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ONE OF POSSIBILITIES OF INFLUENCE QUANTIFICATION OF VIBRO-ISOLATION ON LIFESPAN OF COMPRESSOR STATION YARD PIPING

JEDNA Z MOŽNOSTÍ KVANTIFIKÁCIE VPLYVU VIBROIZOLÁCIE NA ŽIVOTNOSŤ POTRUBNÝCH DVOROV KOMPRESOROVÝCH STANÍC

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In piping systems of compressor stations there are rigid supporting foots replaced by vibro-isolated elements with metallic shock absorbers. Because significant part of working life of piping systems is already exhausted, the suggested modifications allowed increasing their residual lifetime. Proposed methodology of theoretical and experimental analysis of quantification of vibro-isolation effect permits to tune up very complex dynamical system that without doubt piping system of compressor station represents. Methodology was verified in working conditions of the compressor station and the results of measurements were confirmed by modern numerical methods.

V potrubných systémoch kompresorových staníc sú tuhé podporné pätky nahradené vibroizolačnými prvkami s kovovými absorbérmi rázov. Pretože podstatná časť životnosti potrubného systému je vyčerpaná, navrhnuté úpravy zvyšujú zostatkovú životnosť. Popísaná metodika teoretickej a experimentálnej analýzy kvantifikácie vibroizolačných efektov umožňuje naladiť tento zložitý dynamický systém. Metodika bola overená v pracovných podmienkach kompresorovej stanice a výsledky meraní boli potvrdené modernými numerickými metódami.

Keywords: *vibro-isolation, vibration, piping system, finite element method, tensometry*

Kľúčové slová: vibroizolácia, kmitanie, potrubný systém, metóda konečných prvkov, tenzometria

Introduction

Compressor stations are very important parts of transit gas lines. The compressor station overcomes the gas pressure drop in the pipe and ensures transport of natural gas to one or more delivery points. Consequently, compressor station is critical for the ability of the gas pipeline system to deliver natural gas to the end user. Under normal operating conditions, compressor station engines run 24 hours a day, seven days a week, 365 days a year. While stations vary according to the number and types of engines they use, most compressor stations consist of

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piping, engines, compressors, fuel gas systems, lube oil systems, engine jacket water systems, electrical generators, safety systems, and personnel to maintain and operate these elements.

Operation of compressor station is accompanied by dynamical loading of station yard piping. This is caused by unbalanced rotors of turboblowers, aerodynamical forces, oscillation of pressure, vibration excited by electromagnetic fields, vibration of neighborhood, effects of closed pipes and so on [3, 4]. Even though various protective elements are used on compressor stations for elimination of above described influences (antipumping regulation, placement of piping systems on elastic support foots and so on), those often induce enormous loading in some parts of station yard piping. This significantly influences amount of damage cumulating and accordingly also lifespan of the station yard piping.

Assessment of extremely loaded parts on elements of station yard piping by numerical methods (e.g. FEM) requires perfect knowledge of material characteristics as well as boundary conditions and forces. Because of this there was proposed methodology for experimental determination of internal force quantities [1, 2]. On the base of experimentally defined resultant force quantities in sections which cut part of station yard piping, it was possible to determine reaction forces in support of piping [15, 18] and consecutively by FEM to compute stress fields and their extreme values and positions. Proposed methodology also permits to consider suitability of vibro-isolation elements (metallic dampers) used for elastic support and vibro-isolation of dynamically loaded parts of piping.

In this paper the first results gained by above-mentioned treatment are described.

Using of metallic dampers for vibro-isolation of machines and devices

In operation of machines and devices the dynamic loadings arise in the form of harmonic cycles (Figure 1a), impacts (Figure 1b), or vibrations (Figure 1c).

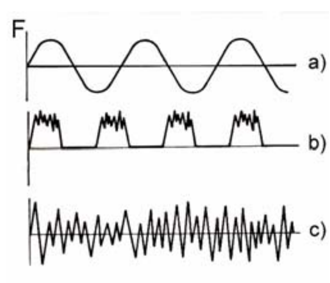


Figure 1. Dynamic loading of machines and devices

The influence of dynamic effects on machines and devices is decreased by vibro-isola-tion realized by elastic support with damping. Elements by which the vibro-isolation is provi-ded are called dampers (or shock absorbers). The dampers can be used for

passive damping (we protect equipment against dynamic influences of neighborhood),

active damping (we decrease dynamic influences of machine in operation).

According to specific cases there are used hydrodynamic dampers, springs with integrated dampers, dampers with elastic members made of plastic or rubber, metallic dampers and so on. Choice of damper type depends on its working conditions and desired parameters of machine or equipment. According to today’s knowledge of operation of compressor stations it is known, that there are no dynamic influences of periodical character there. Station yard piping cannot be considered as discrete linearly damped system excited by solitary forces, but it is a very complex mechanical system with complicated forcing functions. It is necessary to find relations between individual sources of forcing and dynamical response of the system, to develop mathematical model of damping, to use Fourier transformation for finding continuous response spectra of stochastic vibration and so on.

In spite of facts that vibro-isolation elements used for elastic support of station yard piping work in heavy conditions of dynamical loading we use for determination of their vibro-isolation characteristics simple computational model for forced vibration with one degree of freedom (Figure 2). Damper has elastic properties represented by a spring with stiffness k and damping characteristic represented by linear viscous damping coefficient b.

In the case of harmonic forcing function (kinematic excitation is $x_z(t)=0$) by force

$F(t)=F_0 \sin \omega t$, the coefficient of vibro-isolation for steady state is expressed as

$$\xi_R = \frac{R_0}{F_0} = \frac{\sqrt{1+4\,b_p^2\,\eta^2}}{\sqrt{\left(1-\eta^2\right)^2+4\,b_p^2\,\eta^2}}, \tag{1}$$

where R_0 is amplitude of force transmitted to foundation,

$\eta = \dfrac{\omega}{\omega_0}$ - magnification factor,

$\omega_0 = \sqrt{\dfrac{k}{m}}$ - natural frequency of vibration (eigenfrequency of system),

$b_p = \dfrac{b}{2\,m\,\omega_0}$ - damping ratio.

In the case of kinematic harmonic excitation of foundation by displacement $x_z = z_0 \sin \omega t$ (forcing function is $F(t)=0$), for steady state the dynamic coefficient χ_k is defined as ratio of maximal displacement of the body to maximal displacement of the frame

$$\chi_k = \frac{x_{\max}}{z_0} = \frac{\sqrt{1+4\,b_p^2\,\eta^2}}{\sqrt{\left(1-\eta^2\right)^2+4\,b_p^2\,\eta^2}}. \tag{2}$$

Relations $\xi_R = \xi_R(\eta)$ and $\chi_k = \chi_k(\eta)$ given by (1) and (2) represent amplitude characteristics of vibration. Because of $\xi_R = \chi_k$, the amplitude characteristics have identical graphic representation (Figure 3).

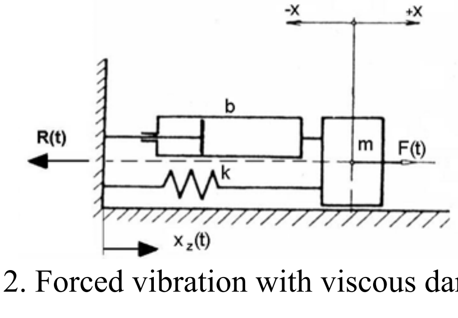


Figure 2. Forced vibration with viscous damping

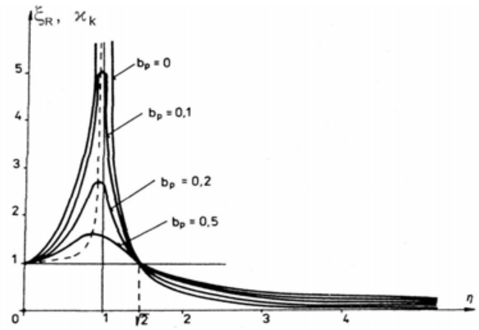


Figure 3 Amplitude characteristics $\xi_R = \xi_R(\eta), \chi_k = \chi_k(\eta)$

Effective damping is ensured under conditions

$$\xi_R \leq 1, \quad \chi_k \leq 1. \tag{3}$$

This is fulfilled for arbitrary damping ratio b_p if

$$\eta = \frac{\omega}{\omega_0} \geq \sqrt{2}\,, \tag{4}$$

or $\omega \geq \sqrt{2}\,\omega_0$.

In order to reach high effectivity of vibro-isolation, it is suitable to use for damping the overdamped area with magnification factor $\eta \rangle \sqrt{2}$, so $\omega \rangle \sqrt{2}\,\omega_0$, where ω is frequency of excitation force and ω_0 is eigenfrequency of the system.

For the most cases is chosen relation $\omega = (3 \div 5)\omega_0$, so efficiency of vibro-isolation is expressed by approximate equation

$$\psi = \frac{\eta^2 - 2}{\eta^2 - 1} \cdot 100\% \tag{5}$$

and efficiency reaches 80-90% [8, 20, 22, 23].

Theoretically the highest efficiency is obtained by using of elastic members without damping (see χ_k, ξ_R in Figure 3), but damping expressed by quantity b_p is needed to cross safely through resonant frequency. In most cases damping of dynamic effects in operation of machines and devices is provided by metallic dampers (Figure 4).

Elastic member of metallic damper is made of CrNi – steel wire. Wire is tressed, wavily, coiled and pressed into desired shape (Figure 4) [5, 6, 7].

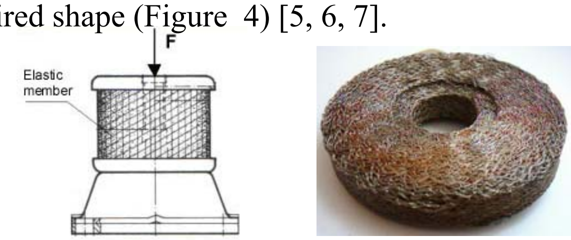


Figure 4. Metallic damper

Stiffness characteristic of metallic damper is shown in Figure 5. It has progressive development, at the beginning is parabolic and then has a sharper increase of force to certain value of compression length $x_{\max}$.

Comparable advantages to linear characteristic of classical spring are:

- small compression length for possible high dynamical overloading,
- stable position of device in spite of elastic support,
- approximately constant eigenfrequency for a wide range of loadings.

Damping is caused by friction between wires of elastic member and damping ratio reaches values 0.15-0.20. The factor of magnification by transition through resonant frequency is then at most 5. Typical shapes of curves $\chi_k = \chi_k(\eta)$ and $\xi_R = \xi_R(\eta)$ for metallic damper are given on Figure 6.

Figure 5. Spring characteristic of metallic damper

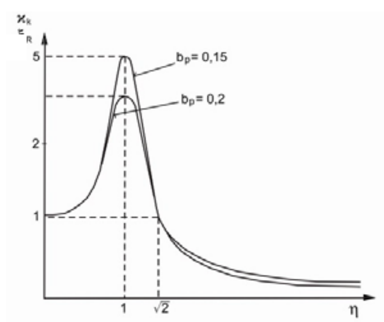


Figure 6. Typical shapes of curves $\chi_k = \chi_k(\eta)$, $\xi_R = \xi_R(\eta)$ for metallic damper

Metallic dampers have also further advantages:

- they have practically unlimited fatigue lifespan,
- their parameters are constant during whole lifespan (without hardening, no plastic deformation, no material aging),
- resistance against temperature influences in range from $-90^0 C$ up to $400^0 C$,
- possibility to overload by dynamic forces that are 5-10 times greater than static forces,

- resistance against influence of the environment (fluids, acids, oils, …).

Moreover, another advantage of using metallic dampers is the possibility to change eigenfrequency by prestress in elastic member and so tune up the system to appropriate dynamical regime.

Upon impact (Figure 1b), there is during very short period of time τ change of exciting force $F(t)$ (or position of frame $x_z(t)$). The aim of isolation is to transform high impact energy of short force impulse $F(t)$, (Figure 7), on longer movement of elastic system with lower value of force level $R(t)$ transmitted to foundation.

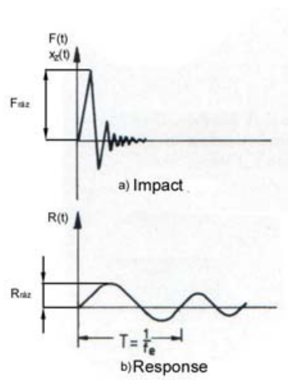


Figure 7. Impact and its response

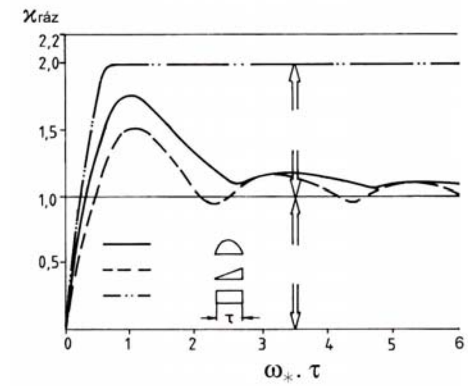


Figure 8. Impact spectrum of vibrated system

Very important parameters by impact isolation are stiffness k^* of elastic support by impact, damping ratio b^* by impact and mass m of isolated body.

The response of the system on impact is characterized by impact spectrum of vibrated system. It represents dependence of maximal force $R_{ráz}$ in foundation under impact on eigenfrequency of vibrated system by impact $\omega_*^2 = k^*/m$, on vibration period, on impulse time τ, or on arbitrary similar parameter. Analytical formulation of this dependence is practically impossible. That is reason why the impact spectrums are represented by graphs or tables. Often it is defined so-called impact coefficient by relation

$$\chi_{ráz} = \frac{R_{ráz}}{F_{ráz}} . \tag{6}$$

Figure 8 shows impact spectrum of vibrated system according to Figure 2 under impact impulse in shape of half sine function, triangle and rectangle. On the figure are expressed dependences of impact coefficient $\chi_{ráz}$ on parameter $\omega_* \cdot \tau$. It results from the Figure 8, that in order to guarantee effective isolation against impacts $(\chi_{ráz} \langle 1)$ it has to be valid inequality $\omega_* \cdot \tau \langle 0{,}25$, that means we have to ensure low level of impact stiffness k^* of the damped member (small ω_*). In the case, there are repeated impacts in the system, in addition to small stiffness k^* it has to be used value of (b^*) big enough, in order to eliminate free vibration after impact to the beginning of the next one.

Application of vibro-isolation elements on compressor station yard piping

Recently, there were carried out reconstructions of supports members on compressor stations, mainly whole station yard piping. On the base of their experience, entrepreneurs of compressor stations come to the result that material of pipes and also other parts of supporting elements exhausted significant part of its lifespan. In order to ensure further safe operation it was necessary to provide steps by which the loading of station yard piping will be decreased. One improvement relied on using elastic supports of station yard piping on foots with metallic dampers. Example of supporting foot of pipe with two metallic vibro-isolation dampers is on Figure 9 and its detail is shown in Figure 10. It must be stressed, that for good operation of dampers it is necessary to use exactly technique of assembly described in [10].

Figure 9. Supporting foot of pipe with metallic dampers

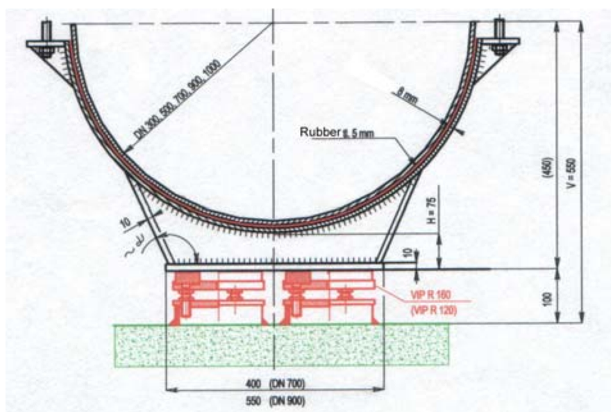


Figure 10. Detail of vibro-isolation support

Aim of the next step was to reduce stress concentrators in areas of saddle supports. Bandages with rubber-textile belt replaced them and they caused that transverse section of pipe was stabilized with local constraining shell vibrations of pipe. In Figure 11 is shown working characteristic of vibro-isolation element that is used for elastic imposition of pipe on supporting foots. Tuning is accomplished by screws that allowed regulation of prestress and by this appropriate working range.

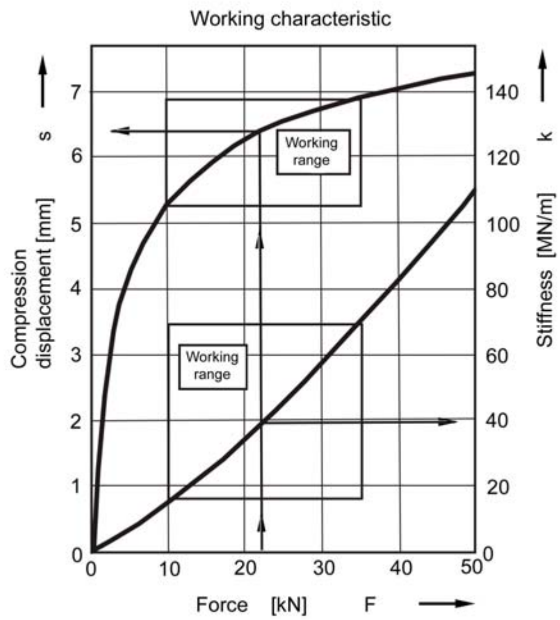


Figure 11. Working characteristic of vibro-isolation element

Methodology for determitation of internal stress quantities in pipes and reaction forces in supporting foots

Methodology for numerical-experimental determination of stresses in pipes serves for determination of residual lifespan of station yard piping [2, 11, 13, 14, 15, 16, 17]. It can identify effect of time-dependent working load (disconnection and switching the equipment on, fluctuation of pressure in pipe and so on) on magnitudes and directions of vectors of bending moment, transversal force (shear), axial (normal) force, and torsion moment in cross-section of pipe.

Assessment of fatigue lifespan in critical segment is possible if we know

- shape and dimensions of pipe in monitored segment,
- actual values of material characteristics,
- time-dependent working load and resultant internal stress quantities on both sides of monitored segment of pipe.

The greatest problems causes assessment of time-depended changes of internal stresses mentioned at the third point above, because during operation of piping there are many working influences (fluctuation of pressure, forces influenced by friction in supports, forces caused by "subside effect" of soil that results in additional bending and torsion, differences between projected parameters and real state, self-weight forces and so on).

Suggested method exploits experimental determination of stress components $\sigma_i, \tau_i \quad i = 1,2,3$ by strain gauge rosettes placed along perimeter of measured cross-section of pipe at angle interval 120^0 (Figure 12). On the base of stress components σ_i, τ_i there is a possibility to determine magnitudes and directions of internal stress quantities at the area of measurement. Locality of measurement is chosen in adequate distance from possible stress concentrators, so we assume plane stress on the surface of pipe induced by internal pressure p, axial force N, bending moment M, transversal force (shear) Q and torsion moment M_k.

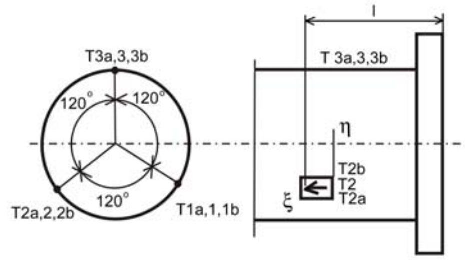


Figure 12. Location of strain gauge rosettes along perimeter of measured cross-section of pipe

From strains ε measured by strain gauge rosettes it is possible to compute normal stresses σ_{vi}, shear stresses τ_i in the areas in question.

We can write relations

$$\sigma_{vi} = E\,\varepsilon_i + \mu \frac{p}{2}\frac{1+n}{1-n}, \quad i = 1,2,3 \tag{7}$$

$$\tau_i = G(\varepsilon_{ia} - \varepsilon_{ib}), \quad i = 1,2,3 \tag{8}$$

where

E, G, μ are Young modulus, shear modulus and Poisson's ratio, respectively, for material of pipe,

$n = \frac{d}{D}$ - ratio of internal diameter d and external diameter D of pipe,

p - internal pressure in pipe,

ε_i - strains in axial direction of pipe,

$\varepsilon_{ia}, \varepsilon_{ib}$ - strains in directions rotated 45^0 around the longitudinal axis of pipe.

Axial force in pipe is

$$N = \frac{\pi D^2}{4}\left(1-n^2\right)\left[\frac{\sigma_{v1}+\sigma_{v2}+\sigma_{v3}}{3} - \mu\, p \frac{1+n}{1-n}\right]. \tag{9}$$

Bending moment is

$$M = \frac{\pi D^3}{32}\left(1-n^4\right)\frac{2\sigma_{v1}-\sigma_{v2}-\sigma_{v3}}{3\sin\alpha}, \tag{10}$$

where α is declination angle of the bending moment vector related to radial vector to strain gage T_1 (Figure 12) and oriented against the direction of numbering of strain gages.

Therefore, we can write

$$tg\alpha = \frac{2\sigma_{v1} - \sigma_{v2} - \sigma_{v3}}{\sqrt{3}\left(\sigma_{v2} - \sigma_{v3}\right)}. \tag{11}$$

Transversal force Q in cross-section area is computed by equation

$$Q = \frac{\pi D^2}{16} \frac{\left(1 - n^4\right)\left(1 - n\right)}{\left(1 - n^3\right)\cos\beta} \cdot \left(2\tau_1 - \tau_2 - \tau_3\right), \tag{12}$$

where $tg\beta = \sqrt{3}\,\dfrac{\tau_3 - \tau_2}{2\tau_1 - \tau_2 - \tau_3}$, (13)

β is angle between line perpendicular to the vector Q and oriented in the same direction as α (see Figure 13).

For bending moment we have equation

$$M_k = \frac{\pi D^3}{16}\left(1 - n^4\right)\frac{1}{3}\left(\tau_1 + \tau_2 + \tau_3\right). \tag{14}$$

Positive orientations of internal quantities N, M, Q, M_k as well as angles α, β are seen in Figure 13, where straight segment of pipe with supporting foot and two cross-sections A, B is depicted.

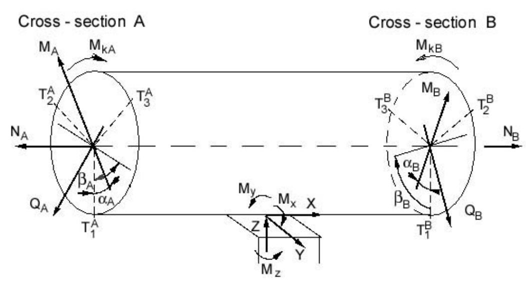


Figure 13. Straight segment of pipe with supporting foot and cross-sections A, B

From equilibrium conditions for the segment we can compute time-dependent components of reactions $X(t), Y(t), Z(t), M_x(t), M_y(t), M_z(t)$ because magnitudes of axial forces, bending moments, transversal forces and torsion moments together with their directions are resultant quantities determined from strain gage measurement in two cross-sections A, B [12, 19, 21].

In case, there is a branch pipe on the main straight pipe, we use for determination of functions $X(t), Y(t), Z(t), M_x(t), M_y(t), M_z(t)$ equilibrium conditions for the segment with supporting foot, where the segment has three cross-sections.

Application of suggested methodology

Methodology described for determination of resultant internal force quantities in piping and computation of reaction forces in supporting foots was applied on the discharge side (high pressure side) of station yard piping. In Figure 14 are described positions of cross-sections A, B, C together with strain gages applied on the discharge side of piping.

Figure 14. Positions of cross-sections A, B, C and strain gages applied on the discharge side of piping

Strain gage measurements were realized in various time intervals. During these intervals there were changed operating conditions on the discharge side of piping by turning on and off various turbo devices, closing gate valves and so on. Examples of measured time-depended axial stresses $\sigma_{vi}(i = 1,2,3)$ are in Figure 15 and in Figure 16 are values of bending moments M in cross-section A.

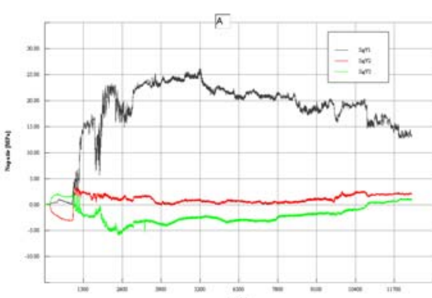


Figure 15.Time-depended axial stresses $\sigma_{vi}(i = 1,2,3)$ in cross-section A

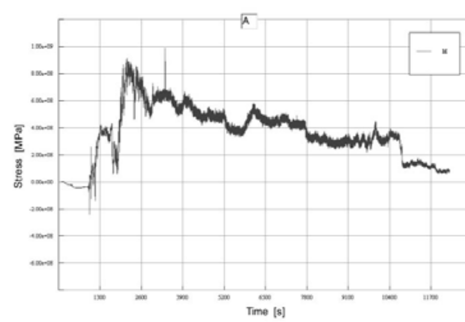
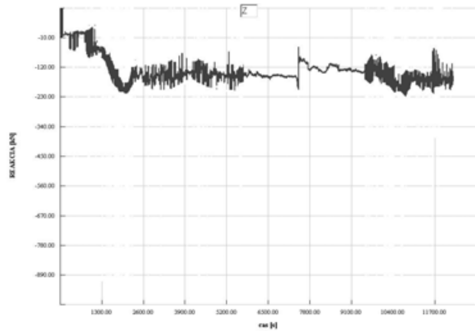


Figure 16.Time-depended bending moment M in cross-section A

In Figure 17 is shown time-dependent component of reaction Z and in Figure 18 are seen time-dependent components of reaction M_x. Both reactions are in supporting foot of piping.



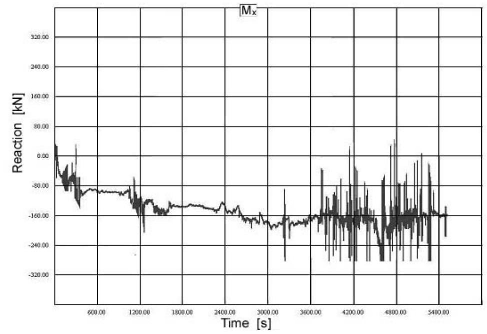
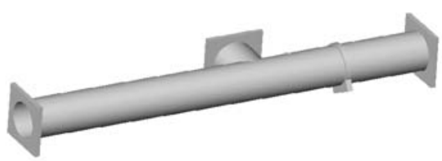


Figure 17.Time-dependent reaction Z

Figure 18.Time-dependent reaction M_x

Using of the finite element method for determination of stresses from resultant internal force quantities

For a chosen time parameter of operation of compressor station stresses in piping were computed by the finite element method. In Figure 19 is shown model of one segment of piping with cross-cestions A, B, C. In Figure 20 is detail of supporting foot and bandage with rubber-textile belt.



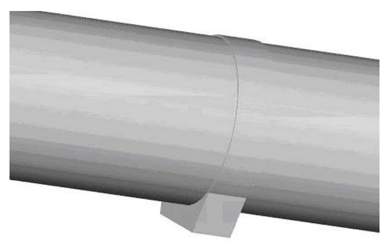


Figure 19.Model of pipe segment

Figure 20.Detail of foot and bandage

Mesh of finite elements developed according to model from Figure 19 is shown in Figure 21. Mesh consists of 17 773 three-dimensional elements and 35 503 nodes.

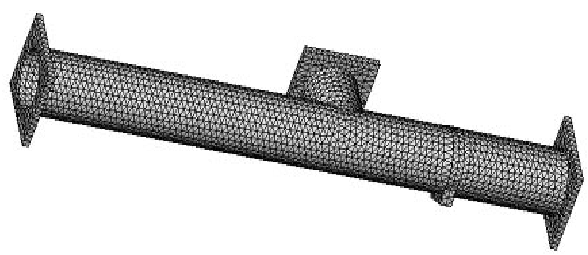


Figure 21.Mesh of finite elements

Fields of equivalent von Mises stresses are in Figs. 22, 23, 24.

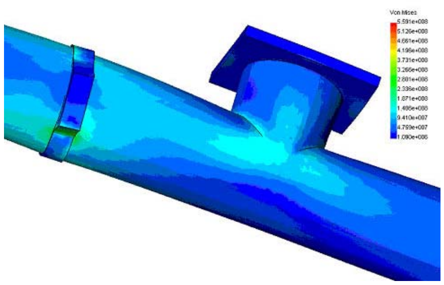


Figure 22. Field of equivalent stresses

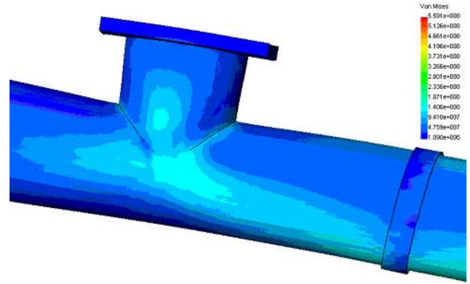


Figure 23. Field of equivalent stresses

Very interesting is detail of stresses and deformations in areas where the pipe is supported (see Figure 25). The deformations in this figure are excessively magnified and it is seen that the belt and pipe have no contact along whole area. One side is loaded more than the other and accordingly the adjacent elastic member has high values of stresses.

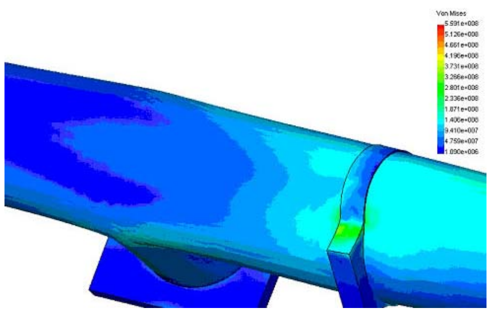


Figure 24. Field of equivalent stresses

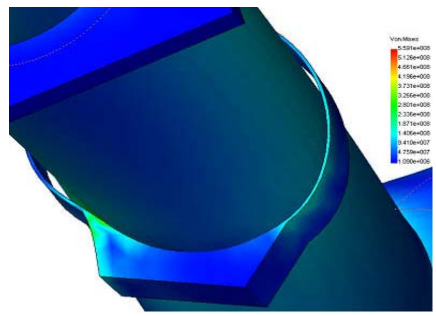


Figure 25. Detail of stresses in foot

Conclusion

On the base of theoretical analysis of vibro-isolation, determination of dynamic coefficient defined as ratio of maximal displacement of the body to maximal displacement of the frame, coefficient of vibro-isolation expressed as ratio of dynamical force transmitted into foundation and excitation force, from analysis of phase and amplitude characteristics it can be stated that conditions of effective damping are from the theoretical point of view ideally fulfilled for chosen metallic dampers. The measurements made by strain gages and from these determined resultant internal force values and reactions in supporting foots, however, documented that these values are much more higher than the values from working characteristic of vibro-isolation element. Deformation analysis of given pipe segment and shape of deformed bandage clearly shows that the pressure on two vibro-isolation elements of the same supporting foot is during operation different. It again increases shifting of working points on one element to points (values) outside of working characteristic of vibro-isolation element.

From detailed analysis of time-dependent resultant internal force quantities - components of reactions as well as stresses - and from equivalent stresses in critical points of station yard piping it can be declared that:

- suggested methodology enables unequivocally define resultant internal force quantities and their time behavior in given cross-sections,

- time dependent quantities in cross-section are consecutively dependences for stresses,
- after measurement in two or three cross-sections we can compute by the finite element method stresses in the pipe,
- suggested methodology enables to determine not only influence of supports on loading, but also to assess behavior of station yard piping as a whole.

Objectification can be provided by changing of support at the same place and by quantification of its influence through resultant internal force quantities, reaction forces and stresses determined by the finite element method.

Vibro-isolation elements by which we replace existing supports should decrease loading because significant part of lifespan of piping system is exhausted. Method suggested above and accordingly realized this aim fulfils, but there are still opened problems there which has to be solved.

It is very important to set up appropriately prestress in springs in order to decrease loads in carried elements of station yard piping to minimum. Criterion according to which the reactions were justified according to self-weight forces cannot be consider as optimal. In addition to this, reactions were justified on some supports for force 80 kN that is outside of working characteristic given in documentation. Tests of working characteristics in our laboratory show that slope of curve by increasing of force over 30-35 kN is nearly constant. As show the results of measurement, reaction force Z on support exceeds without moment influence value 50 kN on one vibro-isolation element.

Material of pipe has according to our measurements the following characteristics: yield stress 391-426 MPa and fracture stress 555 Mpa. From the finite element computations is seen that maximal stresses are near to yield stress. Presented fact invokes necessity to decrease load and on the other hand leads entrepreneurs of compressor stations to correct decisions concerning changes of supporting elements.

If we consider piping on saddle support without bandage and elastic element, than the values of reduced stresses in critical points are higher than yield stress, which can be documented by [9], [18] a [21].

In conclusion can be stated, that methodology suggested by working team allowed unequivocally to rate influence of changes in supports of station yard piping and to quantify changes of operational parameters on reaction forces in supports. Using of numerical methods allow to prepare animation of deformations and to evaluate effect of changes on lifespan of station yard piping. By theoretical analysis, experimental measurement and further evaluation there was concluded correctness of changes in supports and using of vibro-isolation elements. Analysis also shows that it is needed to pay extreme attention to the prestress and accordingly to set up appropriately reaction forces in supports.

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