THE SZEWALSKI INSTITUTE OF FLUID-FLOW MACHINERY
POLISH ACADEMY OF SCIENCES

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

117

WYDAWNICTWO IMP PAN

GDAŃSK 2005

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ISSN 0079-3205

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

No. 117, 2005, 39-58

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Low-power gas turbines

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Abstract

This paper overviews the development of small scale gas turbines. It analyses the possibilities of their potential evolution in the arrangements of gas turbines systems and defines the main tasks and aims for the design of one-stage compressors and radial expanders. The analysis concerns also characteristic features of combustors and regeneration heat exchangers. It finally determines trends, which are necessary for further improvement of small scale gas turbines.

Keywords: Microturbine; Low-power gas turbine cycles; Regeneration

Nomenclature

a_0	–	speed of sound, calculated at total temperature, $a_0 = \sqrt{kRT_0}$, m/s
c_p	–	specific thermal capacity, kJ/kg K
h	–	enthalpy, kJ/kg
k	–	isentrope index, overall heat-transfer coefficient,
l	–	specific work of the gas turbine, $l = \frac{N}{\dot{m}}$, kJ/kg
m	–	mass flux, kg/s
m_z	–	reduced mass flux, $m_z = m \cdot a_0/(p_0 A)$,
p	–	pressure, Pa
t	–	temperature, °C
w_p	–	thermal capacity of the air in the regenerator, kW/K
w_{sp}	–	thermal capacity of the combustion gas in the regenerator, kW/K
A	–	characteristic surface, m^2
D	–	characteristic diameter, m
D_z	–	$D/\sqrt{A}$

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K_n – specific speed
N – number of heat transfer unit (Number of heat Transfer Unit – NTU)
R – individual gas constant, degree of regeneration (recovery ratio)
Re – Reynolds number,
T – temperature, K
β – ratio of the flux of fuel to the compressor downstream flux
δ_n – diameter index,
ε – compression ratio,
ς_0 – $\varsigma_0 = \frac{T_1}{T_0}$, Fig. 1
ς_R – $\varsigma_R = \frac{T_{2a}}{T_0}$, Fig. 1
ς_{SP} – $\varsigma_{SP} = \frac{T_4}{T_1}$, Fig. 1
ς_0 – $\varsigma_0 = \frac{T_0}{T_1}$, Fig. 1
ς_T – $\varsigma_T = \frac{T_3}{T_0}$, Fig. 1
η – $\eta = w_{op}/w_p$
ω – angular velocity
ω_z – reduced angular velocity, $\omega_z = \omega\sqrt{A}/a_0$
ξ_R – thermal efficiency of the regenerator
μ – $\mu = R/c_p$
σ – number of hydraulic losses

Subscripts

max – maximum value
min – minimum value
S – compressor
T – turbine
n – nominal
R – regenerator
P – air
Sp – combustion gas

1 Introduction

Experience gathered in the course of constructing gas turbines for cars and turboloaders as well as small engines for aircraft models or aiding engines applied in aircraft technology, and also in the range of micro-robots, accelerated the development of small-scale gas turbines applied in power engineering. The term "small scale" is not univocal. Generally, it comprises turbines with a power rating of less than 3 MW [1]. However, such a division has not been commonly accepted; very often turbines with a power rating up to 20 MW are classified within this group.

Within the range from 0 to 3 MW micro-gas-turbines may be further divided (μTG). These are high-speed turbines ($n = 30 \cdot 10^3 - 10^6$ rpm) with power ratings up to 100 kW (250 kW). We can also distinguish a mesoscopic scale (the characteristic lengths of the flow system do not exceed scores centimeters). Micro-gas-turbines belonging to this group are characterized by power ratings below 1 kW and encountered rotational speeds are less than $5 \cdot 10^5$ rpm. The handbook [2] disinguishes between μTG – up to 200 kW and mini-gas-turbines (mTG) 200-475 kW.

From among numerous predictions concerning the dissemination of μTG and mTG it seems to be expedient to present data quoted in [2]. Table 1 contains information concerning the predicted number of installations put into service in the years 2003 – 2012; Table 2 provides data about starting such installations in the respective years.

Table 1. Number of installations put into service in the years 2003-2012.

No.	Power rating	Number of installations	Share %	Value of sales, USD	Share of sales %
1.	Up to 100 kW	10150	81 %	338 mln	40
2.	100-475 kW	2350	19 %	511 mln	60

The number of installations with a power rating of up to 100 kW is much higher than those with a power rating ranging from 100 to 475 kW, because there are more consumers within this range of power ratings. The total value of sales in the range of this class of turbines amounts approximately to about 850 million USD. In the group of turbines with up to 100 kW installations with a power rating from the range 30 – 100 kW amounts to about 80 %.

Table 2. Number of produced turbines in the respective years.

No.	Power rating	2003	2004	2005	2006	2007	2008	2009	2010	2011	2012
1.	Up to 100 kW	645	665	690	965	1225	1325	1330	1310	1050	945
2.	100-475 kW	105	90	150	210	300	310	330	300	280	275

The data contained in Table 2 indicate that the maximum demand for the considered installations has been predicted for the years 2008 – 2010. In the following two years the number of turbine units within this range of power rating will drop. In distributed energy installations of combined heat and power generation μTG and mTG compete with piston gas engines. Their predicted dissemination indicates that they are rather successful in this competition, particularly in the range of power rating exceeding 30 kW. They also constitute an elastic engine in hybrid systems with fuel cells (including those with a low power rating).

The present paper deals with the characteristic technological and structural features of μTG and mTG.

2 General characteristics of the technological structures

The considered classes of turbines (gas turbine cycles) usually operate in an open system with internal combustion. The competition with piston engines in the construction of distributed source of the generation of heat and electricity, as well as the difficulties related to cooling of the flow system require a general application of regeneration. The fundamental, applied and considered technological structures, are to be seen in Fig. 1.

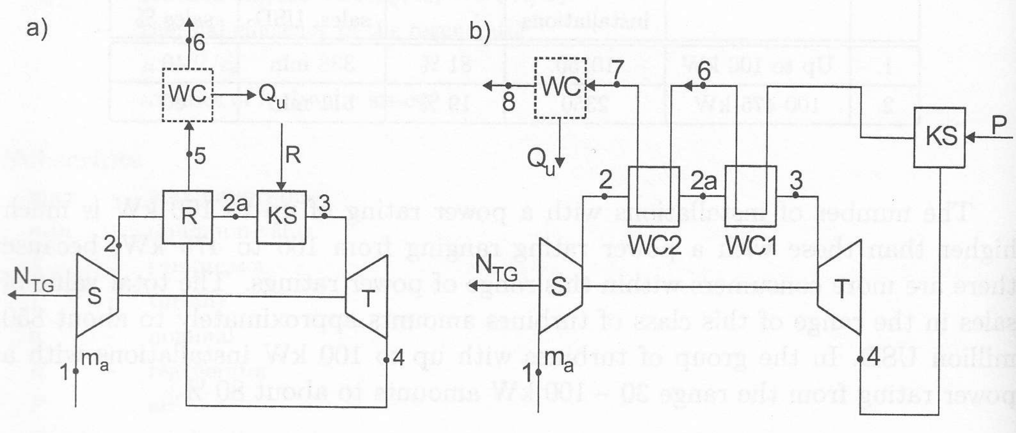


Figure 1. Technological structures of small-scale gas turbines. a) system with internal pressure combustion, b) system with external atmospheric combustion; WC – heat exchangers, KS – combustion chamber, P – fuel, R – regenerator, S – compressor, T – turbine (expander).

Figure 1a presents a system with internal combustion. It may be considered both with regeneration and without a regenerator. Figure 1b illustrates a system with external combustion, which can be extended to a smaller or larger degree. The diagram shows two heat exchanger for the preheating of the air. Such a solution may be justified by material reasons. Both general structures can be applied in the combined co-production of heat and electricity. The combination of these systems of internal and external combustion permits to obtain a new system, which might be expedient if two different kinds of fuel are available (e.g. gas and biomass, Fig. 2). Various possibilities of such combinations have been discussed in [3, 4, 5]. However, these are not the only suggestions. From among various attempts of improving the efficiency the following ones may be taken into account: humidification of the air (HAT systems), introduction of air (steam) into various nodes of installation [6÷9], cooling in the course of compression [8, 11].

As an example of system with intercooling may serve the installation presented in Fig. 3 [11]. In the case of of turbines with a low power rating also the cooling of air at the inlet the compressor may be applied [e.g. 12] or water may be injected in the course of compression [13, 14].

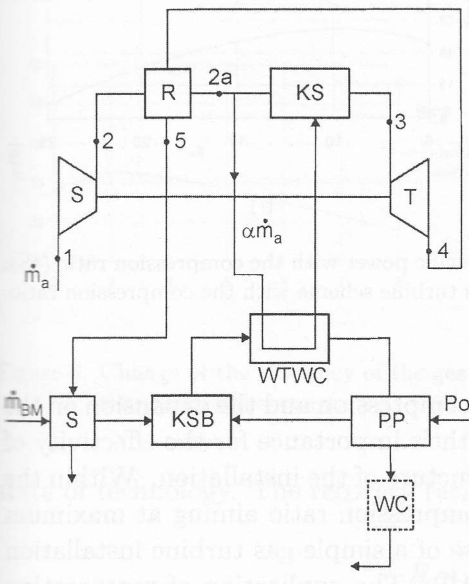


Figure 2. Double-fuel system (gas + biomass). KSB – combustion chamber of the biomass. SP – drying and preparation of the fuel, PP – preheater of air, WTWC – high-temperature heat exchanger.

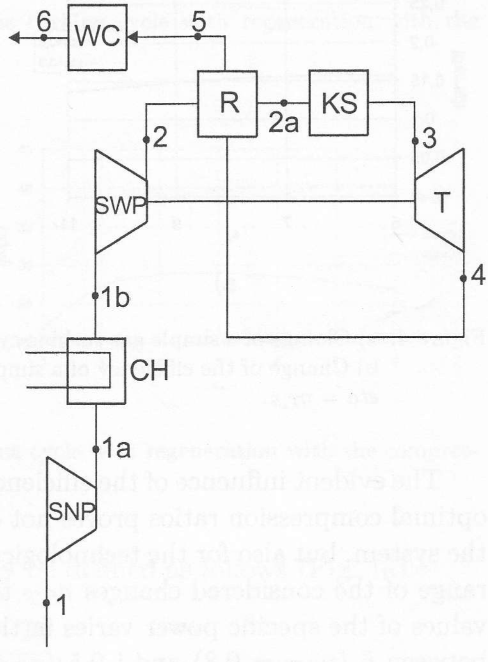
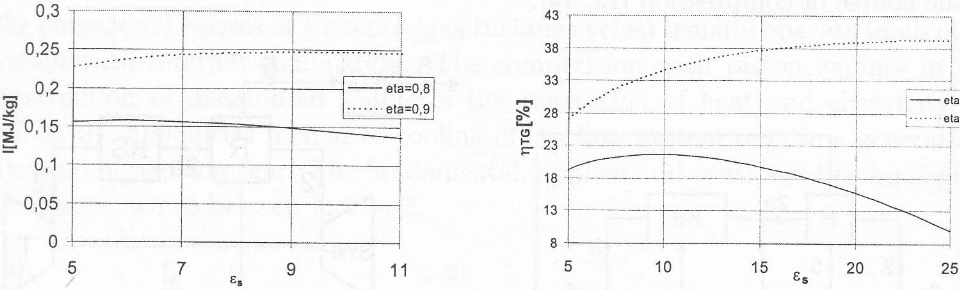


Figure 3. System with internal combustion and intercooling.

In the compression and expansion system μTG or mTG single-stage compressors and expanders are applied. This must be taken into account when choosing the compression ratio. Figure 4 provides the results of calculations concerning the temperature ahead of the turbine (TIT) T = 1173 K (this temperature may be considered to approach the maximum temperature, when the flow system of the turbine is not being cooled down) and two different efficiencies $\eta_{PS} = \eta_{PT} = 0.8$

and 0.9. The first one of these two values may be considered as attainable in near future (which depend on the construction of the block and the possibility of compressing without any inflow of heat); the second one will be possible to be attained later. The calculations presented in Figs. 4-6 were carried out without taking into consideration the hydraulic losses and the losses in the combustion chamber, reflecting some potential values of the efficiency of these systems and their unit power ratings[1].



a) b)

Figure 4. a) Change of a simple gas turbine cycle specific power with the compression ratio (eS); b) Change of the efficiency of a simple gas turbine scheme with the compression ratio, $eta = \eta_{T,S}$.

The evident influence of the efficiency of compression and the expansion on the optimal compression ratios proves not only their importance for the effectivity of the system, but also for the technological structure of the installation. Within the range of the considered changes $\eta_{S,T}$ the compression ratio aiming at maximum values of the specific power varies in the case of a simple gas turbine installation between 6 ($\eta_{S,T} = 0.8$) and i 9.5 ($\eta_{S,T} = 0.9$). The application of regeneration does not affect any change of these values.

The maximum efficiencies are attained in the case of a simple system by ε_S amounting to 10 and 25, respectively. Regeneration reduces the optimal compression ratios considerably. Figure 6 indicated that at high recovery ratios they amount to 3 – 4. In the case of lower recovery ratios higher compression ratios must be applied in order to reach maximum efficiencies. In modern designs of μTG and mTG, and also in parametric analyses the compression ratios are contained within the range of 2 – 6, i.e. they correspond to the maximum efficiency achieved at high values of the recovery ratio. The efficiencies attained for these compression ratios at $R = 0.9$ are potentially possible considering the present

[1]The calculations presented in Figs. 4, 5 and 6 were carried out by S. Lepszy, MSc.

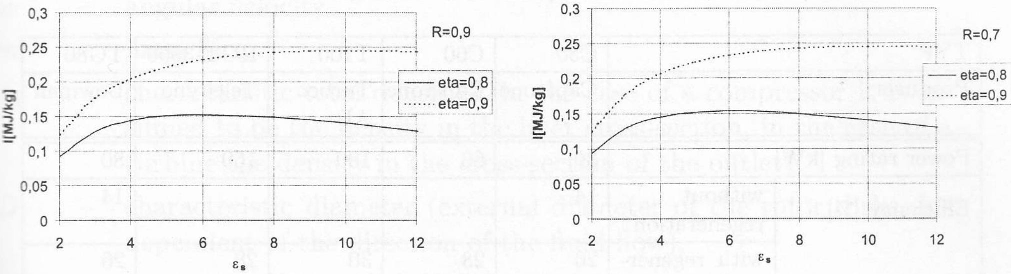


Figure 5. Change of the specific power of the gas turbine cycle with regeneration with the compression ratio, $eta = \eta_{pS,T}$.

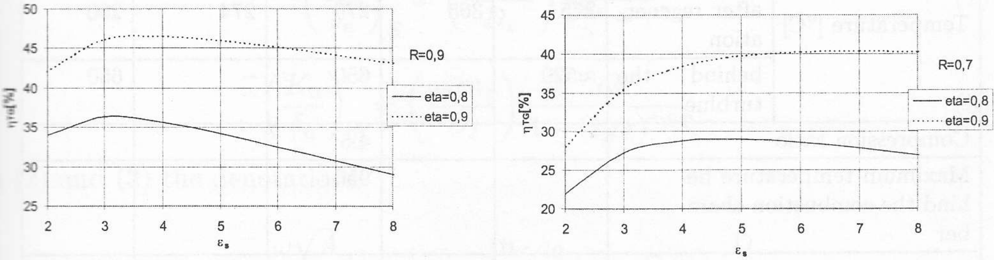


Figure 6. Change of the efficiency of the gas turbine cycle with regeneration with the compression ratio, $eta = \eta_{pS,T}$.

state of technology. The recovery ratio may be defined as follows (Fig. 1a):

$$R = \frac{T_{2a} - T_2}{T_4 - T_2}\,.$$

Table 3. provides data concerning some of the produced installations (taken over from catalogues).

3 Problems concerning the design and flow

3.1 Design of the rotors of compressors and turbines. General problems

As has already been stated, compressors and expanders in μTG and mTG are constructed as single-stage installations. In order to attain adequate compression ratios and degree of expansion, these must be radial or diagonal stages. Assuming

Table 3. Selected types of small-scale gas turbines.

TYP		C30	C60	T100	RGT-3600	TG80
Producer		Capstone	Capstone	Turbec AB	Teledyne	Bowman
Power rating [kW]		30	60	100	150	80
Efficiency [%]	without regeneration	14				14
	with regeneration	26	28	30	28	26
Total efficiency	without regeneration	–	–			
	with regeneration	70	78.4	8.0		75
Temperature [°C]	after regeneration	275	305	270	274	260
	behind the turbine	≈520		650	–	650
Compression ratio				4.5		
Maximum temperature behind the combustion chamber				950		
Angular velocity, thousands per minute		96	96	70	36	68
NOx (ppm, 15 % O_2)		< 9	< 9	<15		< 25
Noise (dBA)		65 (10 m)	70 (10 m)	70 (1 m)		77 (1 m)
Mass flux [kg/sec]		0.31	0.49	0.8		
Bearings		air	air	oil	oil	oil
Number of shafts		1	1	1	2	1

classical definitions of the specific speed K_n and diameter index δ_n [15], we have:

$$K_n = \frac{\omega\sqrt{\frac{m}{\rho_0}}}{(\Delta h_s)^{3/4}} \qquad \delta_n = \frac{D\,(\Delta h_s)^{1/4}}{\sqrt{\frac{m}{\rho_0}}} \tag{1}$$

where:

ω – angular velocity,

m – mass flux,

ρ_0 – characteristic total densities (in the case of a compressor it is assumed to be the density in the inlet cross-section, in the case of a turbine the density in the cross-section of the outlet),

D – characteristic diameter (external diameter of the rotor disks, independent of the direction of the fluid flow),

Δh_s – isentropic change of the total enthalpy,

and express it by means of the following relations (the index S denotes the compressor, the index T the turbine):

$$\left(\frac{K_n}{\delta_n}\right)_S = \left(\frac{\omega_z m_z}{D_z}\right)\frac{\mu_s}{\varepsilon_S^{\mu_s}-1}, \tag{2}$$

$$\left(\frac{K_n}{\delta_n}\right)_T = \left(\frac{\omega_z m_z}{D_z}\right)\frac{\mu_T}{1-(\varepsilon_S\sigma)^{-\mu_T}}. \tag{3}$$

In (2) and (3) the denotations

$$\omega_z = \frac{\omega\sqrt{A}}{a_0}, \qquad m_z = \frac{m \cdot a_0}{\rho_0 A}, \qquad D_z = \frac{D}{\sqrt{A}}$$

have been substituted, where

A – characteristic cross-section,

$a_0 = \sqrt{kRT_0}$, T_0 – total temperature (in the case of the compressor in the cross-section of the inlet, in the case of the turbine in the cross-section of the outlet),

k – izentropy index,

R – individual gas constant,

ε_s – compression ratio of the compressor,

σ – number of hydraulic losses,

$\mu = R/c_p$, c_p – average specific thermal capacity in the course of compression and expansion.

The relation (2) is illustrated in Figs. 7 and 8. The former one presents the minimum ratio K_n/δ_n, corresponding to the assumed limit peripheral speed $u = 500$ m/sec. Calculations have been carried out for different compression ratios and three values of the mass flux and the rotational speed n. An increase of the compression ratio reduces the value of K_n/δ_n and also decreases with the decreasing number of revolutions. For the radial stage K_n ought to be contained

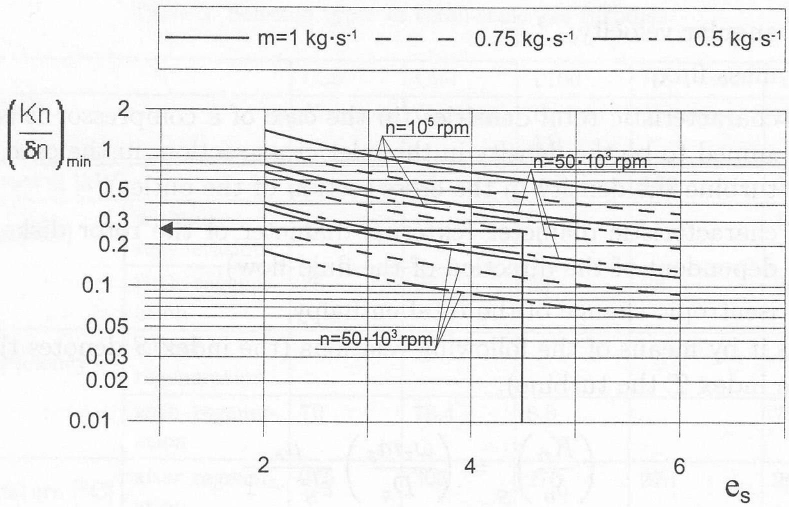


Figure 7. Change of $\left(\frac{K_n}{\delta_n}\right)_{\min}$ with ε_S and m and n.

within the range $0.8 \div 1.0$ [16]. Thus, the diagram permits to assess the maximum value of the diameter index. Similar information results from Fig. 8. In this case it indicates the possible value of one from among three reduced quantities m_z, ω_z, D_z, the two other data remaining unchanged. E.g. if $\varepsilon_S = 4$ and $n = 75 \cdot 10^3$ 1/min, the value of $\frac{\omega_z m_z}{D_z}$ will change within the range $0.32 \div 0.64$, while $0.5 \leq m \leq 1$ kg/sec is being changed.

From the conditions

$$m_T = m_S\,(1+\beta)\,, \qquad \omega_S = \omega_T$$

we obtain the relations

$$\left(\frac{\omega_z m_z}{D_z}\right)_T = \left(\frac{\omega_z m_z}{D_z}\right)_S (1+\beta)\,, \tag{4}$$

$$\left(\frac{K_n}{\delta_n}\right)_T = \frac{\mu_T\,(1+\beta)\left(\varepsilon_S^{\mu_s}-1\right)}{\mu_S\left(1-(\varepsilon_S\delta)^{-\mu_T}\right)}\left(\frac{K_n}{\delta_n}\right)_S . \tag{5}$$

Basing on (5) we may conclude that $\left(\frac{K_n}{\delta_n}\right)_T$ is always larger than $\left(\frac{K_n}{\delta_n}\right)_S$. If we assume that $K_{nT} \approx K_{nS}$, then $\delta_{nT} < \delta_{nS}$

The application of these relations is limited to adiabatic processes of compression and expansion. In design they may serve as the starting point for determination of the relations existing between the geometry and the parameters of operation of construction optimization.

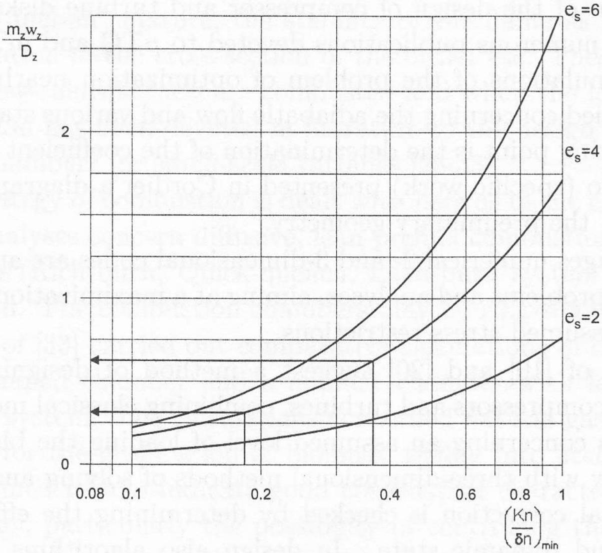


Figure 8. Change of $\frac{m_z \omega_z}{D_z}$ with ε_S and $\left(\frac{K_n}{\delta_n}\right)_{\min}$

In actual constructions we encounter restrictions, due to differences of the Reynolds numbers and to non-adiabatic processes of compression. If we assume that the dimensionless quantity $\frac{m_z \omega_z}{D_z}$ remains at the given compression ratio unchanged in small-scale and large-scale compressors, then (the index n concerns data for conventional turbomachinery)

$$\frac{\mathrm{Re}}{\mathrm{Re}_n} \sim \sqrt{\frac{m}{m_n}}.$$

The non-adiabatic flow is caused by the much higher ratio of the moistened surface to the mass flux in the case of μTG i mTG.

The results are:

- increased losses of mechanical energy,
- difficulties in attaining a high efficiency of compression,
- higher losses at the outlet.

Moreover, due to small dimensions, Euler's equation is not satisfied (the influence of the annular boundary layer). This problem, as well as the effect of the non-adiabatic flow have been dealt with in [17].

3.2 Aerodynamic optimization

The optimization of the design of compressor and turbine disks has been dealt with recently in numerous publications devoted to μTG and mTG (for instance [18-27]). In formulations of the problem of optimization nearly always the assumption is applied concerning the adiabatic flow and various stages of designing. Usually the starting point is the determination of the coefficient of efficiency and compression ratio (specific work) presented in Cordier'a diagram, as well as the determination of the preliminary geometry.

In further stages numerical 2- and 3-dimensional codes are applied for the solution of inverse problems and analyses, aiming at a maximization of the efficiency of the stage at assumed stress restrictions.

The authors of [10] and [20] suggest a method of designing single highly loaded stages of compressors and turbines, combining classical methods of solving inverse problems concerning an assumed level of loading the blading for a non-compressible flow with three-dimensional methods of solving analysed problems. Every geometrical correction is checked by determining the efficiency, stresses, displacements and dynamic state. In design also algorithms may be applied which are used to solve problems of designing any kind of fluid-flow machines [22-23].

The application of advanced procedures of aerodynamic optimization permits to attain for the adiabatic flow efficiencies of about 90 %, concerning both compressing and expanding flow chanels.

In investigations the peripheral speed is restricted to 500-550 m/s, number Ma_u amounting to 1.1-1.3.

3.3 Construction and calculations of combustion chambers

One of the most important problems concerning the construction of μTG and mTG ist the problem of combustion chambers permitting the combustion of fuel with some given composition or various kinds of fuels, without changing the criteria of reliability and disposability. Different requirements concerning those fuels render it difficult to reach a universal design of combustion chambers. In many cases this leads to individual designing of combustion chambers. The applied technological structures of μTG and mTG units (application of regeneration and low compression ratios) cause that in such systems the combustion chambers operate under different conditions from those existing in conventional large-scale systems.

In the case of the applied compression ratios regeneration leads to rather high temperatures ahead of the combustion chamber, which is of essential importance in the combustion process. It must also be taken into account that for the temperature ahead of the turbine the excess air factor exceeds considerably the values which are characteristic for conventional stationary turbines. This may affect the reactivity of the fuel-air mixture, the stability of the flame as well as the temperature distribution in the cross-section of the outlet etc. These problems may intensify when low-calorific fuels are combusted and when the load varies.

Much attention has been devoted in literature to the design and calculations of combustion chambers for small-scale turbines [28-33].

A similar strategy of combustion is dealt with here as in the case of large-scale installations. Analyses concern diffusive, lean premix combustion, combustion in the RQL system (Rich-burn, Quick-quench, Lean-burn system) as well as catalytic combustion. The combustion chambers may be ring-shaped or annular.

The authors of [33] carried out comparative calculations of diffusive combustion in a ring-shaped chamber and a annular chamber with lean premix, and also in the RQL system. The calculations concerned natural gas and fuel, which is characteristic for the gasification of biomass, taking into account the varying loads. The obtained results indicate good combustion characteristics with lean premix technology, particularly the possibility of controlling the NO_x emission. On the other hand, however, an increased emission of CO is to be observed, as well as an unfavourable temperature profile in the cross-section of the outlet.

As far as diffusive combustion in a ring-shaped chamber is concerned, low CO emissions could be attained in spite of worse characteristics of the generation of NO_x. The worst results occurred when RQL combustion was realized applying the adapted geometry of the combustion chamber, which does not mean that this way of combustion was ineffective, although attempts of its optimization ought to be made.

Numerical solutions of flows equations with combustion are an important source of information about physicochemical processes taking place in the combustion chambers. Of much interest are the aerodynamics of combustion chambers, the formation of mixtures (in the case of liquid fuels also processes of the atomization and evaporation of the fuel), the initiation and stability of the flame, phenomena occurring in the flame and on its periphery, including the emission of NO_x, CO and not oxidized hydrocarbons. Proposed models are constantly developed. At present, besides the application of commercial codes, more complex procedures are used, basing, for instance, on "flamelet" conceptions, associated with the LES methods, however, difficult to determine, which of the applied procedures of calculations are most adequate for different techniques of combustion.

3.4 Regeneration exchangers

Regeneration increases the thermodynamic competiveness of small gas turbines. The compression ratio can be reduced and the application of single-stage compressors and expanders is possible. The costs of a regenerator usually amount to 25-35 % of the whole gas turbine unit. For this reason investigations should tend towards the development of such a construction, which would not only warrant a high degree of regeneration, but also be characterized by low costs. This problem has been dealt with in literature in [34-39]. Complex constructions are being looked for, ensuring low hydraulic losses and high durability. Various constructions are nowadays applied, including classical stationary plate and ring-shaped solutions. Besides that also rotary regenerators are applied [38]. The final version of the design of the gas turbines set depends on the structural shape of the regenerator and its mutual configuration with the other modules of the turbine [40]. The costs of its production depend on R, the maximum temperature of preheating (influencing the kind of required material) and the complexity of the construction and assumed pressure losses. The authors of [37] suggest that the costs of constructing the regenerator considered by them would amount to about 100 USD/kW, if $R = 0.9$ and the temperature ahead of the turbine $t_3 = 900^oC$ (ε_s being equal to 4). If $t_3 = 800^oC$, the other data remaining unchanged, the production would cost about 125 USD/kW.

These differences are due to differences in temperature, at which the regenerator must operate. Figs. 9-12 illustrate the values of temperature behind the the air exchanger, corresponding to different compression ratios.

The temperature behind the regenerator is calculated by means of the relation (the denotation T_0 in Fig. 1a means the reference temperature)

$$\frac{T_{2a}}{T_0} = \varsigma_R = \varsigma_0 \varepsilon_S \mu_S \left(1 - \eta \xi_R\right) + \eta \xi_R \varsigma_T \left(\varepsilon_S \sigma\right)^{-\mu_T} \tag{6}$$

where:

$\varsigma_0 = \frac{T_1}{T_0}, \eta = \frac{w_{sp}}{w_p}, w_{sp}, w_p$ – mean thermal capacities in the regenerator on the side of combustion gases and on the side of air, respectively;

$\varsigma_T = \frac{T_3}{T_0}, \xi_R$ – thermal efficiency of the regenerator.

For a counter-flow exchanger we get:

$$\xi_R = \frac{1 - \exp\left(-\alpha\right)}{1 - \gamma \, \exp(-\alpha)} \tag{7}$$

where:
$\alpha = N\left(1-\gamma\right),\ \gamma = \frac{w_{\min}}{w_{\max}};\ N = \frac{kA}{w_{\min}},\ w_{\min} = \min\left\{w_p, w_{sp}\right\}, k$ – overall heat transfer coefficient; A – heat exchange surface.

The thermal efficiency ξ_R is connected with the degree of regeneration by the relation

$$\xi_R = \frac{1}{\eta} R \tag{8}$$

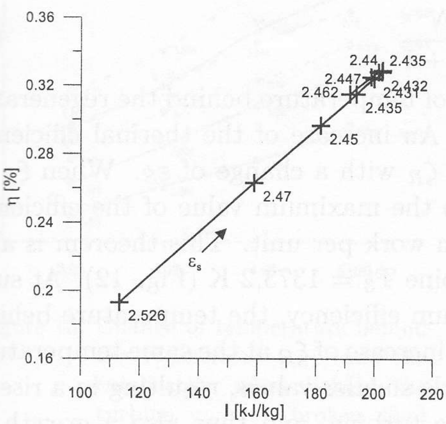


Figure 9. Characteristics of the gas turbine unit with $R = 0.5$, $t_3 = 900°C$. The values of $\zeta_R = T_{2a}/T_0$, $T_0 = 2732°C$. The values of ζ_R are marked in the diagram.

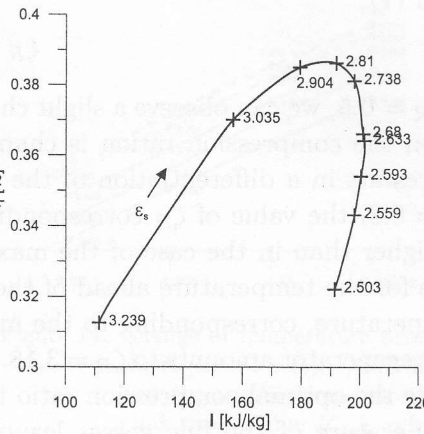


Figure 10. Characteristics of the gas turbine unit with $R = 0.8$, $t_3 = 900°C$. The values of ζ_R have been marked in the diagram.

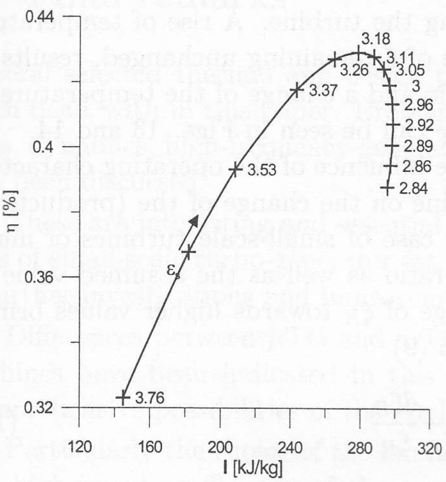


Figure 11. Characteristics of the gas turbine unit with $R = 0.8$, $t_3 = 1100°C$. The values of ζ_R have been marked in the diagram.

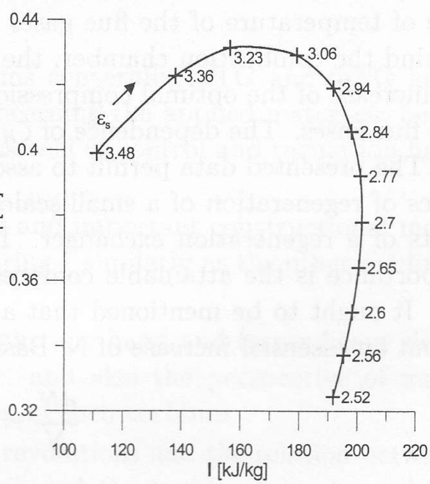


Figure 12. Characteristics of the gas turbine unit with $R = 0.9$, $t_3 = 900°C$. The values of ζ_R have been marked in the diagram.

Calculation of η for a semi-ideal gas model and in characteristic states of regeneration indicate that $w_p \approx w_{sp}$, i.e. $\gamma = 1$. Then $\zeta_R \approx$ R and, in compliance with (7)

$$\xi_R = \frac{N}{N+1} \tag{9}$$

If $\xi_R = 0.5$, we can observe a slight change of temperature behind the regenerator when the compression ration is changed. An increase of the thermal efficiency ξ_R results in a differentiation of the value ζ_R with a change of ε_S. When $\xi_R \approx$ R $= 0.8$, the value of ζ_R corresponding to the maximum value of the efficiency is higher than in the case of the maximum work per unit. This theorem is also true for the temperature ahead of the turbine $T_3 = 1373{,}2$ K (Fig. 12). At such temperature, corresponding to the maximum efficiency, the temperature behind the regenerator amounts to $\zeta_R = 3{,}18$. The increase of ξ_R at the same temperature shifts the optimal compression ratio towards smaller values, resulting in a rise of temperature of the flue gasses leaving the turbine, and thus also a growth of the value ζ_R. The production costs of the regenerator will then depend on the temperature of the combustion gases at the outlet of the turbine (the kind of the applied material), its surface (the values of A) and the applied technology. If the compression ratio warrants an optimal efficiency, the increase of ξ_R at a constant value of ζ_T will also increase the costs of constructing the regenerator due to the enlarged surface of heat exchange and the reduced value of ε_S, which leads to a rise of temperature of the flue gases leaving the turbine. A rise of temperature behind the combustion chamber, the value of ξ remaining unchanged, results in an increase of the optimal compression ratio and a change of the temperature of the flue gases. The dependence of ζ_R on ζ_T can be seen in Figs. 13 and 14.

The presented data permit to assess the influence of the operating characteristics of regeneration of a small-scale turbine on the change of the (production) costs of a regeneration exchanger. In the case of small-scale turbines of much importance is the attainable compression ratio as well as the assumed value of ξ_R. It ought to be mentioned that a change of ξ_R towards higher values brings about an essential increase of N. Basing on (9)

$$\frac{dN}{N} = \frac{1}{1-\xi_R} \frac{d\xi_R}{\xi_R} . \tag{10}$$

And thus, in the case of a change of about $\xi_{RO} = 0.8$

$$\frac{dN}{N} = 5 \frac{d\xi_R}{\xi_R}$$

and about $\xi_{R0} = 0.9$

$$\frac{dN}{N} = 10 \frac{d\xi_R}{\xi_R} .$$

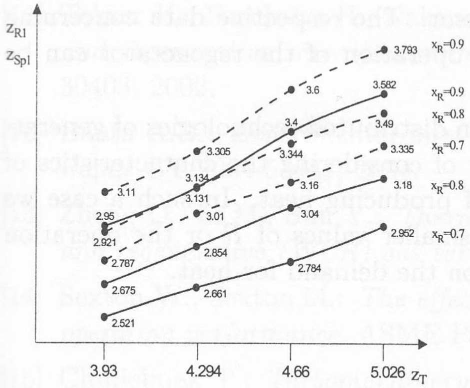


Figure 13. Change of temperature behind the regenerator (ς_{R1}) and the combustion gases behind the turbine ς_{sp1} – (broken line) with the value of ζ_T (T_0 has been assumed as equal to 273,2°) and the value of ξ_R.

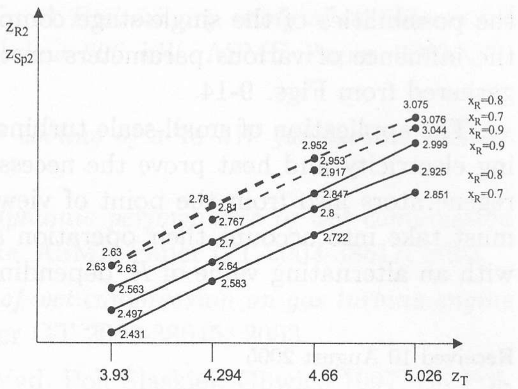


Figure 14. Change of temperature behind the regenerator (ζ_{R2} – full line) and the combustion gases behind the turbine (ζ_{Sp2} – broken line) with ζ_T and ξ_R. Calculations have been carried out for a compression ratio corresponding to the maximum work per unit.

4 Final remarks

Several selected thermal and design problems concerning μTG and mTG have been dealt with in this paper. Problems concerning the applied materials, bearings, dynamics, high-frequency generators, as well as control and regulation have not been discussed.

These are interesting and essential issues and important constructional modules of small-scale turbo-generator set, requiring – similarly as the other modules – further investigations and improvements.

Differences between μTG and mTG on the one hand and larger heavy duty turbines have been indicated in this paper, and also the perspective of more comprehensive possibilities of the application of such turbines.

Particularly the choice of the number of revolutions and the relation between the high-speed coefficients of the compressor and the turbine have been dealt with. An analysis of the operation of a regenerator in a small-scale turbine has shown that there are two main factors affecting the costs of these heat exchangers. The first one concerns the compression ratio, the other one the efficiency effectiveness of the regenerator. The choice of the compression ratio corresponding to the maximum efficiency results in higher temperatures of both media in the regenerator. Practically the choice of the compression ratio is determined by the possibilities of the single-stage compressor. The respective data concerning the influence of various parameters on the operation of the regenerator can be gathered from Figs. 9-14.

The application of small-scale turbines in distributed technologies of generating electricity and heat prove the necessity of considering the characteristics of regenerators also from the point of view of producing heat. In such a case we must take into account their operation at smaller values of R or the operation with an alternating value of R, depending on the demand for heat.

Received 10 August 2005

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