METHOD OF CALCULATING TRANSIENTS IN GAS TURBINE POWER PLANTS

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EDITED MACHINE TRANSLATION

METHOD OF CALCULATING TRANSIENTS IN GAS TURBINE POWER PLANTS

By: V. V. Selin

English pages: 12


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PREPARED BY:
TRANSLATION DIVISION
FOREIGN TECHNOLOGY DIVISION
AFB, OHIO.

Date 14 Sept. 1967
ABSTRACT: A method of finite differences for calculating the transient characteristics of a gas-turbine power plant is offered; the method is claimed to be "simpler and more reliable as compared to existing methods." The following formulas are developed: rate-of-flow in the turbine; same in the compressor; compression in the compressor; expansion in the turbine; balance of total compression ratio and total expansion ratio; rate-of-change of gas quantity in the capacity (regeneration case); heat exchange in the regenerator; power balance in the compressor shaft; compressor efficiency; turbine efficiency. The above formulas, some additional turbine formulas, and ship-propulsion formulas are tabulated. Peculiarities in the calculation of a load-drop case are discussed. Orig. art. has: 2 figures, 33 formulas, and 1 table. English Translation: 12 pages.
## U. S. BOARD ON GEOGRAPHIC NAMES TRANSLITERATION SYSTEM

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* ye initially, after vowels, and after Ь, ь; e elsewhere. When written as Е in Russian, transliterate as Ye or Е.

The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.
METHOD OF CALCULATING TRANSIENTS IN GAS TURBINE POWER PLANTS

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For a well-founded selection of a gas turbine power plant [GTU][ITY] layout, during its designing of great value is a detailed calculated analysis of characteristics in transitional modes (transients), for which a sufficiently simple and exact method of calculation is necessary. In this article a method of calculation is offered which is more simple and reliable, as compared to the existing methods.

Under variable conditions, the number of independent parameters for any layout of a GTU is equal to the number of combustion chambers [1]. During transitions of a GTU, the number of independent parameters, simply determining this or that mode, is equal to the sum of the number of working combustion chambers and the number of rotors in the layout. As an example let us examine the two-shaft layout with a low pressure power turbine [TND](TPf) (Fig. 1). If for this layout we assign, for example, the temperature beyond the combustion chamber and the number of rotor turns, then we will simply determine the mode of the power plant.

Data available in literature permit examining the layout of a high pressure air duct as one which is discrete [3, 5], i.e., to consider that the given capacity is concentrated in one point - beyond the regenerator (Fig. 2).

One may assume that establishing the gas parameters before the turbine and after it occurs instantly [2, 3], which is equivalent to disregarding the gas volumes in the flow-through part of the turbine. Here the working substance can be considered as being inertialless. This assumption will agree with the results of experimental investigations [3].

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Fig. 1. Diagram of a two-shaft ΠПУ with independent low pressure power turbine. ТВД — high pressure turbine; ΗД — low pressure turbine; К — compressor; Р — regenerator; КС — combustion chamber; П — reducer; 1, 2... — indices of the parameters of working substance at a given point.

When determining time t during the transition from one established mode to another the parameters of these conditions and character of load on power shaft are assumed to be well-known.

The problem is solved with the help of the method of finite differences. One may assume that the entire transient process consists of a number of unsteady modes of the power plant, each of which is included in a certain small, finite interval of time Δt₁.

As the condition which disturbs the balanced state we assume the assigned law of change of gas temperature beyond the combustion chamber $T_3 = f(G_T)$, taking into account the dynamic characteristics of the control system. Here and below by a dash we have designated the relative magnitudes, i.e., $\bar{G}_T = G_T/G_{T0}$ — the flow rate ratio through the turbine in a given mode to the flow rate in the nominal mode. Numerical index at parameters $T$ and $p$ corresponds to the designations in Fig. 1.

For determining the parameters of each unsteady mode, it is necessary to solve a system of equations which characterizes the work of all basic ΠПУ elements.

**Equation of the flow rate for a turbine in general has the form**

$\dot{G}_t = f(p_3; p_4; T_3; n; p_i; n_i; T_i; n_i)$,

where $p_3$ and $p_4$ are the pressure before and beyond the turbine; $T_3$ is the temperature of gas before the turbine; $n$ is the number of revolutions.

For multistage turbines it is possible to use the Stodol equation, and for low stage turbines when determining flow rate, we can use the method of I. V. Kotlyar [1]. Using the flow rate equation, for the assigned law of change of gas temperature before the turbine $T_3 = f(G_T)$, it is possible to construct a characteristic of the turbine $p_3 = f(\bar{T}_T)$. 

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Equation of flow rate for a compressor is assumed to be assigned in the form of a universal characteristic.

**Equation of compression in the compressor**

\[
L_e = \frac{H_e}{\eta_e} = \frac{k - 1}{\epsilon} T_1 (\epsilon^{\frac{k-1}{\epsilon}} - 1),
\]

where \(L_e\) is the specific work of compression in the compressor; \(H_e\) is the isentropic heat drop in the compressor; \(T_1\) is the air temperature at the entrance to the compressor; \(\epsilon\) is the compression ratio; \(\eta_e\) is the internal efficiency of the compressor; \(k\) is the index of isentropy.

Air temperature beyond the compressor is

\[
T_3 = T_1 + \frac{H_e}{\epsilon \eta_e}.
\]

**Equation of expansion for a turbine is**

\[
L_T = H_T \eta_T = \frac{\eta_T c_p}{\epsilon} \left( 1 - \frac{1}{\phi} \right).
\]

where \(L_T\) is the specific work of expansion in the turbine; \(H_T\) is the isentropic heat drop in the turbine; \(\eta_T\) is the internal efficiency of the turbine; \(c_p\) is the specific heat capacity of gas; \(\phi\) is the expansion ratio.

Gas temperature beyond the turbine

\[
T_4 = T_3 - \frac{H_T}{c_p};
\]

\(H_e\) and \(H_T\) are determined by the thermal diagram [1].

In certain cases one should consider the nonstationary heat exchange in the flow-through part of the turbine.

The heat which is imparted by the gas to the metal (by the metal to the gas) during the time \(\Delta T_1\),

\[
\Delta Q = \frac{3 F \Delta e}{3600} (T_1 - T_{c_1}),
\]

where \(\alpha\) is the average (along the flow-through part) coefficient of heat transfer; \(F\) is the surface, washed by the gas; \(T_1\) and \(T_{c_1}\) are, correspondingly, the average temperatures of gas and the walls of the blade channels.
A change of gas temperature
\[ \Delta T_g = \frac{\Delta Q}{G_M \Delta T_g} . \]  

A change of temperature of the channel walls
\[ \Delta T_h = \frac{\Delta Q}{c_M} . \]  

where \( G_M \) and \( c_M \) are, correspondingly, the weight of metal, participating in the heat exchange, and its specific heat capacity.

As a result the temperature of gas beyond the turbine will be determined
\[ T'_i = T_i - \Delta T_g . \]  

A change of heat drop in the turbine as a result of the heat exchange of gas with metal will be proportional to the change of temperature
\[ \Delta H = \frac{H_i \Delta T_g}{T'_i} . \]  

The influence of thermal accumulation in the metal of the combustion chamber and the gas conduits can be disregarded due to the low coefficients of heat transfer and the low metal content of the combustion chamber. In many cases it's also possible not to consider the heat exchange in the flow-through part.

Equation of balance of the general compression ratio and expansion is
\[ \frac{\xi_{in}}{\xi_{out}} = \frac{1}{\xi_{out}} \]  

where \( \xi \) is the loss factor of pressure.

Equation of the speed of change of the quantity of gas in capacity. In case of the presence of regeneration and the gas-air capacities connected with it, flow rate through the turbine in unsteady modes will be different than the flow rate through the compressor. This difference in flow rates \( \Delta Q_i \); at the end of any assigned interval of time \( \Delta t_i \) will be directly proportional to the increase of pressure \( p \) and inversely proportional to the increase of gas temperature \( T \) in the tank. As a result,
\[ \Delta Q_i = \frac{R_{v_{i-1}} T_{i-1}}{\Delta t_i} \left( \frac{\Delta t_i}{\Delta Q_{i-1}} T_{i-1} - 1 \right) . \]  

where index \( i \) pertains to the desired mode, while \( (i-1) \) pertains to the preceding mode; \( R_v \) is time of capacity, i.e., the time in which the tank \( V \) will be emptied in the presence of flow rate \( Q_v \).
With a sufficient degree of accuracy, the temperature of gas in the tank can be determined as the temperature of air beyond the regenerator $T_1 = T_5$. Value $\varepsilon_1$ can be found by the trial-and-error method, by the characteristics of the compressor and the turbine.

Equation of heat exchange in the regenerator. As has been already noted by many authors [2, 4], the accumulation of heat in the metal of a regenerator renders a very significant influence on transitions of a MTY. This should be considered when determining the air temperature beyond regenerator $T_5$.

The heat which is imparted by the gas to the regenerator wall (by the wall to the gas),

$$\Delta Q_1 = \frac{a_r f_1 A_j}{3600} (T_r - T_1).$$

(12)

the heat, imparted by the regenerator wall to the air (air to wall),

$$\Delta Q_5 = \frac{a_p f_1 A_j}{3600} (T_r - T_5).$$

(13)

A change of thermal potential of the wall

$$\Delta Q_r = \Delta Q_5 - \Delta Q_1.$$ 

(14)

In equations (12) and (13), $a_r$ and $a_p$ are coefficients of heat transfer from the gas to the wall and from the wall to the air, which in the transient process can be, with a sufficient degree of accuracy, taken as depending only on the flow rates of gas and air, if we were to consider the low variability of coefficients of thermal conduction $\lambda$ and coefficients of viscosity $\mu$:

$$a_r = \alpha_{fg} G^m;$$

$$a_p = \alpha_{pg} G^n.$$  

(15)

where $m$ and $n$ are indices of degrees which enter into equation $Nu = a Re^m$ and depend on the type of heat exchanger; $F$ is the surface of the heat exchanger; $T_{CT}$, $T_r$, $T_B$ are averaged temperatures of the wall, gas and air.

Gas temperature beyond the regenerator is

$$T_1 = T_5 - \frac{\Delta Q_1}{C_{p,g} \varepsilon_1}.$$ 

(16)

Air temperature beyond the regenerator is

$$T_5 = T_1 + \frac{\Delta Q_5}{C_{p,a} \varepsilon_5}. $$ 

(17)

A change of wall temperature of the regenerator

$$\Delta T_r = \frac{\Delta Q_r}{C_u \varepsilon_r}.$$ 

(18)
where $G_M$ and $c_M$ are, correspondingly, the weight of metal, participating in the heat exchange, and its specific heat capacity.

Value $\Delta T_{CT}$ is usually low, and in many cases the temperature of the wall of the heat exchanger in transient processes can be considered as being constant.

**Equation of balance of powers of the compressor shaft.** Excess moment of the compressor shaft

$$\Delta M_c = M_{TH} - M_B - M_p$$

(19)

where $M_{TH}$, $M_B$, $M_p$ are moments of the turbine, compressor and friction.

A change of revolutions of the compressor shaft

$$\Delta n_c = \frac{30}{I_c} \Delta M_c \Delta t_c$$

(20)

where $I_c$ is the moment of inertia of the compressor shaft.

**Equation of balance of powers of the power shaft.** Excess moment of the power shaft

$$\Delta M_p = M_{TH} - M_B - M_p$$

(21)

A change of revolutions of the power shaft

$$\Delta n_p = \frac{30}{I_p} \Delta M_p \Delta t_p$$

(22)

where $I_p$ is the moment of inertia of the power shaft (including the reductor and the consumer), applied to revolutions of the power turbine. $M_B$ is the moment on the consumer shaft, which can be determined by characteristics, taking into account the delay of the control system. For example, for a variable pitch screw ($\text{BPW}$) it is sufficient to have dependences

$$M_B = f(\tilde{n}_e; \frac{H}{D}; v);$$

$$P_r = f(H; v);$$

$$R = f(v),$$

applied to revolutions of the power turbine, and a program of screw pitch variation.

Here $H/D$ is the pitch ratio; $v$ is the speed of vessel in knots; $R$ is water drag on vessel motion; $P_r$ is total thrust of the screws.
A change of speed of a vessel for the given interval of time \( \Delta t_1 \) will be determined from equation

\[
m \frac{dv}{dt} = p_0 - R.
\]

(23)

where \( m \) is vessel mass with additional mass of the water.

Internal efficiency of the compressor can be found by its characteristic.

Efficiency of the turbine can be calculated by well-known dependences \( \eta_{\text{turb}} = f(x) \) in the variable mode, if \( x = \frac{v}{c_0} \) is defined as the given ratio of speeds

\[
x = x_{0} \bar{\eta} \sqrt{\frac{\eta_{\text{turb}}}{\eta_{0}}}. \]

After determining all the parameters of the \( i \)-th mode, we should set as our goal the time interval \( \Delta t(i+1) \) and find parameters of the \( (i+1) \)-th mode. The calculation should be conducted up to achievement of the parameters of assigned conditions. An example of the calculation sequence is represented in Table 1, where two unsteady modes are determined during reception of a load for the \( \Gamma TV \) (Fig. 1), working on BPII. The law of temperature change before the turbine is accepted as

\[
T_s = \text{const} = 1098 \, \text{K},
\]

which indirectly considers the accumulation of heat in the regenerator.

According to calculated data it is possible to construct dependence \( \bar{\eta} = f(\tau) \) and the line of modes of the compressor in the transient process. Time of the transient process \( \tau \) will be determined by the sum of time intervals accepted in this calculation

\[
\tau = \sum \tau_i. \quad (24)
\]

The given method can be applied not only to the simplest, but also to any complicated layouts of \( \Gamma TV \). In the presence of layouts of \( \Gamma TV \) of several compression ratios, division of compression ratios by steps should be made by proceeding from the condition of their joint work [1]

\[
B_j = B_0 \frac{\eta_{(j-1)} \eta_{(j-1)}}{\eta_{(j-1)}}, \quad \sqrt{\frac{T_{\text{in}}}{T_{\text{out}}}}. \quad (25)
\]

where \( B \) is the parameter of flow rate, equal to \( \frac{G_{\text{in}}}{G_{\text{out}}} \sqrt{\frac{T_{\text{in}}}{T_{\text{out}}}} \); \( j \) is the ordinal number of the compression ratio.

Here

\[
\tau = \tau_1 \tau_2 \ldots. \quad (26)
\]
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<thead>
<tr>
<th>Designation of value</th>
<th>Formula</th>
<th>Dimension</th>
<th>Value</th>
</tr>
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<tbody>
<tr>
<td>Interval of time</td>
<td>( t - \text{we assign} )</td>
<td>( s )</td>
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<tr>
<td>Number of revolutions of compressor shaft</td>
<td>( n_0 - n_{0(t-1)} + 3n_{0(t-1)} )</td>
<td>( \text{r/min} )</td>
<td>2666</td>
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<tr>
<td>Relative number of revolutions of compressor shaft</td>
<td>( \frac{n_0}{n_{0}} )</td>
<td>( - )</td>
<td>0.444</td>
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<tr>
<td>Compression ratio</td>
<td>( \frac{\rho G}{R(T_r/T_i)} \left( \frac{T_i}{T_r} \right) )</td>
<td>( - )</td>
<td>1.65</td>
</tr>
<tr>
<td>Relative flow rate through the compressor</td>
<td>( \frac{G_i}{(n_i; \tilde{t})} )</td>
<td>( (\text{by characteristic}) )</td>
<td>0.2575</td>
</tr>
<tr>
<td>Relative flow rate through the turbine</td>
<td>( \frac{G_t}{G_i} - 3G )</td>
<td>( - )</td>
<td>0.25</td>
</tr>
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</table>
| Pressure beyond the 
\( \text{KB} \)    | \( \rho_s = \rho_s \tilde{t}(G_t) \)         | \( \text{kg/m}^2 \) | 1.55  |
| Compression ratio (check)                    | \( \frac{\rho_s}{\rho_i} \)                 | \( - \)   | 1.65  |
| Pressure beyond the 
\( \text{TB} \)          | \( \rho_2 = \sqrt{\rho_2^2 - 5G_2(\rho_2^2 - \rho_2^2)} \) | \( \text{kg/m}^2 \) | 1.094 |
| Expansion ratio of 
\( \text{TB} \)       | \( q_t = \frac{\rho_t}{\rho_2} \)              | \( - \)   | 1.51  |
| Heat drop of 
\( \text{TB} \)            | \( H_t = f(\tilde{t}; t_3) \)                | \( \text{keal/kg} \) | 29.5  |
| Coefficient                                | \( x_s = x_s \tilde{n}_s \frac{H_m}{H_t} \) | \( - \)   | 0.312 |
| Efficiency of 
\( \text{TB} \)          | \( \eta_t = f(x_t) \)                         | \( - \)   | 0.685 |
| Power of 
\( \text{TB} \)                  | \( N_t = 4.19 G_2 G_t H_t q_t \)             | \( \text{kw} \) | 1176  |
| Moment of 
\( \text{TB} \)                 | \( M_t = 973 N_t \tilde{n}_t \)              | \( \text{kg-m} \) | 430   |
| Efficiency of compressor                    | \( \eta_n = \text{by characteristic} \)     | \( - \)   | 0.78  |
| Heat drop of compressor                    | \( H_n = f(\tilde{t}; t_1) \)                | \( \text{keal/kg} \) | 10.6  |
| Air temperature beyond the compressor      | \( T_2 = \frac{T_2}{
\text{Cpt}^\text{K}} \)     | \( \text{K} \) | 344   |
| Power of the compressor                    | \( N_n = 4.19 G_2 G_n H_n \tilde{\eta}_n \) | \( \text{kw} \) | 834   |
| Moment of the compressor                   | \( M_n = 973 N_n \tilde{n}_n \)              | \( \text{kg-m} \) | 305   |
| Excess moment of the compressor shaft      | \( \Delta M_h = M_t - M_n - M_{tr} \)       | \( \text{kg-m} \) | 75    |
Table 1 (continued)

<table>
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<tr>
<th>Designation of value</th>
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<th>Dimension</th>
<th>Value</th>
</tr>
</thead>
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<tr>
<td>Change of revolutions of compressor shaft</td>
<td>$\Delta n_k = \frac{30}{n_c} \Delta M_c \Delta z$</td>
<td>r/min</td>
<td>276 298</td>
</tr>
<tr>
<td>Number of revolutions of power shaft</td>
<td>$n_c = n_{c(u-1)} - \Delta n_{c(u-1)}$</td>
<td>r/min</td>
<td>1700 1951</td>
</tr>
<tr>
<td>Relative number of revolutions of power shaft</td>
<td>$\tilde{n}<em>c = n_c / n</em>{c_0}$</td>
<td>-</td>
<td>0.34 0.39</td>
</tr>
<tr>
<td>Expansion ratio of THD</td>
<td>$\varphi_2 = \frac{P_2}{P_1}$</td>
<td>-</td>
<td>1.094 1.14</td>
</tr>
<tr>
<td>Temperature before THD</td>
<td>$T_3 = T_8 - \frac{H_3 n_8}{c_p}$</td>
<td>°K</td>
<td>1023   1012</td>
</tr>
<tr>
<td>Heat drop of THD</td>
<td>$H_z = f(\varphi_2, \varphi_3)$</td>
<td>kcal/kg</td>
<td>6      9</td>
</tr>
<tr>
<td>Coefficient</td>
<td>$x_8 = x_9 \tilde{n}_c \sqrt{ \frac{H_3}{\tilde{H}_2} }$</td>
<td>-</td>
<td>0.451 0.432</td>
</tr>
<tr>
<td>Efficiency of THD</td>
<td>$\eta_2 = f(x_9)$</td>
<td>-</td>
<td>0.83   0.82</td>
</tr>
<tr>
<td>Power of THD</td>
<td>$N_3 = 4.19 G_9 \tilde{G}_1 H_3 \varphi_2$</td>
<td>kw</td>
<td>268    477</td>
</tr>
<tr>
<td>Moment of THD</td>
<td>$I_3 = 973 H_3 \tilde{n}_c$</td>
<td>kg·m</td>
<td>140    238</td>
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<td>Speed of the vessel</td>
<td>$v = v_{(u-1)} + \Delta v_{(u-1)}$</td>
<td>km/ht</td>
<td>0.057 0.06</td>
</tr>
<tr>
<td>Relative screw pitch</td>
<td>$H/D$ - by program</td>
<td>-</td>
<td>0.3    0.3</td>
</tr>
<tr>
<td>Moment of the screw</td>
<td>$M_s = f \left( \tilde{n}_c; \frac{H}{D}; v \right)$</td>
<td>kg·m</td>
<td>40     52</td>
</tr>
<tr>
<td>Excess moment of power shaft</td>
<td>$\Delta M_s = M_s - M_s - M_{wp}$</td>
<td>-</td>
<td>64     141</td>
</tr>
<tr>
<td>Change of revolutions of power shaft</td>
<td>$\Delta n_c = \frac{30}{n_c} \Delta M_c \Delta z$</td>
<td>r/min</td>
<td>251    561</td>
</tr>
<tr>
<td>Thrust of the screw</td>
<td>$P_s = f \left( \frac{H}{D}; v \right)$</td>
<td>kPa</td>
<td>3000   5500</td>
</tr>
<tr>
<td>Water drag to vessel travel</td>
<td>$R = f(x)$</td>
<td>-</td>
<td>0      0</td>
</tr>
<tr>
<td>Change of vessel speed</td>
<td>$\Delta v = \frac{R_s - R_{(u-1)}}{0.515 \times 3t}$</td>
<td>knots</td>
<td>0.03   0.055</td>
</tr>
<tr>
<td>Temperature beyond the THD</td>
<td>$T_8 = T_3 - \frac{H_3 \varphi_2}{c_p}$</td>
<td>°K</td>
<td>1065   989</td>
</tr>
</tbody>
</table>
Table 1 (continued)

<table>
<thead>
<tr>
<th>Designation of value</th>
<th>Formula</th>
<th>Dimension</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature beyond the regenerator (^1)</td>
<td>(T_s = T_1 \cdot r(T_4 - T_3))</td>
<td>(°K)</td>
<td>807 790</td>
</tr>
<tr>
<td>Specific gravity of gas in the tank</td>
<td>(\gamma = 34/5 \frac{\rho}{T_4})</td>
<td>(\text{kg/m}^3)</td>
<td>0.698 0.77</td>
</tr>
<tr>
<td>Time of capacity (tank)</td>
<td>(R_p = \frac{R_{10T}}{G_{10}})</td>
<td>(\text{________})</td>
<td>3.07 3</td>
</tr>
</tbody>
</table>

\(^1\)In this case temperature \(T_s\) is determined by a simplified formula, since it is used only when determining \(\gamma\) and \(R_p\); \(r\) is the degree of regeneration.

In case of a \(\text{TVW}\) layout without regeneration, the calculation is simplified. For layouts with intermediate heat feed, it is necessary to assign a law of change of gas temperature in the intermediate combustion chamber.

**Peculiarities of calculating \(\text{TVW}\) layouts for load-drop.** In order to determine the dynamic stoking of \(\text{TVW}\) revolutions during full load-drop (for example, cutoff of a generator from the network), it is necessary to know the value of time lag of the control system \(\tau_p\) (the time which has passed from load-drop up to cessation of fuel feed). After cessation of fuel feed, pressure of gas before the turbine will be determined by the pressure of gas in the tank. For the case of isentropic expansion, the connection between the parameters of gas in the tank will be expressed as

\[
p_T^{-\alpha} = p_0^{-\alpha},
\]

where \(\gamma\) is the specific gravity of gas.

By differentiating this equation we will obtain

\[
\frac{d\rho}{\rho} = \gamma^{-1} - \delta.
\]

In the presence of small changes of \(p\), ratio

\[
\frac{\delta}{p} \approx 1.
\]

Thus,

\[
d_T = \frac{dp}{\rho}^{-\alpha}.
\]
A change of weight quantity of gas in the tank can be determined as

\[ dA = V dt, \]

or

\[ dA = (G_t - G_i) dt. \]

If we consider that in the normal mode \( G_t = G_{t0} \), then, crossing to finite increments, after certain transformations, we can determine the relative change of pressure in the tank as a result of expansion for the time interval \( \Delta \tau_1 \).

\[ \Delta p = \frac{\gamma}{\gamma - 1} (G_t - G_i). \]  (27)

One can assume that after cessation of fuel feed into the combustion chamber, gas temperature before turbine \( T_3 \) at any moment of time is equal to the exit temperature from the tank. In turn, gas temperature in the tank at the end of time interval \( \Delta \tau_1 \) will be determined by the temperature of displacement \( T_{CM} \) of gas which is anew proceeding to the tank from the regenerator, during the time \( \Delta \tau_1 \) with the remaining gas in the tank and the change of gas temperature as a result of expansion \( \Delta T \).

\[ T_t = T_{CM} - \Delta T. \]  (28)

If during interval of time \( \Delta \tau_1 \) into the tank quantity of gas \( G_K \Delta \tau_1 \) with temperature \( T_5 \) proceeded from the regenerator, and from the tank quantity of gas \( G_T \Delta \tau_1 \) with temperature \( T_3 \) departed into the turbine, then from the equation of heat balance, if one were to accept the process of mixing as being instantaneous, we can obtain the temperature of mixing in the tank

\[ T_{eq} = \frac{(R_g - 1) \frac{G_t G_i}{G_t + G_i} T_5 \Delta \tau_1}{(R_g - 1) G_t + G_i}. \]  (29)

A change of gas temperature in the tank, as a result of isentropic expansion, will be expressed by equation

\[ \bar{T} = \bar{p}^{\frac{1}{k - 1}}. \]

or

\[ d\bar{T} = \frac{k - 1}{k} \bar{p} \frac{1}{dp}. \]

Considering that in the presence of small changes of pressure value \( \bar{p} \approx 1 \), by crossing to finite increments, we can obtain

\[ \Delta \bar{T} = \frac{k - 1}{k} \Delta \bar{p}. \]  (30)
For any intermediate mode the capacity time is

\[ R_s = \frac{R_{st}}{G_{ts}} \]  \hspace{1cm} (31)

According to the data of detailed calculation for load-drop, it is possible to construct dependence \( \bar{R} = f(\tau) \) and to determine the value of dynamic stoking of turns \( \Delta n_{\text{max}} \).

It is possible to prove that for two \( \Gamma TV \), carried out according to the same layout, with identical gas parameters but having different power and different design execution (different moments of inertia of the rotors and number of revolutions) this equality is true

\[ -\frac{1}{n_0} = \bar{R} = \text{const} \]  \hspace{1cm} (32)

where \( \bar{R} \) is the parameter of pickup.

For two similar \( \Gamma TV \) of different power, dynamic stoking of revolutions will be identical

\[ \Delta n_{\text{max}} = \Delta n_{\text{max}}' \]  \hspace{1cm} (33)

Equations (32) and (33) permit (in necessary cases) producing a reduction of dynamic \( \Gamma TV \) characteristics.

**Literature**


Submitted by the Department of Industrial Power Plants