Leakage Evaluated and Controlled from Industrial Process Pipeline by Optimum Gasket Assembly Stress

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Abstract

Air pollution studies for hazardous gases in Taiwan generally consider the impact on the human body, and always put the emphasis on large immobile and mobile pollution sources (like chimneys and vehicles), but they have ignore the leakage between pipelines in process industries. When hazardous materials leak imperceptibly, they increase the pathogenic risk to the workers at the workplace. Since 1937, traditional designers of gasket bolted joints have always utilized the “y” factor for seating yield, and the “m” factor for maintaining the gasket load under operating conditions as per ASME Boiler and Pressure Vessel Code. If industries still continue to utilize these simple and obsolete methods to seal flanges and attempt to reduce emission, how can they correctly select the right gasket materials and determine optimal loading as well as operating conditions to ensure required gasket sealing performance? During the last 30 years, more new gasket materials have been introduced to achieve required emission performance. The old factors are not appropriate for the new gasket materials, because there are no comparable test methods for “m” and “y”, and claims for reduced emission based on “m” and “y” cannot be verified. This study will ultimately result in a complete proposal to prevent any hazardous gas leaks in the process industries, to protect workers involved at the source of the hazardous gases, and to analyze the risk for workers.

Keywords: leakage, gasket, flange, risk

1. Introduction

In the past few decades, evaluation of tightness property in gaskets has been considerably poor. , Visual observation is usually the way to determine whether leakage occurs. If on-scene workers determine the happening of leakage through eye-seeing, then leakage should not have occurred in gasket being tightly locked in flange. In long-term practical applications, a large amount of liquid leakage, however, still takes place, and these leaking fluids can hardly be seen simply by eyes or sensed by their odor. In practice, the occurrence of these leakage incidents has something to do with false selection of gaskets, inappropriate setting and repair, as well as poor maintenance.

In the traditional gasket design, obtaining the stress of a setting is based on a simple calculation. Since the 1950s, the “m” and “y” factors have been the most vital and widely accepted, designed and used by persons who set the gaskets **[1][2]**. In the present design and model, the “m” and “y” factors have played a crucial role because they can be used to realize some traits of closeness in the gasket. In practical applications, the real meaning, however, lies in the fact that liquid cannot leak out. Unfortunately, the standard on leakage has not yet been clearly defined. “To see is to believe” and “to smell is to know” are merely the ways to evaluate the leakage. Such methods lack the idea of measuring leakage by quantity, that is, the meanings that “m” and “y” stand for. Under such circumstances, they are only concerned if pipelines have leakages or not in normal operating conditions. Since leakage is judged by the senses, for low odor threshold and highly toxic materials, workers near pipelines have taken in poisonous material unconsciously. By the time that these toxin materials are sensed and leakage is seen, the concentration in the surroundings has long already reached the degree of damage to workers’ health. Judging from the design scale of gaskets by the American Society of Mechanical Engineers (ASME), these documents of design models do not over-demand the tightness property of tightness joints, and they only require tightness that will not lead to large amount of leakage due to the influence of manufacturing conditions, damage of gasket or crushing by too high process pressure and so on **[3]**.

In general, an industrial manufacturing process is made up of many pipelines with different materials and operating conditions. In view of real situations, flanges used to connect pipelines likely have liquid leaks between pipelines without proper ways to set gaskets. Taking 1.3-Butadiene (BD) for an example, the earliest TWA-PELs is required to be 1,000 ppm. However, many animal tests have shown that it causes cancer to rats and mice**[4][5]**. Thus, the Occupational Safety and Health Administration (OSHA) lowered TWA-PELs to 2 ppm and to 1 ppm in 1992 and 1997, respectively. Even under well-controlled situations, in the manufacturing environment of Taiwan it is hard to meet the stringent standard of 1 ppm for BD. From a practical standpoint currently in Taiwan, TWA-PELs of 13-BD can only be restricted to 10 ppm.

In the early 1990s, based on the Clean Air Act Amendment (CAAA), the USA put forth the flange, which has potentially a large amount of liquid leakage by regular control. In practice, it mainly targets dangerous air-polluting materials. In the long term, “m” and “y” factors have not yet satisfied the accuracy and applicability that the closeness property demands. Thus, they have been questioned and discussed since then. After the mid-1970s, with a view to the above-mentioned problem, ASME developed various technologies to predict seal performance for proper evaluation of gaskets so as to propose a more exact and efficient measurement standard of closeness property in gaskets and operating limits of asbestos-free gasket material. Since the 1970s, some new seal researches have been developed by industrial groups including the Pressure Vessel Research Council (PVRC) and the Materials Technology Institute of the Chemical Process Industries (MTI), that have started to work on the development of testing research in the gaskets. Accordingly, ROTT testing was developed in this period **[6]**.

From a practical viewpoint, through a flange joint, it is technically impossible to stop completely the leakage between flanges. Since flange joints cannot seal a fine surface seam, the design of the gasket actually allows a minute amount of leakage to occur. Leakage scale and rate in different gasket materials will be discussed as to the tightness because they are directly related to the amount of leakage.

PVRC and other American industrial organizations have come up with a series of closeness evaluation techniques that can replace the old “m” and “y” factors. These measurement techniques can more effectively and accurately describe the closeness property of gaskets and the differences in application. Based on the testing analysis of ROTT, seal parameters are developed to assist in more accurate gasket design. These newly developed parameters are “Gs”, “Gb”, and factor “a”. “Gb” and factor “a” mainly reflect the closeness property of the gaskets during the setting, and “Gb” is the degree of sensitivity. While inner pressure decreases, Tp (tightness parameter) and

Sg (gasket stresses) would monotonically go down. In particular, three parameters are based on the analysis of ROTT and testing and evaluated by engineering calculations. Processed by standard linear recycling, every parameter differs with the material, thickness, density and processing ways of the tested gaskets, to name a few. These parameters, however, do not directly reflect the closeness property of gaskets but will be used in the mathematical operation induced by ROTT testing. As a result, these new parameters differ greatly with the old “m” and “y” factors **[7][8]**.

2. Room Temperature Tightness (ROTT)

ROTT mainly has pressurized nitrogen, 400 psig and 800 psig, as the testing condition. In practice, testing results will illustrate the seal performance of gaskets and reflect the circular stress-varying situation of the gaskets. Essentially, ROTT tests are mainly divided into two parts. The first part can be obtained from ROTT testing, as shown in Fig. 1. Gaskets under testing take a stress rating, 1,000~10,000 psi, to observe the changes in seal parameters. Normally, gaskets of better seal properties get higher $Tp_{(max)}$. On the other hand, they also help to classify the seal properties and serve as a reference for the choice of material. In the second part, after the fluid pressure goes through repeated switch-on and switch-off, and most gaskets under testing undergo three thorough cyclings; gasket stresses (Sg) vary between 1,000 ~ 10,000 psi. These two parts of data are shown in Fig. 2, and seal properties of gaskets are to follow the fixed-amount analysis of leakage rate and mathematical deduction. Therefore, the main purpose in analyzing Fig. 2 is to obtain the values "Gb" and "a" which both indicate characteristics of tightness, along with "Gs" of seal traits **[9][10]**.

To evaluate the leakage rate under any inner-stress condition, the following mathematical formulas are developed:

$$T_P = \frac{P_D}{P*} \times \left(\frac{L*}{L} \right)^d \quad (1)$$

$$L = L_{rm} \times D_t \quad (2)$$

where：

T_p dimensionless tightness parameters

P_D fluid pressure in the manufacturing process (psia or Kpa)

$P*$ refer to atmospheric pressure (one atmospheric pressure = 14.7 psia or 101.325 Kpa)

$L*$ standard leakage rate which functions on gaskets with 150 mm (5.9 in) in O.D.(mg/sec/mm or lb_m/min/in. Generally assumed as 1 mg/sec/mm)

L predicted leakage which functions on gaskets with 150mm (5.9 in) in O.D. between pipeline flange. It can be expressed as equation (2)

L_{rm} where evaluated leakage rate of pipelines (lbm/min/in or mg/sec/mm)

Dt O.D. of gaskets. Normally set as 150 mm (5.9 in)

d empirical value (if fluid in pipelines is liquid, d=1; as fluid in pipelines is vapor, d=0.5)

Considering that the flow is vapor-liquid two-phase, different degrees of dryness can be used for the calculation:

$$d = \frac{1}{2} \times (2 - X) = 1 - \frac{1}{2} X \tag{3}$$

If the fluid is all vapor, X = 1.0; if it is all liquid, X =0.0. To calculate the leakage rate of the pipelines (L), equation (1) can be rewritten as:

$$L = \left[\frac{P_D}{p^*} \times \frac{1}{T_p} \right]^2 \tag{4}$$

From equation (4), it is clear that leakage rate diminishes 100 times with the increase of “Tp” to 10 times. Comparatively, when “Tp” decreases to 10 times, the leakage rate rises to 100 times.

By applying equation (4), plant operators usually do certain proper simplifications so as to avoid complex calculations. Since every pipeline requires different leakage rate, it is categorized for the convenience of specific application (as shown in Fig. 3).

Simplified equations (1) and (4) can be expressed as follows:
In the general environment, quality X = 1.0 (under normal pressure):

$$Tp = \frac{P_D}{14.7} \times \left(\frac{1}{0.002 \times 150} \right)^{0.5} \times C = 0.1242 \times (C) \times (P_D) \tag{5}$$

Under different sets of pressure,

$$Tp = \frac{P_D}{P^*} \times \left(\frac{1}{0.002 \times 150} \right)^{0.5} \times C = 1.8275 \times (C) \times ({}^{P_D}\!/_{P^*}) \tag{6}$$

Where C is a designed parameter. New paragraph varies with the required seal effects, which can be referred to by Table 1. Accumulated leakage length and time can be assumed and, under various seal situations, total amount of leakage can thus be predicted **[8][10]**.

3. ROTT Testing

From the ROTT testing data, as shown in Fig. 3, part of the data indicates the tightness parameters at which function on gaskets goes up with the increase of Sg. After the calculation by linear regression analysis, the slope “a” and the intercept against y-axis of the curve can be created. With the log-log figure, the information of qualitative analysis in gaskets can be therefore established.

While ROTT testing data is evaluated, the seal property of the gasket increases considerably with the small amount of rising Sg, if tightness curve “a” becomes less mild. “Gb” reflects the needed amount of Sg at the occasion of setting the gasket. “Gb”, which is smaller, means that with the smaller Sg, the effect of expected closeness can be acquired. The last parameter “Gs” represents the relationship between Sg and leakage rate. Smaller "Gs" shows the difficulty of increasing Tp by raising Sg. As a result, it is necessary to retain a higher scale of seal. On the other hand, smaller "Gs" also implies that, with the decrease of inner pressure, changes in leakage rate of pipelines are not so significant.

From Figure 3, data in part A can be found that the $Tp_{(max)}$ varies with that of different materials. The meaning of $Tp_{(max)}$ lies in the fact that as Sg continues increasing to 10,000 psi, the tightness of the gasket influences $Tp_{(max)}$, due to the scale of leakage rate. If Sg is set to 10,000 and $Tp_{(max)}$ is not so high, the gasket material cannot meet with the needs of proper tightness or fit in the low-leakage environment. On the contrary, the gasket has higher seal property.

From the intercept against y-axis, when Tp=1, the tightness capacity of the gasket is “Gb”. With Tp=1, the parameter “Gb” value can be seen as the smallest tightness needed to produce the effect of closeness, similar to but not yet equal to y.

Data in part B shows that, during ROTT testing, to stop pressuring the fluid within pipelines loses the tightness parameters of gaskets as shown in Figure 1. According to the regression analysis of three-time on-and-off fluid pressure curves, the intercept against the y-axis is “Gs”.

The quality of the gasket made by the specific material can be evaluated simply taking a close look at the ROTT data analysis. For example, a comparison of Figs. 4 and 5 can readily differentiate two different gasket materials and the differences of their seal properties. When Sg increases to 10,000 psi, $Tp_{(max)}$ of a PTFE insertable gasket is nearly 43,000, and $Tp_{(max)}$ of a graphite spiral wound gasket is about 11,000. What these two data represent is that, under the same gasket pressure, Sg=10,000 psi, the PTFE insertable gasket has higher Tp. In equation (1), it can be seen that with higher Tp, leakage rate "L" gets lower. Therefore, under the same Sg, the PTFE insertable gasket will attain higher Tp, demonstrating better tightness and lower leakage rate than the graphite spiral wound gasket.

Comparing the relatively linear parts of curves in Figs. 4 and 5, it can be found that, as the PTFE insertable gasket is between 1,000 and 7,000 psi, Tp increases 278 times in total (Tp=90~25,000). In contrast, while the Sg of graphite spiral wound gasket is between 1,000 and 7,000 psi, it increases only 60 times (Tp=30~1,800).

From the loss of Tp, while Sg decreases to 1,000 psi from 10,000 with the fluid pressure decrease within pipelines, the loss of PTFE insertable gasket is not significant. Compared with the graphite spiral wound gasket, Tp decreases considerably. From another perspective, as the inner pressure stops functioning, the PTFE insertable gasket can maintain higher Tp than the graphite spiral wound gasket.

4. Establishing a Seal

According to the required gasket stress by pipe process, using the new gasket parameters, these gasket parameters are applied by the following equations **[9]**.

1. Within operating pressure range, minimum tightness parameter ($Tp_{(min)}$) expressed as follows：

   Under normal pressure (psi)：

   $$Tp_{(min)} = 0.1242 \times (C) \times (P_D) \tag{7}$$

   With different pressure unit：

   $$Tp_{(min)} = 1.8257 \times (C) \times (P/P^*) \tag{8}$$

2. Tightness ratio (T_r)

   $$T_r = \log(1.5 \times Tp_{(min)}) / \log(Tp_{(min)}) \tag{9}$$

3. Required theoretical seating stress (S_{ya}，psi)

   （During the initial operating state, it can get the Required tightness）

   $$S_{ya} = \frac{Gb}{e}(1.5 \times Tp_{(min)})^a \tag{10}$$

4. Minimum design stress (choice the bigger value, S_{m2}、S_{m1} or $2P_D$, psi)

   Seating stress component：

   $$S_{m2} = \frac{S_{ya}}{1.5} - P_D \frac{A_H}{A_G} \tag{11}$$

   Operating stress component：

$$S_{m1} = Gs\left(e \times \frac{S_{ya}}{Gs}\right)^{\frac{1}{T_r}} \tag{12}$$

5. Optimum bolt load (W_m，lb_f)

$$W_m = P_D \times A_H + S_m \times A_G \tag{13}$$

6. Actual bolt load (W_{mo}，lb_f)

(A factor of 0.85 is applied to modify the optimum bolt load for bolt torque loss during pipe operation in ASME proposed suggestion.)

$$W_{mo} = {W_m}/{0.85} \tag{14}$$

7. Required bolt torque (T，ft-lb_f /bolt)

(bolt friction factor is 0.15 for well-lubricated bolts, 0.25 for non-lubricated bolts. This equation uses a mean value of 0.2)

$$T = {[(\text{Force/bolt}) \times 0.2 \times D]}/{12} \tag{15}$$

Notation：

Gb	New ASME gasket factor, under Tp = 1, psi
Gs	New ASME gasket factor, remaining Tp = 1, psi
a	New ASME gasket factor, slope for gasket sealing curve
e	Method of bolts seating, for manual is 0.75 and for machine is 1.0
A_H	Hydrostatic area, $A_H = (3.14/4)\times(G)^2$, in^2
G	Mean diameter of gasket，G = (OD+ID)/2, in
A_G	Gasket area，$A_G = (3.14/4)(OD^2 - ID^2)$, in^2
D	Normal bolt diameter, in

5. Example

1. Gasket type and process conditions

(1) 1/16" graphite spiral wound gasket

(2) Flange：ASME/ANSI 150 class 16" pipe

OD = 20.25"
ID = 16.0"
G = 18.125"

(3) Bolts：20 bolts for grade 5, diameter = $1\frac{5}{8}''$ (Manual seat bolt, e = 0.75)

(4) Process design condition：

P_D = 2,000 psig = 2,014.7 psia

Air-Liquid phase：dryness (X) = 0.6,

$$d = 1 - \frac{1}{2}(0.6) = 0.7$$

(5) ASME seal factors：

Gb = 550 psi

Gs = 0.46 psi

a = 0.314

(6) Required leakage rate ：

Maintain T_3 seal

2. According to the following data, the optimum load stress on the gasket can be estimated as：

(1) During operating pressure, minimum tightness parameter ($Tp_{(min)}$)

$$Tp = \frac{P_D}{14.7} \times \left(\frac{1}{0.002 \times 150} \right)^{0.5} \times C = \frac{2{,}014.7}{14.7} \times \left(\frac{1}{0.002 \times 150} \right)^{0.7} \times 10 = 3{,}184$$

(2) Tightness ratio (T_r)

$$T_r = \log(1.5 \times Tp_{(min)}) \big/ \log(Tp_{(min)}) = \log(4{,}775) \big/ \log(3{,}184) = 1.05$$

(3) Required theoretical seating stress *(S_{ya} ，psi)*

$$S_{ya} = \frac{Gb}{e}(1.5 \times Tp_{(min)})^a = \frac{550}{0.75}(4{,}775)0.314 = 10{,}484\,(psi)$$

(4) Minimum design stress (choice the bigger value, S_{m2}、S_{m1} or $2P_D$,*psi*)

Seating stress component：

$$S_{m2} = \frac{S_{ya}}{1.5} - P_D \frac{A_H}{A_G} = \frac{10{,}484}{1.5} - 2{,}014.7 \left[\frac{\frac{\pi}{4} \times 18.125^2}{\frac{\pi}{4} \times (20.25^2 - 16^2)} \right]$$

= 6,989 - 2,014.7(257.9/120.9)

= 6,989 - 4,298= 2,691 (psi)

Operating stress component：

$$S_{m1} = G_s \left(e \times \frac{S_{ya}}{G_s} \right)^{\frac{1}{T_r}} = 0.46 \left(0.75 \times \frac{10{,}484}{0.46} \right)^{\frac{1}{1.05}}$$

= 4,943 (psi)

Double design pipe pressure：

$2P_D$ = 2014.7×2 = 4,029.4 (psi)

Because S_{m1} is the largest value between S_{m1}、S_{m2} and $2P_D$, using S_{m1} to estimate the optimum stress.

(5) Optimum bolt load *(W_m，lb_f)*

$$W_m = P_D \times A_H + S_m \times A_G$$
$$= (2{,}014.7) \times (257.9) + (4{,}943) \times (120.9)$$
$$= 1{,}117{,}224 \text{ (lbf)}$$

(6) Actual bolt load *(W_{mo}，lb_f)*

$$W_{mo} = {W_m}/{0.85} = 1{,}117{,}224/0.85 = 1{,}314{,}381 \text{ (lbf)}$$

(7) Tightness per bolt，(lb_f/bolt)

1,314,381 /20 = 65,719 (lb_f/bolt)

(8) Required bolt torque (*T*，*ft-lb_f /bolt)*

$$T = {[(\text{Force/bolt}) \times 0.2 \times D]}/{12} = [65{,}719 \times 0.2 \times 1.625]/12 = 1{,}800 \text{ (ft - lbf/bolt)}$$

3.Data analysis:

Under the design pressure of 2,000 psig，each bolt can provide a torque of about 1,800 ft-lb_f and create a compressive force of 1,117,224 lb_f. In addition, this force can be uniformly distributed on an area of 120.9 in^2 (780 cm^2), which can accordingly provide the initial stress of 9,241 psi (1,117,224 lb_f / 120.9 in^2) on the graphite spiral wound gasket.

After operating under internal pressure at 2,000 psig, the gasket stress will reduce to 4,943 psi，indicating T_3 seal can still be reached.

6. Summary

Evaluation methods of gasket leakage have not greatly developed over the past few decades. Process industries still widely adopt the previous design norm that is simpler, but not accurate enough to comply with more stringent regulations. However, such a situation raises inevitable concerns. With the rapid growth of modern technology, fluid conditions inside industrial pipelines have become more complicated. Historically, various leakages have been worldwide problems. Unfortunately, they were paid little attention until the awakening to the need for environmental protection and the increased worries about health and safety as well.

The control of pipeline leakage, in reality, includes complex variables. Adequate usage of gaskets and working in admittable operating surroundings can basically ensure inherent safety and get rid of worry with such approaches. Viewed from the standard of decreasing leakage rate, it is definitely a future trend for plant personnel to apply technical information and knowledge of tightness in measuring leaks. For the time being, the present skills of sealing technology and methodology to measure leakage cannot completely simulate the on –the-scene high pressure, due to the inner pressure of fluid and stress of gaskets. Thus, the measurement work of leakage rate inevitably has its failings and shortcomings.

ROTT, the technique of seal to measure leakage, involves implementing gaskets in the most strict environment of inner stress 400 psi、800 psi and gasket stress 1,000 ~10,000 psi, which are relatively super-high pressures in the process industries. Through repeated operations, the lowest seal properties and the leakage rate that it can retain are thus proved. Therefore, the testing method fits considerably the control of toxic chemical leakage inside manufacturing pipelines. In fact, data from the techniques of measuring leakage, along with the stress in processing procedure and tightness degrees of gaskets, can be used to demand various leakage rates so as to effectively control the leaking of toxic chemical materials.

The application of design and testing norms of newly developed gaskets has progressed and matured to a large degree. In practice, this study is one type testing method on gaskets. One point that designers must realize is that, during the evaluation process of actual operation, one single testing alone cannot determine the seal properties of the gaskets. Instead, different measurement methods of seal properties are executed for crisscross analysis, contrast and verification. As a result, for the new generation of testing norms in gaskets, this method validates a product more scientifically and theoretically, unlike those old gaskets of which the seal properties are determined merely by “m” and “y” values.

7. References

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Table 1. Seal classification and leakage rate [8][10]

Tightness classes		
Seal classification	Leakage rate (mg/sec/mm)	Constant C
T_1	2×10^{-1}	0.1
T_2	2×10^{-3}	1
T_3	2×10^{-5}	10
T_4	2×10^{-7}	100
T_5	2×10^{-9}	1,000

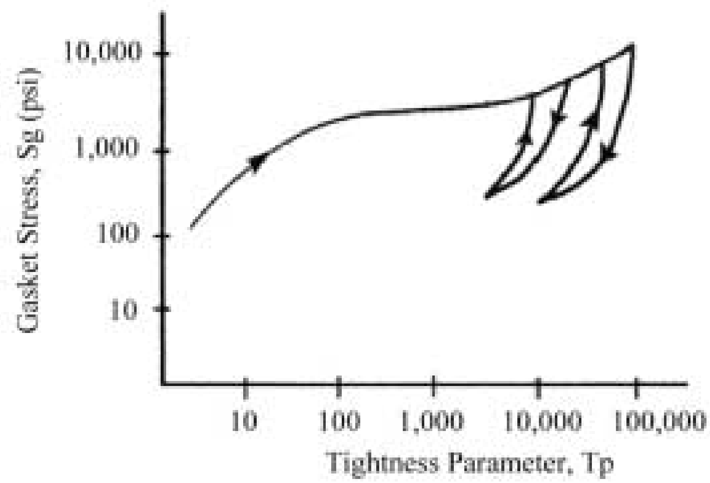

Figure 1. ROTT Test: Sg and Tp [8]

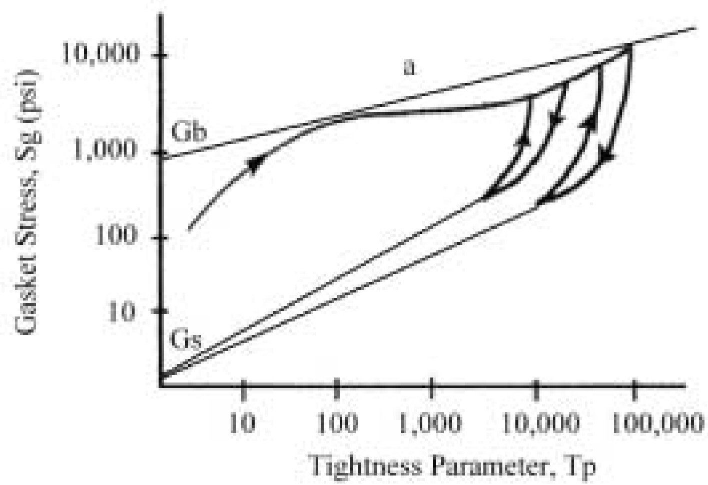

Figure 2. ROTT Test: the seal factor [8]

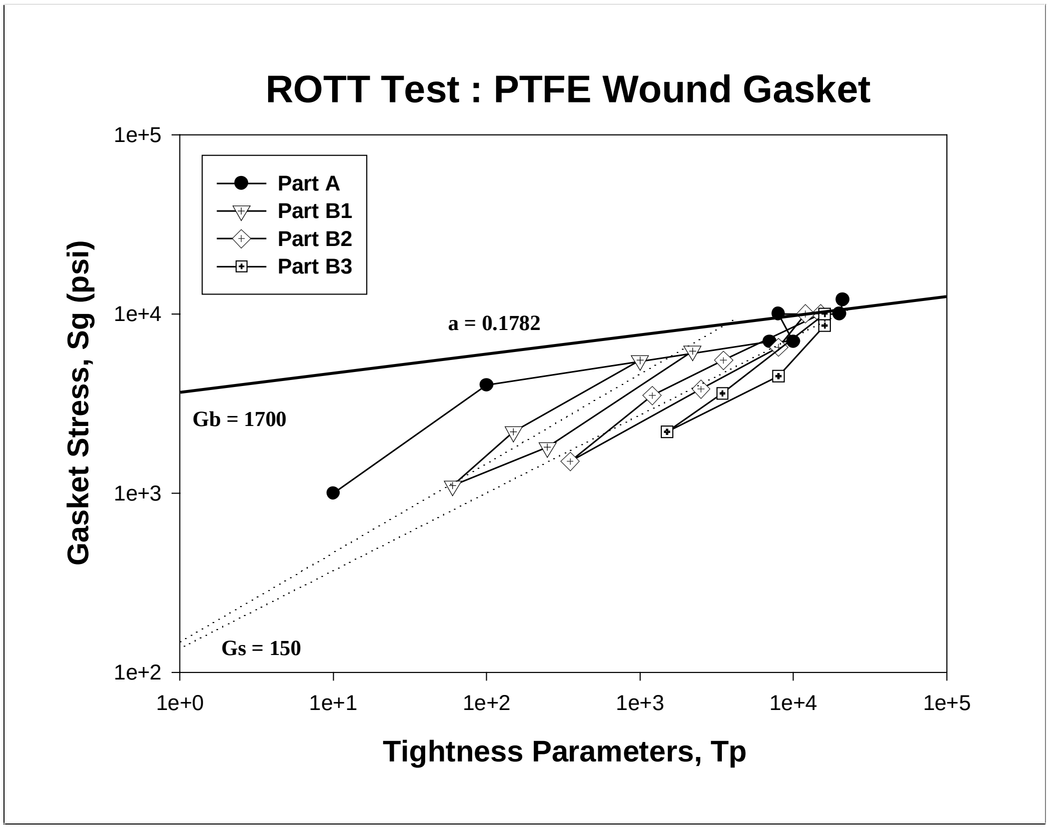

Figure 3. ROTT Test by PTFE Wound Gasket.

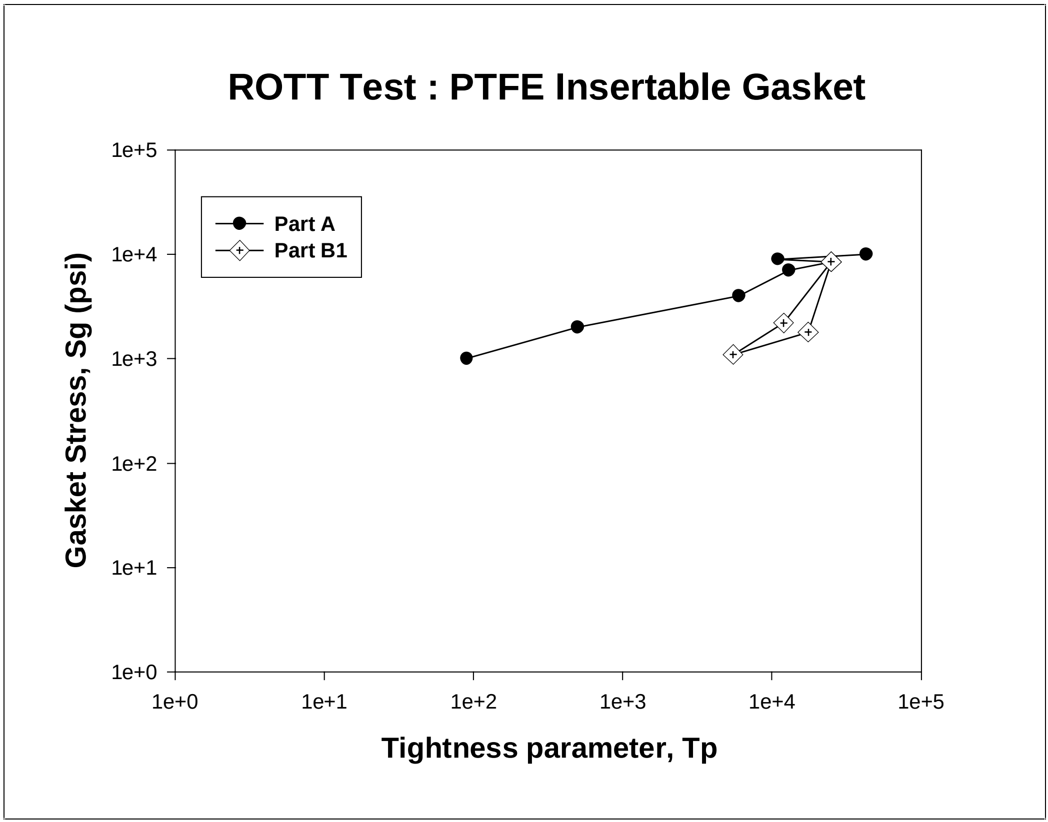

Figure 4. ROTT Test by PTFE Insert able Gasket.

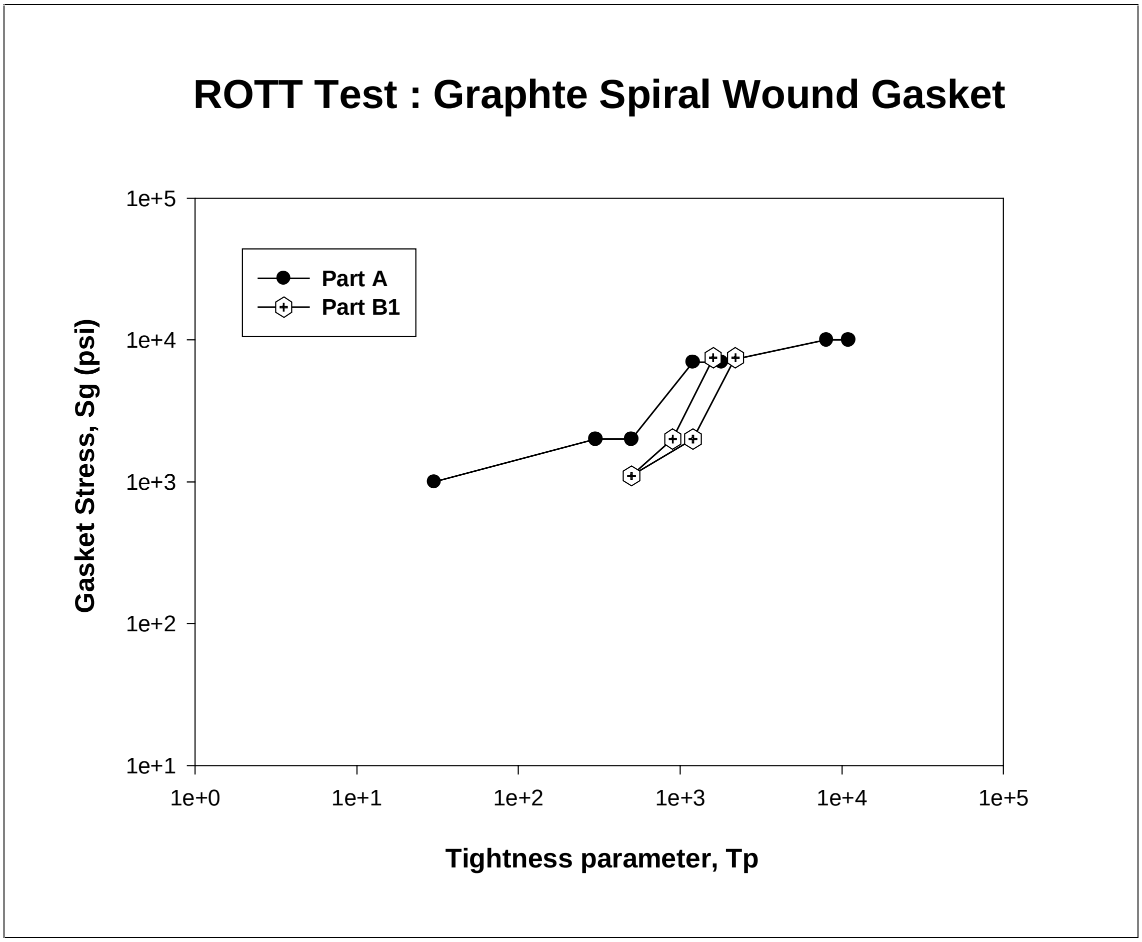

Figure 5. ROTT Test by Graphite Spiral Wound Gasket.