Tutorial 4: Design of Bushed-Pin Flexible Coupling

Design the coupling for rated power of 20 kW at 720 rpm. Draw it with the designed dimensions, and report all design calculations on the reverse side of the sheet. Refer the given table for material properties:

Component	S_{yt} (MPa)	S_{yc} (MPa)	S_{ys} (MPa)
Shaft	380	---	$0.5\ S_{yt}$
Flange/Hub	---	---	15
Key	400	$1.5\ S_{yt}$	$0.5\ S_{yt}$
Pin	400	---	35

- Factor of safety may be taken as 2.
- Bearing pressure between cast iron and rubber bush should not exceed 1 MPa.
- Missing dimensions, if any, may be adjusted within 5 – 6mm.

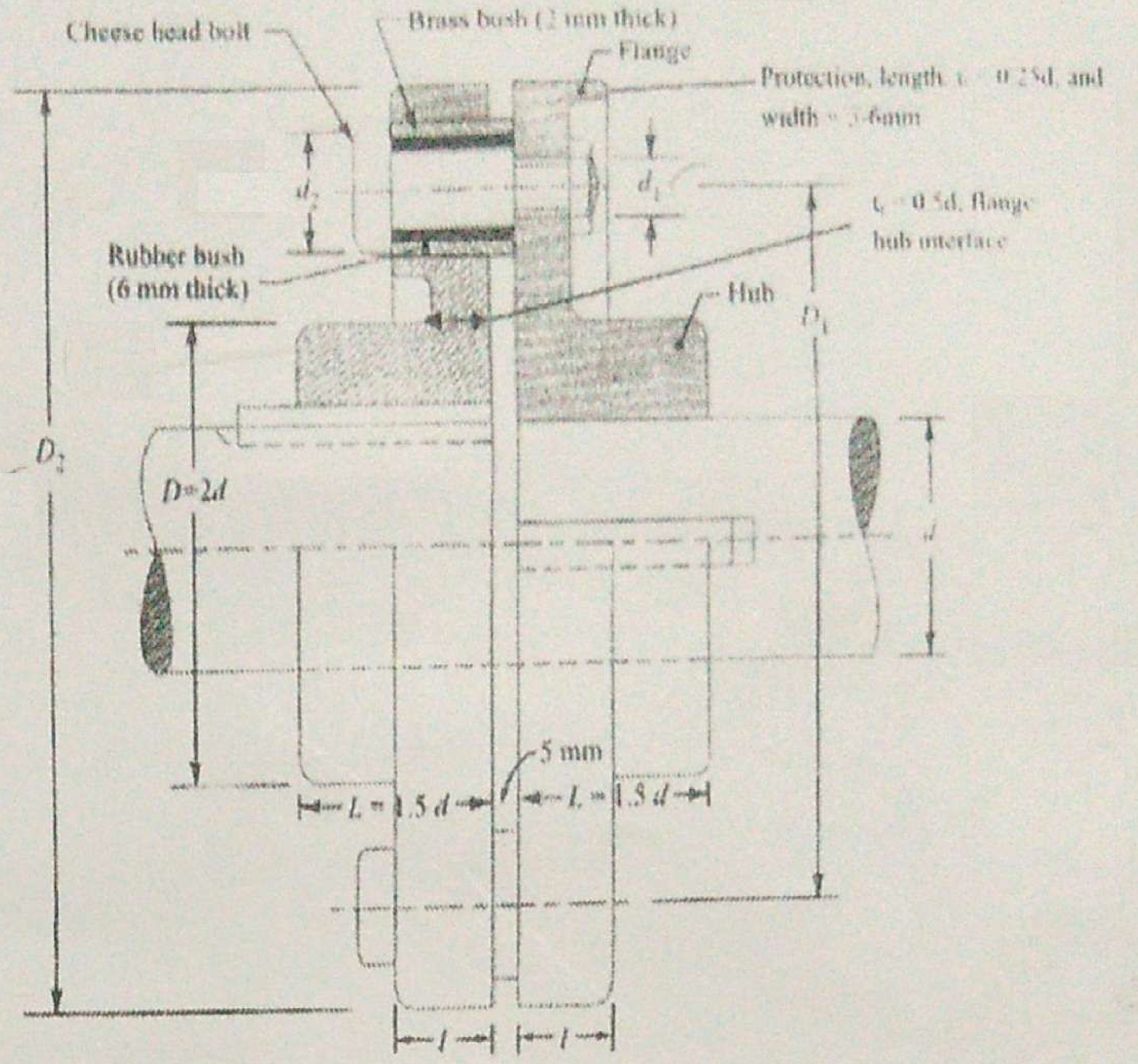


Figure: Bushed-Pin Flexible Coupling

Design Steps: (List out allowable stresses for each component).

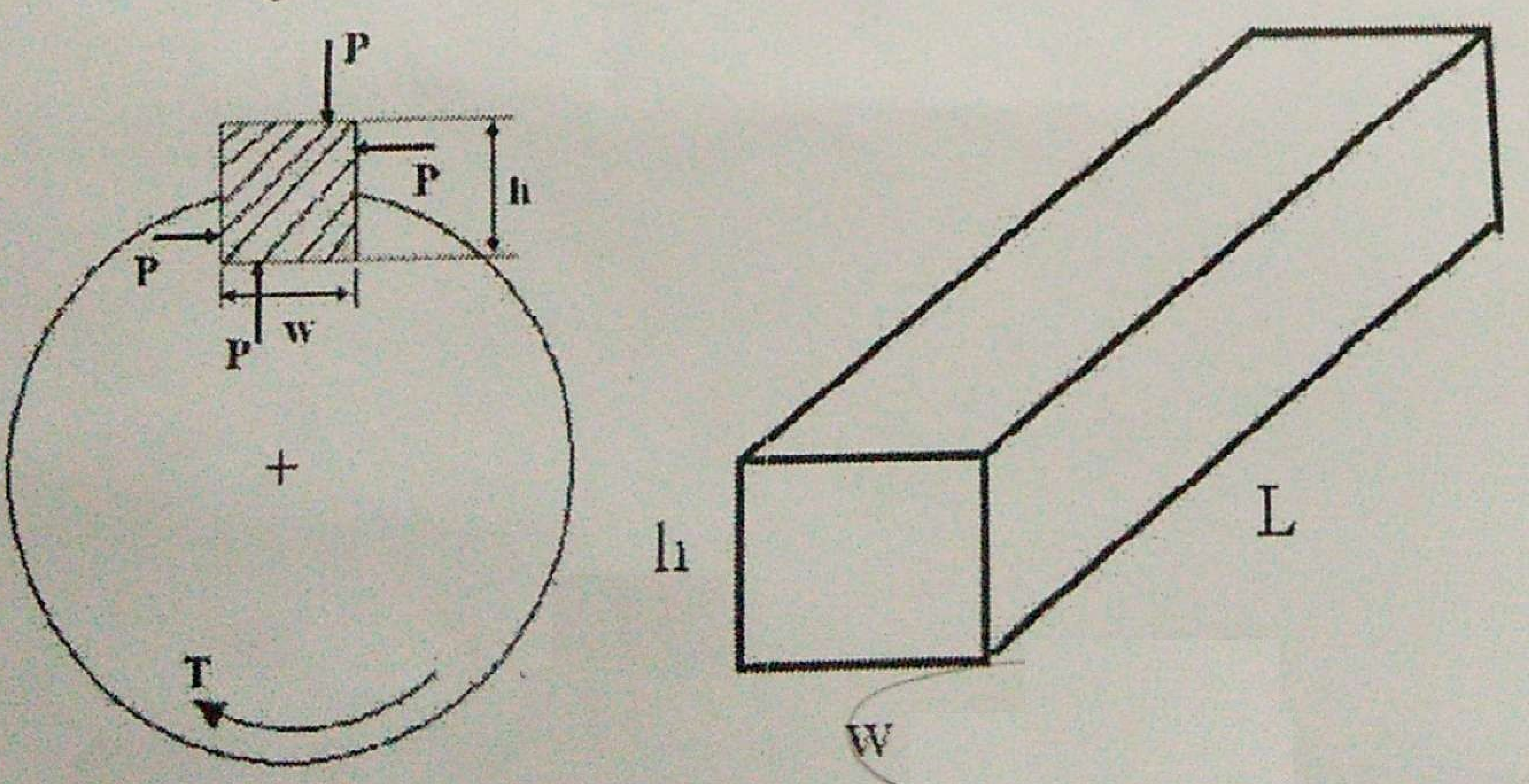
Design of Shaft: Take the designed power to be 50% more than the rated power. Use the following formulas: $P = T.\omega$ and $\tau = 16T/\pi d^3$.

Standard shaft sizes are: 25 mm to 60 mm in steps of 5 mm; 60 mm to 110 mm in steps of 10 mm; and 110 mm to 140 mm in steps of 15 mm.

Design of Hub:

Hub diameter D=2d, and its length L = 1.5d. Check the stress limit in the hub by considering the hub as if a hollow cylinder is transmitting power. Therefore, $\tau = 16TD/\pi\left(D^4 - d^4\right)$

Design of key: Length of the key L =1.5 d; the key is normally designed for equal strengths in shear and crushing. Crushing may occur due to severe compressive stresses.

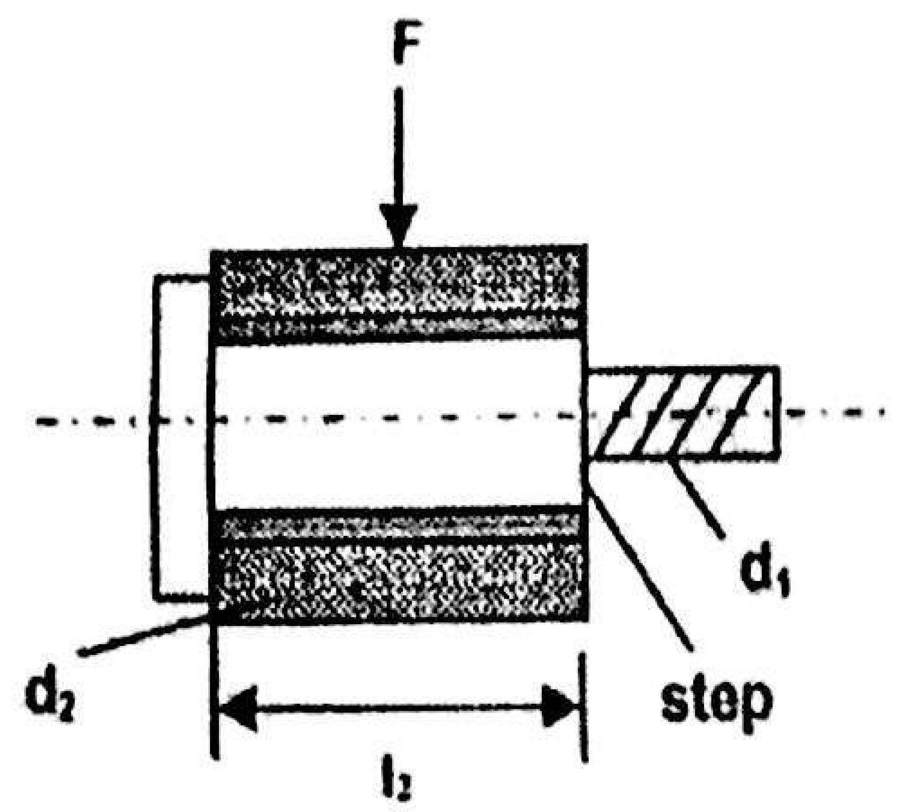


- In shear: $\tau = 2T/d\,w\,L$
- In crushing: $\sigma_c = 4T/d\,h\,L$
- Key width may be taken as 1/4th of the shaft diameter.
- Since the exact location of P on key surface is unknown; hence, for simplification, it may be taken as tangential to the shaft.

Design of flange: Thickness of flange, $t_f = 0.5d$. Check for shear failure at the junction of flange and hub. Failure may occur due to direct shear stress over the interfacial area.

Thereby, $\tau = T/(\pi D t_f \times (D/2))$

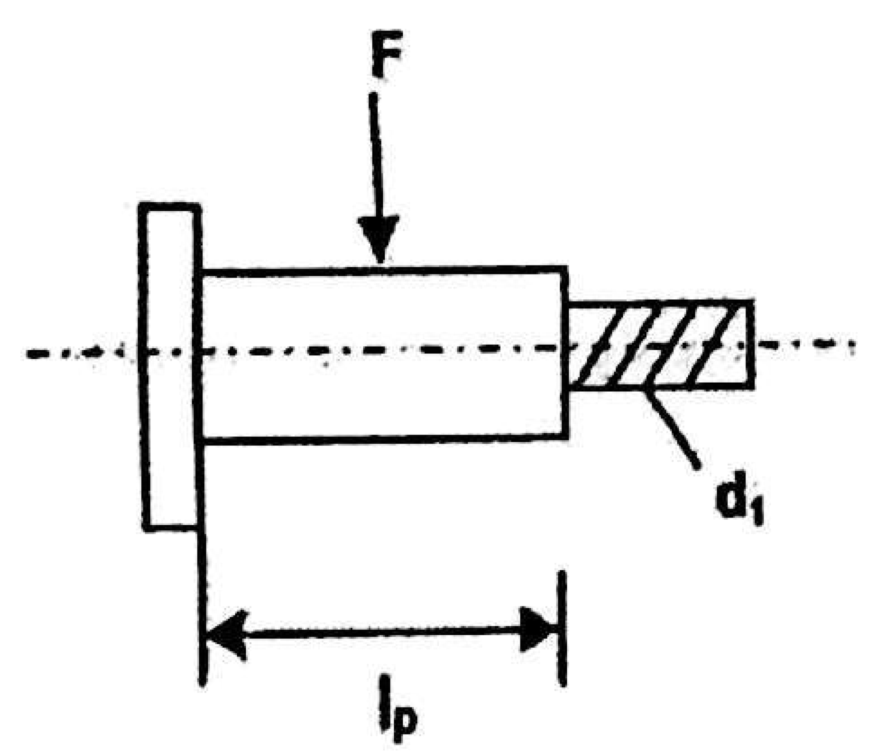
Design of the rubber bushes and pins:



- Number of pins/bushes, N = 6 to 8 or more; however, start with 6.
- Load on each rubber bush/pin: $F = T/(N \times (D_1/2))$, where D_1 – pin pitch circle diameter of the pins and may taken within 3d to 4d.
- Pin diameter: $d_1 = 0.5d/\sqrt{N}$.
- Check for the shear strength of the pin. $\tau = F/\left(\pi d_1^2/4\right) \le 35$ MPa
- Outer diameter of the bush: d_2 = d_1 + 2mm (2xstep) + 4mm (2xbrass bush thickness) + 12mm (2x rubber bush thickness)

Check for the bearing pressure on rubber bush: $P_b = F/(d_2 \times l_2) \le 1\,\text{MPa}$, where l_2 is the effective length of the bush, usually, $l_2 = d_2$.

Check for the strength of pin under combined bending and shear: Length of the pin $l_p = l_2 + 5$ mm



- Maximum bending moment: $M = F\ l_p/2$
- Bending stress: $\sigma_b = M/Z$, Z – section modulus of the pin cross section of diameter d_1
- Shear stress is already obtained.
- Estimate the principal stresses, and use maximum principal stress theory to check the strength.

Note: If any of the conditions do not satisfy, then increase the number of bushes/pins, and reiterate the process. However, while increasing the number of bushes/pins, ensure that the circumferential gap between the holes (d_2) to be not less than 8 mm (space requirement to engage spanner for fitting). In addition, one may also vary pin pitch circle diameter to satisfy the design.