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Evaluation of the flange rigidity index J - versus the k - factor approach for large diameter integral type shell girth flanges

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ABSTRACT: *Large diameter shell girth (body) flanges subject to low or moderate pressures are prone to extensive rotation. The gasket serves as a pivot point for the rotation. It is even possible for leakage to increase with increased bolt loading as the flange rotates off the gasket. In order to limit potential flange leakage, a flange rigidity criterion has been developed to set limits on flange rotation. The current ASME Code includes such a flange rigidity criterion and it is claimed that it has been proven through extensive user experience for a wide variety of joint designs and service conditions. As in the USA, the flange calculation methods in the UK and EU are also based on the Taylor Forge analysis. However, no rigidity criterion is found in either the UK or the EU flange calculation standards. In view of this anomaly, a "k - Factor" was introduced with the intention of essentially lowering the allowable stresses in a proportional manner thus compensating this inconsistency. The sections below consider both the impact of the flange rigidity criterion and the flange stress reduction factor approach for large diameter integral body flanges. In addition, it has been endeavored to give the designer a better understanding of the meaning of reduced allowable flange stresses and the use of a rigidity criterion for the flange. Appropriate consideration of flange stresses and flange rigidity will ultimately lead to a prudent flange design. The aim of this article is to give an important impetus to this issue.*
KEYWORDS: *rotation, leakage, flange rigidity, flange stresses, stress reduction factor.*

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I. INTRODUCTION

Flanges that have been designed based on allowable stress limits alone may not be sufficiently rigid to control leakage. To address this problem , a rigidity criteria has been included which depends upon the type of flange. The rigidity criterion became mandatory since the 2007 issue of ASME BPVC Section VIII - Division 1[1] (Table 2-14). A somewhat more extensive version of the flange rigidity criterion can be found in ASME BPVC Section VIII - Division 2 [2] (Table 4.16.10). Adherence to the flange rigidity criterion and associated rigidity index tend to limit flange rotation within acceptable limits and to prevent potential leakage. The ASME flange calculation method is based on the Taylor Forge [7] analysis and as such has been also the basis for the traditional flange calculation methods as included in PD 5500 [3] and EN 13445-3 [4]. However, it appears that the flange rigidity criterion added to the ASME code has not been adopted by PD 5500 and EN 13445. The responsible code committees of both standards have opted for an alternative approach instead of the rigidity criterion, namely by incorporating a so-called stress reduction factor - k. This means that within the validity range of the k - factors the flange design stresses should be divided by a factor - k, which in fact leads to lower allowable design stresses and hence more rigid flanges. The flange rigidity criterion has no limitation in terms of flange diameters, while the flange stress reduction factor is exclusively intended for flanges with a relatively large diameter in excess of 1000 mm. Moreover, the article will emphasize the effects of the use of a rigidity criterion - J and the application of a stress reduction factor - k.

II. SCOPE OF RESEARCH

This article is limited to research that focuses on custom designed body flanges with a large diameter that falls outside the range of standard flanges. The research question focuses on the k-factor approach as

applied in PD 5500 and EN 13445 and whether this leads to equivalent results (in terms of flange thickness) compared to the J-index approach according to ASME.
The flanges selected for this purpose are depicted in Table 2. A typical drawing of the analyzed flanges is shown in the appendix which also contains a table with the main dimensions of the relevant flanges.

III. BACKGROUND FLANGE RIGIDITY CRITERION

The background of the flange rigidity criterion stems from the analysis developed by Waters et al.[5] which included a calculation of the hub ring junction rotation of a hubbed integral flange. The equation developed for the angular rotation of an integral flange is:

$$\boxed{\theta = \frac{52.4\,V\,M_0}{L\,E\,g_0^2\,h_0}\ \text{(degrees)}}$$

This equation has been successfully applied by many designers to limit flange rotation to 0.3° for integral flanges. This equation, in modified form has been added to the ASME BPVC Section VIII, Division 1 and 2 [1] [2], where the flange rotation was changed to a rigidity index (J). The ultimate goal of the ASME J-index is to control the angular rotation of flanges within acceptable limits in order to avoid flange leakage.

IV. FLANGE RIGIDITY ASPECTS AND ANALYSIS

Table 1: Overview of flange rigidity index factor "J" and stress reduction factor "k"

Provision	Equation
Integral type shell girth flange (all diameters) (Symbols conforming ASME Code)	$J = \frac{52.4\,V\,M_0}{L\,E\,g_0^2\,K_I\,h_0} \leq 1.0$
Inside diameter of flange ≤ 1000 mm Inside diameter of flange (mm): 1000 > B or D < 2000 Inside diameter of flange ≥ 2000 mm (Symbols conforming PD 5500 and / or EN 13445-3)	$k = 1.0$ $k = \frac{2}{3}\left(1 + \frac{B \text{ or } D}{2000}\right)$ $k = 1.333$

Table 2: Data of selected shell girth flanges

Flange Identification	Flange # 1000	Flange # 1600	Flange # 2000	Flange # NPS 60
Design pressure (bar)	20	27.5	10	21.5
Hydrostatic test pressure (bar)	As per code	As per code	As per code	As per code
Design temperature (°C)	250	250	250	250
Corrosion allowance (mm)	0	0	0	0
Thickness tolerance (mm) (%)	0	0	0	0
Geometric flange data				
Outside diameter of flange [A] (mm)	1160	1825	2175	1675
Inside diameter of flange [B] (mm)	980	1560	1980	1494
Bolt-circle diameter [C] (mm)	1100	1725	2110	1615
Thickness of hub at large end [g_1] (mm)	20	30	20	22
Thickness of hub at small end [g_0] (mm)	10	20	10	15
Hub length [h] (mm)	30	30	30	22.5
Flange thickness [t] (mm)	As per code	As per code	As per code	As per code
Flange material	**A 105**	**A 105**	**A 105**	**A 105**
Gasket data: (Semi-Confined)	**Solid flat metal**	**Solid flat metal**	**Solid flat metal**	**Spiral Wound**
Material:	Monel	Monel	Monel	Graphite filled
Outside diameter gasket (mm)	1050	1660	2050	1570 - 3 (bead) = 1567
Inside diameter gasket (mm)	1018	1628	2018	1519.2
Gasket thickness (mm)	1.6	1.6	1.6	4.5
Bolting data				
Size of stud bolts (inch)	1 1/8" - 8UN	1 1/2" - 8UN	1 1/4" - 8UN	1 1/8" - 8UN
Number of stud bolts (-)	48	60	68	64
Material of stud bolts	A 193 B7	A 193 B7	A 193 B7	A 193 B7

Adjoining shell data				
Outside diameter (mm)	1000	1600	2000	1524
Shell thickness (mm)	10	20	10	15
Material Grade	A 515 - 65	A 515 - 65	A 515 - 65	A 515 - 65

Table 3: Calculation results

Design code	ASME VIII - 1 Appendix 2	PD 5500 Clause 3.8	EN 13445 - 3 Clause 11
Required flange thickness: (mm) Flange # 1000	99	97	97
Flange thickness in conjunction with a flange rigidity index : J = 1.0	≈ 96.0		
Required flange thickness: (mm) Flange # 1600	157	179	173
Flange thickness in conjunction with a flange rigidity index : J = 1.0	≈ 150		
Required flange thickness: (mm) Flange # 2000	160	159	163
Flange thickness in conjunction with a flange rigidity index : J = 1.0	≈ 160		
Required flange thickness: (mm) Flange # NPS 60	134	149	143
Flange thickness in conjunction with a flange rigidity index : J = 1.0	≈ 134		

Note: J-index calculated according to formula from Table 2-14 of ASME VIII-1 for determining condition

Table 4: Overview of stress reduction factor - k

Flange identification	Flange # 1000	Flange # 1600	Flange # 2000	Flange # NPS 60
B or D (mm)	**980**	**1560**	**1980**	**1494**
k - factor	**1.0**	**1.1876**	**1.3267**	**1.1647**

Table 5: Overview basic allowable stresses "S" (MPa)

Material	A 105 Flange	A 515 Grade 65 Shell barrel	A 193 Grade B7 Stud bolts	Design Code / Standard
Temperature (°C)	"S" (MPa)	"S" (MPa)	"S" (MPa)	
20	138.0	128.0	172.0	ASME VIII-1
250	136.0	128.0	172.0	ASME VIII-1
20	166.67	160.0	215.0	EN 13445
250	136.0	132.0	204.67	EN 13445
20	165.0	161.0	193.0	PD 5500
250	127.0	124.0	158.0	PD 5500

Table 6: Overview k-factor corrected allowable stresses

Flange Identification	Code Material Temperature		Flange # 1000 Allowable Stress (MPa)		Flange # 1600 Allowable Stress (MPa)		Flange # 2000 Allowable Stress (MPa)		Flange # NPS 60 Allowable Stress (MPa)	
Design Code	ASME VIII-1									
Material	A105									
Temperature [°C]	20	250	138	136	138	136	138	136	138	136
Material	A515 - 65									
Temperature [°C]	20	250	128	128	128	128	128	128	128	128
Material	A193 - B7									
Temperature [°C]	20	250	172	172	172	172	172	172	172	172
Design Code	PD 5500									
Material	A105									
Temperature [°C]	20	250	165	127	138.94	106.94	124.37	95.73	141.67	109.04
Material	A515 - 65									
Temperature [°C]	20	250	161	124	135.57	104.41	121.35	93.46	138.23	106.47
Material	A193 - B7									
Temperature [°C]	20	250	193	158	193	158	193	158	193	158

Design Code	EN 13445									
Material	A105									
Temperature [°C]	20	250	166.67	136	140.34	114.52	125.63	102.51	143.10	116.77
Material	A515 - 65									
Temperature [°C]	20	250	160	132	134.73	111.15	120.60	99.49	137.37	113.33
Material	A193 - B7									
Temperature [°C]	20	250	215	204.67	215	204.67	215	204.67	215	204.67

Note: ASME allowable stresses are not corrected. Bolt stresses (A193 - B7) do not need to be corrected.

Graph 1 displays visually the differences in calculated flange thicknesses

GRAPH 1: REQUIRED FLANGE THICKNESSES AS PER CODE

Table 7: ASME flange stress summary including stress utilization factor (U)

Flange identification	t (mm)	S_H (MPa)	S_R (MPa)	S_T (MPa)	$(S_H + S_R)$ /2 (MPa)	$(S_H + S_T)$ /2 (MPa)
#1000 (Oper)	99	159.979	6.831	81.386	83.405	120.682
#1000 (Seating)	99	180.968	7.727	92.063	94.347	136.515
U (Oper)		0.784	0.050	0.598	0.613	0.887
U (Seating)		0.874	0.056	0.667	0.684	0.989
#1600 (Oper)	157	183.838	8.731	85.900	96.284	134.869
#1600 (Seating)	157	136.075	6.462	63.583	71.269	99.829
U (Oper)		0.901	0.064	0.632	0.708	0.992
U (Seating)		0.657	0.047	0.461	0.516	0.723
#2000 (Oper)	160	114.361	1.637	63.194	57.999	88.778
#2000 (Seating)	160	147.374	2.109	81.436	74.741	114.405
U (Oper)		0.561	0.012	0.465	0.426	0.653
U (Seating)		0.712	0.015	0.590	0.542	0.829 *
#NPS 60 (Oper)	134	178.328	6.290	82.440	92.309	130.384
#NPS 60 (Seating)	134	130.676	4.609	60.411	67.642	95.543
U (Oper)		0.874	0.046	0.606	0.679	0.959 *
U (Seating)		0.631	0.033	0.438	0.490	0.692

Note: The stress utilization factor "U" can be defined as the ratio of the actual stress and the allowable stress.
* Both the stress - and the rigidity criterion are decisive.

Table 8: PD 5500 flange stress summary including stress utilization factor (U)

Flange identification	t (mm)	$k \cdot S_H$ (MPa)	$k \cdot S_R$ (MPa)	$k \cdot S_T$ (MPa)	$k (S_H + S_R)/2$ (MPa)	$k (S_H + S_T)/2$ (MPa)
#1000 (Oper)	97	167.72	7.39	83.41	87.55	125.56
#1000 (Seating)	97	207.96	9.16	103.42	108.56	155.69
#1000 (Test)	97	272.54	12.01	135.54	142.27	204.04
U (Oper)		0.8804	0.0582	0.6568	0.6894	0.9887
U (Seating)		0.8402	0.0555	0.6279	0.6458	0.9436
U (Test)		0.7708	0.3539	0.5750	0.6036	0.8656
#1600 (Oper)	179	164.19	6.44	88.44	85.31	126.32
#1600 (Seating)	179	141.67	5.56	76.31	73.61	108.99
#1600 (Test)	179	266.88	10.47	143.76	138.68	205.32
U (Oper)		0.8619	0.0507	0.6964	0.6717	0.9946
U (Seating)		0.5724	0.0337	0.6009	0.2974	0.6605
U (Test)		0.7548	0.0444	0.6100	0.3922	0.8711
#2000 (Oper)	159	154.35	2.23	84.69	78.29	119.52
#2000 (Seating)	159	212.09	3.07	116.38	107.58	164.23
#2000 (Test)	159	250.82	3.63	137.63	127.22	194.22
U (Oper)		0.8102	0.0176	0.6669	0.6165	0.9411
U (Seating)		0.8569	0.0186	0.7053	0.6520	0.9953
U (Test)		0.7094	0.0154	0.5839	0.5397	0.8240
#NPS 60 (Oper)	149	165.39	5.00	85.96	85.19	125.68
#NPS 60 (Seating)	149	141.89	4.29	73.75	73.09	107.82
#NPS 60 (Test)	149	269.24	8.13	139.94	138.69	204.59
U (Oper)		0.8682	0.0394	0.6769	0.6708	0.9896
U (Seating)		0.5733	0.0260	0.4470	0.4430	0.6535
U (Test)		0.7615	0.0345	0.5937	0.3101	0.8680

Note: The stress utilization factor "U" can be defined as the ratio of the actual stress times k and the allowable stress

(*) J-index calculated according to formula from Table 2-14 of ASME VIII-1

Table 9: EN 13445 flange stress summary including stress utilization factor (U)

Flange identification	t (mm)	$k \cdot S_H$ (MPa)	$k \cdot S_R$ (MPa)	$k \cdot S_T$ (MPa)	$k (S_H + S_R)/2$ (MPa)	$k (S_H + S_T)/2$ (MPa)
#1000 (Oper)	97	167.72	7.39	83.41	87.55	125.56
#1000 (Seating)	97	221.43	9.75	110.12	115.59	165.78
#1000 (Test)	97	257.45	11.34	128.03	134.39	192.74
U (Oper)		0.8222	0.0543	0.6133	0.6438	0.9232
U (Seating)		0.8857	0.0585	0.6607	0.6935	0.9947
U (Test)		0.7209	0.0476	0.5377	0.5644	0.8095
#1600 (Oper)	173	177.28	7.31	92.03	92.29	134.66
#1600 (Seating)	173	153.22	6.31	79.54	79.77	116.38
#1600 (Test)	173	271.72	11.20	141.06	141.46	206.39
U (Oper)		0.8690	0.0538	0.6767	0.6786	0.9901
U (Seating)		0.6129	0.0379	0.4772	0.4786	0.6983
U (Test)		0.7608	0.0470	0.5924	0.5941	0.8668
#2000 (Oper)	163	144.95	2.02	81.66	73.49	113.31
#2000 (Seating)	163	211.59	2.95	119.21	107.27	165.40
#2000 (Test)	163	222.50	3.10	125.35	112.80	173.93
U (Oper)		0.7105	0.0149	0.6004	0.5404	0.8332
U (Seating)		0.8464	0.0177	0.7152	0.6436	0.9924
U (Test)		0.6230	0.0130	0.5265	0.4735	0.7305
#NPS 60 (Oper)	143	181.09	5.81	89.94	93.45	135.52
#NPS 60 (Seating)	143	154.01	4.94	76.49	79.47	115.25
#NPS 60 (Test)	143	277.96	8.92	138.05	143.44	208.01
U (Oper)		0.8877	0.0427	0.6613	0.6871	0.9965
U (Seating)		0.6160	0.0296	0.4589	0.4768	0.6915
U (Test)		0.7783	0.0375	0.5798	0.6024	0.8736

Note: The stress utilization factor "U" can be defined as the ratio of the actual stress times k and the allowable stress

(*) J-index calculated according to formula from Table 2-14 of ASME VIII-1.

The flange stresses as tabulated in tables 8 and 9 must satisfy the following criteria:

$k \cdot S_H \leq 1.5 \min [S ; S_n]$; $k \cdot S_R \leq S$; $k \cdot S_T \leq S$; $0.5\, k (S_H + S_R) \leq S$; $0.5\, k (S_H + S_T) \leq S$

The flange stresses as shown in Table 7 must also meet the above criteria, however with applying k = 1.0 , and the longitudinal hub stress S_H not greater than the smaller of 1.5 S or 2.5 S_n for integral type flanges with hub welded to the neck (shell barrel). For all selected flanges 2.5 S_n > 1.5 S.
Although PD 5500 divides the flange design stresses by the design stress factor k, which in fact results in a reduction of the allowable flange stress, this basically corresponds to the assessment as described above according to EN 13445. That is why comparable approaches have been chosen in Tables 8 and 9.

Table 10: Overview different notations regarding stress symbols

Design Code	ASME	PD 5500	EN 13445
Longitudinal stress in hub	S_H	S_H	σ_H
Radial stress in flange	S_R	S_R	σ_r
Tangential stress in flange	S_T	S_T	σ_θ

Table 11: Summary of flange rigidity indexes

Flange Identification	# 1000	#1600	# 2000	# NPS 60
Design Code	ASME	ASME	ASME	ASME
Flange Thickness (mm)	99	157	160	134
J - Oper	0.8553	0.9164	0.8204	0.9954
J - Seat	0.9039	0.6337	0.9877	0.6815
Design Code	PD 5500	PD 5500	PD 5500	PD 5500
Flange Thickness (mm)	97	179	159	149
J - Oper	0.892	0.686	0.83	0.79
J - Seat	0.97	0.48	1.00	0.54
Design Code	EN 13445	EN 13445	EN 13445	EN 13445
Flange Thickness (mm)	97	173	163	143
J - Oper	0.892	0.741	0.78	0.86
J - Seat	0.97	0.51	0.94	0.59

Table 12: Overview determining cases for flange thicknesses

Design Code / Flange Identification	**ASME / # 1000**	**PD 5500 / # 1000**	**EN 13445 / #1000**
Flange Thickness (mm)	99	97	97
Thickness determined by:	$(S_H + S_T)/2$ Seating Condition	k $(S_H + S_T)/2$ Operating Condition	k $(S_H + S_T)/2$ Seating Condition
Design Code / Identification	**ASME / # 1600**	**PD 5500 / # 1600**	**EN 13445 / # 1600**
Flange Thickness (mm)	157	179	173
Thickness determined by:	$(S_H + S_T)/2$ Operating Condition	k $(S_H + S_T)/2$ Operating Condition	k $(S_H + S_T)/2$ Operating Condition
Design Code / Identification	**ASME / # 2000**	**PD 5500 / # 2000**	**EN 13445 / # 2000**
Flange Thickness (mm)	160	159	163
Thickness determined by:	J - Index & $(S_H + S_T)/2$ Seating Condition	k $(S_H + S_T)/2$ Seating Condition	k $(S_H + S_T)/2$ Seating Condition
Design Code / Identification	**ASME / # SP 60**	**PD 5500 / # NPS 60**	**EN 13445 / # NPS 60**
Flange Thickness (mm)	134	149	143
Thickness determined by:	J - Index & $(S_H + S_T)/2$ Operating Condition	k $(S_H + S_T)/2$ Operating Condition	k $(S_H + S_T)/2$ Operating Condition

V. DISCUSSION AND OBSERVATIONS

Both the ASME J-index and the k-factor approach pursue a common goal, namely controlling the angular rotation of flanges. From analysis of the calculation results of all selected flanges, hardly any substantial differences can be observed with regard to the calculated flange thicknesses. The maximum variation between the extreme values of the required flange thicknesses is successively approximately: 2%, 14%, 2.5% and 11% for flange identifications # 1000, # 1600, # 2000 and # NPS 60. The differences are shown visually in Graph 1.Table 3 shows that the required flange thickness calculated according to the ASME code for flange # 1000 and flange # 1600 is determined by the flange stress rather than on rigidity, while (more or less coincidentally) for flange # 2000 and flange # NPS 60 both the stress - and the rigidity criterion are decisive. It appears that in all cases, the average of the hub and tangential flange stress compared to the other calculated flange stresses to be the highest. A remarkable difference in ASME's approach compared to that of PD 5500 and EN 13445 is that, according to ASME, the J-Index criterion is flange diameter independent, that is, it also applies to flanges with a smaller diameter, while PD 5500 and EN 13445 apply the k-factor approach from a flange diameter of 1000 mm. It is unknown to the author on what grounds PD 5500 and EN 13445 apparently deviated intentionally from the J-Index approach outlined by ASME. Instead they have implemented the k-factor approach, despite the fact that the basis of the flange calculation was also based on the Taylor Forge method. The generally accepted view among experts is that implementing the rigidity criterion in PD 5500 and EN 13445 is preferable to that of the current k-factor approach. PD 5500 and EN 13445 code committees therefore should consider this issue as extremely relevant for large diameter flanges and should reconsider alignment with the ASME approach as suggested in Table 13 displayed below.

Table 13: Overview of suggested equations for implementation in PD 5500 and EN 13445

Suggested equations for the rigidity criteria in PD 5500 and EN 13445 to replace the k-factor approach	
PD 5500	EN 13445
$J = \frac{52.14\ V\ M}{\lambda\ E\ g_0^2\ K_I\ h_0} \leq 1.0$	$J = \frac{52.14\ \beta_V\ M}{\lambda\ E\ g_0^2\ K_I\ l_0} \leq 1.0$
Used symbols are in accordance with the relevant code	
K_I = 0.3 = Rigidity factor for integral flanges E = Modulus of elasticity for the flange material at design temperature (operating condition) or at atmospheric temperature (gasket seating condition)	

VI. CONCLUSIONS

The calculations have demonstrated that the k-factor approach does not entirely compensate for the flange rigidity index approach. In some cases the k-factor approach over-compensates for the J-index approach which may lead to a less economic flange design and should therefore be avoided. It is recommended that this issue be further evaluated on a less limited scale, whereby this paper can serve as a starting point. I hope that the relevant code committees responsible for the content of the code will include this serious topic in future updates to PD 5500 and EN 13445.

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APPENDIX

Typical geometry of analyzed flanges

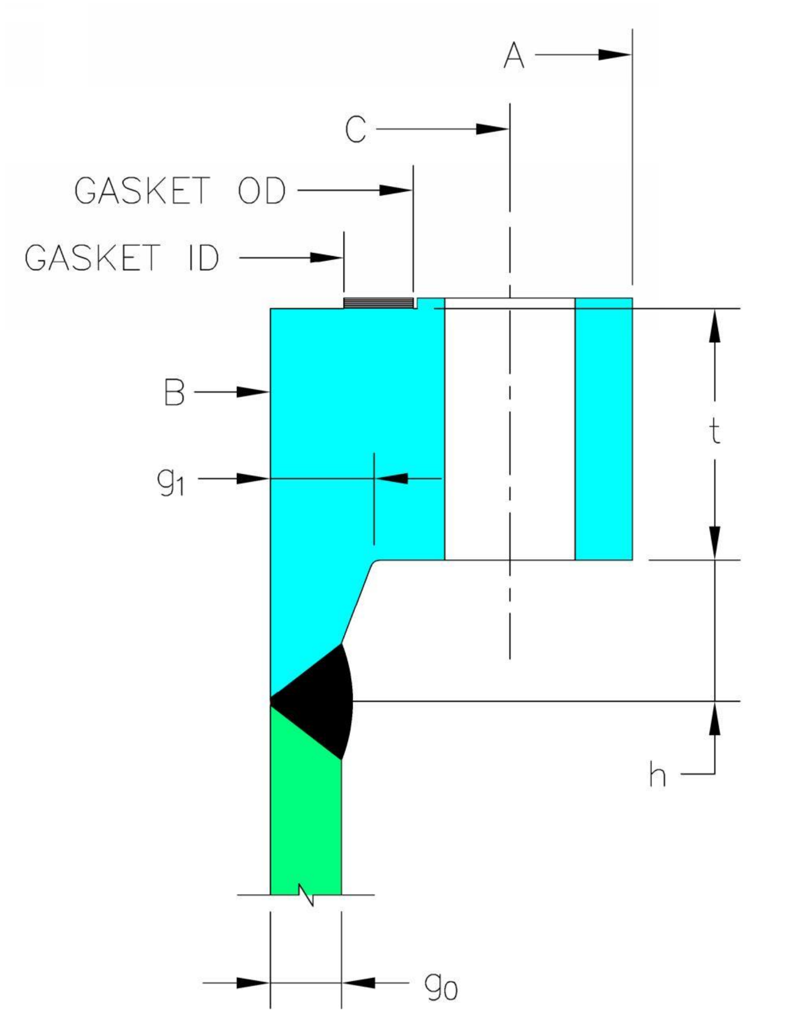

Table A-1: Main dimensions of analyzed flanges

Flange identification	# 1000	# 1600	# 2000	# NPS 60
Flange OD, A (mm)	1160	1825	2175	1675
Flange ID, B (mm)	980	1560	1980	1494
Bolt Circle, C (mm)	1100	1725	2110	1615
Gasket OD, (mm)	1050	1660	2050	1567
Gasket ID, (mm)	1018	1628	2018	1519.2
Flange Thickness, t (mm)	Per code	Per code	Per code	Per code
Hub Thickness, g_1 (mm)	20	30	20	22
Hub Thickness, g_o (mm)	10	20	10	15
Hub Length, h (mm)	30	30	30	22.5
Number of Bolts, (-)	48	60	68	64
Bolt Size, series 8 (inch)	1.125	1.5	1.25	1.125

For completeness, the required flange thicknesses have also been determined in accordance with ASME Section VIII Division 2, clause 4.16 [2]. As is known, the calculation method is identical to ASME Section VIII Division 1, Appendix 2 [1], with the exception of the allowable flange stresses. An overview of the allowable stresses (stress intensities) are shown in Table A-2. The calculation results are shown in Table A-3. It is striking that there are no or only slight and few differences compared to those determined according to ASME Section VIII Division 1[1].

Table A-2: Overview design stress intensities (MPa)

Material	A 105 Flange	A 515 Grade 65 Shell barrel	A 193 Grade B7 Stud bolts	Design Code / Standard
Temperature: 20°C	161.0 (S_{fg})	150.0 (S_{ng})	172.0 (S_{bg})	ASME VIII-Division 2
Temperature: 250°C	136.0 (S_{fo})	132.0 (S_{no})	172.0 (S_{bo})	ASME VIII-Division 2

Table A-3: Results of flange calculations

Flange Identification	Required Flange Thickness (mm)	J-Index (Seat)	J-Index (Oper)	Determining quantity
Flange # 1000	96	0.99	0.91	J-Index (Seat)
Flange # 1600	157	0.63	0.91	0.5 ($S_H + S_T$)
Flange # 2000	160	0.99	0.82	J-Index (Seat)
Flange # NPS 60	134	0.68	0.99	J-Index (Oper)

Worth noting that AS1210 [6] clause 3.21.6.8 state that to ensure sufficient rigidity to prevent leakage, flanges that are greater than DN 600 (NPS 24"), or that comply with a European flange Standard (i.e. with a safety factor less than 3.5 against ultimate strength) shall satisfy the following requirement for integral type flanges, under both gasket seating and operating conditions:

$175 \, \frac{M_0 V}{L\,E\,g_0^2 h_0} \leq 1.0$ $\Rightarrow$ This expression is similar to that according to ASME since $[\frac{52.4}{0.3}] \approx 175$

In the case of custom designed integral type flanges that are smaller than DN 600 (NPS 24"), it is strongly recommended they satisfy the rigidity requirement. The rigidity limit is somewhat dependent on gasket type, thickness, and operating temperature. Thicker gaskets in some materials may be more tolerant of flange rotation, and so may tolerate a higher value. Similarly, thinner gaskets may need a lower value to control leakage. Non-metallic gaskets operating close to their temperature limit generally require lower limits of flange rotation.

Walther Stikvoort. ET. Al " Evaluation of the flange rigidity index J - versus the k - factor approach for large diameter integral type shell girth flanges". *American Journal of Engineering Research (AJER)*, vol. 9(03), 2020, pp. 68-76..