API HPHT 6BX Flange Design, Verification & Capabilities: Example use of Methodology

January 22nd, 2018
Summary

1. Determine the suitability of the existing BX gaskets for 20K

2. Use Robert Eichenberg’s formulas from his ASME paper 57-PET-23 “Design Considerations for AWHEM 15,000 psi Flanges” of 1957 and his Journal of Engineering for Industry paper of 1964, to size the Flange and Bolting

3. Use non-linear FEA to optimize and determine capabilities of the flange under combinations of loading
Determine Suitability of Existing BX Gaskets
Determine Suitability of Existing BX Gaskets

• Minimum Material Condition
  – BX Gaskets
  – Seal Grooves

• 316SS Gaskets
  – 30K Yield 75K Tensile
  – True Stress – True Strain
  – Room Temperature Properties

• Friction Factor = 0.1
Determine Suitability of Existing BX Gaskets
Assessing BX Gaskets for 20K WP

Subsequent Gasket Reaction (lbf/in-cir) vs. Pressure Applied (psi)

- BX164 18 3/4"
- BX159 13 5/8"
- BX158 11"
- BX157 9"
- BX156 7 1/16"
- BX155 4 1/16"
- BX154 3 1/16"
- BX153 2 9/16"
- BX152 2 1/16"
- BX151 1 13/16"
Use Eichenberg’s Methods to Size the Flange

1. Hub Thickness (wall of pipe)
2. Pressure Loading
3. Optimal size and number of bolts
4. Raised face diameter
5. Flange thickness
Determine Wall Thickness of Small End of Hub (adjoining pipe)

Eichenberg used Lame’s formula to determine the minimum wall thickness of the hub. The allowable stress at the Working Pressure was half of the yield strength.

\[ g_0 = \frac{B}{2} \left( \sqrt{\frac{s+P}{s-P}} - 1 \right) \]
Wall Thickness of All API 6BX Flanges

Comparison to Eichenberg's Equation (%)

Size of Flange (in)

- 5K
- 10K
- 15K
- 20K
Preliminary Hub Thickness for 18 ¾” 20K

- Yield = 75 000 psi
- Bore = 18.78in
- Thickness $g_o = 7.63in$
- OD = 34.04in
Calculate Pressure Loading

• There are two components to the Gasket Pressure Loading:
  1. Effective Sealing Diameter
  2. Vertical Reaction Force
     – Summed to determine the total pressure loading

• The equations for these pressure forces come from Robert Eichenberg’s papers
Pressure Loading for 18 ¾” 20K

- BX 164 Gasket
- Test Pressure = 30Ksi
- Pressure End Load = 11,842,503 lbf
- Vertical Gasket Reaction = 1,062,815 lbf
- Total Pressure Load = 12,905,318 lbf
Optimal Size and Number of Bolts

• Bolt load must balance the Total Pressure Load
• Criterion is 83% of Yield at Test Pressure
  – Total Pressure Load / Total Root Area
• Minimum even number of bolts
• The Bolt Circle and Flange OD are determined
## Optimal Bolting for 18 ¾” 20K?

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<th>Bolt Diameter</th>
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### Graph
![Graph showing OD vs Bolt Size](attachment:image.png)
Discussion

Taylor Forge Wrench Clearances are 70 years old!

and the table stops at 3”!
# 1 ½" Drive Sockets

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Discussion

• 3 ¼” 8UN Studs
• 5” AF Heavy Hex Nut
• Could use modern literature
  – Hytorc Catalogue
• Bolt Circle 47.7” → 45.5”
• OD 53.9” → 51.7”
Raised Face Diameter

• Eichenberg neglected the small contact area inside the gasket groove

• Eichenberg sized the contact area outside the gasket groove based on a contact pressure of 30 000 psi due to bolt tensions

• Actual 6BX Flange contact pressures vary from 15 000 to 30 000 psi with most around 22 500 psi
Raised Face Bearing Stress

Value chosen for example

Bearing Stress (psi)

Size of Flange (in)
Preliminary Raised Face Ø for 18 ¾” 20K

• Raised Face Diameter = 31.0”
• Bolt Tension = 47.5 Kσi
• Contact Pressure = 23 000 psi
Flange Thickness

- Eichenberg’s equations are based on Taylor Forge and ASME
- They are awkward and iterative
- API HPHT Flange TG have used Roark’s Formulae
- The Flange Thickness can be re-assessed during the non-linear FEA phase
1. Outer edge free, inner edge fixed

\[ y_b = 0, \quad \theta_b = 0, \quad M_{r_3} = 0, \quad Q_a = 0 \]

\[ M_{rb} = -\frac{wa}{C_a} \left( \frac{r_0 C_9}{b} - L_3 \right) \]

\[ Q_b = \frac{w r_0}{b} \]

\[ y_a = -\frac{wa^3}{D} \left[ \frac{C_2}{C_8} \left( \frac{r_0 C_9}{b} - L_3 \right) - \frac{r_0 C_3}{b} + L_3 \right] \]

\[ \theta_a = -\frac{wa^2}{D} \left[ \frac{C_6}{C_8} \left( \frac{r_0 C_9}{b} - L_3 \right) - \frac{r_0 C_5}{b} + L_5 \right] \]

If \( r_0 = a \) (load at outer edge),

\[ y_{\text{max}} = y_a = -\frac{wa^4}{b D} \left[ \frac{C_2 C_9}{C_8} - C_3 \right] \]

\[ M_{\text{max}} = M_{rb} = -\frac{wa^2}{b} \frac{C_9}{C_8} \]

(For numerical values see case 1b after computing the loading at the inner edge.)

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Table 11.2 Formulas for Flat Circular Plates of Constant Thickness (Continued)
Flange Thickness: Membrane + Bending Stress

value chosen for example

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<th>Size of Flange (in)</th>
<th>Membrane + Bending Stress (psi)</th>
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Preliminary 18 ¾” 20K Flange Design

- 22 x 3 ¼” Studs
- 45.5” Bolt Circle
- 51.7” Outside Diameter
- 31.0” Raised Face Diameter
- 12.0” Flange Thickness
- 34.04” Hub Outside Diameter
Preliminary 18 ¾” 20K Flange Design
Preliminary 18 ¾” 20K Flange Design
22 x 3.25" Studs, 95K Bolt, 0 Tension

Pressure (ksi) vs. Bending Moment (ft-kip)

- Red line: Flange Stress Criterion
- Green line: Bolt Stress Criterion
### Optimal Bolting for 18 ¾” 20K?

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#### Graph

- **X-axis**: Bolt Size (in)
- **Y-axis**: OD (in)
- The graph shows the OD (in) decreases as the bolt size increases.
Optional Bolt Circle

- **20 x 3 1/2” Studs** (was 22 x 3 ¼”)
- **44.3” Bolt Circle** (was 45.5”)
- **51.0” Outside Diameter** (was 51.7”)
- **31.0” Raised Face Diameter** (was 31.0”)
- **12.0” Flange Thickness** (was 12.0”)
- **34.04” Hub Outside Diameter** (was 34.04”)

Preliminary 18 ¾” 20K Flange Design

- 22 x 3 ¼” Studs
- 45.5” Bolt Circle
- 51.7” Outside Diameter
- 31.0” Raised Face Diameter ??
- 12.0” Flange Thickness
- 34.04” Hub Outside Diameter
Raised Face $\varnothing$ Increased: 31.0” → 35.0”

22 x 3.25” Studs, 35.0” RF, 95K Bolt, 0 Tension

Pressure (ksi)

Bending Moment (ft-kip)

- Red: Flange Stress Criterion
- Green: Bolt Stress Criterion
Conclusions

• Standard API 6BX Gasket Geometries will be acceptable for 20K
  – 18 ¾” will require an increase in yield strength
• Eichenberg’s methods for sizing 6BX Flanges get us very close to an optimized design