BULETINUL Universităţii Petrol – Gaze din Ploieşti	Vol. LX No. 1/2008	37 - 44	Seria Tehnică

Aspects Regarding Noise in the Pumping Stations

Ion Pană

Universitatea Petrol-Gaze din Ploieşti, Bd. Bucureşti, 39, Ploieşti
e-mail: ion.pana@mail.upg-ploiesti.ro

Abstract

The article deals with the possibility of using the numerical methods in the design of a pumping room to reduce the level of noise inside and outside of the building. We use Comsol program, acoustic section, to generate and analyze the model of a pumping room with six generators. In conclusion, the advantages and the limits of using numerical methods in this purpose are presented comparatively with the actual method of calculus used for the noise appreciation.

Key words: *noise, pumping station, numerical simulation.*

The Sources of Noise in a Pumping Station

In the design of a pumping station, it is important to consider the problem of noise and vibrations generated by engines and generators. Variations in noise amplitude and frequency that result from malfunction or improper operating conditions depend on the type of the hydraulic generator and the problem which causes the noise. Noise in pumping systems can be generated by the mechanical motion of the elements of the pump and by the liquid motion in the pump and piping system. Liquid noise sources can result from vortex formation in high-velocity flow, from pulsating flow in the case of reciprocating pumps and from cavitation [1]. Noise from the sources enumerated before, can be propagated to the environment by piping, the liquid in the piping, the pump and support structure, and the surrounding air.

The main problems regarding pump noise are: noise levels that do not meet applicable environmental criteria [2-4]; noise that can be used to diagnose faulty pump operation or failure. As we said before, the sources of noise in a pumping station are: mechanical sources and liquid sources. Examples of mechanical sources: pistons, bearings, valves, belt transmissions, rotating unbalanced assemblies, and vibrating pipe walls. In centrifugal machines, improper installation of couplings causes mechanical noise. If pump speed is near or passes through the critical speed, noise can be generated by high vibrations resulting from imbalance or by the rubbing of bearings, seals, or impellers. In positive displacement pumps, noise depends on the speed of the pump and the number of pump pistons. Incorrect crankshaft counterweights will also cause shaking at running speed. Other mechanical noises are associated with worn hearings on the connecting rods, worn wrist pins, or slapping of the pistons.

Liquid pulsations are the primary mechanically induced noise, and these in turn can excite mechanical vibrations in components of both pump and piping system. Some of the pressure fluctuations are produced directly by liquid motion. The fluid dynamic sources include turbulence, flow separation, cavitation, waterhammer, and impeller interaction with the pump cutwater. The resulting pressure and flow pulsations may be either periodic or broad-band in frequency and generally excite the piping or the pump into mechanical vibration [5,6]. These mechanical vibrations can then radiate acoustic noise into their environment. The pulsation sources in pumps can be classified in [7]: discrete-frequency components generated by the pump impeller or pistons; broad-band turbulent energy resulting from high flow velocities; impact noise consisting of intermittent bursts of broad-band noise caused by cavitation, waterhammer; flow-induced pulsations caused by periodic vortex formation when flow is past in the piping system. A variety of secondary flow patterns that produce pressure fluctuations are possible in centrifugal pumps, in the case of operation at off-design flow [1]. When a centrifugal pump is operated at flows less than or greater than best efficiency capacity, noise is usually heard around the pump casing. The magnitude and frequency of this noise depends of type of pump and are dependent on the value of the pressure of the pump, the ratio of *NPSH* required to *NPSH* available, and the amount by which pump flow deviates from nominal flow [6,7]. Some formulas and values of frequency for different sources of noise are indicated in table 1 [5-7,9].

Table 1. Excitation frequency induced by generators and flow

No.	**Source of noise/vibration**	**Value of frequency** [Hz]	**Generated frequency**[1] [Hz]
1.	Reciprocating compressor	$n \cdot f$	25,50,75…
2.	Reciprocating pumps	$n \cdot f$ $n \cdot r \cdot f$	25,50,75 50,75,100,125…
3.	Centrifugal pumps and compressors	$n \cdot f$ $n \cdot z \cdot f$ $n \cdot v \cdot f$	25,50,75… 150,300,450… 25,50,250,500…
4.	Screw pumps, compressors	$n \cdot f$ $n \cdot t \cdot f$	25,50,75… 125,250,375…
5.	Flow through restrictions	$\frac{0,2V}{D} \dots \frac{0,5V}{D}$	3…50
6.	Cavitation	0 –1000	
7.	Flow turbulence due to quasi –steady flow	0 –30	
n =1,2,3, …; *f* running frequency; *r* number of pump pistons; *z* number of blades ; *v* number of volutes of diffuser vanes; *t* number of teeth of the rotor; *V* velocity of fluid; *D* diameter of pipe; [1]for engine speed *n* =1500 rpm, and frequency *f* = 25 Hz.			

A special mention goes to the cavitation, which produced a wide-band shock that excited many frequencies s. table 1 [1, 6]. Pressure regulators or flow control valves may produce noise associated with both turbulence and flow separation. These valves, when operating with a severe pressure drop, have high flow velocities that generate significant turbulence. If the frequency of induced noise coincides with the one of the eigenfrequencies of the pumping room, the effects grow up. It is recommended that the lowest resonant frequency of the pumping room should be well below the minimum operating frequency, and none of the higher resonant modes should be coincident with running speed or multiples thereof [5]. But as we see from table 1, there are many induced frequencies and the problem is very complicated. Following we present an application in Comsol [8] concerning the evaluation of eigenfrequencies of a pumping room.

Eigenfrequencies of a Pumping Room

We will use for our application a model of a pumping station (26 x 13 x 8), in which 6 pumps and their engines are mounted. Each pumping units is approximated with three boxes: mounting plate (3 x 1.5 x 0.5); electric engine (1 x 0.5 x 1); centrifugal pump (1.5 x 0.5 x 1), all dimensions are in meters. The corner of the first mounting plate is situated at 3 m from each walls of the room, and the distance between units is 1.5 m. On the wall there is an air outlet conduit for ventilation with a square section (1 x 1). In the room it is also situated an office for accredited personnel. We modeled the geometry of the pumping station, using the program Comsol 3.2 [8], figure 1,*a*. Following we define an acoustic study in the 3D created geometry. We keep for the study only the free volume of the room, figure 1,*b*. Sound propagation in free air is described by the wave equation:

$$-\Delta p + \frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} = 0, \tag{1}$$

where *p* is the pressure; *c* is the speed of sound, and *t* is the time. If the air is brought into the motion by a harmonically oscillating source, we will have a solution of the form:

$$p = \hat{p} \cdot e^{i\omega t}, \tag{2}$$

and the wave equation becomes:

$$-\Delta p + \frac{\lambda}{c^2}\hat{p} = 0, \tag{3}$$

where an eigenvalue is:

$$\lambda = \left(2\pi \cdot f\right)^2 . \tag{4}$$

We realized the study in two assumptions: sound hard boundary condition, the normal component of the velocity is zero on the boundary; that means the normal derivative of the pressure is zero on the boundary:

$$\frac{\partial p}{\partial n} = 0, \tag{5}$$

and a soft wall condition, which means:

$$p = 0. \tag{6}$$

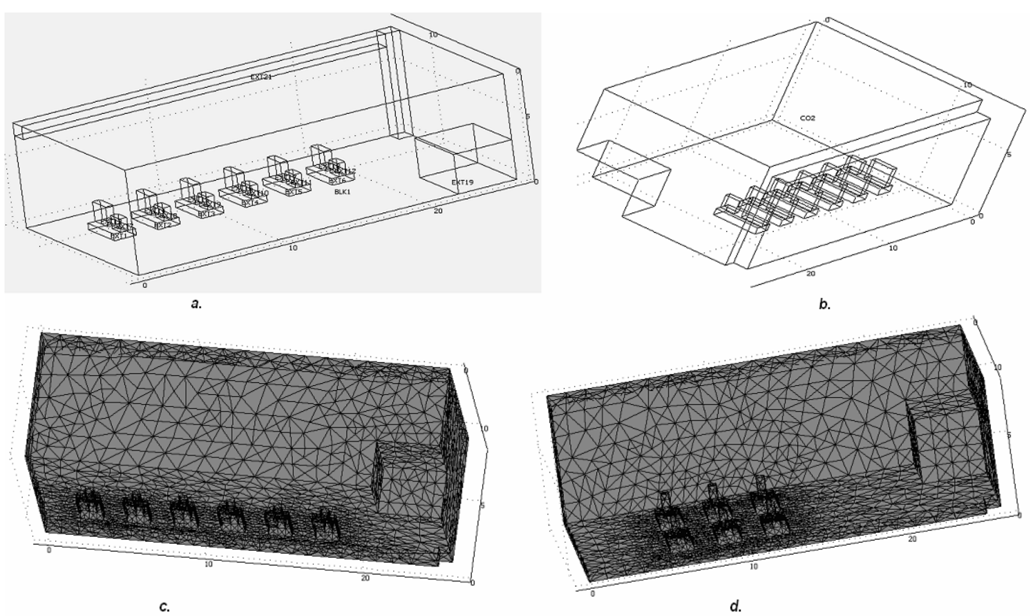
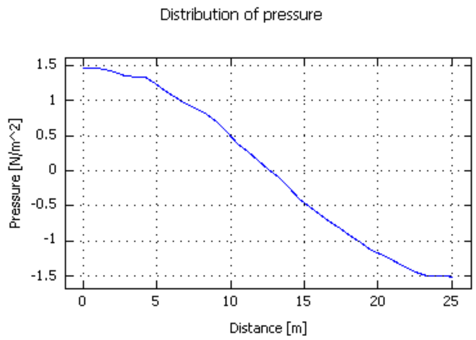
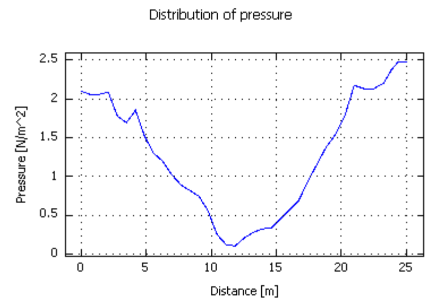


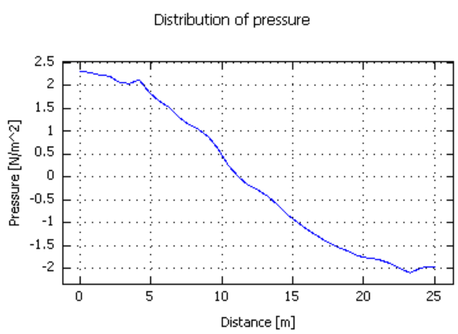
Fig.1. The model of pumping room: a) geometry 3D; b) compound object obtained by removing of office and pumping groups [8]; c) the model with mesh of finite elements in the case of six pumps inline; d) the model in the case of six grouped pumps.



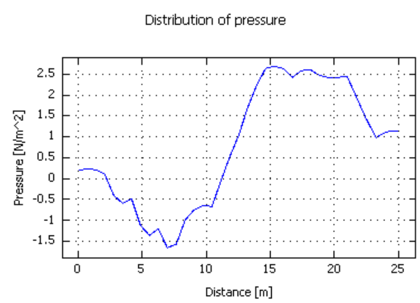
f = 6.65 Hz



f = 13.19 Hz



f = 15.05 Hz



f = 23.70 Hz

Fig.2. Distribution of acoustic pressure on the edge of the pumping room.

We apply a mesh to our model, figure 1,*c*, and solve the model to obtain the eigenvalues for the pumping room. The eigenvalues for this arrangement of pumping room are indicated in table 2, column 2. We plotted the variation of the acoustic pressure on the long edge of the floor of the room for some eigenfrequencies, in figure 2. The noise is direct with the value of pressure. To see the effect of changing the arrangement of pumps into the room, another simulation was made with the pumps on two rows in parallel.

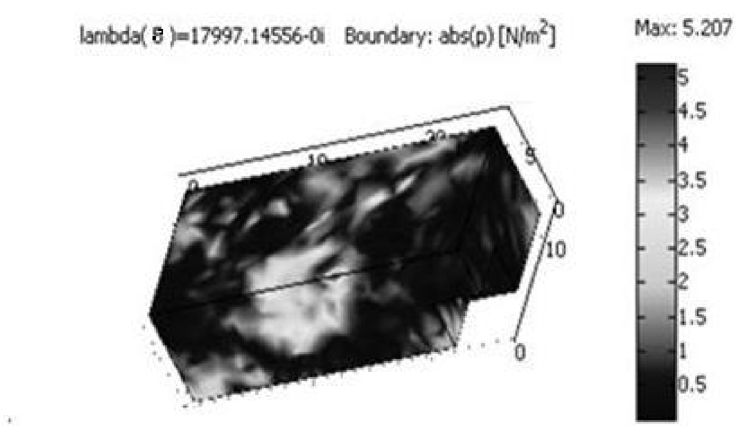


Fig.3. Distribution of absolute pressure abs(p) on boundary for Eigen frequency of 21.36 Hz; *sound hard boundary (wall),* boundary condition for all boundaries.

The corner of the mounting plate of the first pump is situated at 3 m from long wall and at 6 m from the short wall. The distance between pumps is 1.5 m. The eigenfrequencies for this situation with hypotheses expressed by relations (5) and (6) are presented in table 2, columns 3 and 4. We observe that there are no significant differences generated by the modifications of the arrangement of pumps into the room (table 2, columns 2 and 3). But an important change appears when we modify the condition on the boundary, columns 3 and 4. In the physical sense a soft wall means a greater possibility of absorption of the noise. We think that the real situation is between these two hypotheses, and the program Comsol is not able yet to simulate selectively the effect of different anti vibration products introduced in the structure of wall.

We represented also the variation of acoustic pressure on the walls of the pumping room. We observe that in the case on a hard wall the acoustic pressure has important values near the walls figure 3, and in the hypothesis of a soft wall, there are only some peaks of pressure near the generators figure 4.

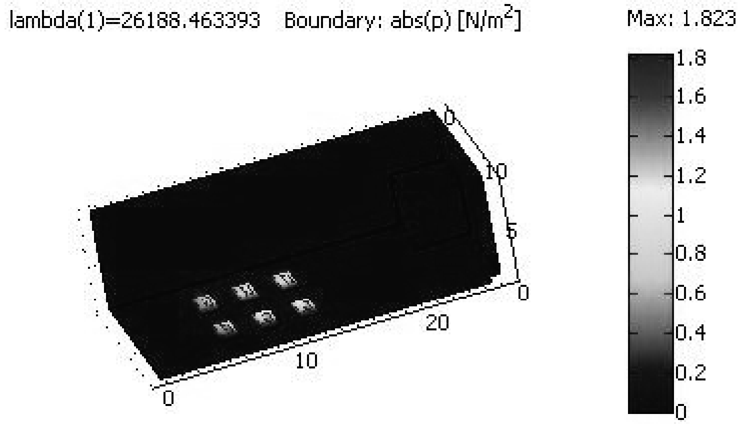


Fig.4. Distribution of absolute pressure abs(p) on boundary for Eigen frequency of 25.75 Hz; *sound soft boundary (wall),* boundary condition for boundaries of the pumping room.

Conclusions

The approaches for reducing noise from pumps and piping systems after it is airborne generally consist of either interrupting the transmission path or controlling the reverberation characteristics of the pump room. A highly reflective pump room or enclosure can increase pump noise levels with several decibels by reflections of the noise back and forth in the enclosure. The effect is more intense in resonance conditions. So we have to design the pumping room to avoid the reverberant conditions.

Practically the maximum reduction that can be achieved by the application of acoustic absorption material to the interior surfaces is about 10 dBA [9]. Using a program like Comsol we can obtain the distribution of the pressure into the pumping room and the eigenfrequencies. If we use some anti vibration materials on the wall, we

Table 2. The values of eigenfrequencies of pumping room

Case	Six pumps nline/ hard walls [Hz]	Six pumps grouped / hard walls[Hz]	Six pumps grouped/ soft walls [Hz]
1	0	0	25.76864
2	6.654320	6.687898	30.48355
3	13.07049	13.33878	32.62505
4	13.19010	13.80032	34.55414
5	15.05805	14.75059	36.50493
6	18.60126	18.61832	37.60390
7	19.35868	19.11690	40.38971
8	22.56365	21.36193	42.07861
9	22.82651	21.63438	44.79761
10	23.70357	22.68312	45.01387

could presume a soft wall condition. Following we determine the frequencies of sources for noise from formulas presented in table 1, or in the case of engines or pumps from manufacturer data. It is necessary to avoid working at these frequencies. The distribution of the acoustic pressure into the room is an indicator of the level of noise *SL* which could be obtained into it. Introducing this value in relation:

$$SL = 20\lg\left(\frac{p}{0.02}\right), \tag{7}$$

we express the value of noise in decibels. So we obtain an image of the noise level. For the maximum value of the pressure shown in figures 3 and 4, the value of noise are 48 dBA and 39 dBA. The admissible value of allowable exposure in time is presented in figure 5 [2-4]. The program cannot appreciate actually the influence of different materials disposed on the wall. The procedure is quick, and considers the exact geometry of the pumping room. Comparatively we present shortly the actual procedure [7].

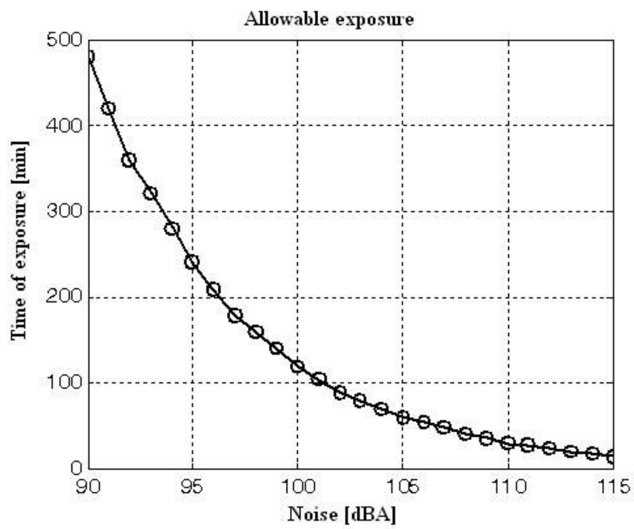


Fig. 5. Allowable exposure time.

An enclosed or partially enclosed room, noise radiating from equipment is reflected from the room boundaries and builds up to a higher noise level than if the equipment were located outside. This buildup of noise caused by acoustical energy being contained within the room creates a reverberant area or field in the room. The noise level in the reverberant field can be approximated with the relation:

$$L_{pri} = L_{pi}(1m) - 10\lg\left(NRC \cdot S\right) + 17 \ , \tag{8}$$

where L_{pi} (1m) is the sound pressure level at one meter from the source *i*, *S* is the total surface area of the interior surface of the room and *NRC* is the average noise reduction coefficient of the room surface exposed to the noise. Total noise level for multiple sources L_p (in number of *ns*) is:

$$L_p = 10\lg\sum_{i=1}^{ns} 10^{N_i} \ , \tag{9}$$

$$N_i = \frac{L_{pri}}{10} \ . \tag{10}$$

The value of noise reduction through walls of the room NR is:

$$NR = STC_c - 5, \quad (11)$$

$$STC_c = -10\lg\left[\frac{1}{s}\sum_{i=1}^{n} A_i 10^{T_i}\right], \quad (12)$$

$$T_i = \frac{STC_i}{10}. \quad (13)$$

The total level of noise L_{pt} is:

$$L_{pt} = L_p - NR. \quad (14)$$

where STC_c is composite class of sound transmission, A_i exposed surface area; STC_i sound transmission class for material i, s is the total surface area of all components in the direction of the listener, n number of materials used in isolation. Also we can reduce the noise by: **s**ource modification, transforming the basic pump design or operating condition to minimize the generation of acoustic energy [10]; interrupting the path between the energy source and the listener. Some of the source modification approaches for pump applications are: increase or decrease pump speed to avoid system resonances of the mechanical or liquid systems; increase liquid pressures (*NPSH* dispose) to avoid cavitation or flashing [1,11]; decrease suction lift; balance rotating or oscillating components; change drive system to eliminate noisy components; correct acoustic resonance to minimize liquid-borne energy; modify centrifugal pump casing vanes so clearance between impeller diameter and casing cutwater (tongue) or diffuser vanes is increased; modify centrifugal pump impeller discharge blade configuration; modify centrifugal pump casing cutwater (tongue) by slanting or adding holes; replace pump with different model or type to permit operation at reduced speed and the least number required [9-13].

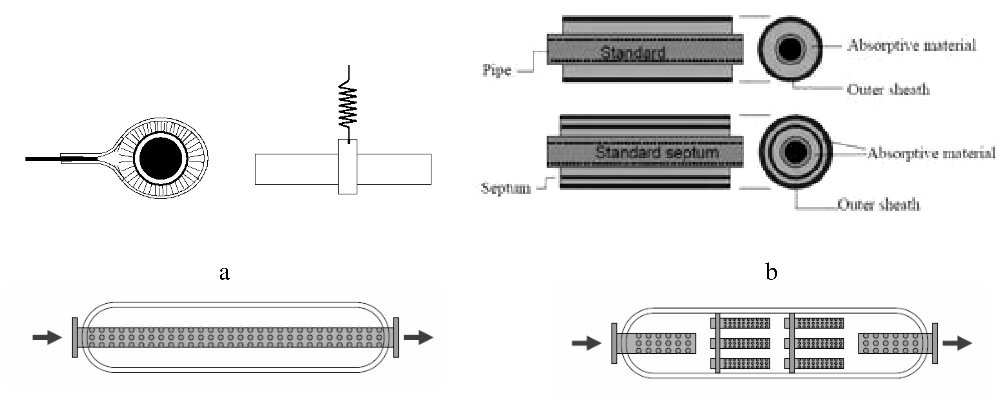


Fig. 6. Methods used in reduction of noise: a) elastic pipe hangers; b) cover absorptive material: standard; sandwiched septum layer; c) inlet pipe perforated with many small holes; d) multiple expansion chambers.

If noise is due to operation of a centrifugal pump at flows less than designed and recirculation is the problem, we install a flow recirculation system bypass to increase total pump flow. If several pumps are operating in parallel, we operate all pumps at the same speed. Others methods frequently use are: heavier bearing lubricant or increased number of bearing rolling elements; inject small quantity of air into the suction of a centrifugal pump to reduce cavitation noises; use adequate valve sizes, avoid high velocities and obstructed flows; keep pressures above the vapor pressure of the fluid being pumped, degasify the fluid; provide adequate pulsation control equipment [6]. Noise generated in the region of the pump is conducted both upstream and downstream by the liquid in the piping system. Such paths can be effectively reduced by acoustic filters or other pulsation control equipment [5, 7], such as side branch accumulators figure 6. The application of elastomeric coatings figure 6,*b,* to the exterior surface of the pump or piping to damp pipe wall vibrations is normally ineffective, except on very thin conduit. However, such coatings may have a small acoustic effect in confining or absorbing high-frequency noise that would otherwise be radiated by the pipe wall vibrations [12]. Oscillatory energy generated near the pump by pulsation, cavitation, turbulence, can be conducted as solid-borne noise for substantial distances before it is radiated as acoustic noise into the atmosphere. Some success in controlling solid borne noise propagation can be achieved by the use of flexible couplings in the piping systems, mechanical isolation for the pump and drive systems, and pipe hangers and supports [1, 6]. Techniques for the vibration isolation of mechanical equipment are well known and available from many suppliers of vibration isolators or mounts [1]. These may consist of resilient supports at each mounting point of the machine, although it is often necessary to assure alignment, to mount the pump and drive system on a single rigid landing skid and then isolate it from its support system [13].

References

1. Săvulescu, P. – *Maşini şi utilaj de transport hidraulic*, Editura Universităţii din Ploieşti, 2005.
2. * * * . EN ISO 4871:1999 – *Acustica – Declararea si verificarea valorilor emisiei de zgomot a masinilor si echipamentelor.*
3. * * * . EN 27574- 1-4:1994– *Acustica – Metode statistice pentru determinarea si controlul valorilor declarate ale emisiei acustice ale masinilor si echipamentelor.*
4. * * * . EN ISO 12001:1999– *Acustica – Zgomotul emis de masini si echipamente. Reguli pentru elaborarea si prezentarea unui cod de incercarea a zgomotului.*
5. Pană, I. – *Sisteme de transport şi distribuţie a hidrocarburilor*, Editura Universităţii Petrol –Gaze din Ploieşti, 2006.
6. Ionel, I. – *Pumps and Pumping*, Elsevier, Amsterdam, 1986.
7. Sanks, R. – *Pumping Station Design*, Butterworth- Hainemann, Boston, 1998.
8. * * * . *Comsol Multiphysics – Modelling Guide,* Comsol Inc. 2006.
9. Horodincă M. – Cercetări teoretice privind localizarea spaţială a surselor de emisie vibroacustică, *în lucrările celei de a VII-a Conferinţe internaţionale de inginerie managerială şi tehnologică, TEHNO'95*, Universitatea Tehnică Timişoara**,** Catedra TCM, p. 79-86,1-2 iunie, 1995.
10. Wachel, J. C., Szenasi F. R., Denison, S. C. – Analysis of Vibration & Failure Problems in Reciprocating Triplex Pumps for Oil Pipelines, *ASME Paper 85-PET-10*, 1985.
11. Smith, D. R., Price, S. M., Kunz, F. K. – Centrifugal Pump Vibration Caused by Supersynchronous Shaft Instability,*13th International Pump Users Symposium*, The Turbomachinery Laboratory, Texas A&M University, Houston, TX, March 1996.
12. Lewis, A. L., Szenasi, F. R., Roll, D. R. – Control-Valve-Induced Pipeline Vibration Corrected by Variable-Speed Pumping Pumping Technology*, The Journal of Advanced Fluid Handling*, A Putnam Publication, Volume 2, No. 1, pp. 27-31, June 1998.
13. Smith, D.R., Price, S.M., Schiavello, B. – Abnormal Vertical Pump Suction Recirculation Problems Due to Pump-System Interaction*, Proceedings of the 21st International Pump Users Symposium,* March, 2004.

Aspecte referitoare la zgomot în staţiile de pompare

Rezumat

Acest articol se ocupă de posibilitatea de utilizare a metodelor numerice la proiectarea unei staţii de pompare, astfel încât să reducem nivelul zgomotului în interiorul şi în exteriorul staţiei. Se utilizează programul Comsol, modulul acustic, pentru a genera şi a analiza un model al unei staţii de pompare cu şase generatoare. In concluzii, sunt prezentate comparativ cu metoda actuală de calcul utilizată pentru aprecierea zgomotului, avantajele şi limitele de utilizare ale metodelor numerice pentru acest scop.