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Prispevek k analizi sil na vodilni lopatici modelske reverzibilne črpalke - turbine v črpalnem obratovalnem režimu

Contribution to the Analysis of Forces on the Guide-Vane of the Pump-Turbine Model in Pumping Regime

ANDREJ PREDIN

V prispevku so obravnavane tokovne razmere v vodilniku modelske reverzibilne črpalke - turbine, analiza sil in momentov ter merilni sistem za merjenje sil in momentov na osi vodilne lopatice. Podan je predlog matematičnega modela nihanja vrtilnega momenta na osi vodilne lopatice in primerjava merilnih rezultatov z izračuni.

Ključne besede: turbine - črpalke reverzibilne, vodilniki turbin, razmere tokovne, sistemi merilni

In this contribution the flow conditions at the guide apparatus of the reversible pump-turbine model, the analysis of forces and torque at guide-vane shaft, and measurement system for force and torque measurement at guide-vane are treated. A new mathematical model is given for guide-vane shaft torque modeling, comparison of the measurement results with theoretical calculations, and comments on the transfer form model to original properties.

Keywords: reversible pump turbines, guide apparatus, flow conditions, measurements systems

0 UVOD

Pri sodobnem optimiranju vodilnih sistemov, predvsem vodilnih lopatic, je potrebno natančno poznavanje obremenitev na osi vodilne lopatice. Premer osi omejuje optimalno oblikovanje oziroma izbiro ustreznega profila vodilne lopatice. Zaradi velike koncentracije obremenitev na mestu prehoda osi v lopatico mora biti os na tem mestu nekoliko tanjša od profila. Znano je, da so tanjši profili hidravlično ugodnejši od debelejših. Z raziskovanjem na modelih je mogoče določiti obremenitve, ki jih lahko ob upoštevanju pravil podobnega obratovanja uporabimo na izvedbah.

1 OBLIKA DELUJOČE HIDRODINAMIČNE SILE

Iz dosedanjih raziskav [1] in dostopne literature je znano, da je fluidni tok [2] na izstopu iz radialnega rotorja črpalk - turbin izrazito utripne in deloma tudi naključne narave. Utrip toka je izrazit predvsem v črpalnem obratovalnem režimu, kjer je tok pojemajoč in se z večanjem premera proti izstopu iz vodilnika umirja. Utripni tok nastane zaradi relativnega vrtinca v rotorskih kanalih s končnim številom lopatic končne debeline [3]. Posledica delovanja takšnega toka [4] na vodilne lopatice je utripajoča hidro-dinamična sila in njen vrtilni moment na osi [5]. Ta obremenjuje os vodilne lopatice z utripno, deloma tudi z izmenično obremenitvijo pri pretokih, manjših od optimalnega, vendar takšna ocena še ni popolna. Obremenitve vodilne lopatice in njene osi je treba obravnavati kompleksneje, kot dinamični sistem, pri katerem so upoštevana lastna, vsiljena in superponirana nihanja celotnega osnovnega vodilnega sklopa.

0 INTRODUCTION

For present-day optimisation of the guide systems, especially guide vanes, knowledge of the exact strength loads on the guide-vane shaft is necessary. The guide-vane shaft diameter limits the optimal modelling or selection of the suitable guide-vane foil. Because of the great load's concentration on the place where the guide-vane shaft enters the guide-vane foil, the shaft diameter must be thinner than the guide-vane foil at this place. It is known that the thinner guide-vane foils are hydraulically more suitable than the equal thicker guide-vane foils. By means of model research, and by considering the similarity laws, the acting loads on the prototype may be determined.

1 PROPERTIES OF ACTING HYDRO-DYNAMIC FORCE

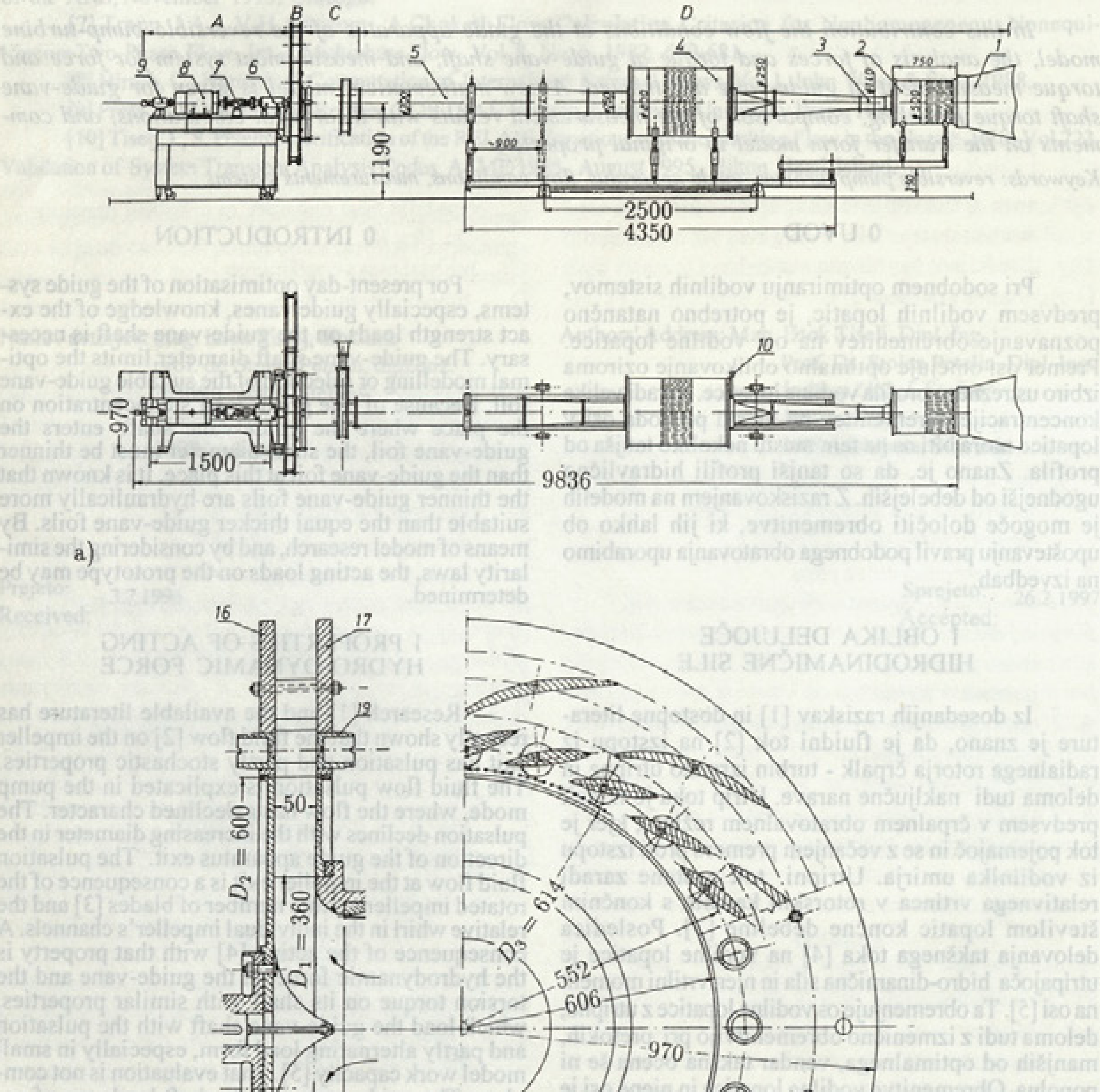
Research [1] and the available literature has recently shown that the fluid flow [2] on the impeller exit has pulsation and partly stochastic properties. The fluid flow pulsation is explicated in the pump mode, where the flow has a declined character. The pulsation declines with the increasing diameter in the direction of the guide apparatus exit. The pulsation fluid flow at the impeller exit is a consequence of the rotated impeller a finite number of blades [3] and the relative whirl in the individual impeller's channels. A consequence of the acting [4] with that property is the hydrodynamic force on the guide-vane and the torsion torque on its shaft with similar properties, which load the guide-vane shaft with the pulsation and partly alternating load form, especially in small model work capacity [5]. That evaluation is not complete. The guide-vane and its shaft loads are of necessity treated as a complex dynamic system by considering the eigen, forced and superposed oscillations of the whole guide system - the guide-vane with its shaft.

2 MERILNA PROGA IN TESTNI MODEL

Eksperimentalne raziskave so bile izvedene v Laboratoriju za turbinske stroje Fakultete za strojništvo v Mariboru na instaliranem merilnem sistemu (sl. 1a), ki obsega poenostavljeni model (sl. 1b) izvedene reverzibilne črpalke turbine, prirejenem za obratovanje v črpalnem režimu z zrakom kot pretočnim fluidom [6]. Model ima radialni rotor konstantne širine b_2=0,05 m z enajstimi lopaticami, vodilnik s štiriindvajsetimi profiliranimi in z dvanajstimi podpornimi vodilnimi lopaticami. Na izstopu je nameščen spiralni vodilnik pravokotnega pretočnega prereza konstantne širine b_{spv} = 0,12 m. Vstopni premer rotorja D_1 = 0,36 m, izstopni pa D_2 = 0,6 m. Vstopni robovi vodilnih lopatic leže na premeru D_3 = 0,624 m.

2 EXPERIMENTAL SYSTEM AND TESTED MODEL

The experimental research was carried out in the Laboratory for turbine machines at the Faculty of Mechanical Engineering in Maribor where the installed measurement system [6] (Fig. 1.a), which occupies the simplified reversible pump-turbine model (Fig. 1.b), is adapted for work with air as the fluid medium. The model has a radial impeller constant width (b_2 = 0.05), eleven impeller blades , twenty-four foiled guide-vanes, twelve foiled support vanes , and a spiral volute with a rectangular form, constant width (b_{spv} = 0.12 m) on the model exit. The impeller intake diameter is D = 0.36 m and the exit diameter is D_2 = 0,6 m. Blade vanes intake diameter is D_3 = 0.624 m.



Sl. 1. *Eksperimentalni sistem [11] a) in poenostavljeni model reverzibilne črpalke - turbine b)*

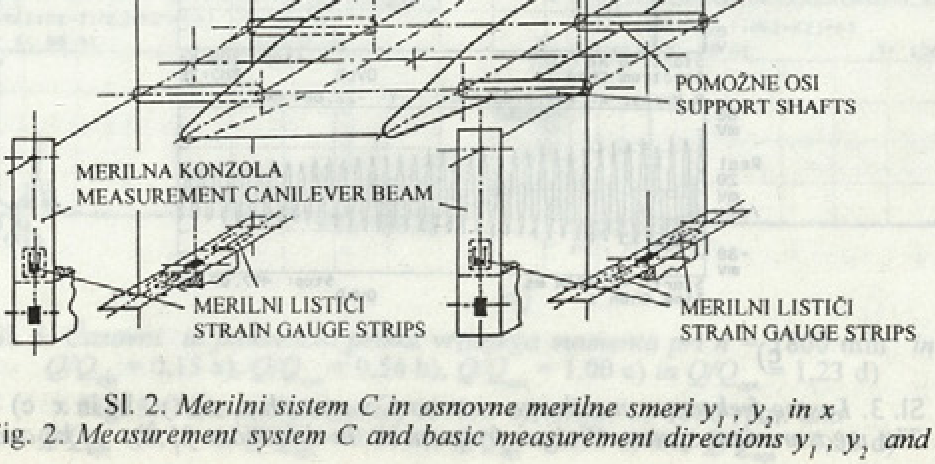
Fig. 1. *Experimental system [11] a) and simplified reversible pump-turbine model b)*

3 MERILNI SISTEMI

V okviru raziskovalnega dela je bilo razvitih več merilnih sistemov [7]. Zadnji izvedeni je merilni sistem, ki temelji na načelu tehtnice sil, imenovan sistem C (sl. 2). Na dveh pomožnih oziroma merilnih oseh je na prednapetih merilnih konzolah vpeta merilna vodilna lopatica. Na merilnih konzolah so nameščeni merilni lističi (HBM 3/350 LY13) povezani v tripolni Wheatstonov mostič, s čimer je dosežena temperaturna izravnava. Trije merilni kanali zagotavljajo merjenje pomikov in komponent hidrodinamične sile v dveh smereh y_1 in y_2, v smeri pravokotno na skeletnico profila in v smeri skeletnice profila, smer x. Predpostavljena je linearna interakcija merilnih konzol med osnovnimi smermi delovanja y_1, y_2 in smeri x [8]. Ker je profil vodilne lopatice nesimetričen v smeri y, ima merilni sistem tri lastne frekvence. Lastne frekvence sistema so določane eksperimentalno z impulznim vzbujanjem merilnega sistema in s spektralnim analizatorjem HP 323 60C. Lastna frekvenca vodilnega sistema v smeri y_1 je 94 Hz, v smeri y_2 je 56 Hz in v smeri x 74 Hz (sl. 3). Iz upadajočih amplitud nihajnih zapisov lahko sklenemo, da ima merilni sistem C majhno dušenje, kar je ugodno za spremljanje časovno odvisnih obremenitev. Umerjanje merilnega sistema je izvedeno vsakokrat neposredno pred merjenjem z utežmi na merilnih oseh, v vseh treh merilnih smereh.

3 MEASUREMENT SYSTEM

An earlier measurement system was developed in previous research work [7]. The latest measurement system, which is based on the force scale principle, is being developed. The measurement system is briefly named as system C (Fig. 2). The measurement guide-vane is fixed on two support measurement shafts, which are fixed on the measurement cantilever beams. The measurement strain gauge strips (HBM 3/350 LY 13) are placed on cantilever beams and connected to the full Wheatstone bridge. In this way the temperature compensation is achieved. The three measurement channels assure the hydrodynamic force component measurement in two directions y_1 and y_2 in normal direction on the guide-vane foil and in direction x in the tangential direction on the guide-vane foil. The linear interaction [8] of the measurement cantilever beams among acting directions (y_1, y_2 and x) is proposed. The measurement system C has three eigen frequencies because of the non symmetrical guide-vane foil in the y direction. The eigen frequencies are determined experimentally by the impulse force action on the measurement guide-vane and by spectral analyser HP 323 60C. The eigen frequency of the measurement system in direction y_1 is 94 Hz, in direction y_2 is 56 Hz, and in direction x is 74 Hz (Fig. 3). From the amplitudes of the oscillation record the low system damping may be determined. This is suitable for time-dependent changeable value measurements. The measurement system is calibrated each time before the performed measurement with the weights placed on the measurement shafts in all three measuring directions.



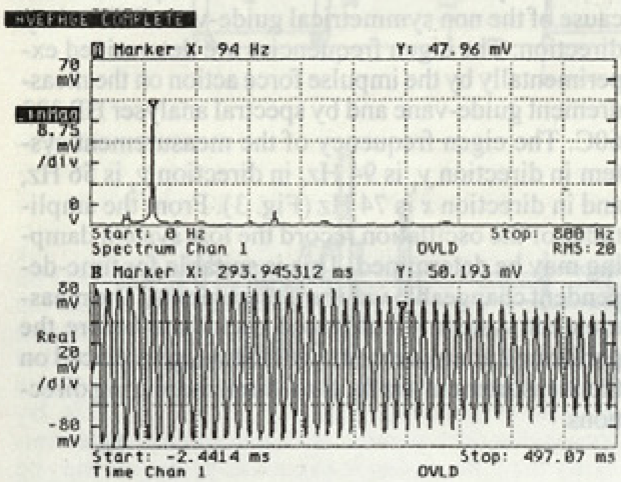
Sl. 2. *Merilni sistem C in osnovne merilne smeri y_1, y_2 in x*
Fig. 2. *Measurement system C and basic measurement directions y_1, y_2 and x*

4 REZULTATI MERITEV

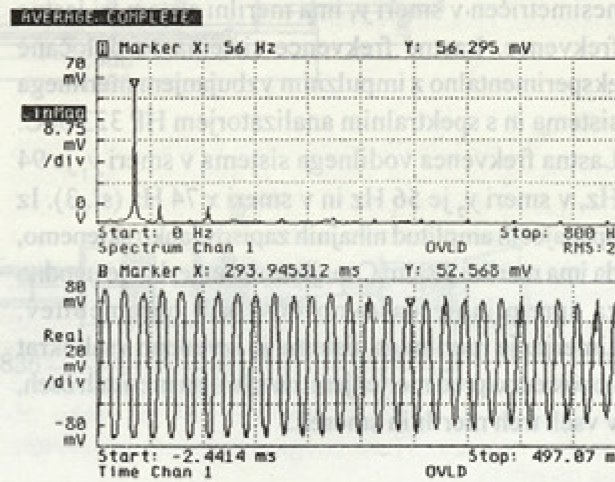
Zapis srednjih komponent hidrodinamične sile F_n in F_t oziroma koeficienta sile c_{Fn} in c_{Ft} kaže na zmanjšanje obeh proti optimalnemu pretoku Q/Q_{opt} = = 1,00 (sl. 3a). Absolutno je komponenta sile v normalni smeri večja od komponente v obodni smeri, kar je tudi razumljivo, ker vodilna lopatica preusmerja tok predvsem v vzdolžni smeri. V področju pretokov Q/Q_{opt} = 0,15 do 0,60 je komponenta sile v vzdolžni smeri razmeroma konstantna, kar kaže na obstoj "mrtvega" cirkulacijskega toka na izstopu iz rotorja. Zapis srednjih vrtilnih momentov (sl. 4b) kaže na zvečanje vrtilnega momenta v območju pretokov Q/Q_{opt} = 0,15 do 0,80, celo večjih kakor pri obratovanju modela z optimalnim pretokom Q/Q_{opt} =1,00 in skoraj linearno zvečevanje vrtilnega momenta v območju pretokov Q/Q_{opt} =1,00 do 1,23.

4 MEASUREMENT RESULTS

The records of force components F_n and F_t, and, force coefficients c_{Fn} and c_{Ft} have shown that both forces declined in the defection of optimal work capacity (Q/Q_{opt} = 1.00) (Fig. 3a). The force component in normal direction is absolute more than the component in the tangential direction. This is understandable because the guide-vane guides the fluid flow more in this direction than in tangential direction. In the area of under optimal capacities (Q/Q_{opt} = 0.15 to 0.60) the force component in normal direction is relatively constant. This is shown by the existing "dead" circulation flow around the impeller's exit. The mean torque record (Fig. 4b) has shown on the torque an increase in the area of under optimal capacities (Q/Q_{opt} = 0.15 to 0.80) which are even bigger than the torque by the optimal capacity (Q/Q_{opt} = 1.00) and almost linear torque increasing in the area of over optimal capacities (Q/Q_{opt} = 1.00 to 1.23).



a)



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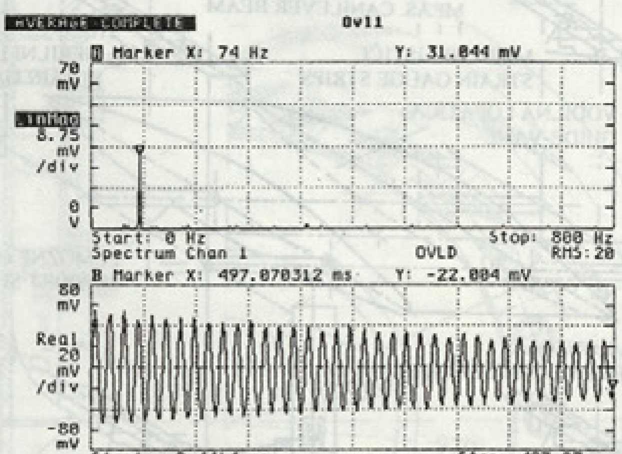
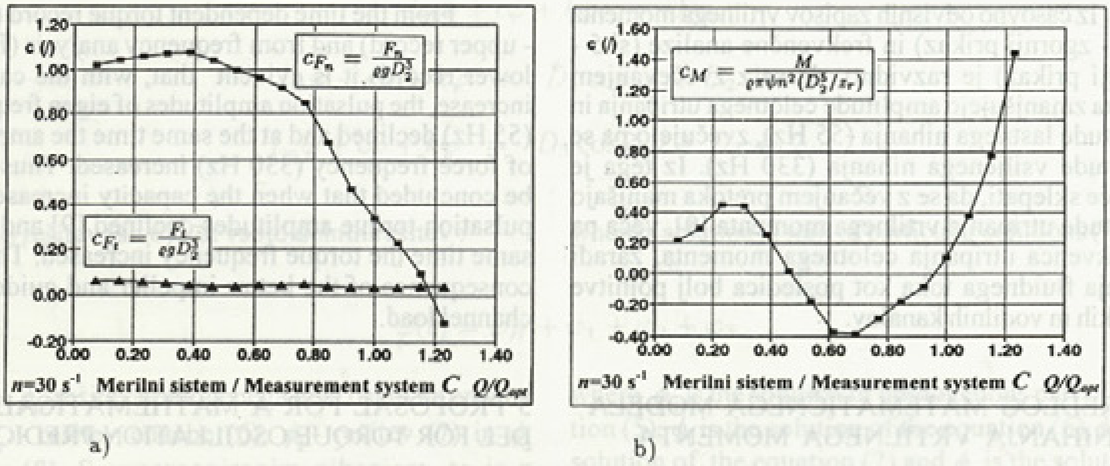


c)

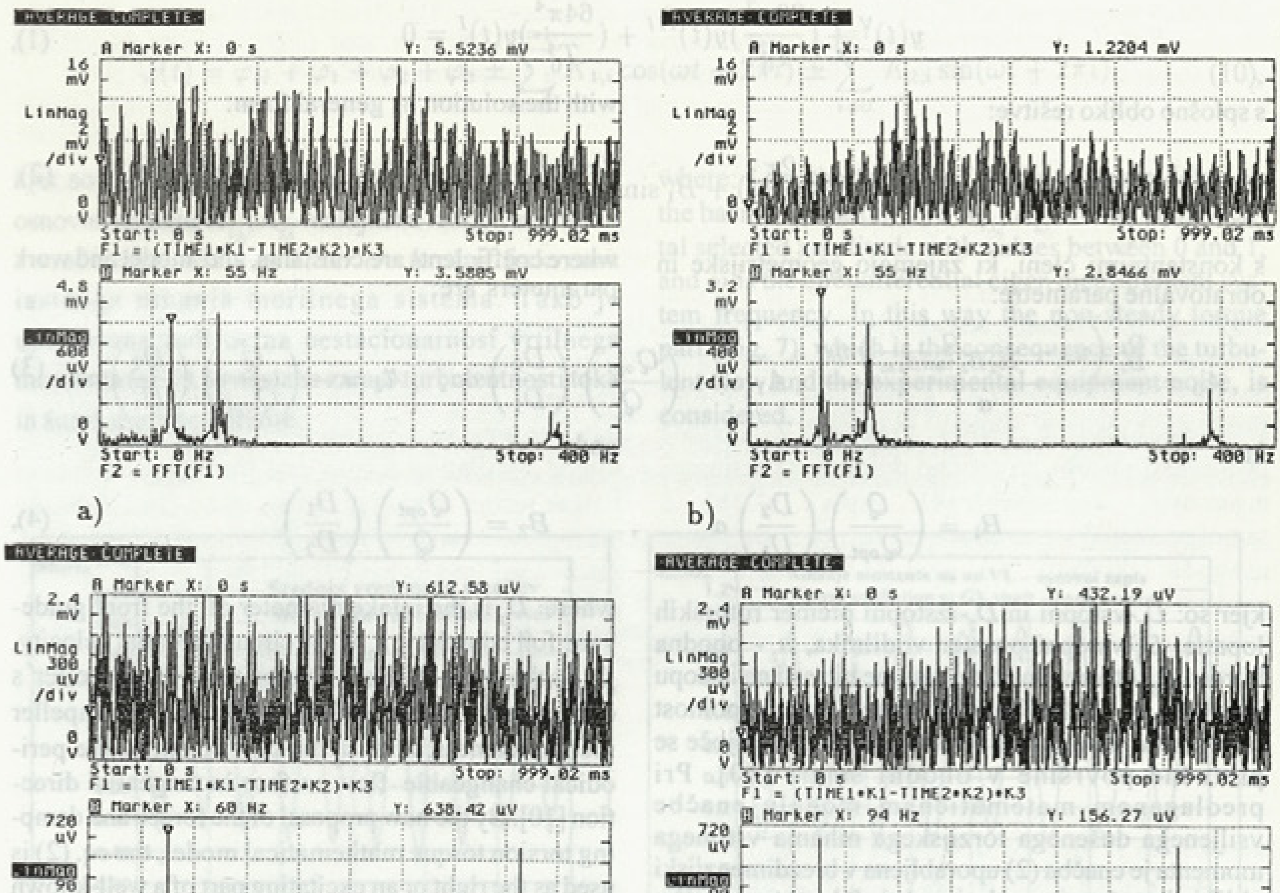
Sl. 3. *Lastne frekvence merilnega sistema C v smereh* y_1 a), y_2 b) in x c)

Fig. 3. *The measurement system C eigen frequences: in directions* y_1 a), y_2 b) *and* x c)



Sl. 4. *Koeficient vzdolžne* c_{Fn} in *obodne* c_{Ft} *komponente hidrodinamične sile* a) *in koeficient vrtilnega momenta* c_M b)

Fig. 4. *Hydrodynamic force coefficient in normal* c_{Fn} *and tangential direction* c_{Ft} a), *with torque coefficient* c_M b)



Sl. 5. *Časovni in frekvenčni prikaz vrtilnega momenta pri* n = 1800 min^{-1} *in* Q/Q_{opt} = 0,15 a), Q/Q_{opt} = 0,54 b), Q/Q_{opt} = 1,00 c) *in* Q/Q_{opt} = 1,23 d)

Fig. 5. *Time and frequency review of torque at* 1800 rpm. *and* Q/Q_{opt} = 0,15 a), Q/Q_{opt} = 0,54 b), Q/Q_{opt} = 1,00 c) *and* Q/Q_{opt} = 1,23 d)

Iz časovno odvisnih zapisov vrtilnega momenta (sl. 5 - zgornji prikaz) in frekvenčne analize (sl.5 - spodnji prikaz) je razvidno, da se z zvečevanjem pretoka zmanjšujejo amplitude celotnega utripanja in amplitude lastnega nihanja (55 Hz), zvečujejo pa se amplitude vsiljenega nihanja (330 Hz). Iz tega je mogoče sklepati, da se z večanjem pretoka manjšajo amplitude utripanja vrtilnega momenta [9], veča pa se frekvenca utripanja celotnega momenta, zaradi urejanja fluidnega toka kot posledica bolj polnitve rotorskih in vodilnih kanalov.

From the time dependent torque record (Fig. 5 - upper record) and from frequency analysis (Fig. 5 - lower record) it is evident that, with the capacity increase, the pulsation amplitudes of eigen frequency (55 Hz) declined and at the same time the amplitude of force frequency (330 Hz) increased. Thus it can be concluded that when the capacity increased, the pulsation torque amplitudes declined [9] and at the same time the torque frequency increased. This is a consequence of the better impeller and guide-vane channel load.

5 PREDLOG MATEMATIČNEGA MODELA NIHANJA VRTILNEGA MOMENTA

5 PROPOSAL FOR A MATHEMATICAL MODEL FOR TORQUE OSCILLATION PREDICTION

Z namenom, da bi natančneje modelirali nihanje vrtilnega momenta, je podan predlog matematičnega modela utripnih hitrosti fluidnega toka, ki temelji na linearni diferencialni enačbi petega reda:

In the research work the proposal for a mathematical model of pulsation flow velocity has been given. The basis of the model is the fifth order differential equation:

$$y(t)^{V} + \left(\frac{20\pi^2}{T_0^2}\right)y(t)^{III} + \left(\frac{64\pi^4}{T_0^4}\right)y(t)^{I} = 0 \qquad (1),$$

s splošno obliko rešitve:

with the solution in general form:

$$y(t) = c_u(t) = A_0 + A_1\cos\left(\frac{2\pi}{T_0}t\right) + B_1\sin\left(\frac{2\pi}{T_0}t\right) + A_2\cos\left(\frac{4\pi}{T_0}t\right) + B_2\sin\left(\frac{4\pi}{T_0}t\right) \qquad (2),$$

s konstantnimi členi, ki zajemajo geometrijske in obratovalne parametre:

where coefficients are constants, and model and work parameters are:

$$A_0 = \frac{\frac{D_2}{D_3}\left(u_2 - \frac{Q}{\pi b_2 D_2 \tan\beta_{2\infty}}\right)\eta_h}{a}, \quad A_1 = -\left(\frac{Q_{opt}}{Q}\right)\left(\frac{D_2}{D_1}\right)a a_s, \quad A_2 = -\left(\frac{Q}{Q_{opt}}\right)\left(\frac{D_2}{D_1}\right)a \qquad (3)$$

in

and

$$B_1 = \left(\frac{Q}{Q_{opt}}\right)\left(\frac{D_2}{D_1}\right)a \quad , \quad B_2 = \left(\frac{Q_{opt}}{Q}\right)\left(\frac{D_1}{D_2}\right) \qquad (4),$$

kjer so: D_1-vstopni in D_2-izstopni premer rotorskih lopatic, D_3-vstopni premer vodilnika, u_2 - obodna hitrost, $\beta_{2\infty}$- teoretični kot relativne hitrosti na izstopu iz rotorja, η_h - hidravlični izkoristek, a - nepopolnost rotorja in a_s-koeficient periodično spreminjajoče se pretočne površine v obodni smeri [10]. Pri predlaganem matematičnem modelu enačbe vsiljenega dušenega torzijskega nihanja vrtilnega momenta je enačba (2) uporabljena v brezdimenzijski obliki, kot desni - vzbujevalni del znane enačbe dušenega torzijskega nihanja:

where: D_3 is the intake diameter of the front guide-vane foil boundary, u_2 is the circumferential velocity, $\beta_{2\infty}$ is the angle of relative velocity on the impeller's exit, η_h is the hydraulic efficiency, a is the impeller incompleteness, and a_s the coefficient of the periodical changeable flow surface in tangential direction [10]. By the new proposal of the forced and damping torsion torque mathematical model, the eq. (2) is used as the right or an excitating part of a well-known equation:

$$I_m\ddot{\varphi} + c\dot{\varphi} + k\varphi = M_s\left(1 + \frac{A_1}{A_0}\cos\omega t + \frac{B_1}{A_0}\sin\omega t + \frac{A_2}{A_0}\cos 2\omega t + \frac{B_2}{A_0}\sin 2\omega t\right) \qquad (5),$$

kjer je M_s srednji vrtilni moment. S postavitvijo koeficientov v nove konstante in razdružitvijo zgornje diferencialne enačbe v tri diferencialne enačbe:

where M_s is the mean torque. With the coefficients setting into the new constants and with the above equation (5) separating into the three following differential equations:

$$I_m\ddot{\varphi} + c\dot{\varphi} + k\varphi = D_0 \quad (6),$$

$$I_m\ddot{\varphi} + c\dot{\varphi} + k\varphi = D_1 \cos \omega t + D_2 \sin \omega t \quad (7),$$

$$I_m\ddot{\varphi} + c\dot{\varphi} + k\varphi = D_3 \cos 2\omega t + D_3 \sin 2\omega t \quad (8)$$

dobimo rešitev (sl. 6) kot vsoto delnih rešitev:

where a solution is sum of particular solutions (Fig.6):

$$\varphi(t) = \varphi_H + \varphi_1 + \varphi_2 + \varphi_3 \quad (9),$$

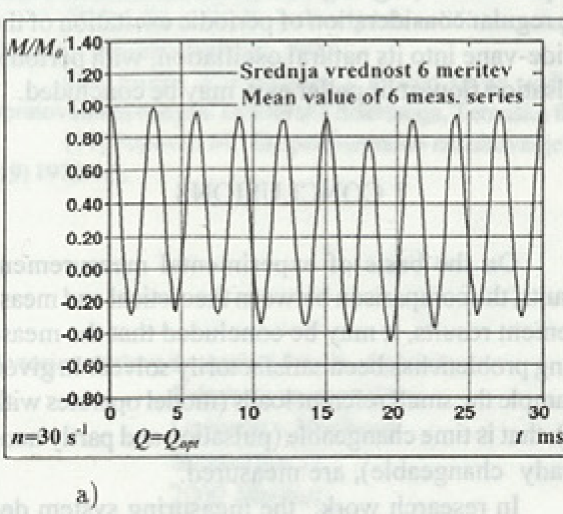
kjer so: ϕ_H-rešitev homogenega dela osnovne enačbe (5), ϕ_1 - rešitev enačbe (6), ϕ_2 - rešitev (7) in ϕ_3 rešitev (8). S superponiranim nihanjem, to je z upoštevanjem ponavljajočega se vzbujanja lastnega nihanja vodilne lopatice z utripnim tokom, lahko simuliramo vrtilni moment z enačbo:

where ϕ_H is the solution of homogenous part of equation (5), ϕ_1 is the solution of the equation (6), ϕ_2 is the solution of the equation (7) and ϕ_3 is the solution of the equation (8). With superposed oscillations it can be considered that the repeated eigen oscillation is agitated with the pulsation fluid flow and the torque can be simulated by the equation:

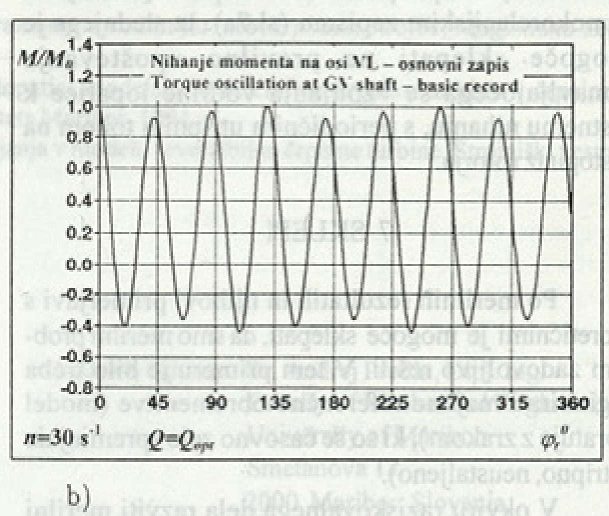
$$\varphi(t) = \varphi_H + \varphi_1 + \varphi_2 + \varphi_3 \pm \sum_{i=0}^{i=1} K_{1,i} \cos(\omega t + 2\pi i) \pm \sum_{i=0}^{i=1} K_{2,i} \sin(\omega t + 2\pi i) \quad (10),$$

kjer so i - faktor kotnega faznega odmika glede na osnovno nihanje, $K_{1,i}$, $K_{2,i}$ - naključno izbrani amplitudi z vrednostjo med 0 in 1, ter ω - krožna frekvenca lastnega nihanja merilnega sistema. Tako je upoštevana tudi delna nestacionarnost vrtilnega momenta (sl. 7), ki nastane zaradi turbulentnosti toka in šuma merilne opreme.

where: i are the factors of angle or phase delay on the basic oscillation. The $K_{1,i}$, $K_{2,i}$ are the coincidental selected amplitude with values between 0 and 1, and ω is the circumferential eigen measurement system frequency. In this way the non-steady torque part (Fig. 7), which is the consequence of the turbulent flow and the experimental equipment noise, is considered.



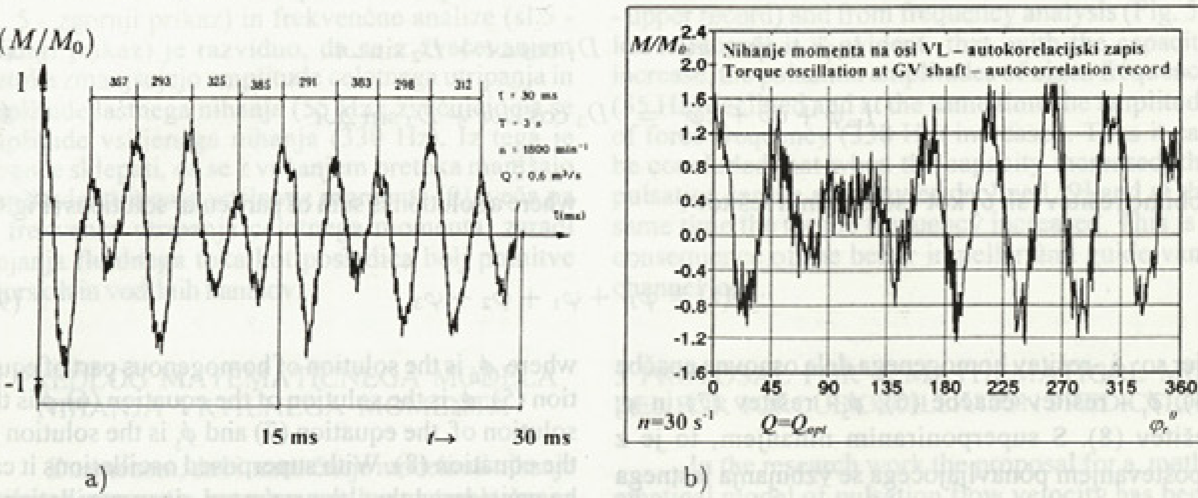
a)



b)

Sl. 6. *Nihanje vrtilnega momenta na osi vodilne lopatice pri* n = 1800 min^{-1} in Q/Q_{opt} = 1,00 a) *ter izračunani grafični prikaz* b)

Fig. 6. *Guide-vane torque oscillations at* 1800 rpm. *and* Q/Q_{opt} = 1,00 a), *with calculated torque graphic review* b)



Sl. 7. *Samokorelacijski zapis vrtilnega momenta na osi vodilne lopatice pri* n = 1800 min^{-1} in Q/Q_{opt} = 1,00 a) *ter izračunani superponirani grafični prikaz* b)

Fig. 7. *Auto-correlation guide-vane torque record at* 1800 rpm. *and* Q/Q_{opt} = 1,00 a), *with calculated superposed torque graphic record* b)

6 PRIMERJAVA IZRAČUNANIH MOMENTOV Z IZMERJENIMI

Iz diagramskih prikazov (sl. 6a, 6b) je razvidno ujemanje izračunanih vrednosti nihanja vrtilnega momenta na osi vodilne lopatice z eksperimentalno določenimi vrednostmi oz. s srednjimi vrednostmi šestih ponovitvenih meritev. Prav tako je razvidna podobnost zapisov teoretično določanega vrtilnega momenta, superponirani zapis (sl.7b) z samokorelacijskim zapisom (sl.7a). Iz slednjega je mogoče sklepati na pravilno upoštevanje ponavljajočega se vzbujanja vodilne lopatice k lastnemu nihanju, s periodičnim utripnim tokom na izstopu iz rotorja.

7 SKLEPI

Po merilnih rezultatih in njihovi primerjavi s teoretičnimi je mogoče sklepati, da smo merilni problem zadovoljivo rešili. V tem primeru je bilo treba registrirati majhne referenčne obremenitve (model obratuje z zrakom), ki so se časovno zelo spreminjale (utripno, neustaljeno).

V okviru raziskovalnega dela razviti merilni sistem C podaja dobre rezultate, saj je z njim moč registrirati neposredno hidrodinamično silo na vodilni lopatici kakor tudi njene komponente in prijemališče, kar je bistvena prednost pred dosedanjimi merilnimi sistemi.

6 COMPARISON OF CALCULATED AND MEASURED RESULTS

The satisfactory agreement between the calculated and measured guide-vane torque mean oscillatory values, determined from six repetition measurements, is evident from diagrams (Fig. 6a and 6b). Also the great similarity between theoretically determined oscillatory torque and the auto-correlation torque record (Fig. 7a) can be seen. On this basis, the regular consideration of periodic excitation of the guide-vane into its natural oscillation, with periodic pulsation flow at impeller exit, may be concluded.

7 CONCLUSIONS

On the basis of experimental measurement results, the comparison between theoretical and measurement results, it may be concluded that the measuring problem has been satisfactorily solved. In given example the small referent loads (model operates with air), that is time changeable (pulsation and partly non-steady changeable), are measured.

In research work, the measuring system developed (system C) gives good results. The hydrodynamic force, its components and it acting handle, can be directly recorded. This is the main priority in comparison with the latest developed measuring systems.

Predlagani matematični model za teoretično napovedovanje nihanja momenta na osi vodilne lopatice daje sprejemljive rezultate, ki jih je mogoče uporabiti pri dimenzioniranju premera osi vodilne lopatice.

Glede na prenos razmer z modela na izvedbo je mogoče upoštevati samo kinematično podobnost, saj gre v konkretnem primeru za obratovanje z dvema različnima fluidoma (zrak - model, voda - izvedba).

The given mathematical model for theoretical torque on guide-vane shaft prediction gives acceptable results. This can be used for calculation of guide-vane shaft diameter dimensions.

Concerning model to original parameter transfer, only the kinematics similarity can be used, while in this case two different fluid mediums appear. The model operates with air, and the original facility operates with water.

6 LITERATURA
6 REFERENCES

[1] Popović, M.: Poročila raziskovalne naloge RSS od leta 1977 do leta 1992. Tehniška fakulteta, Univerza v Mariboru.

[2] Velenšek, B., K. Oberdank : Flow kinematics in a stay vane/vicked gate cascade of a low head Kaplan turbine. Experimental heat transfear. Fluid Mechanics, and Thermodynamics 1991, Elsevier Science Publishing Co. Inc. Amsterdam,1991.

[3] Predin A., M. Popovič: Contribution to the experimental analysis of fluid flow in the guide wheel of the reversible pump-turbine in pump mode. Heat Transfer and Fluid Mechanics Institute Proceedings, Sacramento California State University, 1993, 41-54.

[4] Rus T., M. Jamnik: Tokovno polje v kaskadi radialne črpalke - turbine. Kuhljevi dnevi '92, Portorož, 1992.

[5] Predin A., M. Popovič: The pivoted guide-vane oscillating in the turbocompressor. International Symposium on Gas Turbines and Gas Cycle Plants Proceedings. Faculty of Mechanical Engineering, University of Ljubljana, 1993, 119-130.

[6] Popović, M.: Proučavanje pulzacionog strujanja na izlazu iz obrtnog kola reverzibilne pumpe-turbine u pumpnom radnom režimu i njegovog uticaja na oscilacije obrtnog momenta lopatice sprovodnog kola. Doktorska disertacija, Mašinski fakultet, Beograd, 1979.

[7] Predin A.-M. Popovič: Contribution to the Measurements and Analysis of Pulsatory Hydrodynamic Forces and Torque on Guide Vanes of Pump-Turbine Systems. IMECO Technical Commitee Series; No.31 1993, 137-144.

[8] Hottner T.: Dreikomponenten-Windkanalwaage für interferenzfreie Messungen von Luftkraftkomponenten. Messtechnische Briefe 16, Heft 2, 1980.

[9] Predin A.:Vpliv pulzacijskih veličin na strujne razmere v vodilniku turbočrpalnih strojev. Magistrsko delo, Tehniška fakulteta Maribor, 1990.

[10] Predin A.:Prispevek k analizi sil na vodilni lopatici modelske reverzibilne črpalke turbine v črpalnem obratovalnem režimu. Doktorska disertacija, Tehniška fakulteta Maribor, 1994.

[11] Popovič, M.: Eksperimentalno raziskovanje strujanja v modelu reverzibilne črpalne turbine. Strojniški vestnik (19) 1973/4-5.

Avtorjev naslov: dr. Andrej Predin, dipl. inž.
Fakulteta za strojništvo
Univerze v Mariboru
Smetanova 17
2000 Maribor

Author's Address: Dr. Andrej Predin, Dipl. Ing.
Faculty of Mechanical Engineering
University of Maribor
Smetanova 17
2000 Maribor, Slovenia

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