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Gas path analysis based approach to evaluation of technical conditions of the passages of naval gas turbines in operation

SUMMARY

The paper presents general possibilities of identification of technical state of the flow elements in a marine gas turbine. The identification was set on the basis of relative change of gas thermodynamical parameters. The paper contains some results of the research on the flow elements of a marine gas turbine in operation. The research contained measurements of thermodynamical parameters in a marine gas turbine and endoscopic inspection of its passages.

INTRODUCTION

Gas turbine engines are used on various types of vessels: from cutters to aircraft carriers. At sea they operate in unique conditions, differing considerably from their use in aviation, which influence their operating characteristics as well as increase wear and tear and failure rate.

Gas turbines are particularly vulnerable because of: high air mass flow rate, high degree of engine mechanical load, complexity (and sometimes impossibility) of carrying out thorough surveys or replacements of sub-assemblies, necessity of high quality assurance and purity of fuel and oil.

The operation of marine engines is characterized by a relatively low average load and short maximum load operating time. For this reason the efficiency of marine gas turbines is considerably lower than predicted. The use of marine engines requires professional technical supervision which cannot be provided by crews of small vessels. Therefore, it was decided to support the crews of such vessels by the "Basic Diagnostic System of a Vessel's Turbine Engines (BDS)" created in the Naval Academy in Gdynia.

The use of a vessel's gas turbine engines for long time periods of heel and trim, and continual rocking of the ship, causes, in addition to normal wear and tear of bearings, considerable fouling of engine passages. This fouling causes changes in passage shape and introduces roughness elements. As a result, a change in kinematic, dynamic and thermodynamic characteristics takes place and a drop of engine efficiency is observed.In the multi-shaft engines, most frequently used on naval vessels, the influence of fouling on engine characteristics is a particularly complicated phenomenon. It is the result of interaction between the shafts and shift of the operating point of the high pressure and low pressure compressors from the optimal operating line in the direction of the surge line.

In order to carry out a thorough examination of operational technical conditions of passages a computational diagnostic model has been elaborated. The model contains thermodynamic and gas dynamic parameters of the working medium, engine parameters and details of entry conditions, control and outside disturbances. By determining changes of the parameters for a particular machine, it is possible to evaluate the condition of passages and pinpoint where the highest drop of efficiency takes place.

The model was validated against data taken from the turbine engines of COGAG type high speed vessels. The turbine operating conditions obtained by numerical evaluation were compared with those obtained by computerized data logger, and were further confirmed by endoscopic inspection of the passages which gave a clear indication of the remedial action required.

BACKGROUND

Use of propulsion turbines in adverse sea states requires their capacity for continuous work in conditions of heel, trim and continual motion of the ship. In these conditions, apart from adverse reactions acting on shaft bearings, there is also a danger that particulate impurities are admitted to the turbine through its intakes during intensive ship motion in waves. As a consequence of the irreversible process of normal wear and tear and the reversible process of fouling, the geometry of blades and the shape of passages are changed, and roughness of their surfaces increases.

The classification of impurities both according to their source and to their influence on passages is presented in Fig. 1. Classifying impurities according to their sources, it is possible to distinguish impurities from external sources (i.e. from outside the engine) and those from internal sources (i.e. from the engine itself). Moreover, the following consequences of the impurities, classified according to influence of the impurities on passages: erosive, corrosive, thermal, geometric and superficial, can be also distinguished.

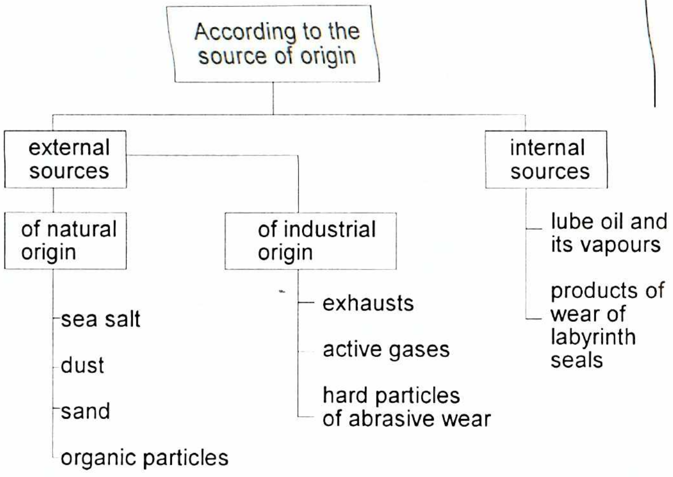


Fig. 1 Classification of possible sources of air impurities

As a result of passage fouling, a change in engine parameters is observed together with a drop in its efficiency. A computer simulation of this process was run for the complex multi-shaft free power turbine arrangement commonly found in naval vessels [2]. Many cases were analyzed such as fouling of the compressor and of the HP turbine. The example analyzed here assumes fouling of the blade passages of the LP compressor only, and modelling a fuel control system which adjusts fuel mass flow rate in order to maintain constant HP spool speed. In this situation a shift of the operating point from the optimal operating line in the direction of the surge line is observed on characteristics of both compressors (from point A to point B in Fig. 2).

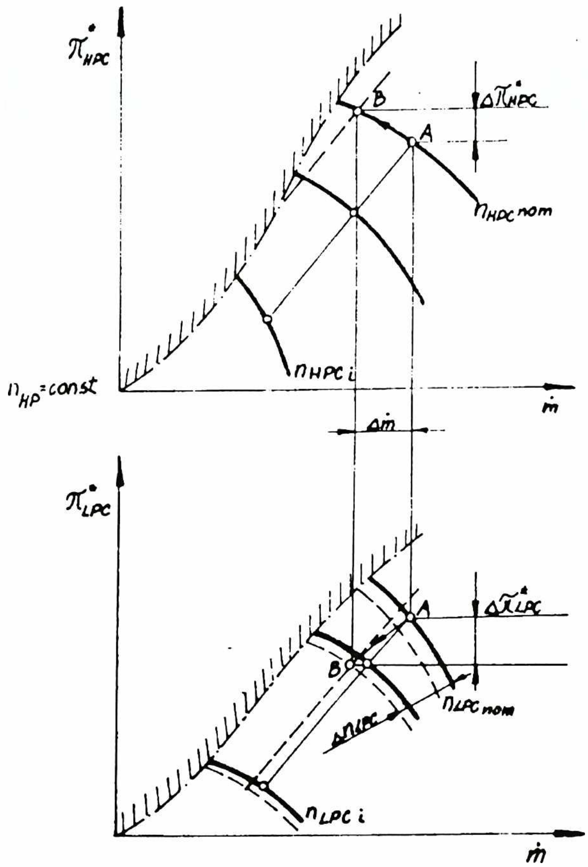
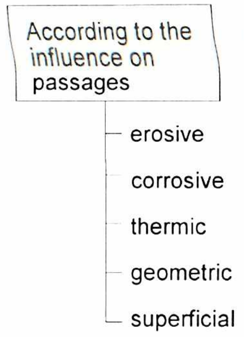


Fig. 2 Shift of operating points of the LP and HP compressors in a two-shaft engine with a free power turbine caused by fouling of LP compressor blade channels

One of the critical problems in the analysis is the determination and correct interpretation, both quantitatively and qualitatively, of the fouling state of the passages. It is very often necessary to differentiate changes in result of fouling from those due to secondary damage that produce the same net effect.



In order to make a thorough examination of the conditions of gas passages during operation, the passages are divided into component sections and energy balance carried out. Application of known dependencies between basic gas dynamic parameters of the engine, knowledge of relations between thermodynamic and geometric parameters, and knowledge of entry conditions, control mechanisms and outside disturbances enable the diagnosis of the actual operating state of the engine to be made together with identification of the point in the gas path where the drop of efficiency is taking place.

It is particularly important to carefully analyze the passages of the gas generator since this is the most critical part of the engine.

EXPERIMENTAL EXAMINATIONS

The analysis of changes in gas path parameters of an engine was carried out for a two-spool machine with free power turbine of a type most commonly used in naval propulsion systems. The tested engine was a part of COGAG propulsion system which is currently used on high speed vessels. Fig. 3 shows a schematic diagram of the engine with control sections marked.

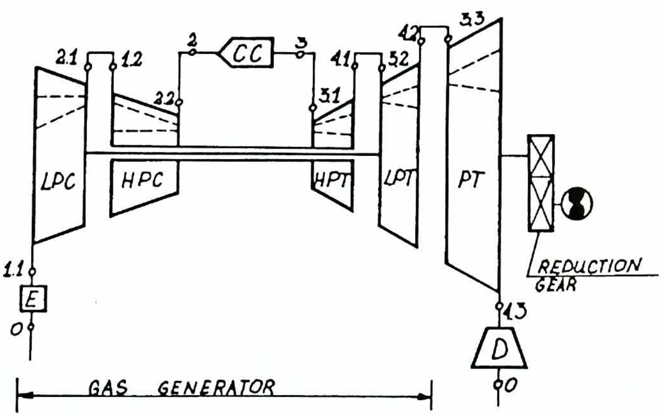


Fig. 3 Schematic diagram of a two-shaft engine with a free power turbine: E - intake dust extractor, D - exhaust diffuser manifold, LPC - low pressure compressor, HPC - high pressure compressor, HPT - high pressure turbine, LPT low pressure turbine, PT - power turbine, CC - combustor

Within the gas generator, a single stage LP turbine drives an eight stage LP compressor, and a single stage HP turbine drives a nine stage HP compressor. The turbo-annular combustor has ten flame tubes arranged around the HP compressor, thus necessitating a reversal of flow, but giving the benefit of a shorter engine. Unfortunately, this creates a restricted access to the HP compressor, making measurments difficult.

A three stage free power turbine and a single stage reverse turbine are driven by the gas generator without any mechanical connection. Tab. 1 shows basic technical data of the engine.

Experimental testing carried out on the engine, while the ship was operating, was designed to check the adequacy of the chosen diagnostic parameters as indicators of the technical condition of the gas path of a turbine in operation [3].

Tab. 1 Basic technical data of the engine

Range of operation	Power	Rotational speed of shafts			Specific fuel consumption	LP turbine outlet temp.	Compression ratio	Mass flow rate
		LP	HP	PT				
	kW	rpm			g/kWh	°K		kg/s
Maximum	8830	12000	14760	7100	266.5	1003	16.77	34.59
Nominal	7358	11320	14320	6400	274.0	945	14.81	32.27
0.8 Nom.	5886	10620	13880	5700	291.2	888	13.44	29.94
Idle	210	5720	10060	2100	1547.8	673	3.52	9.64

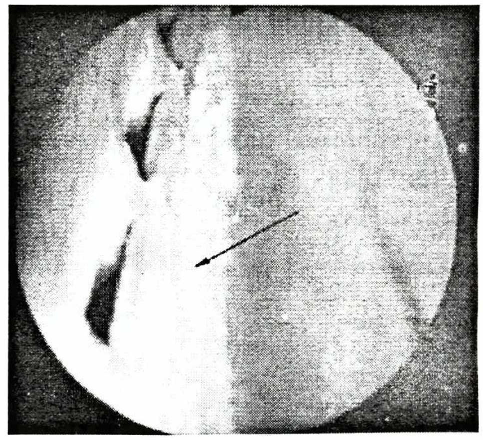
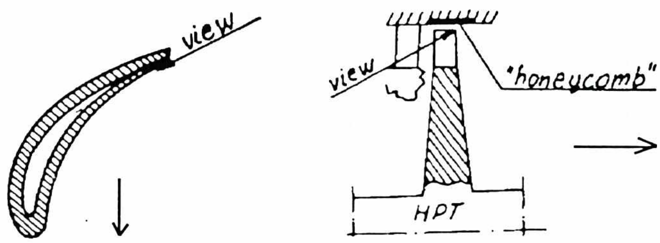


Fig. 5 Trailing edge of the rotor blade of the HP turbine (after washing)

To carry out the test program, it was necessary to put the ship to sea twice in each series of tests, and several cycles of tests were made.

Two different states of the engine were tested:
- state «1» - before washing, after approximately 500 hours of operation;
- state «0» - after washing [4], directly after cleaning the passages of compressors and turbines.

Results of endoscopic examination show the differences between these two states (Fig. 4 and 5) [5]. In Fig. 4, salt sludge on the blade surfaces and cooling passages of the rotor blade of the HP turbine is easily visible. In Fig. 5, traces of sludge are still visible, confirming the necessity of several wash repetitions in order to restore the engine to its former technical state. In Fig. 6, a defect on the surface of the HP turbine blades is visible. This was detected after approximately 1000 hours of operation.



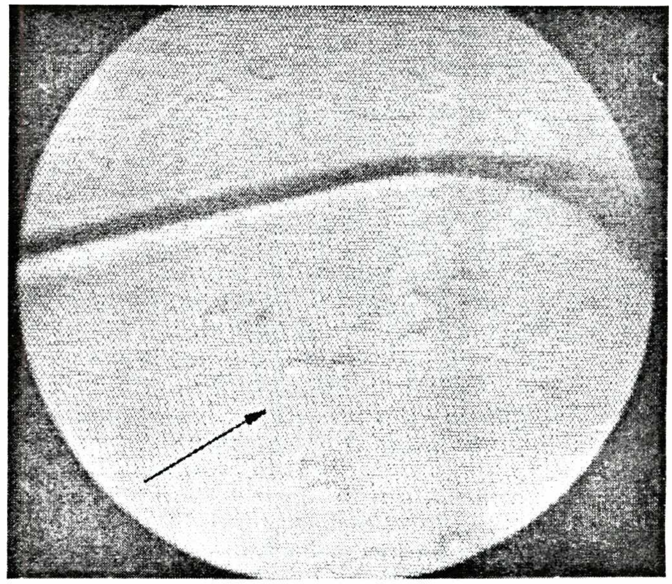


Fig. 6 Surface defects on blades of the HP turbine

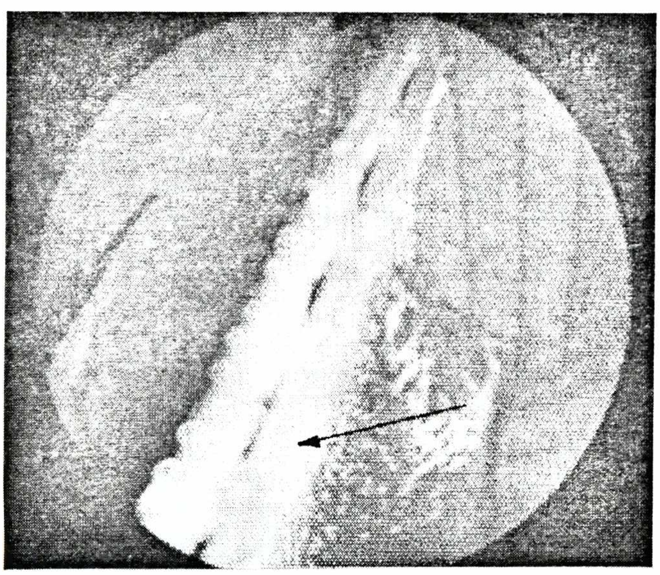


Fig. 4 Trailing edge of the rotor blade of the HP turbine (before washing)

The tests contained measurements of the following magnitudes:
- parameters of sea atmosphere (pressure,temperatue and humidity, respectively): p_0, T_0, ϕ_0 ,
- shaft speeds (LP, HP, and PT, respectively): n_{LP}, n_{HP}, n_{PT} ,
- angular position of the variable inlet of the first stage of the compressor: α_{KW} ,
- LP and HP compressor delivery pressures, respectively: p^*_{21}, p^*_{22} ,
- pressure in the control section 1.1 : p_{11st} ,
- total pressure loss in engine inlet ducts: Δp^*_{11} ,
- air temperature in engine inlet ducts: T^*_{11} ,
- exhaust gas temperature in the control section 4.2 : T^*_{42} ,
- fuel delivery pressure: p_f ,
- fuel temperature: T_f .

These measurements enabled to calculate changes of those parameters which describe the energy states along the passages. The changes were then used to estimate the quantitative and qualitative influence of engine condition (before and after washing) on the compression and expansion processes.

The examination of the conditions of engine passages showed, that in order to make a diagnosis, it is necessary to consider the following parameters:
- changes of compression ratio of the engine: $\delta\pi_C^*$,
- changes of compression ratio of LP and HP compressors,

respectively: $\delta\pi_{LPC}^{\ *}$, $\delta\pi_{HPC}^{\ *}$,
- changes of fuel consumption: $\delta\ G_e$,
- changes of specific fuel consumption: $\delta\left(\frac{G_e}{N_{HPT}}\right)$
- changes of fuel injector pressure: $\delta\, p_f$,
- changes of difference between HP and LP shaft speed: $\delta\ (\Delta\, n_{LP})$
- changes of temperature distribution of working medium at all i-points along passages: δT_i^* ,
- changes of distribution of the enthalpy flux at each control i-section of the engine [7]: $\delta\ \dot{J}_{it}^{\ *}$,
- changes of output power of the HP and LP turbine, respectively: $\delta\ N_{LPT}$, $\delta\ N_{HPT}$.

These parameters fully characterize the condition of gas generator passages. The percentage values of the changes are presented in Tab. 2. The values were calculated using the following formula:

$$\delta\ X = \Delta\, X / X_1 = (X_0 - X_1)/X_1$$

where:
0 and 1 - indices of the analyzed states of the engine,
X - diagnostic parameter of the engine.

Tab. 2 Values of relative changes of the engine parameters due to passage fouling (at full power)

Diagnostic parameter	Value %	Diagnostic parameter	Value %	Diagnostic parameter	Value %
$\delta\ N_{LPT}$	4.88	$\delta\ \dot{J}_{3t}^{\ *}$	3.78	$\delta\ \pi_{LPC}^{\ *}$	1.34
$\delta\ N_{HPT}$	4.47	$\delta\ \dot{J}_{31t}^{\ *}$	3.81	$\delta\ \pi_{HPC}^{\ *}$	1.42
$\delta\ \dot{J}_{11t}^{\ *}$	3.56	$\delta\ \dot{J}_{41t}^{\ *}$	3.39	$\delta\ G_e$	2.66
$\delta\ \dot{J}_{21t}^{\ *}$	4.80	$\delta\ \dot{J}_{32t}^{\ *}$	3.73	$\delta\ p_f$	1.03
$\delta\ \dot{J}_{12t}^{\ *}$	5.09	$\delta\ \dot{J}_{42t}^{\ *}$	3.03	$\delta\ T_{42}^{\ *}$	-0.53
$\delta\ \dot{J}_{22t}^{\ *}$	5.12	$\delta\ \dot{J}_{33t}^{\ *}$	3.02	$\delta\ (\Delta\, n_{LP})$	16.6
$\delta\ \dot{J}_{2t}^{\ *}$	5.18	$\delta\ \pi_C^{\ *}$	3.04	$\delta\left(\frac{G_e}{N_{HPT}}\right)$	-1.75

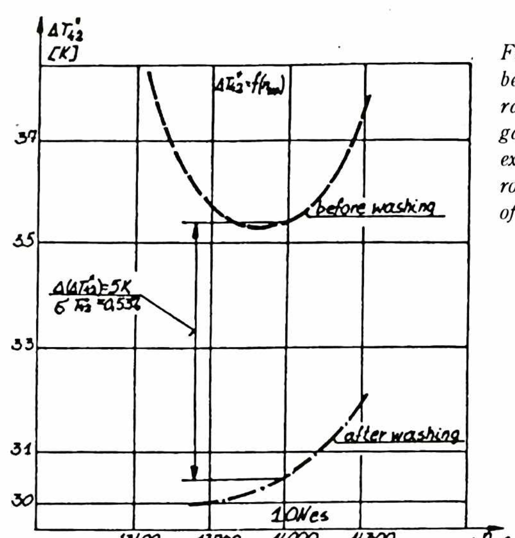


Fig. 7 Relation between temperature rise of gas generator exhaust and the rotational speed of the HP shaft

Analysis of the presented diagnostic parameters, when compared with internal endoscopic examinations, shows that changes of the condition of the passages, caused by fouling, have a measurable effect on the engine performance. The engine may be restored to its original state by washing the passages [3].

Changes of temperature rise at the gas generator outlet, plotted against rotational speed of the HP shaft, are presented in Fig. 7. The changes are due to fouling of passages. A rise in exhaust gas temperature indicates a drop of the engine efficiency caused by fouling.

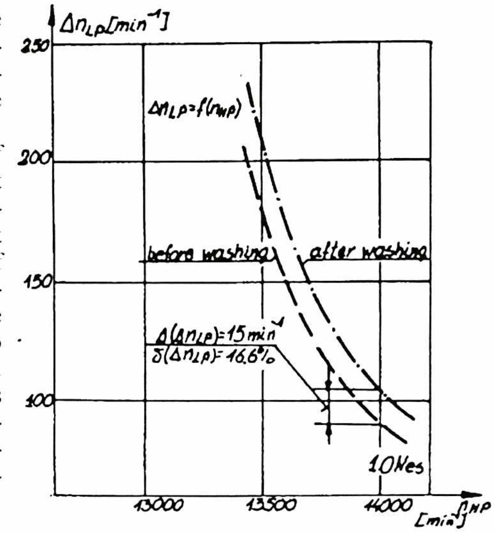


Fig. 8 Changes of differences between rotational speeds of the LP and HP shafts as a function of the HP shaft speed

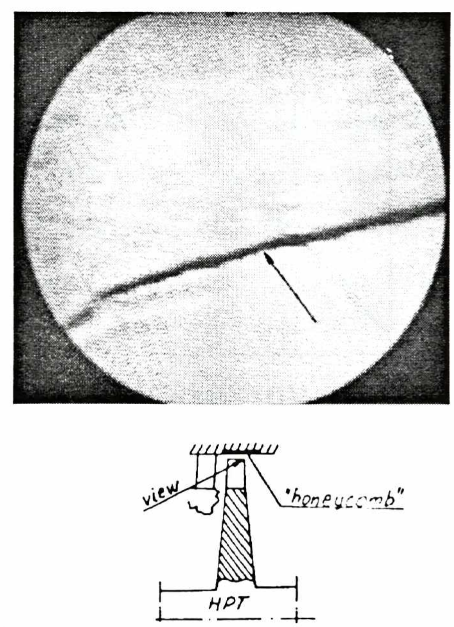


Fig. 9 The enlarged radial clearance of the HP turbine shaft resulting of its seizure

Changes of differences between LP and HP shaft speeds are shown in Fig. 8. As this parameter was positive (approx. 100 min^{-1}) for the rated rotational speed, additional parametric and endoscopic analyses of the HP turbine were carried out. During the endoscopic examination an enlarged radial clearance of the HP turbine shaft was found, shown in Fig. 9. Its existence had been indicated by the value of the measured parameter. The photograph taken during the endoscopic examination shows an enlarged radial clearance of the rotor blade of the HP turbine with clearly visible signs of seizing. The enlarged radial clearance had been caused by seizure of the shaft in the engine body.

Such situation could occur in the case of an emergency stoppage of the engine operating at full power (or close to it), followed by attempts to prematurely restart it (without allowing it to cool down). Thus the tip seals were immediately worn so enlarging the radial clearance. The efficiency of the HP turbine dropped considerably changing the distribution of enthalpy drop in the turbines. Sincb the operating point is controlled by the HP shaft speed, to achieve its set value, an apparent rise of LP shaft speed over its analytical value and a positive value of the change in shaft speed difference is needed.

The passages of the LP compresssor were fouled and therefore washing of them made the consequences of the change in the radial clearance more visible: a decrease in efficiency of the HP turbine, manifested as a positive change of the relative value of shaft speed difference $\delta\left(\Delta\ n_{LP}\right)$ amounting 16.6 %.

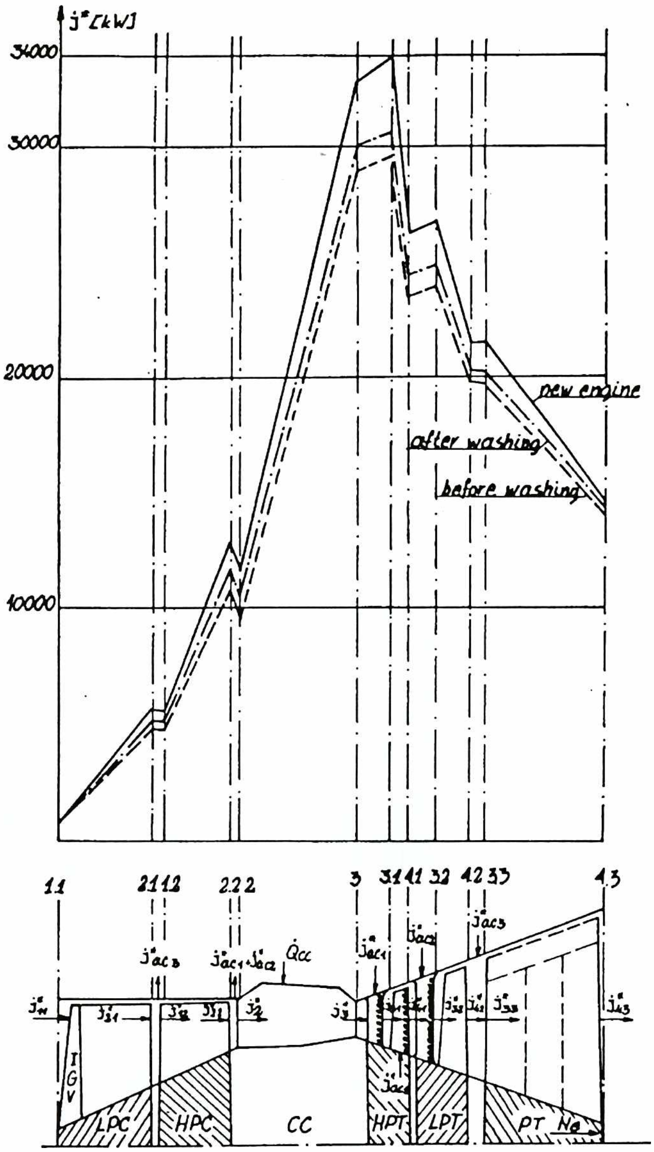


Fig. 10 Distribution of enthalpy flux in control sections of the engine

Examination of changes of enthalpy flux distribution along passages, calculated applying a specially written computer program and using collected information from a data logging system, makes it possible to identify qualitatively and quantitatively the operating technical conditions of passages.

Comparative analysis of the distribution of enthalpy flux in control sections of an engine in commision, an engine after washing, and a brand new engine confirms that the distribution of enthalpy drop was changed due to the enlarged radial clearance in the HP turbine. In particular, the difference between the enthalpy drop in the HP turbine and that in the LP turbine decreased (Fig. 8). The output governor of the engine is coupled with the HP shaft and maintains constant rotational speed of that shaft, therefore a lower enthalpy drop in the HP turbine causes a rise of it in the LP turbine. In result, the gas generator shaft speed difference changes as the speed of the LP shaft rises above the analytical value. The condition of the working gas in each subsequent component is disturbed leading to a decrease of compressor surge margin and a drop of engine efficiency.

CONCLUSIONS

Changes of enthalpy flux distribution together with changes of gas generator shaft speed differences give valuable quantitative and qualitative diagnostic information on engine fouling, which has been previously lacking. It also enables to early detect primary defects in passages, which may cause an engine failure.

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NOMENCLATURE

i (= 0, 1.1, 2.1, 1.2, 2.2, 2.3, 3.1, 4.1, 3.2, 4.2, 3.3, 4.3) - control sections of the passages;
CC - combustor;
D - exhaust diffuser manifold;
E - intake dust extractor;
f - fuel;
G - fuel consumption;
HPC - high pressure compressor;
HPT - high pressure turbine;
IGV - inlet guide vanes;
$\dot{J}$ - enthalpy flux;
$\dot{J}_{aci}$ - enthalpy flux removed from the compressor part and delivered to the turbine part of the system for its cooling;
LPC - low pressure compressor;
LPT - low pressure turbine;
$\dot{m}$ - mass flow rate;
n - shaft rotational speed;
N - power;
PT - power turbine;
p, p_0 - pressure;
Q - heat flow;
T, T_0 - temperature;
δ, Δ - changes of a magnitude;
ϕ, ϕ_0 - relative humidity;
π - compression ratio.

Note:
Some of the above explained symbols when used as indices denote relation to the designated quantity, system component or section.