MODELING ANALYSIS OF THE VALVE FLANGE WITH OCTAGONAL RING GASKET UNDER THE INFLUENCE OF TEMPERATURE

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In flanges, which operate at high temperatures and high pressures, such as in the petrochemical industry, relief of material stresses and thermal deformation are produced due to temperature increase, and as a result, the sealing pressure of the gasket is weakened, and leaks can occur. In this paper, using ANSYS 19.2, the tightening forces and torques of the flange bolts were determined according to the temperature changes for each flange material. The optimal functions for calculating the tightening force and torque of the bolt according to the temperature change were determined using the analysis data and method of least squares in Excel.

Keywords: Parallel gate valve, Flange, Tightening force of the bolt, Gasket, Torque.

1. Introduction

Bolted flange connections are widely used in the process industries as a connecting element due to their simple structure and light construction. As the tightening force of the bolt at these flanges has a direct effect on the leakage, various conditions must be taken into account when calculating this tightening force.

From this, many researchers have conducted studies to determine the tightening force of the flange using the finite element method, and good research results have been introduced in this process. In paper [1], the flange leakage was evaluated using the finite element method (FEM). Using thermal limitation conditions, they confirmed that, in the flanges of different sizes subjected to internal pressure and thermal load, the 4-inch flange has the highest leakage. *Vishwanath et al*. [2] analyzed the stress distribution of the flange assembly elements according to the tightening force of the bolt and the thickness of the flange using ANSYS and verified that the deviation of the ANSYS analysis results is in the range of 10-20 % through comparison with numerical calculations.

In paper [3] is investigated the influence of operating conditions on the compression pressure of the flange gasket, which was designed in accordance with the standard EN 13445-3 Annex G, using ANSYS. Through analysis, they

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confirmed the improper assembly of the flange gasket that is currently in operation and also found that if the temperature distribution is not uniform, the compression pressure of the gasket decreases.

Wenbo et al. [4] evaluated the structural strength and sealing performance of the DN600 bolt flange using a finite element analysis model provided by ABAQUS software. Through analysis, they confirmed that the seal stress of gasket decreases significantly after creep and the stress at this time increases almost linearly from the inside of the gasket to the outside.

In this way, many studies were performed to analyze the sealing performance and structural strength of a flange with a gasket using the finite element method, and positive research results were introduced in this process, but the relationship between the working temperature and the tightening force and torque of the flange was not observed.

Thus, this paper aims to reveal the relationship between the working temperature and the tightening force of the bolt that directly affects the flange leakage, using the finite element method offered by ANSYS. For this, at first, using the calculation equations that reflect the relationship between temperature and mechanical properties of metallic materials, proposed by *Luecke et al*. [5], the values of change in the mechanical properties of the flange assembly elements depending on the temperature increase is determined. Based on these changes, the internal stresses of the flange assembly elements depending on the temperature increase are also analyzed. Using the method of least squares offered by Excel, the relationships between the tightening force of the bolt and the internal stress and between the tightening force of the bolt and the working temperature are determined.

2. Determination of flange material parameters depending on temperature

In general, parameters that evaluate the mechanical properties of a material, such as tensile strength, yield strength, and Young's modulus, are dependent on the temperature. From this, *Luecke et al*. [5] performed an experiment to reveal changes in the mechanical property parameters of steel materials in accordance with the temperature change using nine types of steel materials commonly used in industrial construction and introduced the calculation expressions related to these.

The values of the changes of the flow limits depending on temperature increase can be calculated with the equation [5, 6]:

$$R = \frac{S_y}{S_{y(T=20°C)}} = \exp\left(-\frac{1}{2}\left(\frac{T^*}{r_3}\right)^{r_1} - \frac{1}{2}\left(\frac{T^*}{r_4}\right)^{r_2} \right) \tag{1}$$

where $S_{y(T=20°C)}$ is the yield strength of a steel material when the temperature is 20 °C, [MPa], R is the ratio between the yield strength at temperature 20 °C and the yield strength at actual temperature, T^* is equal to 20 °C less the real temperature: $T^* = T - 20$ °C.

The parameters for equation (1) are [5, 6]:

$$r_1 = 5.708;\ r_2 = 1.000\ ;\ r_3 = 590\ °C;\ r_4 = 919\ °C.$$

The values of the yield strength for the flange assembly elements depending on temperature increase, calculated using equation (1), are shown in Table 1.

Table 1

Calculation results of the yield strength according to the temperature increase

Temperature [°C]		20	100	200	300	400	500	600
Value R		1	0.957	0.916	0.852	0.781	0.66	0.463
Parts	Marks	Yield strength [MPa]						
Flange	ASTM A487 4C	415	397.2	380.1	353.6	324.1	273.9	192.2
Bolt	ASTM A193 B7	515	492.9	471.7	438.8	402.2	339.9	238.5
Gasket	Soft iron	235	224.9	215.3	200.2	183.5	155.1	108.8
Gasket	4-6%Cr0.5%Mo	275	263.2	249.2	234.5	214.7	181.6	127.4
Gasket	AISI 304	205	196.3	185.8	174.8	160.1	135.4	95
Gasket	AISI 316	205	196.3	185.8	174.8	160.1	135.4	95
Gasket	AISI 347	205	196.3	185.8	174.8	160.1	135.4	95
Gasket	AISI 410	290	277.6	262.8	247.3	226.5	191.5	134.4

The Young's modulus of the steel material is also affected by temperature, and its size can be calculated with the equation [5, 6]:

$$E = E_0 + e_1 \cdot T + e_2 \cdot T^2 + e_3 \cdot T^3 \tag{2}$$

where E_0 is the Young's modulus of the material at room temperature.

Of course, the Young's modulus is slightly different depending on the type of steel, but because the difference is so small that it has no effect on the analysis, to simplify the analysis, it was assumed that the Young's modulus is almost constant regardless of the type of steel and is 190 GPa at temperature 20 °C. The values of the Young's modulus of the flange parts according to the temperature increase, calculated using equation (2), are presented in Table 2.

Table 2

Calculation results of the Young's modulus according to the temperature increase

Temperature [°C]	20	100	200	300	400	500	600
Young's Modulus [GPa]	190	185.26	179.42	172.09	162.87	151.38	137.2

On the one hand, according to the law of elasticity, the tensile strength of the material is determined by the equation:

$$\sigma = E \cdot \varepsilon \tag{3}$$

That is, the tensile strength is dependent on the elasticity of the material.

The values of the tensile strength depending on the temperature increase, calculated using equation (3), are presented in Table 3.

Table 3

Calculation results of the tensile strength according to the temperature increase

Temperature [°C]		20	100	200	300	400	500	600
Parts	Marks	Tensile strength [MPa]						
Flange	ASTM A487 4C	620	603.9	584.9	561	531	493.5	447.3
Bolt	ASTM A193 B7	690	672.5	651.3	624.7	591.2	549.5	498
Gasket	Soft iron	370	361.3	349.9	335.6	317.6	295.2	267.5
Gasket	4-6%Cr0.5%Mo	485	472.4	457.5	438.8	415.3	386	349.9
Gasket	AISI 304	515	502.1	486.2	466.4	441.4	410.2	371.8
Gasket	AISI 316	515	502.1	486.2	466.4	441.4	410.2	371.8
Gasket	AISI 347	550	535.4	518.5	497.3	470.7	437.5	396.5
Gasket	AISI 410	510	496.5	480.8	461.2	436.5	405.7	367.7

3. Flange model design and discrete composition

The model of the flange for the analysis of stress was designed by AutoCAD based on the standard ASME PCC-1:2019 and the design drawing of the gate valve. The nuts and bolt heads that will be installed on the flange have been designed in a hexagonal shape. To shorten the analysis time, only the flange portion of the gate valve was modeled. The values of the physical properties of the flange material are presented in Table 4.

Table 4

Values of physical properties of materials for flange parts

Parts	Marks	Density	Young's modulus	Poisson's Ratio	Specific Heat	Thermal Conductivity
		[kg/m^3]	[GPa]	-	[J/kg·°C]	[W/m·°C]
Flange	ASTM A487 4C	7700	190	0.29	480	30
Bolt	ASTM A193 B7	7850	190	0.29	480	53
Gasket	Soft iron	7800	190	0.29	434	60.5
Gasket	4-6%Cr0.5%Mo	7850	190	0.29	434	60.5
Gasket	AISI 304	7900	190	0.29	500	16.3
Gasket	AISI 316	7900	190	0.29	500	16.3
Gasket	AISI 347	7960	190	0.29	500	16.3
Gasket	AISI 410	7740	190	0.29	460	24.9

For the finite elements, the tetrahedral and hexahedral solid elements were used, and the discretization was performed by the automatic division function. The geometric shape and dimensions of the valve flange are presented in Figures 1, 2 and 3.

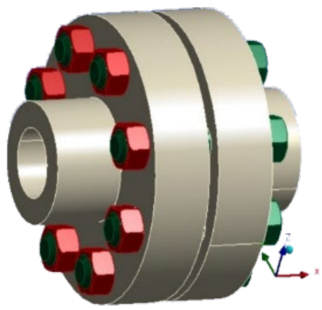


Figure 1. Geometric shape of the flange assembly

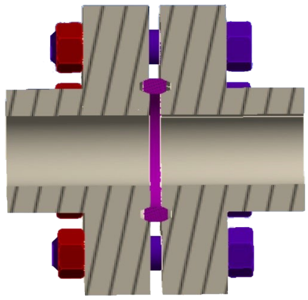


Figure 2. Flange with octagonal ring gasket

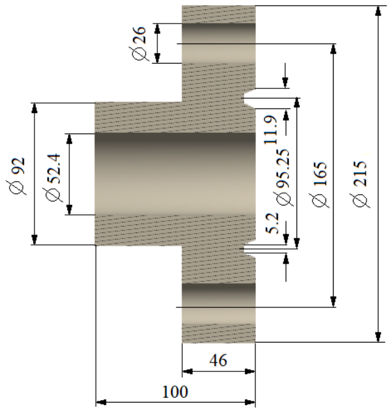


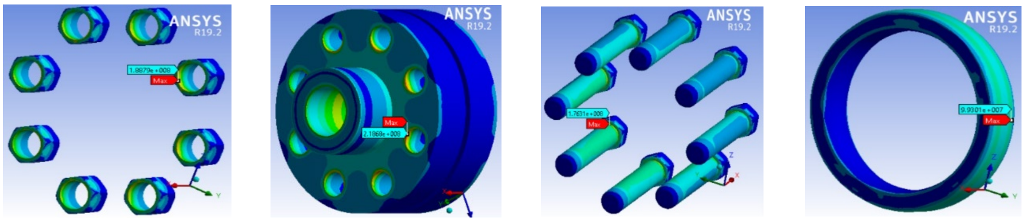
Figure 3. Geometric size of the flange

For the flange gasket, R24 from Table 3 of the ASME16.20: 2012 standard was selected, taking into account the dimensions of the valve design. In addition, for the analysis of flange stresses, M27 type alloy steel bolts were used, in accordance with the valve design and ASME PCC-1: 2019 and ASME 16.5: 2013 standards. At the discretization of the flange model, the total number of finite elements is 63232, and the total number of nodes is 195766. For structural analysis, only one of the two flanges was fixed, and the other was connected by bolts. Between the flange and the gasket and between the bolt and the nut, the “Bonded” conditions were established, and between the nut and the flange and between the bolt and the flange, the friction conditions were established. For the bolt applied with lubricant, the coefficient of friction was set to 0.16, for the bolt not applied with lubricant up to 0.2. To give the axial tightening forces to the bolts, relative coordinates were established according to each bolt. The tightening force of each bolt was applied as a volumetric load. For the internal pressure, it has been set at 35 MPa, which is the pressure of the valve completely closed which will be subjected to the maximum load.

4. Determination of the maximum tightening forces and torques of bolt

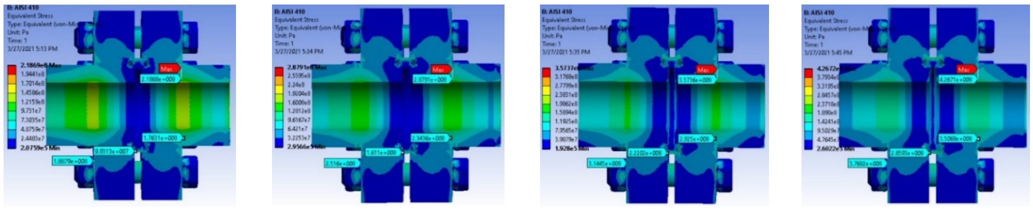
4.1. Flange with the gasket made with 4-6%Cr0.5%Mo

In order to determine the maximum tightening force of the flange with the gasket made with 4-6%Cr0.5%Mo [7], the stress analysis of the flange assembly was performed, changing the tightening force from 30 kN to 60 kN and, at this time, the working temperature from 20 °C to 600 °C. First, at a temperature of 20 °C, the stress distribution for the coated and uncoated bolts is shown in Figure 4, and the results of the analysis are shown in Table 5 and Figures 5 and 6.



Nut Flange Bolt Gasket

a) Stress distribution of each piece when the tightening force of 30 kN is applied



30 kN 40 kN 50 kN 60 kN

b) Section of the flange model according to the increase of the tightening force

Figure 4. Stress distribution in the flange model at 20 °C when the friction coefficient is 0.2

Table 5

Results of the stress analysis of the flange parts according to the tightening force of the bolt, at a temperature of 20 °C, [MPa]

Flange parts	Flange		Gasket		Bolt		Nut	
Coefficient of friction / Tightening force [kN]	0.2	0.16	0.2	0.16	0.2	0.16	0.2	0.16
30	218.7	218.8	99.3	99.3	176.3	176.2	188.8	188.7
40	287.9	288.1	161.1	161.1	234.4	234.1	251.6	251.5
50	357.4	357.7	222	222	292.5	292.3	314.5	314.4
60	*426.7*	*427.2*	*285.9*	*286*	350.7	350.5	376.8	377.1

(In the table, the values indicated in Italic font mean values that exceed the yield strength of the corresponding material.)

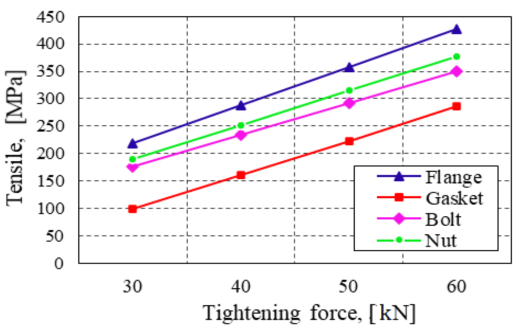


Figure 5. Maximum stresses of the Parts according to the tightening force, at a temperature of 20 °C when the friction coefficient is 0.2

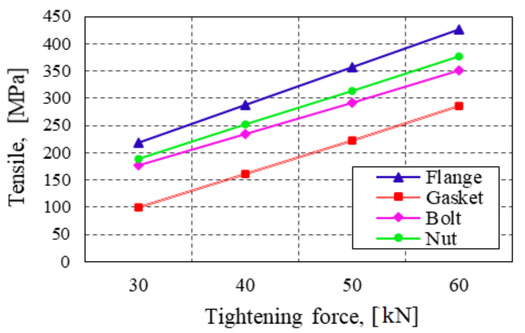


Figure 6. Maximum stresses of the Parts according to the tightening force, at a temperature of 20 °C when the friction coefficient is 0.16

As can be seen from Figures 5 and 6, the results analysis suggest that the internal stresses of the flange Parts change almost linearly as the tightening force of the bolt increases. The results also show that there is very little difference between the coated and uncoated bolts. Also, as can be seen in Table 5, the internal stresses of the flange and gasket exceed first the permissible limit of the material. In this paper, for the facilitation of analysis, for other gaskets, only the case of the uncoated bolts was analyzed. In addition, only the analyses results of the inner tension of the gasket and flange were used to determine maximum tightening forces and torques of the bolt. The internal stresses of the gasket and flange according to the temperature increase are presented in Table 6.

Table 6

Internal stresses of the gasket and flange according to the temperature increase, [MPa]

Temperature [°C]		20		100		200		300		400		500		600	
Parts		Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.
Tightening force, [kN]	30	218.7	99.3	218.7	99.2	218.8	99.2	218.8	99	218.9	98.9	219	98.7	219.1	98.3
	40	287.9	161.1	288	160.9	288	160.7	288.1	160.4	288.2	160	288.3	159.5	288.5	158.7
	50	357.4	222	357.4	221.7	357.5	221.4	357.6	220.9	357.8	220.3	357.9	219.4	358.2	218.2
	60	426.7	285.9	426.8	285.5	426.9	285	427	284.3	427.2	283.3	427.4	282	427.7	280.3

Then, using the data in Table 6 and the method of least squares offered by Excel, the equations reflecting the relationship between the tightening force and the internal stress were determined. From Figure 5, it was assumed that there is a linear relationship between the tightening force and the internal stress, and the optimal functions were calculated using the function:

$$f(\mathrm{x}) = a + b \cdot X \tag{4}$$

where $f(\mathrm{x})$ is the internal stress, MPa, X is the tightening force of the bolt, kN.

The maximum tightening force of the bolt can be calculated by the equation:

$$X_{\mathrm{max.}} = min.\ (X_{\mathrm{flange}}\ , X_{\mathrm{gasket}}) \tag{5}$$

The calculation results are presented in Tables 7 and 8 and in Figures 7 and 8.

Table 7

Coefficient values *a*, *b* and R^2 for the flange with the gasket made with 4-6%Cr0.5%Mo depending on the temperature

Temperature [°C]		20		100		200		300		400		500		600	
Parts		Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.
Coefficient	a	10.6	-87.24	10.56	-87.04	10.59	-86.57	10.53	-86.23	10.5	-85.45	10.49	-84.51	10.4	-83.6
	b	6.935	6.207	6.937	6.197	6.938	6.181	6.941	6.164	6.945	6.135	6.948	6.098	6.955	6.055
	R^2	1	0.999	1	0.999	1	0.999	1	0.999	1	0.999	1	0.999	1	0.999

Table 8

Maximum tightening forces and torques for the flange bolt with the gasket made with 4-6%Cr0.5%Mo depending on temperature

Temperature, [°C]	20	100	200	300	400	500	600
X_{flange} [kN]	58.31	55.74	53.26	49.43	45.15	37.91	26.14
X_{gasket} [kN]	58.36	56.52	54.32	52.03	48.92	43.64	34.85
Maximum tightening force, [kN]	58.31	55.74	53.26	49.43	45.15	37.91	26.14
Maximum tightening torque, [N·m]	314.9	301	287.6	267	243.8	204.7	141.2

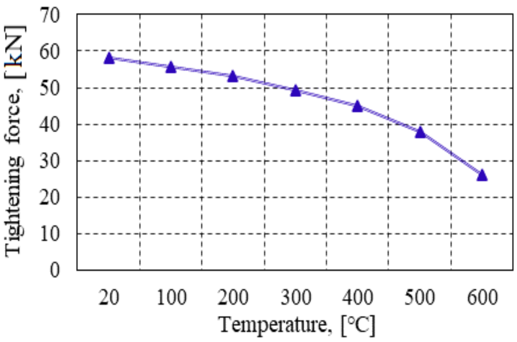


Figure 7. Maximum tightening forces of the flange bolts with gasket of 4-6%Cr0.5%Mo according to the temperature increase

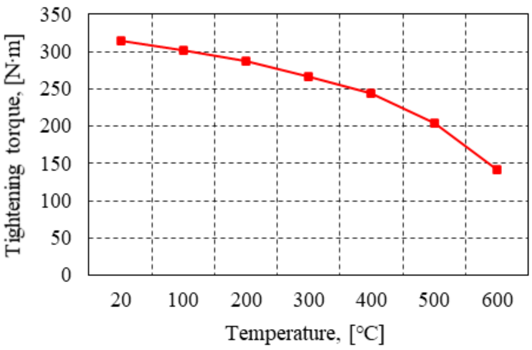


Figure 8. Maximum tightening torques of the flange bolts with gasket of 4-6%Cr0.5%Mo according to the temperature increase

In Table 8, the maximum tightening torques of the bolts were calculated by equation [8, 9]:

$$M_b = (k_b \cdot D_b \cdot F_T)/1000 \tag{6}$$

where M_b is the tightening torque on a bolt, [N·m], D_b is the nominal diameter of the bolt, [mm], F_T is tightening force on a bolt, [kN], k_b is friction coefficient for the bolted joint ($k_b = 0.2$).

The results show that the maximum tightening forces and torques of the bolt decreases nonlinearly with increasing temperature. Then, the optimal equations to calculate the maximum tightening forces and torques of the bolt depending on temperature increase, determined using the method of least squares offered by Microsoft Excel and the data in Table 8, are:

$$y_{force} = -9 \cdot 10^{-5} \cdot x^2 + 0.0013 \cdot x + 57.299 \tag{7}$$

$$y_{torque} = -5 \cdot 10^{-4} \cdot x^2 + 0.0068 \cdot x + 309.44 \tag{8}$$

where y_{force} is the maximum tightening force of a bolt, [kN], y_{torque} is the maximum tightening torque of a bolt, [N·m], and x is the working temperature, [°C].

4.2. Flange with other types of gaskets

In the same way, for flanges with other types of gaskets, the analyses were performed. First, the results of the stress analysis for the flange with the gasket made with soft iron [7] and the values of the coefficient for the optimal equations determined using the method of least squares are presented in Tables 9 and 10.

Table 9

Internal stresses of the flange with gasket of soft iron depending on temperature, [MPa]

Temperature [°C]		20		100		200		300		400		500		600	
Parts		Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.
Tightening force, [kN]	30	218.7	99.2	218.7	99.2	218.8	99.2	218.8	99	218.9	98.9	219	98.7	219.1	98.3
	40	287.9	160.9	288	160.7	288	160.7	288.1	160.4	288.2	160	288.3	159.5	288.5	158.7
	50	357.4	221.7	357.4	221.4	357.5	221.4	357.6	220.9	357.8	220.3	357.9	219.4	358.2	218.2
	60	426.7	285.5	426.8	285	426.9	285	427	284.3	427.2	283.3	427.4	282	427.7	280.3

Table 10

Coefficient values *a*, *b* and R^2 for the flange with gasket of soft iron depending on the temperature

Temperature [°C]		20		100		200		300		400		500		600	
Parts		Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.
Coefficient	a	10.6	-87.24	10.56	-87.04	10.59	-86.57	10.53	-86.23	10.5	-85.45	10.49	-84.51	10.4	-83.6
	b	6.935	6.207	6.937	6.197	6.938	6.181	6.941	6.164	6.945	6.135	6.948	6.098	6.955	6.055
	R^2	1	0.999	1	0.999	1	0.999	1	0.999	1	0.999	1	0.999	1	0.999

The maximum tightening forces and torques of the bolt calculated based on Table 10 are shown in Figures 9 and 10 and Table 11.

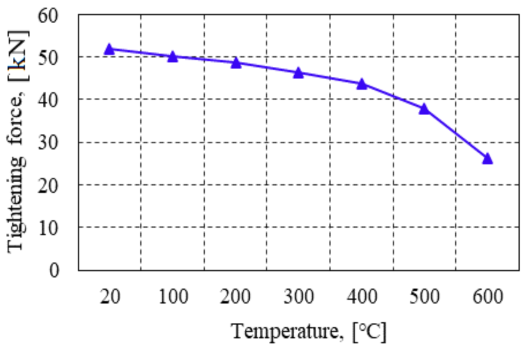


Figure 9. Maximum tightening forces of the flange bolts with gasket of soft iron depending on the temperature increase

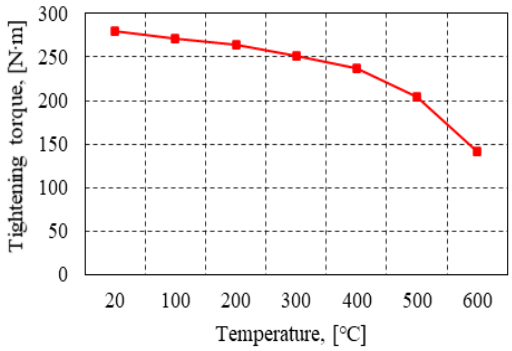


Figure 10. Maximum tightening torques of the flange bolts with gasket of soft iron depending on the temperature increase

Table 11

Maximum tightening forces and torques for the flange bolt with the gasket made with soft iron depending on temperature increase

Temperature, [°C]	20	100	200	300	400	500	600
X_{flange} [kN]	58.31	55.74	53.26	49.43	45.15	37.91	26.14
X_{gasket} [kN]	51.92	50.34	48.84	46.47	43.84	39.29	31.79
Maximum tightening force, [kN]	51.92	50.34	48.84	46.47	43.84	37.91	26.14
Maximum tightening torque, [N·m]	280.4	271.8	263.7	250.9	236.7	204.7	141.2

The optimal equations to calculate the maximum tightening forces and torques of the bolt depending on temperature increase, determined using the method of least squares and the data in Table 11, are:

$$y_{\text{force}} = -9\cdot 10^{-5}\cdot x^2 + 0.0168\cdot x + 50.402\text{, [kN]} \tag{9}$$

$$y_{\text{torque}} = -5\cdot 10^{-4}\cdot x^2 + 0.0901\cdot x + 272.2\text{, [N·m]} \tag{10}$$

As with the gasket made with 4-6%Cr0.5%Mo, the results show that the tightening forces and torques of the bolt decreases nonlinearly with increasing temperature. But, as can be seen in Table 11, the results suggest that, unlike the gasket made with 4-6%Cr0.5%Mo, when using the soft iron gasket, up to a temperature of 400 °C, the internal stress of the gasket exceed first the permissible stress of the material, but at a temperature above it, the internal stress of the flange exceed first.

Next, the results of the stress analysis for the flange with the gasket made with AISI 304, 316 and 347 [7] and the values of the coefficient for the optimal equations determined using the method of least squares are presented in Tables 12 and 13.

Table 12

Internal stresses of the flange with the gasket made with AISI 304, 316 and 347 depending on temperature increase, [MPa]

Temperature [°C]		20		100		200		300		400		500		600	
Parts		Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.
Tightening force, [kN]	30	218.7	99.2	218.7	99.2	218.8	99.2	218.8	99	218.9	98.9	219	98.7	219.1	98.3
	40	287.9	160.9	288	160.7	288	160.7	288.1	160.4	288.2	160	288.3	159.5	288.5	158.7
	50	357.4	221.7	357.4	221.4	357.5	221.4	357.6	220.9	357.8	220.3	357.9	219.4	358.2	218.2
	60	426.7	285.5	426.8	285	426.9	285	427	284.3	427.2	283.3	427.4	282	427.7	280.3

Table 13

Coefficient values *a, b* and R^2 for the flange with the gasket made with AISI 304, 316 and 347 depending on temperature increase

Temperature [°C]		20		100		200		300		400		500		600	
Parts		Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.
Coefficient	*a*	10.6	-87.24	10.56	-87.04	10.59	-86.57	10.53	-86.23	10.5	-85.45	10.49	-84.51	10.4	-83.6
	b	6.935	6.207	6.937	6.197	6.938	6.181	6.941	6.164	6.945	6.135	6.948	6.098	6.955	6.055
	R^2	1	0.999	1	0.999	1	0.999	1	0.999	1	0.999	1	0.999	1	0.999

The maximum tightening forces and torques of the bolt calculated based on Table 13 are shown in Figures 11 and 12 and Table 14.

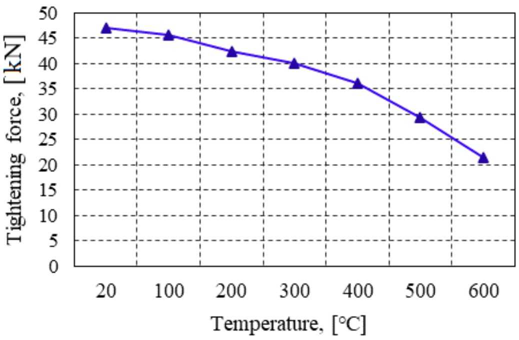


Figure 11. Maximum tightening forces of the flange bolts with gasket of AISI 304, 316 and 347 depending on the temperature increase

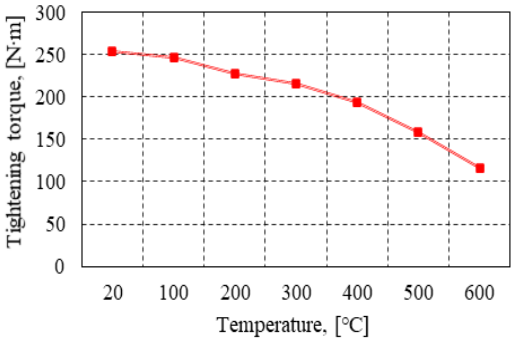


Figure 12. Maximum tightening torques of the flange bolts with gasket of AISI 304, 316 and 347 depending on the temperature increase

Table 14

Maximum tightening forces and torques for the flange bolt with gasket of AISI 304, 316 and 347 depending on the temperature increase

Temperature, [°C]	20	100	200	300	400	500	600
X_{flange} [kN]	58.31	55.74	53.26	49.43	45.15	37.91	26.14
X_{gasket} [kN]	47.08	45.72	42.29	39.96	36	29.44	21.40
Maximum tightening force, [kN]	47.08	45.72	42.29	39.96	36	29.44	21.40
Maximum tightening torque, [N·m]	254.2	246.9	228.4	215.8	194.4	159	115.6

The optimal equations to calculate the maximum tightening forces and torques of the bolt depending on temperature increase, determined using the method of least squares and the data in Table 14, are:

$$y_{force} = -6 \cdot 10^{-5} \cdot x^2 - 0.0041 \cdot x + 46.759, \quad [kN] \tag{11}$$

$$y_{\text{torque}} = -3 \cdot 10^{-4} \cdot x^2 - 0.0222 \cdot x + 252.48, \text{ [N·m]} \quad (12)$$

When applying the stainless-steel gasket, internal stresses of the gasket exceeded first the permissible limit of the material. Of course, the tightening forces and torques of the bolt decreased nonlinearly with increasing temperature.

Next, the results of the stress analysis for the flange with the gasket made with AISI 410 [7] and the values of the coefficient for the optimal equations, determined using the method of least squares, are presented in Tables 15 and 16.

Table 15

Internal stresses of the flange with gasket of AISI 410 depending on the temperature, [MPa]

Temperature [°C]		20		100		200		300		400		500		600	
Parts		Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.
Tightening force, [kN]	30	218.7	99.3	218.7	99.2	218.8	99.2	218.8	99	218.9	98.9	219	98.7	219.1	98.3
	40	287.9	161.1	288	160.9	288	160.7	288.1	160.4	288.2	160	288.3	159.5	288.5	158.7
	50	357.4	222	357.4	221.7	357.5	221.4	357.6	220.9	357.8	220.3	357.9	219.4	358.2	218.2
	60	426.7	285.9	426.8	285.5	426.9	285	427	284.3	427.2	283.3	427.4	282	427.7	280.3

Table 16

Coefficient values *a, b* and R^2 for the flange with gasket of AISI410 depending on the temperature

Temperature [°C]		20		100		200		300		400		500		600	
Parts		Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.	Flan.	Gas.
Coefficient	a	10.6	-87.24	10.56	-87.04	10.59	-86.57	10.53	-86.23	10.5	-85.45	10.49	-84.51	10.4	-83.6
	b	6.935	6.207	6.937	6.197	6.938	6.181	6.941	6.164	6.945	6.135	6.948	6.098	6.955	6.055
	R^2	1	0.999	1	0.999	1	0.999	1	0.999	1	0.999	1	0.999	1	0.999

The maximum tightening forces and torques of the bolt calculated on the basis of Table 16 are shown in Table 17 and in Figures 13 and 14.

Table 17

Maximum tightening forces and torques for the flange bolt with gasket of AISI 410 depending on the temperature increase

Temperature, [°C]	20	100	200	300	400	500	600
X_{flange} [kN]	58.31	55.74	53.26	49.43	45.15	37.91	26.14
X_{gasket} [kN]	58.36	56.52	54.32	52.03	48.92	43.64	34.85
Maximum tightening force, [kN]	58.31	55.74	53.26	49.43	45.15	37.91	26.14
Maximum tightening torque, [N·m]	314.9	301	287.6	267	243.8	204.7	141.2

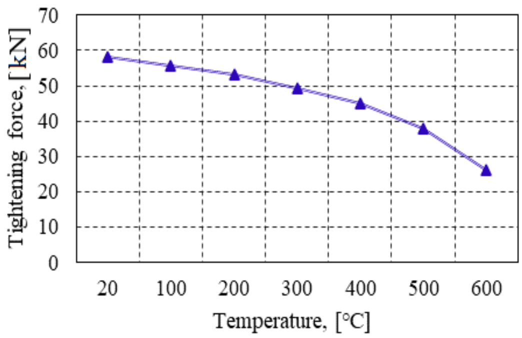


Figure 13. Maximum tightening forces of the flange bolts with gasket of AISI 410 depending on the temperature increase

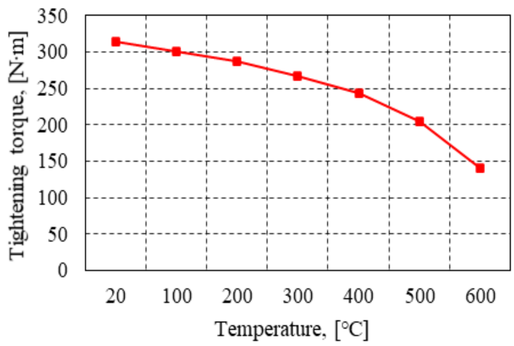


Figure 14. Maximum tightening torques of the flange bolts with gasket of AISI 410 depending on the temperature increase

The optimal equations to calculate the maximum tightening forces and torques of the bolt depending on temperature increase, determined using the method of least squares and the data in Table 17, are:

$$y_{\text{force}} = -9\cdot 10^{-5}\cdot x^2 + 0.0013\cdot x + 57.299, \quad [\text{kN}] \tag{13}$$

$$y_{\text{torque}} = -5\cdot 10^{-4}\cdot x^2 + 0.0068\cdot x + 309.44, \ [\text{N·m}] \tag{14}$$

Through the results of the analysis, it was confirmed that the use of the gasket made with AISI 410 is the same as the use of the gasket made with 4-6% Cr0.5%Mo.

5. Conclusions

This paper presented the process of revealing the relationship between the working temperature and the tightening force of the bolt that directly affects the flange leakage, using ANSYS 19.2. Using the calculation equations reflecting the relationship between the temperature and the mechanical properties of metallic materials proposed by *Luecke et al* [7], the changing values in the mechanical properties of the flange assembly elements depending on the temperature increase were determined, and, on the basis of these changes, the internal stresses of the flange assembly elements were analyzed. In addition, the optimal equations reflecting the relationships between the tightening force of the bolt and internal stress and between the tightening force of the bolt and working temperature were determined using the method of least squares offered by Excel. Through the stress analysis of the flange according to the working temperature and the type of gasket, the following conclusions can be drawn:

a) As the tightening force of the bolt increases, the internal stresses of the flange assembly elements increase almost linearly;

b) The presence or absence of coating on the bolt surface has a very small influence on the internal stress of the flange assembly elements;

c) For all types of gaskets, the tightening forces and torques of the bolt decreases nonlinearly with increasing temperature;

d) The assembly elements that first exceed the permissible stress of the material (yield strength) are flange and gasket, and which of these assembly elements exceed first the permissible limit depends on the type of gasket.

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