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UPDATE OF THE TABULATED « M AND Y » VALUES IN THE NEW REVISIONS OF FRENCH CODES CODAP® AND CODETI® - DEVELOPMENT OF A TESTING PROCEDURE TO DETERMINE « M AND Y » VALUES FOR SEVERAL TIGHTNESS CLASSES

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ABSTRACT

The existing tables giving "m and y" values, used in the Taylor Forge method for bolted flange connections calculation, have remained unchanged through years. Some gasket types do not appear in these tables and no reference is made to a Tightness Class associated to these values. The need for an update of the exiting tables has been raised by the supplier of French codes (CODAP® [1] and CODETI® [2]).

A survey about the recommended values of "m and y" and their associated expected leakage rates for the gasket types available on the market has been performed. The wide discrepancy in the test procedures and the gasket parameter values showed the need for the development of a common test procedure.

The new test procedure giving tables of "m and y" values depending on the tightness class is presented here. The application of this procedure on several gasket types, lead to the publication of new tables for "m and y" values, in the last French codes revision.

NOMENCLATURE

A	Parameter of the developed model (equation (3))
A_1	Gasket area [mm²]p
A_2	Inside area of gasket [mm²]
A_m	The total required cross sectional area of bolts [mm²]
a	Exponent of gasket assembly-loading curve
B	Parameter of the developed model (equation (3))
b	Effective gasket width [mm]
C	Parameter of the developed model (equation (3))
G	Diameter of gasket load reaction [mm]
G_b	Gasket property used to describe the assembly-loading curve [MPa]
G_s	Gasket property used to describe the unloading curve [MPa]
K_{12}	Ratio of the total required cross sectional area of bolts in design condition to bolt-up condition (equation (7)).
m	Maintenance factor
mI	Value of m at intercept of a specific leakage rate value corresponding to a tightness class (Figure 7)
n	Parameter of the developed model (equation (3))
P	Internal pressure [MPa]
$Q_{min(L)}$	Minimum level of surface pressure required for leakage rate class L on assembly [MPa]
$Qs_{min(L)}$	Minimum level of surface pressure required for leakage rate class L after off-loading [MPa]
S_a	Allowable bolt stress at ambient temperature [MPa]
S_b	Allowable bolt stress at design temperature [MPa]
SI	Value of initial gasket stresse for step I of the procedure (Figure 1) [MPa]
W	Total fastener force [N]
W_{m1}	Minimum bolt load for design condition [N]
W_{m2}	Minimum bolt load for bolt-up condition [N]
y	Yield factor [MPa]

INTRODUCTION

The method for the design of bolted flange joints needs gasket coefficients as inputs of the calculation. Tables of gasket coefficients exists for the different calculation method as “m” and “y” according to Taylor Forge based methods, G_b,G_s and a for the ASME BFJ new appendix or Q_{min} and Q_{smin} for the European calculation method EN1591 [3]. Whereas the tabulated values for the two last methods have been obtained by performing tests according specified existing procedures (ROTT [4] for G_b,G_s and a or EN13555 [5] for $Q_{min,(L)}$, $Q_{smin,(L)}$), the origin of the tabulated “m” and “y” is less precisely identified and no official valid test procedure is currently defined to generate these values.

The need for an update of the exiting “m” and “y” tables has been raised by the SNCT (Syndicat National de la Chaudronnerie, de la Tuyauterie et de la maintenance industrielle), a French association, supplier of the construction codes CODAP® [1] and CODETI® [2]. A lot of codes still use a Taylor Forge based method for their flange assembly calculation section. Even the European standard for pressure vessels EN13445 [6] is still referring to a Taylor Forge based method in the body of the documents whereas EN1591 [3] is only introduced as an alternative method in a specific annex. The initial request was concerning the integration of new gasket types. Moreover, the interest for linking these updated values to tightness classes has also been emphasized during the project development.

In a first step, a survey about the recommended values of “m and y” and their associated expected leakage rates, for the gasket types available on the market, has been performed. The wide discrepancy in the test procedures and the gasket parameter values showed the need for the development of a common test procedure.

The new test procedure giving tables of “m and y” values depending on the tightness class has been applied on several gasket types. The updated tables have been published in the last revisions of [1] and [2].

DEFINITON OF “M” &”Y”

The standard for the determination of the “m” and “y” values is ASTM F586 [7]. This standard initially published in 1979 and re-issued in 1989 has been withdrawn in 1998 with no replacement. This document defines “y” and “m” as follows:

$$y = W / A_1 \qquad (1)$$

$$m = (W - A_2 * P)/(A_1 * P) \qquad (2)$$

With:

- W: Total fastener force [N]
- A_1: Gasket area [mm²]
- A_2: Inside area of gasket [mm²]
- P: Internal pressure [MPa]

The equation (1) gives the value of “y” by dividing the force value applied on the gasket by its compressed area. Equation (2) gives the value of “m” by dividing the force applied on the gasket reduced with the end thrust force, by the compressed area of the gasket and the internal pressure.

SURVEY WITH GASKET MANUFACTURERS

In 2003, CETIM has first conducted a survey with European gasket suppliers (27 firms) for recording the “m” and “y” values and test procedures used on a day-to-day basis.

An analysis of the answers received by 20 gasket suppliers has been conducted using several criteria as the gasket type, the gasket thickness, the test procedure reference and/or condition (pressure, temperature, fluid,…). Several levels of details have been taken into account to issue the gasket type classification as the presence and the shape of metallic insert, the material of the filler and/or of the metallic part, etc…

The analysis has shown a huge lack of homogeneity in the procedures used to determine the gasket coefficients. For example, the procedure parameters has revealed the use of liquid (water) and gases (nitrogen, helium,…), a wide variation in the testing pressures (less than 1 bar to 80 bar) and in the leak rate measurements method (bubble detection, pressure decay, Helium mass spectrometry,…). In some cases the procedures and conditions were not clearly defined or the given values were directly extracted from the existing tables.

The lack of homogeneity in the test procedures involves a wide spread in the “m” and “y” values within a given gasket type, even after applying a correction of the measured values in order to take into account the different test conditions.

Due to the withdrawal of [7], there is a lack of existing valid procedure for the determination of “m” and “y” values. This leads to the impossibility to issue a revision of gasket coefficients based on the results of the survey. Moreover, this absence makes the comparison between several gaskets very difficult or practically impossible for the gasket user. On the basis of this analysis, developing a method for evaluating these coefficients seemed necessary.

The new test procedure proposed by CETIM (CEntre Technique des Industries Mécaniques – Technical CEnter for Mechanical Industry) aims at filling the lack of valid procedure for the determination of "m" and "y" values. The objective is also to connect the obtained "m" and "y" values to tightness classes.

DEFINITION OF A NEW TEST PROCEDURE

Test procedure principle

The proposed procedure consists in loading a gasket at various tightening levels and having it undergo different internal fluid pressure levels. The leak is measured at each stress and pressure level. The procedure integrates gasket compression / decompression cycles so as to take account of both the seating and operating conditions. The fluid used is helium, which is the reference gas for in-lab sealing tests.

Test sequence

The test procedure involves helium leak rate measurement for various gas pressures (P) and gasket stresses (S). As shown in the diagram below (Figure 1), the gasket stress sequence enables to study the gasket sealing behaviour at loading (gasket load increase) and unloading (gasket load decrease) for several internal gas pressure values.

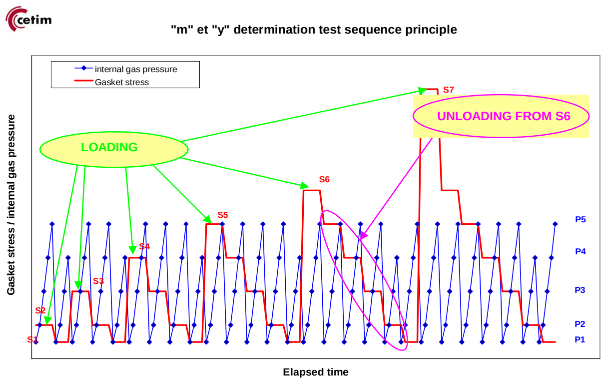

Figure 1 : Test procedure principle

Test rig

The tests can be performed using the ROTT [4] test rig or any other compression press. The gasket leak rate is measured using a flow meter, the pressure decay or the helium mass spectrometry method depending on the leak rate value.

Figure 2: ROTT test rig

Data analysis

Applying the test sequence described in (Figure 1) enables to get a leak rate value for each of the unloading steps associated to the considered initial load (S1, S2, S3,…).and for each tested internal pressure value as shown in red in Figure 3. The leak rate values are given in mass unit normalized by the external diameter of the gasket expressed in mm (i.e. mg/s/mm).

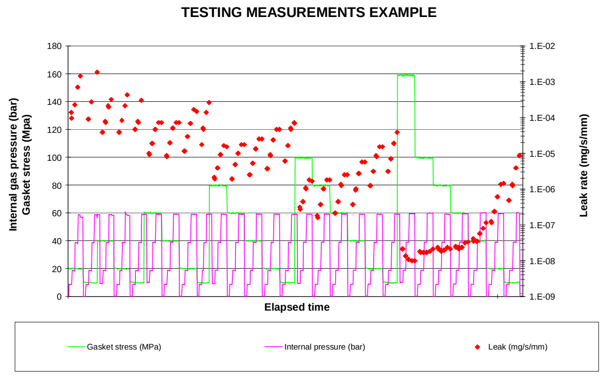

Figure 3: Test measurements example

From the data measured during the test (Figure 3), the value for the ratio of the gasket stress / internal fluid pressure is plotted versus leak rate in a Log-Log format (as shown in Figure 4).The measurements performed during unloading are plotted in different colours depending on their initial gasket load value (S1, S2,…).

As defined in equation (1), the value of “y” is corresponding to the initial gasket stress i.e. to the values refrenced S1,S2,…S6. The equation (2), defines “m” as the ratio of the gasket stress ([tightening force – end thrust force]/gasket surface) divided by the internal fluid pressure. So the graph shown in Figure 4 can be interpreted as the variation of “m” versus leak rate for several initial gasket stresses (S1 to S6).

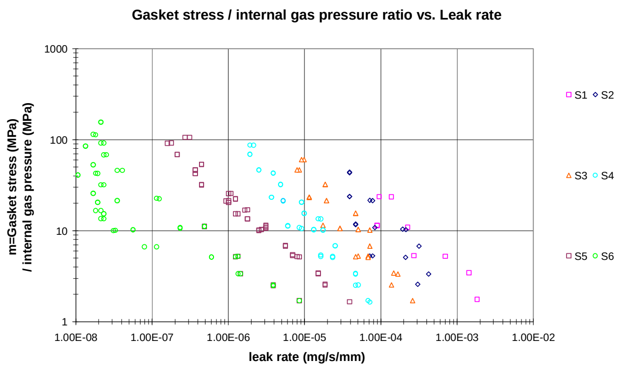

Figure 4: Gasket stress/internal pressure (« m ») vs leak rate

For the next step of the data analysis, a model for the variation of “m” versus the leak rate is determined for each intial gasket stress (S1 to S6) as shown in Figure 5. This model enables to know the “m” value associated to a given leak rate for each initial gasket stress level tested. The selected model here involves the following relation between the leak rate and the “m” value, but other model forms could be investigated if necessary at this step:

$$LOG_{10}(L) = A + B*(LOG_{10}(m) + C)^{n} \quad (3)$$

The values of A, B, C and n that enable the best fitting are selected for each initial gasket stress. For the determination of these parameters, a penalty on the error between the measured values and the model can be applied to the data points where the modelled leak rate is lower than the measured one, in order to be more conservative (as shown on the green curve for S6 in the Figure 5.

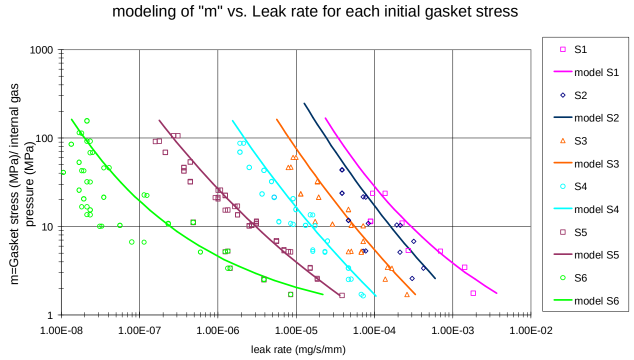

Figure 5 : Modelling of “m” versus leak rate dependence for each initial gasket stress.

Using the modelling explained above, the values of “m” for the specified leakage rate corresponding to the Tightness Classes defined in Figure 6 are determined as shown in Figure 7. This enables to generate a raw version of the “y” and “m” table depending on the leakage rate for each initial gasket stress level as shown in Figure 8.

Tightness Type	Tightness Class	Leak rate (mg/s/mm)
ECONOMY	T1	2 E-01
	T1.5	2 E-02
STANDARD	T2	2 E-03
	T2.5	2 E-04
HIGH	T3	2 E-05
	T3.5	2 E-06
EXCEPTIONNAL	T4	2 E-07
	T4.5	2 E-08

Figure 6: Tightness Classes definition

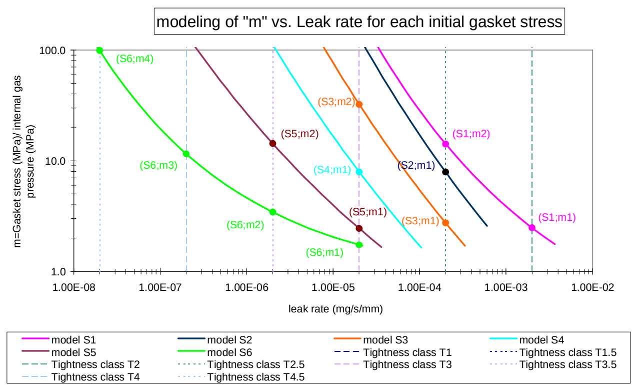

Figure 7: Determination of the “m” values versus Tightness Class for each initial gasket stress

TIGHTNESS TYPE		ECONOMY		STANDARD		HIGH		EXCEPTIONNAL	
TIGHTNESS CLASS		T1	T1,5	T2	T2,5	T3	T3,5	T4	T4,5
Leak rate (mg/s/mm)		2.00E-01	2.00E-02	2.00E-03	2.00E-04	2.00E-05	2.00E-06	2.00E-07	2.00E-08
Initial gasket stress level	S1			m1	m2				
	S2				m1				
	S3				m1	m2			
	S4					m1			
	S5					m1	m2		
	S6					m1	m2	m3	m4

The Tightness Class can not be reached with the defined value of initial gasket stress

Figure 8: « m » & « y » table model

USE OF THE NEW “M” & “Y” TABLES

New tables explanation

On the basis of the raw version of table (Figure 8), a more user oriented version is issued as shown in Figure 9. This new table gives several (y;m) pairs enabling to fulfil the tightness criteria of the associated Tightness Class. It assumes that a pair valid for a tightness class is also valid for all the lower tightness classes.

A colour code is also added in order to help the user to choose between the available pairs.

The green cell indicates that the value of “y” is within the typical range of initial stress for this gasket type and that the value of “m” is not too high.

The yellow cells indicates that either the initial gasket stress is in the upper level area for the considered the gasket or that the high value of m will tend to generate a high value of bolt force for the design condition when performing the calculation.

The orange cells indicates that these data must be used with special care to perform a calculation for a field application where the parameters would be less controlled than during the test in laboratory. They generally indicate that either the initial gasket stress is higher than the typical values or that the induced “m” value is very high. The orange cells suggest that the chosen gasket type may not be the best choice regarding the sealing performance requirements and that another gasket type should be investigated to optimize the design.

The grey cells indicate that there is no available data.

<table>
<tr><td>TIGHTNESS TYPE</td><td colspan="2">ECONOMY</td><td colspan="2">STANDARD</td><td colspan="2">HIGH</td><td colspan="2">EXCEPTIONNAL</td></tr>
<tr><td>TIGHTNESS CLASS</td><td>T1</td><td>T1,5</td><td>T2</td><td>T2,5</td><td>T3</td><td>T3,5</td><td>T4</td><td>T4,5</td></tr>
<tr><td>Leak rate (mg/s/mm)</td><td>2.00E-01</td><td>2.00E-02</td><td>2.00E-03</td><td>2.00E-04</td><td>2.00E-05</td><td>2.00E-06</td><td>2.00E-07</td><td>2.00E-08</td></tr>
<tr><td rowspan="6">GASKET TYPE XX</td><td colspan="3">(S1;m1)</td><td>(S1;m2)</td><td rowspan="3">(S3;32.3)</td><td rowspan="5">(S5;14.3)</td><td rowspan="6">(S6;11.5)</td><td rowspan="6">(S6;98.9)</td></tr>
<tr><td colspan="4">(S2;m1)</td></tr>
<tr><td colspan="4">(S3;m1)</td></tr>
<tr><td colspan="5">(S4;7.9)</td></tr>
<tr><td colspan="5">(S5;2.4)</td></tr>
<tr><td colspan="5">(S6;1.7)</td><td>(S6;3.4)</td></tr>
</table>

Recommended value
Possible value
Not recommended value
No available value

Figure 9: "m" and "y" table for the user

Optimizing the choice of (y;m) pairs

As shown in Figure 9, several pairs of (y;m) are available for a given Tightness Class. For a given Tightness Class, the greater the "y" value, the lower the "m" value. In the calculation, the value of "y" is governing the bolt load in assembly condition (W_{m2}) whereas the value of "m" is governing the bolt load in design condition (W_{m1}) as shown in equations (4 & 5). The greater the value of $W_{m2,}$ is the lower the value of W_{m1} is.

The total required cross sectional area of bolts (A_m) is depending on the values of the minimum bolt load for design and bolt-up conditions (6). The aim of the optimization is to get the lowest value of A_m enabling to fulfil the tightness criteria. A_m being defined as the maximum of two parameters varying in an opposite way, its minimum value will be reached when the values of the two parameters are the closest. In order to quantify the proximity of these two parameters, the variable K_{12} is defined as the ratio of the first parameter to the second (see equation (7)). Then the optimal (y;m) pair will be defined as the pair leading to a value of K_{12} as close as possible to 1.

So the choice of the (y;m) pair can be optimized depending on which of the bolt-up or the design condition is the most critical. This gives a flexibility on the calculation and can enable to reduce the bolt area or make an existing assembly fulfil the criteria by choosing the best (y;m) pair.

$$W_{m2} = b * \pi * G * y \quad (4)$$

$$W_{m1} = \frac{\pi * G^2 * P}{4} + 2 * b * \pi * G * m * P \quad (5)$$

$$A_m = MAX\left(\frac{W_{m1}}{S_b};\frac{W_{m2}}{S_a}\right) \quad (6)$$

$$K_{12} = \frac{W_{m1}}{S_b} \Big/ \frac{W_{m2}}{S_a} \quad (7)$$

EXAMPLES AND APPLICATIONS

The table below (Figure 10) gives result examples for specific tests performed on a fibre gasket and a flexible graphite filled spiral wound gasket references.

The whole program has involved the major generic flat gasket types available on the market (

- Modified PTFE (2 and 3 mm thick)
- Fibre based (2 and 3mm)
- Flexible graphite (with metal inserts)
- Spiral wound gaskets (flexible graphite and PTFE filler)
- Kammprofile
- Metal-jacketed (covered or not)

The obtained tables values can be found in [1] and [2]. A practical application of the new table values has been performed by using them on selected industrial cases calculations. The result of these calculations has revealed lower mechanical stresses in the bolted flange connections for 13 cases on the 16 investigated by using the new tables. A major reason for that is, the possibly of choosing between several (y;m) pairs for a given leak rate that enables to optimize the design.

<table>
<tr><td></td><td colspan="8">TABLE FOR (y[MPa];m)</td></tr>
<tr><td>TIGHTNESS TYPE</td><td colspan="2">ECONOMY</td><td colspan="2">STANDARD</td><td colspan="2">HIGH</td><td colspan="2">EXCEPTIONNAL</td></tr>
<tr><td>TIGHTNESS CLASS</td><td>T1</td><td>T1,5</td><td>T2</td><td>T2,5</td><td>T3</td><td>T3,5</td><td>T4</td><td>T4,5</td></tr>
<tr><td>Leak rate (mg/s/mm)</td><td>2.00E-01</td><td>2.00E-02</td><td>2.00E-03</td><td>2.00E-04</td><td>2.00E-05</td><td>2.00E-06</td><td>2.00E-07</td><td>2.00E-08</td></tr>
<tr><td rowspan="6">FIBRE</td><td colspan="3" rowspan="6">(20;2.5)</td><td>(20;14.2)</td><td rowspan="3">(60;32.3)</td><td rowspan="5">(100;14.3)</td><td rowspan="6">(160;11.5)</td><td rowspan="6">(160;98.9)</td></tr>
<tr><td>(40;7.9)</td></tr>
<tr><td rowspan="4">(60;2.7)</td></tr>
<tr><td>(80;7.9)</td></tr>
<tr><td>(100;2.4)</td></tr>
<tr><td>(160;1.7)</td><td>(160;3.4)</td></tr>
<tr><td rowspan="3">GRAPHITE FILLED SPIRAL WOUND GASKET</td><td colspan="5">(20;8)</td><td>(80;7)</td><td rowspan="3">(120;8)</td><td></td></tr>
<tr><td colspan="5">(40;4)</td><td rowspan="2">(160;4)</td><td></td></tr>
<tr><td colspan="5">(60;2.5)</td><td></td></tr>
</table>

(y;m): y value in (MPa)

Recommended value
Possible value
Not recommended value
No available value

Figure 10: Result table example

CONCLUSION

Due to the withdrawal of [7], there is currently a lack of existing valid procedure for the determination of "m" and "y" values. This results in a huge heterogeneity in the used procedures and the associated "m" and "y" values. Moreover, the comparison between several gaskets is very difficult or practically impossible for the gasket user

CETIM has developed a new test procedure enabling to link the values of "m & y" to Tightness Classes. This procedure has been applied on the major types of flat gasket to update the exiting "m & y" tables in the last revisions of CODAP® [1] and CODETI® [2].

Moreover, the proposed procedure is offering the possibility of choosing between several (y;m) pairs of values for a required Tightness Class. It is to be noted here that for a given Tightness Class, the greater the value of "y", the lower the value of "m". This choice between several (y;m) enables to optimize the bolted flange assembly calculation.

REFERENCES

[1]: SNCT Publications, 2005, CODAP® 2005 (last revision 09/07): Code for construction of unfired pressure vessels

[2]: SNCT Publications, 2006, CODAP® 2006 (last revision 09/08): Code for construction of industrial piping

[3]: CEN TC 74, 2001, NF EN1591-1: Flanges and their joints - Design rules for gasketed circular flange connections - Part 1: Calculation method

[4]: PVRC (Pressure Vessel Research Council), ROTT: ROom Temperature Tightness test

[5]: CEN TC 74, 2005, NF EN13555: Flanges and their joints Gasket parameters and test procedures relevant to the design rules for gasketed circular flange connections

[6]: CEN TC 54, NF EN13445-3: Unfired Pressure Vessels - Part3:Design

[7]: ASTM, (initial version 1979, last revision 1989, withdrawn with no replacement in 1998), ASTM F586: Test method for leak rates versus y stresses and m factors for gaskets.