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ДИПЛОМНА РОБОТА
(ПОЯСНЮВАЛЬНА ЗАПИСКА)

ВИПУСКНИКА ОСВІТНЬОГО СТУПЕНЯ
“БАКАЛАВР”

Тема: ***Газотурбінна установка простого циклу з розробкою системи вібраційного контролю***

Виконав: _____________ Акван Омар (Сирія)

Керівник: канд. техн. наук, доцент ________________ Якушенко О.С.

Нормоконтролер з ЄСКД : __________________________/ __________________/

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MINISTRY OF EDUCATION AND SCIENCE OF UKRAINE

NATIONAL AVIATION UNIVERSITY

To move all this gas across the country we needed a source of power which is gas turbines are used in the gas transport industry as compressor stations, gas turbines are used in different power outputs and designs and types.

Gas turbines just like any engine they can have faults during the working hours of the engine so for that a monitoring system is required to make sure the engine is running smooth and without any issues.

Such monitoring system is used in gas turbines to check the vibrations of the engines as they are pulse of the engine with vibration monitoring the system can detect any defects or faults and also can predict maintenance time also, the system has alarm trigger when vibration sensors detect any abnormal change in the vibrations, and if the change exceeds the critical point the system can shut down the engine automatically.

Vibration monitoring systems are very important especially for Gas compressor stations as the engines are in very sensitive environment any faults in the engine could lead to a critical damage and losses in the engine and the gas transportation system.

Aim of research is to develop an efficient and accurate vibration monitoring system and chose and mount vibration sensors for the designed GTP to ensure accurate reading of vibration values

Tasks of research are

1. To calculate the operating characteristics of the designed GTP in different working regimes.

2. To calculate the stresses of forces on the turbine elements (Blades, Disc) of the designed GTP.

3. To choose efficient and accurate vibration sensors as part of the vibration monitoring system for the designed GTP.

1 GAS DYNAMIC CALCULATION OF GTU

The purpose of the gas-dynamic calculation is to determine the diametrical dimensions in the characteristic sections of the flowing part of the installation, the number of rotors and their rotational frequencies, the number of stages of the compressor and the turbine, the distribution of the compression (expansion) between cascades and stages, and the specification of the parameters of the GTP.

As output data, the results of the thermodynamic calculation of the actual installation cycle are used.

In the course of the gas - dynamic calculation, based on statistical data of the executed designs of the GTP, the axial component of the air velocity at the inlet to the compressor and the speed of the outer diameter of the first wheel of the compressor unit is selected. These parameters to a large extent determine the diameter of the GTP, the number of stages of the compressor and turbine, as well as the axial dimensions and weight of the installation.

1.1 Design of the plant

The GTP is made according to a three-shaft scheme with an axial fourteen-stage compressor, an intermediate casing, an annular combustion chamber, two single-stage compressor turbines, a three-stage power turbine and an exhaust device.

The engine power on the shaft of the power turbine when it is operating in nominal mode is 6300 kW at an air inlet temperature of the engine +15 ^{o}C and atmospheric pressure 101325 Pa.

The engine compressor is an axial, two-stage, consists of a supersonic low pressure compressor (LPC) and a subsonic high pressure compressor (HPC) and is connected to the input device by a spacer.

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Author							
Chief	Yakushenko O.S.						
Adviser							
N.Supervis							
D.Chairm.	Kulyk M.S.						

LPC - seven-stage consists of a front housing, a rotor and a stator. In the front case, an input guide vane of LPC and an assembly of the front ball bearing of the LPC rotor are mounted. Blades of guide vanes made with pins. The input edges of the seven racks of the front housing, the vanes are heated with hot air (if necessary); the input edge of one rack of the front housing is constantly heated with hot oil, merging from the front support of the low pressure switch.

The compressor rotor is a disk-drum design, the disks are connected to the front and rear shafts by bolts, the working blades of the first two steps are connected to the disk by a dovetail type shank, the LPC rotor is connected to the rotor of the LPT by means of splines and forms a low pressure rotor.

HPC - seven-stages, consists of an input guide vane, a rotor, a stator. Guide vanes are mounted on an intermediate housing and have the ability to rotate the blades for tuning the engine on the stand.

HPC rotor - drum-disk construction, it consists of a welded, bolted to the front shaft, the HPC rotor is connected to the high-pressure turbine by means of coupling bolts and forms a high-pressure rotor mounted on two supports. The front ball bearing is mounted in an elastic support mounted in an intermediate housing.

To ensure the stable operation of the engine at start-up and in low modes, the low pressure and high pressure valves have air bypass valves for the third stage of low pressure and the fourth stage of high pressure.

Combustion chamber- annular type combustion chamber consists of an external casing, a diffuser with an HPC straightener, a flame tube, 24 nozzles, 2 igniters, a fuel manifold with fuel supply pipes. Two-channel gas nozzles are used on the combustion chamber.

Engine turbine - axial, jet, four-stage, and turbine consists of a single-stage high-pressure turbine, a single-stage low-pressure turbine, and a three-stage power turbine.

1.2 Calculation of working body parameters

Principal scheme of gas-turbine power plant (GTPP) with power turbine is shown on Fig.1.1, where there are cross sections in which working body parameters are determined, all parameters were calculated using Mathcad and with Help of Methodical Guide.[1],[2]

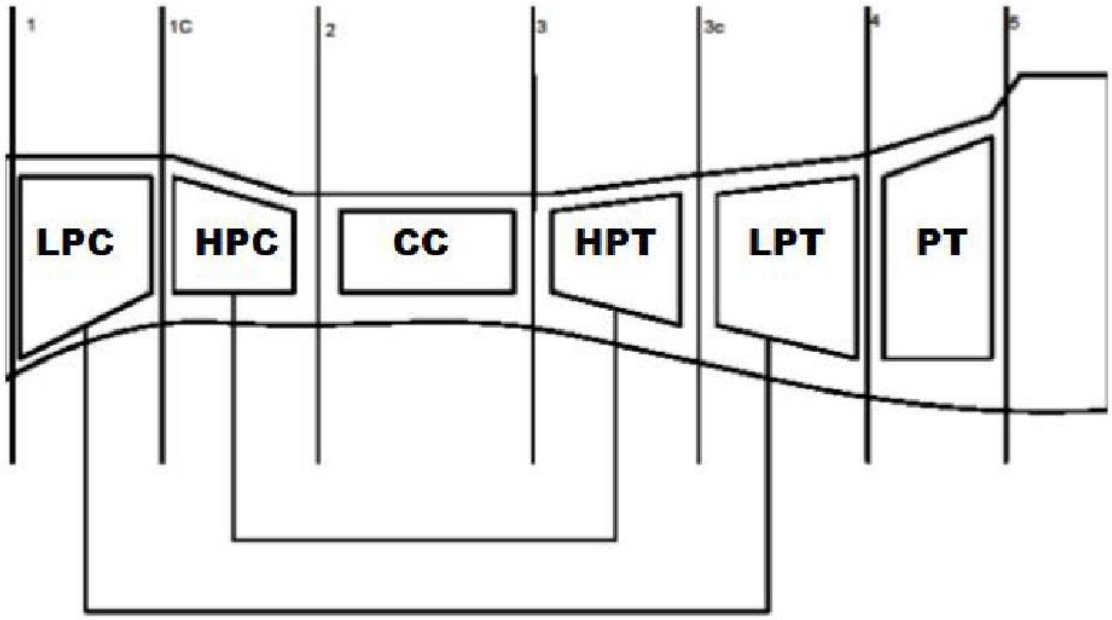

Fig.1.1 GTPP cross-sections

$$T_1 := T_H = 288 \ \ (K)$$

Determination of air parameters in section 1-1 (at the entrance of the power plant)

$P_1 := P_H \cdot \sigma_{in} = 99298.5 \qquad (Pa)$,

where δin - coefficient that takes into account losses of total pressure in an air suction system (before the compressor); δin= 0,97 ... 0,98.

Determination of work that is necessary to compress 1 kg of air in compressor and air parameters in section 2-2 at the exit from the compressor

$\eta_{st} := 0.90$

$$L_c := \frac{K_{air}}{K_{air} - 1} \cdot R_{air} \cdot T_1 \cdot \left(\pi_c^{\frac{K_{air}-1}{K_{air}}} - 1 \right) \cdot \frac{1}{\eta_c} = 438971 \qquad \left(\frac{J}{KG} \right)$$

$$\eta_c := \frac{\pi_c^{\frac{K_{air}-1}{K_{air}}} - 1}{\pi_c^{\frac{K_{air}-1}{\eta_{st}\cdot K_{air}}} - 1} = 0.8531$$
,

where k=1.4 R=287 J/(kg*k), η_{st}= 0,89 ... 0,91 — compressor stage efficiency.

Air temperature and pressure at the exit from the compressor are calculated according to the formulas:

$$T_2 := T_1 + \frac{K_{air} - 1}{K_{air}} \cdot \frac{L_c}{R_{air}} = 732.7525 \qquad (K)$$

$$P_2 := \pi_c \cdot P_1 = 1787373 \qquad (Pa)$$

Determination of working body parameters in section 3-3 (before turbine)

Thermal capacity of combustion product in the temperature range T3-T2 with high accuracy average thermal capacity of gases near combustion chamber GTP is determined by generalized equation:

$$CP := 878 + 0.208 \cdot (T_3 + 0.48T_2) = 1221.558 \quad \left(\frac{J}{KG \cdot K}\right)$$

Relative fuel consumption in combustion chamber is calculated in such way:

$$g_f := \frac{CP \cdot (T_3 - T_2)}{\eta_{cc} \cdot H_u} = 0.0166$$
,

where Hu - lower fuel combustion heat; η_{cc} - coefficient, that takes into account

Incompleteness of fuel burning and heat losses through the combustion section walls. Usually η_{cc} = 0,97 ... 0,98. For liquid hydro carbonic *fuel H*u = (42,5 - 43,5) •106

Specific supplied heat in the combustion chamber:

$$q_1 := CP\cdot(T_3 - T_2) = 692926 \quad \left(\frac{J}{Kg}\right)$$

Pressure on the exit from the combustion section:

$$P_3 := P_2\cdot\sigma_{cc} = 1733752 \quad (Pa)$$

Value σ_{cc} = 0,97 ... 0,99 characterizes total pressure losses in the combustion chamber.

Determination of expansion work of 1 kg gas in turbine, that drives compressor and gas parameters in section 4-4 (after compressor turbine or compressor in case when it is two-shaft)

$$L_{tc} := \frac{L_c}{(1 + g_f)\cdot(1 - g_{cool})\cdot\eta_{tm}} = 463788 \quad \left(\frac{J}{Kg}\right),$$

where g_{cool} - value of specific air waste, that is extracted at the exit from the compressor of turbine for turbine elements cooling - specific air waste that is extracted for technological necessitate .

Value g_{cool} is determined according to temperature level at the exit from the combustion chamber and chosen cooling way, temperature and pressure at the exit from the compressor turbine are determined with help of following equations:

$$T_4 := T_3 - \frac{K_{gas} - 1}{K_{gas}}\cdot\frac{L_{tc}}{R_{gas}} = 900.43 \quad (K)$$

$$P_4 := P_3 \cdot \left(1 - \frac{T_3 - T_4}{T_3 \cdot \eta_{tc}}\right)^{\frac{K_{gas}}{K_{gas}-1}} = 329330 \quad (Pa)$$,

where K_{gas}=1.33, R_{gas}=288(j/kg.k) η_{tc}=0.9…0.91-compressor turbine efficiency.

Determination of expansion work in power turbine and gas parameters at the exit from it.

Pressure at the exit from the power turbine:

$$P_5 := P_H \cdot 1.05 = 106391 \quad (Pa)$$

Power turbine work:

$$L_{PT} := \frac{K_{gas}}{K_{gas} - 1} \cdot R_{gas} \cdot T_4 \cdot \left[1 - \left(\frac{P_5}{P_4}\right)^{\frac{K_{gas}-1}{K_{gas}}}\right] \cdot \eta_{PT} = 229976 \quad \left(\frac{J}{KG}\right)$$,

where power turbine efficiency η_{PT}=0.89….0.91 in transitional channel :

Gas temperature at the exit from the turbine:

$$T_5 := T_4 - \frac{K_{gas} - 1}{K_{gas}} \cdot \frac{L_{PT}}{R_{gas}} = 702.302 \quad (K)$$

1.3 Calculation of main parameters of gas-turbine power plant

$$N_{es} := \eta_m \cdot L_{PT} \cdot \frac{\left(1 + g_f\right)}{1000} = 231.464 \quad (KW)$$,

where η_m -power turbine mechanical efficiency (chose from the range) η_m=(0.99…0.995).

GTP cycle efficiency:

$$\eta_T := \frac{L_{PT}}{q_1} = 0.3319$$

Specific fuel consumption:

$$C_e := \frac{3600 \cdot g_f}{N_{es}} = 0.2588$$

For given power air expenses through the compressor are equal:

$$G_{air} := \frac{N_e}{N_{es} \cdot 1000} = 27.218 \quad \left(\frac{Kg}{s}\right)$$

Absolute fuel consumption per hour (nm^3/h) for measured gas with density ρ_{fuel}=0.682 Kg/m^3

$$Q_{fuel} := \frac{C_e \cdot N_e}{\rho_{fuel}} = 2390269 \quad \left(\frac{m^3}{Kg}\right)$$

1.4 Determination of diametrical sizes on the entry of the compressor

For modern GTP with power turbine C_{a1}=150…180 m/s assume that C_{a1}=160 m/s. Reduced velocity λ_1 and flux density function $q_{\lambda 1}$ are calculated

$$C_{cr} := 18.3\sqrt{T_1} = 310.56 \quad \left(\frac{m}{s}\right)$$

$$\lambda_1 := \frac{C_{a1}}{C_{cr}} = 0.5152$$

$$q_{\lambda.1} := \left(\frac{K_{air} + 1}{2}\right)^{\frac{1}{K_{air}-1}} \cdot \lambda_1 \cdot \left(1 - \frac{K_{air} - 1}{K_{air} + 1} \cdot \lambda_1^{\,2}\right)^{\frac{1}{K_{air}-1}} = 0.7254$$

Channel area at the entry of the compressor is defined:

$$F_1 := \frac{G_{air} \cdot \sqrt{T_1}}{q_{\lambda.1} \cdot P_1 \cdot m_{air}} = 0.1591 \qquad \left(m^2\right)$$

,

where $-m_{air}=0.0403$ and relative diameter is chosen , $d_1=0.5$ and external diameter of working wheel at the entry of the compressor if defined

$$D_{t1} := \sqrt{\frac{4 \cdot F_1}{3.14\left(1 - d_1^{\ 2}\right)}} = 0.5199 \quad (m)$$

Then hub diameter and mean diameter are calculated:

$$d_{H1} := \sqrt{D_{t1}^{\ 2} - \frac{4F_1}{3.14}} = 0.2599 \quad (m)$$

$$h_1 := 0.5\left(D_{t1} - d_{H1}\right) = 0.13 \qquad (m)$$

,

where h_1 is vane height of compressor first stage at the entry.

1.5 Determination of stages and air compression work distribution in two-spool compressors

Compressor is made as two-spool, that consists of low pressure compressor (LPC) and high pressure compressor (HPC).

$$L_{LPC} := 0.42L_c = 184368 \qquad \left(\frac{J}{Kg}\right)$$

$$L_{HPC} := 0.58L_c = 254603 \qquad \left(\frac{J}{Kg}\right)$$

Then pressure ration is calculated according to the formula:

$$\pi_{LPC} := \left[1 + \frac{\eta_{LPC} \cdot L_{LPC} \cdot \left(K_{air} - 1\right)}{K_{air} \cdot R_{air} \cdot T_1}\right]^{\frac{K_{air}}{K_{air}-1}} = 4.603$$

LPC efficiency assumed greater on 1-2% then taken in thermodynamic calculation:

$$\eta_{LPC} := 1.01 \cdot \eta_c = 0.8616$$

Air pressure and temperature after LPC are calculated by formula:

$$P_{1c} := \pi_{LPC} \cdot P_1 = 457097 \quad (Pa)$$

$$T_{1c} := T_1 \cdot \left(1 + \frac{\pi_{LPC}^{\frac{K_{air}-1}{K_{air}}} - 1}{\eta_{LPC}}\right) = 474.796 \quad (K)$$

Then air speeds after LPC are determined:

$$C_{a1c} := \frac{\left(C_{a1} + C_{a2}\right)}{2} = 145 \quad \left(\frac{m}{s}\right),$$

where C_{a1},C_{a2} - reduced velocities at the entry , exit from the compressor, after the calculation of critical and reduced velocities after LPC , flux density function

$$\lambda_{1c} := \frac{C_{a1c}}{18.3\sqrt{T_{1c}}} = 0.3636$$

$$q_{\lambda.1c} := \left(\frac{K_{air} + 1}{2}\right)^{\frac{1}{K_{air}-1}} \cdot \lambda_{1c} \cdot \left(1 - \frac{K_{air} - 1}{K_{air} + 1} \cdot \lambda_{1c}^{2}\right)^{\frac{1}{K_{air}-1}} = 0.5421$$

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BACHALOR THESIS

(Explanatory Note)

Topic: Gas turbine plant of simple cycle with development of vibration monitoring systems

Performed by : _______________ **Akwan Omar** ***(Syria)***

Supervisor:
Candidate of science (Engineering), assoc.prof. __________________ **Yakushenko O.S.**

Standards Inspector: _____________________ (___________________)

Kyiv 2020

Channel area after LPC is defined:

$$F_{1c} := \frac{G_{air} \cdot \sqrt{T_{1c}}}{q_{\lambda.1c} \cdot P_{1c} \cdot m_{air}} = 0.0594 \qquad \left(m^2\right)$$

Tip and hub diameters after LPC are determined in such way:

$$D_{t1c} := D_{t1} = 0.5199 \qquad (m)$$

$$d_{H.1c} := \sqrt{D_{t1c}{}^2 - \frac{4F_{1c}}{3.14}} = 0.4412 \quad (m)$$

$$D_{t2} := \sqrt{d_{H2}{}^2 + \frac{4F_2}{3.14}} = 0.4807 \qquad (m)$$

$$h_2 := 0.5\left(D_{t2} - d_{H2}\right) = 0.0198 \qquad (m)$$

$$d_{H2} := d_{H.1c} = 0.4412 \quad (m)$$

After that air velocities after HPC are defined C_{a2} =130 m/s and, after calculation of critical and reduced velocities after HPC, flux density function

$$\lambda_2 := \frac{C_{a2}}{18.3\sqrt{T_2}} = 0.2321$$

$$q_{\lambda.2} := \left(\frac{K_{air} + 1}{2}\right)^{\frac{1}{K_{air}-1}} \cdot \lambda_2 \cdot \left(1 - \frac{K_{air} - 1}{K_{air} + 1} \cdot \lambda_2{}^2\right)^{\frac{1}{K_{air}-1}} = 0.3577$$

Channel area after HPC F2 is defined:

$$F_2 := \frac{G_{air} \cdot \sqrt{T_2}}{q_{\lambda.2} \cdot P_2 \cdot m_{air}} = 0.0286 \qquad \left(m^2\right)$$

Let's determine number of stages of LPC and HPC. Moreover angular velocity of HPC tip must be greater than LPC on 15-20%.

$$u_{t.1chpc} := 1.25 \cdot u_{t1} = 437.5 \qquad \left(\frac{m}{s}\right),$$

where u_{t1}=(250…350) chosen as 350(m/s).

Angular velocity near the hub of LPC working wheel is defined:

$$u_{Hl} := u_{t1} \cdot \frac{d_{Hl}}{D_{t1}} = 175 \qquad \left(\frac{m}{s}\right)$$

Vane cascade density diameter is assumed as $\frac{b}{t} = 2$ air flow spin on diameter is determined according to the formula:

$$\Delta Wu_{\ Hl} := \frac{C_{al} \cdot 1.55}{\left[1 + \frac{1.5}{\left(\frac{b}{t}\right)}\right]} = 141.7143 \qquad \left(\frac{m}{s}\right)$$

Work that is transferred by vanes to the vanes to the air is calculated according to the Euler's equation:

$$L_{1lpc} := u_{H1} \cdot \Delta Wu_{\ H1} = 24800 \quad (J)$$

Effective work of last stage of LPC is determined as:

$$L_{c1lpc} := u_{H1c} \cdot \Delta Wu_{\ H1c} = 33376 \quad (J)$$

$$\Delta Wu_{H1c} := \frac{C_{a1c} \cdot 1.55}{1 + \left(\frac{1.5}{\frac{b_{c1}}{t_{c1}}}\right)} = 112.375 \quad \left(\frac{m}{s}\right)$$

Average value of compressor stage work:

$$Lm_{LPC} := \frac{L_{1lpc} + L_{c1lpc}}{2} = 29088 \quad \left(\frac{J}{KG}\right)$$

Number of LPC stages:

$$Z_{LPC} := \frac{L_{LPC}}{Lm_{LPC}} = 6.3383$$

Now we must round off the obtained value to the integer number, hence in my case there are 7 stages of LPC

Let's determine number of HPC stages, by calculating work near hub of HPC working wheel.

$$u_{t.1chpc} := 1.25 \cdot u_{t1} = 437.5 \quad \left(\frac{m}{s}\right)$$

$$u_{h.c1hpc} := u_{t.1chpc} \cdot \frac{d_{H1c}}{D_{t1c}} = 371.25 \quad \left(\frac{m}{s}\right)$$

$$\Delta W_{u.H1} := \frac{C_{a1c} \cdot 1.55}{\left[1 + \frac{1.5}{\left(\frac{b}{t}\right)}\right]} = 128.4286 \quad \left(\frac{m}{s}\right)$$

Effective work of first stage of HPC is determined as:

$$L_{1hpc} := u_{h.c1hpc} \cdot \Delta W_{u.H1} = 47680 \quad (J)$$

Effective work of last stage of HPC is determined as:

$$u_{H2} := u_{t.1chpc} \cdot \frac{d_{H2}}{D_{t1c}} = 371.25 \quad \left(\frac{m}{s}\right)$$

$$L_{c2hpc} := u_{H2} \cdot \Delta Wu_{H2} = 33088 \quad (J)$$

$$b_2 := 3 \qquad t_2 := 2$$

$$\Delta Wu_{H2} := \frac{C_{a2} \cdot 1.55}{1 + \left(\frac{1.5}{\frac{b_2}{t_2}}\right)} = 89.125 \left(\frac{m}{s}\right)$$

Average value of compressor stage work:

$$Lm_{HPC} := \frac{L_{1hpc} + L_{c2hpc}}{2} = 40384 \quad \left(\frac{J}{Kg}\right)$$

Number of HPC stages:

$$Z_{HPC} := \frac{L_{HPC}}{Lm_{HPC}} = 6.3046$$

Let's assume that I have HPC which contain 7 stages.

1.6 Determination of diametrical size of the entry of the compressor turbine and number of its stages

In GTPP with two-spool compressor LPC rotates due to low pressure turbine (LPT), and HPC - due to high pressure turbine (HPT), geometrical sizes on the entry to the HPT (section 3) and on the entry to the LPT (section 3c) are determined to the same as for one-spool engine. For determine geometrical sizes between HPT and LPT, at first, their work L_{HPT} and $L_{.LPT}$ are calculated by the formula [1]:

$$L_{HPT} := \frac{L_{HPC}}{(1+g_f)\cdot(1-g_{cool})} = 263617 \qquad \left(\frac{J}{Kg}\right)$$

$$L_{LPT} := L_{tc} - L_{HPT} = 200171 \qquad \left(\frac{J}{Kg}\right)$$

$$G_g := G_{air}\cdot(1+g_f)\cdot(1-g_{cool}) = 26.287 \qquad (Kg)$$

Temperature T_{3c} and pressure P_{3c} are defined as:

$$T_{3c} := T_3 - \frac{L_{HPT}\cdot(K_{gas}-1)}{K_{gas}\cdot R_{gas}} = 1072.89 \qquad (K)$$

$$P_{3c} := P_3\cdot\left(1 - \frac{T_3 - T_{3c}}{\eta_{HPT}\cdot T_3}\right)^{\frac{K_{gas}}{K_{gas}-1}} = 748174 \quad (Pa)$$

HPT efficiency value is take greater then value (chosen on thermodynamic calculation) by 0, 01...0, 02.

Cross sectional area at the entry of the HPT is defined by the formula:

$\sigma_{cc} = 0.97 \qquad \alpha := 20° \qquad q\lambda_3 := 1 \qquad m_g := 0.0396 \qquad \sigma_{NV} := 0.99$

$$F_3 := \frac{G_g\cdot\sqrt{T_3}}{P_3\cdot\sigma_{cc}\cdot\sigma_{NV}\cdot q\lambda_3\cdot m_g\cdot\sin(\alpha)} = 0.042 \qquad \left(m^2\right)$$

Mean diameter on the entry of HPT is equal to:

$$D_{tm3} := 1.1\cdot D_{t2} = 0.5287 \quad (m)$$

$$h_3 := \frac{F_3}{3.14 \cdot D_{tm3}} = 0.0253 \quad (m)$$

$$D_{t3} := D_{tm3} + h_3 = 0.5541 \quad (m)$$

$$d_{H3} := D_{tm3} - h_3 = 0.5034 \quad (m)$$

Cross sectional area at the exit from the HPT it’s defined by the formula:

$$F_{3c} := \frac{G_g \cdot \sqrt{T_{3c}}}{\sigma_{NV} \cdot q\lambda_{3c} \cdot P_{3c} \cdot \sin(\alpha) \cdot m_g} = 0.0858 \quad \left(m^2\right)$$

Choosing a type of turbine we can find mean diameter, vane height, tip and hub diameters.

$$D_{tm3c} := D_{tm3} = 0.5287 \quad (m)$$

$$h_{3c} := \frac{F_{3c}}{3.14 \cdot D_{tm3c}} = 0.0517 \quad (m)$$

$$D_{t3c} := D_{tm3c} + h_{3c} = 0.5804 \quad (m)$$

$$d_{H3c} := D_{tm3c} - h_{3c} = 0.4771 \quad (m)$$

After determination the value of angular velocity on the mean diameter compressor turbine:

$$U_{tm3c} := u_{t1} \cdot \frac{D_{tm3}}{D_{t1}} = 355.97 \quad \left(\frac{m}{s}\right)$$

And choosing loading coefficient y=0,45..0,6 Let’s determine approximate number of compressor turbine stages by the formula :

$$Z_{LPT} := \frac{2 \cdot y^2 \cdot L_{LPT}}{\eta_{LPT} \cdot U_{tm3c}^2} = 0.7586 \qquad Z_{HPT} := \frac{2 \cdot y^2 \cdot L_{HPT}}{\eta_{HPT} \cdot U_{tm3}^2} = 0.6338$$

By approximation we have 1 stage of each compressor turbine type. Values of Lt and nc are taken from thermodynamic calculation.

1.7 Determination of diametrical sizes at the entry of the power turbine and at the exit from it, number of power turbine stages

Cross sectional area F_4 at the entry to the power turbine is defined by the formula:

$$F_4 := \frac{G_g \cdot \sqrt{T_4}}{\sigma_{NV} \cdot q\lambda_4 \cdot P_4 \cdot \sin(\alpha) \cdot m_g} = 0.1786 \quad \left(m^2\right),$$

where gas expenses through the power turbine is equal to (78.46).

T_4, P_4-gas temperature and pressure before power turbine that is taken from thermodynamic calculation:

$q_{\lambda 4}$- coefficient of pressure losses between turbines. If there is super critical pressure differences in the power turbine assume that compressor turbine gas shape with constant mean diameter.

$$D_{tm4} := D_{tm3} = 0.5287 \quad (m)$$

$$h_4 := \frac{F_4}{3.14 D_{tm4}} = 0.1076 \quad (m)$$

$$D_{t4} := D_{tm4} + h_4 = 0.6363 \quad (m)$$

$$d_{H4} := D_{tm4} - h_4 = 0.4212 \quad (m)$$

Cross sectional area F_5 at the exit from the power turbine is defined by the formula:

$$F_5 := \frac{G_g \cdot \sqrt{T_5}}{q\lambda_5 \cdot P_5 \cdot m_g} = 0.2417 \quad \left(m^2\right)$$

,

where value λ_5 is range 0,5..0,7 (assume that λ_5 =0.5) Let's determine values

$$q\lambda_5 := 1.527 \cdot \lambda_5 \cdot \left(1 - 0.143\lambda_5^{\,2}\right)^{3.02} = 0.684$$

Let determine diametrical sizes at the exit from the power turbine: tip diameter D_{t5}, mean d_{5m} and hub d5h and vane height h_5.

$$d_{H5} := d_{H4} = 0.4212 \quad (m)$$

$$D_{t5} := \sqrt{d_{H5}^{\,2} + \frac{4F_5}{3.14}} = 0.6966 \quad (m)$$

$$h_5 := \frac{D_{t5} - d_{H5}}{2} = 0.1377 \quad (m)$$

Moreover, for providing the enough strength of turbine last stage vane, it is necessary to fulfill the inequality $\frac{d_{t5}}{h_5} > 4$

Let's check this $\frac{D_{t5}}{h_5} = 4.9482$ condition (m) condition is satisfied.

At first, angular velocity on mean diameter at the entry to the power turbine is chosen (for non-coolant turbine value U_{tm4} is taken in the range 220...280 m/s, assume U_{tm4} =240 m/s and power turbine stage number is found according to the formula :

$$Z_{PT} := \frac{2 \cdot y^2 \cdot L_{PT}}{\eta_{PT} \cdot U_{tm4}^{\,2}} = 1.9599$$

That's mean we have 2 stage of power turbine, value y for power turbine is chosen in the same range as compressor turbine (y=0.47)

1.8 Determination of powers and frequencies of rotation of rotors in compressor and turbine

We determination the capacities spent on the rotation of the compressor $N_к$ (compressors of low $N_{кн}$ and high preassure $N_{кв}$ at a two-shaft compressor) and the power produced by the turbine of the compressor $N_{тк}$ (turbines of compressors of high $N_{т.в}$ and low $N_{т.н}$ pressure), the rotor speed of the compressor $n_к$ (compressors of low $n_{к.н}$ and high $n_{к.в}$ pressure) and power turbine $n_{с.т}$[1]:

$$N_{LPC} := G_{air} \cdot L_{LPC} = 5018129 \quad (W)$$

$$N_{HPC} := G_{air} \cdot L_{HPC} = 6929797 \quad (W)$$

$$N_{LPT} := G_{g} \cdot L_{LPT} = 5261964 \quad (W)$$

$$N_{HPT} := G_{g} \cdot L_{HPT} = 6929797 \quad (W)$$

$$N_{PT} := G_{air} \cdot L_{PT} = 6259498 \quad (W)$$

$$n_{LPC} := \frac{60 \cdot u_{t1}}{3.14 \cdot D_{t1}} = 12864 \quad (rpm)$$

$$n_{LPT} := \frac{60 \cdot U_{tm3c}}{3.14 \cdot D_{tm3c}} = 12864 \quad (rpm)$$

$$n_{HPC} := \frac{60 \cdot u_{t.1chpc}}{3.14 \cdot D_{t1c}} = 16080 \quad (rpm)$$

$$n_{HPT} := \frac{60 \cdot U_{tm3}}{3.14 \cdot D_{tm3}} = 16080 \quad (rpm)$$

$$n_{PT} := \frac{60 \cdot U_{tm4}}{3.14 \cdot D_{tm4}} = 8673 \quad (rpm)$$

As a result of the calculation it is necessary to depict the following part GTP on the appropriate scale and specify the appropriate sizes, as shown in Fig.1.2

GTP parameters, obtained as a result of thermodynamic and gas-dynamic calculations are compared with the parameters of the serial GTU, given in the

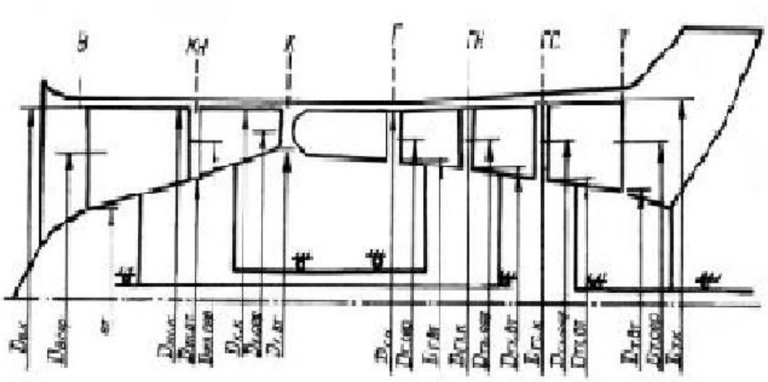


Fig.1.2 Scheme of the flowing path of the gas turbine

1.9 Calculation of operating characteristics

Operation of the GTP during operation is assessed by throttle and climatic characteristics.

Throttle characteristic is the dependence of effective power N_e and specific fuel consumption C_e from the rotational speed of the turbocharger rotor (Rotor turbocharger of high pressure in the GTP with a two-way compressor) with constant parameters of air at the entrance to the GTP $P_н$ and $T_н$, the climatic characteristic is the dependence of the quantities N_e and C_e from the air temperature at the entrance to the GTP $T_н$ at constant physical speed of rotation Rotor of Turbocharger, proceeding from the definitions of characteristics, it is obvious that the calculation of characteristics is a thermodynamic calculation for values of node parameters that depend on n or $T_н$ [1].

Analysis of real characteristics of nodes GTP shows that the parameters of the compressor: the degree of pressure increase $\pi^*_к$ and efficiency $*к$ when changing n and $T_н$ vary to a much greater extent than the parameters of the combustion chamber and the turbine, and therefore, in order to simplify the calculations, only the dependence of the quantities is taken into account in this method $\pi^*к$ and $*к$ on n and

NATIONAL AVIATION UNIVERSITY

Faculty: Aerospace Faculty
Department: The Aero-engines department
Educational degree: Bachelor
Speciality: 142 Power Machinery
Major: Gas Turbine Plants sand Compressor Stations

APPROVED BY

Head of the Department

____

"____"_____________2020

Bachelor Degree Graduation Work Assignment

Student Name: ***Akwan Omar (Syria)***

The Topic of work: Gas turbine plant of simple cycle with development of vibration monitoring systems

1. Approved by the Rector's order of «21 » _05._2020 _ № 630/ст.

2. The paper to be performed from _18. 05. 2020 to 21.06.2020

3. Initial data for the work (thesis): *Calculations of gas turbine plant should be done for standard ambient conditions (H = 0, T = + 15 °C), subject of study: Analyses and development of improved development of vibration monitoring systems;*

4. The content of the explanatory note (the list of problems to be considered): *Introduction, analysis of modern and prospective vibration monitoring systems for gas turbine plants, selecting parameters of working body of gas turbine plant; thermal and gas-dynamic calculation of engine; strength calculations of engine elements; development of vibration monitoring system of designed power turbine unit*

5. The list of mandatory graphical (illustration) materials: engine design, *graphics to the special part of diploma work. Graphic materials to be performed using computer software.*

$T_н$, and the parameters of the combustion chamber and the turbine are considered to be unchanged. Characteristics calculated by this method, qualitatively reflect the actual nature of their change [1].

In this case, the initial parameters for the calculation of characteristics are selected and calculated parameters in the process of thermodynamic calculation, which is assigned an index "p" (calculated): $p_{н.р}$, $T_{н.р}$, $\sigma_{вх.р}$, $\pi^*_{к.р}$, $\eta^*_{к.р}$, $\sigma_{к.з.р}$, $\eta_{г.р}$, $\pi^*_{т.к.р}$, $T^*_{г.р}$, $\eta^*_{т.к.р}$, $g_{пал.р}$, $g_{ох.р}$, $g_{в.р}$, $\eta^*_{м.р}$, $\eta^*_{т.к.р}$, $\eta^*_{м.с.р}$, $G_{к.р}$, $L_{к.р}$, $p^*_{т.р}$.

After determining the output parameters, select 3-4 values of the variable parameters: for the throttle - the combined rotational speeds in range 0,75–1,0; for climatic - air temperature within -30 - +30 oC. Calculations are conveniently carried out in the form of tab.1. In order to control the process of calculation of characteristics in Table.1 recorded results of thermodynamic calculation in the calculation mode[1].

Table 1– Calculation of operating characteristics

№ п/п	Parameter and calculation formula	Parameters in the calculation mode	Throttle characteristics			Climatic Characteri-stics		
1	2	3	4	5	6	7	8	9
1	$\overline{n}$		0,95	0,9	0,85	1,0	1,0	1,0
2	$T_н = T^*_в$, К		288	288	288	268	298	320
3	$p_н$, МПа		0,101	0,101	0,101	0,101	0,101	0,101
4	$\overline{n}_{пр} = \overline{n}\sqrt{288/T^*_в}$		0.95	0.9	0.85	1.037	0.983	0.949
5	$\overline{\pi}^*_к = \overline{n}_{пр}{}^a$, де $a = \pi^{*\,(0.2\cdot\pi^*_{к.р}/\overline{n}_{пр}{}^{-2.15})}_{к.р}$		0.913	0.829	0.748	1.066	0.97	0.91
6	$\pi^*_к = \pi^*_{кр}\overline{\pi}^*_к$		16.427	14.918	13.473	19.193	17.461	16.387
7	$\overline{\eta}^*_к = n_{пр}{}^b$, де		0.997	0.987	0.973	0.998	1	0.997

	$b = \pi_{к.р}^{*\ (0.2 \cdot \pi_{к.р}^{*} / \bar{n}_{пр}^{-2.15})}$							
8	$\eta_{к}^{*} = \eta_{к.р}^{*} \bar{\eta}_{к}^{*}$		0.85	0.842	0.83	0.852	0.853	0.85
9	$L_{к} = \frac{kRT_{в}^{*}}{k-1}\left(\pi_{к}^{*(k-1)/k} - 1\right)\frac{1}{\eta_{к}^{*}}$, Дж/кг		394168.13	379113.958	365181.5	395065.8	419074.50	437503
10	$T^{*}_{к} = T^{*}_{в} + L_{к}/[kR/(k-1)]$, К		627	614	602	608	659	696
11	$p_{г}^{*} = p_{н}^{*}\pi_{к}^{*}\sigma_{к.з.р}\sigma_{вх.р}$, Па		158225 6.8	1436876 .7	129768 3.9	184861 7	1681800 .5	1578349 .6
12	$T_{г}^{*} = T_{г.р}^{*} L_{к} / L_{к.р}$, К		1167	1122	1081	1169	1241	1295
13	$c_{к.з} = 878 + 0{,}208\left(T_{г}^{*} + 0{,}48 T_{к}^{*}\right)$,Дж/(кг·К)		1183	1172	1163	1182	1201	1217
14	$g_{пал} = c_{сер.кз}\left(T_{г}^{*} + T_{к}^{*}\right)/\left(H_{u}\eta_{г.р}\right)$		0.015	0.014	0.013	0.015	0.016	0.017
15	$L_{тк} = L_{к}[(1+g_{пал})(1-g_{ох.р}-g_{в.р})\times$ $\times\eta_{м.р}]$, Дж/кг		416986	401466	387070	417687	442697	461847
16	$T_{гс}^{*} = T_{г}^{*} - \left(k_{г}-1\right)/\left(k_{г}R_{г}\right)$, К		808	776	748	810	859	897
17	$p_{гс}^{*} = p_{г}^{*}\left(1 - \frac{T_{г}^{*} - T_{т.с}^{*}}{T_{к}^{*}\eta_{т.к.р}^{*}}\right)^{\frac{k_{г}-1}{k_{г}}}$, Па		762380	691698	624169	891199	811396	761957
18	$L_{с.т} = \frac{k_{г}}{k_{г}-1} R\left(1 - \left(p_{т.р}^{*} / p_{гс}^{*}\right)^{k_{г}-1/k_{г}}\right)$, Дж/кг		326294	301526	277647	346850	355585	362429
19	$N_{е.п} = L_{с.т}\eta_{м.с.р}\left(1 + g_{пал}\right)$, Вт·кг/с		327985	302782	278546	348855	357942	365083
20	$G_{к} = G_{к.р}\frac{p_{г}^{*}}{p_{гр}^{*}}\sqrt{\frac{T_{г.р}^{*}}{T_{г}^{*}}}$, кг/с		23.53	20.96	18.58	27.53	25.79	24.73
21	$C_{е} = 3{,}6 g_{пал} / N_{е.п}$, кВт/год		0.16	0.17	0.17	0.16	0.16	0.17
22	$N_{е} = N_{е.п} G_{к}$, кВт		7.7	6.3	5.1	9.6	9.2	9.03
23	$\overline{C}_{е} = C_{е} / C_{е.р}$		0.65	0.657	0.66	0.63	0.65	0.66
24	$\overline{N}_{е} = N_{е} / N_{е.р}$		1.41	1.30	1.20	1.50	1.54	1.57

1.10 GTP Systems

1.10.1 Lubrication system

Engine lubrication system - circulating, pressurized. Provides a constant supply of oil to the surfaces of the bearings of the friction rotor bearings, rotor bearing seals, rotating central drive parts, the upper drive box and the speed limiter PT and its drive for lubrication and cooling.

The engine lubrication system monitors the pressure and temperature of the oil at the engine inlet.

The engine uses early detection of malfunctions of the lubrication system, parts and components washed with oil.

Alarms emit signals to the ACS when the following limit parameters are reached:

minimum oil pressure at the engine inlet;

clogging of the fine oil filter;

the appearance of ferromagnetic chips or exceeding the allowable temperature in the mains of oil pumping from the cavities of the LPC bearing, bearings of turbines HP and LP, bearings of the power turbine;

the appearance of ferromagnetic chips in the oil pumping line from the upper gear box, the central drive, the support of the HPC and the lower gear box.

The engine lubrication system includes the following main components:

oil tank installed on the GTP;

the block of oil coolers established on GTP;

oil unit consisting of a supercharger pump, four pumping pumps, non-return and pressure reducing valves, fine oil filter with bypass the valve and the signaling device of pressure difference, the valve of release of air. On the oil unit the receiver of system of measurement of temperature of oil at an entrance to the engine is mounted:

air separator with coarse oil filter;

chip alarm;

thermal chip detector for the presence of ferromagnetic particles and oil overheating in the mains for pumping oil from the rotor supports;

Protective filters for oil injectors;

Protective filters for pumping pumps;

pipelines, channels and nozzles, oil drain valve.

The engine blowing system includes:

centrifugal prompter;

pipelines and channels of the blowing system.

Oil from the oil tank flows by gravity into the pump supercharger of the oil unit, where it is fed under pressure into the fine filter. The oil pressure at the inlet to the engine is maintained within the specified limits by the pressure reducing valve.

The oil, having passed the fine filter, is supplied via external pipelines to the LPC support, turbine supports and oil pressure control devices, and to the HPC support, the central drive and the upper gearbox - through the channels made in the intermediate housing. Other parts and components are lubricated by bubbling.

Oil is pumped from the LPC bearing cavity by a pump, from the HPT and LPT bearing cavity by a pump, and from the free turbine bearing cavity by a pump.

On the way to the pumping pumps, the oil washes the thermal chip detectors and is filtered by the pump safety filters.

The pumped oil from the above cavities is poured into the pan of the lower gearbox. Here the oil from a cavity of the top box of drives, the HPC bearing and the central drive on internal cavities of edges of the intermediate case merges by gravity.

From the carter of the lower box of drives all oil, having passed the chip signaling device and the filter, is pumped out by the main pumping out pump and on the channel in the lower box of drives goes to the central air separator. The oil separated in the air separator from the air enters for cooling in the oil cooler unit.

The cooled oil is returned to the oil tank.

The oil pressure at the inlet to the engine is measured by a pressure transducer, and the minimum pressure is shown by an alarm signal

A temperature receiver is installed at the engine inlet to control the oil temperature

Installed in the oil pumping mains, thermal chip detectors give signals to the ACS when ferromagnetic particles appear in the oil or the maximum temperature of the pumped oil is exceeded.

The appearance of ferromagnetic particles in the oil pumped from the gearboxes, the central drive and the cavity of the HPC bearing, is detected by the chip. The control of the amount of oil in the tank is carried out by a level indicator.

Draining oil from the lubrication system is carried out through taps, which are located:

on the carter of the lower box of drives;

on the oil tank;

on the block of oil coolers;

through the drain boxes located on the oil unit and air coolers

Prompting of all oil cavities is necessary for ensuring normal work of systems of lubrication and sealing.

Oil cavities of the LPC bearing and turbine bearings are blown into the cavity of the upper gearbox via external pipelines.

The central drive and the lower drive box are connected to the cavity of the upper drive box through channels in the ribs of the intermediate housing.

The air-oil mixture from the upper gearbox enters the centrifugal prompter, where the separated oil is drained by an external pipeline into the pan of the lower drive box, and the purified air exits into the exhaust device.

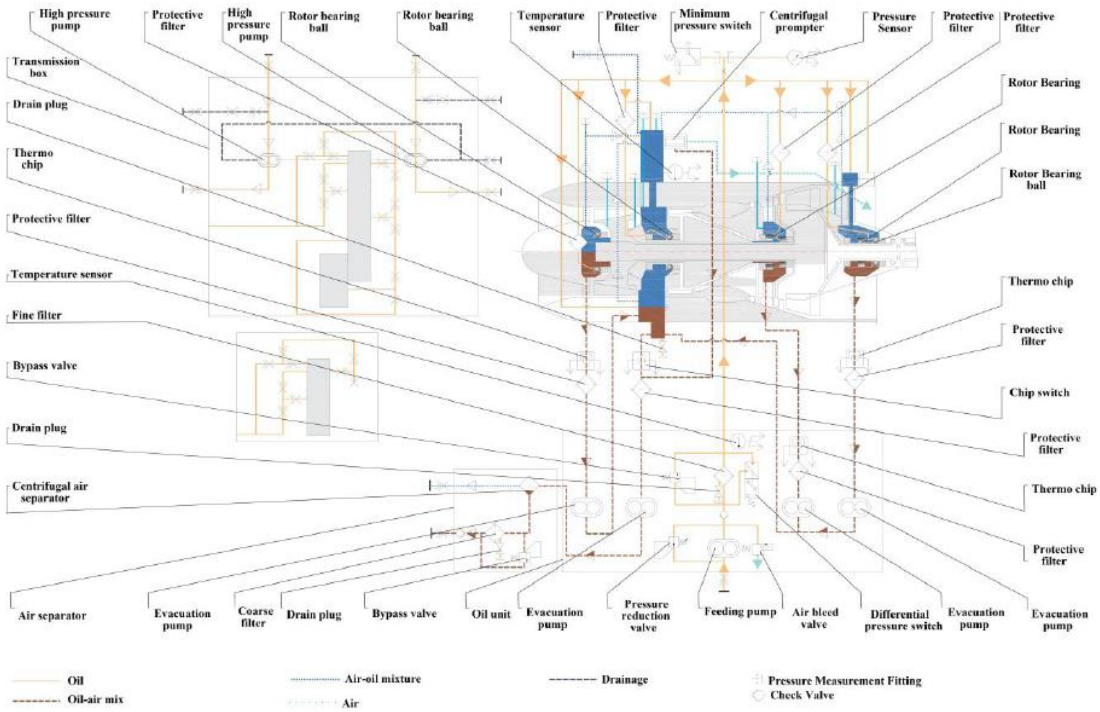


Fig.1.3 GTP Oil system

1.10.2 Fuel system

The fuel system provides the supply of gaseous fuel to the starting and operating injectors in quantities and with a certain purity, pressure, temperature, ensuring the operation of the engine in all modes and in all operating conditions.

The fuel system includes: shut-off valve, gas dispenser, fuel collector, fuel injectors, solenoid valve for starting fuel, starting injectors, fuel gas leak detector, fuel pressure converter in front of the injectors, air pressure converter according to HPC.

Working fuel from the GTP fuel preparation system with a pressure of (2.4 - 0.1) MPa, fine filtration of 40 μm and a temperature of 15 to 50 °C is supplied to the shut-off valve and then to the gas dispenser , in which it is dosed in quantities determined by ACS.

The metered fuel from the dispenser is fed to the fuel collector and then to the working injectors. Fuel from the GTP fuel preparation system is supplied via a throttle to the starting fuel solenoid valve, which opens (closes) the access of fuel with a pressure of (0.25 - 0.02) MPa to starting nozzles on an electric signal of ACS.

The fuel gas leak detector in the presence of gas pressure in the combustion chamber of the engine before starting gives a signal to the ACS about the gas leak.

Fuel regulating units carry out:

automatic fuel dosing at start-up and at steady modes;

maintaining the constancy of the speed of the rotor of the free turbine when changing operating conditions;

protection of the engine against excess of speed of rotations of high and low pressure, free turbine and temperature of gases behind the turbine of low pressure and against a surge.

1.10.3 GTP ACS

The control system of engine parameters is designed to collect information from sensors signalling device, which are installed in the engine, its processing, analysis, issuance of commands to control the engine and display information on the operator's panel.

The control system includes:

ДТА-10 sensors of rotation frequency of rotors of HP, LP, PT and the СВ-36Г rotor;

gas temperature measurement system;

П-109 fuel temperature sensor before the injectors;

the converter П319-03 of fuel pressure before work nozzles;

the converter П319-02 of air pressure on HPC;

П-109 oil temperature sensor at the engine inlet;

converter П319-01 oil pressure at the inlet to the engine;

signaling device мств-1,6А of the minimum gas pressure before turbine nozzle apparatus СВ-36Г;

gas leak detector СВПГ;

the signaling device of the minimum pressure of oil at an entrance to the engine МСТВ-1,6;

signaling device of difference of pressure of oil on the oil filter СП-0,6Е;

chip signaling CC-36;

thermal chip detectors;

signaling devices of open position air bypass valve мств-2,1;

vibration measurement system IB-Д-ПФ-0, Ч.М;

unit ACS;

information equipment on the operator's panel.

Signals from sensors and alarms come to ACS, then ACS processes information, issues commands to the actuators of the engine, monitors the operation of the actuators, monitors the electrical circuits of sensors and alarms.

ACS provides information about the current value of the parameters of the engine to the operator's panel; issues a message to the operator's panel when the actuators are triggered, when the parameters limit values of the engine are reached; issues a message about the nature of faults in case of their occurrence. All information from ACS is issued to the operator's panel.

Information about vibrations arrives on the operator's panel bypassing ACS.

Conclusion

This part was considered under on the basis of data acquired by different calculations. In calculations different parameters are considered. For instance, in thermodynamic calculation parameters such as pressure, temperature and their effect on different elements of the engine were taken into account. Similarly, in gas-dynamic calculation geometrical parameters of GTE which includes, number of stages in compressor, turbine and power turbine were calculated along with other geometrical configurations.

2 STRENGTH CALCULATION OF GTE ELEMENTS

2.1 Strength calculation of the turbine rotor blade

The strength calculation of the rotor turbine blade is executed by simplified methods. It is necessary to determine the tension stress from centrifugal forces and the bending stress from gaseous and centrifugal forces. The torsion stress from gaseous forces due to their insignificant value in the given computation is not taken into account [2].

The tension stress from centrifugal forces necessary to calculate in four cross sections: on root and on distances 25 %, 50 % and 75 % of blade length h_B from the root to cross section. Taking in account that the area of blade cross-section changes for exponent law [2]:

$$F(R) = F_r - (F_r - F_0)\left[[\frac{R - R_r}{R_0 - R_r}] \right]^q,$$

where Fr, F_o = root and tip cross section areas accordingly (are determined after profiling [9]), m^2; R = current radius, m; q = exponent, which for turbine rotor blades equal 0,5 ...0,6 [2].

For condition the tension stress from the action of centrifugal forces in any blade cross section is:

$$\sigma_{tens} = \rho\omega^2 \left\{ \frac{R_0^2 - R^2}{2} - h\left(1 - \frac{F_0}{F_r}\right)\left[\frac{R_r}{1+q} + \frac{h}{2+q} - \left(\frac{R_r}{1+q} + \frac{R-R_r}{2+q}\right)\left(\frac{R-R_r}{h}\right)^{(1+q)} \right] \right\}$$

where ρ = blade material density, kg/m^3; ω = angle rotation speed, rad/s; $\overline{F} = F_O/F_r$ (in computations it is necessary to take $\overline{F} = 0{,}25\ldots0{,}35$); h_B= blade height, m [2].

6. TIME and WORK SCHEDULE

#	***Stages of Graduation Paper Completion***	***Stage Completion Dates***	***Remarks***
1	Introduction, analysis of the current state of the problem of vibration monitoring of gas turbine plants	18.05-22.05.20	
2	Choice of working process parameters for the projecting GTP	23.05- 25.05 .20	
3	Thermodynamic and gas-dynamic calculation of GTP	26.05-29.05.20	
4	Development of gas turbine plant design	30.05– 5.06.20	
5	Special part	6.06 – 7.06.20	
6	Calculations on the strength of the main parts of the engine. Making the explanatory note	8.06 - 10.06.20	
7	Anti plagiarism check	11.06-12.06.20	
8	Fulfillment of drawings	12.06 – 16.06.20	

8. Date of assignment issue: “_____"___________ 2020

Supervisor of diploma work ________________ Yakusenko O.S.
(supervisor`s signature) (name & initials)

Assignment is accepted for execution ________________ Akwan Omar
(graduate student`s signature) (name & initials)

The blades bending stresses from gaseous and centrifugal forces usually are small, that is why they are almost quite compensated due to displacement of the mass centers of blade cross sections relatively to the root cross section. The bending stress from gaseous forces can be determined by the intensity of gaseous load that act on rotor blades, however, with the aim of computations reduction in the course project is supposed to adopt the total bending stress in all blade cross section [2]:

$$\sigma_{\text{bend}\Sigma} = (0{,}05...0{,}1)\sigma_{\text{tens}}.$$

The summary stress in the calculation cross-section:

$$\sigma_{\Sigma} = \sigma_{\text{bend}\Sigma} + \sigma_{\text{tens}} = (1{,}05...1{,}1)\sigma_{\text{tens}}$$

The final computation task is determination of the strength safety margins *k* in the calculation cross-sections of blade, comparison of them with admissible margins and formulation of the deductions about the efficiency of rotor blade [2].

For determination of the safety margin *k* is necessary to determine the dependences of total stresses change on the centrifugal and gaseous forces, the blade temperature *T* and the long-term tensile strength on 100 h base along the blade height [2].

During the determination of blade material temperature in the calculation cross sections it is necessary to take into account that the temperatures field of the combustion chamber outlet is formed so, that the maximum temperature value must be reached on the front blade flange on the distance 50...70 % of blade length from its root cross section. The blade temperature in root cross section in dependence on cooling type can be taken as 150…200K and maximum value as 120…170 K lower from the calculation gas temperature on the middle radius before stage [2].

The values of tensile long-term strength limits for the certain blade material in each calculation cross section may be determined by the graph (fig.2.1)[2]

The strength safety margins *k* in the calculation cross sections are determined by comparing of the values of tensile long-term strength limits of the select material to the total stresses [2].

$$k = \sigma_{100}^{T} / \sigma_{\Sigma} \geq 1{,}3\ldots1{,}5$$

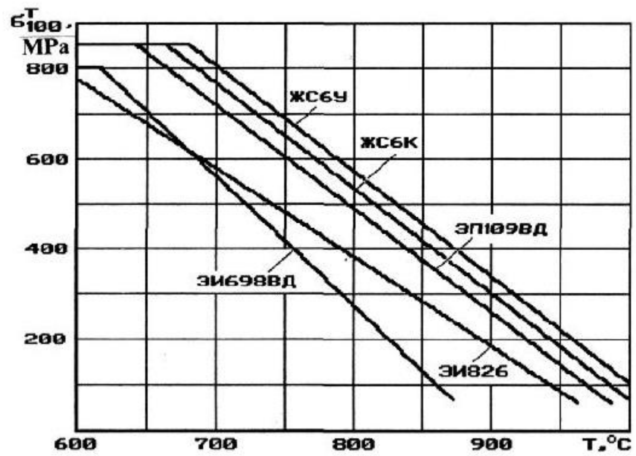


Fig.2.4 The dependence of the tensile long-term strength limits of heatproof alloys from the temperature [2]

2.2 Blade Stress calculations

Data for blade of the modified turbine:

b=27mm

h=45mm

C1=5mm

C2=4mm

Fr=94.5

F0=75.6

Material used: ЭИ826

Density=8470 kg/m3

n_{LPT}=12864 rpm

W=77184 ^{rad}/sec

Results of calculation are shown in Table 2.3.

Table 2.3- Result of blade stress calculation

Parameter	Units	$\bar{L}$=0	$\bar{L}$=0,25	$\bar{L}$=0,5	$\bar{L}$=0,75
$\bar{r}$	-	0,859	0,894	0,926	0,965
$\sigma_{від}$	МПа	167	137	100	54
σ_c	МПа	192	157	115	62
T	К	965	1114	1164	1112

$\sigma_{\text{тм}}$	МПа	557	273	179	276
К	-	2.9	1.7	1.6	4.4

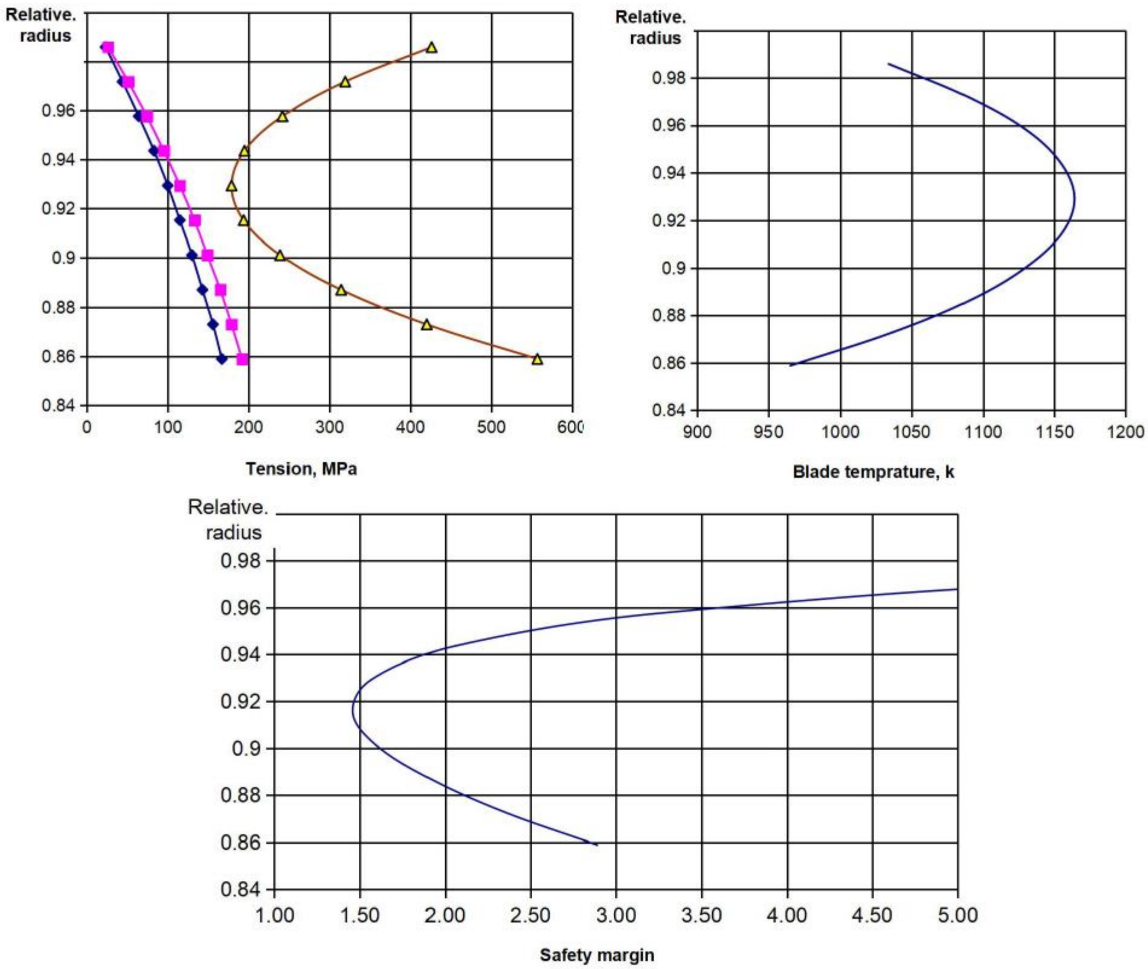


Figure 2.5 – Result of blade stress claculation

2.3 Strength calculation of the turbine disc

The strength calculation of turbine disc is necessary to execute as test after drawing of the turbine unit and a material choice, the computation is executed for the turbine stage disc for which the blade computation done, the disc computation of other turbine or compressor stages can be made in separate cases, by concordance with course project supervisor [2].

The aim of computation is to change determination of the equivalent stresses and safety factors of long-term material strength on bases of 100 and 1000 h attached to suitable temperature values along the disc radius [2].

For the strength computation of turbines discs it is necessary to execute a big number of monotonous computing operations, that results in non-productive time expenses of students and does not allow to estimate influence on the discs strength of various engines exploitation conditions and to execute computation in the elastic deformation area. In this case, for above of diploma and course projects, mentioned computation is necessary to perform on a computer [2].

For correcting computation a student have to know the types of disc loading and construction principles of its calculation scheme, to acquaint with peculiarities of various calculation modes conditioned by engines exploitation conditions; be able to calculate stresses distribution along the disc radius, clearly understand the equations of strained state and deformations community, to know the boundary condition for the determination of these equation, to right estimate the discs safety margins and to make the deductions about their capacity [2].

The basic questions associated with turbine disk computations on strength, with preparation of the outgoing data for computation, algorithms and programs for the computation of resilient and resiliently - plastic disc stresses on computer are considered in work. That's why the sequence of outgoing data preparation for disc computation on strength by method of final differences on a computer is presented below [2].

The outgoing data preparation is executed in such sequence.

The calculation scheme is divided into cylindrical cross sections with taking in to account of such recommendations.

1. To divide the calculation disc scheme by the medium of main cross sections on row of typical domains I, II, III and etc., which describe the disc form. Each domain of the disk scheme inside to divide on cross sections with certain pitch; in domains II and I pitches can be used approximately identical. In III domain pitches must be chosen as two times bigger, and in IV and V domains less in two-three times, than in I domain [2].

2. For solid disc the first cross section is taken near the center on distance (0,05...0,10) Rк from axis (Rк is the external disc radius), for disc with central hole - on distance equal to hole radius R_C [2].

3. For each cross-section to determine the values of radius Ri and disc width Bi [2].

4. The action of centrifugal forces from the masses of rotor blades and disc comb (the lock part of a disk rim) is replaced by the action of radial contour load, evenly divided on the cylindrical rim surface on the radius Rк. The value of radial contour stress σ_{RK} from this load is:

$$\sigma_{RK} = \frac{Z_{RB}\sigma_{TR}F_R + \sum_{i=1}^{3} P_i}{2\pi R_K B_K},$$

where Z_{RB} = amount of rotor blades; σ_{TR} = tension stress from centrifugal forces in root blade section, Pa; F_R = area of root blade cross section, m^2; B_K = disc comb thickness, m; $P_i = m_i R_i \varpi^2$ - centrifugal mass forces of internal blade shelf, blade butt and disc comb; m$_i$, R$_i$= masses and gravity centers radiuses of blade shelf, blade butt and disc comb accordingly, kg, m; $\varpi = \pi n/30$ - angle rotor rotation speed, rad/s; n = rotor rotation frequency, RPM [2].

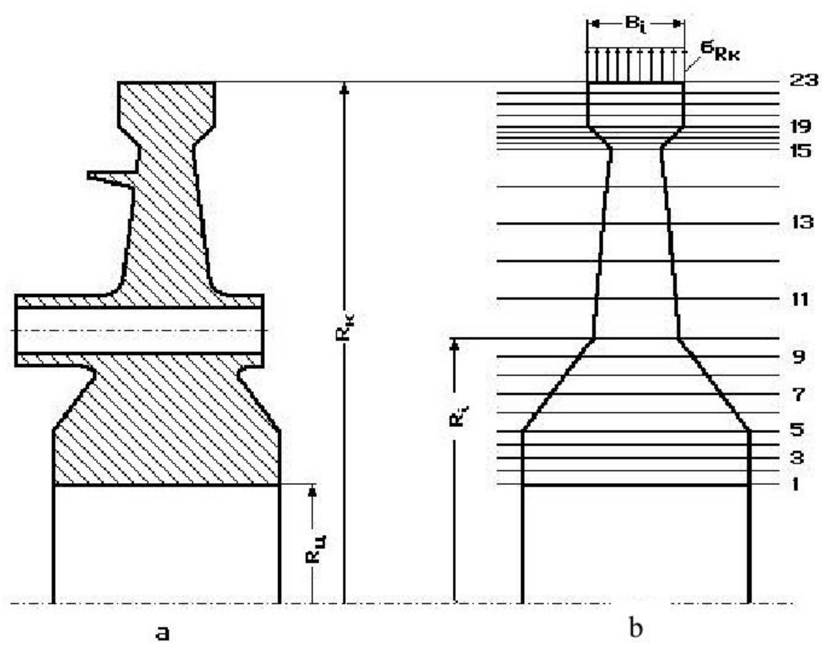


Fig.2.6 Radial disc cross sections (a) and its calculation scheme (b)

5. For the calculation mode to determine the values of disc temperature on rim T_R ˚C and in center T_C ˚C (or on hole radius). The temperature change along the disc radius can be approximated by power dependence [2]:

$$T = T_C + (T_K - T_C)\left(\frac{R - R_C}{R_K - R_C}\right)^S,$$

2.4 Disk strength Calculation results

Results of calculation are shown in table 2.4, and in fig.

Table 2.4 – Table of disk loads

RADIAL LOADS			
0	5	15	25
35	44	55	63
75	88	99	114
182	213	239	246
210	175	155	134
117	116	114	112
110	107		
CIRCULAR LOADS			
1456	1450	1439	1428
1417	1406	1397	1393
1385	1377	1368	1306
1249	1194	1136	1073
1054	1029	1011	992
974	969	959	950
941	932		
EQUIVALENT LOADS			
1456	1448	1431	1415
1400	1385	1370	1363
1348	1334	1321	1244
1168	1103	1037	973
966	954	943	932
921	916	907	899
891	883		

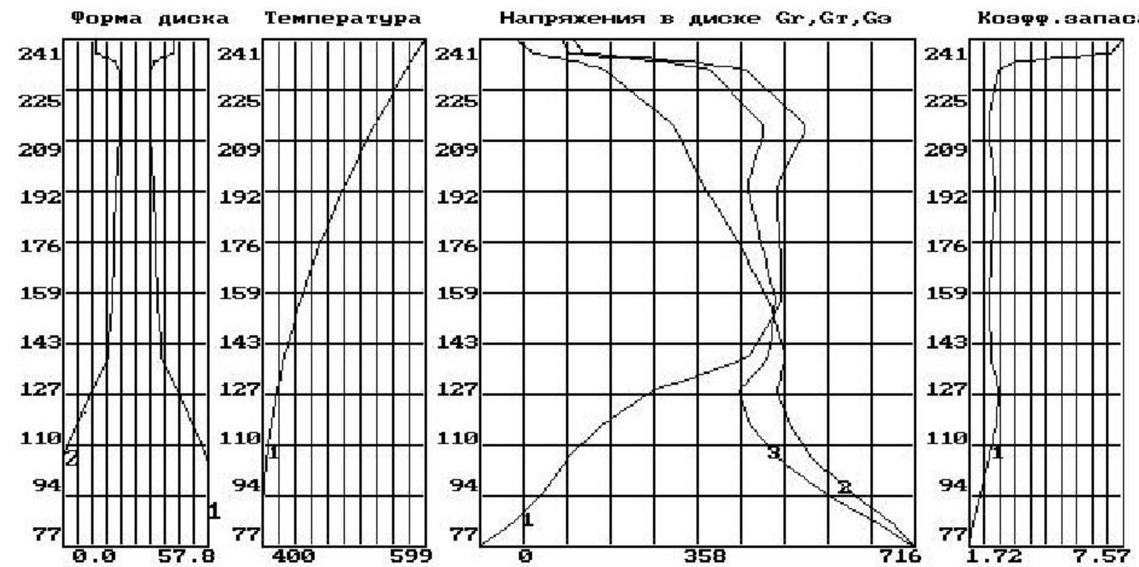


Fig.2.7 Disk strength results

From fig. 2.7 we can see that strength coefficient in all section of the disk grater than 1.72.

Conclusion

In this part a Strength calculation was performed on turbine elements disc and blades taking into account the data from the thermodynamic calculation and gas-dynamic calculation the calculation included a tension stress from centrifugal forces and the bending stress from gaseous and centrifugal forces, excluding the torsion stress from gaseous forces due to their insignificant value. The tension stress was calculated in four sections: on root and on distances 25 %, 50 % and 75 % of blade length. The material was chosen as ЭИ826.

3 VIBRATION MONITORING OF DESIGNED GTP

3.1 Vibration and its measuring

Vibration is a physical phenomenon that presents itself in operational rotating machineries and moving structures, regardless of the condition of their health. Vibration can be induced by various sources, including rotating shafts, meshing gear-teeth, rolling bearing elements, rotating electric field, fluid flows, combustion events, structural resonance and angular rotations [3].

The significance of vibration levels in the operation of rotating machinery is well established. They are a direct indicator of machine's operating condition. High amplitudes, or changing amplitude pattern, can indicate to an equipment failure. They can also produce discomfort and ultimately safety hazard to operating personnel.

Vibration measuring instrumentation is now commonly employed on rotating machinery to identify characteristic levels and trends. In order to successfully full-fill their purpose, however, the vibration instruct-mentation must measure amplitudes and frequencies which are meaningful and truly indicative of the machinery's condition. These amplitudes and frequencies must, in turn, be compared against «standard" reference values which establish the condition on status of the machinery [4].

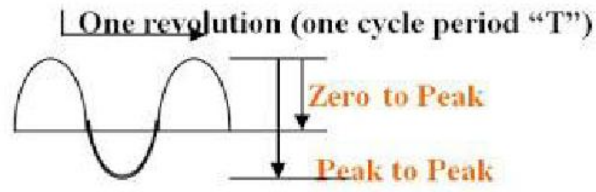
The main characteristics of the vibration signals measured are

1. Amplitude
2. Frequency
3. Phase

Author					Sign		List	Lists
Chief	Yakushenko O.S.							
Adviser								
N.Supervis								
D.Chairm.	Kulyk M.S.							

Amplitude

Amplitude is a measure of how severe the vibration is and can be expressed in 3 different ways: Peak to peak, Zero to peak , depending on what signal we are measuring.



Vibration measurement either in means of displacement, velocity or acceleration, vibration displacement is mostly measured as Peak to Peak, a measure of the absolute excursion of the rotor or machine casing in micrometers. Vibration velocity and Acceleration are measured as Zero to Peak.

Units used are "inches per second" or "millimeters per second" for velocity or in terms of "G" or "meters per second per second" for acceleration [5].

Frequency

Frequency is the measurement of speed of an object vibrating and it is also used to identify the location of vibration source of vibration. Normally Frequency is expressed in shaft rotative speed. If the vibration is at the same frequency as the shaft speed, this will be 1 time shaft speed (1X), if it is twice then it's (2X). Another definition of frequency is cycles per unit of time or Hertz.

Phase

Phase is a simple timing relationship between 2 events which may be 2 vibration signals for Relative Phase measurements or a vibration signal and a key phase reference signal for Absolute measurements. Both these are important vibration signal properties.

To measure the phase between 2 signals, both vibration signals should be at the same frequency and in the same units of measurement. both displacements, both velocity and both acceleration. Both signals may be taken as the reference and the relative phase is expressed as an angle between 0o and 180o leading or lagging.

ABSTRACT

Object: Gas turbine plant of simple cycle with development of vibration monitoring systems

Aim: to develop an efficient and accurate vibration monitoring system and chose and mount vibration sensors for the designed GTP to ensure accurate reading of vibration values

Gas turbine plants operation parameters are affected by the design of the GTP and the ambient conditions temperature and pressure, a thermodynamic calculation were done on the designed GTP to calculate the change in temperature and pressure through different sections of the designed GTP.

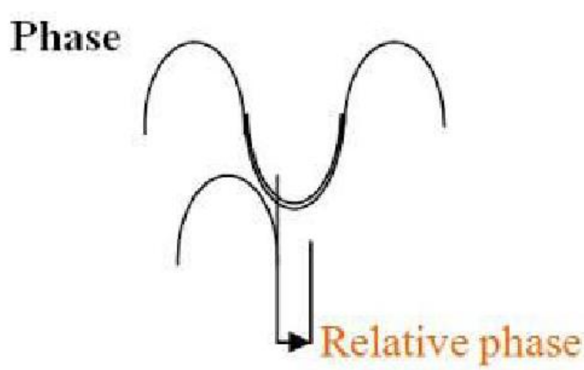
Based on the thermodynamic calculations a gasdynamic calculation were done to determine the optimal diametrical sizes on each section of GTP, and to calculate the compression work distribution between LPC and HPC and to determine the number on stages for each of LPC, HPC, LPT, HPT and PT.

A strength calculation were done on GTP elements to determine the tension stresses from the centrifugal forces and the bending stress from gaseous and centrifugal forces the first strength calculations were done on turbine rotor blade on four cross sections on root 25%, 50% and 75% of the blade length, the second strength calculations were done on the turbine disc.

As any rotary machine faults in any of the GTP elements can happen but with a good system for monitoring the health of the GTP parts the faults can be either predicted or prevented before happening this system is Vibration Monitoring system this part is focused on better understanding of vibration Monitoring and the development of the system itself. Vibration sensors were chosen taking into account the vibration limits of the GTP elements and the purpose of the sensor with working limits of the sensor itself.

Key words: Gas turbine plant, Vibration monitoring system

					Sign	List	Lists
Author							
Chief	Yakushenko O.S.						
Adviser							
N.Supervis							
D.Chairm.	Kulyk M.S.						



Types of Sensors used in Vibration Measurement

1. Acceleration Sensors (Accelerometers)
2. Velocity Sensors
3. Proximity Probes (capacitance or eddy current)

Acceleration sensors (Accelerometers)

Capacitive accelerometers:

Used generally in those that have diaphragm supported seismic mass as a moving electrode and one/two fixed electrodes the signal generated due to change in capacitance is post-processed using LC circuits etc., to output a measurable entity[5].

Piezoelectric accelerometers:

Acceleration acting on a seismic mass exerts a force on the piezoelectric crystals, which then produce a proportional electric charge. The piezoelectric crystals are usually preloaded so that either an increase or decrease in acceleration causes a change in the charge produced by them. But they are not reliable at very low frequencies [5].

Advantages:

1. Simple to install
2. Good response at high frequencies
3. Stand high temperature
4. Small size

Disadvantages:

1. Sensitive to high frequency noise

2. Require external power
3. Require electronic integration for velocity and displacement

Velocity sensors

Electromagnetic linear velocity transducers:

Typically used to measure oscillatory velocity a permanent magnet moving back and forth within a coil winding induces an emf in the winding this emf is proportional to the velocity of oscillation of the magnet, this permanent magnet may be attached to the vibrating object to measure its velocity [5].

Electromagnetic tachometer generators:

Used to measure the angular velocity of vibrating objects they provide an output voltage/frequency that is proportional to the angular velocity, DC tachometers use a permanent magnet or magneto, while the AC tachometers operate as a variable coupling transformer, with the coupling coefficient proportional to the rotary speed [5].

Potentiometric accelerometers:

Relatively cheap and used where slowly varying acceleration is to be measured with a fair amount of accuracy.

In these, the displacement of a spring mass system is mechanically linked to a viper arm, which moves along a potentiometric resistive element, various designs may have either viscous, magnetic or gas damping [5].

Reluctive accelerometers:

They compose accelerometers of the differential transformer type or the inductance bridge type, AC outputs of these vary in phase as well as amplitude. They are converted into DC by means of a phase-sensitive demodulator [5].

Servo accelerometers:

These use the closed loop servo systems of force-balance, torque-balance or null-balance to provide close accuracy, acceleration causes a seismic mass to move, the motion is detected by one of the motion-detection devices, which generate a signal that acts as an error signal in the servo-loop [5].

The demodulated and amplified signal is then passed through a passive damping network and then applied to the torqueing coil located at the axis of rotation of the mass. The torque is proportional to the coil current, which is in turn proportional to the acceleration [5].

Advantages:

1. Simple to install
2. Good response in middle range frequencies
3. Stand high temperature
4. Does not require external power
5. Low cost

Disadvantages:

1. Low resonant frequency and phase shift
2. Cross noise
3. Big and heavy
4. Require electronic integration for displacement

Proximity sensors

Eddy Current Sensor Probe:

Eddy currents are formed when a moving (or changing) magnetic field intersects a conductor, or vice-versa, the relative motion causes a circulating flow of electrons, or currents, within the conductor, these circulating eddies of current create electromagnets with magnetic fields that oppose the effect of the applied magnetic field [5].

The stronger the applied magnetic field, or greater the electrical conductivity of the conductor, or greater the relative velocity of motion, the greater the currents developed and the greater the opposing field Eddy current probes sense this formation of secondary fields to find out the distance between the probe and the target material [5].

Advantages:

1. Measures static and dynamic displacements
2. Good response at low frequencies
3. Small and low cost

Disadvantages:

1. Electric and mechanical noise
2. Bounded by high frequencies
3. Not calibrated for unknown metal materials
4. Requires external power and difficult to install

It is not possible to avoid mechanical oscillations in the technical praxis. Specialist estimating importance of oscillation manifestation, issues from the need of machine reliable performance. Every assembled machine has unique vibration signature, when it works. The changes of this signature in time indicate abnormal behavior or structural stiffness changes. Changes of vibration allow revealing changed health of machine.

Bearings

Monitoring of rolling bearing vibrations is generally possible only on bearing house. It can be distinguished between three stages of bearing life. Let us consider bearing to be perfect, immediately after production and trial run. This first stadium can last for relatively longtime, in case of right operation and maintenance.

Beginning of wear or fault development starts the second stage. Fault development can be slow, in case of right using (loading) of a bearing vibrations of bearing components copy trend curve. Slow development of wearing is guided with small variance of vibration parameters. Components are visible only when using special methods developed for rolling bearing diagnostics.

Growth of variance and quantities them self indicates serious development of fault. It indicates the third stage. Bearing components start to be visible in the spectrum of vibrations measured on the bearing house .When vibrations are measured far from bearing housing, there is small probability of bearing frequency com-opponents visibility. If some occurs, particular part is in unsatisfactory condition – it is very close to failure.

With aim to detect the second (it means an earlier) stage of fault development, it is suitable to implement special methods, as for example, enveloping of filtered signal.

It is not guaranteed, but more probable, that in the spectrum of enveloped signal will be visible bearing fault component with small amplitude, what indicates fault development. Growth of this component during the work hours depends on the fault development, on the signal transfer path properties and on distance from the bearing to the sensor.

Bearing frequencies are important indicator of health of engine shaft support. Their occurrence in spectrum of enveloped signal indicates the second stage of wear/fault development. Bearing frequencies occurrence in spectrum of raw signal indicates the third – dangerous development stage.

Gas turbine necessarily need for their performance lubrication and cooling, fuel supply, working regimes control. Pressure and pumps, regulators, speed sensors and other equipment are driven from the engine shafts.

They have almost the same importance, as above mentioned bearings. Their faults are, by experiences, developed continuously and they are early detectable by vibration measurement.

Blades/vanes or whole/completed stages are threatened mostly by solid material intrusion from atmosphere. Grains of sand do not damage them immediately, but long term abrasion results in damage.

It can be detected early, because they are well accessible for visual inspections. Following compressor and turbine stages are controllable through visual inspection holes.

This check can be done during maintenance. Vibration diagnostics of higher compressor stages that have large number of small vanes, needs larger frequency range. Higher sampling frequency together with requirement of high resolution leads to big size of files, where measured data are recorded.

Gas turbine changes ongoing during the time of duty can lead to decreasing of rotating mass by singe, melting, breaking, etc., or to increasing by pollution, soot, welding of material occurring in hot gas stream, etc. It leads to changes in engine main shaft balance. Unbalance is not dangerous, until it reaches standardized value according to ISO 1940. However, the higher unbalance is harmful for shaft support (it means bearings).Rotor vibrations can have many harmonics. They excite vibrations of surrounding components.

If some harmonic has frequency close to natural frequency of engine component, resonance can occur. Shaft vibrations absolute size evaluation is possible according to engine producer directions for operation. When there is known mass of rotor, then technical standard can be implied, in the meaning of rigid rotor balance criteria. In some cases, elastic shaft supports are used in gas turbine.

Relations between vibration signal properties and engine health are expressed by health symptoms or indicators. Symptoms can be signal parameters itself, quantities calculated from raw signal and/or recalculated signal. Time signal waveform is meant as a raw signal some kind of instrumentation gives direct transformation to the frequency domain in real time during measurement.

Original signal from accelerometer is acceleration, and it is disadvantage that any integration and/or differentiation introduce some signal deformation. Basically, vibration velocity (acceleration integrated in the time) represents good correlation to energy carried by vibration motion. However, vibration acceleration can disclose some phenomena that are non-visible in signal of velocity. Both quantities are useful in complex approach to vibration analysis. Particular technique prefers one of these parameters.

3.2 Assign of vibration limit value

The vibration limits are the physical limits for any equipment where if the vibration exceeds a fixed value the operation of the equipment becomes dangerous and may lead to serious issues in the structure and the components of the equipment. The vibration Limits are usually set in factory in accordance with the International Standards Organization (ISO) recommendations standards and after running tests and also after installation of the equipment as if the equipment is not correctly installed it can lead to increase of vibration.

It is recommended to shut down the compressor if there is a sudden change in vibration velocity at a speed of two bearings of one rotor, adjacent bearings or two components of the vibration velocity of one support at 1 mm / s and more from any entry level.

The compressor is not allowed to operate if there is a continuous accelerating in any component of the vibration velocity of the bearing support at 2 mm / s; and more.

In the presence of low-frequency vibration (sub-harmonic rotation speed) exceeding 0.5 mm / s: compressor operation is not valid.

In Table 3.5 limit vibration levels of different GTU are shown [6]. Using them limit vibration levels of designed GTU was chosen. The warning is issued if vibration is reached level 30 mm/s. If vibration is reached level 40 mm/s GTU must be stopped.

Table 3.5 – Vibration limits of gas turbine drives

№ п/п	Type of equipment	Type of GTP	Normative document	Estimation and value of the parameter			
				Ok	Reasonable	Admissable***	Unacceptable
Vibration of bearings of mm / s bearings (in a frequency band of 10, .. 1000 Hz)							
1	ГТ-750-G	ГТ-750-6	ТУ 24-2-017-69	<4,5	4,5...<7,l	7,1...<11,2	a11,2
2	-ГТ-6-750	ГТ-6-750	ТУ 24-2-382-72	-	<6.7	6.7...<13	>_13
3	ГТН-6	ГІЗ-І-6	Тц 24-2-382-72	-	<6,7	6,7...<13	>13
4	f7K-f0	ГТК-10	ТцI08.1011-81	54,5	4,5...<_7,1	7,1..51,2	1 1 , 2
5	Г7К-10I	iV153002	Контракт No 50-4 1/52196	-	512,7	12,7...<25,4*)	~25,4*
6	ГТН-16	ГТН-16	Б-735300ИЭ	-	<12.7	12.7...Q5,4	225,4
7	ГТК-25I	MS5002	Контракт Nя 50-0401/22008	-	512,7	(2,7...~25,4")	>35,4*
8	ГТН-25	ГТН-25	ТУ24-03.1412-9i	-	<12,7	12,7...<25,4*1	5,4*
Vibration of GTP cases, mm / s (according to the readings of automatic control systems in the frequency band 50 ... 500 Hz)							
9	Г1IA-Ц-6,3	НК-12СТ	Тц1-01-0843-88	<28	28...<45	45*)...<60*1	>_60
10	ГПА-Ц-6,3А	Д-336-2	336-2.00.00.000 ТУД	<28	28...<45	45*)...<60*)	>_60*
11	П1А-Ь,3С	дТ7l	К061108000	<28	28...<45	45*)...<60*)	?60*
12	ТТIA-Ц-8А	АИ-336-8	3380000000-б-ТУ	<28	28...<45	45*)...<6f1*)	Z60*
13	ТТК-10IA	АИ-336-l0	(скспсриментапьний)	<28	28...<45	45*)...<60*)	>_60*
14	ГГ1С-1оС	дН70	751108000РЭ	<13	13...<l6	16*),, <1g	>_18•
15	ГПУ-10	ДЖ-59Р	ТУ5-433.9657-87	<18	18...<25	25*1...<35*7	235*
16	ГIIA-Ц-l6	I3К-16СТ	ТУ 26-12-632-84	<28	28...<45	45s1...<60*1	260*
I7	ГПУ-16	ДЖ-59Л	ТУ5-433.9778-88	<(8	18...<25	25*)...t35*	~35*
18	гПУ-IЬК	Дц-7i	К1Ы08000РЭ	<13	13...<15	15*)(16)**)...<!8	r18*•
19	ГГL4-ц-16С	ДГ-9оЛ2	Г90I08000ИЗ	<17	t7...<20	20*)...<26	?26*)
20	ГТ1А-25С	ДН-80	8021080007У	<13	13,..<l7	17*)(20)'*) ..c22*	r22*)

* Warning and alarm alarm activation levels

** Power limit level.

3. 3 Vibration monitoring of designed GTP

3.3.1 Choice of sensors

Every rotating machine has its own unique vibration characteristics. Problems such as unbalance or rolling element bearing damage cause the vibration characteristics to change. In most cases the manufactures of such machinery will be able to tell you what the normal vibration characteristics are.

As with most all other industrial sensors and transducers, it is critical that the vibration sensor, or accelerometer selected is well suited to condition monitoring applications. This is necessary to ensure that the acquired data is accurate and fit for purpose. Using an inappropriate device could result in unexpected plant failure or a reduction in the working life of a piece of machinery.

For this Vibration Monitoring system i want to continuously monitor the vibration levels of the GTP so i will choose the Sensor type Accelerometer I will be using Accelerometers with Multi-Axis measuring to reduce the number of sensors used and as result reduce the cost of the system.

Such sensors can be used in designed machine.

Biaxial Industrial ICP® Accelerometer Model 605B01 [7]

Biaxial accelerometer contain two sensing elements, each sensing axis contains a dedicated, Built in, low noise, microelectronic signal amplifier whose output signal is delivered to an independent cable lead or connector pin. A technical specification of this sensor included in Table3.5.

Table 3.5-Technical Specifications of chosen sensors [7]

Technical Specifications

Model Number	604B31	605B01	629A31
Performance			
Sensitivity (±20 %)	100 mV/g 10.2 mV/(m/s²) [2]		N/A
Sensitivity (±5 %)	N/A		100 mV/g 10.2 mV/(m/s²) [2]
Measurement Range	±50 g ±490 m/s²		
Frequency Range (±5%)	N/A		144 to 120,000 cpm 2.4 to 2 kHz [4]
Frequency Range (±10%)	N/A		102 to 300,000 cpm 1.7 to 5 kHz
Frequency Range (±3 dB)	30 to 300,000 cpm 0.5 to 5 kHz [4]		48 to 480,000 cpm 0.8 to 8 kHz
Resonant Frequency	600 kcpm 10 kHz [5]		1,200 kcpm 20 kHz [5]
Broadband Resolution (1 to 10 kHz)	350 µg 3,434 µm/sec² [5]		100 µg 981 µm/sec² [5]
Non-linearity	±1 % [6]		
Transverse Sensitivity	≤5 %		≤7 %
Environmental			
Overload Limit (Shock)	5,000 g pk 49,050 m/s² pk		
Temperature Range	-65 to 250 °F -54 to 121 °C		
Enclosure Rating	IP68		N/A
Electrical			
Settling Time (within 1% of bias)	≤ 2.0 sec		≤ 3.0 sec
Discharge Time Constant	≥ 0.3 sec		≥ 0.2 sec
Excitation Voltage	18 to 28 VDC		
Constant Current Excitation	2 to 20 mA		
Output Impedance	<150 ohm		<100 ohm
Output Bias Voltage	8 to 12 VDC		
Spectral Noise (10 Hz)	8 µg/√Hz 78.5 (µm/sec²)/√Hz [5]		7 µg/√Hz 68.7 (µm/sec²)/√Hz [5]
Spectral Noise (100 Hz)	5 µg/√Hz 49.1 (µm/sec²)/√Hz [5]		2.8 µg/√Hz 27.5 (µm/sec²)/√Hz [5]
Spectral Noise (1 kHz)	4 µg/√Hz		1 µg/√Hz

Model Number	604B31	605B01	629A31
Physical			
Size - Diameter	1.38 in 35.1 mm		N/A
Size - Length	N/A		1.5 in 38.1 mm
Size - Width	N/A		1.5 in 38.1 mm
Size - Height	1.00 in 25.4 mm		0.82 in 20.8 mm
Weight	4.4 oz 124 gm	3.9 oz 110.6 gm	4.9 oz 139 gm
Mounting	Through Hole		
Mounting Thread	1/4-28 Male	1/4-28 UNF	1/4-28 Male [1]
Mounting Torque	2 to 5 ft-lb 2.7 to 6.8 N-m		
Sensing Element	Ceramic Shear		
Housing Material	Stainless Steel		
Sealing	Welded Hermetic		
Electrical Connector	4 pin MIL-C-26482	3 pin MIL-C-5015	4 pin MIL-C-26482
Electrical Connection Position	Side		
Electrical Connections (Pin A)	X-axis		
Electrical Connections (Pin B)	Y-axis		
Electrical Connections (Pin C)	Z-axis	Ground	Z-axis
Electrical Connections (Pin D)	Ground	N/A	Ground

3.3.2 Choice of points for vibration measuring

The mounting of a vibration sensor (accelerometer) directly impacts on its performance. Incorrect mounting may give readings that relate not only to a change in conditions but also to the instability of the sensor itself – therefore making the sensor's readings unreliable. As such, correct mounting of a vibration sensor is vital to ensure true readings. To detect any faults in machine components, vibrations sensors should be mounted in locations that ensure horizontal, vertical and axial movement are measured effectively.

The sensor Biaxial Industrial ICP® Accelerometer Model 605B01 is mounted to the first bearing house for vertical measurement, this measures velocity mm/sec (Peak or RMS) to detect looseness and problems with structural rigidity and/or foundation.

The sensor Biaxial Industrial ICP® Accelerometer Model 605B01 is mounted to the power turbine housing for horizontal and axial measurement this measures velocity mm/sec (Peak or RMS) to detect unbalance, and problems with structural rigidity and misalignment.

Accelerometers should be mounted onto a surface that is free from oil and grease as close as possible to the source of vibration. The surface should be smooth, unpainted, flat and larger than the base of the accelerometer itself. For best results, sensors should be mounted via a drilled and tapped hole directly to the machine housing.

LIST OF CONVENTIONAL ABBREVIATIONS

GTP- Gas Turbine Plant

GTU- Gas Turbine Unit

LPC-Low Pressure Compressor

HPC-High Pressure Compressor

CC- Combustion Chamber

LPT-Low Pressure Turbine

HPT-High Pressure Turbine

PT-Power Turbine

GTPP-Gas Turbine Power Plant

ACS-Automatic Control System

LP-Low Pressure

HP-High pressure

AC-Alternative Current

DC-Direct Current

					Sign	List	Lists
Author							
Chief	Yakushenko O.S.						
Adviser							
N.Supervis							
D.Chairm.	Kulyk M.S.						

Conclusions

After research of vibration properties for better understanding of the nature of the vibration measurement the limits of the vibration were chose for the designed GTP, the warning is issued if vibration is reached level 30 mm/s, if vibration is reached level 40 mm/s GTU must be stopped. Taking into account these limits the sensors were chosen, the sensor Biaxial Industrial ICP® Accelerometer Model 605B01 is mounted to the first bearing house for vertical measurement, and the sensor Biaxial Industrial ICP® Accelerometer Model 605B0 is mounted to the power turbine housing for horizontal and axial measurement. Accelerometers with Multi-Axis measuring were chosen to reduce the number of sensors used and as result reduce the cost of the system.

CONCLUSIONS

1. In Part one was considered under on the basis of data acquired by different calculations. In calculations different parameters are considered. For instance, in thermodynamic calculation parameters such as pressure, temperature and their effect on different elements of the engine were taken into account. Similarly, in gas-dynamic calculation geometrical parameters of GTE which includes, number of stages in compressor, turbine and power turbine were calculated along with other geometrical configurations.

2. in Part two a Strength calculation was performed on turbine elements disc and blades taking into account the data from the thermodynamic calculation and gas-dynamic calculation the calculation included a tension stress from centrifugal forces and the bending stress from gaseous and centrifugal forces, excluding the torsion stress from gaseous forces due to their insignificant value. The tension stress was calculated in four sections: on root and on distances 25 %, 50 % and 75 % of blade length. The material was chosen as ЭИ826.

3. After research of vibration properties for better understanding of the nature of the vibration measurement the limits of the vibration were chose for the designed GTP, the warning is issued if vibration is reached level 30 mm/s, if vibration is reached level 40 mm/s GTU must be stopped. Taking into account these limits the sensors were chosen, the sensor Biaxial Industrial ICP® Accelerometer Model 605B01 is mounted to the first bearing house for vertical measurement, and the sensor Biaxial Industrial ICP® Accelerometer Model 605B0 is mounted to the power turbine housing for horizontal and axial measurement. Accelerometers with Multi-Axis measuring were chosen to reduce the number of sensors used and as result reduce the cost of the system.

Author					Sign	List	Lists
Chief	Yakushenko O.S.						
Adviser							
N.Supervis							
D.Chairm.	Kulyk M.S.						

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Author					Sign	List	Lists
Chief	Yakushenko O.S.						
Adviser							
N.Supervis							
D.Chairm.	Kulyk M.S.						

CONTENT

					Sign	List	Lists
Author							
Chief	Yakushenko O.S.						
Adviser							
N.Supervis							
D.Chairm.	Kulyk M.S.						

					Sign	List	Lists
Author							
Chief	Yakushenko O.S.						
Adviser							
N.Supervis							
D.Chairm.	Kulyk M.S.						

INTRODUCTION

Ever since the first industrial revolution and transformations of the factories from hand production methods to machines, it was journey about looking for the best source of energy to power up the factories there was steam or water powered machines using the heat from coal or wood till the first successful gas turbine was built back in 1903.

Due to that invention the industrial revolution developed very fast and with this growth the demand on the fuels increased even higher we started using different types of fuels for all purposes for factories, electricity generation, transportation, etc… this high demand lead to a development in the methods of fuel transportation from stationary facilities such as gas wells or import/export facilities, and deliver to a variety of locations, with time these transportation methods became wide and more complex and we know today as Gas Pipelines or Gas Transport Systems.

These systems start including more consumers which lead to more complexity in the systems and bigger systems which included smaller systems like gathering systems, transmission systems, and distribution systems. Gas gathering system gathers the raw gas from production wells. It is then transported with large lines of transmission system pipelines that move gas from facilities to refiners, ports, and cities all over the country.

Lastly, the distribution systems consist of a network that distributes the product to houses and businesses, which also consist of two smaller systems the main distribution line, which are bigger lines that move the products of the gas close to cities, and the distribution pipelines, which are smaller pipelines that connect main lines into homes and businesses.

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Author							
Chief	Yakushenko O.S.						
Adviser							
N.Supervis							
D.Chairm.	Kulyk M.S.						