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Analiza obratovalnega hrupa in vibracij okrova radialne črpalke

Radial Pump Operating Noise and Casing-Vibration Analyses

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V prispevku je podana teoretična in eksperimentalna študija dinamičnih tekočinskih vibracij in hrupa. Eksperimentalne meritve so izvedene na enostopenjski radialni črpalki, ki obratuje s čisto hladno vodo. Pri različnih obratovalnih režimih (vrtilnih frekvencah rotorja) so izvedene meritve naslednjih obratovalnih karakteristik: dušilna krivulja, pretok – izkoristek, pretok – obratovalni hrup in pretok – amplituda vibracij okrova. Rezultati meritev so podani v časovnem in frekvenčnem prostoru.

(Ključne besede: črpalke radialne, hrup črpalk, meritve vibracij, minimiranje hrupa)

This paper surveys theoretical and experimental studies of fluid-dynamic vibration and noise. The experimental measurements were carried out on a radial one-stage pump which operates with clean cold water. Several experimental measurements on the operating characteristics such as capacity-head, capacity-efficiency, capacity-operating noise and capacity-pump casing vibration amplitudes under different operating regimes (different impeller speed) were performed. Measurement results are given in time and frequency domains.

(Keywords: radial pumps, pump noise, vibration measurements, noise minimization)

0 UVOD

Obratovalni hrup in vibracije okrova sodobnih črpalk morajo biti minimalni. V ta namen moramo reducirati vse vire. Obratovalni hrup in vibracije okrova so posledica pulzirajočih tokovnih veličin na izstopu iz rotorja. Hrup se sprošča vedno, ko imamo opravka z relativnim gibanjem dveh tekočin (kavitacija – voda in para) ali tekočine in trdne stene (lopatice). Značilni viri hrupa črpalk vsebujejo časovno spremenljiv sistem sil, ki delujejo na eno ali več komponent črpalke. Hrup je rezultat reakcije tekočine na omenjene sile in vsiljenega nihanja teles, ki so v stiku s tokom. Vibracije okrova in njegovih mirujočih ter gibajočih se delov so posledica tokovno vzbujenih vibracij teles v ustaljenem ali pulzirajočem in turbulentnem toku. Pri tokovno vzbujenih vibracijah je prav tako treba upoštevati tudi različne vibracijske oblike kakor pri nihajočih trdnih telesih. Pulzacija toka na izstopu iz rotorja je posledica nepopolnosti rotorja in še posebej relativnega vrtinčnega toka v posameznem rotorskem kanalu, ki je pri obratovanju zunaj preračunske točke (točka največjega izkoristka) še močnejši in povzroča nepravilni natok

0 INTRODUCTION

Modern pumps are expected to have their operating noise and casing vibrations minimized. In this way the noise and vibration pollution of the surroundings are reduced. The operating noise and casing vibrations of radial pumps are a consequence of the pulsating fluid flow properties at the impeller exit. Noise may be emitted whenever there is a relative motion of two fluids (cavitation – water and vapour) or a fluid and a solid surface (blades, vanes). Typical sources of noise from pumps involve a time-varying system of forces affecting one or more components of the pump. Noise results from the fluid's reactions to this force as well as from the forced vibration of structures in contact with the flow. The vibration of the pump casing and its static and dynamic (moving) parts are a consequence of the flow-induced vibration of solid structures in stationary or pulsating and turbulent flow. Therefore, the subject of flow-induced vibrations must also consider the vibration of structures of a single mode or of many modes as well. The pulsating flow at the impeller exit is the result of impeller imperfections, especially, of the relative flow whirl in an individual impeller channel. The whirl increases by shifting the operating mode out of the optimum regime (out of the best v rotorske in vodilne kanale. Tako nastane neustaljeni pulzacijski turbulentni tok, ki povzroča vibracije okrova. Moč opisanih tokovnih pulzacij v posameznih obratovalnih režimih (obratovanje pri majhnih, pod optimalnih pretokih) se spreminja. Odvisna je od natočnega kota na rotorske lopatice, ki se spreminja v odvisnosti od pretoka in/ali vrtilne frekvence.

Študija hidravličnih lastnosti črpalke je zasnovana na modelu črpalke, na katerem so opazovane vibracije okrova in hrup v odvisnosti od vrtilne frekvence. Pri meritvah so določeni tudi različni mehanizmi tokovnih motenj. Za minimizacijo hrupa in vibracij moramo vedeti, da se te povečujejo s povečanjem vrtilne frekvence rotorja in tudi kadar je obratovalni pretok ali obratovalna točka črpalke zunaj točke največjega izkoristka. Hrup in vibracije okrova so minimalne, ko črpalka obratuje z optimalnim pretokom, ki je največkrat kar preračunski pretok. Če pa želimo dosegati zahtevano črpalno višino in pretoke pri manjših vrtilnih frekvencah rotorja, morajo biti izstopne hitrosti toka večje. Te lahko dosežemo le s povečanjem izstopnega kota rotorskih lopatic ali s povečanjem izstopnega premera rotorja. Na žalost sta oba postopka omejena s trdnostjo materiala in kavitacijskimi problemi. Zaradi tega je za minimizacijo hrupa in vibracij priporočena standardna tridimenzionalna metoda optimizacije med vrtilno frekvenco rotorja, izstopnim kotom rotorskih lopatic in izstopnim premerom rotorja.

efficiency point – BEP), and causes the irregular loading of the impeller and guide-vane channels. As a result, the non-stationary pulsating and turbulent flow that causes the casing vibrations is created. The intensity of the flow pulsations in some particular pump regimes (operating with small under-optimum capacities) is changed. It depends on the angle of flow attack to the impeller blade that is changed by the operating capacity or/and by changing the impeller speed.

The study of pump hydraulics behaviour is based on the pump-scale model in which the noise and casing vibrations observed for a given configuration as a function of the impeller speed are tested. With measurements, the particular disturbance mechanisms are determined. For minimizing the pump operating noise and casing vibrations it is necessary to know that the noise intensity and pump-casing vibration amplitudes increase as the impeller speed increases. They also increase when the operating capacity, or pump operating point is out of the BEP. The minimum noise intensity and pump casing vibration amplitudes exist when the pump operates at optimum capacity, that is in most cases the pump design capacity. However, if we want to satisfy the required head and capacity with a smaller impeller speed, the flow velocities at the impeller exit must be larger. This can only be achieved by increasing the impeller-blade exit angle or by increasing the impeller-exit diameter. Unfortunately, both approaches are limited by impeller material strength and by cavitation problems. For these reasons the classic 3D optimisation plan of the impeller speed, the blade's exit angle and the impeller-exit diameter for the minimization of the pump operating noise and casing vibration are recommended.

1 DIMENZIJSKA ANALIZA NASTAJANJA ZVOKA

Za analizo tokovno vzbujenih motenj razdelimo tok na časovno povprečno in oscilirajočo komponento (Reynoldsov postopek). V skladu s tem je lokalna hitrost v določeni točki definirana kot vsota povprečne vrednosti in trenutnega odmika od tega časovnega povprečja. Hitrost v točki tekočinskega toka je tako predstavljena z:

1 DIMENSIONAL ANALYSIS OF SOUND GENERATION

The first feature of flow-induced disturbance to note is that flow is generally usefully regarded as a mean plus a fluctuating part (Reynolds' idea). Therefore, the local velocity at a particular point may be regarded as a superposition of an average value and an instantaneous fluctuating part. Thus the velocity at a point in the fluid flow may be described by:

$$U = \overline{U} + u(x,y,z,t) \qquad (1),$$

kje sta - $\overline{U}$ povprečna vrednost in $u(x,y,z,t)$ - njena oscilirajoča vrednost, ki je odvisna od časa in lege v toku in jo poenostavljeno lahko določimo kot:

where $\overline{U}$ is the average value and $u(x,y,z,t)$ is the unsteady value which depends on time and location in the flow, and can be determined in a simplified way as:

$$\overline{u^2}(t) = \frac{1}{t_0}\int_0^{t_0} u^2(t)\,dt \qquad (2),$$

kjer je t_0 - opazovan časovni trenutek. V dinamično podobnih tokovih, pri katerih je zahtevana enakost tako amplitude kakor faze, ostane zveza med silami in premiki nespremenjena, kar je pri primerjavi modela s prototipom upoštevamo v razmerju:

where t_0 is the observation time. In similar dynamic flows, this requires the relationship of both the magnitude and the phase, among forces and motions to remain fixed, for example in a model-to-full-size comparison, the ratio:

$$\frac{u(t)}{\bar{U}} \quad (3),$$

ki je nespremenljivo, ne glede na vrednost $\bar{U}$, ta pomeni stopnjo hitrostnih sprememb glede na povprečno hitrost [1] in [2]. Da bi lahko bilo zgornje razmerje nespremenljivo, morajo biti v ravnotežju tudi različne napetosti, ki delujejo na tekočinske delce. Ponavadi je to kombinacija vztrajnostnih in viskoznih sil, katerih razmerje je definirano z Reynoldsovim številom. Zgoraj omenjena podobnost je zelo pomembna, saj je vzbujevalna tlačna napetost, označena s $|p|$, ki vzbuja hrup ali vibracije v danem toku v neposrednem razmerju:

is a constant, regardless of the value of $\bar{U}$, that is, the distribution of velocity fluctuation's scales on the mean velocity, [1] and [2]. For maintenance of this constancy through the flow, the balance of the various types of stress that act on fluid particles must also be maintained. Generally these are the combinations of the inertial and viscous stress and a measure of the ratio of the inertial to viscous stress in the flow is the Reynolds number. The above-mentioned similitude is important because of the exciting stress, denoted here by $|p|$, that produces sound or vibration in a given type of flow which is in direct proportion as:

$$|p| \approx \frac{1}{2}\rho_0 U^2 = p_d \quad (4),$$

kjer pomenita: ρ_0 - gostoto tekočine, p_d - dinamični tlak. Sorazmernost velja tako dolgo, dokler sta sorazmerni tudi povprečna in oscilirajoča komponenta hitrosti. Ker sta napetosti, ki povzročata hrup in vibracije sorazmerni dinamičnemu tlaku p_d, lahko le-tega privzamemo za merilo moči vzbujanja. Merilo ujemanja hidrodinamičnih ali aerodinamičnih gibanj in hitrosti delčkov glede na hitrost širjenja zvoka je Machovo število, ki pomeni razmerje med hidrodinamično hitrostjo in hitrostjo zvoka. Reynoldsovo in Machovo število predstavljata relativen pomen vztrajnostnih, viskoznih in tlačnih napetosti v tekočini. Za dinamično in akustično podobnost modela in prototipa morata biti poleg podobne geometrijske zato enaki tudi vrednosti Reynoldsovega in Machovega števila.

where ρ_0 is the fluid mass density and p_d is the dynamic pressure. The proportionality may hold as long as the fluctuating velocity and the mean velocity are also proportional. Since the sound- and vibration-producing stresses are proportional to p_d, this can be taken as a measure of the intensity of the magnitude of the excitation. A measure of the matching of fluid inertial motions of hydrodynamics or aerodynamics and the particle velocities related to the propagation of sound is the Mach number, which expresses the ratio of the hydrodynamic velocity to the acoustic particle velocity. The Reynolds and Mach numbers express the relative importance of inertial, viscous, and compressive stress in the fluid. Fluid dynamics and acoustic similitude therefore ideally require, in addition to similar geometries, equal values of Reynolds and Mach numbers for model and prototype.

1.1 Stopnja zvočnega tlaka

Osnovna merjena veličina zvoka v določeni točki je tlak p. Ker je zvok dinamični pojav je tudi akustično vzbujen tlak časovno spremenljiva veličina. Običajno merilo akustičnega tlaka je njegova časovno povprečena kvadratna vrednost:

1.1 Sound pressure level

The principal measured property of sound is the pressure (p) at a point. Since sound is a dynamic phenomenon, the acoustically induced pressure is also a time-varying quantity. The measure of acoustic pressure that is conventionally reported is the time average of a pressure squared, that is:

$$\overline{p^2} = \frac{1}{T}\int_{-T/2}^{T/2} p^2(t)\,dt \quad (5)$$

s časovnim povprečjem enakim nič, $\bar{p} = 0$, ki je v preprosti zvezi z intenziteto in stopnjo jakosti zvoka. Stopnja zvočnega tlaka je določena z:

with the time average equal to zero, $\bar{p} = 0$. This is simply related to sound intensity and power levels. The sound pressure level is determined from the above as:

$$L_S = 10\log\left(\overline{p^2} / p_{ref}^2\right) \quad (6),$$

kjer je $p_{ref} = 2 \cdot 10^{-5}$ N/m^2, ali 20 μPa za zvok v plinih, in 10^{-6} N/m^2, ali 1 μPa za zvok v kapljevinah. Če obravnavamo širjenje zvoka na močnostni bazi, je stopnja moči zvoka podana z:

where $p_{ref} = 2 \cdot 10^{-5}$ is N/m^2, or 20 μPa for sound in gases, and 10^{-6} N/m^2, or 1 μPa for a sound in liquids. Generally, if the sound transmission is considered on a power basis, the sound power level is defined as:

$$L_n = 10\log\left(P / P_{ref}\right) \quad (7),$$

kjer sta P - moč zvoka, prenesena prek določene površine in P_{ref} - referenčna veličina, ponavadi enaka 10^{-12} W. Moč zvoka, prenesenega prek krogelne površine A_s tvori točkast izvor, ki je v naslednji zvezi z zvočnim tlakom:

where P is the sound power transmitted across a specified surface and P_{ref} is a reference quantity conventionally taken as 10^{-12} W. The sound power radiated across a spherical surface of the area A_s, forms an omni-directional source that is related to the sound pressure as:

$$L_n = L_S + 10\log\left(\frac{p_{ref}^2 A_S}{\rho_0 c_0 P_{ref}}\right) \qquad (8),$$

kjer je c_0 - hitrost zvoka. Intenzivnost zvoka lahko določimo iz:

where c_0 is the acoustic speed. The sound intensity level may be found from:

$$L_I = 10\log\left(\frac{I}{I_{ref}}\right) \qquad (9),$$

kjer je zveza akustične intenzivnosti s povprečno kvadratno vrednostjo tlaka definirana kot:

where the acoustic intensity is related to the mean-square pressure by:

$$I = \frac{\overline{p^2}}{\rho_0 c_0} \qquad (10)$$

in $I_{ref} = 10^{-12}$ W/m^2. Akustična intenzivnost je dejansko vektorska veličina. Če je dovolj daleč od vira, je njena smer normalna na krogelno površino, ki točkasti izvor obdaja. Smer I je na dovolj veliki oddaljenosti, torej radialno iz središča izvora.

V črpalki veljajo med dimenzijskimi parametri naslednje zveze: tipske hitrosti $U_T \propto D n_s$, tlačna razlika $\Delta p \propto \rho_0 D^2 n_s^2$, prostorninski pretok skozi črpalko $Q \propto D^3 n_s$. n_8 je specifično število vrtljajev rotorja. Tako je za podani medij (npr. zrak) odvisnost skupne nastale moči zvoka od padca tlaka in volumskega pretoka podana kot:

and $I_{ref} = 10^{-12}$ W/m^2. The acoustic intensity is a vector property. However, far enough from the source the acoustic energy intensity across a spherical surface surrounding the source is directed normal to the surface. Therefore in the far field the direction of I is radial from the acoustic centre of the source.

In a pump the following relationships apply between dimensional parameters: the tip speed $U_T \propto D n_s$, the pressure drop across the pump $\Delta p \propto \rho_0 D^2 n_s^2$, and the volumetric flow rate (capacity) through the pump $Q \propto D^3 n_s$. n_8 is the shaft rotation rate or specific impeller speed. Thus for a given working fluid (e. g., air) the overall radiated sound power has the following dependence on the pressure drop and flow rate:

$$P_{rad}[\mathrm{W}] = a_P \left(\Delta p[\mathrm{Pa}]\right)^2 Q\left[\mathrm{m^3/s}\right] \qquad (11),$$

ki jo lahko upoštevamo kot sorazmerno v neki frekvenčni stopnji pa:

and the proportional band levels:

$$P_{rad}(f,\Delta f) = a_P (\Delta p)^2 Q\, F\left(\frac{fD}{U_T}\right) \qquad (12),$$

kjer je a_P - konstanta, odvisna od tipa črpalke. Normni spekter pasovnih stopenj je:

where a_P is a constant that depends on the type of pump. The normalized spectrum of band levels is:

$$\frac{P_{rad}(f,\Delta f)}{P_{rad}} = f\left(\frac{fD}{U_T}\right) \qquad (13),$$

iz česar je razvidna odvisnost od tipa črpalke in frekvenčne širine pasu.

Za grobo oceno lahko moč nastalega zvoka določimo sorazmerno P_{rad}, Δp, Q ter preostalim obratovalnim parametrom ([3] in [4]). Preproste enačbe (12) in (13) s podanimi vrednostmi a_P in $F(fD/U_T)$ lahko uporabimo za različne vrste črpalk [5].

which exhibits a dependence on both the type of pump and the frequency bandwidth.

For rough estimations the sound power outputs can be determined with the sizing process, P_{rad}, Δp and Q and and all working parameters ([3] and [4]). Simple formulas such as (12) and (13) with given values of a_P and $F(fD/U_T)$ can thus be used to estimate the sound power for different types of pumps [5].

2 MERILNO POSTROJENJE Z RADIALNO ČRPALKO

Meritve obratovalnega hrupa in vibracij okrova so bile izvedene na radialni črpalki

2 RADIAL PUMP EXPERIMENTAL SET-UP

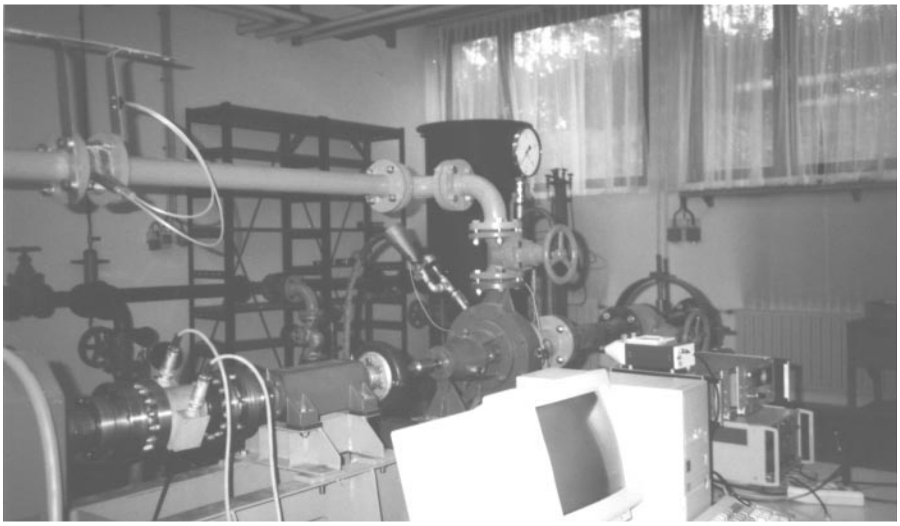
To perform the experimental measurements on the pump operating noise and pump-casing vi-

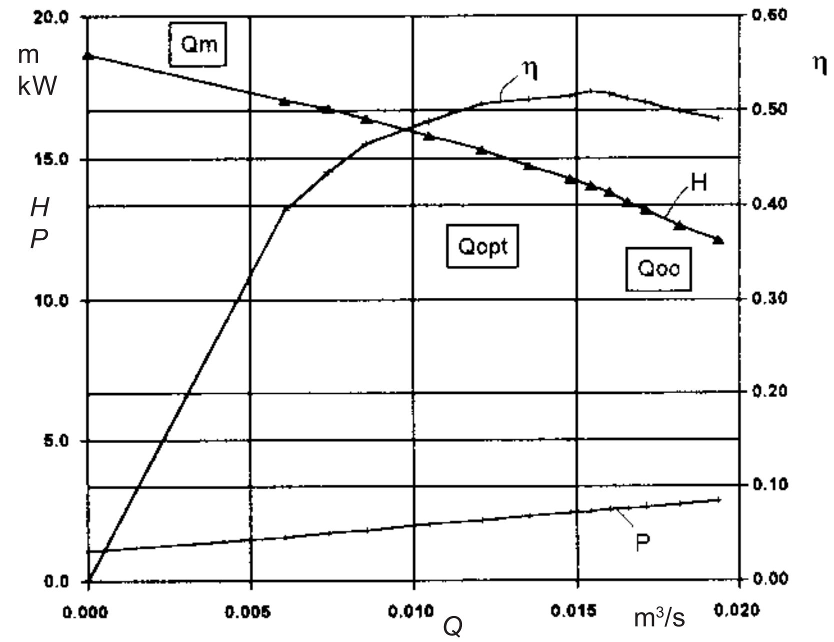
Litostroj CN50/250 (sl. 1). Črpalka ima nizko specifično število vrtljajev n_q = 24 min^{-1}, osem lopatic, spiralni okrov in stabilno dušilno krivuljo (sl. 2).

Vibracije okrova smo merili v treh smereh: pozitivna smer y – osna smer vstopnega cevovoda, pozitivna smer z – radialna smer vstopnega cevovoda, pozitivna smer x – smer tangente na izstopni premer. Za merjenje vibracij smo uporabili merilnik B&K tip 4321 za merjenje hrupa pa merilni sistem RFT 2218 [6]. Za frekvenčni spekter smo uporabili sistem za zbiranje podatkov (DAQ), sestavljen iz osebnega računalnika in več funkcijske kartice Intelligent Instrumentation PCI-2048W. Podatke smo obdelali z Visual Designerjem s frekvenco zajemanja enako 6 kHz na kanal.

brations a radial pump, type CN50/250, manufactured by Litostroj was used (Fig. 1). The pump has a low specific speed (n_q = 24 rpm), eight impeller blades and a volute casing. Its operating characteristics, capacity-head (*Q-H*) are stable (Fig. 2).

Casing vibrations were investigated in three directions: Positive y–direction in the pump axial direction (in the direction of the intake pipe axis), positive z–direction in the pump radial direction, and positive x–direction in a direction tangentional to the impeller exit diameter. A B&K system type 4321 was used for vibration measurements, and an RFT 2218 measuring system for the noise measurements [6]. For the frequency spectrum (and the power spectrum) a data acquisition system (DAQ) was used. The system consists of a PC with a Multifunctional PCI-2048W board card – Intelligent Instrumentation, and software – Visual Designer. The sampling frequency used in the experiment was 6 kHz per channel.





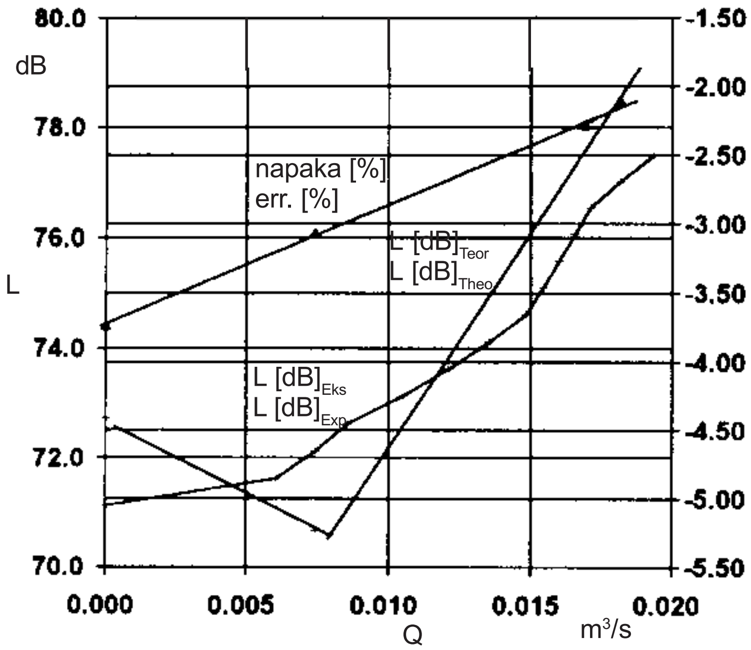
Sl. 2. *Obratovalne karakteristike radialne črpalke*
Fig. 2. *Tested radial pump operating characteristics*

2.1 Obratovalni hrup črpalke pri delni obremenitvi

Povprečne vrednosti stopnje hrupa so razvidne iz rezultatov meritev, prikazanih na sliki 3, od koder je razvidno, da se hrup zvečuje s pretokom. Domnevamo lahko, da sta za to dva vzroka: prvič: dušilna krivulja je stabilna z majhno spremembo črpalne višine (od H = 23 m pri najmanjšem pretoku, do H = 15 m pri največjem pretoku), zato je majhna tudi sprememba hitrosti na izstopu iz rotorja; drugič: absolutna sprememba pretoka je razmeroma majhna (v razponu od 0 do 0,02 m^3/s), zato so hitrosti toka skozi črpalko tudi majhne, kar povzroča spremembo jakosti zvoka.

2.1 Pump operating noise during part-load operation

The mean values of noise level are evident from the common-noise measurement results (Figure 3). The noise level increases with capacity increase. The are two possible reasons for this. First, the pump characteristic *Q-H* is stable and the change of the head by the increase in capacity is small (from H=23 m at zero capacity, up to H = 15 m at maximum capacity). Therefore, the flow velocity changes at the impeller exit are small. Second, the absolute capacity changes are also relatively small (in the range 0 up to 0,02 m^3/s), so the flow velocities across the pump are also small, which causes the change in the noise level.



Sl. 3. *Povprečni karakteristični obratovalni hrup radialne črpalke*
Fig. 3. *Mean-common operating noise of the radial pump*

2.1.1 Močnostni spekter

V nizkofrekvenčnem močnostnem spektru vibracij črpalke do 2000 Hz pri obratovanju z najmanjšim pretokom (sl. 4a) sta v obodni smeri (x) opazni frekvenca vrtenja rotorja (I1) – prvi vrh z leve in prva višja harmonska frekvence lopatice (B2) – drugi vrh z leve. V zgornjem intervalu (do 3000 Hz) je razvidno večje število vrhov, ki pripadajo toku, povzročajo jih tokovno vzbujene vibracije. S porastom pretoka do optimalnega (sl. 4b) in nad optimalnega (sl. 4c) ostane frekvenčni spekter nespremenjen. V vseh posnetkih (sl. 4a, b,c) prevladuje prva višja harmonska frekvence lopatic oz. pulzacije toka (B2), [7] in [8].

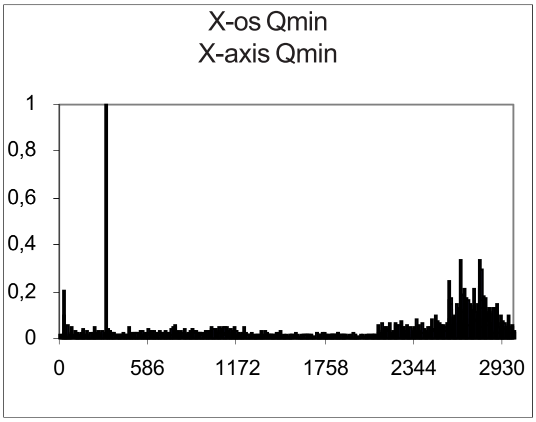
Pri obratovanju z najmanjšim pretokom prevladuje v osni smeri (y) frekvenca vrtilne frekvence rotorja (I1) – prvi vrh z leve proti desni. Iz spektra je razvidna tudi frekvenca lopatic (B1) in

2.1.1 Power spectra records

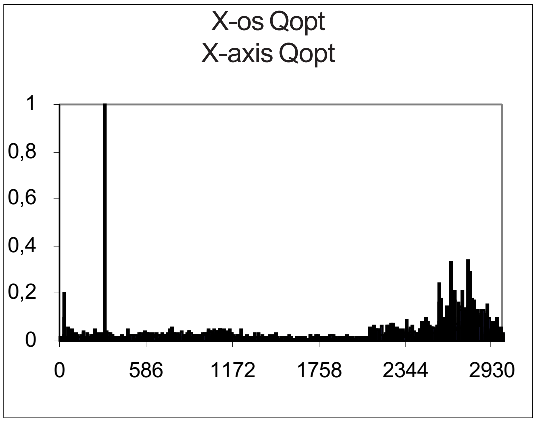
In the power-spectra record of pump-casing vibrations with operation at minimum capacity the impeller rotation frequency (I1), first left peak (Figure 4.a), and the first higher harmonics of the blade frequency (B2), second left peak (Figure 4.a), are evident in the tangentional *x*–direction of the pump-casing vibration in the lower frequency range up to 2000 Hz. In the upper frequency range (up to 3000 Hz) a larger crowd of higher peaks are evident. These vibration frequencies belong to the flow. The vibration may be caused by the flow-induced vibrations. With a capacity increase up to the optimum capacity (Figure 4.b) and a further capacity increase up to over-optimum capacities (Figure 4.c), practically the same frequency situation as observed for minimum capacity is evident. In all three records, Figure 4.a,b and c, the first higher harmonic of the flow pulsation or blade frequency (B2) is dominant, [7] and [8].

njenih sedem višjih harmonskih (B2-B8). Posebej prevladujoča je prva višja harmonska (B2) – tretji vrh z leve. Amplitude drugih vibracij v višjem frekvenčnem intervalu so višje kakor v primeru obodne smeri. To dokazuje, da vplivajo vstopne tokovne vibracije na vibracije okrova v osni smeri, [9]. S povečanjem pretoka do optimalnega (sl. 4e) in nadoptimalnega (sl. 4f) zasledimo podobni frekvenčni spekter. Edina večja razlika je v osnovni pulzacijski frekvenci toka (frekvenca lopatice (B1)), katere amplituda je najmanjša pri optimalnem pretoku skozi črpalko.

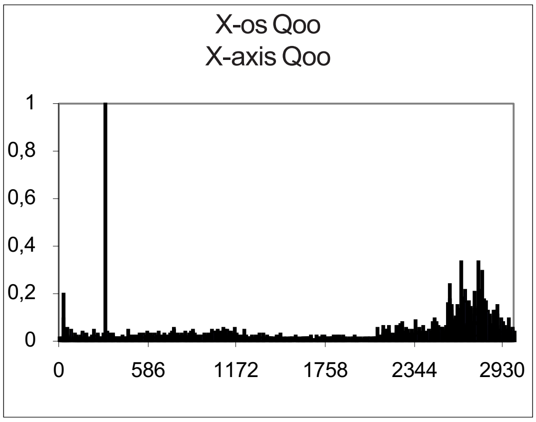
In the axial direction (*y*–direction) at minimum pump capacity operation the impeller speed frequency (I1) amplitude (first peak from left to right) in the power spectrum's record (Figure 4.d) is dominant. The blade frequency (B1), second peak from left to right, and its seven higher harmonics (B2 – B8) are present in the record. This proves that the intake flow vibration has an influence on the pump casing vibration in the axial direction, [9]. With capacity increases up to the optimum (Figure 4.e) and with over-optimum (Figure 4.f), a similar frequency situation as with the minimum capacity is evident. The only difference is the basic flow pulsation frequency (blade frequency B1), the amplitude of which is a minimum at the optimum pump capacity.



a)



b)



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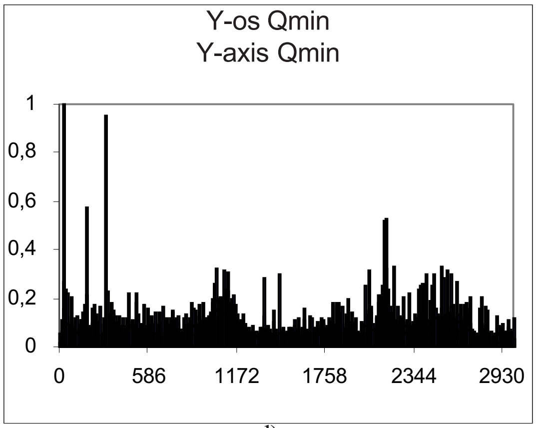

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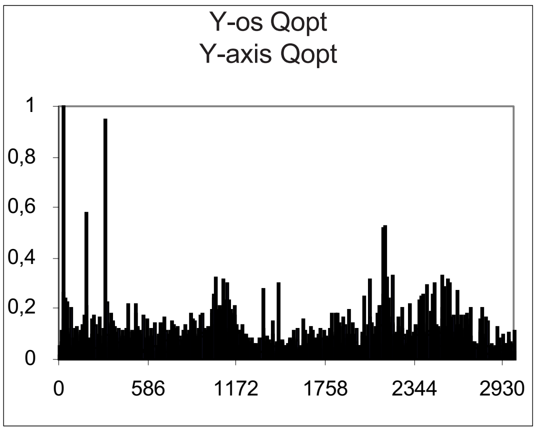

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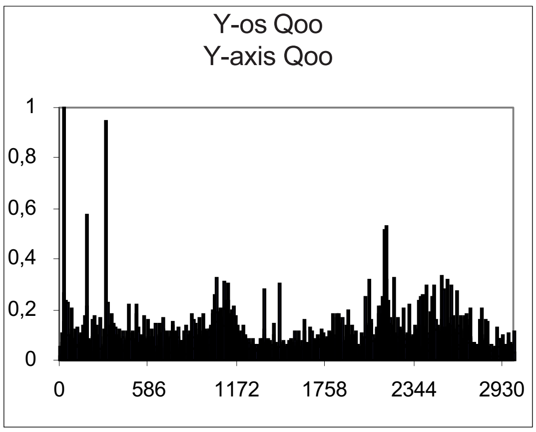

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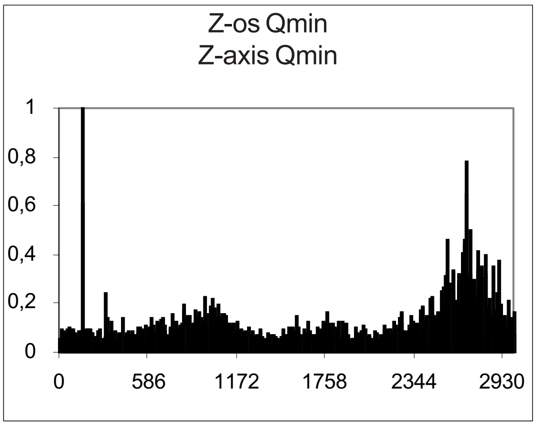
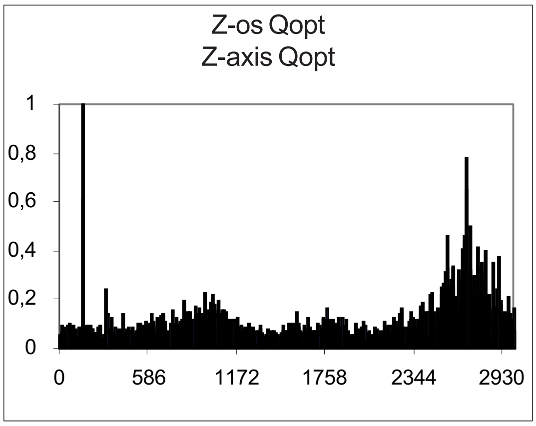


g)



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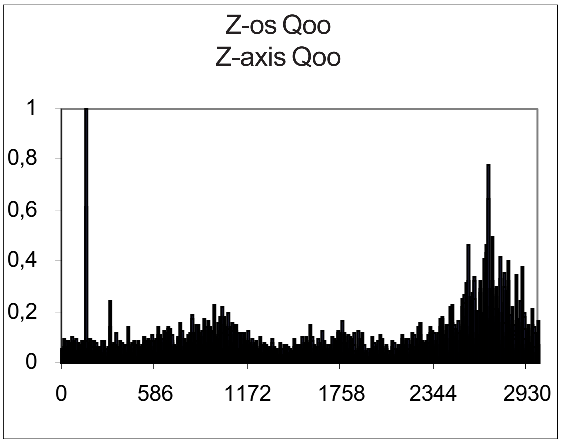


i)

Sl. 4. *Močnostni spektri vibracij okrova v treh koordinatnih smereh*
Fig. 4. *Power spectrum records of the pump casing vibrations in three directions*

V radialni smeri (x) je pri obratovanju z najmanjšim pretokom (sl. 4g) prevladujoča frekvenca lopatice (B1). V močnostnem spektru so opazne tudi njene višje harmonske (B2-B8). V zgornjem frekvenčnem območju je večja množica frekvenc, ki pripadajo tokovno vzbujenim vibracijam. S povečevanjem pretoka skozi črpalko do optimalnega (sl. 4h) in nadoptimalnega (sl. 4i) dobimo podobne rezultate, [10].

Močnostni spekter obratovalnega hrupa, prikazanega na sliki 5 je šibkejši od močnostnega spektra vibracij okrova. V vseh treh koordinatnih smereh (sl. 5a, b, c) prevladuje prva višja harmonska frekvence lopatic (B2). Amplituda te frekvence je pri spremembi pretoka od najmanjšega do največjega domala konstantna. Zato sta frekvenca pulzacije toka oziroma lopatična frekvenca (B1) in njena prva višja harmonska (B2) glavna vira zvoka.

Zdaj je problem ustrezno določiti amplitudo prvega višjega harmonika pulzacije toka. Obratovalni hrup lahko določimo z enačbo:

In the radial direction (*x*–direction) during minimum pump operating capacity the blade frequency (B1) amplitude is dominant in the power spectrum's record (Figure 4.g). Also all higher harmonics of the blade frequency (B2 – B8) are present in the record. In the upper frequency range, the larger frequency amplitude crowd that belongs to flow-induced casing vibrations in the radial direction is evident. By increasing the capacity up to the optimum (Figure 4.h) and up to over-optimum (Figure 4.i) a similar frequency situation is evident [10].

From the power spectra's records (Figure 5) of pump operating noise, a much weaker record in comparison with casing vibration is evident. For all three records (Figure 5.a,b,c) the first higher harmonic's amplitude of the blade frequency (B2) is dominant. The amplitude of this highest peak in the record is practically constant during the capacity change from minimum up to maximum capacity. Therefore, the flow pulsation frequency, or blade frequency (B1), and its first higher harmonic (B2) are the main noise sources.

However, the problem is how to estimate the first higher harmonic's amplitude of the basic flow pulsation frequency amplitude properly. The pump operating noise can be evaluated by the equation:

$$P_{rad} = a_P \cdot \rho \, c_2^3 \left(\frac{c_2^2}{c_0} \right)^2 D_2^2 \qquad (14),$$

kjer so c_2 - absolutna hitrost toka na izstopnem premeru rotorja D_2, in a_P funkciji tipa črpalke in vrtilne frekvence rotorja, podana z:

where c_2 is the absolute flow velocity at the pump impeller exit diameter D_2, and a_P is a function of the pump type and impeller speed in the following form:

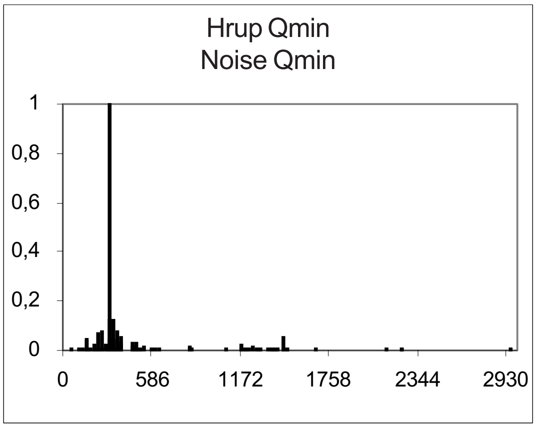
$$a_P = 4.8 \cdot 10^{-5} \left[\frac{n}{n_{des}} \right]^{\frac{n}{n_{des}}} \qquad (15),$$

kjer sta n - število vrtljajev rotorja, n_{des} - preračunsko število vrtljajev rotorja. Številčna konstanta v enačbi je eksperimentalno določena srednja vrednost merilnih rezultatov.

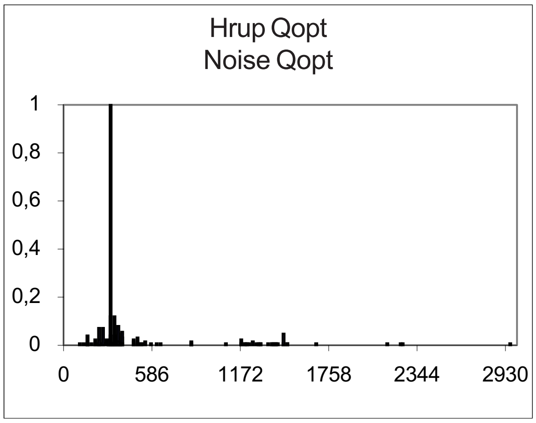
S slike 3 je razvidno dobro ujemanje med teoretično določenim obratovalnim hrupom in eksperimentalnimi rezultati.

where n is the impeller speed and n_{des} is the impeller design speed. The numerical constant in equation is experimentally determined as the mean-common value of the measuring results.

There is a good agreement between the theoretically determined pump operating noise and that of the experimental measurements (Figure 3).



a)



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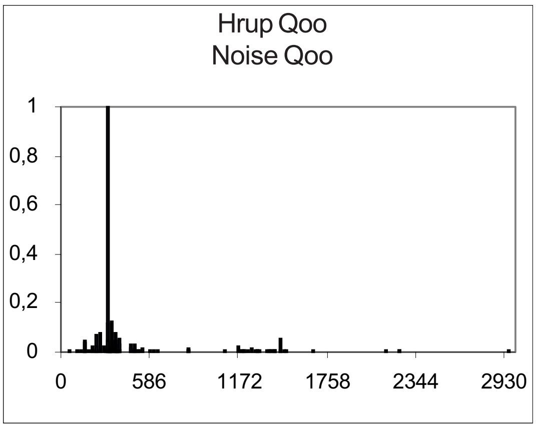

c)

Sl. 5. *Močnostni spekter obratovalnega hrupa radialne črpalke*
Fig. 5. *Power spectrum records of the radial pump operating noise*

3 SKLEP IN RAZLAGA

V večini testiranih obratovalnih režimov sta prevladujoči frekvenci vrtenja rotorja (I1) in prva višja harmonska frekvence lopatice (B2). Tako lahko sklenemo, da sta prevladujoča vira obratovalnega hrupa in vibracij okrova vrtilna frekvenca rotorja in pulzacija toka na izstopu rotorja s frekvenco lopatic.

3 CONCLUSIONS AND COMMENTARY

In almost all tested operating regimes the first higher harmonic of the impeller speed frequency (I1) and the first higher harmonic of the blade frequency (B2), are dominated by the amplitude in the power-spectra records. Therefore, the dominant sources of the operating noise and casing vibrations are the impeller speed and pulsating flow with the blade frequency at the impeller exit diameter.

Amplituda obratovalnega hrupa in vibracij okrova pada z manjšanjem pretoka skozi črpalko in manjšanjem vrtilne frekvence rotorja.

Ker v močnostnem spektru prevladujeta prva višja harmonika vrtilne frekvence in frekvence lopatice, lahko povzamemo, da je za zmanjšanje obratovalnega hrupa treba znižati vrtilno frekvenco rotorja. Če želimo dosegati zahtevano črpalno višino pri znižani vrtilni frekvenci, moramo povečati izstopno hitrost. To lahko dosežemo na dva načina: prvič: s povečanjem izstopnega kota rotorske lopatice, ki pa je na žalost omejena s trdnostjo materiala na 25° do 40°; drugič: s povečanjem izstopnega premera, ki pa je omejen z optimalnim razmerjem vstopnega in izstopnega premera. Tako moramo poiskati najugodnejšo rešitev z optimizacijskim procesom.

With a capacity and impeller speed decrease the noise and casing-vibration amplitudes also decrease.

Finally, in almost all power-spectra records the first higher harmonics of the impeller speed frequency and of the blade frequency dominate. Therefore, for minimizing operating noise the impeller speed must be decreased. However, if the required head and the capacity at lower impeller speed is to be satisfied, the flow velocities at the impeller exit must be increased. There are two possibilities to achieve this: first, by increasing the impeller-blade exit angles, but unfortunately they are limited in the range of 25 to 40 degrees by material strength; and second, by an exit diameter increase, which is also limited by an optimum intake / exit diameter ratio. So, for the best solution the optimising process must be used.

4 SIMBOLI
4 SYMBOLS

konstanta (funkcija tipa črpalke)	a_p	-	constant (function of the type of the pump)
sferična površina	A_s	m^2	spherical surface area
hitrost zvoka	c_0	m/s	acoustic speed
absolutna hitrost toka na izstopnem premeru	c_2	m/s	absolute flow velocity at exit diameter
premer rotorja	D	m	impeller diameter
izstopni premer rotorja	D_2	m	impeller exit diameter
frekvenca	f	Hz	frequency
črpalna višina	H	m	pump head
zvočna intenzivnost	I	W/m^2	acoustic intensity
referenčna zvočna intenzivnost	I_{ref}	W/m^2	reference acoustic intensity
stopnja zvočne intenzivnosti	L_I	-	sound intensity level
stopnja zvočne jakosti	L_n	-	sound power level
stopnja zvočnega tlaka	L_S	-	sound pressure level
število vrtljajev rotorja	n	l/s	impeller speed
preračunsko število vrtljajev rotorja	n_{des}	l/s	impeller design speed
zvočni tlak	p	Pa	acoustic pressure
vzbujevalna tlačna napetost	$\|p\|$	Pa	exciting stress
dinamični tlak	p_d	Pa	dynamic pressure
referenčni tlak	p_{ref}	Pa	reference pressure
zvočna moč	P	W	sound power
sevajoča zvočna moč	P_{rad}	W	radiated sound power
referenčna zvočna moč	P_{ref}	W	reference sound power
prostorninski pretok	Q	m^3/s	capacity
najmanjši prostorninski pretok	Q_{min}	m^3/s	minimum capacity
optimalni prostorninski pretok	Q_{opt}	m^3/s	optimum capacity
nadoptimalni prostorninski pretok	Q_{00}	m^3/s	over optimum capacity
čas	t	s	time
opazovan časovni trenutek	t_0	s	observing time
čas periode	T	s	period time
neustaljena, oscilirajoča komponenta hitrosti	u	m/s	unsteady, fluctuating part of velocity
hitrost v točki tekočinskega toka	U	m/s	velocity at a point in fluid flow
povprečna hitrost	$\bar{U}$	m/s	average value of velocity
vršna hitrost	U_T	m/s	tip speed
padec tlaka	Δp	Pa	pressure drop
gostota	ρ_0	kg/m^3	density

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