PAPER • OPEN ACCESS

Temperature Deformation Influence on the Locking Equipment Destruction in Water Supply Systems at High-Rise Buildings Upper Floors

To cite this article: A Antimonov and N Pushkareva 2020 *IOP Conf. Ser.: Mater. Sci. Eng.* **877** 012046

IOP Conf. Series: Materials Science and Engineering **877** (2020) 012046 doi:10.1088/1757-899X/877/1/012046

Temperature Deformation Influence on the Locking Equipment Destruction in Water Supply Systems at High-Rise Buildings Upper Floors

A Antimonov[1] and N Pushkareva[2*]

[1] Department of Mechanical Engineering, Ural Federal University, Institute of New Materials and Technologies, Mira Street 19, 620002 Ekaterinburg, Russia

[2] Department of Physics, Ural Federal University, Institute of Fundamental Education, Mira Street 19, 620002 Ekaterinburg, Russia

*E-mail: nbpush@mail.ru

Abstract. The locking equipment in water supply systems destruction problem on the upper floors of high-rise buildings is being solved. After destruction products quality study did not reveal defects of a metallographic nature or rejects in manufacture. Therefore, the hypothesis that the destruction cause is low-cycle material fatigue under the dynamic loads influence, arising due to temperature deformations in the pipeline, was proposed. In this paper, we consider a pipeline scheme in which a temperature gradient arises in the fluid flow direction. At different temperature strains of the pipeline parts bending stresses emerge. These stresses cyclic nature is determined by the hot- and cold-water consumption frequency. The pipeline model where the locking equipment was installed is a beam with pinched ends. The bending stresses calculation for this beam was performed using the initial parameter method. Plots of moments and transverse forces in the beam section are plotted. The permissible stress calculation under the dynamic loads action was made. The pipeline section size, within which shut-off equipment can be installed without destruction, was determined.

1. Introduction

The metal structures destruction cause in many cases is material fatigue [1-7]. This phenomenon essence is the appearance and development cyclic loads fatigue crack under the action [8-14]. As the crack grows, the part cross-section gradually decreases. When the stress in this section reaches the ultimate strength, the part suddenly collapses. This process feature is the fact that the calculated static stresses are in the range of elasticity and significantly less than the permissible. Sudden destruction can lead to structural bearing capacity loss and a major technological disaster of varying severity and damage degree [15-18].

2. Problem statement

A significant amount of work has been devoted to problems of strength under the dynamic loads action [19-21]. In many respects, this concerns the high-speed machines operation, in which loads changes with a high frequency. At a low frequency for a relatively short period of time, their action is less dangerous, while with prolonged exposure, structures destruction is also possible. This phenomenon is called low-cycle fatigue [22-25]. The technical systems reliability that, at first glance, are in a static state without visible external forces acting on them, are particularly unpredictable. In this paper, we consider just such case when it was necessary to establish the structural elements destruction cause, which service purpose were no requirements for their bearing capacity. Such objects are brass ball valves, which are used as locking equipment in the water supply systems of high-rise buildings. These faucets body destruction on the upper floors during operation was the cause of flooding the lower floors with hot- or cold-water.

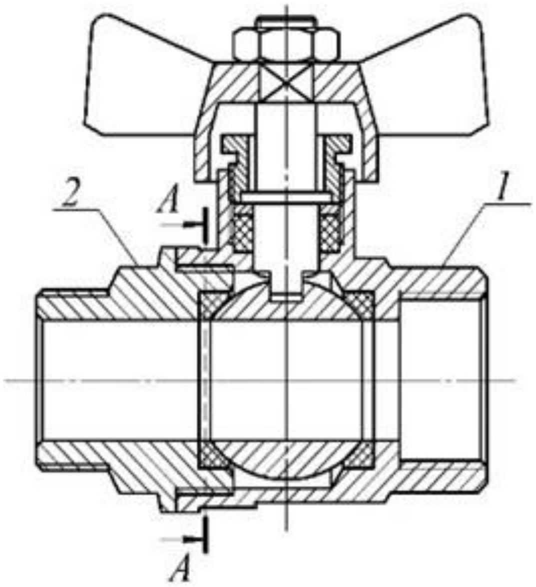

Figure 1. Faucet drawing.

Figure 2. The appearance of the destroyed faucet.

The faucet drawing is shown in ‘Figure 1’. The crane consists of a body 1 and a fitting 2 connected by a thread. The analysis shows that the hazardous section A-A of the crane body is located in the threaded connection area between the body and the fitting. The body wall thickness in this section is 1.5 mm. The destroyed faucet is shown in ‘Figure 2’, from which it follows that the faucet breakdown occurred in this section.

3. Process dynamics

The faucet destruction occurs mainly on the pipelines of the upper-floors some time after the buildings commissioning. This allows to suggests that the failure cause is material fatigue due to dynamic loads that arise in the pipeline thermal deformations case.

A pipeline diagram with a faucet, which is installed at the upper floor level, is presented on ‘Figure 3’. The pipeline consists of two racks H hight and traverse length l_0. Traverse has two sections separated by a faucet, with length l_1 and l_2, which may vary in length. The faucet is used to control the flow of hot or cold-water.

When users begin to make a water consume, a temperature gradient arises in the pipeline in the fluid flow direction. With hot-water supply, one of the racks is heated more strongly than the other. It is known that with fluctuations in temperature, body sizes change.

Therefore, one of the racks lengthens more than the other, and the pipeline cross-section undergoes bending deformation, as shown in Figure 4.

When supplying cold water, the reverse process takes place. One of the racks cools faster than the other and, accordingly, is shortened more. In this case, the pipeline traverse bending deformation also occurs. With a water flow variable mode, the pipe racks temperature periodically changes, and accordingly, deformations and stresses in the pipeline system change over time.

IOP Conf. Series: Materials Science and Engineering **877** (2020) 012046 doi:10.1088/1757-899X/877/1/012046

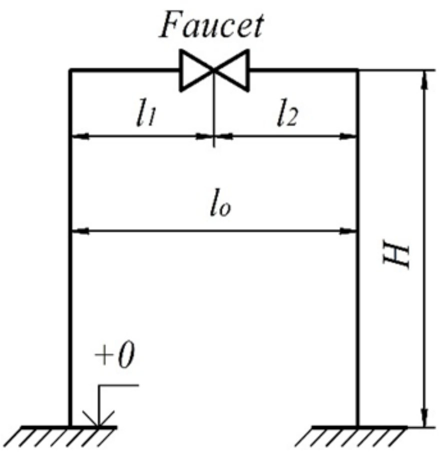

Figure 3. The pipeline scheme.

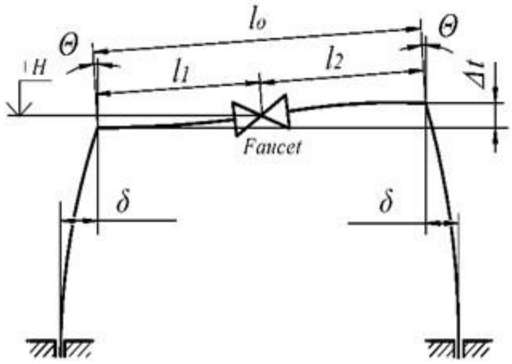

Figure 4. The pipeline deformation scheme.

4. Stress Calculation Method

The bending stresses calculation in the pipeline cross-section is performed for the pipeline without thermal compensators, when the cross-beam fully accepts the loads from the racks temperature deformations. The geometric model of the pipeline for calculating stresses is presented in Figure 5.

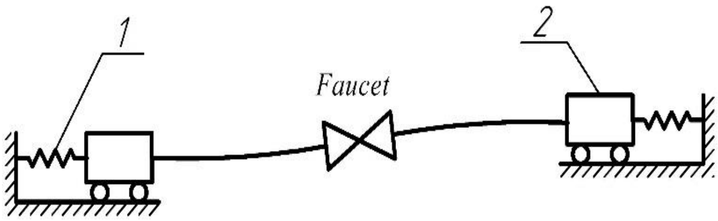

Figure 5. Pipeline model for stress analysis: 1 - elastic element, 2 – roller.

The traverse is presented as a beam with movable pinched ends. The ends mobility is ensured by the rollers movement 1 in the horizontal direction, which simulates the horizontal movement δ the pipe racks ends when they are bent (Figure 4). The elastic elements stiffness 2 is analogous to the stiffness of the uprights bending, which creates tensile stresses in the traverse. In the calculations, we will neglect the elastic elements stiffness, which eliminates the influence on the tensile stress strength in the traverse. We take this effect into account due to an increase in bending stresses, taking the rotation angles sections at the ends of the traverse Θ equal to zero (Figure 4).

Bending stress σ in traverse we define by the formula:

$$\sigma = \frac{M}{W} \tag{1}$$

Where: M – bending moment; W – cross section resistance axial moment.

For a beam with an annular cross section, calculation W produced by the formula:

$$W = \frac{\pi D^3}{32}(1 - \chi^4);\ \chi = \frac{d}{D} \tag{2}$$

Where: D and d – outer and inner diameter of the ring.

To calculate the bending moment, we use the initial parameter method [26]. The reverse diagram in the beam form with pinched ends for calculating stresses is presented in 'figure 6'. Under the external forces action of in the beam cross section, bending moments and transverse forces arise, the which distribution along the beam axis is represented by diagrams $M_x$and Q_x.

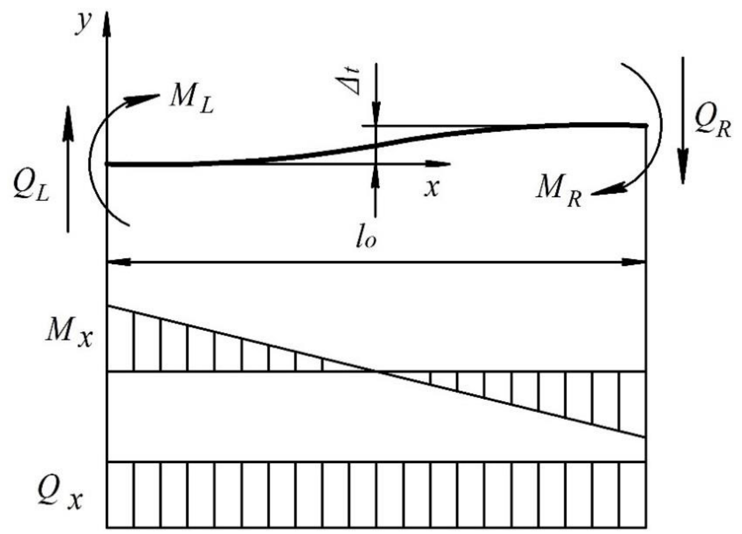

Figure 6. The traverse diagram in the beam form with pinched ends for calculating stresses.

Let's combine the origin with the left end of the traverse. The rotation angle and deflection for its right end are defined as follows:

$$\Theta_R = \Theta_L + \frac{1}{EI}\left(M_L l_0 + Q_L \frac{l_0^2}{2}\right),\ \ y_R = y_L + \Theta_L l_0 + \frac{1}{EI}\left(M_L \frac{l_0^2}{2} + Q_L \frac{l_0^3}{6}\right) \tag{3}$$

Where: Θ_L, Θ_R, y_L, y_R, M_L, Q_L – accordingly, rotation angles, deflections, moments and transverse forces in sections for the left and right ends of the beam; E - elastic modulus; I - inertia axial moment

$$I = \frac{\pi D^4}{64}(1-\chi^4); \chi = \frac{d}{D}. \tag{4}$$

Assuming from the boundary conditions in these equations $\Theta_L = \Theta_R = 0$; $y_L = 0$;$y_R = \Delta t$, where Δt – the difference in the length of the pipe racks, heated to various temperatures, we obtain for the transverse force and bending moment in the cross section at the right end of the beam:

$$Q_L = -\frac{2M_L}{l_0},\ M_L = \frac{2EI\Delta t}{l_0^2} \tag{5}$$

The minus sign in the expression for Q_L means that this force has direction opposite to that chosen in 'figure 6'. Then in an arbitrary pipeline beam section with the coordinate x will haveQ_x:

$$Q_x = Q_L;\ M_x = M_L + Q_L \cdot x \tag{6}$$

Whence for a bending moment at the right end of the pipeline beam when $x = l_0$ we obtain

$$M_R = -\frac{2EI\Delta t}{l_0^2}. \quad (7)$$

Thermal deformation value Δt determined by the formula:

$$\Delta t = \alpha \cdot H \cdot \delta t \quad (8)$$

Where: α – linear expansion coefficient; H – pipeline rack height; δt– racks temperature difference.

Calculation resultsQ_L, M_L, σ and δt depending on the racks without heat compensators temperature difference are presented in table 1.

Table 1. Calculation results Q_L, M_L, σ and δt depending on the racks without heat compensators temperature difference.

Temperature difference in racks δt, degrees	1	5	7	15	20	25	30	35	40
Deformation Δt, mm	0.4	1.9	2.7	5.7	7.5	9.4	11.3	13.2	15
Transverse force Q, N	22	102	144	304	400	501	602	703	799
Bending moment M, Nm	11	51	72	152	200	251	301	352	400
Bending stress σ, MPa	9	40	57	120	158	198	238	278	315

When carrying out the calculations it is accepted: for steel – $\alpha = 125 \cdot 10^{-7} deg^{-1}$. Elastic modulus $E = 1 \cdot 10^5$ MPa, rack height at 10 floors $H = 30$ m; traverse length $l_0 = 1000$ mm; the outer and inner faucet body diameters in a dangerous section are respectively equal $D = 35$ mm, $d = 32$ mm.

5. Fatigue strength calculations.

Regardless of the current load type, the condition for bending strength has the form

$$\sigma \leq [\sigma] \quad (9)$$

Where: σ and $[\sigma]$ – calculated and allowable stresses.

In fatigue calculations, the permissible stress is determined by the following formula [27]:

$$[\sigma] = \frac{2[\sigma_{+1}][\sigma_{-1}]}{(1-r)[\sigma_{+1}]+(1+r)[\sigma_{-1}]} \quad (10)$$

Where: $[\sigma_{+1}]$ – permissible stress under static load; $[\sigma_{-1}]$ – endurance limit for a product with a symmetrical stress cycle taking into account the stress concentration influence, scale factor and surface roughness; r – cycle characteristic defined by the formula

$$r = \frac{\sigma_{min}}{\sigma_{max}} \quad (11)$$

Where: σ_{min} and σ_{max} – minimum and maximum cycle stress.

The stresses in the pipeline traverse vary in time along a pulsating cycle, for which $\sigma_{min} = 0$. Then $r = 0$ and:

$$[\sigma] = \frac{2[\sigma_{+1}][\sigma_{-1}]}{[\sigma_{+1}]+[\sigma_{-1}]} \quad (12)$$

Values $[\sigma_{+1}]$ and $[\sigma_{-1}]$ are determined from the following relations:

IOP Conf. Series: Materials Science and Engineering **877** (2020) 012046 doi:10.1088/1757-899X/877/1/012046

$$[\sigma_{+1}] = \frac{\sigma_T}{n}; [\sigma_{-1}] = \frac{\sigma_{-1}}{k} \quad (13)$$

Where: σ_T– material yield strength; σ_{-1} – material endurance limit for a smooth reference material; n – safety factor under static load; k – coefficient that taking into account the stress concentration influence, scale factor and surface roughness on reducing the fatigue limit.
For most materials, the value of these coefficients is such that $[\sigma_{+1}]$ and $[\sigma_{-1}]$ decrease in relation to σ_T and σ_{-1} on 30-50%. For brass L68, close in chemical composition to brass faucet, we have $\sigma_T = 91$ MPa; $\sigma_{-1} = 120$ MPa [1]. Let's accept a decrease in these values by 40%. Then, rounding off the results, we have $[\sigma_{+1}] = 55$ MPa and $[\sigma_{-1}] = 70$ MPa. As a result, for permissible stresses, we obtain $[\sigma] = 62$MPa. Comparing the calculated stresses from the table with the permissible stress, we find that even with a pipe racks heating temperature difference in 10 degrees, the stress at the beam ends can exceed the permissible.

6. Conclusions

From the bending moment analysis diagram, presented in 'figure 6', it follows that in accordance with the bending moment, the bending stresses vary linearly along the pipeline crosshead length, are equal to zero in the middle and change sign when going through zero. Thus, within the traverse there is a length section l_S with ends equally spaced from the middle where safe stress apply. Faucet installation within this area reduces the chance of its destruction. This section length is determined by the formula:

$$l_S = \frac{[\sigma]}{\sigma_{CE}} l_0 \quad (14)$$

Where: σ_{CE} - calculated stress at the ends of traverse.

The safe section length value, depending on the temperature difference between the racks for the pipe traverse length $l_0 = 1$ m presented in table 2.

Table 2. The safe section length value, depending on the temperature difference between the racks for the pipe traverse length $l_0 = 1$ m.

Rack temperature difference δt, degree	1	5	10	15	20	25	30	35	40
Safe section length l_S, mm	1000	1000	1000	1000	1000	940	775	667	591

From the table it follows that with a temperature difference in the racks up to 7 degrees, the locking equipment can be installed anywhere in the traverse. With a greater temperature difference in the racks, its installation must be done closer to the beam center, and the safest place is its center.

7. References

[1] Ekobori T 1971 *Fizika i mehanika razrushenija i prochnosti tvjordyh tel* [*Physics and mechanics of fracture and solid strength*] (Moscow: Metallurgija) p 264

[2] Ivanova V S 1979 *Razrushenie metallov* [*Metal destruction*] (Moscow: Metallurgija) p 168

[3] Meshkov J J 1981 *Fizicheskie osnovy razrushenija stal'nyh konstrukcij* [*Physical foundations of the destruction of steel structures*] (Kiev: Naukova dumka) p 238

[4] Astaf'ev V I 2001 *Nelinejnaja mehanika razrushenija* [*Nonlinear fracture mechanics*] (Samara: Samara University) p 630

[5] Terent'ev V F and Petuhov A N 2013 *Ustalost' vysokoprochnyh metallicheskih materialov* [*Fatigue of high strength metal materials*] (Moscow: IMET RAN – CIAM) p 515

[6] Cherepanov G P 2012 *Mehanika razrushenija. Serija Matematika i mehanika* [*The mechanics of destruction. Series Mathematics and Mechanics*] (Moscow-Izhevsk: Izdatel'stvo «IKI») p 872
[7] Terent'ev V F and Korableva S A 2015 *Ustalost' metallov* [*Metal fatigue*] (Moscow: Nauka) p 479
[8] Starokon' I V 2012 *J. Sovremennye problemy nauki i obrazovanija* **4** 100-103
[9] Fishgojt A V 2003 *J. Izvestija Rossijskoj akademii nauk. Serija fizicheskaja* **6**(**67**) 881–86
[10] Den'gaeva O V *J. Tehnika i tehnologija* **5** 34–40
[11] Vasjuta B N. 2003 *J. Izvestija vysshih uchebnyh zavedenij. Stroitel'stvo*. **10(538)** 4–13
[12] Shi K, Cai L and Chenet L al 2014 A theoretical model of semi–elliptic surface crack growth. *Chi. J. Aeronaut.* **3**(**27**) 730–34
[13] Fatemi A and Vang L 1998 *Int. J. Fatigue* **20**(**1**) 9–34
[14] Lautrou N, Thevenet D and Cognard J–Y 2009 *J. Fatigue Fract Eng Mater Struct* **32**(**5**) 403–17
[15] Suresh S 2003 *Fatigue of Materials*. 2nd. ed. (Cambridge: Cambridge Uniiversity Press) p 679
[16] Bathias S 2005 *Gigacycle fatigue in mechanical practice* (Yerlag: Magsel Dekker) p 304
[17] Zshbu M and Dzhons D 2010 *Konstrukcionnye materialy. Polnyj kurs. Uchebnoe posobie* [*Construction materials. Full course. Tutorial*] (Dolgoprudnyj:Izdatel'skij dom «Intellekt») p 672
[18] Gilmor R 1984 *Prikladnaja teorija katastrof* [*Applied Catastrophe Theory*] (Moscow: Mir) p 350
[19] Dubrovskij V A and Podvolockaja NI 2005 *J. Problemy mashinostroenija i nadezhnosti mashin* **4** 119-124
[20] Kolesnikov K S 2002 *Problemy dinamiki i prochnosti ispolnitel'nyh mehanizmov i mashin: Materialy nauch. konf* [*Problems of dynamics and strength of actuators and machines*] *In: Proc. of Sci. conf.* (*Astrahan*), (Astrahan': Inform.-izdat. centr AGTU) p 315
[21] Svetlickiy V A and Narajkin O S 2005 *Problemy prikladnoj mehaniki, dinamiki i prochnosti mashin : sbornik statej* [*Problems of applied mechanics, dynamics and strength of machines: collection of articles*] (Moscow: Izd -vo MGTU) p 315
[22] Collins G 1984 *Povrezhdenie materialov v konstrukcijah. Analiz, predskazanie, predotvrashhenie* [*Damage to materials in structures. Analysis, prediction, prevention*] (Moscow: Mir) p 624
[23] Troshhenko V T 2009 *Prochnost' materialov i konstrukcij*. T. 2. *Ustalost' metallov* [*Strength of materials and structures. V 2. Fatigue of metals*] (Kiev: In-t problem prochnosti) p 664
[24] Il'inyh A V 2016 *Malociklovaja ustalost' konstrukcionnyh splavov pri peremennyh parametrah cikla. V sbornike: Mehanika, resurs i diagnostika materialov i konstrukcij Sbornik materialov* [*Low-cycle fatigue of structural alloys with variable cycle parameters. In the collection: Mechanics, resources and diagnostics of materials and structures Materials collection*] (Ekaterinburg: IMASh UrO RAN) pp 292–293
[25] Vansovich K A and Ljagova AA 2015 *J. Gazovaja promyshlennost'* **12(731)** 96–99
[26] Pisarenko G S, Jakovlev A P and Matveev V V 1988 *Spravochnik po soprotivleniju materialov Izd. 2-e, pererab* [*Handbook of materials resistance*] (Kiev: Nauk dumka) p 736
[27] Beljaev N M 2014 *Soprotivlenie materialov: ucheb. posobie* [*Resistance of materials: Tutorial*] (Moscow: Al'jans) p 607