BULETINUL Universităţii Petrol – Gaze din Ploieşti	Vol. LVIII No. 3/2006	7 - 12	Seria Tehnică

Equipment for the Vibration Resistance Test under Bending Loads of the Joints of Copper Pipes

Mihail Minescu, Vlad Ulmanu, Andrei Dumitrescu, Dragoş G. Zisopol

Universitatea Petrol-Gaze din Ploieşti, Bd. Bucureşti, 39, Ploieşti
e-mail: mminescu@upg-ploiesti.ro

Abstract

This paper presents the structural and functional description of the stand conceived by the authors in order to test the vibration resistance under bending loads of the joints of copper pipes and fittings used in the natural gas interior installations. This stand has been built in order to perform the tests required to assess the quality of the assembly technology of copper pipes used for natural gas installations.

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

Introduction

The results of the research work presented in this paper were aimed at conceiving, designing and building of an equipment to be used for testing the vibration resistance under bending loads of the joints obtained by pressing or hard soldering of the copper pipes and fittings used within the natural gas installations.

The tests accomplished on the presented stand are part of the testing procedures conceived by the authors – presented in another paper [2] – in order to certify 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 these technologies have also been presented by the authors in a previous paper [1].

The Construction and the Operation of the Stand

The stand designed to test the vibration resistance under symmetrically alternating bending loads of the joints resulted from pressing or hand soldering of the copper pipes was conceived to submit an experimental model shaped as "U" (made up of several pipes and four fittings – two 90 degrees bends and two couplings) at vibrations (under bending loads), at room temperature and at atmospheric pressure .One end of the model is firmly fixed, while the other end is imprinted with an alternating movement with a constant amplitude, having the frequency and the duration (number of cycles) indicated in the testing procedure (presented in [2]). The stand (fig. 1) has an electrical engine provided with a controllable device with an eccentric, which induces – by means of a bar – the alternating bending of one end of the model, the other end being rigidly fixed by means of a retaining device.

The main elements of the conceived testing equipment are as follows:

- the frame of the stand on which all the other components are rigidly fixed by means of retaining devices;
- the electrical driving motor provided, at the driving shaft, with an eccentric device fitted with a link with a controllable position; the motor is also endowed with a device to register the number of the working cycles;
- the mechanism for the movement transmission from the driving shaft of the motor to the right end of the experimental model, which is made up of:
    - the device with eccentric, on whose link there is a bearing which represents the first kinematic coupling of the mechanism,
    - the kelly, fixed at one end to the bearing housing and having at other end a special screw, representing the second kinematic coupling,
    - the device with rectilinear reciprocating motion, which conveys the motion from the bar to the experimental model and is made up of two blades which provide the fixing of the model by means of two interchangeable bushings; the right blade is provided with a bar which is connected to a special screw,
    - the guiding blade of the device that was described above, provided with a bronze slide bearing, which is fixed to the frame;
- the device for the rigid fixing of the left end of the experimental model, made up of two fishing sockets and two blades, the left one being fixed to the frame by means of a support.

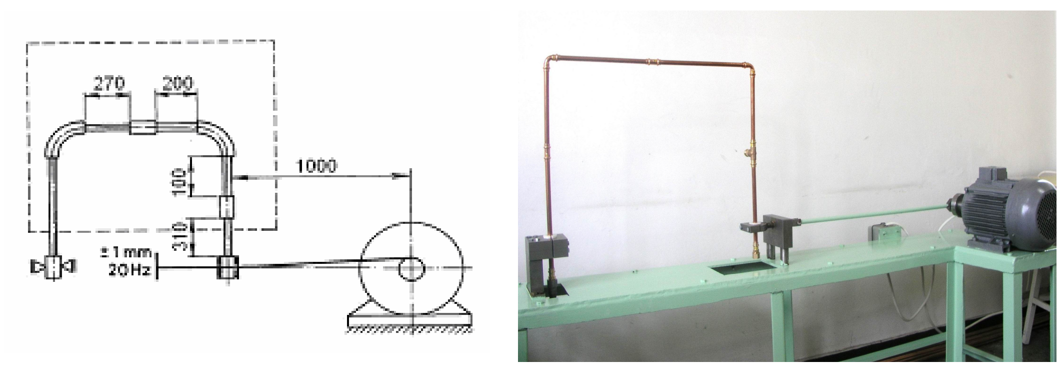

Fig. 1. Stand conceived for the vibration resistance test under bending loads of the copper pipes joints

An electrical actuator mechanism was chosen for the conceived stand, in the most convenient version (from an economic point of view, taking also into account the reliability and the easy utilization) of a three-phased induction motor. The large vibrations frequency necessary for the testing of the experimental model does not require the use of a speed reduction unit, the actuator mechanism being directly fixed to the shaft of the motor. While the stand is functioning, the right end of the experimental model is imprinted with a rectilinear reciprocating motion by means of the actuator mechanism, while the left end is remains fixed. The amplitude of the movement can be adjusted by modifying the position of the link with respect to the semicoupling fitted to the shaft of the actuator mechanism.

In order to design the stand for the resistance and tightness testing under bending loads, the following criteria were taken into account:

1. The adoption of the simplest kinematic scheme (an actuator mechanism with the smallest number of kinematic elements) and at the same time of the most reliable one.

2. The construction of the simplest stand possible, on which the experimental model can be mounted rapidly and easily.
3. The construction of the retaining devices of the experimental model on the stand, so that they should not produce a fracture or a crack in this area; practically, the length of the bushings and of the blades was initially established by assessing the contact pressure exercised on the copper pipes of the experimental model, so that the surface of these ones should not be damaged.
4. The assurance of a high rigidity to the stand; to this aim, for example, the retaining blades of the fixed end of the experimental model were oversized.
5. The selection of the materials from which the components of the stand were manufactured aiming to obtain the necessary resistance characteristics in the cheapest way possible; as the stress to which these elements are submitted is quite low, carbon steel has been used.

After the elaboration of the draft of the stand, its main elements were designed, using the typical formula for simple loads and taking into account the main forces that act upon them – those resulting from the weight of the sustained elements and those induced by the actuator mechanism (assessed by means of the kinematic and dynamic analysis of this one, presented in the next paragraph).

The standardized elements (bearing, bolts, nuts) were adopted on the basis of the indications of the standards, taking into account the forces that act upon them.

In order to accomplish a vibration resistance test under symmetrically alternating bending loads on the conceived stand, the following steps must be performed:

- the components of the experimental model are prepared and assembled;
- the position of the link in the channel of the semicoupling is adjusted so that we obtain the prescribed eccentricity of the actuator mechanism (1 mm);
- the retaining blades are removed, first on the left side of the stand and then on its right side;
- the experimental model is placed in its established position and it is kept as firmly as possible;
- at the left end of the model, the two fishing sockets are placed (their dimension must correspond to the diameter of the pipes which made up the model), after which the right blade is set;
- at the right end of the model, the two fishing sockets are placed and then the left blade is set;
- the correctness of the experimental model assembling on the stand is checked;
- the electrical motor is set in and the model is submitted to the symmetrically alternating bending loads for the established number of cycles (10^6);
- the experimental model is removed from the stand, following in a reserved order the operations undergone at the assembling stage.

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

The Kinematic Analysis of the Motion Transmission Mechanism

In order to analyse the actuator mechanism conceived for the stand, this one was assimilated to a crank and connecting rod assembly as the kinematic scheme from figure 2 shows.

The three kinematic elements of the considered model of the mechanism are the following:

1) The leading element ("the crank" *AB* – see fig. 2) which corresponds to the device with eccentric fitted to the driving shaft of the motor, made up of semicoupling, link and retaining elements. The total mass of these parts was considered to be the mass of the "crank" with an approximate value: m_1=3.5 kg. The length of the leading element, *r*, is given by the distance between the axis of the driving shaft and the axis of the link, which is eccentrically fitted. Following the requirements of the testing procedure, this length must have the value: $r = 1$ mm. The angular speed of the "crank", ω_1, is constant and corresponds to the rotational speed of the motor, *n*. According to the elaborated testing procedure, it must correspond to a frequency $f = 20 \pm 2$ Hz and therefore it has the highest value: $\omega_1 = 2\pi f_{max} =$ 138.3 rad/s ≈ 140 rad/s.

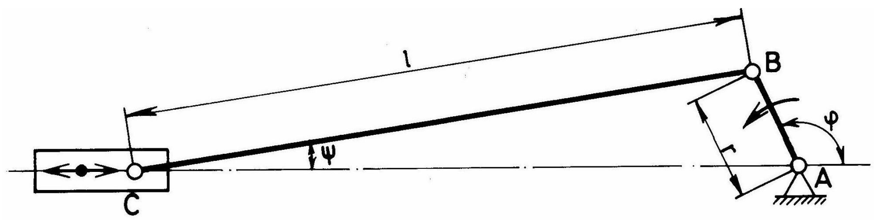

Fig. 2. The kinematic scheme of the actuator mechanism of the stand

The necessary rotational speed of the driving engine also corresponds to the established frequency, having the value: $n = 1200 \pm 120$ rot/min.

2) The "connecting rod" element (*BC* from fig. 2) which practically corresponds to the kelly and to the elements connected to this one, and has the approximate total mass: m_2=3 kg. The length of the "connecting rod" was adopted (l = 1000 mm) on the basis of the kinematic analysis of several possible values.

3) The "link" element (corresponding to point C from fig. 2), made up of: blades and bushings, fixing elements and bracket, with total approximate mass: m_l = 2 kg. The law of the motion of the "link" is the same with the one of the right end of the experimental model which is firmly fixed to the former.

The motion transmission mechanism has the following kinematic couplings (see fig. 2):

a) the rotation coupling *A*, between the leading element and the fixed element of the stand (in fact, the driving motor);
b) the rotation coupling *B*, between the leading element and the connecting rod, coupling which is materialised in the ball bearing;
c) the rotation coupling *C*, between the "connecting rod" and the "link", coupling that corresponds to the special screw; in order to avoid the functional complications due to the fitting of a bearing into this coupling, it was preferred the solution of a kelly; with a relatively great length *l*, so that the relative rotation between the "connecting rod" and the "link" becames practically negligible;
d) the translation coupling between the link and the fixed elements of the stand (in fact, the guiding blade)

The kinematic analysis of the mechanism was accomplished by means of the application of the closed vectorial contour method and had as an aim to establish its motion law and its comparison to the initial data. Several possible versions have been studied, the presented solution being finally adopted.

As a result of the kinematic analysis, it has been established, taking into account the value of the angle of the leading element φ: the displacement of the "link" (the variation of the distance *AC*), s_x; the variation of the angle between the "connecting rod" and the horizontal axis *AC*, ψ; the variation of the speed, v_x, and of the acceleration, a_x, of the "link".

To this purpose, the vectorial equation

$$\overrightarrow{AB}+\overrightarrow{BC}+\overrightarrow{CA}=0 \ , \tag{1}$$

was projected on two orthogonal directions, from where the following two scalar relations result:

$$-r\,cos\varphi + l\,cos\Psi - s_x = 0 \tag{2a}$$
$$r\,sin\varphi - l\ \ sin\Psi = 0 \tag{2b}$$

Solving the system made up of the equations (2a) and (2b) above, we obtain:

$$s_x \cong r\cdot\left(-cos\,\varphi+\frac{1}{\lambda}\right), \tag{3a}$$
$$\Psi = arcsin(\lambda\ sin\varphi) \tag{3b}$$

where $\lambda = r / l$, ratio that has a value much lower than one, as r = 1 mm and for l a much greater value was finally selected: l = 1000 mm, for which $\lambda = 10^{-3}$. As a result, the value of λ^2 was considered negligible.

By the successive differentiation of the equation (3a) in its simplified form and taking into account the fact that the angular speed of the leading element, ω_1, is constant, we obtain v_x and a_x as follows:

$$v_x = r\,\omega_1 \sin\varphi \tag{4}$$
$$a_x = r\,\omega_1^2 \cos\varphi \tag{5}$$

By applying the relations (3a, 3b, 4, 5) for r = 1 mm, $\lambda = 10^{-3}$, ω_1 = 140 rad/s and the representative values of the angle φ, we obtain the result synthesized in table 1.

Table 1. The results of the kinematic analysis

φ (grade)	0	90	180	270	360
Ψ (grade)	0	0.06	0	- 0.06	0
s_x (mm)	999	1000	1001	1000	999
v_x (m/s)	0	0.14	0	- 0.14	0
a_x (m/s^2)	19.6	0	- 19.6	0	19.6

The results of the accomplished kinematic analysis confirm the fact that, at the right end of the experimental model, firmly fixed to the "link", we obtain an alternating displacement with the required amplitude (±1mm). Furthermore, for the adopted length of the "connecting rod", l = 1000 mm, we obtain very small values for the angle ψ (maximum 0,06 degrees) and therefore there is no need for a joint between the kelly and the bracket.

The maximum calculated values of the speed and acceleration of the "link" are respectively: $v_{x,max}$ = 0.14 m/s and $a_{x,max}$ = 19.6 m/s^2. The latter value is useful for the dynamic analysis of the mechanism aiming at determining the forces and the moments of inertia which appear during its functioning. As the leading element has an invariable angular speed, ω_1, there are no moments of inertia acting upon it. The same is valid for the "connecting rod", due to the very small values of the angle ψ. The maximum value of the force of inertia which appears in the "link" area can be estimated conveniently with the relationship:

$$F_{i,max} = (m_2 + m_3)\ a_{x,max} \approx 0{,}1 \text{ kN} \ . \tag{6}$$

Due to the relatively small value of this force, there is no need for special elements in order to counterbalance the analysed system. In order to establish the minimum power output necessary for the driving motor, the value of the mechanical work, L_m, necessary for the proper functioning of the mechanism, has been calculated. To this aim, it was assessed the value of the propelling force which must be produced by the mechanism at the level of the link, taking into account the mass and the inertia of the elements of the stand which are moving, and the rigidity of the experimental model at the level of the mobile end: $F_m = 4$ kN.

The minimum required power output was then calculated with the relation:

$$P = \frac{L_m}{t} = \frac{F_m \cdot d}{\eta \cdot t}, \tag{7}$$

where t is the minimum duration of a load cycle, whose value corresponds to the maximum admitted frequency of 22Hz, d is the displacement distance of the "link" for a full load cycle (a complete rotation of the leading element), $d = 4$ mm, and η is the efficiency of the mechanism estimated to the value: η = 0.7.

Conclusions

This paper presents the structural and functional description of the stand conceived by the authors to test the vibration resistance and tightness, under symmetrically alternating bending loads of the pipes and fittings joints, manufactured from copper, to be used within natural gas installations.

In order to test the adopted solution for the stand, preliminary testings have been performed by applying the presented methodology, using the experimental models made up of pipes and fittings with a nominal diameter of 15 mm, assembled by pressing.

References

1. Ulmanu, V., Minescu, M., Zisopol, D. G., Dumitrescu, A. - Cercetări privind folosirea ţevilor şi fitingurilor din cupru în instalaţiile de utilizare a gazelor naturale, *Buletinul U.P.G. Ploieşti, Seria Tehnică*, Vol. LVII, Nr. 2/2005, pp. 221-226.
2. Dumitrescu, A., Ulmanu, V., Zisopol, D. G., Minescu, M. - Procedures for the Quality Assessment of the Assembly Technologies of Copper Pipes for Natural Gas Installations, *Buletinul U.P.G. Ploieşti*, *Seria Tehnică*, Vol. LVIII, Nr. 3/2006.
3. * * * VP 614, *Unlösbare Rohrverbindungen für metallene Gasleitungen; Pressverbinder*, DVGW – Deutsche Vereinigung des Gas-und Wasserfaches (e.V. EN 1254, *Copper and copper alloys – Plumbing fittings – part 7: Fittings with press ends for metallic tubes*).
4. * * * *Sanitary and heating systems, Planning and application: Profipress, Profipress XL, Profipress THERM, Profipress G*, First edition, Viega, Germany, May 2001.

Echipament pentru încercarea la încovoiere alternant simetrică a asamblărilor nedemontabile ale ţevilor din cupru

Rezumat

În lucrare se prezintă construcţia şi funcţionarea standului, conceput de către autori, pentru efectuarea încercării de rezistenţă şi etanşeitate la încovoiere alternant - simetrică (vibraţii) a asamblărilor nedemontabile ale ţevilor şi fitingurilor confecţionate din cupru destinate instalaţiilor de distribuţie a gazelor naturale. Încercările efectuate pe standul prezentat sunt cuprinse în procedurile de încercare pentru atestarea calităţii asamblărilor nedemontabile ale ţevilor şi fitingurilor confecţionate din cupru.