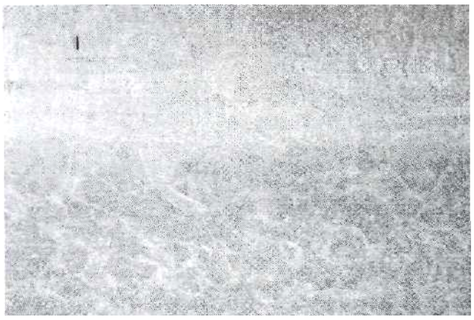

Fig. 6. Transitional area, × 100

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STEAM TURBINES WITH A LOW-BOILING WORKING AGENT

The subject of the article is the assembly of a steam-generator plant with a natural working agent. A method of calculation for steam turbines with a low-boiling working agent is offered, which accounts for the correlation between the adiabatic curve indication, pressure and temperature in the overheated vapor area.

Keywords: steam turbine, Freon, adiabatic curve indicator, low-grade heat, coolant.

Nowadays the problems of electric power production are becoming more and more essential. This is the result of the rising cost of energy sources (oil, gas, coal) and consequently their consumer prices are growing.

In the routine work of industrial enterprises (using various heat-carrying agents) a great amount of heat which could be used in some other production cycles is dumped and lost. These heat-carrying agents can vary from slightly warm sewage water with low temperatures to coal coking gases with the temperature up to four hundred degrees (Celsius). Other sources of heat can come from all kinds of plants and systems that dump heat in their working cycle. Some of these sources and their temperatures are listed in the table.

Heat-carrying agents

Heat-carrying agents	Temperature, °C
Sewage water	15–19
Industrial gases flow	250–300
Heating equipment temperature	30–100
Coal coking gas temperature	400–430
TV3-117 engine oil	80–150
VR-14 reductor oil	70–80

Thermal power could be accumulated by means of thermal pumps. But thermal pumps cannot convert heat into other kinds of energy. And it is desirable to get from this heat the power that is easy to be transmitted for long distances, i. e. electric power.

Rather low temperature of most heat sources, their non-gaseous state, such as that of oil in helicopter systems, for example, as well as rather low pressure in gaseous sources do not allow direct application of their thermal energy. The energy is to be extracted by means of low-boiling working agents, such as in thermal pumps. After the heat is extracted from the source, its thermal energy is converted into mechanical energy by a rotodynamic machine-steam turbine.

Taking into consideration the ecological and economic requirements, the working circuit of the plant should be a closed system, i. e. the working agent is to be used many times over. As coolants are usually chemically active and dangerous compounds, this system permits to avoid environmental pollution by the direct ejection of the used coolant from the system. The possible harmful influence on human health is ten times less.

To put this closed system into practice it is necessary to install a component for circulation of the working agent inside the system. The component can be a pump (for the circulation of a liquid working agent) or a compressor (for the circulation of the agent in the form of overheated vapour). Unlike the pump, the compressor system doesn't require conversion of the working agent into liquid phase on its outlet from the turbine.

The installation (fig. 1) consists of the reservoir filled with a working agent, the pump, the evaporator, the turbo generator and the condenser. The coolant Freon R22 (the chlorinedifluoromethane) is considered as a working body. This Freon is the most suitable to the given system under its physical and chemical characteristics and it is widely used in the modern refrigerating equipment.

The system working cycle diagram shows four sections (fig. 2). The first section *1–2* shows the feeding of the working agent to the evaporator, thus increasing the Freon pressure in the system and slightly raising the temperature because of the losses in friction. The second section *2–3* shows the evaporation and overheating of the working agent in the evaporator at a constant pressure and the extraction of heat from the source. The third section *3–4* corresponds to the conversion of the working agent thermal energy into kinetic energy of the turbine shaft, while the temperature and pressure are getting lower. The fourth section *4–1* shows the condensation of the working agent to the initial parameters that were at the pump input, which is necessary for the next cycle.

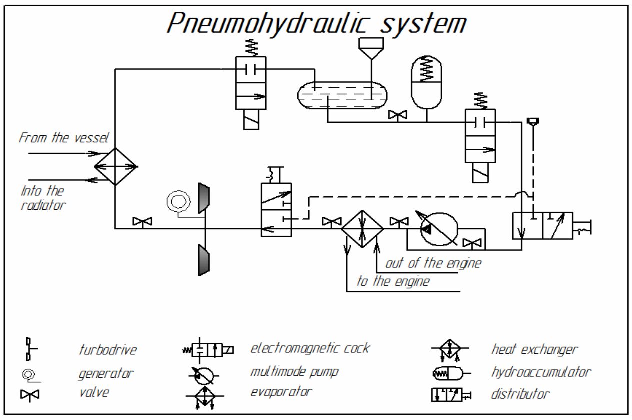

Fig. 1. Pneumohydraulic system using helicopter oil system as a source of heat

The basic points influencing the system power output are points *3* and *4* (fig. 2). Point *4* is the starting point of the working agent condensation and is determined by the type of the condenser and the substance used for heat extraction. Point *4* can be in a low temperature sector if the heat is extracted by a different coolant. But it is more effective to use the environmental media, such as air and, if possible, water. Point *3* is the point of optimal overheat of the working agent. It reflects the optimal overheating of the agent at the given evaporation pressure and is determined by the position of point *4*. Maximum temperature at point *3* is limited by the highest possible temperature of a given coolant. The most suitable heat sources can be determined by evaluation of the working agent parameters at point *3*. When air is used as a condenser (steam parameters at point *4* are 11.8 bar and 36 °C) the system's optimal temperature at point *3* will be 100 °C at 30 bar pressure. For that variant of the system all sources with the temperature of 110 °C and more are suitable. To make the system work more effectively a part of the working agent can be converted into liquid so that there is more used energy to be transferred at heating of the working agent. Modern steam turbines can work with 20 % of liquid agent, but there are still some problems to be solved, as there are no reliable methods of calculation of the percentage of phase correlation for the flow of the axial turbine.

For pipelines with a small amount of working agent and large pressure break the optimal efficiency can be achieved when using an axial-flow turbine with partial working agent feeding.

The capacity at the turbine and compressor shaft is derived from adiabatic work $L_{ад}$, working agent consumption, turbine efficiency η_m or the compressor $\eta_к$ [1–3]:

$$N = G \cdot L_{ад} \cdot \eta_m,$$

where N is the capacity at the turbine shaft, W; $L_{ад}$ is adiabatic work, kJ/kg; G is working agent consumption, kg/s; η_m is turbine efficiency.

Adiabatic work at the turbine and compressor shaft is derived from working agent parameters

$$L_{ад.m} = \frac{k}{k-1} \cdot R \cdot T^*_{Г} \cdot \left(1 - \frac{1}{\left(P_1/P_2\right)^{\frac{k-1}{k}}}\right),$$

$$L_{цбк.s} = \frac{k}{k-1} \cdot R \cdot T_{вх.к} \cdot \left(\pi_{цбк}^{\frac{k-1}{k}} - 1\right),$$

where $L_{ад.m}$ is turbine adiabatic work, kJ/kg; R is gas constant, J/kg·K; $T^*_{Г}$ is gas temperature at the turbine input, K; P_1 and P_2 are gas pressure before and after the turbines, Pa; k is isentrope index; $L_{цбк.s}$ is compressor adiabatic work, kJ/kg; $\pi_{цбк}$ is degree of pressure increase in the compressor; $T_{вх.к}$ is gas temperature at the compressor input, K.

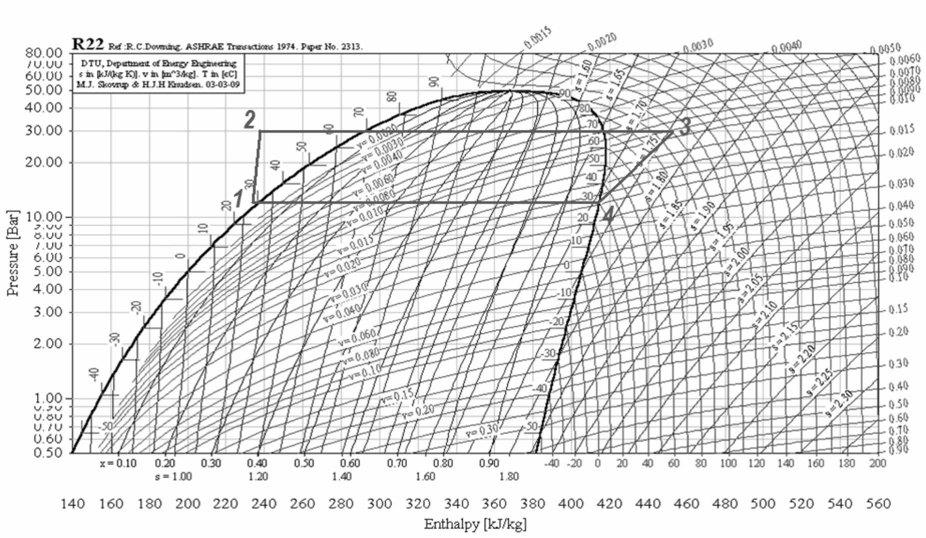

Fig. 2. Plant cycle in *lg P–I* coordinates

Adiabatic work also determines adiabatic speed, sound speed and the temperature at the turbine or compressor outlet:

$$c = \sqrt{2 \cdot L_{\text{ад}}}\ ; \quad a_{\text{кр}} = \sqrt{2 \cdot \frac{k}{k+1} \cdot R \cdot T_{\text{вх}}}\ ;$$

$$T_{\text{вых.т}} = T_{\Gamma} - \frac{L_{\text{ад}.m}}{R\frac{k}{k-1}}\ ; \quad T_{\text{вых.к}} = T_{\text{вх.к}} + \frac{L_{\text{цбк}.s}}{R\frac{k}{k-1}}\ ,$$

where c is adiabatic speed, m/s; $\alpha_{\text{кр}}$ is sound speed in the nozzle, m/s; $T_{\text{вых.к}}$ is gas temperature on its outlet from compressor, K; $T_{\text{вых.т}}$ is gas temperature on its outlet from turbine, K.

These parameters are essential for the kinematic calculation and the design of the directing unit blades, and the compressor driving wheel, and the nozzle assembly, and the turbine driving wheel.

The processes taking place in the compressor and the gas turbine with traditionally used working agents are studied in full and calculation algorithms are valid to obtain the precise parameters for the flow section.

The particular feature of the low-boiling working agents is their variable adiabatic curve indicator depending on the temperature and pressure [4].

There is no analytical dependence of the adiabatic curve indicator on the temperature, there are only tabular values.

Using scientific literature [4; 5] we can draw a surface showing the adiabatic curve indicator changes depending on the temperature and pressure (fig. 3).

The calculation of the adiabatic work at the turbine shaft and the temperature on the outlet, taking into account the variable factor of the adiabatic curve, is also given:

$$L_{\text{ад}.m} = \frac{k'}{k'-1} \cdot R \cdot T_{\Gamma}^{*} \cdot \left(1 - \frac{1}{\left(\frac{P_1}{P_2}\right)^{\frac{k'-1}{k'}}}\right),$$

$$T_{\text{вых}.m} = T_{\Gamma} - \frac{L_{\text{ад}.m}}{R\frac{k'}{k'-1}}$$

where $k' = f(P, T)$.

As the surface has a complex profile with numerous peaks, it cannot be described taking into account only one analytical dependence.

The calculation is based on final differences method, the sector being divided into small pressure values, with the following summing up of the adiabatic work values for each segment.

The calculation results are given in fig. 4 and 5.

The most influencing changes of the adiabatic curve indicator are found in the temperature zone which is close to the working agent decomposition temperature, up to 5 % of adiabatic work and 20 % of the working agent temperature on its outlet. The influence of the adiabatic curve indicator also increases with the growth of pressure at the input.

Similar correlations can be worked out for the compressor. In compressor performance the influence of the variability indicator of the adiabatic curve will be more significant, as the working agent is the state conversion zone.

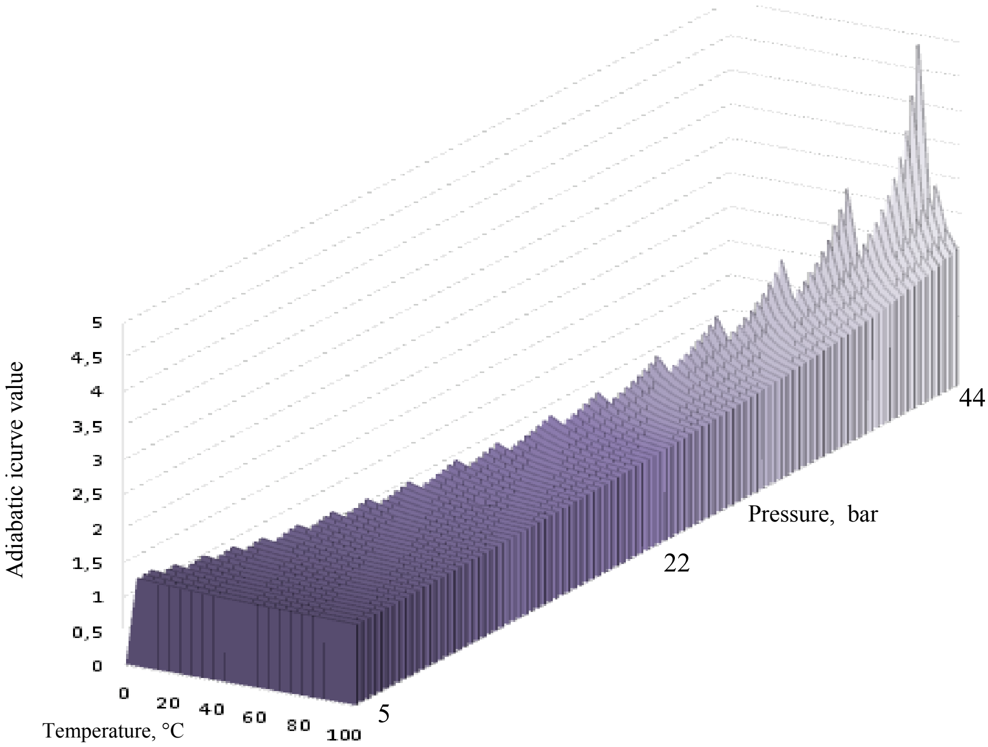

Fig. 3. Adiabatic curve indicator changes in dependence on the pressure and temperatures

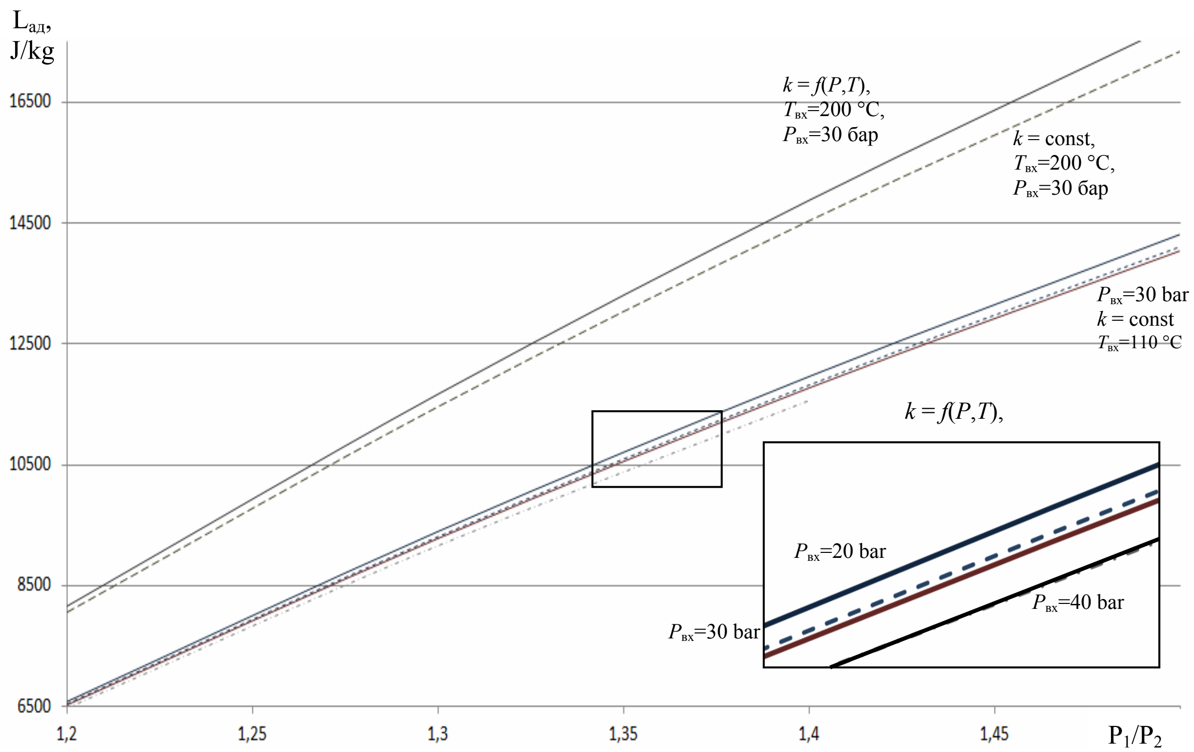

Fig. 4. Adiabatic work value at different parameters:
- - - - – k = const; ——— – $k = f(P, T)$

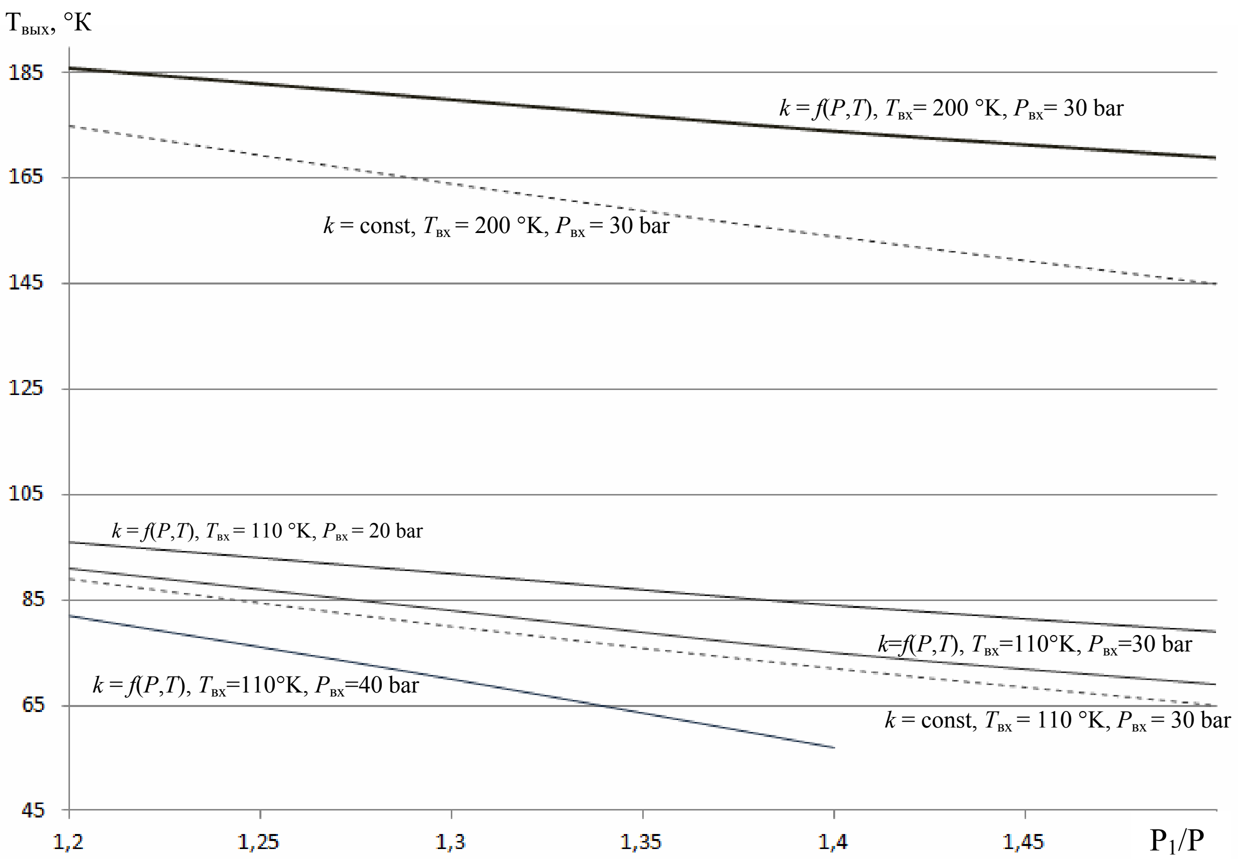

Fig. 5. Flow temperature values at the turbine escape at various adiabatic parameters:
- - - - – k = const; ——— – $k = f(P,T)$

The given calculations show that the registering of the adiabatic curve indications is necessary for more accurate determination of the thermodynamic parameters for the compressor and turbine performance. The calculation of the kinematic parameters of the flow in the system and the design of the flow sector and turbine or compressor blades should also be carried out taking into account the variability of the adiabatic curve indicator.

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COMPUTERIZED TEST STAND FOR INVESTIGATION OF STRAIN WAVES IN COMPOSITE CONSTRUCTIONS

The paper presents a test stand for investigation of ramp loading of composite constructions such as beams and plates. This ramp loading was performed using a striker with a piezoelectric force transducer. The velocity of the striker is up to 40 mps. The registration of the measurements results is carried out using a computerized measurement system. The experimental results of contact shock force acquisition are stated with appropriate calculations of non-stationary bending and shearing strain waves in glass-fiber-reinforced plastic beams and plates.

Keywords: composite constructions, shock loading, strain waves, computer measuring system.

The percentage of composite materials content in modern military and civil aircrafts does not exceed 20. Increasing composites percentage up to 50 and more, like in Boeing-787 (introduced in 2007) revealed a number of drawbacks because of the insufficient shock strength of composite materials being used. The matter doesn't concern the damage caused by some extreme situations inside an aircraft but it concerns normal shock loads for constructions made of traditional materials. Thus, at present time it is necessary to create crash-proof construction elements made of fibrous-layered composite materials like glass-fiber and coal-plastic. The application of these materials is not possible without reliable expert evaluations obtained by numerical simulation combined with experiments.

Carrying out shock tests gives more empirical data than carrying out static tests. Moreover, the experiments on shock loading and non-stationary deformation even for simple constructions like beams are complicated by the lack of complete standards for test stands, sensors, instruments and test techniques. So, experimental data are required to be accurate and reliable.

The present paper proposes an automated shock loading test stand with the computerized measuring system based on the LabVIEW virtual instrumentation from the National Instruments Corporation. The developed impact force and dynamical strain acquisition procedure provide sufficiently accurate experimental data to design crash-proof composite constructions.

Automated shock loading test stand. The scheme of the automated test stand for studying beams and plates non-stationary deformation processes under lateral shock loading is shown in fig. 1. It consists of loading device, impact rate acquisition system, and strain gauging system, dynamic force measurement system and computerized measuring system.

The loading device is a light gas ballistic installation for ramp loading of objects with the strikers having mass from 5 to 100 g providing shock loading rate up to 40 mps [1]. The striker rate was measured with drag-type sensors during its flying between magnets N1 and N2 with the inductance coils L1 and L2. Beams and plates non-stationary surface deformations were measured by foil strain gauges KF-4P1-5-100-B-12.

The novelty of the dynamic force measuring system is in using the striker sensor [1] to measure impact contact force including its magnitude, duration and form change in the contact zone between the striker and an object. This sensor is made of piezoelectric tablets that have the diameter 8 mm and are 2.8 mm thick.