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COMPUTER DESIGN OF THERMOELECTRIC AUTOMOBILE STARTING PRE-HEATER OPERATED WITH DIESEL FUEL

The results of computer design of a 70 – 90 W thermoelectric automobile heater operated with diesel fuel for start heating of engine under low ambient temperatures are presented.

Key words: computer design, physical model, starting pre-heater, thermoelectric generator.

Introduction

The specific feature of diesel internal combustion engine is that compression stroke parameters (pressure and temperature) determine the reliability of fuel self-ignition. For a reliable start-up of diesel engine the temperature of the end of the compression stroke must exceed the temperature of fuel self-ignition. At engine start-up under low ambient temperatures the temperature of the end of the compression stroke is reduced for a number of reasons. Thus, reduction of engine crankshaft speed leads to a reduction of average piston speed. As a result, the time allotted for compression process is increased. Low temperature of engine cylinder walls is responsible for higher intensity of heat exchange between the air and cylinder walls, which leads to increased heat loss in the process of diesel fuel compression. Moreover, the temperature of the end of the compression stroke is reduced due to cold air coming to cylinders.

Under low ambient temperatures the reduction of compression temperature is affected by the non-uniform piston speed. In this case not only the time of heat exchange between “air charge” and cylinder walls is increased, but there is also maximum temperature difference between them, so heat loss due to heat exchange becomes greater. Due to reduced piston speed and insufficiently firm adherence of piston rings to cylinder walls, there is a loss of air flowing in the gaps between the piston and cylinder. So, in addition to reduced temperature of the end of the compression stroke, there is also an additional pressure reduction.

Low air temperature also has a negative effect on the quality of fuel spraying by engine nozzles, which also complicates start-up of internal combustion engine. It is primarily related to increase in diesel fuel viscosity, increase in its surface tension forces and, as a result, formation of waxes. Inferior quality of fuel spraying, low temperature and pressure values under compression increase the time of fuel ignition, complicating start-up of diesel engine. Sometimes a combination of these factors does not guarantee at all the conditions for diesel fuel self-ignition and engine start-up becomes impossible [1].

So far, the most efficient method for pre-heating of diesel engines and their reliable start-up is autonomous start heating. Such heaters are suitable practically for all types of transport means, so they are used in the cars and trucks, as well as in the buses, planes, yachts and motor boats. Moreover, the use of such equipment provides for a reduction of toxic discharge with automobile exhaust gases by a factor of 5, increase in engine service life by 50 – 60 thousand km and saving of 90 –150 l of fuel during only one winter season [2].

However, one of the main constraining factors for a wide practical application of such equipment is the need for electric energy to power components of starting pre-heaters, such as a fuel pump, a fan for air delivery to combustion chamber, a circulating pump for pumping of liquid heat carrier. For instance, in operation of liquid heater of thermal power 4 kW and electric power consumption 37 – 40 W, and together with a standard fan of automobile heating system – 60 W, a storage battery of capacity 60 A·h in as few as 4.5 hours loses 50 % of capacity. To prevent from a "deep" discharge of storage battery by strong frosts it is recommended not only to switch off completely the function of interior heating, but also to refuse from using additional equipment installed in the car (audio- and video complexes, GPS-navigators, signaling systems). As practice shows, drivers who drive a car for less than 30 min throughout the day (home-office-home) and operate the heater prior to each engine start -up for 20 – 30 min, will not avoid weekly recharging of storage battery [3].

Note that so far none of known models of starting pre-heaters have solved the problem of battery discharge. Most common methods of heating cold automobile engine without the use of storage battery energy include electric heating and heating by means of thermal accumulators. However, in this case a driver is constantly confined to external power supply.

As shown in [4], said problem is solved by means of thermoelectric generator. As well as start heating process becomes fully autonomous, without the use of storage battery electric energy, the excess energy of thermal generator can be used for recharging storage battery and power supply to other automobile equipment.

In [5], analysis of technical specifications of starting pre-heaters for various kinds of transport means is made and electrical parameters of thermal generators necessary for the autonomous operation of such heaters are determined. Thus, total electric power of thermal generator for starting pre-heaters of transport means with engine displacement to 4 l must be 70 – 90 W; for transport means with engine displacement 4 – 10 l and over 10 l – 130 – 150 W, 230 – 250 W, respectively. Besides, such electric powers of thermal generator will allow power supply to a standard fan heater of transport means and storage battery recharging.

Therefore, *the purpose* of the work is to design a 70 – 90 W thermoelectric automobile starting pre-heater operated with diesel fuel for start heating of engine under conditions of low ambient temperatures.

Design selection of thermoelectric starting pre-heater

Fig. 1. shows a schematic of automobile starting pre-heater with thermoelectric power supply. Structurally, such thermoelectric pre-heater is composed of the hot heat exchanger 1 accommodating in its internal space the source of heat 2 for flame combustion of liquid or gas fuel and air delivery fan 3 for fuel combustion. Fuel delivery to the source of heat is done by fuel pump 4. On the external surface of the hot heat exchanger there are thermoelectric modules 5 the heat from which is removed by liquid heat exchangers 6.

Liquid heat exchangers are combined into one hydraulic loop connected to engine cooling system with connecting pipes 7. Circulation of liquid heat carrier in the "heater-engine" loop is done by pump 8. Combustion products are removed to the environment by exhaust pipe 9. Start-up and operation control of all pre-heater devices (fan, fuel and circulating pumps) is done by electronic unit 10.

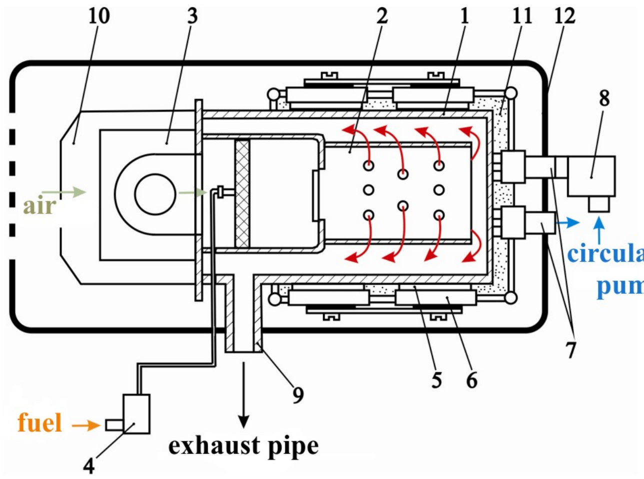
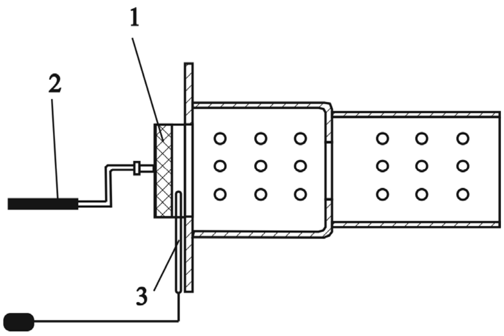


Fig. 1. Schematic of automobile starting pre-heater with thermoelectric power supply: 1 – hot heat exchanger; 2 – source of heat; 3 – fan; 4 – fuel pump; 5 – thermoelectric module; 6 – cold heat exchanger; 7 – inlet and outlet connecting pipes; 8 – circulating pump; 9 – exhaust pipe; 10 – electronic unit; 11 – thermal insulation; 12 – case [6].

Free space between the hot and cold heat exchangers is filled with thermal insulation 11. Automobile heater with a fan, electronic unit, heat exchangers and thermoelectric modules is placed in case 12.

The heater works as follows. Thermal energy from fuel combustion heats the hot heat exchanger, passes through thermoelectric converter and is removed by liquid heat carrier circulating in the heat exchanger of the heater and engine cooling system. Due to temperature difference between the hot and cold sides the thermal converter generates electric current. Thermal energy removed by heat carrier from the thermal converter is used for engine warm-up and heating of car interior.

As a source of heat, use was made of a diesel burner of maximum thermal power 4 kW of liquid starting pre-heater “Thermo Top Evo 4” (Webasto). The schematic and appearance of the burner is presented in Fig. 2.



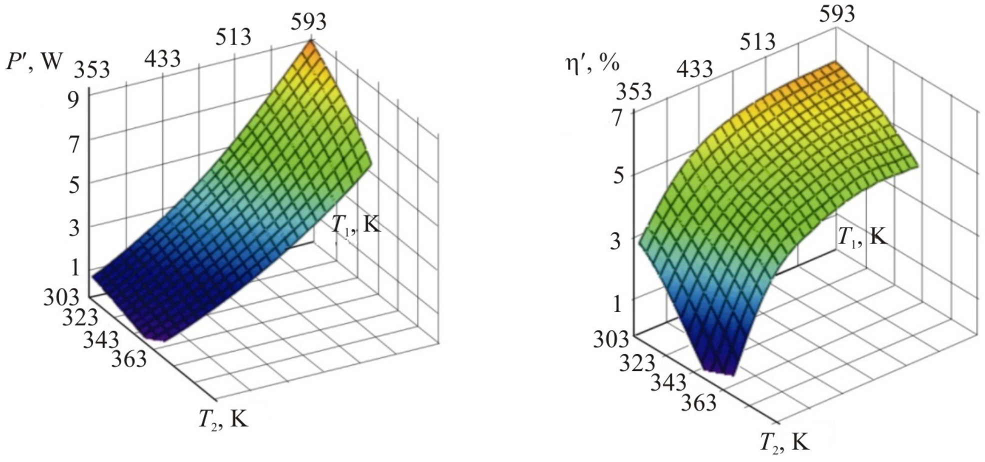
a)

b)

Fig. 2. Schematic (a) and appearance (b) of diesel burner Ersatzbrenner D TT-C MB (Webasto): 1 – evaporator; 2 – fuel pipe; 3 – starting electrode [7].

The most rational shape of the hot heat exchanger in terms of efficient heat exchange with the source of heat is a cylinder pipe with one end closed and the other end accommodating a diesel burner. The external surface of the heat exchanger has the form of planes where thermoelectric modules are arranged.

The thermoelectric converter is composed of thermoelectric modules based on bismuth telluride of the type “ALTEC-1061” which on arrival of the necessary amount of heat to the hot side and on achievement of optimal operating temperatures assure generation of the assigned electric power. Fig. 3 shows a three-dimensional graphical image of the dependence of electric power $P^{/}$ and efficiency $\eta^{/}$ on the hot side T_h and cold side T_c temperature of thermoelectric module "ALTEC-1061".



a) *b)*

Fig. 3. Dependence of electric power $P^{/}$ (a) and efficiency $\eta^{/}$ (b) on the hot side Th and cold side Tc temperature of thermoelectric module "ALTEC-1061" [8].

From the analysis of data presented in Fig. 3 it follows that to provide the output electric power of the heater 90 – 70 W, it is necessary to have 12 modules "ALTEC-1061". From these considerations, the most efficient design of the hot heat exchanger is a regular hexahedron, with 2 thermoelectric modules arranged on each side thereof. In so doing, the hot side temperature of modules must be 330 – 280 °C, the cold side – 30 – 70 °C.

For diesel fuel delivery use was made of a pulse fuel pump (Fig. 4) of liquid starting pre-heater "Thermo Top E" (Webasto). The experimentally found dependence of fuel combustion g_n and thermal power Q of diesel burner on pulse period t of fuel pump is given in Fig. 5. From the data given in Fig. 5 it is seen that the heat source maximum power 4 kW is achieved with a pulse period of fuel pump 0.5 s. In so doing, fuel consumption is ~ 350 g/h.

Fig. 4. Appearance of fuel pump BTL.DP30.02.12V DAEMPFLER E-TEIL (Webasto) [7].

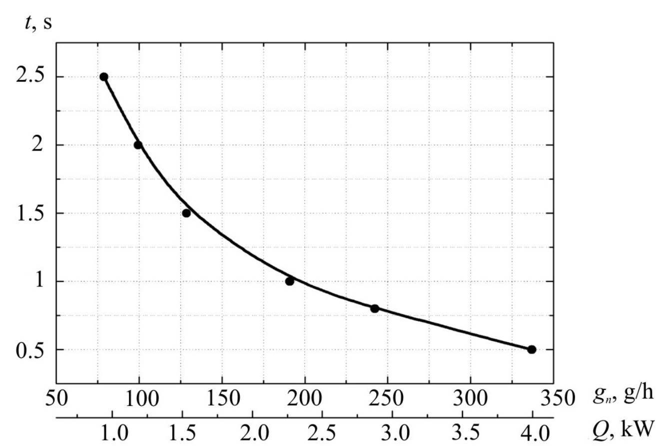


Fig. 5. Dependence of fuel consumption g_n and thermal power Q on pulse period t of fuel pump.

Taking into account that the efficiency of thermoelectric generators using single-stage modules based on bismuth telluride is 3 – 3.5 % [9], to assure the output electric power of the heater 90 – 70 W, it is necessary to spend ~ 2.3 – 2.5 kW of heat which corresponds to fuel consumption 180 – 220 g/h.

As a circulating pump, the liquid pump (Fig. 6) of staring pre-heater "Thermo Top C" (Webasto) was used. This type of the pump is specially designed for pumping of heat carrier in transport means with engine displacement 2.5 – 4 l. Fig. 7 shows the experimentally determined dependence of liquid fuel consumption q_T on the supply voltage U of the pump.

Fig. 6. Appearance of liquid pump 12V U4847 TT C/E (Webasto) [7].

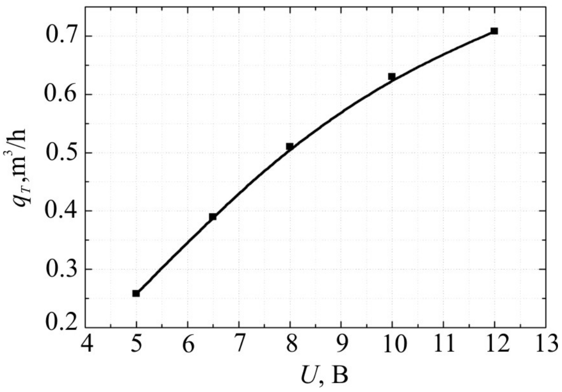


Fig. 7. Dependence of liquid heat carrier consumption q_T on the supply voltage U of the pump.

Fuel pump starts pumping heat carrier at supply voltage 5 V with consumption 0.25 m^3/h. At a voltage of 12 V heat carrier consumption is maximum – 0.7 m^3/h.

Physical model of thermoelectric starting pre-heater and its description

A physical model of thermoelectric automobile starting pre-heater is presented in Fig. 8.

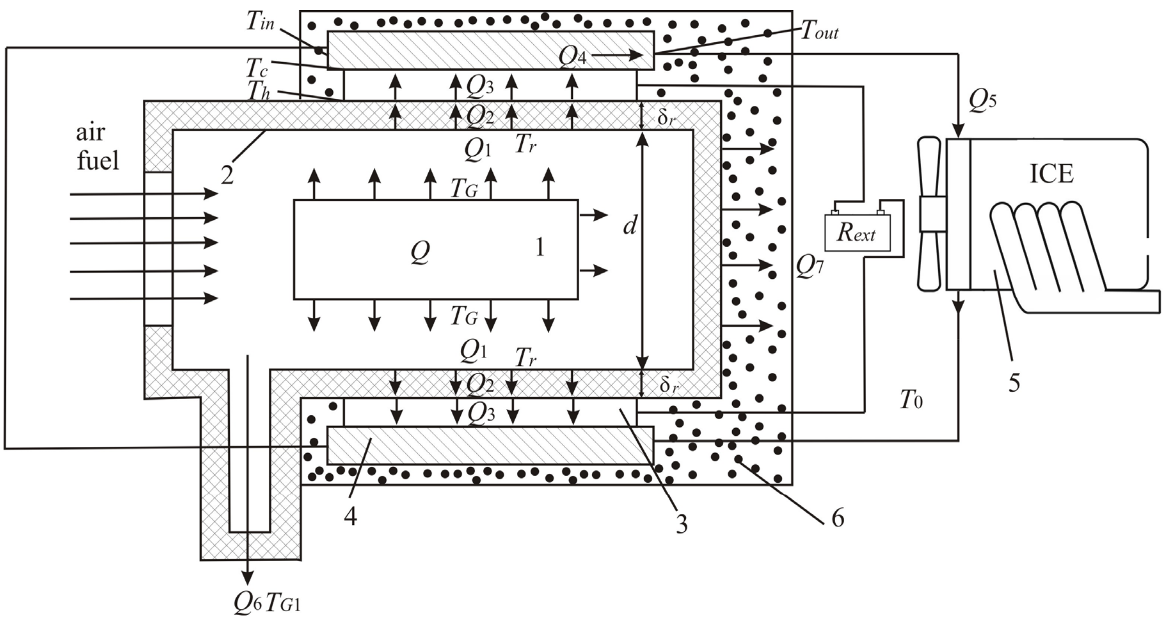


Fig. 8. Physical model of thermoelectric automobile heater:
1 – heat source; 2 – hot heat exchanger; 3 – thermopile;
4 – cold heat exchanger; 5 – automobile engine; 6 – thermal insulation.

The source of heat 1 in thermoelectric automobile heater is chemical reaction of diesel fuel flame combustion resulting in the release of thermal power Q.

$$Q = g_n G_n, \tag{1}$$

where G_n is calorific power of diesel fuel; g_n is diesel fuel consumption.

Heat Q_1 coming from the heat source to the hot heat exchanger 2 is transferred due to convection and radiation:

$$Q_1 = \alpha \cdot (T_G - T_r) \cdot S_R + \varepsilon \cdot \sigma_0 \cdot \left(\varepsilon_G \cdot \left(\frac{T_G}{100} \right)^4 - A_r \cdot \left(\frac{T_r}{100} \right)^4 \right) \cdot S_R, \tag{2}$$

where α is convective coefficient of heat transfer from the hot gas to heat-absorbing surface of the hot heat exchanger;

T_G is average gas temperature in the hot heat exchanger;

T_r is average temperature of the heat-absorbing surface of the hot heat exchanger;

S_R is the area of the heat-absorbing surface of the hot heat exchanger;

$\varepsilon = (\varepsilon_r+1)/2$ is effective emissivity factor of the "hot gas-heat-absorbing surface" system of the hot heat exchanger;

ε_r is emissivity factor of the heat-absorbing surface of the hot heat exchanger;

σ_0 is Stephan-Boltzmann constant;

ε_G is gas emissivity factor;

A_r is absorbing ability of the heat-absorbing surface of the hot heat exchanger.

Heat Q_2 is transferred due to thermal conductivity from the heat-absorbing surface of the hot exchanger to the hot side of thermopile 3:

$$Q_2 = \frac{S_r \cdot \lambda_r}{\delta_r} \cdot (T_r - T_h), \tag{3}$$

S_r is the area of the heat-releasing surface of the hot heat exchanger;

λ_r is thermal conductivity of heat exchanger material;

δ_r is thickness of heat exchanger base;

T_h is temperature of the thermopile hot side.

Useful heat Q_3 coming to thermoelectric modules was calculated from the power P' of one module with determined hot side T_h and cold side T_c temperatures of thermopile, its efficiency η' and the number of modules n:

$$Q_3(T_h, T_x) = n \cdot \frac{P'(T_h, T_c)}{\eta'(T_h, T_c)}. \tag{4}$$

Heat Q_4 is removed from the cold side of thermopile by heat carrier flow circulating in the cold liquid heat exchanger 4:

$$Q_4 = g_T \cdot c_{pT} \cdot (T_{in} - T_{out}), \tag{5}$$

where g_T is heat carrier consumption c_{pT} is heat carrier heat capacity T_{in}, T_{out} are heat carrier temperatures at inlet to and outlet of thermopile cooling system, respectively.

As long as cold liquid heat exchangers are combined into one hydraulic loop with engine cooling system 5, the heat removed by heat carrier from the modules is spent on start heating of engine:

$$Q_5 = c_{eng} \cdot m_{eng} \cdot (T_{out} - T_0), \tag{6}$$

where c_{eng}, m_{eng} are heat capacity and mass of automobile engine, respectively; T_0 is ambient temperature.

Basic heat losses:

1) Q_6 – with reaction products (water H_2O, carbon dioxide CO_2 and nitrogen N_2):

$$Q_6 = C_c \cdot m_c \cdot \left(T_{G1} - T_0\right), \tag{7}$$

where C_c is average heat capacity of reaction products, m_c is mass of reaction products, T_{G1} is temperature of reaction products.

2) Q_7 – on thermal insulation :

$$Q_7 = \frac{\lambda S_{ph}}{L}(T_B - T_0), \tag{8}$$

where λ is thermal conductivity of insulating material; S_{ph} is the area of the hot heat exchanger surface which is not occupied by thermopile; L is thickness of thermal insulation layer.

Thus, heat balance equation for the selected model of thermoelectric automobile heater can be written as:

$$\begin{cases} Q = Q_1 + Q_6, \\ Q_1 = Q_2 + Q_7, \\ Q_2 = P + Q_4. \end{cases} \tag{9}$$

Solving the system of heat balance equations (9) enables one to determine the basic energy and design parameters of thermoelectric automobile heater.

Computer design of thermoelectric automobile heater was conducted in two stages aimed at determining:

– gas temperature T_G in the hot heat exchanger and the effective area of heat-absorbing surface of the hot heat exchanger S_R to assure on the hot side of the thermopile the temperature T_h = 330 – 280 °C.

– thermal power Q_5 removed by heat carrier to estimate the warm-up rate of automobile engine.

Calculation of hot gas temperature

The process of heating gases in the hot heat exchanger that formed due to diesel fuel combustion is described by equation:

$$Q = C_p \cdot (T_G - T_o), \tag{10}$$

where C_p is total heat capacity of gases that formed due to diesel fuel combustion:

$$C_p = C_p(CO_2) + C_p(H_2O) + C_p(N_2) + C_p(air), \tag{11}$$

where $C_p(CO_2)$, $C_p(H_2O)$, $C_p(N_2)$, $C_p(air)$ is heat capacity of carbon dioxide, water, nitrogen and air.

Gas heat capacity is given by:

$$C_p = (\frac{i}{2} + 1) \cdot R \cdot \frac{M}{\mu}, \tag{12}$$

where i is the number of degrees of freedom of gas, R is the Mendeleyev-Klapeyron constant, M is the weight of gas, μ is molar weight of gas.

The weights of CO_2 and H_2O are defined by the ratio:

$$M(CO_2) = \frac{0.87 \cdot g_n \cdot \mu(CO_2)}{\mu(C)}, \tag{13}$$

$$M(H_2O) = \frac{0.13 \cdot g_n \cdot \mu(H_2O)}{\mu(H_2)}, \tag{14}$$

where coefficients 0.87 and 0.13 determine the content of C and H in diesel fuel; μ (CO_2), μ (C), μ (H_2O) and μ (H_2) are molar weights of carbon dioxide, carbon, water and hydrogen, respectively.

The weight of nitrogen is calculated through the weight of oxygen necessary for diesel fuel combustion:

$$M(N_2) = 4 \cdot M(O_2), \tag{15}$$

where

$$M(O_2) = \frac{0.87 \cdot g_n \cdot \mu(O_2)}{\mu(C)} + \frac{0.13 \cdot g_n \cdot \mu(O_2)}{2 \cdot \mu(H_2)}, \tag{16}$$

where $M\ (O_2)$ is the weight of oxygen necessary to form CO_2 and H_2O.

The weight of air that formed as a result of fuel combustion:

$$M(air) = 5 \cdot (\kappa - 1) \cdot M(O_2), \tag{17}$$

where $\kappa > 1$ is the excess coefficient that determines the amount of excess air that must be spent on complete fuel combustion. Ideally, $\kappa = 1$, however, in real conditions, using only the stoichiometric amount of oxygen, complete combustion cannot be achieved.

We substitute (13) – (17) into (10), equate (10) and (1) and get a ratio for the determination of κ:

$$\kappa = \frac{\dfrac{G_n}{T_G - T_0} - 18.53}{16.85} + 1. \tag{18}$$

Coefficients 18.53 and 16.85 determine the content of carbon dioxide, water, nitrogen and air that formed as a result of complete combustion of diesel fuel and were derived with regard to specific values of i, μ of gases and R.

On the other hand, the velocity of air delivery v to heat exchanger is determined from the ratios:

$$v = \frac{g_{air}}{\rho_{T_o} \cdot S_R}, \tag{19}$$

$$g_{air} = 5 \cdot \kappa \cdot M(O_2), \tag{20}$$

where g_{air} is air consumption, ρ_{T_o} is air density at given ambient temperature.

Substituting (16) and (18) into (19) we obtain a dependence of velocity v on the temperature of hot gases T_G in the heat exchanger:

$$v = 5 \cdot \left[\frac{\frac{G_n}{T_G - T_0} - 18.53}{16.85} + 1 \right] \cdot \frac{3.36 \cdot g_n}{\rho_{T0} \cdot \pi \cdot d^2 / 4}, \tag{21}$$

where 3.36 is coefficient that determines the amount of oxygen required for complete combustion of diesel fuel.

The Mathcad application software package was used to determine the inverse dependence of hot gas temperature T_G on air velocity v (Fig. 9) at $G_n = 42.7$ MJ/kg, $T_0 = 0$ °C, $g_n = 220$ g/h, $\rho_{T_o} = 1.29$ kg/m^3 and combustion chamber diameter $d = 70$ mm.

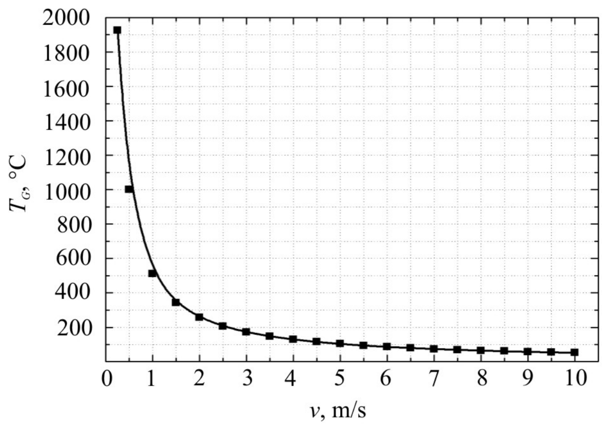


Fig. 9. Dependence of hot gas temperature T_G in the heat exchanger on air velocity v.

From the analysis of data given in Fig. 9 it follows that for further calculations of the hot exchanger design it is not reasonable to use $T_G < 300$ and $T_G > 500$ °C: in the former case it is impossible to achieve the necessary hot side temperatures of module, in the latter case the rise in temperature leads to increase in the heat exchanger dimensions due to possible overheating of thermopile.

Calculation of the hot heat exchanger

To determine a dependence of module hot side temperature T_h on the area of the heat-absorbing surface, we used hot gas temperature T_G = 500 – 300 °C which corresponds to air velocities 1 – 2 m/s.

Design was made with the use of “Comsol Multiphysics” software [10] by the numerical finite element method.

In the process of computer design the following values were used as the input data:

– hot gas temperature T_G = (300; 400; 500) °C;

– gas velocity in the heat exchanger v = (1; 1.5; 2) m/s;

– heat-absorbing surface area S_R = (0.025; 0.045; 0.07; 0,09) m^2;

– thermal conductivity of the hot heat exchanger material λ_r = 140 W/m·K;

– thermal conductivity of thermoelectric material λ_m = 1.4 W/m·K.

In so doing, it was considered that heat sinks are at the outlet of combustion products from the heat exchanger and at places of location of modules, and thermal adiabatic isolation conditions are imposed at the rest of the boundaries.

Fig. 10 shows computer designed dependences of the module hot side temperature T_h on the area of heat-absorbing surface S_R at hot gas temperature T_G within 500 – 300 °C.

From the data presented in Fig. 10 it is seen that in the case of undeveloped area of the heat-absorbing surface ($S_R \sim 0.025$ m^2) at $T_G = 400$ °C, T_h approaches the optimal value and makes 290 °C. The increase in heat exchange area leads to the rise in temperature T_h and at $S_R = 0.09$ m^2 the necessary temperature level of 330 °C is achieved on the hot side of module.

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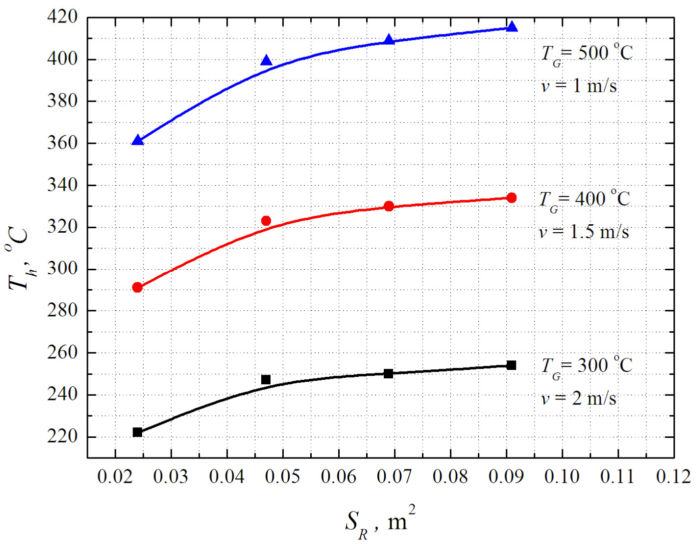

Fig. 10. Dependence of the module hot side temperature T_h on the area of the heat-absorbing surface S_R of heat exchanger.

Fig. 11 shows temperature distribution in the “hot heat exchanger-thermoelectric modules” system, Fig. 12 – finite element method mesh.

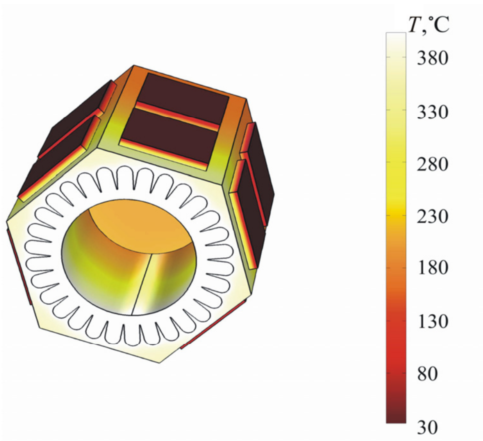

Fig. 11. Temperature distribution in “hot heat exchanger-thermoelectric modules” system T_G = 400 °C, S_R = 0.09 m.^{2}

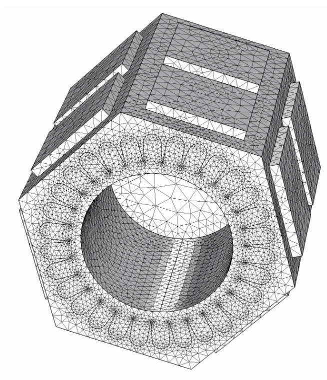

Fig. 12. Finite element method mesh.

Note that computer model of the heater is a somewhat simplified version of a physical model. To avoid awkward calculations aimed at determining heat flows due to radiation and convection and heat losses with combustion products, we consider that heat transfer process occurs due to passing through the heat exchanger of gas flow with known inlet temperature T_G So, as is evident from Fig. 11, as the gases flow to heat sink, T_G is reduced, which results in the reduction of temperature T_h from 330 to 250 °C. However, as long as in a real design the source of heat is directly in the heat exchanger, it can be considered that the hot side temperature of modules is equal.

Similar calculations to determine T_G at constant $S_R = 0.09$ m^2 and $g_n = 180$ g/h show that to assure the hot side thermopile temperature 280 °C, the temperature of hot gases in the heat exchanger must be 350 °C. In this case air delivery velocity must be 1.2 m/s.

Calculation of the cold heat exchanger

For quick heating of automobile engine it is necessary to assure maximum transfer of thermal power from the modules to the cold heat carrier. On the other hand, to assure the efficient operation of modules, it is important to create such conditions whereby the difference in heat carrier temperature at the inlet to and outlet of heat exchangers would be minimal. With this aim, at the present simulation stage the effective channel area of cold heatexchangers and optimal consumption of li -quid heat carrier were determined. The coldheat exchanger simulation was done by finite element method with the use of "Comsol Multiphysics" application software package.

In the process of computer design the following values were used as the input data:

– cold heat carrier temperature at inlet to cold heat exchanger $T_{in} = 30$ °C;
– thermal power removed from the modules $Q_4 = 1.4$ kW;
– heat carrier consumption $g_T = (0.25; 0.5; 0.7)$ m^3/h;
– channel area of the cold heat exchanger $S_c = (10 – 70)$ cm^2;
– thermal conductivity of cold heat exchanger material $\lambda_{r1} = 105$ W/m·K;
– heat carrier heat capacity $c_{pT} = 3151$ J/kg·K;
– heat carrier thermal conductivity $\lambda_T = 0.34$ W/m·K.

In the design, heat sink was assigned at points of heat carrier outlet from the heat exchanger, and adiabatic thermal insulation conditions were imposed at the rest of the boundaries.

Fig. 13 shows a dependence of thermal power Q_5 removed from thermoelectric modules to aut omobile engine on the total area of channels S_c of the cold heat exchangers with different heat carrier consumption.

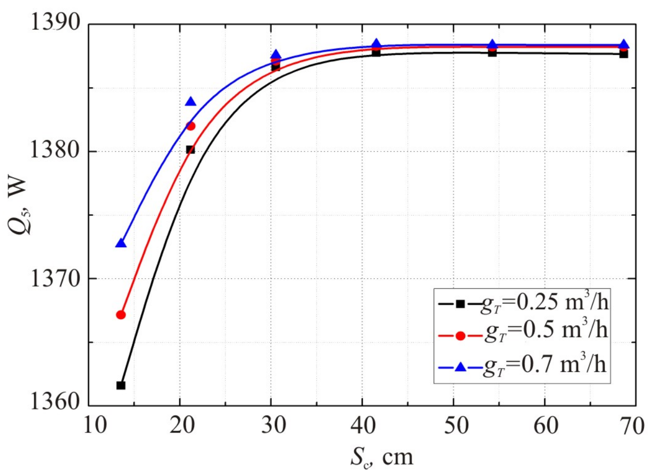

Fig. 13. Dependence of thermal power Q_5 on the channel area S_c of the cold heat exchangers.

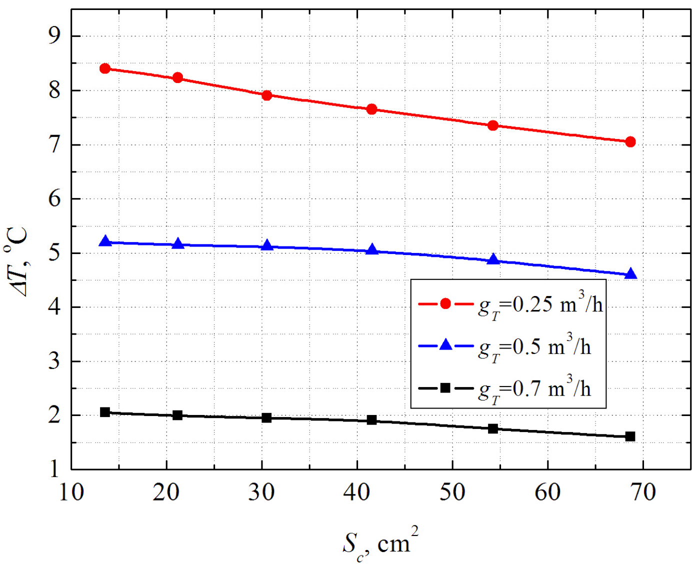

Fig. 14. Dependence of heat carrier temperature difference ΔT between the inlet to and outlet of the cold heat exchangers on the channel area S_c.

From the analysis of data represented in Fig. 13 it follows that for complete transfer of thermal power from the modules to engine the area of channels in the cold heat exchangers must be at least 40 cm^2. With such channel area Q_5 is practically independent of heat carrier consumption, which in turn allows reducing the losses of the output electric power of the heater on power supply to circulation pump. However, as follows from Fig. 14, heat carrier consumption must be 0.7 m^3/h. In this case heat carrier temperature difference between the inlet to and outlet from the heat exchangers is minimal and equal to ~ 2 °C.

Fig. 15 and Fig. 16 show temperature distribution and finite element method mesh for the cold heat exchanger:

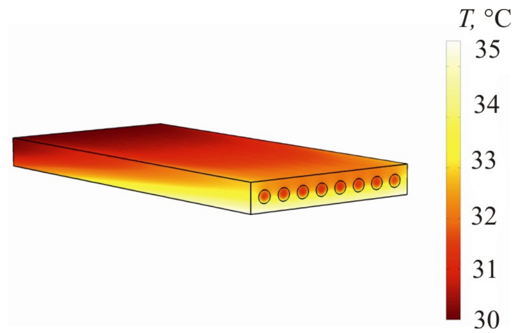

Fig. 15. Temperature distribution in the cold heat exchanger. S_c = 40 cm^2. g_T = 0.7 m^3/h.

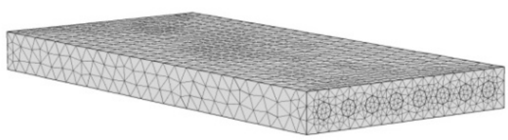

Fig. 16. Finite element method mesh.

According to (6), the amount of heat that must be spent on heating of engine with heat capacity 0.462 kJ/(kg K) (engine material – steel) and mass 200 kg from 0 °C to 30 °C will make ~ 2.8 MJ. Taking into account that total thermal power removed from the thermopile by heat carrier with temperature 30 – 70 °C is ~ 1.39 kW (Fig. 17), start heating of engine requires ~ 40 min.

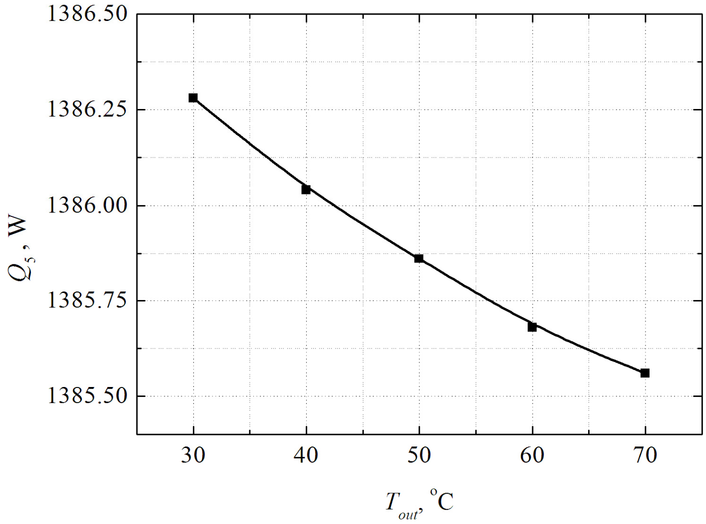

Fig. 17. Dependence of thermal power Q_5, removed by heat carrier from heat carrier temperature T_{out} at the outlet of the cold heat exchangers.

Under real conditions the time of engine heating will be longer which is related to heat losses due to engine heat exchange with the environment, losses of heat transfer along cooling circuit, as well as due to the availability of hydraulic resistances in "heater-engine" system.

Conclusions

1. It is established that the output electric power of the heater 90 – 70 W is attained through use of 12 modules "ALTEC-1061" with the hot and cold side temperatures 330 – 280 °C and 30 – 70 °C, respectively. With the heater efficiency within 3 – 3.5 %, to attain such electric power, it is necessary to spend ~ 2.3 – 2.5 kW of heat, which corresponds to fuel consumption 180 – 220 g/hour.
2. It is shown that the necessary temperature level on the hot side of thermopile is attained at gas temperature in the hot heat exchanger in the range of 400 – 350 °C, the velocity of air delivery to combustion chamber 1.5 – 1.2 m/s and the area of the heat-absorbing surface of the hot heat exchanger 0.09 m^2.
3. It is established that total thermal power transferred from thermopile to heat carrier is ~ 1.39 kW. With the heat carrier consumption 0.7 m^3/h and channel area of the cold heat exchangers 40 cm^2 both complete transfer of thermal power from heat carrier to automobile engine and minimal temperature difference of heat carrier between the inlet to and outlet from the heat exchangers is assured. Under these conditions, at heat carrier temperature 30 – 70 °C the start heating of engine from 0 °C to 30 °C requires ~ 40 min.

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