the band, Ɵ, from the force applied by the T-bolt, F_β, subtended half angle of the V-section, β, half angle of V-band, Ø, and flange radius, R_2. Incorporated into this equation was the coefficient of friction μ acting between the contacting surfaces with the effects of friction being accounted for in both the circumferential and transverse directions. However, a previously presented version of this equation [5] neglected the transverse friction effect. To allow the circumferential and transverse effects to be investigated separately, the equation from [7] was modified to separate the transverse coefficient of friction, μ_T, and circumferential coefficient of friction, μ_c:

$$q = \frac{F_\beta exp\left[\frac{-\mu_C(\beta-\theta)}{(\mu_T cos\emptyset + sin\emptyset)}\right]}{2R_2(\mu_T cos\emptyset + sin\emptyset)} \quad (1)$$

This modification of Equation1 allowed the impact of including transverse friction to be readily investigated. The results from FEA and Theory (with and without transverse friction) are compared for a V-band size of radius 257mm with coefficient of friction 0.05 and 0.25 are shown in Fig. 5. It can be seen here that for the lower coefficient of friction, the impact of including transverse friction is negligible and both theories agree well with the FEA results. However, for the higher coefficient of friction, the transverse friction effect is significant and for this large band diameter, the FEA results give a good match to the theory neglecting transverse friction.

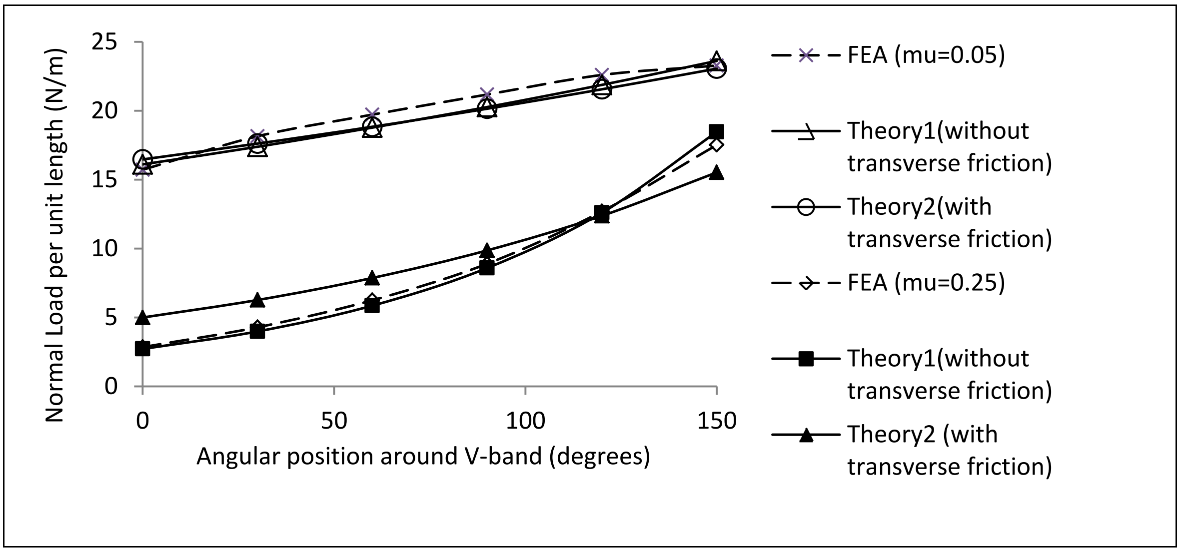

Figure 5. FEA and Theory results for V-band size of 257mm (radius) and coefficient of friction 0.05 and 0.25

Fig. 6 shows the results for a small band with a radius of 57 mm but an identical T-bolt load applied. For the lower coefficient of friction, there is again a good match between the FEA and both theories. However, there is more distinction between the two theories with the inclusion of transverse friction giving more variation in contact pressure around the band. With the higher coefficient of friction, in contrast to the larger band, the FEA model gives a better fit to the transverse friction theory.

A further significant difference between the two bands is that the contact pressure for the larger band is significantly lower than for the smaller band. This difference suggests that in the larger band the circumferential force is sufficient to overcome friction in the circumferential direction. Hence, friction provides no resistance in the transverse direction. However, the much larger contact pressure in the smaller band means that the circumferential force is insufficient to overcome friction in the circumferential direction. Hence, the transverse friction effect is still active.

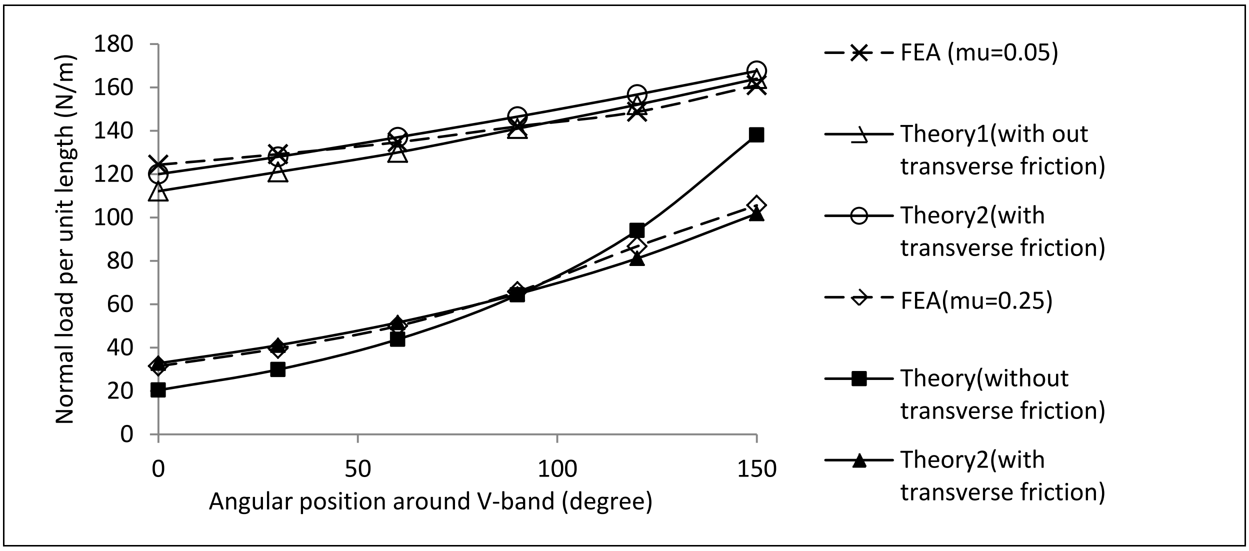

Figure 6. FEA and Theory results for V-band size of 57mm (radius) and coefficient of friction 0.05 and 0.25

Conclusion

There is significant variation in the contact pressure distribution around the V-band joint, particularly when the coefficient of friction is high. This variation can only be captured in a finite element analysis using a 3D model.

The effect of transverse friction will be removed if friction in the circumferential direction is overcome. This effect will be significant for larger bands.

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