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COMPUTER DESIGN OF THERMOELECTRIC STARTING PRE-HEATER ON GAS FUEL

This paper presents the results of computer design of thermoelectric heater on gas fuel with electric power output 230 – 250 W for start heating of vehicle engines under low ambient temperatures.

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

Introduction

According to International Energy Agency, the number of automobiles in the world by 2015 had exceeded 1 milliard. 22.5 million of them use liquefied propane-butane as fuel. In Ukraine ~ 10 % of the automobile park, i.e. 800 thousand vehicles, are equipped with compressed natural gas equipment (GBO) of 3 and 4 generations. It is expected that within 2 – 3 years not less than 20 % of the park will be models that run on gas fuel [1].

Keen demand for such automobiles is caused by a number of advantages in the use of propane-butane as compared to petrol:

– higher octane number (112 for propane-butane, 80 – 98 for petrol) assuring reliable engine operation for all modes almost without detonation;

– reduced load on engine pistons;

– increased engine turnaround time by a factor of 1.5;

– increased engine oil service life by a factor of 1.5 – 2;

– reduced noise level with the engine running by 3 – 8 dB (at least twice);

– increased service life of spark plugs by 40 %;

– reduced level of toxic exhaust gases: CO – by a factor of 2 – 3, NO – by a factor of 1.2, CH – by a factor of 1.3 – 1.9.

One of the main disadvantages of using liquefied propane-butane as automobile fuel is a negative influence of low temperatures which results in formation and freezing of condensate in the reducer and pipes of GBO. As a result, the engine stalls, and to start it again is practically impossible. Moreover, icing of reducer decreases considerably the service life of its membranes.

To prevent from premature failure of reducer membranes and valves and in general to assure a reliable automobile start at low temperatures, the engine must be pre-heated.

Today the most efficient pre-heating method is autonomous start heating [2].

However, the main factor restricting the use of gas starting pre-heaters for automobiles equipped with GBO-3 and GBO-4 is operating instability and low reliability of gas system. In this case it is necessary to start and heat the engine on liquid fuel with a gradual transition to gas fuel, which, although saving fuel, neutralizes all advantages of start heating.

At the present time, new vistas in the use of gas starting pre-heaters are related to the use of automobiles of compressed natural gas equipment of "new generation" – GBO-5 and GBO-6. The distinctive feature of these developments lies in the fact that gas delivery to inlet manifold is done in the liquid, rather than the gas state, namely through special nozzles. So, these systems are of a completely new design. Thus, a reducer is substituted by pressure controller and gas fuel pump which pumps liquid gas directly to nozzles. As long as this design eliminates the need for evaporation of gas in the reducer, this enables one to start the engine at any low ambient temperatures.

With the advent of GBO-5 and GBO-6, models appeared in the market (Webasto LGW 300, Trumatic E2400-E4400) that run on propane-butane and provide start heating of gas-fuelled vehicles.

To avoid automobile battery discharge during start heating, it is reasonable to use thermoelectric generator as a power supply source [3 – 5].

In [6] it is shown that total electric power of thermal generator for starting pre-heaters of vehicles with engine displacement up to 4 l must be 70 – 90 W; for vehicles with engine displacement 4 – 10 l and more than 10 l – 130 – 150 W, 230 – 250 W, respectively. Such electric powers of thermal generators assure autonomous power supply to heater components and will allow additional battery charging and power supply to other automobile equipment.

In [7, 8], the construction, operating principle and results of computer design of thermoelectric automobile starting pre-heater on liquid fuel of electric power 70 – 90 W and 130 – 150 W are presented.

The purpose of this work is to design thermoelectric automobile starting pre-heater on liquefied gas fuel of electric power 230 – 250 W for start heating of engine under low ambient temperatures.

Design selection of thermoelectric starting pre-heater

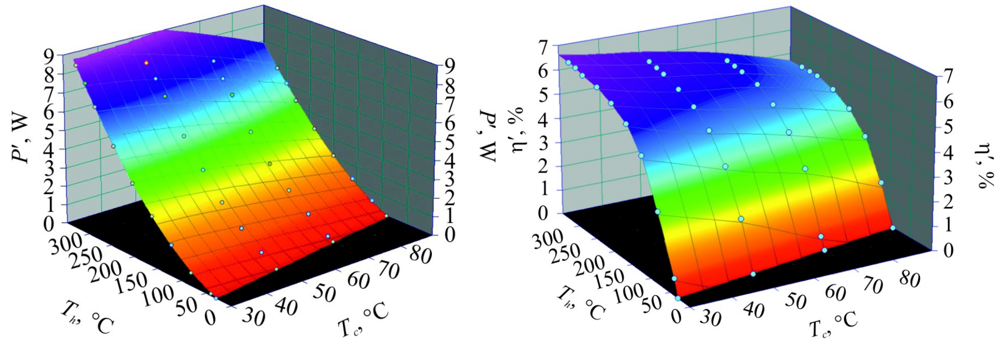
As a source of heat, use was made of a gas burner (Fig. 1) of liquid starting pre-heater "Webasto LGW 300". With regard to burner design, the most rational shape of the hot heat exchanger in terms of efficient heat exchange with the source of heat is a cylinder pipe accommodating a burner in its internal space, and its external surface having the form of planes where thermoelectric modules are arranged.

The thermoelectric converter is composed of thermoelectric generator modules "ALTEC-1061" based on bismuth telluride 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. 1. Appearance of gas burner of liquid starting pre-heater "LGW 300" (Webasto) [9].

Fig. 2. Appearance of circulating pump of liquid starting pre-heater "LGW 300" (Webasto) [9].

Fig. 3 shows a three-dimensional graphical image of the dependence of electric power P' and efficiency η' 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 η' (b)on the hot side T_h and cold side T_c temperature of thermoelectric module "ALTEC-1061" [10].

From the analysis of the data presented in Fig. 3 it follows that to provide the electric power output of the heater 230 – 250 W, it is necessary to have 32 modules "ALTEC-1061". From these considerations, the most efficient design of the hot heat exchanger is a regular octahedron, with 4 thermoelectric modules arranged on each side thereof. In so doing, the hot side temperature of the modules must be 280 – 330 °C, the cold side – 30 – 70 °C.

Taking into account that the efficiency of thermoelectric generators using single-stage modules based on bismuth telluride is 3 – 3.5 % [11], to assure the electric power output of the heater 230 – 250 W, it is necessary to spend ~ 6.6 – 7.2 kW of heat which corresponds to propane-butane consumption 510 – 550 g/h. It should also be stated that fuel delivery to the burner is done by a pump of GBO system, so in the design of gas heater, unlike the petrol and diesel analogs, there is no individual fuel pump, and consumption is controlled by changing nozzle holes.

As a circulating pump, the liquid pump (Fig. 2) of starting pre-heater "LGW 300" (Webasto) was used. This type of the pump is specially designed for pumping of heat carrier in transport means with engine displacement more than 10 l. Nominal supply voltage of the pump is 24 V, maximum heat carrier consumption is 5.2 m^3/h.

Physical model of thermoelectric starting pre-heater and its description

In [7] it is shown that for the calculation of the basic energy and design parameters of the heater it is convenient to use a physical model (Fig. 4) where the process of heat transfer from the source of heat to thermopile takes place as a result of gas flow passing through the hot heat exchanger at velocity v and temperature T_G.

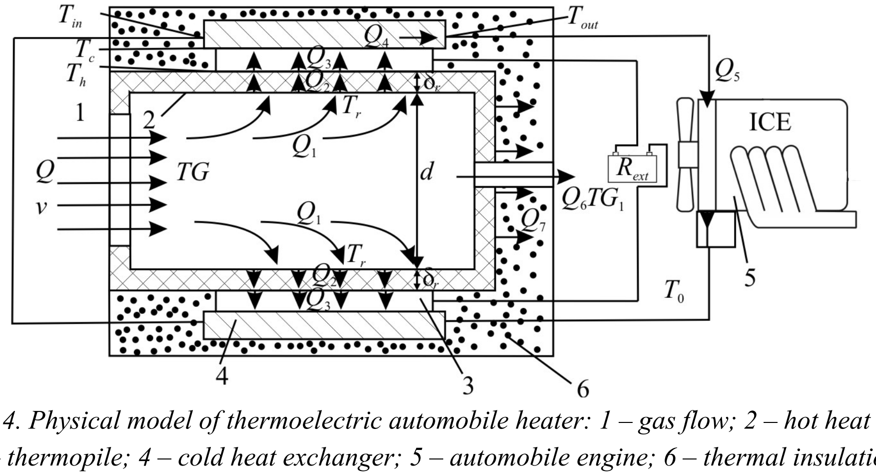


Fig. 4. Physical model of thermoelectric automobile heater: 1 – gas flow; 2 – hot heat exchanger; 3 – thermopile; 4 – cold heat exchanger; 5 – automobile engine; 6 – thermal insulation.

In so doing, the amount of heat Q released due to petrol combustion is used for heating of gases that formed as a result of complete fuel combustion:

$$g_n G_n = C_p (T_G - T_o)\,, \tag{1}$$

where g_n and G_n are consumption and calorific power of petrol fuel, C_p is total heat capacity of gases (CO_2, H_2O, N_2 and air) that formed as a result of fuel combustion, T_0 is ambient temperature.

Heat Q_1 coming from the heated gases to the hot heat exchanger is transferred due to convection

$$Q_1 = \alpha(T_G - T_r)S_R\,, \tag{2}$$

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

T_r is the 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.

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

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

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

λ_r is the thermal conductivity of heat exchanger material;

δ_r is the thickness of heat exchanger base;

T_h is the hot side temperature of thermopile.

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:

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

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

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

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 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:

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

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

Basic heat losses:

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

$$Q_6 = C_c m_c(T_{G_1} - T_0), \tag{7}$$

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

2) Q_7 – on thermal insulation:

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

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

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

$$\begin{cases} Q = Q_1 + Q_6, \\ Q_1 = Q_2 + Q_7, \\ Q_2 = Q_3, \\ Q_3 - P = Q_4, \end{cases} \tag{9}$$

where P is output electric power of the heater.

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

Computer design aimed at determining:

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

– thermal power Q_5 removed by heat carrier to estimate pre-heating rate of automobile engine.

Calculation results

Calculation procedure of gas temperature T_G is described in detail in [7].

As a result of appropriate calculations, ratio (10) was obtained to determine air delivery velocity v to the hot heat exchanger as a function of temperature T_G:

$$v = 5\left[\frac{\dfrac{G_n}{T_G - T_0} - 20.29}{18.15} + 1\right] \cdot \frac{3.62 g_n}{\rho_{T0} S_R}, \tag{10}$$

where 20.29 and 18.15 are coefficients that determine the content of carbon dioxide, water, nitrogen and air that formed as a result of complete combustion of propane-butane, 3.62 is coefficient that determines the amount of oxygen necessary for complete combustion of propane-butane, ρ_{T_0} is the density of air at given ambient temperature.

The Mathcad application software package was used to determine the inverse dependence of hot gas temperature T_G on air velocity v (Fig. 5) at G_n = 46.8 MJ/kg, g_n = 550 g/h, T_0 = 0 °C, $\rho_{T_0} = 1.29\ \text{kg/m}^3$ and combustion chamber diameter d = 80 mm.

From the analysis of data in Fig. 5 it follows that for further calculations of the hot exchanger design it is not reasonable to use T_G < 300 °C 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.

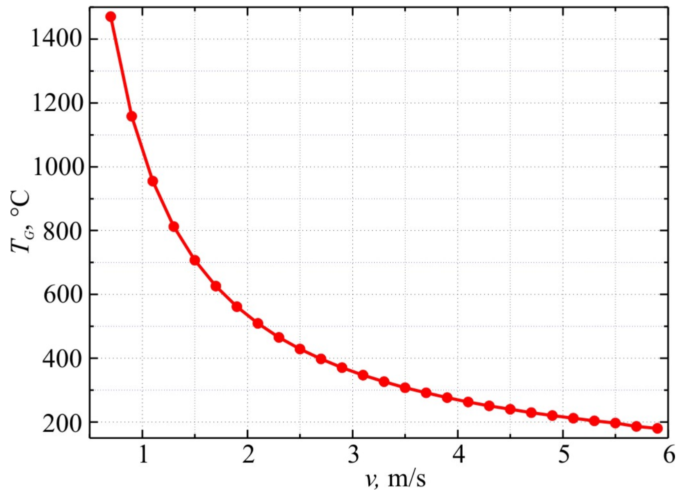


Fig. 5. Dependences of gas temperature T_G on air delivery velocity to the hot heat exchanger.

So, to determine the dependence of the module hot side temperature T_h on the area of the heat-absorbing surface, we used hot gas temperature $T_G = 300 – 500$ °C which corresponds to air velocities 2.1. – 3.5 m/s.

Simulation was done by finite element method with the use of "Comsol Multiphysics" application software package [12].

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

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

– the velocity of gas in the heat exchanger $v = (3.5; 2.6; 2.1)$ m/s;

– the area of the heat-absorbing surface $S_R = (0.05; 0.085; 0.115; 0.185)$ m^2;

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

– the thermal resistance of thermoelectric module $\kappa_m = 0.7$ m·K/W.

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

Fig. 6 shows computer designed dependences of the module hot side temperature T_h on the area of the heat-absorbing surface S_R at hot gas temperature T_G within 300 – 500 °C.

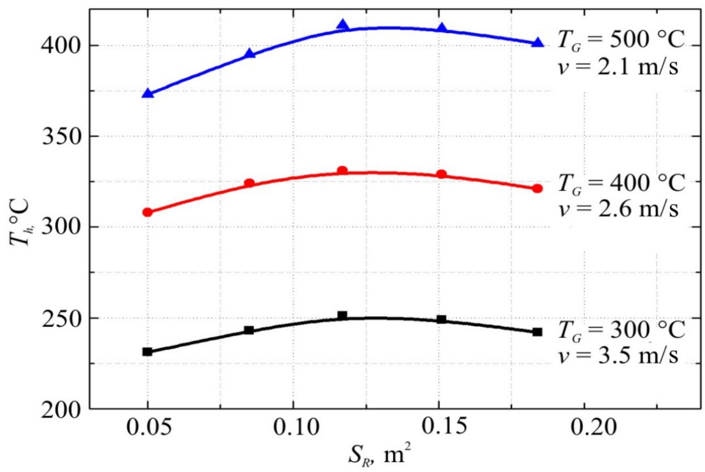


Fig. 6. Dependences of the module hot side temperature T_h on the area of the heat-absorbing surface S_R of heat exchanger.

From the data presented in Fig. 6 it is seen that the hot side temperature of module 330 °C is achieved at gas temperature T_G = 400 °C and the area of the heat-absorbing surface of the hot heat exchanger S_R = 0.12 m^2. Further increase in the number of channels on the internal surface of heat exchanger leads to creation of additional air resistances owing to which there is a drop in gas motion velocity and, hence, in the hot side temperature of thermopile.

Fig. 7 shows temperature distribution in the "hot heat exchanger-thermoelectric modules", Fig. 8 – the finite element method mesh.

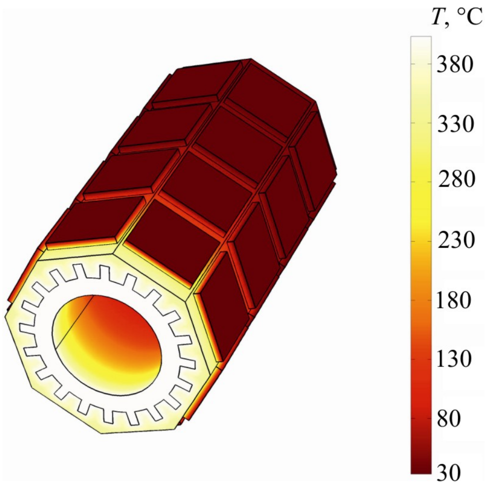


Fig. 7. Temperature distribution in the "hot heat exchanger – thermoelectric modules" system. T_G = 400 °C. S_R = 0.12 m^2.

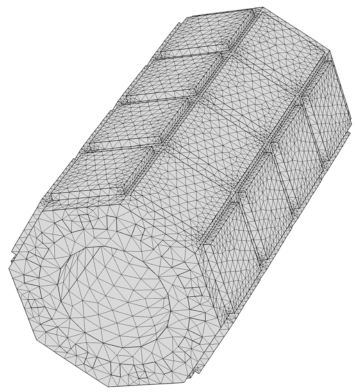


Fig. 8. Finite element method mesh.

As it follows from the analysis of temperature distribution, as the gases flow to heat sink, T_G is reduced, which results in the reduction of temperature T_h by ~ 200 °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 identical.

Similar calculations to determine T_G at S_R = 0.12 m^2 and g_n = 510 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 2.8 m/s.

For quick heating of automobile engine and the efficient operation of thermoelectric converter, it is necessary, on the one hand, to assure maximum transfer of thermal power from the modules to the cold heat carrier, on the other hand, 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, the effective channel area of the cold heat exchangers and optimal consumption of liquid heat carrier were determined. The cold heat 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:

– the temperature of cold heat carrier at inlet to cold heat exchanger T_{in} = 30 °C;

– total thermal power removed from the modules in operating mode Q_4 = 3.68 kW (115 W based on one module);

– heat carrier consumption (heat carrier – antifreeze) g_t = (1.6; 2.6; 3.6; 4.6; 5.2) m^3/h;

– channel area of the cold heat exchanger S_c = (55 – 300) cm^2;

– the thermal conductivity of cold heat exchanger material λ_{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. 9 shows a dependence of thermal power Q_5 removed from thermoelectric modules to automobile engine on the total area of channels S_c of the cold heat exchangers with different heat carrier consumption.

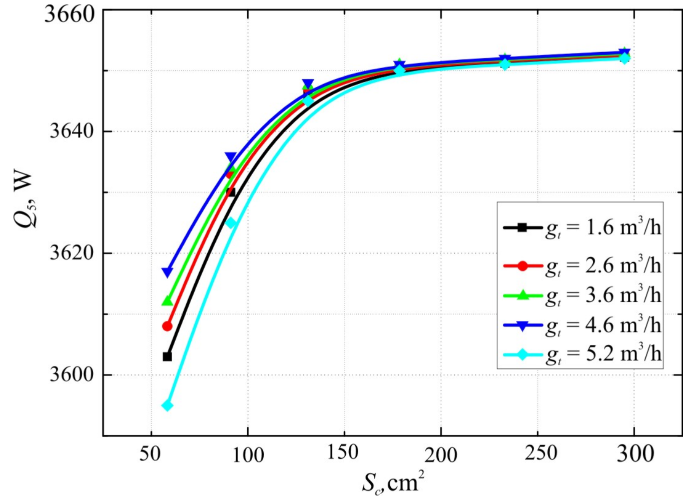


Fig. 9. Dependences of thermal power Q_5 on the channel area S_c of the cold heat exchangers.

From the analysis of data represented in Fig. 9 it follows that for complete transfer of thermal power from the modules to the engine the area of channels in the cold heat exchangers must be at least 170 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.

Moreover, as is seen from Fig. 10, at S_c = 40 cm^2 heat carrier consumption has no essential effect on the difference in heat carrier temperature at the inlet to and outlet of cold heat exchangers (ΔT = 1 – 2 °C). So, the choice of optimal g_t will be determined only by engine displacement and automobile cooling circuit.

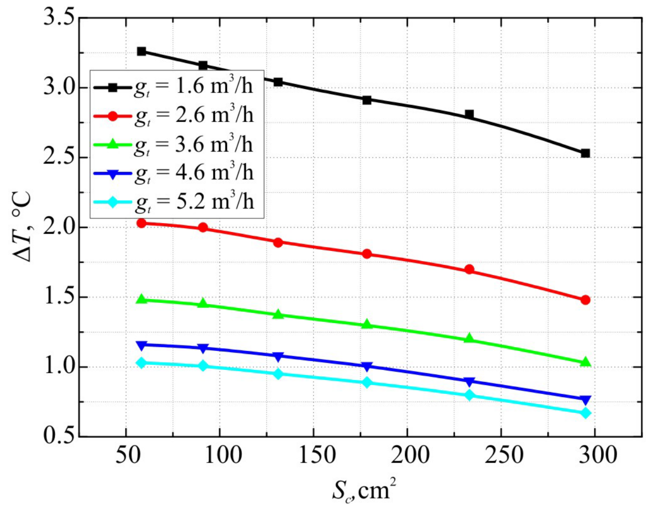


Fig. 10. Dependences of heat carrier temperature difference ΔT between the inlet to and outlet of the cold heat exchangers on the channel area S_c.

Fig. 11 and 12 show the finite element method mesh and temperature distribution for the cold heat exchanger.

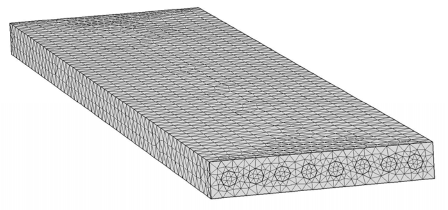


Fig. 11. Finite element method mesh.

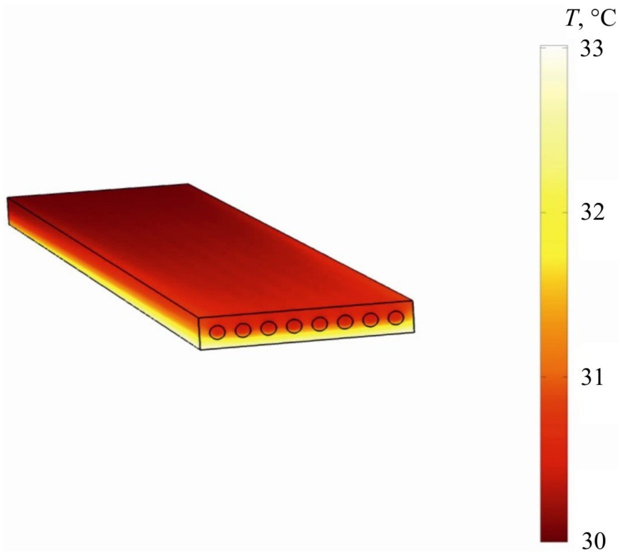


Fig. 12. Temperature distribution in the cold heat exchanger. S_c = 170 cm^2, g_t = 5.2 m^3/h.

Start heating of the engine of displacement more than 10 l (modern 8 – 12 cylinder internal combustion engines), heat capacity 0.462 kJ(kg K) from 0 °C to 30 °C will require ~ 13.9 MJ of thermal energy. Taking into account that total thermal power removed from the thermopile by heat carrier with temperature 30 – 70 °C is ~ 3.65 kW (Fig. 13), start heating of such engine will require ~ 1 hour.

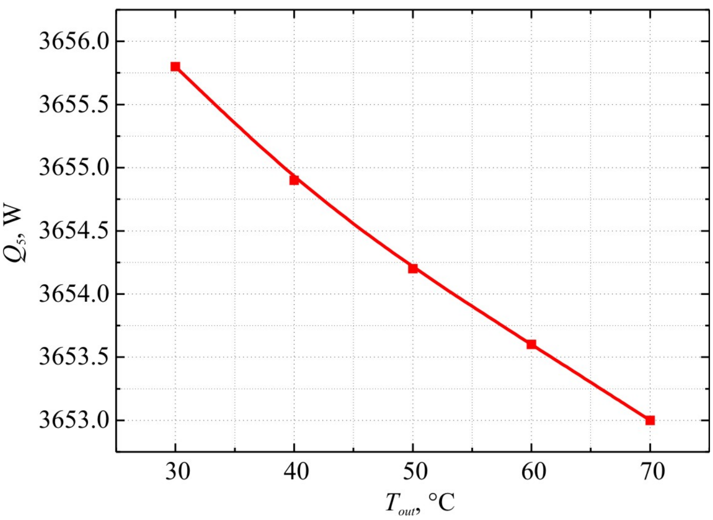


Fig. 13. Dependences of thermal power Q_5 removed by heat carrier on heat carrier temperature T_{out} at the outlet of the cold heat exchangers.

Under real conditions the time of engine heating will be somewhat 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 the "heater-engine" system.

Conclusions

1. It was established that the electric power output of the heater 230 – 250 W is attained through use of 32 modules "ALTEC-1061" with the hot and cold side temperatures 280 – 330 °C and 30 –70 °C,

respectively. With the heater efficiency 3 – 3.5 %, to attain such electric power, it is necessary to spend ~ 7.2 – 6.6 kW of heat, which corresponds to fuel consumption 510 – 550 g/hour.

2. It was shown that the necessary temperature level on the hot side of thermopile 280 – 330 ºC is attained at gas temperature in the hot heat exchanger in the range of 350 – 400 °C, the velocity of air delivery to combustion chamber 2.6 – 2.8 m/s and the area of the heat-absorbing surface of the hot heat exchanger 0.12 m^2.
3. It was established that total thermal power transferred from the thermopile of the heater to heat carrier of the engine is ~ 3.65 kW. At heat carrier temperature 30 – 70 °C the start heating of engines of displacement more than 10 l from 0 °C to 30 ^oC will require ~ 60 min.

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