THEORETICAL DESIGN AND STRESS ANALYSIS FOR FLANGE COUPLING

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Abstract

Flange couplings are used into pressurizing piping systems that require the connection of tubing ends or two pipes. Flange couplings are characterized by strong connecting methods. This paper aims to design and analyze the stresses of a gray cast iron protected flange coupling, that connects a motor to a pump. The flange will transfer 15KW at 900rpm to momentum (215KN.m) on the right side with a service factor of 1.35. The flange design was based on the theoretical structure analysis and was modeled into a three-dimensional model. Results have shown that the shaft and key were the most components affected by the applied torque. The applied torque had a relatively insignificant effect on the resulting parameters of the flange, and the only significant effect on the bolt was at the contact points with the flange. The theoretical results were compared to the allowable stresses of the chosen material. The results of this comparison have shown that the flange coupling's material can endure the applied torque without failure or deformation and within an acceptable safety margin.

Keywords: Flange; Shear Stress.

Introduction

A coupling is a device used to connect the shafts together to transmit power and torque. Generally, couplings are used for connection of shafts' unit that are manufactured separately, such as a motor and a generator, an electric motor and a centrifugal pump. Due to the inconvenience in transportation of shafts of greater length, it becomes necessary to join two or more shafts by means of coupling. The shafts that are connected by coupling needs to be easily assembled and dismantled when repaired and altered. The severe failure due to shearing of bolts head, key head, nuts and other projecting parts may cause accidents (Kondru et al,2017). Therefore, it should be covered by developing a suitable shape to the flange or by providing guards. Flange coupling is classified into two types namely Rigid Coupling and Flexible Coupling. Rigid flange coupling consists of two separate gray cast iron flanges. One keyed to the driving shaft and the other to the driven shaft by means of nuts and bolts arranged in a circle concentric with the axes of the shafts. There are two subtypes of rigid flange couplings; protected rigid flange coupling and unprotected rigid flange coupling. In a protected rigid flange coupling, a protective circumferential rim covers the nut and bolt head. So in case of bolts failure during operation the broken piece of the bolt will dash against this rim and eventually fall down, and hence the operator is protected from possible injuries. In unprotected rigid flange, the protective circumferential rim is absent so in case of failure of the bolt, it may hit and harm the operator (Kondru et al,2017). Rigid flange is usually manufactured by casting as it consists of projection and recess. The commonly used material for flange coupling is gray cast iron, which is characterized by graphitic microstructure that cause fracture of the material to have a gray appearance (Kondru et al,2017, p 400). It is one of the most commonly used forms of cast iron, and the widely used cast material is based on the casting properties. Most alloys of iron contain 2.5-4% carbon, 1-3% silicon and iron constitutes the rest (Saurav et al 2015, p 9600). Compared to its compressive strength, its tensile strength and shock resistance are less. Its mechanical properties are controlled by the size and morphology of the graphite flakes, which deflect a passing crack and initiate counter fewer new cracks as the material breaks due to which it has good wear resistance and damping capacity. It also experiences less solidification shrinkage than other cast iron that does not form a graphitic microstructure during the casting process. The silicon promotes good corrosion resistance and increases fluidity while casting. It also offers good weld ability (Saurav et al 2015, p 9600).

The rigid flange coupling is specially designed and developed for horizontal shaft mounted gear unit applications (Dhaval, 2013). These applications require a rigid link between the low-speed shaft of the gear unit and the machine, such as for conveyor drive, bucket elevator, travel drive applications. In rigid flange coupling, flanges are brought together and bolted into the annular space between the hub and the protecting flange. The protective flange is provided to guard the projecting bolt heads and nuts. The bolts are placed equi-spaced on a bolt circle diameter, and the number of bolts depends on the shaft diameter (Dhaval,2013). A spigot on one flange and a recess on the opposing face are provided for ease of assembly. During design procedure, generally, the shaft diameter for a given torque transmission is determined and then the empirical relationships of other dimensions of the coupling are obtained.

Mathematical Modeling

Figure (1) shows the model used to represent a flange coupling that connects a motor to a pump where d, L, D, D1 and D2 represent the shaft diameter, flange length, hub diameter, bolt pitch circle diameter and flange outer diameter respectively whereas tp and tf represent the thickness of the protective circumferential flange and thickness of the flange respectively. The model is presented under some assumptions such as no time-varying loads are applied, the load is static, material properties are continuing, it is homogeneous and isentropic, there are no internal stresses before loading and accelerations equal zero. Tables (1) and (2) show gray cast iron allowable stresses and material properties respectively.

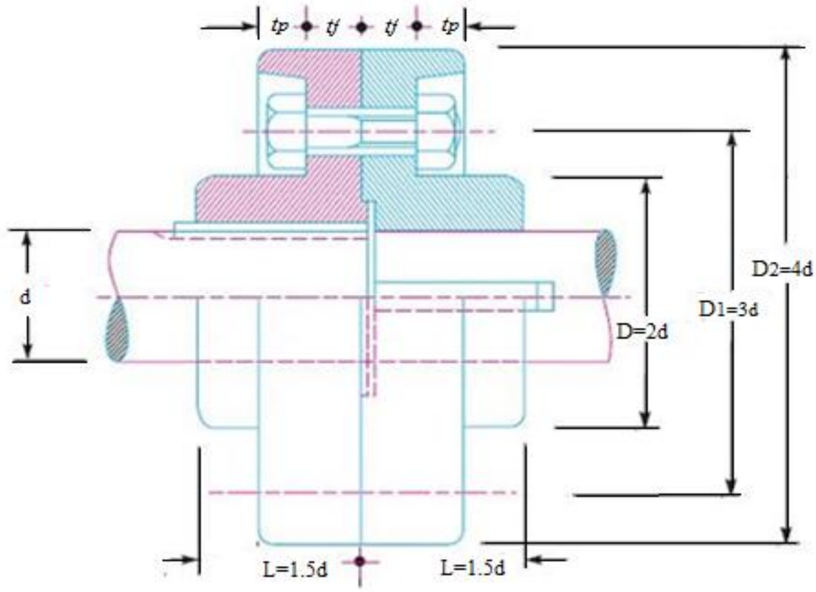

Figure 0): Flange Coupling Model (R.S Khurmi; J.K Gupta, 2005).

Table (1): Gray Cast Iron Allowable Stresses.

Allowable shear stress in shaft	40 MP
Allowable shear stress for key material	40 MPa
Crushing stress for bolt and key	80 MPa
Shear stress for bolt material	40 MPa
Gray cast iron shear stress.	8 MPa

Table (2): Gray Cast Iron Material Properties.

Density (Kg/m3)	7200
Young modulus (MPa)	1.1x105
Poisson's ratio	0.28
Bulk modulus (MPa)	83333
Shear modulus (MPa)	42969
Tensile yield strength	0
Compressive yield strength	0
Tensile ultimate strength (MPa)	240
Compressive ultimate strength (MPa)	820

Hub Design

As mentioned above, the flange coupling includes the hub, key, flange and bolts, so each part needs to be designed separately. As shown below, all equations used in the design were according to R. Khurmi (2005).

Before designing the hub shown in Figure (2); it is necessary to calculate the diameter of the shaft (d). For shaft diameter, it is essential to calculate the torque transmitted by the shaft.

$$\boldsymbol{T = \frac{P \times 60}{2\pi N}} \quad \textbf{(1)}$$

Where;

T torque.

P Power transmitted.

N revolution speed.

τ_{shaft} allowable shear stress

τ_{hub} induced shear stress

Since the service factor is 1.35, the maximum torque transmitted by the shaft is shown below;

$$\boldsymbol{T_{max} = T \times 1.35} \quad \textbf{(2)}$$

$$\boldsymbol{d = \sqrt[3]{\frac{T_{max} \times 16}{\pi \times \tau_{shaft}}}} \quad \textbf{(3)}$$

$$\boldsymbol{D = 2d} \quad \textbf{(4)}$$

$$\boldsymbol{L = 1.5d} \quad \textbf{(5)}$$

$$\boldsymbol{\tau_{hub} = \frac{T_{max} \times 16 \times D}{\pi \times (D^4 - d^4)}} \quad \textbf{(6)}$$

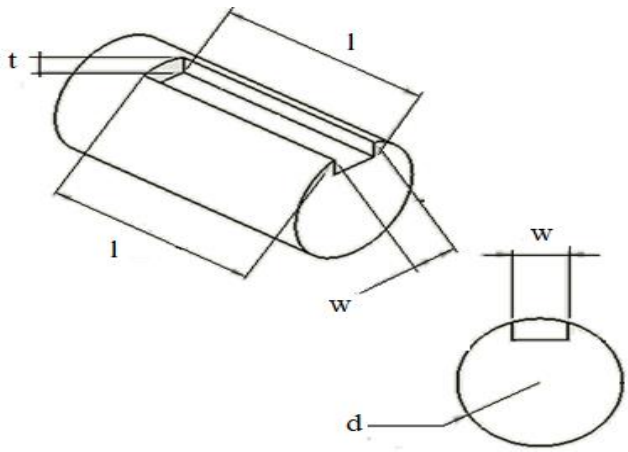

Figure (2): The Shaft Schematic Used in Hub.

Key Design

The permissible crushing stress for the key material is twice of its permissible shear stress i.e.:

$$\boldsymbol{\sigma_{ckmax} = 2\tau_{kmax}} \quad \textbf{(7)}$$

Where σ_{ckmax} is the permissible crushing stress and τ_{kmax} is the permissible shear stress. A square key will be used for the flange. For the calculated shaft diameter of 35 mm diameter, a key width of 12 mm and key thickness of 12mm will be used i.e.:

$$\boldsymbol{t = w} \quad \textbf{(8)}$$

The key length (l) equals the length of hub i.e.:

$$\boldsymbol{l = L} \quad \textbf{(9)}$$

Induced shear stress is calculated by using the maximum torque transmitted.

$$\boldsymbol{\tau_k = \frac{T_{max} \times 2}{l \times w \times d}} \quad \textbf{(10)}$$

where τ_k is the key's induced shear stress.

Induced crushing stress is also calculated using the maximum torque transmitted.

$$\boldsymbol{\sigma_{ck} = \frac{T_{max} \times 4}{l \times t \times d}} \quad \textbf{(11)}$$

where σ_{ck} is the key induced crushing stress. If the induced shear and crushing stresses in the key are less than the permissible stresses, the key design is considered safe. Figure (3) shows the key schematic.

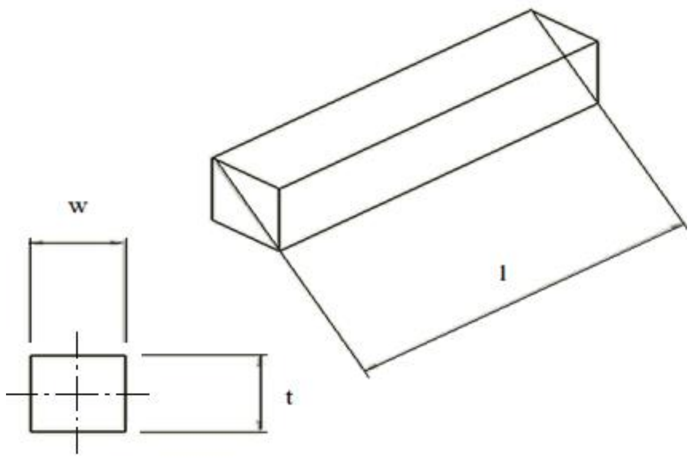

Figure (3): Key Schematic.

Design For Flange

The thickness of flange (tf) is half of shaft diameter i.e.:

$$tf = 0.5d \qquad (12)$$

The induced shearing stress in the flange is calculated by considering the flange at the junction of the hub in shear stress.

$$\sigma_f = \frac{T_{max} \times 2}{\pi D^2 \times tf} \qquad (13)$$

where σ_f is the flange induced shear stress. The flange design is considered safe if the induced shear stress is less than the permissible shear stress. Figure (4) shows the flange scheme.

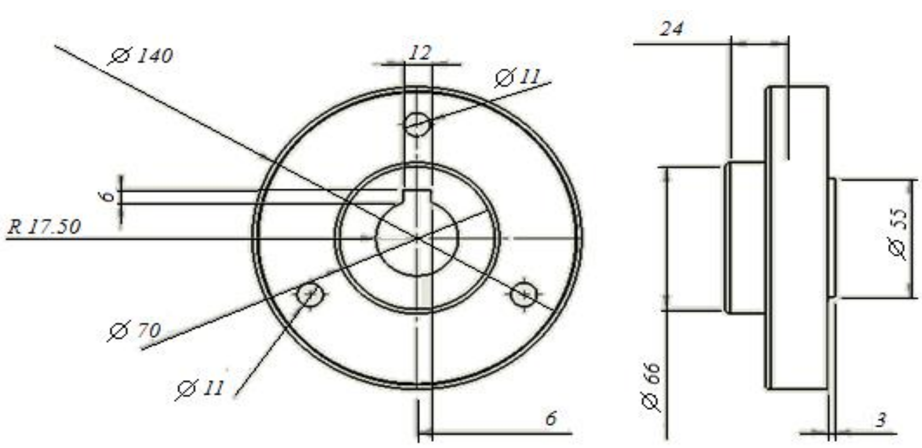

Figure (4): Flange Scheme.

Bolts Design

The number of bolts needed depends on the diameter of the shaft.

$$\mathbf{D1=3xd} \qquad \textbf{(14)}$$

To calculate the bolt diameter, equation (15) the maximum transmitted torque is used.

$$d_1 = \sqrt[2]{\frac{T_{max}\times 8}{n\times\pi\times\sigma_b\times D_1}} \qquad \textbf{(15)}$$

Where n is the number of bolts and d1 is bolt diameter. Assuming coarse threads, the closest standard size to the bolt diameter is chosen. To calculate the bolt's crushing stress, the equation (16) is used.

$$\sigma_C = \frac{2\times T_{max}}{n\times d_1\times D_1\times tf} \qquad \textbf{(16)}$$

Figure (5) shows the scheme of the bolt, while figure (6) shows the scheme of the nut.

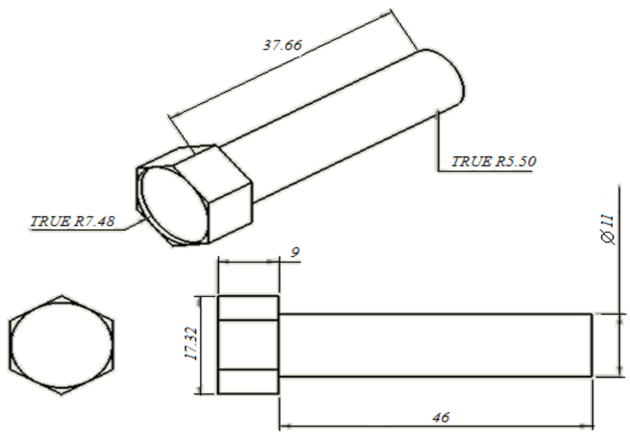

Figure (5): Bolt Schematic.

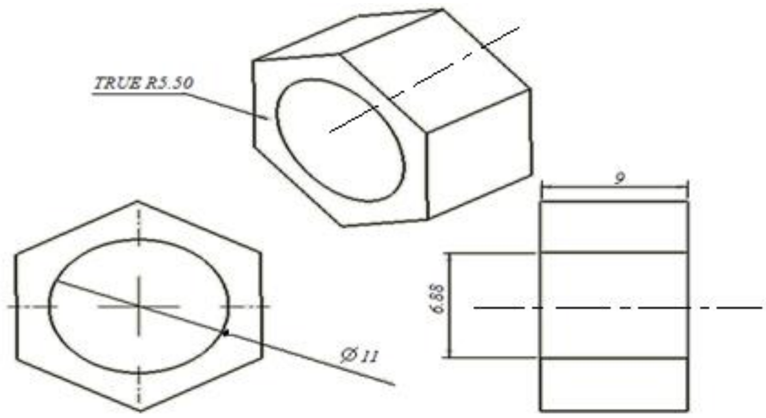

Figure (6): Nut Schematic.

The outer flange diameter is calculated using the equation:

$$D2 = 4d \qquad (17)$$

Thickness of the protective circumferential flange is calculated using the equation:

$$t_p = 0.25d \qquad (18)$$

Design and Theoretical Results

The maximum transmitted torque was calculated to be 215000 N.m. . The flange dimensions based on the theoretical calculations are shown in Table (3), while Table (4) shows the induced shear and crushing stresses.

Table (30 Flange Dimensions Based on the Theoretical Calculations.

Part specification	Dimensions (mm)	Standard size (mm)
Shaft diameter / inner hub diameter (d)	30.13	35
Hub diameter (dh)	70	70
Hub length (lh)	52.5	NA
Bolt circle diameter (D)	105	NA
Flange thickness (t)	17.5	NA
Protective flange thickness (tp)	8.75	9
Diameter of spigot and recess (dr)	52.5	NA
Outside diameter of flange (Do)	10	140
Bolt diameter	8.7	11

Table (4):Induced Shear and Crushing Stresses.

	Allowable (MPa)	**Theoretical result (MPa)**
Shear stress in shaft material	40	25.53
Shear stress in key	40	19.5
Shear stress in bolt	40	3.5
Crushing stress in key	80	39
Crushing stress in bolt	80	7

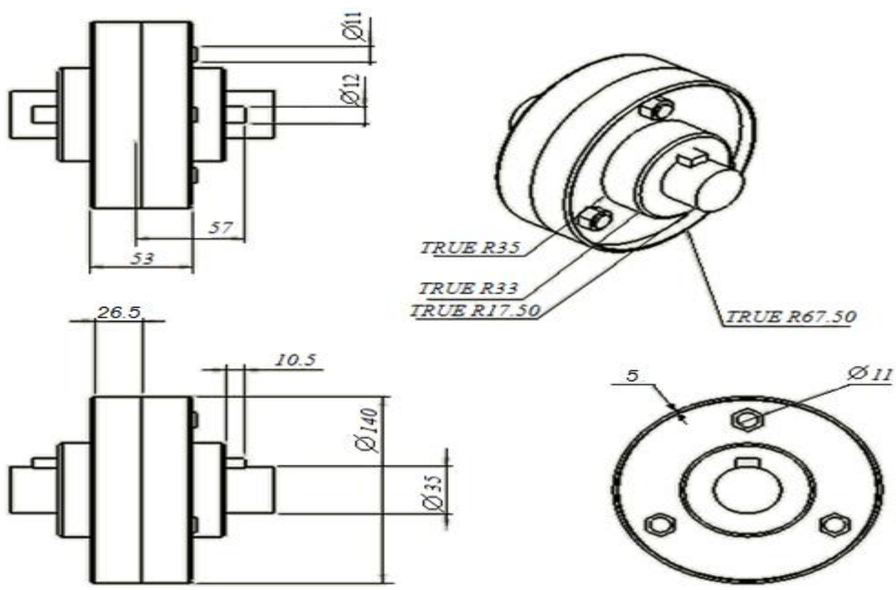

Figure (7): Show the Completed Flange Schematic.

Conclusion

To design and analyze a protected flange coupling, a three-dimensional model of the flange coupling was modeled and based on the results of the theoretical analysis of flange coupling. The flange coupling components included a shaft, a key, a hub and bolts. Gray cast iron was selected as the flange coupling's material.

Result parameters were based on a previous study by Shivaji G. Chavan, which included maximum principal stress, minimum principal shear stress, total deformation, maximum shear stress, normal stress, shear stress and equivalent stress.

Results have shown that the shaft and the key were the most affected components by the applied torque, especially at the contact points between the shaft, the key and the hub. This effect decreased steadily as it moved away from the contact points. The applied torque had a relatively insignificant effect on the resulting parameters of the flange. However, the only significant effect it had on the bolt was at the contact points with the flange. It is worth mentioning that in the theoretical analysis bolt stresses were lower than the permitted stresses. To check the safety of the flange coupling and ensure that the flange coupling would not fail or permanently deform under the applied torque, the results were compared to the allowed stresses of the chosen material. Results of this comparison revealed that the flange coupling's material can endure the applied torque without failure or deformation and it was within an acceptable margin of safety.

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