EXAMINATION OF THE IDLING AND STARTING OF LOW CAPACITY GAS TURBINES

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1. The steadily growing use of low capacity gas turbines is also partly due to their advantageous starting qualities. In most cases there is no need of warming up when starting: at the latest within 30—35 sec, the gas turbine reaches its max. output. In cold or by frost endangered weather these advantageous qualities of gas turbines become especially evident.

The idling and starting conditions of gas turbines might considerably be modified in case of altering the working cycle of the gas turbines. This study deals only with the most simple working cycle of two adiabatic curves and constant pressure, without heat exchanger. The calculations refer to a gas turbine with single-stage centrifugal compressor and separated axial power turbines. Evidently, the calculations and principles as presented in this study may be adequately applied on gas turbines of any working cycle and operating with any kind of machine parts.

The working cycle under examination is shown in Fig. 1. The indices and markings as used in this study are the following:

Indices:

"1" Before the compressor.
"2" After the compressor (before the combustion chamber).
"3" After the combustion chamber (before the turbine).
"4" After the turbine.

Markings:

T_3 ; t_3	Maximum permissible temperature before the turbine from the point of view of operational safety and duration of life
p_2/p_1	pressure ratio of the compressor
$\varkappa_l$: c_{pl}	adiabatic exponent and specific heat measured at constant pressure, resp. of the air passing through the compressor
$\varkappa_g$; c_{pg}	adiabatic exponent and specific heat of the overheated air passing through the turbine and of its combustion products deriving from the fuel
$\eta_{ad\,K}$ $\eta_{ad\,T}$	adiabatic efficiency of the compressor and the turbine resp.
u_k [m/sec]	periphery speed of the compressor rotor
d_k [m]	outer diameter of the compressor rotor
$\sigma = p_3/p_2$	pressure loss factor of the combustion chamber *i. e.* the, ratio of absolute pressures after and before the combustion chamber
G_l [kg/sec]	quantity of air passing through the turbine (irrespective of the loaded fuel)

N [Le]	output of the turbine at working speed
N_K; N_T; [Le]	output of the compressor and the turbine, resp.
N_h [Le]	useful power during the starting of the gas turbine
N_m [Le]	power output of the starting motor
t [sec]	acceleration time of the gas turbine
Θ [kg m sec²]	moment of inertia of the gas turbine rotor (compressor and turbine, this latter driving a compressor having a common shaft with the former)

2. A remarkable feature in the starting of the gas turbines is that their rotor must be accelerated to a definite number of revolutions and periphery speed, resp., by means of external power input.

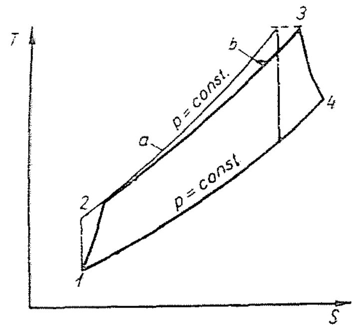

Fig. 1. TS diagram of the examined cycle. *a)* ideal cycle, *b)* actual (losing) cycle

To the given t_3 temperature before the turbine of each gas turbine belongs a minimum number of revolutions (pressure ratio), i. e. at a revolution low enough the gas turbine can be kept in operation only at a very high t_3 temperature. Practically, the gas turbine can, thus, keep itself rotating — also without external power input — only beyond a speed to be precisely determined by its characteristics.

The equality of the compression and expansion works gives a starting point to the study of the idle running conditions of the gas turbine. Accordingly, all the expansion work is used merely to cover the losses of the compression work and of the process. In an ideal case:

$$c_p\, T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{\varkappa-1}{\varkappa}} - 1\right] = c_p\, T_3 \left[1 - \frac{1}{\left(\frac{p_2}{p_1}\right)^{\frac{\varkappa-1}{\varkappa}}}\right]$$

therefrom:

$$T_3 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\varkappa-1}{\varkappa}}, \text{ i. e. } T_3 = T_2$$

Thus, as it was to be expected, the ideal idling gas turbine is able to keep itself rotating at any pressure ratio without external heat transfer ($T_3 = T_2$).

T_3 temperature necessary for idling, in case of actual gas, taking the two main losses of the gas turbine (compressor and turbine losses) into consideration :

$$T_3 = \frac{c_{pl}}{c_{pg}} \frac{T_1}{\eta_{adK}\,\eta_{adT}} \left(\frac{p_2}{p_1}\right)^{\frac{\varkappa_g-1}{\varkappa_g}} \frac{\left[\left(\frac{p_2}{p_1}\right)^{\frac{\varkappa_l-1}{\varkappa_l}} - 1\right]}{\left[\left(\frac{p_2}{p_1}\right)^{\frac{\varkappa_g-1}{\varkappa_g}} - 1\right]}$$

As a limit transition, T_3 temperature pertaining to $\frac{p_2}{p_1} = 1$:

$$T_3 = \frac{c_{pl}}{c_{pg}} \frac{T_1}{\eta_{adK}\,\eta_{adT}} \cdot \frac{\frac{\varkappa_l - 1}{\varkappa_l}}{\frac{\varkappa_g - 1}{\varkappa_g}}$$

Considering, therefore, the usual losses of gas turbines, neither in case of very low pressure conditions does the idling T_3 temperature increase, moreover, even in case of $\frac{p_2}{p_1} = 1$ definite, finite values are obtained. Of course, at a pressure ratio of unit the whole process looses its physical meaning and has a formal significance only.

Let us now consider the pressure loss of the combustion chamber, a loss very important from the point of view of idling conditions. In this case

$$T_3 = \frac{c_{pl}}{c_{pg}} \cdot \frac{T_1}{\eta_{adK}\,\eta_{adT}} \cdot \left(\frac{p_2}{p_1}\sigma\right)^{\frac{\varkappa_g-1}{\varkappa_g}} \frac{\left[\left(\frac{p_2}{p_1}\right)^{\frac{\varkappa_l-1}{\varkappa_l}} - 1\right]}{\left[\left(\frac{p_2}{p_1}\sigma\right)^{\frac{\varkappa_g-1}{\varkappa_g}} - 1\right]}$$

During reduction of the compressor pressure ratio (p_2/p_1) — due to the pressure loss of the combustion chamber — the expansion-indicating nominator [ ] as shown in brackets, proceeds quicker towards 0, than the numerator of the same formula, this latter, however, in consideration of the compression. As a consequence of the pressure losses the work of the turbine decreases quicker than that of the compressor, thus, under a certain pressure ratio, the equibalance cannot be realized but at the price of the increase of the T_3 temperature only. In case the pressure increase in the compressor is equal to the pressure loss in the combustion chamber, T_3 temperature rises beyond measure. From the above formula the value of T_3 is infinite if :

$$\left[\left(\frac{p_2}{p_1}\sigma\right)^{\frac{\varkappa_g-1}{\varkappa_g}} - 1\right] = 0$$

that means:

$$\left(\frac{p_2}{p_1}\right)_{T_3=\infty} = \frac{1}{\sigma}$$

Since at higher pressure conditions the value of T_3 begins rising again, it has to take up a minimum value at a certain pressure ratio. After having completed the extreme value calculation, we find that the grade of the idling minimum T_3 temperature is determined by all the losses together, whereas that of the pertaining pressure ratio, exclusively by the pressure loss factor.

$$\sigma = \left[\frac{\left(1+\frac{a}{b}\right)\left(\frac{p_2}{p_1}\right)^{a+b-1} - \left(\frac{p_2}{p_1}\right)^{b-1}}{\frac{a}{b}\left(\frac{p_2}{p_1}\right)^{a+2b-1}}\right]^{\frac{1}{b}}$$

where

$$a = \frac{\varkappa_l - 1}{\varkappa_l} \text{ and } b = \frac{\varkappa_g - 1}{\varkappa_g}$$

The results of the calculations are shown in Fig. 2; the starting values are the following: $T_1 = 288$ K°; $\eta_{adK} = 0.72$ $\eta_{adT} = 0.7$; $c_{pg} = 0.26 \frac{\text{kkal}}{\text{kg C°}}$ $c_{pl} = 0.24 \frac{\text{kkal}}{\text{kg C°}}$; $\varkappa_g = 1.35$; $\varkappa_l = 1.4$. In the present case the value of σ is constant, in reality it changes continuously in the function of the pressure ratio. Thus, the value of T_3 does not necessarily reach the infinite value, the strong increase of the t_3 temperature, however, takes place without fail in case of low pressure ratio. Among the starting values, the efficiency value of the turbine and of the compressor is intentionally somewhat unfavourable, though such efficiencies are very frequent with low capacity gas turbines (40—60 HP).

It can, therefore, be especially well seen that the pressure ratio of gas turbines (their revolution) cannot be reduced to any extent, only so far as the idle running temperature does not rise above the maximum temperature.

Fig. 3 shows the variation of the idle running T_3 temperature and of the combustion chamber pressure loss factor, in the function of the pressure ratio, of a test jet gear made by the Department of Gas Turbines, Polytechnic University, Budapest. As can be seen, the pressure loss factor is reduced by the value of the pressure ratio, the exact change of same may, however, be determined only experimentally.

At a value lower than the pressure ratio pertaining to the minimum idling temperature, the gas turbine can be kept at continuous running but only with the greatest precaution. The quicker change of the fuel quantity

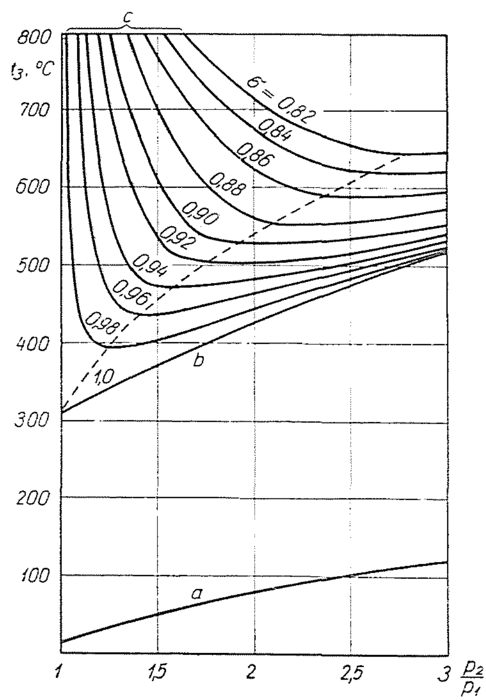

Fig. 2. Idling temperature of gas turbines in the function of the pressure ratio. *a)* ideal process, *b)* actual η_{adK} = const ; η_{adT} = const ; *c)* actual η_{adK} = const ; η_{adT} = const ; σ = const

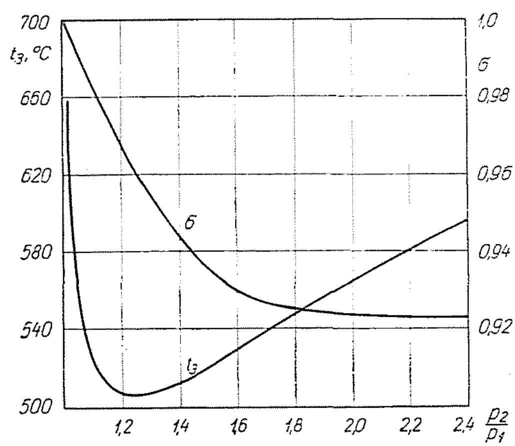

Fig. 3. Alteration of the t_3 temperature of the test gas turbine and of the σ pressure loss factor in the function of the pressure ratio

or modifications in any external condition entail respectively the separation of the compressor, the ceasing of equibalance between turbine and compressor and the stoppage of the gas turbine. Another interesting experience has been that, though at 1,02—1,03 very low pressure conditions the gas turbine was still operating, in such a working order, however, it could be accelerated but very slowly, by an extraordinarily cautious increase of the fuel quantity. By the quicker increase of fuel quantity, exactly as the above mentioned, ceasing of the thermic equibalance has been reached. During the experiments the maximum t_3 temperature was 700° C, anyway, the experiments fully confirmed the theoretical statements made up till now:

a) Depending on the characteristics of the gas turbine, there can always be found such a pressure ratio where the idling t_3 temperature is minimum.

b) This temperature grows particularly rapidly at low pressure conditions and, in an unfavourable case, at the value of $\frac{p_2}{p_1} = \frac{1}{\sigma}$ it can even be infinite.

c) At the minimum speed (pressure ratio) the gas turbine can still be operated, though its accelerating capacity is very small.

Most of the literatures on gas turbines only consider the efficiency of the turbine and of the compressor when determining the idling temperature. By taking into consideration merely these two losses, in case of appropriate heat transfer, the gas turbine should start by itself after moving the rotor, since according to curve *b)* of Fig. 2 an idling temperature of decreasing tendency belongs to low pressure conditions.

The procedure of starting can be examined in a concrete way only by considering the pressure loss arising in the individual machine parts. Thus there is a very tight, unambiguous connection between the idling temperature and the thermic process of the starting. Then starting, the gas turbine must be accelerated so long by external power input, until the t_3 temperature necessary to idling does not decrease to the permissible value at the given machine. The rate of the permissible t_3 temperature is mutually determined by the turbine blade material and its design, as well as by the required working time. At this minimum speed, though with a very high t_3 temperature, the gas turbine rotates by itself, without external assistance. At minimum speed the co-operation of the turbine, and of the compressor can be ensured only by means of the maximum t_3 temperature, thus the reserve power necessary for the acceleration can only be obtained by a further increase of the t_3 temperature. The excessive rising of temperature before the turbine endangers the integrity of the structural parts of the gas turbine and — as a consequence of the increased volume of gas streaming into the turbine — incidents of separation may appear in the compressor. For this reason gas turbines are not only accelerated up to the minimum revolution, but still higher by means of external power input, for the sake of ensuring a further acceleration capacity. According to theoretical calculations and to the practice (see later) with gas turbines having a single-stage centrifugal compressor, at a temperature of $t_3 = 750$ to $800\,°C$ the necessary minimum periphery speed (min. rev.) of the compressor impeller is about 50 to 60 m/sec, its idling periphery speed being abt. the double of same.

3. Accordingly, the procedure of starting is the following : To ensure adequate pulverization and heating possibilities, the gas turbine must be first accelerated up to a certain speed by external power input, without heating (coldly). This so-called ignition speed mainly depends on the applied heating system and on the layout of the combustion chamber. This revolution is at maximum the 14 to 17% of the working speed. Within the greater limits an appropriate heating in the combustion chamber is ensured by the heating system, so much more reduction of the cold acceleration is possible.

After reaching the ignition revolution, the turbine in an ever increasing extent is supplying useful power, due to the effect of the heat energy disengaged in the combustion chamber; this power partly covers the work requirement of the compressor. By exceeding the minimum speed, the gas turbine turns into a useful work-rendering machine, instead of being a power consumer. Thereafter the effective output of the starting motor and of the gas turbine are already together accelerating the motor, whereas, prior to reaching the minimum revolution, only the difference between the output of the starting motor and that of the cold gas turbine could accelerate the turbine. "Warm" acceleration with external power assistance continues up to such a speed, whereby the gas turbine gears up by itself, in a satisfactorily short time, to the working speed, depending solely on its reserve power (useful work). The procedure of starting may thus be divided into two main parts : the cold and the warm starting. These two periods cannot be sharply parted from each other, the more so, as at the beginning of the warm rotation there is not yet an even combustion, the still cold and the already warm parts of the air get into the turbine only very imperfectly mixed with each other.

4. From the point of view of practice, the following questions are raised : How much time is needed for the starting of a gas turbine of given output with a starting motor of known power, or vice-versa, what is the minimum starting-motor power required at a given gas turbine, which still ensures a satisfactorily short starting. At low capacity, auxiliary or reserve gas turbines the following question arises as a consequence of the efforts to obtain the most simple construction : what is the capacity of gas turbines that can still be started by hand power?

The solution of these tasks consists of two parts : First of all the power requirement or the reserve power of the turbine must be determined. Thereafter, with the knowledge of the starting output, the free output at disposal for the acceleration can be determined, where of, knowing the inertia moment of the gas turbine, the acceleration time can be calculated.

The starting conditions of the gas turbines are especially influenced by the following characteristics :

a) the narrowest outlet section (F) of the fixed row of blades of the compressor driving turbine, measured at the discharge edge of the fixed blade.

b) Reaction grade of the turbine (ϱ).

c) Maximum permissible temperature before the turbine (t_3; T_3). The change in the efficiency of the turbine and of the compressor is negligible in the period of starting, according to the above-mentioned experiment of the Department of Gas Turbines. *This means that in the case of a single-stage centrifugal compressor, the working condition of the compressor in the starting period can be satisfactorily calculated without compressor characteristics, exclusively on the basis of Euler's equation, of some special coefficients and of the approximate compressor efficiency.* This approximation is the more permissible, as the compressor characteristics do not include, in most of the cases, the starting condition.

The above characteristics (F, ϱ, T_3) are determining not only the process of starting but also are essentially fixing the output of the turbine, as the other data of the gas turbine $\left(\eta_{adK};\ \eta_{adT};\ \left(\frac{p_2}{p_1}\right)\right)$ etc. — especially in the case of low outputs — (50 to 200 HP) can be registered, by means of adequate approximation, as being identical. Thus, a direct comparison between the output data and the starting conditions of the gas turbines is possible. By means of Fig. 4 the output and the P_t thrust of a gas turbine with given t_3, ϱ and F values can be determined. In the course of calculation the more important characteristics, considered as being constant ones, were the following ; $\eta_{adK} = 0.77$; $\eta_{adT} = 0.79$; $\left(\frac{p_2}{p_1}\right) = 3.3$. Starting from quadrant No. 1 with the t_3 temperature, the reaction grade of the turbine, then, by means of the narrowest outlet section of the fixed row of blades the air absorption of the turbine (G_l) and, after a further intersection with t_3 temperature, its output can be determined. In case of a jet gear with gas turbine, the air absorption must be projected onto the dot line (—.—.—.), thereafter the thrust is obtained by means of the t_3 lines being in the fourth quadrant.

5. The more important steps to obtaining the time required for the cold acceleration, are the following :

a) In the function of the compressor periphery speed the pressure ratio and the specific power consumption (falling on one kg air) of the compressor can be determined.

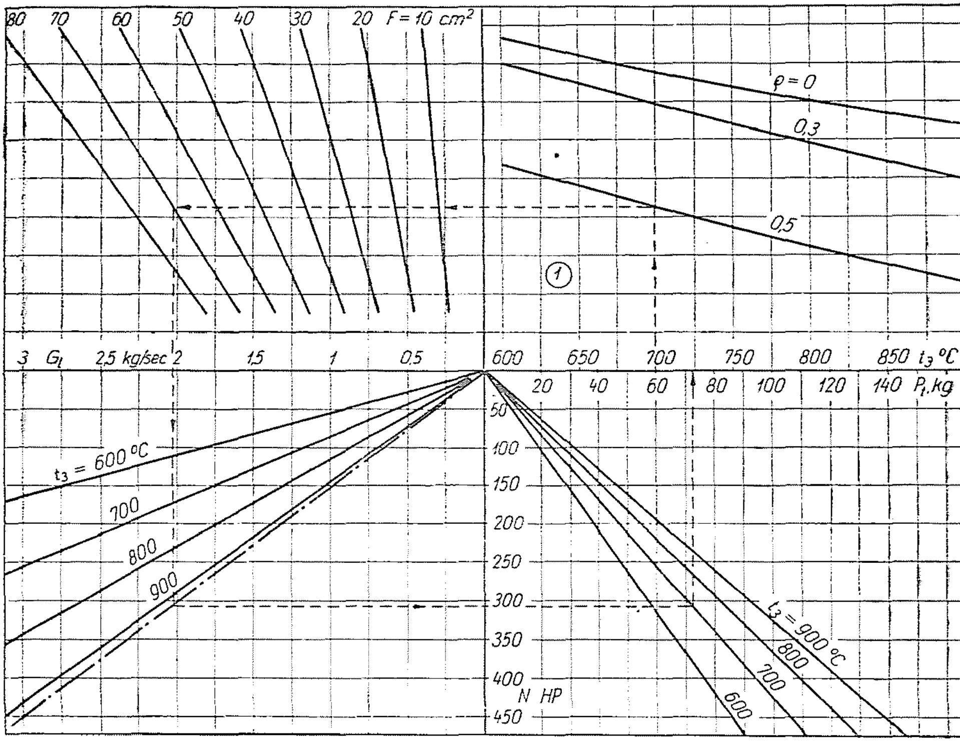

Fig. 4. Diagram for determining the output and thrust of gas turbines

b) By considering the pressure loss factor of the combustion chamber (in the present study the experimental data of the Department have been made use of) and the loss factor of the fixed blade, the air quantity streaming through the gas turbine can be determined :

$$G_l = \frac{F \cdot p_2 \cdot \sigma \sqrt{\frac{2g}{A} c_{pg}\, \varphi^2 (1-\varrho)\left[1-\left(\frac{p_4}{p_3}\right)^{\frac{\varkappa_g-1}{\varkappa_g}}\right]\left\{1-(1-\varrho)\left[1-\left(\frac{p_4}{p_3}\right)^{\frac{\varkappa_g-1}{\varkappa_g}}\right]\right\}^{\frac{\varkappa_g}{\varkappa_g-1}}}}{\left(\sqrt{T_3 R}\right)\left\{1-(1-\varrho)\,\varphi^2\left[1-\left(\frac{p_4}{p_5}\right)^{\frac{\varkappa_g-1}{\varkappa_g}}\right]\right\}}$$

c) With the knowledge of the air quantity streaming through the gas turbine, the power consumption of the compressor can be determined in the function of its periphery speed.

d) In the case of a cold acceleration, though the air as furthered by the compressor flows and expands resp. through the turbine, the speed triangles of the turbine, dimensioned for warm operation, changes according to Fig. 5. As can be seen, the turbine not only fails to render work, but is doing a considerable quantity of agitation work. According to the experiments of the Department, the approximate change of the $N_k + N_T/N_K$ ratio (i. e. the ratio of the full power requirement to that of the compressor) is the following in the function of the reaction grade :

ϱ	0	0,3	0,5
$N_k + N_t/N_k$	$\sim$ 2,1	$\sim$ 1,7	$\sim$ 1 4

Of course, this ratio can vary according to the layout of the turbine, these values are of informative character only. The relative amelioration at the increase of the reaction grade can be explained by the more regular streaming among the rotating blades of the reaction turbine. (The ratio pertaining to $\varrho = 0.5$ is an extrapoled value.)

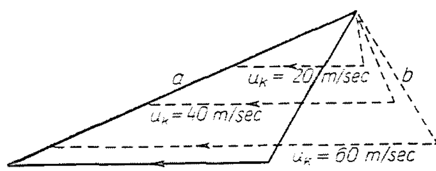

Fig. 5. Modification of the turbine speed triangles in case of cold and warm condition resp. *a)* warm condition, *b)* cold condition

After multiplying the power taken up by the compressor by modifying the ratio above, we obtain the power requirement of the cold rotation. The results of calculation are shown in Fig. 6. On calculating both the cold and the warm acceleration, the following starting values were taken into consideration : $\eta_{\text{adK}} = 0{,}72$; $\eta_{\text{adT}} = 0{,}73$. The pressure loss factor of the combustion chamber, according to the experiments of the Department, has been approached by quadratic parabole $\sigma = 1 - \frac{u^2\,[\text{m}^2/\text{sec}^2]}{7{,}2 \cdot 10^5}$ Similarly, also the pressure ratio change of the power turbine (blow pipe) has been approximated by quadratic parabole. Starting from the compressor periphery speed, by measuring ϱ and F, in cold condition furthered air quantity can be seen at the given periphery speed, then, intersectioning anew the value of u_k, the power requirement of the compressor and, again with the aid of ϱ, the full power requirement of the gas turbine can be determined. This output is, therefore, necessary to the even rotation at the u_K periphery speed as given to the gas turbine.

e) With the knowledge of the starting motor output and of the power requirement of the "cold" gas turbine, the Δt acceleration time, necessary

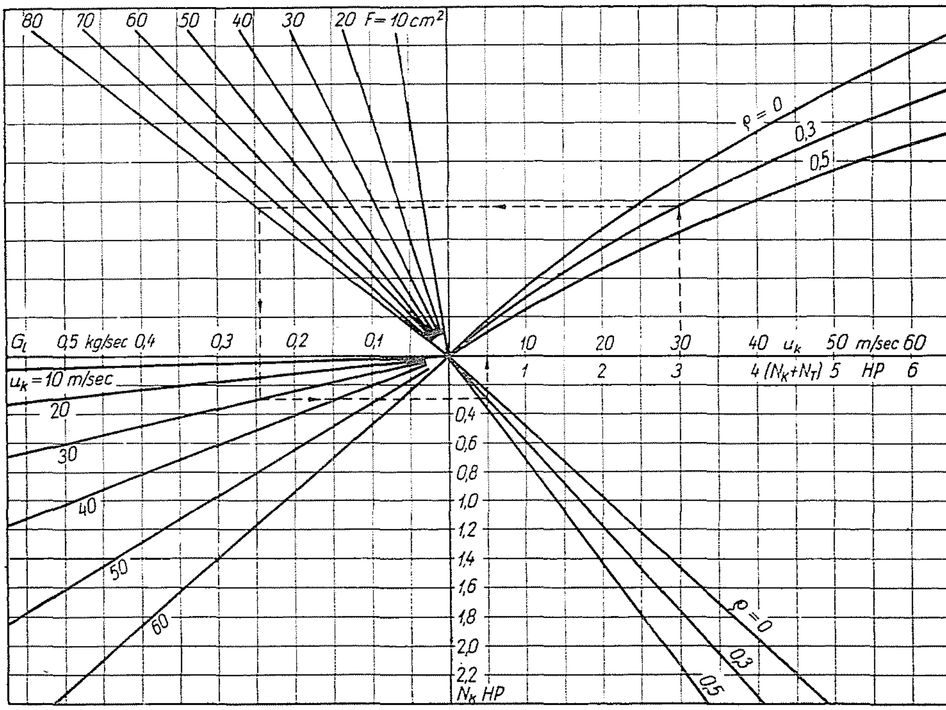

Fig. 6. Alteration of the starting of power requirement of a "cold" gas turbine

for reaching the periphery speed increase of a given Δu_k compressor can be determined.

$$\Delta t = 0{,}0533 \frac{\Theta \cdot u \cdot \Delta u}{d_k^2 [N_m - (N_k + N_T)]} \text{ [sec]}$$

In our calculations (Fig. 7) a periphery speed increase of $u_k = 10$ m/see has been considered. Accordingly, the time necessary for reaching, after starting any periphery speed, is :

$$t = \sum_{u=o}^{u=u} \Delta t$$

Obviously, with the increase of the periphery speed the acceleration time of the "cold" gas turbine rises more and more quicker, it is, therefore, advisable to begin at the possibly lowest periphery speed when firing, taking at the same time the given possibilities into consideration.

6. The acceleration time of the "warm" gas turbine can be calculated on the basis of similar principles with the difference that by now the turbine, too, is rendering work. Namely, due to the heat transfer, the speed of

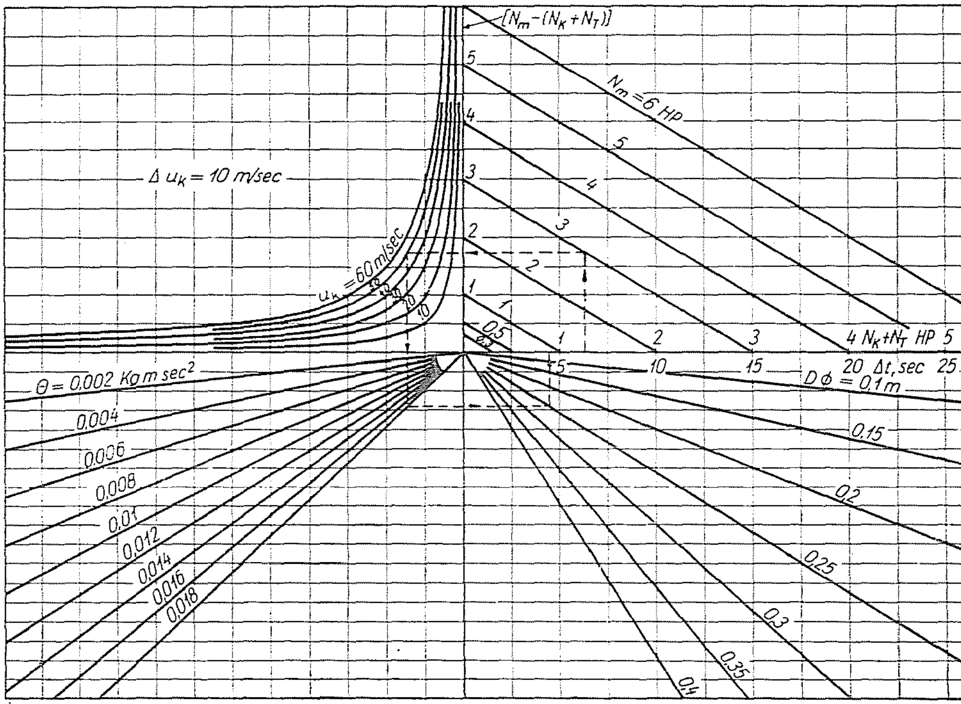

Fig. 7. For the determination of the speed-up time of the "cold" gas turbine

the gases has increased in the fixed blade of the turbine and the speed triangles of the warm acceleration are similar to that of the working speed triangles. At warm starting beyond a speed (min. rev.) depending on the t_3 temperature, the turbine work is higher than that of the compressor, thus the gas turbine turns into a useful work-rendering machine. This work is added to that of the motor, thus the acceleration conditions of the warm gas turbine become ever better, whilst speed is increasing.

With the aid of Fig. 8 the power requirement and the useful power, resp. of the gas turbine can be determined. Starting from the compressor periphery speed, in the fourth quadrant the change in the specific useful power (HP/kg) of the gas turbine can be seen, this being negative at the beginning (power requirement) then it becomes positive. The straight line starting from point O of the HP/kg scaled ordinate and the intersection points of the specific useful works are giving — in conformity with our statements up till now — the minimum revolutions. In accordance with the afore-said, with the increase of the t_3 temperature the minimum revolution is decreasing.

The output and the power requirement, resp. of the gas turbine can be determined if the specific output and air consumption are known. The

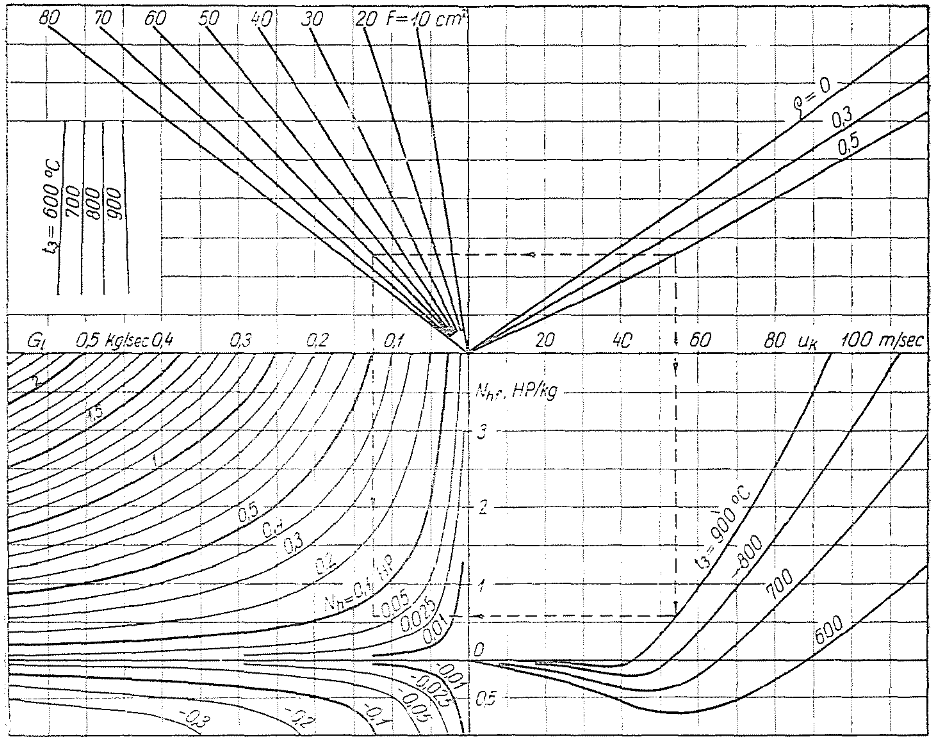

Fig. 8. Alteration on the starting of the useful power of the "warm" gas turbine

air consumption can also be determined with the aid of Fig. 7 in the function of the periphery speed with the intersection of ϱ and F being in the first and the second quadrant and with the consideration of the t_3 temperature correction. When taking the temperature correction in consideration, the ordinate of the second quadrant is parallel with the straight line of the respective temperature correction. The hyperboles of the third quadrant show the products of the conjugated values of the air consumption and of the specific output, *i. e.* the full output.

The acceleration time can be similarly calculated to that of the cold gas turbine, with the difference that now not only negative but also positive outputs are reckoned with. Here, too, the full acceleration time is given by the amount of the partial acceleration times. (Fig. 9.)

One of the characteristics of the warm starting is that depending on its dimensions and its t_3 temperature, a maximum power requirement pertains to each gas turbine. This maximum power requirement at the same time determines the minimum starting power by means of which the gas turbine, though theoretically during an infinitely long time, but can still be started.

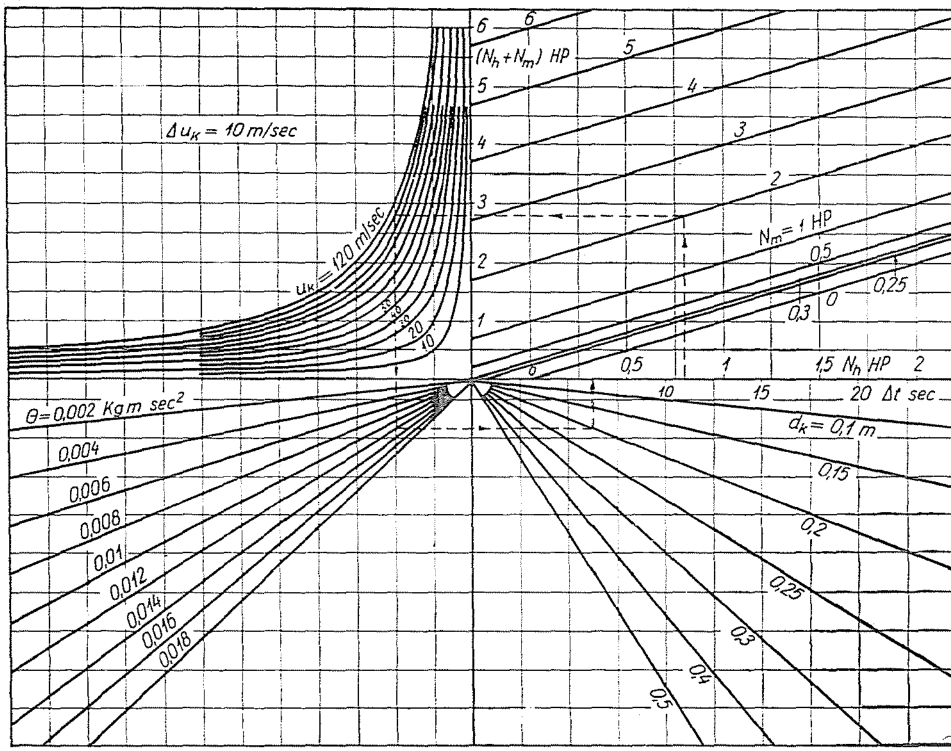

Fig. 9. For the determination of the speed-up time of the "warm" gas turbine

7. The full acceleration time of the gas turbine is the amount of the cold and warm acceleration times. The aim, as already mentioned, is that with the beginning of the firing at the poossible lowest revolution, the time of the cold acceleration is to be reduced, since thereby also the full acceleration time decreases. The ratio of the cold and warm acceleration times can be only approximately determined, as this in a high degree varies according to the types and the design.

The experiments effected in our Department in connection with the starting may serve as a base in this field. Fig. 10 shows, in the case of different starting motor outputs, the rated cold starting time of the Department's test gas turbine. Also the cold starting power requirement is shown in Fig. 10. The output of an up-to-date gas turbine of similar dimensions is abt. 250 HP.

Fig. 11 shows the rated warm starting time of the test gas turbine, its experimentally determined (measured) starting time and the change of the useful power arising at the warm starting. At the starting of the test gas turbine the switching in of the starting fuelpump and of the incentive

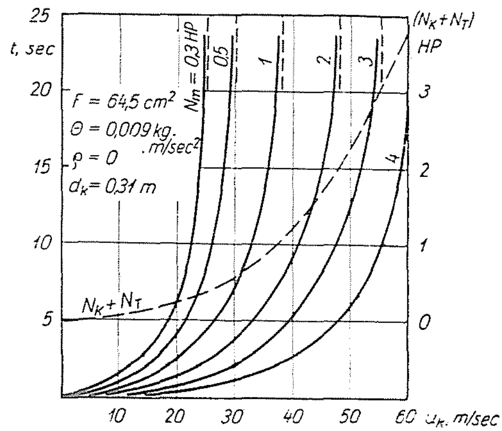

Fig. 10. Rated "cold" starting time of the test gas turbine

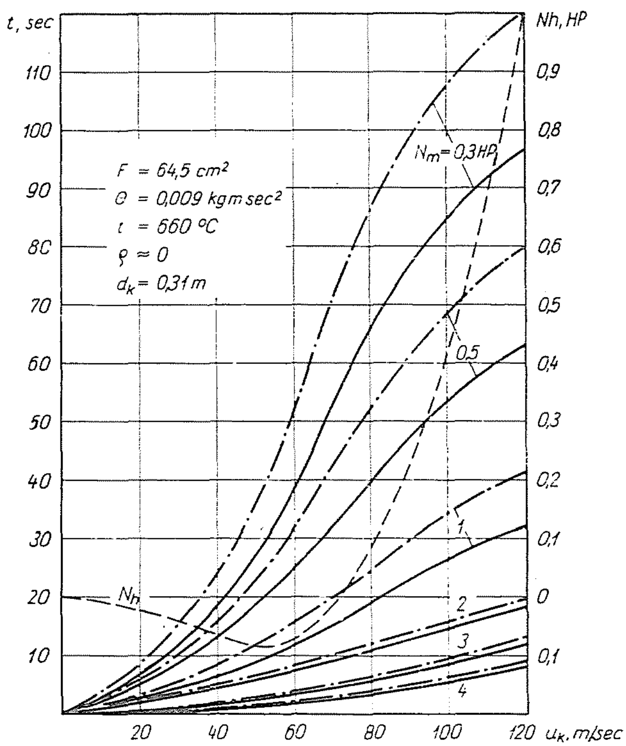

Fig. 11. "Warm" starting conditions of the test gas turbine ——— rated —·—·—· measured (experimental) — — — — useful power

has been effected at abt. 6 to 8 m/sec compressor periphery speed, whilst the main fuelpump has begun operation at abt. 12 to 14 m/sec periphery speed. The firing, corresponding to working conditions, had set in at abt. 30 to 40 m/sec speed.

Under such circumstances, when starting with a higher (2—3—4 HP) starting motor power, the measured starting time almost conforms with the time of the rated "warm" starting. At a lower (1 to 0,5 HP) and especially at 0,3 HP starting power, the measured starting time is substantially higher than the rated one. This is easy to percieve if we assume that the periode of the "cold" starting makes its starting time increasing effect felt especially at low starting powers. Also during the experiments we observed that at low starting power, especially with 30 to 50 m/sec compressor periphery speed, the acceleration of the gear was particularly difficult. If it succeeded in reaching a periphery speed of abt. 60 m/sec (approx. 3700 r. p. m.) the rotor of the turbine speeded up ever so vigorously. At any rate, the so-called "throttling down" experienced more than once at an exceedingly low starting power, took place always at 30 to 50 m/sec periphery speed.

The starting motor power of low capacity gas turbines realized according to the statistics of technical literature, is abt. 1% of the operating power. This ratio is somewhat increasing until 100—120 HP is reached, and rather falls at a power beyond the afore-mentioned one.

The usual acceleration time of a running low capacity gas turbine is 10 to 16 sec. As it will be discerned from the above experiments, at a starting-motor power of identical percentage the starting time is by some seconds higher; this can be explained in the first line with the lower temperature $t_3 = 660$ C°) before the turbine of the test machine. Also during operation not more than maximum $t_3 = 700$ °C has been permitted.

From these experiments it may be seen that the actual speed-up time of gas turbines as measured started with the abt. 1% of the operating power, is by approx. 5 to 10% longer than the rated "warm" acceleration time.

As conclusion, let us examine a problem of present interest: gas turbine of what output can still be started by man power? The computed starting power has been 0.25 HP. According to literature data, up to 1—1,5 minute, also man power suffices to produce 0,03 HP power, by adopting, however, 0,25 HP, the difference experimentally stated between the measured and the rated results is compensated. Practically, gas turbines of an output under 50 HP are rarely being built, therefore we, too, begin our experiment with a gas turbine of such a power. Starting values as considered when determining the dimension: $t_3 = 810°$ C, $\varrho = 0{,}3$; $p_2/p_1 = 3{,}3$. The Fig. below shows the statistical average of the diameter of gas turbine compressors and of the inertia moment of their rotors.

N_h	Le	50	75	100	125	150	200
Θ	kgmsec²	0,001	0.0015	0,0025	0,004	0,006	0,01
d_k	m	0,15	0,17	0,19	0,21	0,24	0,3

In Fig. 12 one can see the alteration of the starting time in the function of the periphery speed at different powers in case of $t_3 = 800°$ C, whereas in Fig. 13 the full starting time has been illustrated in the function of the

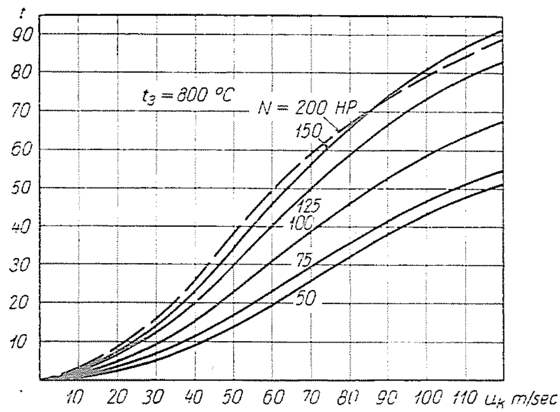

Fig. 12. Speed-up time of handpower started gas turbines of various outputs in the function of the compressor periphery speed

t_3 temperature. It is to be seen that an 50 HP gas turbine can still be started by man power, though taking a rather long time. Of course, by reducing the inertia moment of the rotor, this starting time may still be reduced, nevertheless, the calculations have proved that the starting of turbines of an output higher than 50 HP by man power, is not any more expedient. It can be seen from Fig. 13 that the increase of the t_3 temperature substantially reduces the starting time especially at a higher output. This can be explained by the fact that the change of the inner output resulting from the alteration of the t_3 temperature, strongly prevails only at high capacity gas turbines, beside the output of the starting motor. An interesting experience is that with such a consideration of ϱ and d the 200 HP gas turbine speeded up sooner than the 150 HP gas turbine. It can be seen from Fig. 12 that especially not long before reaching the idling speed that a sudden acceleration takes place. Accordingly, from between the increments of the useful surplus power and that of the inertia moment, the surplus power is more effective. Of course, by modifying the moment of inertia — within certain limits — another result can also be obtained, the tendency of the process, however, can well be seen. The starting time does not increase necessarily together with the increase of gas turbine dimensions and, generally, it even decreases, in the case of a carefully designed inertia rotor.

All that has been said may be applied also when the acceleration conditions of turbo chargers are examined. In such a case, however, when determining the free power of the turbo charger, the piston engine thermically connecting the turbine and the compressor must also be taken into consideration.

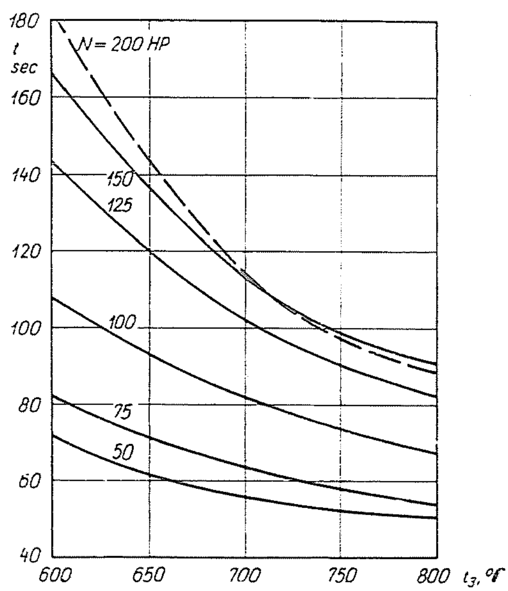

Fig. 13. The decrease of the starting time as a result of the t_3 temperature-increase of gas turbines started by hand power

Finally, let us sum up the more important results and conclusions referring to the starting of low capacity gas turbines.

a) For the right examination of the idling and starting problems also the pressure loss of the combustion chamber must be reckoned with.

b) There is a pressure ratio pertaining to each turbine, having a minimum t_3 idling temperature. This temperature grows especially quickly at pressure conditions pertaining to the minimum idling temperatures, in the case of lower pressure conditions.

c) Starting principle for gas turbines : to provide, by means of external power input, for such a working stage, in which the idling temperature does not exceed the permissible maximum temperature and the gas turbine disposes of further acceleration capacity resp.

d) The time of starting can be determined by the amount of the "cold" and "warm" starting times.

e) On gas turbines including single-stage centrifugal compressor, the process of both the warm and the cold starting can be satisfactorily computed, merely on the basis of Euler's equation, without compressor characteristics.

f) The measured starting time of gas turbines started with at least the 1% of the operating power is by only a few seconds longer than the rated "warm" starting time.

g) On starting by hand power the reaching of a sufficiently short (30—50 sec) starting time is already difficult in case of gas turbines having a capacity beyond 50—75 HP.

h) The starting time of gas turbines with a higher output generally decreases at an identical percentage of the starting power.

Summary

The present study discusses the problems of the starting of low capacity (50 to 200 HP) gas turbines. In the interest of clearing up theoretical connections, it examines the starting of gas turbines of the simplest cycle. According to experiments carried out at the Budapest Polytechnical University, Department of Gas Turbines, the process of starting can be computed with sufficient accuracy already when the approximate efficiency of the individual machine parts is known. This study deals with the theory of starting and particularly with the determination of the time necessary for reaching, from stillstand, the idling operating stage.

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