Bulletin of the *Transilvania* University of Braşov • Vol. 9 (58) No. 2 - 2016
Series I: Engineering Sciences

DIESEL FUEL HEATER USING ENGINE COOLANT FOR COLD WEATHER OPERATION

Veneția SANDU[1]

Abstract: *The paper studies diesel fuel heating in vehicle operation at below-zero temperatures by means of engine coolant and a helical coil heat exchanger placed in the fuel tank. A preliminary prototype of the helical coil heat exchanger was manufactured and investigated on a diesel engine truck operating in real winter condition being measured the diesel fuel and coolant temperatures. The experimental data on coolant temperature variation and heating time were used to improve the final thermostatically controlled design.*

Key words: *diesel fuel preheating, helical coil heat exchanger.*

1. Introduction

The behavior of fuel in diesel engines running at low temperatures is strongly influenced by its paraffin content which may crystallize and clog the fuel filter and block the fuel supply. The phenomenon known as waxing may occur under 0 °C, so in order to avoid paraffin precipitation the diesel cold flow performance must be managed either depressing the cold fuel temperatures (Cloud Point, Pour Point, Cold Filter Plugging Point) with diesel flow improvers or by using fuel heating.

The gradual replacement of the petroleum diesel with renewable fuels such as biodiesel accentuates that difficulty due to higher temperatures at which crystallization occurs in biodiesel blends.

The need of fuel heating led to the implementation of several techniques described in literature either as electric heating of fuel filter, fuel pump or fuel duct [4], [5] or heat exchangers based on the heat released by engine hot sources such as coolant [6], lubricant or exhaust gas. A patent survey in this field was presented in [9] showing that the most spread solutions opt for electric and coolant heating.

Especially for heavy duty diesel vehicles, the high demand of electricity on board and the opportunity of energy harvesting led to the implementation of special purpose designed heat exchangers mounted either on the fuel tank or on the low pressure fuel ducts.

The present study reveals the design, testing and improvement of a fuel heater used in a 100 liter tank of the 7 tonne diesel truck using the heat of the engine coolant.

2. Design Demands and Phases

For diesel engines fitted with mechanical injection systems the optimum viscosity of the fuel in fuel filter is reached in the

[1] Mechanical Engineering Dept., *Transilvania* University of Braşov.

interval 20-40 °C [9]. The fuel heater was intended to be placed in the fuel tank being heated with engine coolant taken from the cabin heating system, upstream thermostat.

The coolant circuit passes through a hose to fuel tank heater and returns in the cooling system upstream coolant pump.

In summer time the system is not used and the circuit is closed manually with a tap or electrically with a thermostat. The system has no influence on the engine cold starting thus being performed by a glow plug.

The design was performed in three phases:

1. Preliminary design of a experimental prototype (heat exchanger and fuel heating system);
2. Prototype testing;
3. Final design.

The testing was required to evaluate in real operation condition during winter the efficiency of the heat exchanger measured as fuel temperature increase and the time needed to reach the desired temperature.

2.1. Preliminary Design

Paradoxically, the engine coolant becomes the heating fluid of the fuel in preheating system. The appropriate type of heat exchanger was considered to be the helical coiled one, with higher heat transfer coefficients than straight tubes due to the secondary flow produced by centrifugal force induces in curved tube [3].

The helical coil heat exchanger was designed to operate immersed in a high capacity tank according to the procedure described in [2]. The fluid characteristics were approximately in the first iteration as those of pure water and diesel fuel for automotive use.

The flow rates of the agents (coolant, diesel fuel) were provided by technical documents of the engine and auxiliaries [10]; the coolant temperatures were collected from test reports on cabin heating [11] of a similar truck and are indicated in Table 1.

Heat exchanger parameters Table 1

	Pure water	**Diesel fuel**
Inlet temperature, t_i [°C]	72	–5
Outlet temperature, t_e [°C]	70	20
Heat capacity, c_p [J/kg·K]	4190	1800

The fuel flow rate passing through the heater was around three times higher than the engine fuel consumption [12] due to the pumping technique which delivers more fuel to cool the injectors, implying some warmed excess fuel which is returned in the tank.

The heat exchanger was made of a cylindrical copper pipe of 0.018/0.015 mm (outer and inner diameter), resistant to corrosion, having a very high thermal conductivity.

The calculation was based on classical heat transfer between two fluids separated by a cylindrical wall, neglecting the heat transfer of the tank with the air and considering the heat exchanger ideal. The subscript 1 refers to the heater agent and the subscript 2 to the heated one.

<u>On the coolant side</u>, when the coolant passes through the circular section of the coil, its velocity, ω_1, is calculated with Equation:

$$\omega_1 = \frac{\dot{m}_1}{\rho_1 \cdot \frac{\pi d_i^2}{4}}. \tag{1}$$

The Reynolds number, Re, calculated for the water viscosity at the mean temperature between inlet and outlet, considered to be 70 °C, indicated a turbulent flow:

$$\mathrm{Re}_1 = \left(\frac{\omega_1 \cdot d_i}{\upsilon_1}\right). \tag{2}$$

For Reynolds number higher than 10^4 it is recommended the following criterial equation which correlates Nusselt number, *Nu*, with Reynolds number and Prandtl number, Pr:

$$Nu_1 = 0.021 \cdot \left(\frac{\mathrm{Pr}_1}{\mathrm{Pr}_w}\right)^{0.25} \cdot \mathrm{Re}_1^{0.8} \cdot \mathrm{Pr}_1^{0.43} . \qquad (3)$$

Prandtl number at the wall temperature, Pr_w, is assumed to be the same with Pr_1 number of the fluid. The convection coefficient α_1 from water to the coil wall can be written in the form:

$$\alpha_1 = Nu_1 \cdot \frac{\lambda_1}{d_i} . \qquad (4)$$

<u>On the fuel side</u>, diesel fuel in the tank is supposed to be heated from –5 to 20 °C and the flow is laminar due to high tank capacity and gravity, being recommended the criterial equation which correlates Nusselt number with Grashof number, *Gr*, and Prandtl number:

$$Nu_2 = 0.4 \cdot (Gr_2 \cdot \mathrm{Pr}_2)^{0.25} , \qquad (5)$$

$$Gr_2 = \frac{g\beta d_e^3 \cdot (t_{2w} - t_{2bl})}{\upsilon_2^{\,2}} . \qquad (6)$$

The convection coefficient α_2 from the coil wall towards the fuel is similar to Equation (4). The calculations considered water properties (mass flow rate, $\dot{m}_1 = 0.08$ kg/s, density, ρ_1 = 977 kg/m^3, viscosity at 70 °C, υ_1 = $0.42 \cdot 10^{-6}$ m^2/s, Prandtl number Pr_1 = 2.4, thermal conductivity λ_1 = 0.663 W/mK), wall properties (copper thermal conductivity, λ_w = 300 W/mK) and diesel fuel (density, ρ_2 = 838 kg/m^3, viscosity at t_{bl} = 33.2 °C, $\upsilon_2 = 6 \cdot 10^{-6}$ m^2/s, Prandtl number Pr_2 = 83.3, thermal conductivity λ_2 = 0.11 W/mK, gravity g = 9.81 m/s^2, $\beta = \frac{1}{T_{bl}} = 0.0033$ K^{-1}).

The wall temperature on fuel side t_{2w} was initially considered 70 °C and the t_{bl} = 32.5 °C resulting Gr_2 = 196959, Nu_2 = 25.4 and α_2 = 155 W/m^{2}K.

By introduction of α_2 in the equation of heat transfer per unit length of pipe:

$$\pi d_1 \alpha_1 (t_1 - t_{1w}) = \pi d_2 \alpha_2 (t_{2w} - t_2) , \qquad (7)$$

it can be calculated the temperature of the wall on the water side, resulting t_{1w} = 67.8 °C. The temperature drop in the wall is considered negligible due to very low thermal resistence, so $(t_{1w} - t_{2w}) \approx 0$, in the first iteration.

In the second iteration starting with t_{1w} = 67.8 °C Grashof number becomes Gr_2 = 189944 and α_2 = 153.7 W/m^{2}K with an error towards previous calculation less than 1%.

By equalizing Equation (7) with heat transfer in the pipe wall:

$$\pi d_1 \alpha_1 (t_1 - t_{1w}) = \frac{2\pi\lambda_w}{\ln \frac{d_1}{d_2}} (t_{1w} - t_{2w}) , \qquad (8)$$

it results the real temperature drop in the wall of 0.06 °C.

For the diesel fuel, the properties were calculated at the temperature of the thermal boundary layer, t_{2bl}, as arithmetic mean of t_{2w} and t_{2i}.

The rate of heat $\dot{Q}_1$ transmitted from water is calculated as follows:

$$\dot{Q}_1 = \dot{m}_1 c_{p_1} (t_{1i} - t_{1e}) . \qquad (9)$$

Finally, the linear overall heat transfer, *k*, takes into account the thermal resistances for each environment: water, copper wall and diesel fuel:

$$k_l = \frac{\pi}{\frac{1}{\alpha_1 \cdot d_i} + \frac{1}{2 \cdot \lambda_w} \ln \frac{d_e}{d_i} + \frac{1}{\alpha_2 \cdot d_e}}. \quad (10)$$

The linear rate of heat between fluids is calculated with the formula:

$$\Phi_l = k_l (t_{1i} - t_{2i}). \quad (11)$$

For the intermediate values α_1 = 3207 W/m^{2}K, α_2 = 153.8 W/m^{2}K, k_l = 8.22 W/mK, it results Φ_l = 633W/m.

The total minimum length, L, of the helical coil results from the formula:

$$L = \frac{\dot{Q}_1}{\Phi_l}. \quad (12)$$

The total length results to be minimum 1.06 m and is divided in a helical length L_h and a straight length L_s:

$$L_h = N\left[(\pi D)^2 + s^2\right]^{0.5}. \quad (13)$$

The dimensions of the helix are: D – diameter; H – height; s - pitch and, implicitly; N - number of coils.

These parameters are dimensionally constrained by two factors: the diameter of the filler neck of the tank (0.105 m) which will allow easier retrofit mounting and the depth of the tank of 0.4 m.

It was chosen D = 0.07 m with the pitch s = 0.04 m, resulting N = 4 and L_h = 0.9 m.

The helix was mounted inclined being positioned to reach the bottom of the tank, folding the fuel suction pipe. The pitch between coils has the smallest value to keep the height H low, thus leading to total immersion of the helical coil even when the tank is one quarter of capacity full. The tank is a rounded corner right angle parallelepiped having dimensions 570x550x400 mm, so the minimum measure of the straight portion, L_s, becomes 170 mm.

2.2. Prototype Testing

The heat exchanger designed according to values from Chapter 2.1 was manufactured and then the heating installation was assembled and tested on the facilities of Road Vehicle Institute (INAR Braşov).

The heating system was adapted to the configuration of the truck AB 7120F manufactured at Roman Truck company, series 0205, which was equipped with a 798-05 diesel engine, series 04598, derated to 120 HP@2500 rpm.

Prior mounting, the injection pump was demounted and readjusted on the pump test bench, the fuel filters were replaced, the fuel tank was replaced with a newly modified one and the engine fuel supply and cooling system were deaerated. The engine coolant was made of 50% (v/v) distilled water and 50% propylene glycol based antifreeze liquid, to withstand to temperatures of –30 °C.

The layout of the heating and measurement system was made of 100 liter fuel tank, helical coil heat exchanger placed in fuel tank around fuel line supplies, T-type connectors to engine cooling system and coolant hoses, as represented in Figure 1.

The equipment included calibrated instruments: data registrator, voltage inverter, temperature sensors, type Pt 100, class B, in range (–50 - 150 °C) with accuracy varying from ±0.3 °C at 0 °C and ±0.8 °C at 100 °C.

The diesel fuel temperature sensors were mounted inside the lateral tank walls and to the inlet of fuel filter; two sensors were welded on the lateral walls, close to the bottom of the tank.

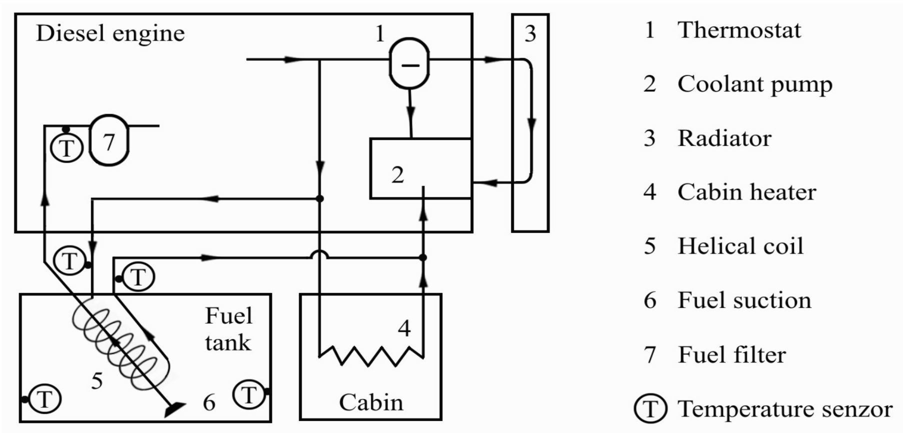

Fig. 1. *Layout of testing installation*

The coolant temperature sensors were mounted on direct and return line from water pump to cabin heater as can be seen in Figure 2.

The routing Brașov - Hărman - Prejmer Teliu - Brădet - Întorsura Buzăului -Vama Buzăului and return was run three times on dry asphalt.

The mean atmospheric temperature was -2 °C,the load of the truck was 1500 kg and the fuel tank full.

The mean speed was 40 km/h and the mean values of measurements are enclosed in Table 2 [10].

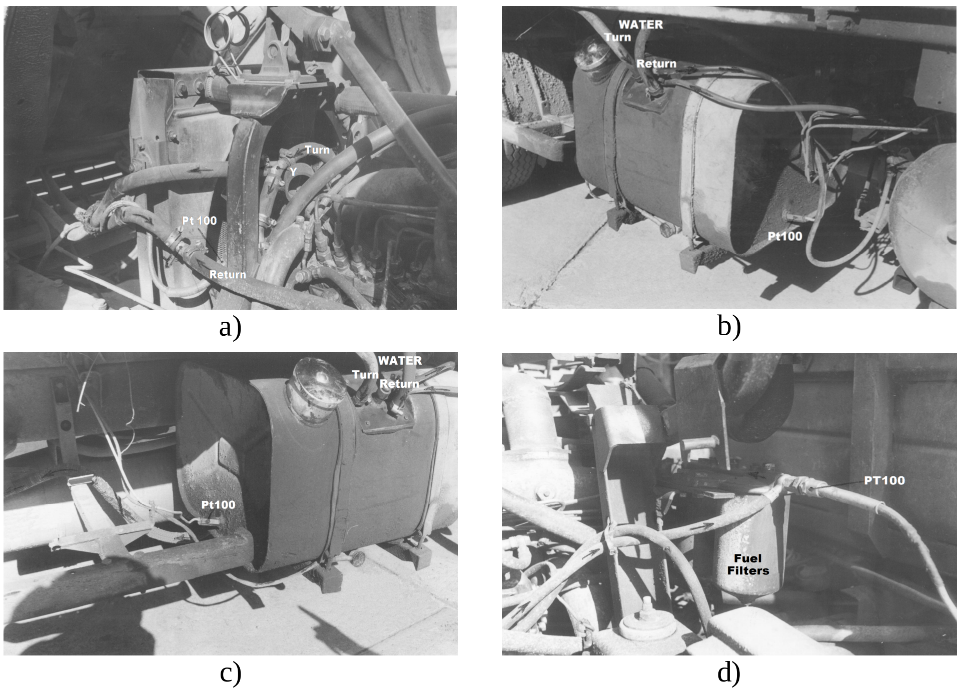

Fig. 2. *a*) *coolant hoses to water pump and cabin and temperature sensor;*
b) *back side tank view with coolant hoses and temperature sensor;*
c) *front side tank view with coolant hoses and temperature sensor;*
d) *fuel filter with temperature sensor*

The final measurements were performed 45 minutes after the initial ones. Later on the testing day, at very short stops, less than 1 hour, it was observed that around the helical coil subsisted a warm fuel blanket with positive effect on subsequent engine start.

Test results Table 2

Temperatures, [°C]	**Initial**	**Final**
Fuel temperature in tank	3.5	15
Coolant temperature in direct circuit	66	75
Coolant temperature in return circuit	63	72

Therefore the design was reiterated aiming to lower the operation temperature under −5 °C, to shorten the heating period and to consider heat loss of the tank towards environment.

The fuel temperature within tank has been increased too slowly due to high thermal inertia of the fuel mass. The results were appreciated as rather fair, but the time interval of heating could be improved.

2.3. Final Design

The most important data of the testing was that the coolant temperature variation in steady state was 3 °C instead of 2 °C as supposed in preliminary phase.

For a better accuracy of the results, the properties of the coolant were modified due to the presence of antifreeze agent. The major changes were the inlet temperature of the diesel fuel (−15 °C), the section of the coolant pipe and the properties of the coolant.

A more rapid heating of the fuel requires a higher rate of heat than in preliminary design calculation described by Equation (9).

The difference of fuel temperature becomes 35 °C instead of 25 °C in preliminary situation. As the fuel flow rate should be kept constant, a solution can be the rising of coolant flow rate with (35/25 = 1.4) ratio, using a wider pipe and helical coil section, thus meaning to pass from d_i = 0.015 to 0.018 m and to thin the pipe wall from 1.5 to 1 mm.

The calculations considered coolant as a mixture of 50% distilled water and 50% propylene glycol with the following properties [7]:

- mass flowrate $\dot{m}_1 = 0.112$ kg/s;
- density, $\rho_1 = 930$ kg/m^3;
- viscosity at 70 °C, $\upsilon_1 = 1.5 \cdot 10^{-6}$ m^2/s;
- Prandtl number, $Pr_1 = 10.62$;
- thermal conductivity, $\lambda_1 = 0.44$ W/mK.

The wall properties remained unchanged and diesel fuel properties were the same, just the thermal boundary layer temperature dropped from 32.5 to 30 °C.

On the coolant side the Reynolds number dropped under 10000 and for a laminar flow rate the criterial equation indicated in [1] is:

$$Nu_1 = 3.66 + \frac{0.065 \mathrm{Re}_1 \mathrm{Pr}_1 \dfrac{d_i}{L}}{1 + 0.04 (\mathrm{Re}_1 \mathrm{Pr}_1)^{2/3}} . \quad (14)$$

In Equation (14) L is the length of the pipe resulted from preliminary design, L = 1.06 m. The calculations were repeated according to Equations (1), (2), (4), replacing Equation (3) with Equation (14).

On the fuel side, diesel fuel in the tank is supposed to be heated from −15 to 20 °C and the flow remains laminar, being used the preliminary criterial Equations (5), (6).

For the intermediate values Re_1 = 5678, Nu_1 = 16.8, α_1 = 411 W/m^{2}K, α_2 = 159 W/m^{2}K, it results Φ_l = 627 W/m.

The new length of the heat exchanger, L', should be calculated taking into consideration not only $\dot{Q}_1$, but also the rate of heat loss of the tank towards environment, $\dot{Q}_{loss}$:

$$L' = \frac{\dot{Q}_1 - \dot{Q}_{loss}}{\Phi_l}. \qquad (15)$$

In the formula of the rate of heat loss:

$$\dot{Q}_{loss} = k_t S \Delta t. \qquad (16)$$

k_t is overall heat transfer between the diesel fuel and air through the tank wall, S is total area of the tank, approximated to 0.4 m x 0.55 m x 0.57 m, in form of rectangular parallelepiped and Δt is the temperature difference between diesel fuel and outside air.

For the calculation of k_t it was used the formula from plane wall fuel tanks:

$$k_t = \frac{1}{\frac{1}{\alpha_{1t}} + \frac{\delta_t}{\lambda_t} + \frac{1}{\alpha_{2t}}}, \qquad (17)$$

with α_{1t} - convection coefficient from fuel to tank wall, 9 W/m^{2}K; δ_t - thickness of the steel wall, 0.0015 m; λ_t - thermal conductivity of the tank wall, 50 W/mK; α_{2t} - convection coefficient from tank wall to atmospheric air at the speed of 40 km/h, 35 W/m^{2}K [8]. For k_t = 7.16 W/m^{2}K, the rate of heat loss becomes $\dot{Q}_{loss}$ = 381 W and finally the minimum length of the pipe $L' = 1.18$ m. By applying Equation (13) at the same dimensions of the helix as mentioned in Chapter 2.1, it results the number of coils N = 5 and the length L_h becomes 1.125 m.

By applying the energy conservation on fuel tank volume it was calculated the time till the fuel is heated up to 20 °C; the heating speed of the preliminary model was around 4 minutes per degree Celsius while for the final model was shortened to 3.2 minutes per degree Celsius.

The heat exchanger requires a temperature controller which will turn on the coolant circuit when the fuel temperature in the filter is below 5 °C and will turn off when is over 20 °C. That imposed conditions prevent the overheating of the fuel in the tank and driver intervention at the change of ambient temperatures.

3. Conclusions

The preliminary design of the heat exchanger introduced significant errors when water properties were used instead of water-propylene glycol blend. The differences of fluid properties, mainly viscosity, changed the type of fluid flow with significant modifications on Reynolds and Nusselt numbers.

The testing of the experimental prototype provided valuable data on real operation heat loss and heating time.

The diesel fuel heating system using engine coolant confirmed the efficiency of the on board energy harvesting solution; on long term, the advantage of saving on board energy tends to counterbalance the constructive extension.

Acknowledgements

The author is grateful towards Vehicle and Engine Testing team from Road Vehicle Institute Brașov for their kind co-operation and support.

References

1. Eduards, D.K., Denny, V.E., Mills, A.F.: *Transfer Processes: An Introduction to Diffusion, Convection, and Radiation.* New York, Holt, Rinehart and Winston, 1973.
2. Ștefănescu, D., Marinescu, M., Dănescu, Al.: *Transferul de căldură în tehnică*

(*Heat Transfer in Technics*). București. Technical Publishing House, 1982.
3. Kumar, V., Faizee, B., Mridha, M., Nigam, K.D.P.: *Numerical Studies of A Tube-In-Tube Helically Coiled Heat Exchanger*. In: Chemical Engineering and Processing **47** (2008), p. 2287-2295.
4. http://www.diesel-therm.com/diesel-therm.htm. Accessed: 29.07.2016.
5. www.partssystems.com/fuel-fluid-warming/in-line-fuel-warmers/. Accessed: 29.07.2016.
6. http://www.arctic-fox.com/products/details-new/hot-fox-in-linein-tank-fuel-warmers. Accessed: 29.07.2016.
7. http://dowac.custhelp.com/app/answers/detail/a_id/7472/~/propylene-glycols---viscosities-of-aqueous-solutions. Accessed: 29.07.2016.
8. https://www.chromalox.com/-/media/files/training-manuals/en-us/tm-pt501-tank-heat.pdf. Accessed: 29.07.2016.
9. *** *Studiu cu privire la posibilitățile de încălzire a motorinei pe circuitul de alimentare a motorului* (*Study Regarding Diesel Fuel Heating on the Engine Fuel Supply System*). INAR Study No. 33297.
10. *** *Standard Tehnic Român - Motoare diesel 798-05* (*Romanian Technical Standard Diesel Engines 798-05*).
11. *** *Studiu în vederea elaborării procedurii de încercare a autovehiculelor în ceea ce privește eficiența încălzirii cabinei* (*Study for Elaboration of an Automotive Testing Procedure Regarding the Efficiency of Cabin Heating*). INAR B31.3/8965.
12. *** *Experimentarea prototipului instalației de alimentare cu combustibil, cu încălzire* (*Experimentation of the Heated Fuel Supply Prototype*). INAR Study No. 33574.