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Institute of mechanics and power engineering

The department of heat power engineering systems

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MASTER'S THESIS

ANALYSIS AND CALCULATION OF HEAT DISSIPATION PROCESS OF OSCILLATIANG AIR STREAM

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THE TASK FOR FINAL QUALIFYING WORK
(in the form of master's thesis)

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4 The content of the final qualifying work

4.1 Introduction of heating system of engine

4.2 The heat transfer enhancement technology

4.3 Analysis and calculation of heat dissipation process of oscillating air stream engine

4.4 The equipment of the experiment

4.5 The energy chain of the experiment equipment

4.6 The analysis of the experiment results

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ABSTRACT

Master's work contains 85 pages, 34 figures, 7 tables, 56 equations, 30 references.

HETER, AIR STREAM, THE HIDRAULIC PIPING HEATER,

Almost all engineering the systems of the building, providing a comfortable the parameters of the microclimate in the room, different types of heat exchangers are used apparatus'. This applies not only to air conditioning systems, but also to radiator heating and air conditioning systems heating, zone heaters, systems heat recovery, etc. In the Central they are most often found the use of heat exchangers (heaters) of the "liquid – air" type, practically completely replacing outdated systems "steam-air". So without exaggeration we can say that the heater, as the rule is the main element of systems air conditioning. In the simplest supply systems have at least one heater, in which the hot water gives its heat supply air. Any heater is an object with distributed parameters with significant inhomogeneity of air temperatures and coolant. The temperature changes along the tube within one stroke, between moves and between rows. Unevenness of the air temperature at the outlet of the heater can be tens of degrees and approach to half of the temperature difference of the coolant.

Due to the distribution of parameters mathematical description and analysis of dynamic characteristics of heaters are extremely complicated. Given the fact that in systems air conditioning supply air, forced driven by a fan, it moves after the heater is turbulent and active mixed, then for this reason in the future it is appropriate and reasonable to consider the heater as an object with concentrated parameters.

As a rule, the fan heater operates in several modes. Each is characterized by water consumption, cooling rate, and air consumption. It is possible to connect to a European-class Central heating line (60 ° C) and a special 150 ° C steam boiler.

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A heater is a device used for heating air. According to the principle of operation, it is a heat exchanger that transfers energy from the heat carrier to the flow of the supply jet. It consists of a frame, inside of which there are dense rows of tubes connected in one or more lines. A heat carrier — hot water or steam-circulates through them. The air passing through the frame section receives heat energy from the hot tubes, so that it is transported through the ventilation system already heated, which does not create the possibility of condensation or cooling of the premises.

In this paper, tests of heaters are carried out for the purpose of definition valid heating capacities and their resistance and comparisons of these indicators with design or catalog data. Exact heat technical and aero-dynamic tests of heaters for the purpose of definition of surface-area factor and resistance are made in laboratories.

CONTENT

INTRODUCTION

The current stage of development of the world economy is directly related to increasing energy efficiency and reducing energy consumption in all spheres of activity, due on the one hand to energy security, and on the other-to environmental factors of nature pollution by waste and greenhouse gases [1].

Energy consumption of buildings and structures depends to a large extent on indicators that characterize the microclimate of premises and affect the health, productivity and comfort of people in them. At the same time, a well-designed heating and ventilation system of the building allows saving up to 40% of the resources spent on maintaining the micro-climate [2].

Energy-saving measures in relation to them are measures that ensure their minimum possible consumption of fuel and other energy sources.

To date, various technologies for heating buildings and structures have been developed and widely used, based on the use of numerous heating devices that differ in ways of transferring heat to the room air:

- convectors (water, electric, etc.) installed on the walls of rooms (usually under the window) and designed to heat the room air by using the effect of natural convection;

- underfloor heating (water, electric, etc.), providing floor heating and intended for heating the room air both by using the effect of natural convection and by radiation exchange with surrounding objects and fences;

-fan heaters that provide heating of the air and its artificial convection by inducing the fan to move;

- infrared heaters installed, as a rule, under the roof and providing radiation heating of surrounding objects and fences with subsequent convective transfer of heat to the air of the room, etc.

1 Introduction of heater system of engine

A heater is a device that serves to heat the air flow in a heating and ventilation system. It is an indispensable device that can heat large warehouses, offices and homes. The main advantages are its power and performance. Contemporary heater is composed of the side flaps (which should be removable), tube sheets and the heat-transfer elements.

Heaters are classified into two types:

Air heaters-these include the following types of heaters: water, steam, electric and oil.

Figure 1.1 – The tubular heat exchanger

Types:

Water heaters are most often used in heating systems of huge premises. Its main task is to heat the building in a short time. It is also highly safe and economical. A water heater with a fan is used in heating systems that are installed in greenhouses, farms, sports halls and huge stores.

Steam heater-used in air conditioning, ventilation and heating systems. The temperature of the coolant should not exceed 180 C degrees. Steam heaters are made of carbon and stainless steel. Electric heaters-have less power than their water counterparts. These heaters are not suitable for the heating system of large rooms no more than 100 m2. The electric heat exchanger dries the air very much, but there is also a positive quality of this device, it is compact and it can be easily installed in a residential apartment. Freon and water coolers are heat exchangers for supply ventilation. This equipment is used for air cooling.

Today, industrial heaters and water heaters with a fan are widely used: in enterprises, factories, warehouses, offices and retail halls, which are installed in heating and air systems. There are two types of heaters: Industrial steam CPSC; industrial water Heater. The oil heater is used in heating systems. Use it for heating small rooms. The advantage of this heater is that it does not dry the air, and dry air is known to be harmful to health, especially for young children and the elderly. The disadvantage of such a heater is that it will take more time to heat the room than for example a thermal convector. Heaters for ventilation are widely used in enterprises and warehouses in ventilation systems for heating the outdoor air flow, in the winter season. [3]

1.1 Current situation and progress of internal and external research of heater system of air- heater engine

1.1.1 External research situation

At this stage, the task is to review modern heat exchange systems. A brief history of heat exchangers is given. The content of such concepts as heat exchange and heat exchangers is revealed. The types of heat exchangers and their principle of operation are also considered. Heat exchangers are an integral part of our daily life. They are used where cooling or heating of liquids or gases is necessary. Heat exchange systems take place in machine, hardware and shipbuilding. Heat exchangers are also used in the chemical and pharmaceutical industries for heat treatment of products or for maintaining a constant temperature of process water. In recent years, the role of heat exchangers in the areas of efficient energy use and the application of new types of energy has significantly increased. At the end, the prospects for the development, improvement of heat exchangers and their application are considered. Keyword: modern heat exchange systems; heat exchange; types of heat exchangers; prospects for the use of heat exchange systems. All heat exchange systems are based on the principle of heat exchange. Heat transfer is a self-voluntary (i.e., performed without compulsion) process of heat transfer that occurs between bodies with different temperatures. Heat exchangers are devices in which heat is transferred from one heat carrier to another. The number of heat exchangers and their technical characteristics is calculated for each system (boiler room, ventilation, heating, etc.) depending on the required output parameters (required temperature, operating pressure, average temperature head). There are three ways to transfer heat: thermal conductivity, convection, and radiation. Let's look at the history of heat exchangers to understand how they have changed and improved over time. Despite the fact that the rapid introduction of plate heat exchangers began only in the last century, the first mention of them were found in frescoes that date back to the sixth century BC. Heat exchangers of that time were used in ancient Greek baths. Since the soldiers in the baths were washed together with their lo-shad, it was unacceptable to cool the water by mixing cold and hot water to exclude the possibility of infection. The solution to this problem was the use of concave metal sheets that were immersed in containers with cold water. It is worth noting that even then the Romans were able to improve the device almost immediately by cutting metal sheets, which accelerated the cooling process of water. This is because the presence of barriers that cause the liquid to mix during its movement has a positive effect on the heat transfer process. Then, only hundreds of years after the fall of the Roman Empire, the exchangers returned, only this time to Mongolia, India, and the countries of Tibet. Here, a new property of the prototypes of heat exchangers appeared. They decided to neglect the coinage, because they thought it was just a decoration, but instead they introduced enamel coating, which led to higher anti-corrosion properties of the metals that were used to construct the plates. Until the end of the thirteenth century, the heat exchanger consisted of a single plate. The first multi-plate heat exchanger was manufactured in Russia. The new design made the device more compact and suitable for transport. Now we should move on to modern heat exchangers. After all, any production process causes thermal reactions and, consequently, the process of heat exchange. Therefore, efficient use and energy savings in enterprises are currently an important area of application of heat exchangers. For example, the use of heat energy transferred from cogeneration plants and heat production using solar and geothermal plants. According to the principle of operation, heat exchangers are regenerative, regenerative and mixing. In recuperative heat exchangers, heat from the hot medium is transferred to the cold through the dividing wall. Examples of recuperative heat exchangers are heaters, evaporators, and condensers. In regenerative heat exchangers, the same surface of a certain body (nozzle) is washed either with a hot or cold coolant. In the first period, the nozzle is heated by the heating medium, and in the second – it is cooled, giving previously accumulated heat to the heated medium. Examples of regenerative heat exchangers: air heaters for blast furnaces and open-hearth furnaces, boiler plants, gas turbine plants, heat recovery systems for ventilation emissions). Mixing heat exchangers are designed for heat and mass transfer processes in direct contact with heat carriers. These include: hollow, filling and bubbling scrubbers, foam machines, irrigation chambers of air conditioning systems, and others. Liquid-liquid, steam – liquid, gas – liquid, steam – steam, steam – gas, steam – gas, etc. devices are divided according to the phase state of heat carriers. According to the design features-shell-and-tube, spiral, plate, coil, capacitive, pipe-in-pipe, hollow, packing and Poppet columns, special, etc. In the relative direction of movement of heat carriers-direct – current, counter-current, with cross and mixed current. According to the mode of operation, there are devices of continuous and periodic action. In devices of the first type, the flow rates and parameters of heat carriers at the inlet and outlet are maintained constant in the established thermal mode. If the parameters and flow rates of heat carriers in such devices change over time, this mode is called transient. In recuperative devices of periodic action, one of the heat carriers is first loaded and when the set parameter values are reached, it is unloaded.

1.1.2 Internal research situation

In 1992, Yang Jianhua, Tang Weixin et al. [5] based on the development of a new test universal centrifugal fan and vane angle measuring instrument, an optimizing experimental study on the A type cooling fan vane of model 175 diesel, and on a B type vane designed in the light of other prototypes as well as an optimum designed C type vane, has been carried out with variation of vane numbers, inner and outer diameter ration, inner and outer fitting angle, air blowing pattern as well as air guiding device. The optimum designed fan and its air guiding device obtained through the experiments haven been shown thoroughly improved performance. In comparing with old ones, 27% increase of air blowing capacity, 57% raise of static pressure, 22% reduction of power consumption and 3dB(A) down of noise have been achieved. The temperature field contrast test shows that temperature of the engine cylinder and the cylinder cover have been lowered 8-10 ℃ and 10-12 ℃ separately, which effectively reduced the engine heat load.

In 1994, Lu Meng Li [6] discussed the reliability research of air-cooled diesel engine, and analyzes and introduces the reliability research of 285F air-cooled diesel engine cooling system from the aspects of reliability design and reliability experiment. It is considered that the cooling system performance of the 285F air-cooled diesel engine can ensure that the diesel engine can work reliably under high temperature conditions. It also shows that the diesel air-cooled system has a high reliability potential and can adapt to the requirements of the 295F diesel engine with a cylinder expansion.

In 1994, Yan Zhaoda, Hu Zhangqi et al. [7] made a comparison upon the ways of fan installed on the flywheel on the single cylinder diesel engine. They got the position of the fan installed on the flywheel has a great influence on the cooling effect, when the cooling fan is integrated with the flywheel. They find the heat transfer effect of air intake from the inside flywheel is better than air intake from the outside flywheel. In addition, the way that air intake from the outside flywheel need enough larger intake surface.

In 2007, Xiao Henglin [8] invented a new cooling system of an inclined cylinder engine. Particularly a cooling device for a lubricating oil storage portion of a small air-cooled diesel engine, The cooling device includes a body, a cylinder head, a crankcase end cap, a lubrication system, an air guiding system. The purpose of this invention is that a closed cooling system is provided at the bottom of the body. The wind shield and the bottom of engine body form a cooling air duct which is a closed ventilation chamber. When passing compressed air to forced to cool the lubricating oil which will increase the cooling speed of oil in the oil pool. At the same time, the cooling effect will be also improved.

In 2010, Wang Benliang [9] and others make an optimized design on the cooling fan on the air-cooled engine. Aim at the heat load of air-cooled diesel engine is very difficult to reduce, by syncretizing the merits of suck air cooling manner and blow air one, the omnipotence fan which can install different shape and size vanes. Experiments had been operated with the original vanes and the optimized vanes. Through contrast, the conclusion that the optimized vanes can enhance the air flux by 34%, advance the pressure by 58% can be drawn. Also, the distribution of cooling air flow field is much sounder than that of original vanes, and the heat load of the air cooled diesel engine can be reduced effectively.

In 2011, Zhang Yongjiu [10] investigated experimentally the design and temperature control of the general engine forced air cooling system. The centrifugal force air through the structure of the heat sink parameter, the cooling air system and its subsidiary bodies adjust the experimental data before and after contrast, concluded a universal approach to design the engine cooling system and method. Study has two parts, the first part is investigate the distance between a flat plate heat sink module of the louvers and heat sink C, as well as the root round R on the cooling effect; the second part of a vertical displacement of 452CC shaft engine-based models to explore the arrangement of the shroud and different impeller, wind system cooling effect. By comparing the experimental data, the distance between louvers and the heat sink will affect the heat sink and thermal distribution of heat flow values, engine shroud arrangement different ways along the circumference of the cylinder will affect the temperature distribution and uniformity; and different impeller, wind system will directly affect the overall. Through this study, data analysis reasonable the distance between wind deflector and heat sink C, larger heat sink roots round R, a reasonable layout of the engine shroud and appropriate different impeller, wind system can effectively reduce engine temperature and the cylinder temperature difference along the circumference. Improving the general reliability of the engine has an important role.

In 2013, Xu Gang, Jiang Shuli et al. [11] took single-cylinder air-cooled diesel engine of X170F as the object. The CFD software called SC/ Tetra software is used to simulate the three-dimensional numerical of flow in its cooling system. According to the distribution of cooling air rate on the cylinder head, they improve and optimized the structure of cylinder head which results in redistribution of the cooling air on the cylinder head. The result shows that the improved scheme increase the cooling air flow that loads into the heat area of cylinder head zone and the cooling flow field is more reasonable to improve the cooling effect of the bottom of the cylinder head.

In 2013, Tian Jie an et al. [12] designed the cooling system for air-cooled engine with CFD, it determines the cooling air flow and the radiating power required by the engine by the traditional way and sets the preliminary design plan for radiating fins, analyzes the design reliability of radiating fins using CFD, determines the relationship between the resistance and flow rate according to the structure of radiating fins, selects the fan on the basic of the relationship, finally, figures out the bench test for the design plan under the conditions that the plan meets the CFD calculation. The results indicate that the combination the theoretical calculation and simulation can be used well in design of air-cooled engine.

In 2014, Tang Gangzhi et al. [13] Aiming at the problem of too high temperature in certain locality of cylinder head, they proposed three modification schemes to intensify its heat dissipation and low its temperature: adding internal cooling ducts in high-temperature region, adding deflectors in cooling duct, or changing the size of fins and the structure of outer ribs. 3D fluid-solid coupling simulation technique is applied to the comparative analyses on the heat dissipation of modification schemes of cylinder head. Based on the comparison results. The third scheme is selected as the optimal scheme and a corresponding prototype is built for carrying out the temperature measurement with far infrared thermography. The results show that with the best modification scheme the overall heat dissipation performance of cylinder head is intensified with the temperatures at the most heated area obviously lowered. The experimental highest temperature of cylinder head falls from 459.1K to 435.9, lowering by 23.2K. The simulated highest temperature of cylinder head falls from 456.1K to 436.0 K, lowering by 20.1K; The simulation results good agree with measured one with the largest error less than 7%.

1.2 Development trends

It is noticed from the above literature that many scholar hope to improve the efficiency of air heater methods. The research work was already carried out widely on cylinder fins with various geometry and material and so on. In the study of heat transfer, fins are generally placed on the surface that extend from an object to increase the rate of heat transfer. Thus, adding a fin to an object increases the surface area and can sometimes be an economical solution to heat transfer problem [14]. So, many experimental methods are available in literature to analyze the effect of fins on the heat transfer rate. It is found that present studies on the heat transfer of air cooling system of engine mainly focus on changing the velocity of the air, the fin geometry and materials, such as the size of fins, spacing between fins, the numbers of fins and so on. In addition, as we know there are three basic modes of heat transfer: heat conduction, thermal convection and heat radiation. The main purpose of enhanced heat transfer research is to increase the rate of heat transfer, in order to achieve improved energy efficiency, according to heat transfer basic heat transfer formula:

$$Q = kA\Delta T. \tag{1.1}$$

It can be found that there are three ways to increase the efficiency of energy transfer. They are to increase the overall heat transfer coefficient, increase the heat transfer area, and increase the average heat transfer temperature difference between hot and cold fluids ΔT .

(1) Expanding the heat transfer area A. Increasing the heat transfer area is one of the most effective ways to enhance heat transfer. However, it is not simply to increase the heat transfer area by increasing the volume of the device, but to increase the heat transfer area by improving the structure of the heat transfer surface. For air-cooled engine cylinder, in order to increase the heat exchange area of an air-cooled

engine cylinder, on the one hand, people add a heat sink to it. On the other hand, more and more scholars have studied the enhanced heat transfer by changing the shape of the heat sink. So far, the shape of heat sinks are rectangular, trapezoidal, triangular, etc.

(2) Increasing the heat transfer temperature difference method ΔT. There are two methods, one is to provide the temperature of the hot fluid or to decrease the temperature of the cold fluid. However, in actual projects, the types and temperatures of the hot and cold fluids are constrained by factors such as the production process and economy, and cannot be changed. There is another way to change the arrangement of hot and cold fluid on the heat transfer surfaces to increase the heat transfer temperature difference. For example, to arrange the hot and cold fluids counter flow instead of parallel flow to increase the temperature difference.

(3) Improving heat transfer coefficient K. Increasing the heat transfer coefficient to increase the amount of heat transfer is an important way to enhance heat transfer, and is also the focus of current research. The heat transfer process of heat exchange equipment is a complex process composed of three basic methods of heat transfer. The heat transfer coefficient K reflecting the heat transfer capacity of the heat exchange equipment is affected by every heat transfer process. In the absence of fouling in the heat exchange equipment, the heat transfer coefficient of the heat exchange equipment can be calculated by the following formula:

$$\frac{1}{K} = \frac{1}{h_1} + \frac{\delta}{\lambda} + \frac{1}{h_2}. \tag{1.2}$$

Where λ is the thermal conductivity of solid， W/(m·K).

So, except increasing the heat transfer area, we can also enhance the heat transfer by increasing the heat transfer coefficient. In this paper, i will investigate the air cooling system of air-cooled engine from view of improving the heat transfer coefficient. And, recently, some researchers find that pulsating flow can enhance heat transfer to some extent by increasing the heat transfer efficiency and do a larger number of investigation.

1.3 The literature review of pulsating flow enhanced heat transfer, external research situation

1.3.1 Internal research situation

In 1929, E G Richardson [15] and other people applied hot wire anemometer to measure the velocity of the tube steady flow and pulsating flow, comparing with the average velocity gradient theoretical values and measured values in the cross-section of the tube found that the pulsating speed "annular effect". That is, the velocity of the fluid at the wall is higher than the tube center and the entire cross-section of the velocity distribution is more flat than steady flow velocity distribution. This also marks the beginning of pulsating heat transfer.

In 1980, Robert C. Herndon et al. [16] used two types of pulsator to pulse the flow in a double-pipe heat exchanger. Condensing stream in the shell was used to heat water, which flowed in the center pipe at rates corresponding to Reynolds numbers from 6600 to 28000. The frequency of the pulsations was varied from 50 to 1000 cycles/min, and the results showed that there was an increase in the heat transfer coefficient on the water side upon pulsing. In addition, they found that there was a maximum in the average enhancement as a function of the pulsation frequency. At lower flow rates showed two maximum value in the average enhancement as a function of the pulsation frequency. However, as the flow rate of the pulsed stream was increased, the two maximum value disappeared and only one maximum value.

In 1990, M. R. Mackley [17] and other people performed an experimental heat transfer measurements for the flow of a lubricating oil on the tube side of a shell and tube heat exchanger. The investigation are made for the case where there is a steady net flow through the smooth walled tube and where periodically spaced baffles have been inserted. The results showed when flow oscillations are superimposed on the net flow. The Nusselt number increase significant for the system when both flow oscillation and baffles are present.

In 1997, Moschandreou et al. [18] analyzed the pulsating flow in the circular pipe full hot development zone of the constant wall temperature heat flux and concluded that the difference between the average Nusselt number (Nu_o) and the corresponding Nusselt number (Nu_{st}) is increasing for the high frequency pulsating flow .

In 2008, Elsayed A.M. Elshafei et al. [19] investigated the heat transfer characteristic of pulsating turbulent air flow in a pipe which was heated at uniform heat flux. The experiments were performed over a ranges of 10^4<Re<$4*10^4$ and 6.6≤f≤68Hz. The experiments indicate that Nu is strongly affected by pulsation frequency and Reynolds number. Its local valve either increases or decreases over that of the steady flow. The higher valves of the local heat transfer coefficient occurred in the entrance of the test tube. It is observed also that the relative mean Nu either increases or decreases, depending on the frequency range. The percentage of maximum enhancement in η_m is about 9% at Re=37100, f=13.3 ; and the percentage of maximum reduction in η_m is about 12% at detected for Re=13350, f=42.5.

In 2012, Zohir [20] investigated the effect of pulsation on the heat transfer rates, for turbulence water stream with upstream of different amplitudes, in a double-pipe heat exchanger for both parallel and counter flows. The experiment runs at pulsation frequency over a range of 140 cpm < f < 260 cpm, Reynolds number: 3855 < Re < 11570, length of stroke: 6 cm < Ls< 18.5 cm. They found the heat transfer of counter heat exchanger can be higher than that of parallel flow. The results reveal the enhancement in relative average Nusselt number up to 10 times and 8 times for higher amplitude and higher pulsation frequencies, respectively, for counter flow and parallel flow.

1.3.2 Internal research situation

In 1989, The effect of pulsating flow on the heat transfer of sugar juice and its mechanism were studied by Deng Xiqun [21], Yang Zhuo and Li Xingren. They obtained pulsating flow is an effective heat transfer method. It was more suitable to strengthen viscous fluid. In turbulent region, the effect of pulsating flow on the heat transfer will be weakened with the increasing of Reynolds number; the intensity of turbulence increasing, the boundary layer thinning and cavitation are the main mechanism of pulsating heat transfer.

In 1996, T. S. Zhao and P. Cheng [22] simulated the laminar pulsating heat transfer in the finite length tube with uniform wall temperature. The four parameters- the pulsating amplitude, the pulsating frequency, the aspect ratio (the ratio of pipe length to diameter) and Pr are the important factors that affect pulsating heat transfer .

In 2003, Zheng Jun et al. [23] Pulsating flow in a pipe was experimentally investigated to determine the effect of pulsation on the rate of heat transfer. The influence of water flow rate, pulsation properties and pressure drop was carefully studied. In order to adjust the pulsating parameters, a self-oscillator was designed in which the length of the resonator and the length of outlet nozzle could be adjusted. The results show that the rate of heat transfer is strongly affected by both the hydrodynamic parameters and the configuration of the resonator. With the increase of the water flow rate and the length of the chamber, the heat transfer is enhanced. When water flow rate $G<0.3$kg/s, the dimensionless heat transfer factor $E<1$. When $G>0.3$kg/s, $E>1$, which enhanced the heat transfer. And the maximum valve of the increment of the dimensionless heat transfer factor $\Delta E=0.21$. When no-dimension length of the resonator L ranges from 0.25 to 0.30, there is fluctuation of the heat transfer. When $L>0.3$, the E is increasing significantly. And the L=0,35, the E reach to maximum valve. In addition, there exist an optimal length of the outlet nozzle at which the heat transfer can be enhanced most effectively.

In 2005, Yu Yielong and Li Zhixin [24] investigated numerical analysis on convective heat transfer of laminar tube with constant heat flux and constant wall temperature. The simulation results of constant heat flux boundary condition were in agreement with the analytical solution very well. The numerical results showed that the pulsating flow in a circular tube under constant heat flux or under constant wall temperature conditions can result in the fluctuation of velocity ,temperature of fluid and Nusselt number around the steady ones, the large pulsating amplitude and the smaller the pulsating frequency, the large fluctuation will be. But the overall effect does not strengthen the pulsating flow, the pulsating flow neither strengthen the heat transfer nor weaken the heat transfer. Their time-averaged are equal to the steady ones at the same Reynolds number.

In 2009, Gao Hong and Liu Juan-fang [25] performed an experimental study of heat transfer enhancement by pulsating fluid and designed a self- oscillator where the continuous flow of the fluid was transferred into pulsation flow. The pulsating flow water passes through a single pipe heat electrically. They changed the hydraulic parameters and the length of the resonator and outlet nozzle. The water velocity ranges from 5.0 m/s to 7.5 m/s; the length of the resonator ranges from 10 mm to 14 mm; the length of outlet nozzle ranges from 12 mm to 25 mm. The experiments revealed that the scale of the enhancement ratio is 1.01~1.07 and the stable enhancement ratio ranges from 1.1 to 1.4. In other words, under the appropriate hydraulic parameters and configuration of the resonator, the heat transfer coefficient can increase 10%-40% comparing with the normal case when the oscillator was removed.

In 2011, Xu Jie [26] investigated the pulsating flow on pipe flow heat transfer by experimental testing and numerical simulation and analyzed the effect of Reynolds number, the pulsation frequency and amplitude to heat transfer. Reynolds number ranges from 500 to 3200, the pulsation frequency ranges from 0 to 12 Hz, as well the distance of the pulser to inlet of the testing tube changes from 0.5 to 1.5m. The results show that: in the laminar flow, the pulsation flow enhances the heat transfer of the pipe flow and the enhancement trend of heat transfer is increasing with the increasing of the frequency. However, with the increasing of the Reynolds number, the enhancement trend of the heat transfer weakens. When the flow state is in turbulent circumstances, the pulsation flow weakens the heat transfer of the pipe flow. With the increasing of the Reynolds number, the effect is increasing gradually.

In 2013, Li Si-wen, Li Hua, Yang Zang-jian et al. [27] investigated experimentally pulsating flow in a pipe under turbulent conditions to determine the effect of pulsation on the rate of heat transfer. The experiments were performed over a ranges of $16 \leq \upsilon \leq 40$ m/s ($4.6 \cdot 10^4 < Re < 11.6 \cdot 10^4$) and $0 \leq f \leq 100$Hz. The results show that the pulsation frequency, pulsation amplitude and Reynolds number have a significant impact on the heat transfer enhancement. The heat transfer is increasing as the increasing of Reynolds number and pulsation amplitude. However the strengthening effect is not infinitely increased. When the flow velocity reaches a certain degree, the strengthening effect will weaken. At the same time, as the pulsation amplitude is beyond the certain value which can enhance the heat transfer. While the best pulsation frequency exists respectively in the range of 0 to 20 Hz and 80 to 100 Hz. The maximum percentage of enhancement heat transfer is 36.1% at f=10 Hz.

In 2016, Hongsheng Yuan et al. [28] performed a theoretical studies on heat transfer of pulsating laminar flow in pipes with wall thermal inertia. They found that pulsating laminar flow can weaken heat transfer and the effects are small under the high frequency, large Prandtl number, small pulsation amplitude and small heat capacity. In addition, comparing the result of pulsating flow in circular pipe with in two parallel plates, the circular pipes have lower temperature but higher temperature gradient. The maximum temperature gradient of fluid in pipe is away from pipe wall while the maximum gradient of parallel plates is near the plate wall. The pulsating effects are a little more remarkable for circular tube which are caused by the distinct boundary conditions.

From the literature review. It is found that studies on pulsating flow to some certain extent could increase heat transfer. Although, some literature have a disagreement with this point. On the one hand, some of them are caused by the relatively narrow scope of the parameters. On the other hand, some of them are not. So far, the mechanism of heat transfer enhancement by pulsed flow is still ambiguous. However. This supplies a new idea to investigate the influence of heat transfer by imposing pulsation on the steady flow. So, it is necessary to study the influence of pulsating flow on the cooling system. In this paper, we will conduct this research to investigate pulsating heat transfer for the air cooling system of petrol engine. Meanwhile, this work will be taken up on these aspects to cover the research gaps.

1.4 Research content

This paper is mainly to study the performance of pulsating air heater system. From the previous literature, we get that the fluctuating flow can affect heat transfer to some extent and increase the efficiency of heat transfer. Relatively speaking, in the air heating system of a room with air heating, the air passes through the heater where it increases the temperature. However, based on the research results, in this paper I will introduce vibrations in a stationary flow to change the steady flow of air in the oscillating flow and add a vortex that is expected to reduce the thickness of the sublayer and increase heat transfer. In this way, the drive and blinds will create air vibrations before it starts to leave the heating pipe, which will increase heat transfer and increase the heat transfer coefficient using this method.

The purpose of this study is as follows.

(1) experimentally investigate the effect of heat transfer of the heater by creating a fluctuating air flow.

(2) experimentally determine the effect of the frequency of vibrations on the heat transfer of the air heating system in the ventilation system.

This experiment will be performed by both stationary flow and oscillating flow using two methods. Comparing a stationary flow with a fluctuating flow, we will conduct an experimental analysis of the influence of the frequency of vibrations on heat transfer.

Conclusion

In this present dissertation, I will try to work from the point of view of improving the heat transfer coefficient. The effect of fluctuating flow on the air heating system will be experimentally investigated. And work will be done on these aspects to cover research gaps and present results based on systematic research.

Thus, the purpose of this study is to determine the degree of increasing the efficiency of heat exchange in the mode of pulsating air flow.

To achieve this goal, the following tasks are set:

- perform an information search for existing ways to increase the efficiency of heat exchange on the surface of heaters;
- develop a prototype of an experimental installation including a heater and a device for organizing the fluctuating flow of air, select the required measuring equipment, assemble and adjust its operating modes;
- develop a mathematical model of the energy chain …;
- select measurement methods and tools;
- perform experimental studies of the operation of an experimental heater with a fluctuating air flow.

2 The heat transfer enhancement technology

As a rule, the fan heater operates in several modes. Each is characterized by water consumption, cooling rate, and air consumption. It is possible to connect to a European-class Central heating line (60 ° C) and a special 150 ° C steam boiler.

A heater is a device used for heating air. According to the principle of operation, it is a heat exchanger that transfers energy from the heat carrier to the flow of the supply jet. It consists of a frame, inside of which there are dense rows of tubes connected in one or more lines. A heat carrier — hot water or steam-circulates through them. The air passing through the frame section receives heat energy from the hot tubes, so that it is transported through the ventilation system already heated, which does not create the possibility of condensation or cooling of the premises.

In this paper, tests of heaters are carried out for the purpose of definition valid heating capacities and their resistance and comparisons of these indicators with design or catalog data. Exact heat technical and aero-dynamic tests of heaters for the purpose of definition of surface-area factor and resistance are made in laboratories.

2.1 The classification of pulsation source

The pulsation source is divided into self-oscillation and forced pulsation. Self-oscillation is a device that generates pulsation by changing the structural shape of the pipeline. This pulsation requires no external force. At present, there are self-oscillating sources, including a non-return valve system and a self-oscillation system. Forced pulsation sources are devices that pulsates fluids by consuming energy. Need to use extra force, such as mechanical force, electromagnetic force. The more common forced pulsation source are diaphragm pumps, solenoid valves, reciprocating pumps, combined rotary vane valves, etc, (table 1).

Table 1 - Some generating pulsation method used by scholars

Date	Researcher	Pulsation source	Frequency
1943	Martinelli	Reciprocating pump	13-265 cycles/min
1952	West	Reciprocating pump	100 cycles/min
1961	Lemlich	Solenoid valve	0-210 cycles/min
1971	Keil	Positive displacement pump	0.4-1.1Hz
1979	Karamercan	Piston and cylinder	0-300 cycles/min
1980	Robert C. Herndon	Ball valve and shower head	0-1000 cycles/min

2.2 Pulsating flow enhancement heat transfer mechanism

So far, the mechanism of pulsating flow enhancement is not clear, however, there are some factors that have been recognized by research field.

2.2.1 The thinner boundary-layer

Convection heat transfer mostly refers to the heat transfer between the fluid and the solid wall, and its heat transfer rate is closely related to the fluid properties and the condition of the boundary layer. The temperature difference is very large in this thin layer next to the wall. Meanwhile, the thermal resistance of convection heat transfer is mainly concentrated in a thin layer. In this thin layer, the fluid maintains the characteristics of laminar flow and is therefore called the viscous laminar flow bottom layer. It is generally conceded that the heat transfer through the laminar film occurs by means of conduction. The fluid in the viscous laminar bottom layer has a large temperature gradient. Once the heat passes through the layer, it is easily carried away by the core fluid. Therefore, the heat transfer strength mainly depends on the thickness and characteristics of the viscous laminar bottom layer. As we found in the case of turbulent flow, the heat transfer intensity is much greater than the heat transfer intensity in the case of laminar flow. Because the turbulence can thin the boundary layer to a certain extant. So the greater the turbulence of the fluid, the better the heat transfer effect, the thinner the viscous laminar flow bottom layer, the smaller the convection heat resistance.

2.2.2 Increasing turbulence

With the development of the boundary layer and the increase in the thickness of the boundary layer, the relative viscous force of the inertial force in the boundary layer will gradually increase, resulting in loss of stability in the flow in the boundary layer, from laminar flow to turbulent flow. It has been found that pulsating flow is associated with a periodic pressure gradient reversal, which causes an increase in radial and longitudinal mixing during part of the pulse cycle which in turn decreases the film thickness in the pulsed stream. The disturbance and mixing of fluids reflect the degree of turbulence. The increase of turbulence intensity will increase the convective heat transfer efficiency, and the turbulence velocity has a close relationship with the velocity shear layer. The radial mixing has an important influence on the convection heat transfer to a large extent. It is the presence of flow in the radial direction of the fluid motion that causes the heat exchange between the side wall and the fluid core and takes away the heat. The flow of the pulse causes the pressure of the fluid at various locations in the tube flow to change periodically as the flow rate changes. When the flow is strengthened, the fluid will intensity turbulence in the direction of movement and in the diameter direction. At the same time, the Reynolds number instantaneously increases in the direction of movement, so the turbulence intensity also increases. When the flow weakens, the rate of change of pressure and velocity in the direction of motion will change significantly, which will also increase the turbulence intensity of the flow. When this theory is applied to the laminar flow of tubes, it will have a relatively large effect on the heat transfer. When the laminar flow of tubes is posed to pulsating, the heat transfer enhancement effect is most obvious. However, this state does not continue. When the Reynolds number increases to a certain extent, the pulsating flow will reduce the strengthening of the pipe flow. This is due to the fact that the increased turbulence of the flow and the originally high turbulence partially cancel each other out. Therefore, the pulse-enhanced heat transfer is most effective for laminar flow, and transitional flow and turbulent flow are worse. For the Reynolds number-large flow, the effect of pulsation is not obvious.

2.2.3 Generating cavitation

Some scholars found the pulsations can cause the fluid next to the tube wall to cavitate in the water cooling system. And they claim cavitation is very important factor to enhance heat transfer for pulsating flow. Cavitation is the formation of many small bubbles on the tube surface of the heat exchanger during the low-pressure portion of the pulsation cycle, the flow is suddenly interrupted or forced to change direction which result to the pressure in fluid pressure drops below the vapor pressure of the fluid in the boundary layer next to the tube wall [4]. So the bubbles will originate where the pressure drops below the vapor pressure and occur cavitation. The bubbles act as carriers of latent heat by moving toward the bulk and giving up their latent heat as sensible heat upon collapsing. The periodic formation and collapse of these bubbles agitates the boundary layer and thus increases the rate heat transfer.

Lemlich [29] has also proposed that cavitation increases heat transfer by transferring the heat of vaporization necessary to produce the vapor bubbles from the tube wall. The bubbles then detach from the tube wall and travel into the bulk of the fluid stream where they collapse, giving up their latent heat.

Therefore, it is believed that cavitation is able to increase heat transfer in the laminar sub-layer next to the tube wall and enter the main fluid stream they distrupt the laminar sub-layer, causing mixing to occur in the layer and the renewable of laminar sub-layer. This may be an additional mode of energy transfer occurring as a result of cavitation in the pulsating flow [27].

2.2.4 Introducing forced convection

In steady flow, the heat transfer is conduction rather than convection in the boundary layer. However, when imposing the pulsation to steady flow, the pressure periodic variation produce forced circulation in the fluid and increase the effective heat transfer by promoting the formation of eddies, thus introducing convective in the boundary layer [4].

2.2.5 Field synergy principle

In 1998, professor Guo [30] from Tsinghua university proposed a novel concept that heat exchange between fluid and wall has a close relationship with the included the angle between the velocity filed and temperature gradient vectors. When the velocity field and temperature gradient filed reach full field coordination, the heat transfer gets optimal result. The Nusselt number exists the upper and lower limit, respectively, Nu=Re·Pr and Nu=0, depending on the coordination degree of the velocity filed and temperature gradient filed. Generally, the heat transfer efficiency is between the upper and lower limit. This theory is called field synergy principle. This theory compares the process of convection heat transfer to the addition of an internal heat source in the heat transfer and compares the fluid flow with an equivalent heat source. The theory shows that the strength of the convection heat transfer is directly proportional to the strength of the equivalent heat source. It is not only related to the temperature difference between fluid and the pipe wall surface, the speed of the fluid flow, and the fluid's own medium characteristics. It is also related to the angle between the fluid flow velocity and the heat flow vector. Because the value of the equivalent heat source is indefinite, the flow can not only enhance the heat transfer effect, but also weaken the heat transfer effect.

Generally, the energy equation:

$$\rho Cp\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+\dot{q}. \tag{2.1}$$

The energy conservation equation of 2-D boundary-layer steady flow:

$$\rho Cp\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right). \tag{2.1a}$$

The energy conservation equation of 1-D steady-state heat conduction with heat sources:

$$-\dot{q}=\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right). \tag{2.1b}$$

It is easy to find that the convective term in the energy equation for the boundary layer flow corresponds to the heat source term in heat conduction equation.

The integral of the equation(2.1a) over the thermal boundary layer thickness leads to

$$\int_0^{\delta_t} \rho Cp\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)dy=-k\frac{\partial T}{\partial y}. \tag{2.2}$$

The equation (2.2) may be rewritten with the convection term in the vector form:

$$\rho Cp\int_0^{\delta_t}\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)dy=-k\frac{\partial T}{\partial y}. \tag{2.3}$$

Where the thermophysical properties ρ, Cp are assured constant.

Introducing the non-dimension array

$$\overline{\vec{U}}=\frac{\vec{U}}{U_\infty}\ ,\quad \nabla\overline{T}=\frac{\nabla T}{(T_\infty-T_w)/\delta_t},\quad \overline{y}=\frac{y}{\delta_t},\quad T_\infty>T_w, \tag{2.4}$$

$$\frac{U_\infty\cdot x}{\nu}\cdot\frac{\nu}{\frac{\lambda}{\rho Cp}}\int_0^1\overline{\vec{U}}\cdot\nabla\overline{T}d\overline{y}=\frac{-\frac{\lambda}{T_\infty-Tw}\frac{\partial T}{\partial y}}{\lambda}\cdot x. \tag{2.5}$$

The non-dimensionalization of equation (2.3) leads to

$$\mathrm{Re}_x\,\mathrm{Pr}\int_0^1\left(\overline{\vec{U}}\cdot\nabla\overline{T}\right)d\overline{y}=Nu_x. \tag{2.6}$$

$$\mathrm{Re}_x\,\mathrm{Pr}\int_0^1\left(\overline{\vec{U}}\cdot\nabla\overline{T}\right)d\overline{y}=Nu_x\overline{\vec{U}}\cdot\nabla\overline{T}=\left|\overline{\vec{U}}\right|\cdot\left|\nabla\overline{T}\right|\cdot\cos\beta. \tag{2.7}$$

$$\mathrm{Re}_x \Pr \int_0^1 \left(\left| \vec{\overline{U}} \right| \cdot \left| \nabla \overline{T} \right| \cdot \cos \beta \right) d\overline{y} = Nu_x, \tag{2.8}$$

where β - is the included angle between the dimensionless velocity and temperature gradient vector.

From the view of equations above, we can see the strength of convection heat transfer is not only related with Re and Pr, but also related with the included angle between the dimensionless velocity and temperature gradient vector. When β<90˚, the smaller the β, the more the strength of convection heat transfer. When β=0, there are maximum strength of convection heat transfer.

So, there are three ways to raise the strength of heat sources/convective terms, and consequently to enhance the heat transfer [42]:

(1) Increasing Reynolds number and / or Prandtl number;

(2) Increasing the fullness of dimensionless velocity and / or temperature profiles;

(3) Reducing the included angle between the dimensionless velocity and temperature gradient vector (when β < 90˚).

Conclusion

There are many ways to enhance the heat transfer, we can choose suitable method according to the specific experiment condition. So far, the mechanism of the pulsating flow enhance the heat transfer is not completely clear. However, the above five statements have been universally recognized. From the existing research mechanism, we obtain the some new ideas to enhance heat transfer and get a deeper understanding of heat transfer enhancement.

3 Analysis and calculation of heat dissipation process of air-heat engine

Using these devices, the room temperature is controlled. In mathematical modeling of air conditioning systems, non-linear models should not be neglected properties of heating installations, as this will inevitably lead to inadequacy the resulting model and the inaccuracies of the synthesized automatic control system. Given the large number of types of heaters, a variety of implementation schemes air conditioning systems and methods for managing heat transfer were highlighted in the work main types of disturbing and controlling effects for air conditioning systems air's. Based on the block diagram of the generalized supply air conditioning system systems of the equation of heat and mass balance of the heat carrier and the air that most accurately reflects all the processes taking place. The properties of heating installations that change dynamically over time were shown in a wide range of coolant flow control in heat exchanger tubes. Recommendations on the choice of the method of hydraulic strapping of the heater are also formed. Air conditioning systems in terms of improving the regulator setting and increasing the accuracy of the automatic control system

3.1 Analysis of heat transfer process of heater wall

For an air-heat engine, the process of heat transfer of a high-temperature working fluid through the wall of the heat exchanger to the cooling air is a more complex and heat-transfer process, as shown in figure 3-1.to enhance heat transfer, air-heat engines are equipped with radiators-heat exchangers. The entire heat transfer process includes the following three aspects.

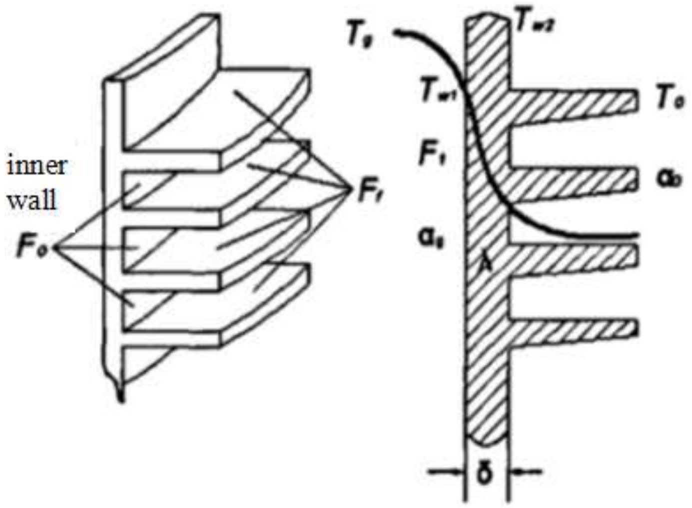


Figure 3.1 - The heat transfer process of the air heat engine

3.1.1 Heat transfer from the high-temperature working substance to the cylinder wall

When the engine is in operation, the high-temperature working fluid in the cylinder continuously flushes the peripheral wall of the cylinder with different temperatures, and strong convection heat transfer is performed between the two.

Convection heat transfer.

The heat flow calculation of the convection heat transfer, currently using Newton cooling formula, its mathematical expression is:

$$Q_c = h_1^{'}(T_{f1} - T_{w1})F_1, \quad (3.1)$$

where $h_1^{'}$ - is the average convection heat transfer coefficient;

T_{f1} is the instantaneous average temperature of the working medium in the cylinder, K, which is a function of the crank angle, can be determined by using the gas state equation according to the engine indicating diagram;

T_{w1} is the average temperature of the wall, K;

F_1 is the heat exchange surface area in contact with the working medium, m^2, which is also a function of the crank angle.

Radiation heat transfer.

The radiation heat exchange between the high temperature working medium in the heater and the wall has two phenomena: gas radiation and flame radiation. The radiant heat flux is the sum of the two.

$$Q_r = Q_s + Q_f, \tag{3.2}$$

where Q_s - is the heat of gas radiation, W.

Q_f is the heat of flame radiation, W.

In summary, the amount of heat transfer Q_w from the high-temperature working medium to the peripheral wall of the cylinder is the sum of the convection heat transfer amount Q_c, the gas radiation heat exchange amount Q_s and the flame radiation heat exchange amount Q_f.

$$\begin{aligned} Q_w &= Q_c + Q_s + Q_f \\ &= h_1(T_{f1} - T_{w1})F_1. \end{aligned} \tag{3.3}$$

where T_{w1} - is the average temperature of the cylinder wall, K. Experiments have shown that the magnitude of its change over time is very small and can be approximated as a fixed value;

h_1 - is the average heat transfer coefficient, W/(m^2·K).

Since there are many factors affecting the heat transfer process of the working fluid to the peripheral wall of the cylinder, when the heat transfer calculation is actually carried out, the complexity of the heat exchange process and the difficulty in calculation are comprehensively concentrated on the average heat transfer coefficient. The engine heat transfer data are comprehensively considered. Since the radiation heat transfer quantity is much smaller than the convection heat transfer quantity, in order to facilitate, the radiation heat transfer is often calculated by increasing the convection heat transfer coefficient.

3.1.2 Heat conduction from the cylinder inner wall to the outer wall

Assuming that the cylinder height and the circumferential direction have the same temperature, the heat flow only changes in the radial direction. In fact, regardless of the temperature of height or circumferential direction are different, the heat flow consists of radial, axial and tangential heat flow. However, under normal circumstances, the sum of the latter two is less than 1/5-1/10 of the former and can be ignored. Secondly. The ratio of the inner diameter and the outer diameter of the cylinder is generally not more than 1/2, and it can be treated as a flat wall conduction heat. The error does not exceed 1%. Therefore, the flow of heat transfer through the peripheral wall of the cylinder can be obtained according to the law of heat transfer proposed by Fourier in 1882, that is, the heat flow is proportional to the wall area F1 perpendicular to the direction of heat flow and the temperature difference of the outer wall surface and inner wall surface (T_{w1}-T_{w2}). It is inversely proportional to the wall thickness δ(m). which is

$$Q_w = \lambda \frac{T_{w1} - T_{w2}}{\delta} F_1, \tag{3.4}$$

where the scale factor λ [W/(m·K)] - is called thermal conductivity. For each substance, λ has a specific value obtained by checking documents, which is generally determined experimentally.

3.1.3 Heat transfer from the outer wall of the cylinder to the cooling air

The heat transfer between the cooling fins of the outer wall of the cylinder and the cooling air is a convective heat transfer process in which the fluid directly contacts with the solid wall with different temperatures. Since the temperature of the heat sink is not high, the radiation to the surrounding environment is negligible. Therefore, the amount of heat transfer can be calculated according to Newton's cooling formula, which is

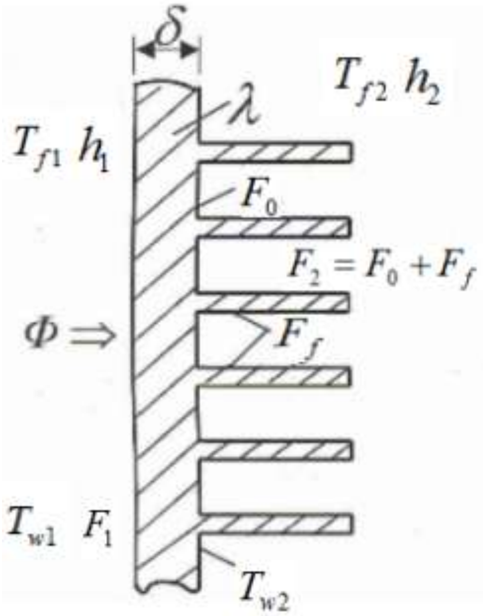


Figure 3.2 - The heat transfer of heat sinks with cooling air

$$Q_w = h_2(T_{w2} - T_{f2})F_2, \qquad (3.5)$$

where h_2 - is the average heat transfer coefficient of heat sink and cooling air;

T_{w2} - is the temperature of the heat sink wall;

T_{f2} - is the average temperature of the cooling air;

F_2 - is the surface area of the heat sink wall, [m^2], is the sum of the surface area of the root between the two heat sinks F_0 and the surface area of the heat sink F_f .

If the engine is running under a stable load and speed, the entire heat transfer process can be considered as a stable heat transfer process, and the energy conservation law should be

$$Q_w = h_1(T_{f1} - T_{w1})F_1 = \frac{\lambda}{\delta}(T_{w1} - T_{w2})F_1 = h_2(T_{w2} - T_{f2})F_2. \qquad (3.6)$$

Assuming that the temperature of the entire fin surface is equal to the base surface temperature, ie, the fin efficiency η_f is equal to 1.

$$T_{f1} - T_{w1} = \frac{Q_w}{h_1 \cdot F_1}. \qquad (3.7)$$

$$T_{w1} - T_{w2} = \frac{Q_w}{\frac{\lambda}{\delta}F_1}. \qquad (3.8)$$

$$T_{w2} - T_{f2} = \frac{Q_w}{F_2 \cdot \alpha_0}. \qquad (3.9)$$

$$T_{f1} - T_{f2} = Q_w\left(\frac{1}{h_1 \cdot F_1} + \frac{1}{\frac{\lambda}{\delta} F_1} + \frac{1}{F_2 \cdot h_2}\right). \tag{3.10}$$

$$Q_w = \frac{T_{f1} - T_{f2}}{\frac{1}{h_1 \cdot F_1} + \frac{1}{\frac{\lambda}{\delta} F_1} + \frac{1}{F_2 \cdot h_2}}. \tag{3.11}$$

Rib effect coefficient β is the ratio of ribbed surface to smooth surface area, ie.

$$\beta = F_2 / F_1. \tag{3.12}$$

$$Q_w = F_1 \frac{T_{f1} - T_{f2}}{\frac{1}{h_1} + \frac{1}{\frac{\lambda}{\delta}} + \frac{1}{\beta \cdot h_0}}. \tag{3.13}$$

The larger rib effect coefficient β, the small the thermal resistance, the larger heat transfer coefficient.

$$F_2 = F_0 + F_f, \tag{3.14}$$

where F_0 - is the root area between fins and fins;

F_f - is the area of the protruding fins.

When the temperature of the entire fin surface is not equal to the base surface temperature, ie, the fin efficiency η_f is not equal to 1.

$$Q_w = h_1(T_{f1} - T_{w1})F_1, \tag{3.15}$$

$$Q_w = \frac{\lambda}{\delta}(T_{w1} - T_{w2})F_1, \tag{3.16}$$

$$Q_w = h_2(T_{w2} - T_{f2})F_0 + h_2(T_{w2} - T_{f2})F_f\eta_f = h_2(T_{w2} - T_{f2})F_2\eta_0, \tag{3.17}$$

$$\eta_0 = \frac{F_0 + F_f\eta_f}{F_2}. \tag{3.18}$$

So

$$Q_w = F_1 \frac{T_{f1} \text{-} T_{f2}}{\frac{1}{h_1} + \frac{1}{\frac{\lambda}{\delta}} + \frac{1}{\beta \cdot h_2 \cdot \eta_0}}, \tag{3.19}$$

where η_f - is fin efficiency;

η_0 is overall fin surface efficiency;

β is the rib effect coefficient;

δ/λ is the conduction resistance;

$1/h_1$ is the heat exchange resistance between high temperature fluid and cylinder;

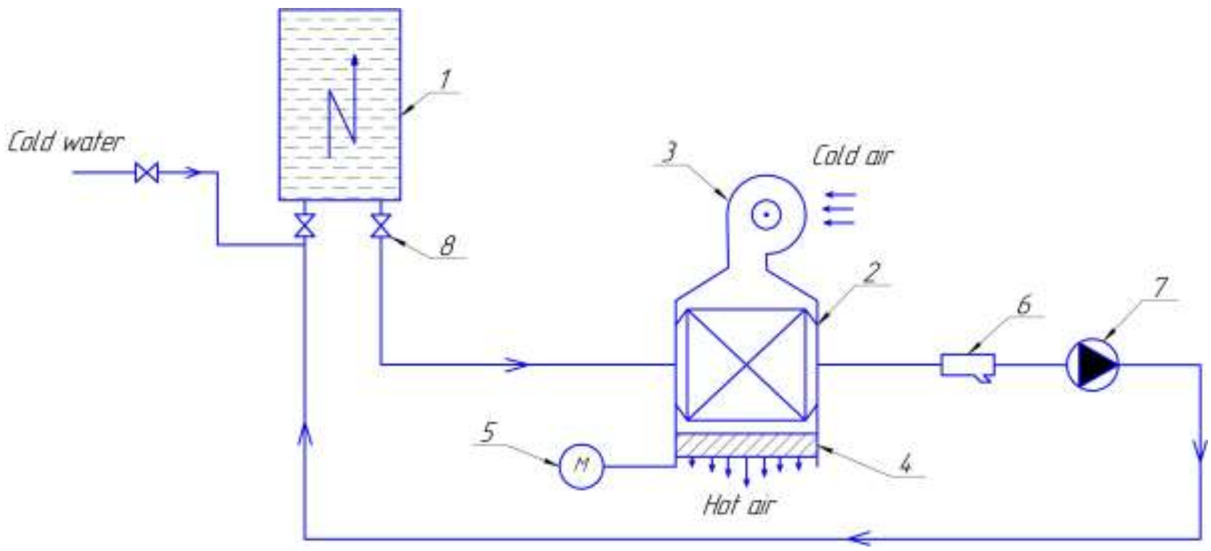
$1/h_2\beta\eta_0$ is the heat exchange resistance between heat sinks and cooling air.

Due to $\beta > 1$, the thermal resistance of the finned surface $1/(h_2\beta\eta_t)$ is smaller than the thermal resistance without finned surface. This is the reason that putting fins on the side where the heat transfer coefficient is small can enhance the heat transfer when there is a large gap between h_1 and h_2. To increase the coefficient of rib effect coefficient, in principle, two approaches can be followed: one is to use thin ribs, narrows the spacing of the fins to increase the number of fins, and the second is to use long fins to enlarge the surface area of each fin. However, the thinner, denser, and higher fins of the design, will bring more difficulties to the processing of the heat sinks, and too dense heat sinks will hinder the flow of air between the heat sinks and even make the flow dead zone, which will weaken the heat transfer effect.

4 The energy chain of the experiment equipment

4.1 The equipment of the experiment

In this paper, we consider the influence of the pulse mode of the coolant flow on the heat transfer "heater-environment". To study this issue, the main thermal scheme of the experimental installation is shown in Figure 4.1.



Convention: 1– boiler, 2– heat exchanger, 3– fan, 4– blinds, 5 – closer, 6 – sump, 7 – pump, 8 – crane.

Figure 4.1– Schematic diagram of the experimental setup

The unit is designed to heat the air using a heat exchanger in which the heat carrier circulates, hot water.

The installation works as follows: cold water enters the boller 1, where it is heated to a certain temperature and enters the circuit, a heat exchanger 2 is installed in the circuit, which is blown by a hair dryer 3. The pulse mode of hot water supply to the heat exchanger and the cleanliness of the opening and closing of the blinds 4 will help us to increase efficiency. With the closer 5, we adjust the opening of the blinds, a sump 6 and a circulation pump 7 are installed in the circuit for rust, and we use a crane 8 to stop the operation of the circuit.

Experimental research includes: design and construction of the air tube. In which thermodynamic processes will take place. the main task will be to find the optimal frequency of opening and closing blinds and pulse supply of coolant. There will be similar optimal parameters: the speed of the coolant, pressure, temperature, and length of the pipeline. In the course of the study, for a better understanding of the scheme, it was decided to study 2 characteristics of hydraulic and thermal, in order to better understand the nature of the forces arising and to more accurately determine the required parameters on the obtained model.

In the first power circuit the hydraulic characteristic at the moment of closing of the shock valve is considered. The main interest is the energy converter throughwhich the pressure is converted into force, and the flow rate in the flow velocity.

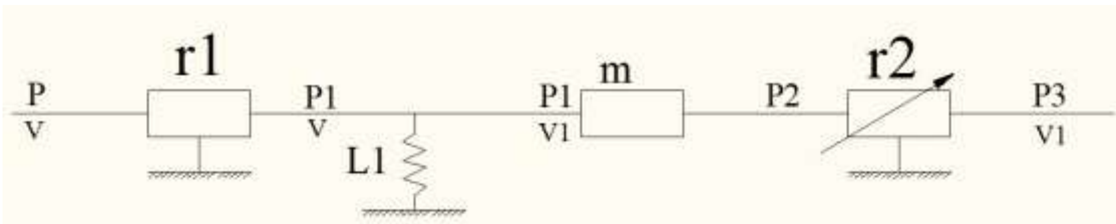


Figure 4.2– Energy circuit 1

Considered the second energy diagram, the backward wave surge with the opening of the valve.

The resulting circuit of the first stage is divided into links. Next, at each level we find the values that affect the basic parameters and change them depending on various factors.

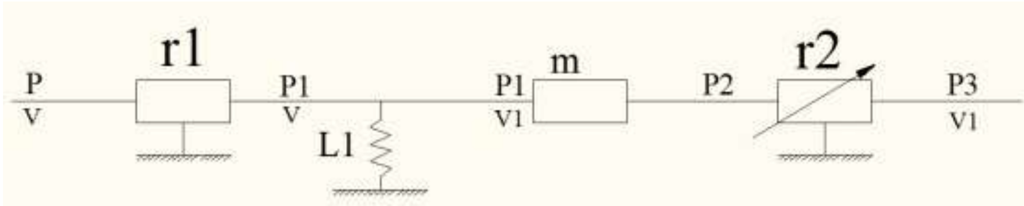


Figure 4.3– Energy circuit with variable parameters

We get links in the energy circuit 1. For each we make a system of equations. For the first link:

1. Step

$$\begin{cases} p = r_1 v^2 + m\dot{v}_1 + r_2 v_1^{\,2} + p_3 \\ v = l\,\dot{p}_1 + v_1 \end{cases} .$$

2. Step

$$\begin{cases} p_3 = \bar{p}_3 + p_{30} \\ v_1 = \bar{v}_1 + v_{10} \end{cases} ;$$

$$\begin{cases} p_1 = m\dot{v}_1 + r_2 v_{10}^{\,2} + 2r_2 v_{10}\,\bar{v}_1 + \bar{p}_3 + p_{30} \\ \dot{p}_1 = m\ddot{\bar{v}}_1 + 2r_2 v_{10}\,\dot{\bar{v}}_1 + \dot{\bar{p}}_3 \end{cases} ;$$

$$\begin{cases} p = 2lr_1 v_{10} m\ddot{\bar{v}}_1 + (m + 4lr_1 r_2 v_{10}^{\,2})\dot{\bar{v}}_1 + (2r_1 v_{10} + 2r_2 v_{10})\bar{v}_1 + r_1 v_{10}^{\,2} + 2r_1 l v_{10}\,\dot{\bar{p}}_3 + \bar{p}_3 + p_{30} \\ v = lm\ddot{\bar{v}}_1 + 2lr_2 v_{10}\,\dot{\bar{v}}_1 + l\dot{\bar{p}}_3 + v_{10} + \bar{v}_1 \end{cases} .$$

3. Step

$$p = 2lr_1v_{10}m\ddot{\bar{v}}_1 + (m + 4lr_1r_2v_{10}{}^2)\dot{\bar{v}}_1 + (2r_1v_{10} + 2r_2v_{10})\bar{v}_1 + r_1v_{10}{}^2 + 2r_1lv_{10}\,\dot{\bar{p}}_3 + \bar{p}_3 + p_{30}$$

$$= a_1\,\ddot{\bar{v}}_1 + a_2\,\dot{\bar{v}}_1 + a_3\,\bar{v}_1 + a_4 + b_1\,\dot{\bar{p}}_3 + b_2\,\bar{p}_3 + b_3$$

$$(a_1p^2 + a_2p + a_3 + 1)v_1(p) = -(b_1p + b_2 + 1)\,p_3(p)\,.$$

4. Step

$$Z\ (p) = \frac{p_3(p)}{v_1(p)} = -\frac{a_1p^2 + a_2p + a_3 + 1}{b_1p + b_2 + 1}\,.$$

5. Step

$$Z\ (j\Omega) = -\frac{-a_1\Omega^2 + a_2\,j\Omega + a_3 + 1}{b_1\,j\Omega + b_2 + 1}\,;$$

$$Z\ (j\Omega) = \frac{-a_1b_2\Omega^2 - a_1\Omega^2 + a_2b_1\Omega^2 + a_3b_2 + a_3 - b_2 - 1 + (a_1b_1\Omega^3 + a_2b_2\Omega\ + a_2\Omega\ - a_3b_1\Omega\ + b_1\Omega)\,j}{-b_1{}^2\Omega^2 - (b_2+1)^2}$$

$$R_e(j\Omega) = \frac{-a_1b_2\Omega^2 - a_1\Omega^2 + a_2b_1\Omega^2 + a_3b_2 + a_3 - b_2 - 1}{-b_1{}^2\Omega^2 - (b_2+1)^2}\,;$$

$$I_m(j\Omega) = \frac{a_1b_1\Omega^3 + a_2b_2\Omega\ + a_2\Omega\ - a_3b_1\Omega\ + b_1\Omega}{-b_1{}^2\Omega^2 - (b_2+1)^2}\,j\quad .$$

We obtain the amplitude-frequency function of the energy circuit:

$$A(j\Omega) = \sqrt{R_e(j\Omega)^2 + I_m(j\Omega)^2}\ .$$

Get the phase-frequency function of the energy circuit:

$$\varphi(j\Omega) = -\arg tg\,\frac{I_m(j\Omega)}{R_e(j\Omega)}\ .$$

The resulting circuit of the first stage is divided into links. Next, at each level we find the values that affect the basic parameters and change them depending on various factors.

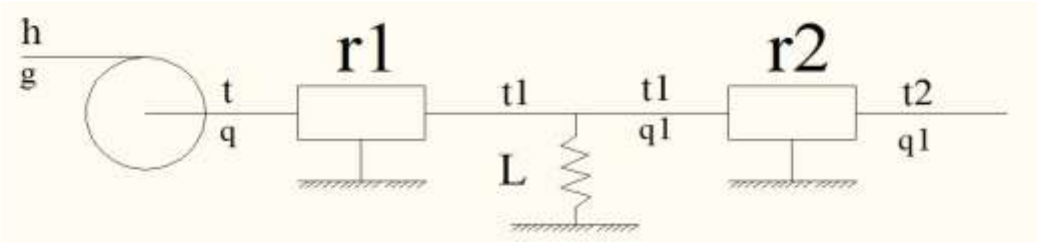


Figure 4.4 – Heat transfer circuit

1. Step

$$\begin{cases} t = r_1 q + r_2 q_1 + t_2 \\ q = l\,\dot{t}_1 + q_1 \end{cases} .$$

2. Step

$$\begin{cases} t_2 = t_{20} + \bar{t}_2 \\ q_1 = q_{10} + \bar{q}_1 \end{cases} ;$$

$$t_1 = r_2 q_1 + t_2$$

$$\dot{t}_1 = r_2 \dot{\bar{q}}_1 + \dot{\bar{t}}_2 \quad .$$

3. Step

$$q = l\dot{\bar{t}}_2 + lr_2 \dot{\bar{q}}_1 + \bar{q}_1 + q_{10}$$

$$= a_1 \dot{\bar{t}}_2 + b_1 \dot{\bar{q}}_1 + b_2 \bar{q}_1 + b_3 \quad ;$$

$$a_1 p t_2(p) = -(b_1 p + b_2 + 1) q_1(p) \ ;$$

$$Z\ (p) = \frac{t_2(p)}{q_2(p)} = -\frac{b_1 p + b_2 + 1}{a_1 p} .$$

4. Step

$$Z\ (j\Omega) = -\frac{b_1 j\Omega + b_2 + 1}{a_1 j\Omega} = \frac{-b_1\Omega + (b_2 + 1) j}{a_1\Omega} .$$

5. Step

$$R_e(j\Omega)=\frac{-b_1\Omega}{a_1\Omega}\,;$$

$$I_m(j\Omega)=\frac{(b_2+1)\,j}{a_1\Omega}\,.$$

6. Step

$$A(j\Omega)=\sqrt{R_e(j\Omega)^2+I_m(j\Omega)^2}\,;$$

$$\varphi(j\Omega)=-\arg tg\,\frac{I_m(j\Omega)}{R_e(j\Omega)}\,.$$

Table 4.1 – The parameters of the energy chain.

V0	r_1	m	P0	l	r_2
0,27	111	1	0,01	0,000009	111

Table 4.2 – The output parameters of the energy chain 1

	$\boxed{\mathrm{Re}(j\Omega)}$	$\boxed{\mathrm{Im}(j\Omega)}$	$\boxed{A(j\Omega)}$	$\boxed{\varphi(j\Omega)}$
1	-59,4398	-0,59889	59,44285	-0,01008
2	-59,4393	-1,19777	59,45141	-0,02015
3	-59,4385	-1,79665	59,46567	-0,03022
4	-59,4374	-2,39554	59,48563	-0,04028
5	-59,4359	-2,99442	59,51127	-0,05034
6	-59,4341	-3,5933	59,54261	-0,06039
7	-59,4319	-4,19218	59,57962	-0,07042
8	-59,4295	-4,79106	59,62229	-0,08044
9	-59,4267	-5,38994	59,67062	-0,09045
10	-59,4236	-5,98882	59,72459	-0,10044

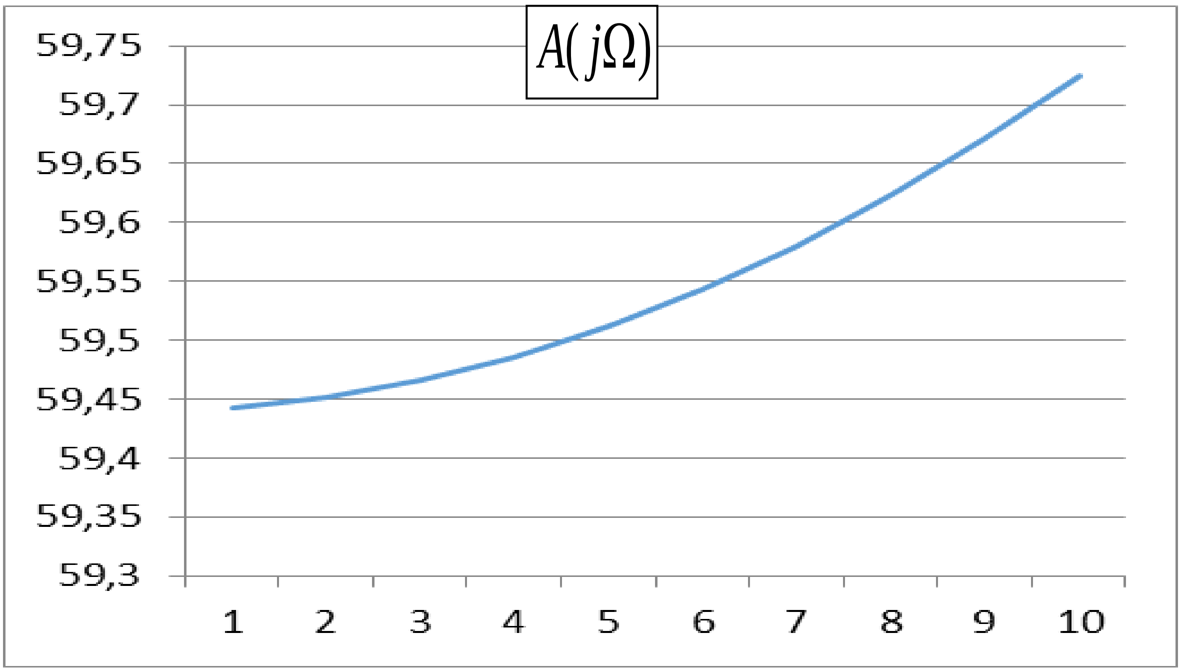


Figure 4.5 – Amplitude-frequency function of the energy circuit 1

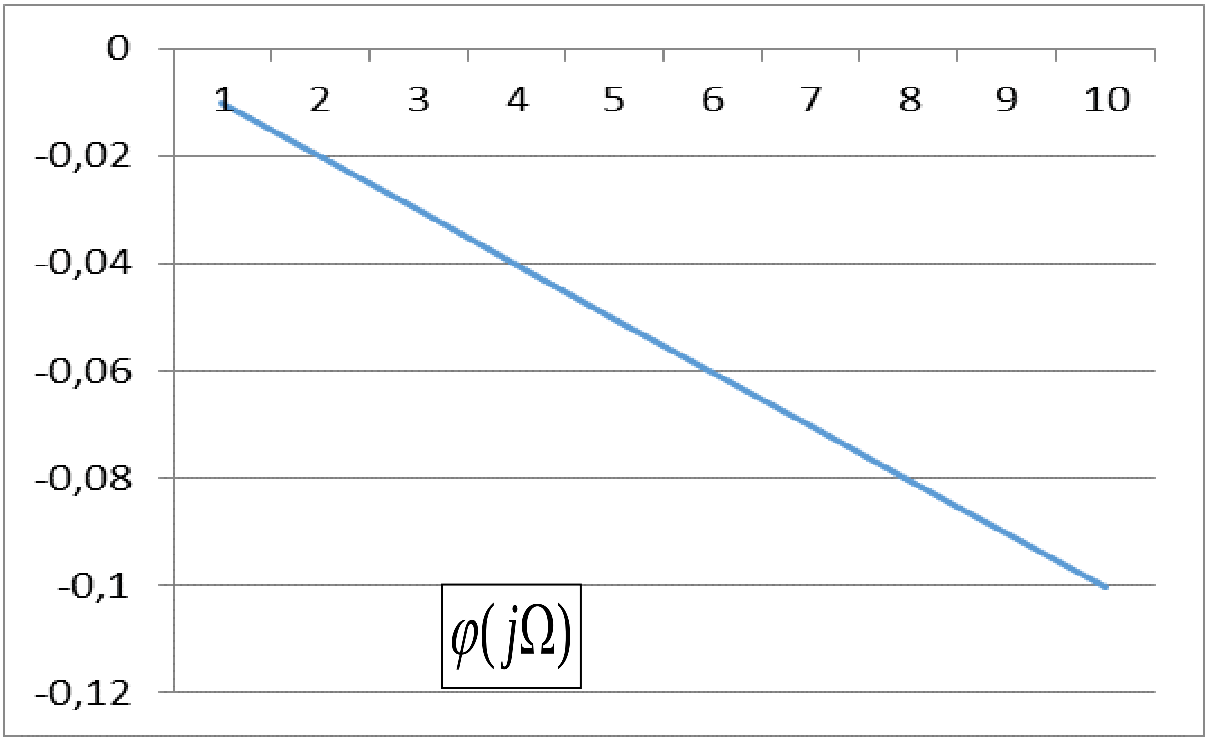


Figure 4.5 – Phase-frequency function of the energy circuit 1

Table 4.3 – Выходные параметры энергетической цепи

№	$\boxed{Re(j\Omega)}$	$\boxed{Im(j\Omega)}$	$\boxed{A(j\Omega)}$	$\boxed{\varphi(j\Omega)}$
1	-111	222222,2	222222,2	1,570297
2	-111	444444,4	444444,5	1,570547
3	-111	666666,7	666666,7	1,57063
4	-111	888888,9	888888,9	1,570671
5	-111	1111111	1111111	1,570696
6	-111	1333333	1333333	1,570713
7	-111	1555556	1555556	1,570725
8	-111	1777778	1777778	1,570734
9	-111	2000000	2000000	1,570741
10	-111	2222222	2222222	1,570746

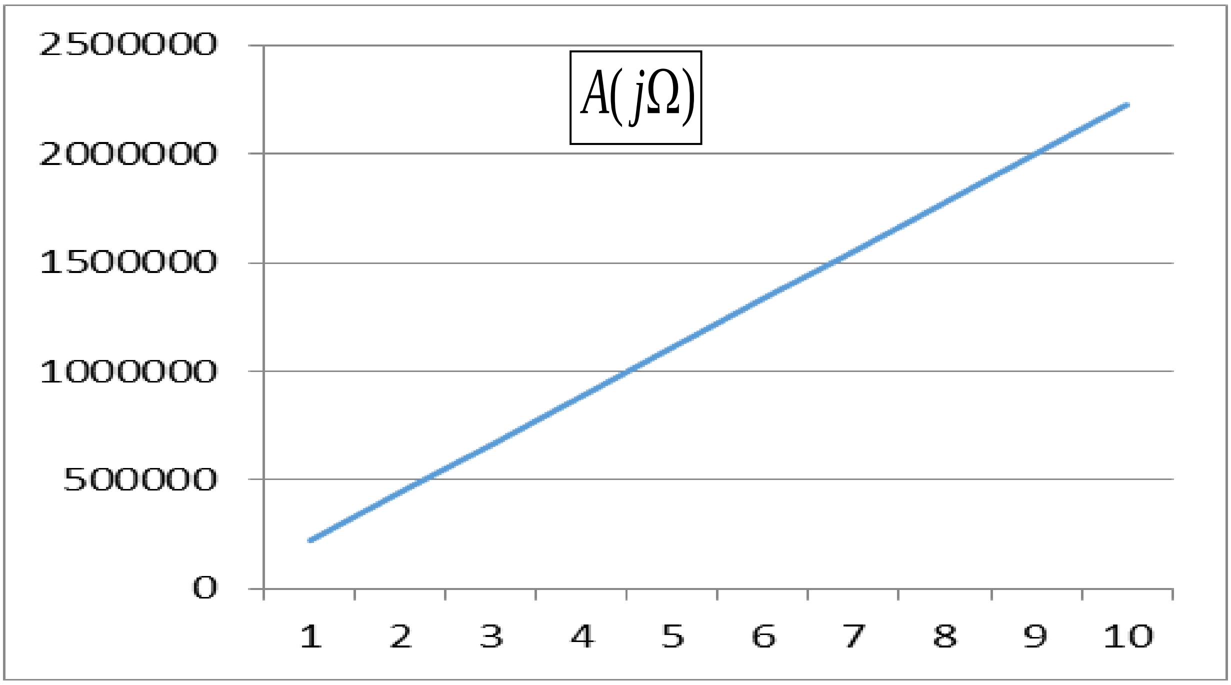


Figure 4.6 – Amplitude-frequency function of the energy circuit 2

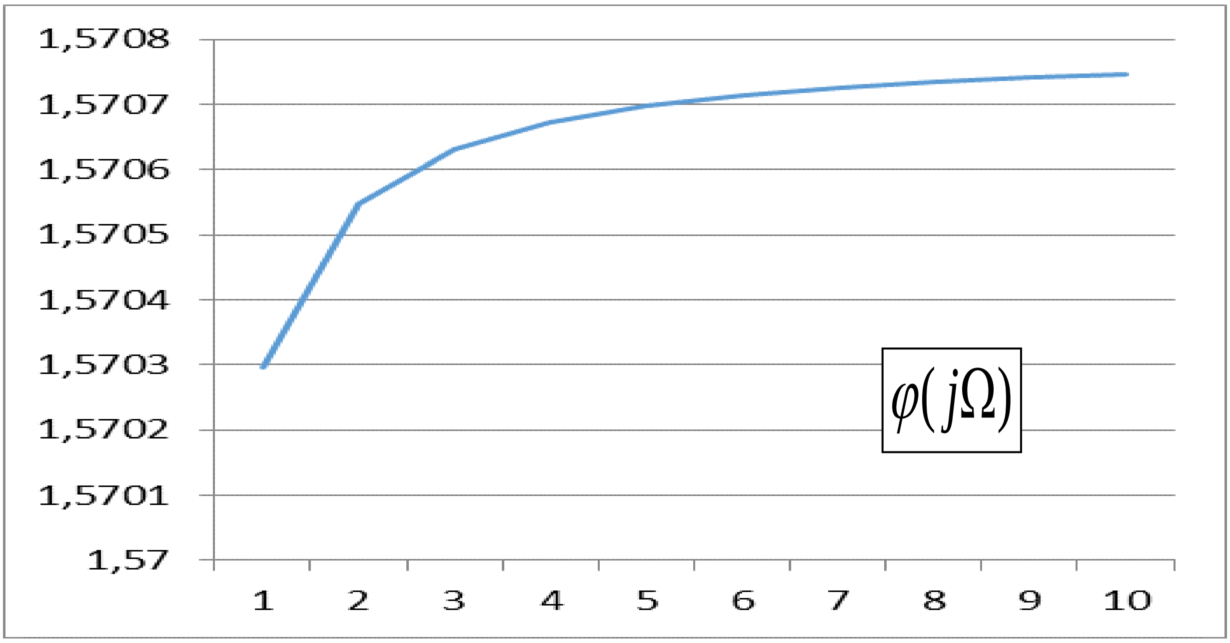


Figure 4.7 – Phase-frequency function of the energy circuit 2

5 The equipment of the experiment

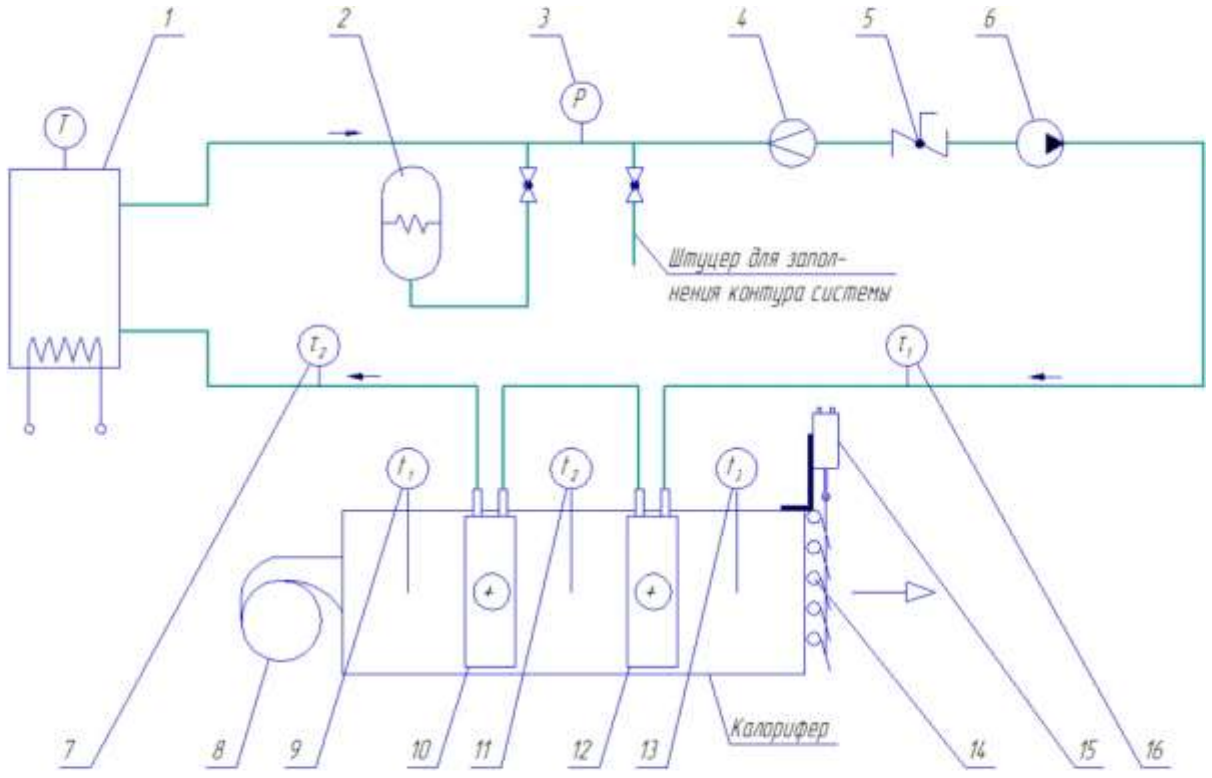
This chapter will present the experimental principle of a heater with a fluctuating air flow, as well as an experimental circuit diagram, basic tests and devices. The device, connection, and pyrometer will be described, as well as experimental methods and procedures. At the same time, a method for processing experimental data is described.

5.1 The principle of the experiment

In this installation, cold air will flow through the vent holes and blow into the space of the air tube, then it will pass through the walls of the heated calorimeter and increase the temperature, at the exit we will get heated air with a changed temperature, the air flow at the exit will be regulated by means of blinds with a drive.

5.2 The introduction of experimental device

In this paper, we consider the influence of the pulse mode of the coolant flow on the heat transfer "heater-environment". To study this issue, the main thermal scheme of the experimental installation is shown in Fig. 5.1.



Convention: 1 – electric boiler, 2 – hydraulic accumulator, 3 – pressure gauge, 4 – electromagnetic flow converter, 5 – manual balancing valve, 6 – circulation pump, 7 and 16 – thermal resistance converters designed to measure the temperature of the coolant after and before the heater, 8 – fan of the heater block, 9, 11 and 13 – thermal resistance converters designed to measure the temperature of the air flow at different points of the heater block, 10 and 12 – water - air heat exchangers of the heater block 1st and 2nd heating stages, 14 – air damper, 15 – electric air damper.

Figure 5.1 – Experimental setup diagram

5.3 The content and step of the experiment

The experimental installation consists of a block of a heater, a source of thermal energy-an electric boiler and a measuring complex (figure 5.2). The heater unit is a frame-panel construction unit that includes a fan that provides air flow in the range of 50-350 m3/h and two water-air heat exchangers installed in series. The coolant flow is selected according to the counter-current scheme.

Figure 5.2 – Photo of the appearance of the heater unit and the complex, including a circulation pump, a hydraulic accumulator, a balancing valve and an electromagnetic fluid flow converter.

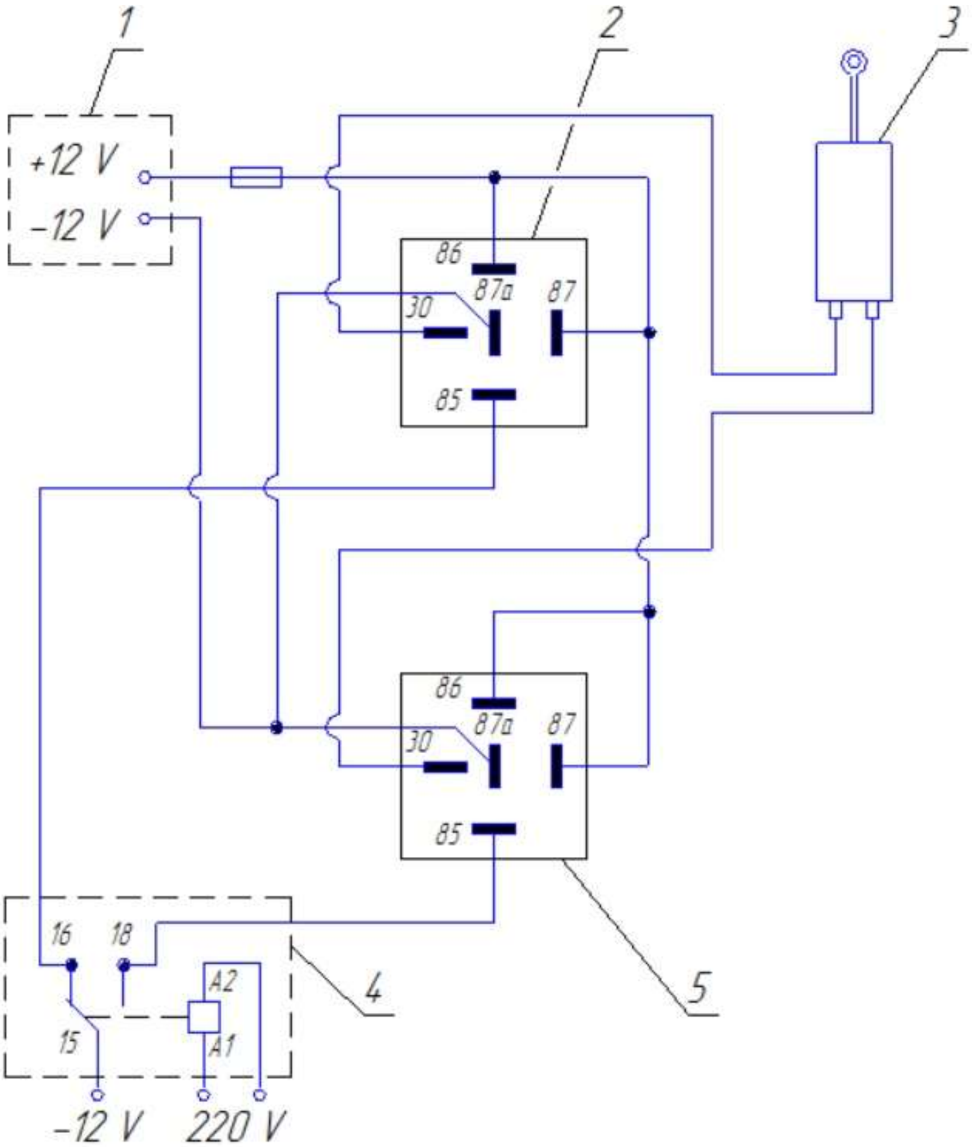
The measuring complex consists of an electromagnetic pre-educator of the liquid flow Masterflow 4, thermal converters of the resistance 7 and 16 for measuring the temperature of the coolant after and before the heat exchangers of the heater and 9, 11 and 13 for measuring the temperature of the air flow at different points of the heater block (figure 5.3).

Figure 5.3 – The part of the appearance of the experimental installation

To enure the circulation of the heated medium, the circuit has a Wilo 6 pump. Changing the flow rate of the heated medium is possible with the help of a manual balancing valve 5. Smoothing possible hydraulic shock in the system is perceived by the membrane of the hydraulic accumulator 2, also included in the scheme (figure 5.4).

Figure 5.4 – The part of the appearance of the experimental installation

Creating a fluctuating air flow through the section of the heater block is organized by us using a standard ventilation grate and an electric drive, the frequency of which is set by the setpoints of the operating periods and the time relay. The electrical power supply diagram of the electric drive is shown in figure 5.5.



Convention: 1 – DC power supply 12 V, 2 and 5 – voltage relay, 3 – electric drive, 4 – time relay RVC-1M

Figure 5.5 – Electrical power supply diagram for the ventilation grid electric drive

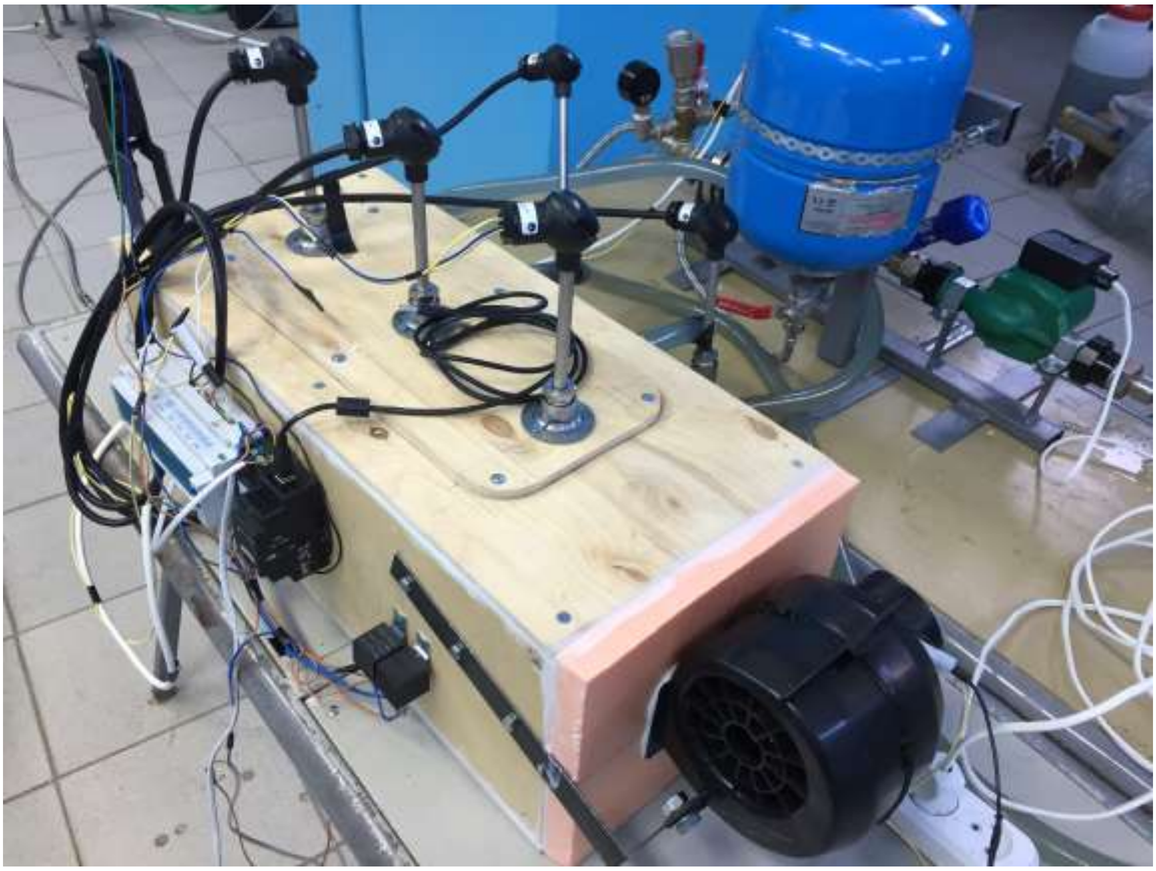


Figure 5.6 – General view of measurement sensors

DS sensors are designed for continuous temperature measurement of liquid, vapor and gaseous media, bulk materials and solids in various industries, as well as in heating, ventilation and air conditioning systems.

There are two types of thermal converters: General industrial and specialized. The sensors can be cable-connected or with a switching head, in various design versions that allow them to be mounted on a pipe, on a wall, immersed in the medium, etc.

Sensors convert a change in temperature into a change in electrical resistance to direct current. They consist of one or two sensing elements (hereinafter CE) connected to the switching head or cable terminal and placed in a protective armature. Depending on the measured temperature range, the PE can be: platinum wire 100 P or platinum film Pt 100 (Pt 500, Pt 1000), copper wire 50 M or 100 M. The main technical characteristics of the DTS XX4 and DTS XX5 type of sensors are shown in table 5.1.

Operating conditions of switching units: rooms with non-regulated climatic conditions and (or) sheds, at atmospheric pressure from 84 to 106.7 kPa, with a temperature in the range of at least minus 40 to plus 85 °C and a relative humidity of no more than 95 % at +35 °C and lower temperatures without condensation.

Table 5.1 – Main technical characteristics of DTS XX4 and DTS XX5 sensors

Characteristic	Value			
	DTSHX4		DTSHH5	
1	2	3	4	5
Nominal static characteristic (NSC)	50M; 100M	50P; 100P; Pt 100 Pt 500 Pt 1000	50M; 100M	50P; 100P; Pt 100 Pt 500 Pt 1000
Measured temperature range, °C	-50…+150	-50…+250	-50…+180	-50…+500
Class of admission	B; C	A; B; C	B; C	A; B; C
The index of thermal inertia	10…30			
Number of sensitive elements, PCs.	1 2			
The scheme of internal connections of the conductors	two-wire three-wire four-wire			
The performance of the sensor relative to the housing	isolated			
Cable output length	0.2 m-standard up to 20 m-on request		-	
The execution of the switching head	-		plastic, metal	
Type of threaded connection	metric thread, pipe thread			
Material of protective reinforcement	steel 12X1810T brass		steel 12X1810T	
Degree of protection (according to GOST 14254)	IP54; IP65			
Average time to failure, 1 hour, at least	35000			
Average service life, years, not less	8			

We accept as a means of measuring the temperature of the supply air and the circulating coolant temperature Converter DTS035-50 M. B3. 60 (figure 4). To measure the temperature of the cooling medium, we will use a thermal Converter DTS314-50 M. B3. 40/2 (figure 5.7).

Figure 5.7 – General view of the dts035-50M thermal Converter.V3. 60

Figure 5.8 – General view of the dts314-50M thermal Converter.V3. 40/2

To measure the flow of coolant in the system, the scheme includes a primary flow Converter-an electromagnetic Converter " mA-sterflou "(modification with a current output of 4...20 mA) produced by NPO"Prompribor". The Converter converts the liquid flow rate to an analog signal of 4.20 mA in the range of 0.020. 5.0 m3 / h (figure 5.9).

Figure 5.9 – Photo of the appearance of the Masterflow flow Converter

To input analog signals to a personal computer, we used the MVA 8 input module (figure 5.10). It can also be used to build automatic systems for monitoring and regulating production and technological processes in various areas of industry, transport, agriculture, utilities and other sectors of the national economy.

The housing contains a printed circuit Board on which the elements of the device's circuit are located.

On the front panel of the device there are two LEDs that serve to indicate the power connection and indicate the operation of the RS-485 network interface.

Figure 5.10 – Input module analog measuring MVA 8

The device performs the following main functions:

- measurement of physical parameters of the object controlled by input primary converters;
- digital filtering of the measured parameters from industrial im-pulse interference;
- correction of measured parameters to eliminate errors of primary converters;
- formation of an alarm signal when a fault is detected in the primary converters;
- transmitting information to the computer about the values of the values measured by the sensors or the values obtained after converting these values;
- changing the values of its programmable parameters using the configuration program;
- saving the set programmable parameters in non-volatile memory when the power supply voltage is switched off MVA 8;
- reading of position sensors (resistive and current type) and contact discrete sensors.

Operating conditions:

- closed explosion-proof rooms without aggressive vapors and gases;

- ambient temperature from + 1 °C to + 50 °C;

- upper limit of relative humidity-80 % at 25 °C and lower temperatures without condensation;

- atmospheric pressure - from 86 to 106.7 kPa.

Primary transducers (sensors) are designed to control the physical parameters of an object (temperature, pressure, flow, etc.) and convert them into electrical signals that are optimal for further processing.

As input sensors of the device can be used:

- thermal resistance converters;

- thermocouples (thermoelectric converters);

- active converters with an analog output signal in the form of a constant voltage or current;

the position sensors of the actuators;

- dry contacts of the relay or switch.

Thermal resistance converters (hereinafter referred to as TS) are used for measuring the ambient temperature at the sensor installation site. The principle of operation of such sensors is based on the existence of a reproducible and stable dependence of the active resistance on temperature for a number of metals. As a material for manufacturing vehicles in the industry, specially processed copper (for TSM sensors), platinum (for TSP sensors) or Nickel (for TSN sensors) wire is most often used.

The output parameters of the vehicle are determined by their nominal static characteristics (hereinafter NSC), standardized by GOST R 50353-92. The main parameters of the NSC are: the initial resistance of the sensor R0, measured at 0 °C, and the temperature coefficient of resistance W100, defined as the ratio of the resistance of the sensor, measured at 100 °C, to its resistance, measured at 0°C. Due to the fact that the NSC of thermal resistance converters are non – linear functions (for TSM in the region of negative temperatures, and for TSP in the entire range), the device provides means for linearization of readings.

In order to avoid the influence of the resistances of the connecting wires on the results of temperature measurement, the sensor should be connected to the device using a three-wire circuit. With this scheme, two wires connecting it to the device are connected simultaneously to one of the terminals of the vehicle, and a third connecting wire is connected to the other terminal. To fully compensate for the influence of connecting wires on the measurement results, it is necessary that their resistances are equal to each other (it is sufficient to use single wires of equal length).

In some cases, it is necessary to connect the vehicle using a two-wire scheme rather than a three-wire one, for example, in order to use existing communication lines on the object.

Thermoelectric converters (hereinafter referred to as TP) as well as thermal resistance converters are used for temperature measurement. The principle of operation of thermocouples is based on the Seebeck effect, according to which the heating of the junction point of two dissimilar conductors causes the appearance of an electromotive force at the opposite ends of this chain-Ter – moeds. The value of the thermal EMF is initially determined by the chemical composition of the conductors and also depends on the heating temperature.

NSCS of various types of thermocouples are standardized by GOST R 8.585-2001. Since the characteristics of all thermocouples are more or less non-linear functions, the device provides means for linearization of readings.

The junction point of dissimilar conductors is called the working junction of the thermocouple, and their ends are free ends or, sometimes, cold junction. The working junction of the thermocouple is located in the place selected for temperature control, and the free ends are connected to the measuring device. If it is not possible to connect the free ends directly to the contacts of the MVA 8 (for example, due to their distance from each other), then the connection of the thermocouple with the device must be performed by means of compensating thermoelectric wires or cables, with the mandatory observance of the polarity of their inclusion. The need to use such wires is due to the fact that the EMF of a thermocouple depends not only on the temperature of the working junction, but also on the temperature of its free ends, the value of which is controlled by a special sensor located in the device. In this case, the use of thermoelectric cables allows you to increase the length of the conductors of the pair and "transfer" its free ends to the terminal block MVA 8.

Active converters with an analog output signal are used in accordance with the purpose of the sensor for measuring physical parameters such as pressure, temperature, flow, level, etc. The output signals of such sensors can be either the DC voltage changing according to a linear law, or the value of the current itself.

Active sensors can be powered either from a built-in DC source with an output voltage of 24 ± 3 V, or from an external power supply.

Connection of sensors with signal output at a constant voltage (-50,0...50,0 mV or 0...1.0 volt) may be connected to the device contacts, and sensors with output in the form of current - only after the installation of shunt resistor 100 Ohm (tolerance not more than 0.1 %). As a shunt, it is recommended to use highly stable resistors with a minimum value of the temperature coefficient of resistance, for example, type C2-9V.

Position sensors are designed to determine the current position (degree of opening or closing) of shut-off and control valves, valves, scrapers, etc.when adjusting process parameters.

Resistive position sensors are most commonly used in the industry. In sensors of this type, an alternating resistance resistor is used as a sensing element, the slider of which is mechanically connected to the regulating part of the actuator.

The MVA 8 is capable of processing signals from resistive sensors with a resistance of up to 900 Ohms or 2.0 kOhm.

Sensors are also used that generate an output signal in the form of a linearly varying current, the value of which depends on the position of the actuator at the moment.

The device is capable of processing sensor signals with a current output of 0...5 mA, 0.. .20 mA and 4 ... 20 mA.

Up to 16 discrete sensors, called "Dry contacts", can be connected to the MVA 8. As sensors can be vyklyuchat Lee, button, contact group relay, etc. Each analog input can be used to connect two discrete sensors.

Any resistors with the same nominal value of 60-90 Ohms can be used as shunt resistors.

The device can be used simultaneously to work with various types of sensors-resistance thermal converters, thermocouples, etc. In this case, it is not important to which input of the MVA 8 will be connected to a sensor of one type or another, since all eight inputs of the device are absolutely identical.

After connection, the sensors are assigned serial numbers of the device inputs to which they are connected (input 1 corresponds to sensor # 1, input 2-sensor # 2, etc.). the Type of each sensor is set by the user in the form of a digital code in the in-t programmable parameter when preparing the device for operation.

During operation, the device monitors the operability of the primary converters connected to it and sends an error message via the RS-485 network interface if any of them is detected (figure 5.11).

Figure 5.11 – RS-85 Network interface

Errors are generated by:

- when working with thermal resistance converters in the event of their failure or short circuit;

- when working with thermoelectric converters in the event of their break, as well as when the temperature of the free ends of the thermocouples increases above 90 °C or when it decreases below +1 °C;

- when working with any type of primary transducers, if the measurement results are obtained that go beyond the limits of the control range set for this sensor.

To program the MVA 8 device, you must connect it via the RS-485 ARIES AC 4 interface adapter to a personal computer and connect power to the device.

Programming of the analog MVA8 input module was performed using the program "MVA 8 Configurator".

Programming procedure for the device:

1) Launch the program "MBA Configurator 8".

2) Setting the program connection with the device.

3) Create a new configuration and then open it from the file.

4) Setting the sensor type for each input used, the polling period, and other characteristics;

5) Setting the upper and lower limits of the measured range of sensors;

6) Recording the configuration in the device.

After starting, the program establishes a connection with the device. The connection is determined when sending commands to switch to the ARIES Protocol. The connection is established based on the network parameters that were set when the program was last launched.

If the connection is established, the Main program window opens. If the program was not able to establish a connection with the device and transfer the MVA 8 to work using the ARIES Protocol, the window for setting network settings opens (figure 5.12).

The table (in the upper part of the window) shows information about the current network settings of the device (a description of the setting, the name of the corresponding parameter and its value). Parameter values can be set directly in the table.

When the "Connect" button is clicked, the device search procedure is started to establish a connection with the device.the search process uses all available communication protocols at the set network settings and speed. When the device responds for the first time, the search procedure is stopped.

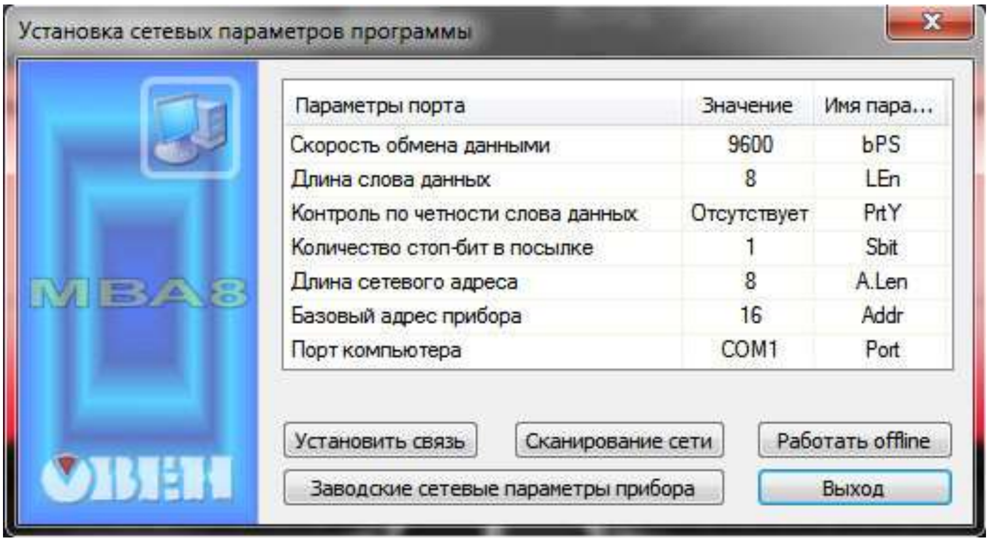


Figure 5.12 – Network settings Window

When you click the network Scan button, the network scan procedure starts to establish communication over all available protocols, with an interruption in exchange rates, starting from the exchange rate of 2400 and then up to the speed of 115200. Other network settings (parity, data word length, etc.) do not change during scanning. Thus, the "network Scan" mode is an extended communication setup mode. The first time the device is clicked, the scan stops.

When the factory network settings button is clicked, the Factory network settings are set and the connection attempt is repeated.

When you click the "Work offline" button, the program stops trying to communicate with the device; the main window of the Configurator opens. This disables the automatic reading of the network parameters.

When you click Exit, the program exits and the window closes.

After the connection is established (immediately or after changing the network settings), or after the offline mode is selected, the main window of the configuration program opens (figure 5.13), containing the menu, toolbar, and workspace. In the workspace, as a hierarchically organized list, the configuration of MBA 8 is displayed, which contains two main sections (two branches of the program tree): Device parameters and program Parameters.

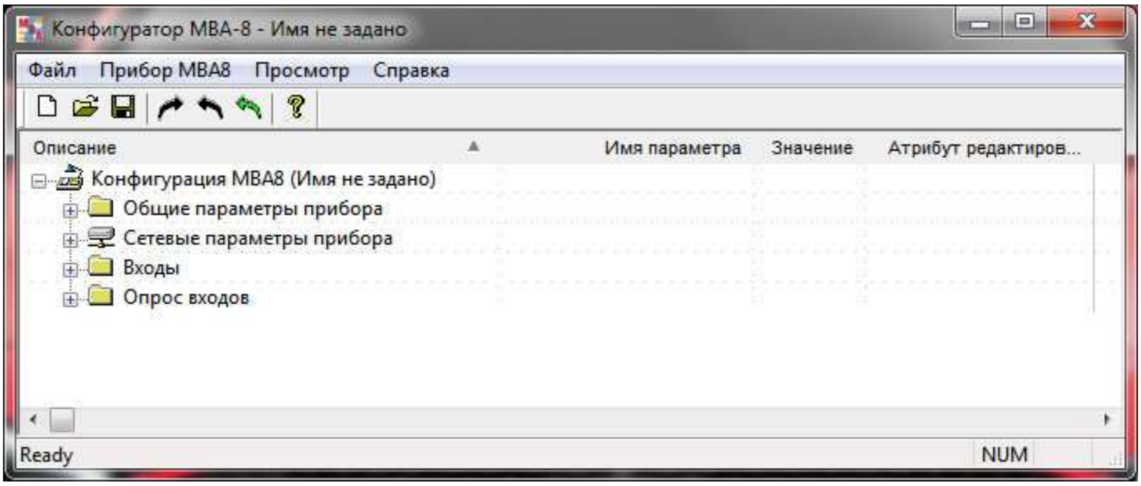


Figure 5.13 – Main window of the Configurator program

The main window includes a title line that shows the "Configurator MVA 8" and the name of the current configuration of the device, menu and toolbar, the display area of the list of sections and program settings and device settings (in the left part of the window) and the display of parameter values (on the right).

The "device Parameters" section includes four folders:

- General parameters of the device-contains non-editable information parameters (device name, SOFTWARE version, and a message about the reason for restarting the device);

- network parameters of the device-contains a list of network parameters that determine the operation of the device via the RS-485 interface;

- inputs-contains 8 subfolders Input #1, Input #2 ... Input #8 with individual parameters of the MVA 8 inputs and a common CJ -. C parameter for all inputs;

- input polling - allows you to view and save the values measured by the device. These values are displayed in the program window in a converted form: for thermal converters and thermocouples, the temperature measured in degrees Celsius is output; for active sensors, the values are recalculated in accordance with the units of the measurement range.

Figure 5.14 – VAZ 2108 stove fan

The centrifugal fan the snail combines reliability of design, excellent indicators of power and simplicity of operation. The basic purpose of the supercharger consists in movement of big air masses

Table 5.3.2 – Main technical characteristics of VAZ 2108 stove fan

Parameters	Current values for operation
Rated voltage	12 V
Rated power	90 W
Maximum current	14.0 A
Rated speed	4100 min -1
Weight	1,0 kg

Figure 5.15 – Copper and aluminum heat exchangers

Recuperative heat exchangers are characterized by the fact that in them processes of heat transfer from the one environment to another proceed through the wall separating these Wednesdays. liquid-air heat exchangers, in which the heat carrier is the hot or cold water In such heat-exchange apparatus, transfer of warmth from more heated environment to less heated proceeds at the same time through their change wall. The specified heat exchangers working at hot and cold water.

a)

• Core type: prefabricated

• Core material: aluminum

• The size of the core, mm 200*191*42

b)

• Core type: prefabricated

• Core material: copper

• The size of the core, mm 200*191*42

Figure 5.16 – Hot water boiler handmade

The boiler in this installation maintains a constant temperature in the circuit. The heating range is from 0 to 100 degrees. A constant temperature is maintained using a thermostat. Powered by a 220 V network.

When planning experiments, they are usually divided into active ones, when the study is carried out at specially set values of input parameters and, as a rule, on a special experimental installation, and passive ones, when the measurement results are obtained on working objects, usually without installing additional devices or assigning special tests. As a result of planning experiments, the number and sequence of experiments that ensure the achievement of the set research goals with the required accuracy and reliability is determined. The experimental setup is always specialized, unique and is built taking into account the specific features of the object and specific research tasks.

The accuracy of the measurement is determined by the degree to which the measurement results correspond to the actual value of the measured value. The difference between these two values is called the absolute measurement error. The ratio of absolute error to the true value, expressed as a percentage, is called the relative error.The reasons for errors are changes that occur during testing, the temperature of the outdoor air or in the object. Can you name the three sources of errors:

sensor of the primary measuring signal;

- measuring devices for air temperature and liquid flow in the system;

- the experimenter himself, who, due to various circumstances, may not correctly take the readings of the devices.

To solve this problem, we will conduct an active experiment with the use of an experimental installation, the scheme of which is given in the third Chapter. We will record the results of the experimental study in a table.

An experiment to determine the efficiency of exhaust air heating in pulsating and continuous modes of air flow was conducted on June 12, 2020, under the following climatic conditions-internal air parameters in the laboratory: the temperature varied from 26.0 to 26.3°C, relative humidity - from 28 to 31 %, the initial temperature of the heated medium varied from 11.6 to 15.7 °C.

The experiment was performed under two operating modes of the fan 226 and 278 m3 / h by means of a switch of operating modes.

The air flow rate through the experimental unit was determined as a result of measurements of the flow rate in two different modes of operation of the ventilator using the MES-200A meteometer (figures 5.17, 5.18).

Figure 5.17 – Photo of the process of measuring the air flow velocity in the section of the experimental installation

Figure 5.18 – Photo of the air flow speed reading on the MES-200A weather meter display

The air flow velocity at the first fan speed was 1.19 m / s, and at the second speed it was 1.45 m/s. The internal dimensions of the live section of the experimental unit are 260x205 mm. The calculation determines the air flow rate when working at the first speed of 226 m3 / h, at the second speed-278 m3/h.

The pulsating air flow mode was created using a standard ventilation grate with an adjustable live section. The grating has a flap, which, by means of a rigid rod, makes reciprocating movements by means of an electric drive, thereby opening and closing the way for air movement.

At the initial stage of the experiment, the unit was switched on with the fan running at the first speed. The liquid flow regime varied from a minimum flow rate of 50 kg/h to a maximum flow rate of 400 kg/h. Under the same operating conditions of the fan and the liquid flow mode, the parameters of the air and the heated liquid were recorded in pulsating and continuous air flows. Then the fan operation was switched to the second speed and the display was repeated.

6 The analysis of the experiment results

Имя параметра	Период	Значение
Вход №1	1000	25.2
Вход №2	1000	29.8
Вход №3	1000	37.2
Вход №4	1000	
Вход №5	1000	43.1
Вход №6	1000	40.5
Вход №7	1000	
Вход №8	1000	190.6

Имя параметра	Период	Значение
Вход №1	1000	25.2
Вход №2	1000	28.6
Вход №3	1000	34.9
Вход №4	1000	
Вход №5	1000	40.8
Вход №6	1000	36.0
Вход №7	1000	
Вход №8	1000	52.8

Имя параметра	Период	Значение
Вход №1	1000	25.2
Вход №2	1000	28.0
Вход №3	1000	34.0
Вход №4	1000	
Вход №5	1000	39.6
Вход №6	1000	35.8
Вход №7	1000	
Вход №8	1000	102.2

Имя параметра	Период	Значение
Вход №1	1000	25.2
Вход №2	1000	28.0
Вход №3	1000	34.8
Вход №4	1000	
Вход №5	1000	46.2
Вход №6	1000	37.8
Вход №7	1000	
Вход №8	1000	102.1

Имя параметра	Период	Значение
Вход №1	1000	25.3
Вход №2	1000	29.2
Вход №3	1000	37.2
Вход №4	1000	
Вход №5	1000	41.8
Вход №6	1000	38.8
Вход №7	1000	
Вход №8	1000	147.9

Имя параметра	Период	Значение
Вход №1	1000	25.4
Вход №2	1000	31.6
Вход №3	1000	41.8
Вход №4	1000	
Вход №5	1000	49.0
Вход №6	1000	46.1
Вход №7	1000	
Вход №8	1000	199.6

Figure 6.1 – Fan operation mode-1 speed

Имя параметра	Период	Значение
Вход №1	1000	25.5
Вход №2	1000	30.3
Вход №3	1000	39.3
Вход №4	1000	
Вход №5	1000	46.6
Вход №6	1000	42.7
Вход №7	1000	
Вход №8	1000	197.1

Имя параметра	Период	Значение
Вход №1	1000	25.5
Вход №2	1000	29.8
Вход №3	1000	38.4
Вход №4	1000	
Вход №5	1000	45.1
Вход №6	1000	41.6
Вход №7	1000	
Вход №8	1000	197.1

Имя параметра	Период	Значение
Вход №1	1000	25.5
Вход №2	1000	28.8
Вход №3	1000	36.3
Вход №4	1000	
Вход №5	1000	42.7
Вход №6	1000	38.9
Вход №7	1000	
Вход №8	1000	153.7

Имя параметра	Период	Значение
Вход №1	1000	25.5
Вход №2	1000	28.1
Вход №3	1000	35.8
Вход №4	1000	
Вход №5	1000	49.3
Вход №6	1000	39.2
Вход №7	1000	
Вход №8	1000	102.8

Имя параметра	Период	Значение
Вход №1	1000	27.1
Вход №2	1000	28.7
Вход №3	1000	30.4
Вход №4	1000	
Вход №5	1000	42.2
Вход №6	1000	33.4
Вход №7	1000	
Вход №8	1000	46.6

Figure 6.2 – Fan operation mode-2 speed

Имя параметра	Период	Значение
☑ Вход №1	1000	27.4
☑ Вход №2	1000	29.6
☑ Вход №3	1000	31.6
☐ Вход №4	1000	
☑ Вход №5	1000	46.7
☑ Вход №6	1000	36.9
☐ Вход №7	1000	
☑ Вход №8	1000	44.9

Имя параметра	Период	Значение
☑ Вход №1	1000	27.4
☑ Вход №2	1000	30.1
☑ Вход №3	1000	32.3
☐ Вход №4	1000	
☑ Вход №5	1000	46.3
☑ Вход №6	1000	38.7
☐ Вход №7	1000	
☑ Вход №8	1000	102.9

Имя параметра	Период	Значение
☑ Вход №1	1000	27.7
☑ Вход №2	1000	31.3
☑ Вход №3	1000	34.0
☐ Вход №4	1000	
☑ Вход №5	1000	50.2
☑ Вход №6	1000	42.9
☐ Вход №7	1000	
☑ Вход №8	1000	154.7

Имя параметра	Период	Значение
☑ Вход №1	1000	27.7
☑ Вход №2	1000	32.5
☑ Вход №3	1000	35.5
☐ Вход №4	1000	
☑ Вход №5	1000	46.7
☑ Вход №6	1000	42.5
☐ Вход №7	1000	
☑ Вход №8	1000	197.4

Figure 6.3 – Fan operation mode-3 speed

The efficiency of the heater in pulsating and continuous air flows was determined by the formula:

$$E = \frac{\tau_2 - \tau_1}{t_1 - \tau_1} \cdot 100\% , \qquad (6.1)$$

where is the temperature of the heated medium before the heater, °C;

- temperature of the heated medium after the heater, °C;

- exhaust air temperature at the inlet to the heater, °C.

The results of calculating the efficiency of a prototype heater with continuous and pulsating air supply modes are summarized in tables 6.1 and 6.2.

Table 6.1 – Results of calculating the effectiveness of a multi-factor experiment with a continuous flow of air

A continuous stream of air							
L, m3 / h	G, kg / h	τ1, ℃	τ2, ℃	t1, ℃	t2, ℃	t3, ℃	Temperature difference , ºC
226	50	40,8	36	25,2	28,6	34,9	9,70
226	100	46,2	37,8	25,2	28	34,8	9,60
226	150	41,8	38,8	25,3	29,2	37,2	11,90
226	200	49	46,1	25,4	31,6	41,8	16,40
278	50	42,2	33,4	27,1	28,7	30,4	3,30
278	100	49,3	39,2	25,5	28,1	35,8	10,30
278	150	42,7	38,9	25,5	28,8	36,3	10,80
278	200	46,6	42,7	25,5	30,3	39,3	13,80
343	50	46,7	36,9	27,4	29,6	31,6	4,20
343	100	46,3	38,7	27,4	30,1	32,3	4,90
343	150	50,2	42,9	27,7	31,3	34	6,30
343	200	46,7	42,5	27,7	32,5	35,5	7,80

Table 6.2 – Results of calculating the efficiency of a multi-factor experiment with a fluctuating air flow

Pulsating air flow							
L, m3 / h	G, kg / h	τ1, °C	τ2, °C	t1, °C	t2, °C	t3, °C	Temperature difference, °C
226	50	40,8	36	25,2	28,7	35,1	9,90
226	100	46,2	37,8	25,2	28,1	34,9	9,70
226	150	41,8	38,8	25,3	29,2	37,3	12,00
226	200	49	46,1	25,4	31,6	42	16,60
278	50	42,2	33,4	27,1	28,8	30,5	3,40
278	100	49,3	39,2	25,5	28,2	36	10,50
278	150	42,7	38,9	25,5	28,9	36,5	11,00
278	200	46,6	42,7	25,5	30,4	39,5	14,00
343	50	46,7	36,9	27,4	29,7	31,7	4,30
343	100	46,3	38,7	27,4	30,2	32,4	5,00
343	150	50,2	42,9	27,7	31,4	34,2	6,50
343	200	46,7	42,5	27,7	32,6	35,7	8,00

The results obtained are shown graphically in figures 6.4 and 6.5 for illustrative purposes.

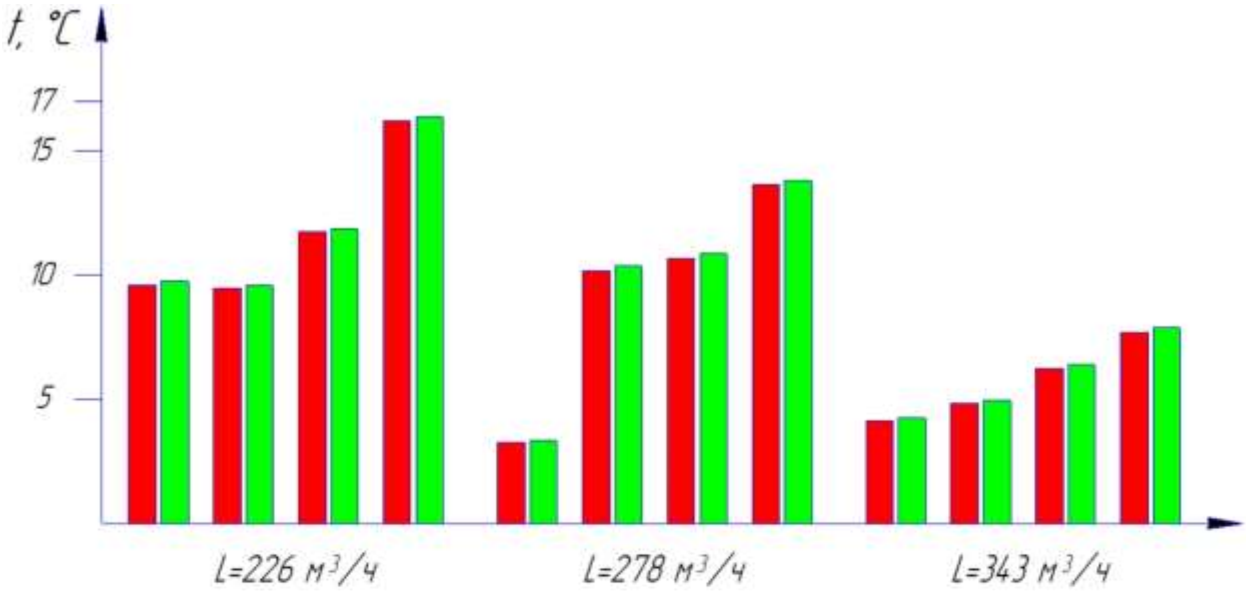


Figure 6.4 – Comparative graph of the heating efficiency dependence in continuous and pulsating air flows with a valve capacity of 226 – 343 m3/h

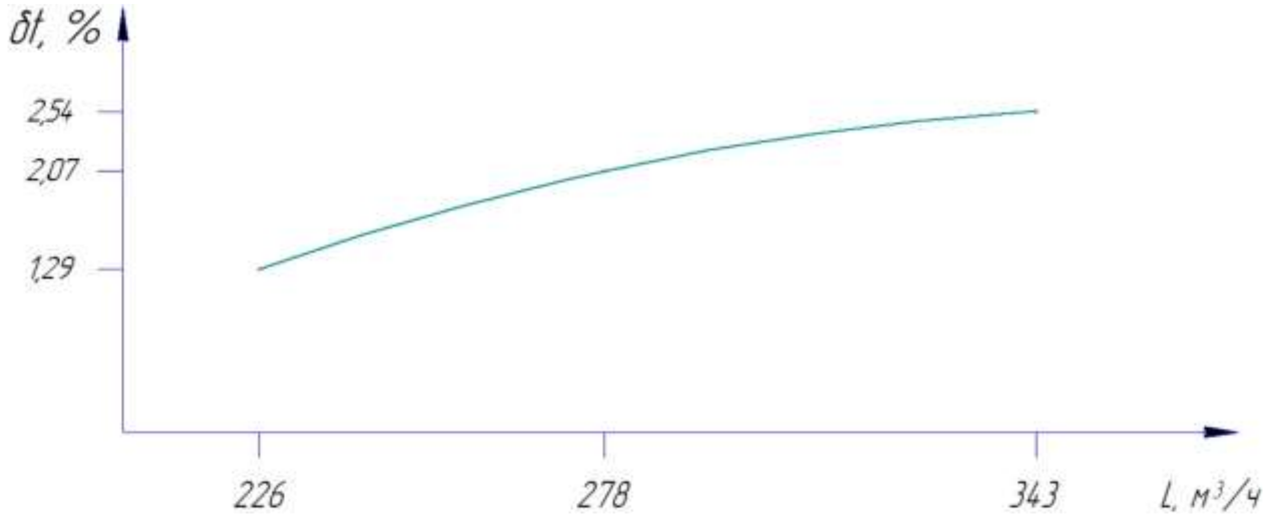


Figure 6.5 – Comparative graph of the heating efficiency dependence in continuous and pulsating air flows

As can be seen from the results of processing the experiment, the heating efficiency in the pulsating mode increases slightly from 1.29 to 2.54%.

CONCLUSION

In this way, the characteristics of the air-conditioning unit will change with different coolant flow rates and outdoor air temperature. Despite the universal prevalence of heaters in indoor air conditioning systems and the simplified mathematical description, for the synthesis of an adequate control system, it is necessary to take into account clearly nonlinear, dynamically changing over time parameters of the heater. This circumstance should be mandatory in cases where the mathematical model is not of a conceptual theoretical nature, but requires practical testing on an object with a wide range of regulation of the system's heating capacity.

In the course of this work, the problems related to this work, possible ways to solve it were described. The scheme of experimental installation is developed and its detailed description and the principle of work is stated. The power circuits of the installation were composed, and individual links were described. Mathematical transformations were done with power circuits and complex resistances, frequency functions, amplitude – frequency and phase – frequency characteristics were obtained. The frequency characteristics of the circuit are constructed.

Based on the obtained graphs, it is possible to trace the relationship of two different characteristics. The graphs show that in $\Omega=1$ we get the peak of the hydraulic characteristic and the minimum of the thermal characteristic. This effect is due to the fact that at the moment there is a water hammer. And it turns out a surge in pressure and a decline in heat flow.

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