UDK 621.438

Značilnice gonilnika radialne plinske turbine

Rotor Characteristics of Radial Gas Turbine

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V prispevku so predstavljene značilnice gonilnika turbine. Definirane so kot odvisnost tlačnega in temperaturnega padca v gonilniku od vpadnega kota in masnega pretoka. Za dejansko turbino jih določimo tako, da z iteracijskim postopkom preoblikujemo konvencionalne značilnice turbine. Pri tem najprej na podlagi modela pretoka v spiralnem vodilniku določimo razmere tik pred gonilnikom. Nato z modelom vpadnih izgub rešimo prehod plinskega toka v gonilnik. Tako izračunamo vpadni kot in razmere v vstopnem prerezu gonilnika. Z dobljenimi rezultati preoblikujemo značilnice turbine v značilnice gonilnika. Primeri uporabe značilnic gonilnika v enodimenzionalnih modelih dvonatočne radialne plinske turbine in radialne plinske turbine s spremenljivo geometrijsko obliko so pokazali, da odlikujeta značilnice gonilnika preprostost uporabe in velika natančnost.

This paper describes turbine rotor characteristics defined as a dependence of pressure and temperature drop in the rotor on the incidence angle and mass flow. A numerical procedure is used to transform the conventional turbine characteristics of an actual radial turbine into rotor characteristics. First the thermodynamic state of gases ahead of the rotor is computed by means of a spiral volute flow model. An incidence losses model is then applied to predict the incidence angle and flow conditions in the rotor entry. Finally, turbine characteristics are transformed into rotor characteristics. Rotor characteristics have been successfully applied in one-dimensional models of twin-turbines and variable area turbines. Good accuracy has been achieved.

0 UVOD

Pretok skozi radialno plinsko turbino brez vodilnih lopatic lahko razdelimo v pretok skozi vodilnik, medprostor in gonilnik turbine. Pretok v vodilniku in medprostoru z dokajšnjo zanesljivostjo in razmeroma preprosto simuliramo z enodimenzionalnimi modeli, ki so le delno [1], [2] ali pa sploh niso [3] podprti z izkustvenimi podatki. V enodimenzionalnih modelih pretoka skozi gonilnik pa je potrebna vrsta pomožnih modelov, predpostavk in izmerjenih konstant. Z njimi določamo energijske izgube pretoka, npr. vpadne izgube, izgube zaradi trenja v kanalih med lopaticami gonilnika, izgube ventiliranja in iztočne izgube [4], [5]. Zanesljivost dobljenih rezultatov je zato vprašljiva, saj je v veliki meri odvisna od pravilne izbire izmerjenih konstant. Negotovost lahko zmanjšujemo tako, da vpeljemo značilnice gonilnika, s katerimi določamo le makro parametre pretoka v gonilniku. Tako je mogoče različne primere pretoka skozi radialno turbino enodimenzionalno simulirati z ustreznima enodimenzionalnima modeloma vodilnika in medprostora ter z uporabo značilnic gonilnika.

0 INTRODUCTION

The flow in the spiral volute, through the interspace and in the rotor, have to be solved successively when the flow through a vaneless radial gas turbine is simulated. The flow between the turbine and the rotor entry may be simulated successfully by onedimensional models, requiring only a few [1], [2] or no empirical data [3], while a great number of sub-models, assumptions and empirical data are needed in one-dimensional models of flow through the rotor. These are necessary in order to predict the rotor losses as incidence losses, fluid friction losses, disk friction losses, clearance losses and exit losses [4], [5]. The accuracy of the results of computation therefore depends first of all on the certitude of the empirical data. To avoid possible uncertainty, especially when no additional data are available, rotor characteristics prescribing the macro parameters of rotor flow can be used. Introducing the rotor characteristics, one-dimensional turbine flow models can be simplified, simulating only the flow between the turbine and rotor entry, while the flow through the rotor is replaced by the rotor characteristics.

1 ZNAČILNICE GONILNIKA

Značilnice gonilnika določajo makro parametre pretoka skozi gonilnik, tako da povezujejo razmere pretoka pred in za gonilnikom. Za znano stanje plina v vstopnem prerezu gonilnika morajo torej določati stanje na izstopu iz njega. Zato smo kot odvisni spremenljivki značilnic gonilnika izbrali razmerje tlakov p_{0r}/p_4 in temperatur T_4/T_{0r}. Podatke o stanju plinov na vstopu v gonilnik smo povezali v dve neodvisni spremenljivki. Prva je parameter masnega pretoka $\dot{m}\sqrt{T_{0r}}/p_{0r}$, ki določa intenzivnost pretoka. Druga neodvisna spremenljivka pa je vpadni kot i, ki določa karakteristične primere pretoka skozi gonilnik. Formalno lahko torej značilnice gonilnika zapišemo:

1 ROTOR CHARACTERISTICS

Rotor characteristics connect the macro parameters of the flow upstream and downstream of the rotor. The state of the gases at the rotor exit can be determined by the rotor characteristics when the flow conditions at the rotor entry are known. The pressure ratio p_{0r}/p_4 and temperature ratio T_4/T_{0r} are therefore chosen as dependent variables of the rotor characteristics, while the rotor inflow conditions are used as independent variables. These independent variables are the rotor mass flow parameter $\dot{m}\sqrt{T_{0r}}/p_{0r}$, prescribing the flow intensity, and the incidence angle i, determining the characteristic flow conditions within the rotor. The rotor characteristics may be formally expressed as:

$$\frac{p_{0r}}{p_4} = f_{p_r}\left(\frac{\dot{m}\sqrt{T_{0r}}}{p_{0r}}, i\right) \tag{1}$$

$$\frac{T_4}{T_{0r}} = f_{T_r}\left(\frac{\dot{m}\sqrt{T_{0r}}}{p_{0r}}, i\right) \tag{2}$$

S transformacijsko iteracijskim postopkom jih je mogoče določiti iz turbinskih karakteristik, ki jih podajajo izdelovalci:

Using a numerical procedure, these are then determined from turbine characteristics supplied by the manufacturer:

$$\frac{p_{03}}{p_4} = f_{p}\left(\frac{\dot{m}\sqrt{T_{03}}}{p_{03}}, \frac{N}{\sqrt{T_{03}}}\right) \tag{3}$$

$$\eta_{sT} = f_{\mathrm{I}}\left(\frac{\dot{m}\sqrt{T_{03}}}{p_{03}}, \frac{N}{\sqrt{T_{03}}}\right) \tag{4}$$

Zaradi enostavnosti je postopek prikazan z diagramom poteka (sl. 1). Vstopni podatki so značilnice turbine, osnovni geometrijski podatki (sl. 2) in obe neodvisni spremenljivki značilnic gonilnika. Za podan parameter masnega pretoka $\dot{m}\sqrt{T_{0r}}/p_{0r}$ lahko z iteracijo izračunamo Machovo število pretoka v vstopnem prerezu gonilnika:

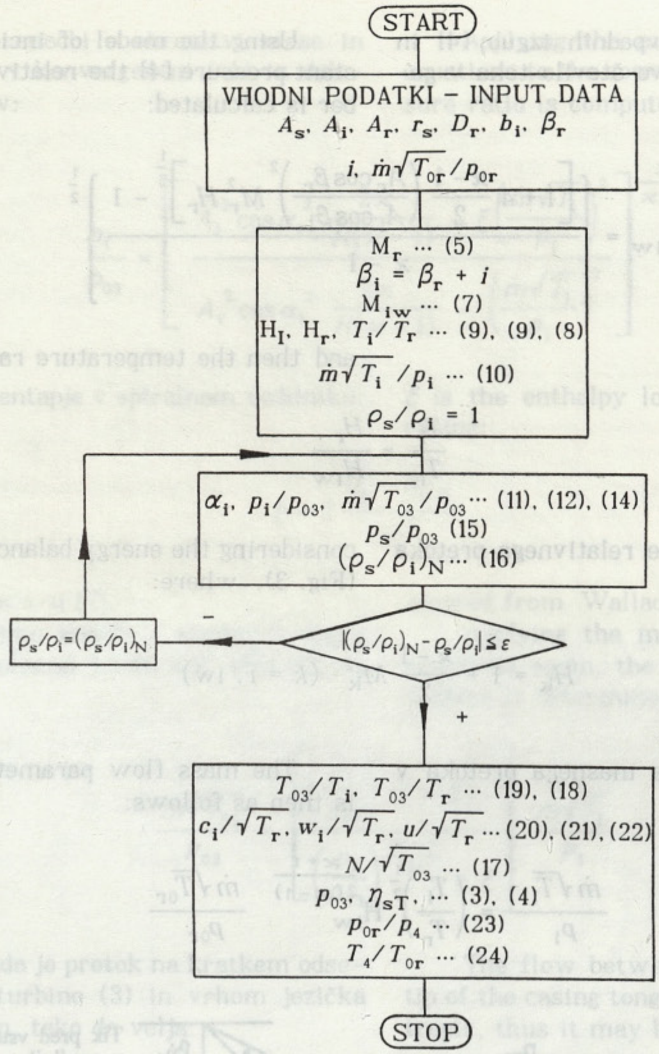
The flowchart to compute the rotor characteristics is shown in figure 1. The input data are the turbine characteristics, turbine geometry (Fig. 2) and the two independent variables of the rotor characteristics. For a given rotor mass flow parameter $\dot{m}\sqrt{T_{0r}}/p_{0r}$, the Mach number at the rotor entry is computed by a simple iteration procedure:

$$M_{r,k+1} = \frac{\dfrac{\dot{m}\sqrt{T_{0r}}}{p_{0r}} \left(\dfrac{R}{\varkappa}\right)^{\frac{1}{2}} \left(1 + \dfrac{\varkappa - 1}{2} M_{r,k}\right)^{\frac{\varkappa + 1}{2(\varkappa - 1)}}}{A_r \cos\beta_r} \tag{5}$$

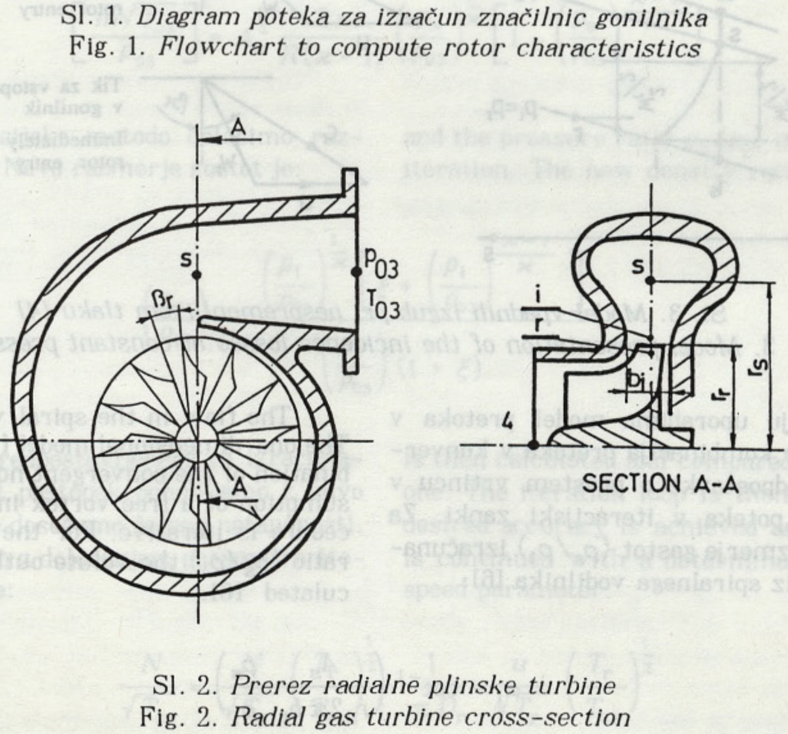
Za podani vpadni kot i sledi relativni kot fluida tik pred gonilnikom:

The relative inflow angle for a given incidence angle is:

$$\beta_1 = \beta_r + i \tag{6}$$



Sl. 1. *Diagram poteka za izračun značilnic gonilnika*
Fig. 1. *Flowchart to compute rotor characteristics*



Sl. 2. *Prerez radialne plinske turbine*
Fig. 2. *Radial gas turbine cross-section*

Nato uporabimo model vpadnih izgub [4] in izračunamo relativno Machovo število toka v gonilnik:

Using the model of incidence losses at constant pressure [4] the relative inflow Mach number is calculated:

$$M_{1w} = \left\{ \frac{\left[1 + 4\,\frac{\varkappa - 1}{2}\left(\frac{A_r \cos\beta_r}{A_1 \cos\beta_1}\right)^2 M_r^2 H_r\right]^{\frac{1}{2}} - 1}{\varkappa - 1} \right\}^{\frac{1}{2}} \qquad (7)$$

in še razmerje temperatur:

and then the temperature ratio:

$$\frac{T_1}{T_r} = \frac{H_r}{H_{1w}} \qquad (8),$$

ki izhaja iz energijske enačbe relativnega pretoka (sl. 3), pri čemer velja:

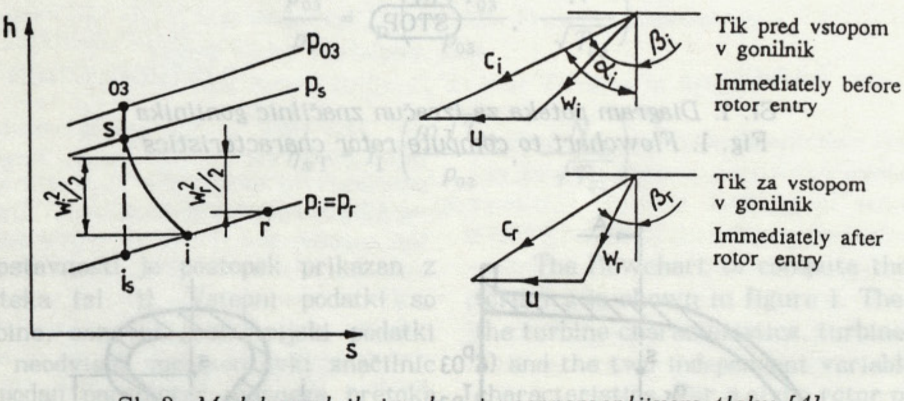
considering the energy balance of the relative flow (Fig. 3), where:

$$H_k = 1 + \frac{\varkappa - 1}{2} M_k^2 \quad (k = \mathrm{r, iw}) \qquad (9).$$

Sledi izračun parametra masnega pretoka v medprostoru:

The mass flow parameter in the interspace is then as follows:

$$\frac{\dot m \sqrt{T_1}}{p_1} = \left(\frac{T_1}{T_r}\right)^{\frac{1}{2}} H_{1w}^{\frac{\varkappa + 1}{2(\varkappa - 1)}} \frac{\dot m \sqrt{T_{0r}}}{p_{0r}} \qquad (10).$$



Sl. 3. *Model vpadnih izgub pri nespremenljivem tlaku [4]*

Fig. 3. *Model presentation of the incidence losses at constant pressure [4]*

V nadaljevanju uporabimo model pretoka v vodilniku [2], ki je kombinacija pretoka v konvergentni šobi in predpostavke o prostem vrtincu v spirali. Postopek poteka v iteracijski zanki. Za predpostavljeno razmerje gostot (ρ_s/ρ_1) izračunamo kot iztekanja iz spiralnega vodilnika [6]:

The flow in the spiral volute is next solved. The one-dimensional model [2] is based on a combination of the convergent nozzle flow and the assumption of a free vortex in the volute. The procedure is iterative. For the given initial density ratio (ρ_s/ρ_1) the volute outlet flow angle is calculated [6]:

$$\alpha_1 = \cot^{-1}\left(\frac{A_s}{r_s 2\pi b_1}\frac{\rho_s}{\rho_1}\right) \qquad (11).$$

Nato uporabimo enačbi o ohranitvi mase in energije pri pretoku v konvergentni šobi in določimo razmerje tlakov:

Applying the mass and energy conservation equations to the convergent nozzle flow, the pressure ratio is computed:

$$\frac{p_1}{p_{03}} = \left[\frac{A_1^2 \cos\alpha_1^2 \dfrac{2\varkappa}{R(\varkappa-1)} - \xi\left(\dfrac{\dot m\sqrt{T_1}}{p_1}\right)^2}{A_1^2 \cos\alpha_1^2 \dfrac{2\varkappa}{R(\varkappa-1)} - \left(\dfrac{\dot m\sqrt{T_1}}{p_1}\right)^2}\right]^{\frac{\varkappa}{\varkappa-1}} \qquad (12).$$

ξ je koeficient izgub entapje v spiralnem vodilniku:

ξ is the enthalpy loss coefficient of the volute casing:

$$\xi = 2\,\frac{h_1 - h_{1,s}}{c_1^2} \qquad (13)$$

in je povzet po Wallace-u [7].

Ponovno uporabimo enačbi o ohranitvi mase in energije in izračunamo koeficient pretoka na vstopu v turbino:

adopted from Wallace [7].

Applying the mass and energy conservation equations again, the mass flow parameter of the turbine is determined:

$$\frac{\dot m\sqrt{T_{03}}}{p_{03}} = \left[\frac{\left(\dfrac{p_1}{p_{03}}\right)^2 (1+\xi)}{\left(\dfrac{p_1}{p_{03}}\right)^{\frac{\varkappa-1}{\varkappa}} + \xi}\right] \frac{\dot m\sqrt{T_1}}{p_1} \qquad (14).$$

Predpostavimo, da je pretok na kratkem odseku med vstopom v turbino (3) in vrhom jezička spirale (s) izentropen, tako da velja:

The flow between turbine entry (3) and the tip of the casing tongue (s) is assumed to be isentropic, thus it may be written:

$$\left[\frac{\dot m\sqrt{T_{03}}}{p_{03}}\right]^2 = A_s^2\,\frac{2\varkappa}{R(\varkappa-1)}\left(\frac{p_s}{p_{03}}\right)^{\frac{2}{\varkappa}}\left[1 - \left(\frac{p_s}{p_{03}}\right)^{\frac{\varkappa-1}{\varkappa}}\right] \qquad (15)$$

in s preprosto iteracijsko metodo določimo razmerje tlakov p_s/p_{03}. Novo razmerje gostot je:

and the pressure ratio p_s/p_{03} may be computed by iteration. The new density ratio:

$$\left(\frac{\rho_s}{\rho_1}\right)_N = \frac{\left(\dfrac{p_1}{p_{03}}\right)^{\frac{1}{\varkappa}}\left[\xi + \left(\dfrac{p_1}{p_{03}}\right)^{\frac{\varkappa-1}{\varkappa}}\right]}{\left(\dfrac{p_1}{p_{03}}\right)(1+\xi)} \qquad (16).$$

Po primerjavi predpostavljenega in izračunanega razmerja gostot postopek ponavljamo z novo vrednostjo, dokler ne dosežemo želene natančnosti. Preračun nadaljujemo z določanjem parametra števila vrtljajev turbine:

is then calculated and compared with the assumed one. The iteration loop is then repeated until the desired accuracy is achieved and the computation is continued with a determination of the turbine speed parameter:

$$\frac{N}{\sqrt{T_{03}}} = \frac{N}{\sqrt{T_r}}\left(\frac{T_r}{T_{03}}\right)^{\frac{1}{2}} = \frac{1}{\pi D_r}\,\frac{u}{\sqrt{T_r}}\left(\frac{T_r}{T_{03}}\right)^{\frac{1}{2}} \qquad (17),$$

Najprej izračunamo razmerje temperatur:

First, the temperature ratio is calculated:

$$\frac{T_{03}}{T_r} = \frac{T_1}{T_r}\,\frac{T_{03}}{T_r} \qquad (18),$$

pri čemer velja:

where:

$$\frac{T_{03}}{T_1} = \frac{1+\xi}{\xi + \left(\frac{p_1}{p_{03}}\right)^{\frac{\varkappa-1}{\varkappa}}} \qquad (19).$$

Nato izračunamo dejansko:

Then follows the computation of actual:

$$\frac{c_1}{\sqrt{T_r}} = \left\{\frac{1}{1+\xi}\,\frac{2\varkappa}{\varkappa-1}\,R\,\frac{T_{03}}{T_r}\left[1-\left(\frac{p_1}{p_{03}}\right)^{\frac{\varkappa-1}{\varkappa}}\right]\right\}^{\frac{1}{2}} \qquad (20)$$

ter relativno hitrost pred vstopom v gonilnik:

and relative flow velocity upstream of the rotor:

$$\frac{w_1}{\sqrt{T_r}} = \frac{w_1}{w_r}\,\frac{w_r}{\sqrt{T_r}} = \frac{T_1\,A_r\cos\beta_r}{T_r\,A_1\cos\beta_1}\,M_r\sqrt{\varkappa R} \qquad (21).$$

Nazadnje sledi iz trikotnikov hitrosti (sl. 3) še obodna hitrost vstopnega roba lopatic gonilnika:

Finally the velocity triangle (Fig. 3) is used to determine the rotor blade tip velocity:

$$\frac{u}{\sqrt{T_r}} = \frac{c_1}{\sqrt{T_r}}\cos\alpha_1 - \frac{w_1}{\sqrt{T_r}}\cos\beta_1 \qquad (22).$$

Za znani vrednosti parametra masnega pretoka in parametra števila vrtljajev lahko iz značilnic turbine določimo razmerje tlakov p_{03}/p_4 in izentropski izkoristek turbine η_{sT} (3) in (4), nato pa napovemo še iskani odvisni spremenljivki značilnic gonilnika, razmerje tlakov:

Once the turbine mass flow parameter and turbine speed parameter are evaluated, the turbine pressure ratio p_{03}/p_4 and turbine isentropic efficiency η_{sT} are determined using turbine characteristics (3) and (4) respectively. It follows the prediction of unknown dependent variables of the rotor characteristics, the pressure ratio:

$$\frac{p_{0r}}{p_4} = \left(\frac{p_1}{p_{03}}\right)\left(\frac{p_{03}}{p_4}\right)H_r^{\frac{\varkappa}{\varkappa-1}} \qquad (23)$$

ter razmerje temperatur:

and the temperature ratio:

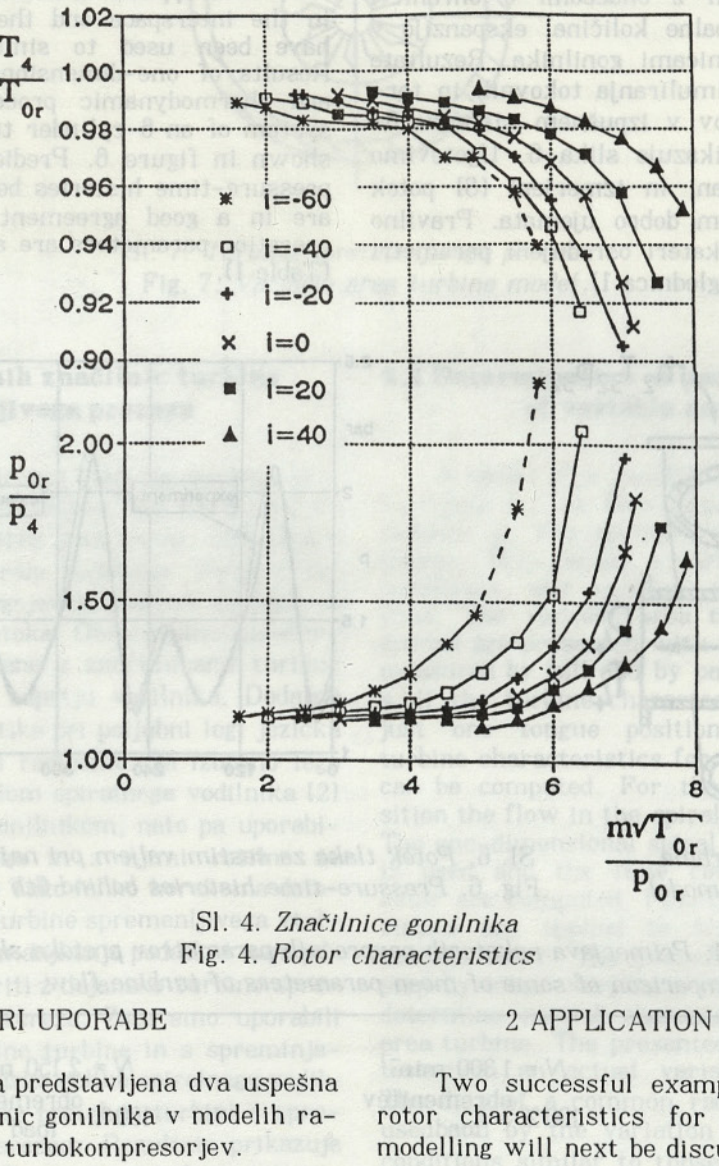
$$\frac{T_4}{T_{0r}} = \left\{1-\eta_{sT}\left[1-\left(\frac{p_4}{p_{03}}\right)^{\frac{\varkappa-1}{\varkappa}}\right]\right\}\frac{T_{03}}{T_r}\,\frac{1}{H_r} \qquad (24)$$

in iteracijski postopek je končan.

and the iteration procedure is terminated.

Na sliki 4 so prikazane značilnice gonilnika, ki smo jih izračunali z opisanim postopkom. Podane so v obliki odvisnosti razmerja tlakov in temperatur od parametra masnega pretoka pri nespremenljivih vrednostih vpadnega kota *i*. Za večje vrednosti vpadnega kota *i* smo karakteristike ekstrapolirali (črtkano), saj stacionarne značilnice turbine, ki jih podaja izdelovalec, za njihov izračun niso zadostovale (vrednost $N/\sqrt{T_{03}}$ je presegla zgornjo mejo).

Rotor characteristics, computed by using the presented iteration method are presented in figure 4, showing the dependence of rotor pressure and temperature ratio on rotor mass flow parameter and the incidence angle as parameter. For the higher values of incidence angle, the characteristic curves are extrapolated (dotted lines), due to insufficient turbine characteristics given by the manufacturer $N/\sqrt{T_{03}}$ exceeds the upper value).



Sl. 4. *Značilnice gonilnika*

Fig. 4. *Rotor characteristics*

2 PRIMERI UPORABE

V nadaljevanju sta predstavljena dva uspešna primera uporabe značilnic gonilnika v modelih radialnih plinskih turbin turbokompresorjev.

2.1 Model robnih pogojev dvonatočne turbine

Večnatočne turbine uporabljamo v primerih impulznega tlačnega polnjenja motorjev z notranjim zgorevanjem. Značilno dvonatočno turbino prikazuje slika 5. Spiralni vodilnik turbine je aksialno pregrajen. Tako je zmanjšan medsebojni vpliv natočnih vej, ki lahko negativno vpliva na procese izpuha in izplakovanja izpušnih plinov iz

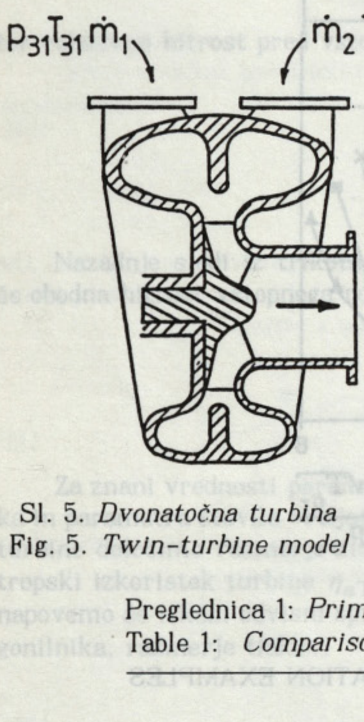
2 APPLICATION EXAMPLES

Two successful examples of application of rotor characteristics for turbocharger turbine modelling will next be discussed.

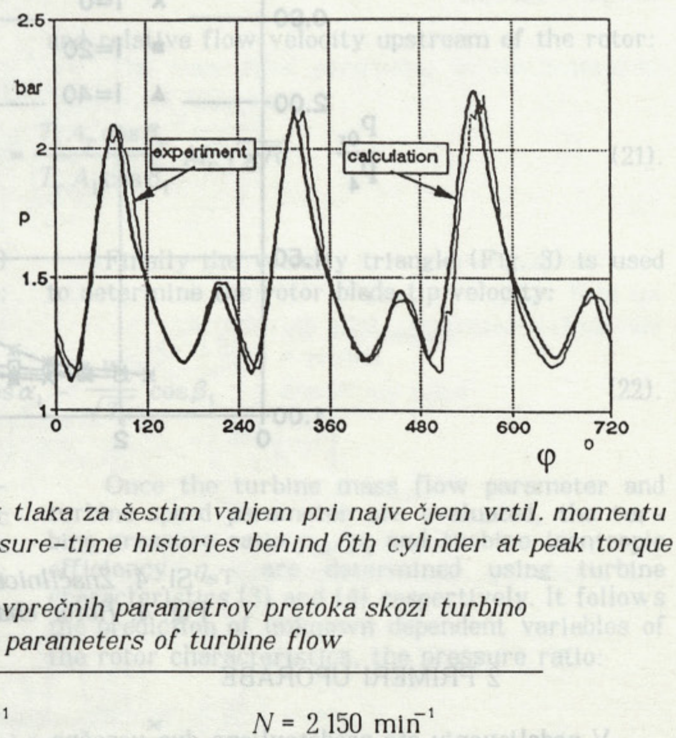
2.1 Twin-turbine boundary conditions model

Multi -entry turbines are usually used when impulse system turbocharging is applied. A so called twin turbine (Fig. 5) is a typical double-entry radial turbine. The volute casing is axially divided to reduce interference between separated inlet branches and to avoid an undesirable influence on the exhaust and scavenging process of the motorja. Zaradi pregrajenega vodilnika se pritekajoča plinska tokova združita šele po ločeni ekspanziji v vodilniku, ko so tlaki že precej nižji in je medsebojni vpliv natokov zmanjšan. Da bi v matematično fizikalnem modelu izpušnega sistema tlačno polnjenega motorja zapisali robne pogoje na turbinskih priključkih, smo izdelali preprost model pretoka v turbini. Vsakega obeh vodilnikov smo simulirali z enodimenzionalnim modelom spiralnega vodilnika [2]. Stekanje plinov v medprostoru smo zapisali z enačbami o ohranitvi mase, energije in gibalne količine, ekspanzijo v gonilniku pa z značilnicami gonilnika. Rezultate enodimenzionalnega simuliranja tokovnih in termodinamičnih procesov v izpušnem sistemu 6-valjnega motorja prikazuje slika 6. Ugotovimo lahko, da se izračunani in izmerjeni [8] potek tlaka za šestim valjem dobro ujemata. Pravilno so napovedani tudi nekateri osrednjeni parametri delovanja turbine (preglednica 1).

engine. The two gas flows, divided by the volute casing, join after substantial area reduction, where pressures are lower, and the undesired interference is moderated. A simple twin-turbine flow model determining the boundary conditions at both turbine entries and at the turbine exit has been developed and incorporated in an engine exhaust system simulation program. Each volute halve has been separately simulated by a one-dimensional spiral volute flow model [2]. Mass, energy and momentum conservation equations have been applied in connection with the mixing in the interspace and the rotor characteristics have been used to simulate rotor operation. Results of one-dimensional simulation of flow and thermodynamic processes in the exhaust system of an 6-cylinder turbocharged engine are shown in figure 6. Predicted and measured [8] pressure-time histories behind the 6-th cylinder are in a good agreement. Some mean turbine operation parameters are also correctly predicted (Table 1).



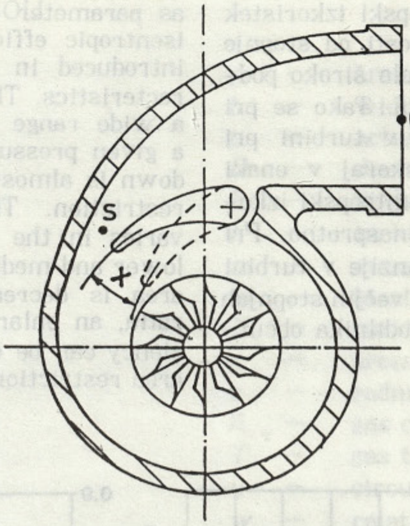
Sl. 5. *Dvonatočna turbina*
Fig. 5. *Twin-turbine model*



Sl. 6. *Potek tlaka za šestim valjem pri največjem vrtil. momentu*
Fig. 6. *Pressure-time histories behind 6th cylinder at peak torque*

Preglednica 1: *Primerjava nekaterih povprečnih parametrov pretoka skozi turbino*
Table 1: *Comparison of some of mean parameters of turbine flow*

		$N = 1\,300$ min^{-1} 100 % obremenitev load		$N = 2\,150$ min^{-1} 100 % obremenitev load	
		meritev experiment	račun calculation	meritev experiment	račun calculation
p_3	bar	1,424	1,411	2,133	2,120
p_4	bar	1,007	1,010	1,050	1,036
T_3	K	906,5	903,9	853,5	850,1
T_4	K	842,0	846,7	745,0	748,6
$\dot{m}_T$	kg/s	0,211	0,210	0,411	0,409



Sl. 7. *Turbina spremenljivega prereza*
Fig. 7. *Variable area turbine model*

2.2 Izračun delnih značilnic turbine spremenljivega prereza

Turbino spremenljivega vtočnega prereza prikazuje slika 7. S spreminjanjem lege gibljivega jezička se spreminja prerez *s* in zaradi tega stanje fluida na vstopu v spiralo vodilnika. To povzroči spremembo vtočnih pogojev v gonilnik turbine in omogoča regulacijo pretoka. Obratovalne karakteristike turbine so podane z značilnicami turbine pri polnem in delnem odprtju vodilnika. Dodatne obratovalne karakteristike pri poljubni legi jezička je mogoče določiti tudi računsko. Za izbrano lego jezička najprej z modelom spiralnega vodilnika [2] določimo stanje pred gonilnikom, nato pa uporabimo značilnice gonilnika in z njimi rešimo še ekspanzijo v gonilniku. Tako lahko korakoma določimo delne značilnice turbine spremenljivega vtočnega prereza. Zaradi pomanjkanja podatkov nam postopka ni uspelo preveriti z dejansko turbino spremenljivega vtočnega prereza. Zato smo uporabili podatke navadne radialne turbine in s spreminjanjem vstopne geometrijske oblike spiralnega vodilnika simulirali pogoje za pretok v turbini s spremenljivim vtočnim prerezom. Rezultate prikazuje slika 8. Pretočne značilnice so pri nekem nespremenljivem parametru števila vrtljajev, podane kot odvisnost stopnje ekspanzije v turbini od parametra masnega pretoka in odprtosti vodilnika turbine – relativnega pretočnega prereza:

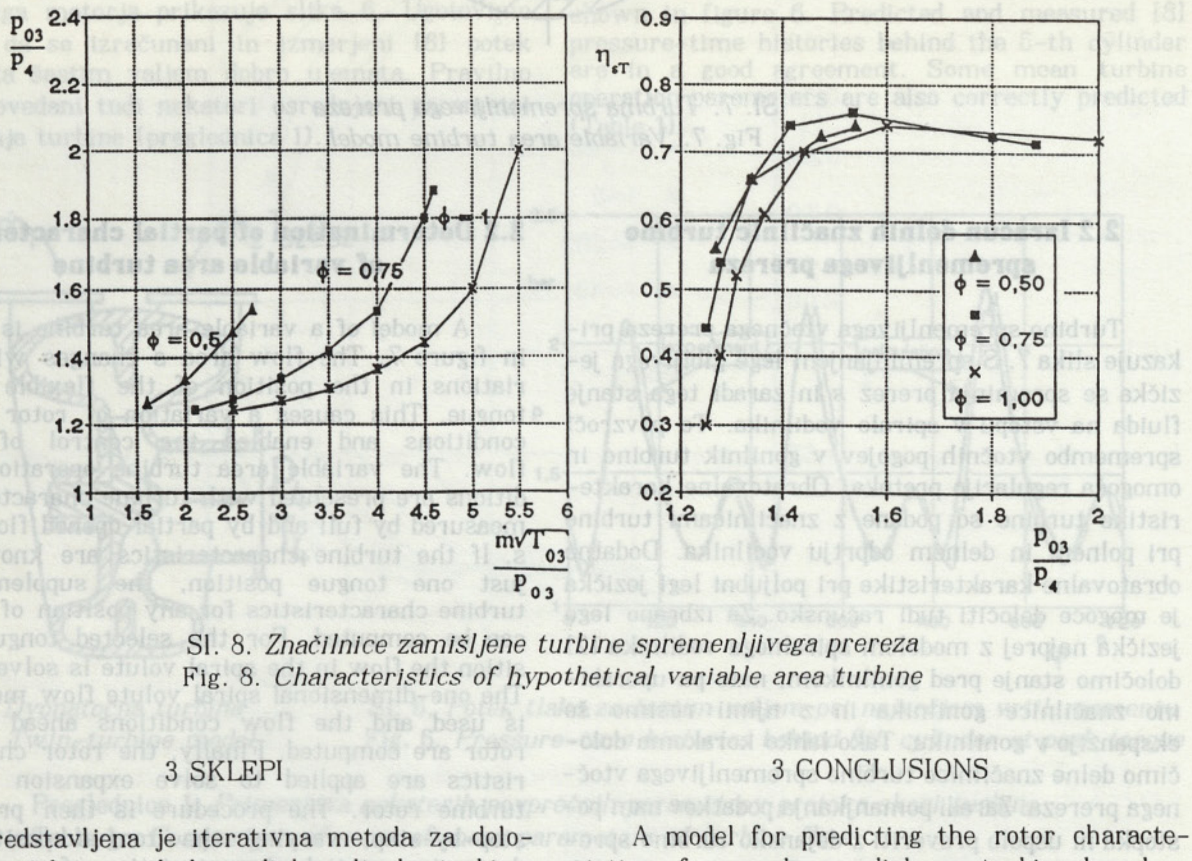
2.2 Determination of partial characteristics of variable area turbine

A model of a variable area turbine is shown in figure 7. The flow area s changes with variations in the position of the flexible volute tongue. This causes a variation of rotor inflow conditions and enables the control of mass flow. The variable area turbine operation conditions are presented with turbine characteristics measured by full and by partial opened flow area s. If the turbine characteristics are known for just one tongue position, the supplementary turbine characteristics for any position of tongue can be computed. For the selected tongue possition the flow in the spiral volute is solved first. The one-dimensional spiral volute flow model [2] is used and the flow conditions ahead of the rotor are computed. Finally, the rotor characteristics are applied to solve expansion in the turbine rotor. The procedure is then proceeded step-by-step, varying the tongue position to determine partial characteristics of a variable area turbine. The presented method has not been tested on an actual variable area turbine yet. The data of a common radial turbine have been used and by the variation of flow area s, flow conditions similar to those in an actual variable area turbine have been achieved. The results are shown in figure 8. Turbine flow characteristics are presented as dependence of the turbine pressure ratio on turbine mass flow parameter and the relative flow area:

$$\Phi = \frac{A_s(x)}{A_{s,max}} \qquad (25),$$

ki se pojavi kot parameter. Izentropski izkoristek turbine je nato prikazan v odvisnosti od stopnje ekspanzije v njej. Opazimo lahko zelo široko področje spreminjanja masnega pretoka. Tako se pri nespremenljivi stopnji ekspanzije v turbini pri prepolovljeni odprtosti vodilnika skoraj v enaki meri zmanjša tudi masni pretok. Izentropski izkoristek turbine se spreminja prav nasprotno. Pri majhnih in srednjih stopnjah ekspanzije v turbini se s pripiranjem vodilnika veča, pri večjih stopnjah ekspanzije pa povzroči pripiranje vodilnika občutnejše zmanjšanje izkoristka.

as parameter at constant turbine speed, while isentropic efficiency—pressure-ratio curves are introduced in connection with efficiency characteristics. The mass flow parameter varies in a wide range for different tongue settings. At a given pressure ratio, the corresponding turndown is almost as high as the volute flow area restriction. The turbine isentropic efficiency varies in the opposite way. It is increased at lower and medium pressure ratia when the flow area is decreasing, while at higher pressure ratia, an enlarged inclination of isentropic efficiency can be observed for increased volute flow area restrictions.



Sl. 8. *Značilnice zamišljene turbine spremenljivega prereza*

Fig. 8. *Characteristics of hypothetical variable area turbine*

3 SKLEPI

Predstavljena je iterativna metoda za določitev značilnic gonilnika radialne plinske turbine, ki temelji na uporabi zakonitosti termodinamike stisljivih tekočin in podana sta dva primera uporabe značilnic gonilnika. V prvem primeru nam je uspelo pravilno popisati robne pogoje na priključkih dvonatočne turbine in model tudi eksperimentalno potrditi. V drugem primeru smo predstavili metodo za določitev delnih karakteristik turbine spremenljivega prereza. Postopka z meritvami sicer nismo potrdili, vendar so dobljeni rezultati prikazali pravilno usmeritev in so dobra spodbuda za nadaljevanje dela.

3 CONCLUSIONS

A model for predicting the rotor characteristics of a vaneless radial gas turbine based on the application of the thermodynamics of compressible fluids has been described and application examples have been given. In the first case, twin-turbine boundary conditions have been correctly predicted and the proposed turbine flow model has been experimentally validated. In the second example, a method for predicting partial characteristics of a variable area turbine has been presented. This method has not been experimentally validated yet. However, the obtained predictions are promising and encourage further development of this model.

4 UPORABLJENI SIMBOLI

A	–	prerez
b	–	širina
c	–	dejanska hitrost
D	–	premer
h	–	entalpija
i	–	vpadni kot
m	–	masni pretok
M	–	Machovo število
N	–	število vrtljajev turbine
p	–	tlak
r	–	polmer
R	–	plinska konstanta
T	–	temperatura
u	–	obodna hitrost
w	–	relativna hitrost
α	–	dejanski kot pretoka
β	–	relativni kot pretoka
η	–	izkoristek
$\varkappa$	–	razmerje specifičnih toplot
ξ	–	koeficient izgub
ρ	–	gostota
Φ	–	relativni pretočni prerez
φ	–	kot zavrtitve ročične gredi

INDEKSI

$_i$	–	medprostor, stanje pred gonilnikom
$_N$	–	nov
$_r$	–	vstop v gonilnik
$_s$	–	izentropno
$_T$	–	turbina
$_w$	–	relativno
$_0$	–	zajezna veličina
$_3$	–	pred turbino
$_4$	–	za turbino

4 NOTATION

A	–	area
b	–	width
c	–	actual velocity
D	–	diameter
h	–	enthalpy
i	–	incidence angle
m	–	mass flow
M	–	Mach number
N	–	turbine rotational speed
p	–	pressure
r	–	radius
R	–	gas constant
T	–	gas temperature
u	–	circumferential velocity
w	–	relative velocity
α	–	actual flow angle
β	–	relative flow angle
η	–	efficiency
$\varkappa$	–	specific heat ratio
ξ	–	enthalpy loss coefficient
ρ	–	density
Φ	–	relative flow area
φ	–	crank angle

SUBSCRIPTS

$_i$	–	interspace, before rotor
$_N$	–	new
$_r$	–	rotor entry
$_s$	–	isentropic
$_T$	–	turbine
$_w$	–	relative
$_0$	–	stagnation conditions
$_3$	–	upstream of turbine
$_4$	–	downstream of turbine

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