Design and Vibrational Analysis of Flexible Coupling (Pin-type)

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ABSTRACT

Torsional vibration is an oscillatory angular twisting motion in the rotating members of a system. It can be deemed quite dangerous in that it cannot be detected as easily as other forms of vibration, and hence, subsequent failures that it leads to are often Abrupt and may cause direct breakage of the shafts of the drive train. The need for sufficient analysis during the design stage of a rotating machine is, thus, well justified in order to avoid expensive modifications during later stages of the manufacturing process.The pin-type coupling discussed in this thesis transmits torque through steel pins which are fitted with barrel shaped become more and more flattened there by cushioning the load. It also permits torsional elasticity, angular elasticity and the deviation between the axis, displacement of the axis with respect to each other as well as longitudinal changes of the shaft with limits.

Keywords: *Design, Solidworks, Flexible coupling, Rubber and brass bushed, single bush.*

I. INTRODUCTION

Bush pin type flange coupling is used to connect of shafts which having a small parallel misalignment, angular misalignment or axial misalignment. This is a modification of the protected type flange coupling which has pins (covered by rubber or leather bushes) and it works with coupling bolts. Generally it is used to assemble electric motors and machines. [1].

In the engines there is a cylindrical flange coupling to union assembled parts. The sensitive piece is a flange to the parameters like moment, torque, etc. Normally the coupling problems treated as a beam theory. As we know in mechanical engineering the coupling is used for connection of two shafts to transmit the power. In gear unit applications the rigid coupling is designed especially for this purpose. [2].

The parameters that effect to the flange and nut-bolts deformation are force and contact stiffness factor. To study effect of parameters like normal stiffness, the pretension force and friction coefficient under external loads the simulations of model bolted joint were carried out, ANSYS 14 software used for this simulation. To obtain accurate results we need a predefined process in this program. In flanged and nut-bolted jointed we can see the force and stress have direct proportional relation. [3].

In this study, the flanged join is modeled and simulated by using Solidwork v. 2016 .the finite element analysis procedure required in Solidwork simulation is presented as a predefined process to obtain accurate results.

For the first procedure the coupling is designed as a solution of the given example which finds the dimensions for the main coupling body and its parts, the simulation is applied on the designed coupling by using Solidwork model designer and solidwork Simulation add-on for better results on the main stress and deformation areas that obtained from main acting forces, as a results from the solution of the problem.

2. METHODOLOGY:

A. CAD-Models

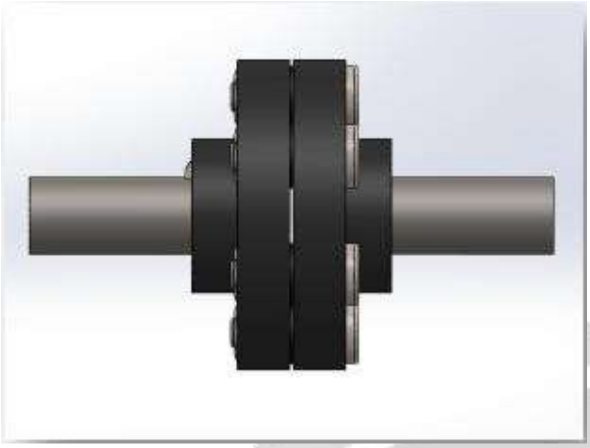


Fig. 1: Front view of CAD model of bushed pin flexible coupling

Fig. 2: Isometric view of CAD model of bushed pin flexible coupling

The solid model of bearing component is created in SOLIDWORKS V. 2016 software.

B. Analytical Design of Bushed pin flexible coupling

To design a bushed-pin type flexible coupling for alloy steel shaft transmitting 40 Kw at 1000 r.p.m. The bearing pressure in the rubber bush and allowable shear stress in the pins are to be 0.45 N/mm^2, 25 Mpa and the Diameter of shaft is 50 mm. [4]. To calculate different Stresses in it we will follow:

Given data,

$P = 40\times 10^3$ W; N = 1000 r.p.m.; d = 50 mm

The torque transmitted by the shaft,

$T = (p\times 60)/(2\pi N) = (40\times 1000\times 60)/(2\pi\times 1000) = 382.16\times 10^3$ N.mm

Considering the shaft in shearing

$$T = \frac{\pi}{16}\times \tau_s \times d^3$$

$$382.16\times 10^3 = \frac{\pi}{16}\times \tau_s \times 50^3$$

$\tau_s = 15.57$ MPa (shear stress induced in shaft)

Design for hub

Outer diameter of the hub (D) = 2d = 100 mm

Length of hub (L) = 1.5 d = 75 mm

Shear stress induced in hub by considering it as hollow shaft.

$$T = \frac{\pi}{16}\times \tau_c \times \frac{D^4 - d^4}{D}$$

$$382.16\times 10^3 = \frac{\pi}{16}\times \tau_c \times \frac{100^4 - 50^4}{100}$$

$\tau_c = 2.0797$ Mpa

Design for key:

Considering the key in crushing

$T = L\times \frac{t}{2}\times \sigma_{ck}\times \frac{d}{2}$

$382.16\times10^{3} = 75\times \frac{10}{2}\times \sigma_{ck}\times \frac{50}{2}$

σ_{ck} = 40.763 MPa

Design for flange

Thickness of flange (t_f) = 0.5d = 25 mm Shear stress in flange

$T = \frac{\pi D^2}{2}\times \tau_c \times t_f$

$382.16 \times 10^3 = \frac{\pi\ 100^2}{2}\times \tau_c \times 25$

τ_c = 0.97 MPa

Design for bolt

d_1 = Nominal diameter of bolts = $\frac{0.5\,d}{\sqrt{n}}$ = 10.2 mm

In order to allow for the bending stress induced due to the compressibility of the rubber bush, the diameter of the pin (d_1) may be taken as 20 mm.

d = diameter of the shaft = 50 mm

Number of pins (n) = 6

The length of the pin of least diameter d_1 = 20 mm is threaded and secured in the right hand coupling half by a standard nut and washer. The enlarged portion of the pin which is in the left hand coupling half is made of 24 mm diameter. On the enlarged portion, a brass bush of thickness 2 mm is pressed. A brass bush carries a rubber bush. Assume the thickness of rubber bush as 6 mm. Overall diameter (d_2) of rubber bush,

$d_2 = 24 + 2\times2 + 2\times6 = 40$ mm

Diameter of the pitch circle of the pins (D_1) = 2 d + d_2 + 2× n = 152 mm

Outside diameter of flange (D_2) = 4d = 4×50 = 200 mm

$W = p_b \times d_2 \times l$, where (l) is length of the bush in the flange

W = 0.45 × 40 × l =18 l N

$T = W\times n\times D_1/2$

$382.16\times10^3 = 18\,l\times6\times152/2$

l = 46.5 mm say 50 mm

W =18l = 900 N

Direct stress due to pure torsion in the coupling halves,

$\tau = \frac{W}{\frac{\pi}{4}(d_1)^2} = \frac{900}{\frac{\pi}{4}(20)^2}$ = 2.86 N/mm^2

Since the induced shear stress in the shaft is less than 25 MPa therefore the design is safe.

3. Meshing of bushed pin flexible coupling

In this work, SOLIDWORK SIMULATION is used for a meshing of bushed pin flexible coupling. It creates sufficient smooth meshing as shown in figures below.

Fig. 3: Meshing of hub

Fig. 4: Meshing of shaft

The flange coupling body is made of Gray Cast iron C.(A48) with maximum shear and tensile strength 50000 N/mm^2 , 151.658N/mm^2 respectively the two part are connected with pin bushed bolts and fixed on the shaft with key .

The shaft used on the test is made of alloy steel type (SS) with max tensile stress of 723.8 N/mm^2 is connected with the coupling by using gib head key.

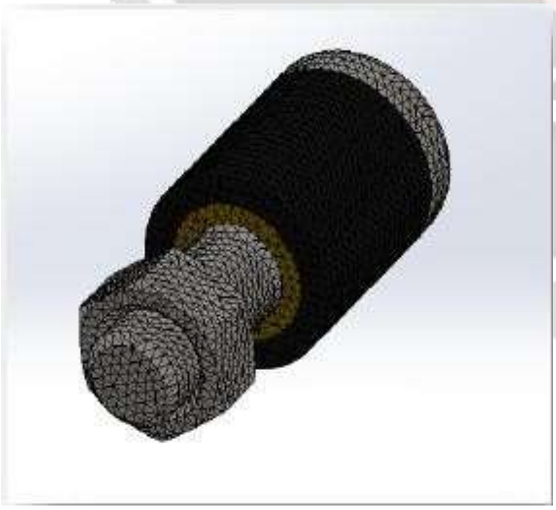


Fig. 5: Meshing of Nut-Pin- Bushed assembly

Fig. 6: Meshing of keys

The assembly of Nut-Pin contains two bushed where made out of Brass and Rubber with thickness of 2 mm and 6 mm consequently which absorbs the main share force acting on the pin neck, and pin itself is made of steel which can withstand a high shear stress.

The used Gib Head Key is rectangular in cross section having a head at the large end. The head makes it easier to remove the key from the hub and shaft. The slot for gib head key must have an open end to permit assembly. For this reason it is placed at the end of a shaft.

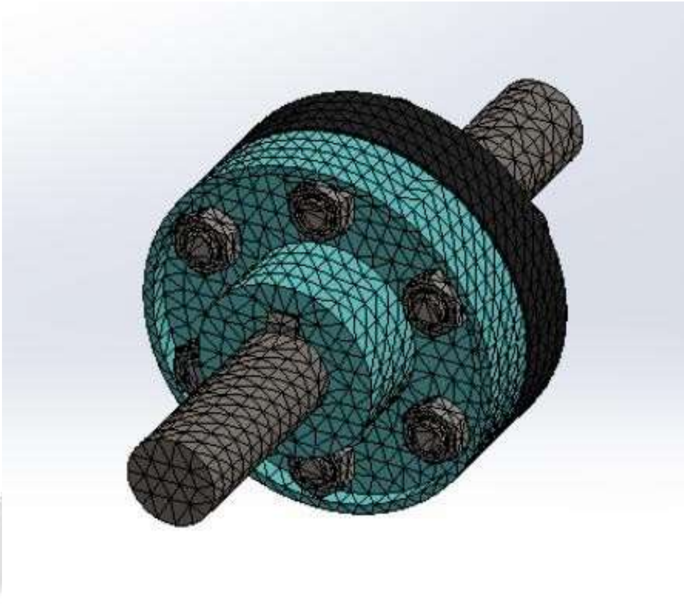


Fig. 7: Meshing of bushed pin flexible coupling

C. Boundary condition

A fix support is used to fix a flange from one end.

Fig. 8: Fixed point

D. Analysis:

Test of bushed pin flexible coupling (Both Brass and Rubber bushed).
Test by applying torque at (T = 382.16 N.m obtained from the calculation).

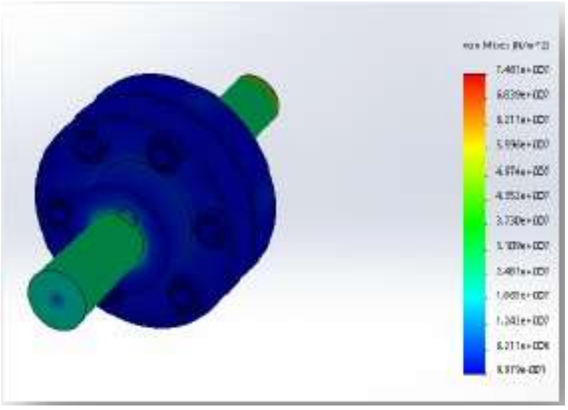


Fig. 9: Equivalent stress of bushed pin flexible coupling

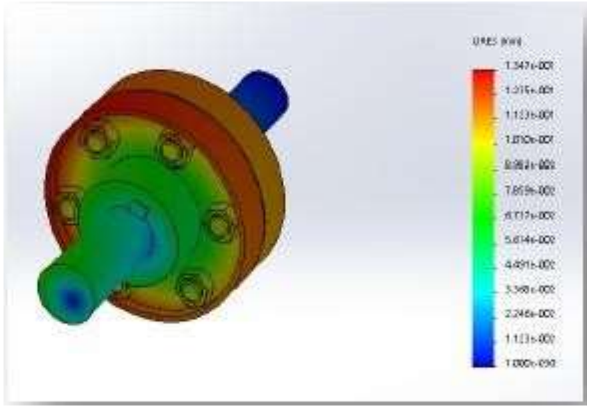


Fig. 10: Total deformation of bushed pin flexible coupling

E. Further Analysis test

Failure test by applying high torque at (T = 1000 N.m)

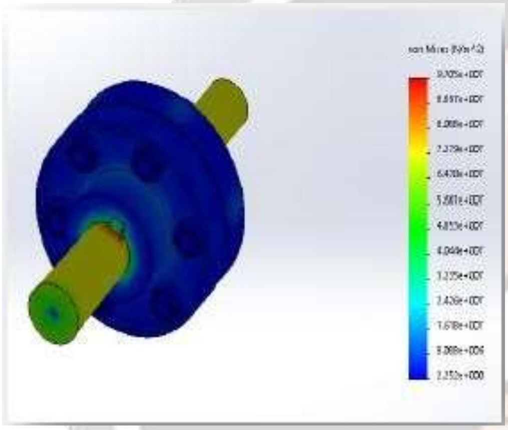


Fig. 11: Equivalent stress of bushed pin flexible coupling

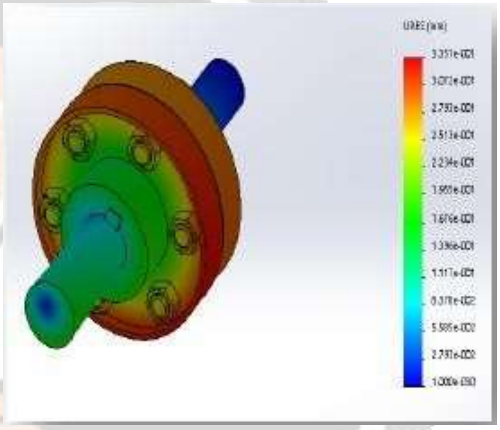
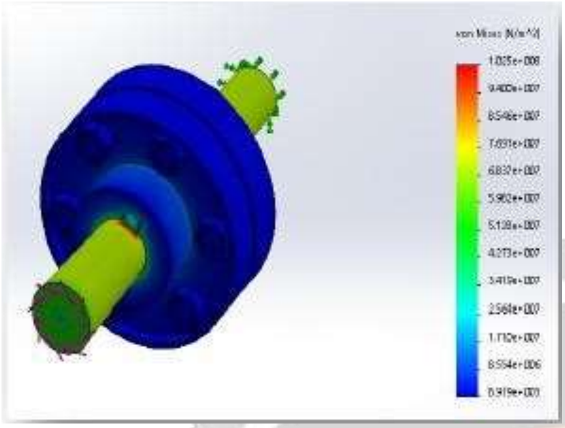


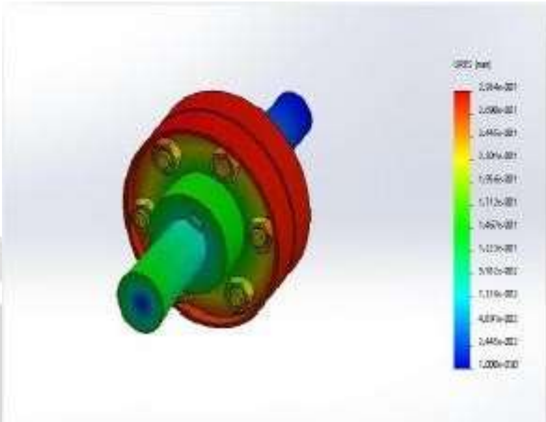
Fig. 12: Total deformation of bushed pin flexible coupling

Replacing Both Brass and Rubber bushed with

Only Brass bush with thickness of 8 mm shown in Fig.13
Only Rubber bush with thickness of 8 mm shown in Fig.14
Using Aluminum bush with thickness of 8 mm Fig.15
Solid pin attached to the pin hole shown in Fig.16

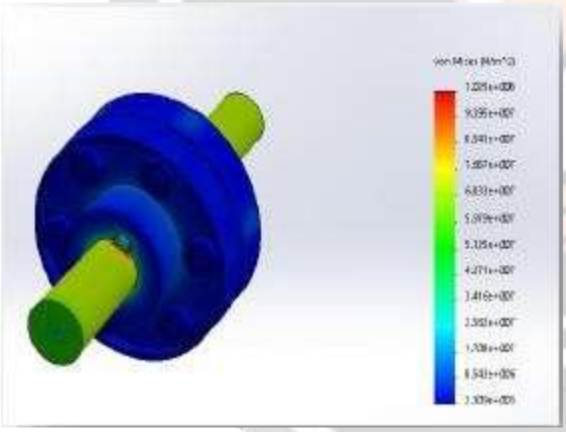


Equivalent stress

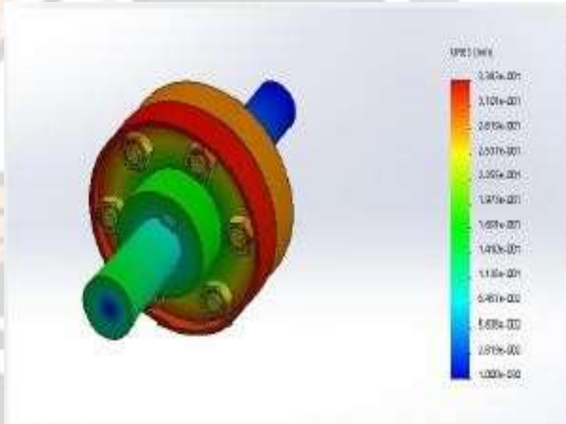


Total deformation

Fig. 13: Brass Bush on high Torque test (T = 1000 N.m)

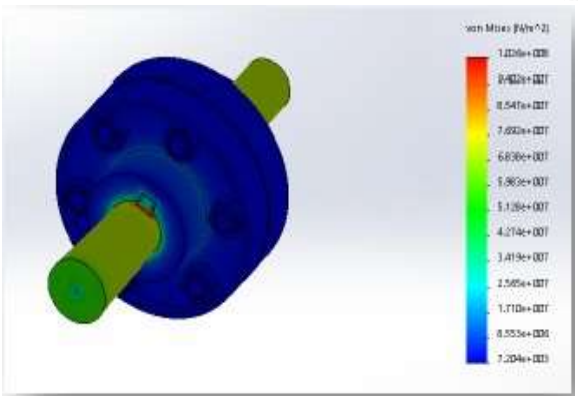


Equivalent stress

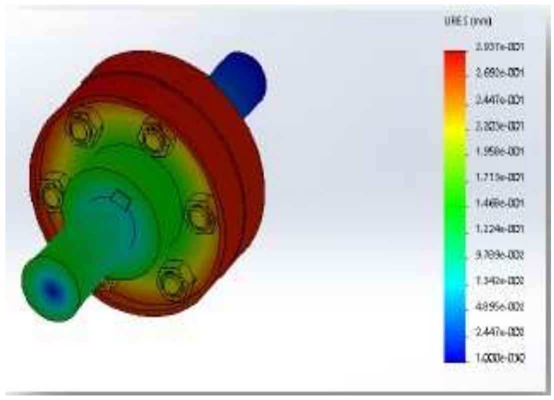


Total deformation

Fig. 14: Rubber Bush on high Torque test (T = 1000 N.m)

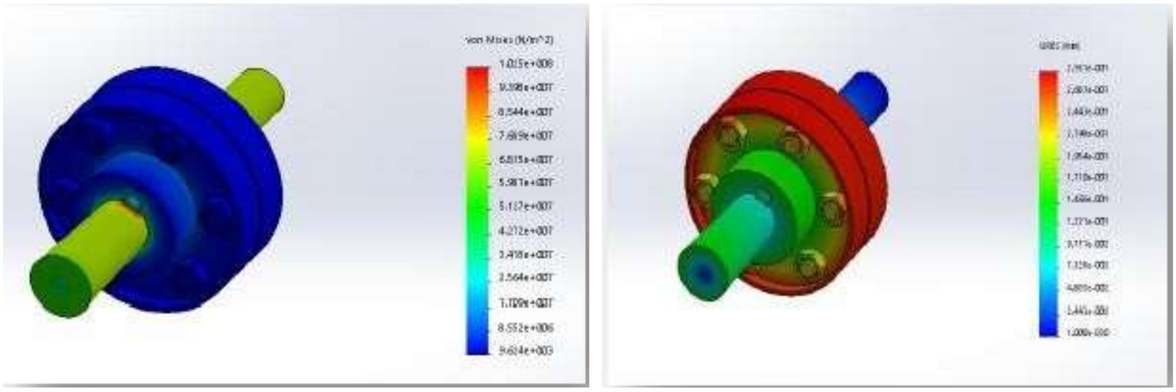


Equivalent stress



Total deformation

Fig. 15: Aluminum Bush on high Torque test (T = 1000 N.m)



Equivalent stress **Total deformation**

Fig. 16: Solid Bolt on high Torque test (T = 1000 N.m)

4. Results and Discussion

Further test applied on the couplings by accessing torque applied on the free end, the material limit is showing the optimum results on the failure possibilities when change of the bush types or removing it, thus to find the main deformation that occurs in the main coupling body that happens when using different bush like Brass, Rubber and Aluminum or removing it completely. Brass bush shows that the both sides of the coupling deforms with small amount of change due to twisting, while in the Rubber bush the deformation is less and the torque loss is high as acts like dumper and spring which causes vibrations, while the Aluminum do not shows any difference compared to the brass as it also causes much deformation and stress for the coupling, thus the optimum solution is using both Brass and Rubber together to get the best result.

5. REFERENCES

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