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Vpliv sekundarnega toka na obratovalno karakteristiko radialnega rotorja normalne širine

Influence of Secondary Flow on the Operating Characteristics of a Radial Impeller with Normal Width

Andrej Predin

V članku so podani rezultati eksperimentalne analize vpliva dodanega sekundarnega toka na vstopu v radialni rotor normalne širine. Sekundarni tok je dodajan skozi vodilnik, ki se končuje s kolobarjasto rego ob pokrovni steni rotorja. V kolobarjastem vodilniku so vzdolžno nameščene vodilne lopatice, ki zagotavljajo osno simetrični nevrtinčni sekundarni tok na vstopu v rotor. Izvedeno je atmosfersko (neprisilno) polnjenje s spiralnim vodilnikom, nameščenim pred kolobarjastim vodilnikom. Po eksperimentalnih rezultatih lahko sklepamo, da dodani sekundarni tok ugodno vpliva na obratovalne karakteristike radialnega rotorja normalne širine ($13 < n_q < 35$), ker poveča stabilni del obratovanja proti manjšim obratovalnim pretokom (poveča se možnost širše regulacije), pri tem pa se še nekoliko izboljša skupni izkoristek, predvsem v področju manjših, podoptimalnih pretokov.

(Ključne besede: rotorji radialni, tok sekundarni, karakteristike obratovalne, analize eksperimentalne)

In the paper an analysis of the experimental results of a secondary flow addition at the radial impeller entrance (normal width impeller) is given. The secondary flow is added through the guide system, which ends with the circle-ring at the impeller front shroud. The guide vanes are set in the circle-ring guide system in the direction of the intake pipe axis. The guide-vanes assure the axi-symmetrical non-whirl secondary flow at the impeller intake. The atmospheric (non-forced) secondary flow loading is performed with the volute guide system placed at the entrance of the circle-ring guide system. Based on the experimental results, it can be concluded that the suitable secondary flow favourably influences the operating characteristic of the impeller ($13 < n_q < 35$) width. The stable operating area increases in the direction of the small operating capacities (thus giving the possibility of wider pump control). At the same time the overall efficiency is increased, especially in the area of small,suboptimum capacities.

(Keywords: radial impellers, secondary flow, operating characteristics, experimental results)

0 UVOD

Radialni rotorji normalnih širin s specifičnimi števili vrtljajev med 20 in 35 min^{-1} imajo nestabilno obratovalno karakteristiko. To dejstvo omejuje uporabo takšnih rotorjev glede dosegajočih pretokov, predvsem z vidika manjše možnosti stabilnega obratovanja v širšem področju pretokov, ki jih dosega tak rotor. Nestabilni del leži v področju majhnih obratovalnih pretokov, kjer je tudi skupni izkoristek razmeroma slab. Znano je, da se pri takšnih rotorjih izkoristek znatno zmanjšuje od optimalnega tako v področje večjih (nadoptimalnih) kakor tudi v področje manjših (podoptimalnih) pretokov. Zato so z obratovalnega vidika zanimivi vsi ukrepi, ki bi ob minimalni spremembi geometrije (majhni stroški), razširili področje obratovalnih pretokov, pri katerih bi izkoristek bil še zadovoljiv.

0 INTRODUCTION

The normal width radial impellers, with specific speed numbers between 20 and 35 rpm, have (in most cases) a non-stable operating characteristic. This fact limits the employability of such impellers regarding the achieved capacities, because of their very narrow stable operating area. The impeller operates in a non-stable manner in the area of small operating capacities, in which the overall impeller efficiency is also low. It is known that (with this type of impellers) the overall efficiency significantly drops when the impeller operates out of the optimum operating point (best efficiency point) regardless of whether the operating capacities are increased up to the over-optimum or decreased down to the suboptimum operating capacities. Therefore, from the impeller operating aspect, all the arrangements are interesting that may increase the area of the operating capacities when the overall operating efficiency is satisfactory.

1 RAZVOJ DODAJANJA SEKUNDARNEGA TOKA

Prvi je vpliv dodanega sekundarnega toka proučeval Eck [1], vendar predvsem na radialnih rotorjih povečane oz. velike širine, pri čemer je dodajal tudi do tri sekundarne tokove, tudi ob pestni steni rotorja in v sredini vstopne širine rotorja. Vodilo mu je bilo predvsem doseganje mejnih obratovalnih zmogljivosti, tako po pretoku (Q_{maks}) kakor tudi po dosegajočih energijskih oziroma tlačnih razlikah (Δp_{maks}), manj pa s strani razširitve ugodnega obratovalnega območja. Znana sta dva načina dodajanja sekundarnega toka, in sicer:

a) prisilna polnitev z dovajanjem sekundarnega toka na vstop rotorja oz. rotorskih lopatic ob pokrovni steni iz tlačnega cevovoda ali z drugim virom (dodatni ventilator – manj ugodno zaradi večjih izgub dodatnega pogona) in
b) prosta polnitev - dodajanje sekundarnega toka s prostim vsesavanjem pri tlaku okolice – atmosferskem tlaku (konkretno uporabljeno pri pričujočem testiranju).

Glede skupnega izkoristka celotnega sistema ima druga varianta b) prednost pred prvo a), v primerjavi največjih – mejnih zmogljivosti pa je v prednosti druga varianta. Tako se je treba tudi v tem primeru ustrezno odločiti, kaj želimo dosegati s konkretnim sistemom.

1 DEVELOPMENT OF THE SECONDARY FLOW ADDITION METHOD

The first person who studied the influence of the secondary flow addition was Eck [1], but his studies were primarily based on impellers of large width. He then added up to three secondary flows: at the impeller hub, at the middle of the impeller width, and at the impeller front shroud. His purpose was to find a way of increasing the maximal impeller performance in order to increase maximal operating capacities (Q_{max}) and/or maximal energy or pressure differences (Δp_{max}), and less directed towards increasing the area of the suitable operating capacities. Two different principles for secondary flow addition are already known:

a) forced loading, supplying the secondary flow at the impeller entrance at the front shroud from the discharge pipe or by other sources (an additional fan – is less suitable, because additional energy losses occur) and
b) atmospheric or non-forced loading with the secondary flow supplied by free suction from the surrounding ambient at atmospheric pressure (as used in our investigation).

Regarding the overall efficiency of the whole testing system, the second option b) is more advantageous than the first one a). However, when maximal capacities or maximal pressure differences are the goal, variant a) is preferable. Therefore, the decision, as to which option is optimal must made from case to case.

2 SPLOŠNE KARAKTERISTIKE TESTNEGA SISTEMA

Sistem obsega štiri osnovne sklope; pogonski, modelni, sklop oziroma sistem za dovajanje sekundarnega toka in dotočni sklop. Poenostavljeni model z radialnim rotorjem (n_q = 24 min^{-1}) je podrobneje predstavljen v prispevku [2].

2.1 Karakteristike sklopa sekundarnega toka

Sistem za dovajanje sekundarnega toka na vstopu v rotor ob pokrovni steni je sestavljen iz kolobarjastega vodilnika z dvajsetimi osno simetrično postavljenimi vodilnimi lopaticami v njem, ki se končuje v obliki kolobarjaste reže širine s = 0,003 m in iz spiralnega vodilnika, ki usmerja sekundarni tok v kolobarjasti vodilnik iz radialne v aksialno smer (sl. 1). Vodilne lopatice v kolobarjastem vodilniku so namenjene za osno simetrično usmerjanje toka na vstopu v rotor (preprečevanje vrtinčnosti toka), ker se dodani sekundarni tok preusmerja iz spiralnega vodilnika v kolobarjasti, pri čemer se spremeni smer toka iz radialne v aksialno smer.

2.2 Karakteristike radialnega rotorja

Rotor je izveden v radialni obliki z enajstimi (z_r = 11) nazaj ukrivljenimi rotorskimi lopaticami

2 GENERAL CHARACTERISTICS OF THE TESTING SYSTEM

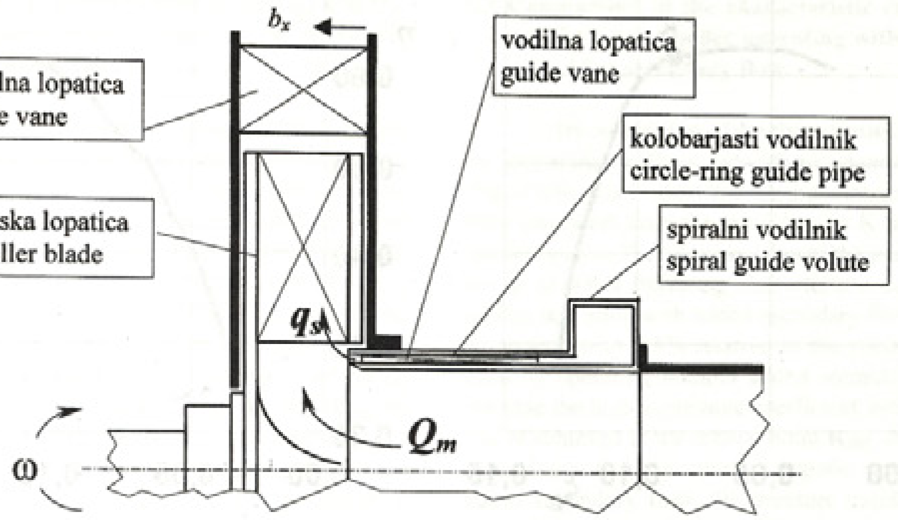
The testing system includes four basic parts: driving part, impeller fan-model, the system for the secondary flow addition, and the main intake pipe system (part). The simplified fan model has the radial impeller (n_q = 24 rpm), and has already been presented in [2].

2.1 Characteristics of the secondary-flow system

The system for the secondary flow addition to the impeller intake diameter at the impeller front shroud is composed of the circle-ring guide pipe with twenty guide-vanes placed in axial-symmetric direction. The circle gap width of the circle-ring guide pipe is s = 0.003 m. The spiral guide volute guides the secondary flow from the radial to the axial direction at the entrance of the circle-ring guide pipe (Fig. 1). The guide vanes in the guide-ring guide the flow in the axial direction (attempting to prevent the flow vortex), since the entrance flow from the spiral guide volute has a radial direction and has to be directed into the axial direction.

2.2 The radial impeller characteristic

The fan-model has a radial impeller with eleven back-curved (z_r = 11) cylindrically shaped (in



Sl. 1. *Sklop za dovajanje sekundarnega toka na vstopu v rotor ob pokrovni steni*
Fig. 1. *Secondary flow addition system at the impeller entrance at the front shroud*

valjaste oblike (prostorsko enkrat ukrivljene lopatice) konstantne širine ($b_1 = b_2 = 0{,}05$ m). Vstopni in izstopni kot lopatic sta $\beta_1 = 23°$ oziroma $\beta_2 = 25°$. Razmerje premerov je $D_2/D_1 = 0{,}6/0{,}36 = 1{,}66$. Preračunska točka rotorja je pri $n = 1800$ min^{-1} in pretoku $Q_{prer} = 0{,}65$ m^3/s. Obratovalna karakteristika rotorja je podana v brezdimenzijski obliki in je nestabilne oblike (sl. 2). Brezdimenzijski koeficient - tlačno število je določeno z:

space once curved) blades with a constant width ($b_1 = b_2 = 0.05$ m). The intake and exit blade angles, respectively, are $\beta_1 = 23°$ and $\beta_2 = 25°$. The impeller calculation point (operating point) is at $n = 1800$ rpm and at $Q_{cal} = 0.65$ m^3/s. The operating characteristic (head-capacity curve), given in non-dimensional form, is presented in Figure 2. As shown, it is non-stable. The non-dimensional coefficient - pressure number is given by:

$$\psi = \frac{2\Delta p}{\rho u_2^2} \quad (1),$$

kjer so: Δp - izmerjena tlačna razlika, ki jo rotor dosega, ρ - gostota toka in u_2 - obodna hitrost na rotorjevem izstopnem premeru D_2. Brezdimenzijski koeficient - specifični pretok je določen z:

where Δp is the measured pressure difference achieved by the impeller, ρ is the density, and u_2 is the circumferential velocity at the impeller exit diameter D_2. The non-dimensional coefficient - specific capacity is given by:

$$\xi_Q = \frac{Q}{nD_2^2} \quad (2),$$

kjer je Q - izmerjeni obratovalni pretok, n - vrtilna hitrost (število vrtljajev na sekundo) in D_2 - izstopni premer rotorja.

Izkoristki so določani po:

where Q is the measured operating capacity, n the impeller speed (rps) and D_2 is the impeller exit diameter.

The impeller efficiency is determined by:

$$\eta = \frac{Q\Delta p}{2\pi n M} \quad (3),$$

kjer je M moment na gredi rotorja. Največji izmerjeni izkoristek znaša $\eta = 0{,}62$ pri pretoku $\xi_Q = 0{,}118$. Kritična točka, ki loči stabilni del od nestabilnega leži pri pretoku $\xi_Q = 0{,}089$, kjer znaša doseženo tlačno število $\psi = 0{,}860$ (točka **K**, sl. 2 a).

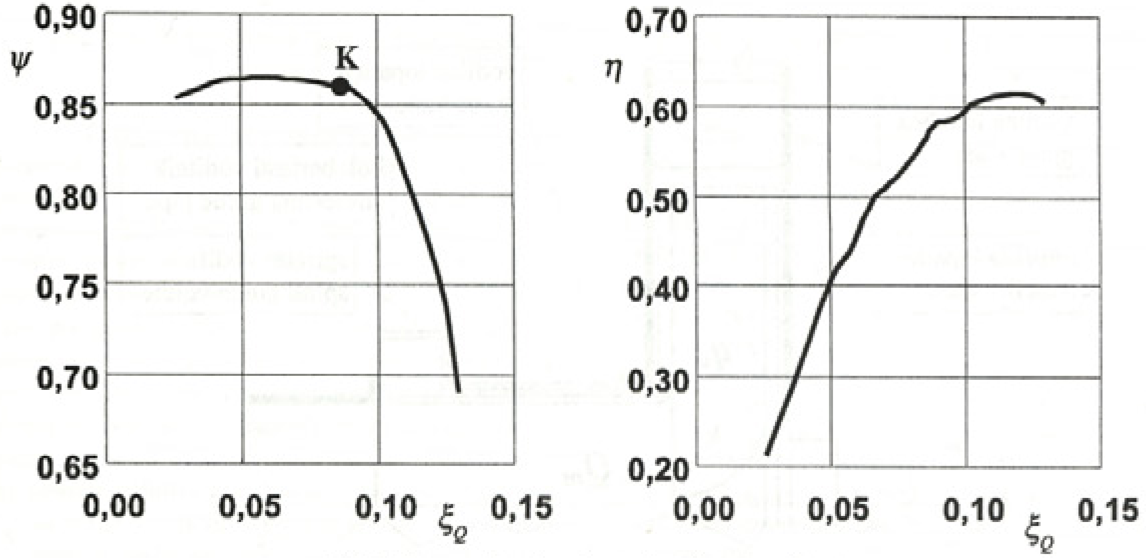
where M is the impeller shaft torque. The maximal measured efficiency is $\eta = 0.62$ at the specific capacity $\xi_Q = 0.118$. The critical point that separates the stable and non-stable operating area (capacity-head curve) is at $\xi_Q = 0.089$, where the pressure coefficient is $\psi = 0.860$ (point **K**, Fig. 2 a).

2.2 Karakteristike radialnega rotorja z dodanim sekundarnim tokom

Iz posnete karakteristike ξ_Q-ψ - dušilne krivulje (sl. 3 levo) je razvidna sprememba oblike . Kritična točka **K** (sl. 3 levo) se je pomaknila v področje manjših pretokov, konkretno na $\xi_{Q,K,SST} = 0{,}66$. S tem se je območje stabilnega dela obratovalne karakteristike ($\Delta\xi_Q = \Delta\xi_{Q,K} - \Delta\xi_{Q,maks}$) povečalo iz $\Delta\xi_{Q,BST} = 0{,}041$, pri

2.3 The impeller characteristic with added secondary flow

From the measured ξ_Q-ψ characteristic capacity-head curve (Fig. 3 left), the shape change is evident. The critical point **K** (Fig. 3 left) is toward smaller capacities (in our case to $\xi_{Q,K,SF} = 0.660$). With this critical point movement the stable characteristic is enlarged ($\Delta\xi_Q = \Delta\xi_{Q,K} - \Delta\xi_{Q,max}$) from $\Delta\xi_{Q,WSF} = 0.041$ (no added



Sl. 2. *Obratovalna karakteristika* (ξ_Q-ψ) *in* (ξ_Q-η)
Fig. 2. *Capacity-Head* (ξ_Q-ψ) *and Capacity-Efficiency curve* (ξ_Q-η)

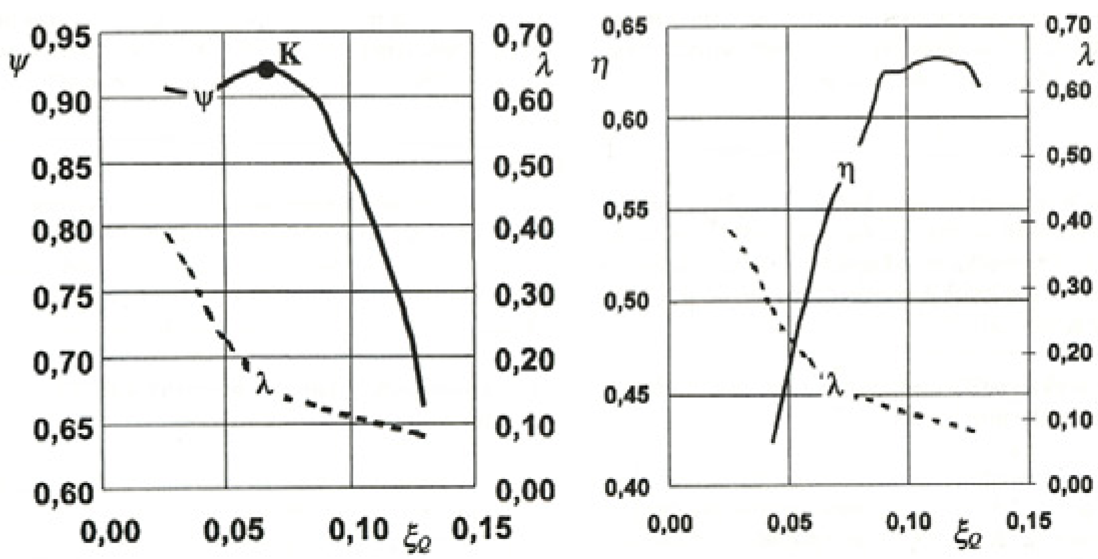
obratovanju brez dodanega sekundarnega toka, na $\Delta\xi_{Q,SST} = 0{,}064$, pri obratovanju z dodanim sekundarnim tokom, kar konkretno pomeni 56,1% relativno povečanje intervala. Z dodajanjem sekundarnega toka se je povečalo tudi maksimalno tlačno število (v kritični točki) iz $\psi_{K,BST} = 0{,}86$ in $\psi_{K,SST} = 0{,}923$, oz. za 7,3 %. Z uvedbo razmernika pretokov:

secondary flow) to $\Delta\xi_{Q,SF} = 0.064$ when the fan operates with added secondary flow. In our case the stable part of the characteristic was increased by 56 % relatively. By adding the secondary flow the maximal pressure number (at the critical point) was increased from $\psi_{K,WSF} = 0.86$ to $\psi_{K,SF} = 0.923$. This is a 7.3 % increase. By introducing the capacity ratio in the form of:

$$\lambda = \frac{q_s}{Q_m} \qquad (5),$$

kjer sta q_s - pretok dodanega sekundarnega toka in Q_m - glavni pretok po osnovnem dovodnem cevovodu (po sesalnem cevovodu) lahko določimo delež dodanega sekundarnega toka v skupnem pretoku. Iz razmernika pretokov (sl. 3) je razvidno, da pri majhnih pretokih poteka 38% celotnega pretoka $Q_c = Q_m + q_s$ kot dodani sekundarni tok, pri največjih pretokih pa le še samo okrog 7%.

where q_s is the capacity of the added secondary flow and Q_m is the main capacity from the intake pipe system (suction pipe system) the portion of the added secondary flow in the common capacity $Q_c = Q_m + q_s$ can therefore be determined. As it is clear from the capacity ratio curve (Fig. 3), this portion is 38% at low operating capacities and is only 7% at high operating capacities.



Sl. 3. *Obratovalna karakteristika* (ξ_Q-ψ) *in* (ξ_Q-η), *dodan sekundarni tok*
Fig. 3. *Capacity-Head* (ξ_Q-ψ) *and Capacity-Efficiency curve* (ξ_Q-η), *added secondary flow*

2.4 Primerjava karakterističnih krivulj ξ_Q-ψ in ξ_Q-η pri obratovanju brez in z dodanim sekundarnim tokom

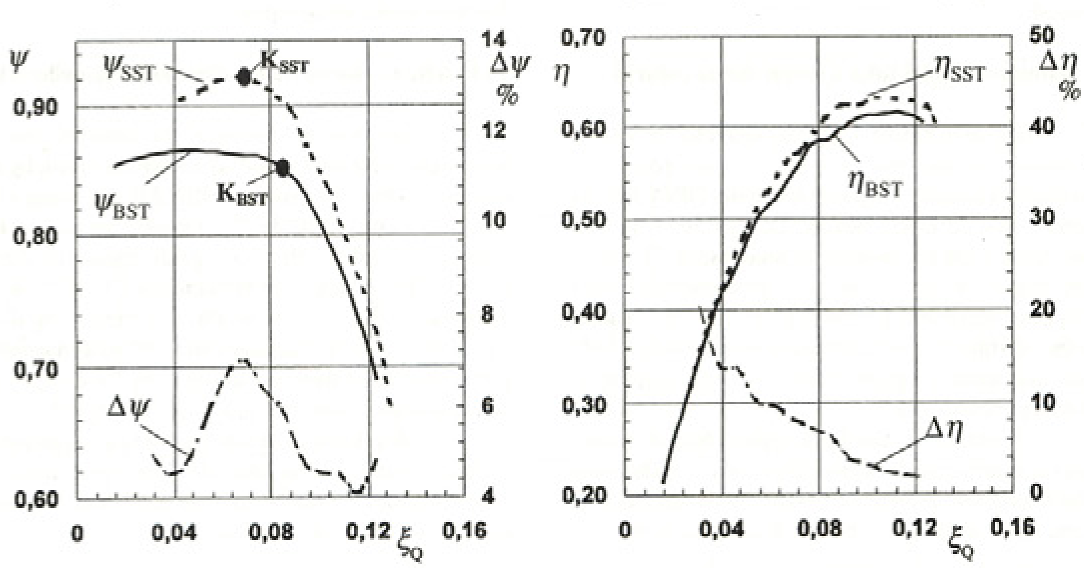
Iz primerjave karakterističnih krivulj (ξ_Q-ψ) pri obratovanju brez in z dodanim sekundarnim tokom (sl. 4 levo) je razvidno povečevanje tlačnega števila ψ proti kritični točki **K** v stabilnem obratovalnem delu krivulje. V povprečju je doseženo tlačno število ψ pri obratovanju na stabilnem delu karakteristične krivulje (ξ_Q-ψ) z dodanim sekundarnim tokom na vstopu v rotor, večje za 5,3 % oz. višje od tlačnega števila kakor pri obratovanju brez dodanega sekundarnega toka. Največje povečevanje tlačnega števila je doseženo prav v kritični točki $\mathbf{K_{SST}}$, in sicer približno 7%. Na celotnem področju obratovalni pretokov, pri obratovanju s sekundarnim tokom, je doseženo najmanj 4% večje tlačno število, glede na obratovanje brez dodanega sekundarnega toka. Na podlagi tega lahko sklepamo, da dodajanje sekundarnega toka na vstopu v rotor ugodno vpliva na celotno obratovalno karakteristiko (ξ_Q - ψ).

Iz primerjave krivulj izkoristkov (sl. 4 desno) med obratovanjem v obeh režimih je razvidno povečanje izkoristka predvsem v področju stabilnega obratovanja. Povečanje izkoristka pri obratovanju z dodanim sekundarnim tokom je med 20,3% pri najmanjših pretokih in 1,9% pri največjih pretokih. Veliko povečanje izkoristka v področju majhnih pretokov je relativno, ker pri majhnih številih(vrednostih) odstopanja hitro naraščajo. Realno lahko podamo povečanje skupnega izkoristka na področju stabilnega obratovanja, in sicer konkretno med 9,7%, v kritični točki, in 1,9% pri največjem pretoku, povprečno torej okrog 7%. V točki optimalnega obratovanja, kjer je izkoristek največji, znaša povečanje izkoristka 3,2% pri obratovanju z dodanim sekundarnim tokom na vstopu v rotor.

2.4 Comparison of the characteristic curves ξ_Q-ψ and ξ_Q-η by impeller operating with and without added secondary flow.

By comparison of the characteristic curves (ξ_Q-ψ), at operating without and with added secondary flow (Fig. 4 left), it is evident that the pressure number ψ has been increased from the critical point K in the stable operating area. The average value of the pressure number ψ, at stable operating characteristic curve (ξ_Q-ψ) part by operating with added secondary flow, has been increased up to 5.3% relative to the pressure coefficient by operating without added secondary flow. In our case the higher-pressure coefficient increase, up to 7%, is achieved at the critical point $\mathbf{K_{SST}}$. In the whole area of operating capacities, at impeller operating with added secondary flow, the pressure number was increased up to 4% relative to the operating, without an added secondary flow. From this fact the suitable secondary flow influence on the impeller operating characteristics (ξ_Q-ψ) could be concluded.

From the efficiency curves (Fig. 4 right), it is evident, with both operating regimes, that the efficiency increases with the added secondary flow, especially at the stable operating part. With secondary flow added, the efficiency of the fan is increased up to 20.3% at minimal capacities, and up to 1.9% at maximal capacities. The higher efficiency increase is achieved in the area of smaller capacities, but this increase is relative because of the small efficiency values. The real efficiency increase has been achieved in the area of the stable impeller operation between the critical point of capacity, where the efficiency increases up to 9.7%, and at the maximal capacities where the efficiency increases up to 7%. The overall efficiency at the optimal impeller-operating regime (at best efficiency point - BEP) by adding secondary flow is increased by up to 3.2%.



Sl. 4. *Primerjava karakteristik (ξ_Q-ψ) in (ξ_Q-η), brez in z dodanim sekundarnim tokom*

Fig. 4. *Capacity-Head (ξ_Q-ψ) and Capacity-Efficiency curve (ξ_Q-η) comparison, without and with added secondary flow*

3 PREDLOG FIZIKALNE RAZLOGE POVEČANJA IZKORISTKA IN STABILNEGA DELA KARAKTERISTIKE

Glede na priporočila [1], da so priporočene hitrosti dodanega sekundarnega toka do dva in polkrat večje ($c_s = 2,5\ c_v$), je mogoče, da se celotne pretočne hitrosti, vsaj ob pokrovni steni rotorja povečajo. S tem pa se doseže boljša polnitev rotorja po celotni širini. Ker sta po Eulerjevi glavni enačbi odločilni obodni hitrosti u_2 in u_1, ki pa sta ob konstantni vrtilni frekvenci rotorja konstantni, ter absolutni hitrosti toka na izstopu in vstopu v rotor v obodni smeri c_{2u}, c_{1u} za doseganje energijske razlike

3 PROPOSED PHYSICAL EXPLANATION OF THE INCREASED EFFICIENCY AND STABLE OPERATING AREA

According to the [1] recommendation, the velocities for the secondary flow should be 2.5 times higher than entrance velocities ($c_s = 2.5c_v$), the common velocities should increase least near by the front shroud. With this increase of velocity the increase of the resultant velocity could be expected to be least near by the front shroud. In the Euler's main equation the circumferential velocities u_1 and u_2 are constant at the impeller constant speed. The absolute flow velocities in circumferential direction c_{1u} and c_{2u} have a high impact in achieving the energy difference.

$$Y_t = u_2 c_{2u} - u_1 c_{1u} \tag{6}$$

je potrebno natančneje preučiti prav absolutni obodni hitrosti toka na vstopu in izstopu iz rotorja. Absolutne hitrosti v obodni smeri na izstopu iz rotorja je moč dokaj preprosto izmeriti z uporabo sonde z vročo nitko oziroma anemomentrijske metode. Na vstopu v rotor, tudi zaradi dodatnega sistema za dovajanja sekundarnega toka, pa so meritve z anemometrijsko metodo zelo omejene, praktično neizvedljive (mogoča bi bila uporaba laserske anemometrijske metode). Zato smo se lotili določanja teh hitrosti z uporabo anemometra z ravnimi lopaticami na razdalji treh vstopnih premerov cevovoda od samega vstopa v rotor [3]. Ob upoštevanju dejstva, da se dodaja sekundarni tok ob stenah vstopnega cevovoda oz. po notranjem obsegu tega lahko pričakujemo, da se bo vstopna vrtinčnost toka umirila. Sekundarni tok je usmerjan na vstop z vodilnimi lopaticami (sl. 1) in je zato tudi v veliki meri vrtinčno nevtralen glede na pretok sekundarnega toka ali glavnega pretoka skozi sesalni cevovod.

Considering that fact, the absolute flow velocities at impeller entrance and exit must be explicitly studied. The absolute flow velocities at the impeller exit could be simply measured using the anemometer method with hot-wire probe. Because of the limited measuring space and the place for the system to adding secondary flow at the impeller entrance is needed, measurement of the entrance flow velocity using a hot-wire anemometer is difficult, practically impossible. The laser-anemometer method is therefore recommended. This problem was the main reason for introducing another method to measure the absolute flow velocity (in hoop direction) at the impeller entrance. The anemometer with straight blades in axial direction is used. It is placed into the main suction pipe at a distance of three intake diameters from the impeller entrance [3]. If the secondary flow is added near by the suction pipe boundary the intake flow whirl at the impeller intake diameter will be reduced. The secondary flow is guided into the impeller eye by guide-vanes (Figure 1). The consequence is mainly the production of a neutral secondary flow whirl relative to the main flow from the main intake suction pipe.

3.1 Izstopne hitrosti toka v smeri širine rotorja

Izstopne hitrosti tekočinskega toka na izstopu iz rotorja so merjene v smeri širine rotorja z enokanalnim anemometrijskim sistemom DISA 55M01 s sondo z vročo nitko. Sonda, DISA 55P11, je bila postavljena v tok po navideznih tokovnicah T1-T10 v smeri širine rotorja od pokrovne proti stojinski steni rotorja, na merilnem premeru D_{3m} = 0,614 m skozi merilno izvrtino. Te so označene kot merilne točke (MT) in so razvrščene s kotnim odmikom dveh stopinj po merilnem premeru (sl. 5).

Iz izmerjene absolutne hitrosti toka v obodni smeri c_{3u}, so določene absolutne hitrosti toka v obodni smeri c_{2u} na rotorjevem izstopnem premeru D_2 ob upoštevanju zakona o potencialnem vrtincu oz. konkretno po zvezi:

3.1 Exit flow velocities in the direction of impeller width

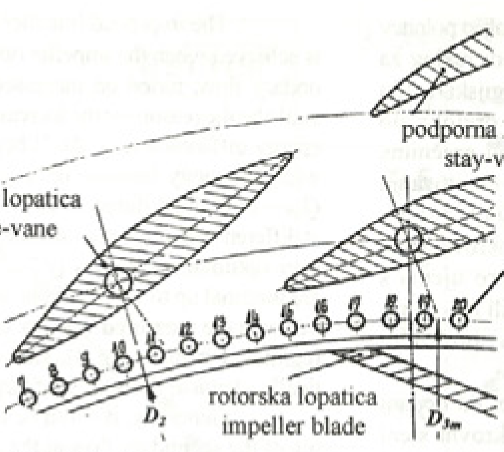
The flow velocities at the impeller exit are measured in the direction of the impeller width by one channel anemometer system DISA 55M01, using a hot-wire probe DISA 22P11. The hot-wire probe is placed through the hole at the front guide apparatus shroud into the flow, at apparent streamlines T1 – T10, in the direction of the impeller width, at a measuring diameter of D_{3m} = 0.614. The holes are marked as measuring points (MP) and they are arranged by division of two degrees at the measuring diameter D_3 (Fig. 5).

Flow velocity at the impeller exit diameter D_2 is determined from the absolute flow velocities in the hoop direction c_{3u}, which are then measured. Considering the law of the potential whirl:

$$\frac{D_2}{2} c_{2u} = \frac{D_{3m}}{2} c_{3u} = r c_u = \text{konst} \tag{7}$$

oziroma:

and



Sl. 5. *Merilni sektor na izstopu iz rotorja na merilnem premeru D_{3m}*
Fig. 5. *Measuring sector at the impeller exit at the measuring diameter D_{3m}*

$$c_{2u} = \frac{D_{3m}}{D_2} c_{3u} \tag{8}$$

in so podane v brezdimenzijski obliki:

or in non-dimensional form:

$$\xi_{Cu} = \frac{c_{2u}}{u_2} \tag{9}$$

v odvisnosti od širine rotorja oz. brezdimenzijskega koeficienta širine:

depending on the impeller width or non-dimensional coefficient of the impeller width:

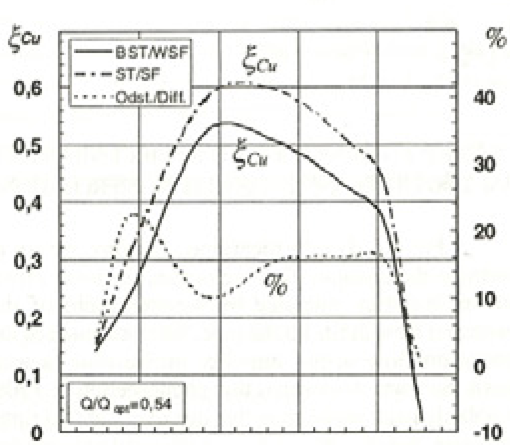
$$\xi_b = \frac{b_x}{b_2} \tag{10}$$

kjer je b_x razdalja od pokrovne stene rotorja do navidezne tokovnice (T1 do T10), na katerih so bile merjene hitrosti.

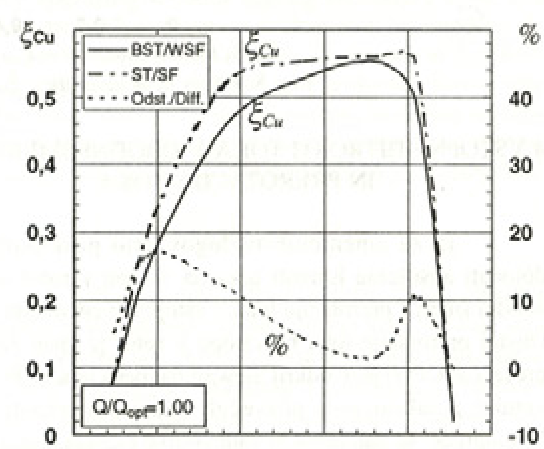
Iz rezultatov meritev hitrosti je razvidno občutno povečanje hitrosti (tudi do 22%, pri obratovanju s pretokom Q/Q_{opt} = 0,54) ob sprednji - pokrovni steni rotorja, kakor tudi po celotni širini rotorja pri obratovanju z dodanim sekundarnim tokom.

where b_x is the distance from the impeller front shroud relative to the apparent current streamline (T1 up to T10) where the velocities are measured.

From the measuring data the velocity increase is evident (up to 22% at the operating capacity ratio Q/Q_{opt} = 0.54). The increased velocities are evident at the front shroud as well as at the total impeller width, when the impeller operates with an added secondary flow.



Sl. 6. *Prečni hitrostni – ξ_{Cu} profil pri Q/Q_{opt} = 0,54 in $n = n_{opt}$*
Fig. 6. *Cross flow velocity – ξ_{Cu} profile at Q/Q_{opt} = 0.54 and $n = n_{opt}$*



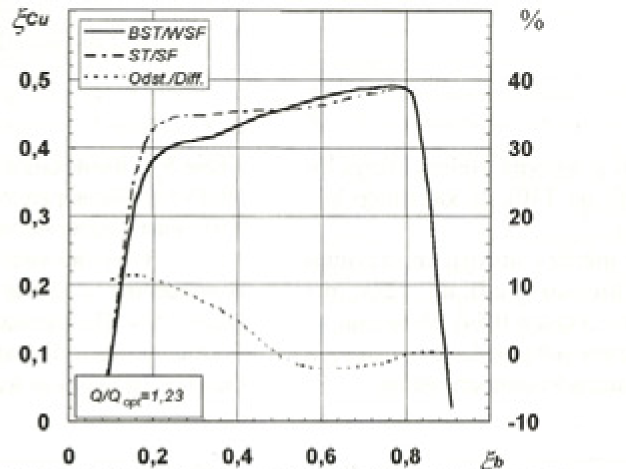
Sl. 7. *Prečni hitrostni – ξ_{Cu} profil pri Q/Q_{opt} = 1,00 in $n = n_{opt}$*
Fig. 7. *Cross flow velocity – ξ_{Cu} profile at Q/Q_{opt} = 1.00 and $n = n_{opt}$*

Na tej osnovi lahko sklepamo na boljšo polnitev rotorja v smeri širine, kar bi lahko bil tudi razlog za povečanje izkoristka in večji doseženi energijski razliki (sl. 4). Z zvečevanjem obratovalnega pretoka do optimalnega Q/Q_{opt} = 1,00 je razlika med prečnima hitrostnima profiloma vse manjša in je pri obratovanju s pretokom, večjim od optimalnega Q/Q_{opt} = 1,23, praktično najmanjša (največ do 11% ob pokrovni steni rotorja). Omenjen potek hitrosti se dobro ujema s potekom izmerjenih karakterističnih krivulj (ξ_Q-ψ) pri obeh obratovalnih režimih, ki se v področju nadoptimalnih pretokov praktično združita.

Splošno lahko torej povzamemo, da dodani sekundarni tok na vstopu v rotor ob pokrovni steni rotorja ugodno vpliva na polnitev rotorja oz. na izkoriščenost celotne razpoložljive širine rotorja. Ker se ob izboljšani polnitvi rotorja absolutno tudi povečajo hitrosti toka, predvsem v obodni smeri, se poveča tudi dosežena energijska razlika oziroma tlačno število rotorja, tako da lahko prav tako splošno sklenemo, da dodajanje sekundarnega toka ugodno vpliva tudi na povečano doseganje energijske razlike rotorja, še prav posebej v področju manjših, podoptimalnih pretokov.

The improved impeller load in width direction is achieved when the impeller operates by an added secondary flow, based on increased flow velocities. This could be the reason for the increased efficiency and larger energy difference (Fig. 4). When the impeller operates with a capacity increase up to the optimal capacity Q/Q_{opt} = 1.00, the differences between the velocity profiles at different operating regimes decrease. By operating with over optimum capacities Q/Q_{opt} = 1.23, the differences are minimal up to 11 % at front shroud. Good agreement between the measured velocity curves at both operating regimes is evident (Fig. 8). The velocity curves are practically identical in the area of large operating capacities.

Generally, it could be concluded that the adding of the secondary flow at the impeller intake, nearby the front shroud, has a positive influence on the impeller-operating load. By better use of the all - impeller width a larger operating load is achieved. At the same time the absolute flow velocity at the impeller exit - in the tangential direction - increases, and consequently, the pressure difference and the energy difference also increases. The added secondary flow has a beneficial effect on larger energy differences, especially at the area in suboptimum operating capacities.



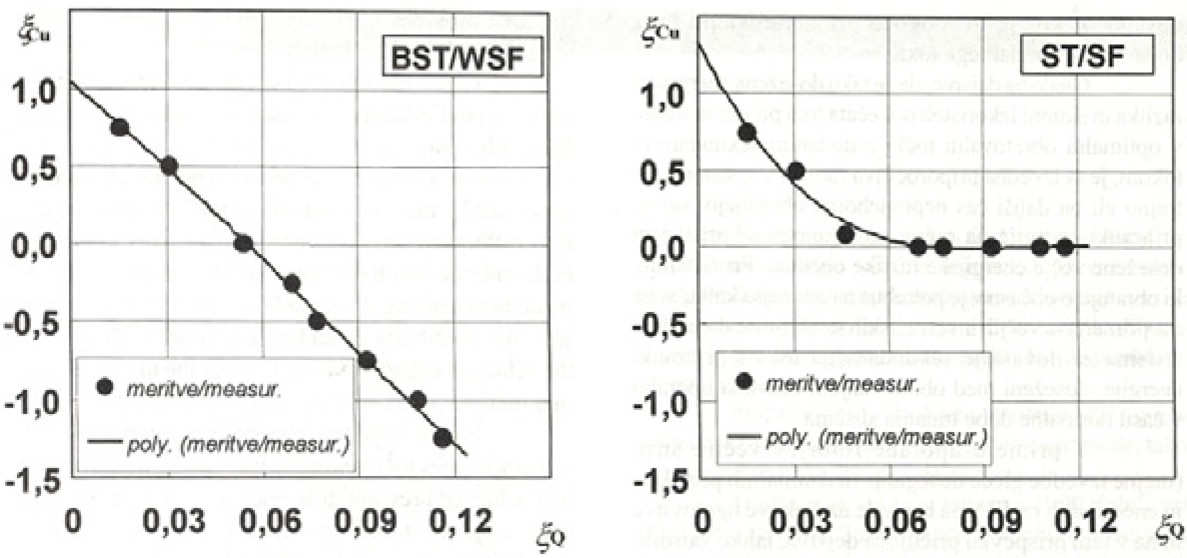
Sl. 8. *Prečni hitrostni – ξ_{Cu} profil pri Q/Q_{opt} = 1,23 in $n = n_{opt}$*
Fig. 8. *Cross flow velocity – ξ_{Cu} profile at Q/Q_{opt} = 1.23 and $n = n_{opt}$*

4 VSTOPNE HITROSTI TOKA V OBODNI SMERI IN PREROTACIJA TOKA

Iz že omenjenih razlogov smo poskušali določati absolutne hitrosti toka na vstopu v rotor v obodni smeri s prerotacijo toka v vstopnem cevovodu. Obstoj prerotacije toka na vstopu v rotor je znan že precej časa. Prvi je to odkril Stewart davnega leta 1909, vendar ji takrat niso posvečali večje pozornosti. Mimogrede, še dandanes ni zadovoljive razlage, zakaj se prerotacijski tok pojavi, spremeni smer in jakost vrtenja glede na smer vrtenja rotorja. Leta 1957 je ta pojav nekoliko podrobneje opisal Stepanoff [4], in sicer z Eulerjevimi vstopnimi trikotniki ob upoštevanju zakonov potencialnega vrtinca, ni pa opisal, zakaj do tega prihaja. Po mojem mnenju je razlog prerotacije v

4 INTAKE FLOW VELOCITIES IN CIRCUMFERENTIAL DIRECTION AND INTAKE FLOW PREROTATION

For the above - mentioned reasons, we try to determine the absolute flow velocities (in hoop direction) at impeller entrance by measurement of the prerotation flow in the intake pipe. The existence of the prerotation flow in the impeller intake pipe is well known. Steward discovered this phenomenon in 1909, but nobody paid attention to this discovery at that time. Nowadays, we still do not have a physical explanation as to why the prerotation flow appears, and changes the direction and the strength of rotation relative to the direction of the impeller rotation. In 1957 this phenomena was explained by Stepanoff [4] by using the Eulerian entrance flow velocity triangles, considering the law of potential whirl, but he did not explain the reasons for this phenomenon. In my opinion, the prerotation flow



Sl.9. *Hitrosti toka na vstopu v rotor v obodni smeri brez dodanega sekundarnega toka (levo), z njim (desno)*
Fig. 9. *Flow velocity in circumferential direction at impeller eye, without (left) and (right) with added secondary flow*

tem, da se tok podobno kakor tok okoli ovire, preusmeri že prej preden zadane v oviro. To pomeni, da se pri določenem obratovalnem pretoku pojavi "zamašitev" rotorskih kanalov, ki pa jo tok lahko odpravi tako, da začne prerotirati v nasprotni smeri vrtenja rotorja, s čimer se pretočna hitrost poveča (absolutno gledano) in s tem omogoči doseganje večjih pretokov. Iz primerjave meritev (sl. 9 levo) prerotacijskih hitrosti toka na vstopu v rotor oz. koeficienta ξ_c je razvidno, da se prerotacija toka ustavi pri obratovanju z dodanim sekundarnim tokom v področju višjih obratovalnih pretokov, konkretno nad $\xi_Q = 0,048$ (sl. 9 desno). Pri obratovanju brez dodanega sekundarnega toka, pri katerem je razvidna prerotacija toka v nasprotni smeri vrtenja rotorja (sl. 9 desno), pri pretokih, večjih od $\xi_Q = 0,05$, pa pri obratovanju z dodanim sekundarnim tokom ne prihaja. To bi lahko bil eden izmed ključnih razlogov, zakaj se dosežena energijska razlika v področju večjih obratovalnih pretokov ne veča v primerjavi z obratovanjem brez dodanega sekundarnega toka.

5 SKLEPI

Dodajanje sekundarnega toka na vstopu v rotor pomembno izboljša polnitev rotorja in s tem tudi celotni izkoristek. Ob tem razširi tudi področje stabilnega dela obratovanja rotorja, ki ima nestabilno karakteristiko, v področje manjših, podoptimalnih obratovalnih pretokov.

Glede na ugodne vplive dodanega sekundarnega toka lahko priporočimo izvedbo s prostim vsesavanjem sekundarnega toka povsod tam, kjer so potrebe po regulaciji oziroma po spreminjanju trenutnega obratovalnega pretoka večje. Razširitev stabilnega dela obratovanja to omogoča v širšem

occurs in the impeller intake pipe similarly to the flow around a solid body, where the flow is diverted before it reaches the body. The flow is arranged in the direction of minimal flow resist, which means that at a certain operating capacity some kind of "choking" occurs in the impeller channels, and at that moment the flow at the intake pipe starts to be prerotated in the opposite direction of the impeller rotation. The flow velocity increases absolutely, and the operating capacities are enlarged. From measuring the results of prerotation velocities at the impeller entrance, as well as from the non-dimensional coefficient ξ_{Cu} (Fig. 9 left and right), it is evident that the prerotation was almost made to disappear by operating with an added secondary flow in the area of larger operating capacities. In our case at $\xi_Q = 0.048$ (Fig. 9 right). By operating without adding secondary flow, the prerotation flow becomes evident with the opposite direction of the impeller rotation, in our case at $\xi_Q = 0.05$ and larger capacities. That could be the key reason why the energy differences do not increase by adding secondary flow in the area of larger operating capacities, in comparison with operating without adding a secondary flow.

5 CONCLUSIONS

The secondary flow addition at the impeller entrance significantly improved the impeller load and overall efficiency. At the same time the area of the stable part of the impeller operating is extended. The area of the non-stable operating characteristic is thereby decreased.

The adding of the secondary flow could be recommended, because of its beneficial influence when wider control - in the sense of the frequent change of the operating capacity - is needed. It is also recommended to use the variant insolving free suction (non-forced) of the secondary flow. The wider stable operatobsegu, kakor je to mogoče pri obratovanju brez dodanega sekundarnega toka.

Glede na dejstvo, da se tako dosežena energijska razlika in skupni izkoristek povečata tudi pri obratovanju v optimalni obratovalni točki z dodanim sekundarnim tokom, je ta izvedba priporočljiva tudi za vse sisteme, ki trajno ali pa daljši čas neprenehoma obratujejo, saj so prihranki energije na račun povečanega izkoristka in dosežene večje energijske razlike občutna. Pri sistemih, ki obratujejo občasno, je potrebna natančnejša kalkulacija oz. primerjava večjih investicijskih stroškov ob dograditvi sistema za dovajanje sekundarnega toka s prihranki energije, doseženi med obratovanjem oziroma uporabo v času normalne dobe trajanja sistema.

V primeru uporabe rotorjev večjih širin (mejne izvedbe glede doseganja maksimalnih pretokov in energijskih razlik), pa bi glede na Eckove ugotovitve in na v tem prispevku pričujoča dejstva, lahko zatrdili, da je tak sistem že skorajda neobhoden oz. nujno potreben.

ing area provides for better control, as it is possible without the added secondary flow.

Given the achieved energy difference, as well as the overall efficiency increase with added secondary flow, this design is recommended for all systems that operate over a longer time period. The energy savings generated by the increased efficiency and achieved energy difference, are significant. For systems which operate only occasionally, detailed calculations must be made between larger investment costs (for the variant, with the possibility of adding a secondary flow), and the achieved energy savings, in order the to ensure normal lifetime of the system.

For the impellers with larger width (higher performance types for maximal capacities, as well as maximal achieved pressure differences) - and with the conclusions given in this article, as well as Eck's recommendations - it could be affirmed that the system for secondary flow addition is almost certainly necessary.

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