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Quantification of Force Effects in Dynamically Loaded Pipe Systems

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Abstract In the assessment process of compressor station pipe yard lifespan it is necessary to taken into account many facts that are connected with instant operation conditions and history of loading. In the paper are described possibilities of methods of experimental mechanics for verification of safe operation of chosen compressor station pipe yard elements.

Keywords: *experimental methods of mechanics, safe operation, dynamic loading*

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1. 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 work in non-stop regimes. While stations vary according to the number and types of engines they use, most compressor stations consist of 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 Figure 1. This is caused by unbalanced rotors of turbo blowers, aerodynamic forces, oscillation of pressure, vibration excited by electromagnetic fields, vibration of neighbourhood, effects of closed pipes and so on [1,2,3]. 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.

In the paper is described procedure for identification of possible failures on the pipe branch leading to compressors on the pipe yard of compressor station.

2. Causes of Pipe Yard Vibrations of Compressor Stations

As results from existing studies, initial reason of unwanted vibration of pipe yards of some compressor stations are resonance pressure vibrations of gas in the branches of pipe collectors [4]. Analytical and numerical analysis has shown that the sources of vibrations are invoked pressure oscillations in the T-tubes Figure 2 by the gas stream that under certain velocity and dimension conditions in the critical locations lead to significantly amplified periodical resonance acoustic (pressure) oscillations in the blind (closed) pipe branches. The energy and amplitudes of such pressure impacts that are connected with so-called Helmholtz standing acoustic wave are big enough to cause vibration of a part of pipe yard in the critical location. This effect makes matters worse in the case of parallel branches of identical lengths (i.e. volumes with identical resonance acoustic frequency) that are connected to a pipe with streaming medium. Qualitatively, the reason of vibration is similar for all compressor stations (quantitatively, the critical frequencies are different on individual compressor stations due to differences in lengths of collector branch pipes and other geometric parameters).

Figure 1. Pipe yard of compressor station

Figure 2. T-tube on the pipe

With respect to complex boundary conditions on the pipe yards (different support methods Figure 3a, b, c, different positioning of pipe with respect to soil background and so on) there was made decision to provide vibration measurement on the most exposed locations of the pipe yard.

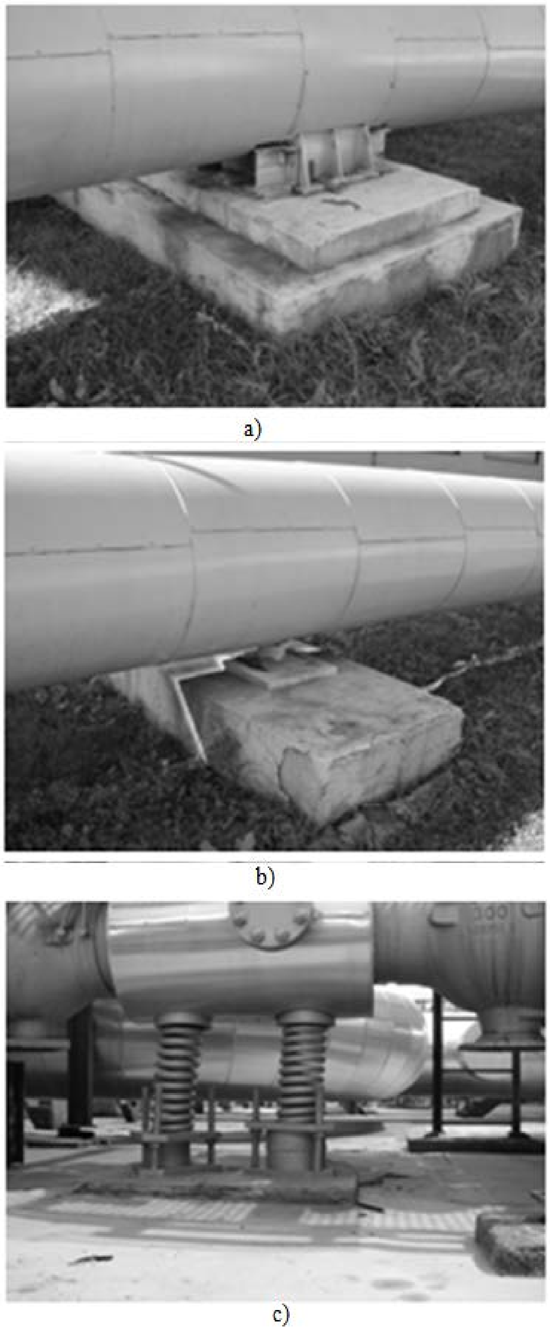

Figure 3. Different types of pipe positioning. a) fixed, b) on the elastic vibroisolation washer, c) on the helical springs

For the measurement was used measurement system PULSE 6 [5] and one or three component acceleration sensors positioned in the chosen locations of pipe yard. During the measurement the velocity of gas in the pipe was continuously changed. Extreme vibrations invoked by pulsation were registered for gas velocities that corresponds to the range of critical stream velocity of gas connected with self-exciting Helmholtz standing wave in the locations of closed T-tube branches of a gas pipe.

As a prove of standing wave initiation in closed branches can be considered measured time Figure 4a and frequency Figure 4b dependency of acceleration in T-tube.

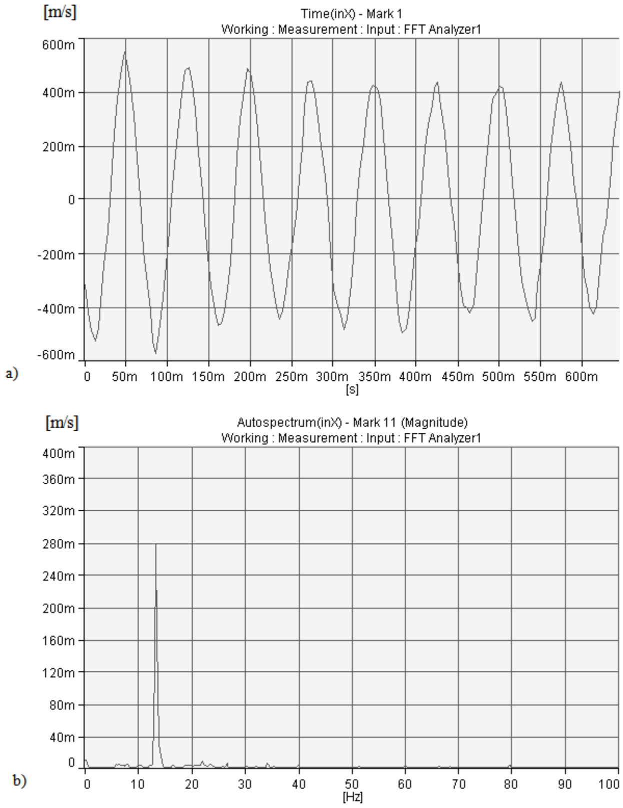

Figure 4. Standing waves on the T-tube. a) time-dependent chart of acceleration, b) frequency dependence of acceleration

Maximum stress amplitude in the pipe under resonance frequency of vibration reached value approximately 42 MPa while according to [4] for given types of pipe branches maximum stress amplitude can reach magnitude up to 90 MPa.

3. Methodology for Determination of Internal Forces in Pipes and Reaction Forces in Supporting Feet

Methodology for numerical-experimental determination of internal forces in pipes serves for determination of residual lifespan of station yard piping [6,7,8,9]. It can identify effect of time-dependent working load (shut down and switch on the equipment, fluctuation of pressure in pipe and so on) on magnitudes and directions of vectors of bending moment, transversal force (shear), axial (normal) force, and torsional 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 forces on both sides of monitored segment of pipe,
- loading history.

The main problems are connected with assessment of the time-dependent changes of internal forces 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 uses the experimental determination of stress components $\sigma_i, \tau_i \quad i = 1,2,3$ by strain gauge rosettes placed along perimeter of the measured cross-section of pipe at angle interval 120^0 (see Figure 5). On the basis of the stress components σ_i, τ_i, it is possible to determine magnitudes and directions of internal forces at the area of measurement. Position of the measurement devices 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 .

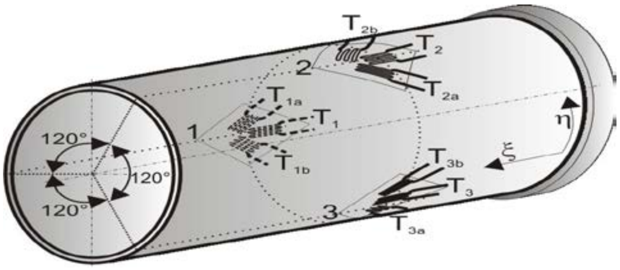

Figure 5. Location of strain gauge rosettes along perimeter of measured cross-section of pipe

From strains ε_i measured by strain gauge rosettes (see Figure 5 it is possible to compute normal stresses σ_{vi} , shear stresses τ_i axial force *N*, bending moment *M*, transversal force (shear) *Q* and torsion moment M_k in the regions of interest [6,8].

Positive orientations of internal quantities N, M, Q, M_k are depicted in Figure 6, where straight segment of pipe with supporting foot and two cross-sections A, B is shown.

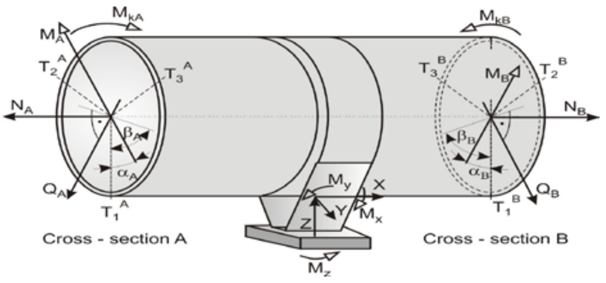

Figure 6. Straight segment of pipe with supporting foot and cross-sections *A*, *B*

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 gauge measurement in two cross-sections A, B .

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 of pipe has three cross-sections.

4. Application of Suggested Methodology

The methodology described above for determination of resultant internal force quantities in piping and computation of reaction forces in supporting feet was applied to the discharge side (high pressure side) of station yard piping. In Figure 7 are shown the positions of the cross-sections A, B, C together with the strain gauges applied to the discharge side of piping.

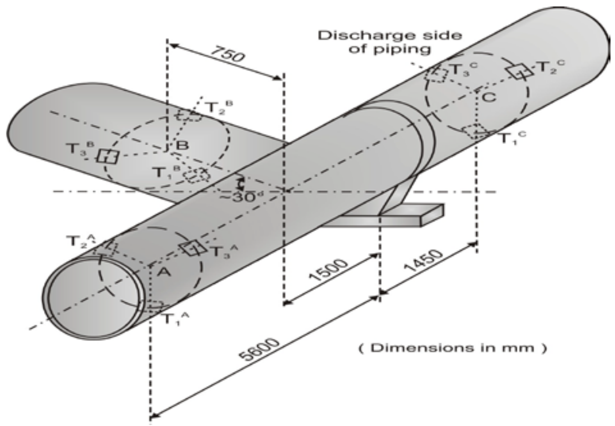

Figure 7. Positions of cross-sections *A*, *B*, *C* and the strain gauges applied to the discharge side of piping

Strain gauge 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-dependent axial stresses $\sigma_{vi}\left(i = 1,2,3\right)$ are shown in Figure 8 while in Figure 9 are values of bending moments *M* in the cross-section *A*.

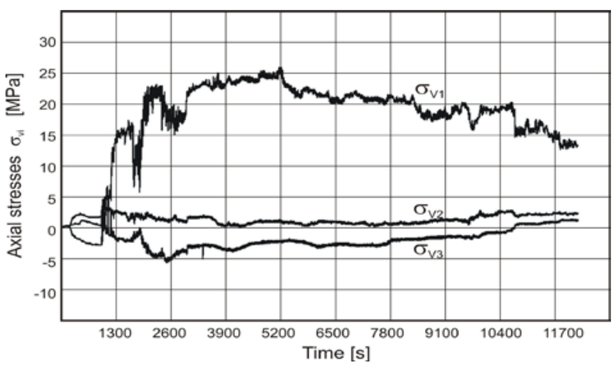

Figure 8. Time-dependent axial stresses $\sigma_{vi}\left(i = 1,2,3\right)$ in cross-section A

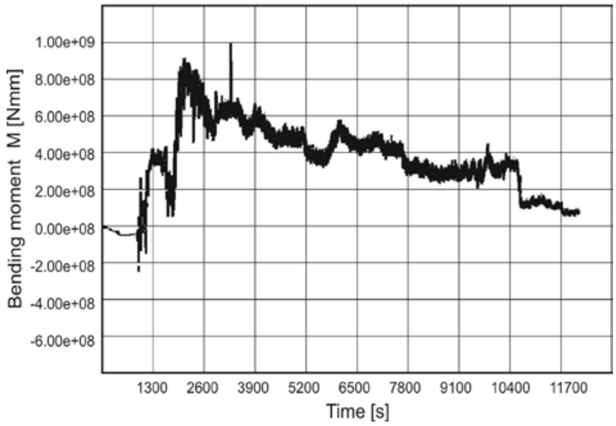

Figure 9. Time-dependent bending moment *M* in cross-section *A*

In Figure 10 is shown time-dependent component of reaction Z while in Figure 11 are seen time-dependent components of reaction M_x . Both reactions are in supporting foot of piping.

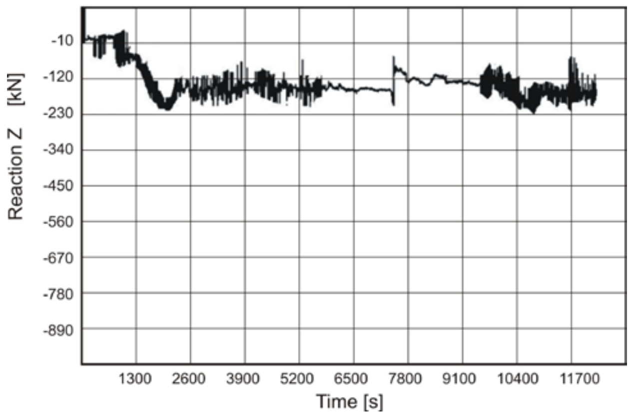

Figure 10. Time-dependent reaction *Z*

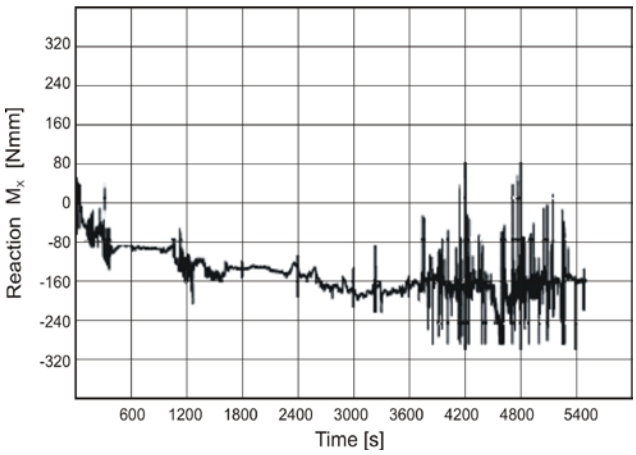

Figure 11. Time-dependent reaction M_x

5. Using 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 12 is shown model of one segment of piping with cross-sections A, B, C . In Figure 13 is detail of supporting foot and bandage with rubber-textile belt.

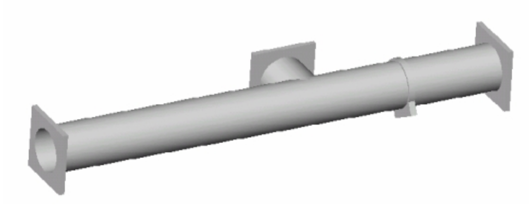

Figure 12. Model of pipe segment

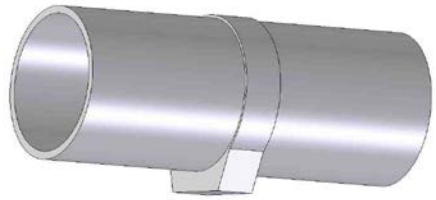

Figure 13. Detail of foot and bandage

In Figure 14, Figure 15 are the fields of equivalent von Mises stresses. Very interesting is detail of stress values and deformations in the areas where the pipe is supported (see Figure 16). The deformations in this figure are excessively magnified and it can be seen that the belt and the pipe have no contact along whole area. The one side is loaded more than the other and accordingly the adjacent elastic member has high values of stresses.

Figure 14. Field of equivalent stresses

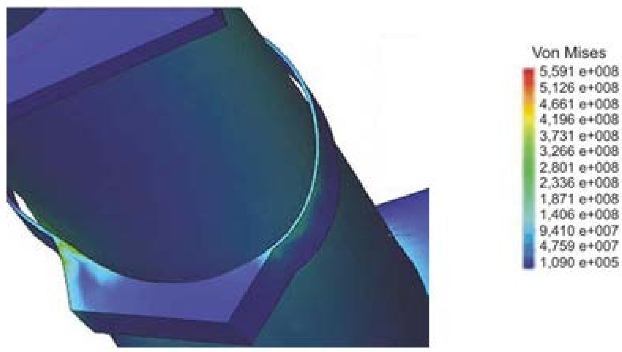

Figure 15. Detail of stresses in foot

6. Conclusions

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 behaviour in given cross-sections,

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

The objectification can be carried out 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 the loading because significant part of lifespan of piping system is exhausted. The method suggested above and accordingly realized fulfils this aims, but there are still opened problems there which has to be solved.

Acknowledgement

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