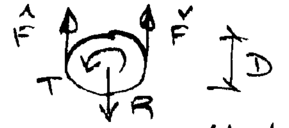
Approach: find the bending moment and torque at various cross-sections then implement (11) in the form $Z = M_E/\sigma_d$ where the design stress σ_d ($= S/n$) is given as 100 MPa.

Power $P = \omega T = 2\pi n T$ where $n = 900/60 = 15$ Hz.
so for $P = 20, 30, 50$ then $T = 212, 318, 530$ Nm.

A torque of 212 Nm is thus transmitted along CDE, and 318 from C to B. The torque at C will be taken as 318 Nm as an approximation.

At any pulley, diameter D

$T = (\hat{F} - \check{F})D/2$ and $\hat{F} = 3\check{F}$ here



So the shaft reaction is

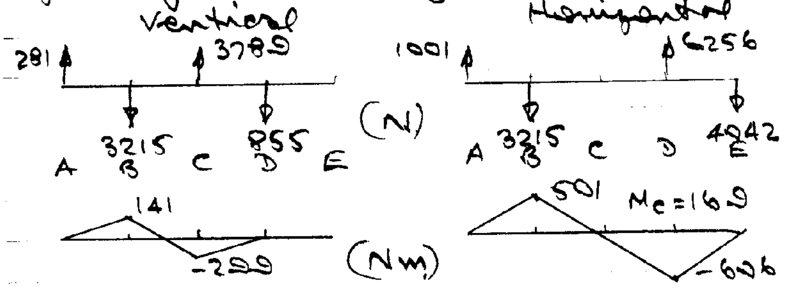
$R = \hat{F} + \check{F} = 4T/D$ — which appears as a shaft load.

$\therefore \quad R_B = 4 \times 318/0.28 = 4547$ N

$R_C = 4 \times 530/0.56 = 3789$ N

$R_E = 4 \times 212/0.21 = 4042$ N.

Resolving these loads into vertical & horiz'l planes, determining the support reactions at the bearings and finding the bending moments leads to :-



The results of the load problem enable us to restrict search to load points B, C, D

	Point	B	C	D
(Nm)	T	318	318	212
	M_V	141	299	–
	M_H	501	169	606
	$M_E(\lambda=1)$	610	468	642
	$M_E(\lambda=3/4)$	589	440	633

$M_E = \sqrt{(M^2 + \lambda T^2)}$

$\lambda = 1$ max shear stress

$= 3/4$ distortion energy

$M^2 = M_H^2 + M_V^2$, resultant

$\therefore M_E = \sqrt{[M_H^2 + M_V^2 + \lambda T^2]}$

cross-section D is thus the most critically loaded, since shaft diameter is constant.

Applying design equation

$$\underset{mm^3}{Z = \frac{\pi}{32} D^3} = M_E/\sigma_d = \underset{Nm\ \frac{mm}{m}\ \frac{mm^2}{N}}{642 \times 10^3 / 100} \quad ; \quad D = 40 \text{ mm}$$

(both theories).

In practice we would also have to consider local stresses due to pulley/shaft connections (i.e. geometric singularities → stress concentration).