Investigation of Nonlinear Numerical Mathematical Model of a Multiple Shaft Gas Turbine Unit

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The development of numerical mathematical model to calculate both the static and dynamic characteristics of a multi-shaft gas turbine consisting of a single combustion chamber, including advanced cycle components such as intercooler and regenerator is presented in this paper. The numerical mathematical model is based on the simplified assumptions that quasi-static characteristic of turbo-machine and injector is used, total pressure loss and heat transfer relation for static calculation neglecting fuel transport time delay can be employed. The supercharger power has a cubical relation to its rotating velocity. The accuracy of each calculation is confirmed by monitoring mass and energy balances with comparative calculations for different time steps of integration. The features of the studied gas turbine scheme are the starting device with compressed air volumes and injector's supercharging the air directly ahead of the combustion chamber.

Key Words: Nonlinear, Numerical Mathematical Modeling, Gas Turbine, Performance, Multi-shaft, Injector, Surge Margin

Nomenclature

A : Heat transfer area, m^2
c : Specific heat, J/(kg K)
E : Energy, J
f : Cross sections of an injector, m^2
G : Moment, N-m
H : Enthalpy, J/kg
h : Heat transfer coefficient, W/(m^{2}K)
J : Polar moment of rotor inertia, kg-m^2
k : Thermal conductivity, W/(m K)
K : Feedback coefficient
m : Mass flow rate, kg/s
m : Mass of wall, kg
n^* : Exponential order for heat transfer coefficients
P : Power, W
p : Pressure, Pa

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Pr : Compressor and turbine pressure ratio
Q : Heat flow rate, W
R : Specific gas constant, J/(kg K)
S : Relative stability margin coefficient
T : Total temperature, K
t : Temperature, ℃
U_s : Control action signal
U : Overall heat transfer coefficient, W/(m^2 K)
u : Internal energy per unit mass, J/kg
u_{inj} : Injection coefficient
V : Volume, m^3

Greek

γ : Specific heat ratio
Δp : Pressure loss, Pa
ΔS : Compressor stability margin $\Delta Ss^H = Ss^H - 1.0$
$\Delta\tau$: Time step, s
δ : Channel wall thickness, m
η : Efficiency
μ : Relative displacement of actuator servo
ν : Kinematic viscosity, m^2/s
ξ : Correlation coefficient
ρ : Density, kg/m^3

σ : Coefficient for temperature distribution
τ : Time variable, s
ω : Rotor angular speed, rad/s

Subscript

a : Air
acc : Accumulated
av : Average
bra : Rotor breakaway moment
cool : Cooling
df : Direct feedback coefficient
e : Exit
f : Fuel
g : Gas
ga : Gain feedback coefficient
i : Index for parameters
in : Inlet
inj : Injection coefficient
int : Intermediate
mech : Mechanical loss
n, n+1 : Old and new time steps
out : Outlet
p : Constant pressure
s : Stability margin
sm : Servo-motor
sur : Anti surge valve
v : Constant volume
w : Wall
wor : Working
set : Pilot signal for regulator
0 : Nominal condition
1 : Outer surface
2 : Inner surface
3 : Injector mixing chamber

Superscript

* : Injector nozzle throat area
^ : Reduced parameter
- : relative parameter

Abbreviation

C : Compressor
CC : Combustion chamber
DE : Differential equation
GTU : Gas turbine unit
HP : High pressure
HPC : High pressure compressor
HPT : High pressure turbine
HPTC : High pressure turbo compressor
IC : Intercooler
LHV : Low heating value
LP : Low pressure
LPC : Low pressure compressor
LPT : Low pressure turbine
LPTC : Low pressure turbo-compressor
MM : Mathematical model
PC : Pipeline compressor
PT : Power turbine
T : Turbine
TC : Turbo-compressor

1. Introduction

The modern trends for enhancing the performance of gas turbine unit(GTU) are the improvement of cooling systems, employment of new concepts and materials and the revision of complicated technological schemes. They base on mathematical models(MMs) of static and dynamic regimes in all design stages. Such modeling is necessary not only for automatic control systems design (Cohen et al., 1996; Kim et al., 1998), estimation of tip clearances, calculation of a thermal state, definition of dynamic characteristics of elements, but also for the improvement of structural design, and analysis of the interdependent operating conditions of all components.

The numerical nonlinear MM to study the static and dynamic regimes (normal and emergency operation without loss of heat carrier) of a multi-shaft GTU with a free power turbine (PT), intercooler, intermediate and high pressure regeneration is considered in the paper. The effectiveness of the MM and used numerical integration method is illustrated in the example of a 2.5 MW three-shaft, driving the pipeline compressor(PC) GTU with an intercooler. The distinctive feature of the scheme is the employment of an injector-starting device with an air volume for direct supercharging of air in front of the combustion chamber(CC).

2. Physical and Mathematical Model

The MM of the GTU is based on the following simplified assumptions: a quasi-static flow condition in turbo-machine and gas injector; a calculation of total pressure losses and heat transfer coefficients for static methods, neglecting the transport delay through the fuel flow path and cubic dependence on rotating velocity of supercharger power consumption. The MM is described: the mechanical energy accumulated in the turbo-compressors (TCs) rotor elements, PT and PC; mass and internal energy accumulated within the volumes of all air and gas channels heat accumulated in walls of channels, air-coolers and regenerators. Mass and internal energy of air and heat accumulated in metal parts of starting system are also accounted. In a general view, it is possible to introduce the MM by a system of ordinary differential equations (DEs),

$$a_i \frac{dx_i}{d\tau} = F_i(x_i, x_2, \cdots, x_i, \cdots, x_I, x_{I+1}, \cdots, x_K, \tau); \quad (1)$$

$$i = 1, 2, \cdots, I; I \leq K$$

and closing functions,

$$f_j(x_1, x_2, \cdots, x_K) = 0, j = I+1, I+2, \cdots, K, \quad (1')$$

where, a_i are coefficients of DEs; $x_1, x_2, \cdots, x_K$ are parameters of a state; τ is a time; F_i is function of known operators $x_1, x_2, \cdots, x_K, \tau$; f_j are closing functions; I and K are number of DEs and unknowns depending on the degree of the MM. In Table 1, the main parameters and functional relations of equations included in Eq. (1) and Eq. (1′), and procedures describing the component characteristics of GTU are introduced. They correspond to the cases of both rotation and beginning of breakaway moment for turbo-machines rotors. The model or actual compressor characteristics are described by reference points of base speed lines. The estimation of compressor characteristics for low frequency regions and pressure losses at stop or slow rotating condition is carried out according to the work (Gittelman, 1974). The turbine characteristics are specified using the tables of reference points or calculated under an ellipse formula with updated coefficients for rotation speed (Kim and Soudarev, 2000). Air-coolers and regenerators are presented by sets of DEs for volumes of materials and cooling and heating heat carriers. The number of these volumes is assumed equal or multiple to the largest number of passes for one of two mediums. The air and gas ducts are similarly described.

In general, time constants T_{sm} of the regulator actuators can be reassigned depending on the sign and magnitude of the signal U and also with other control actions.

3. Algorithm of Numerical Calculations

The approach to a numerical integration of DEs shall be shown on an example of an equation of mechanical-energy accumulation for the rotor. The finite difference analog of this equation in an implicit form looks like

$$J\omega^*_{n+1}(\omega_{n+1} - \omega_n)/\Delta\tau = \Delta P, \quad (2)$$

where, ω_{n+1} and ω_n are the values of angular speed of rotation at time $n+1$ and preceding instant n; ΔP is the power imbalance in the shaft during the step of integration $\Delta\tau$. The solution with respect to the speed of rotation ω_{n+1} can be described as,

$$\omega_{n+1} = \omega_n + \Delta P\, \Delta\tau/(J\omega^*_{n+1}), \quad (2')$$

where ω^*_{n+1} is a predictable value of rotation speed in the $n+1$ instant. For irreversible machines, $\omega_{n+1} \geq 0$.

The value of a speed of rotation ω_n at an instant n is known from initial conditions or calculation of previous instant. The value, ω^*_{n+1}, by which the characteristics and power imbalance P on the shaft of the turbo-machine or PT are defined, can be found by a universal procedure of iterative radicals searching for nonlinear equations (Kovalevski, 1992) until the following discrepancy condition is satisfied:

$$|\Delta\omega_{n+1}| = |(\omega_{n+1} - \omega^*_{n+1})| \leq \varepsilon_\omega(\omega^*_{n+1}), \quad (3)$$

where, $\varepsilon_\omega(\omega^*_{n+1}) \geq 0$ is the accuracy of the solution in an iterative cycle of ω_{n+1}. Procedure of finite difference analogs construction and integration for other DEs of the MM is implemented similarly. For a numerical integration of DE for complex scheme, such $\Delta\tau$ as multi-shaft GTU, errors induced by truncation is unavoidable and

Table 1 Components of the MM for a GTU

Element		a_i	x_i	F_i	f_j
Rotors of turbo machine	When $\omega>0$	$J\omega$	ω	$P_T-P_C-P_{mech}$	$P_T=c_{pg}T_g(1-r_T^{(1-\gamma)/\gamma})\eta_T$; $c_{pg}(T_{g\,av})$; $c_v=c_p-R$; $\gamma=c_p/c_v$; $m^*=m\sqrt{T}/p$; $\omega^*=\omega/\sqrt{T}$; $r_T=r_T(m^*_g,\ \omega^*_T)$; $\eta_T=\eta_T(\omega_T/c^0_g)$; $P_C=c_{pg}m_a T_a(r_C^{(\gamma-1)/\gamma}-1)\eta_c$; $c_{pa}=c_{pa}(T_{a\ av})$; $r_C=r_C(m^*_g,\ \omega^*_c)$; $\eta_C=\eta_C(m^*_a/\omega^*_c)$; $P_{mech}=P_{o\ mech}(\omega/\omega_o)^{1.8}$; $T_{av}=(T_{in}+T_{out})/2$;
	$\omega=0$	J	ω	$m_T-m_C-m_{mech}$	$m_{mech}=m_{bra}$
Elementary volumes of air-gas path : air ducts, gas ducts, CC, intercoolers and regenerators ; starting bottles		V	ρ	$m_{in}-m_{out}$	$p=R\rho T$; $p_{av}=(p_{in}+p_{out})/2$; $T_{av}=\sigma T_{in}+(1-\sigma)T_{out}$; $p_{out}=p_{in}-\Delta p$; $\Delta p=\Delta p_o\ \xi_R\ \xi_T\ \xi_m$; $\xi_R=R/R_o$; $\xi_T=T_{av}/T_{0av}$; $\xi_P=p_{0av}/p_{av}$; $\xi_m=m_{av}/m_0$ $u=c_v t$; $t=T-273$; $Q_w=U_2A_2(t-t_w)$; $Q_{in}=(m\ c_P t)_{in}$; $Q_{out}=(m\ c_P t)_{out}$
			ρu	$Q_{in}-Q_{out}-Q_w$	
Walls of elementary volumes		$c_w m_w$	t_w	$Q_{w1}-Q_{w2}$	$Q_{w1}=U_1A_1(t_1-t_w)$; $Q_{w2}=U_2A_2(t_2-t_w)$ $U_1=[1/h_1+(1-\sigma_w)\ \delta_w/k_w]$; $\xi_k=k_{av}/k_{0av}$; $U_2=[1/h_2+\sigma_w\ \delta_w/k_w]$; $\xi_k=\nu_{0av}/\nu_{av}$; $h=h_o\ \xi_k[\xi_R\ \xi_T\ \xi_m\ \xi_v]^{n*}$
speed regulator		T_{sm}	μ	$U\,\|\,U\|\le 1$	$U=k_{f\ ga}(w_{set}-w_{pc})-k_{f\ df}\ \mu_f$; $m_f=m_{f0}(\mu_f)$; $Q_{cc}=Q_{cc}(m_f)\ \eta_{cc}$
Regulators of an anti-surge protection					$U=k_{s\ ga}(S_{s\ set}-S_s)-k_{s\ df}\mu_{sur}$; $f_{sur}=f_{sur}(\mu_{sur})$; $m_{sur}=m_{sur}(f_{sur}, p, T)$

Note 1. The subscripts : 0 indicates the nominal condition ; av, in, out indicate averaged, input and output parameters ; a, g indicate air and gas. c^0_g is an ideal efflux velocity.

Note 2. The subscripts 1 and 2 for A, h, U, t, Q_w correspond to inner and outer heat transfer surfaces of a considered volume.

Note 3. For a control signal U of the speed and anti-surge protection regulators, main terms are recorded only. The additional, connected with a particular for concrete operation regime, limitations of control algorithm will be specified in the text. The index set is entered for a pilot signal of the regulator setting device.

selection of integration time step as well as iteration numbers are very important. The integration step is selected in view of minimum value of time constants of members in dynamic system. Further it is updated by the results of comparative calculation of different $\Delta\tau$ and remains constant when further decreasing does not influence on the resultant output. Integration time step for static and dynamic calculation was between 0.01s~0.05s and imbalances in mass and thermal energy δM and δQ were $2.4\times10^{-4}\%$, $5.3\times10^{-2}\%$, respectively. The numerical integration of the equation set of the MM is carried out under the implicit iterative finite difference scheme. The main global cycles of dynamic calculations are the cycle for rotation speed of the PC (or the

PT) ω_{PC}, cycle for inlet flow rate of air main (Zhuravlev, 1993), and (for a regenerative scheme) a cycle for air temperature in front of the CC. The main, inserted cycles are the cycles for speeds of rotation of TCs ω_{HP} and ω_{LP}, pressure behind turbo-machines and moving of the anti-surge valves μ_{sur}. The equations for heat accumulated in the heat exchanger walls have a larger time constant as compared with those for the rotors, and are much larger than for the air-gas path volume. This enables calculation with the wall temperature from the previous time n and without a large loss of accuracy, energy equation for exchanging heat mediums can be solved without iteration with respect to the outlet temperature directly. The equations for conservation of mass and energy for the accumulate volumes of gas-air channels are solved with respect to the outlet parameters of working mediums: flow rate and temperature. It, and also the solution of equations for compressors and turbines with respect to outlet pressure allows using the trough running solving scheme for all chain of accumulating volumes and turbo-machines and to find current distribution of parameters: the flow rate, temperatures and pressures at all points of the channel inlet to the outlet section.

The calculation of static performances is thus carried out using an iterative finite difference scheme. In this case, the external cycle for rotation speed of the PC (or the PT) ω_{PC} is substituted with the external cycle for the fuel flow rate mf. The scheme of the remaining iterative cycles completely coincides with the scheme described above. The formal similitude in non-stationary and stationary writing of the finite difference analog equations at $\Delta\tau \to \infty$ (see finite difference analog in Eq. (2)) allows calculation of static and dynamic conditions practically within the framework of one algorithm. Mass imbalance in static calculation is calculated as,

$$\Delta M = M_{input} - M_{output} \tag{4}$$

$$M_{input} = M_{inlet} + \Sigma M_f,\ M_{output} = M_{leak,i} + M_{out} \tag{5}$$

M_{inlet} is air flow rate into the GTU and M_f is the fuel input for both primary and additional burner. M_{leak} and M_{out} are the flow leaked in the whole GTU and exhaust gas product at exit channel respectively. Imbalance of the thermal energy for static calculation is calculated as,

$$\Delta Q = Q_{input} - Q_{output} \tag{6}$$

$$Q_{input} = M_{in} \times H_a + \Sigma M_{f,i} (Q_{LHV} + H_f)\, \eta_{CC,i} \tag{7}$$

$$\begin{aligned} Q_{output} = {} & \Sigma M_{leak,i} \cdot H_{leak,i} + \Sigma Q_{in.cool,i} + \Sigma Q_{cool,air,i} \\ & + \Sigma N_{mech,i} + \Sigma N_{GTU} + M_e . H_e \end{aligned} \tag{8}$$

$$\Sigma Q_{int.cool,i} = \Sigma M_{cool,i} (H_{int.cool,i} - H_{out.cool,i}) . \tag{9}$$

For relative error of mass and energy conservation, following equations are used.

$$\delta M = \Delta M / M_{in} \tag{10}$$

$$\delta Q = \Delta Q / Q_{in} \tag{11}$$

Integral dynamic mass imbalance for time interval $\tau_{end} - \tau_{begin}$ is defined as,

$$\Delta M = M_{in} - M_{acc} - M_{out} \tag{12}$$

$$M_{in} = \int M_{in}(\tau)\, d\tau + \int M_{start}(\tau)\, d\tau + \Sigma \int M_{f,i}(\tau)\, d\tau \tag{13}$$

$$\begin{aligned} M_{acc} & = \Sigma \int V_i (\bar{\rho}_i(\tau)/d\tau)\, d\tau = \Sigma V_i \int d\bar{\rho}_i(\tau)/d\tau d\tau \\ & = \Sigma V_i (\bar{\rho}_{i,end} - \bar{\rho}_{i,begin}) = \Sigma (M_{i,end} - M_{i,end}) \\ & = M_{i,end} - M_{i,begin} \end{aligned} \tag{14}$$

$$M_{out} = \Sigma \int M_{leak,i}(\tau)\, d\tau + \int M_{e,i}(\tau)\, d\tau + \int M_e(\tau)\, d\tau . \tag{15}$$

For dynamic imbalance of thermal and mechanical energy, following equations hold,

$$\Delta E = E_{in} - E_{acc} - E_{out} \tag{16}$$

$$\begin{aligned} E_{in} = {} & \int M_{in} H_a\, d\tau + \int M H_{start}\, d\tau + \Sigma \int P_{start,i}\, d\tau \\ & + \Sigma \int M_{f,i} (Q_{LHV} + H_T)\, \eta_{KC,i}\, d\tau \end{aligned} \tag{17}$$

$$E_{acc} = E_{Uacc} + E_{Cacc} + E_{Jacc} \tag{18}$$

$$\begin{aligned} E_{Uacc} & = \Sigma \int V_i (d\rho u_i / d\tau)\, d\tau = \Sigma V_i \int (d\rho u_i / d\tau)\, d\tau \\ & = \Sigma V_i (\rho u_{i,end} - \rho u_{i,beg}); \end{aligned} \tag{19}$$

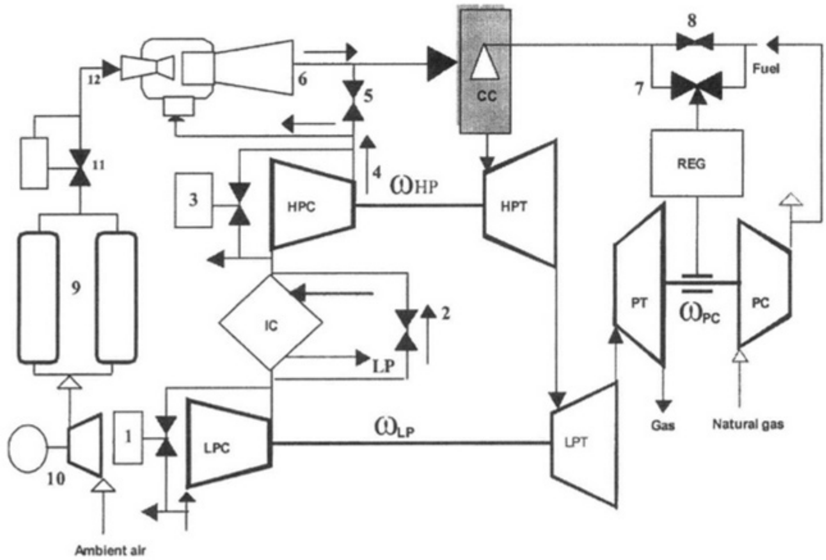

Fig. 1 The principal scheme of the GTU
LPC and HPC are compressors of low and high pressures; IC is an intercooler; CC is a combustion chamber; LPT and HPT are turbines of low and high pressures; PT is a power turbine; PC is a pipeline compressor; ω_{LP} and ω_{HP} are the rotation speeds of turbo-compressors of low LPTC and high HPTC pressures of the gas generators; ω_{PC} is rotational speed of the PC and PT; 1-LPC anti-surge device; 2-bypass valve; 3-HPC anti-surge device; 4-main pipe line; 5-starting slide valve; 6-starting pipe line; 7-fuel valve; 8-ignition valve; 9-accumulating bottles; 10-compressor of air re-pressuring; 11-reduction gearbox; 12-injector.

$$E_{Cacc}=\Sigma\int m_i(\mathrm{d}cT_{wi}/\mathrm{d}\tau)\,\mathrm{d}\tau=\Sigma m_i\int(\mathrm{d}cT_{wi}/\mathrm{d}\tau)\,\mathrm{d}\tau =\Sigma m_i(cT_{wi,end}-cT_{wi,beg}); \tag{20}$$

$$E_{Jacc}=\Sigma\int J\omega_i(\mathrm{d}\omega_i/\mathrm{d}\tau)\,\mathrm{d}\tau=\Sigma J_i\int(\mathrm{d}(\omega_i^2/2)\,\mathrm{d}\tau)\,\mathrm{d}\tau =\Sigma J_i(\omega_{i,end}^2-\omega_{i,beg}^2)/2; \tag{21}$$

$$E_{out}=\Sigma\int M_{leak,i}H_{leak,i}\,\mathrm{d}\tau+\Sigma\int Q_{in,cool,i}\,\mathrm{d}\tau +\Sigma\int Q_{cool,a,i}\,\mathrm{d}\tau+\Sigma P_{mech,i}\,\mathrm{d}\tau +\int P_{GTU}\,\mathrm{d}\tau+\int M_eH_{e,i}\,\mathrm{d}\tau \tag{22}$$

4. Description of the GTU

In order to demonstrate the possibilities of designed MM, calculation results of static and dynamic regimes for a 2.5 MW pipeline GTU (see Fig. 1) with intercooling, are shown above. The GTU has a gas generator with two radial type turbo-compressors of low (LPTC) and high (HPTC) pressures and a free PT working with PC. Initial parameters of a nominal condition at ambient temperature $t_a=15$℃ are: pressure $p_{HPT}=1.5$ MPa, temperature $t_{HPT}=1170$℃, airflow rate $m_{air}=6.3$ kg/s, and efficiency $\eta_{GTU}=35\%$. The moment of rotor inertia of the HPTC and the LPTC are equal to 0.014 and 2.1 kg-m^2, and the PT with PC is 2.4 kg-m^2, respectively. Air from the LPC of LPTC (see Fig. 1) with a compression ratio of 3.9 passes through the pipeline where an anti-surge device 1 is installed. Then air enters four-pass tubular counter-flow IC. The IC has bypass valve 2, which will be opened for softening of thermal shock during emergency shutdown. Air from the IC enters suction connection of the HPC. HPC as well as LPC has an anti-surge device 3 and a compression ratio of 3.9 at inlet temperature 40℃. From HPC, high-pressure air through the air duct 4 which has a valve 5 and a branch 6 toward the starting injector, goes to the two-stage CC with premixing of fuel. Fuel is provided through a fuel valve 7 and an ignition valve 8. The combustion products sequentially pass common (with the purposes of reducing the lengths) high-temperature channels of turbo-compressors and go to the PT, activating the PC-supercharger. For start-up of the GTU with axial turbo-machines and turbo-compressors of small moments of inertia, supercharging the air via the injector from starting bottles at inlet of HPC or LPC (Dajneko, 1984) is made. The start-up of the GTU, in this case, proceeds with a sufficient compressor stability margin. The ignition of burners becomes practically possible immediately follwing air provision. It essentially shortens the startup time, though energy consumption is higher than that used by an electric starter motor. The presence of the pipe line 4 between the HPC and the CC, as different from an axial GTU with annular type CC, has allows the possibility of an original injector starting system with accumulated bottle 9 (total volume 2 m^3) for supercharging of air in this pipe line directly ahead of the CC. The initial filling of the volume and its consequent replenishment can be accomplished by repressuring the compressor 10. The flow rate m_{wor} with starting air pressure 4.0 MPa goes in the Laval nozzle of jet injector 12 after the pressure is lowered to 1.6 MPa in the reduction gearbox 11. The acceptance chamber of the injector is connected to the pipeline 4 up to the valve 5, and the delivery line-after valve 5. Upan starting, the valve 5 is closed. The injector sucks in air through the compressors, IC and air ducts. The flow rate m_{inj} is determined by the geometrical characteristic of the injector and the value of injection coefficient $u_{inj}=m_{inj}/m_{wor}$, which is functionally related to its relative pressure ratio.

Supercharging of air in the CC allows the ignition to start almost immediately once working air has been supplied. The ejected (through the compressors) flow rate of air is substantially less than the total flow rate (through the turbines). With a decrease in the compressor's consuming power (with acceleration of rotor speed), the stability margin will accordingly be decreased. Considering the lack of a similar starting system, it will be necessary to study the characteristics and optimization of the starting algorithm with a prescribed limit on temperature overshoot at ignition point and stability margin factor S_s of the HPC. Stability margin is defined as, $S_s=\left(\frac{Pr}{m}\right)_{surge}/\left(\frac{Pr}{m}\right)_{working}$. Below, a typical, calculated results of the executed example is given.

5. Modeling of Basic Operational Modes

The calculation results of start-up regime are obtained with the following control algorithm. The fuel rate through the ignition valve 8 is 0.015 as a nominal value. The valve 5 is closing upon switching-off the starting air. The opening speed of a fuel valve 7 is constant. The time constant required for its drive system was set equal to 10s. At $S_s<1.10$, the opening of the fuel valve 7 for HPC is suspended, and at $S_s<1.08$, a maximum pilot signal for opening the bleed valve 3 next to the HPC is sent additionally.

In Fig. 2, calculation results of one start-up regime for supercharging the air into the pipeline by an injector in front of the CC are introduced. Here, only the relative values of parameters are presented for startup regime. They are obtained by dividing the parameter values by its value at the nominal load. In the figures, these parameters are marked upper-dashed. The flow rate of

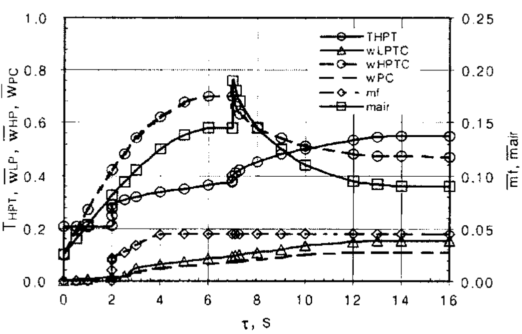

Fig. 2 Time variation of the GTU relative parameters at an injector start-up THPT ; High pressure inlet temperature, wLPTC, wHPTC, wPC are rotor speed of low pressure spool, high pressure spool and power turbine compressor, mf, mair are fuel flow rate and airflow rate, respectively

starting air is determined by a critical efflux and varies lightly in time. It is about 1.1kg/s. For prevention of HPC's hitting the surge range, a convenient mechanism to delay the starting burner ignition with respect to the moment of starting air provision is established. In this way, a two-seconds delay allows startup to be executed with a bleed valve 3 behind the HPC closed. The minimum stability margin coefficient S_s at this moment is not less than 1.08. The opening of bleed valve 3, in view of rarefaction in the beginning and minor pressure ratio in the early moments of starting, has small effectiveness. The temperature overshoot before the HPT with ignition of the starting burner (τ=2s) does not exceed 100℃. With a sudden cutoff of starting air (τ=7s), the temperature increase becomes about 50℃. Displacement of the fuel valve μ_f from time=4s and up to the instant of the starting air cutoff is confined due to a decreased HPC stability margin of less than 1.1. After cutoff of the starting device, this movement becomes stabilized to be a steady-state condition at exit.

After the moment of starting air cutoff, slowdown of the HPTC rotation speed is noticed. This is caused by a jump of airflow in compressors (approximately on 20%) at the cutoff and corresponds to an increase in power consumption by them. The unit goes out to static conditions at ω_{PC}=0.1. The consumed for start-up, total mass of air is 7.5 kg. The pressure of the starting bottles is reduced from 4.0 to 3.6 MPa, and accordingly, the temperature is reduced from 15 to 6℃. In the example above, further increase of speed up to the nominal rotation was not considered. Only the actual start-up process was considered here.

For the study of static and dynamic regimes, the following control algorithm was assumed. At temperature overshoot before HPT, a signal controlled by a prescribed data is issued to open or close the fuel valve. At S_s<1.10, the signal to open the compressor anti-surge valve is issued proportional to the magnitude of deviation, and at S_s<1.08, as in the case of start-up, the maximum control signal is issued. The GTU static characteristics depending on a relative rotation speed of the PC shaft ω_{PC} at partial loads for the case of ambient temperature t_a=15℃ are shown in Fig. 3. High spool turbo-compressor rotor speed is higher than low spool. Air flow rate, power, and efficiency are all increasing with rotor speed. Excess airflow rate, however, decreases with the rotor speed. Combustion chamber exit temperature has the highest increasing rate with the rotor speed compared with HPC exit and GTU. In Fig. 4, the characteristics of the transient process are shown. Effect of signal changing in the speed regulator is shown. In the beginning, the signal was in steady state condition and the value is 1.0. At τ=0s, this value is dropped to 0.7 and

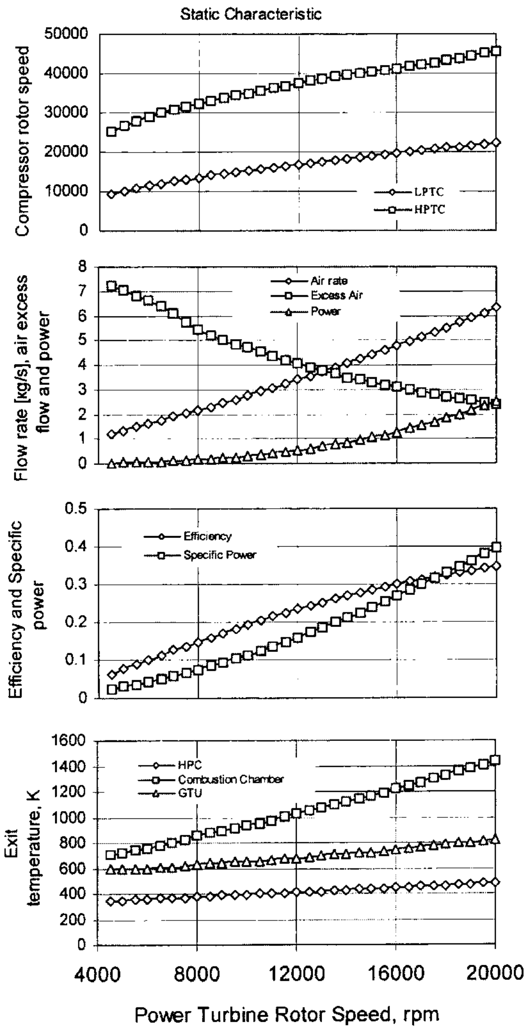

Fig. 3 Static characteristic of the GTU with variation of the PT(PC) rotation speed

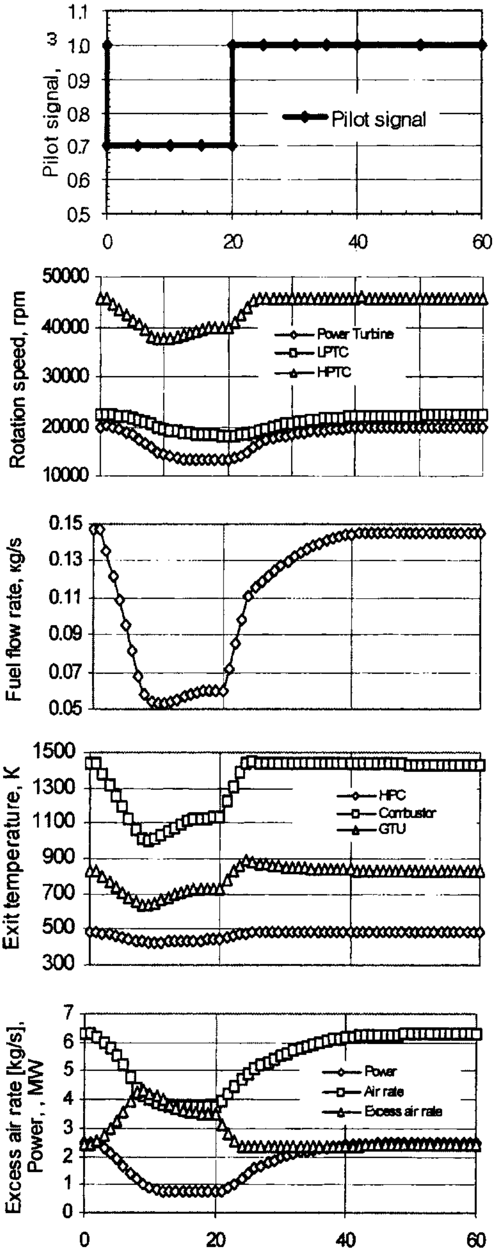

Fig. 4 Transient processes of rotor speeds at changing of the pilot signal of the speed regulator device under the law 1.0

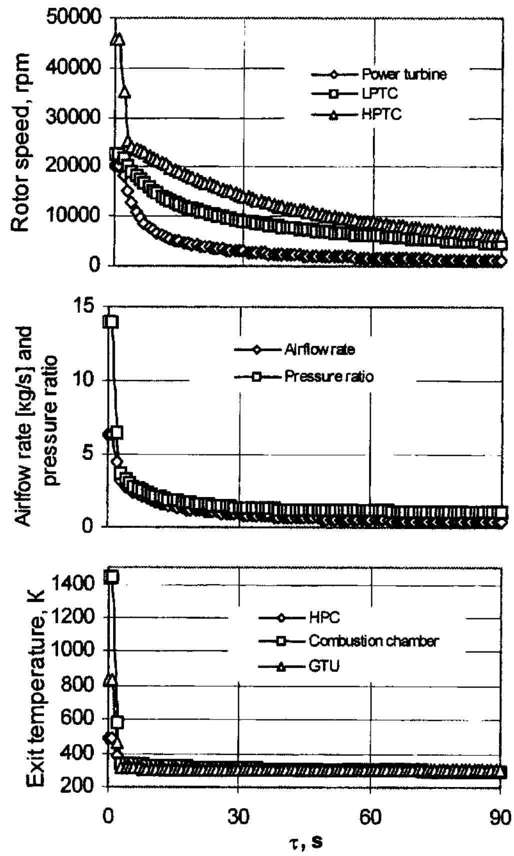

Fig. 5 Time variation of rotation speeds during emergency shutdown of the GTU under the nominal load

remained constant for the next 20s, and then eturns back to 1.0. The time step used in the numerical calculation varies from 0.01 to 0.05s depending on the calculation regime. The time process of the supercharger and LPTC rotation speeds are aperiodic. For the rotation speed of the HPTC shaft, an overshoot is observed (0.07 at a drop and 0.03 at a pick up). The transient period is about 14s. Here and further in Fig. 5, effect of change in rotation speeds is discussed.

In Fig. 5, the transient process during an emergency stop of nominal operating regime is submitted. At the instant τ=0s, the fuel cut-off is implemented during 1s. For the next 4s, a sharp decrease in HPTC shaft rotation speed is observed. This can be explained due to a small moment of inertia in the HPTC rotor as compared with that in LPTC (0.014 kg-m^2 of HPTC as compared with 2.1 kg-m^2 for the LPTC). After that, there is a decrease of deceleration in the HPTC rotor speed due to the flow impetus from the LPTC.

6. Start-Up Characteristics

Considering the importance of the start-up process of the GTU, a review of starting characteristics with an injector start-up device has also been studied. Special attention is given to HPTC that has a tendency to approach the surge line. The necessity to delay the ignition moment relative to the point of compressed air supply has already been discussed. Here, the influence of injector performance and starting airflow rate is studied. The design parameter f_3/f^*, which defines the pressure-flow rate characteristic of the injector with a Laval nozzle for supercritical efflux of the working medium, is the ratio of cross-sectional area of injector mixing chamber f_3 to the critical nozzle throat area f^* (Sokolov and Singer, 1970). In general, one unique pressure ratio in jet nozzle with small f_3/f^* corresponds to a higher injection coefficient u_{inj}.

In Fig. 6, comparative calculation results of start-up process for two design characteristics of injectors with $f_3/f^*=34$ (variant 1) and $f_3/f^*=47$ (variant 2) during the initial 20s are presented. The initial values of the compressed air flow rate in this case are about 0.56 and 1.31 kg/s. It can be seen that in start-up variant 1, is executed with a large margin of gas-dynamic stability ($S_s>1.13$) without opening the bleed off air valve 3. This is due to higher airflow $m_{air}=m_{inj}$ sucked in through the compressors. Cutting off of starting air (opening of the valve 5) will be instantaneous. This moment corresponds to the point of air flow jump in the cycle. The increase in airflow through the compressors will require higher power consumption. This, in turn, will reduce the HPTC rotation speed (relative values are 0.06 for variant 1 and 0.18 for variant 2). By the end of the 20s following startup, the HPTC rotation speed becomes $\omega_{HP}=0.77$ (variant 1) and $\omega_{HP}=0.42$ (variant 2). This difference in variant 2, based on the adopted control algorithm, can be explained by the limitation in fuel valve opening motion and the presence of discharged air. After switching-off the starting system, the fuel valve position in calculation become stabilized. The expended total mass of starting air becomes 4kg for variant 1 (at $f_3/f^*=34$). It corresponds to 60% of the nominal air flow rate of the cycle. This is considerably less than that of a similar estimation (110....150%) for an injector start-up with supercharging at inlet of LPC (Dajneko, 1984).

It also important to investigate start-up regimes for lower and higher ambient temperatures.

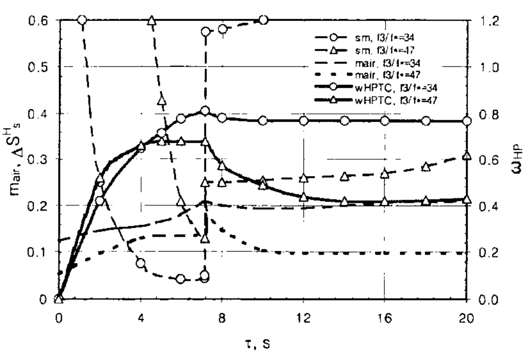

Fig. 6 Time variation of HPTC parameters at an injector start-up of the GTU with two different variants of flow rates and injector characteristics ; ○ : $m_{wor}=0.56$ kg/s, $f_3/f^*=34$; △ : $m_{wor}=1.31$ kg/s, $f_3/f^*=47$; sm : $\Delta S_s^H=S_s^H-1.0$ is an absolute stability margin of the HPC ; mair is a relative air flow rate in the cycle ; wHPTC is a relative rotation speed of HPTC

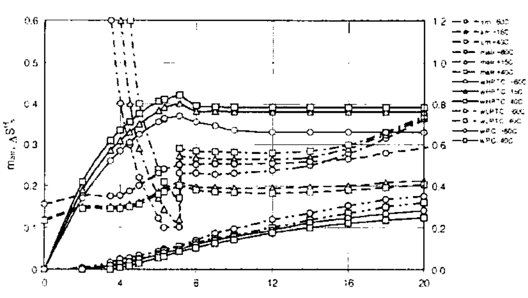

Fig. 7 Time variation of the GTU parameters at injector start-up with $m_{wor}=0.56$ kg/s ; $f_3/f^*=34$ and of different ambient air temperatures : sm is the stability margin in HPC for $t_a=-60$℃, $t_a=+15$℃ and $t_a=+40$℃ ; mair is air flow rate, wHPTC, wLPTC are rotor speed of hipg pressure and low pressure spool, wPC is power turbine compressor, respectively

These are reviewed in example of variant 1 with m_{wor}=0.56 kg/s ; f_3/f^*=34, too (see Fig. 6). In Fig. 7, the results of these calculations are presented at ambient air temperatures of −60℃, +15℃ and +40℃.

The start-up at t_a=40℃ is quite similar to the start-up at t_a=15℃. At t_a=−60℃, decrease in the stability margin of HPC is observed. This restricts the fuel supply and as a result, leads to a slow down of the HPTC rotation speed (at 20^{th} seconds, it is only 0.66 against 0.76 and 0.78 at temperatures +15℃ and +40℃, respectively). For a rotation speeds of LPTC and the supercharger, the opposing characteristic has been noticed : the higher rotation speeds correspond to lower temperatures.

The calculation results presented above are the most typical cases of static and dynamic conditions of the GTU, and are executed by the program SADGTU (calculation of Static And Dynamic regimes of GTU), which one was written on the COMPAG VISUAL FORTRAN 6.5. The accuracy of the integration is confirmed by monitoring the observance of mass and energy (calorific and mechanical) balances, and comparative calculations by different time steps for integration.

7. Conclusions

The numerical nonlinear MM to study static and dynamic behavior of a multiple-shaft GTU during all operation conditions as well as emergency conditions without loss of heat carrier is introduced. In general, this model is designed to study four shafts GTU with three TCs working with a free PT driving the PC. The MM enables the calculations with the intermediate air coolers and with the intermediate and tip regenerators. The MM can also be used in a structure of more complicated MMs of combined power plants.

The convenient, iterative finite difference scheme for integration of the system equations for static and dynamic problem is offered. The numerical study of the GTU processes with a starting injector and with direct air supply to the CC is shown :

(1) The system ensures a sufficient reserve of compressors gas dynamic stability and permissible operating temperature conditions for turbines

(2) For a purpose of increasing the gas dynamic stability margin by decreasing energy consumption and temperature overshoot upon ignition, it is necessary to have a limited starting airflow rate and to have higher injector pressure ratio.

(3) During the start up process for ambient temperature range from 15 to 40℃, parameters' value do not differ significantly ; however, at a reduced temperature, acceleration rate of the HPTC rotor is observed at some increased rotation speed of LPTC and PT ;

(4.) The computed value of a total starting air mass (4 kg at f_3/f^*=34) corresponds to 60% of the nominal air flow rate of the cycle. This is considerably less than that of a similar estimation (110, ⋯, 150%) for an injector start-up with supercharging at inlet of LPC (Slobodianiuk and Dajneko, 1983).

The further development of the MM is needed to be executed in the directions of: description of the PT operation in generator regime, description refinement of control system and actuators, addition of a complementary CC and consideration not only recuperative heat exchangers, but also regenerators.

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