

Design & Development Calculations

Design Package No.: 2L-DP1620-03

Package start date: 08/06/2019

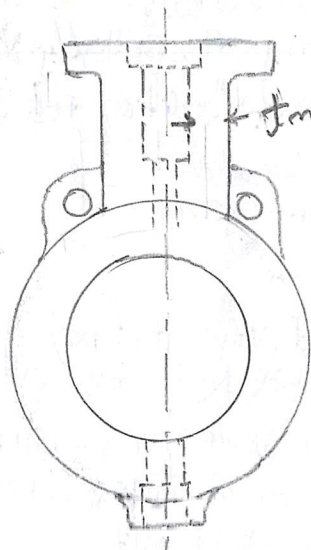
Customer:	2L Engineers
Work Order No., if any:	2L19-002 (PRO)
Valve Type:	Category - A, Butterfly Valve
Design Specification/Standard:	API 609, 8 th Edition Feb 2016, Errata April 2017
Operation:	Gear Operated
Size:	DN 150 (NPS 6)
Bore:	148.4 mm
Pressure Class:	Designated Valve at 10 bar
End Connections:	Welder to Bolt Class 150
Drilling / End Connection as per Standard:	As Per ASME B16.5 - 2017
Face to face as per Standard:	56 mm, As Per Table-2, API 609, 8 th Edition Feb 2016 Errata April 2017.
Inspection / Testing Standard:	API 548, 10 th Edition, Oct-2016.
Material- Body/Bonnet/Cover/Flap/Ball:	ASTM A216 Gr. WCB
Trim Material:	ASTM A479 TP 304
Overlay/Hard facing:	Not Required
Unidirectional/Bidirectional:	Bi-directional
Dead-end Service (API 609)	YES / NO Not Applicable
Reference Specifications:	Refer Format No:- 2L-D&D-02-609, Rev 00.

Reference:

- 1 API 609, 8th Edition Errata April 2017
- 2 ASME B16.34-2017, ASME B16.34 Series I Part 1-2013
- 3 ASME Sec VIII, Div-1-2013
- 4 Crane Nuclear Technical Paper.

Design Calculation

A) Body:



$t_m = \text{shell thickness}$

shell thickness,

The maximum bore considered in Body,
is, $D = 192 \text{ mm}$

a) maximum shell thickness as per ASME B16.34, Table 3A

$$t_{m1} = 8 \text{ mm}$$

b) Minimum shell as per ASME B16.34; Appendix VI.

$$t_{m2} = 0.0163 (D) + 4.7$$

$$= 0.0163 (192) + 4.7$$

$$t_{m2} = 7.8 \text{ mm}$$

c) Alternative shell thickness calculation:

Alternative shell thickness calculation is performed
as per ASME BPVC Section VIII Div-1, UG-27.

$$R = \frac{1}{2} = \frac{192}{2} = 96 \text{ mm}$$

Body Material Consider for Calculation is 'NCB', working pressure as per ASME B16.34 is,

$$P_o = 19.6 \text{ bar} = 1.96 \text{ MPa} \\ = 2 \text{ MPa}$$

$$\text{Shell Rating Pressure} = P = P_o \times 1.5 \\ P = 2 \times 1.5 \\ P = 3 \text{ MPa}$$

Allowable Stress as per ASME BPVC Section-II, Part-1.
 $S = 138 \text{ MPa}$.

Joint Efficiency = $E = 1$, for bolted joints,

Shell thickness Calculated by below Equation,

$$t = \frac{P \times R}{2SE + 0.4P} = \frac{3 \times 96}{(2 \times 138 \times 1) + (0.4 \times 3)} = 1.04 \text{ mm}$$

$$\underline{t = 1.04 \text{ mm.}}$$

Thus, the 2L selected shell thickness is 11mm & is greater than Calculation & ASME B16.34, Table-5A, Hence the Design is Safe.

Valve Torque :

Butterfly Valve torque are analysis of both total Seating/unseating torque & total /dynamic Torque with minimum required operator torque being the large of the two Times,

Seating / Unseating torque,

Total Seating & Unseating Torque (T_t) is Calculations as follows;

T_s = Seat Torque

T_b = Bearing Torque,

For Calculation of Seat Torque, the Seal Factor is considered as below;

Seat factor ;

$$(i) \text{ i.e. OD} = \phi 152.5 \text{ mm} = 15.25 \text{ cm}$$

$$\text{Sleeve ID} = \phi 148.4 \text{ mm} = 14.84 \text{ cm}$$

$$\text{Area of Sealing} = \frac{\pi}{4} \times (OD^2 - ID^2)$$

$$A = \frac{\pi}{4} \times [(15.25)^2 - (14.84)^2] = 9.2 \text{ cm}^2$$

$$\text{Seat factor} = \frac{\text{elongation} + \text{Tensile strength}}{\text{Hardness} + \text{Area of Sealing}}$$

$$\text{Seat factor} = \frac{454 + 48}{70 + 9.2} = 6.8$$

$$\text{For Safety factor} = 1.5 \times \text{Seat factor} = 1.5 \times 6.8 = 9.45$$

$$\therefore \text{Seat factor} \approx 10$$

A) Seat Torque : $T_s \Rightarrow$

The Seat Torque for both interference related, & differential pressure related effects, The Seating factor of 10 has been found to Conservatively bound, the Required torque to account for Seat material, Seat Contact, thickness & the Service application.

$$T_s = AD^2, \text{ where } A = \text{int factor} = 10$$

$$D = \text{Disc diameter} = 152.5 \text{ mm} = 6 \text{ inch}$$

$$T_s = AD^2$$

$$= 10 \times 6^2$$

$$T_s = \underline{360 \text{ m-lbs}}$$

B) Bearing Torque (T_b) :

$$T_b = \mu \left(\frac{\pi}{4} \times D^2 \right) \frac{d}{2} \times P$$

where, T_b = Bearing Torque

$$\mu = \text{friction coefficient} = 0.15$$

$$D = \text{Disc diameter} = 152.5 \text{ mm} = 6 \text{ inch}$$

$$d = \text{Stem diameter} = 19 \text{ mm} = 0.75 \text{ in}$$

$$P_0 = \text{Working Pressure} = 19.6 \text{ bar} = 300 \text{ PSI}$$

$$P = \text{Shell testing Pressure} = 1.5 \times P_0 = 1.5 \times 300$$

$$P = 450 \text{ PSI.}$$

$$\tau_b = \mu \left(\frac{\pi}{4} \times D^2 \right) \left(\frac{d}{2} \right) P$$

$$= 0.15 \left(\frac{\pi}{4} \times 6^2 \right) \times \left(\frac{0.75}{2} \right) 450$$

$$\tau_b = 716 \text{ in-lbs.}$$

Total Torque \Rightarrow

$$\tau_{td} = \tau_s + \tau_b$$

$$= 360 + 716$$

$$= 1076 \text{ in-lbs}$$

$$\tau_{td} = 121.5 \text{ N-m}$$

Stem diameter Calculation:

For safety factor, the design torque for all trains calculation, shall considered 1.5 times of calculated torque.

$$\text{Therefore, } \tau_{stem} = 1.5 \times \tau_{td} = 1.5 \times 121.5$$

$$\tau_{stem} = 182.25 \text{ N-m}$$

For safety factor conditions the stem is calculated by using the definitions of SS304 as the base material, other material can be used, stem material without verification as long as the minimal yield strength (σ_y) of the selected material is 205 MPa.

 2L Engineers	Design & Development Calculations	
	Design Package No.:	2L-DP1920-03
	Package start date:	08/06/2019

For Safety Factor the maximum primary shear stress under design conditions, exclusive of stress concentration at the periphery of solid concentration at the periphery of solid Circulation Section is torsion shall be limited to 0.8 γ_s , Therefore the nominal torque applied on stem shall be calculated by this factor of lower Yield strength.

$$\tau = 0.8 \times \gamma_s = 0.8 \times 205$$

$$\tau = 164 \text{ MPa}$$

The Maximum torsional shear stress induced due to torsional loading is calculated by below Equation,

$$\tau = \frac{16 \times T_{stem} \times 1000}{\pi d^3}$$

Simplifying above Equation, minimum stem diameter required as,

$$d = \sqrt[3]{\frac{16 \times T_{stem} \times 1000}{\pi \times \tau}} = \sqrt[3]{\frac{16 \times 21.5 \times 1000}{3.142 \times 164}}$$

$$d = 15.56 \text{ mm}$$

The adapted stem diameter is 19mm,

where, T_s = Valve torque (Nm).

γ_s = nominal Yield strength

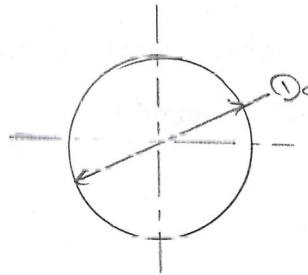
τ = maximum Primary shear stress (MPa)

d = minimum stem diameter required (mm).

Weakest Section Outside the Pressure Boundary,

According to ASME-B09, Clause 5.5.1, the shaft to disc connection at all parts of the shaft within the pressure boundary shall, under torsional load, exceed the strength of the shaft that lies outside the pressure boundary by more than 10%, the determination of shaft strength shall be Calculation or Testing.

at Circular Sealing Section of Stem,



① = Stem diameter

r = Stem Radius.

According to Roark's Stress & Strain handbook, Table 10.1, Shear stress induced in shaft with circular section is given by below equation,

$$\tau_{max} = \frac{2 \times T_{stem} \times 1000}{\pi \times r^3}$$

$$\text{Stem Radius at sealing section} = r = \frac{D}{2} = \frac{19}{2}$$

$$r = 9.5 \text{ cm}$$

Maximum torque carrying Capacity of Circular Section;

$$T = \frac{S_m \times \pi \times r^3}{2 \times 1000} = \frac{164 \times \pi \times (9.5)^3}{2 \times 1000}$$

$$T = 220.89 \text{ Nm}$$

Top Section of Stem (key section)

According to Rankine's stress & strain hand book Table 10.1, Shear stress, induced in shaft with single key way, is Calculated by Equation.

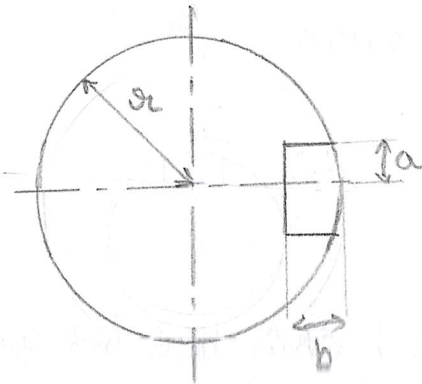
$$\tau_{\text{induced}} = \frac{T_{\text{stem}} \times 1000 \times B}{r^3}$$

Where, B varies with b/r as follows.

$$\text{For } 0.2 \leq \frac{b}{r} \leq 1.5$$

$$B = k_1 + k_2 \left(\frac{b}{r} \right) + k_3 \left(\frac{b}{r} \right)^2 + k_4 \left(\frac{b}{r} \right)^3$$

where, for $0.5 \leq \frac{a}{b} \leq 1.5$



key dimension i.e,

$$a = 3 \text{ mm}$$

$$b = 3.5 \text{ mm}$$

$$d_{\text{coupling}} = \text{Coupling diameter} = 19 \text{ mm}$$

$$a = \frac{d_{\text{coupling}}}{2} = \frac{19}{2} = 9.5 \text{ mm}$$

$$a = 9.5 \text{ mm}$$

where for,

$$k_1 = 1.1690 - 0.3168 \left(\frac{a}{b} \right) + 0.0490 \left(\frac{a}{b} \right)^2$$

$$= 1.1690 - 0.3168 \left(\frac{9.5}{3.5} \right) + 0.0490 \left(\frac{9.5}{3.5} \right)^2$$

$$k_1 = 0.93 \text{ mm}$$

$$k_2 = 0.4344 - 1.5046 \left(\frac{a}{b} \right) + 0.8677 \left(\frac{a}{b} \right)^2$$

$$= 0.4344 - 1.5046 \left(\frac{9.5}{3.5} \right) + 0.8677 \left(\frac{9.5}{3.5} \right)^2$$

$$k_2 = -0.22 \text{ mm}$$

$$k_3 = -1.1830 + 4.2762 \left(\frac{a}{b} \right) - 1.7024 \left(\frac{a}{b} \right)^2$$

$$= -1.1830 + 4.2762 \left(\frac{9.5}{3.5} \right) - 1.7024 \left(\frac{9.5}{3.5} \right)^2$$

$$k_3 = 1.62 \text{ mm}$$

$$k_4 = 0.8812 - 0.2627 \left(\frac{q}{b}\right) - 1.1897 \left(\frac{q}{b}\right)^2$$

$$= 0.8812 - 0.2627 \left(\frac{3}{3.5}\right) - 1.1897 \left(\frac{3}{3.5}\right)^2$$

$$k_4 = 1.53 \text{ mm}$$

$$B = k_1 + k_2 \left(\frac{b}{a}\right) + k_3 \left(\frac{b}{a}\right)^2 + k_4 \left(\frac{b}{a}\right)^3$$

$$= 0.93 + \left[(-0.22) \left(\frac{3}{3.5}\right)\right] + \left[1.62 \left(\frac{3}{3.5}\right)^2\right] + \left[1.53 \left(\frac{3}{3.5}\right)^3\right]$$

$$B = 2.9 \text{ mm}$$

$$\text{Hence, } \tau_{\text{induced}} = \frac{15200 \times 1000 \times B}{a^3} = \frac{152.75 \times 1000 \times 2.9}{(3.5)^3}$$

$$\tau_{\text{induced}} = 616.45 \text{ MPa.}$$

The maximum allowable torque at key section is,

$$MAST_{\text{key}} = \frac{\sigma_m \times a^3}{B \times 1000} = \frac{164 \times (3.5)^3}{2.9 \times 1000} = 48.48 \text{ Nm}$$

$$MAST_{\text{key}} = 48.48 \text{ Nm}$$

Conclusion: The minimum mast of the stem is found at the keyway section which is outside the pressure boundary area, thus the Design is Safe.

 2L Engineers	Design & Development Calculations	
	Design Package No.:	2L-DP1920-03
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Key Strength Calculation:

Key strength calculation is made by the torque applied on the key which is subjected to pure shear, the shear key strength is calculated by considering C45, as key material which has yield strength of 530 MPa.

For safety factor the maximum primary shear stress under design condition exclusive of stress concentration at the periphery of solid circular section in torsion shall be limited to 0.8 σ_y ,

$$\tau_{max} = 0.8 \times \sigma_y = 0.8 \times 530$$

$$\tau_{max} = 264 \text{ MPa}$$

- Coupling diameter = $d = 19 \text{ mm}$
- Width of key = $w = 6 \text{ mm}$
- Thickness of key = $t = 6 \text{ mm}$
- Length of key = $l = 30 \text{ mm}$
- No. of key = $n = 1$.

Tangential Shearing force acting at key is calculated by

$$F = \text{Area Resisting Shearing} \times \text{Shear Stress}$$

$$F = l \times w \times \tau \rightarrow \text{---} \text{---} \text{---}$$

Torque transferred by the shaft:

$$\tau = \frac{F \times d}{2}$$

i.e., $F = \frac{2 \times \tau}{d} \rightarrow \textcircled{b}$

Substituting \textcircled{b} in \textcircled{a} Simplifying we get induced shear stress as,

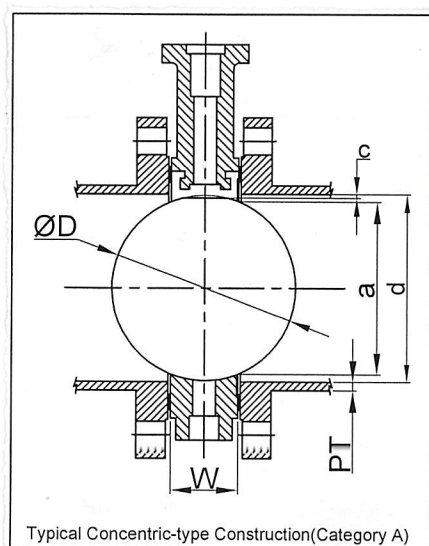
$$\tau_{ind} = \frac{T_{stem} \times 2 \times 1000}{d \times w \times d \times \pi} = \frac{182.25 \times 2 \times 1000}{30 \times 6 \times 19 \times \pi}$$

$$\tau_{ind} = 106.5 \text{ MPa.}$$

Conclusion:

Since the Induced Shear Stress (τ_{ind}) is less than the yield strength adopted keys are safe.

Disc :



Typical Concentric-type Construction (Category A)

As Per ASME B31.1, Appendix D, we have,

a = Chord of disc (mm), (To be Calculated)

c = Nominal Radial Clearance = 1.5 [As per Table D1, of ASME B31.1, 8th Edition]

D = Maximum disc diameter (mm), [To be Calculated].

W = Face to Face dimension (mm), = 56 mm (As per Table 2 of ASME B31.1, 8th Edition)

As Per ASME B31.1, we have

For Schedule 40, & nominal diameter 150 mm (NPS 6)

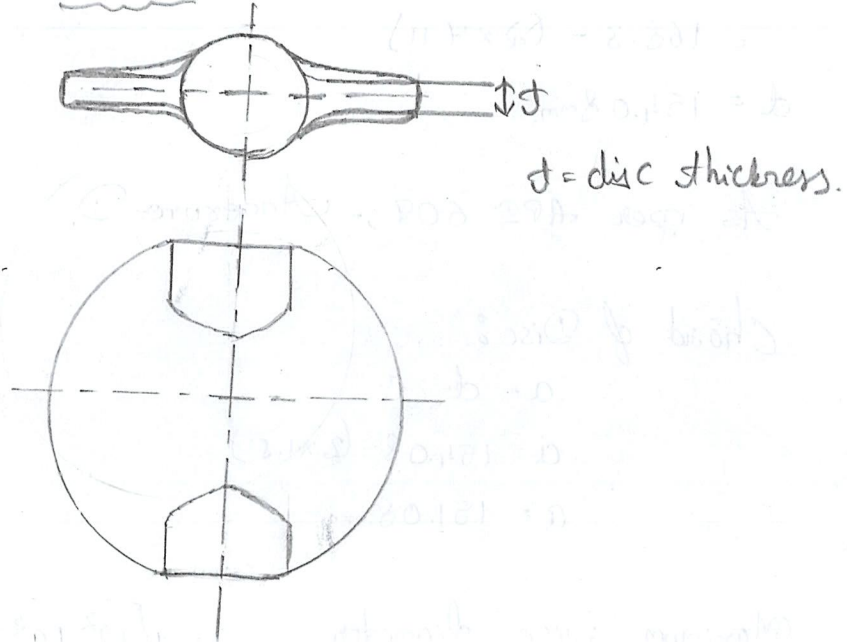
Pipe diameter = 168.3 mm

Pipe wall thickness = 7.11 mm

$\therefore d$ = Inside diameter of Connecting Pipe or Flange

$$d = (\text{Pipe diameter}) - 2 \times \text{Thickness.}$$

Disc thickness Calculation:-



The disc thickness calculation will be carried out by the below equation,

$$\tau = \frac{0.39 \times W}{t^2}$$

Simplifying above equation, Minimum disc thickness required as,

$$t = \sqrt{\frac{0.39 \times W}{\tau}}$$

Where;

$t = \text{Disc thickness in (mm).}$

$W = \frac{\pi}{4} \times D^2 \times P.$

$\tau = \text{Design shear stress in (MPa).}$

$D = \text{Disc diameter in (mm)}$

$P_w = \text{Working Pressure in (MPa).}$

$P = \text{Shell test Pressure in (MPa).}$

$$d = (\text{Pipe diameter}) - 2 \times \text{Thickness}$$

$$= 168.3 - (2 \times 7.11)$$

$$d = 154.08 \text{ mm}$$

As per API 609, Annexure-D.

Chord of Disc:

$$a = d - 2c$$

$$a = 154.08 - (2 \times 1.5)$$

$$a = 151.08 \text{ mm}$$

$$\text{Maximum Disc diameter} = D = \sqrt{w^2 + a^2} = \sqrt{(56)^2 + (151.08)^2}$$

$$D = 161.12 \text{ mm}$$

$$\text{Maximum disc diameter} = D = 161.12 \text{ mm}$$

Thus adapted disc diameter = 152.5 mm.

The Disc OD of 152.5 mm is adapted & Construction of Valve is made accordingly then the bore of diameter 148.4 mm is maintained after construction

Thus the adapted Valve bore ⇒ 148.4 mm

$$W = \frac{\pi}{4} \times D^2 \times P$$

$$= \frac{\pi}{4} \times (152.5)^2 \times 3$$

$$W = 54803.35 \text{ N}$$

$$d = \sqrt{\frac{0.39 \times W}{\tau}} = \sqrt{\frac{0.39 \times 54803.35}{164}}$$

$$d = \underline{\underline{5.24 \text{ mm}}}$$

The adapted disc thickness is 6mm, hence the design is safe.

The Bearing Calculation:

Bearing durability is calculated by the applied thrust force on the bearing torque surface area that is in contact with the disc.

Thrust force of Disc:

As there is one bearing on disc, the force is equal to 'F_{thrust}'

$$\therefore F_b = F_{\text{thrust}}$$

$$F_{thrust} = P \times \frac{\pi}{4} \times D^2 = 3 \times \frac{\pi}{4} \times (152.5)^2 = 54803.35 \text{ N}$$

$$F_b = F_{thrust} = 54803.35 \text{ N}$$

The DD of the Stem/Disc bearing is $\phi 19 \text{ mm}$ & the maximum allowable stress on H1-A5 (moderate), bearing is 200 MPa ,

$$B = \frac{F_b}{A \times \sigma_b} = \frac{54803.35}{19 \times 2000} = \underline{14.42 \text{ mm}}$$

The actual Bearing height is 30 mm , therefore the stress on bearing is,

$$\sigma_b = \frac{F_b}{A \times B_{act}} = \frac{54803.35}{19 \times 30} = 96.14 \text{ MPa}$$

$$\sigma_b = 96.14 \text{ MPa}$$

The Disc / Stem Bearing factor is the ratio between actual Bearing height & Required Bearing Height, therefore,

$$S_{bearing} = \frac{B_{act}}{B} = \frac{30}{14.42} = \underline{2.08}$$

Conclusion:

The Calculated bearing length & stress on bearing is lesser than actual Bearing length & maximum allowable stress on bearing, hence the adapted bearing are safe.

Body shell thickness Calculation at Derated Pressure,

Shell thickness,

The maximum bore considered in Body is,

$$D = 192 \text{ mm.}$$

a) Minimum shell thickness as per ASME B16.34, Table 5A,

$$t_{m1} = 8 \text{ mm}$$

b) minimum shell as per ASME B16.34, Appendix A.1.

$$t_{m2} = 0.0163 (D) + 4.7$$

$$= 0.0163 (192) + 4.7$$

$$t_{m2} = \underline{7.8 \text{ mm}}$$

c) Alternative shell thickness Calculation:

Alternative shell thickness Calculation is performed as per ASME B31C Section VIII, Div-1, CG-27.

$$R = \frac{D}{2} = \frac{192}{2} = 96 \text{ mm}$$

Body material considered for calculation is 'NCB'

Working pressure as per Derated pressure is,

$$P_D = 10 \text{ bar} = 1.0 \text{ MPa}$$

$$\text{Thus } P_D = 1.0 \text{ MPa.}$$

Shell Rating pressure = $P = P_w \times 1.5$

$$P = 1 \times 1.5$$

$$P = 1.5 \text{ MPa.}$$

Allowable Stress as per ASME BPVC Section-VI, Part-D.

$$S = 138 \text{ MPa}$$

Joint Efficiency = $E = 1$, for bolted joints,

Shell thickness Calculated by below Equation,

$$t = \frac{P \times R}{2SE + 0.4P} = \frac{1.5 \times 96}{(2 \times 138 \times 1) + (0.4 \times 1.5)}$$

$$t = \underline{0.52 \text{ mm}}$$

Thus, the 2L Adopted Shell thickness is 11mm.

Gear Box Selection:

$$\text{Valve Torque} = T_d = 121.5 \text{ Nm}$$

$$\text{FOS} = 1.25$$

$$\text{Gear Box Model} = \text{QA-150}$$

$$\text{Mechanical Advantage} = 8$$

$$\text{Handwheel Diameter} = 110 \text{ mm}$$

$$\text{Gear Box Input Torque} = \frac{\text{Valve Torque} \times \text{FOS}}{\text{Mechanical Advantages}}$$

$$\text{Gear Box Input Torque} = \frac{121.5 \times 1.25}{8} = 18.98 \approx 19 \text{ Nm}$$

$$\text{Maximum force at Handwheel} = \frac{\text{Gear Box Input Torque} \times 2000}{\text{Hand wheel diameter}}$$

$$= \frac{19 \times 2000}{110}$$

$$= 345.45 \text{ Nm}$$

$$\therefore \text{Maximum force at Handwheel} = 345.45 \text{ Nm}$$

The Selected Gear Box over minimum Handwheel force is lesser than 560 Nm, Hence the Selected Gear box is safe.

Prepared By : Design Engineer

(Raghu)
18-6-19

(Raghu)

Calculations are found : Ok / Not Ok

(Raghu)
19/6/19
Aditya.Raghu

Calculations are reviewed by : Design Engineer

weakest section outside the pressure boundary shall, under torsional load, exceed the strength of the shaft that lies outside the pressure boundary by more than 10%, as per Clause 5.5.1 of API 609,

We have the yield strength of shaft SS304, is 205 MPa,

$$\sigma = 61\% \sigma_y$$

$$\sigma = 1.1 \times 205$$

$$\sigma = 225.5 \text{ MPa}$$

$$\tau = \frac{16 \times T_{\text{stem}} \times 1000}{\pi \times d^3}$$

$$d = \sqrt[3]{\frac{16 \times 121.5 \times 1000}{3.142 \times 225.5}}$$

$$d = 13.99 \text{ mm}$$

Thus the 2L Adapted Stem diameter is d = 19 mm.