PERIODICA POLYTECHNICA SER. TRANSP. ENG. VOL. 29, NO. 1-2, PP. 117–135 (2001)

NONLINEAR MODEL-BUILDING OF A LOW-POWER GAS TURBINE

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Received: October 1, 2001

Abstract

The development of a nonlinear gas turbine model is presented in this paper for loop-shaping control purposes. The nonlinear dynamic equations of the gas turbine are based on the first engineering principles. In order to complete the model, constitutive algebraic equations are also needed. These equations describe the static behavior of the gas turbine at various operating points. The complete, substituted nonlinear model is presented along with its model verification results based on a simulator and measured data.

Keywords: nonlinear models, simulation, energy balance, gas turbines.

1. Introduction

Gas turbines are important and widely used prime movers in transportation systems, such as aircraft, cars. Besides of this area gas turbines can be found in power systems where they are the main power generators [7].

The investigation of steady-state behavior of gas turbines in terms of static gas turbine characteristics is a traditional area in engineering [4]. This kind of models is based upon the characteristics of the component parts of the engine [5], [3].

The static characteristics can be given in the form of polynomials reflecting the results of the preliminary calculations or the measurements. They can be described by several methods as can be seen in [6].

The steady-state detailed model of a gas turbine at a given operating point helps to understand the dynamic behavior of a gas turbine. It also indicates that a nonlinear dynamic model is needed if we want to capture the dynamic behavior of a gas turbine.

This nonlinear dynamic model can then be effectively used to develop non-linear controller to improve the dynamic response of aircraft engines [6].

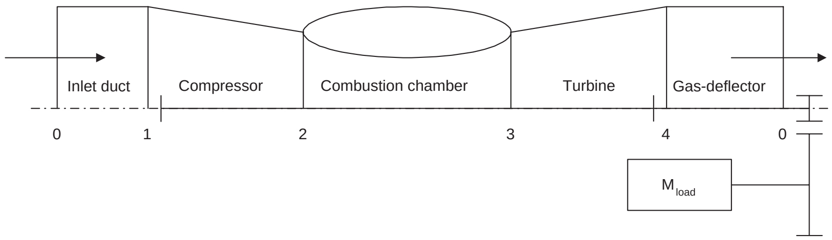

Fig. 1. The main parts of the gas turbine

2. Nonlinear State-Space Model of the Gas Turbine

2.1. The Modelling Problem

We develop a dynamic simplified model of the gas turbine for control purposes. The modelling problem statement then consists of the description of the system and that of the control aim.

System description

Gas turbines are one of the prime movers. The aim of using a gas turbine can be twofold: either generating thrust force for an aircraft, or generating power for any other machine, for example a generator, propeller and so on.

The main parts of a gas turbine include the inlet duct, compressor, combustion chamber, turbine and nozzle or gas-deflector. The interactions between these components are fixed by the physical structure of the engine.

The operation of these two types of gas turbines is basically the same. The air is drawn into the engine through the inlet duct by the compressor, which compresses it and then delivers it to the combustion chamber. Within the combustion chamber the air is mixed with fuel and the mixture ignited, producing a rise in temperature and hence an expansion of the gases. These gases are exhausted through the engine nozzle or the engine gas-deflector, but first they pass through the turbine, which is designed to extract sufficient energy from them to keep the compressor rotating, so that the engine is self-sustaining. The main parts of the gas turbine are shown schematically in *Fig. 1*.

In this paper we analyze a low-power gas turbine, which is installed in the Budapest University of Technology and Economics, Department of Aircraft and Ships on a test-stand.

Control aim

The following control aims are set for our low-power gas turbine:

- The number of revolutions has to follow the position of the throttle and should not be affected by the load and the ambient circumstances (the disturbance vector).
- The temperatures (basically the total temperature after the turbine) and the number of revolutions has to be limited, their values are constrained from above by their maximum values.

2.2. Modelling Assumptions

In order to get a low order dynamic model suitable for control purposes simplifying modelling assumptions should be made.

General assumptions

(a) Constant physico-chemical properties are assumed in each main part of the gas turbine, such as specific heat at constant pressure and at constant volume, specific gas constant and adiabatic exponent.
(b) Heat loss (heat transmission, heat conduction, heat radiation) is neglected.

Other assumptions

(c) In the inlet duct a constant pressure loss coefficient (σ_I) is assumed. It means that the total pressure loss in the inlet duct is a fixed percentage of its inlet total pressure (p_0^*).
(d) In the compressor:
  - the mass flow rate is constant: $\dot{m}_{C\text{in}} = \dot{m}_{C\text{out}} = \dot{m}_C$,
  - there is no energy storage effect: $U_2^* =$ constant.

(e) In the combustion chamber:
  - constant pressure loss coefficient (σ_{Comb}) is assumed,
  - constant efficiency of combustion (η_{comb}) is assumed,
  - the enthalpy of fuel is neglected,
  - and the combustion chamber is assumed to be a perfectly stirred region (balance volume). It means that a finite dimensional concentrated parameter model is developed and the value of the variables within this balance volume is equal to that at its outlet.

(f) In the turbine:
  - the mass flow rate is constant: $\dot{m}_{T\text{in}} = \dot{m}_{T\text{out}} = \dot{m}_T$,
  - there is no energy storage effect: $U_4^* =$ constant.

(g) In the gas-deflector a constant pressure loss coefficient (σ_N) is assumed.

2.3. The Development of the Model Equations

The nonlinear state equations are derived from the laws of conservation principles. Dynamic equations come from the conservation balances constructed for the overall mass m and internal energy U [1]. The notation list and the units of the parameters are given separately in the Appendix. The development of the model equations is performed in the following steps.

1. Conservation balance of the total mass:

$$\frac{\mathrm{d}m}{\mathrm{d}t} = \dot{m}_{\mathrm{in}} - \dot{m}_{\mathrm{out}}. \tag{1}$$

2. Conservation balance of the internal energy, where the heat energy flows and the work terms are also taken into account:

$$\frac{\mathrm{d}U^*}{\mathrm{d}t} = \dot{m}_{\mathrm{in}} i^*_{\mathrm{in}} - \dot{m}_{\mathrm{out}} i^*_{\mathrm{out}} + Q + W. \tag{2}$$

We can transform the above energy conservation equation by considering the dependence of the internal energy on the measurable temperature:

$$\frac{\mathrm{d}U^*}{\mathrm{d}t} = c_v \frac{\mathrm{d}}{\mathrm{d}t}\left(T^* m\right) = c_v T^* \frac{\mathrm{d}m}{\mathrm{d}t} + c_v m \frac{\mathrm{d}T^*}{\mathrm{d}t}. \tag{3}$$

From the two equations above we get a state equation for the temperature as state variable:

$$\frac{\mathrm{d}T^*}{\mathrm{d}t} = \frac{\dot{m}_{\mathrm{in}} i^*_{\mathrm{in}} - \dot{m}_{\mathrm{out}} i^*_{\mathrm{out}} + Q + W - c_v T^* \left(\dot{m}_{\mathrm{in}} - \dot{m}_{\mathrm{out}}\right)}{c_v m}. \tag{4}$$

The ideal gas equation ($p^* V = m R T^*$) is used together with two balance equations above to develop an alternative state equation for the pressure:

$$\frac{\mathrm{d}p^*}{\mathrm{d}t} = \frac{p^*}{m}\left(\dot{m}_{\mathrm{in}} - \dot{m}_{\mathrm{out}}\right) + \frac{p^*}{T^*}\left(\frac{\dot{m}_{\mathrm{in}} i^*_{\mathrm{in}} - \dot{m}_{\mathrm{out}} i^*_{\mathrm{out}} + Q + W - c_v T^* \left(\dot{m}_{\mathrm{in}} - \dot{m}_{\mathrm{out}}\right)}{c_v m}\right). \tag{5}$$

3. Conservation balance of the mechanical energy of the compressor-turbine shaft:

$$\frac{\mathrm{d}E_{\mathrm{shaft}}}{\mathrm{d}t} = \dot{m}_T c_{p\mathrm{gas}} \left(T^*_3 - T^*_4\right) \eta_{\mathrm{mech}} - \dot{m}_C c_{\mathrm{pair}} \left(T^*_2 - T^*_1\right) - 2\Pi \frac{3}{50} n M_{\mathrm{load}}. \tag{6}$$

2.4. Model Equations in Intensive Variable Form

These dynamic equations have to be transformed into the intensive variable form to contain measurable quantities. Therefore the set of transformed differential balances include the dynamic mass balance for the combustion chamber, the pressure form of the state equation derived from the internal energy balance for the combustion chamber and the intensive form of the overall mechanical energy balance expressed for the number of revolutions n.

$$\frac{\mathrm{d}m_{\mathrm{Comb}}}{\mathrm{d}t} = \dot{m}_C + \dot{m}_{\mathrm{fuel}} - \dot{m}_T, \tag{7}$$

$$\frac{\mathrm{d}p_3^*}{\mathrm{d}t} = \frac{p_3^*}{m_{\mathrm{Comb}}}(\dot{m}_C + \dot{m}_{\mathrm{fuel}} - \dot{m}_T) + \frac{p_3^*}{T_3^* c_{v\mathrm{med}} m_{\mathrm{Comb}}}(\dot{m}_C c_{\mathrm{pair}} T_2^* - \dot{m}_T c_{p\mathrm{gas}} T_3^* + Q_f \eta_{\mathrm{comb}} \dot{m}_{\mathrm{fuel}} - c_{v\mathrm{med}} T_3^* (\dot{m}_C + \dot{m}_{\mathrm{fuel}} - \dot{m}_T)), \tag{8}$$

$$\frac{\mathrm{d}n}{\mathrm{d}t} = \frac{1}{4\Pi^2 \Theta n}\left(\dot{m}_T c_{p\mathrm{gas}}\left(T_3^* - T_4^*\right)\eta_{\mathrm{mech}} - \dot{m}_C c_{\mathrm{pair}}\left(T_2^* - T_1^*\right) - 2\Pi \frac{3}{50} n M_{\mathrm{load}}\right). \tag{9}$$

2.5. Constitutive Relations

Some constitutive equations are needed to complete the nonlinear gas turbine model.

1. Two equations come from the modelling assumptions ((c),(e.1),(g)) for the total pressures after the compressor and the turbine:

$$p_2^* = \frac{p_3^*}{\sigma_{\mathrm{Comb}}}, \tag{10}$$

$$p_4^* = \frac{p_1^*}{\sigma_I \sigma_N}. \tag{11}$$

2. The second is the ideal gas equation which is used for the combustion chamber:

$$T_3^* = \frac{p_3^* V_{\mathrm{Comb}}}{m_{\mathrm{Comb}} R_{\mathrm{med}}}. \tag{12}$$

3. The third type of constitutive equations describes the total temperature after the compressor (T_2^*), and the total temperature after the turbine (T_4^*).

- The total temperature after the compressor is found by using the isentropic efficiency η_C in the following manner:

$$T_2^* = T_1^* \left(1 + \frac{1}{\eta_C}\left(\left(\frac{p_2^*}{p_1^*}\right)^{\frac{\kappa_{\text{air}}-1}{\kappa_{\text{air}}}} - 1\right)\right). \tag{13}$$

- The total temperature after the turbine is found similarly by using the isentropic efficiency η_T:

$$T_4^* = T_3^* \left(1 - \eta_T\left(1 - \left(\frac{p_4^*}{p_3^*}\right)^{\frac{\kappa_{\text{gas}}-1}{\kappa_{\text{gas}}}}\right)\right). \tag{14}$$

4. The fourth type of constitutive equations describes the mass flow rate and the isentropic efficiency of the compressor and the turbine.

$$\dot{m}_C = \text{const (1) } q(\lambda_1)\frac{p_1^*}{\sqrt{T_1^*}}, \tag{15}$$

$$\dot{m}_T = \text{const (2) } q(\lambda_3)\frac{p_3^*}{\sqrt{T_3^*}}. \tag{16}$$

In the equations above $q(\lambda_1)$ and $q(\lambda_3)$ (the dimensionless mass flow rate of the compressor and the turbine) can be calculated as follows:

$$q(\lambda_1) = f_1\left(\frac{n}{\sqrt{\frac{T_1^*}{288.15}}}, \frac{p_2^*}{p_1^*}\right), \tag{17}$$

$$q(\lambda_3) = f_2\left(\text{const (3)}\frac{n}{\sqrt{T_3^*}}, \frac{p_3^*}{p_4^*}\right). \tag{18}$$

In the equations above $q(\lambda_1)$ is the function of the corrected number of revolutions and the compressor pressure ratio, and $q(\lambda_3)$ is the function of the dimensionless velocity and the turbine pressure ratio.

The equations of the isentropic efficiencies of the compressor and the turbine can be calculated as follows:

$$\eta_C = g_1\left(\frac{n}{\sqrt{\frac{T_1^*}{288.15}}}, q(\lambda_1)\right), \tag{19}$$

$$\eta_T = g_2\left(\text{const (3)}\frac{n}{\sqrt{T_3^*}}, \frac{p_3^*}{p_4^*}\right). \tag{20}$$

The parameters, the constants of these functions can be determined with the help of the results of the measurements, the compressor and turbine characteristics.
For these characteristics we have *two types of approximations.*
The first approximation:
The dimensionless mass flow rate of the compressor is the linear function of the corrected number of revolutions and linear function of the compressor pressure ratio:

$$q(\lambda_1) = a_1 \frac{n}{\sqrt{\frac{T_1^*}{288.15}}} \frac{p_2^*}{p_1^*} + a_2 \frac{n}{\sqrt{\frac{T_1^*}{288.15}}} + a_3 \frac{p_2^*}{p_1^*} + a_4. \qquad (21)$$

The isentropic efficiency of the compressor is the linear function of the corrected number of revolutions and linear function of the dimensionless mass flow rate of the compressor:

$$\eta_K = b_1 \frac{n}{\sqrt{\frac{T_1^*}{288.15}}} q(\lambda_1) + b_2 \frac{n}{\sqrt{\frac{T_1^*}{288.15}}} + b_3\, q(\lambda_1) + b_4. \qquad (22)$$

The dimensionless mass flow rate of the turbine is the linear function of the dimensionless velocity and linear function of the turbine pressure ratio:

$$q(\lambda_3) = c_1 \, (\text{const}\,(3)\frac{n}{\sqrt{T_3^*}})\, \frac{p_3^*}{p_4^*} + c_2 \, (\text{const}(3)\frac{n}{\sqrt{T_3^*}}) + c_3 \frac{p_3^*}{p_4^*} + c_4. \quad (23)$$

The isentropic efficiency of the turbine is the linear function of the dimensionless velocity and linear function of the turbine pressure ratio:

$$\eta_T = d_1 \, (\text{const}\,(3)\frac{n}{\sqrt{T_3^*}})\, \frac{p_3^*}{p_4^*} + d_2 \, (\text{const}\,(3)\frac{n}{\sqrt{T_3^*}}) + d_3 \frac{p_3^*}{p_4^*} + d_4. \quad (24)$$

The second approximation is:
The dimensionless mass flow rate of the compressor is the linear function of the corrected number of revolutions and parabolic function of the compressor pressure ratio:

$$q(\lambda_1) = a_1 \frac{n}{\sqrt{\frac{T_1^*}{288.15}}} (\frac{p_2^*}{p_1^*})^2 + a_2 \, (\frac{p_2^*}{p_1^*})^2 + a_3 \frac{n}{\sqrt{\frac{T_1^*}{288.15}}} \frac{p_2^*}{p_1^*}$$
$$+ a_4 \frac{p_2^*}{p_1^*} + a_5 \frac{n}{\sqrt{\frac{T_1^*}{288.15}}} + a_6. \qquad (25)$$

The isentropic efficiency of the compressor is the linear function of the corrected number of revolutions and parabolic function of the dimensionless mass flow rate of the compressor:

$$\eta_K = b_1 \frac{n}{\sqrt{\frac{T_1^*}{288.15}}} (q(\lambda_1))^2 + b_2 (q(\lambda_1))^2 + b_3 \frac{n}{\sqrt{\frac{T_1^*}{288.15}}} q(\lambda_1) + b_4 q(\lambda_1) + b_5 \frac{n}{\sqrt{\frac{T_1^*}{288.15}}} + b_6. \tag{26}$$

The dimensionless mass flow rate of the turbine is the linear function of the dimensionless velocity and parabolic function of the turbine pressure ratio.

$$q(\lambda_3) = c_1 (\text{const } (3)\frac{n}{\sqrt{T_3^*}}) (\frac{p_3^*}{p_4^*})^2 + c_2 (\frac{p_3^*}{p_4^*})^2 + c_3 (\text{const } (3)\frac{n}{\sqrt{T_3^*}}) \frac{p_3^*}{p_4^*} + c_4 \frac{p_3^*}{p_4^*} + c_5 (\text{const } (3)\frac{n}{\sqrt{T_3^*}}) + c_6. \tag{27}$$

The isentropic efficiency of the turbine is the linear function of the dimensionless velocity and parabolic function of the turbine pressure ratio:

$$\eta_T = d_1 (\text{const } (3)\frac{n}{\sqrt{T_3^*}}) (\frac{p_3^*}{p_4^*})^2 + d_2 (\frac{p_3^*}{p_4^*})^2 + d_3 (\text{const } (3)\frac{n}{\sqrt{T_3^*}}) \frac{p_3^*}{p_4^*} + d_4 \frac{p_3^*}{p_4^*} + d_5 (\text{const } (3)\frac{n}{\sqrt{T_3^*}}) + d_6. \tag{28}$$

The constants of these functions (a_i b_i c_i d_i) can be calculated with the help of the least squares method. For the comparison of the two types of approximations the minimum distance-norm of the least squares method is capable.

Table 1. The constants of the first approximation and its minimum distance-norm

i	a_i	b_i	c_i	d_i
1	0.00035319	−0.00059576	−0.03248	0.144
2	0.0011097	0.00028848	0.0018218	0.0021314
3	−0.4611	0.5265	0.047843	−0.19685
4	0.16635	0.42051	0.16026	1.07
norm	0.026003	0.054047	0.01534	0.11549

Table 2. The constants of the second approximations and its minimum distance-norm

i	a_i	b_i	c_i	d_i
1	0.00026257	0.0028255	0.50085	−0.59478
2	−0.22273	−3.9674	−0.3687	0.32512
3	−0.00076179	−0.0014457	−2.0632	2.9127
4	0.49936	3.052	1.5469	−1.7322
5	0.0022535	0.0000014215	2.0335	−3.1926
6	−0.84134	0.15366	−1.345	2.8633
norm	0.025632	0.053865	0.013467	0.11448

As it can be seen in the last rows of *Table 1* and *Table 2*, there is no important difference between the minimum distance-norms of the two types of approximations. For the sake of simplicity of the analysis and control design of the nonlinear model in the following we will use the first approximation of the compressor and the turbine characteristics. (The original measured points of the compressor and turbine characteristics and the approximated points of the two types of approximations of the characteristics can be seen in the Appendix.)

It is an interesting and important property of the nonlinear differential and algebraic equations that all constitutive relations can be substituted into the dynamic equations [8]. So the resulted model consists of 3 independent dynamic equations, therefore the gas turbine can be described by only 3 state variables.

2.6. *Range of Operation of the Nonlinear Model*

On the gas turbine test-stand the following operation domain can be investigated experimentally for the gas turbine expressed in terms of the *measurable intensive set of state variables*:

$$\overline{x} = \left[\, m_{\text{Comb}} \;\; p_3^* \;\; n \, \right]^T, \tag{29}$$

$$0.0021 \le m_{\text{Comb}} \le 0.011, \quad 101334 \le p_3^* \le 357894, \quad 650 \le n \le 833.33.$$

The value of the only *input variable* $\dot{m}_{\text{fuel}}$ is also constrained by:

$$0.00367 \le \dot{m}_{\text{fuel}} \le 0.027.$$

The set of possible *disturbances* includes:

$$d = \left[\, p_1^* \;\; T_1^* \;\; M_{\text{load}} \, \right]^T, \tag{30}$$

$$60000 \le p_1^* \le 110000, \quad 243.15 \le T_1^* \le 308.15, \quad 0 \le M_{\text{load}} \le 363.$$

Finally we construct the *set of output variables* by noticing that the pressure p_3^* and the number of revolutions n in the state vector above can be measured but the mass m_{Comb} cannot:

$$y = \left[\; T_4^* \;\; p_3^* \;\; n \; \right]. \tag{31}$$

3. Model Verification

The verification of the developed model is performed by extensive simulation experiments using the MATLAB/SIMULINK model of the gas turbine against engineering intuition and operation experience on the qualitative and order of magnitude behavior of the system.

3.1. Simulation Results

The simulation investigations were carried out using the computed parameters of the gas turbine based on primary engineering data and the results of the static characteristics.

A typical operating point has been selected

$$\overline{x}^* = [\; 0.0043 \;\; 208270 \;\; 730.7095 \;]^T \; . \tag{32}$$

The value of the disturbance variables has been kept constant

$$d^* = [\; 305.45 \;\; 98711 \;\; 99.2 \;]^T \; . \tag{33}$$

A step of magnitude $\Delta u = 0.00119$ has been added to the reference value of the fuel flow rate $u^* = 0.0119$.

The resulted dynamic step response (in number of revolutions, in the total pressures: (p_2^* and p_3^*) and in the total temperatures: (T_2^*, T_3^* and T_4^*)) is shown in the Appendix. The responses live up to the engineering expectations: the number of revolutions, the total pressures and the total temperature after the compressor are increasing, but according to the characteristics the total temperatures before and after the turbine are decreasing. The figures also indicate that the gas turbine is stable at this operating point.

4. Conclusion and Future Work

The nonlinear state space model of the gas turbine based on first engineering principles is presented in this paper. A simulator of the gas turbine model in MATLAB/SIMULINK has been prepared. The model is verified against engineering intuition and qualitative operation experience.

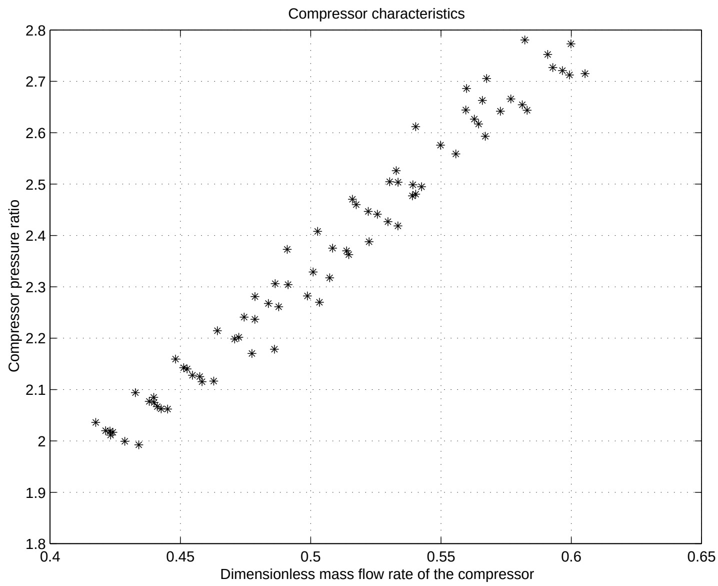

Fig. 2. Dimensionless mass flow rate of the compressor (measured points)

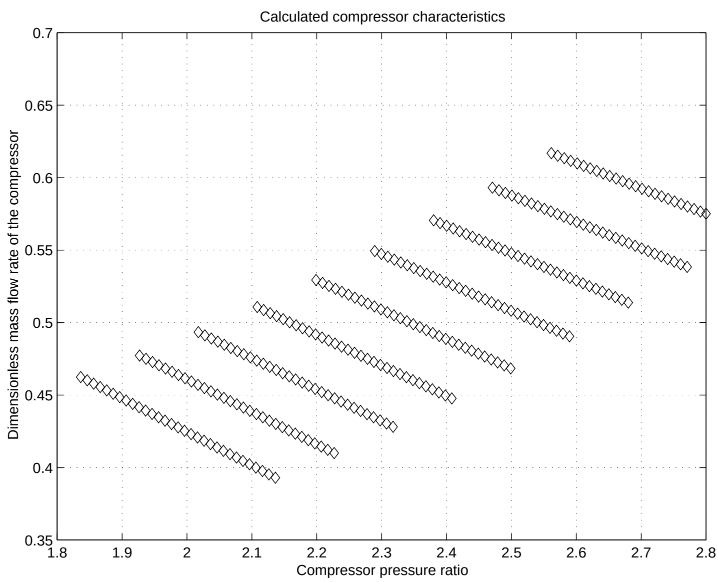

Fig. 3. Dimensionless mass flow rate of the compressor (1. approximation)

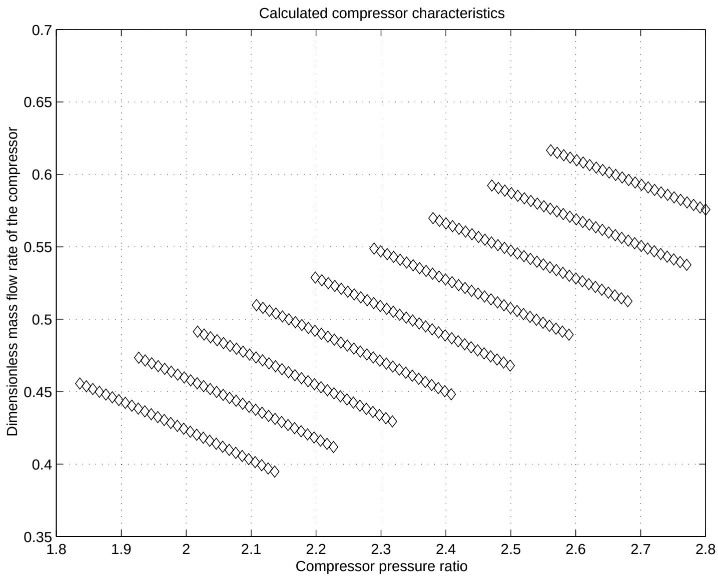

Fig. 4. Dimensionless mass flow rate of the compressor (2. approximation)))

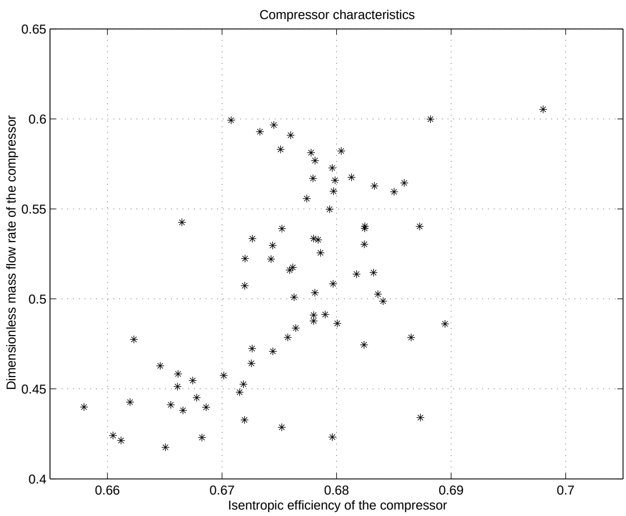

Fig. 5. Isentropic efficiency of the compressor (measured points)

With the help of this nonlinear dynamic model we can analyze the open-loop properties of the gas turbine, such as reachability, observability and stability of the

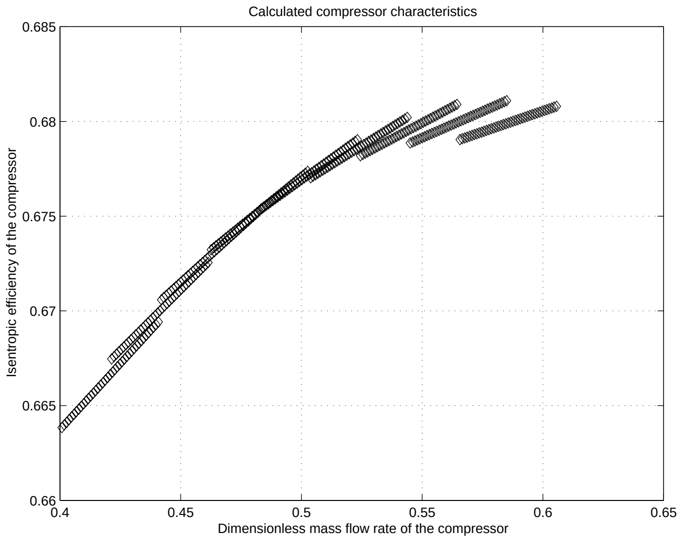

Fig. 6. Isentropic efficiency of the compressor (1. approximation)

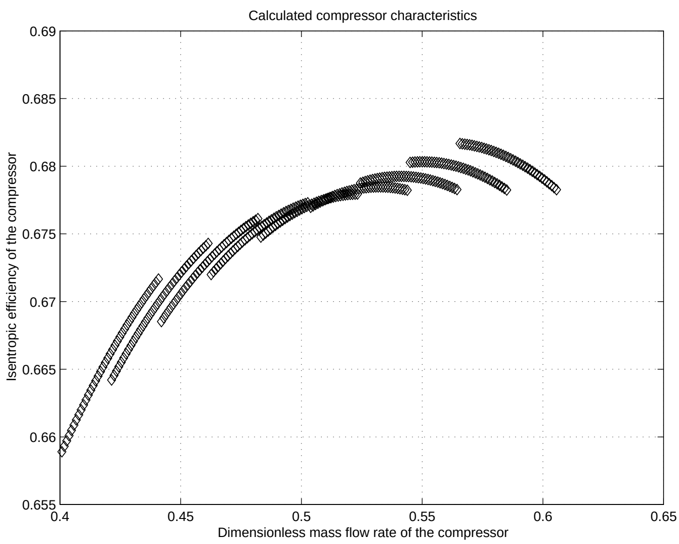

Fig. 7. Isentropic efficiency of the compressor (2. approximation)

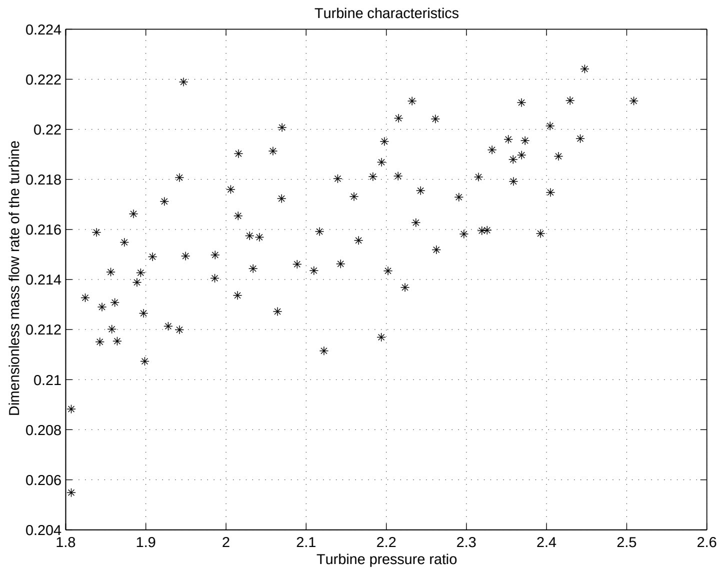

Fig. 8. Dimensionless mass flow rate of the turbine (measured points)

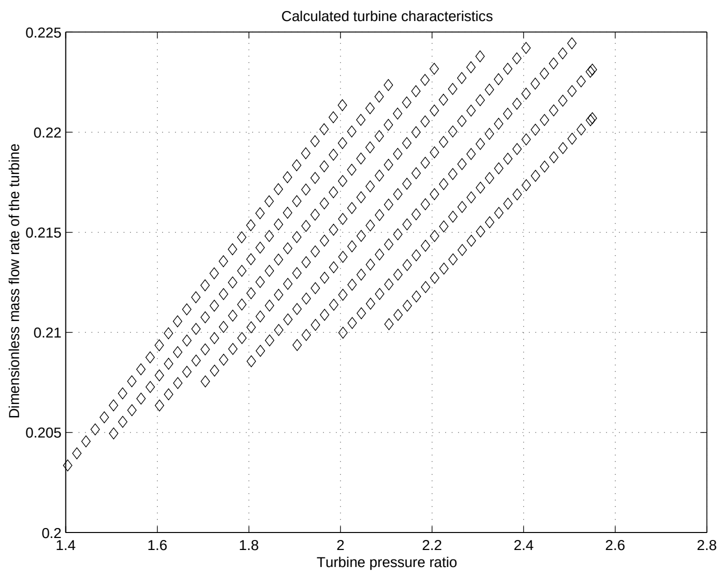

Fig. 9. Dimensionless mass flow rate of the turbine (1. approximation)

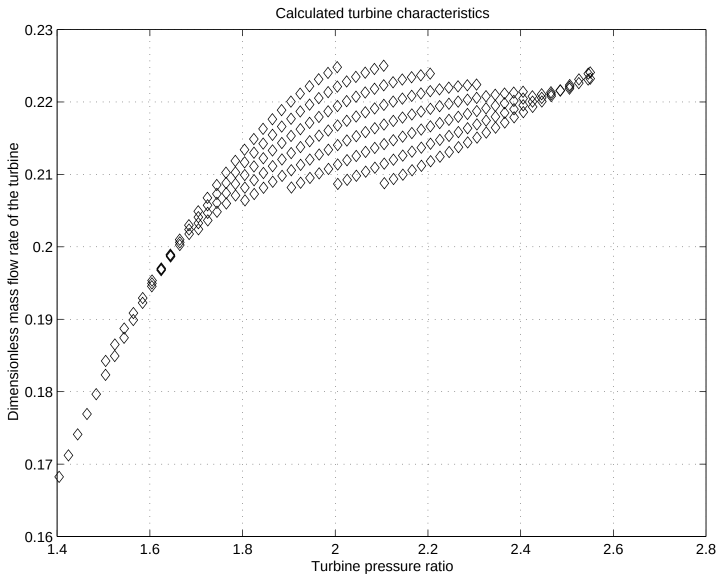

Fig. 10. Dimensionless mass flow rate of the turbine (2. approximation)

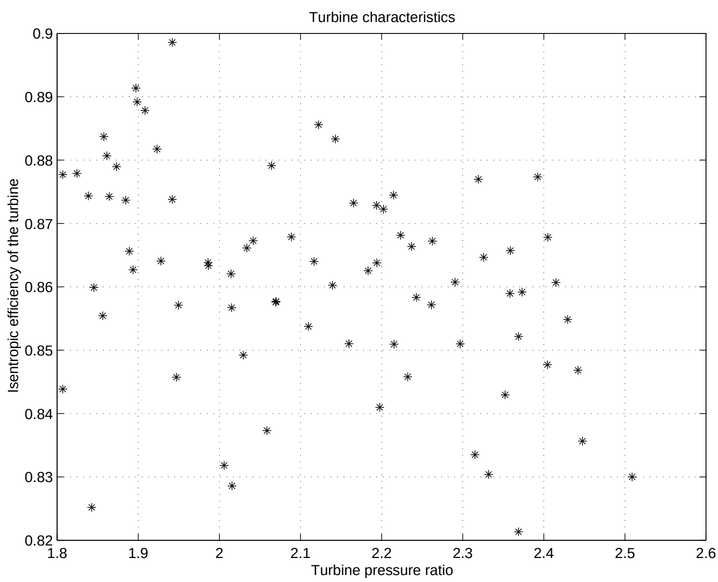

Fig. 11. Isentropic efficiency of the turbine (measured points)

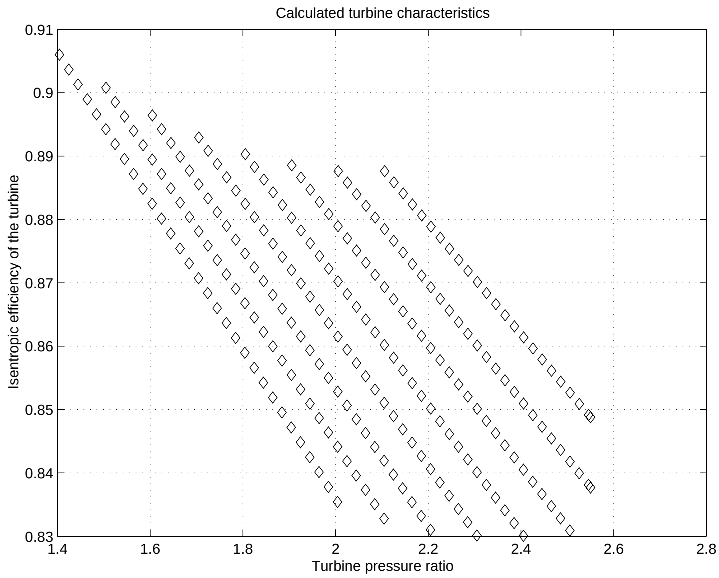

Fig. 12. Isentropic efficiency of the turbine (1. approximation)

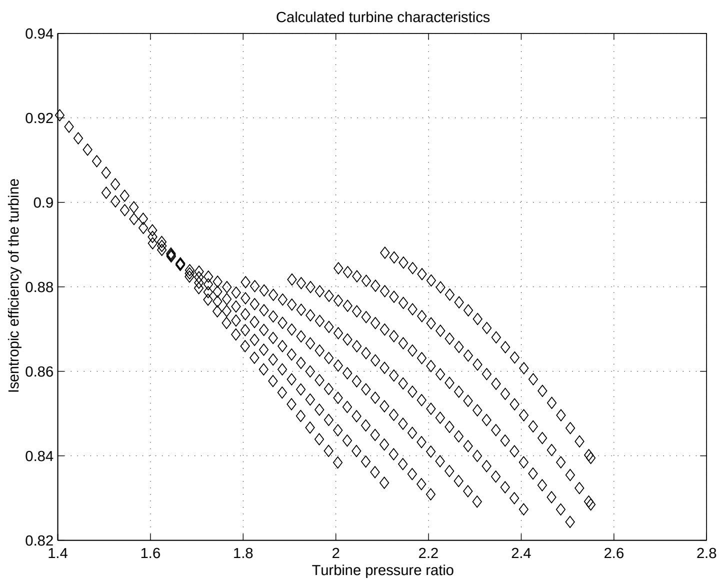

Fig. 13. Isentropic efficiency of the turbine (2. approximation)

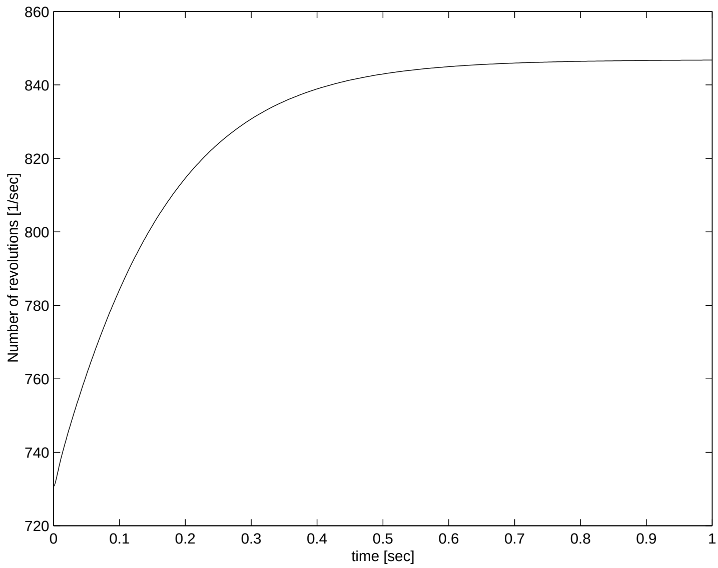

Fig. 14. Dynamic response of the number of revolution (simulation result)

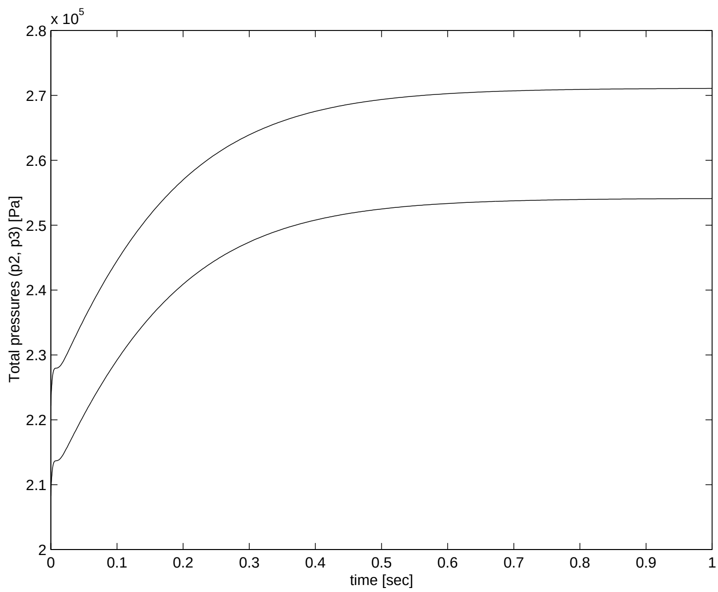

Fig. 15. Dynamic response of the total pressures (simulation result)

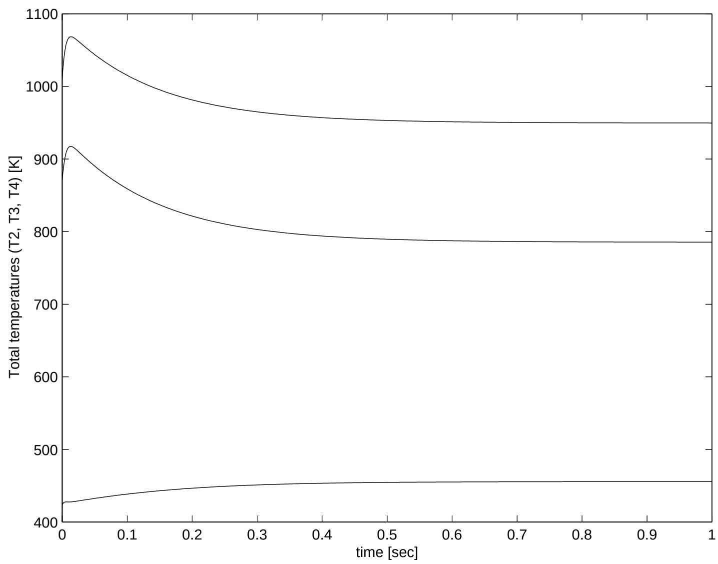

Fig. 16. Dynamic response of the total temperatures (simulation result)

operating points within the application area.

Using the results of the standard nonlinear analysis [2], we are able to design an appropriate nonlinear controller according to the control aims.

References

[1] HANGOS, K. M. – CAMERON, I. T., *Process Modelling and Model Analysis*, Academic Press, 2001.

[2] HANGOS, K. M. – BOKOR, J. – SZEDERKÉNYI, G., Analysis and Control of Nonlinear Process Systems, *Research Report of Computer and Automation Research Institute*, **SCL-1** (2000).

[3] SÁNTA, I., Non-linear Mathematical, Thermal Models of Gas Turbine Engines and their Application in Operation, *ICAS Proceedings* (1990), pp. 2264–2270.

[4] CZÉDLI, GY. – SÁNTA, I., Az NK8-2U típusú hajtómű termikus folyamatainak matematikai modellezése, *Járművek, Mezőgazdasági Gépek*, **11** (1985), pp. 422–426.

[5] SÁNTA, I., Repülőgép hajtóművek matematikai modellezése és a modell alkalmazása a diagnosztikában, *Járművek, Mezőgazdasági Gépek*, **11** (1985), pp. 426–431.

[6] QI, O. F. – MACCALLUM, N. R. L. – GAWTHROP, P. J., Improving Dynamic Response of a Single-Spool Gas Turbine Engine Using a Nonlinear Controller, *ASME 92-GT-392*.

[7] EVANS, C., Testing and Modelling Aircraft Gas Turbines: Introduction and Overview. *Prep. UKACC International Conference on Control'98*.

[8] AILER, P. – SZEDERKÉNYI, G. – HANGOS, K. M., Modeling and Nonlinear Analysis of a Low-Power Gas Turbine, *Research Report of Computer and Automation Research Institute*, **SCL-1** (2001).

Appendix: Notation List

Variables

c	specific heat [J/kg K]
i	specific enthalpy [J/kg]
m	mass [kg]
n	number of revolutions [1/sec]
p	pressure [Pa]
q	dimensionless mass flow rate [$-$]
t	time [sec]
Q_f	lower thermal value of fuel [J/kg]
M	moment [Nm]
Q	heat [J]
R	specific gas constant [J/kg K]
T	temperature [K]
U	internal energy [J]
V	volume [m^3]
W	work [J]
η	efficiency [$-$]
κ	adiabatic exponent [$-$]
λ	dimensionless speed [$-$]
$\dot{m}$	mass flow rate [kg/sec]
σ	pressure loss coefficient [$-$]
Θ	inertial moment [kg m^2]

Indices

*	total, stagnation
0	inlet duct inlet
1	compressor inlet
2	compressor outlet
3	turbine inlet
4	turbine outlet
I	refers to inlet duct
C	refers to compressor
Comb	refers to combustion chamber
T	refers to turbine
N	refers to gas-deflector
air	refers to air
gas	refers to gas
med	refers to medium parameters
comb	refers to combustion
mech	mechanical
load	loading
fuel	refers to fuel
in	inlet
out	outlet
p	refers to constant pressure
v	refers to constant volume