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Equipment for the Resistance Testing under Dynamic Torsion Loads of the Copper Pipes Joints

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Abstract

This paper presents the structural and functional description of the testing equipment conceived by the authors in order to investigate the resistance under dynamic torsion loads of the joints for copper pipes and fittings used within the natural gas installations. Such equipment has been designed in order to perform the tests required to assess the quality of the assembly technology used for natural gas installations made of copper pipes.

Key words: *copper pipes, press fit joints, hard soldered joints, dynamic torsion loads.*

Introduction

The research work of the authors, presented in this paper, have been materialised in conceiving, designing and constructing a stand required for the resistance and tightness testing under dynamic torsion loads of the joints resulted from pressing or hard soldering of the copper pipes used for the natural gas interior installations.

The tests to be performed using the designed equipment are part of the tests – performed according to procedures conceived by the authors and presented in another paper [2] – required in order to assess the quality of the assembly technology (with press fit or hard soldered joints) used for the natural gas installations made of copper pipes. The advantages of using such technologies have been presented by the authors in a previous paper [1].

The Constuction and the Operation of the Stand

The stand designed to investigate the resistance under dynamic torsion loads of the joints resulted from pressing or hard soldering of the copper pipes, has been conceived – according to the testing procedure developed by the authors [1, 2] – to strain an experimental model (see fig. 1) made up of a fitting, *2*, and two copper pipes, *3*,*4*, at a symmetrical alternating torsional stress, at the room temperature and at the atmospheric pressure.

One end of the model, *3*, is rigidly fixed, and the other end, *4*, is imprinted with an altenating rotation at a constant amplitude with a duration (number of cycles) and a frequency specified in the testing procedure indicated in [2].

The stand (figs. 1 and 2) is made up of an electrical engine provided with an actuator mechanism which performs an alternating rotation of the end *4* of the experimental model, while the other end *3* is firmly fixed by means of a holding device *1*.

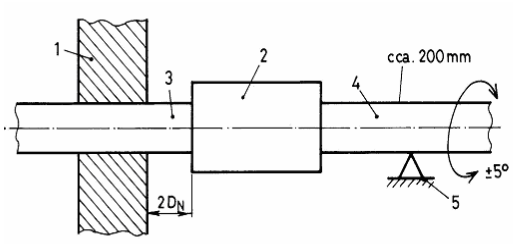

Fig. 1. Scheme of the stand for the resistance test under symmetrical altenating torsion of the joints for copper pipes: *1* - pipe retaining device; *2* – fitting; *3* - pipe 1; *4* - pipe 2; *5* - support.

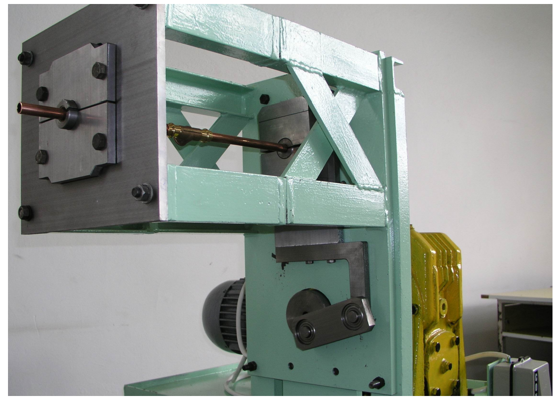

Fig. 2. Photo of the stand for the resistance test

The main elements of the conceived stand are the following:

- the frame of the stand on which all its components are fixed;
- the electrical actuator mechanism (a motoreductor) at whose driving shaft is attached the movement transmission device;
- the retaining device with the role to rigidly fix the left end *3* of the experimental model, made up of two fishing sockets fixed on a guide plate;
- several welded subassemblies playing the part of sustaining elements for the retaining device above and for the movement transmission device; the main element is the foot plate which directly sustains the motion transmission device and – by means of four spacers – the retaining device of the end *3*, being fixed at the frame of the stand by means of a support;
- the motion transmission mechanism from the driving shaft of the motoreductor at the right end *4* of the model, made up of:
  - a bushing fitted on the driving shaft of the motoreductor whit the function of leading element, and having an eccentric catch pin with bearing corresponding to a first kinematic coupling of the mechanism,
  - a fishing plate having the function to transmit the motion from the bushing to the driving pin,
  - a driving pin which transmits the motion to the retaining device of the end *4* of the experimental model; the driving pin has an eccentric catch pin with bearing, representing another kinematic coupling of the mechanism,
  - the retaining device for the end *4* of the model made up of a guide plate, two fishing sockets and two blades; the fishing sockets also have the function to make up – togetter with a bushing guide fitted on a bed plate – a friction kinematic coupling.

As a driving mechanism for the concieved stand, an electrical driving system was adopted in the most convenient version (from both the economic and the reliability points of view) of a three-phased induction motor. The relatively high rotational speed of such a motor requires the use of a rotational speed reductor, and therefore practically the use of a motoreductor was necessary.

The stand functions as follows: the end *4* of the experimental model is imprinted with an alternative rotational motion by the means of the driving mechanism, while the end *3* remains fixed. The amplitude of the motion is determined by the dimensions of the elements of the driving mechanism and it corresponds to the one defined within the elaborated testing procedure (presented in [2]).

During the design of the stand for the investigation of the resistance under dynamic torsion loads, the following issues have been considered:

- the adoption of the simplest possible kinematic scheme (a driving mechanism with the minimum number of kinematic elements) and of the most reliable one;
- the creation of a stand with the smallest dimensions possible and very easy to handle, especially from the point of view of the easy way to assemble the experimental model;
- the design of the retaining devices of the experimental model on the stand so that it is avoided the event of a break or a slit in this area; to this purpose, the length of the fishing sockets and of the holding blades was determined by assessing the contact pressure exercised on the copper pipes of the experimental model, so that their surface should not be damaged;
- the assurance of a high rigidity of the stand; as a result, among other things, the holding blades of the end *3* of the experimental model have been oversized;
- the selection of the materials from which the compenents of the stand were made in order to obtain the necessary resistance characteristics at the lowest price; the stress to which these elements are submitted having relatively low values, plain carbon steel was used.

After conceiving a first constructional scheme of the stand, its main elements have been designed taking into account the typical formula for simple stress and considering the main forces that act upon them, that is those that result from the weight of the sustained elements and those developed from the driving mechanism (assessed on the basis of the kinematic and dynamic analysis presented in the next paragraph).

In order to perform a the resistance test under symetrical alternating torsion on the concieved stand, the following steps have to be followed:

- the components of the experimental model are prepared and assembled;
- the fixing elements of the blades that sustain the left end of the experimental model are removed; the superior blade is demounted and the two fishing sockets are removed (if they were previously mounted);
- the elements that fix the blades that sustain the right end of the model are removed and the placing elements for the superior blade are weakened;
- the two fishing sockets of the right end are removed and then also the blade;
- the inferior fishing socket from the rigid end of the model is placed on the inferior blade;
- the fishing sockets (with the dimension coresponding to the pipe diameter of the experimental model), are placed at the right end of the model, after which they are introduced together with this one in the bushing guide;
- the left end of the model is fixed on the inferior fishing socket after which the superior fishing socket is fixed;
- the superior blade from the right end of the model is set up, after which the guide plate and the inferior blade are firmly fixed;
- the superior blade from the left end of the model is set up, after which it is firmly fixed on the inferior blade;
- the motoreductor is set to motion and the experimental model is submitted to the symmetrical alternating torsional stress for the established number of stress cycles (10);
- the experimental model is demounted from the stand following in a reversed order the operations performed at the set-in stage.

The result of the test is considered to be positive and the tested type of assembly is considered to be adequate if, after the test, the experimental model keeps its tightness. To this purpose, the experimental model is finally submitted to a tightness test at internal pneumatic pressure, at the room temperature, following the procedure presented in [2].

The Kinematic Analysis of the Movement Transmission Mechanism

In order to analyse the driving mechanism conceived for the stand, it was assimilated to a quadrangular mechanism, according to the kinematic scheme in figure 3. The three kinematic elements of the model considered for the mechanism, are as follows:

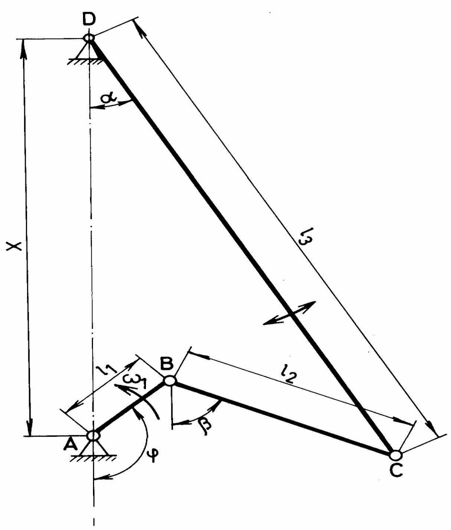

Fig. 3. The kinematic scheme of the driving mechanism of the stand

1) The conducting element (*AB* in fig. 3), which corresponds to the eccentric device, fitted to the driving shaft of the motoreductor, made up practically of the bushing with eccentric catch pin. The length of the conducting element, given by the distance between the axis of the shaft of the motoreductor and the axis of the catch pin, was adopted at the value: l_1 = 30 mm. The total mass of the two parts was considered to be the mass of the conducting element, having the approximate value: m_1 =5 kg. The angular speed of the conducting element, ω_1, is constant and is given by the rotational speed of the motoreductor, *n*; according to the testing procedure (which requires that the duration of a stress cycle should be of approximately one second), it must correspond to a frequency f = 1 Hz, and as consequence it has the following value:$\omega_1 = 2\pi f \approx$ ≈ 6.0 rad/s. The necessary rotational speed of the motoreductor also corepónds to the established frequency with the value: $n = 60\,f = 60$ rot/min.

2) The drived element (*BC* in fig. 3) which coresponds, practically, to the fishing plate with a total mass of aproximatelly m_2 = 1.5 kg. The length between the two axes of the fishing plate was adopted at the value of l_2 = 110 mm after the analysis from the kinematic point of view of several possible values.

3) The “balancer” element (*CD* in fig. 3), which must perform a rotative swinging motion with the amplitude established in the testing procedure ($\pm 5^0$), is made up of the driving pin with eccentric catch pin and the fixing device of the mobile end of the model and has the total mass of approximately m_3 = 16 kg. The length of this element corresponds to the distance between the axis of the catch pin and the axis of the fishing sockets and was adopted (after the analysis from thr kinematic point of view of several possible values) at the value l_3 = 327 mm. On the basis of this value, the dimensions of all the components were established.

The motion transmission mechanism has the following kinematic couplings, all being rotating couplings (fig. 3):

a) the coupling *A*, between the conducting element and the fixed elements of the stand (in fact of the motoreductor on whose shaft this element is fixed);
b) the coupling *B*, between elements *1* and *2*, coupling that corresponds in fact to the first bearing; the element *1* is the conducting element;
c) the coupling *C*, between element *2* and the “balancer”, coupling that corresponds to the second bearing;
d) the coupling *D*, between the “balancer” and the fixed elements of the stand (the foot plate), coupling which is a friction one; for this coupling, a bearing was not used, due to the complications that might have appeared and to the relatively low amplitude of the rotation of the “balancer” ($\pm 5^0$).

The kinematic analysis of the mechanism has been performed by applying the closed vector contour method and had as a purpose to establish its movement law and to compare it with the initial data. Several possible construction versions have been studied, finally the above presented solution being adopted.

As a result of the kinematic analysis and taking into account the value of the angle of the conducting element φ, the followings were determined: the variation of the angle between the "balancer" and the vertical axis *AD*, α; the variation of the angle between the element *2* and the vertical, β; the way the speeds and the angular accelerations of the two conducted elements (*2* and *3*) vary. We underline that, for the dimensions finally adopted for the elements that build up the "balancer", the exact vertical position of these ones corresponds to the value of the angle $\alpha = 20^o$. To this purpose, the following vectorial equation has been projected on two orthogonal directions:

$$\overrightarrow{AB} + \overrightarrow{BC} + \overrightarrow{CD} + \overrightarrow{DA} = 0 \quad , \tag{1}$$

from where the following two scalar relation result:

$$-l_1 \cos\varphi - l_2 \cos\beta + l_3 \cos\alpha - X = 0 \tag{2a}$$

$$l_1 \sin\varphi + l_2 \sin\beta - l_3 \sin\alpha = 0 \tag{2b}$$

where X is the fixed distance betweend A and D, whose finally adopted value is: $X = 308$ mm.

By eliminating the angles β from the equations (2a) and (2b) above, the following equation results:

$$\frac{l_1^2 + l_3^2 - l_2^2 + X^2 + 2 \cdot l_1 \cdot X \cdot \cos\varphi}{2 \cdot l_3} = (l_1 \cdot \cos\varphi + X)\cos\alpha + l_1 \cdot \sin\varphi \cdot \sin\alpha \,, \tag{3}$$

where the unknown is the angle α (for a given representative value of the angle φ) and whose solution was determined by numerical methods.

After the determination of α for the given φ, from equation (3) we can calculate the value of the angle β,using the equation(2a). Furthermore, by the differentiation of the equations (2a) and (2b), we obtain the following system of equations, whose unknowns are the angular speeds of the conducted elements ω_2 and ω_3:

$$l_1\, \omega_1 \sin\varphi + l_2\, \omega_2 \sin\beta - l_3\, \omega_3 \sin\alpha = 0 \tag{4a}$$

$$l_1\, \omega_1 \cos\varphi + l_2\, \omega_2 \cos\beta - l_3\, \omega_3 \cos\alpha = 0 \tag{4b}$$

Finally, by the differentiation of the equations (4a) and (4b) and taking into account the fact that the angular speed of the conducting element, ω_1, is constant (so its angular acceleration, ε_1, is null), we obtain a third system of equations with the unknowns – the angular accelerations of the conducted elements, ε_2 and ε_3. By solving the equations (3) and (2a) and applying the numerical methods for the finally adopted values of the lengths (l_1 = 30 mm, l_2 = 110 mm, l_3 = 327 mm, X = 308 mm) and the representative values of the angle φ,we have obtained the results of the kinematic analysis of the mechanism synthetised in table 1.

Table 1. The results of the kinematic analysis

φ (grades)	0	21.3	60	94.6	130	159.4	180	200.6	230	265.4	300	338.7
α (grades)	*17.6*	20.0	24.4	*27.4*	23.7	20.0	*17.6*	20.0	23.7	*27.4*	24.4	20.0
β (grades)	*105.1*	105.0	103.2	96.3	84.5	75.6	*73.3*	75.6	84.5	96.3	103.2	105.0

The results from table 1 confirm the fact that, at the right end of the experimental model firmly fixed to the "balancer",we obtain an alternating rotation movement with the required amplitude. Furthermore, the other systems of equations were solved numerically with the purpose to determine the maximum values of the angular accelerations of the conducted elements,ε_2 and ε_3.

Those values proved to be relatively low and as a consequence they were not included in the table. For the same reason, there is no need to perform a dynamic analysis, the moments of inertia which appear while the mechanism is working, having in their turn, low values.

In order to establish the required minimum power of the motoreductor, the value of the power input, P_m, necessary to set in the mechanism, was calculated. To this purpose, the motor movement which must be produced by the mechanism at the level of the "balancer" was carefully assessed by taking into account not only the mass and the inertia of the components of the stand which are moving, but also the rigidity of the experimental model at the level of the mobile end; the value, $M_m = 80$ N m resulted in this way.

The necessary minimum power was calculated by means of the relation below:

$$P = \frac{P_m}{\eta_t} = \frac{M_m \cdot \omega_1}{\eta_t} \cong 0.68\,\mathrm{kW}\ , \tag{5}$$

where η_t is the total mechanical efficiency of the mechanism and of the rotational speed reductor assessed at the value: $\eta_t = 0.7$.

Conclusions

This paper presents the construction and operation of the stand conceived by the authors for the resistance and tightness testing under dynamic torsion loads of the joints of the pipes and fittings made of copper, designed to be used within the natural gas installations. In order to verify the adopted solution for the stand, preliminary tests have been performed by applying the above presented methodology to experimental models made up of copper pipes and fittings with a nominal diameter of 15 mm, assembled by pressing.

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Echipament pentru încercarea la torsiune alternantă a asamblărilor nedemontabile ale țevilor din cupru

Rezumat

În lucrare se prezintă construcţia şi funcţionarea standului, conceput de către autori, în vederea efectuării încercării de rezistență şi etanşeitate la torsiune alternantă a asamblărilor nedemontabile ale țevilor şi fitingurilor confecţionate din cupru destinate instalaţiilor de distribuţie a gazelor naturale. Încercările efectuate pe standul prezentat sunt cuprinse în seria procedurilor de încercare pentru atestarea calităţii asamblărilor nedemontabile ale ţevilor şi fitingurilor confecţionate din cupru.