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Experimental Stress Analysis

Verification Possibilities of Safe Operation of Pipe Yards on Compressor Stations by the Methods of Experimental Mechanics

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Abstract: In the assessment process of compressor station pipe yard lifespan 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, Residual stresses

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 (Fig. 1). This is caused by unbalanced rotors of turboblowers, aerodynamic forces, oscillation of pressure, vibration excited by electromagnetic fields, vibration of neighborhood, effects of closed pipes and so on [1]. 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.

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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 [2]. Analytical and numerical analysis has shown that the sources of vibrations are by the gas stream invoked pressure oscillations in the T-tubes (Fig. 2) 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).

Fig. 1. Pipe yard of compressor station

Fig. 2. T-tube on the pipe

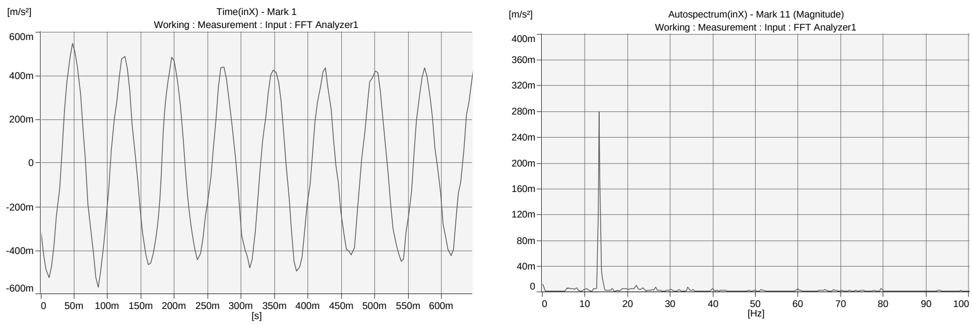
With respect to complex boundary conditions on the pipe yards (different support methods (Fig. 3a, b, c), different positioning of pipe into soil and so on) there was made decision to provide vibration measurement on the most exposed locations of the pipe yard.

a) b) c)

Fig. 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 [3] 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 (Fig. 4a) and frequency (Fig. 4b) dependency of acceleration in T-tube.



a) b)

Fig. 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 [2] for given types of pipe branches maximum stress amplitude can reach magnitude up to 90 MPa.

3. Time-dependent increments of integral internal force quantities in cross-sections of pipe yard

For determination of time-dependent increments of integral internal force quantities in the cross-sections of pipe yards was chosen location with a T-tube on the delivery pipe (Fig. 2). In the location of T-tube in the axial direction were applied in every from cross-sections A, B, C three strain-gages, with angle 120° to each other (Fig. 5).

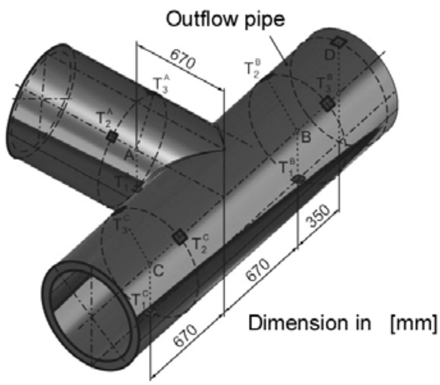


Fig. 5. Location of strain-gages on the T-tube

The measured strains ε_1, ε_2, ε_3 in locations 1, 2, 3 of individual cross-sections A, B, C of T-tube (Fig. 5) allow us to determine axial forces N, bending moments M and their directions in cross-sections A, B, C [4].

As the previous strain-gage measurements on the compressor stations showed that the stresses caused by shear forces and torsion moments are small in examined cross-sections (order of several MPa), those were not determined during measurement. In Fig.6a, b are given for illustration examples of time-dependent charts of internal force quantities N and M in cross-sections A, B, C of T- tube.

From the measurement on the pressured pipe results that for increment of internal pressure 7.35 MPa the circumferential stress has magnitude 112 MPa and axial stress magnitude 52 MPa.

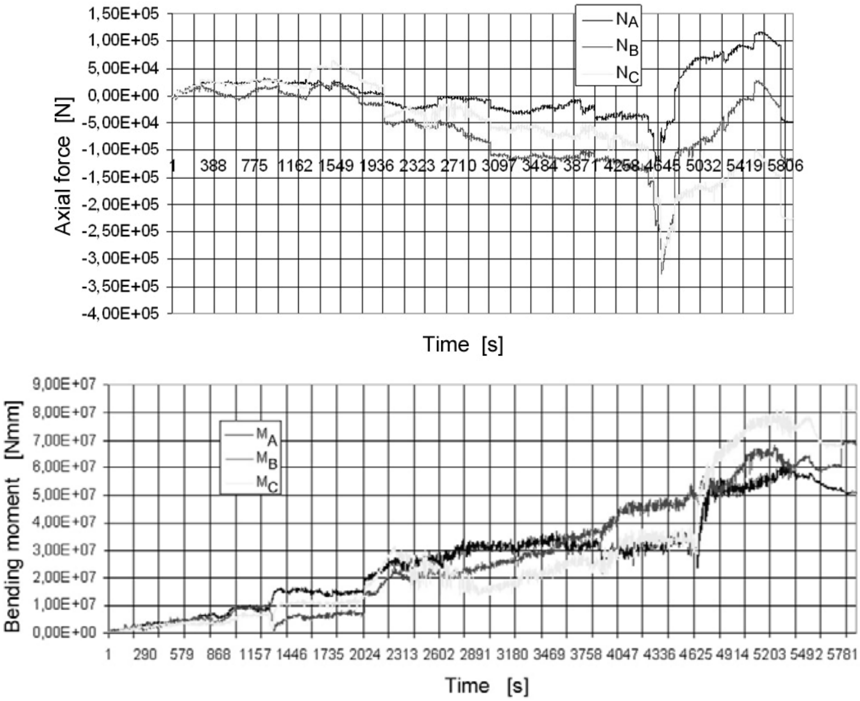


Fig. 6. Time-dependent increments of internal forces in cross-sections A,B,C for axial forces N, and bending moments M

4. Residual stresses in the pipes of pipe yard

Measurement of magnitudes and directions of principal stresses was accomplished by the hole-drilling method [5] on the specimen taken from T-tube (Fig. 7), which was damaged by breakdown.

Fig. 7. Location of material sampling for measurement of residual stresses

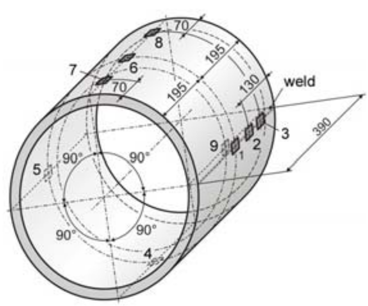


Fig. 8. Positioning of strain-gages for the hole-drilling method

Quantification of magnitudes and directions of principal residual stresses was accomplished in equidistant four locations on outer perimeter of pipe (locations 1 to 8) and in one location on the inner side of pipe in the axis of axial weld (location 9) (Fig. 8). The magnitudes of principal residual stresses and their directions defined by angle φ of declination σ_{max} from circumferential direction are determined according to Standard ASTM E 837-01 [6] for individual locations of measurement. Maximum principal residual stresses on outer surface of specimen are tensile and they reach in circumferential direction magnitudes approximately 130 to 140 MPa, and in axial direction magnitudes approximately 65 to 85 MPa.

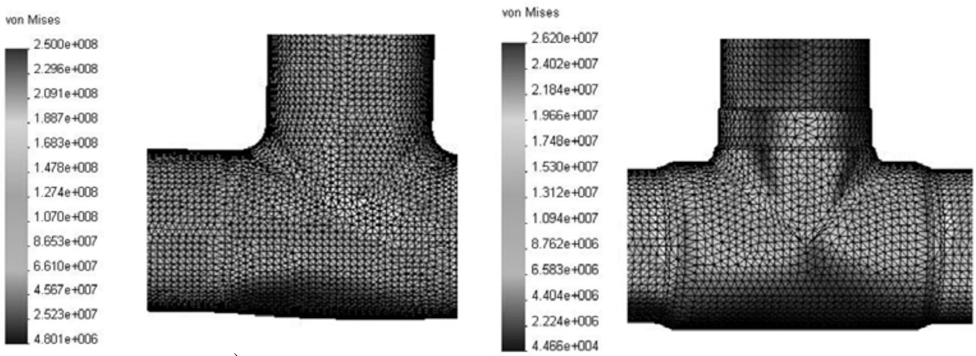
5. Determination of stress concentrations in T-tubes by the finite element method

The computation of internal force quantities in pipe yards of compressor station and analysis of stress concentration in T-tubes was carried out by the finite element method.

Two computational models that represent two types of pipes connections used in pipe yards were analyzed. Computation of stress fields was realized for two cases:

- loading of T-tubes by inner pressure 7.35 MPa,
- loading of T-tubes by internal force quantities determined experimentally in locations of cross-sections while in agreement with above-mentioned, there were applied only internal bending moments and axial forces.

In Fig. 9a) is shown field of equivalent stresses in the most loaded locations of T-tube I under internal pressure while maximum stress magnitude was 207 MPa for T-tube I and 206 MPa for T-tube II. In Fig. 9b) is field of equivalent stresses in T-tube II loaded by internal bending moments and axial forces while the maximum stresses are 22.2 MPa for T-tube I and 24.0 MPa for T-tube II. The analysis concludes that for chosen load cases can be from the point of view of stress concentration both T-tubes considered to be equivalent in spite of differences in geometry and dimensions. Given values can be considered as stress amplitudes resulting from internal pressure as well as from initiated vibration during operation of pipe yard.



a) b)

Fig. 9. Equivalent stresses in the most loaded locations of T-tubes. a) loading by internal pressure – T-tube I, b) loading by internal force quantities in locations of cross-sections – T-tube II

6. Assessment of pipe yard safe operation

The parameters of mechanical properties of material were determined from specimen taken from T-tube (Fig. 7). In the frame of analysis were realized the following tests:

- static tensile tests of basic material and material of weld connection,

- tests for determination of Young modulus,
- Charpy flexural impact test (notch of type Charpy V),
- tensile fatigue tests of basic material as well as welded connection,
- flat bending fatigue tests of welded connection,
- metallographic analysis.

According to producer the pipe was made of material St 52-3 (equivalent steel 11 524.1), thickness 22.2 mm. It corresponds to steel S 355 J2 according to EN 100 25-2:2004 with mechanical properties $R_{e\ \min}$ = 345 MPa, R_m = 470 – 630 MPa, $A_{5\ \min}$ = 20%, for specimens in transverse direction to direction of rolling, Charpy impact energy KV = 27 J is prescribed for specimen in longitudinal direction under 20ºC. The mechanical tests gives the following results: R_e = 345 – 350 MPa, R_m = 533 – 545 MPa, A_5 = 28,6 – 34,0 %, KV = 75 – 77 J transverse direction, KV = 145 – 155 J longitudinal direction, so that all values prescribed by the Standard were fulfilled. The value of Young modulus determined by experimental measurement was in interval (2.091 – 2.175) . 10^5 MPa, which slightly exceeds value 2.06 . 10^5 MPa given in the Standard. The fatigue tests were realized for pulsed tensile loading and alternating symmetric flat bending. In accordance with State research institute in Prague [7] for the determination of fatigue limit can be used empirical relations from which results values σ_C = 192 MPa for alternating symmetrical tensile-pressure test, σ_{hC} = 336 MPa for vanishing tensile test, and σ_{Co} = 240 MPa for alternating symmetric bending.

Fatigue tests were accomplished in two stages. In the first one were chosen stresses on the level of fatigue limit of material. Pulsing tensile loading had average stress σ_m = 250 MPa (value was determined by summation of residual stress and internal pressure) and stress amplitude σ_a = 90 MPa (maximum stress amplitude of dynamic loading of pipe during operation determined in [2]). The stress amplitude for alternating symmetric bending was chosen to be $\sigma_{ao} = \sigma_{Co}$ = 240 MPa. Neither from both specimens was damaged under such loading for $2.6.10^6$ cycles. At the second stage of fatigue tests were chosen for pulsing tensile loading stresses σ_m = 360 MPa, σ_a = 90 MPa, while the average stress σ_m respects, in contrast to the first stage, stress concentration in T-tubes. The specimens without weld were damaged under approximately 9.10^5 cycles, specimens with the weld under approximately 8.10^4 cycles. Situation for fatigue tests with pulsing tensile loading and both stages is depicted in Smith diagram in Fig. 10.

As results from Smith diagram, the second stage represents high deformation fatigue, because in every loading cycle plastic deformation of material occurred. In the second stage the tests with pulsing symmetric loading were realized with stress amplitude σ_{ao} = 450 MPa that corresponds to magnitude of upper stress for test with pulsing tensile loading. The specimens were damaged after approximately 8.10^3 cycles. Fatigue tensile and bending tests showed that the basic material as well as welding metal has satisfactorily good fatigue properties and they fulfill criteria of State research institute in Prague. The tests last without failure 2.6 .10^6 cycles under stresses that define limit states for fatigue.

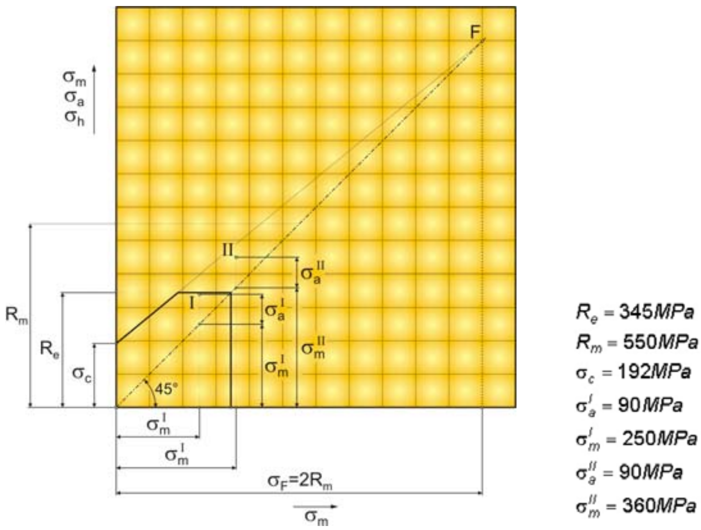


Fig. 10. Smith diagram for the material of pipe

7. Conclusions

In the paper is described analysis of possible reasons of failures that occurred on branch pipes leading to compressors on pipe yard of compressor station. On the basis of reached results can be stated :

- vibration measurements in exposed locations of pipe yard showed that the resonant states are invoked by self-exciting Helmholtz standing waves in locations of closed (blind) branches of gas pipes and they are initiated by valve closing after stopping of corresponding turbo-compressor; self-excited vibration is result of fact that critical gas stream velocity lay inside working gas velocities during operation,
- experimentally determined maximum amplitudes of stress changes in the pipe resulting from pressure changing and pressure impacts reaches magnitude approximately 42 MPa,
- internal pressure 7.35 MPa invokes circumferential stress 112 MPa and axial stress 52 MPa in the pipe,
- residual tensile stresses on the outer pipe surface reached in circumferential directions 130 – 140 MPa and in axial directions 65 - 85 MPa,
- analysis by the finite element method showed that maximum equivalent stresses in T-tubes due to internal pressure 7.35 MPa reach 207 MPa and influence of operation and vibration invokes additional magnitudes about 24 MPa,
- analysis of mechanical properties of pipe including impact and fatigue tests showed that material of pipe fulfill all demands for static, impact and fatigue properties according to relevant standards and regulations,
- on the basis of analysis can be stated that probable reason of branch pipe failure were uncontrolled resonance states invoked by self-excited Helmholtz standing waves that initiate inadmissible levels of dynamic loading,

- decreasing of pipe yard loading can be reached by e.g. change of pipe supports (vibro-isolation washer), regulation of gas stream velocity in the pipes, optimization of working regimes of turbo-compressors and so on.

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