THE EFFECT OF WATER INGESTION ON THE OPERATION OF THE GAS TURBINE ENGINE

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Abstract

Gas turbine engines often operate in such meteorological circumstances, when the air entered the intake of the engine has significant amount of water content. The ingestion of water by gas turbine engines changes the process which takes places in the compressor, turbine, combustion chamber, turbine and nozzle. In unfavourable conditions an engine flameout is also possible.

In this paper, the effect of water ingestion on the engine operation is analysed. It explores the physical content of the processes occurred in the engine and deals with the mathematical modelling of them.

The method of examination is based on the application of the mathematical model of the engine. Finally the results of investigations are demonstrated.

Introduction

During training flights, at touch-and-go manoeuvre performed in heavy rain, one of the engines of a 2-engine aircraft (AN-26) - immediately after touching - stopped. After a short period of time at the same manoeuvre both engine of an another AN-26 aircraft also stopped. Restart of the engines a few minutes later was successful.

It was heavy raining and on the runway water was gathered in puddles within the region of touching. The engine stop occurred at the same part of the runway.

The paper analyses the processes caused by the water ingestion.

1. Analyses of processes occur in the engine

When the effects of the air water content on the processes taking place in the compressor is examined, here are two fundamental cases to be distinguished:

- water is present in vapour phase,
- water is present in liquid and vapour phase in the air to be ingested by the compressor.

The two cases induce processes essentially different from each other in the compressor, as well as in the power-plant elements placed behind it in the flow. It should be noted that prior to the inlet duct, the saturated or nearly saturated humid air free of liquid drops nevertheless may contain condensed drops of liquid in the cross-sectional area prior to compressor. This condensation process is initiated in the case of proper condensation cores (e.g. grains of dust), and due to the release of heat of vaporisation, it will increase the stagnation temperature prior to compressor thus reducing (with control rule n = const.) the corrected number of revolution of the first stages, since (in case gas constant R and specific humidity x are identical)

$$\bar{n}_{corr} = \bar{n}\sqrt{\frac{T_0}{T_1^*}} \qquad (1)$$

where - $\bar{n}$ - is the relative physical number of revolution, T_0 - is the temperature of correction, T_1^* - is the stagnation temperature at the inlet of the compressor or compressor section.

In case water is contained in vapour phase across the inlet cross-sectional area of the compressor, i.e. the relative humidity is $\varphi \leq 1$, the effect of water content will manifest itself through the change in the physical parameters of the humid air as mixture of gases [1],[5]. By applying the method of small deviations, from the change of adiabatic exponent $\delta\kappa$ and specific gas constant δR at $\lambda_{u1} = \frac{u_1}{c_{1cr}} = const$. (where u_1 and c_{1cr} – peripheral and critical velocities respectively) can be determined the changes in the corrected mass flow rate, pressure ratio and efficiency [1],[5].

With yielded values can be calculated on the basis of the following relationships, as described here:

The gas constant of humid air by:

$$R_{h.a} = \frac{R_a + x_1 R_v}{1 + x_1} \qquad (2)$$

its adiabatic exponent by:

$$\kappa = \frac{c_{pa} + x_1 c_{pv}}{c_{pa} + x_1 c_{pv} - R_a - x_1 R_v} \qquad (3)$$

where

R_a , and R_v - is the gas constant of dry air and water vapour,

c_{pa}, c_{pv}, c_{pw} - the specific heat at constant pressure of dry air, water vapour and water,

x , x_1 - the specific humidity and water content at the inlet, respectively, the formula of which are:

$$x = \frac{\varphi\, p_v}{p_1 - p_v} \qquad x_1 = \frac{\dot{m}_w}{\dot{m}_{air}} \qquad (4)$$

Here,

p_v - the partial pressure of water vapour, p_1 - pressure at the inlet of the compressor, φ - the relative humidity, $\dot{m}_w$ - the water mass flow rate, $\dot{m}_a$ - air mass flow rate.

The change in the corrected mass-flow rate on the basis of relationship:

$$\delta\left(\frac{\dot{m}\sqrt{T_1^*}}{p_1^*}\right) = -0.5\delta R - \left\{\frac{\kappa}{(\kappa-1)^2} ln\left(1 - \frac{\kappa-1}{\kappa+1}\lambda_1^2\right) + \right.$$
$$\left. + \frac{\frac{\kappa+1}{\kappa-1}\lambda_1^2 - 1}{2\left[\kappa + 1 - (\kappa-1)\lambda_1^2\right]}\right\}\delta\kappa$$

where $\lambda_1 = c_1/c_{1cr}$;

the change in pressure-ratio by:

$$\delta\pi_c^* = \left\{\frac{4\kappa^2}{(\kappa-1)^2(\kappa+1)}\frac{B}{2+B} - \right.$$
$$\left. - \frac{\kappa}{(\kappa-1)} ln(1+B) - \frac{(1+B)^{\frac{\kappa}{\kappa-1}} - \pi_c^*}{(\kappa+1)\pi_c^*}\right\}\delta\kappa$$

where

$$B = \left(\pi_c^{*\frac{\kappa-1}{\kappa}} - 1\right)\frac{1}{\eta_c^*}$$

the change in efficiency by:

$$\delta\eta_c^* = \left\{\frac{\pi_c^{*\frac{\kappa-1}{\kappa}}}{B\eta_c^*} ln\frac{\pi_c^{*\frac{1}{\kappa}}}{(1+B)^{\frac{\kappa}{\kappa-1}}} + \frac{4\kappa}{(\kappa^2-1)}\frac{1-\eta_c^*}{(2+B)\eta_c^*} - \right.$$
$$\left. - \frac{\kappa-1}{\kappa(\kappa+1)}\frac{(1+B)^{\frac{\kappa}{\kappa-1}} - \pi_c^*}{B\eta_c^*\pi_c^{*\frac{1}{\kappa}}}\right\}\delta\kappa$$

The deviations calculated according to the above relationships, in the case of compressor map having the form: $\pi_c^*, \eta_c^* = f[q(\lambda), \overline{n}_{corr}]$ should be interpreted with the corrected number of revolution:

$$\overline{n}_{corr} = \overline{n}\sqrt{\frac{T_0 R_a \frac{\kappa_a}{\kappa_a+1}}{T_1^* R_{h.a}\frac{\kappa_{h.a}}{\kappa_{h.a}+1}}} \qquad (5)$$

taking into consideration the variation of the new gas characteristics, too. (Here subscripts of *a* and *h.a*. symbolise parameters according to dry air and humid air respectively).

At low temperatures, p_v and x are small, so that according to our calculations, the effect of vapour-content on the compressor characteristics will be negligibly small [2],[5].

At higher temperatures (T > 300 K) and also with a relative humidity of about $\varphi = 1$, only the change of the gas constant will be considerable, the change of κ will be negligible due to the adjacent values of the adiabatic exponents of water vapour and dry air.

With all this taken into consideration, the corrected number of revolution will be reduced due to the increase in the humidity of inlet air. As a consequence of those said above, and taking into consideration relationship (5), both the mass of inlet air and the pressure ratio of compression decreases (downward motion on the operating-line can be observed).

In case water is contained in the air also in the form of water drops (e.g. preliminary condensation in the inlet duct, or take-off and landing in the rain, or flight at law altitude, respectively), it will evaporate partially in the compressor. Consequently, in the downstream stages of compressor, the partial pressure of vapour will increase relative to that of dry air, which involves additional reduction π_c^* and $\dot{m}_1$ (due to the change in composition). At the same time, due to the amount of the heat absorbed by evaporation, the temperature of air will also reduce.

This, in turn, results in the increase of the corrected number of revolutions of the compressor stages concerned (controlling rule is n = const.), which involves the increase of π_c^* . As a result of this double effect, according to our examinations, if a small amount of water is evaporated in the compressor, its pressure ratio will generally decrease, while in case of the evaporation of a large amount of water, its pressure ratio may increase. When analysing the process, taking place in the compressor, we should examine the very important circumstance that the volume of water will increase to its multiple value in the course of evaporation.

The evaporation process itself, due to its complex character, cannot be exactly calculated. Calculations can be carried out only on the basis of appropriate measurement results. However, on the basis of steam tables it can be determined in a simple way that - in our case - with a corresponding pressure of $p_2 \approx 8$ bar, the volume of the superheated steam will increase to the 280-290-fold value of water-volume, depending on temperature [2].

Due to the increase in volume involved by the phase-change of water, ever smaller amount of air can get into the burner with a constant fuel mass-flow rate. This increase of volume results in the increase in the volume flow-rate of the vapour-air mixture, and as a consequence, in the increase of axial velocity c_a in the final stages of the compressor. The change of the axial velocity, in this way, will modify the velocity triangles of the final stages so that the angle of attack will decrease, then the permeability of these stages will also decrease, which will result in the reduction of the volume of ingested air, the increase in the angle of attack of the first stages and the reduction in the pressure ratio of the compressor. As a consequence of this, the operating-point will move towards the surge line on the compressor map.

With the gas-turbine aircraft engines, a considerable part of the ingested water in liquid phase can be transferred into the burner, as our experiences show, and there it will be evaporated. This means that at the outlet cross-sectional area of the burner, the gas-steam mixture will contain more steam than at the inlet area, i.e. the outlet mass flow-rate of the gas-steam mixture will be higher than the inlet one.

In case the temperature of the burner is kept constant, the mass-flow rate of the gas-steam mixture to pass through the turbine will decrease due to the considerable increase of the volume flow-rate (thermal throttling).

(At a burner-temperature of $T_3 \approx 1060$ K, and at a pressure of ~ 8 bar [2], the volume of the superheated steam is about 400-fold value of the water volume).

This reduction in the mass flow-rate will, in turn, bring about a further reduction in the amount of ingested air, as well as in the pressure ratio of the compressor in the first stages, the increase in the angle of attack, the approach of the operating-point to the surge-line, and the enrichment of the fuel-air mixture.

With a determined air-water ratio, due to the over-enrichment of the combustion-mixture, the combustion will be extinct, or else the compressor will get into an unstable operational duty. It can be detected by the occurrence of one of the following phenomena as a function of the properties of the relevant engine, as well as the cause of defective operation:

- combustion will be extinct due to the over-enrichment of the combustion-mixture brought about by the abrupt reduction of the ingested air amount and the pressure ratio of the compressor
- the rpm cannot be increased due to the increase of the turbine inlet temperature
- vibration effecting the whole power-plant, and a high fluctuation in the pressure and in the amount of air will occur leading to flame-separation in the burner.

2. Processes taking place in the engine during an abrupt increase of power

The motion of equation of the rotor in the turbojet engine can be written in the following form:

$$P_T\eta_m - P_c - \Delta P = 4\pi^2 n\theta \frac{dn}{dt} \qquad (6)$$

in the turboprop it will be

$$P_T\eta_m - P_c - P_{pr} - \Delta P = 4\pi^2 n\theta \frac{dn}{dt} \qquad (7)$$

where

P_T - the power developed by the turbine overcoming the friction and driving the compressor, propeller and the auxiliary equipment, Θ - the inertial moment of the rotor, n- rotational speed.

In the steady-state operational duty equations are:

$$P_T\eta_m - P_c - \Delta P = 0 \qquad (8)$$

$$P_T\eta_m - P_c - P_{pr} - \Delta P = 0 \qquad (9)$$

In case we should like to increase the rpm or keep it constant because of the change in the angle of attack of the propeller, the torque of the turbine, or its power should be increased. The excess power

$$\delta P = P_T\eta_m - P_c - \Delta P \qquad (10)$$

or

$$\delta P = P_T\eta_m - P_c - P_{pr} - \Delta P \qquad (11)$$

required e.g. by the change in the angle of attack of the propeller can be attained through the stocking of excess fuel, - in case of constant cross-sectional areas of flow - through the increase of the turbine inlet temperature as compared to that of the steady-state operational duty.

The increase of the turbine inlet temperature will involve the modification of the operating-line plotted on the compressor performance. (It is assumed that the map of the compressor is unchanged during the transient processes). By increasing the turbine inlet temperature, the operating-line will move to the left, towards the greater T_3^* / T_0^* = const. lines, and as a result, towards the surge-line (Fig. 1). As a consequence of the changes in temperature and in the co-operational duty between the turbine and compressor, the pressure ratio of the compressor and the ingested amount of air will also be modified. The increase in temperature as compared to the previous steady-state phase will result in thermal throttling in the burner, and due to it, an instantaneous reduction in the ingested amount of air is involved, while the pressure ratio of the compressor will increase. The reduction of the amount of air and the excess fuel intake will bring about the enrichment of the combustion-mixture. Consequently, the limitation imposed on the speed-up, or the power-increase is marked by reaching the surge line, but in the course of the process, the enrichment of the combustion-mixture can also occur, which, in turn, can lead to the extinction of combustion.

We should like to note that this process - due to the shortage of the few data available – can not be calculated even with approximate accuracy.

3. Analyses of the joint effect of abrupt power-increase and the atmospheric conditions.

The most probable cause of the stalling of engines should be looked for - with the knowledge of circumstances - in the joint occurrence of the special (accelerating or power-increasing) operational duty, on the one hand, and the weather conditions prevailing during the flight training, on the other.

The detailed analysis of the two effects as performed separately from each other will confirm this assumption. Such situations may occur during a touch-and-go manoeuvre performed in heavy rain, and on the runway, water gathered in puddles within the region immediately prior to landing.

The liquid phase contained in the ingested air due to rain, as well as a part of amount of the liquid splashed up and entered the compressor at the touching the runway and during running was evaporated in the compressor, while the remainder (greater part) of it in the burner.

Both cases of evaporation involved the reduction of the air amount delivered by the compressor. In the final stages of the compressor the mass of steam-air mixture was increased due to evaporation, as well as the volume flow-rate was also increased to a considerable extent. Here the increase of the volume flow-rate resulted in the increase of axial velocity, and - at the same time - the reduction of the angle of attack. Thus, the blades became more and more the obstacle of flow, and this, in turn, led to the reduction of the air amount ingested into the compressor, and to the increase of the angle of attack in the first stages. The operating-point was shifted towards the surge-line, the pressure-ratio of the compressor was reduced.

The steam generated in the burner increased additionally the steam-content of the gas-steam mixture and its volume at the same time so that the given flow area of the turbine proved to be narrow to ensure the flow of steam through it. Due to it, the amount of ingested air continued to decrease, the angle of attack of the first compressor stage continued to increase, and the operating-point the more approached the surge-line. In the meanwhile, to cover the power-requirement raised by the increase of the angle of attack of the propeller, excess fuel had to be charged into the burner. This, in turn, resulted in an increase of temperature in the burner, thermal throttling and thus further decrease in the amount of ingested air. The operating-point approached the more the surge-line. At the same time, due to the thermal throttling and the steam volume developed in the course of the phase-change of water, the relatively smaller amount of air ingested into the burner formed an over-enriched mixture with an increasing amount of fuel, and as a consequence, combustion became extinct.

In the last analysis, we can state that the stalling of the engine was caused with great probability by the joint effect of the great amount of water ingested into the compressor, and the abrupt power-increase taking place at the same time. This is characterising feature of the given type of engines.

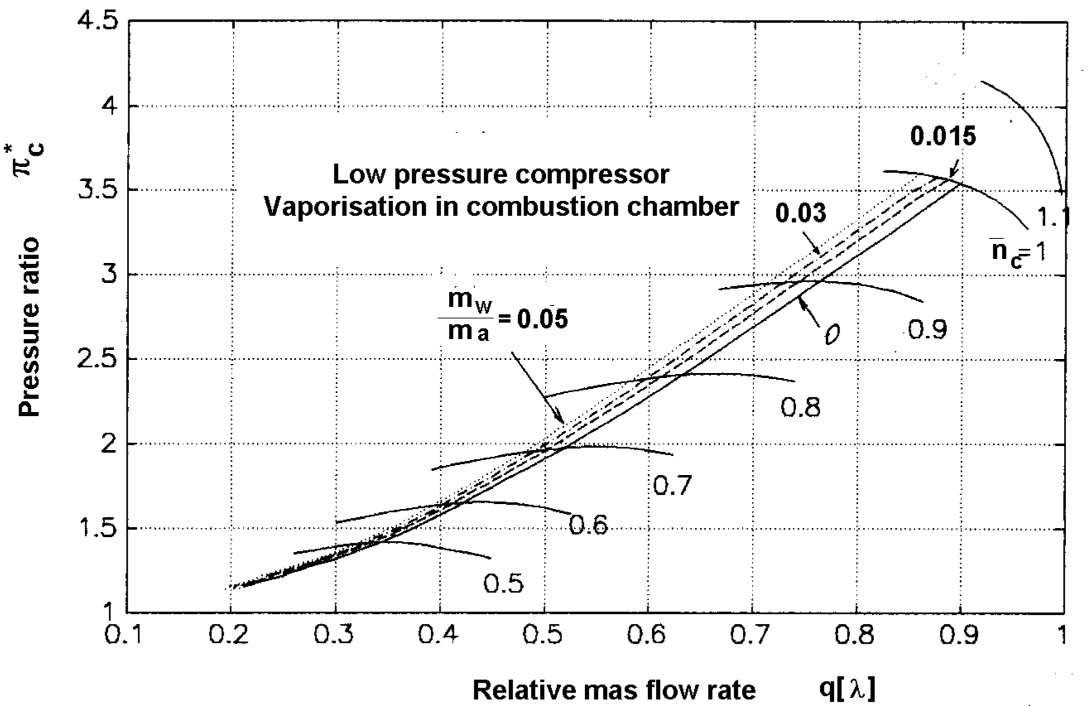
4. Modelling of processes

Analyses of processes occur in gas turbine engines - among other methods - is possible by mathematical model, described in [3], [4]. Model built up by this way is the basis of investigation of effect of water ingestion on the processes in the engine too. In mathematical model such operational modes we should take into account characteristic of the processes mentioned above

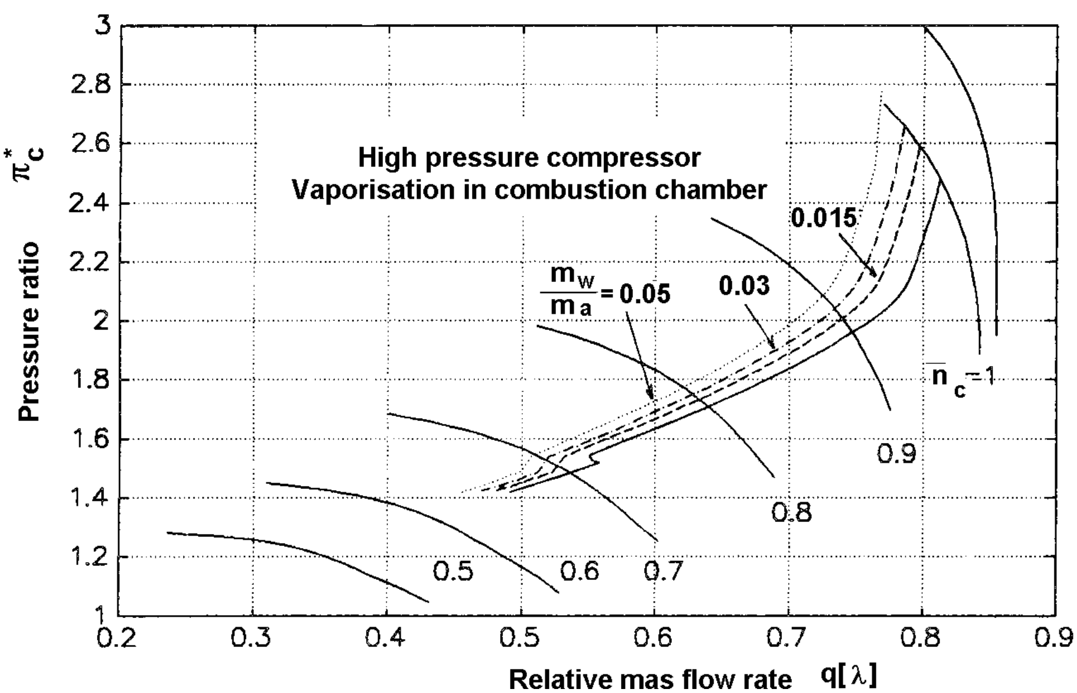
The process occurred in compressor as a result of vaporisation becomes a cooled polytropic one, instead of adiabatic. The water is considered as a separate parallel stream. It means, the specific work required for compression can be determined by equation

$$w_c = \frac{1}{1+x_{1in}}\left(c_p - c_n\right)\left(T_2^* - T_1^*\right) + \frac{z x_{1in}}{1+x_{1in}}\frac{u^2}{2} \tag{12}$$

where



4.5
4
3.5
3
2.5
2
1.5
1
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1
Pressure ratio
π*c
Low pressure compressor
Vaporisation in combustion chamber
0.015
0.03
mw/ma = 0.05
0
0.5
0.6
0.7
0.8
0.9
1.1
nc=1
Relative mas flow rate
q[λ]



3
2.8
2.6
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2
1.8
1.6
1.4
1.2
1
0.2
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0.5
0.6
0.7
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0.6
0.7
0.8
0.9
nc=1
Relative mas flow rate
q[λ]

Fig.1.

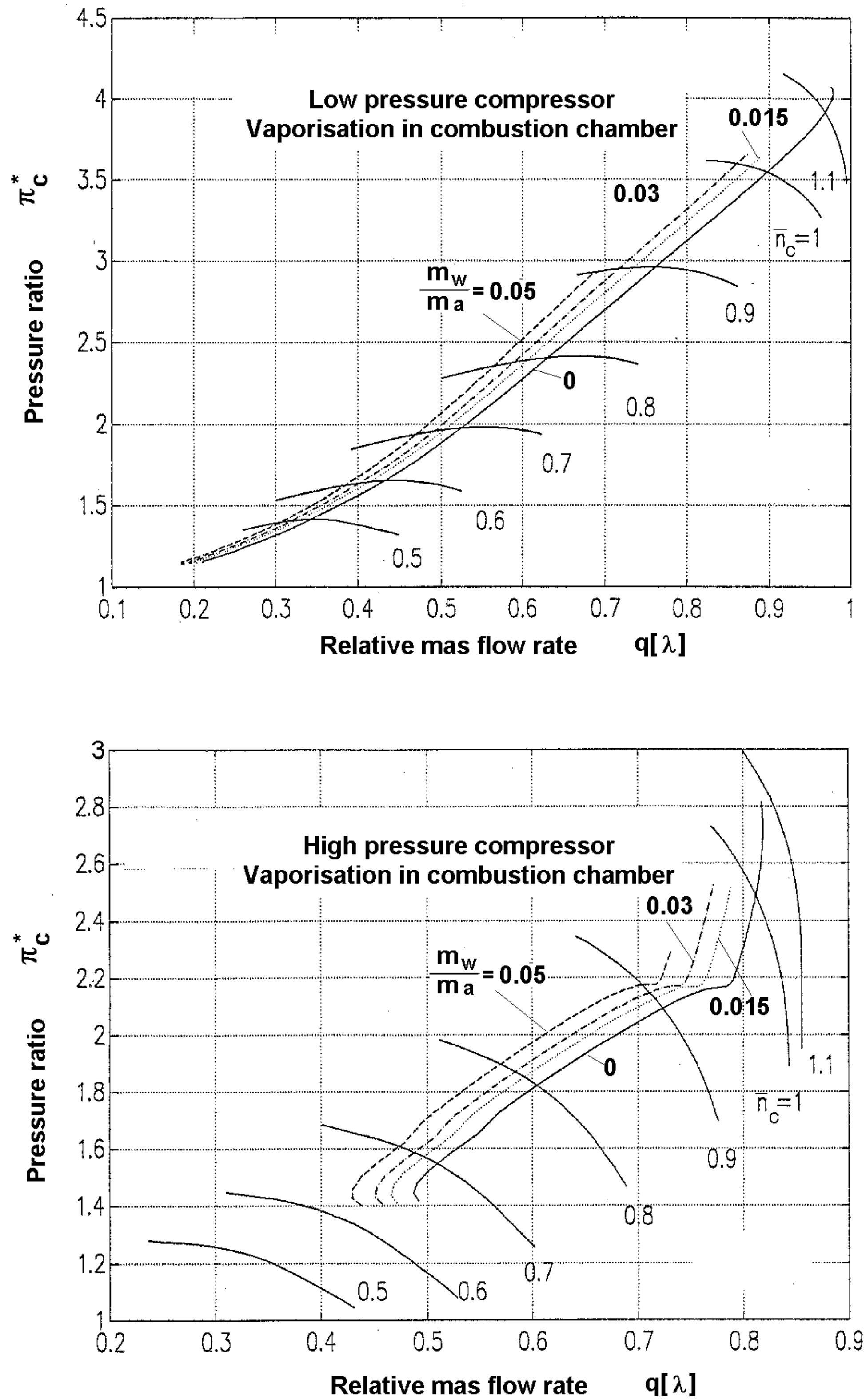
Low pressure compressor
Vaporisation in combustion chamber
Pressure ratio π_c^*
Relative mas flow rate q[λ]
$\frac{m_w}{m_a}$ = 0.05
0.03
0.015
0
$\bar{n}_c$=1
1.1
0.9
0.8
0.7
0.6
0.5
High pressure compressor
Vaporisation in combustion chamber
Pressure ratio π_c^*
Relative mas flow rate q[λ]
$\frac{m_w}{m_a}$ = 0.05
0.03
0.015
0
$\bar{n}_c$=1
1.1
0.9
0.8
0.7
0.6
0.5

Fig.2.

$$c_n = c_v \frac{n-\kappa}{n-1} \tag{13}$$

c_v - is the specific heat of air at constant volume; n - is the mean polytropic exponent of the process, κ - is the adiabatic exponent of air; u – is the tip peripheral velocity for the stage; z – number of stages where the water in liquid state [5].

$x_{1in} = \frac{\dot{m}_w}{\dot{m}_a}$ at the inlet of given compressor section.

These expressions should be applied to that section of multispool compressor, where the vaporisation takes place.

The mean polytropic exponent can be determined from the heat balance of ingested water and air in compressor. The heat absorbed by water formed from friction of compression process and polytropic heat.

$$RT_1^*\left(\pi_c^{*\frac{n_f-1}{n_f}}-1\right)\left(\frac{\kappa}{\kappa-1}-\frac{n_f}{n_f-1}\right)-$$
$$-c_nT_1^*\left(\pi_c^{*\frac{n-1}{n}}-1\right)=x_{1in}\left(i_{wout}^*-i_{win}^*\right) \tag{14}$$

where

$i_{wout}^*-i_{win}^*$ - the change in stagnation enthalpy of water in the given compressor section including phase change.

$$n_f = \frac{\ln\pi_c^*}{\ln\pi_c^*-\ln\left(1+\frac{\pi_c^{*\frac{\kappa-1}{\kappa}}}{\eta_s}\right)} \tag{15}$$

Enthalpy balance for the combustion chamber

$$c_{pa}T_2^*+fH_L\eta_b+x_{1in}\left(i_{win}^*-i_{wout}^*\right)=$$
$$=\left(1+x_{1in}+f\right)c_{pmix}T_3^* \tag{16}$$

where c_{pmix} - the isobaric specific heat of the air- vapour mixture.

The effect of increase in volume flow rate due to vaporisation, should be taken into account by proportional decrease of the cross-sectional areas downstream the place of vaporisation. $\alpha = \frac{A_{new}}{A_{old}} = \frac{\frac{\dot{m}_{air}RT}{p}+\dot{m}_{water}v_{water}}{\frac{\dot{m}_{air}RT}{p}+\dot{m}_{water}v_{vapour}}$

(17)

where

A_{old} ,A_{new} - original and new cross-sectional areas; T,p -static temperature and pressure of air at the given cross-sectional area; v_{vapour} ,v_{water} - specific volume of vapour and water respectively.

The results of calculations by mathematical model are demonstrated in Fig. 1 and Fig. 2. Fig.1 shows the twin-spool engine behaviour at steady operational modes in case of water ingestion. Diagrams in Fig.2 show a twin-spool engine accelerating process in case of increasing water content on original compressor maps.

The diagrams confirm the shifting of operating points and operating lines to the surge line.

Summary

The behaviour of the aircraft gas turbine engine changes, due to water ingestion by the engine. This paper examines the origin and characteristics of these changes. It can be stated, that while the ingestion of water in vapour phase hardly modifies, the increase of water content in liquid phase shifts operating line both in steady and transient duties to surge line. In case of acceleration or power increase of the engine the ingestion of water may cause the flame out of the combustion chamber.

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