Mechanics and Mechanical Engineering
Vol. 15, No. 3 (2011) 207–216

Algopithm for Design Calculation of Axial Flow Gas Turbine Compressor – Comparison with GTD–350 Compressor Design

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Received (10 August 2011)
Revised (25 September 2011)
Accepted (28 September 2011)

The paper presents a simple algorithm for design calculation of axial flow compressor of gas turbine. The algorithm enables calculation of characteristic dimensions and gas angels in compressor stages using real gas model. Preliminary and detailed calculation base on data from GTD–350 gas turbine, also calculated and measured dimension of bleeds are compared.

Keywords: Axial compressor, gas turbine, design

1. Introduction

Continuous development of gas turbine technology is strongly connected with the aviation and power industry. The gas turbine cycles are usually designed for combustion of fossil fuels. However, it is also possible to replace the natural gas or petroleum with the renewable resources and biomass fuels [4]. The aim of many scientific researches in this area is to determine the optimal configuration of the thermal cycles with the external biomass combustion as well as development and tests of the gas turbine combustion chambers which can be adapted for the combustion of relatively low calorific value fuels and with low emission of pollutants [1,5].

The process of application of gas turbine based on biomass combustion is slow. The commercial gas turbine systems are not well available. Although, many small and large scale test gas turbine installations arise.

The power generation units based on gas turbine have many advantages which in particular applications could be important. In case of installations with gas turbines the main advantages are:

– high efficiency of combined gas–steam installations (gas turbine of medium and high power output);

– fuel flexibility;

– large quantity of high temperature heat;

– low emission;

– fast start–up process.

The electricity production based on fuel from biomass in micro gas turbine installations may involve necessity of design new concepts of thermal cycles due to different fuel to air ratio (especially in case of low calorific gases). The new design concepts have to take into account also different working conditions of gas turbine based on the alternative fuels dedicated for electricity generation. The different working conditions during the year have to include partial loads, season work or peak loads. Moreover, quality and parameters of the specified fuel as well as additional heat generation adjusted to the requirements may have a great influence for the whole gas turbine system. The application of gas turbine based on alternative fuels in the industry may be especially interesting. This technology may be applied in sugar factory or corn and biomass driers. In these cases the large consumption of electricity and heat is desirable which is the advantage of the gas turbine technology. The important feature of gas turbine technology is the ability of design the installation which can work with almost every fuel type without limitations due to the octane or methane number or fuel temperature. The new and innovative concepts of application of gas turbine technology require development of new algorithms which can offer the calculation of thermodynamic and economy characteristics. The algorithms should take into account proper selection of necessary machine and devices which allows direct connection between the design calculations and economy analysis.

Economic effectiveness is the main feature which decide about the implementation possibility of new energy technology. In order to obtain economic characteristic of systems with gas turbines the determination of compressor maps and main dimensions is necessary. In this paper the algorithm of the design process of the axial compressor for the gas turbine is presented. These program will be a part of a complex algorithm which covers the design and off design calculations of the main devices of the gas turbine installation. Applied non–dimensional model for preliminary calculations and 1-dimensional model for stage by stage calculations are so simple that it does not require the use of sophisticated computational techniques and at the same time allow to obtain the necessary data for further calculations of technical and economic characteristics.

Results of compressor design presented in this paper base on parameters of GTD–350 helicopter turbine.

2. Compressor Design Algorithm

2.1. *Preliminary calculations*

The preliminary calculations of the axial compressor allows to estimate inlet and outlet cross-section area and number of the compressor stages. The main input parameters are the air inlet temperature, pressure and mass flow, compression ratio, isentropic efficiency and rotational speed. Tab. 1 presents the assumed input data for the preliminary calculation process.

Table 1 Input data for preliminary calculations

Inlet total pressure	$p_{01},\ bar$
Inlet total temperature	T_{01}, K
Axial velocity	$c_a, m/s$
Inlet velocity radial component	$c_{w1}, m/s$
Compression ratio	p_{02}/p_{01}
Mass flow rate	$m_p,\ kg/s$
Isentropic efficiency	η_i
Limited circumferential velocity of the blade tip	$u_t, m/s$
Rotational speed	$n,\ rev/s$

The inlet pressure and temperature are assumed according to the ambient air parameters. The axial velocity for the axial compressor is in range 150–200m/s. This velocity is constant in compressor and for this reason this is one of most important assumption in design process using this algorithm. The assumed circumferential velocity of the blade tip is connected with the acceptable stress value in the blade or high Mach number which have a great influence for the flow losses. The impact of inlet guide vanes can be taking into account using inlet velocity radial component of air.

The preliminary calculation algorithm is based on a few main steps [2, 3]:

1. Inlet static temperature and pressure calculations.
2. Inlet cross–section calculations.
3. Determine the characteristic inlet radius dimension.
4. Estimate outlet temperature and pressure.
5. Calculate characteristic outlet radius.
6. Preliminary estimation of the number of stages.

Temperature rise in one stage of the compressor (constant mean radius) can be expressed as:

$$\Delta T_{0S} = \frac{\lambda \cdot u_m \cdot c_a \cdot (\tan\beta_1 - \tan\beta_2)}{c_p} \tag{1}$$

Outlet angle is calculated using de Haller number defined as:

$$H = \frac{w_2}{w_1} \tag{2}$$

where $w_{2,1}$ – relative stator air velocities, using the rule H>0.72.

The number of stages in the compressor was estimated as follows:

$$l_s = \frac{T_{02} - T_{01}}{\Delta T_{0S}} \tag{3}$$

Output and input dimensions were calculated using continuity equation.
To avoid excessive losses, Mach number at rotor tip should not exceed 1.1.
The results obtained In case of preliminary calculations are presented in Tab. 2.

Table 2 Preliminary calculation results

	Compr. inlet	Compr. outlet
Total temperature		T_{02}
Total pressure		p_{02}
Static temperatura	T_1	T_2
Static pressure	p_1	p_2
Density	ρ_1	ρ_2
Outer blade radius	r_{t1}	r_{t2}
Inner blade radius	r_{r1}	r_{r2}
Inner to outer blade radius ratio	r_{r1}/r_{t1}	r_{r2}/r_{t2}
Relative tip blade Mach number	M_{1t}	
Mean radius	r_m	
Circumferential velocity on mean radius	U_m	
Temperature rise for one stage	ΔT_{0S}	
Estimated number of stages	l_s	

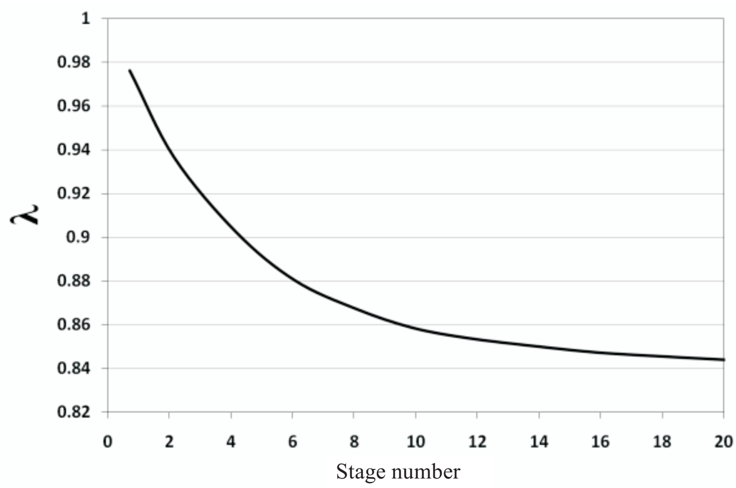

Figure 1 Work done factor in function of stage number

2.2. *Compressor stage design algorithm*

The presented algorithm is able to calculate the basic flow and geometry parameters in specified cross–sections of the compressor as well as the power output and velocity triangles of the individual stages are calculated. The input data are similar to those assumed in case of preliminary calculations. Additionally, temperature rise and reaction of the stages was assumed. The mean temperature rise and number of stages was estimated on the basis of preliminary calculations.

The work done factor λ which decrease the theoretical value of work carried out by fluid was taken into account. The work done factor is strictly connected with the non–uniform velocity profile in cross-section of the compressor. This parameter is a function of stage number (Fig. 1)[3]

The basic relations applied in the presented algorithm include calculations of:

– circumferential velocity component rise

$$\Delta c_w = \frac{c_p \cdot \Delta T_{0S}}{\lambda \cdot u_m} \tag{4}$$

– circumferential velocity component c_{w2}calculation:

$$c_{w2} = c_{w1} + \Delta c_w \tag{5}$$

– β_1 angle:

$$\beta_1 = \arctan\left(\frac{u_m - c_{w1}}{c_a}\right) \tag{6}$$

– β_2 angle:

$$\beta_2 = \arctan\left(\frac{u_m - c_{w2}}{c_a}\right) \tag{7}$$

– α_1 angle:

$$\alpha_1 = \arctan\left(\frac{c_a}{c_{w1}}\right) \tag{8}$$

– α_2 angle:

$$\alpha_2 = \arctan\left(\frac{c_{w2}}{c_a}\right) \tag{9}$$

– De Hallera number calculation for the rotor blade of the first stage

$$H_w = \frac{w_2}{w_1} = \frac{\frac{c_a}{\cos\beta_2}}{\frac{c_a}{\cos\beta_1}} = \frac{\cos\beta_1}{\cos\beta_2} \tag{10}$$

Total pressure and temperature in the first stage outlet was calculated as follows:

$$p_{03} = p_{01} \cdot \left(1 + \frac{\eta_i \cdot \Delta T_{0S}}{T_{01}}\right)^{\frac{\kappa}{\kappa-1}} \tag{11}$$

$$T_{03} = T_{01} + \Delta T_{0S} \tag{12}$$

Stage power output:

$$N_s = m_p \cdot u_m \cdot \Delta c_w \tag{13}$$

Calculations of next stages were differ in case of β_1 and β_2 angles:

$$\beta_1 = \arctan\left(\frac{\Delta\beta_{1-2} + \Delta\beta_{1-2}}{2}\right) \tag{14}$$

$$\beta_2 = \arctan\left(\frac{\Delta\beta_{1-2} - \Delta\beta_{1-2}}{2}\right) \tag{15}$$

$$\Delta\beta_{1-2} = \tan\beta_1 - \tan\beta_2 = \frac{\Delta T_{0S} \cdot c_p}{\lambda \cdot u_m \cdot c_a} \tag{16}$$

$$\Delta\beta_{1+2} = \tan\beta_1 + \tan\beta_2 = \frac{2 \cdot \Lambda \cdot u_m}{c_a} \tag{17}$$

Additinally α_1 and α_2 angles are calculated as follow:

$$\alpha_1 = \arctan\left(\frac{u_m}{c_a} - \tan\beta_1\right) \tag{18}$$

$$\alpha_2 = \arctan\left(\frac{u_m}{c_a} - \tan\beta_2\right) \tag{19}$$

Deflection of the rotor blades:

$$D = \beta_1 - \beta_2 \tag{20}$$

De Hallera number for the rotor blades of n stage:

$$H_w(n) = \frac{w_2(n)}{w_1(n)} = \frac{\frac{c_a}{\cos\beta_2(n)}}{\frac{c_a}{\cos\beta_1(n)}} = \frac{\cos\beta_1(n)}{\cos\beta_2(n)} \tag{21}$$

De Hallera number for the stator blades of $n-1$ stage:

$$H_k(n-1) = \frac{c_3(n-1)}{c_2(n-1)} = \frac{\frac{c_a}{\cos\alpha_3(n-1)}}{\frac{c_a}{\cos\alpha_2(n-1)}} = \frac{\cos\alpha_2(n-1)}{\cos\alpha_3(n-1)} = \frac{\cos\alpha_2(n-1)}{\cos\alpha_1(n)} \tag{22}$$

The n–1 stage outlet angle is equal to n stage inlet angle. Additionally, inlet total pressure p01 and total temperature T01 to the n stage is equal to the outlet total pressure p03 and temperature T03 in case of n-1 stage.

Total pressure and temperature for the outlet section of n stage is expressed as follow:

$$p_{03} = p_{01} \cdot \left(1 + \frac{\eta_i \cdot \Delta T_{0S}}{T_{01}}\right)^{\frac{\kappa}{\kappa-1}} \tag{23}$$

$$T_{03} = T_{01} + \Delta T_{0S} \tag{24}$$

For calculation of compressor with constant outer diameter, for each stage different values of average circumferential velocity were assumed.

3. Validation of the compressor design algorithm

The assessment of the presented algorithm was done on the basis of GTD-350 helicopter gas turbine design data. The selected gas turbine is a part of a test rig in the Institute of Power Engineering and Turbomachinery of Silesian University of Technology (Fig. 2). The turbine is connected with the eddy current brake. It allows to investigate the influence of different turbine loads for the main gas turbine parameters. The test rig is supplied by the aviation kerosene. Parameters of gas turbine at starting conditions were the source of data for preliminary design calculation (Tab. 3). Results of preliminary calculation are shown in Tab. 4.

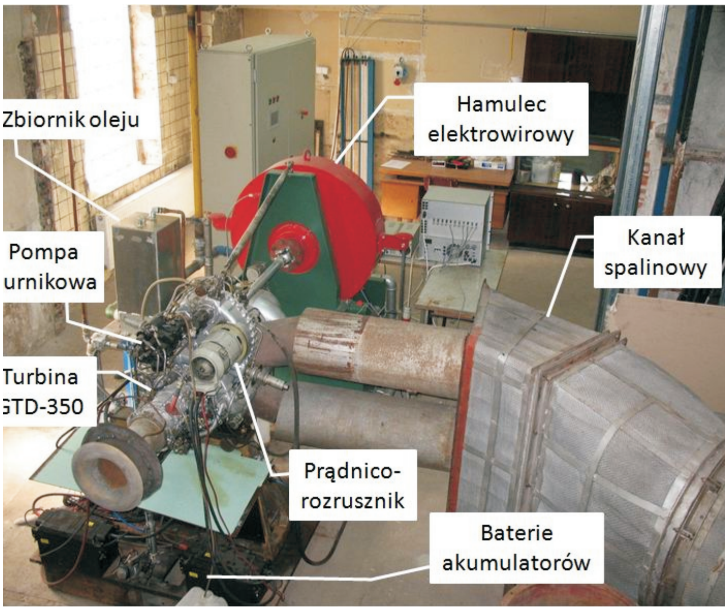

Figure 2 Fotography of GTD–350 gas turbine test rig

Table 3 The main input data assumed for the calculations

Inlet total pressure	p_{01}	1.01	bar
Inlet total temperature	T_{01}	288.00	K
Axial velocity – constant for all stages	c_a	151.00	m/s
Outlet radial velocity	c_{w1}	20.00	m/s
Pressure ratio		5.60	-
Inlet air mass flow	m_p	2.19	kg/s
Isentropic efficiency	$?_{is}$	0.85	-
Limited circumferential velocity of the blade tip	Ut	305.2	m/s
Rotational speed	n	720.00	obr/s

Results of preliminary design are shown on Tab. 4

Table 4 The main preliminary design results

	Compressor inlet		Compressor outlet	
Total tempeature K	T_{01}	288.0	T_{02}	513.91
Total pressure, bar	p_{01}	1.010	p_{02}	5.656
Static temperature	T_1	287.8	T_2	502.56
Static pressure	p_1	1.008	p_2	5.231
Density	ρ_x	1.220	ρ_2	3.627
Tip bled radius	r_{t1}	0.0675	r_{t2}	0.0675
Hub blead radius	r_{r1}	0.0277	r_{r2}	0.0572
Hub tip radius ratio	r_{r1}/r_{t1}	0.410	r_{r2}/r_{t2}	0.849
Blead tip reative Mach number	M_{1t}	0.949		
Mean diameter	r_{m1}	0.0476	r_{m2}	0.0623
Bled Speed at mean diameter	u_{m1}	215.23	u_{m2}	276.54
Estimated stage total temperature increase	ΔT_{0S}	19.57		
Estimated number of stages	ls	12		

For detailed calculation it was assumed that compressor has 11 stages and static temperature increase is equal in each stage. Reaction for first two stages was lower then 0.5. Selected results for first, seventh and eleventh stag are shown in Tab. 5.

Table 5 The main design results for selected stages

Stage number	1	7	11
Total inlet pressure, bar	1.010	2.988	5.186
Total inlet pressure, bar	1.247	3.463	5.656
Total inlet tempeature, K	288.0	414.0	498.0
Total outlet tempeature, K	309.00	435.00	512.71
β_1, deg	51.81	44.43	41.41
β_2, deg	32.98	16.22	21.26
α_1, deg	0.00	16.22	21.26
α_2, deg	31.89	44.43	41.41
Rotor de Haller number	0.74	0.74	0.80
Stator de Haller number	0.89	0.74	
Work coeficient λ	0.97	0.87	0.86
Deflection, deg	18.824	28.203	20.159
Power, kW	47.719	52.866	37.823

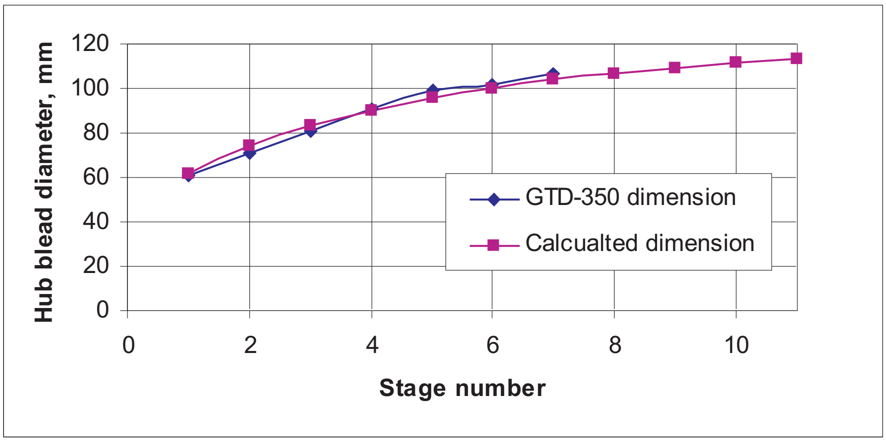

Figure 3 Comparision of calculated and mesuered bled hub diameter

In the next step length of rotor bleeds were compared. This results are presented in Fig. 3. It can be seen that calculated and measured values are quite similar. This comparison is made only for first 7 stages because compressor from GTD–350 consist with seventh stages of axial compressor and on stag of centrifugal compressor.

4. Conclusions

Presented algorithm allows calculation of main axial flow compressors parameters. Crucial advantages of presented algorithm is the possibility of calculations of relatives Mach number at blade tip and de Haller number in stators and rotors. Results obtained for modelling, based on GTD–350 parameters, corresponds with reference data of axial velocity and relative Mach number at bleat tip. Results of blades high calculations are comparable with GTD-350 blades high.

Acknowledgements The results presented in this paper were obtained from research work co–financed by the National Centre of Research and Development and ENERGA SA in the Strategic Research Programme – "Advanced technologies for energy generation" – "Development of integrated production technology of fuels and energy from biomass, agricultural waste and others".

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