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LRE Axial Booster Pump Hydrodynamic Vibration Source

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Abstract: Reduction of hydrodynamic vibration in the pump units of the fuel supply system is crucial for improving the reliability and service life of a Liquid propellant Rocket Engine (LRE). Pressure pulsations in the pump working path is the source of hydrodynamic vibration. It is useful to model pressure pulsations in the flow path and hydrodynamic vibration of the pump casing during the early stages of design. The study reports numerical modeling of three-dimensional non-stationary turbulent flow in a liquid-propellant axial screw booster pump unit. The spectral composition of pressure pulsations in the flowpath of the pump unit is analyzed. It is shown that the spectrum of pressure pulsations at the exit of the working screw impeller consists of tonal components at the Blade Passing Frequency (BPF) and its higher harmonics. At the entrance to the screw the spectrum of pressure pulsations is dominated by tonal components at the rotor frequency and BPF.

Key words: Axial screw pump, pressure pulsations, finite volume method, LRE booster pump forced oscillations, BPF

INTRODUCTION

LRE bladed pump operation is accompanied by pressure pulsations in flow ducts and by vibrations of structure members. Great attention is paid to measurements and control of a level of pressure pulsations and vibrations both in operation and during development stage. Pressure pulsations and vibrations bear information on the dynamic stresses the pump structure members are subjected to. Dynamic stresses imposed on the effective static stresses result in fatigue failure of structure members. The higher are static and dynamic stresses, the more likely is the fatigue failure of a structure member and the shorter is non-failure operation life of the pump unit. A unique dependence between dynamic stresses, acting on specific members and measured pressure pulsations and vibrations is usually unknown. A permissible level of pressure pulsations and vibrations is determined experimentally in bench tests. Many studies both in Russia and abroad were dedicated to investigation of working fluid pressure pulsations, vibration and noise of centrifugal and other bladed pumps. The experimental data have shown direct relationship between pressure pulsations in the pump flow path and its vibration-noise characteristics. The presence of intensive pressure pulsations is typical for all types of bladed pumps. Under certain conditions the pressure pulsations in the pump outlet, for example, may reach the values hazardous for the structure integrity. The study of pressure pulsations in the pump cavity gives information on dynamic loads, acting on structure members. The calculation of amplitudes of pressure pulsations in bladed pumps at the early development stage is an urgent need.

Vibration of hydrodynamic nature is caused by the peculiarities of pump fluid flow in the flow parth (Pokrovsky and Yudin, 1966; Borovsky *et al*., 1975): non-stationary interaction of the flow, leaving the rotor blade passages with an outlet device, vortex formation including small-scale turbulence and large-scale vortex structures (reverse flows), cavitation processes in the pump flow path.

Spectral composition of pressure pulsations, vibration and noise in the bladed pump is represented by broadband background and by well-defined discrete components whose level generally determines pump vibration activity and dynamic deformation of structure elements.

These oscillations can be of hazard in case of coincidence with natural resonant frequencies of structural elements. In an actual structure, it is necessary to take into consideration “Integral” stator (pump case) response to the action of pressure pulsations that is non-stationary pressure field within a volume of the pump flow part. This problem was described in detail as applied to the pumps of propulsive devices.

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Research objective: High-speed screw-centrifugal pumps (Ovsyannikov and Borovsky, 1986) and booster axial screw pumps are used to feed fuel components to the combustion chamberin heavy-duty LREs. The turbo-pump units are the chief source of pressure pulsations and vibration in the propulsion system. One such booster pump unit was studied with a view to increasing its reliability and service life. Pressure pulsations in the working fluid delivered by the pump generate disturbance forces that leading to mechanical vibration of the components of the pump casing. Numerical modeling of the three-dimensional non-stationary flow was carried using the FlowVision Software package (CAPVIDIA, 2007) in order todetermine amplitude of the pulsations in the A working fluid pressure which arise in the flow path of the booster pump. This requires the construction of a three-dimensional geometric model of the main elements of the flow path. The model, Fig. 1 consists of a rotor and a stator. The pump stator is represented by the internal wall of the pump casing, to which are connected eight pylons and guide output vanes. The rotor of the pump is a three-blades screw with a hub of variable diameter and the turbine disk attached to the exit shroud ring of the screw.

The geometrical model takes account of radial clearance between the pump casing and the screw rotor and the computations set boundary conditions for the model operation mode of the booster pump during water tests. Following accepted model conditions, static pressure at the entrance to the pumpis set at one atmosphere and a flow rate of 324 kg/sec is set at the pump exit (the computations assumes that the working fluid is water). A “moving body” interface which provides rotation of the screw at 5720 rpm is used for computations. Following usual practice in modeling of three-dimensional non-stationary turbulent flow in a pump, gas phase and leakage through the seals are not taken into account.

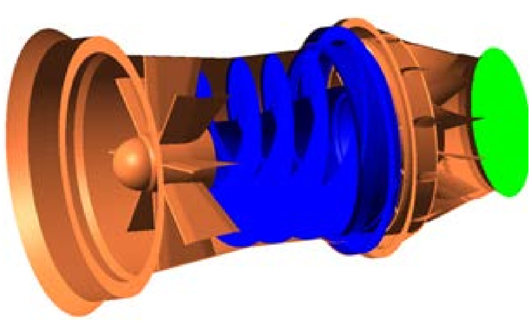

Fig. 1: Geometric model of the flow part of the booster screw pump

MATERIALS AND METHODS

Computational method: An incompressible fluid model is used. The computational modelis for flows of an incompressible fluid based onthe solution of the Navier-Stokes Eq. 1, taking account of the continuity Eq. 2 and the system of equations included in the k-aturbulence model.

The mathematical model of incompressible liquid flow based on the Navier-Stokes equation:

$$\frac{\partial U}{\partial t}+\nabla(U\otimes U)=-\frac{\nabla P}{\rho}+\frac{1}{\rho}\nabla((\mu+\mu_t)(\nabla U+(\nabla U)^T)) \quad (1)$$

And the continuity equation for an incompressible liquid:

$$\nabla\cdot\mathbf{U}=0 \quad (2)$$

The model of turbulence (Wilcox, 1994) is used to determine turbulent viscosity by means of the equation:

$$\mu_t=0.09\cdot\rho\cdot\frac{k^2}{\varepsilon} \quad (3)$$

Initial values of kinetic energy and the dissipation rate are calculated automatically in the first step of the computation.

The finite volume method is used for the FlowVision calculations. An initial computational grid in the form of parallel epipeds is constructed to allocate the finite volumes. The computational grid automatically adapts to specifics of the geometry, so that, geometric features of the elements of the pump flow path are properly taken into account in the calculations. This is achieved by splitting the cells of the initial grid into smaller cells in zones of intensive flow deformation (each cell of the initial grid is divided into eight cells at the next adaptation level). Buil-in functions of thesub-gridresolution of the geometry where rectangular cells that are by intersected a curvilinear boundary are split into arbitrary polyhedra, enable complex geometry of the pump flow path to be taken into account. This substantially reduces the number of cells that are needed in order to ensure a high level of accuracy.

Points were set in the resulting computation domain (the pump flow path) at which pressure pulsations were recorded with time sampling of $0.5*10^{-5}$ sec of physical time. This enabled an understanding of the change in static pressure over time and provided all of the data necessary for analyzing the spectral composition of the pressure pulsationsat points 1-7 (Fig. 2).

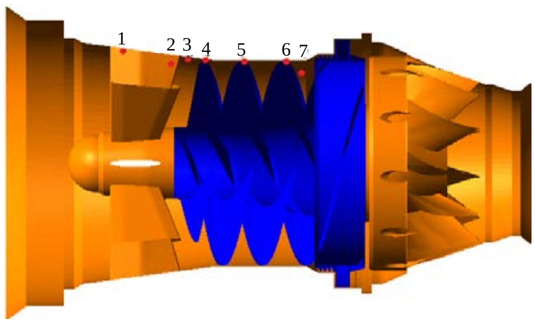

Fig. 2: Fixation points of the pressure pulsations

RESULTS AND DISCUSSION

Computational results: The numerical procedure of spectrum analysis is developed to obtain spectral composition of pulsations of the working fluid pressure in the booster pump flow path (Korn and Korn, 1974). The data show that pulsations at point 7 (Fig. 2) arising at the exit from the screw rotor are mainly due to load on the working blade. Pressure pulsations atblade passing frequency (BPF, 273 Hz) and its higher harmonics are clearly visible in the spectrum of pressure pulsations of the working fluid (Fig. 3).

Such pulsations are characteristic for pump units and have been described in sufficient detail in the literature (Pokrovsky and Yudin, 1966).

A more complex spectral composition (Fig. 4) was obtained at the inlet to the booster pump unit. Iinteraction of the screw with vortex traces after the inlet pylons and tipflow through the radial clearance between the pump casing and the screw blade screate a complex flow structure which causes a pressure pulsation of the working fluid, both in rotor frequency and blade passing frequency (Fig. 4).

It shows the importance of the inlet design of the axial pump to ensure a low level of pressure pulsations and vibration. This result matches with vibration measurements of the booster pump showing a clear rotor spectrum component. Analysis of flow parameters at the inlet to the screw impeller shows the formation of a pressure field that does not have circular symmetry and that is a source of pressure pulsations in the rotor frequency. This corresponds to the results of (Lyudvinitskaya and Ayupov, 2014) (Fig. 5).

The computational results indicate that the uneven pressure field at the inlet to the screw (the emergence of pressure drop zones) depends not only on the number of pylons and the ratio of the number of pylons to the number of screw blades but also, on the flow pattern of the fluid around the pylons and the axial clearance

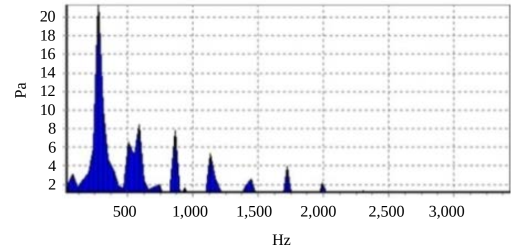

Fig. 3: Spectrum of pressure pulsations at the screw exit, point 7

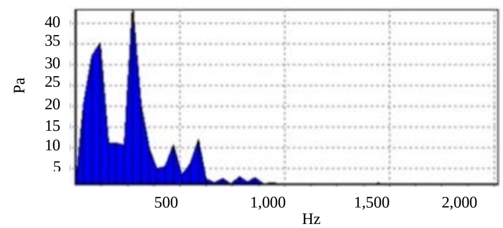

Fig. 4: Spectrum of pressure pulsations at point 2 before the screw blades

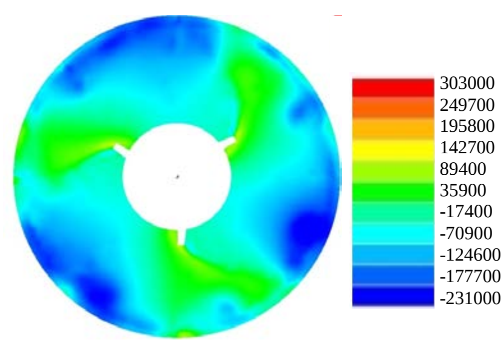

Fig. 5: Inlet screw pressure filed

between the pylons and the screw. The computations show that reduction in the number of pylons from 8-3 substantially reduces the rotor pressure pulsations. This may also be due to the restoration of circular symmetry of the inlet flow. The installation of a shroud ring at the inlet to the screw in order to reduce intensity of the tip flow as well as increase in the axial clearance between the pylons and the screw are also recommended.

Defining eigen modes of vibration: Modeling of vibration of the pump casing begins with determination of its eigenmodes and vibration frequencies. Modeling of forced oscillations of the structural elements and pump casing is obtained by assigning distributed dynamic loads from pressure pulsations (Pitolin, 2012).

Examination of the spectral composition of pressure pulsations enables a review of possible resonance oscillations in the pump components caused by the

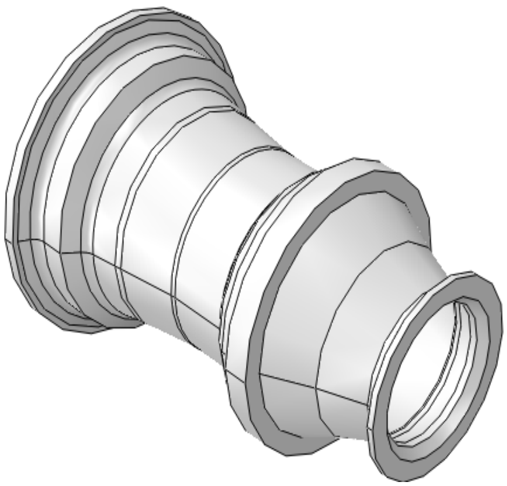

Fig. 6: Geometric model of the booster casing

pulsations. A computational experiment was carried out for this purpose using the commercial finite volume software package. A geometric model of the pump housing was created, consisting of cylindrical and conical sections connected with each other in a flange (Fig. 6).

It should be noted that the computations use a simplified geometric model in which the bolted connection is not taken into account. It is usual practice to consider the oscillation eigenmodes of the pump casing, treating the latter as a complex envelope with variable cross-section.

It is assumed in the computations that the pump casing is made of steel with density of 7700 kg/m^3, Poisson ratio of 0.28 and a Young modulus of 210 GPa. The calculation takes account of flange fixing of the pump casing as a boundary condition that prevents the movement of finite elements located on the surface of the side flanges with 6 degree of freedom. The finite element method is used for the calculations. Finite elements in the form of tetrahedra are usedfor discretization of the geometric model of the housing. This enables sufficiently accurate account of the complex forms of the geometric model.

The oscillations obtained for the pump casingare compared with the pulsation frequencies obtained in the modeling. The results of computation of the oscillation modes are used to find the two modes which have the closest frequencies.

The first of these, the oscillation eigen mode (Fig. 7) is obtained at a frequency of about 90 Hz which is close to the pressure pulsations at the rotor frequency (95 Hz). It should be noted that not all three flange connections are involved when there is eigenmode at frequency close to 90 Hz. The oscillations occur in the cylindrical and conical parts of the casing. A similar picture is

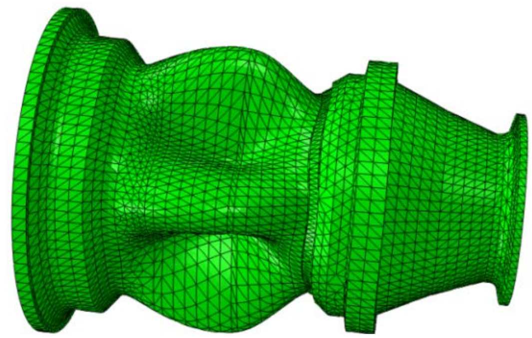

Fig. 7: Eigenmode at frequency of 90 Hz

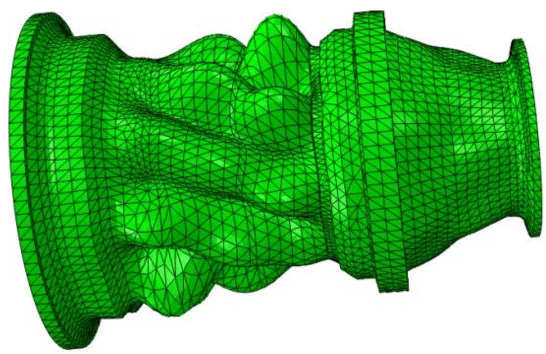

Fig. 8: Eigenmode of the housing at frequency of 274 Hz

observed at frequency of 273 Hz. The resulting eigenmode coincides with the blade passing frequency of pressure pulsations in the pump flow path (Fig. 8).

There is no dynamic load on the flange connections in this mode of vibration. Based on the results obtained, it can be concluded that the main load in these conditions is on the pylons located in front of the screw. To assess the effect of pressure pulsations on the pump pylons, further computations was carried out in which the cylindrical section of the pump housing was considered together with the pylons.

Computations of the cylindrical section of the pump casing with pylons: A computational study was carried out of vibrations arising from the effect of pressure pulsations on the cylindrical study of the casing. The effect of the pylons on the eigenmode and the form of the oscillations was determined. It was found that the pylons influence the eigenmodes of the pump, increasing resilience of the structure. However, the frequencies at which the eigenmodes appearare almost unchanged which suggests that there is a resonance effect, accompanied by a significant increase in the amplitude of oscillation of the pump components (Fig. 9).

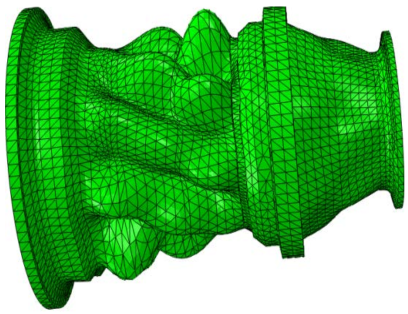

Fig. 9: The eigenmode of oscillations of a cylindrical body section with pylons (264 Hz)

To assess influence from magnitude of the movements caused by pulsations an additional computation was carried out in which distributed forces from the pressure pulsations were applied to the inner walls of the cylindrical section of the pump housing. The results show that pressure pulsations can cause vibrations of certain sections of the pump casing with amplitude up to 7 μm.

CONCLUSION

The source of booster pump hydrodynamic vibration is studied. It is shown that pressure pulsations at the exit of the screw are the result of a rotor-stator hydrodynamic interaction. Pressure pulsations at the inlet to the screw are generated by an uneven pressure field intensified by the hydrodynamic interaction of the vortex traces of the pylons with the tip flow through the radial clearance between the pump casing and the blades of the screw.

Numerical modeling of forced vibration of the pump unit, generated by pressure pulsations in the flow path is completed.

RECOMMENDATIONS

Based on the results of the numerical modeling, recommendations are made for reducing the level of pressure pulsations in the booster pump. This can be done by fitting a shroud ring increasing the axial clearance and reducing the number of pylons to a multiple of the number of the blades ofthe screw.

Eigenmodes of the pump unit casing at frequencies of 274 and 95 Hz are determined which can cause resonance in the frequency of the working blades and the rotor frequency under the influence of pressure pulsations.

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