Design & Coding of Weld Neck Flange for Shell and Tube Heat Exchanger

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Abstract : In case of pressure vessel and heat exchanger two ends of shell are joined with the channel or any other mating component by means of flanged coupling or any other but if joined by flanged coupling sometimes it is also called as body flange. For design purpose, we need to check whether it is necessary to design a body flange first. Reason is custom flange will be much more costly as such kind of custom flanges are fabricated according to needs of customer in terms of dimensions, materials , quantity etc. So, if required size is not obtained in various standards like ASME B16.5, ASME B16.47 then we can think about designing a custom flange.

This study has been undertaken to investigate design of body flange in components as mentioned and design procedure is suggested based on various loads like Minimum design bolt load for operating and seating condition, flange design bolt load for gasketing condition, hydrostatic end load as well as moments created because of these loads. Also stresses are analyzed based on which it can be predicted that whether designed flange will sustain applicable loads or not.

***IndexTerms* - Component, pressure vessel, loads , moments.**

I. INTRODUCTION

Heat transfer equipments can be categorized by various ways like type or by its function or by its phase. Various examples are air-preheater, chiller, condenser, cooler, heater, reboiler, vaporizer, superheater etc.. The type shell-and-tube heat exchanger, which needs to selected also depend on various factors and parameters like provision for thermal expansion or differential movement between shell and tubes, design pressure & temperature, minimum design metal temperature, thermal properties of fluids, service conditions also like fouling nature of the fluids. Although all heat exchanger types can perform any function but still correct type selection is crucial. The various shell-and-tube heat exchanger types, and their variations, are described by a letter relating to the particular designation system. All above types and class of heat exchanger has flange component to connect various sections of exchanger, which can be designed or standard dimensions can be adopted from various codes.

In addition to above it should be clear that custom components are always costly compared to standard components and same in case of flange also. Reason is standard flanges are manufactured in large scale by many companies around the world and custom flanges are fabricated according to the customer needs (dimensions, materials, quantity, etc) So, if right and correct size of flange is not obtained in ASME B16.5 (flanges from 1/2″ to 24″) and ASME B16.47 (flanges from 26″ to 60″) then custom flange design becomes mandatory.

II. TYPE OF FLANGE

Various types of flanges are used with piping, pressure vessel or any other industrial equipment. The various types are: welding neck, slip on, blind, lap joint, threaded and socket weld. Various types of flange can be used in various conditions in terms of pressure withstanding capacity or pressure rating, service conditions, possible connection and application. All types of flanges are described as below.

Welding Neck (WN): When we can direct weld flange this type of flange is used and weld which can be used is butt weld.. another application of this flange is in case of cyclic stress as it can, due to the “smooth” transition between the flange and the pipe and do better distribute the stresses. But out of all this flange is more expensive. Also, it can’t be easily adjusted as the other types and their ends need to be chamfered which also increases cost.

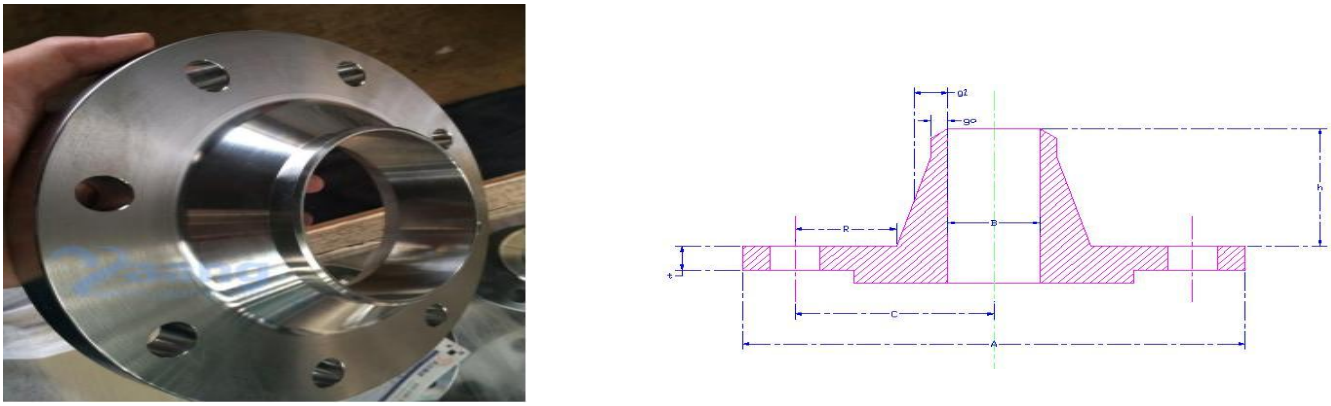

Fig.1 Welding Neck

Slip-On (SO): This type of flange first choice of any designer because of its low cost and easily adjustable, also pipe which will joined to flange by welding doesn't need to be cut with great accuracy. It also has small hub which helps in the stress distribution but overall this flange is weak compared to the welding neck. In terms of strength its strength [1] compared to the welding neck is only 2/3 and in case of fatigue loading or cyclic stresses it is only 1/3. So, In case of high residual stresses or cyclic stresses and stress concentrations due to the discontinuity (flange/pipe) which can lead to failure if this flange is used so it is not advisable.

Threaded (TH): This type of flange is not considered more reliable and it has limited use in industry, it is used only for pipes p that can't be welded or non-metallic pipes (plastic). Also it is not recommended to use this type of flange with severe toxic/dangerous fluids, high temperature, high pressure conditions etc. Also, there are chances of leakage as well as stress concentration.

Fig.2 Threaded (TH)

Socket Weld (SW): This type of flange is quite similar to slip-on but it has a socket to pipe due o which it doesn't require inner weld. It is usually used in small-size pipes (up to 1 1/2″). This type of flange is better choice for fatigue loading as socket weld has fatigue strength greater than slip on flange.

Lap Joint (LJ): Function of this flange is very much different to other joint as this type of flange is used for the purpose of blocking up fluid. This type of flange have advantageous over other in many cases like less expensive materials can be used due to lack of contact with fluid also it can be reused (because it's not welded). The downside of this type of flange is that it doesn't have a good fatigue resistance (around 1/10 of the welding neck) not being used in severe conditions systems.

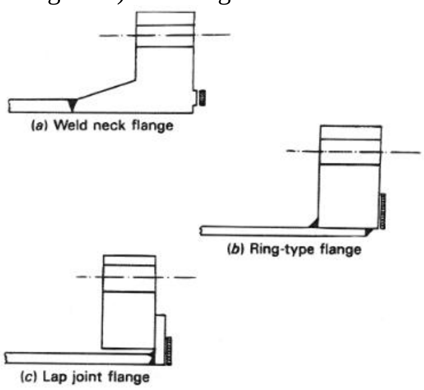

Fig.3 Lap Joint (LJ)

Blind (BD): this type of flange is used to block the flow, so its most common in equipments, which is used in man way nozzles, rather than pipelines and other applications.

III. FLANGE DESIGN

The two ends of shell are joined with the channel by means of flanged coupling known as body flanged. Standard flanged can also be utilized but it will lead to more wastage of material. For that purpose the design of flanged is been done as per the ASME SECTION VIII, DIV.1, APPENDIX-2[32]. The different parameters are listed as below.

While designing Flange some assumptions are done as listed below.

1. Materials of all of the elements are assumed to be homogenous and remain elastic under the loading conditions assumed in the design & effect of the bolt holes in the flanges is neglected.
2. Axial symmetry is used to reduce the problem to consideration of the conditions on a single flange hub and shell cross section, neglecting variations due to location of bolts.
3. All loading applied to the flange is reduced to a `couple' involving a pair of equilibrating loads located at the extremities of the flange.
4. Stretching of the middle surface of the flange ring due to the applied couple is negligible.
5. Displacements of the joint are small such that the theorems of superposition are valid.
6. When a ring moment is applied to the flange, the point of connection between the flange and the hub is assumed to have zero radial displacement.
7. Hub and shell are assumed to act as thin shells.
8. The inside bore of the hub and shell is used in the shell theory analysis instead of the mean thickness diameter & Effects due to interaction of elements are neglected.

Design specification for flange

For designing flange for different conditions, first some of the parameters needs to be fixed.

Flange inner diameter (Df) = 600 mm

Flange corroded inner diameter (Dfc) = ID+ 2C= 606 mm

Flange outer diameter (Dfo) = 812 mm

Flange thickness(c) [17] = (1.88-0.0629) Inch. = (47.75-1.6) = 46.152 mm

Thickness of hub at small end = go=10 mm

Corroded thickness of hub at small end= gocorr -c = 7 mm

Thickness of hub at large end = g1= 20 mm

Corroded thickness of hub at large end= g1corr= g1- c= 17 mm

Length of hub = 30 mm

Bolt circle diameter(C) = 708 mm

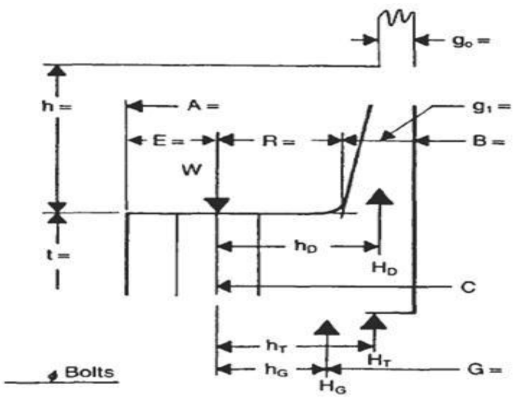

Fig 4: Flange Parameters

Required thickness due to internal pressure at small end of hub:

Considering hub as a cylinder, the minimum required thickness and maximum allowable thickness is to be calculated from equation (1) and (2).

$$t = \frac{PR}{SE - 0.6P} + C.A \tag{1}$$

$$P = \frac{SEt}{R + 0.6t} \tag{2}$$

Following the same terminology as shell, for internal pressure of 1.41 N/mm^2, the mini-mum required thickness can be obtained as 5.22 mm and for this thickness the maximum allowable working pressure should not exceed 4.43 N/ mm^2

Loading condition for flange:

Stresses acting on the flange depend upon the basic flange loads and bolt loads. The flange should be of sufficient strength so that it resists the moments produced due this loads. Below mentioned equations determine these loads. The various loads acting on flange is described below:

Minimum required bolt load for operating condition.

Minimum required bolt load for gasket seating.

Flange design bolt load for gasket seating.

Minimum required bolt load for operating condition (Wm1):
Thickness of flange should be provided in such a way that it prevent leakage and also gives proper sealing ability. It should be able to resist bolting forces.

$$Wm1 = H + Hp = 0.785G^2P + 2bpGmP \qquad (3)$$

Equation (3) contains hydrostatic end force and total joint contact compressive force. From above equation bolt load is calculated as 43188.461 kgf

Minimum required bolt load for gasket seating (Wm2):
Gasket should not be displaced at atmospheric condition for that sufficient amount of load on gasket is required to keep gasket in position. Equation for bolting load is given as below.

$$Wm2 = pbGy \qquad (4)$$

Minimum bolt load can be calculated as 110938.06 kgf.

Required bolt area (Am):
To withstand bolting load for operating condition bolt cross-sectional area of bolt at root of thread or section of least diameter under stress is required to be calculated.

$$Am = \max\left(\frac{Wm\,1}{sb}, \frac{Wm\,2}{sa}\right) \qquad (5)$$

Required bolt area is maximum of above two terms and can be obtained as 63.11 cm^2.

Bolt area (Ab):
It is the bolt cross-sectional area of bolt using the root diameter of the thread or least diameter of unthreaded position if less.
TEMA Section-9, Table D-5 is been followed for selecting the bolt, based on Am and bolt Spacing which is more than required bolt area.

Flange design bolt load for Gasket seating (W):
Bolt load can be calculated by using size and number of bolts as follows:

$$W = \frac{(Am + Ab)Sa}{2} \qquad (6)$$

Flange design bolt load from above equation is 112927.8 kgf.

Calculation of moments due to bolt load
Flange should be verified for different bending moments and bending moment is multiplication force to its perpendicular distances.Equivalent forces are found as below.

1. Hydrostatic end load.
2. Pressure force on flange face.
3. Gasket load.

Above mentioned loads are calculated for both operating and seating conditions and maximum bending moments can be calculated by considering maximum loads.

Hydrostatic end load at flange ID (Hd):
Factor responsible for this load is fluid pressure, it is propotional to pressure of fluid and diameter of flange. The hydrostatic end load appendix-2 in ASME SECTION VII, DIV. 1 is used.

$$Hd = \pi * Dfc * \frac{p}{4} \qquad (7)$$

Hydrostatic end load can be calculated as 30457.8 kgf.

Pressure force on flange Face (Ht):
Moment produced due to resultant force is responsible for pressure force on and it can be calculated as,

$$Ht = H - Hd \qquad (8)$$

In above equation 1st term is total hydrostatic end force and second is hydrostatic end force on area inside of flange. Pressure force on flange face is the subtraction of these two, obtained as 2746.50 kgf.

Gasket load for operating condition (HG):
Between two flanges gasket is used. It is used for avoiding direct contact and to prevent leakage but it will exertGasket is present a force on the flanges.

$$HG = Wm1\text{-}H \qquad (9)$$

Gasket load for operating condition is calculated as 9984.38 kgf.
Moment arm length needs to be evaluate after finding forces acting on flanges, flange is designed for the maximum moment, to avoid the failure of flange.
The moment arm length or equivalent perpendicular distances are:

1. Distance to Gasket load reaction (hg)
2. Distance to face pressure reaction (ht)
3. Distance to end pressure reaction (hd)

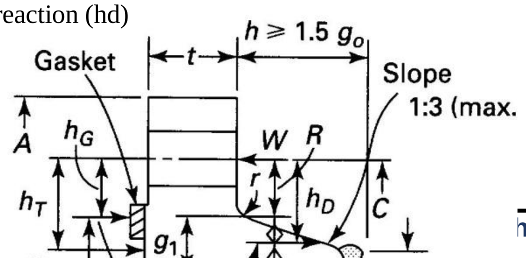

Fig. 5: Location of moment arm distances [2]

Distance to Gasket Load reaction (hg):

It is radial distance from gasket load reaction to the bolt circle as seen from fig. (6). Although the location of gasket load reaction could be different based on effective gasket seating width as shown.

$$\mathrm{hg} = \frac{C-G}{2} \tag{10}$$

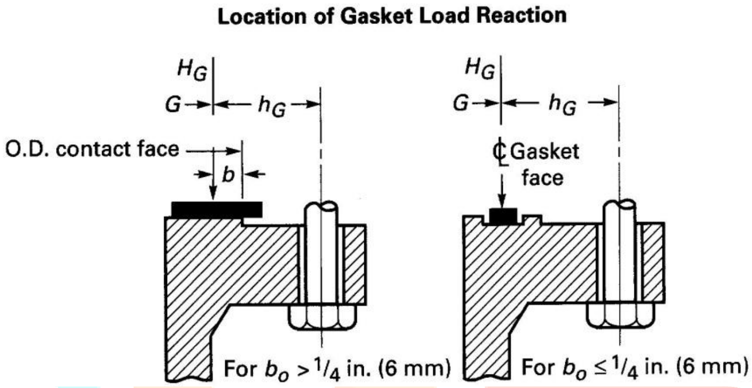

Fig. 6: Location of gasket load reaction [2]

Distance to Face Pressure Reaction (ht):

It is the radial distance from the bolt circle to the circle on which Ht acts.

$$\mathrm{ht} = \frac{R + g1.corr + hg}{2} \tag{11}$$

Distance to End Pressure Reaction (hd):

It is the radial distance from the bolt circle to the circle on which Hd acts.

$$\mathrm{hd} = R + \frac{g1.corr}{2} \tag{12}$$

From above equations distance to gasket load reaction, distance to face pressure reaction, distance to end pressure reaction can be obtained as 37.42 mm, 44.11 mm and 42.4 mm respectively. Due to this distances, moment obtained are as shown in table (4.14).

Table 1: Different Moment on flange

Sr. No.	**Parameter**	**Obtained Value (kgf.m)**
1	Face pressure moment (Mt)	121
2	Gasket load moment (Mg)	374
3	Gasket seating moment (Mgs)	4226

Stresses in Flange

Due to above Bending moments & internal pressure, following stresses are developed in flange.

1. Longitudinal Hub Stress.
2. Radial Flange Stress.
3. Tangential Flange Stress.

These three stresses taken in order to determine the acceptability of flange for operating as well as seating conditions and its is limited by,

SH < SHO or 1.5 SF O
or SH < SHA or 1.5 SF A
R & ST < SF O or 1.5 SF A
0.5(SH + SR) < SF O or SF A

0.5(SH + ST) < SF O or SF A

where, SF O, SF A, SHO, SHA , are the allowable design stresses for the flange and hub materials at the operating and ambient temperature respectively.

It can be seen that as the allowable design stress is usually about 2/3 of the material yield, this allows the hub to be stressed up to the material yield point, allowing yielding in the hub during hydro test. The flange stress limits are set to a level which should keep the main flange bodies elastic under all conditions, providing the joint is not over tightened during bolting-up. The latter two stress limits are the application of a Tresca type criterion to the bi-directional stresses at the interface between the flange and hub.

Table 2: Stress Result for Flange

Sr. No.	Stress Category	Magnitude (kgf/cm^2)
1	Longitudinal Hub Stress, Operating [SHo]	728.86
2	Longitudinal Hub Stress, Seating [SHa]	1724.51
3	Radial Flange Stress, Operating [SRo]	49.27
4	Radial Flange Stress, Seating [SRa]	116.57
5	Tangential Flange Stress, Operating [STo]:	369.64
6	Tangential Flange Stress, Seating [STa]:	874.58
7	Average Flange Stress, Operating [SAo]:	549.25
8	Average Flange Stress, Seating [SAa]:	1299.55
9	Bolt Stress, Operating [BSo]:	660.57
10	Bolt Stress, Seating [BSa]:	1696.81

IV. MAT LAB CODING FOR FLANGE

```
%% Flange %%
clc;    clear all;  close all;  commandwindow;
%% Required Thickness t
P1 = input ('Enter pressure P :: ');
R = input ('Enter radius R :: ');
S = input ('Enter stress S :: ');
E = input ('Enter joint efficiency E :: ');
CA = input ('Enter corrosion allowance A :: ');
t = ((P1*R)/((S*E) - ((0.6)*(P1)))+CA);
fprintf ('\nRequired Thickness is %0.2f',t);
%% Pmax
Pmax = ((S*E*t)/(R+(0.6*t)));
fprintf ('\nPmax is %0.2f\n\n',Pmax);
%% Min Req Bolt Load for Operating Condition
P = input ('Enter pressure P :: ');
G = input ('Enter Diameter at gasket load reaction G :: ');
b = input ('Enter effective gasket seating stress b :: ');
M = input ('Enter gasket factor M :: ');
H = (0.785*G^2*P);
Hp = (2*b*pi*G*M*P);
Wm1 = (H + Hp);
fprintf ('\nWm1 is %0.2f\n\n',Wm1);
%% Min Req Bolt Load for Gasket Seating
y = input ('Enter gasket contact factor y :: ');
Wm2 = (pi*b*G*y);
 fprintf ('\nWm2 is %0.2f\n\n',Wm2);
%% Reqd Bolt Area
Sa = input ('Enter allowable bolt stress at design temperature Sa :: ');
 Sb = input ('Enter allowable bolt stress at design temperature Sb :: ');
Am = max ((Wm1/Sb),(Wm2/Sa));
fprintf ('\nAm is %0.2f\n\n',Am);
%% Flange Design Bolt Load
Ab = input ('Enter area of bolt Ab :: ');
W = (((Am + Ab)*Sa)/2);
```

```
fprintf ('\nW is %0.2f\n\n',W);
%% Hydrostatic End Load at Flange
Dfc = input ('Enter corrugatated flange diameter Dfc :: ');
Hd = ((pi*(Dfc^2)*P)/4);
fprintf ('\nHd is %0.2f\n',Hd);
%% Pressure Force on Flange Face Ht
Ht = (H - Hd);
fprintf ('Ht is %0.2f\n\n',Ht);
%% Gasket Load for Operating Condition
HG = (Wm1 - H);
fprintf ('\nHG is %0.2f\n\n',HG);radial distance from bolt circle to point of intersection R1 :: ');
ht = ((R1 + g1corr + hg)/2);
fprintf ('\nht is %0.2f\n',ht);
%% Distance to End Pressure Reaction
hd = (R1 + (g1corr/2));
fprintf ('hd is %0.2f\n\n',hd);
%% Longitudinal Hub Stress (Operating)
f = input ('Enter integral flange f :: ');
M = input ('Enter moment M :: ');
Bcor = input ('Enter corrugated diameter Bcor :: ');
L = input ('Enter gasket factor L :: ');
g = input ('Enter thinkness of hub g :: ');
%% Longitudinal Hub Stress (Operating)
f = input ('Enter integral flange f :: ');
M = input ('Enter moment M :: ');
Bcor = input ('Enter corrugated diameter Bcor :: ');
L = input ('Enter gasket factor L :: ');
g = input ('Enter thinkness of hub g :: ');
SHO = ((f*M*Bcor)/(L*(g^2)));
fprintf ('\nSHO is %0.2f\n\n',SHO);
%%
f = input ('Enter integral flange f :: ');
M = input ('Enter moment M :: ');
Bcor = input ('Enter corrugated diameter Bcor :: ');
% L = input ('Enter gasket factor L :: ');
% g = input ('Enter thickness of hub g :: ');
SHd = ((f*M*Bcor)/(L*(g^2)));
fprintf ('\nSHd is %0.2f\n\n',SHd);
%%
Beta = input ('Enter gasket constant Beta :: ');
M = input ('Enter moment M :: ');
Df = input ('Enter corrugated diameter Df :: ');
% L = input ('Enter gasket factor L :: ');
t = input ('Enter flange thickness t :: ');
SRo = ((Beta*M)/(Df*L*(t^2)));
fprintf ('\nSRO is %0.2f\n\n',SRO);
%%
% Beta = input ('Enter gasket constant Beta :: ');
M = input ('Enter moment M :: ');
% Df = input ('Enter corrugated diameter Df :: ');
% L = input ('Enter gasket factor L :: ');
% t = input ('Enter flange thickness t :: ');
SRd = ((Beta*M)/(Df*L*(t^2)));
fprintf ('\nSRd is %0.2f\n\n',SRd);
  %%
Y = input ('Enter gasket factor Y :: ');
M = input ('Enter moment M :: ');
t = input ('Enter gasket thickness t :: ');
Dfcorr = input ('Enter corrugated diameter Dfcorr :: ');
Z = input ('Enter gasket factor Z :: ');
STo = (((Y*M)/((t^2)*Dfcorr))-(Z*SRO));
STa = (((Y*M)/((t^2)*Dfcorr))-(Z*SRA));
BSo = (Wm1/Ab);
BSa = (Wm2/Ab);
SAo = (SHO + max (SRO , STO))/(2);
SAa = (SHA + max (SRA , STA))/(2);
```

```
fprintf ('\nSTo is %0.2f\n\n',STo);
fprintf ('\nSTa is %0.2f\n\n',STa);
fprintf ('\BSo is %0.2f\n\n',BSo);
fprintf ('\BSa is %0.2f\n\n',BSa);
fprintf ('\nSAo is %0.2f\n\n',SAo);
fprintf ('\nSAa is %0.2f\n\n',SAa);
```

V. MAT LAB OUTPUT FOR FLANGE

Enter pressure P :: 1.02
Enter radius R :: 300
Enter stress S :: 138
Enter joint efficiency E :: 0.85
Enter corrosion allowance A :: 3

Required Thickness is 5.62mm
Pmax is 2.17N/mm^2

Enter pressure P :: 10.54
Enter Diameter at gasket load reaction G :: 633
Enter effective gasket seating stress b :: 6.42
Enter gasket factor M :: 3

Wm1 is 43188.46 kgf

Enter gasket contact factor y :: 703.07

Wm2 is 110938.06 kgf

Enter allowable bolt stress at operating temperature Sa :: 1757
Enter allowable bolt stress at design temperature Sb :: 1757

Am is 63.116 cm^2

Enter area of bolt Ab :: 65.38

W is 112927.84 kgf

Enter corrugated flange diameter Dfc :: 606

Hd is 30457.58 kgf
Ht is 2746.5 kgf

Enter radial distance R1 :: 34
Enter corrugated diameter g1corr :: 16.8

HG is 9934.38 kgf
ht is 40.11mm

Enter radial distance R1 :: 34

hd is 42.40mm

Enter integral flange f :: 2.0996
Enter moment M :: 1786
Enter corrugated diameter Bcor :: 606
Enter gasket factor L :: 3
Enter thinkness of hub g :: 16.8

SHo is 728.86 kgf

Enter integral flange f :: 2
Enter moment M :: 4226
Enter corrugated diameter Bcor :: 606

SHa is 1724.5 kgf/cm^2

Enter gasket constant Beta :: 2.11
Enter moment M :: 1786
Enter corrugated diameter Df :: 606
Enter gasket factor L :: 3
Enter flange thickness t :: 64.8

SRo is 49.27 kgf/cm^2

Enter moment M :: 2.1
Enter corrugated diameter Df :: 4226
Enter gasket factor L :: 606
Enter flange thickness t :: 64.8

SRa is 116.57 kgf/cm2

Enter gasket factor Y :: 8.2653
Enter moment M :: 4226
Enter gasket thickness t :: 64.8
Enter corrugated diameter Dfcorr :: 606
Enter gasket factor Z :: 4.266

STo is 369.64 kgf/cm^2
STa is 874.58 kgf/cm^2
BSo is 660.57 kgf/cm^2
BSa is 1696.81 kgf/cm^2
SAo is 549.25 kgf/cm^2
SAa is 1299.55 kgf/cm^2

VI. Conclusion

Above design procedure based on ASME appendix 2 can be used for design of such kind of custom flange when standard flange is not utilized. Also generalized program for matllab is developed which can be directly used for design of such flange. In this case weld neck flange is selected and as the results of stresses obtained by ASME codes and MATLAB programming shows that weld neck flange is safe for given operating and seating condition and as the results is same for both ASME and MATLAB so that MATLAB programming can be used as a generalized solution for design of weld neck flange.

VII. References

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