Master’s Degree in Mechanical Engineering

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Evaluation of New Design Methods for Assessment of Flange Leakage Rate in Gas Media Application

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Abstract

Bolted joints flange plays an important role in the industries, specifically oil and gas sector. Considering environmental pollution, the new methods require to avoid leakage. Hydrogen is a new fuel in transportation and industries, then the old methods cannot guarantee the minimum leakage required by upgraded rules. Previous works mainly focused on ASME and Taylor Forged method. In the Taylor forged method the properties of gasket have not been upgraded and the leakage class is not involved. Gasket behavior is complicated due to nonlinear material properties linked with permanent deformation. The current work concentrated on EN standard methods in which the tightness level is the base of calculation, and the elastic behavior of gasket is modeled precisely. In this investigation, a comparison between the EN 13445-3 Annex G and EN 1591-1 (for designing bolted flanges) has been done to assess the behavior of the joint. Three types of common gasket: Spiral-wounded, Kamprofile and metal Jacketed, have been investigated. In addition, 6 size of flanges have been considered. Both Design by Analysis (DBA) and Design by Finite Element (DBF) have been researched to verify the behavior of the joint.

Key words: Bolted joints flange; Leakage; Gasket; EN standard; Design by Analysis; Design by Finite Element; Leakage class; Tightness level.

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1.Introduction

In the industries, there are numerous obstacles that remain to receive a significant amount of study interest, mainly in bolted flanged joint (B.F.J.) design. Two problems that are crucial in bolted flanged joint design are strength of the joint and leakage. The former has been studied since the 1920s for metallic joints with a general agreement on the existing resolution [1]. The latter issue has been studied for approximately long while, still leakage assessment continues to be the subject of a lot of studies [2-4]. The bolted flanged joints shall be designed to ensure leak tightness at the operating situation. The initial tightening of the bolts is crucial for the effective operation of the joint. Regardless of the fact that the joint should be leak-tight, the design methods are generally related to stress analysis to satisfy the structural integrity [5-6].

Here, an analysis for leak detection for a given pressure is described that can be employed in design formulations. Bolt load, flange stiffness, internal pressure, and gasket material seem to be the most essential of the many factors that affect joint leakage (bolt load, internal pressure, gasket material, flange geometry, enclosed medium, etc.). [7]. The wedge model is computationally less effective than the axisymmetric model, though. The pressure penetration option in ABAQUS, which is a crucial option required to accurately simulate the leakage, is not supported in three-dimensional elements, which is another reason axisymmetric modeling is prioritized.

A semi-empirical method was employed to analyze the loss of touch in Ref. [6]. Researcher employed potentiometric gauges to quantify the loss of contact between the two plastic plates using a single bolt plastic model, which was made of two round plastic washers held together by a bolt. After loading, the residual contact area was calculated using the interfacial pattern of light reflected from the separated surfaces (i.e., Newton rings). This outcome was utilized to determine the stress distribution around the bolt in an axisymmetric finite element model of the single bolt system. This was further expanded upon to investigate the bolt spacing in a model of a flanged-joint connection. The model does not account for the effects of the pipe/hub, gasket, or internal pressure. Later, using a dimensional analysis, the same researchers [8] went back to their model and expanded it to incorporate the impacts of the pipe/hub and internal pressure. Finite element analysis was performed by Nishioka et al. [9, 10] to study the loss of contact in a flanged-joint. However, due to limitations in the computing tools that were available at the time, they were only able to apply an iterative process to calculate the loss of contact after formulating the problem as previously explained. The flange and gasket were modeled using an axisymmetric triangular element with constant strain. Additionally, when the usual stress approached the gasket yield stress, it was believed that the gasket would give. The authors looked at the impact of bolt count and hub taper on gasket contact stresses. Sawa et al experimental examination of leakage was combined with an analytical investigation of the contact stresses using an axisymmetric elasticity formulation [11]. When the gasket material is loaded over its elastic limit point and fills the imperfections on the flange face, the leakage problem is made worse. Additionally, when the rotation rises, the bolt load relaxes; this relaxation is thought to be insignificant since a mechanic generally retightens the bolts to maintain a constant bolt load throughout the joint's lifespan. The progression of leakage has been examined in Ref [7] utilizing a thorough contact finite element analysis. The findings of the leakage study were compared to well values obtained under the contact circumstances outlined in Section VIII, Division 1 of the ASME Boiler and Pressure Vessel Code (hereafter referred to as the ASME code) [12].

Either the Taylor Forge approach or the DIN 2505 method is the main foundation for all national standards pertaining to flange design. Both approaches start with roughly the same presumptions, which means that two distinct bolt loads are computed independently for two different conditions: first, the force at bolting-up, required for the gasket seating, and subsequently, the load required to achieve leak tightness in service. In this manner, the flange calculation is done for two separate, unrelated condition, bolting-up and service, and maybe a hydrostatic test condition. However, a bolted joint is a complicated construction made up of two flanges (which may be different), a gasket, and bolts; As a result of the initial bolt tightening and the elastoplastic (and creep) behavior of all the associated components, the bolt load changes from one situation to another. For the first time, a method based on a more intricate model of the bolted assembly has been created in accordance with European Standard EN 1591 [13].

The approach has been further refined in Annex G of EN 13445 Part 3 [14] (the European Unfired Pressure Vessel Standard), which is now being revised in light of experiment findings and considering the various behaviors of liquids and gases in terms of leak tightness. The purpose of [15] is not to fully explain the theory behind EN 1591, but rather to merely outline the guiding principles of the approach and contrast the outcomes of its application as an alternative to the other ways using a number of real-world examples.

Several bolts, a gasket, and two flanges that are welded to a shell or a domed end make up a bolted junction; the two flanges can be equal or different from one another, or one of them might be a fixed flat cover. The gasket is made of a somewhat soft substance (in any case softer than the gasket contact surfaces on the flanges). When an appropriate tensile force is given to the bolts, the gasket is compressed up to a contact pressure that is specific to the particular gasket material and form (the "seating stress" y, as stated in the ASME Code [16] and in Clause 11 of EN 13445.3 [14]). The gasket will flex as a result of this pressure, allowing it to fill in any ridges, crevices, or other imperfections in the seats. The first need that must be met in order to ensure leak tightness is this one. The second condition is that in order to ensure leak-tightness while in operation, the gasket must endure adequate residual compressive stress once pressure has been built up inside the vessel. This is accomplished by using a separate factor m, which is also specific in [16] and [14] above and is also distinct for each type of gasket material and form.

As a consequence, in the operational cycle of a bolted joint the first design condition to be considered is the initial tightening of the bolts without internal pressure (bolting-up or assembly condition). Since every pressure vessel shall be tested with water at the end of the fabrication cycle and before putting it into service, the design condition following the bolting-up is the hydrostatic test condition (generally made at room temperature, with an internal pressure higher than the design pressure). Then there is the real service condition (at operating temperature and operating pressure). Further conditions may exist, because of temperature and pressure modifications occurring in service. Of course, the load initially applied to the bolts will change when going from the bolting-up condition to the subsequent service (and/or testing) conditions. In fact, the presence of the internal pressure would cause an increase of the bolt load and a decrease in the gasket compression, but the rotations due to the moments acting on the two flanges tend to balance this effect, by decreasing the bolt load and shifting outwards, the actual contact surface force between the gasket and the seats. Ref. [15] considers only bolted joints, that is bolted joints having the gasket entirely contained within the bolt circle, with exclusion of the so called "full face" gaskets.

Bolted flange connections that have been improperly designed, assembled, or maintained may produce fugitive emissions, which make for 8% of all emissions in the petrochemical industry [17]. Absolute tightness may be expensive to achieve, thus regulated leak rate that does not go beyond a specific point is desirable. However, engineering equations used to determine leak rate are complicated and call for specific inputs [18].

The difficulty in bolted flange connections is brought on by the extremely nonlinear gasket behavior and the lack of, incompleteness of, or idealization of gasket test data. This creates a circumstance that need frequent monitoring and modifications, such as tightening bolts or changing gaskets. On the most basic level, bolted flange connection tightness may be related to existence of continuous gasket contact around the joint circumference. Estrada and Parsons [19] presented analysis of GFRP flange joint where fluid pressure was considered on flange-gasket interface areas with lost contact.

Vinod et al. [20] investigated leakage of steam generator bolted flange joint at elevated temperatures, and found that minimum contact pressure cannot be achieved (there may be no contact at all) due to higher stud thermal expansion. Leakage was mitigated using longer studs and sleeves, thus changing mechanical behavior of the joint. Somewhat improved approach considers fluid and gasket contact pressures, assuming that when the former is greater than the latter, fluid will propagate trough the seal. This is a basis for the pressure penetration criterion. It was used by Aljuboury et al. [21] to estimate the extent to which the fluid is spread on the seal surface.

Both above-mentioned approaches are only able to estimate if there is a leak or not. However, in case of gases, there is always a leak, and the real question is its magnitude [22]. Authors in [23] tested theoretical models against measured gas leak rates and found a good correlation for the capillary model with the second order slip flow for the widest range of parameters. Main parameters during testing were gasket contact pressure and internal fluid pressure, while the initial seating pressure was constant.

In bolted flange connections, the gasket contact pressure is typically at its greatest during assembly before decreasing following the introduction of internal fluid pressure. It was discovered that the initial gasket seating pressure during assembly also affects the leak rate, in addition to the current gasket contact pressure and internal fluid pressure. This idea is applied in the EN 1591-1 standard [13], where the mechanical model enables accurate prediction of the gasket contact pressure even in situations when the gasket face is only partially in touch. Calculated contact pressure history may be utilized to estimate the leak rate along with gasket leak rate test results in accordance with EN 13555 [24]. The mechanical model is applicable to both comparable and standard flanges in line with EN 1092-1 [25]. Gasket testing under constant internal pressure demonstrates that the rate of leakage is dependent on both the current contact pressure and the greatest contact pressure in the past. The tests are carried out under idealized circumstances, where the gasket contact pressure is always constant, which is typically not the case in real-world situations.

In order to address leak issues, a better design process is necessary given the technical and environmental demands of today. The validity of the m and y gasket factors, their inability to forecast tightness, and the relative difficulty of sealing some types of flange joints are of great concern [26]. New tightness-based gasket constants were created by the PVRC in an effort to enhance the design methodology, and the ASME BPVC-SWG-BFJ is now adopting them [27]. The new constants, on the other hand, come from ROTT experiments done on rigid platens where the gasket contact stress is evenly dispersed across the whole region. The gasket contact stress is not consistent because flanges with ring-type gaskets might rotate. Lower bolt loads would result from the addition of a correction factor as a consequence. Research demonstrates that the leak rates of bolted joints rely on both the average contact stress on the gasket and the distribution of that stress throughout the gasket's breadth [28]. Experiments on sheet gasket materials conducted in Europe [29] revealed that the tightness increases with rotation. This conclusion was supported by a recent PVRC research [30] on the impact of flange rotation on the leakage tightness of flexible joints.

Flange rotation is restricted to 0.3 degrees for integral or optional-type flanges and 0.2 degrees for lap-type flanges under the new non-mandatory ASME code flange design method. However, the resulting nonuniform radial distribution of the gasket contact stress can significantly affect the sealing performance depending on the gasket and the joint design. To create an ideal design that prevents excessive flange rotation or overtightening of bolts without sacrificing the high level of leak-tightness integrity, a good evaluation of this distribution is of utmost importance. The gasket contact stress was examined by Sawa et al. [11] using a three-dimensional theory of elasticity. Their analysis was further streamlined by considering the pipe flange and the gasket as hollow cylinders as the mechanical behavior of the gasket was assumed to be linear. While an attempt was made to incorporate the hub, the gasket material behavior was likewise thought to be linear in the research [31]. The anticipated linear elastic behavior of the gasket did not stand out as an oversimplification until experiments on large-diameter flanges [31] showed partial separation at the gasket-flange contact.

Analytical models are difficult to build or too challenging to solve for complex geometries like gaskets, and experimentation is quite challenging. Simulation and numerical modeling of SWGs can, nevertheless, acquire certain mechanical qualities. even on a desktop computer, easily and affordably. The gasket's plasticity parameters and orthotropic characteristics are necessary for simulating the effectiveness of gasketed joints under applications that need three dimensions include bending and other variables that are elastic and plastic. A way to determine the relationship between the macroscopic mechanical and microscopic deformation Homogenization of behavior is a concept. By using an analogous homogeneous continuum model, it is utilized to substitute a heterogeneous material [32].

FE analysis of a raised flange with a nonlinear gasket was published by Shoji and Nagata [33] utilizing 2-D axisymmetric and 3-D solid element FE models. In article [34], this kind of flange was described. The analysis has been completed in two stages, pre-load and pressured, depending on the load state. Due to the gasket's nonlinearity, two values of the modulus of elasticity (compression and decompression) have been employed, depending on whether the gasket is in compression or decompression. According to the outcomes of the numerical simulations, the stresses are greater under pre-load settings than under pressure situations, and they grow from the inner radius of the gasket toward the outside radius of the gasket for both the 2-D and the 3-D models.

Aljuboury et al [35]'s numerical analysis of the sealing effectiveness of a glass fiber reinforced plastic (GFRP) bolted flange connection with a rubber gasket. The GFRP materials' orthotropy and the rubber gasket material's non-linear behavior under both loading and non-loading circumstances were both taken into consideration in the FEA model that was created using ANSYS. Additionally, the leakage propagation between the flange and the gasket has been modeled using the pressure-penetration criteria PPNC in ANSYS. The findings demonstrate that the minimum contact pressure is located at the gasket's inner radius and rises in the radial direction. The contact pressure at the bolt hole is higher than it is in the center of the circumferential direction between the bolts. The results show that the leakage development began at the inner radius of the gasket, where the contact pressure is lower than other places, and rises towards the outer radius for the distribution of the fluid pressure penetration (FPP) between the flange and the gasket. Additionally, the leakage increases at the midway between the bolts is greater than at the bolt center due to the unequal distribution of the bolt stresses in the circumferential direction.

This research consists of four main parts. First part is an overview of the previous research relevant to my work and introduction of the motivation and goals of my proposal.

In the second chapter are introduced the analytical methods of EN 13445 Annex G and EN 1591 for dealing with gasket modeling and software for performing the simulation in this work, which is VVD.

In the third chapter are introduced the FE methods of gasket modeling according to EN 13445 Annex G and EN 1591 and software is Axipro and Ansys.

In the fourth chapter, a comparison between the analytical model and FE model is formed to demonstrate the advantages and disadvantages.

Finally, because of the effects of geometrical and characteristics of different gaskets on the performance of the bolted flange joints, so detailed discussion about them is performed within the fifths chapter. the consequences of different values of bolt size and flange thickness are analyzed.

2. Analytical model and Setup

2.1 Analytical method

Assessment of analytical model has been investigated by 3 types of gaskets and 6 size of flanges. The temperature, material and pressure are same in all cases. In Annex G [14], Gasket parameters are available, but they shall be selected from www.gasketdata.org for EN 1591 [13]. Gasket data follows EN 13555 [24] for testing and parameters. Keeping consistency of investigation, gasket characteristics have been selected, for EN 1591, as much possible same as Annex G.

All design data are listed in Table 1:

Temperature (°C):	100	Flange Size:	DN50,100,250,400,600,800
Pressure Class:	PN16	Gasket types:	-Spiral Wound -Kamprofile -Metal Jacketed
Flange Material:	P355NH	Design Code:	EN 13445-3 Annex G, EN 1591-1

Table 1. Design Data

Due to the complexity of two methods, evaluation of effects of each parameter of gasket on bolting-up process is sophisticated. Hence, the size of bolts and total weight of flanges are considered. The other geometrical parameters of flange and gasket are kept fixed except flange thickness. In some cases, some changes, in other geometrical parameter of flange and nozzle, are inevitable, then their effects can be considered in weight. Considering weight and bolt size, the man cost, and total cost of the flange manufacturing will be predictable.

2.2 Introduction of VVD Software

VVD is a software to design pressure vessels, heat exchangers and towers according to ASME, EN 13445 and EN 1591 code. Design method by software is By Formula. First steps in software are selecting the method of design, gasket design method, flange size, material, and bolt size. Then software is checking the formulas and reporting the limitation and ration designed parameters with the required ones.

2.3 EN 13445-3 Annex G

In this normative, apart from the Clauses which are discussing on designing the pressure vessels, we have 24 Annex which Annex G is focused on an alternative method for designing flanges and gasketed connections.

This annex uses a method according to EN 1591. Hence this method is more detailed, and it tries to conquer the assumptions of the clause 11 (Taylor Forge Method) of EN 13445-3. This annex considers 6 parameters instead of 2 parameters in clause 11. Then the stress analysis is more accurate than clause 11. In addition, elastic-plastic behavior has been involved for the gasket. Furthermore, temperature of engaged parts, all conditions from assembly to operating and service have been considered as well as additional loads.

The design procedure for gasket in this annex is not straight away, because some iteration processes are needed to be done. Specifically, when the calculation for modulus of elasticity and effective width of gasket shall be done.

Design steps and only important formula are reported in Table 2. Very briefly.

1) a random value for the applied force on gasket, F_{G0}, shall be chosen:

$F_{G0} = A_B \times \frac{f_{B0}}{3} - F_{R0}$ (1)

2) effective gasket width will be calculated according to assumption in (1). Noting that b_{Gi} depends on type of flange (Table G.5.1 Annex G has mentioned about the different formula regarding the type of gasket)

$b_{Ge} = min\{b_{Gi}; b_{Gt}\}$ (2)

3) effective area of gasket shall be calculated:

$A_{Ge} = \pi \times d_{Ge} \times b_{Ge}$ (3)

4) to guarantee the minimum force on gasket, it shall be always greater than $F_{GI:}$

$F_{GA} = \max_{\text{all } I \neq 0}\left\{F_{GI,min} \cdot Y_{GI} + \left[F_{QI} \cdot Y_{QI} + (F_{RI} \cdot Y_{RI} - F_{R0} \cdot Y_{R0}) + \Delta U_I\right]\right\}/Y_{G0}$ (4)

5) two conditions need for seating the gasket perfectly (gasket required force and bolt load):

$$\begin{aligned} F_{G0,\,req} &= max\{F_{GO,min}; F_{G\Delta}\} \\ F_{BO,\,req} &= F_{GO,\,req} + F_{RO} \end{aligned} \quad (5), (6)$$

6) if bellow formula is not satisfied, then the calcuation must be repeated:

$F_{GO,\,req} \leq F_{GO}$ (7)

7) but if the $F_{G0,\,req}$ is lower than F_{G0}, the calculation will be accepted

8) when the bellow required is satisfied, the precision is adequate, then the itration will be terminated.

$F_{G0,\,req} \approx F_{GO}$ (8)

Table 2. the main steps for designing gasket according to Annex G [14]

Gasket unloading compression modulus E_G is linear equation which its variable is maximum compression stress:

$$E_G = E_0 + K_1 \cdot Q_{(max)} \quad (9)$$

2.4 EN 1591

EN 1591-1 is based on the minimum bolt load required to fulfill the chosen leakage rate. The gasket sealing coefficients must be calculated using EN 13555 or directly from EN 1591-2. However, the leakage must be checked in accordance with EN 13555.

Annex D describes all method stages and, like EN 13555, a factor (PQR) for gasket creep is addressed, as well as the plastic deformation scenario. Annex B the scatter calculation is explained, Annex C also considers rotation for flanges with certain limitations.

All sealing calculation performance in this standard is based on elastic load or deformation for all relations between joint parts and is compensated for expected plastic deformation in gasket material. Resistance is calculated using a combined flange-shell plastic limit analysis that takes into account both internal and exterior loads.

This standard's load calculation is based on the minimum bolt load that must be applied to the gasket in order to fulfill the requisite tightness class. Increasing bolt load within the permitted load range of flanges, bolts, or gaskets in order to prevent leakage and have a safe design. The designer can select a bolt load between the obtained load for the tightness class and the load ratios.

Calculation order is as bellow steps:

1) Calculating parameters according to clause 6: both side of flange; bolts and washers; gasket.

2) Forces calculation according to clause 7: applied loads; compliance of the joint; minimum gasket forces; internal forces in assembly condition; internal forces due to subsequent conditions.

3) Load limits according to clause 8: bolts; gasket; first and second flanges.

2.2.3 Requirements for using the method:

1.1. General:

-special testing

-proven practice

-using of standard flange with permitted condition

1.2. Geometry:

-flanges section shall be according/similar to permitted configuration

-four or more bolts shall be distributed uniformly

-gasket section shall be as in a given shape

-flange dimension shall be meet following condition:

$0.2 \leq b_F I e_F \leq 5.0; 0.2 \leq b_L I e_L$

$cos\varphi \geq 1/\left(1 + 0.01\frac{d_s}{e_s}\right)$

Note: No need to use $0.2 \leq b_L I e_L$ limitation for collar in combination with loos flange.

Two situations are out of scope of this normative:

-non-axisymmetric geometry

-direct or indirect metal to metal contact between flanges inside or/and outside the gasket, inside or/and outside the bolt

1.3. Material

No nominal design stresses are specified in calculation method.

1.4. Loads

Calculation method are valid for the following load types:

-fluid pressure: internal or external

-external loads: axial, lateral, torsion and bending moments

-axial expansion of flanges, bolts, and flanges, specially owing to thermal effects

Designing shall consider all conditions as: start-up, test, service, cleaning, maintenance, shut down.

2.2.3 Checking the assembly for a Specified initial tightening bolt force or torque

EN 1591-1 method is established on a selected leakage-rate to be achieved. However, if aim is checking design for a given value of the tightening bolting force at assembly ($F_{BO,specified}$) the calculation shall be started using

$$F_{G0} = F_{B0,specified} \times (1 - \varepsilon_{-}) - F_{R0} \quad (1)$$

Note: this formula is considering scatter instead of minimum value for tightening force.

then continue from (55) to (110). Using formula $F_{G0} \approx F_{G0req}$ (110) two cases shall be considered:

-If $F_{G0,req}$ in (110) > F_{G0} in Formula (1), the value of $F_{a0,specified}$ is not sufficient to guarantee the tightness criteria. So, the value of $F_{B0,specified}$ shall be increased to meet the tightness criteria. The calculation procedure from Formula (55) to Formula (110) shall be applied again.

- If $F_{G0,req}$in formula $110 < F_{G0}$ in Formula (1), the value of $F_{B0,specified}$ is sufficient to guarantee the tightness criteria and therefore the calculation can be continued using the value of F_{G0} calculated by Formula (1) as the gasket force in assembly condition (I=0). In that case, the initial bolt force at assembly can be very much greater than the required one, and the Formula (119) shall be replaced by Formula (103), considering the lower bound of the applied initial bolt force at assembly phase.

$$F_{G0d} = max\left\{F_{B0min} - F_{R0}; (2/3) \times \left(1 - \frac{10}{N_R}\right) \times F_{B0max} - F_{R0}\right\} \quad (2)$$

2.2.3 Calculation parameters

3.1. General

All dimensions, area and stiffness are effective.

3.2. Flange Parameters

Special flange types are considered in this normative:

-Integral Flange

-Blank Flange

-Loose Flange

-Screwed Flange

-Collar

Very detailed geometric parameters which shall be considering, are listed in normative.

3.3. Gasket Parameters

The effective width of gasket is relative to applied force.

$$b_{Ge} = \min\{b_{Gi}; b_{Gt}\} \quad (55)$$

Effective gasket area:

$$A_{Ge} = \pi \times d_{Ge} \times b_{Ge} \quad (56)$$

Table 1, in normative, is used as the initial approximation, then the gasket width and effective area will be calculated through a few iterations.

In addition, initial gasket stress is calculated:

$$Q_{G0} = F_{G0} / A_{Ge} \quad (57)$$

$$E_{G0} = E_G(Q_{G0}) \quad (58)$$

2.2.3 Forces

4.1. General

Different conditions shall be considered by abbreviation “I”. Thermal load, Compliance of the joint, Additional external loads, Fluid pressure, Minimum essential forces for gasket and the other condition for gasket and Internal forces shell be calculated according to normative.

4.2. Applied Loads

I=0	Assembly Condition:
	-fluid pressure P_0 (int. and ext.) =0
	-T=T_0
	-External bending moments and axial force combine to net force F_{R0}
	-Lateral forces and torsion moment=0
I=1, 2, ….	Different test conditions, operating, ……: details are in normative

4.3. Minimum Necessary Load on Gasket

4.3.1. Condition I=0.

$$F_{G0min} = A_{Ge} \times Q_A \tag{103}$$

4.3.2. Condition I=1, 2, …

Required condition for the force:

-leak-tightness.

-no loss of contact permitted at either bolt or nut.

-adequate axial load on gasket due to external torsion and radial force producing by friction.

$$F_{Glmin} = max\left\{A_{Ge} \times Q_{smin(L),l}; -\left(F_{Ql} + F_{RI}\right); \frac{F_{LI}}{\mu_G} + \frac{2 \times M_{TGl}}{\mu_G \times d_{Gt}} - \frac{2 \times M_{Al}}{d_{Gt}}\right\} \tag{104}$$

4.4. Internal forces condition I=0.

The minimum condition for internal forces to guarantee the sealing shall not fall below the value of F_{Glmin}. Hence the adequate internal force on gasket is as bellow:

$$F_{G\Delta} = \max_{all\ l\neq 0}\left\{F_{Glmin} \times Y_{Gl} + F_{Ql} \times Y_{Ql} + \left(F_{Rl} \times Y_{Rl} - F_{R0} \times Y_{R0}\right) + \Delta U_l + \Delta e_{Gc,l}\right\}/Y_{G0} \tag{105}$$

Note: formula (106) does not consider plastic deformation. In the presence of significant plastic deformation, bellow formula shall be considered:

$$F_{G\Delta} = \max_{all\ l\neq 0}\{F_{Glmin} \times Y_{Gl} + F_{Ql} \times Y_{Ql} + (F_{Rl} \times Y_{Rl} - F_{R0} \times Y_{R0}) + \Delta U_l + \Delta e_{Gc,l} + \lfloor e_G(Q_{G0}) - e_{G(A)} \rfloor\}/Y_{G0} \tag{106}$$

4.4.1. Bolt-load scatter at assembly

Limitation for load are as follows:

$$F_{B0min} \leq F_{B0} \leq F_{B0max} \tag{111}$$

Where

$$F_{B0min} = F_{B0\ av} \times (1 - \varepsilon_-) \tag{112}$$

$$F_{B0max} = F_{B0\ av} \times (1 - \varepsilon_+) \tag{113}$$

Detailed calculation on scatter is mentioned in Annex B.

4.5. Internal forces condition I=1 ,2 ,... .

To guarantee the minimum leakage, the F_{Glmin} shall be lower than the formula (105). In condition I=1 , 2, …. The plastic deformation may occur. Specially, in the case of frequently re-assembly the situation is worse. To prevent the plastic deformation at start-up after each assembly, load limit of the joint shall be checked:

$$F_{G0d} = max\left\{F_{G\Delta}; \frac{2}{3} \times \frac{10}{N_R} \times F_{B0\ max} - F_{R0}\right\} \tag{119}$$

Subsequently bolt and gasket load limit calculation are:

$$F_{Gl} = \{F_{G0d} \times Y_{G0d} - [F_{Ql} \times Y_{Ql} + (F_{Rl} \times Y_{Rl} - F_{R0} \times Y_{R0}) + \Delta U_l] - \Delta e_{Gc,l}\}/Y_{Gl} \tag{120}$$

In (120) no plastic deformation is considered, therefore, when significant plastic deformation is presence, the formula shall be replaced:

$$F_{Gl} = \{F_{G0d} \times Y_{G0d} - [F_{Ql} \times Y_{Ql} + (F_{Rl} \times Y_{Rl} - F_{R0} \times Y_{R0}) + \Delta U_l] - \Delta e_{Gc,l} - \lfloor e_G(Q_{G0}) - e_{G(A)} \rfloor\}/Y_{Gl} \tag{121}$$

From formula (120) and (121), bolt load shall be calculated as bellow:

$$F_{BI} = F_{GI} + (F_{QI} + F_{RI}) \tag{122}$$

The admissibility shall be checked according to Clause 8 as following approaches:

-Assembly condition: F_{B0max} and F_{G0max} shall be checked.

-Subsequent condition: F_{BI} and F_{GI} shall be checked.

2.2.3 Load limits

5.1. Bolts

Bolt load ratio.

$$\Phi_B = \frac{1}{f_B \times c_B} \sqrt{(\frac{F_B}{A_B})^2 + 3 \times (c_A \times \frac{M_{t,B}}{I_B})^2} \leq 1.0 \tag{123}$$

5.2. Gasket

Gasket load ration:

$$\Phi_G = \frac{F_G}{A_{Gt} \times Q_{s_{max}}} \leq 1.0 \tag{128}$$

The situation for Integral flange and collar, Blank flange and Loose flange with collar are explained very detailed in normative.

2.5 Results and discussion

In Figure 1. the bolt size and weight of flange against flange size are reported for Annex G and EN 1591 for 3 types of gaskets:

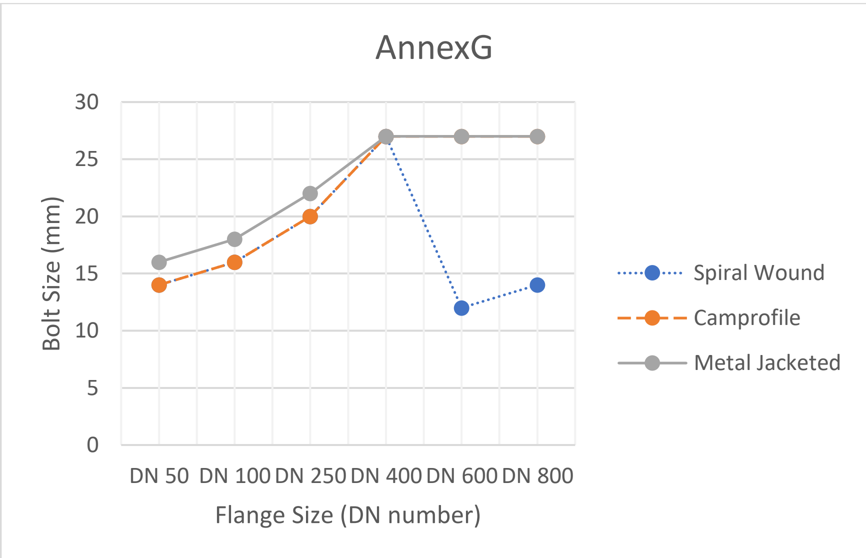


Figure 1. Comparison of bolt size in Annex G

Both Spiral-wounded and Kamprofile are treating very similar to each other. The only deviations are in DN 600 and DN 800 that can be corrected by re-designing. The total trend of Metal-Jacketed is like the others with a bit greater size.

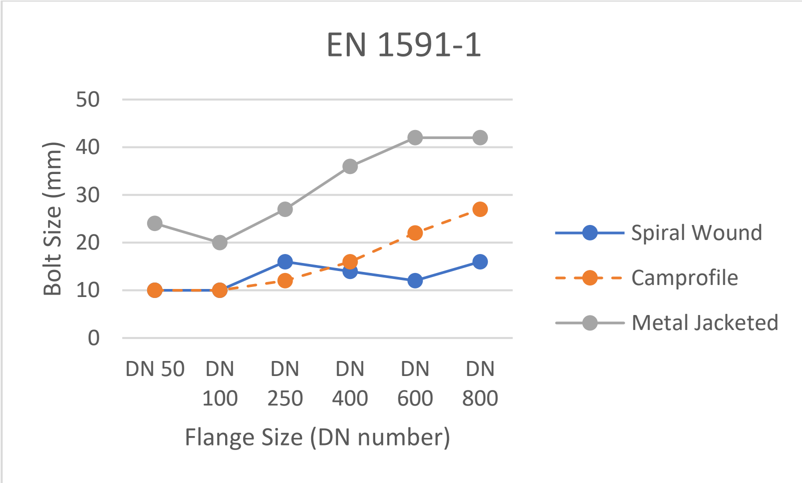


Figure 2.Comparison of bolt size in EN 1591

Also, in Figure 2, for EN 1591, Spiral-wounded and Kamprofile have similar behavior. The size of bolts in Metal-Jacketed are greater than the two others.

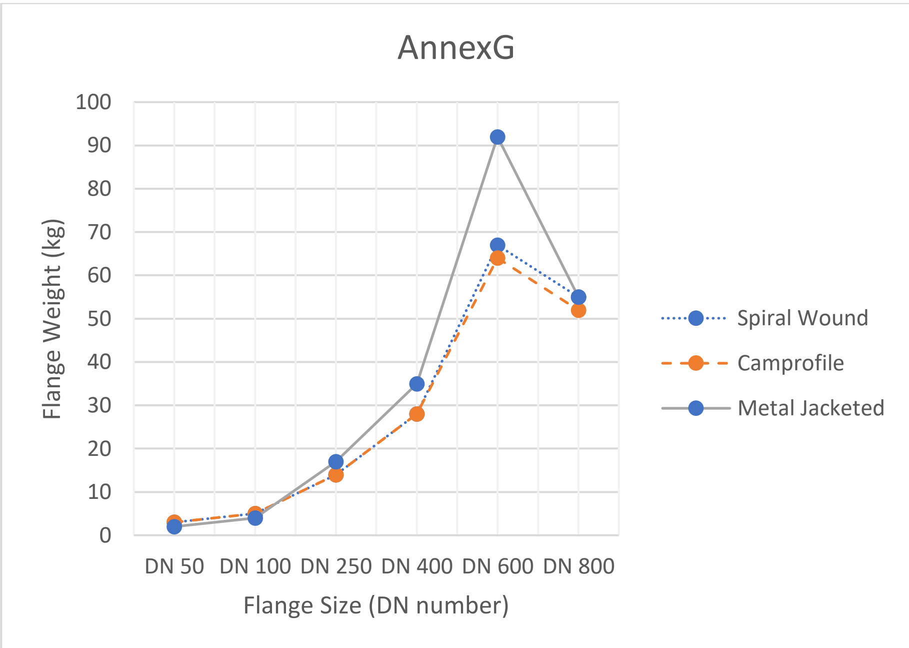


Figure 3. Comparison of flange weight in Annex G

Figure 3 compares the weights. Spiral-wounded and Kamprofile have the very same weights, but the weight of Metal-jacketed design is greater than the other types.

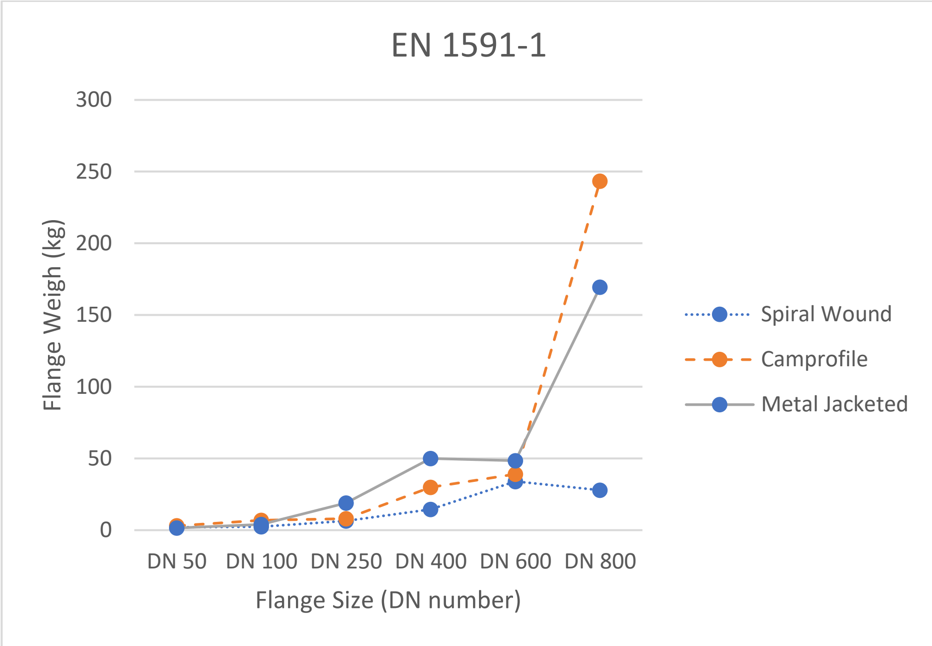


Figure 4. Comparison of flange weight in EN 1591

As it is expected, again Kamprofile and Spiral wounded are acting very similar, and their weight are same. The total trend in metal-Jacketed is increasing and the weight is a bit greater than the two others.

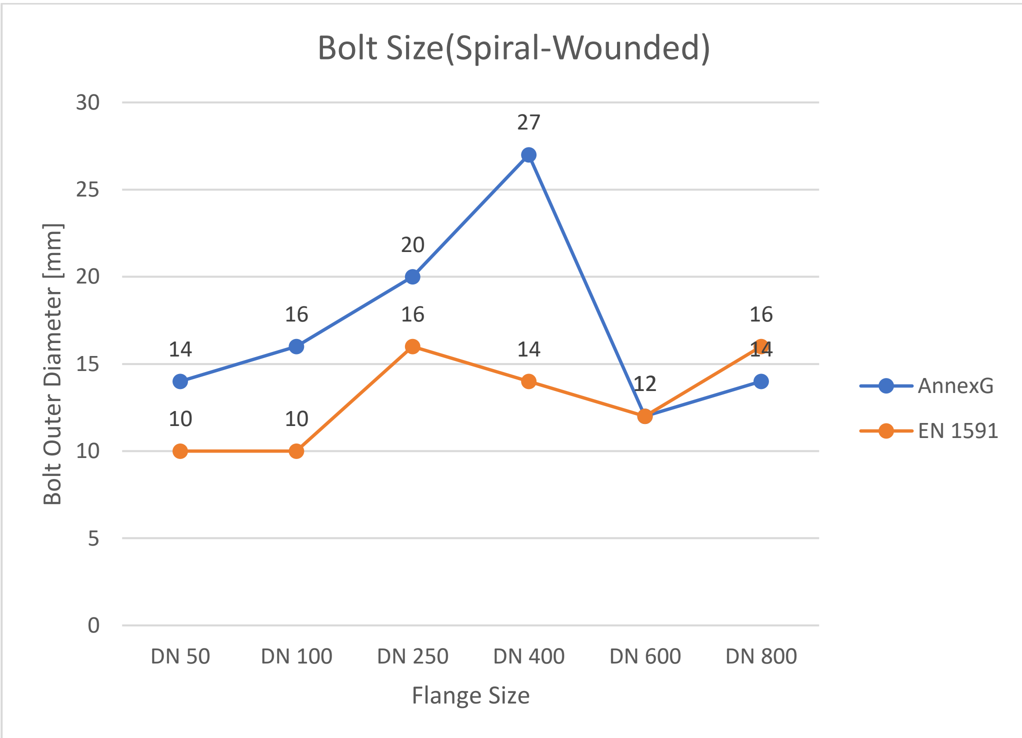


Figure 5. Comparison of bolt size in EN 1591 and Annex G Spiral Wounded

In Annex G bolt sizes are greater than the EN 1591 for Spiral-Wounded, which is represented in Figure 5.

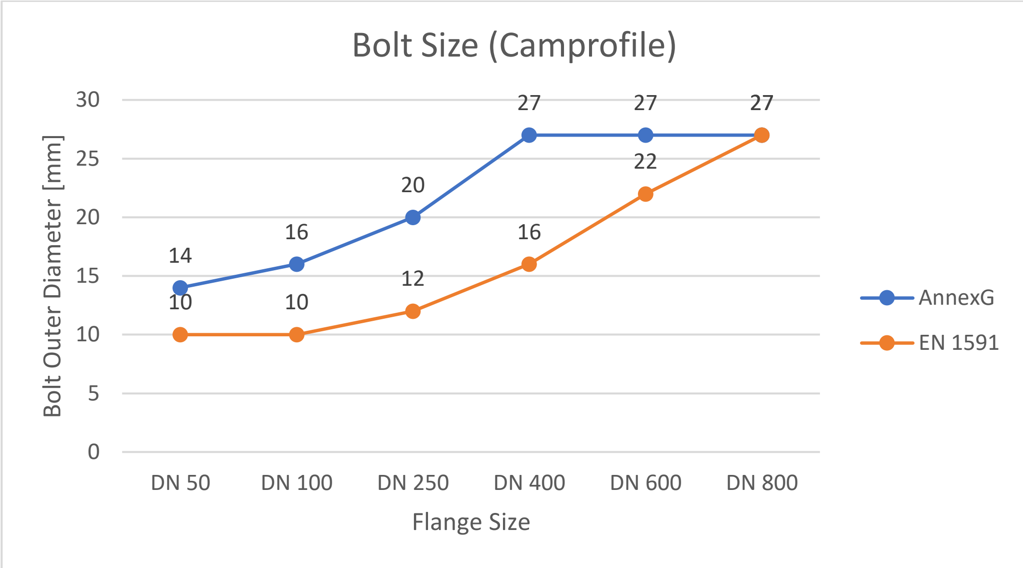


Figure 6. Comparison of bolt size in EN 1591 and Annex G Kamprofile

In Annex G bolt sizes are greater than the EN 1591 for Kamprofile also.

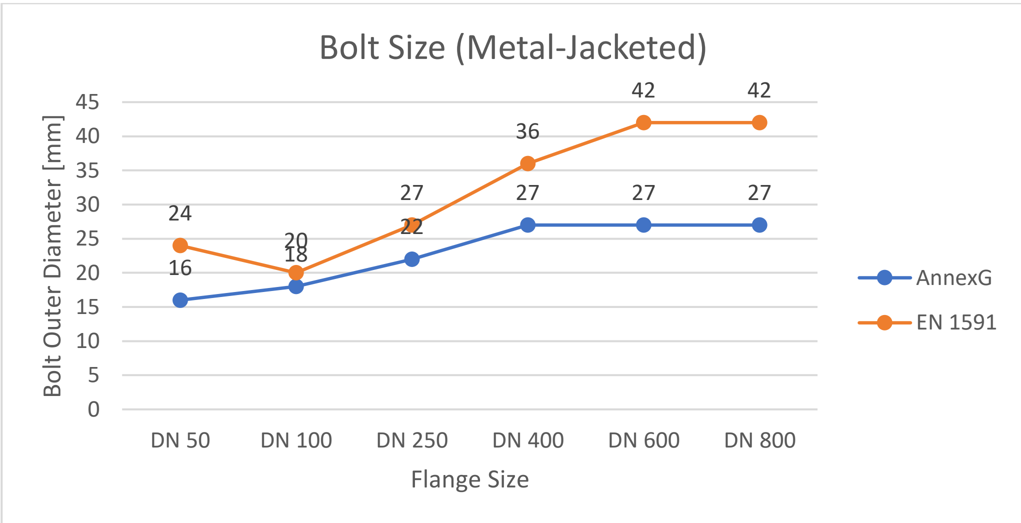


Figure 7.Comparison of bolt size in EN 1591 and Annex G Metal-Jacketed

But as it is shown in Figure 7, in Annex G bolt sizes are smaller than the EN 1591 for Metal-Jacketed. Which is different trend rather than the other cases.

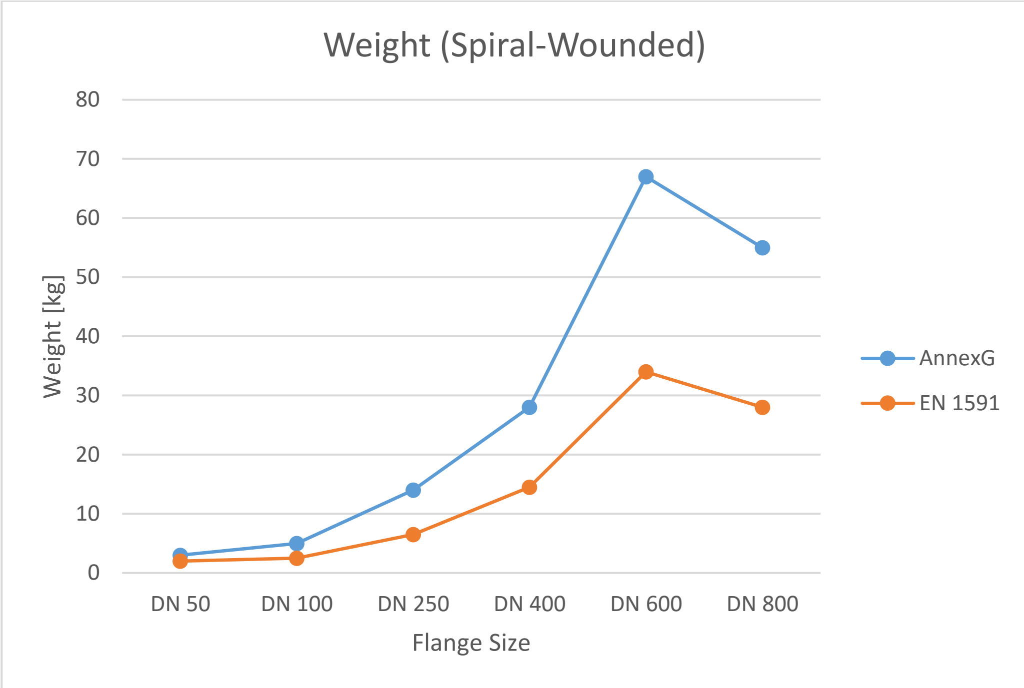


Figure 8. Comparison of weight in EN 1591 and Annex G Spiral Wounded

Weight of flange according to the Annex G method is higher than EN 1591 for Spiral-wounded.

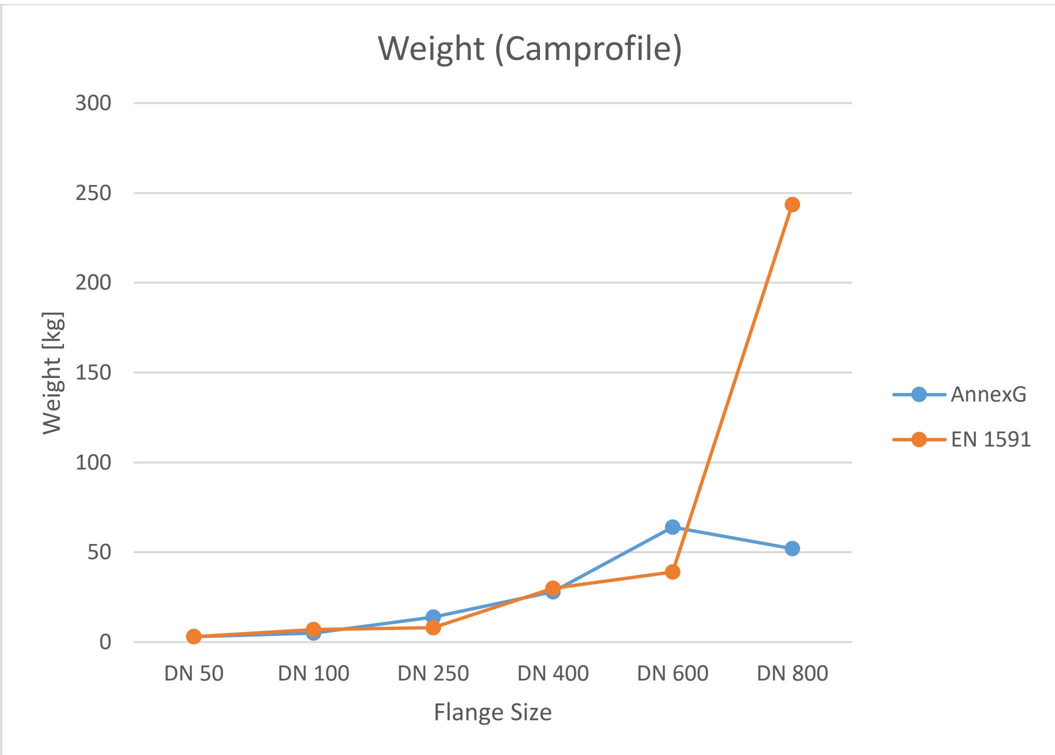


Figure 9. Comparison of weight in EN 1591 and Annex G Kamprofile

But weight of flange for Kamprofile according to both methods are same.

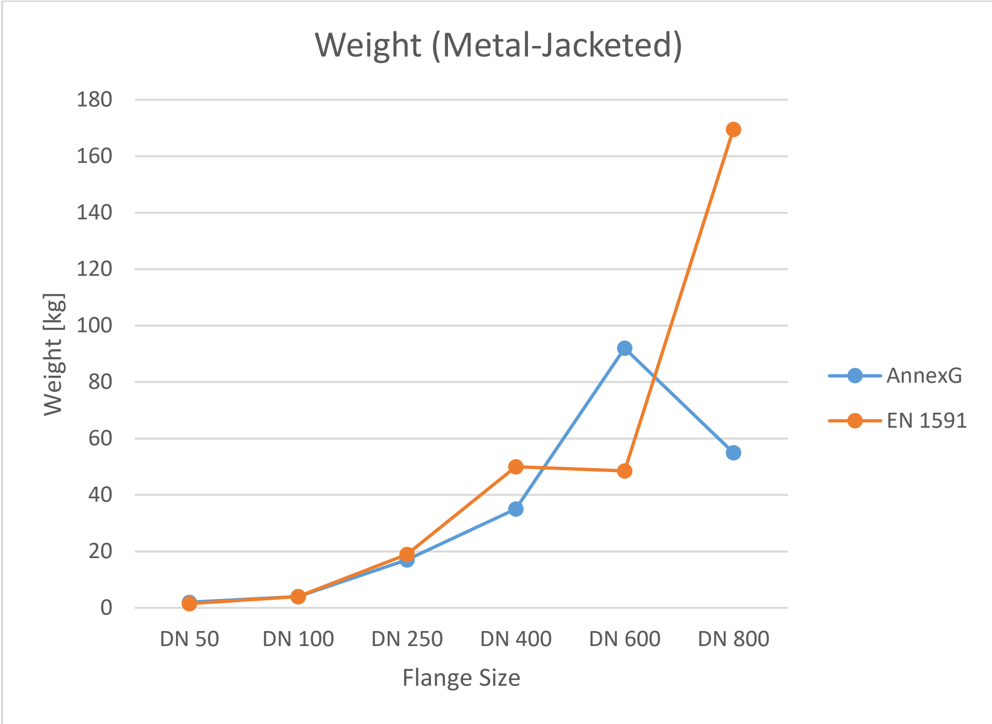


Figure 10. Comparison of weight in EN 1591 and Annex G Metal-Jacketed

Also, for Metal-Jacketed same weight are represented.

Comparing the result of two almost same flanges with Spiral Wound gasket, is representing that the weight of flange is less in design with EN 1591. Thickness of flange is approximately same. Then the only difference is on bolt area, because EN 1591 needs the smaller bolt size.

For Kamprofile the weight of flanges is almost same, but Annex G needs greater bolt size. And again EN 1591 needs smaller bolt size like spiral wounded.

But in metal-jacketed, although the weight in both methods is same, but the bolt size for EN 1591 is greater.

All in all, for spiral wounded and Kamprofile the proposed method is EN 1591, but for metal-jacketed the suggested method is Annex G.

3. FEA model and setup

3.1. Computational setup

Figure 11. represents the complete model which is composed of bolt and nut, gasket, and mating flange. The joint flange has been modeled in Axipro©. Joint geometry is same as the DBA. 3 types of gaskets have been modeled and the pressure and temperature are same.

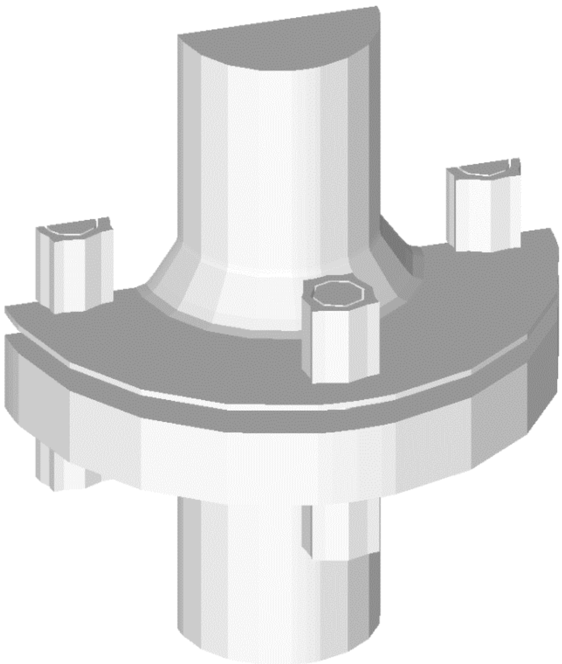
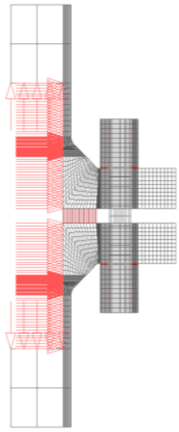
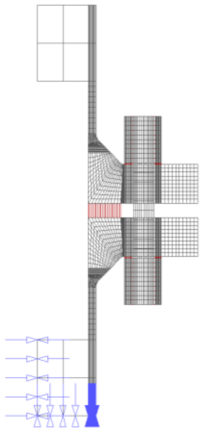


Figure 11. Schematic model of DN 50 Flange in Axipro

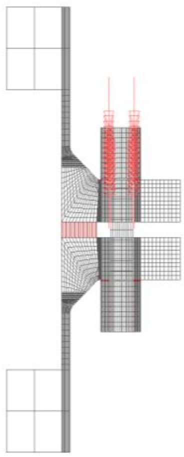
Boundary conditions have been represented in Figure 12. Inside surfaces, have been set as pressure, top surface of Flange is set as Fixed support and the pretension load has been considered for bolt. The symmetric model has been considered for reducing the convergence time. Hence the side walls are set as Fixed support.



Internal Pressure



Fixed support



Pretension Bolt Load bolt

Figure 12. Boundary Conditions

In Axipro, the standard default mesh has been used. The type of mesh is Quadratic Mapped.

and the number of mesh is 1437.

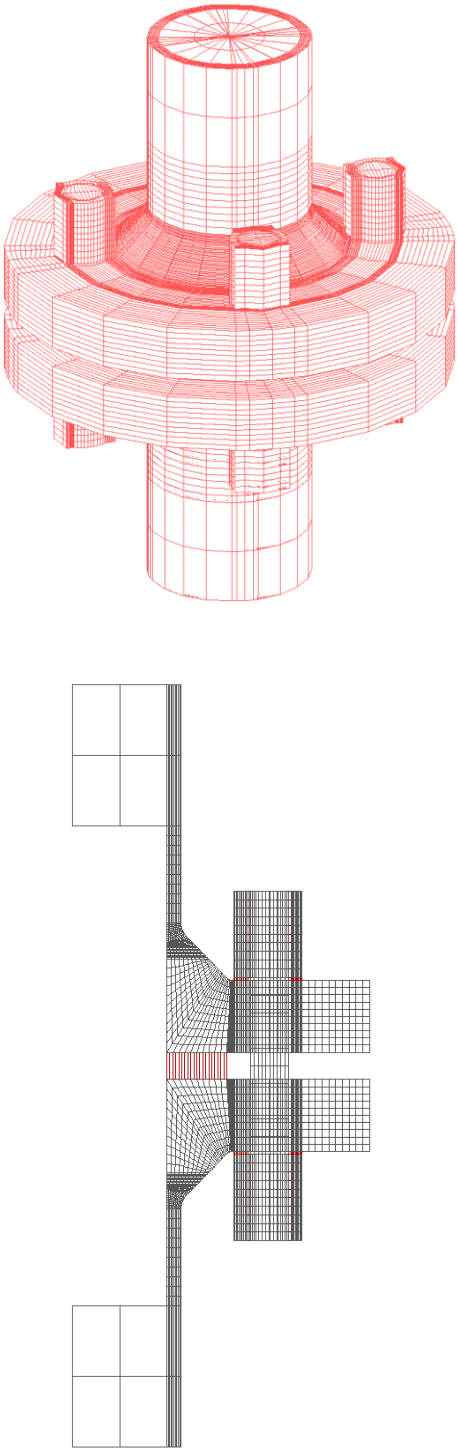
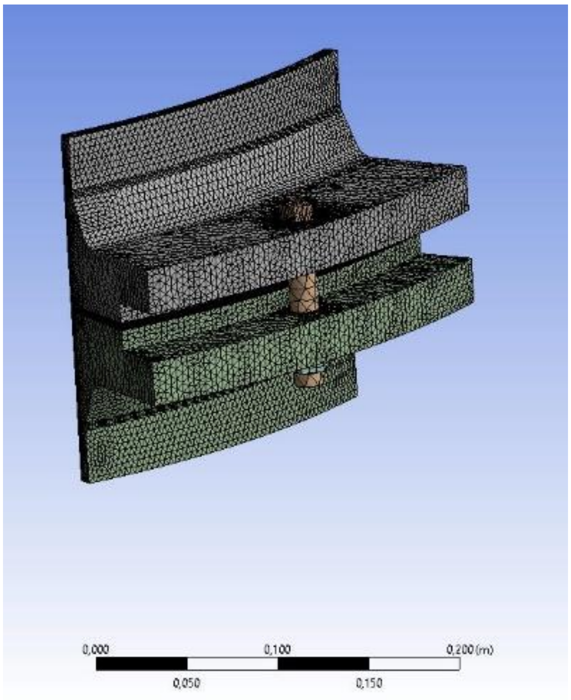


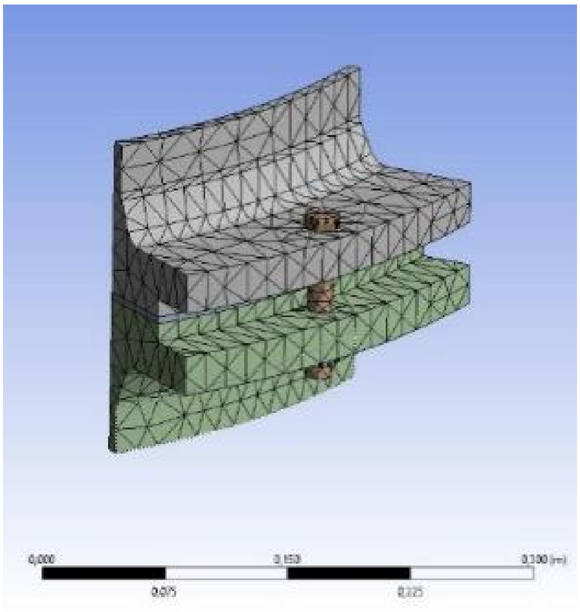
Figure 13. Quadratic Mapped

3.2. Mesh Independency validation

Grid independence verification is an important aspect to evaluate the validity of numerical calculation results. In this case, in order to study the influence of grid number on the calculation results, the whole under two grid numbers is carried out. The calculation results are shown in Figure 14, where the number of grids of (a) Mesh grid number: 160000) is about 160,000, and (b) is when the number of grids is about 80,000. It can be seen apparently when the grid number is doubled, the calculation results are not significantly different, thus verifying the grid independence. To reduce the calculation workload, the number of grid cells in the calculation is controlled at about 80,000. The grid distribution is shown in Figure 14.



a) Mesh grid number: 160000



b) Mesh grid number: 80000

Figure 14. Mesh Independency

3.3. Numerical validation

Symmetrical model has been considered in FED. For the evaluation, DN800 size flange is modeled in Ansys©. Then results of stress compared with Axipro. Error for Spiral wound gasket is approximately less than 5% (max stress in Ansys: 143.31 Mpa, Axipro: 125.281 Mpa, Allowable: 206.700 Mpa). Due to the reasonable error, all results are reported in Axipro©.

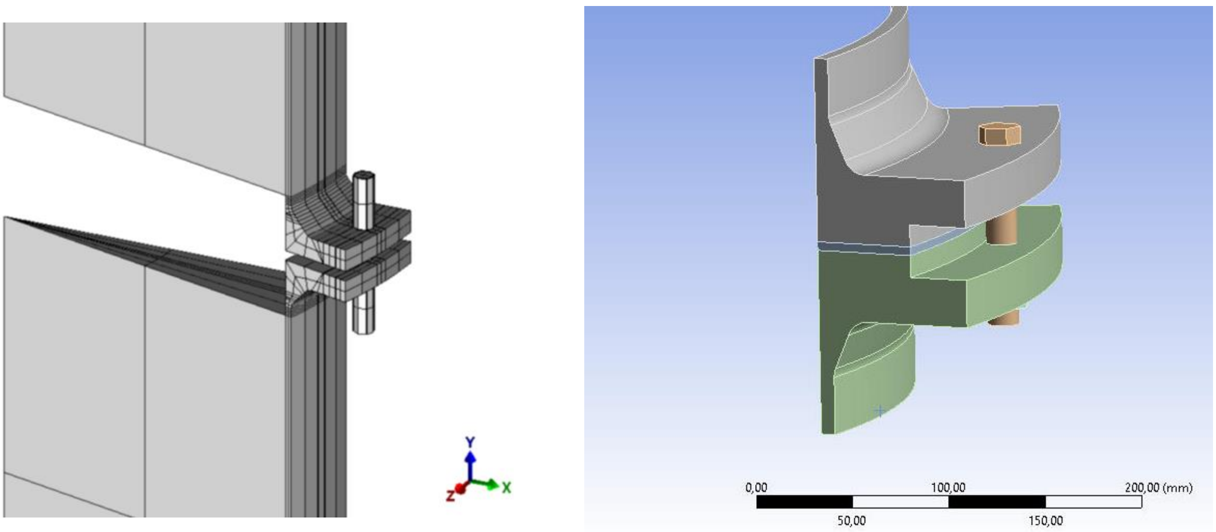
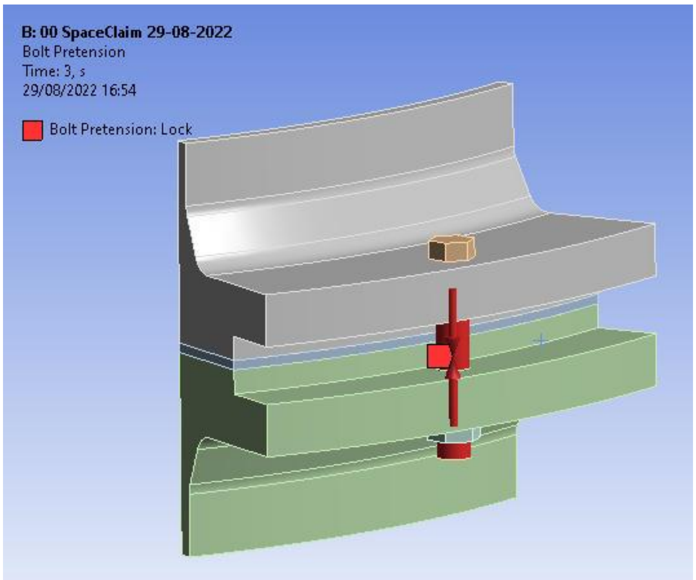


Figure 15. Schematic of Flange in: Right: Ansys left: Axipro

Load and Boundary Condition:

In Ansys, Internal pressure and bolt tightening are considered as the loads. Boundary conditions have been shown in Figure 16. two sides are set as Fixed.



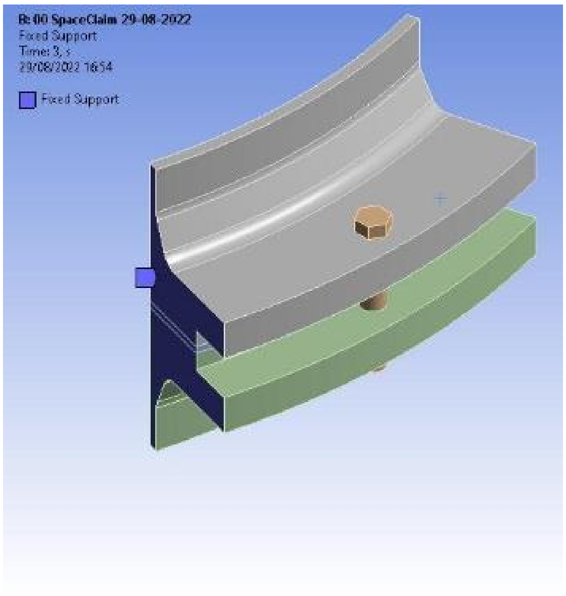


Figure 16. Boundary Conditions and Loads

For modeling gasket in Ansys and Axipro, 3D Swept Mesh shall be used to create INTER195 gasket element. To mesh the structural components on either side of the gasket, SOLID185 elements shall be used. Regarding the gasket's material, Non-linear model with Unloading (Closure vs Unloading slope) is implemented. The gasket data has been selected from www.gasketdata.org data base. Mesh type for the flange and bolt is Hex Dominant, Figure 17.

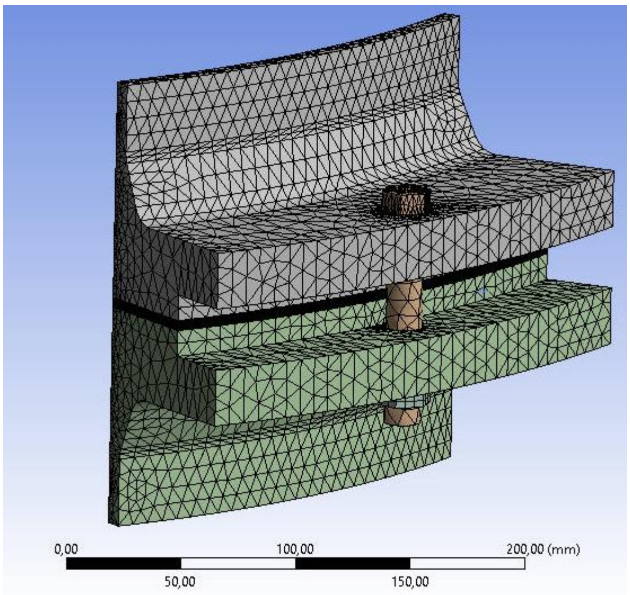
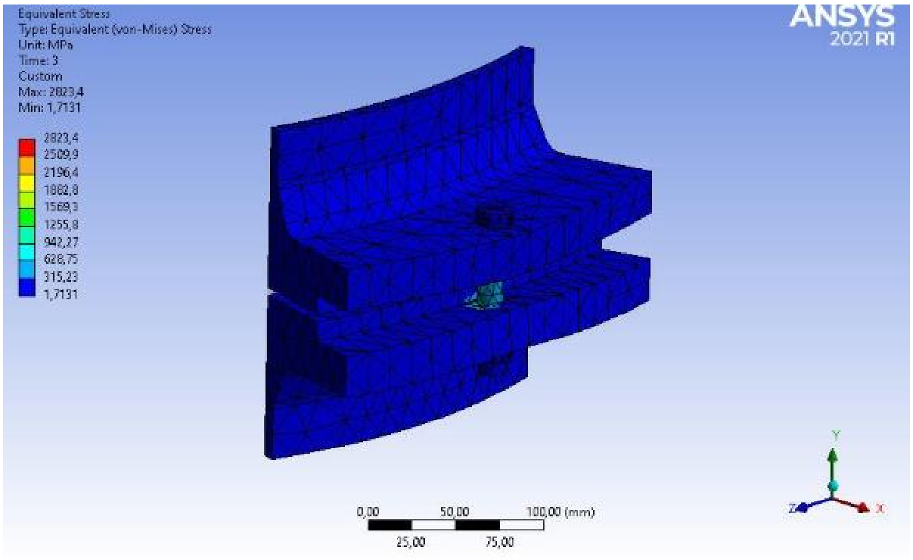
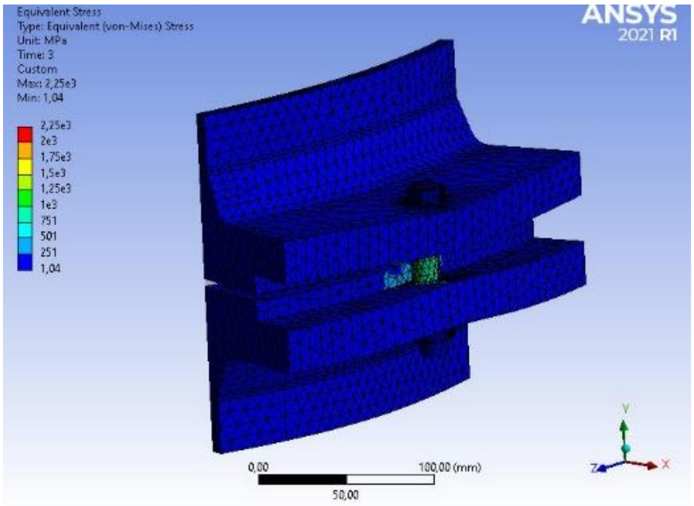


Figure 17. Mesh Schematic

Number of total elements is: 2865. For checking mesh independency, the number of elements is increased to 11951 and the stress result is shown in **Error! Reference source not found.**, then the results are not significantly different.



Number of total elements is: 2865



Number of total elements is: 11951

Figure 18. Mesh Independency

3.5 Results and discussion

Like the analytical method, 3 types of gaskets with two parameters (Bolt size and Flange thickness) are compared.

Considering the spiral wound type, Figure 19. shows the result for different bolt size. Annex G needs higher bolt size compared to EN 1591. therefore, design based on EN 1591 may cost less including material and manpower. But, due to the higher bolt size in Annex G, design based on Annex G may results the higher bolt loads and the consequence is lower gasket and bolt seating which is more beneficial.

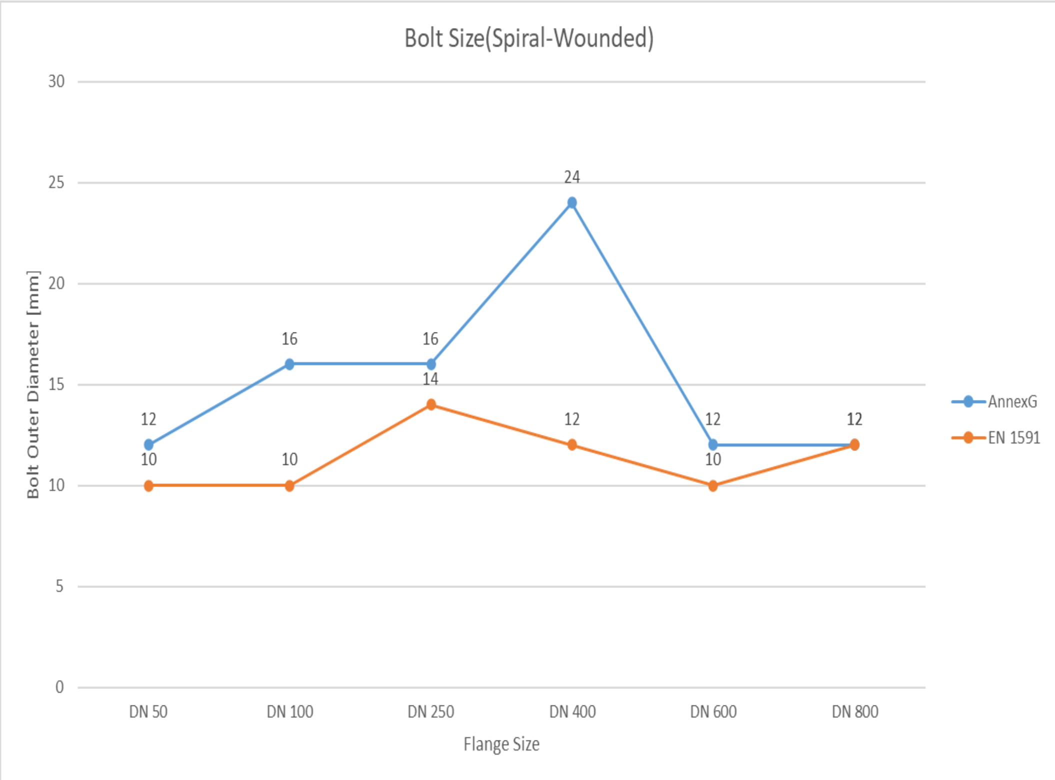


Figure 19. Comparison of bolt size of Spiral Wounded gasket, designed by Annex G and EN 1591

In addition, Figure 20. is represented the weight comparison for Spiral wounded gasket design by Annex G and EN 1591. As it expected from Figure 19. the total weight design based on Annex G is greater than EN 1591 design based. And this is due to the higher bolt size. Then comparatively, the cost of manufacturing is higher than EN 1591, but the flange can tolerate higher loads.

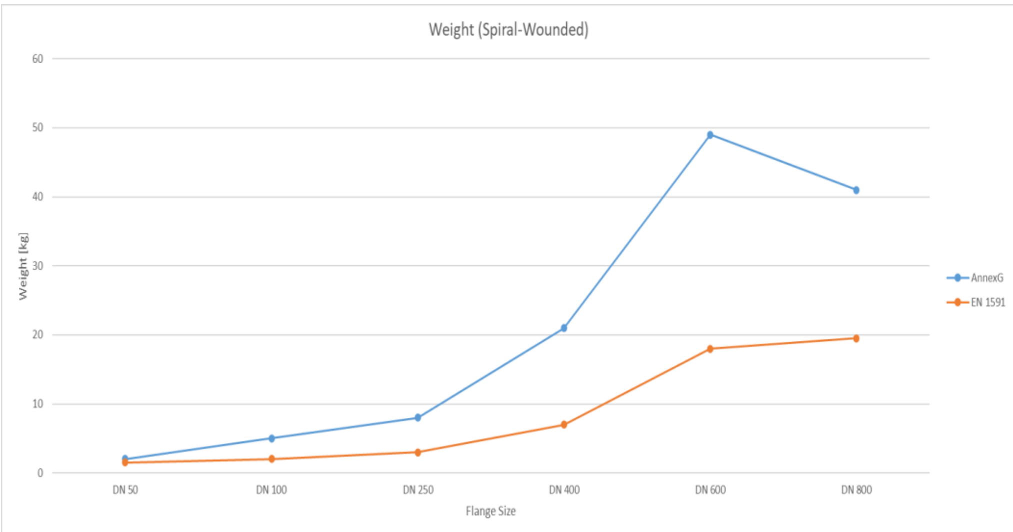


Figure 20. Comparison of weight of Spiral Wounded gasket, designed by Annex G and EN 1591

The outcome for Camprofile gasket is shown in Figure 21. Demonstrating bolt size functionality shows again the higher size bolt for Annex G but the total weight is approximately same as the EN

1591 in Figure 22. This happened because there were some changes in other parameter like flange thickness and hub thickness.

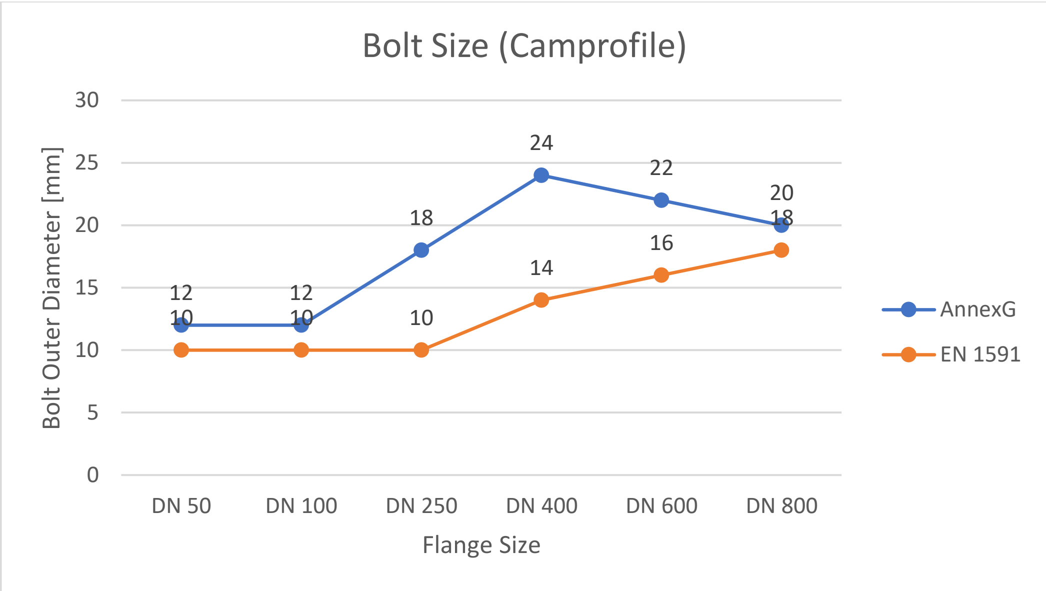


Figure 21. Comparison of bolt size of Camprofile gasket, designed by Annex G and EN 1591

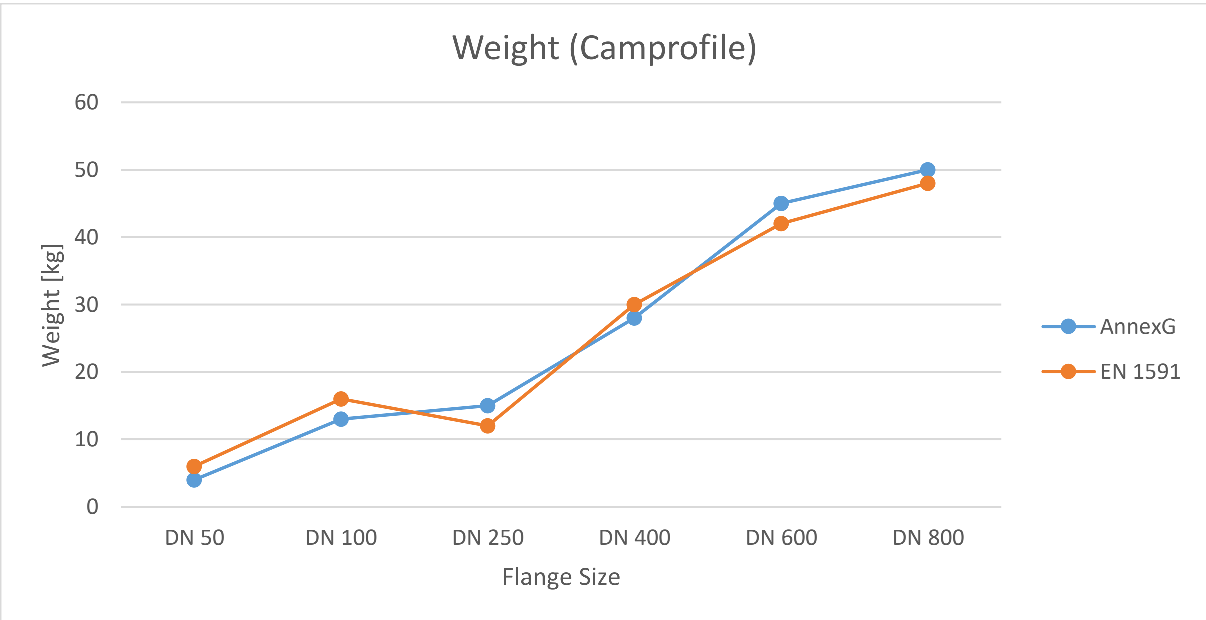


Figure 22. Comparison of weight of Camprofile gasket, designed by Annex G and EN 1591

Considering the cost and manpower, the Camprofile gasket, does not depend on the method of design significantly. But if requires controlling leakage rate according to a normative, for instance VDI 2290, the EN 1591 shall be implemented.

Discussing on Metal Jacketed gasket, shows the versus results than the two other gasket type. The bolt size is less based on design with Annex G than the EN 1591. This is because of the gasket properties.

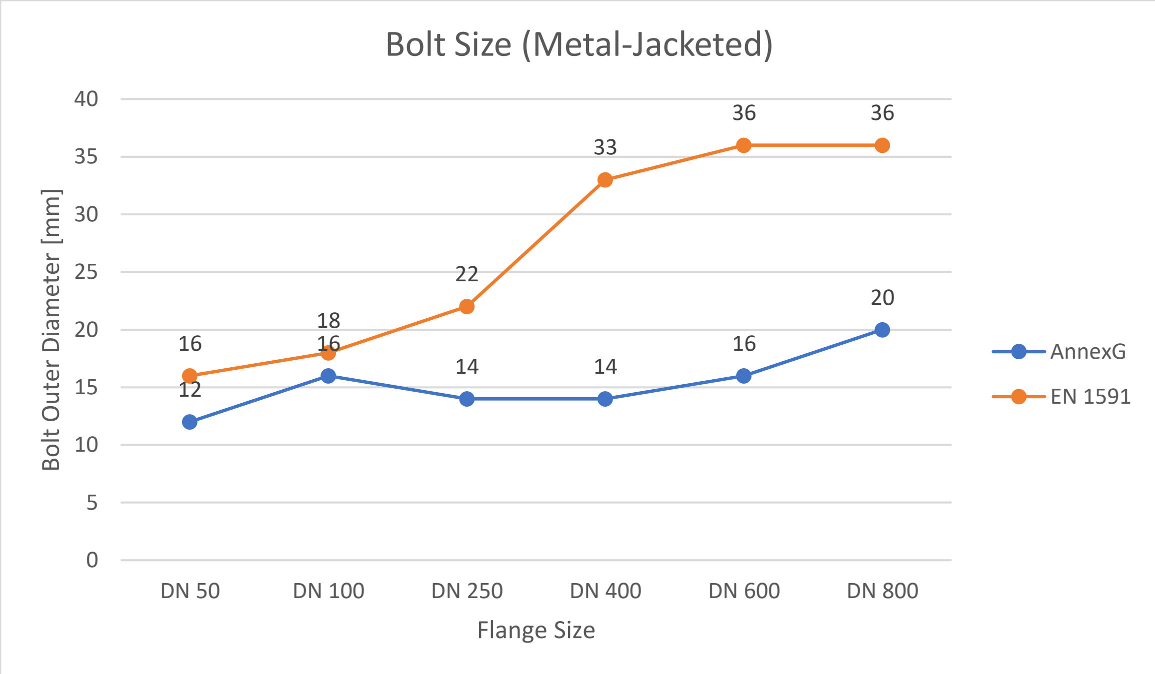


Figure 23. Comparison of bolt size of Metal jacketed gasket, designed by Annex G and EN 1591

Weight comparison for Metal Jacketed is represented in Figure 24. It is shown that there is not any significant change in the weight designed based on both methods.

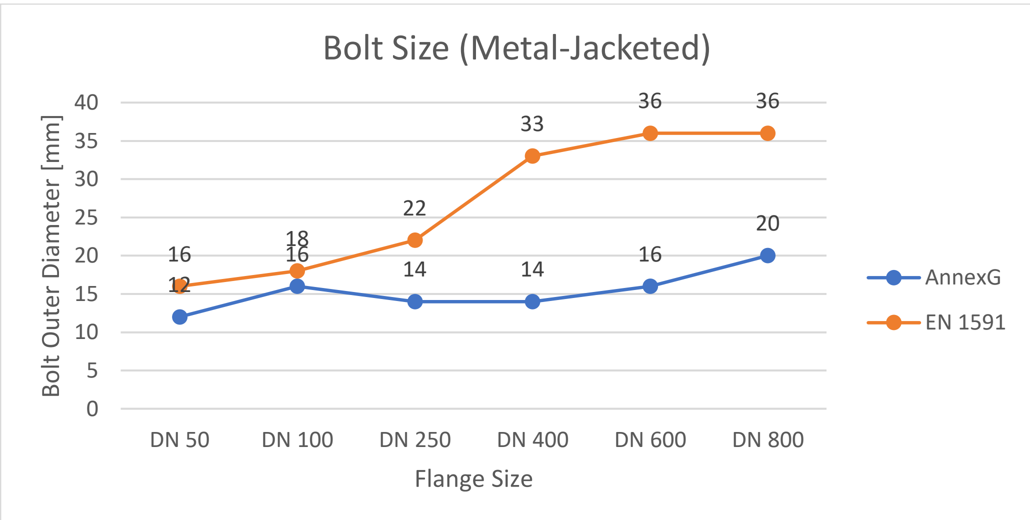


Figure 24. Comparison of weight of Metal jacketed gasket, designed by Annex G and EN 1591

4.Cmparison Between Analytical and FE methodologies

Variety of factors are affecting the bolted joint flange. Then investigation of all parameters together, needs so much effort. In this work the bolt size and flange weigh compared in both analytical and finite element methods. In chapter 2 analytical results discussed, then in chapter 3 finite element discussed and then in this chapter a brief comparison of both analytical and finite element is discussing.

4.1 Influence of the Bolt Size

Bolt size and bolt forces are important because it is influencing the flange seating and bolt seating. In another hand, the effective bolt area is a very important factor in designing the flange.

The other side of design is cost, which is simply function of weight. Then as far as the bolt size is higher, the weight is higher. Then the manhour and cost of design will be higher. Then the design shall be optimized as the manner of cost, but it must satisfy the minimum required for leakage and consequently the seating.

In Figure 25 bolt size comparison for spiral wounded gasket designed according to both DBF and DBA are represented. As it is apparently shown, for both DBA and DBF the bolt size is very similar. A little greater bolt size has been reported for DBF because in some case the seating was not satisfied by the same bolt size. Then all in all, the main issue in designing by DBF is seating, which will affect both bolt size and flange thickness.

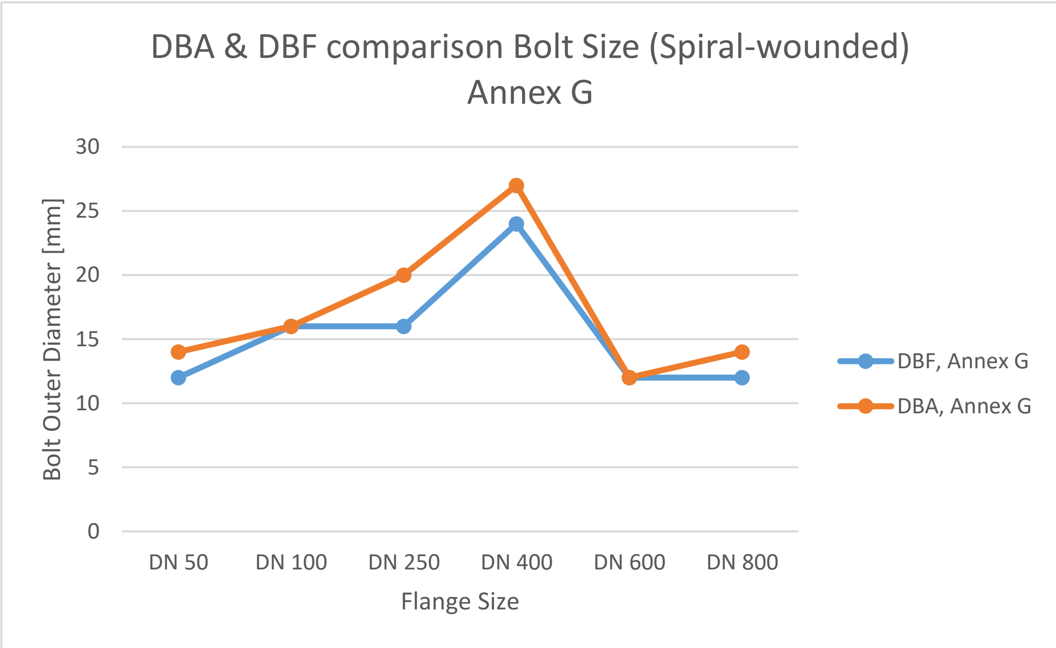


Figure 25. Comparison of Bolt Size for Spiral-wounded gasket designed by DBA and DBF Annex G

Regarding to EN 1591 spiral wounded gasket, in Figure 26 the bolt size for spiral wounded gasket, DBA and DBF for EN 1591 are illustrated. For both methodology the size is very same. Again, the small differences are due to the seating problem which in DBF is faced. Then the higher bolt size needs for solving the seating issues.

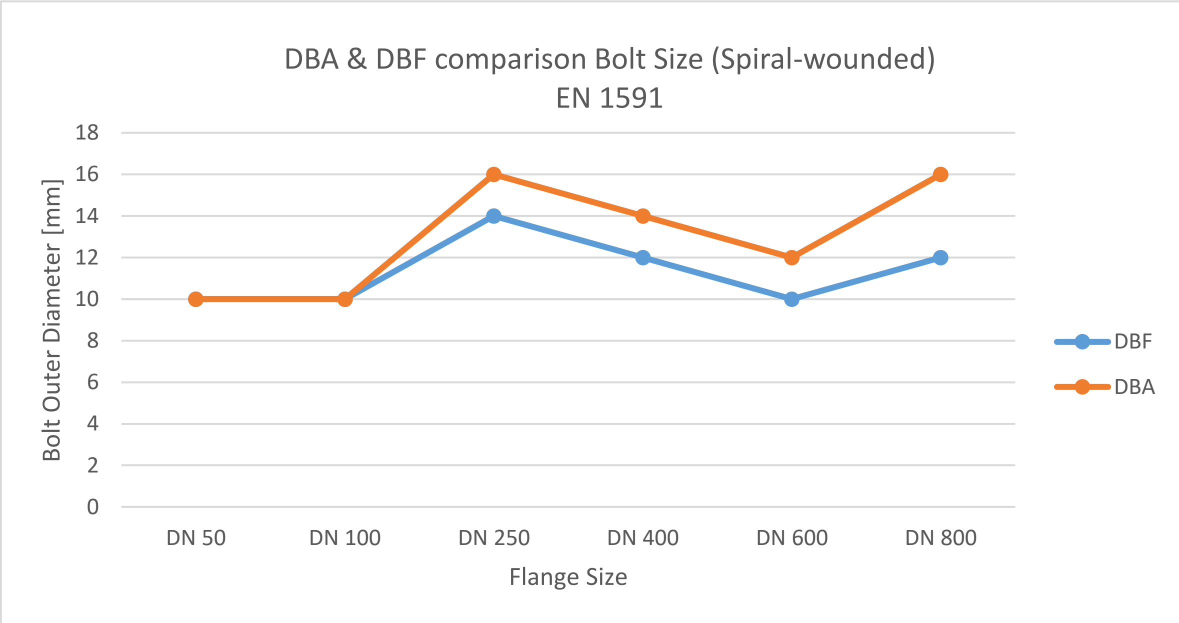


Figure 26. Comparison of Bolt Size for Spiral-wounded gasket designed by DBA and DBF EN 1591

In Figure 27 the results for Camprofile gasket designed by DBA and DBF are demonstrated. Annex G is the methodology, and as it is shown, the bolt size for DBA is higher than the DBF, because the seating issue for the Camprofile gasket is not significant then smaller bolt size can satisfy the seating requirements.

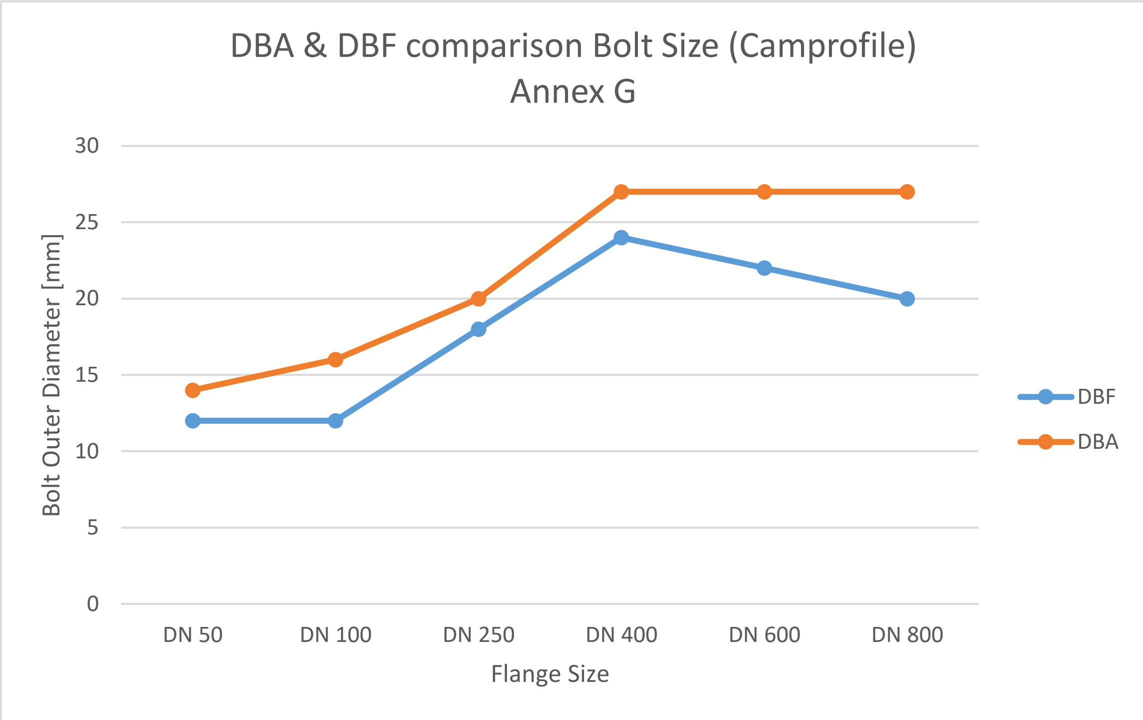


Figure 27. Comparison of Bolt Size for Camprofile gasket designed by DBA and DBF Annex G

In the Figure 28, the bolt size designed by DBF and DBA with EN 1591 is represented. The results are very similar to trend of Annex G. DBA needs higher bolt size to satisfy the seating and leakage requirements, while the DBF is satisfying with the smaller bolt size rather than DBA.

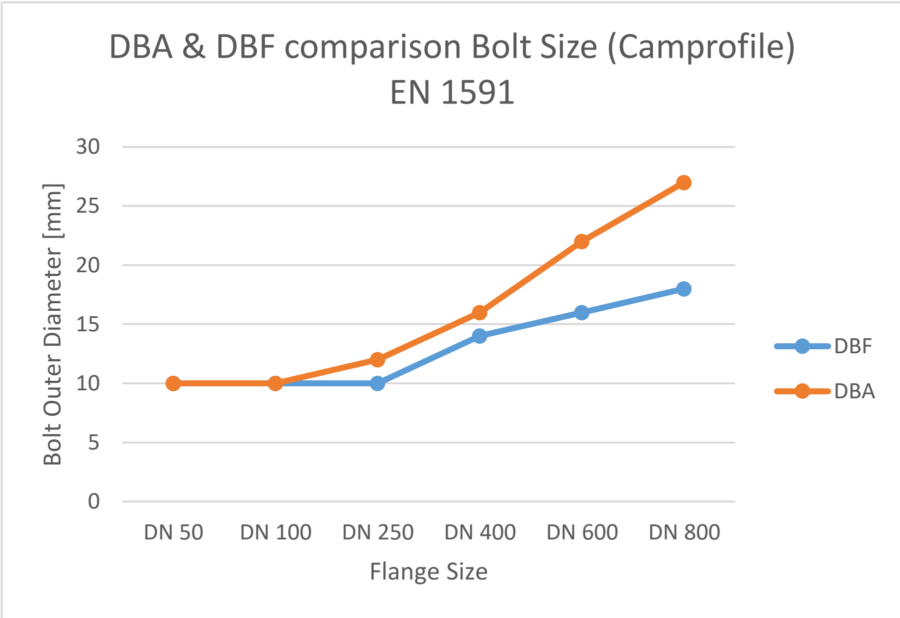


Figure 28. Comparison of Bolt Size for Camprofile gasket designed by DBA and DBF EN 1591

The last type, Metal Jacketed, of gasket bolt size result is shown in Figure 29. As it is shown in Figure 29, the bolt size designed by DBA is greater than the ones are designed by DBF. The trend is almost similar to the Camprofile. The seating issues are not significantly demonstrated by DBF, because of the characteristic of the gasket.

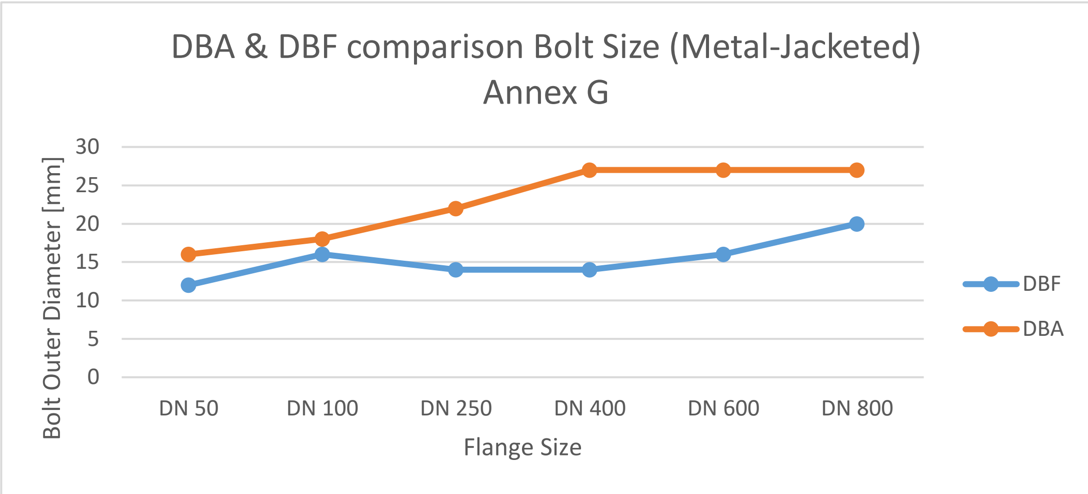


Figure 29. Comparison of Bolt Size for Metal Jacketed gasket designed by DBA and DBF Annex G

As it is shown in Figure 30, the required bolt size for DBA is higher rather than the DBF for metal

Jacketed Gasket with EN 1591.

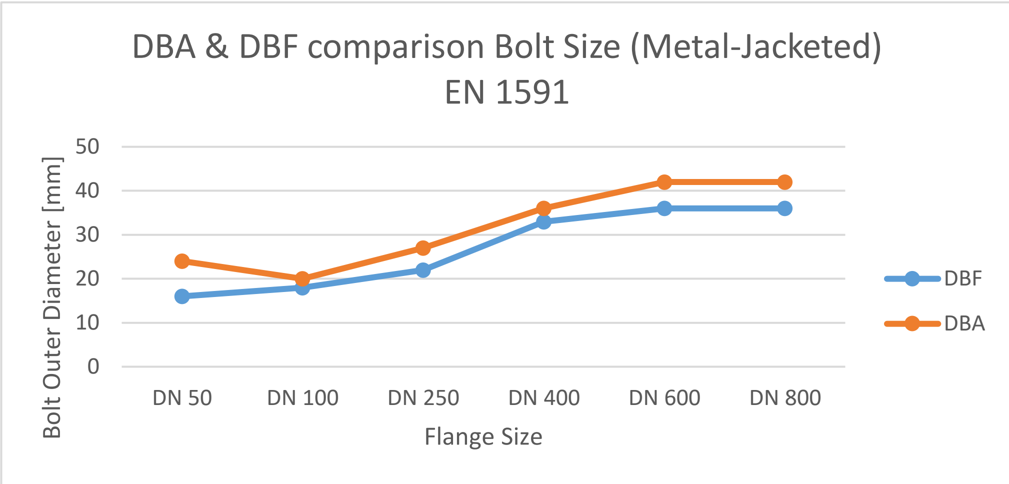


Figure 30. Comparison of Bolt Size for Metal Jacketed gasket designed by DBA and DBF EN 1591

4.2 Influence of the Flange Weight

As it is mentioned, there are versatile parameters which are affecting the elastic behavior of bolted joint flanges. The simple method to investigate the whole parameters are overall weight. Considering the weight, the overview of the construction will be available.

In Figure 31, the weight of the flanges by DBF and DBA with Annex G is represented. The weight in DBA is higher because the bolt size is higher than the DBF. For spiral wounded type, we are not facing any significant seating issues in DBF, but for DBA, the major issue is ratio of the flange thickness and bolt size.

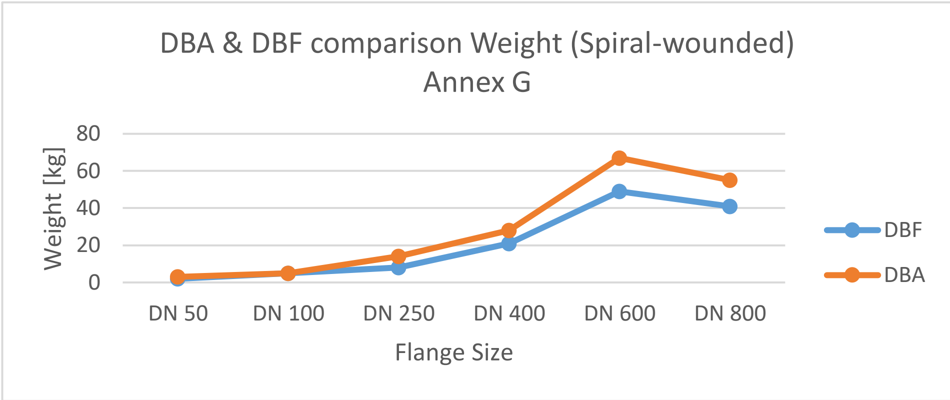


Figure 31. Comparison of weight for Spiral wounded gasket designed by DBA and DBF Annex G

In Figure 32, the weight of the flanges by DBF and DBA with EN 1591 is represented. The weight in DBA is higher because for spiral wounded the bolt size is higher than the DBF. For spiral wounded type, we are not facing any significant seating issues in DBF, but for DBA, the major issue is ratio of the flange thickness and bolt size.

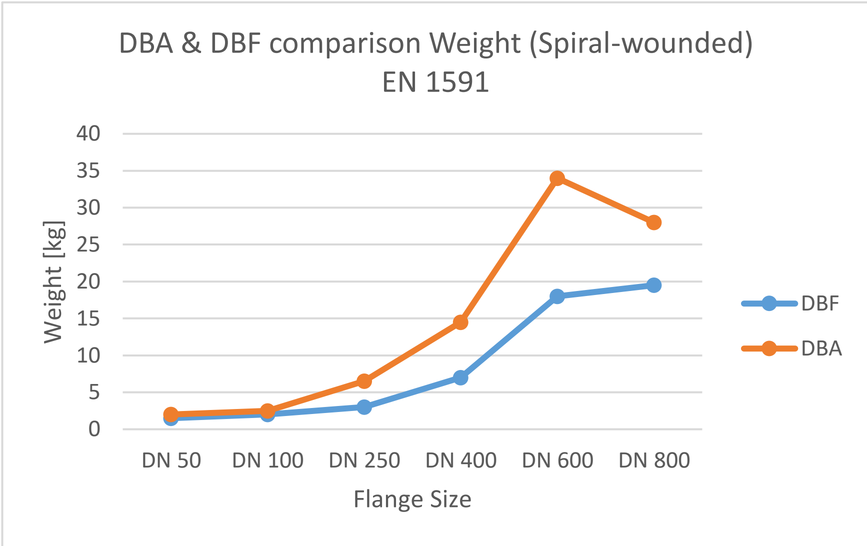


Figure 32. Comparison of weight for Spiral wounded gasket designed by DBA and DBF EN 1591

In Figure 33, the weight of the flanges by DBF and DBA with Annex G for Camprofile gasket is represented. The weight for both method is very similar. The thickness of flange and hub is higher to compensate the seating issues for DBF, then the weight is very similar for both methods.

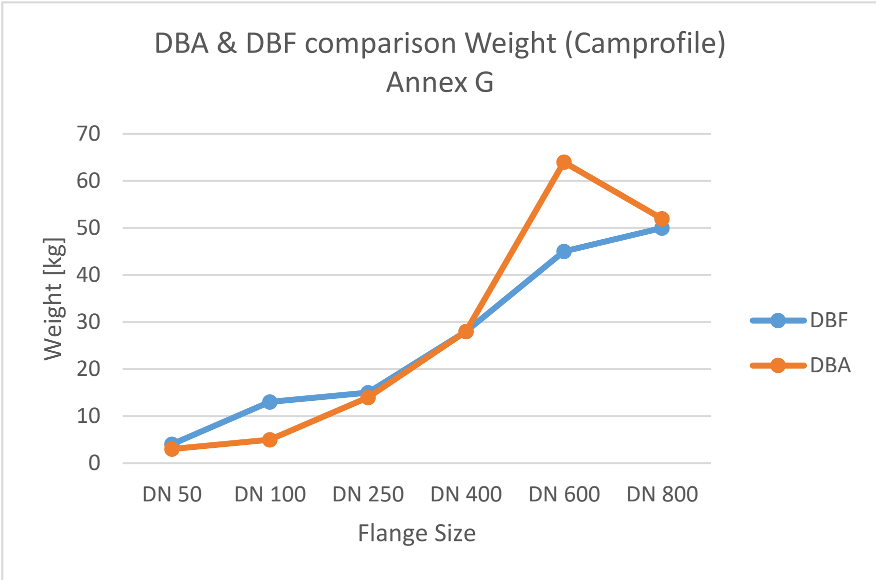


Figure 33. Comparison of weight for Camprofile gasket designed by DBA and DBF

Annex G

In Figure 34, the weight of the flanges by DBF and DBA with EN 1591 for Camprofile gasket is represented. The weight for both method is very similar also. The thickness of flange and hub is higher to compensate the seating issues for DBF, then the weight is very similar for both methods.

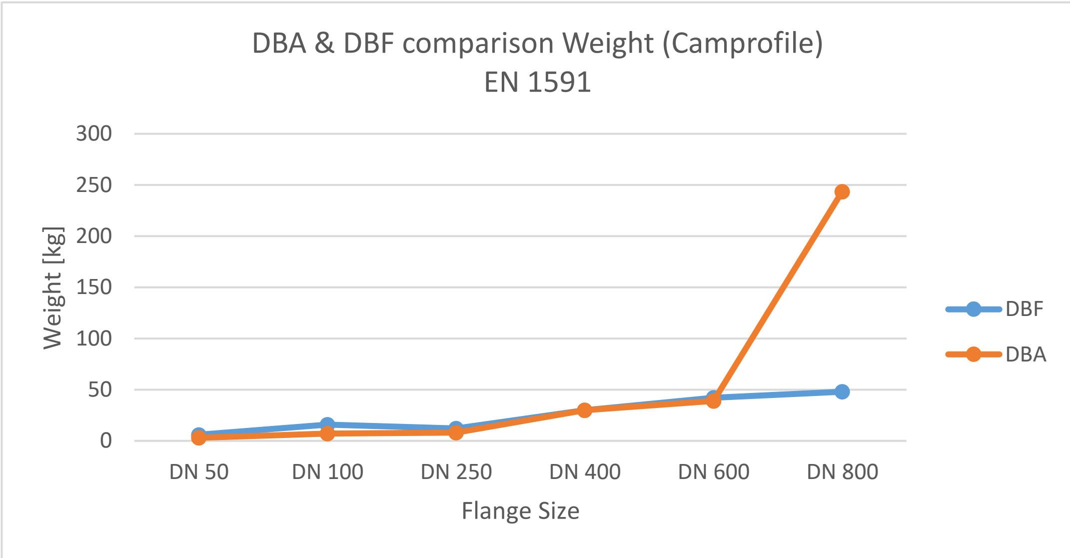


Figure 34. Comparison of weight for Camprofile gasket designed by DBA and DBF EN 1591

In Figure 35, the weight of the flanges by DBF and DBA with Annex G for Metal Jacketed gasket is represented. The weight for both method is very similar also. The thickness of flange and hub is higher than spiral wounded, to compensate the seating issues for DBF, then the weight is very similar for both methods.

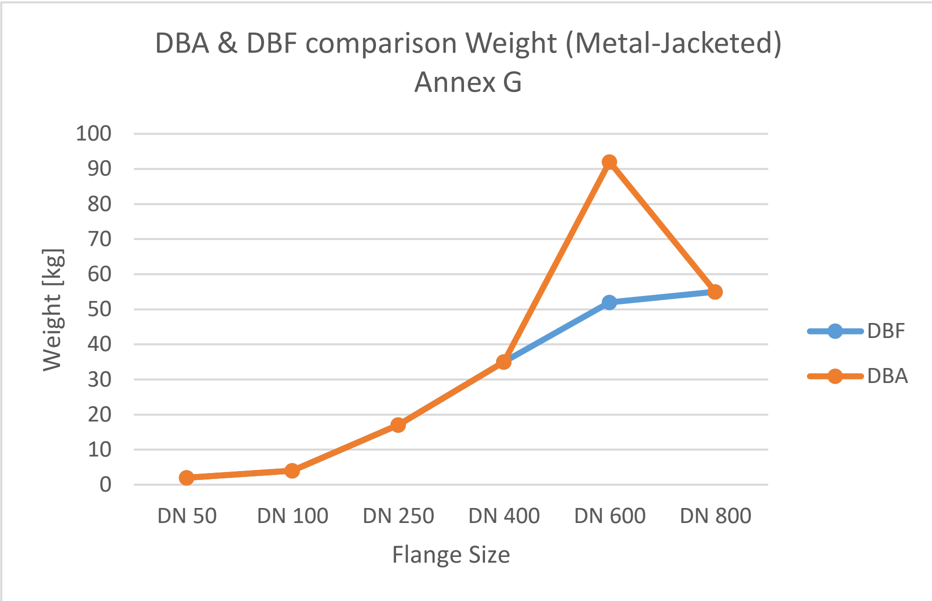


Figure 35. Comparison of weight for Metal Jacketed gasket designed by DBA and DBF Annex G

In Figure 35, the weight of the flanges by DBF and DBA with EN 1591 for Metal Jacketed gasket is represented. The weight for both method is very similar also. The thickness of flange and hub is higher than spiral wounded, to compensate the seating issues for DBF, then the weight is very similar for both methods.

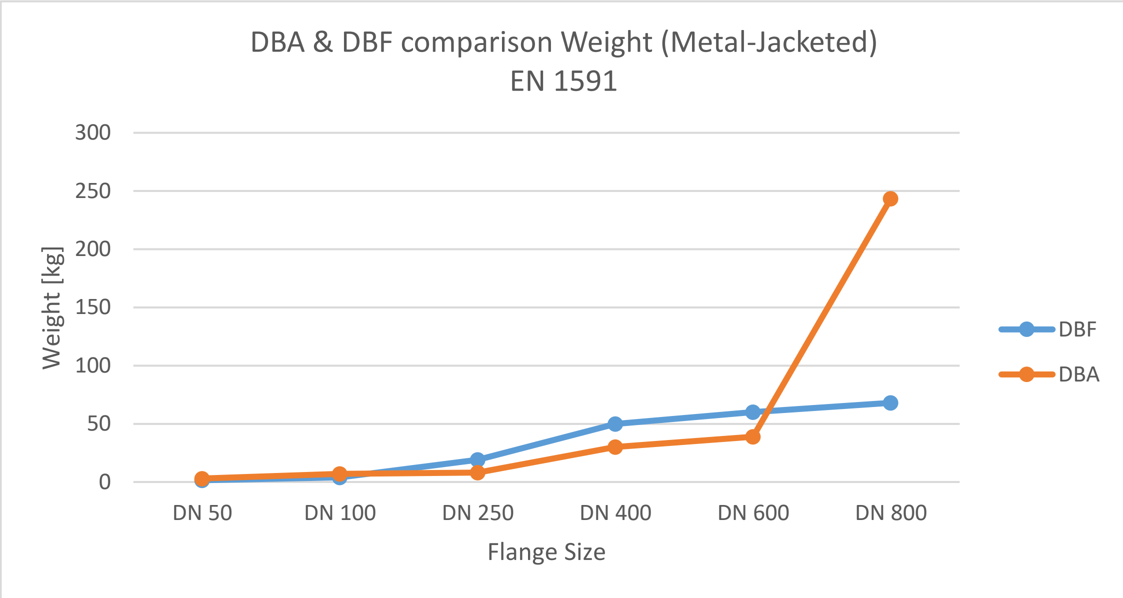


Figure 36. Comparison of weight for Metal Jacketed gasket designed by DBA and DBF EN 1591

5. Conclusions

This work has discussed the performance of designing of flange, comparing EN 13445 Annex G and EN 1591. Performance, regarding both analytical and finite element analysis l methods have been investigated. The main affecting parameters such as bolt size and flange thickness have been discussed. The results obtained allow us to compare the performance of the bolted joint flanges for different applications like heat exchanger and to study the functioning of tightness class at different size and type of gaskets. The main conclusions are:

(1) Both methods achieved almost the same functionality regarding the leakage level.

(2) Concerning the Spiral wound gasket, both normative have been demonstrated the same performance.

(3) The Annex G had better action on designing of Kamprofile gaskets. But with regard to the Metal jacketed, probably using of EN 1591 presents the enhanced results and lower costs.

(4) Design with Annex G proved the less weight than EN 1591 and consequently the cost of manufacturing is lower generally.

(5) The major convergence problem in designing with DBF is gasket seating while in DBA is unity of flange and bolt.

Acknowledgement

On the completion of the thesis, I would like to thank those who helped me with my graduate study and thesis during this period.

Thanks to Prof. Cristiana Delprete, Dr. Michele Ferracci and Mr. Luca Gaetani for their careful and key guidance during my thesis writing.

Thanks to the hard work of the family, thanks to the girlfriend ’s considerate support and heart-warming encouragement, she is almost the source of all my motivation.

Thanks to my colleagues Francesco Faraco, Alessandro Masia and other my coworkers in Eleo2.

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