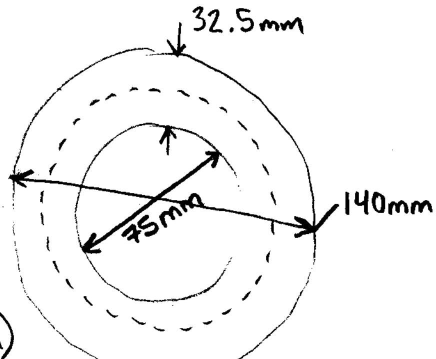
0 – 18.5 kN
10 cm THICK ALUMINUM HEAD
UNCONFINED 1mm COPPER ASBESTOS GASKET
CAST IRON BLOCK
PISTON IS 75 mm dia & CYL IS 140 mm OUTSIDE dia.

→ SPECIFY # OF BOLTS, PRELOAD, TIGHTENING TORQUE, AND BOLT CIRCLE
MINIMUM FACTOR OF SAFETY OF 1.5 AGAINST ALL FAILURE MODES
USE FINE THREAD



$$\frac{32.5}{3} = 10.8$$

$$C_b = \pi(107.5\,mm)$$

$$\#\,bolts = \frac{\pi(107.5\,mm)}{30\,mm} = 11.25$$

FOR CONVENIENCE, PUT BOLT CIRCLE AT CENTER (107.5mm dia). ASSUME 10 mm BOLT, SINCE ANY LARGER BOLT WOULD CAUSE EFFECTIVE CYLINDER TO EXIT THE ENGINE'S CYLINDER WALL (WHICH WOULD VOID CALCULATIONS).
IF THE BOLTS ARE PLACED IN SUCH A WAY TO HAVE THE EFFECTIVE CYLINDERS TOUCH, IT MIGHT BE A CONVENIENT PLACE TO START.

USING THE DESIGN ASSUMPTIONS ABOVE, CALCULATE FACTORS OF SAFETY FOR 12 10mm × 1.25 BOLTS.

FIRST COMPUTE THE BOLT STIFFNESS:

$$K_b = \frac{\pi d^2 E}{4 l} = \frac{\pi (0.010m)^2\, 206.8 \times 10^9\, N/m^2}{4(.115m)} = 1.41 \times 10^8\, N/m$$

(APPROX 1.5 BOLT DIAMETER IN BLIND HOLE IN ENGINE BLOCK)

NEXT CALCULATE THE JOINT AND GASKET STIFFNESS:

STIFFNESS OF ALUMINUM

$$K_{al} = \frac{2\pi d^2 E}{l} = \frac{2\pi (0.010m)^2\, 71.7 \times 10^9\, N/m^2}{(0.10m)} = 4.5 \times 10^8\, N/m$$

STIFFNESS OF GASKET

$$K_g = \frac{2\pi d^2 E}{l} = \frac{2\pi (0.010m)^2\, 93 \times 10^9\, N/m^2}{(0.001m)} = 5.8 \times 10^{10}\, N/m$$

STIFFNESS OF CAST IRON

$$K_{fe} = \frac{2\pi d^2 E}{l} = \frac{2\pi (0.010m)^2\, 103 \times 10^9\, N/m^2}{(0.015m)} = 4.3 \times 10^9\, N/m$$

TOTAL JOINT STIFFNESS

$$\frac{1}{k_m} = \frac{1}{k_{al}} + \frac{1}{k_g} + \frac{1}{k_{fe}} = 4.0 \times 10^8 \ \text{N/m}$$

$$C = \frac{k_b}{k_b + k_m} = \frac{1.41}{4 + 1.41} = 0.26$$

$$F_i = 0.60\, A_t S_p = 0.60(61.20\text{mm})\, 650\ \text{MPa} = 23.8\ \text{kN}$$

* SINCE THE F.O.S. IN YIELD MUST BE AT LEAST 1.5, WE NEED A RELATIVELY LOW PRELOAD, TRY 60%

18.5kN TOTAL EXTERNAL LOAD WITH 12 BOLTS → 1.54 kN/BOLT

$$P_b = C P_{ext} + F_i = 0.26(1.54\text{kN}) + 23.8\text{kN} = 24.2\text{kN}$$

$$N_y = \frac{720\text{MPa}\,(61.20\text{mm})}{24.2\text{kN}} = 1.82$$ (FACTOR OF SAFETY IN YIELD)

CHECK FACTOR OF SAFETY AGAINST JOINT SEPARATION

$$N_{sep} = \frac{F_i(1-C)}{P} = \frac{23.8\text{kN}(1-0.26)}{1.54\text{kN}} = 11.4$$

CHECK FACTOR OF SAFETY IN FATIGUE

$$F_{b,max} = 24.2\text{kN} \quad \& \quad F_{b,min} = 23.8\text{kN}$$

$$\sigma_{b,a} = \frac{F_{b,max} - F_{b,min}}{2A_t} = 3.3\ \text{MPa} \qquad \sigma_{b,m} = \frac{F_{b,max} + F_{b,min}}{2A_t} = 392\ \text{MPa}$$

$$S_e = C_{LOAD} C_{SIZE} C_{SURF} C_{TEMP} C_{RELIAB} \frac{1}{K_f} \frac{S_{ut}}{2} = 0.7(.95).72(1)\,1\,\frac{1}{3}\,\frac{900}{2} = 72\ \text{MPa}$$

$$N_f = \frac{S_e(S_{ut} - \sigma_i)}{S_e(\sigma_m - \sigma_i) + S_{ut}\sigma_{alt}} = \frac{72(900-390)}{72(392-390) + 900(3.3)} = 11.8$$

NOTE: HOW THE JOINT IS VERY OVER DESIGNED WITH MASSIVE FACTORS OF SAFETY, MOST SINGLE CYLINDER GAS ENGINES HAVE 6 OR 7 6mm BOLTS, THE CALCULATIONS ABOVE ARE FOR 12 10mm BOLTS, SO OF COURSE, WE EXPECT LARGE FACTORS OF SAFETY.
THIS PROBLEM HAS AN INFINITE NUMBER OF SOLUTIONS SINCE IT IS A QUESTION OF DESIGN.