

Effects of Brass (Cu_3Zn_2) as High Thermal Expansion Material on Shrink Disc Performance During High Thermal Loading

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Abstract. This research work is focused on shrink disc operation at high temperature. Geometrical and material design selections have been done by taking into consideration the existing shrink disc operating at high temperature condition. The existing shrink disc confronted slip between shaft and shaft sleeve during thermal loading condition. The assessment has been obtained through virtual experiment by using Finite Element Analysis (FEA) -Thermal Transient Stress for 900 seconds with 300°C of thermal loading. This investigation consists of the current and improved version of shrink disc, where identical geometries and material properties were utilized. High Thermal Expansion (HTE) material has been introduced to overcome the current design of the shrink disc. Brass (Cu_3Zn_2) has been selected as the HTE material in the improved shrink disc design due to its high thermal expansion properties. The HTE has shown a significant improvement on the total contact area and contact pressure on the shaft and the shaft sleeve. The improved shrink disc embedded with HTE during thermal loading exhibit a minimum of 1244.1 mm² of the total area on shaft and shaft sleeve which uninfluenced the total contact area at normal condition which is 1254.3 mm². Meanwhile, the total pressure of improved shrink disc had an increment of 108.1 MPa while existing shrink disc total pressure has lost 17.2 MPa during thermal loading.

Keywords: *Shrink disc, thermal expansion, thermal loading, contact pressure, contact area, shaft and shaft sleeve.*

1. Introduction

The interference fit has been used for torque transmitting from one component to other components since a long time ago [1]. It has been used in many industries such as automotive, plantation, medical, oil and gas and army application. It was designed due to drawbacks of other mechanical couplings such as pin, slots and key and screw which have less efficiency during operation compared to interference fit. On the other hand, interference fit has numerous capabilities such as simple structure, high load carrying capacity, excellent concentricity and etc. There are many researchers performing studies in the related field such as Sun et.al on his work, contributed on implementing 3D FEM modelling during shrink fit process of assembled crank shaft. Observation concluded that uniform expansion of crank shaft cannot be achieved by subjecting homogenous temperature distribution



around the cylinder bore [2]. The crankshaft experienced distortion during assembly process which resulted the shrink fit interface undergoing inhomogeneous contact pressure. Meanwhile Bengeri et.al, has focused on the effect of temperature dependent yield stress on contact pressure during thermal assembly of shrink fit by analytical transient stress approach on plastic-elastic model [3]. The authors found that there was no significant effect of contact interface by introducing a temperature dependent and plastic-elastic model in transient stress analysis. Alexandre et.al in their study has widened observation of shrink fit interference pressure which are subjected to centrifugal force and temperature effect. The result revealed that the rotational speed and the existence of steady state thermal loading on the shrink fit has shown significant effect on the reduction of torque capacity [4]. Jianmei et.al investigated the interference stress on shrink fit by including several parameters as influence factors which consist of various range of interference, shaft diameter, wall thickness and mating length that might influence the interference stress on shrink fit. In his study, the various FEA contact analyses have been conducted by utilising orthogonal test approach. This approach used multiple step regression analysis by analysing data obtained from FEA in order to determine relationship toward contact stress. It was discovered that the interference parameter has strong sensitivity towards the interference stress [5]. Another study Strozzi et.al focuses upon determination of the significant contacting shape in a frictionless, keyless, shaft-hub press-fit profile which promotes uniform elastic contact pressure distribution with analytical study. They found that the infinite elastic stress build up at the edge of hub bore, otherwise the stress will reduce as the fillet shape is introduced on the profile. On the other hand, reduction of stress concentration is respected to various change of fillet radius [6]. Cerdán et.al investigated the stress distribution on the interference fit with having circumferential notched on the hub. They found that the stress concentration is higher on the hub notch. By modifying the notch shape, stress analysis was conducted which resulted in a reduction of stress concentration on that region [7].

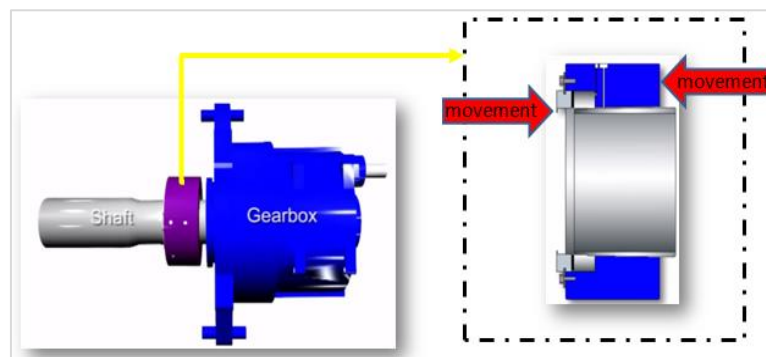


Figure 1 Shrink disc operating condition in pump

Figure 1 shows the current application of shrink disc embedded in pump. The obstacles are normally experienced by the system when operating at high temperature operating condition. The components encompass shaft and shrink disc tend to deform thus losing the grip of the shaft and the shaft sleeve whereby the torque will be unsuccessfully delivered. This study focused on the geometrical concept design and material properties modification of the existing shrink disc in order to enhance the contact area and contact pressure of the shaft and shaft sleeve thus simultaneously eliminating the occurrence of gap of shaft and shaft sleeve during thermal loading condition.

2. Simulation Methods

The simulation was performed by using Ansys Workbench version 17.2. The geometrical model has been designed using CAD Autodesk Inventor. The multilayer interference fit shrink disc consists of shaft, bolt, sleeve, conical ring – ferrule, clamping collar (1) and (2), thermal expansion components and pusher as shown in the **Figure 2**. Shrink disc improved design encompasses material and

geometrical modification to ensure its performance even during high temperature condition. The thermal expansion components are built in shrink disc assembly as one of potential parameters in performance advancement studies. Brass (Cu_3Zn_2) has been selected due to its high thermal expansion properties compared to other materials [8]. Geometrical simulations were conducted with the following parameters, boundaries and mechanical properties setting stipulates in **Figure 3**. The analysis were performed using thermal stress transient analysis to observe the change in contact pressure and contact area as time elapsed [9].

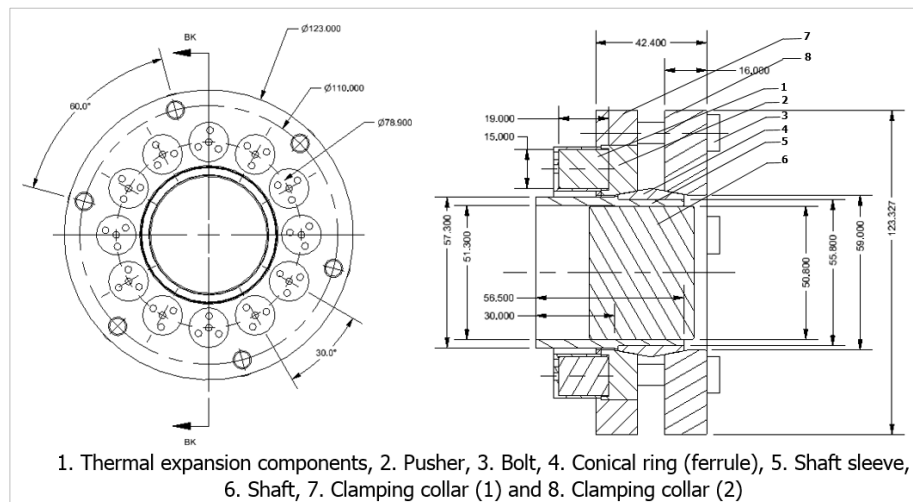


Figure 2 Geometrical Concept Design

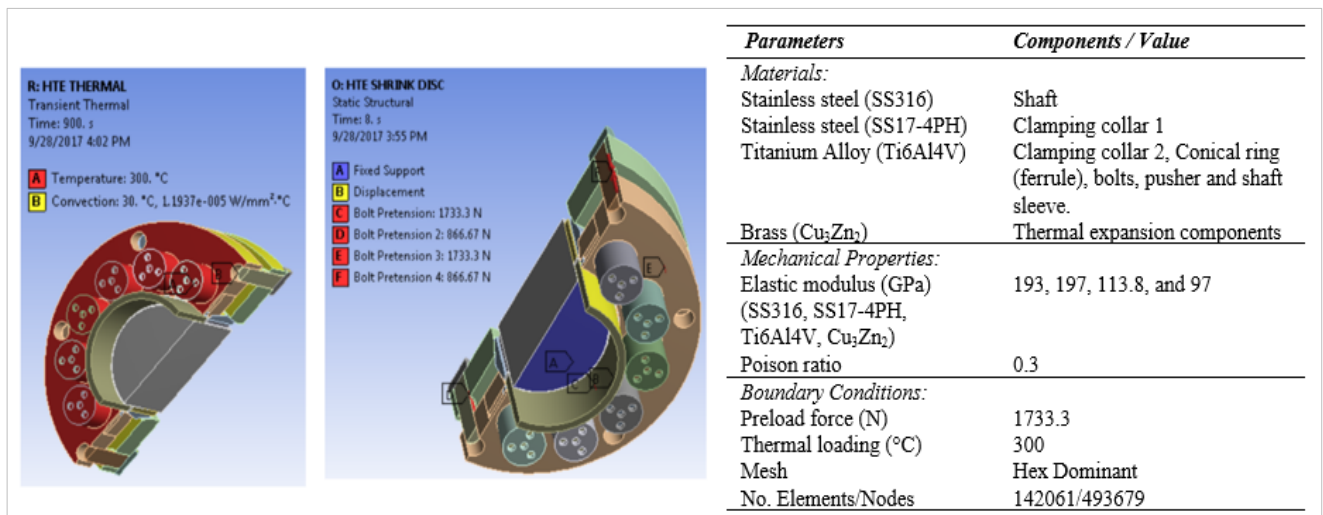


Figure 3 Basic parameters and boundary conditions setting

Total preloading forces were computed by the following equation:

$$T = K \cdot F_{sa} \cdot d \quad \text{Eqn. 1}$$

Where T is the tightening torque of the bolt (N mm); K is the torque coefficient; F_{sa} is the preloading force (N) and d is the nominal size of bolt diameter (mm) [10].

Contact pressure of the shaft sleeve and shaft is calculated by the following equation:

$$P_{min} = \frac{F_{Sa}(1-\mu_2 \tan \beta)}{\pi d_3 l_f (\tan \beta + \mu_2)} \quad \text{Eqn. 2}$$

Where F_{Sa} is the total bolt preload force (N); μ_2 is the friction coefficient of conical ring - ferrule and clamping collar; β is the conical ring - ferrule angle (degree); d_3 is the external diameter of sleeve (mm) and l_f is the length of conical ring – ferrule [11].

3. Results and Discussion

Figure 4, **Figure 5** and **Figure 6** illustrate the results of improved and existing shrink disc design which are further explained in section 3.1 to 3.4 in detail.

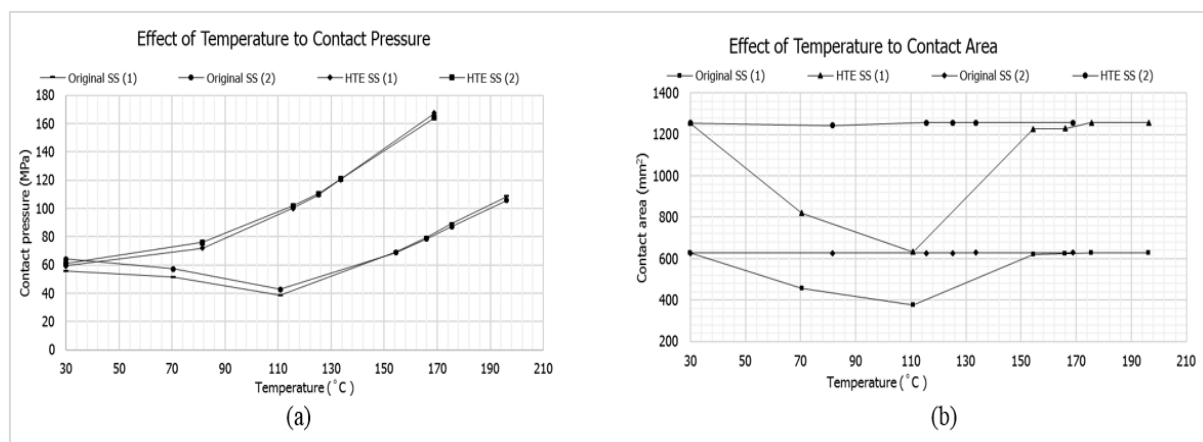


Figure 4(a) Effect of Temperature to Shaft Sleeve Contact Pressure and **4(b)** Effect of Temperature to Shaft Sleeve Contact Area

3.1 Effect of Temperature to Shaft Contact Pressure

The performance of contact area is analyzed by mating parts of shaft and shaft sleeve. This contact pressure is generated as torque is applied to the bolt which acts as an axial preload on the clamping collar. As the clamping collar travelled towards each other, the radial force was produced by conical ring which is located below clamping collar. Shrinkage of conical ring resulted in gripping between shaft and shaft sleeve. In **Figure 4(a)**, increase of contact pressure of 21.5% from 59.29 MPa to 72.04 MPa with High Thermal Expansion Shaft Sleeve 1 (HTE SS1) and 24% of increased to 75.84 MPa from 61.06 MPa are achieved by HTE SS (2) as the temperature of contact pressure between shaft and shaft sleeve elevated to 81.5°C. There was no pressure drop at HTE SS 1 and HTE SS 2 as the temperature continuously increased. On the other side, the existing shrink disc with identical geometrical and material properties resulted in an inverse behaviour, where the contact pressure started to decrease as the temperature increased from 30°C to 70°C. The original shaft sleeve (Ori. SS 1) shows reduction of approximately 7.7 % and 31% of contact pressure at 70°C and 110°C respectively as the temperature continuously increased [12].

3.2 Effect of Temperature to Shaft Contact Area

Contact area of HTE SS 1 in **Figure 4(b)** has shown tremendous improvement with minimal reduction of 0.8% while HTE SS 2 increased to 0.07% during high thermal loading. The Ori. SS 1 and 2 were severely reduced in contact area with approximately 1/2 and 2/3 from the total contact area before the application of thermal loading. The improved design of shrink disc with utilizing high thermal expansion materials have shown to successfully eliminate the occurrence of slip at high thermal loading condition.

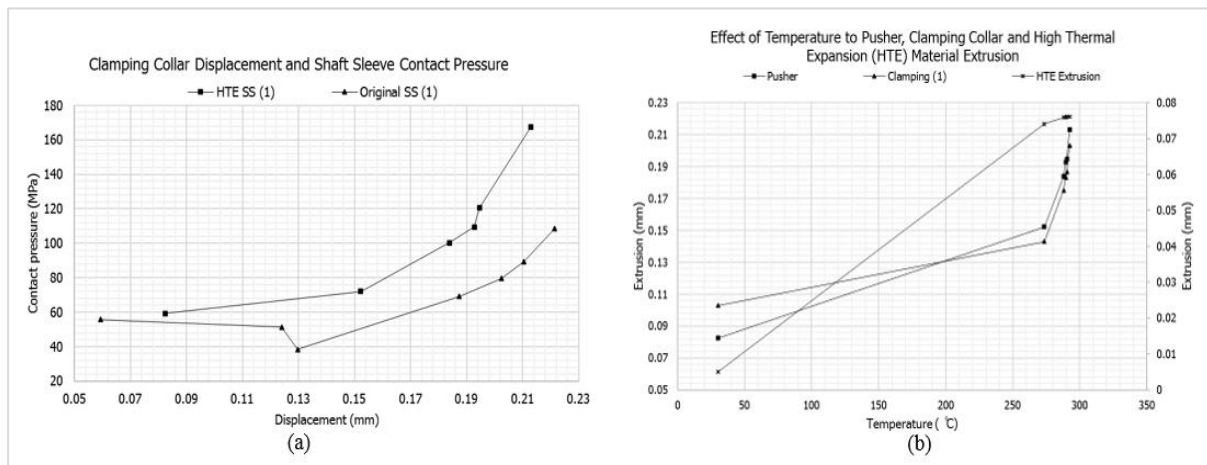


Figure 5(a) Clamping Collar Displacement vs. Shaft Sleeve Contact Pressure and **5(b)** Effect of Temperature to Pusher, Clamping Collar and HTE Material Extrusion

3.3 Clamping Collar Displacement and Shaft Sleeve Contact Pressure

Figure 5(a) shows clamping collar displacement of HTE SS (1) increased as the contact pressure increased and vice versa obtained by the original SS (1) where decline of 13 MPa has been obtained at 0.13 mm of clamping collar displacement, thus this would lead to slip of shaft and shaft sleeve.

3.4 Effect of Temperature to Pusher, Clamping Collar and HTE Material Extrusion

The initial extrusion of clamping collar, pusher and HTE component are due to bolt pretension at normal condition, 30°C were 0.10 mm, 0.082 mm and 0.005 mm respectively. These components were consistently extruded due to thermal load applied. HTE expanded approximately 48% which is 0.069 mm from the normal condition, while pusher and clamping collar also showed the similar behaviour with 84%, 0.07 mm and 39%, 0.04 mm. This condition has successfully overcome the existing model drawbacks where slip of shaft and shaft sleeve occurred during thermal loading. It has been illustrated that with the High Thermal Expansion (HTE) design, the pusher managed to extrude to provide acting force on the shaft sleeve, whereby this was not achieved by the traditional shrink disc, thus this leads to the occurrence of slip between the shaft and the shaft sleeve.

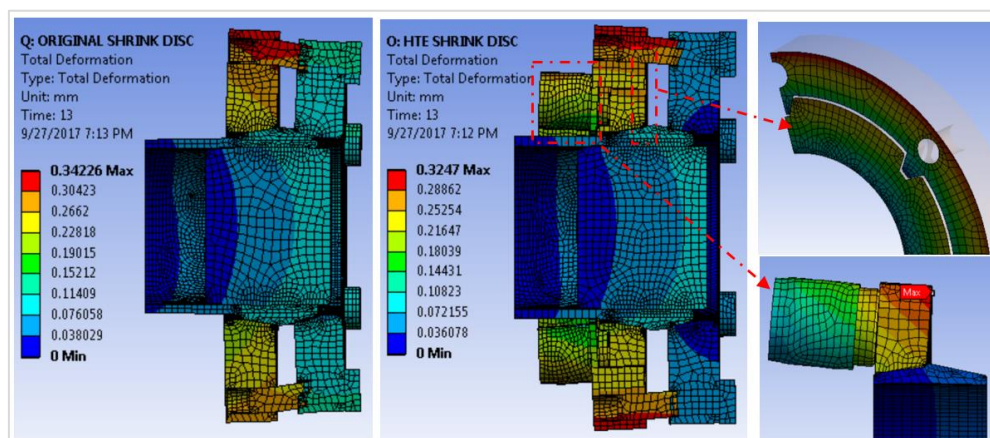


Figure 6 Simulation Results of Original and High Thermal Expansion Shrink Disc

Figure 6 further illustrates the extrusion condition encountered by the pusher, HTE and clamping collar at high temperature. The components experienced deformation and expansion during thermal loading. HTE components expanded and produced axial force which pressed the pusher forward

simultaneously pushing the conical ring (ferrule) to shaft sleeve. This phenomenon has improved the contact pressure and area of shaft and shaft sleeve which are able to maintain the performance and deliverance of torque even at high thermal loading condition.

4. Conclusion

Compared to the existing shrink disc, the HTE concept model has improved the problem confronted by the existing shrink disc. The performance advancement on the contact pressure and contact area achieved by HTE model overcome the occurrence problems of slippage during thermal loading. Instead of producing gap the HTE holds the shaft and shaft sleeve tightly with contact pressure up to 167 MPa and 164 MPa at SS 1 and SS 2 respectively. The contact area also improved significantly with total contact area holds between shaft and shaft sleeve are at 629 mm² at shaft sleeve 1 and 1258 mm² at shaft sleeve 2. The reduction of 10.2 mm² at 81.5 °C deduced total contact area of 1244.1 mm², which is considered unaffected since the full contact area of shaft and shaft sleeve is 1254.3 mm² at normal condition.

5. Acknowledgement

The author gratefully acknowledges ProEight Engineering and also Universiti Teknologi Malaysia for their technical advices and facilities.

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