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Strength of Materials, Vol. 42, No. 1, 2010

INFORMATION TECHNOLOGIES FOR VIBRATION STRENGTH ANALYSIS OF THE ROVENSKAYA NUCLEAR POWER PLANT MAIN STEAM LINE

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UDC 373:534:621.6

Sources of vibration in the main steam line have been determined during steady-state operation of the power-generating unit. Amplitude and frequency of the pressure pulsations in the heat carrier and behavior of forced vibrations of the pipeline system have been assessed using mathematical simulation methods.

Keywords: vibrations in steam pipeline, pressure pulsations, mathematical modeling.

General Formulation of the Problem, Scientific Tasks and Practical Applications. This work is concerned with development of design plans and specifications necessary to replace the main steam safety valves (Fig. 1) by new pulse type valves.

Design solutions involving implementation of new pulse valves in the "closed-loop" steam exhaust pipelines imply changes in the dynamic behavior of these due to anchor support displacement along the "closed-loop" pipe axis and appearance of loose valves. This necessitates assessment of the general pipeline vibration strength at the design stage.

The analysis of the available results of investigations shows that vibration strength evaluation for these pipelines in the design stage is not provided, moreover, it is difficult to carry out any corresponding experiments [1–3]. Therefore, design information technologies were used in this study for assessing vibration strength for steam pipelines [4]. Investigation technique has been developed which consisted of the following two stages: analysis of hydrodynamic impact on the steam pipeline and calculation of vibrations in the main steam line under unsteady-state loading conditions.

Gas dynamic effects arise from pressure pulsations of the heat carrier that are determined by mathematical modeling of the unsteady-state steam flow in the pipeline. Flow Vision software [3] is used for performing computational simulations.

As far as steady-state steam flow in the pipeline was considered, it would appear reasonable to assume that pressure pulsations are mainly caused by pipeline sections with structural deviations (pipe T-joints, gate valves, valves, etc.).

Figure 2 shows the computational model corresponding to flow channel of the pipeline with "closed-loop" steam exhaust pipe. The model was generated using graphical package PK KOMPAS-3D V8.

In order to describe steam flow behavior in the pipeline, we have used an analytical model for compressible gas, which is applicable for steam flow simulation at any Mach numbers (sub-, trans-, super-, and hypersonic flows). Within framework of this model, flow velocities and steam pressure values, as well as turbulent heat carrier flow distribution, have been assessed for a specific geometric volume.

For identification of the origin of the pressure pulsations in the flow channel under study (Fig. 2) we have analyzed variation of turbulent steam flow in time. The calculations show that turbulent energy eddy forms at the leading edge of the lower pipe T-joint connected to the steam exhaust piping (Fig. 3), travels to the trailing edge of

Open Joint-Stock Company "Kiev Research and Design Institute 'ÉNERGOPROEKT'," Kiev, Ukraine.
Translated from Problemy Prochnosti, No. 1, pp. 165 – 170, January – February, 2010. Original article submitted June 21, 2009.

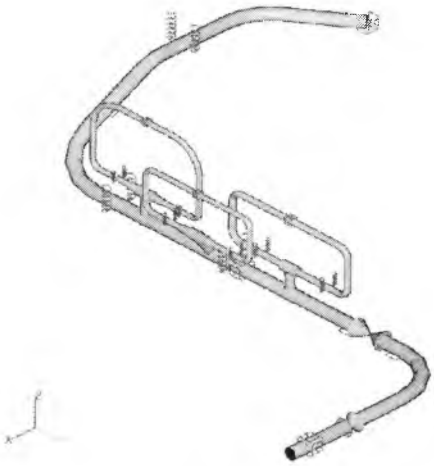

Fig. 1

Fig. 2

Fig. 1. Scheme of the main steam line system TKh50 of "Rovenskaya" nuclear power plant.
Fig. 2. Geometric model of flow channel for the steam exhaust pipeline (fragment).

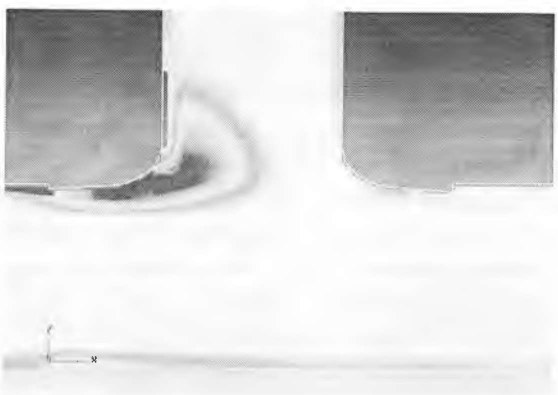

Fig. 3. Occurrence of the turbulent eddy.

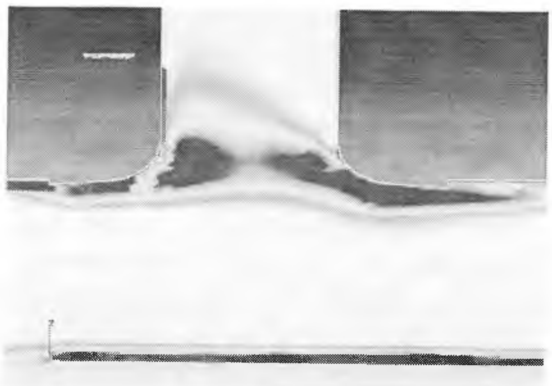

Fig. 4. Formation of the turbulent eddy.

Fig. 5. Separation of the turbulent eddy.

the pipe T-joint (Fig. 4), separates from it (Fig. 5) and follows the steam flow with further dissipation of energy. Then the next turbulent eddy is formed likewise and its kinetics is repeated. The calculations confirm that this effect repeats with time interval of 0.05 s.

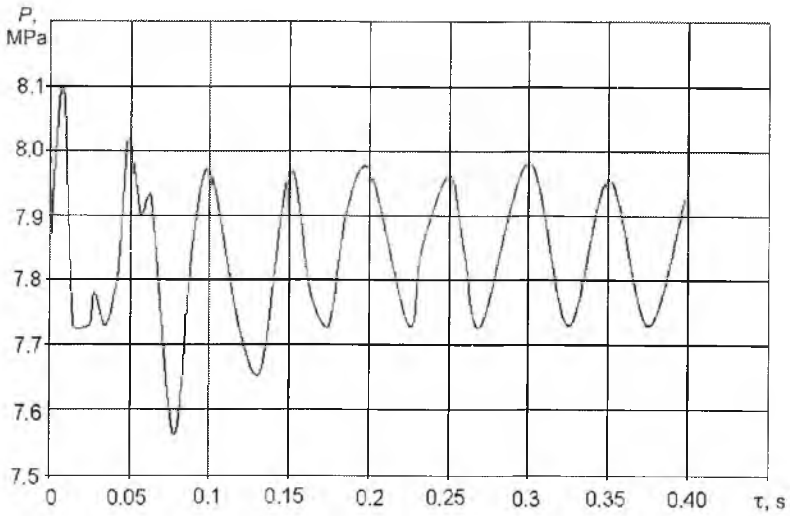

Fig. 6. Variation of pressure P in the lower pipe T-joint with time τ.

Analysis of the numerical results shows that the specified time interval depends on design and technological capabilities of the "closed-loop" steam exhaust pipe. In region of turbulent eddy separation/detachment (Fig. 5), pressure pulsation is observed, which induces cyclic steam flow through the "closed-loop" steam pipeline and gives rise to pressure pulsations in the main steam line. Periodic pressure wave propagation through the "closed-loop" steam exhaust pipe results in pressure pulse variation in its vertical and horizontal sections. Pressure variation is observed in the lower pipe T-joint with time interval of 0.05 s, while the average amplitude is 0.12 MPa (Fig. 6).

To determine possible sources of steam pressure pulsations in the elbows of the main pipeline (Fig. 1), numerical simulation of heat carrier flow with specified parameters (pressure 7.85 MPa and temperature 294°C) was performed. Heat carrier flow velocity, distribution of turbulent heat carrier flow and pressure were determined within framework of the geometrical model shown in Fig. 1.

It has been established that flow turbulization is induced by the difference in velocities of the steam flowing through the elbow. Turbulent energy dissipation has been observed on the inner and outer surfaces of the 90°-elbow (in the steam flow direction). The locations of turbulization origins were stable, while no turbulent eddy separation/detachment was observed. This implies the absence of pressure pulsation sources in such elbows.

In order to assess the pressure pulse decay behavior in the main steam line, the test problem with pressure pulse propagation of 0.012 s duration and 0.2 MPa amplitude along the pipe of 10 m length was analyzed. Results of the numerical simulations performed show that at a distance of 5 m from excitation source significant pressure pulse reduction (90%) due to energy dissipation is observed. This implies that pressure pulse in the range from 5 to 8% of its basic magnitude will further propagate along the pipeline.

We have performed vibrostrength assessment based on several standards (VDI European Criterion, 2004; ASME OMa S/G-2000 Standard; STUK, Finland; Russian Peak Vibration Velocity Standard for steam and hot water pipelines, etc.), according to which the absolute acceptance criteria relating to the peak vibration velocity were defined.

The forth integration method for equations of motion in form of the digital model SCAD [5] was used for measuring vibration displacements, velocities and accelerations of the pipeline under study (Fig. 1).

Two types of loading were considered: static (structure weight) and dynamic (exposure to pressure pulsations from the heat carrier).

The analysis of normal mode and frequency of the pipeline vibrations was performed and, as a result, a very narrow frequency range was obtained, which eliminates the possibility of tuning the forcing frequency 20 Hz.

The calculations were carried out for two variants of the design model for the pipeline: before and after replacement of the main steam safety valves.

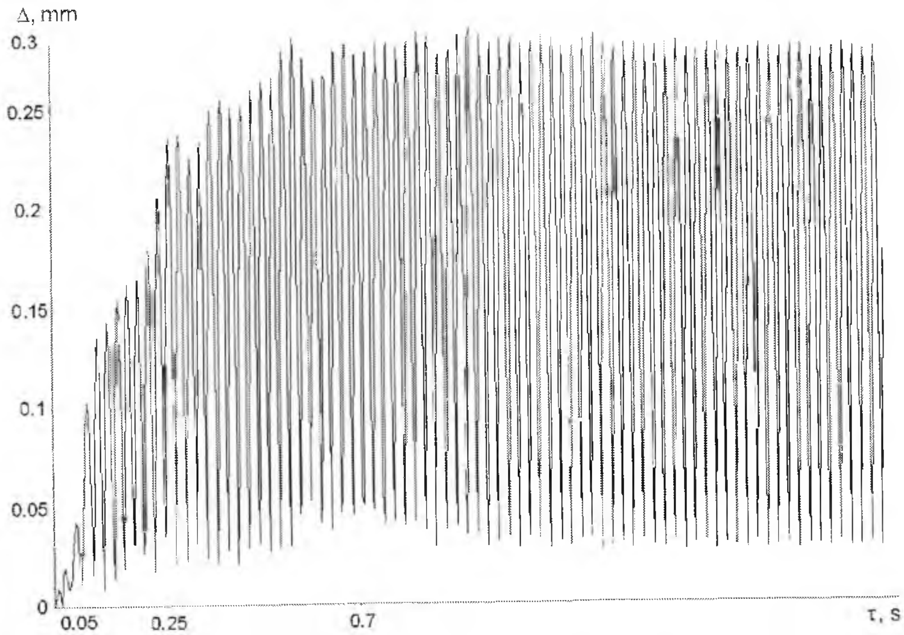

Fig. 7. Vibration displacements Δ of a "dangerous point" of the steam exhaust pipeline.

The analysis of forced vibrations shows that the highest values of velocity and acceleration are observed in the "closed-loop" steam exhaust pipe, whereas the respective (the most dangerous) pipe locations are in the middle of the left section of the "closed-loop" pipe.

A typical diagram of displacements for a "dangerous point" in the left section of the "closed-loop" pipe is shown in Fig. 7.

The analysis of the numerical simulation results demonstrates that before replacement of the main steam safety valves average vibration velocity of dangerous points in the "closed-loop" pipe is 15 mm/s and after replacement = 17.4 mm/s. Therefore, vibration of the "closed-loop" pipe is increased by a factor of 1.2.

Noteworthy is that the calculated vibration velocity values are higher than limit values specified in the standard (12–15 mm/s). Measures reducing vibrations of "closed-loop" pipeline during steady-state operation of the power-generating unit are expedient.

CONCLUSIONS

1. It was found that sources of pressure pulsations are observed at the base of the "closed-loop" steam exhaust pipe; characteristics of pressure pulsations and pressure pulse decay were determined.

2. Modal analysis was performed and, as a result, a narrow frequency range that eliminates the possibility of tuning the forcing frequency was obtained.

3. Dynamic forces caused by steam pressure pulsations were calculated.

4. The analysis of the forced vibrations has been performed. Vibration displacements, velocities, and accelerations of the main steam line points were calculated. The most vibroactive zone in the pipeline has been identified.

5. It was found that design solutions involving replacement of the main steam safety valves would deteriorate the vibration state of "closed-loop" pipeline.

6. Since the calculated values are higher than the limit ones, it is expedient to envisage some special measures to reduce vibrations of "closed-loop" pipelines.

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