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DESIGN AND CHARACTERISATION OF A DOUBLE-PISTON PRESSURE BALANCE IN HIGH PRESSURE

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Abstract – The paper describes the design and the analysis of a new kind of pressure balance intended to operate up to 500 MPa. The design can be considered as a calculation test to evaluate the possibility to produce a pressure balance whose changes in effective area with pressure (and, consequently, the elastic distortions) are small if compared to other pressure balances operating in the same pressure range and, furthermore, it presents an extended measurement range. The method used for the theoretical characterisation is an improvement of a numerical method using a finite element method.

Keywords: Pressure balance, Double-piston

1. INTRODUCTION

The development of innovative high pressure production technologies and the scientific interest in determining the thermophysical properties of materials has increased, in the last years, the attention paid to high pressure metrological traceability. Pressure balances are widely used over a very wide range (from 1 kPa up to 1 GPa and above) for the generation of primary standards of pressure [1]. Currently conventional pressure balances that operate at very high pressure are based on re-entrant and controlled clearance modes, and also moderately on free-deformation (simple) mode up to 500 MPa. Several studies on innovative pressure balances were carried out in the last years in order to extend the metrological performances to very high pressure [2] or to increase the rangeability of pressure balances [3].

Recently, the Warsaw University of Technology investigated and proposed a new kind of pressure balance combining the advantages of both simple and re-entrant balances [4]. The effective area typically increases with pressure in simple units while it decreases in re-entrant units. In mixed units a self-compensation of these two opposite effects can be observed, with the advantage of extending the operating range without using another pressure control system. A mixed unit operating up to 1 GPa was designed and built at Warsaw University and studies based on Zhokhowski's simplified model and Finite Element Analysis to extend the measurement range up to 2.6 GPa were presented in [2, 5-6]. On the other hand, in [3] an innovative dual-range pressure balance was designed and made by Budenberg Gauge Co Ltd. This pressure balance is constituted by two coaxial pistons of different diameter, the one of small diameter fitting closely the larger. At low pressures the larger piston acts in the usual way whereas if the pressure become higher it comes against a stop and the smaller piston takes over. In any case, as regards pressure balances, it is important that the most important design and operative parameters, such as the pressure distortion coefficient, λ, that allows to estimate the change in the effective area, A_e, and the piston fall rate, v, be well understood to keep errors and uncertainties to a minimum.

At higher pressures (over 50 MPa) the change in the effective area due to elastic distortion of the piston and cylinder becomes a significant source of uncertainty [1]. For example, if a piston-cylinder unit has a pressure distortion coefficient l equal to 0.8 10-6 MPa-1 and considering uncertainties of A_e and A_0 of 15 and 10 ppm respectively, for a pressure of 100 MPa, a λ uncertainty of 14% is obtained and from the uncertainty contribution will be 11.2 ppm. Consequently in the primary calibration field, in order to obtain the best uncertainty in the pressure measurement, both low l values and related uncertainty less than 10% have to be obtained [7].

The change in Ae with pressure is described by [8]:

$$A_e(p) = A_0\left[1 + f(p)\right] \quad (1)$$

where A_0 is the effective area at reference conditions and f(p) models the change in area with pressure (generally a linear model $f(p) = \lambda \cdot p$ is used). Recently, different theoretical methods to evaluate the pressure distortion coefficient by the analytical [7, 9-10] or numerical solution [2, 11-13] of a system of differential equations were developed.

In the present work, the authors propose a new kind of pressure balance able to limit the elastic distortions in the clearance gap at high pressures and, in the meantime, to extend the measurement range to lower pressures. The main difficulty in designing the pressure balance was to limit the piston fall rate. Both to design and to theoretically characterise a 500 MPa pressure balance, a FEA method is used, that is an improvement of a numerical method reported in [12].

2. THE DESIGN OF THE DOUBLE-PISTON PRESSURE BALANCE

This new kind of pressure balance proposed differs from a conventional piston-cylinder unit because of the existence of two gaps. In fact, the pressure balance is made up of two concentric pistons (Fig. 1). Figure 1 gives a schematic overview of the 500 MPa pressure balance, showing its most important geometric characteristics.

The unit can indifferently work in high pressure range with the internal cylinder joined to the external one or in low pressure with the internal piston joined to the external one constituting only one piston. In any case, the pressure acts, independently from the pressure range adopted, both in the internal and the external gap. Thus, the characteristic working mode allows one to extend the measurement range also at lower pressures (typically a pressure balance operates with an acceptable repeatability only for measurement pressure higher than 5% of the full scale). Furthermore, the particular geometry allows to obtain very small change of the effective area versus the measurement pressure and a piston fall rate value not very high despite of the presence of two clearance gaps.

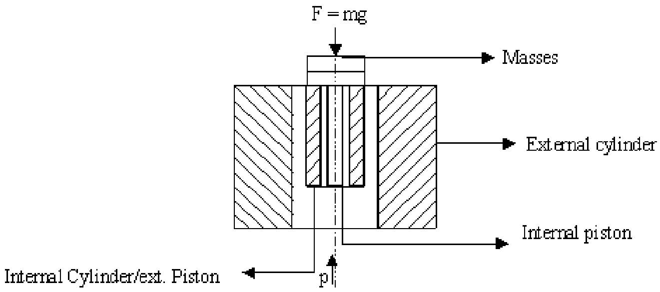


Fig. 1. Schematic view of the 500 MPa pressure balance

Both cylinder and piston are designed in tungsten carbide, selected with a Young modulus E, equal to a 543 GPa and a Poisson modulus ν, equal to 0.238. As working fluid an ethylene (50%) and glycol (50%) mixture is adopted, normally named ethylene-glycol. The ethylene-glycol properties (practically dynamic viscosity η and density ρ versus pressure) are evaluated by using the experimental data reported in [14]:

$$\eta\left(p,\ 20°C\right) = 0.13\ \exp\left(4\cdot 10^{-6}\cdot p\right) \qquad \eta/(\text{Pa s}) \text{ if } p/\text{GPa} \tag{2}$$

$$\rho\left(p,\ 20°C\right) = 1185.9 + 305.73\ \cdot p - 188.3\cdot\ p^2 + 63.194\cdot\ p^3 \qquad \rho/(\text{kg m}^{-3}) \text{ if } p/\text{GPa} \tag{3}$$

2.1. Numerical procedure

A numerical procedure is proposed to design and characterise the pressure balance. This procedure uses the piston-cylinder dimensions, material and operating fluid to calculate the distortions and pressure gradients in the clearance. The above mentioned parameters are then used to calculate the piston fall rate and the change of the effective area versus the measurement pressure and they have been extremely important for the design of the unit and for the selection of the appropriate clearances of the piston-cylinder unit.

In particular, the evaluation of the effective area is based on the following equation obtained from a balance of the forces acting on the piston [13]:

$$A_e = \pi\cdot r_p\left(0\right)^2\cdot\left[1+2\frac{u(0)}{r_p\left(0\right)}\right]+ \\ +\frac{2\,\pi}{p}\cdot\int_0^L r_p\left(x\right)\cdot\left(-\frac{h\left(x\right)}{2}\frac{dp}{dx}\right)dx+\frac{2\,\pi}{p}\cdot\int_0^L r_p\left(x\right)\cdot\left(p\frac{du}{dx}\right)dx \tag{4}$$

where the radial elastic displacements of the piston u(x), the radial gap h(x) and the fluid pressure distribution p(x) are determined using the mechanical theory of elastic equilibrium and fluid and dynamic theory of laminar flow. In particular, the elastic distortions of piston and cylinder, knowing the pressure along the clearance, are obtained using the equilibrium field equations and the compatibility and the design-constitutive (taking care of appropriate boundary conditions) relationships. The solution of the equation system can be numerically obtained via the F.E.A. method allowing the radial clearance dimension of the piston-cylinder engagement to be determined using the following equation:

$$h(x) = h_0 + U\left(x\right) - u\left(x\right) \tag{5}$$

On the other hand, the fluid-dynamic theory of laminar flow allows the distribution of pressure p(x) along the clearance to be obtained, provided the clearance radial dimension h(x) is known. Assuming i) monodimensional laminar flow, ii) stationary regime, iii) Newtonian fluid and iv) constant temperature, one can obtain by means of the Navier-Stokes equation:

$$p(x) = p_0 - \frac{6\cdot\dot{m}}{\pi\cdot r_p}\cdot\int_{x_0}^{x}\frac{\eta(p)}{\rho(p)}\frac{1}{h(x)^3}dx \tag{6}$$

As regards the FEA method, triangular iso-parametric and axial-symmetrical type elements are used. The number of elements (approximately 8000 elements for the cylinder and 3000 for each piston) and the mesh type are determined considering a mesh optimisation criterion. In particular Zienkiewicz-Zhu criterion is used [15]. Finally, as regards to the convergence and consistence of the numerical solution, the moving average criterion is applied.

2.2. Design optimisation

Figure 2 shows the final geometry of the model adopted as result of the optimisation analysis with the corresponding coordinates of the most important points reported in Tab. I.

Respect to the simple double-piston unit, the following modifications were adopted:

- a cross-section reduction is added over the external gap to increase the pressure losses and then to diminish the piston fall rate;
- the internal gap in the lower part of the engagement is increased to avoid interferences.

As regards the boundary conditions, it is necessary to consider that the measurement pressure is applied also on $\overline{sd}$ and $\overline{qg}$ and that the gap between the second piston and the cylinder is composed of two different engagement with two different pressure distribution (with the same pressure value in the section k-t-V-U).

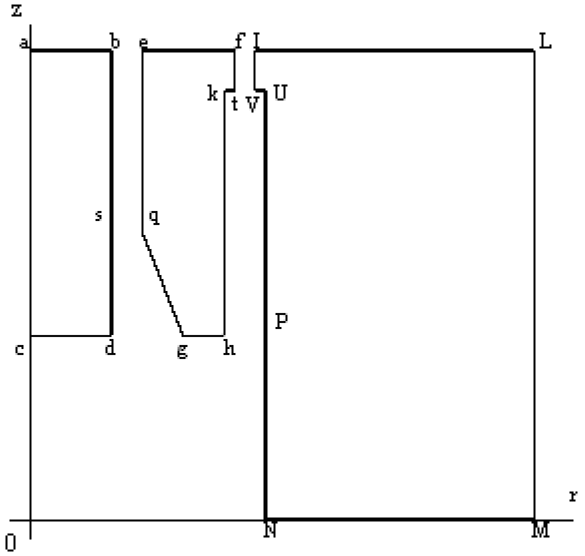


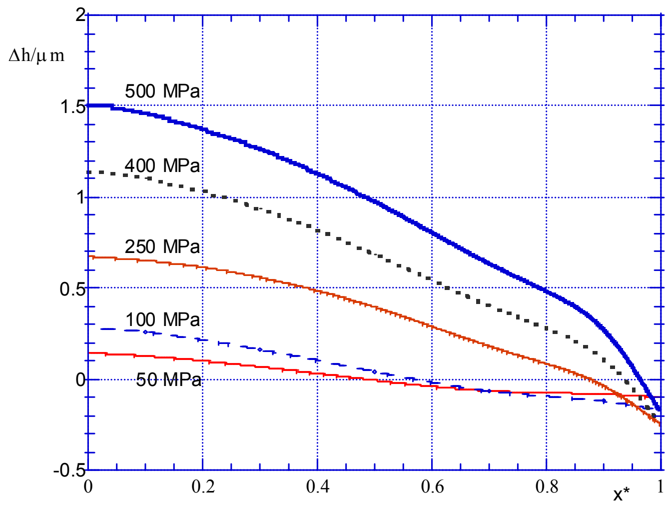
Fig. 2. Geometric model

TABLE I. Coordinates of the most important points

Points	r/mm	z/mm
a	0	38
b	2.5	38
c	0	17
d	2.5	17
s	2.5	28
e	2.5008	38
f	4.5013	38
g	3	17
h	4.5011	17
q	2.5008	28
k	4.5011	35
t	4.5013	35
I	4.5018	38
L	30.5021	38
M	30.5021	0
N	4.5021	0
P	4.5021	17
U	4.5021	35
V	4.5018	35

3. NUMERICAL RESULTS

The dimensionless internal pressure, $p^* = p(x)/p_m$, and the gap radial variation for the internal engagement of the pressure balance proposed are showed in Fig. 3 as the dimensionless axial coordinate, $x^* = x/\overline{sb}$, varies (where $x^* = 0$ corresponds to the point s and $x^* = 1$ to the point b). In particular, the gap profile increases as the pressure increases showing a maximum value of 2.3 µm for a measurement pressure equal to 500 MPa. As regards the pressure profile, it is a parabolic one with a concavity that increases as the pressure increases (see Fig. 3b). This trend is similar to the ones reported in [11, 13].



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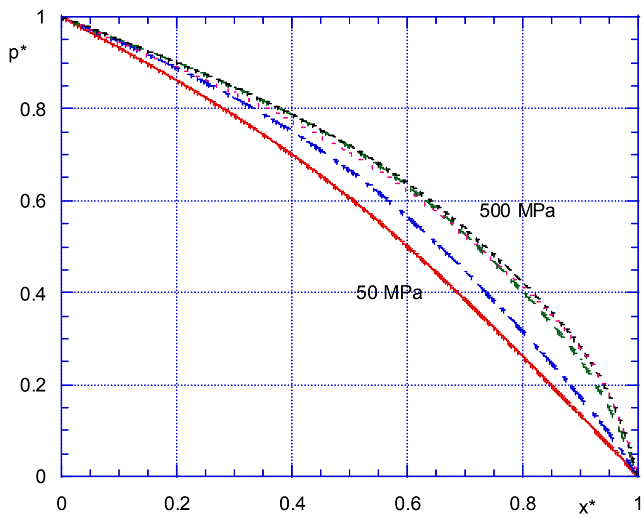
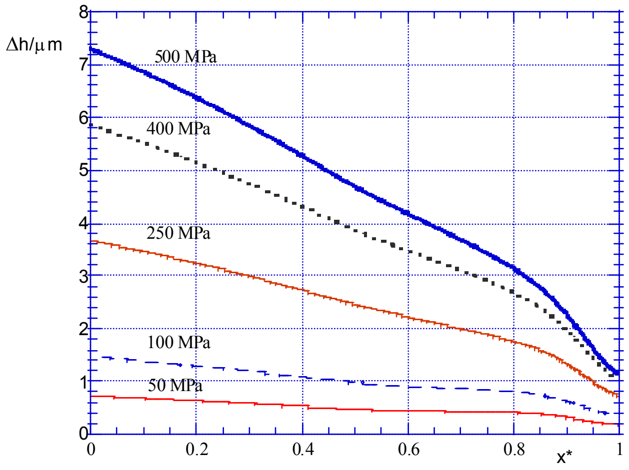


b)

Fig. 3. a) Radial gap variation and dimensionless pressure in the internal gap as the axial coordinates varies.

In Fig. 4, the dimensionless pressure and external gap variation for the pressure balance reported in Fig. 2 are showed as the dimensionless axial coordinate varies (where $x^* = 0$ corresponds to the point h and $x^* = 1$ to the point f). As regards the gap profile, the range can be divided in two parts: a first one ($0 \le x^* \le 0.857$) where the values are high with a maximum value of 8.2 µm for a measurement pressure of 500 MPa and a second one ($0.857 \le x^* \le 1$) with a reduced values of the gap. The relevant part of the pressure drop is found in the second part of the engagement (see Fig. 4b) confirming the importance of the cross section reduction adopted.



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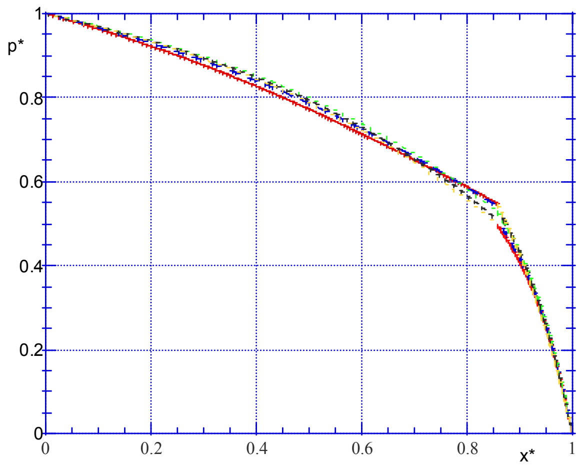
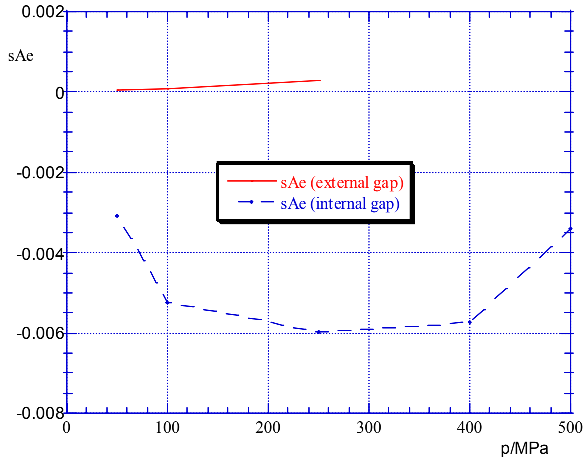


b)

Fig. 4. a) Radial gap variation and dimensionless pressure in the external gap as the axial coordinates varies.



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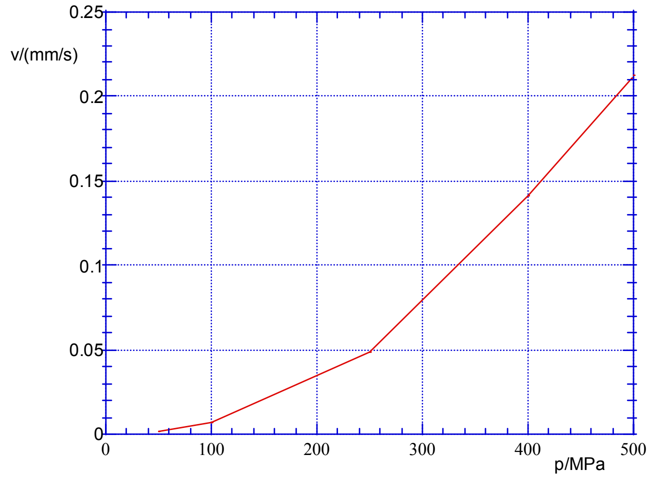


b)

Fig. 5. a) Effective area and b) piston fall rate versus the measurement pressure.

Figs 3 and 4 show a good performance of the pressure balance referred to gap and pressure profile but the geometric optimisation allows to obtain small changes of the effective area. In fact, in Fig. 5a the effective area presents a negative trend and the corresponding variations respect to the effective area at reference condition (defined as $sA_e = (A_e - A_0)/A_0$) are less than 60 ppm. These values were obtained introducing the cross section reduction and in Fig. 5b the piston fall rate is reported as the measurement pressure varies.

The change in the effective area are extremely small respect to the ones of other pressure balances that operate at high pressure. In Fig 6, a comparison between the change in the effective area and the piston fall rate of the pressure balance designed and the ones of two complex units [11, 16] in high pressure range is reported.

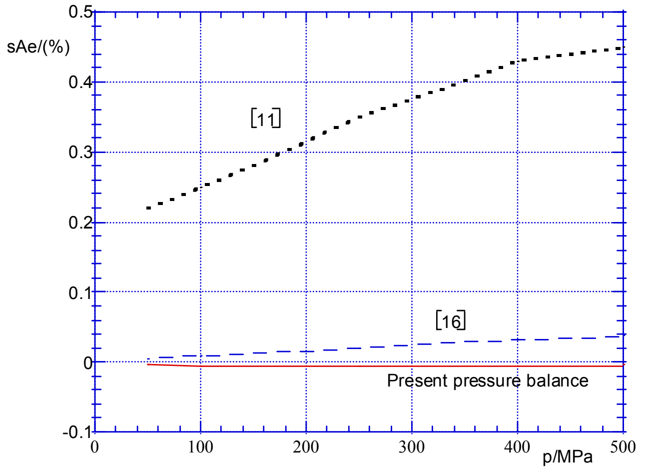
As regards the use of the present pressure balance at lower pressure, the pressure distortion coefficient considering the external gap is equal to 0.94 10^{-6} MPa^{-1}, value that is compatible with the ones usually referred to similar pressure balances [1, 7, 11, 16].

4. CONCLUSIONS

The present study designs and analyses a new kind of pressure balance intended to operate up to 500 MPa, able to limit the elastic distortions in the clearance gap at high pressures and, in the meantime, to extend the measurement range to lower pressures.

The unit, made up of two concentric pistons, can indifferently work in high pressure range with the internal cylinder joined to the external one or in low pressure with the internal piston joined to the external one constituting only one piston.

From the numerical results, the change in the effective area are extremely small respect to the ones of other pressure balances that operate at high pressure. The main difficulty in designing the pressure balance was to limit the piston fall rate.



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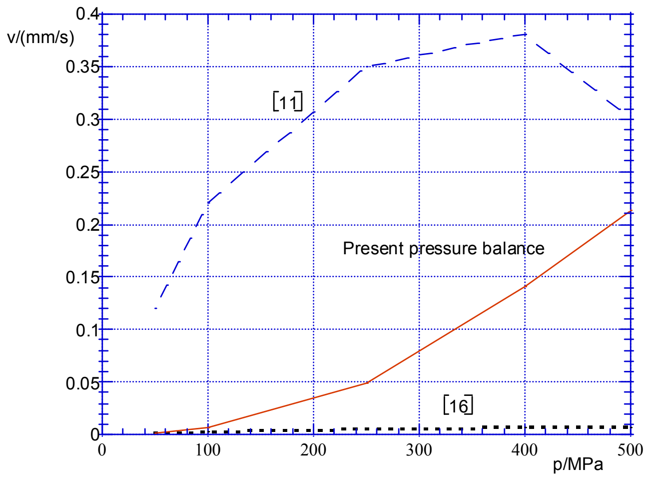

b)

Fig. 6. a) Change of the effective area and b) piston fall rate versus the measurement pressure

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