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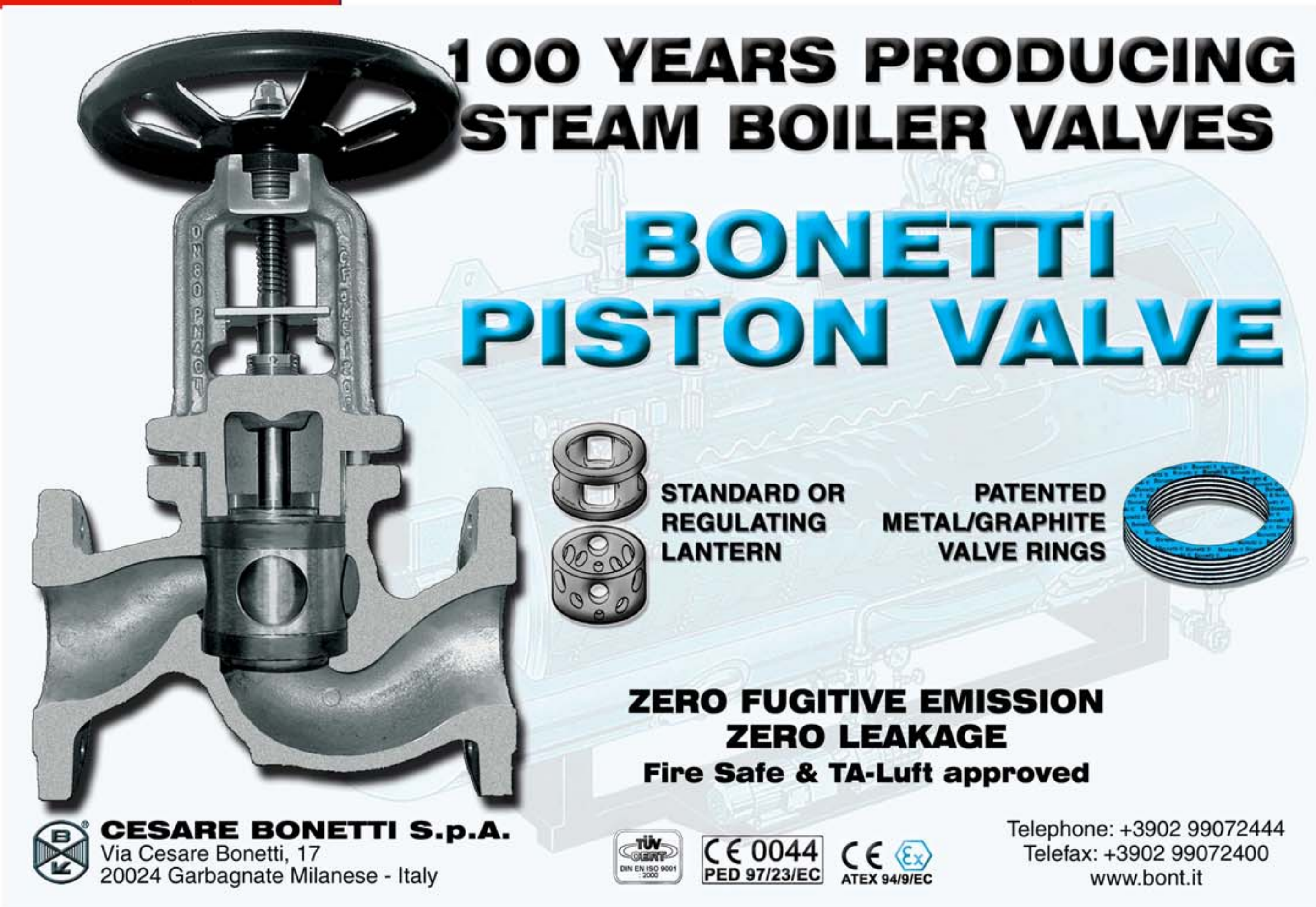

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Severe service ball valves

In this paper, the author discusses potential weaknesses in the ball valve design when used under severe services. As a design improvement he presents a gasket structure based on alternate layers of stainless steel and expanded (then elastic compressed) graphite. The fluid pressure/velocity flows are given a theoretical foundation.

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Nowadays, valve manufacturing is based on tested theories and wide circulated experimental data. However, as per every primary process device, the required function sometimes overcomes the status of the art and therefore difficult problems may be encountered. We are positive that the most frequent and troublesome problem is due to an obvious occurrence, that is the fact that a valve does operate either closed or open. When closed after having been open there is line leakage, whereas having to open after a period of closure there is an external leakage. This occurs, of course, under rather severe operating conditions. Although such conditions are in the minority several types of applications are affected, namely high differential pressures, high or low temperatures and difficult to handle fluids.

Wide range of applications

In terms of flow capacity and ease of operation ball valves are ideal for a wide range of applications. In addition there are several other uses, admissible as far as the theoretical functionality is concerned, but which are currently forbidden due to lack of safety in operation. For example, consider pressures which, although within the valve rating, could cause the erosion of seat in case of operation under high differential pressure on opening the valve and/or such temperatures which could deteriorate the seating in specific operating conditions. Were a ball valve able to meet these operating conditions, therefore a fortiori fit for the remaining applications of the installation, one could reduce the number of different types of devices, and take advantage in the construction and handling of installation. However, in certain severe processes, "valve men" look suspiciously at the idea of stopping a threatening flow or giving free rein to a dangerous fluid just by rotating a stem through ninety degrees. Instead, they prefer the several turns of a handwheel or a motor. The perceptions that in a ball valve the manoeuvring and the modification of stress/strain distribution directly affect the gaskets of the sealing boundary, that opening and closing the valve can cause seat erosion and finally that no clasping, squeezing or turning a lever will do: a ball valve either closes or leaks.

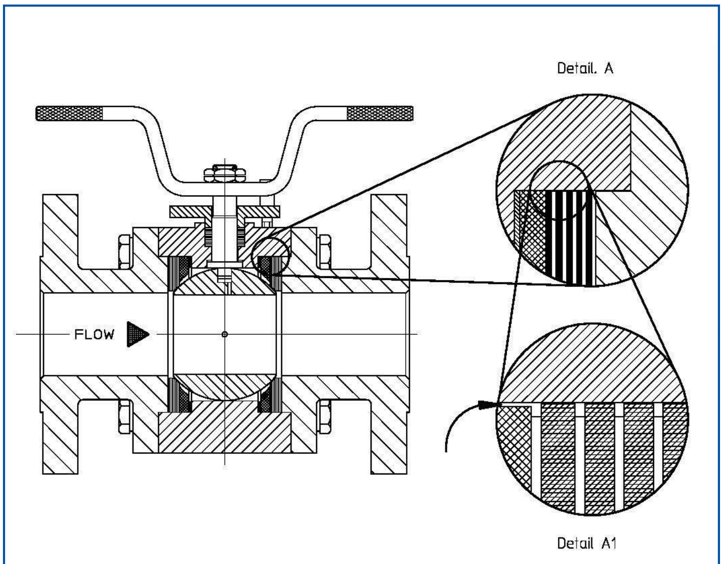

Fig. 1: Gasket structure principles based on alterned material layers

To overcome this distrust a safe valve is required, offering constant and full in-line and environmental tightness, whether after a long opening period or after a protracted closure, or after several on-off cycles both at high and low temperatures, and mainly after having operated under protracted lamination conditions at an high differential pressure. (Some or all of these conditions can be met with various fluids in batch production, see steam in the food industry, particularly cheese production, or saturated water and steam in the rubber industry).

Gaskets and seats

If the problem lies in gaskets and seats then there is a solution. A few years ago a technology was developed which has proven to be very effective. It can also be applied in other types of valves and similar applications. The principle is to develop a gasket structure based on alternate layers of stainless steel and expanded (then elastic compressed) graphite. We will examine this structure below.

In the valve we will consider (an HTBV, short for High Temperature Ball Valve) this technology is used through a sort of topological shrewdness. This device resembles a common three piece ball valve, but includes a double innovation, which first of all reduces the principal construction components from two to one (seat and gasket become a unique integral piece), whilst the second innovation is obviously the seat, consisting of a structure made of alternate layers of expanded graphite and stainless steel (see Figures 1 to 4).

Let us first consider the most foreseeable aspect of our analysis, that is the environment tightness. This tightness is obtained against the internal cylindrical wall of valve body, by means of the micrometric protrusions of graphite layers out of the metallic rings (Figure 1, detail A.1). The shear resistance of the protruding graphite corresponding to the sole shear stress yielded by fluid pressure, exceeds the stress by a factor of ten and more (the topic has been dealt with several times. See, for example, Valve World Volume 8 Issue 3 page 71 and Volume 7 Issue 5 page 25) Also, the standard machining body-gasket allowance, in any case much less than the elastic deformation of graphite, allows a constant adherence along the whole ring contact with the valve body. Finally the connecting parts of valve to piping (flanges, or other type of fittings) do not affect the gaskets, but stand directly on body. Therefore adherence of gaskets to body, hence the environmental tightness, is not dependent on manoeuvring the load and piping stress, and practically independent of pressure or temperature variations. This peculiarity alone is obviously not exclusive, but relates to the safety grade which we have demanded above. Moreover, this very construction component is also the valve seat ring, supposed to ensure the on-line tightness, and for that reason it is worthwhile to perform some analysis.

Pressure/velocity progress

Figure 2 shows the most critical operating condition of valve, particularly of a ball valve, namely the opening start or closing end (choking and lamination conditions). Let us look at the fluid jets flowing through the "slot" between seat and ball mouth (Figure 3), their travel inside the valve ball and their output trough the second "slot".
The pressure/velocity progress along the ball is roughly as follows:

- slight compression nearly before the inlet
- velocity increase and pressure reduction at inlet of hollow plug
- then, at the same time, a sudden trend to increase pressure and reduce velocity
- then the opposite situation, of velocity increase combined with pressure reduction, at the exit of hollow plug.

This is of course only valid for the average progress of flow, but what matters here is the rather different behaviour of the peripheral jets gradually in contact with edges of seat layers (detail of Figure 4). These alternate layers of metal and graphite which force, locally, the flow to several compressions and expansions, thus forming an almost motionless veil. This phenomenon is particularly important here. Giving the increase of flow sections as the fluid progresses after its inlet in the ball (and respectively the reduction towards the ball outlet), the pressure steps of fluid in contact with graphite layers, following the velocity steps corresponding to the metal layers, partially recover the velocity-pressure situation of the preceding stages. Therefore, as far as the fluid jets near the seat ring are concerned, the process is similar to that in a multistep pressure reducing valve, where each graphite layer bears only a fraction of the total differential pressure.
It is worthwhile analysing some further details of the flow internal path. (Note: since we are dealing with erosion, tightness and possible cavitation, the following remarks are in particular related to liquid flow). Let us take into consideration an opening around one to two per cent of total travel, where the involved operating condition is possible, that is a pressure drop higher than the critical differential pressure of the concerned fluid.
Comparing Figures 2 and 3, respectively describing the internal sections of a standard ball valve and a HTB valve, note the usual path of fluid jets, similar to an 'S' bend, under the centrifugal force and the wall reaction. This reveals an area where the cavitation could be trigged off given "proper" flow inlet conditions. The second figure shows that the various alternate layers of graphite and metal in the seat offer a higher resistance against the flow progress, therefore the centrifugal force is partially reduced and balanced, the total valve recovery factor is also reduced and there is a gradual pressure drop along the hollow plug wall. The S bend is not too much stressed, that is higher pressure not only outside the curve but also inside the curve, with less possible evaporation. This means there is a lower possibility of cavitation in the following inverse S curve bend (within the range of PN 100 this means an almost impossible cavitation situation).
We can now develop an approximate algebraic analysis, yielding some very rough numeric figures, to get a consistent idea of the process.

Fluid channels

Looking at Figure 5 consider a fluid channel of consecutive sections, approximately lens shaped, limited as follows:

- one side by the containing surface made of internal wall of the seat ring, then the valve body, then the hollow plug leading the flow till near the outlet
- the other side, the fluid jets of the main line (L).

A second fluid channel can be located as follows:

- one side, by the fluid jets of the main line
- the other side the inner wall of the ball, opposite to the previous one, then valve body wall, then the internal wall of the other seat ring.

The jets of the two channels will mix in the area near the outlet, giving rise to a turbulence increase and homogenisation of conditions in the whole section.
The continuation of both channels, mixed,

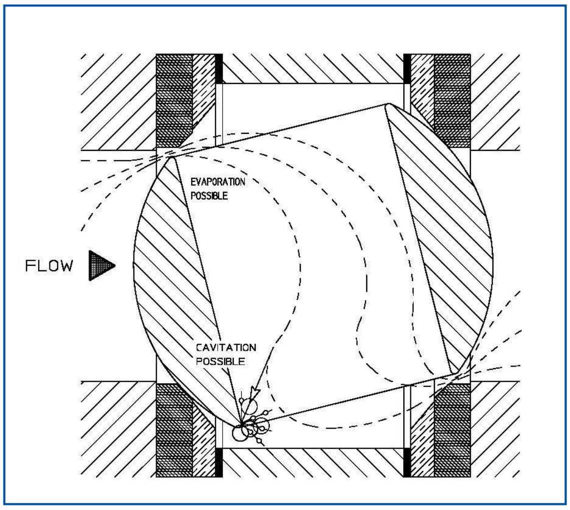

Fig. 2: Most critical valve operating condition

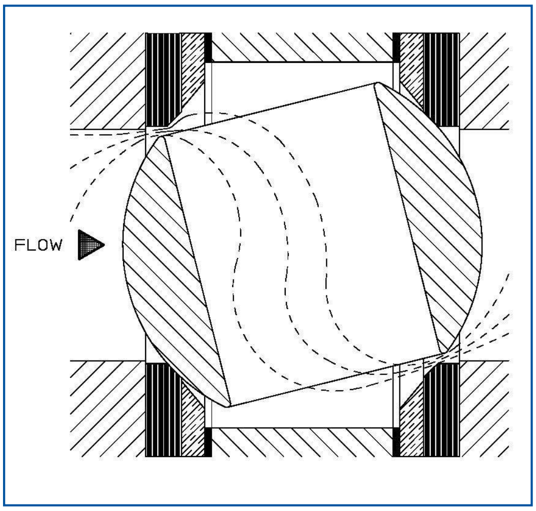

Fig. 3: Fluid flows through the "slots" between seat and ball mouth

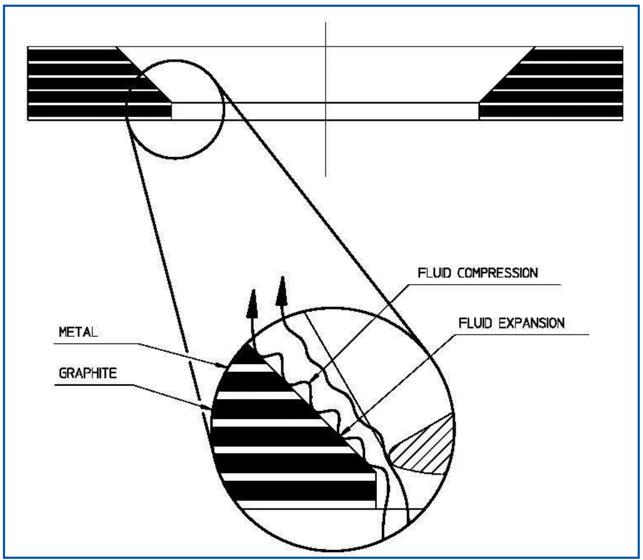

Fig. 4: Impact of seat layers on fluid

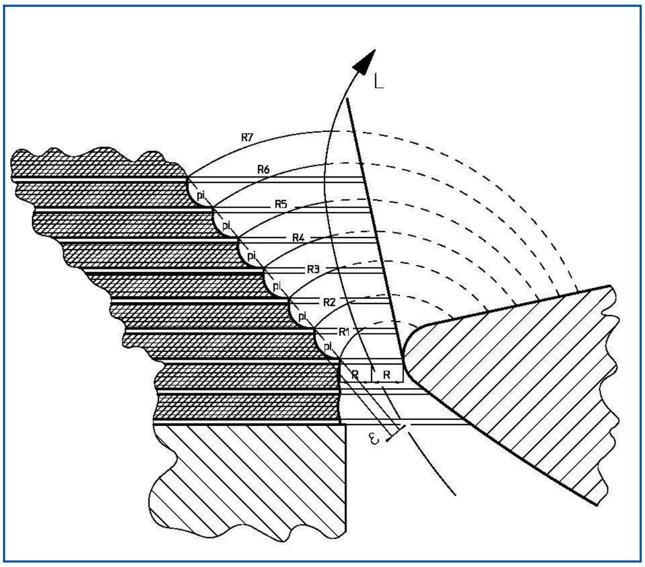

Fig. 5: Consecutive sertains of a fluid channel

will develop in this area, also symmetrical to the first area described. Here we can consider the progress of velocity and pressure, towards the outlet, considering one half of the outlet slot.

Let us analyze the pressure progress in the proximity of the seat ring internal wall. If we take the metal layer as non deformable in comparison with the graphite layer, and assume for graphite a strain value corresponding to about a half of its yield limit, (a situation already examined in another occasion, occurring on PN 100 valves at the maximum allowed operating pressure), we obtain for a fluid jet large one unit, an expansion (with a following compression), "triangular" of size ε .

Let us call Ri the "thickness" of section going from the peripheral area interested to the main fluid jet z the distance between the edges if two adjacent metallic layers of seat

R a half width of inlet slot

f a factor roughly proportional to sin of angle between seat wall and main fluid jet at inlet.

Thus a sort of curvilinear triangle is defined, the base of which is the distance Ri, approximately yielded by

$$R_i = (R + f \cdot i \cdot z)$$

therefore the pressure Pi in the fluid sections corresponding to the metal edges, and not considering local

$$P_i = \frac{P_1 \cdot (R_0 + f \cdot i \cdot z)^2}{R_0^2} \cong$$

$$\frac{P_1 \cdot (R_0 + 2 \cdot f \cdot i \cdot z)}{R_0} \qquad (1)$$

recoveries due to the graphite strain will be:

Let us now evaluate the local expansion degree of the fluid channel. From the continuity equation, with the above approximation which takes into account the radial expansion only, calling the velocity of fluid jet just coming out from metal ring, and the velocity of fluid jet when touching the strained layer of graphite, we have:

$$\omega_i = w_i \cdot \left(\frac{R_i}{R_i + \varepsilon} \right) \cong w_i \frac{R_i - \varepsilon}{R_i}$$

$$\omega_i^2 = w_i^2 - 2w_i^2 \varepsilon / R_i + w_i^2 \varepsilon^2 / R_i^2$$

that is, neglecting the term containing the square of $\frac{\varepsilon}{R_i}$,

$$w_i^2 - \omega_i^2 = 2\omega_i^2 \varepsilon / R_i$$

according to the Bernoulli theorem (conservatively neglecting the head loss)we can write:

$$p_i + \omega_i^2 \cdot \rho / 2 = P_i + \omega^2 \cdot \rho / 2$$

where

p_i is the pressure in the area of strained graphite and

ρ is the fluid density

The pressure recovery factor in the expansion stage is thus:

$$p_i = P_i + w_i^2 \cdot \varepsilon \cdot \rho / R_i \quad (2)$$

Since the choked, inlet section area is very reduced, we take the velocity in the pipe as null and express the relation between fluid velocity soon inside the ball and the inlet ball w_i pressure P1 with:

$$w_1^2 = \frac{P_0 - P_1}{\rho \cdot \varsigma}$$

where ς is a factor above unity, typical of a pipe segment or fitting, which takes into account the head loss included in Bernoulli theorem.

Along the curvilinear triangle the velocity will be:

$$w_i^2 = \frac{P_0 - P_1}{\rho \cdot \varsigma(1 + f \cdot i \cdot z)} (3)$$

from (1) , (2) and (3) we obtain:

$$p_i = P_1 \frac{R_0 + 2 \cdot f \cdot i \cdot z}{R_0} \cdot [1 + \frac{\varepsilon (P_0 - P_1)}{\rho \cdot R_0}]$$

where we see that:

- Pressure recovery, within the involved area inside the ball, is not only proportional to square of passage section offered to flow (factor $\frac{R_0 + 2 \cdot f \cdot i \cdot z}{R_0}$), after the choking "slot" where the fraction $(P_o - P_1)$ of inlet pressure is transformed into kinetic energy, but depends also on a further factor, namely $[1 + \frac{\varepsilon (P_0 - P_1)}{\rho \cdot R_0}]$

  This is determined by the amount of strain in the graphite layers, a value most significant in proximity of inlet section, just where and when the very passage has a size

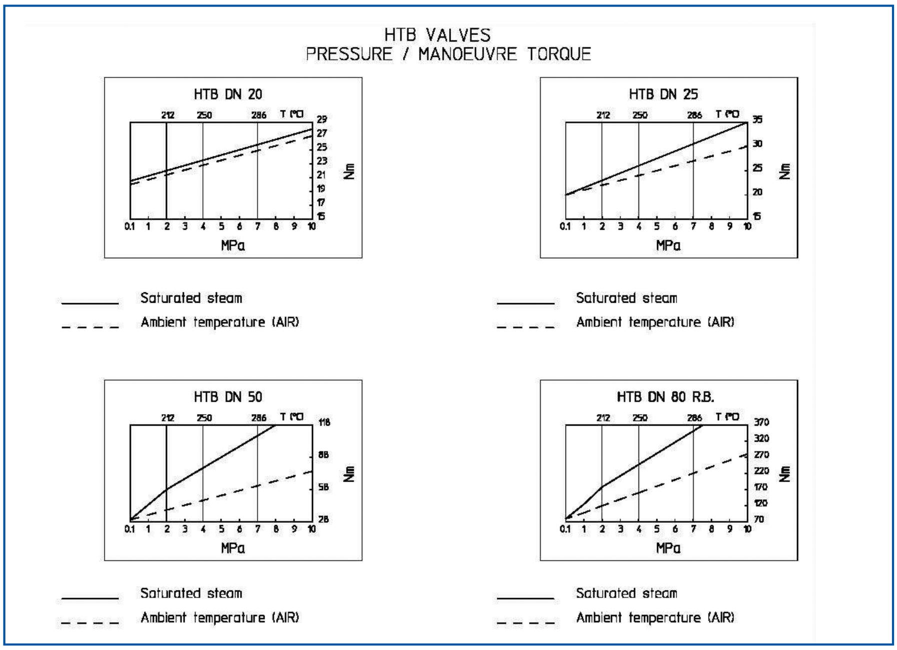

Fig. 6: HTB Valves pressure / manoevre torque

comparable with the graphite strain amount, and just when $(P_o - P_1)$ trends most to increase, corresponding to the opening reduction

- The flow velocity near the metallic layers is reduced as is seat erosion.

Velocity increase

In the following step, near the outlet port, Bernoulli's theorem again shows that a gradual velocity increase is produced. This is related to a gradual pressure decrease as the flow section is being gradually reduced. Again, in the second step, the relationship between pressure reduction and velocity increase is emphasised by partial pressure recoveries corresponding to the area expansions in the points of strained graphite. This resembles that of the preceding step, but with an opposite trend along the flow progress. In this way, sudden pressure drops $(P_o - P_1)$ and velocity increases at the ball outlet (which would occur in a standard ball valve seat) are avoided.

Inlet and outlet pressure/velocity transformations are being added, that means, if we could set the precise numeric values in the found formula, a further halving of the initial pressure drop , the one which sets the inlet velocity and, since we are dealing with liquids, the outlet velocity as well.

As a matter of fact, the pressure soon after the ball inlet reduces the fluid velocity (and flow rate) in the two slots at inlet and outlet as well as along the graphite/metal layers in the inner wall of the seat ring, thus preventing erosion.

Therefore the total pressure drop across the valve is subdivided into two sets of drops, and each set is made of several steps, along the internal walls of seat rings, thus strongly reducing the total recovery factor.

All that produces a liquid veil along the layers of metal and graphite. This serves to protect the choking surfaces from erosion and facilitates a long operating life of gasket/seats in HTB valves, particularly after having operated a long time under lamination conditions.

Another point to be taken into consideration is the operating torque and its variation due to changes in operating conditions, mainly the temperature. Standard seat construction materials allow operating torques to be relatively reduced at given specified temperatures, but require much higher torques when operating under other (typically "high") temperatures. Similarly, a favourable torque setting at a high temperature would not be advisable since it would yield a high probability of leakage should the temperature drop. (Note: the temperature when the valve closes drops by definition).

This aspect favours the geometrical, chemical and mechanical stability of graphite, since this soft component is able to follow the (limited) strain of metal parts, ensuring full tightness at the various operating conditions. Moreover, graphite is suited to varying operating torques caused by temperature variations (see Figure 6).

Case history

Finally, we should consider the scale which can build up on seat rings. This phenomenon, which can occur even "before" erosion, can make a valve useless even before any erosion is actually witnessed. As any other ball valve, the HTB valve offers an advantage at the shut-off stage, namely that the ball is "brushed". The alternate metallic and graphite rings have proven to contribute to very effective brushing since scale is completely removed from seat. The level of cleanliness would not be obtained from a metal seat.

To give a practical example, consider applications of valves in geothermal steam lines. The endogenous steam under treatment carries a noticeable fraction of borax (borax is also a by-product resulting from the exploitation of the reservoir). The thermodynamic transformations inside the valve tend to produce a solid deposit which is concentrated on the seat surface. Consequently, opening and shut-off are rather unsatisfactory. Until the HTB was brought in, the only solution had been periodical replacement of the valve, normally at intervals of less than one month. Since its installation, the HTB has seen more than a year of service without the need for any maintenance. In fact, it is reasonable to estimate a valve operation life at least as long as the installation life time. ■

About the Author

Mr Raffaele Angelini, who graduated in 1959, has been dealing with design of automatic devices such as control and safety valves transmitters, gauges, etc, since 1963. He is currently involved in the development of instrument lines, including low noise control valves high differential pressure valves, glass and magnetic level gauges.