

STROJNÍŠKI VESTNIK

JOURNAL OF MECHANICAL ENGINEERING

LETNIK
VOLUME 40

LJUBLJANA.

MAJ-JUNIJ
MAY-JUNE 1994

ŠTEVILKA
NUMBER 5-6

UDK 519.615:532.54:621.224.24

Numerična analiza in eksperimentalna overitev toka v francisovi turbini

Numerical Analysis and Experimental Validation of the Flow in a Francis Turbine

ANDREJ LIPEJ — DRAGICA JOŠT — MATEJA JAMNIK — BORIS VELENŠEK

Pri razvoju novih in pri obnovi sedanjih vodnih turbin je zelo pomembno poznavanje natočnih razmer na gonilnik. To lahko dosežemo z meritvami na modelu ali pa z numerično analizo toka v delih turbine pred gonilnikom. V članku so predstavljene numerične analize turbulentnega toka pri obtekanju lečastih loput, predvodilnih in vodilnih lopatic in tok v spirali. Primerjava z meritvami je pokazala, da lahko v teh delih turbine z numerično analizo toka nadomestimo meritve.

Boundary conditions at the inlet of the runner are very important for the development of new turbines or for refurbishment. Inlet conditions can be obtained by model measurements or numerical flow analysis in different parts upstream of the runner. Numerical flow analysis of turbulent flow past butterfly valves, stay vanes, guide vanes and in spiral casing is presented. Comparison between the numerical analysis and measurements shows that the experiment can be replaced by computational fluid dynamics methods.

0 UVOD

V zadnjem času se je uporaba računalnikov pri izračunu tokovnih razmer v turbinskih strojih zelo povečala. Numerični rezultati lahko mnogokrat nadomestijo drage in zamudne meritve. Najboljša rešitev je poiskati primerno razmerje med izračuni in meritvami. Meritve so še vedno nujno potrebne za poznavanje robnih pogojev in primerjave rezultatov. Naloga, opisana v članku, je bila izračunati tok skozi francisovo turbino. Zaradi zahtevnosti smo se odločili le za izračun toka okoli loput, v spirali in skozi predvodilnik ter vodilnik. Vsak del smo računali posebej, kajti računalnikov pomnilnik ni omogočal izračuna celotnega pretočnega trakta. Reynoldsovo število je pri toku v turbinskih strojih zelo veliko (reda velikosti 10^6). V takšnih primerih je tok turbulenten in za izračun smo izbrali dvoenačben model turbulence $k - \epsilon$ v programskemu paketu FIDAP. Obtekanje lopute in tok v spirali smo obravnavali tridimensionalno, zato je bilo treba uporabiti metodo ločevanja za reševanje sistema enačb, kar zelo poveča število iteracij in porabo procesorskega časa. Nekatere numerične rezultate smo primerjali z eksperimentalnimi in v večini primerov je bilo ujemanje dobro.

0 INTRODUCTION

Computational fluid dynamics has recently become important for flow analysis in water turbines. Numerical analysis can often replace expensive and time-consuming measurements. The best solution is an appropriate proportion between numerical and experimental analysis. From experimental analysis the boundary conditions can be obtained, and experimental results are important for comparison with numerical analysis. The main task has been calculation of flow through a Francis turbine. But computer capabilities have not allowed such flow analysis and that is why we have decided to calculate each part separately. First the flow around a butterfly valve has been calculated; further on the flow in the spiral casing of a Francis turbine and finally the flow in a guide vane cascade in order to predict hydraulic torque has been obtained. In all cases the Reynolds number is very high - about 10^6 . For high Reynolds numbers it is necessary to use one of the turbulent models. The geometries are very complicated so we were forced to use 3-D analysis. The consequence of this decision was the use of the segregated method, a high number of iterations and huge consumption of CPU time. The results of numerical analysis have been compared with experimentally obtained results and in most cases the agreement has been acceptable.

1 IZRAČUN CORIOLISOVEGA KOLIČNIKA

Če hočemo zadostiti zahtevam, ki jih postavlja Eulerjeva turbinska enačba, mora biti porazdelitev hitrosti pri vstopu v francisovo turbino, to je pri vstopu v spiralno, enakomerna. Vendar pa v hidroelektrarnah porazdelitev hitrosti pri vstopu v turbino ni enakomerna. Vzrok za to so kolena na koncu cevovodov in turbinski varnostni organi (lečaste in dvokrilne lopute ter krogelni zasuni). Takšna motena porazdelitev hitrosti v vstopnem prerezu lahko vpliva na izkoristek turbine. Vstopni pogoji pri modelnih preizkusih, narejeni so v skladu s priporočili mednarodne elektrotehničke komisije, se zaradi enakomerne vstopne hitrosti ne ujemajo z dejanskim stanjem v hidroelektrarni. Vrednosti izkoristka, preračunane z modela na izvedbo, zato ne pomenijo dejanske vrednosti.

Da bi podkreplili razprave med investitorjem in proizvajalcem turbine, je bila narejena študija postopkov ovrednotenja neenakomernosti izstopnega profila.

V Bernoullijevi enačbi za stacionarni tok nestisljive realne tekočine po cevi se pojavi tako imenovani Coriolisov količnik, ki upošteva neenakomernost hitrosti po prerezu cevi. Fizikalno je Coriolisov količnik, nekateri ga imenujejo količnik kinetične energije, definiran kot razmerje med kinetično energijo, izračunano iz dejanskih hitrosti v danem prerezu in kinetično energijo, izračunano iz povprečne hitrosti v istem prerezu.

Glavna naloga naše analize toka pri vstopu v turbino je določiti vpliv hitrostnega profila na izkoristek stroja, in sicer za primer z vgrajeno lečasto loputo in za prazen cevovod. Problem se je pojavil med prevzemnim preizkusom modelne francisove turbine (sl. 1). Sedanj pretočni trakt ima koleno in loputo velikih dimenzij, ki moti tok pri vstopu v turbino. Ob uporabi numeričnih metod za analizo toka smo za različne primere napovedali vpliv Coriolisovega količnika na izkoristek turbine. Kinetična energija toka v cevi je:

$$W_k = \frac{\rho}{2} \int_A v^3 dA \quad (1)$$

Če se pojavlja v enačbi povprečna hitrost \bar{v} , ima enačba za kinetično energijo naslednjo obliko:

$$\bar{W}_k = \frac{\rho}{2} \bar{v}^3 A \quad (2)$$

Razmerje med obema izrazoma je vedno večje kakor ena:

$$\frac{W_k}{\bar{W}_k} = \frac{\int_A v^3 dA}{\bar{v}^3 A} = \delta \geq 1 \quad (3)$$

1 PREDICTION OF CORIOLIS CORRECTION FACTOR

To meet demands posted by Euler turbine equation at the turbine inlet, i.e. at the inlet of the spiral casing, a uniform velocity distribution is necessary. However at the hydro electric power plants, due to the bending penstocks and security valves i.e. butterfly, spherical or biplane valves, the velocity distribution at the inlet of the water turbine is not uniform. Real plant conditions disturb velocity distribution in the inlet cross-section and have a definite influence on turbine efficiency. The model tests, although carried out according to the recommendations of the International Electrotechnical Commission, i.e. with uniform inlet velocity, do not match real plant conditions; scaled-up values may differ from the real values caused by nonuniformity of the flow.

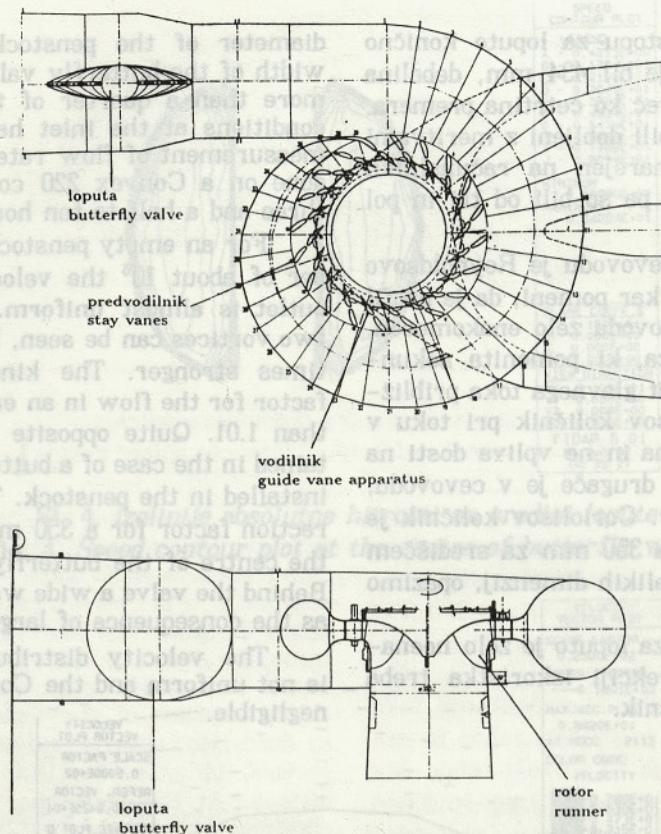
In order to help discussions between plant owners and turbine manufacturers a study was done to examine different procedures of evaluating flow nonuniformity.

In Bernoulli's equation for steady pipe flow of the inviscid real fluid the so-called Coriolis coefficient occurs, which takes into account nonuniformity of the velocity distribution over the pipe cross section. From a physical point of view the Coriolis coefficient, sometimes called the kinetic energy correction factor, can be defined as the ratio of kinetic energy computed through real flow velocities over a given cross-section to the kinetic energy calculated from average velocity in the same cross-section.

The main task of flow analysis at the inlet is to find out the influence of velocity profile on turbine efficiency for two cases: with a butterfly valve and without it. The problem arose during the test of a model of a Francis turbine (Fig. 1). The existing flow tract has a bend and a butterfly valve of large dimensions that can obstruct the flow at the turbine inlet. Using computational fluid dynamics methods the influence of the kinetic energy correction factor on efficiency in different cases was predicted. Kinetic energy for a pipe flow is:

If the average velocity \bar{v} is used, the kinetic energy is:

The ratio between the two different expressions is more than one:



Sl. 1. Model francisove turbine
Fig. 1. Francis turbine model

Količnik kinetične energije δ je Coriolisov količnik. Problem obravnavamo z metodo končnih elementov. Z uporabo majhnega števila izoparametričnih elementov lahko popišemo precej zahtevne geometrijske oblike. Pri izračunu z metodo končnih elementov (MKE) lahko izračunamo povprečno hitrost z upoštevanjem ploščin posameznih elementov:

The coefficient δ is called the kinetic energy correction factor or Coriolis coefficient. Numerically the problem was treated by the finite element method. Complex geometries can be well described by a small number of curvilinear elements. The average velocity is obtained by the finite element (FEM) method where the area of each element is taken into account:

$$\bar{v} = \frac{\sum v_{el} p l_{el}}{p l} \quad (4)$$

V našem primeru smo računali Coriolisov količnik z uporabo naslednje enačbe:

$$\delta = \frac{\sum v_{el}^3 p l_{el}}{\bar{v}^3 p l} \quad (5)$$

Problem cevovoda smo obravnavali prostorsko, ker so tridimenzionalni učinki v toku znatni. Uporabili smo računalniški program FIDAP, ki omogoča računanje turbulentnih tokov z dvoenačbenim modelom $k - \epsilon$.

Pri praznem cevovodu brez lopute je bilo območje obravnavne razdeljeno na 4500 elementov, ko pa smo upoštevali še loputo, se je število elementov povečalo na približno 5000. Pri drugem izračunu smo izstopni del cevovoda malo spremenili,

In our case the Coriolis coefficient was calculated using the following equation:

In the flow through the bending penstock and past a butterfly valve, 3D effects are created. So numerically the problem has been treated three dimensionally with FIDAP package and with the two equation $k - \epsilon$ turbulence model.

The mesh for an empty bending penstock has about 4500 elements and the mesh for the same penstock with installed butterfly valve about 5000. Further on the outlet geometry was changed. In this case the penstock was shorter and the diameter of the penstock behind the valve smaller. The

tako da smo dobili na izstopu za loputo konično obliko. Premer cevovoda je bil 434 mm, debelina lopute pa 109 mm, kar je več ko četrtina premera. Robni pogoji na vstopu so bili dobljeni z meritvami pretoka. Izračun je bil narejen na računalniku Convex 220, računski časi pa so bili od tri in pol do deset ur (CPU).

Pri toku v praznem cevovodu je Reynoldsovo število reda velikosti 10^6 , kar pomeni, da je profil hitrosti pri izstopu iz cevovoda zelo enakomeren. Opazimo lahko dva vrtinca, ki pomenita sekundaren tok, vendar je hitrost glavnega toka približno 25-krat večja. Coriolisov količnik pri toku v prazni cevi je zelo blizu ena in ne vpliva dosti na izkoristek turbine. Precej drugače je v cevovodu, kjer je loputa (sl. 2, 3, 4). Coriolisov količnik je 1,08 na izstopu iz cevovoda 350 mm za središčem lopute. Za loputo, zaradi velikih dimenzij, opazimo tudi močno senco (sl. 5).

Porazdelitev hitrosti za loputo je zelo neenakomerna, zato je pri korekciji izkoristka treba upoštevati Coriolisov količnik.

V Bernoullijevi enačbi za stacionarni tok ne-
staljive realne tekočine po cevi se pojavi tako
imenovan Coriolisov količnik, ki daje neen-
akomernost hitrosti pri izstopu iz cevovoda.
Izkušna je
Coriolisov količnik, ki daje neenakomernost
kinetične energije, definiran kot razmerje
kinetične energije, izvornega cevovoda in teh
hitrosti.



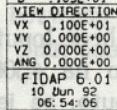
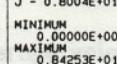
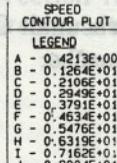
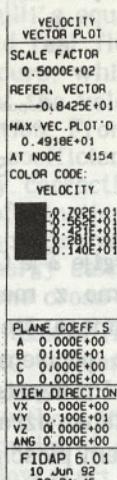
Sl. 2. Vektorji hitrosti v cevovodu z loputo

Fig. 2. Velocity distribution in the penstock with butterfly valve

diameter of the penstock is 434 mm and the width of the butterfly valve is 109 mm which is more than a quarter of the diameter. Boundary conditions at the inlet have been obtained from measurement of flow rate. All calculations were done on a Convex 220 computer and took from three and a half to ten hours of CPU time.

For an empty penstock and a Reynolds number of about 10^6 the velocity distribution at the outlet is almost uniform. Secondary flow with two vortices can be seen, but the main flow is 25 times stronger. The kinetic energy correction factor for the flow in an empty pipe is small, less than 1.01. Quite opposite results have been obtained in the case of a butterfly valve (Fig. 2, 3, 4) installed in the penstock. The kinetic energy correction factor for a 350 mm cross section behind the centre of the butterfly valve was about 1.08. Behind the valve a wide wake (Fig. 5) can be seen as the consequence of large dimensions.

The velocity distribution behind the valve is not uniform and the Coriolis coefficient is not negligible.

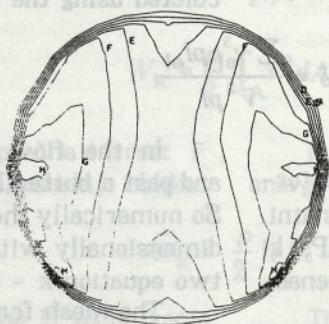


FIDAP 6.01

10 Jun 92

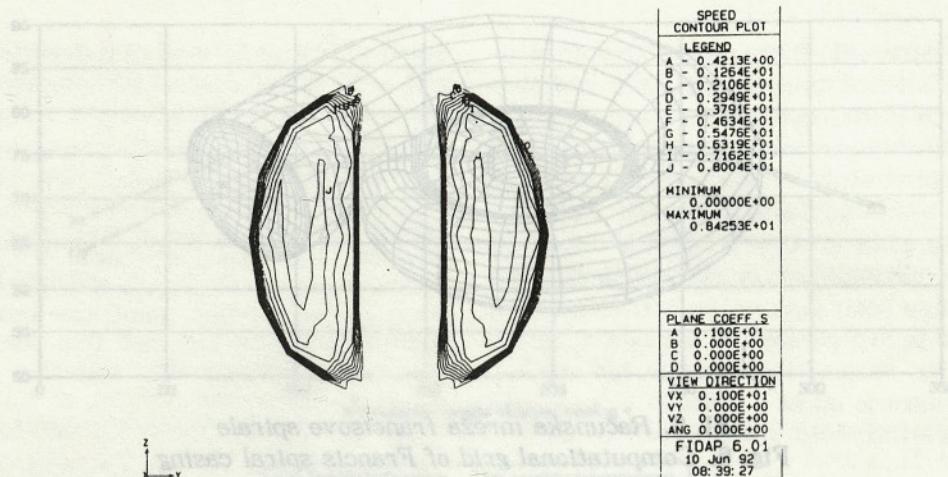
08:54:06

(c)

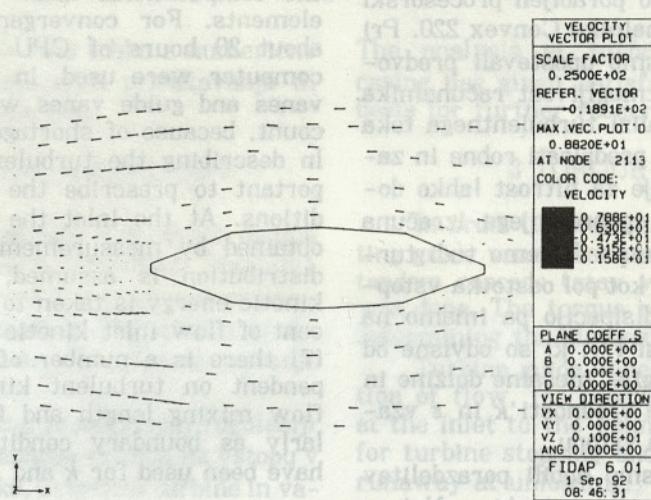


Sl. 3. Izolinije absolutne hitrosti na izstopu iz cevovoda z loputo

Fig. 3. Speed contour plot at the outlet of the penstock with butterfly valve



Sl. 4. Izolinije absolutne hitrosti na sredini lopute
Fig. 4. Speed contour plot at the centre of butterfly valve



Sl. 5. Vektorji hitrosti okoli lopute
Fig. 5. Velocity distribution around butterfly valve

2 TOK V SPIRALI

Spirala z vodilnikom in predvodilnikom poskrbi za optimalni natok vode v rotor. Dobro je, da je čim bolj enakomeren. Tok v spirali se mora izoblikovati tako, da so izgube za različna odprtja vodilnika minimalne. Poleg oblike je, zaradi energetskih karakteristik, pomembna tudi velikost spirale. Treba je poiskati optimalno razmerje med energetskimi karakteristikami in gradbenimi stroški.

Kovinske spirale (sl. 6) so primerne za francisove turbine s srednjim in z velikim padcem. Porazdelitev hitrosti v takšni spirali je odvisna od kota spirale in velikosti prečnih prerezov. S primerno numerično metodo lahko analiziramo tok v različnih delih turbine. Računati moramo celotno spiralno tridimenzionalno, kajti poenostavitev niso mogoče.

2 FLOW IN A SPIRAL CASING

Spiral casing with stay and guide vanes is crucial for optimal angle of attack to the turbine runner. Uniform flow and a high level of symmetry are desirable. The spiral casing has to enable such velocity distribution that energy losses are minimal for different guide vanes opening. Beside the shape of spiral casing the size is important because energy characteristics depend on size. An optimal solution between energy characteristics and construction costs has to be found.

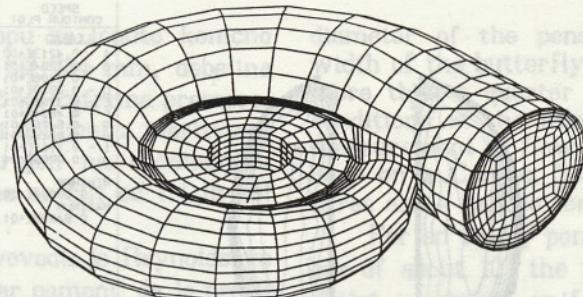
Metal spiral casings (Fig. 6) are suitable for medium and large head Francis turbines. Velocity distribution inside the spiral casing depends on the angle of the spiral casing and the size of the transverse cross-sections. Appropriate numerical methods enable accurate analysis of flow conditions in various parts of turbine machines. Because of the complicated geometry, 3-D calculation was taken into account.

takoj da smo dobili na izstopu na oblike. Premer cevovoda je 434 mm, kar lopute pa 109 mm, kar je več kot dvojnina premera cevovoda. Robni pogoji na vstopu so bili enaki za vse tri modelje preteka. Izračun je bil nareden na Convex 220, računalniški čas pa je bil dosegel ur (CPU).

Pri toku v prazenem cevovodu je stavljen reda velikosti 10^4 , kar predstavlja hitrosti pri izstopu iz cevovoda. Opozimo lehkovo dva vrtinca, ki so v celoti delen tok, vendar je hitrost na izstopu 25-krat večja. Coriolisova sila je na izstopu neznatna, zato je rezultirajoči tok zelo blizu uniformnemu.

Model spirale je bil pri numeričnem izračunu razdeljen na 6 324 elementov. Rešitev je konvergirala po 126 iteracijah; za to porabljen procesorski čas je znašal 20 ur na računalniku Convex 220. Pri izračunu toka v spirali nismo upoštevali predvodilnika in vodilnika, kajti zmogljivost računalnika tega ni omogočala. Pri analizi turbulentnega toka je zelo pomembno pravilno predpisati robne in začetne pogoje. Vstopne pogoje za hitrost lahko dobimo iz meritve ali pa iz poprejšnjega izračuna toka v cevovodu. Pri vstopu predpišemo tudi turbulentno kinetično energijo kot pol odstotka vstopne kinetične energije, za disipacijo pa imamo na voljo več empiričnih formul [7], ki so odvisne od turbulentne kinetične energije, mešalne dolžine in strilžne hitrosti. Za začetne vrednosti k in ε vzamemo vrednosti, podane na vstopu.

Z numerično analizo smo dobili porazdelitev hitrosti znotraj spirale francisove turbine. Najpomembnejši rezultat izračuna toka v spirali je porazdelitev izstopnih vektorjev hitrosti. Primerjava kotov izstopnih vektorjev hitrosti z meritvami je pokazala dobro ujemanje rezultatov. Meritve so bile narejene na modelu z dvouknjično valjasto sondijo. Največja razlika med izmerjenimi in izračunanimi izstopnimi koti (sl. 8) je bila okoli 4° . Analiza velikosti vektorjev na izstopu je pokazala tudi zelo enakomeren tok, saj se vektorji ne razlikujejo po velikosti za več kot 6 odstotkov (sl. 7).



Sl. 6. Računska mreža francisove spirale
Fig. 6. Computational grid of Francis spiral casing

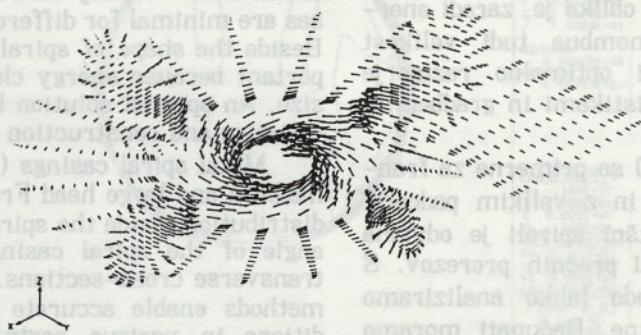
The diameter of the penstock is 434 mm and the boundary valve is 109 mm which is about two times the diameter. Boundary conditions have been obtained from measurements. All calculations were made on a computer and took from 20 to 30 hours of CPU time.

At the outlet the flow in an empty pipe is small, less than 1% of the main flow. Secondary flow with a maximum angle of 25° can be seen, but the main flow is 25 times faster. The kinetic energy correction factor is 0.5.

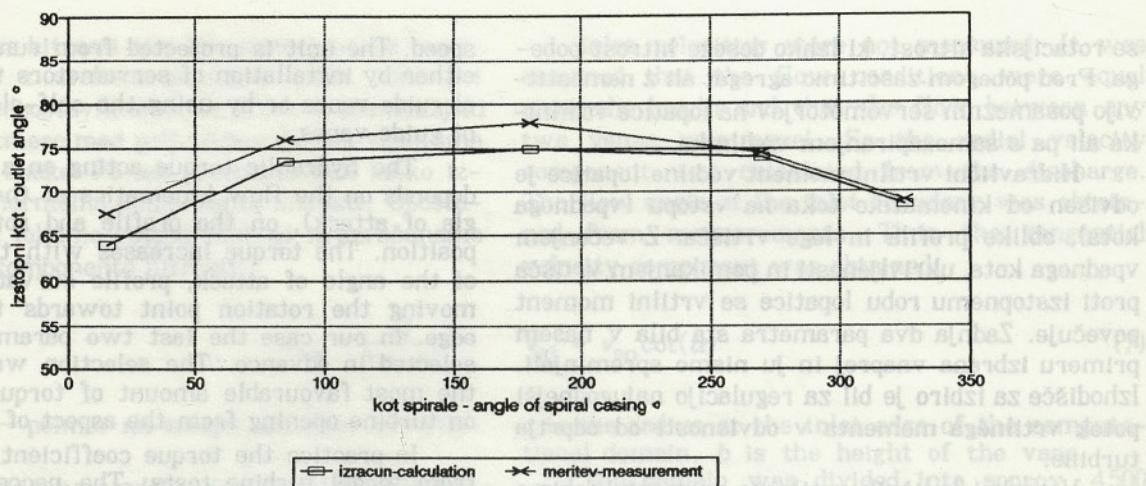
For numerical analysis of turbulent flow in the spiral casing of the Francis turbine model the computational mesh was divided into 6 324 elements. For convergence 126 iterations and about 20 hours of CPU time on a Convex 220 computer were used. In flow analysis the stay vanes and guide vanes were not taken into account, because of shortage of computer memory.

In describing the turbulent flow it is very important to prescribe the correct boundary conditions. At the inlet the velocity distribution is obtained by measurements or constant velocity distribution is assumed, while the turbulence kinetic energy is taken to be one half of one percent of flow inlet kinetic energy. For dissipation [7] there is a number of empirical formulae dependent on turbulent kinetic energy, turbulent flow mixing length and friction velocity. Similarly as boundary conditions, initial conditions have been used for k and ε .

The velocity distribution inside the spiral casing was obtained from numerical flow analysis. Comparison between experimentally obtained distribution of outlet angle and numerical results have shown good agreement. Measurement has been made using a two-hole cylindrical probe. Maximum difference between experimental and numerical results of outlet angle (Fig. 8) reaches about 4° . Absolute velocity vectors at the outlet are almost uniform. The difference between the greatest and the smallest vector is less than 6% (Fig. 7).



Sl. 7. Vektorji hitrosti v spirali
Fig. 7. Distribution of velocity vectors in spiral casing



Sl. 8. Porazdelitev izstopnega kota – Izračun in meritve
Fig. 8. Distribution of outlet angle – numerical and experimental

Primerjava je pokazala, da smo lahko z numerično analizo toka v spirali zadovoljili pričakovanja in je dober temelj za nadaljnje delo.

3 IZRAČUN VRITILNIH MOMENTOV

Numerično smo izračunali tok skozi dvojno kaskado francisove turbine (predvodilnik, vodilnik). Cilj izračuna je bila napoved hidravličnega vritilnega momenta na vodilne lopatice. Poznavanje vritilnega momenta je pomembno za dimenzioniranje servomotorja.

Vodilnik vodne turbine je namenjen regulaciji pretoka, ustvarjanju potrebnega vrtinca na vstopu v gonilnik, prekiniti pretoka, ustaviti turbine in varovati agregata pred pobegom pri razbremenitvi.

Vodilne lopatice se vrtijo s servomotorjem regulatorja okoli svoje osi (sl. 9). V splošnem so čepi vodilnih lopatic razporejeni tako, da hidravlični vritilni moment zapira odprt vodilnik oziroma odpira zaprtega. Tako se v primeru zloma enega regulacijskega organa izognemo vodnemu udaru. Zvezca

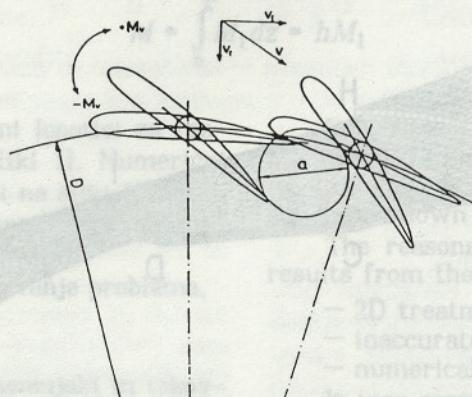
The analysis of turbulent flow in the spiral casing has given satisfactory results and a good basis for further research.

3 TORQUE PREDICTION

In order to predict the hydraulic torque acting on guide vanes, a numerical analysis of flow in tandem cascade (stay vane/wicket gate cascade) was done. The torque prediction is important for determining the dimensions of the servomotor.

Turbine guide vanes are designed for regulation of flow, for producing the necessary vortex at the inlet to the runner, for flow interruption, for turbine stopping and for unit protection from runaway at unloading.

By means of a governor servomotor the guide vanes rotate around their respective axes (Fig. 9). In general the guide vane stems are arranged in such a way that the hydraulic torque acting on the vanes tends to close the open guide vanes or to open the closed ones. Thus in case of one shear-pin failure a water hammer is avoided. The resulting speed rise can reach runaway



Sl. 9. Vodilnik

-M – deluje na zapiranje, +M – deluje na odpiranje

Fig. 9. Guide vane apparatus

-M – closing tendency, +M – opening tendency

se rotacijska hitrost, ki lahko doseže hitrost pobjega. Pred pobegom zaščitimo agregat ali z namestitvijo posameznih servomotorjev na lopatice vodilnika ali pa s samozapiranjem vodilnika.

Hidravlični vrtilni moment vodilne lopatice je odvisen od kinematike toka na vstopu (vpadnega kota), oblike profila in lege vrtišča. Z večanjem vpadnega kota, ukrivljenosti in pomikanjem vrtišča proti izstopnemu robu lopatice se vrtilni moment povečuje. Zadnja dva parametra sta bila v našem primeru izbrana vnaprej in ju nismo spremenjali. Izhodišče za izbiro je bil za regulacijo najugodnejši potek vrtilnega momenta v odvisnosti od odprtja turbine.

V praksi količnik vrtilnega momenta dobimo iz preizkusov na modelni turbini. Potreben pogoj za zanesljivost tako dobljenih količnikov sta geometrična in kinematična podobnost med modelom in prototipom. Število vodilnih in predvodilnih lopatic na modelu je enako. Vrtilni moment smo merili z uporavnimi lističi, nameščenimi na štirih vodilnih lopaticah. Lega vodilnih lopatic, na katerih smo merili moment, je označena na sliki 12.

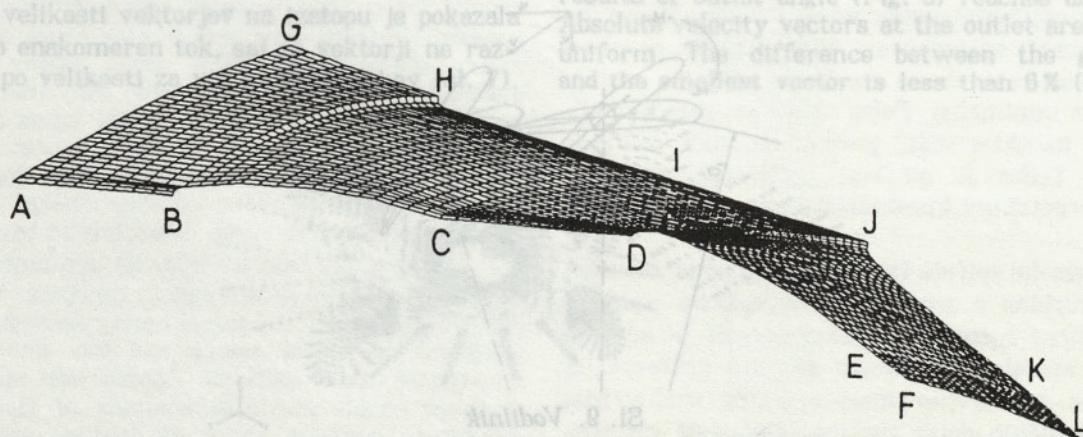
Numerično smo problem reševali z metodo končnih elementov s programskim paketom FIDAP.

Tok skozi kaskado predvodilnika in vodilnika je tridimensijski in turbulenten. Zaradi omejenega računalniškega pomnilnika in časa smo problem obravnavali dvodimensijsko. Računali smo s turbulentnim dvoenačbnim modelom $k - \varepsilon$.

Območje računanja smo skrčili na območje med dvema predvodilnima in dvema vodilnima lopaticama. Na odsekih AB, GH; CD, IJ; in EF, KL smo predpisali antiperiodične robne pogoje (sl. 10):

$$u_{n,AB} = -u_{n,GH}$$

kjer sta: $u_{n,AB}$ — normalna komponenta, $u_{t,AB}$ — obodna komponenta hitrosti na AB.



Sl. 10. Mreža za odprtje vodilnika $A_0 = 1,2$

Fig. 10. Computational grid for guide vane opening $A_0 = 1.2$

speed. The unit is protected from runaway speed either by installation of servomotors to individual guide vanes or by using the self-closing effect of guide vanes.

The hydraulic torque acting on a guide vane depends on the flow kinematics at the inlet (angle of attack), on the profile and rotation point position. The torque increases with the increase of the angle of attack, profile curvature and by moving the rotation point towards the trailing edge. In our case the last two parameters were selected in advance. The selection was based on the most favourable amount of torque depending on turbine opening from the aspect of regulation.

In practice the torque coefficient is obtained from model turbine tests. The necessary condition for the reliability of so obtained coefficients is the geometrical and kinematical similarity of model and prototype. The number of stay and guide vanes is equal. Hydraulic torque was measured by means of strain gauges installed on four guide vane stems [1]. The position of the guide vanes chosen for measurements can be seen in Figure 12.

Numerically the problem was treated by the finite element method using the FIDAP code.

The flow through the stay ring and distributor cascade is three-dimensional and turbulent. Due to limited computer memory and time the problem was treated in 2D. A two equation $k - \varepsilon$ turbulent model was used.

The computational domain was reduced to the region between two stay vanes and two guide vanes. In sections AB, GH; CD, IJ; and EF, KL antiperiodical boundary conditions were prescribed (Fig. 10):

$$u_{t,AB} = -u_{t,GH} \quad (6)$$

where: $u_{n,AB}$ is the normal, while $u_{t,AB}$ is the tangential velocity component on AB.

Vstopne hitrosti niso bile izmerjene. Iz meritev spirale z zrakom poznamo smer toka pri vstopu. Predpostavili smo, da je tok dvodimensijski in da so razmere med poljubnima dvema vodilnima lopaticama enake. Po teh predpostavkah lahko izračunamo normalno komponento hitrosti. Upoštevamo smer toka na vstopu (kot α) in izračunamo še obodno komponento hitrosti:

$$v_{on} = \frac{Q}{2\pi rh},$$

kjer sta: r – polmer na vstopu območja, h – višina območja.

Območje smo razdelili na okoli 4500 štirivzelnih elementov. Računski čas je bil približno 4 ure na računalniku Convex 220.

Hidravlični vrtilni moment izračunamo iz razdelitve tlaka po lopatci. Naj bo sesalna stran vodilne lopatice podana s funkcijo $f^+(x)$, tlačna stran pa s funkcijo $f^-(x)$. Os x naj bo v smeri tretive lopatice, $p^+(x)$ in $p^-(x)$ naj bosta funkciji tlaka za sesalno in tlačno stran, x_v , y_v pa koordinate vrtišča profila. S temi označbami se vrtilni moment profila izraža takole [4]:

$$M_1 = \int_0^1 (x - x_v)(p^-(x) - p^+(x))dx + \\ + \int_0^1 [(f^-(x) - y_v)f'^-(x)p^-(x) - (f^+(x) - y_v)f'^+(x)p^+(x)]dx \quad (8).$$

Da dobimo vrtilni moment na celotno vodilno lopatico, moramo vrtilni moment za profil integrirati po višini lopatice. Vsi profili lopatice so enaki, predpostavili smo, da se tudi tokovne razmere po višini ne spreminjajo. Zato za vrtilni moment na vodilno lopatico dobimo kar:

$$M = \int_0^h M_1 dz = hM_1 \quad (9).$$

Porazdelitev tlaka po vodilni lopatici za različna odprtja je prikazana na sliki 11. Numerični in merilni rezultati so prikazani na sliki 12.

Vzroki za odstopke izračunanih vrtilnih momentov od izmerjenih:

- dvodimensijsko obravnavanje problema,
- nenatančni robni pogoji,
- numerične napake.

Vzeli smo, da je tok dvodimensijski in tokovne razmere med poljubnima vodilnima lopaticama enake (antiperiodični robni pogoji, izračun vstopnih hitrosti). Razlike med izmerjenimi vrtilnimi momenti na posameznih vodilnih lopaticah (sl. 12)

Inlet velocities were not measured. It was assumed that the flow conditions were equal along the height and that the flow between any two vanes was equal. So the radial velocity component was calculated from the discharge. The flow angle at the inlet boundary was obtained from measurements. Thus the tangential velocity component was obtained:

$$v_{ot} = v_{on} \cot(\alpha) \quad (7)$$

r is the radius at the inlet edge of the computational domain, h is the height of the vane.

The domain was divided into approx. 4500 four-nodal elements. The CPU time was about 4 hours on average by CONVEX 220.

Hydraulic torque was calculated from the pressure distribution along the guide vane. The suction side of guide vane was described with the function $f^+(x)$, the pressure side with the function $f^-(x)$. The axis x was in the direction of guide vane chord, $p^+(x)$ and $p^-(x)$ were the pressure functions of the suction and pressure side respectively and x_v , y_v were the coordinates of the profile rotation point. Torque for profile was expressed as follows [4]:

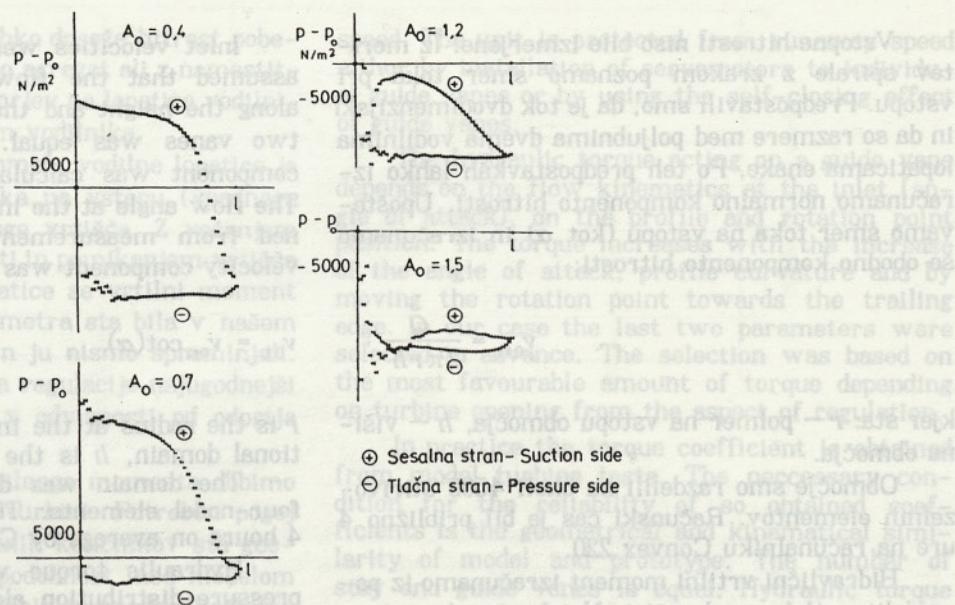
In order to obtain the torque acting on the guide vane the torque for one profile should be integrated along the vane height. All the profiles being equal and assuming that the flow conditions along the height were the same, the following was obtained:

Pressure distribution on the guide vane surface for different guide vane openings is presented in Figure 11. Numerical and experimental results are shown in Figure 12.

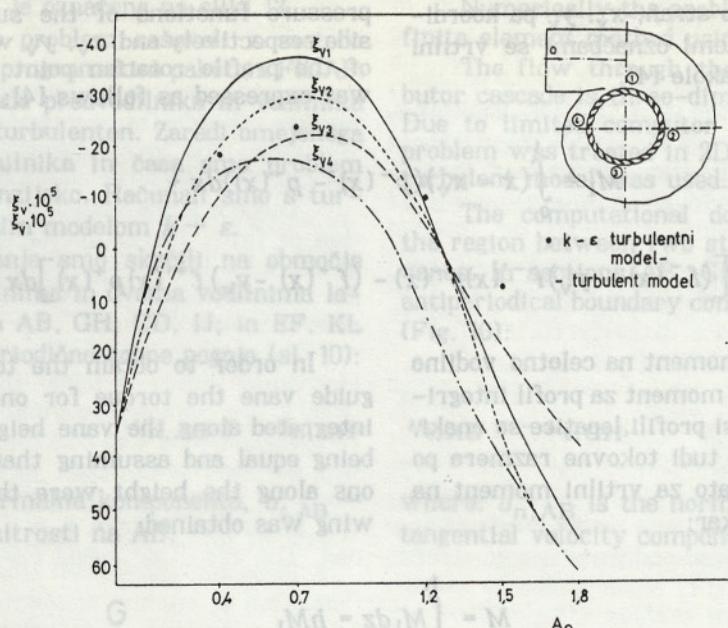
The reasons for the deviation of calculated results from the experimental ones are:

- 2D treatment of the problem,
- inaccurate boundary conditions,
- numerical errors.

It was assumed that the flow was two-dimensional and that the conditions between any two guide vanes were equal (antiperiodical boundary conditions, calculation of the velocity). The differences between torque coefficients obtained



Sl. 11. Porazdelitev tlaka za različna odprtja vodilnih lopatic
Fig. 11. Pressure distribution for different guide vane openings



Sl. 12. Količnik vrtilnega momenta – Izračunane in izmerjene vrednosti
Fig. 12. Torque coefficient – numerical and experimental values

kažejo, da predpostavka o antiperiodičnih pogojih ne velja povsem. Predpisani robni pogoji se torej ne ujemajo povsem z realnim tokom.

K nenatančnosti numeričnih rezultatov prispevajo tudi numerične napake. Pri štirivozelnih elementih tlaka ne dobimo v vozilih, ampak v središču elementov. Zato ne dobimo porazdelitve tlaka po površini lopatice, ampak za pol višine elementa stran. Štirivozelnii elementi tudi ne zadoščajo inf. sup. pogoju [2]. K numeričnim napakam prispeva tudi neortogonalna mreža.

experimentally on four guide vanes are high (Fig. 12). So the prescribed boundary conditions do not satisfactorily match the real flow.

Numerical errors, too, contribute to the deviation of numerical results from the measured ones. With the 4-nodal elements the pressure is not obtained at the nodes but in the centers of elements. Pressure distribution is not obtained on the guide vane surface but half a thickness of the element away. In addition the linear elements do not satisfy the Inf. Sup. condition [2]. A nonorthogonal grid contributes to numerical errors as well.

UDK Numerični rezultati se zadovoljivo ujemajo z izmerjenimi. Odstopki so večji za zelo odprt vodilnik.

Na podlagi numerične napovedi momenta, preverjene z meritvami, lahko optimizamo servomotor. Izberemo kompromisno rešitev med zaželenim samozapiranjem vodilnika in minimalnimi silami servomotorja.

Numerical results correspond satisfactorily with the measured ones. Deviations are greater at larger guide vane openings.

On the basis of such numerical prediction verified by experiment, the servomotor can be optimized and a compromise solution between the required tendency of guide vane self-closing effect and servomotor minimum forces can be selected.

Označbe

a	potek	— odprtje vodilnika,
A_0	v top	— odprtje vodilnika, $A_0 = az/D_0$,
D_0	In turn	— premer vodilnika,
D	the box	— premer gonilnika,
h	display	— višina lopatice,
H	therm	— padec,
I	This is	— tetiva,
M	0.8 mlečev	— hidravlični moment,
M_1	PAZU	— hidravlični vrtilni moment profila,
N	— vrtilna frekvenca,	
p	Nevidljivo	— tlak,
p_0	čašir, povr	— tlak pri vstopu v vodilne lopatice,
p_{el}	plavi	— ploščina elementa,
Q	vezemerj	— pretok,
r	ravnano	— polmer na vstopu v območje,
Re	da fazi	— Reynoldsovo število, $Re = lu/\nu$,
u	skozi zapis	— vstopna hitrost,
v_r, v_t	in enačba	— radialna in obodna komponenta hitrosti,
v_{on}, v_{ot}	Vodilne	— radialna in obodna komponenta hitrosti na vstopu v območje,
W_k	Vodilne	so podane z Navier-Lamejovimi enačbami:
x, y	Grad	— kinetična energija,
x_v, y_v	div	— kartezične koordinate,
z	per SRI	— koordinate vrtišča profila,
α	R	— število vodilnih lopat,
δ	Body forces	— kot toka v kaskadni ravnini, $\alpha = \text{arc cot}_n(v_t/v)$,
ξ	shear	— Coriolisov količnik,
ρ	modul	— količnik vrtilnega momenta, $\xi = M/(\rho \frac{\pi^3}{8} \psi N^2 D^5)$,
ψ	modul	— gostota,
	modul	— tlačno število, $\psi = H/(\frac{u^2}{2g})$

Symbols

a	— guide vane opening,
A_0	— guide vane opening, $A_0 = az/D_0$,
D_0	— diameter of guide vane apparatus,
D	— runner diameter,
h	— guide vane height,
H	— head,
I	— chord,
M	— hydraulic torque,
M_1	— hydraulic torque for profile,
N	— speed,
p	— pressure,
p_0	— pressure at the guide vane entry,
p_{el}	— area of each element,
Q	— discharge,
r	— radius at the domain entry,
Re	— Reynolds number, $Re = lu/\nu$,
u	— peripheral velocity,
v_r, v_t	— radial and tangential velocity components,
v_{on}, v_{ot}	— radial and tangential velocity components at the inlet of computational domain.
W_k	— kinetic energy,
x, y	— Cartesian coordinates,
x_v, y_v	— coordinates of profile rotating point,
z	— number of guide vanes,
α	— flow angle in cascade plane, $\alpha = \text{arc cot}_n(v_t/v)$,
δ	— kinetic energy correction factor,
ξ	— torque coefficient $\xi = M/(\rho \frac{\pi^3}{8} \psi N^2 D^5)$,
ρ	— density,
ψ	— pressure number, $\psi = H/(\frac{u^2}{2g})$

4 LITERATURA

4 REFERENCES

- [1] Gregori, J.-Kercan, V.-Pišljar, M.-Hočevare, D.: PS CATALAN — Dynamic Characteristics of Model Francis Turbine. Turboinstitute Report No. 1859. Ljubljana, 1987.
- [2] Girault, V.-Raviart, P.A.: Finite Element Methods for Navier-Stokes Equations. Theory and Algorithms. Springer-Verlag, Berlin, 1986.
- [3] Jošt, D.-Velenšek, B.: Numerical Prediction of the Hydraulic Torque on Kaplan Turbine Guide Vanes. Forum on Advances in Finite Element Analysis in Fluid Dynamics II. Dhaubhadel, M.N. et al. (eds). Published by ASME, New York, FED-Vol. 137, pp. 81-85.
- [4] Velenšek, B.-Mejak, G.: Theoretical Calculation of Axial Force, Torque on Turbine and Torque on Blade Pivot of Kaplan Runner — HPS Haditha. Proceedings of the Conference on Fluid Flow Machinery and Flow Measurements, Velenšek, B. and Bajd, M. (eds). Ljubljana, 1984, pp. 118-128.
- [5] FIDAP USERS MANUAL, Revision 6.0
- [6] Ruprecht, A.: Finite Element Method, Part I. Notes of CFD '90 — Intensive Course on Computational Fluid Dynamics and Heat Transfer. Ljubljana 1990.
- [7] Zhang, C.-Sousa, A.C.M.: Numerical Simulation of Turbulent Shear Flow in Isothermal Heat Exchanger Model. ASME Journal of Fluids Engineering, Vol. 112/1, March 1990, pp. 48-55.

Naslov avtorjev: mag. Andrej Lipej, dipl. inž.,
mag. Dragica Jošt, dipl. inž.,
Mateja Jamnik, dipl. inž.,
prof. dr. Boris Velenšek, dipl. inž.
Turboinstitut
Rovšnikova 7
61210 Ljubljana

Prejeto: 1.12.1993
Received:

Authors' Address: Mag. Andrej Lipej, Dipl. Ing.,
Mag. Dragica Jošt, Dipl. Ing.,
Mateja Jamnik, Dipl. Ing.,
Prof. Dr. Boris Velenšek, Dipl. Ing.
Turboinstitute
Rovšnikova 7
Ljubljana, Slovenia

Sprejeto: 5.5.1994
Accepted:

kažejo, da predpostavka o vseh elementih pogojih ne velja povsem. Trenutno pa pogoj se tore ne ujemajo povsem.

K nemotanemu numeričnemu rezultatov pripravijo tudi numerične rezulatajte sestavljenih elementov (finih in nadobrih) vrednost, ki je v sredini elementov. Zato ne dobimo porezenitve tlaka po površini lopatice, ampak za pol višine elementa stran. Stirnovzeta elementi tudi ne zadostujejo inf. sup. pogoju [2]. K numeričnim nepraken pripravijo tudi neortogonalna mreža.

Numerični rezultati kažejo, da do končne to do vrednosti numeričnih rezulatajte sestavljenih elementov (finih in nadobrih) vrednost, ki je v sredini elementov. Zato ne dobimo porezenitve tlaka po površini lopatice, ampak za pol višine elementa stran. Stirnovzeta elementi tudi ne zadostujejo inf. sup. pogoju [2]. K numeričnim nepraken pripravijo tudi neortogonalna mreža.

experimentally on four guide vanes are high (Fig. 12). So the prescribed boundary conditions do not satisfy the condition.