Gasket factors derived from EN 13555 for use with EN 1591

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Calculation standard EN 1591-1:2013 applied in flange connections designing is currently the most advanced algorithm describing functioning of such joints. Unlike algorithms using the Taylor-Forge method ASME Code or method: DIN 2505 and AD 2000 Merkblatt, it takes into consideration elastic deformation of the joint and elastic and plastic deformations of the seal. Also, limited plastic deformation of the flange and/or the bolts are acceptable. Thus, it allows flange connection functioning in numerous pressurised conditions, under temperature and in assembly state, to be thoroughly analysed. However, the most important innovation in the approach to flange connections design and calculation is that the tightness class was considered as input data for calculation. **Such an approach guarantees that after calculation of necessary bolts load in each of the designed states the joint shall not demonstrate emissions higher than those assumed in calculations.**

The most important output data from the connection assembly point of view is the gasket compression force and the bolt tightening torque. Based on this information the fitter bolts the joint together. Calculations are based on coefficients characterising the gasket in terms of its mechanical, rheological and tightness properties. The values of those coefficients are determined on test stands in accordance with the methodologies specified in EN 13555:2014. The test stand is constructed based on a diagram presented therein. Since the number of calculation coefficients is significantly higher than in the case of previously applied algorithms, that often presents a challenge for designers, their meanings are generally explained below.

Q_{smax} [MPa] – The maximum surface pressure that can be safely imposed upon the gasket at the service temperature without damage or intrusion into the bore. Values are determined both in ambient and elevated temperatures. In calculations it appears in formulas determining the gaskets effective width and specifies the gaskets maximal permissible load.

E_G [MPa] – The unloading secant modulus of elasticity of the gasket. Value describing elasticity of the gasket or, in other words, it is a degree of recovery change vs. stress. Value determined in the joint test with **Q_{smax}** at ambient and elevated temperatures.

Values of both coefficients are determined during one test for a given assumed temperature, however, not higher than the maximum operating temperature declared by the seal manufacturer. The testing process simulates operation of the flange connection in loading/unloading cycles as shown in Diagram 1.

The gasket is heated to the test temperature and then loading and unloading cycles begin. Maximum permissible stress on the gasket is the stress value preceding the gasket damage. The answer to the question as how to define the moment in which the gasket is damaged constitutes a separate and difficult issue and shall not be discussed here. The testing procedure is described in EN 13555, Figure 2. Diagram 1 also shows the change of the gaskets thickness at the different stress levels.

P_{QR} [1] – The creep relaxation factor, the ratio of the residual and initial surface pressures. The factor is determined both in ambient and elevated temperatures. It determines numerically the ratio of the stress remaining on the gasket after the test is finished to the value of stress applied at its start.

It is always: $P_{QR} \leq 1$. The closer this value is to one, the weaker the phenomenon of the gasket relaxation and creep is. Value = 1 means that the gasket does not relax and creep. Certainly, there is a correlation that the higher the temperature, the lower **P_{QR}** value. In the current edition of the calculation standard the **P_{QR}** value helps calculate the following values:

Δ_{eGc} [mm] – The change in gasket or sealing element thickness due to creep. Values calculated based on the previously determined **P_{QR}** values. Thus, the values depend on temperature, but also on the starting stress used in the **P_{QR}** test. The Δ_{eGc} values can also be measured on the test rig and must not be calculated.

Tightness/leakage -related factors:

Q_A [MPa] – The gasket surface pressure at assembly prior to unloading,

$Q_{min(L)}$ [MPa] – The minimum level of surface pressure required to obtain the tightness class on assembly,

$Q_{smin(L)}$ [MPa] – The minimum level of surface pressure required to retain the tightness class after off-loading.

The meanings of the three values above are best explained in Diagram 2 taken from a test performed with 10 bar pressure of helium. The horizontal axis represents stress on the gasket [MPa], the vertical axis represents leakage level [mg/(m*s)] (also known as tightness classes). The test consists of cyclical loading/unloading of the gasket. The blue

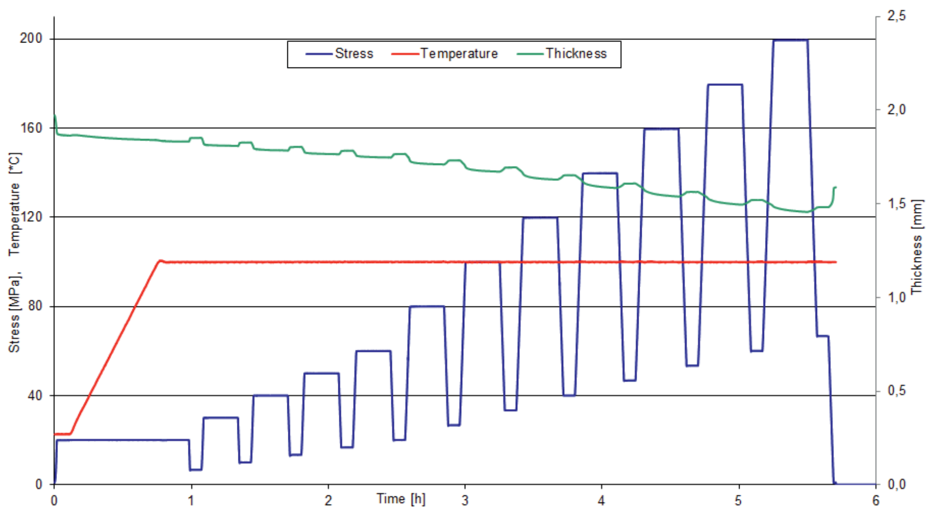

Diagram 1

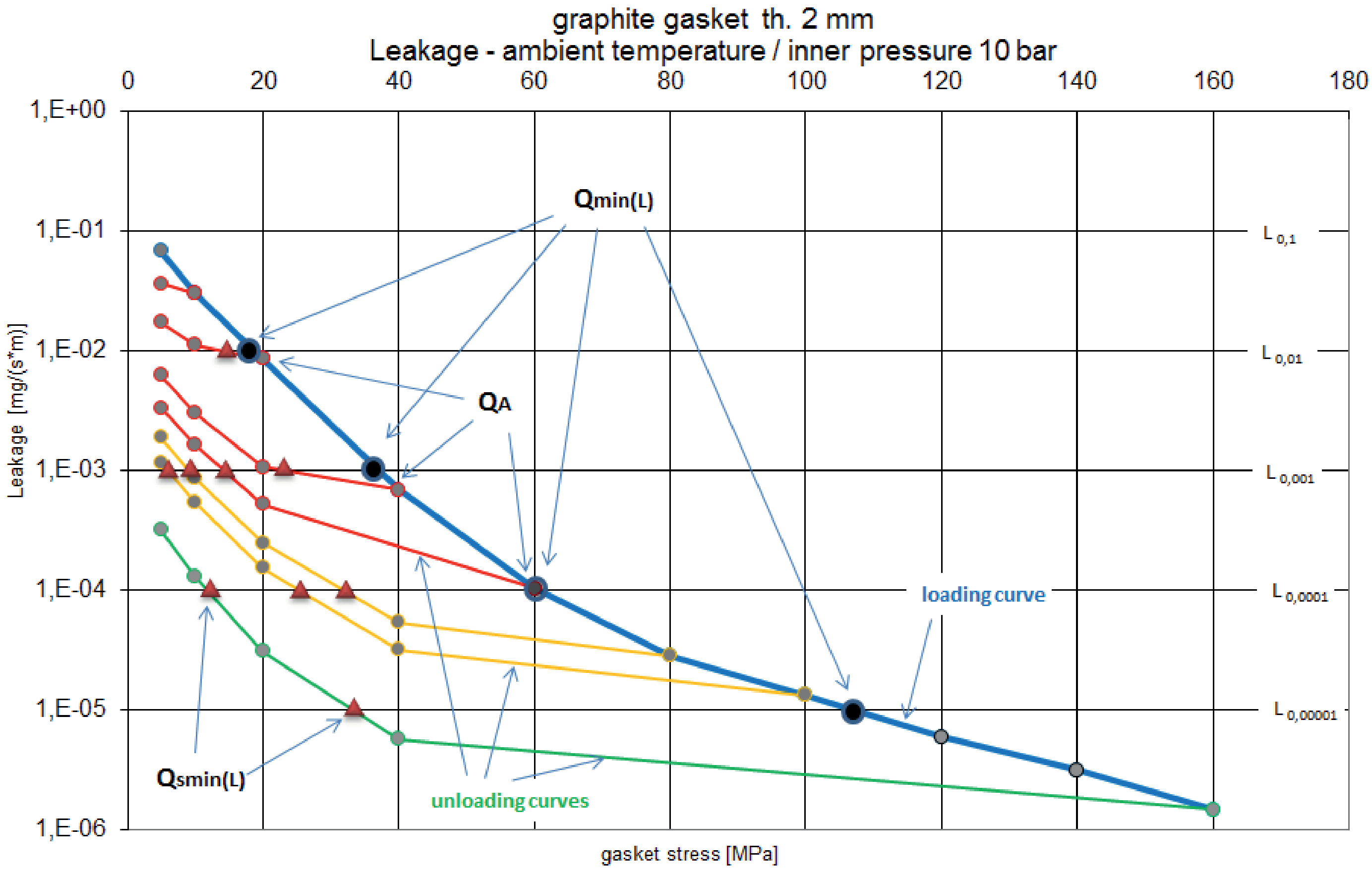

Diagram 2

line illustrates how the loading proceeded, whilst lines in other colours show unloading after points Q_A were reached. So, for example, after point $Q_A = 40$ MPa was reached during loading, the system was unloaded, first to 20MPa, then to 10 and 5 MPa. Such a procedure simulates operation of a real flange joint subjected to various operating conditions, that is, loading and unloading from changes of pressure, temperature, external forces and/ or moments.

Let us assume that during operation, L0.001 [mg/(m*s)] (1e^{-03}) tightness class needs to be maintained. Let us also assume that due to pressure, thermal expansion and external forces the stress acting on the gasket drops to 5MPa. When looking at the diagram we see that only unloading the joint from 160MPa (green line) guarantees the leakage class is maintained under the assumed conditions. Even at the final unloading of 5MPa we are still below the L0.001 mg/m/s class. However, the question arises, whether bolts and flanges can withstand such a high initial stress. The answer can be obviously found by calculations, and if the bolting / flange strength conditions are not met, the initial $\mathbf{Q_A}$ value needs to be reduced and the calculations carried out again. However, it can occur that after reducing the $\mathbf{Q_A}$ value the tightness class condition is not maintained. Then either another gasket needs to be used or new flange/bolting materials are required and the joint needs to be recalculated.

$\boldsymbol{\mu_G}$ – the static friction factor between the gasket and the flange facing. Another value introduced into calculations to ensure that the gasket is not blown out due to the effect of external forces and moments, whose impact can significantly decrease the applied stress on the gasket; even locally.

Knowledge of the axial coefficient of thermal expansion of a gasket material α_G is also necessary for full calculations.

The standard uses several calculation coefficients, used together this amounts to tens, and even hundreds of final values. Conducting full calculations manually with the EN 1591-1 method is practically impossible, but generally available spreadsheet and/or mathematical programs can make this easier. Commercially available programs can make a designers life much easier, yet the complexity of the standard is a difficult learning curve to overcome. Moreover, the designer needs to have access to all the gaskets coefficients. EN 1591-2, which contains factors for calculation, is not fully compatible with calculation standard. The requirement to provide those coefficient values is left to the gasket manufacturer. EN 1591-2 will be updated and re-issued (as a Technical Specification) in the future. The coefficients can be mainly found on www.gasketdata.org or directly on leading manufacturer's websites but not always.

This analysis is only a brief introduction to a complex world of the EN 13555 and EN 1591-1. Those interested in learning more about this subject should contact their regular supplier or another member of the Flange Gasket Division of the ESA, details can be found on www.europeansealing.com.

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