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Xu's Sealing Theory and Rectangular & O-Shaped Ring Seals

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Abstract: The difficulty for a sealing element to create and maintain a leak-free joint is determined by its sealing difficulty factor m_1, m_1 = elastic modulus E_c of its sealing contact layer/elastic modulus E_s of its sealing contact layer substrate. Therefore, theoretically the contact layer of a sealing element shall be soft & inelastic and assembled up to its fully yielded deformation to provide a contact layer with a lower value of active elastic modulus E_c, and the contact layer substrate shall be strong & elastic and assembled up to its fully elastic deformation to provide a contact layer substrate with a higher value of active elastic modulus E_s. It is the most difficult for a rubber sealing element to create a leak-free joint because its $E_c \equiv E_s$, and it is far easier for a metal sealing element than for a rubber sealing element because the metal sealing element can be designed and coated to ensure that assembling can cause its $E_c < E_s$.

Keywords: Seal, Categorization of seals, Circle-based system of O-ring seals, Minimum necessary sealing stress y, Sealing difficulty factor m_1, Leak-free maintenance factor m_2, Self-sealing mechanism for material (Mechanism of self-sealing Poisson's deformation caused by fluid pressure), Self-sealing mechanism for O-rings (Mechanism of self-sealing deformation caused by fluid seepage)

1 Categorization of Seals [1]

Seals are to create either a leak-free butt joint of two flat end surfaces or a leak-free fit of two cylindrical surfaces, and hence can be divided into an end face (butt joint) seal and a cylinder (fit) seal according to a shape of sealing surfaces. As shown in Fig.1a, an O-ring (Φd_{1o}) or a rectangular ring (Φd_{1r}) can be used for a self-sealing joint of two end faces. As shown in Fig.1b, O-ring seals used as a self-sealing joint of two cylinders can be divided into a rod seal (Φd_{12}) for a rod/hole fit and a piston seal (Φd_{13}) for a piston/cylinder fit. As shown in Fig.1c, one of O-ring seals (Φd_{11} ~Φd_{13}) can be substituted for the self-energizing O-ring seal (Φd_{1o}) for two end faces. If the substitute seals indicated by Φd_{12} and Φd_{13} in Fig.1c are respectively regarded as static rod seals and piston seals, the seal indicated by Φd_{11} in Fig.1c and substituted for the seal indicated by Φd_{1o} can be undoubtedly called a port seal for a cylinder port or hole port. Given that the threaded port seal (Φd_{11}) in Fig.1d is the same as the port seal (Φd_{11}) and that the size series and the cavity's fill characteristic of the original face seal (Φd_{1o}) can be the same as those of a port seal (Φd_{11}), the self-energizing seals for a joint and a fit of surfaces can be unifiedly studied and disposed by being divided into a rectangular ring seal (Φd_{1r}) and three O-ring seals called port seal (Φd_{11}), rod seal (Φd_{12}) and piston (Φd_{13}) seal.

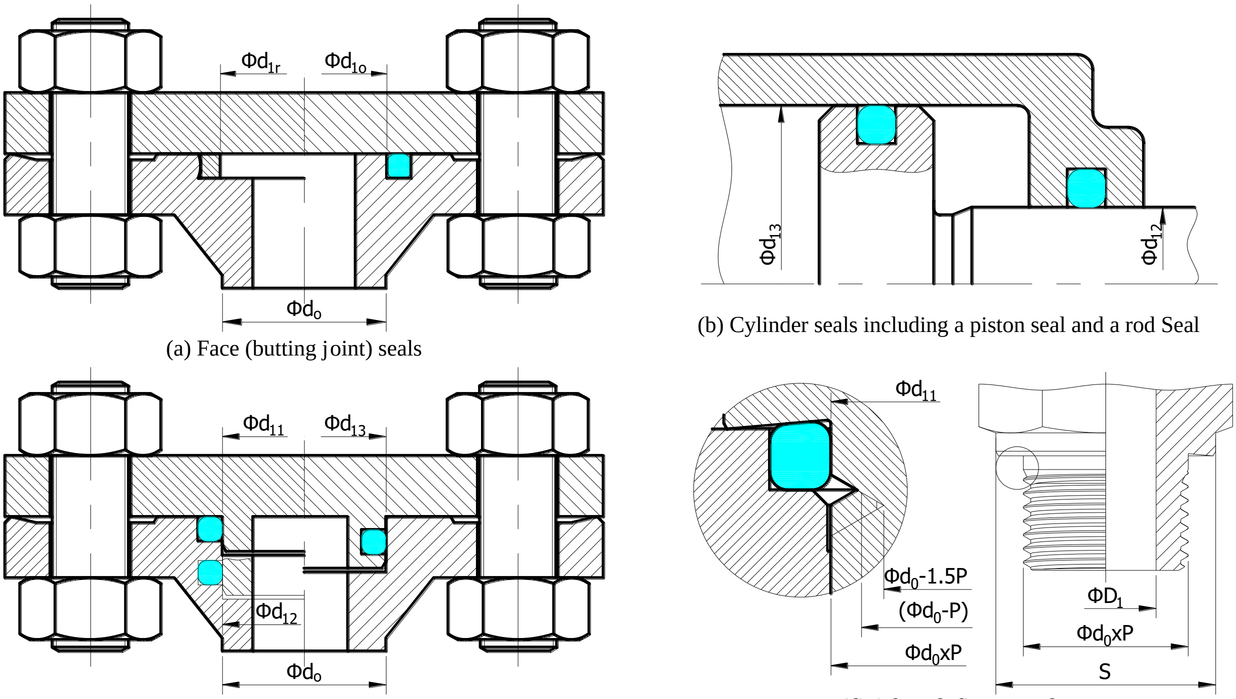
A non-self-energizing seal opposite to a self-energizing seal is called a pressure-tight seal, or a non-self-sealing joint opposite to a self-sealing joint is called a pressure-tight joint.

2 Flange Gasket Seals of the Prior Art [2]

The flange joint of the prior art, achieving their fastening connection by some bolts and their sealing connection by a gasket between two end faces as shown in Fig.2, is a pressure-tight joint that is different from the self-sealing joint with a rectangular ring (Φd_{1r}) in Fig.1. As shown in Fig.3, the two flanges have a wide touch width b due to no bending before tightened, a narrowed touch width b due to some bending after tightened, and a more narrowed touch width b due to more bending under a fluid pressure. Generally speaking, any tightening assembly is causing a gasket to be loaded or resulting in a gasket increasing its sealing stress, and any fluid pressure is causing a gasket to be unloaded or resulting in a gasket decreasing its sealing stress. Therefore, any flange gasket seal of the prior art has the following four inherent fatal problems:

a. It will leak, no matter how it is constructed.

Any sealing needs to deform a sealing surface into some imperfections on a being sealed surface, whereas any leaking is a process for a leaking fluid on the two contact surfaces to output a contact-separating force whose limit equals “contact area x pressure”. Hence, the mechanical condition for a gasket to create a leak-free joint is to enable the gasket to provide both a full deformation and a fully strong support for its sealing contact. The soft and inelastic gasket material, such as grease, can easily provide a full deformation but not a fully strong support for its sealing contact, and the strong and elastic gasket material can only conversely provide the deformation and the support. In the prior art, there is not such a designing idea or method resulting in a both soft and strong sealing contact layer that spiral wound gaskets and Kammprofile gaskets that are inadvertently in good accordance with the designing idea cannot yet have a fully strong support underneath the sealing surface. Therefore, any pressure-tight gasket seal of the prior art, however it is constructed, must leak at a certain temperature and pressure that are not high.



(a) Face (butting joint) seals

(b) Cylinder seals including a piston seal and a rod Seal

(c) Port seals and cylinder seals substituted for face seals

(d) (Threaded) Port seal

***Fig.1** Categorization of seals*

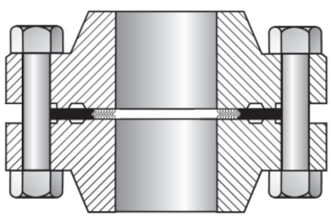


***Fig.2** Bolted flanges of the prior art*

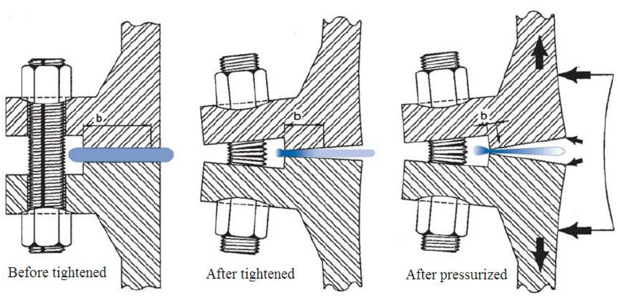


***Fig.3** Flange rotation of the prior art*

b. It will leak, no matter how it is assembled.

Any sealing is deforming a sealing surface into some roughness-caused imperfections on a being sealed surface, whereas an economically machined surface roughness for flange faces is not greater than R_a 3.2 μm; i.e. any flange gasket shall have a compressive deformation with its circumferential uniformity within μms, otherwise any great tightening load is not fully useful or sufficient. Because any torque controlling method of the prior art, as shown in Fig.3, cannot result in a tightened deformation with its circumferential uniformity within μms, thus any pressure-tight gasket seal of the prior art must be a leaking device however it is assembled, even if its sealing contact layer can be as close to being both soft and strong as that of spiral wound gaskets and Kammprofile gaskets.

c. Any fluid pressure is to worsen its sealing performance.

Any fluid pressure acting on an equivalent flange cover, whether it is great or small, is to cause gasket's compressive stress or sealing performance to decrease. Any fluid pressure acting on gasket's internal cylinder is to cause a gasket to be at risk of blowout. To protect gasket from being blown out, the prior standards specify that a flange face shall have an abnormal surface roughness of R_a 3.2 ~ 6.3 μm, which is so against the ordinary technical ideas of "the smoother the butt joint surface, the easier to be sealed" and "the more precise the machining, the better" as to have an extra leaking risk.

d. Any cold flow is to worsen its sealing performance.

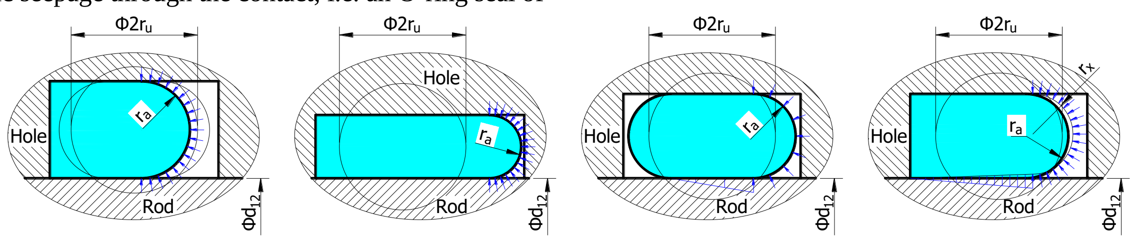
Any sealing material has some cold flow causing gasket's compressive stress or sealing performance to decrease at a more than 0.5 homologous temperature.

Therefore, the flange gasket seal of the prior art is originally only a leaking device under Xu's sealing theory.

3 O-Ring Seals of the Prior Art [1, 3-4]

According to ISO 3601: 2008, an installed O-ring shall not have such a more than 85% fill in its cavity that it can be compressed only in axial direction in face seals and called an axial seal, and compressed only in radial direction in rod and piston seals and called a radial seal; i.e. any installed O-ring of the prior art is compressed at two sides, never at four sides.

As shown in Fig.4a, an installed O-ring shall be flattened to $r_a \geq 0.75r_u$ according to the required sealing power or support or according to the requirement that its sealing actuation area of fluid shall be greater than its sealing contact area (its unsealing actuation area of leaking fluid); and, as shown in Fig.4b, an installed O-ring shall be flattened to $r_a \leq 0.5r_u$ according to the minimum tight contact required to resist an atmospheric seepage through the contact; i.e. an O-ring seal of the prior art is irreconcilable in designs under Xu's sealing theory, or the O-ring of the prior art cannot at the same time meet its assembly squeeze requirements as qualified initial and working seals. An installed standard O-ring of the prior art, as shown in Fig.4c, is flattened to $r_a \geq 0.7 \sim 0.85r_u$ only to have an assembled stress of 0.14 ~ 0.12 MP_a, which is inferior to 0.2 MP_a required to achieve its minimum tight contact, and also to be unable to move or deform against the third wall and start its self-sealing action below a fluid pressure of 0.5 ~ 1.5 MP_a because of its enormous static friction coefficient of 1~4 and its total sealing contact area of 3~5 times its sealing actuation area (see Annex A.4.); or, under Xu's sealing theory, any O-ring seal of the prior art is leaking in low pressure systems and at the starting and ceasing stages of ordinary pressure systems; however, the O-ring seal has been used as the best sealing means at low pressures. Besides, any standard O-ring seal of the prior art at a high pressure, as shown in Fig.4d, can only work as a weak small-great end piston and will leak without stopping once it gets leaking, especially when O-rings deteriorate in elasticity or flexibility, because any leaking will not only cause its unsealing actuation force of leaking fluid on its sealing contact area to get greater than its sealing actuation force of fluid on its O-ring but also cause its sealing actuation surface to lessen in curvature radius (from r_a to r_x) and get more resistant to fluid pressure or cause its O-ring to need a greater power for a self-sealing move or deformation.



(a) As working seal, $r_a \geq 0.75r_u$ (b) As initial seal, $r_a \leq 0.5r_u$ (c) Leaking at low pressure (d) Leaking at high pressure

Fig.4 *Problems of assembly squeeze of O-ring seals of the prior art*

The isotropic softness, incompressibility and elasticity of rubber cause its O-ring to be like such a ring of shaped elastic liquid as to make its internal pressure raise and lower synchronously with its external pressure. Given that there can only be a uniform fluid pressure on a rubber O-ring in its cavity, its fluid compression surface and its extrusion surface are two different radii of single round surfaces tangent to its cavity walls. They can be seen by observing a sectional shape of flattened O-rings and an external surface shape of stretched O-rings. However, US 2180795 (original O-ring seal patent) and ISO 3601: 2008 (standard for O-ring seals) deem the section of an O-ring flattened in its cavity an ellipse (see Annex A.5.). In other words, the prior art has not at all known any sealing mechanism for O-rings and has been unable to properly specify their design, manufacture and assembly by relating their maximum working pressures with their sectional diameter and material strength.

The stronger the strength and the elasticity of a sealing contact layer, the more difficult for its sealing element to create or maintain a fully deformed contact; whereas the stronger the strength and the elasticity of a contact layer substrate, the easier for its sealing element to create or maintain a fully deformed contact. Hence, it is the most difficult for a rubber element to create a leak-free joint whose contact's elastic modulus (E_c) divided by its substrate's elastic modulus (E_s) identically equals one (or $E_c/E_s \equiv 1$), the easiest for a greased hard element whose contact's elastic modulus (E_c) divided by its substrate's elastic modulus (E_s) equals zero (or $E_c/E_s = 0$), and more difficult than for a greased hard element and more easier than for a rubber element for the other sealing elements whose contact layer is yieldable to cause E_c/E_s to be greater than zero and smaller than one (or $0 < E_c/E_s < 1$); i.e. a rubber O-ring seal is the most difficult to create a leak-free joint under Xu's sealing theory, but mistaken for the easiest by the prior art.

The Gough-Joule effect tells us that a stretched rubber piece will not elongate but will shorten in its stretch direction when warmed up, and meanwhile, its tensile modulus will increase but not decrease (see Annex A.3.). Hence, any stretched rubber O-ring in two-wall-touching assembly of its radial seal application will radially shrink in its cross-section to cause a possible leak during a warm start in the morning after further stretched in the colder night. Obviously, the effective means to overcome the Gough-Joule shrinkage influence is to have an assembly of rubber O-rings that uniformly touches their four walls of cavity.

Each solid rocket motor case of Space Shuttle Challenger is made of 11 individual cylindrical weld steel sections about 12 feet in diameter. As shown in Fig.5, the 11 sections of the motor case are joined by tang-and-clevis joints held together by 177 steel pins around the circumference of each joint, and separated by controlled explosion ejection of the pins. Joint sealing is provided by two rubber O-ring piston seals. The installed O-rings were heavily greased and stretched in their cavities, and hence had more circumferential being shortened in time caused both by more cold shrinkage at time of the disaster launch whose ambient temperature was 36 degrees Fahrenheit, or 15 degrees lower than the next coldest previous launch, and by more Gough-Joule circumferential warm shrinkage after ignition; i.e. the O-rings for the disaster launch successively had in cross-sections, because of circumferential being stretched, such an installed radial shrinkage according to the prior standard specification, a more cold radial shrinkage at a colder ambient temperature and a more Gough-Joule warm radial shrinkage after ignition, and had such a gap increase between the tang and the inside leg of the clevis under combustion gas pressure as to jointly make these O-rings be continually off the sealed tang surface and cause the grease to be blown out to form a leak. In other words, from Xu's sealing theory and the records in the Report of Presidential Commission on the Space Shuttle Challenger Accident, it can be seen that the Space Shuttle Challenger disaster was caused by a greased standard O-ring seal, whose O-ring is in a radially clamped assembly that cannot resist any influence of cold and warm radial shrinkages on its tight contact. However, the Report neither pays any attention to the Gough-Joule effect of rubber nor pays any attention to the influence of grease coating on O-rings only to incorrectly deem the disaster to be caused by the O-ring seal design that can result in a possible four-wall-touching assembly or that cannot ensure the O-rings a two-wall-clamped assembly or a two-wall-touching assembly (see Annex A.4.).

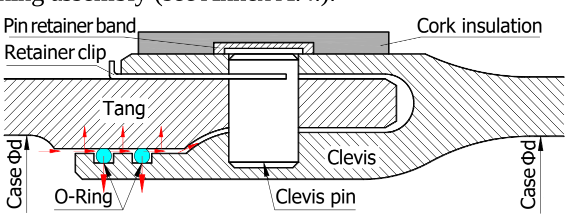


Fig.5 *O-ring seals causing Space Shuttle Challenger Disaster*

Therefore, the O-ring seal of the prior art has been severely mistaken and has a risk causing some major accidents.

4 Designs and Parameters (the minimum necessary sealing stress *y* and the sealing difficulty factor m_1) for a Sealing Element [2, 4]

Any leak-free connection results from loading a sealing contact layer first up to its fully deformed contact and then up to its fully tight contact, or needs first to create a fully deformed contact that can seat a sealing surface into irregularities on a surface to be sealed and then to create a fully tight contact that can resist a seepage through the sealing contact, and any full seepage or leaking is a process for a leaking fluid on the two contact surfaces to output a contact separating force whose limit equals “contact area x pressure”; hence the minimum stress needed to resist an atmospheric seepage through the contact is the minimum necessary sealing stress y for a seal, and the contact whose stress is up to the minimum necessary sealing stress y is called the minimum necessary tight contact. In order to achieve a fully deformed contact without any flaw, it is better to load the sealing contact layer up to its fully yielded deformation. In order to achieve a fully tight contact, it is needed to load the sealing contact layer substrate up to its fully elastic deformation to provide a fully powerful support for the contact. Therefore, the contact stress at loading a sealing contact just up to its fully deformed contact without any flaw at atmospheric pressure is the flaw-blocking or sealing-beginning stress G_b; the assembled stress S_a up to the fully elastic deformation of the substrate shall approach the yield strength S_e of the substrate material; the contact stress at unloading a fully leak-free contact just up to being leaving its fully tight contact is the leakage-starting stress or the minimum necessary sealing stress y; the contact stress at further unloading the contact just up to being leaving its fully deformed contact is the flaw-reproducing or sealing-stopping stress G_s; and $y = y_e + y_a$, where y_e is the component for loading the sealing contact just up to restoring its full deformation along unloading line, and $y_a = 0.1$ MP$_a$ is the component for loading the fully deformed contact just up to restoring its minimum necessary tight contact that can fully resist the atmospheric seepage through contact along unloading line.

The stronger the strength and the elasticity of a sealing contact layer, the more difficult for its sealing element to create or maintain a fully deformed contact; whereas the stronger the strength and the elasticity of a contact layer substrate, the easier for its sealing element to create or maintain a fully deformed contact; i.e. the difficulty for a sealing element to create or maintain a leak-free joint is determined by its sealing difficulty factor m_1, m_1 = contact layer's elastic modulus E_c/ substrate's elastic modulus E_s; or the greater the value of $m_1 = E_c/E_s$ for a sealing element, the more difficult for it to create or maintain a leak-free joint, and also the greater the values of the sealing-stopping stress G_s and the minimum necessary sealing stress y. As shown in Fig.6, the contact layer's elastic modulus (E_c) for a general sealing element can be the residual elastic modulus of its yielded material, and the substrate's elastic modulus (E_s), the elastic modulus of its unyielded material; and for a rubber sealing element without yielding, $E_c = E_s$. Therefore, it is the easiest for a grease coating to create a leak-free joint whose $E_c = 0$ or $m_1 = 0$; it is the most difficult for a rubber sealing element to do whose $E_c = E_s$ or $m_1 = 1$; and it is far easier than for a rubber sealing element for any other sealing element such as a metal element to do whose contact layer can be weakened by designing or coating to cause its $E_c < E_s$ or $m_1 < 1$.

From the fact that grease can be easily deformed and not return to its original shape at atmospheric pressure, it can be seen that the internal pressure caused by the elastic strength of a free grease body is 0 MP_a; i.e. for a grease coating, $G_b = G_s = 0$, or $y_e = 0$. From the fact that rubber can be easily deformed and return to its original shape at atmospheric pressure, it can be seen that the internal pressure caused by the elastic strength of a free rubber body is 0.1 MP_a; i.e. for a rubber element, $G_b = G_s = 0.1$ MP_a, or $y_e = 0.1$ MP_a. Therefore, as to any qualified sealing element, its lower limit of sealing-stopping stresses $G_{smin} = 0$ (equals G_b value of grease coatings), its upper limit of sealing-stopping stresses $G_{smax} = 0.1$ MP_a (equals G_b value of rubber elements), its lower limit of minimum necessary sealing stresses $y_{min} = 0.1$ MP_a (equals y value of grease coatings), its upper limit of minimum necessary sealing stresses $y_{max} = 0.2$ MP_a (equals y value of rubber elements), and $G_{smax} - G_{smin} = y_{max} - y_{min} = y - G_s = 0.1$ MP_a (standard atmospheric pressure); or any sealing element is unqualified whose minimum necessary sealing stress y is more than 0.2 MP_a.

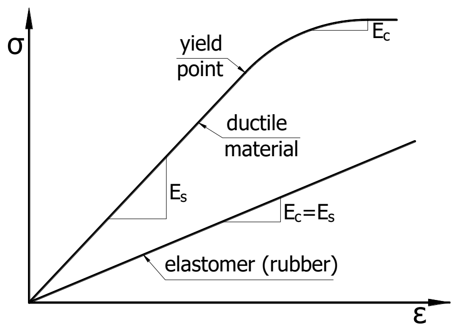


Fig.6 *Xu's sealing difficulty factor* $m_1 = E_c/E_s$

Any leaking through a sealing contact starts at the moment when the contact-tightening stress is less than the fluid pressure, and tries to cause the contact-tightening stress to be counteracted and cause the fluid to be decompressed up to atmospheric pressure by fluid separation of contact. Because the contact separating force increases as the seepage-wetted area increases before a full leaking and lowers as the fluid pressure lowers after a full leaking, thus the moment when the contact-separating force is to be up to the limit of “contact area x fluid pressure” is the moment when the contact-tightening force and stress are to be thoroughly counteracted and cause a full leakage or a full decompression of fluid. If at this moment adding a tightening stress not less than atmospheric pressure to the contact surface, the contact tightening stress will not be less than the fluid pressure that is lowering up to atmospheric pressure and can prevent the sealing contact from leaving its fully tight state. In other words,

a. any sealing contact can never be unloaded to leave its fully tight state and, even if separated by a disturbing force for a moment, will return to its fully tight state right after a full leaking as long as the contact-tightening force is positively greater than the contact-separating force limit of “contact area x fluid pressure”;
b. the contact stress that can resist an atmospheric seepage through the contact is the minimum necessary sealing stress y for a seal;
c. any seal can never leak as long as its sealing stress can be always not less than its minimum necessary sealing stress y or any leaking cannot happen until the sealing stress is less than the stress y, and
d. any seal shall be designed to make its minimum necessary sealing stress y not more than 0.2 MP_a.

Therefore, any sealing element shall be designed to have a sealing difficulty factor m_1 less than 1 or a minimum necessary sealing stress y less than 0.2 MP_a; or ideally, any sealing element shall be designed to have both a soft & inelastic contact layer and a strong & elastic contact layer substrate or support, and assembled to form a fully plastically deformed contact and a fully elastically deformed support at the same time.

5 Total Necessary Fastener Loads for a Leak-Free Connection (the leak-free maintenance factor or the disturbance resistance factor m_2)

According to the definitions of the foregoing minimum necessary sealing stress y and the leak-free maintenance factor m_2, m_2 = joint's sealing actuation force F_s/joint's unsealing actuation force F_u, for a sealing joint, where F_s = the fastener or fluid actuation force causing a tight contact and F_u = contact separating force limit = pA_u, the condition for a pressure-tight joint to maintain its leak-free contact is:

$(F_s - F_u) \geq yA_u \rightarrow (m_2pA_u - pA_u) \geq yA_u \rightarrow m_2 \geq (1 + y/p)$;

the condition for a self-sealing joint to maintain its leak-free contact is:

$(F_s - F_u) \geq yA_u \rightarrow (pA_s - pA_u) \geq yA_u \rightarrow m_2 = A_s/A_u \geq (1 + y/p)$;

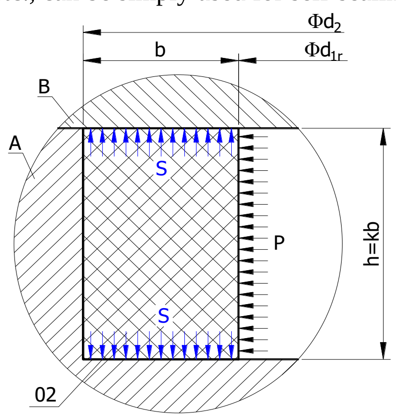
where y = minimum necessary sealing stress, p = fluid pressure, A_s = fluid pressure actuation area of a self-sealing ring, and A_u = sealing contact area of a sealing element.

Given that $y \leq 0.2$ MP_a, thus, if any joint whose pressure rating $p_n \leq 1$ MP_a can be designed according to $p_n = 1$ MP_a, then the condition for any joint to be up to resisting a seepage through the sealing contact at an ultimate pressure of $p_b = 4p_n$ = ultimate strength of its jointed body, whether it is a pressure-tight joint or a self-sealing joint, can be its sealing maintenance factor $m_2 > (1 + y/p_b) = (1 + 0.2/4) = 1$, m_2 being required for $F_s = m_2F_u$; i.e. the total fastener load ($F_\sum$) needed by any sealing joint is $F_\sum \geq \pi p_n d^2$, where p_n = pressure rating, and d = external diameter of a sealing ring. For example, as for the self-sealing joint shown in Figs.7 and 12, $F_\sum = F_s + p_bA_{1r} = m_2p_bA_u + p_bA_{1r} \geq p_b(A_u + A_{1r}) = p_bA_2 = \pi p_nd_2^2$.

6 The Mechanism of Self-Sealing Poisson's Deformation of Material Caused by Fluid Pressure [5-8]

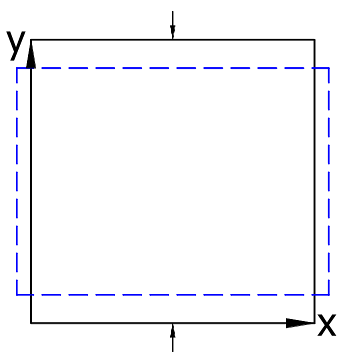
Any self-energizing seal is virtually causing a sealing ring, for example, a face sealing ring (02) in Fig.7, to exactly orthogonally transmit a fluid pressure (p) or to exactly convert the fluid pressure (p) on its internal cylinder into the sealing stress (S) on its end faces, and hence any material with a full liquid behavior can be simply used for self-sealing rings.

As shown in Fig.8, any object will shorten in its compressed direction y and elongate in its non-compressed directions x and z. The orthogonal strain ratio $\varepsilon_x/\varepsilon_y$ or $\varepsilon_z/\varepsilon_y$ of the non-compressive direction to the compressive direction is known as Poisson's ratio υ. The property that a liquid can transmit a pressure equally in each direction originates from its volume incompressibility during flow and deformation under a pressure. It can be seen from bulk modulus $K = E/[3(1\text{-}2\upsilon)]$ that an object whose Poisson's ratio υ is closer to 0.5 has a volume incompressibility closer to infinity. The Poisson's ratio of a general object under normal temperature is greater than zero and smaller than 0.5, but will be close to 0.5 when its homologous temperature, which is the ratio of its absolute temperature to its melting absolute temperature, is higher than 0.5, and the closer to 1 (melting point) its homologous temperature, the closer to 0.5 its Poisson's ratio, and vice versa. Thus it can be said that the Poisson's ratio is an index of liquid behavior and incompressibility of a general object and that the closer to 0.5 its Poisson's ratio, the fuller its liquid behavior; i.e. or a general solid object has both a solid property and a liquid property (see Annex A.1.). Therefore, any material that has a Poisson's ratio close to 0.5 and can be deformed under a fluid pressure, such as rubber, PTFE, lead, gold etc., can be simply used for self-sealing rings.



A Designed (port) end B Fully flat (cover) end

Fig.7 *Behavior of self-energizing seals*



Poisson's ratio $\upsilon = \varepsilon_x/\varepsilon_y = \varepsilon_z/\varepsilon_y$

Fig.8 *Poisson's orthogonal deformation*

Now that the behavior of a self-energizing seal is causing a self-sealing ring to exactly orthogonally transmit a fluid pressure, and determined by whether its orthogonal strain ratio or Poisson's ratio of its material can be up to 0.5 or not under a fluid pressure, thus any pliable solid material, no matter how smaller than 0.5 its Poisson's ratio is, can be used for a self-sealing ring by compensating for its orthogonal strain ratio up to 0.5 by an angle θ_l shown in Fig.9. In short, the compensation of a self-sealing ring for its orthogonal strain is actually compressing a general compressible self-sealing ring from a great room to a small room to make it virtually have the same incompressibility as a liquid and exactly orthogonally transmit a pressure.

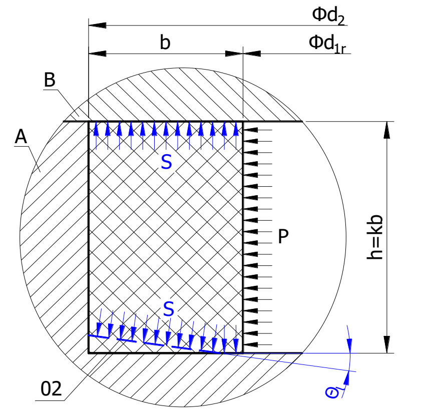


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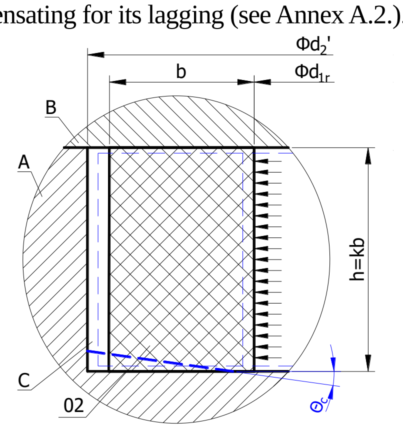
Fig.9 *Compensation for orthogonal deformation*

As shown in Fig.10, any radial clearance (C) between the sealing ring (02) and its bonding wall ($\Phi d_2'$) will result in a Poisson's deformation making it increased in circumference and decreased in height at a certain pressure, and cause it to be away from its previous sealed contact. Actually, it is only when a self-sealing ring with full liquid behavior has no axial and radial contact clearances that it can in time depend on its incompressibility to exactly orthogonally transmit a pressure or function as a self-energizing seal. However, manufacture error and thermal cycling often cause it to have a radial contact clearance, and hence any self-sealing ring with full liquid behavior still needs such an angle (θ_c) compensating for its radial contact as to be able to in time offset its orthogonal deformation that is proportional to its Poisson's ratio and caused by its possible radial clearance. Besides, Poisson's ratio almost changes as synchronously with temperature and time as creep strain does, and also needs compensating for its lagging (see Annex A.2.).



A Designed (port) end B Fully flat (cover) end

Fig.10 *Offset of orthogonal deformation*

The compensation of a sealing ring for its Poisson's orthogonal deformation or for its liquid behavior is aimed at compensating for its insufficient increase in height caused by its Poisson's ratio less than 0.5 when its radial contact has no clearance, and the compensation for its radial contact, aimed at offsetting its decrease in height that is proportional to its Poisson's ratio and caused by its possible radial clearance; i.e. it is necessary for a self-sealing ring to be compensated for its liquid behavior and for its radial contact, and both are to increase its deformation in height under a fluid pressure. Because a general sealing material has a Poisson's ratio ranging from 0 to 0.5, any self-sealing ring needs or has one angle (θ_l) fully compensating for its liquid behavior and one angle (θ_c) fully compensating for its contact whose magnitudes are both determined by the Poisson's ratio limit (0.5) if the compensation for its liquid behavior is done from 0 to 0.5 and the compensation for its contact is done from 0.5 to 0. The two full compensation angles can be unifiedly called an essential Poisson's deformation compensation or offset angle (θ_e):

$$tg\theta_e = \Delta_h/\Delta_r$$
$$= (\Delta_h/\mathbf{h})/(\Delta_r/\mathbf{h})$$
$$= [(\Delta_h/h)]/[(\Delta_r/\mathbf{r})/(h/\mathbf{r})]$$
$$= [(\mathbf{h/r})(\Delta_h/h)]/[(\Delta_r/r)]$$
$$= [(h/r)][(\varepsilon_h)/(\varepsilon_c)]$$
$$= [(h/r)][\upsilon]$$
$$tg\theta_e = (\upsilon h)/r = h/d \quad (\text{when } \upsilon = 0.5),$$

where ε_h = strain of a self-sealing ring in height
ε_c = strain of a self-sealing ring in circumference
h = height of a self-sealing ring
$d = 2r$ = internal diameter of a self-sealing ring
$\upsilon = \varepsilon_h/\varepsilon_c$ = Poisson's ratio definition
$\upsilon = 0.5$ = Poisson's ratio limit.

A self-sealing ring is designed to deform under any pressure. The wedging function of its essential Poisson's deformation compensation or offset angle (θ_e) can only cause it to have some useful sealing deformation. However great the angle θ_e is, what it changes is only the time for the ring material to reach its virtual Poisson's ratio limit 0.5 or 0 but never the magnitude of the two limits, or at most compensates the ring material for its orthogonal deformation ratio from 0 to 0.5 or offsets its orthogonal deformation ratio from 0.5 to 0 as soon as possible, or at most eliminates the lagging of its orthogonal deformation ratio behind its final value. Therefore, any material of self-sealing rings, however great its Poisson's ratio is, can use one angle θ_x greater than θ_e as its liquid behavior compensation angle θ_l and offset angle θ_c when the influence from its thermal coefficient can be ignored.

7 The Mechanism of Self-Sealing Deformation of O-Rings Caused by Fluid Seepage [5-6, 9]

Rubber's softness and Poisson's ratio close to 0.5 mean that rubber is a thickest liquid. Hence, adding its best elasticity and weakest solid behavior, rubber can be regarded as a shaped elastic liquid whose internal pressure can raise and lower synchronously with its external pressure, so that a rubber O-ring can be regarded as a ring of fully liquid-filled tubing whose wall thickness is infinitesimal. Given that there can only be a uniform fluid pressure on a "rubber O-ring tubing" in its cavity, there can be also only such a fluid pressure inside the "rubber O-ring tubing" that causes its external surface at high pressure sides to be uniformly compressed and causes its internal surface at low pressure sides to be uniformly stretched as to make its fluid compression surface and its extrusion surface respectively tangent to its cavity walls by two different radii of single round surfaces. If a rubber O-ring is further regarded as a ring of metallic thin-walled tubing fully filled with pure liquid (see Fig.11a), it can be seen from the strength formula ($p = \sigma t/r$) of thin-walled tubes that:

$p_x r_x = \sigma_k t$ = a constant for a compressed O-ring,

because both the virtual tubing strength σ_k and the virtual wall thickness $t = kr_u$ are invariable for a certain O-ring; i.e.:

$p_x r_x = p_u r_u$ or $p_x = p_u r_u/r_x$,

Where p_x = internal pressure of a compressed O-ring,
p_u = internal pressure of an uncompressed O-ring,
r_x = free extrusion radius of a compressed O-ring,
r_u = free (extrusion) radius of an uncompressed O-ring
σ_k = virtual tubing strength of an O-ring,
t = virtual wall thickness of an O-ring.

Since an externally compressed O-ring always has an external compressing pressure slightly greater than its internal pressure, from the fact that an O-ring can be easily deformed and return to its original shape at atmospheric pressure it can be seen that the assembled stress S_a or internal pressure $p_a = p_u r_u/r_a = 0.1 r_u/r_a$ (MP_a) for an O-ring, supposing the internal pressure (p_u) caused by the elastic strength of an unassembled O-ring is 0.1 MP_a (standard atmospheric pressure), where r_u/r_a = free extrusion radius's ratio of the unassembled to assembled O-ring.

As shown in Fig.11a, given that the virtual tubing is metallic, its wall thickness t, $t = kr_u$, is infinitesimal. Hence, according to the principle that the tensile capacity of the virtual tubing should equal the tensile capacity of the actual O-ring in cross-sections or according to the expression: $(\mathbf{2}\pi r_u \cdot \boldsymbol{k} r_u \cdot \boldsymbol{\sigma_k}) = (\pi r_u^2 \sigma_b) \rightarrow \mathbf{2}\boldsymbol{k\sigma_k} = \boldsymbol{\sigma_b}$, it can be found that:

- the maximum pressure that an unassembled "O-ring tubing" or a free "O-ring tubing" can withstand is
  $p_{um} = \sigma_k(t/r_u) = \sigma_k(kr_u/r_u) = k\sigma_k = 0.5\sigma_b$, and that
- the maximum pressure or the maximum working pressure that an assembled "O-ring tubing" can withstand is
  $p_m = 0.5\sigma_b r_u/r_e$

where r_e = extrusion radius of a rubber O-ring at extrusion gap
r_u = free (extrusion) radius of an uncompressed rubber O-ring
σ_b = tensile strength of rubber O-ring material.

Any compressing of an ordinary soft object will cause it to enlarge in cross-sectional area and get stronger, and any stretching of it will cause it to reduce in cross-sectional area and get weaker; whereas any compressing of a rubber piece will cause it to have more liquid behavior and get weaker, and any stretching of it will cause it to have more solid behavior and get stronger. Thus, as shown in Fig.11b, the solid behavior of rubber O-rings unassembled in its cavity is equivalent to some elastic concentric circles on its cross-sections; as shown in Fig.11c, the solid behavior of rubber O-rings assembled in its square cavity is equivalent to some elastic square curves with four corners rounded by being stretched and strengthened and capable of withstanding a higher internal pressure; and as shown in Fig.11d, a rubber O-ring in service has a compressed region that has more liquid behavior and can exactly transmit fluid pressure and a stretched region that has more solid behavior and can withstand higher internal pressure. Therefore, the greater the rubber O-ring in cross-sectional diameter, the more massive its region (with more solid behavior) out of its region ($\phi 2r_i$) with more liquid behavior, and the higher the fluid pressure that it can withstand (see Annex A.3.). In other words, a rubber O-ring increased in cross-sectional diameter can save an anti-extrusion back ring in high pressure applications.

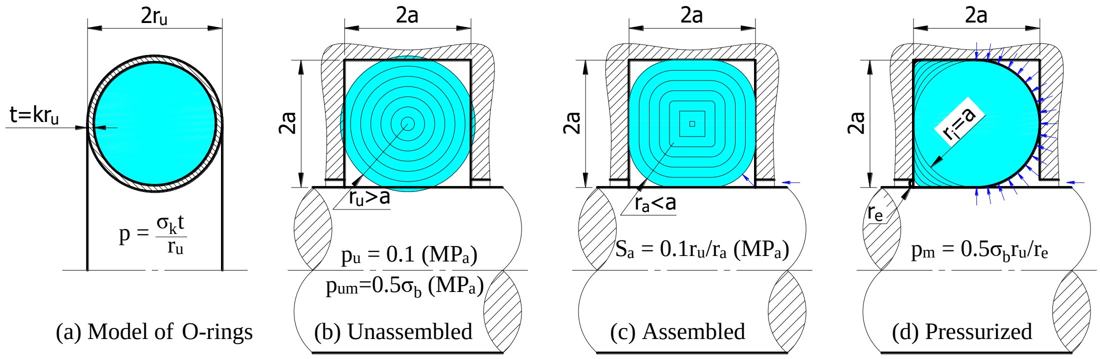


Fig.11 *Sealing behavior of rubber O-rings*

A rubber O-ring in a four-wall-touching assembly, as shown in Fig.11c, has a fluid compression corner (right lower) and a free extrusion corner (left lower) in its cross-sections, and its sealing is to resist a seepage from the compression corner to the extrusion corner. In the compression corner, the O-ring is continuously compressed by fluid from a small room into a great room, which causes it to continuously get off its cavity wall and causes its fluid actuation area and force to get greater and greater. In the extrusion corner, the O-ring is continuously compressed by the increasing actuation force from a great room into a small room and causes its sealing state to get better and better (see Fig.9). Accordingly, if an inscribed arc surface of the square cavity is substituted for the other two corners, the O-ring has only a fluid compression corner and a free extrusion corner, or has not any power consuming unnecessary flow and enormous friction, so that any fluid pressure capable of causing it to start seeping can cause its sealing actuation area and force to start a continuous increase and cause its tightness to start a continuous enhancement, or that any fluid pressure capable of causing an O-ring whose cross-sectional circle is not less than the inscribed circle of its cavity to start seeping can cause it to automatically reach its fully leak-free state and that it does not need to be assembled up to its fully tight contact. Therefore, the key to a self-energizing seal of O-rings is their seepage-accompanied self-sealing behavior caused by a fluid compression corner and a free extrusion corner formed by a uniformly each-side-touching assembly in cavity.

As to the rubber O-ring in radial seal applications, if assembly only causes its two radial sides of cross-sections to touch its cavity wall (see Fig.4c), then the Gough-Joule warm shrinkage causing it to inward shrink will cause its radial external side of cross-sections to get off its sealed surface and cause some leakage. If assembly causes all its four sides of cross-sections to touch its cavity wall (see Fig.11c), then its axial wall will effectively contain its radial inward shrinkage caused by the Gough-Joule warm shrinkage and make it have a fluid compression corner and a free extrusion corner that can start its self-energizing deformation at any time and be thoroughly rid of any possible leakage.

8 The Design of Power for Self-Sealing Rings

A rectangular self-sealing ring whose design and assembly both are qualified can exactly convert a fluid pressure p on its internal cylinder into a sealing stress S on its end faces by deforming as soon as the fluid pressure p arises. As shown in Fig.9, the fluid pressure p on the internal cylinder is causing the ring's height to increase, and the seeping fluid pressure p on the end faces, causing the ring's height to decrease. Therefore, it is only when the ring has an internal cylinder area not less than its end face area that it can be ensured that it has an enough power for its sealing deformation and for its tight maintenance; i.e. the power-designing condition for a rectangular self-sealing ring shall be:

$$\pi d_{1r} h \geq \pi(d_{1r} + b)b$$

$$d_{1r} kb \geq (d_{1r} + b)b$$

$$k \geq (1 + b/d_{1r})$$

where b = wall thickness of a rectangular ring

$h = kb$ = height of a rectangular ring

d_{1r} = internal diameter of a rectangular ring.

Any rubber O-ring in a two-wall-touching assembly of the prior art needs to move a distance before starting its sealing deformation against the third wall, and hence its sealing power shall be designed at least for its fluid action area to be greater than its sealing contact area in order to be able to overcome its static friction. Any rubber O-ring assembled by a uniform touch of its each cavity wall, without enormous static friction, can deform/move or move/deform to automatically reach its most efficient self-sealing state once under somewhat of a fluid pressure; i.e. any initial seeping can cause it to automatically reach a compressed state without any unnecessary touch of its cavity wall or cause it to automatically work as a powerful great-small end piston against its sealing contact surface and does not need to consider its self-energizing power when designed.

9 Xu's Rectangular Ring Seal [2]

Xu's rectangular ring seal is designed for a self-sealing joint of two opposing flat faces or flange faces, and has some more unique advantages over a face seal of O-rings.

As shown in Fig.12, Xu's flange joint includes a designed (port) end (A) and a fully flat (cover) end (B). on the designed end, there are a supporting macrosawtooth ring (05), two sealing microsawtooth rings (04) used to provide a pressure-tight joint, and a rectangular ring cavity (Φd_2) used to provide a self-sealing joint, dually ensuring a safest seal.

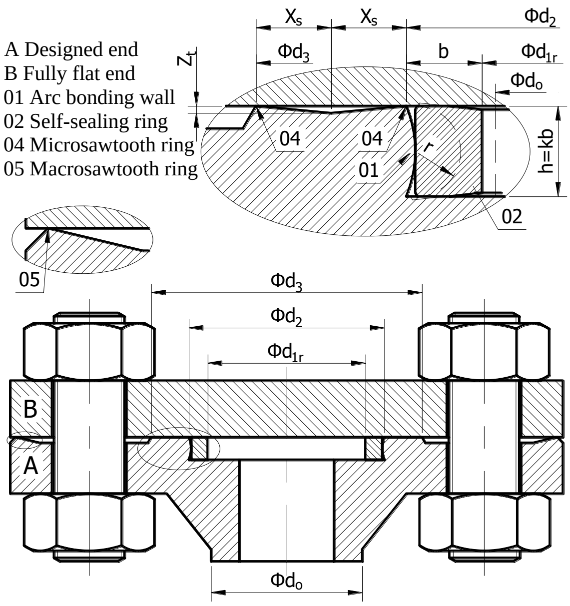


Fig.12 *Xu's Flange Joint*

The microsawtooth rings (04) are equivalent to a surface weakening design for lowering the rigidity of the sealing contact layer and ensuring a sealing difficulty factor less than one. The sawtooth height (Z_t) equals 0.02~0.03 mm, approximately being 10~15 times surface roughness R_a of butt faces. The ratio of the sawtooth pitch (X_s) to the sawtooth height (Z_t) is 20~500 so as to ensure that the sawtooths are both easily deformed into the imperfections caused by a surface roughness R_a not more than 3.2 μm and repeatedly used without any plastic deformation.

The top of the supporting macrosawtooth ring (05) and the top of the sealing microsawtooth rings (04) are on the same plane, and hence the structure that can virtually withstand the tightening compressive load is the microsawtooth but not the macrosawtooth because the former has a substrate far stronger than the later, so that the macrosawtooth ring does not influence any sealing deformation of the microsawtooth rings at all. However, it is impossible for the first round of uniform hard tightening by fingers and for the second round of uniform snug tightening by wrench and more impossible for each following round of uniform hard tightening by wrench to cause the macrosawtooth to be more compressed for more than 1μm in a partial direction in one round when tightened by a sequence of multiple cross-tightening rounds with torque increased by rounds, so that the difference of the finished sealing compressive deformation in circumference is at most within one or two μms and enables the calculated sealing stress to fully approach the actual sealing stress.

It is still necessary to point out that the peripheral macrosawtooth ring (05) is also useful for bolted self-sealing joints and that the macrosawtooth ring (05) and microsawtooth ring (04) are also very easily machined and inspected.

As shown in Fig.13, the arc bonding wall (01) is so diametrally inward bulged as to be capable of providing both an elastic deformation rotation fulcrum with r as the rotating radius and two full Poisson's deformation compensation and offset angles (θ_x) for a rectangular self-sealing ring (02) that touches its cavity wall at the middle, and hence any fluid pressure can cause the ring to work as two rigid self-sealing wedges before the ring yields and as two pliable self-sealing lumps after the ring yields or can cause the ring to get into a full self-sealing state as long as the ring can first deform in the joint system under a fluid pressure.

As shown in Fig.13 again, the self-sealing ring (02) is designed to outwards acuminate or dwindle its both end walls according to "height (h_1) of ring between two weakening ridge bottoms < depth (h) of ring cavity < height (h_2) of ring between two weakening ridge tops", so that any fasteners that have potentiality to cause the ring ends to be crushed may not have any chance to cause the ring body to yield, absolutely ensuring itself a sealing difficulty factor (m_1) less than one. However, any film, such as grease film, and any compressed material, such as water and rubber, without extrusion gaps have such an infinite compressive strength that the ring's sealing contact layer can only be compressed into a film and cannot be forever crushed. Therefore, any bolted flange connection appropriately designed according to "total fastener tensile capacity/area > pipe tensile capacity/area > ring body section area > ring end contact area" can ensure that:

a. fasteners, being stronger than the connected pipe in tensile capacity, have a strength condition that can cause both the ring ends a full plastic deformation and the ring body a full elastic deformation,
b. a deformation that first and more happens during assembly and service is the ring's sealing deformation but not any other deformation of the other components, and thus
c. a flange seal never has any failure caused by its own sealing capability and its own strength when the ring is so designed in accordance with $k > 1 + b/d_{1r}$ as to ensure "its sealing actuation force (F_s) > its unsealing actuation force (F_u)".

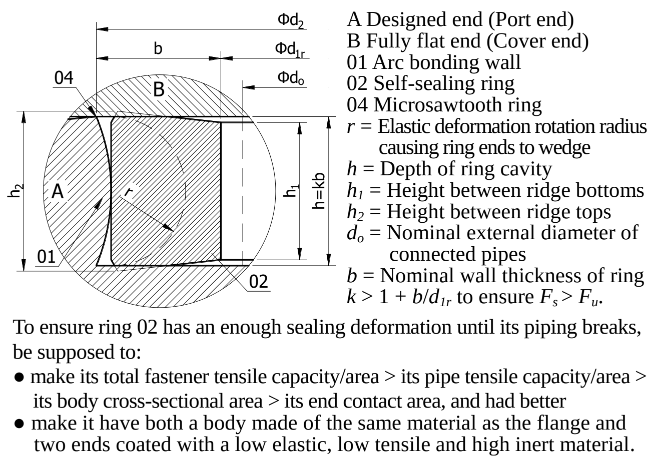


Fig.13 *Rectangular ring seal in Xu's flange joints*

The Xu's self-sealing ring (body) and flange made with a similar material can ensure that there is no thermal expansion coefficient difference that can create any contact clearance therebetween in thermally cycled services, and the Poisson's deformation provided by their assembly can eliminate any radial contact clearance therebetween caused by manufactured errors. Therefore, any Xu's rectangular metal ring in any thermally cycled service can be deformed up to a full self-sealing state once under somewhat of a fluid pressure.

Some coatings of low elastic, low tensile, high inert materials such as gold and nickel can further lower the sealing difficulty factor (m_1) of Xu's rectangular metal ring up to the extent that the value of m_1 is far less than one, and ensure it has such a minimum necessary sealing stress (y) approaching zero as not only to more easily realize and maintain its leak-free joint but also to more easily pass any pressure test up to a burst pressure with it uniformly loosened to a finger-tightened state in the original position after tightened to a fully deformed state.

On the one hand, any fluid pressure on the Xu's self-sealing ring, before its fasteners and its connected body are broken, not only does not cause its compressive stress to decrease but also can cause it to recover its compressive stress decrease caused by cold flow; on the other hand, any ring in elastic compression may not get to a finger-tightened state due to cold flow. Hence, it can be said that Xu's self-sealing ring seal can withstand any relaxation caused by cold flow of material.

Therefore, Xu's rectangular ring seal can be a most ideal face seal because it can be made with a material similar to pressure vessel and not limited by any working temperature and pressure, any thermal expansion coefficient, any corrosion resistance and any manufacturing technology of materials.

10 Xu's O-Ring Seal [2]

10.1 The Circle-Based System of O-Ring Seals

It can be seen from the foregoing self-sealing mechanism for O-rings that an ideal rubber O-ring seal, in cross-sections, shall have only a fluid compression corner and a free extrusion corner formed by a uniformly each-side-touching assembly of its O-ring in its cavity, or shall have a round-wall cavity whose round wall is tangent to the straight walls of its only two corners and concentric with its free O-ring, and so, because its O-ring can have a cross-sectional circle nominally equal to its round-wall circle of cavity, can be called a circle-based system of O-ring seals.

The circle-based system of O-ring seals can be categorized into a square-based round-wall cavity O-ring seal (see Fig.14), a right-triangle-based (or polygon-based) round-wall cavity O-ring seal (see Fig.15) and an isosceles-right-triangle-based round-wall cavity O-ring seal (see Fig.16), if their cavity is regarded as a geometry formed by revolving a closed curve consisting of the incircle and the selected sides of a polygon.

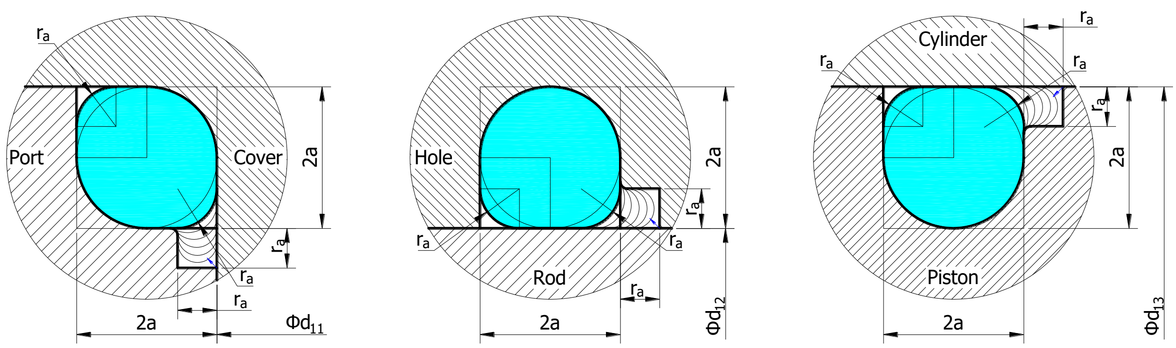
As to the square-based round-wall cavity of O-ring seals shown in Fig.14, excluding its overflow chamber area r_a^2,

its section area $A_c = 2a^2 - 2r_a^2 + 0.5\pi r_a^2 + 0.5\pi a^2$
$= (2 + 0.5\pi)a^2 - (2 - 0.5\pi)r_a^2$

its void section area $A_v = 2r_a^2 - 0.5\pi r_a^2$
$= (2 - 0.5\pi)r_a^2$

its void percentage $C_v = A_v/A_c$
$= (2 - 0.5\pi)r_a^2/[(2 + 0.5\pi)a^2 - (2 - 0.5\pi)r_a^2]$
$= (2 - 0.5\pi)/\pi$ (when $r_a = a$)
$= 2/\pi - 0.5$
$= 14\%$ (maximum void percentage)

The additional overflow chamber (r_axr_a) in Fig.14 is necessary for an O-ring with a severe saturated swell in fluid and a severe thermal expansion relative to its cavity, because the original void of the cavity is too few.



(a) Port seals (b) Rod seals (c) Piston seals

Fig.14 *Square-based round-wall cavity of O-ring seals*

As to the right-triangle-based round-wall cavity of O-ring seals shown in Fig.15, which is also regarded as a cavity based on polygon AEFD or square ABCD (2*a*x2*a*),

its section area $A_c = (1 + \sqrt{3} + \pi/4 + \pi/6)a^2$
$= 4.0410a^2$

its void section area $A_v = r_a^2 - \pi r_a^2/4 + \sqrt{3}r_a^2 - \pi r_a^2/3$
$= (1 + \sqrt{3} - \pi/4 - \pi/3)r_a^2$
$= 0.8994r_a^2$

its void percentage $C_v = A_v/A_c$
$= 0.8994r_a^2/4.0410a^2$
$= 0.2226r_a^2/a^2$
$= 22\%$ (max. void pct., when $r_a = a$)

Generally, the cavity in Fig.15 may not need any additional overflow chamber for a saturated swell and a thermal expansion of O-rings, but its sealing ability at positive and negative pressures are somewhat different from each other.

As to the isosceles-right-triangle-based round-wall cavity of O-ring seals shown in Fig.16,

its incircle radius $a = \sqrt{2}a'\text{tg}22.5° = 0.5858a'$

its half leg side length $a' = a/(\sqrt{2}\text{tg}22.5°) = 1.7071a$

its section area $A_c = 2(2a' - a)a + 0.25\pi a^2$

$= 4a^2/(\sqrt{2}\text{tg}22.5°) - 2a^2 + 0.25\pi a^2$

$= [4/(\sqrt{2}\text{tg}22.5°) - 2 + 0.25\pi]a^2$

$= 5.6138a^2$

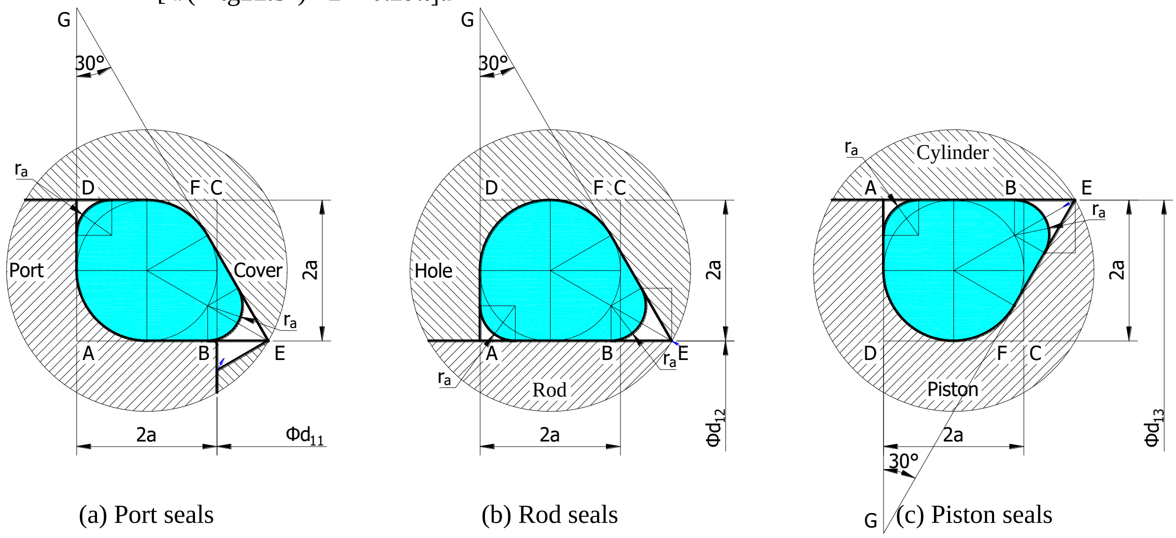
its void section area $A_v = 2r_a^2/\text{tg}22.5° - 0.75\pi r_a^2$

$= 2.4722r_a^2$

its void percentage $C_v = A_v/A_c$

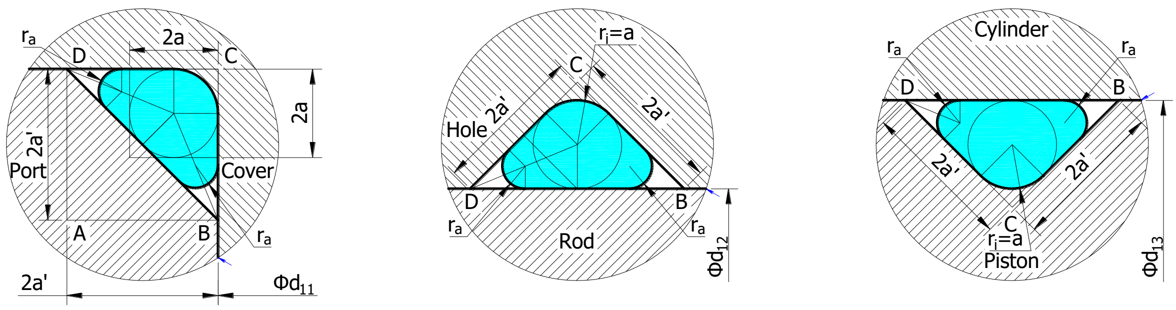
$= 0.4404r_a^2/a^2$

$= 44\%$ (max. void pct., when $r_a = a$)



(a) Port seals (b) Rod seals (c) Piston seals

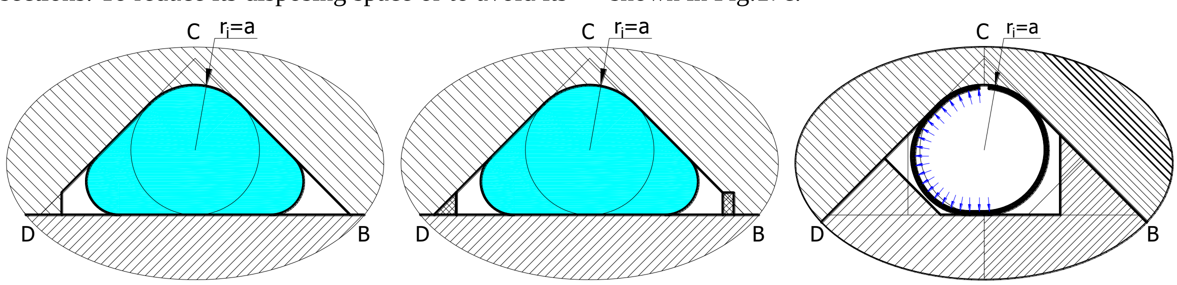
Fig.15 *Right-triangle-based round-wall cavity of O-ring seals*



(a) Port seals (b) Rod seals (c) Piston seals

Fig.16 *Isosceles-right-triangle-based round-wall cavity of O-ring seals*

The cavity in Fig.16 has both an enough void for a saturated swell and a thermal expansion of rubber O-rings and an identical sealing ability at positive and negative pressures, but needs either a greater disposing space or a smaller O-ring in cross-sections. To reduce its disposing space or to avoid its O-ring's excessive deformation in corners, some corner improving designs can be used, such as a mini-truncated corner shown in Fig.17a, a corner filled with anti-extrusion rings shown in Fig.17b and an optional much-truncated corner shown in Fig.17c.

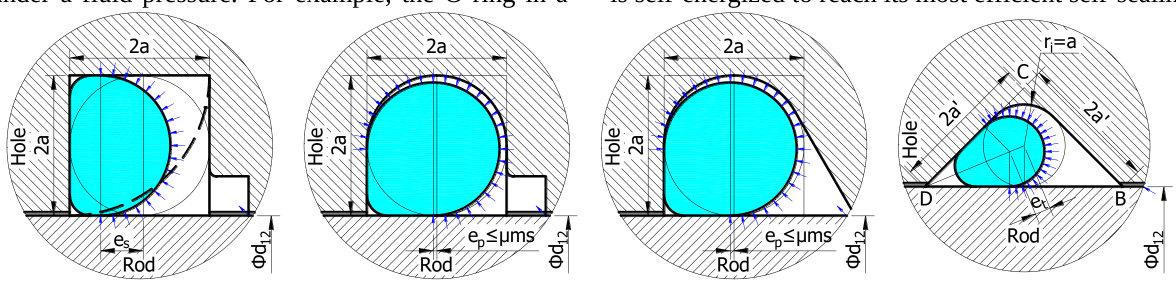


(a) Mini-truncated corners (b) Corners filled with backup rings (c) Optional much-truncated corners

Fig.17 *Truncated or filled corners of isosceles-right-triangle-based round-wall cavities*

Given that any rubber O-ring can only accept fluid compression by one single arc surface tangent to its cavity wall, it will get fully off its round wall of cavity and reach its most efficient self-sealing state without any unnecessary friction shown in Fig.18 once compressed in the compression corner to get to the same as its cavity incircle in diameter or once the seepage is up to the tangent point of the straight wall and the round wall; i.e. as to a circle-based system of rubber O-rings, the fewer its assembled bulge beyond the incircle, the easier to reach its most efficient self-sealing state instantly under a fluid pressure.

It is necessary to further point out that any rubber O-ring whose cross-sectional circle diameter equals its base circle diameter can only deform and move to its sealing state under a fluid pressure as shown in Fig.18, because any fluid pressure that can cause it to deform/move or move/deform a little can cause it to have such an inner pressure increment causing its freer extrusion surface to outward bulge that any O-ring with somewhat of damping or moving or deforming reaction to seepage can reach its most efficient self-sealing state instantly under a fluid pressure. For example, the O-ring in a square cavity in Fig.18a, whose cross-sectional circle diameter equals its base circle diameter, can only deform to its sealing state shown in the continuous wide line but not to its leaking state shown in the dashed wide line under a fluid pressure, because the deformation shown in the dashed line will cause a pressure field with a higher upper pressure that can only cause a deformation shown in the continuous line. In other words, the deformation of any O-ring whose sectional circle diameter is not less than its base circle diameter can only be a sealing deformation but not a leaking deformation in a circle-based system of cavities under a fluid pressure. However, as for the deforming move distance (*e*), e_s (in square cavities) > e_t (in isosceles-right-triangle-based cavities) > e_p (in the other polygon-based cavities) ≤ μms; i.e. as for a circle-based system of O-ring seals, the more the round wall portion of its cavity, the faster the speed at that it is self-energized to reach its most efficient self-sealing state.



(a) Square cavities (b) Square-based cavities (c) Right-triangle-based cavities (d) Isosceles-R.T.B. cavities

Fig.18 *Fully self-energized state of circle-based system O-ring seals*

Therefore, an O-ring of the circle-based system of O-ring seals has an cross-sectional circle used as the cross-sectional size of it and its cavity and nominally equal to the inscribed circle of the base polygon of its cavity, but each cavity of a size can have different compression and extrusion corners, and the rubber O-rings of a size can be grouped according to volume or weight errors. In other words, the designing of any rubber O-ring seal is to simply selects a combination of a cavity having different compression and extrusion corners with an O-ring having a volume error range within a size to ensure that its O-ring can have both an enough expansion space in a possible high temperature service and a safe initial wall-touching state at a possible low temperature.

10.2 Ideal O-Rings

Theoretically, a sealing contact layer of any sealing element shall be soft & inelastic and assembled to its full (plastic) deformation, whereas the contact layer substrate shall be strong & elastic and assembled to its full elastic deformation. Hence, any soft material (such as rubber, PTFE, lead, indium etc.) O-ring with a spring core (07) shown in Fig.19, a plating core (08) shown in Fig.20a, a perforated O-tubing core (09) shown in Fig.20b or a C-tubing core (10) shown in Fig.20c is more ideal than the traditional rubber O-ring without any support core, being capable of raising both its temperature limit and its corrosion resistance in services.

As shown in Fig.18, the softer the O-ring body material, the higher the requirement of its resilience; and the poorer the resilience of the O-ring body material, the higher the requirement of its strength resisting a plastic deformation. Hence, any metal O-ring, with a very poor resilience, shall have an enough tensile strength for resistance to any plastic deformation under a maximum working pressure or a proof test pressure. Because not only some soft coatings can meet the deformation requirements of metal O-rings in interference assembly and in service but also the O-ring body and the cavity body can be such a similar metal that there will be no longer any thermal expansion or saturated swell problem, thus any solid or hollow (11) metal O-ring and any metal C-ring (12), as shown in Fig.21, with a soft coating such as PTFE, indium, lead, nickel, gold etc. can be more ideal than the traditional rubber O-ring, especially in thermally cycled services, as long as it has an appropriate strength.

Therefore, the O-ring in the future widest use will no longer be any traditional rubber O-ring without any strengthening core because a large number of gasket seals will also be replaced by some rubber and non-rubber O-rings with cores and by some coated metal O-rings or C-rings, and the word O-ring will also no longer only refer to the traditional rubber O-ring.

11 Conclusions

Under Xu's sealing theory, the only criterion of a qualified seal is that its minimum necessary sealing stress y ≤ 0.2 MP_a.

However, all the sealing ideas and standards of the prior art are so mistaking black for white and unscientific as to be unable to ensure that seal's y ≤ 0.2 MP_a. The ROTT method and EN 13555 have fully confirmed that there is no leak-free gasket seal in the prior art. From the statement that “conformance to the dimensional information in ISO 6149 does not guarantee rated performance” in ISO 6149 that shall be but cannot be in accordance with ISO 3601, it can be seen that the O-ring seal of the prior art is a severe problem. The Report on the Space Shuttle Challenger Accident cannot find but positively records that the disaster was caused by the sealing idea or standards of the prior O-ring seal.

Note: This paper was initially published in Chinese magazine of Petro-Chemical Equipment, 2013, Volume 42 (2): 75-85.

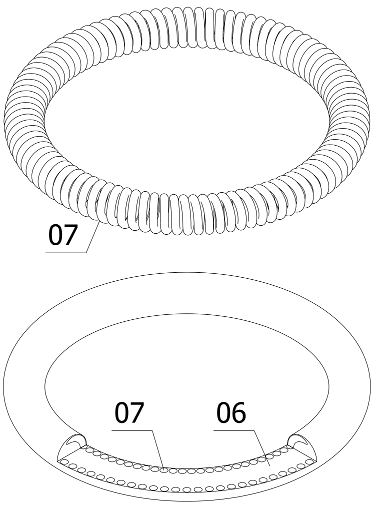


Fig.19a Soft O-rings with spring core

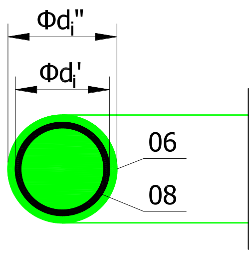


Fig.20a *Soft O-ring with plating core*

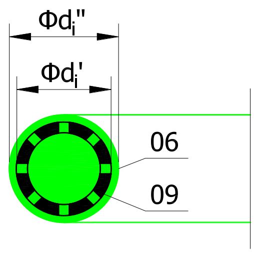


Fig.20b *Soft O-ring with O-tubing core*

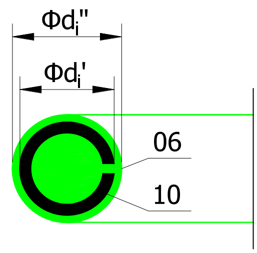


Fig.20c *Soft O-ring with C-tubing core*

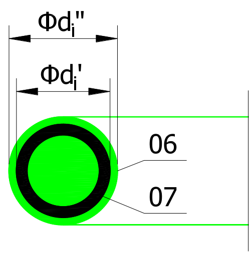


Fig.19b *Soft O-ring with spring core*

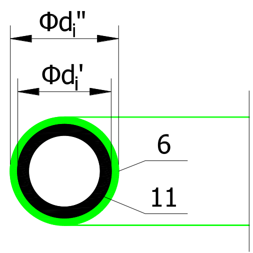


Fig.21a *Metal O-ring with coatings*

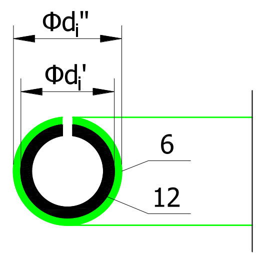


Fig.21b *Metal C-ring with coatings*

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Annex A Further Dissections of the Prior Art

A.1 The Prior Art Has Found that Some Material Has Both Solid Behavior and Liquid Behavior

Professor He etc. (see ref. 5) tell us that amorphous polymers exist in the three mechanical states of glassy state, rubbery state and viscous flow state with temperature changes, and feature both a solid behavior and a liquid behavior in the rubbery state. They adds "This results from each moving situation of two different sizes of moving monomers: each segment of molecular chains behaves like a liquid, and each molecular chain as a whole behaves like a solid. Therefore, the condensed state matter exhibits a duplicity in behavior, or both a solid behavior and a liquid behavior."

From ref. 6, it can be seen that Jin Wang further observes a true liquid behavior on polymer thin film surfaces by instrument.

It can be seen from refs. 5 and 6 that the people have found for a few dozens of years that an amorphous polymer can behave like both liquid and solid, but so far the liquid behavior of polymers has not been intentionally used to transmit a fluid pressure in sealing devices.

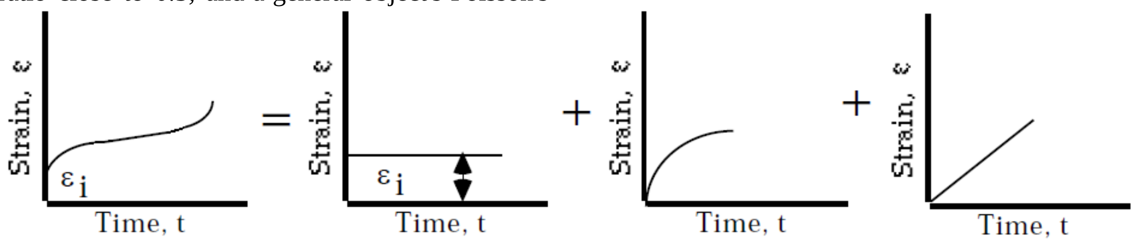
In fact, it can be seen from bulk modulus $K = E/[3(1-2\upsilon)]$ that a general object can behave more or less like a liquid, because any liquid features a flowability or deformability and an incompressibility or a great bulk modulus, or features a Poisson's ratio close to 0.5, and a general object's Poisson's ratio is above 0 and below 0.5. Accordingly, it can be said that Poisson's deformation ratio has been discovered for the sealing application of materials for more than one hundred years.

A.2 The Prior Art Has Found that Poisson's Ratio Changes with temperature & time

Ref. 7 tells us that any object can undergo some delayed creep deformations at a homologous temperature more than 0.5, i.e. any object has some temperature-and-time-dependent creep strains (see Fig.A.1).

Ref. 8 further tells us that Poisson's ratio for a crept material is the total deformation ratio at the creep finish, or Poisson's ratio for an object can almost change as synchronously with temperature and time as creep strain does.

Therefore, it is imaginable that any self-sealing ring needs a Poisson's deformation compensation angle for adding its orthogonal deformation response to fluid pressures so as to instantly fully perform its sealing duty once under a fluid pressure. In other words, the rubber O-ring seal of the prior art has a terrible sealing performance at a start or impulse pressure, or is not so perfect as originally expected, because it cannot instantly produce any deforming movement from a great room into a small room to start its self-sealing actions on being under a fluid pressure.



Total creep curve | Elastic sudden strain | Plastic transient strain | Viscous strain

Fig.A.1 *Superposition of Various Phenomenological Aspects of Creep*

A.3 The Prior Art Has Found that Rubber Has Some Properties that Are Exactly Opposite to a General Material

Ref. 9 tells us that the molecular chains of any rubber, as shown in Fig.A2a, wind and tangle around each other to be elastic when not stretched, and when stretched, as shown in Fig.A.2b, will line up straight in its stretch direction to behave like a crystallizing process causing it to get stronger; and, when compressed, will get more wound and more tangled and weaker. In other words, rubber will get stronger when stretched and will get weaker when compressed. Because the rubber in and out of an O-ring's extrusion corner is separately stretched to get stronger and compressed to get weaker, thus a rubber O-ring greater in cross-sections and capable of causing more rubber to be compressed into its extrusion corner can withstand a higher fluid pressure.

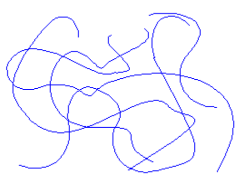


Fig.A.2a *Unstretched Rubber*

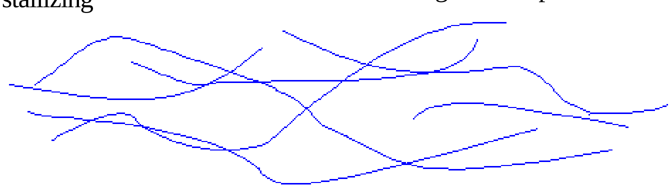


Fig.A.2b *Stretched Rubber*

The Gough-Joule effect tells us that a stretched rubber piece, as shown in Fig.A.3, will not elongate because of thermal expansion but will shorten in its stretch direction when warmed up; meanwhile, its tensile modulus will increase but not decrease; for molecular chains being warmed up are ceaselessly acquiring some energy recovering its original wound and tangled state, and the more being stretched the rubber and the faster the temperature rise, the greater and the more violent the recovering shrinkage force.

Fig.A.3 Gough-Joule Effect

Obviously, a rubber O-ring in a four-side-touching assembly can effectively avoid the influence or leak caused by both any cold shrinkage or cold being stretched and any Gough-Joule warm shrinkage of rubber O-rings.

The Report on the Space Shuttle Challenger Accident cannot pay any attention to the influence that the O-rings have more cold being stretched at a colder ambient temperature and have more Gough-Joule warm shrinkage after ignition, and so is incorrect.

A.4 Dissection of Parker's Leak Test of Rubber O-Rings of the Prior Art

A.4.1 Parker's Test Device and Results for Rubber O-Ring Leak Rate of the Prior Art

Parker's test device for O-ring leak rate is a face seal shown in Fig.A.4 (left), having a Butyl O-ring of 4.850 Inch I.D. at 15%, 30% and 50% squeezes tested under 0.41 MP_a at 25ºC. The tests achieve a squeeze-leak rate curve shown in Fig.A.4 (right).

Parker's leak rate curve shows that grease coatings can reduce the leakage of the O-ring at squeezes less than 50%, but neither the O-ring at 50% squeeze can create a fully leak-free joint nor its sealing performance can be further improved by grease coatings (see Ref. 4).

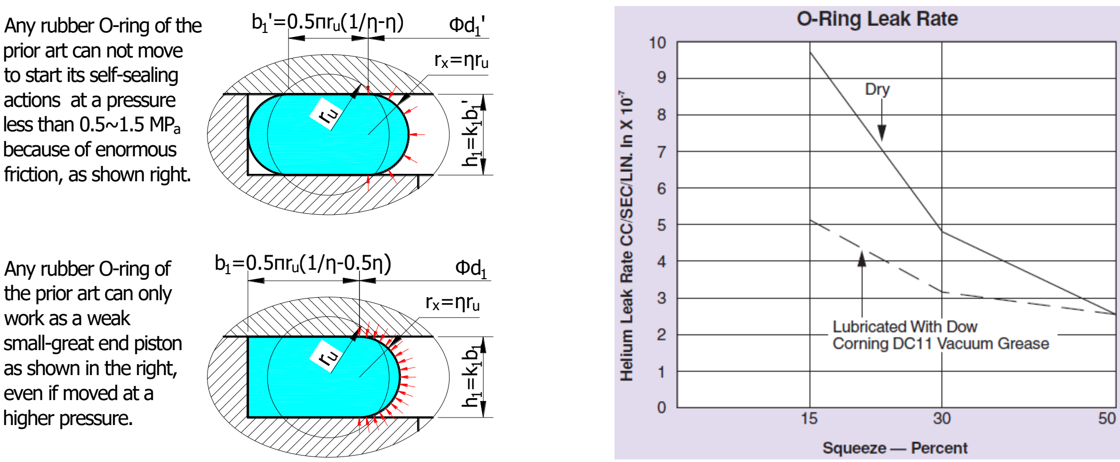


Fig.A.4 *Parker's Leak Test of Rubber O-Rings*

A.4.2 The Structural Problem with Rubber O-Ring Seals of the Prior Art

As to an O-ring just assembled into its cavity (see Fig.A.4, upper left),

$\because$ its assembled stress $S_a = p_u r_u/r_x = 0.1r_u/r_x$ (MP_a), where r_u and r_x are separately the cross-sectional radii of the O-ring before and after assembled into cavity (see Clause 7),

$\therefore$ at 50% squeeze or at $r_u/r_x = 2$, $S_a = 0.1r_u/r_a = 0.1x2 = 0.2$ (MP_a);

in other words, any rubber O-ring in a two-wall-touching assembly shall have an assembly squeeze $(1 – \eta)$ more than 50% in order to ensure that the O-ring at least has an assembled stress not less than 0.2 MPa that can resist atmospheric seepage through the sealing contact, because the minimum necessary sealing stress y for a rubber sealing element is 0.2 MP_a (see Clause 4).

As to an O-ring in its working state (see Fig.A.4, lower left), given that its cross-sectional area is the same as the area before assembled into its cavity, then

$\because$ $\pi r_u^2 = 0.5\pi r_x^2 + 2r_x b_1$, where r_u and r_x are separately the cross-sectional radii of the O-ring before and after assembled into cavity,

$\therefore$ $b_1 = 0.5\pi r_u(1/\eta - 0.5\eta)$;

$\because$ $(h_1 = \mathbf{k_1} b_1 =)$ $\mathbf{k_1} 0.5\pi \mathbf{r_u}(1/\eta – 0.5\eta) = 2\eta \mathbf{r_u}$

or $\eta^2 = \pi k_1/(4 + 0.5\pi k_1) = 1/(0.5 +4/\pi k_1)$

$\therefore$ To ensure $k_1 > (1 + b_1/d_1) > 1$(see Clause 8), be supposed to ensure $\eta^2 > \pi k_1/(4 + 0.5\pi k_1) = 1/(0.5 +4/\pi k_1)$.

It can be seen from the above inequality that at $k_1 = 1$, $\eta > 0.75$, $(1 – \eta) < 25\%$. Hence, be supposed to ensure an O-ring a maximum assembly squeeze less than 25% in order to ensure that it has an enough sealing power. In other words, any rubber O-ring in a two-wall-touching assembly shall have an assembly squeeze $(1 – \eta)$ less than 25% so as to ensure that it shall not work as a weak small-great end piston as shown in Fig.A.4 (left).

Therefore, it is impossible to make a rubber O-ring of the prior art have an assembly squeeze that can at the same time meet the necessary requirements of its initial sealing stress and working power by designing.

A.4.3 Dissections of Parker's test results on Xu's sealing theory

—— *Why can grease coating reduce the leakage of rubber O-rings at less than 50% squeezes?*

Because the minimum necessary sealing stress *y* of a grease coating is 0.1 $\mathrm{MP_a}$, the minimum necessary sealing stress *y* of a sealing element of rubber is 0.2 $\mathrm{MP_a}$, and a rubber O-ring whose assembly squeeze is less than 50% can only reach an assembled stress $S_a < 0.2\ \mathrm{MP_a}$ or does not reach its fully leak-free tight level that can withstand 0.1 $\mathrm{MP_a}$ (atmospheric pressure); i.e. because grease coatings can lower the minimum necessary sealing stress *y* of a rubber O-ring at less than 50% squeeze and improve its initial sealing tightness.

Note: It have been proved that a greased O-ring that was once compressed and decompressed by a fluid pressure is worse than a not greased O-ring in initial sealing tightness at no fluid pressure.

—— *Why can the rubber O-ring seal at 50% squeeze yet leak under 0.41 $\mathrm{MP_a}$?*

Because any rubber O-ring at 50% squeeze just reaches an assembled stress $S_a = 0.2\ \mathrm{MP_a}$ or just reaches its fully leak-free tight contact that can withstand 0.1 $\mathrm{MP_a}$, and a fluid pressure of 0.41 $\mathrm{MP_a}$ cannot yet cause the O-ring to move or deform to automatically reach its fully leak-free tight contact that can withstand 0.41 $\mathrm{MP_a}$.

—— *Why cannot grease coatings further improve the rubber O-ring seal at 50% squeeze?*

Because what the rubber O-ring at 50% squeeze needs for its fully leak-free tight contact under 0.41 $\mathrm{MP_a}$ is only that its cross-section can have an adding movement/deformation that can cause it to reach a tighter contact, but the fluid pressure of 0.41 MPa cannot cause its cross-section to move/deform.

—— *Conclusions*

Parker's leak tests have proved that the minimum necessary sealing stress *y* of a rubber sealing element is 0.2 $\mathrm{MP_a}$, the minimum necessary sealing stress *y* of a grease coating is less than 0.2 $\mathrm{MP_a}$, and a rubber O-ring whose assembly squeeze is up to 50% just reaches its fully leak-free tight level that can withstand 0.1 $\mathrm{MP_a}$ (atmospheric pressure); or Parker's leak tests have proved Xu's sealing theory scientific.

A.5 The O-Ring Seal Standards of the Prior Art Far Away from the ABC of Seals

The isotropic softness, incompressibility and elasticity of rubber cause its O-ring to be like a ring of shaped elastic liquid, whose internal pressure raises and lowers synchronously with its external pressure. Given that there can only be a uniform fluid pressure on a rubber O-ring in its cavity, its fluid compression surface always resists the compression from its external fluid pressure by a single round surface with a uniform compressive ability, and its extrusion surface always resists the stretching from its internal pressure by another single round surface with a uniform tensile ability. In other words, both the fluid compression surface and the free extrusion surface of a rubber O-ring can only be a single round surface that is tangential to its cavity walls.

However, Patent US 2180795 in 1937 explicitly taught us that a rubber O-ring is perfectly circular in cross section prior to its assembly and insertion in a cylinder, and, when slid with the piston into the cylinder, compressed to the shapes with somewhat of an ellipsoidal or rather square or rectangular cross section. As shown in Fig.A.5, ISO 3601:2008 (left) deems the rubber O-rings in a two-wall-touching assembly ellipsoidal in section, and Parker (right) also deems them somewhat rectangular. In other words, for about 75 years the prior art of rubber O-ring seals has not deemed the fluid compression surface and the free extrusion surface of a rubber O-ring in its cavity two different radii of single round surfaces tangent to its cavity wall; i.e. the prior art has not known how for a rubber O-ring to deform or work, or what fluid pressure limit or rating for a rubber O-ring to withstand, or how to design a rubber O-ring seal. Actually, a rubber O-ring is virtually a length of water-filled soft thin wall tubing, which has a definite working pressure limit or rating that can be calculated.

Therefore, all the technical standards of rubber O-ring seals were drafted at will before knowing their relative ABC, having no coordination. For example, Clause 6.3 of ISO 3601-2:2008 specifies, “Housing fill of the installed O-ring should not be more than 85 % to allow for possible O-ring thermal expansion, volume swell due to fluid exposure and effects of tolerances.” This specification is thoroughly neglected in ISO 6149. If the specification in ISO 3601is scientific, there should be no ISO 6149 at all. Perhaps, it is because it is diametrically against the ISO 3601 that it has to declare “conformance to the dimensional information in ISO 6149 does not guarantee rated performance” to evade its responsibility.

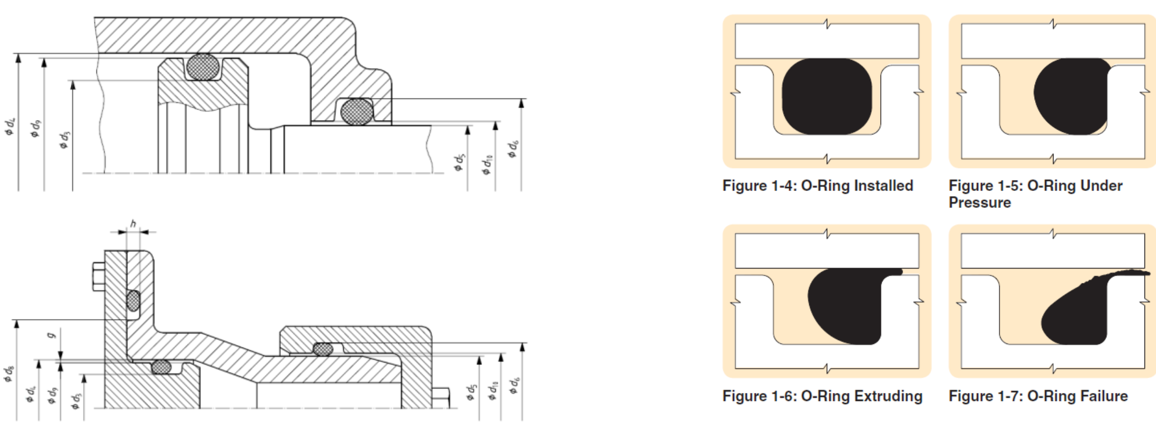


Fig.A.5 *The O-Ring's Sectional Shape Mistaken in ISO 3601 (left) and in Parker's O-Ring Handbook (right)*