STUDIES ON HIGH PRESSURE GASKETS.

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1. Introduction.

With regard to the design of high pressure chemical plant, the prevention of leakage of the working fluid from joints is of the utmost importance. The work was taken up from this point of view.

2. Previous Works.

A glance over earlier works is not out of place here before entering into detailed description of our present work.

A gasket joint of simple design as shown in Fig. 1 is chosen, because it is fundamental and convenient to apply the results obtained therewith to high pressure joints of particular design in which we have particular interests.

Let p_0=working fluid pressure,

t=thickness of gasket before tightening joints,

b=width of gasket before tightening joints,

P_{b1}=initial total bolts load caused by tightening joints,

then, when the working fluid is permitted to exert pressure the force P_{b1} will increase by $\triangle P_b$ and assume the value of P_{b2}. The joint will expand at the same time axially, stretching each bolt by $\triangle l$.

Hence,

$$P_{b2}-P_{b1}=\triangle P_b=A_b E_b \triangle l/l \qquad (1)$$

where A_b=total cross-sectional area of bolts,

l=natural length of each bolt,

E_b=modulus of longitudinal elasticity of bolt materials.

The process will make the gasket recover its thickness by $\triangle l$ which wil reduce the contact force by $\triangle P_g$, making it to assume the new value of P_g, or

$$P_{b1}-P_g=\triangle P_g=A_g E_g \triangle l/t \qquad (2)$$

where $A_g \doteqdot \pi(\gamma_2^2-\gamma_1^2)$=contact area of gasket,

E_g=modulus of longitudinal elasticity of gasket materials.

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Furthermore, the following relation holds:

$$P_{b2}=P_0+P_g \quad \text{...} (3)$$

where P_0 is the axial force exerted by the fluid pressure p_0, acting upon the area A_0 as will be discussed in the next section.

Fig. 2 illustrates the diagram of these relations between loads and elongations.

In an ideal case, even if the residual contact (or gasket) force P_g reduced to nothing under the fluid pressure, the prevention of leakage of the working fluid from joints could be attained; but actually it is not the case unless the force P_g or the so-called residual mean gasket pressure p_g $(=P_g/A_g)$ assumes the value somewhat larger than the limiting one to be found experimentally.

Accordingly, in every actual case, it is necessary to take an adequate value of the initial total bolts load P_{b1} in order to prevent the leakage of the working fluid from joints.

Table 1, compiled from the data found in some recent literatures, gives the values of the residual mean gasket pressure on contact surface and the initial bolts load necessary to prevent the leakage from joints.

3. The authors' theory and its application to design.

As shown in Table 1, the values of the initial total bolts load or the residual mean gasket pressure on contact surface vary so widely that the designer finds it very difficult to choose proper ones.

The fact may certainly be due to the diversities of experimental conditions, especially of the manner of tightening the flanges, the dimension ratio of gaskets to flanges (especially the diameter of pitch circle of bolts) and the number of bolts. Furthermore, the assumption of the effective area on which working fluids act may come into question.

In this connection, the authors are of the opinion that the effective diameter of the circle in which working fluids act is the inner diameter of gasket, and that, with regard to the critical leakage pressure, the fact that the distribution of the initial gasket pressure on contact surface is not uniform must also be taken into consideration. Accordingly, the minimum gasket pressure p_{min} on contact surface will be discussed, being based on the working fluid pressure p_0.
These considerations lead to a set of equations of more general applicability than the usual ones, as follows:—

Let he distribution of the initial gasket pressure on contact surface be expressed by the equation,

Table 1.

No.	Working fluid pressure p_0 kg/cm²	Temperature °C	Gasket: Inner dia. $2r_1$ cm	Gasket: Outer dia. $2r_2$ cm	Gasket: Area A_g cm²	Effective dia. (cm) & area (cm²) on which working fluids act.	Hydraulic end force P_0' kg	Initial tightening force P_{b1} kg	P_{b1}/P_0'	Bolt: Nominal dia.	Bolt: Number	Bolt: Total sectional area A_b cm²	Residual gasket pressure p_0' kg/cm²	p_0'/p_0	Remarks
1	10.0	450	—	—	—	18 (Mean dia.) 254.5	(25,520)	(76,560)	3.0	1⅛″	12	52.08	—	—	An example of design. Steam-pipe joints.
2	17.6*	—	15.2*	24.6*	293.1*	20* (Mean dia.) 314.2	5,522*	8,664*	1.57	¾″	12	—	10.7*	0.61	An example of design. Screwed flanges.
3	17.6*	—	121.9*	129.5*	1505*	Outer dia.	231,650*	244,900*	1.06	—	—	—	8.79*	0.5	An example of design. Pressure vessels.
4	8.79*	—	41.0*	45.4*	(301)	40.6* (Inner dia. of Vessels) 129.7	11,430*	22,860*	2.0	1⅜″	14	—	(38.0)	(4.3)	An example of design. Steam cylinder head. Steel-asbestos gasket.
5	—	—	—	—	—	Outer dia.	—	—	1.5~1.2	⅞″~2½″	—	—	—	—	Proposal based upon experiences.
6	10~200	20~350	12.0	14.0	40.84	12.0 (Inner dia.) 113.1		—	≧2.0 <2.0	— (gum)	—	—	—	(≧2.77) (<2.77)	Experiments. Metal gasket (<350°C) & gum or other non-metal gaskets (<200°C).
7	10~160	400 & 500	12.0 $b=(0.1\sim0.2)r_1$	14.0	— —	12.0 (Inner dia.) 113.1 —	—	—	1.5~3.0	—	—	—	—	—	Experiments. Soft iron gasket & stainless steel V2A gasket. Gasket section is not rectangle.
8	527~* 1477	—	3.81*	4.45*	(4.15)	3.81* (Inner dia.) (11.4)	—	—	(1.46)	—	—	—	—	1.25	Experiments. Copper gasket.
9	352* 457 703	—	18.7*	23.0*	(140.8)	23.0* (Outer dia.) 415.5	146,100* 189,600 292,100	205,000* 243,100 334,300	1.40 1.28 1.14	1¼″	12	—	(418) (380) (300)	(1.19) (0.83) (0.43)	Experiments.
	112 155 232 281	—	19.7*	22.5*	(92.8)	22.5* (Outer dia.) 399.4	44,910* 61,690 92,760 112,300	51,250* 102,500 153,800 174,600	1.14 1.66 1.66 1.35	⅝″	16	—	(68.4) (440) (658) (671)	(0.61) (2.84) (2.84) (2.39)	Akro-metal gasket.
10	—	—	Small width Large width ($r_2/r_1 \geqq 2$)			Inner dia.	—	—	1.5~3.0 ≧3.5	—	—	—	—	— (≧0.83)	Experiments. Lead, aluminum, copper and other metal gaskets.
11	—	—	$b=(0.1\sim0.2)r_1$			Inner dia.	—	—	(1.42~2.76)	—	—	—	—	2~4	General description. Flanges of large dimensions.

Note:—1. Symbol * designates the numerical values converted from ft-lb unit to metric unit.
2. Numerical values bracketed by ( ) are the values presumed.
3. P_0'=(Working fluid pressure)×(Effective area) & $p_g'=(P_{b1}-P_0')/A_g$; in general, they are distinct respectively from P_0 & p_g used in the authors' equations.

$$p=\frac{c}{\rho^n} \qquad \cdots\cdots(4)$$

where ρ is the distance between the centre of any bolt and any point on the elementary corresponding area $A_1\,B_1\,B_2\,A_2$ of contact surface as shown in Fig. 3, namely,

$$\rho=\sqrt{r^2-2Rr\cos\theta+R^2} \qquad \cdots\cdots(5)$$

in which c and n are constants, the former being determined by boundary conditions and the latter experimentally.

From Eq. (4) & (5), it follows that

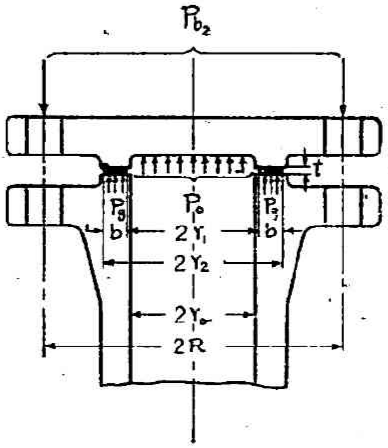

Fig. 1.

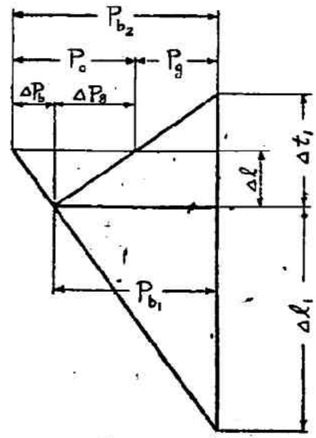

Fig. 2.

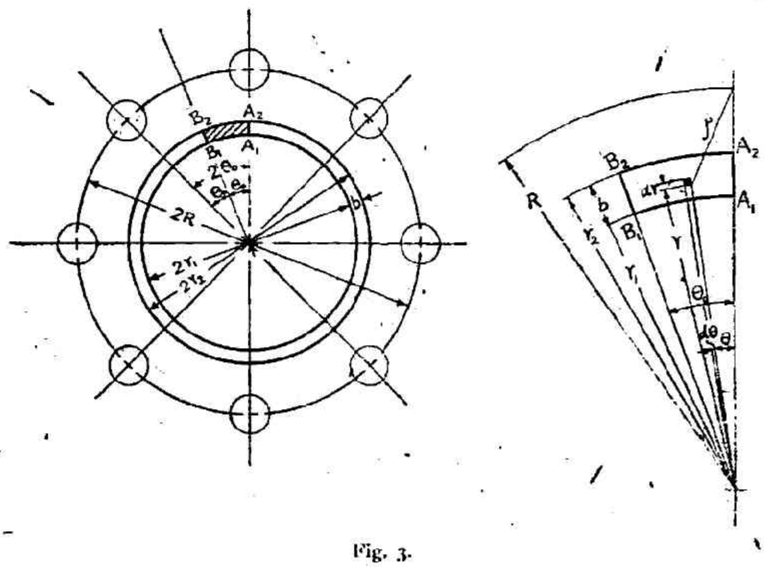

Fig. 3.

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$$p=\frac{c}{\sqrt{(\gamma^2-2R\gamma\cos\theta+R^2)^n}}\quad\cdots\cdots(6)$$

The equation of equilibrium of forces acting upon the area $A_1\,B_1\,B_2\,A_2$, i. e.,

$$\frac{\theta_0}{2\pi}P_{b1}=\int_0^{\theta_0}\int_{\gamma_1}^{\gamma_2}p\gamma d\gamma d\theta$$

and the above equation (6) leads to the following equation,

$$P_{b1}=\frac{2\pi c}{\theta_0}\int_0^{\theta_0}\int_{\gamma_1}^{\gamma_2}\frac{\gamma d\gamma d\theta}{\sqrt{(\gamma^2-2R\gamma\cos\theta+R^2)^n}}\quad\cdots\cdots(7)$$

From this equation (7) the constant c is determined and so Eq. (6) gives the initial gasket pressure at any point. For example, at the point B_1 where the pressure is minimum,

$$p_{B1}=p_{min}=\frac{\theta_0}{2\pi}\cdot\frac{P_{b1}}{I_n\sqrt{(\gamma_1^2-2R\gamma_1\cos\theta_0+R^2)^n}}\quad\cdots\cdots(8)$$

in which

$$I_n=\int_0^{\theta_0}\int_{\gamma_1}^{\gamma_2}\frac{\gamma d\gamma d\theta}{\sqrt{(\gamma^2-2R\gamma\cos\theta+R^2)^n}}\quad\cdots\cdots(9)$$

Table 2 gives some values of the integration of I_n.

Table 2.

γ_1	γ_2	I_1	I_2
4.1	4.3	0.0734	0.0163
"	4.5	0.154	0.0350
"	4.7	0.241	0.0562
"	4.9	0.336	0.0805

($R=8.5$, $\theta_0=\pi/8$)

When the working fluids act, the minimum initial gasket pressure on contact surface caused by tightening joints reduces to the value:

$$p_{min}-(1-a)\,P_0/A_g$$

where a is defined by

$$\left.\begin{aligned}&\triangle P_b=aP_0\\ \text{or}\quad&\triangle P_g=(1-a)\,P_0\end{aligned}\right\}\quad\cdots\cdots(10)$$

Hence, remembering the relations expressed in Eq. (1) & (2), it follows that

$$\frac{a}{1-a}=\frac{A_bE_bt}{A_gE_gl}$$

$$\text{or}\quad a=\frac{A_bE_bt}{A_bE_bt+A_gE_gl}\quad\text{..............................(11)}$$

To prevent entirely the leakage of the working fluid from joints, the next equation must hold, i.e.,

$$\left.\begin{aligned}&p_{min}-(1-a)\frac{P_0}{A_g}\geqq 0\\ \text{or}\quad &p_{min}\geqq(1-a)\frac{P_0}{A_g}\end{aligned}\right\}\quad\text{..............................(12)}$$

On the other hand, from various experiments and experiences as described in the former section, it is apparent that the initial tightening force P_{b1} should be greater than the hydraulic foree P_0; hence, we can put in general

$$P_{b1}=P_0+xp_0A_g\quad\text{..............................(13)}$$

where x is the coefficient of additional force.

In earlier works the coefficient x was regarded as independent of the dimension ratio of joints. But, the theoretical considerations based upon non-uniform distribution, of initial gasket pressure lead to the conclusion that x depends upon the dimension ratio of joints.

Let the ratio A_0/A_g be denoted by a, then from Erom Eq. (13) we obtain

$$P_{b1}=(a+x)p_0A_g\quad\text{..............................(14)}$$

On the other hand,

$$P_{b1}=p_m\cdot A_g\quad\text{..............................(15)}$$

hence, from Eq. (14) & (15) we have

$$a+x=\frac{p_m}{p_0}=\frac{p_m}{p_{min}}\cdot\frac{p_{min}}{p_0}\quad\text{..............................(16)}$$

To prevent entirely the leakage of the working fluid from joints, it must hold at least as the postulate of Eq. (12) that

$$p_{min}=(1-a)\frac{P_0}{A_g}=(1-a)ap_0$$

$$\text{or}\quad\frac{p_{min}}{p_0}=(1-a)a\quad\text{..............................(17)}$$

Accordingly, from Eq. (16) & (17) we obtain following equation,

$$x=\left\{(1-a)\frac{p_m}{p_{min}}-1\right\}a \quad \ldots\ldots(18)$$

For the special case of $a=0$,

$$x=\left(\frac{p_m}{p_{min}}-1\right)a \quad \ldots\ldots\ldots\ldots(18a)$$

or, remembering Eq. (17)

$$x=\frac{p_m-p_{min}}{p_0} \quad \ldots\ldots\ldots\ldots(18b)$$

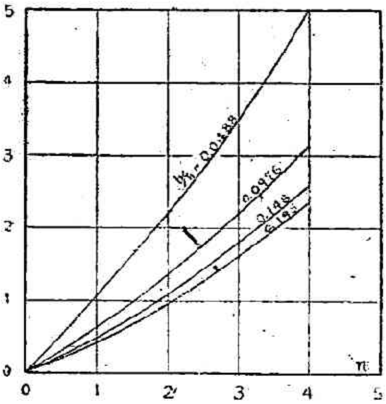

Fig. 4.

Fig. (4) shows how the value of x varies with the dimension ratio of b/γ_1 and the experimental constant n in the case of $R/\gamma_1=2.07$, $\theta_0=\pi/8$ & $a=0$; by which we can readily recognize the facts mentioned above.

With a set of equations thus obtained, we can now easily design the gasket joint of simple type in the following way.

First assume the main dimensions of joints and the number of bolts, as the inside and outside diameter of vessels and the working fluid pressure p_0 are given and also the constant n is known***.

Then we find the value of p_m/p_{min} from Eq. (8) & (15), calculating the value of I_n by Eq. (9). The coefficient of additional force, x, will also be determined by Eq. (18) with the value of a computed from Eq. (11). The minimum value of the initial tightening force P_{b1} will be thus obtained. If the value of P_{b1} thus found is not appropriate in view of the strength of bolt materials, the first assumption should be modified and the calculation be repeated. In this way we can finally decide the adequate main dimensions of joints and the corresponding initial tightening force.

4. Conclusions.

As above mentioned, theoretical considerations based upon non-uniform distribution of initial gasket pressure have led to a set of equations of more general applicability than the usual ones. With the aid of these equations we can find the numerical values requisite in the design of the gasket joint of simple type.

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*** In our preliminary experiments where $R/\gamma_1=2.07$, $b/\gamma_2=0.145\sim0.195$, $\theta_0=\Pi/8$ and $a=0.05$, the value of the contant n fell near 1.8.