HEAT RECOVERY SYSTEM IN A FUTURE DIESEL PARTICLE FILTER

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Abstract

The main problem in conventional diesel engines and HCCI diesel is reduction of solid particles emitted to atmosphere. Applying of particle filters requires special methods for their regeneration after some period of the engine work in a result of closing of substrate pores by soot. These methods require additional energy for combustion of soot (additional fuel or electric energy). The new method takes into account a self-regeneration of diesel particle filter by use of special heat recovery system. The paper shows an example of DPF self-regeneration. The exhaust temperature behind the turbine and catalytic converter is very low and particularly for HCCI engine the emission of particles is low in comparison to conventional diesel engines. For that case an additional energy is required for increase of gas temperature before DPF. The preliminary studies show a possibility of using the special design of DPF with heat recovery system. The paper shows the simulation results of such system and possibility of increasing the heat recovery ratio by change of geometry of DPF. The gas heat exchange formulas between DPF and the heat exchange module are partly included in the paper. The preliminary results of calculations shows the possibility of increase of exhaust gases in front of DPF about 20%, which enables a continuous regeneration of DPF. The work is carried out as part of the European Project Ipsy.

***Keywords**: transport, compression ignition engine, diesel particle filter, regeneration*

1. Introduction

Diesel particulate filter requires a long time for deposit of soot on the walls of the inlet ducts. The total mass of soot in DPF is almost linearly proportional to time for modern compression engines. Filtration of soot and nanoparticles depends on structure of used material of the filter substrate, amount and size of soot in the gas, geometry of channels and thermodynamic parameters of exhaust gases. Filtration is not so effective in the clean DPF and after several hours the pores in the filter are partially clogged and filtration proceeds much more effectively. When the filter walls are filled by soot molecules the pressure drop increases in the filter. The level of the pressure drop is the signal for the beginning of regeneration process. Regeneration process can be done by several methods. However these methods require an additional energy for initiation of soot burning (additional fuel injection in DPF, heating by use of electric energy or post-injection of fuel). In compression ignition engines the exhaust temperature behind the turbine and catalytic converter is very low and particularly for HCCI engine the emission of particles is lower in comparison to conventional diesel engines. Applying of particle filters requires special methods for their regeneration after some period of the engine work in a result of closing of substrate pores by soot. Filtration of nanoparticles in DPF and soot combustion is widely described by many authors: Konstandopoulos et al [1, 5], Bisset [7], Nakatani [8] and also by polish researchers Nagórski, Teodorczyk and Bernhardt [3].

2. Regeneration of DPF

The required temperature of exhaust gases should be higher than 600°C to initiate the burning of soot on the filter walls. In order to increase the temperature of inlet gases in DPF a special heat recovery system (HRS) is needed. During combustion of soot in DPF the outlet gases have higher temperature and this phenomenon can be utilized in the recovery heat system.

The one-dimensional mathematical model can be used to simulate non-uniform distributions of soot along the DPF channels and the filter wall during the regeneration process. The model considers that under most common operating conditions of internal combustion engines, the spatial rates of change of fluid properties are far greater than the temporal ones, so quasi-steady flow conditions can be assumed and steady flow gas dynamic relations may be used. The model is primarily used for regeneration modelling though it is also able to extend to soot loading modelling. It was assumed the one-dimensional Bisset model [6] of filter regeneration shown in Fig. 1 in order to determine the changes of DPF parameters during filtration and heat increase after soot combustion.

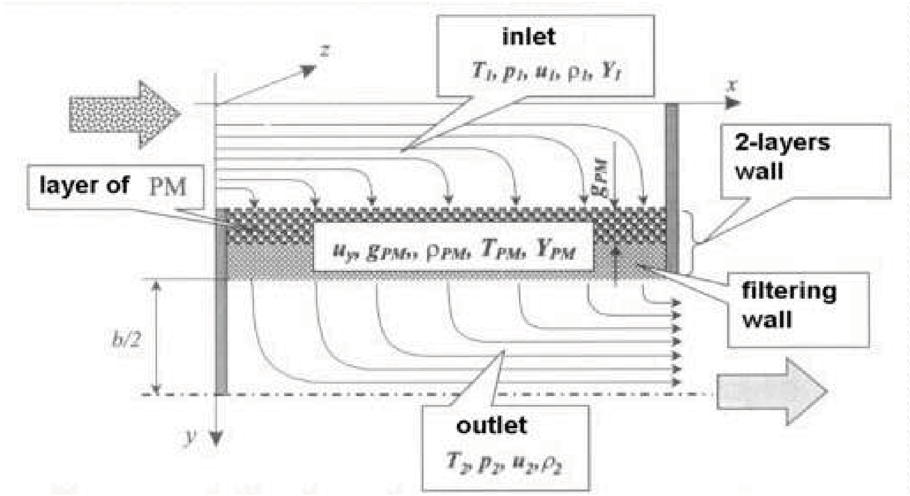

Fig.1. One dimensional model of DPF filtration and regeneration [6]

The model takes into account the following phenomena:
1. Balance of gas mass in the channels,
2. Balance of momentum of gas jets in the channels,
3. Balance of energy conservation,
4. Balance of oxygen,
5. Balance of PM mass,
6. Balance of energy in the filtration wall given for determination of exhaust gas temperature during soot combustion described by author [10].

Full mathematical model of filtration and regeneration of DPF is out of the scope of the paper.

For lower inlet temperature particularly for HCCI engine there is not possible to burn the soot in DPF, because temperature does not exceed 450°C. For that case an additional energy is needed in order to increase the inlet temperature. There are foreseen different regeneration strategies, for example post-injection as a source of soot ignition or increase the temperature in Diesel Oxidation Catalyst (DOC) for burning hydrocarbons (HC) and carbon monoxide (CO). During soot combustion outlet gases have higher temperature, which can be used for heating the inlet gases in the heat exchanger. The first proposition is to use the separate heat exchanger with many heating pipes with cross flow of gases. The task of the heat exchanger is to increase the heat recovery ratio or increase the inlet temperature minimum 20%.

3. Heat recovery in DPF

One of considered design was the simplest recovery model A which is shown in Figure 2. The increase of the exhaust gas temperature from T_1 to T_2 and internal energy caused by the soot combustion effects on the heating of the pipes perpendicularly set to inflow of the gas. Heat exchange between the walls of the pipes and the exhaust gases increases their temperature from T_0 to T_1.

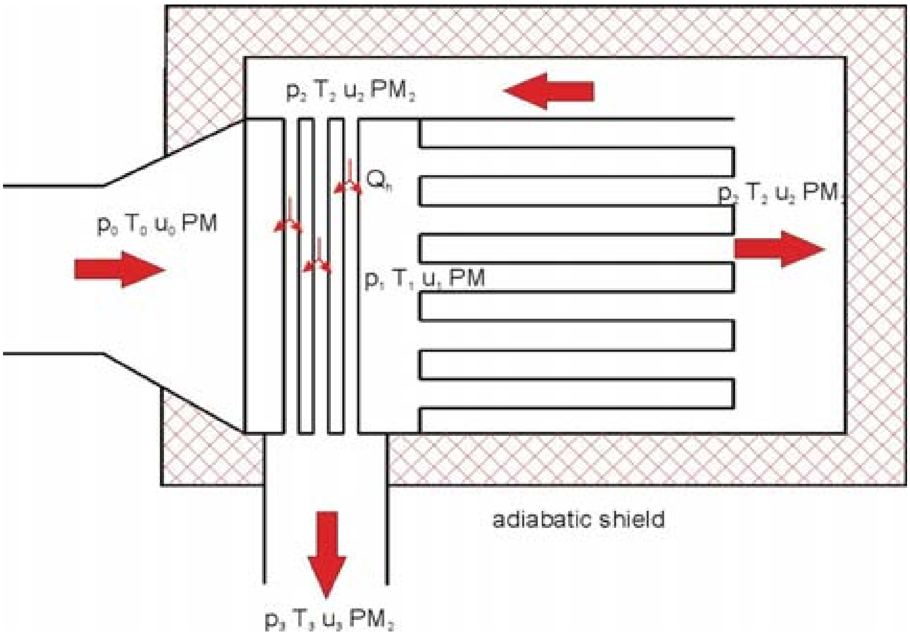

Fig. 2. Diagram of the DPF regeneration model

The heat exchange between the gas behind the outlet channels in DPF flown into the exchange pipe and the gas flown to the DPF the following simple energy equation for ideal gas is given:

$$\dot{m}c_p(T_0)T_0 + \dot{m}\sum Fk(T_2 - T_0) = \dot{m}c_p(T_1)T_1, \tag{1}$$

where: $\dot{m}$ - mass flow rate of the exhaust gas,

T_0 - inflow gas temperature before the exchanger,

T_1 - gas temperature behind the exchanger,

T_2 - gas temperature behind DPF after regeneration,

$\sum F$ - total area of the exchanger,

k - coefficient of heat exchange.

For small changes of the temperature in front of and behind the exchanger the specific heat coefficient at constant pressure can be assumed at the same level. After regeneration the temperature behind the DPF increases with value

$$\Delta T_{DPF} = T_2 - T_1. \tag{2}$$

After simplifications of the equation (1) one obtains the following formula on the increase of the temperature behind the exchanger with value ΔT:

$$\Delta T = \frac{T_0 + \Delta T_{DPF}}{\frac{c_p}{\sum F \cdot k} - 1}, \tag{3}$$

where: $\Delta T = T_1 - T_0$.

The increase of the temperature behind the DPF causes also an increase of the temperature behind the heat exchanger. Value in the dominator should be above zero and the total exchange heat area should meet the requirement:

$$\sum F < \frac{c_p}{k} . \tag{4}$$

At assumption that the exhaust gas in engine with $\lambda = 1.5$ has temperature equal 700 K, then specific heat at constant pressure amounts:

$$c_p = 1146 \qquad \text{[J/(kg K)]}.$$

For engine of capacity V_{ss}= 2000 cm^3 and $\lambda = 1.5$ working at constant speed n=2000 rpm with volumetric efficiency η_v = 1.0 the mass flow rate of the exhaust gas amounts:

$$\dot{m} = \frac{V_{ss}\eta_v\rho_0 n}{120} = \frac{0.002 \cdot 1 \cdot 1.24 \cdot 2000}{120} = 0.04133 \quad \text{[kg/s]}.$$

The density of the exhaust gases at temperature T=700 K and ambient pressure amounts $\rho_{ex} = 0.49$ [kg/m^3]. Then the total flow area of the pipes in the exchanger is achieved with assumed mean gas velocity in the pipes u_2

$$\sum A = \frac{\dot{m}}{\rho_{ex} u_2} \cdot \tag{5}$$

The number of the pipes r is obtained when the inside diameter d is known:

$$r = \frac{4\dot{m}}{\pi d^2 \rho_{ex} u_2} \cdot \tag{6}$$

Assuming the pipes with length L and thickness g the total external exchange area amounts:

$$\sum F = \pi(d + 2g)Lr . \tag{7}$$

The total heat exchange coefficient k is determined from the formula:

$$\frac{1}{k} = \frac{1}{\alpha_1} + \frac{g}{\lambda} + \frac{1}{\alpha_2} , \tag{8}$$

where λ determines heat conductivity of material.

The heat conduction coefficients inside and outside of the pipes depends on the Reynolds and Nusselt numbers, which are functions of the temperature and velocity of the gases.

$$\alpha_{1,2} = f(Nu_{1,2}, d, \lambda_{1,2}) ,$$

$$\mathrm{Nu} = f(\mathrm{Re}, \mathrm{Pr}) .$$

For gas flow inside the pipe Nusselt number is defined as [9]:

$$\mathrm{Nu} = 0.23\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} , \qquad \text{with Re > 2100.} \tag{9}$$

For the gas flow perpendicular to the pipes:

$$Nu = 0.33\,\mathrm{Re}^{0.6}\,\mathrm{Pr}^{0.33}, \qquad \text{with Re} > 2000, \tag{10}$$

$$\mathrm{Re}_1 = \frac{\rho_1 u_1 d}{\mu_1}, \qquad \mathrm{Re}_2 = \frac{\rho_2 u_2 d}{\mu_2}, \tag{11}$$

$$\mathrm{Pr}_1 = \frac{c_{p1}\rho_1\upsilon_1}{\lambda_1} = \frac{c_{p1}\mu_1}{\lambda_1}, \qquad \mathrm{Pr}_2 = \frac{c_{p2}\mu_2}{\lambda_2}, \tag{12}$$

$$\mathrm{Nu}_1 = \frac{\alpha_1(b-d-2g)}{\lambda_1}, \qquad \mathrm{Nu}_2 = \frac{\alpha_2 d}{\lambda_2}, \tag{13}$$

where: μ - gas dynamic viscosity,
υ - gas kinematic viscosity,
λ - gas heat conductivity,
b - distance between pipes.

On the basis of given above equations the simple simulation can be performed to determine the geometrical parameters of the exchanger. Recovery ratio determines how much of heat involved during soot combustion can be utilized for heating of the gas at inlet of DPF and is expressed as:

$$R_{DPF} = \frac{T_{i2} - T_{i1}}{T_{i1}}. \tag{14}$$

The concentrations of the gaseous species in the exhaust gases: CO, CO_2, H_2O, O_2, N_2, NO are calculated from the equation of the diesel oil combustion and equilibrium equations for a given relative air-fuel ratio λ. Mass concentration of the species *p*:

$$g_p = \frac{x_p M_p}{m_{ex}}. \tag{15}$$

Specific heat at constant pressure for mixture of chemical species:

$$c_{pp} = MR\sum_{j=1}^{j=5} a_j T_j^{\,j-1}. \tag{16}$$

The one- dimensional steady model of DPF heat recovery system takes into account:

a) number of the pipe rows and columns,
b) length of the pipes,
c) rectangular shape of the regeneration part (the same length),
d) constant inlet temperature of the pipes,
e) constant velocity and pressure equal the pressure in the DPF outlet channel,
f) change the pressure and temperature in the pipes,
g) change of the exhaust gas temperature in each column of the pipes,
h) constant mass flow rate around the pipes to DPF.

The biggest recovery ratio will be in the space which contains the inlets of pipes. The increase of the temperature will be lowest at the last column of the pipes.

The precise mathematical model of the DPF self-regeneration should take into account the discretization model, because the temperature inside the pipes during heat exchange decreases as a result of heat transfer to the walls. The enthalpy of the mass of gas in the pipes decreases and on the pipes' outflow the temperature is higher than in the case without soot combustion in the DPF. The temperature in DPF depends on amount of the burnt mass of soot and influences on the regeneration ratio. In simulation a different levels of ΔT_{DPF} were assumed.

The different initial temperatures and mass flow rates were assumed at the first step in the calculations of the DPF self-regeneration. It was assumed that system is in adiabatic cover and therefore temperature flowing into pipes T'_2 is equalled the temperature T_2 flowing out from the DPF. The discretization model takes into account the change of the temperature of the regeneration gases and gases in the pipes. The pipes and the recovery volume were divided into small volumes with height Δx with assumption of the same gas parameters in the same length x from the beginning of the pipes where the exhaust gases from the DPF flow in. The discretization model is presented in Fig. 3 with certain number of rows and columns of pipes. At the first step it was taken into account only the heating of the gas by conduction and convection. It was assumed the equal value of the exhaust gas temperature T_o on each level of the discretization model.

The heat exchange between the gas inside the pipes and outside gas for each level Δx can be performed by use the energy balance for ideal gas (1) and for each domain of the discretized model it has the following form:

$$\dot{m}_j c_p(T_0)T_0 + \sum F_j k_j \left(T_{2j} - T_{1j}\right) = \dot{m} c_p(T_{1j})T_{1j}, \tag{17}$$

where: $\dot{m}_j$ - mass flow rate of the exhaust gas in domain j,

T_0 - inflow gas temperature before the exchanger in domain j,

T_{j1} - gas temperature behind the exchanger in domain j,

T_{2j} - gas temperature behind DPF after regeneration in domain j,

$\sum F_j$ - total area of the exchanger in domain j,

k_j - coefficient of heat exchange.

4. Calculations of the heat exchanger efficiency of model A

On the basis of given above equations the simple simulation can be performed to determine the geometrical parameters of the exchanger. The simulation was carried out by use the computer program written by author in Microsoft C++. This was done for different configurations of the exchanger system. The calculations were carried out for the following assumptions:

- radiation neglected,
- engine speed - 2000 rpm,
- square section of the exchanger - 160 x 160 mm,
- circular cross section of the pipes,
- pipe distance (between walls) - 1.5 mm,
- mean effective pressure - p_e = 2 bars,
- engine capacity - V_{ss} = 2.0 l,
- volumetric efficiency - $\eta_v = 1.2$,
- excess air ratio - λ = 1.2,
- soot production (Cummins data) - 0.085 g/kWh.

By change of many geometrical parameters the change of the temperature outside of the pipes and inside the pipes, walls' temperature and exchange heat power was determined by assumptions of the equal inlet temperature T_o in front of the exchanger.

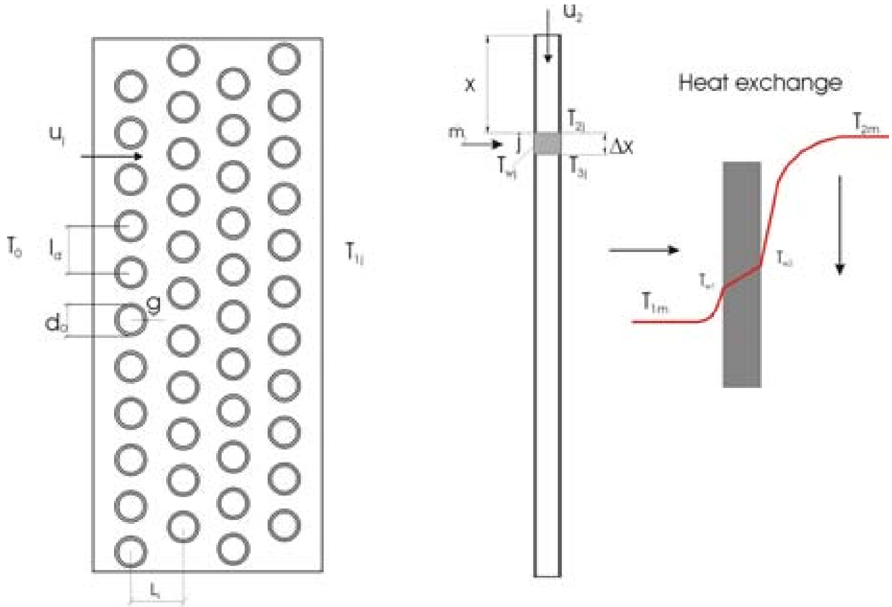

Fig. 3. Heat exchange on the discretization model

The pipes and calculation area were divided into i = 20 equal volumes with different gas parameters. The calculations were performed for pipes made from copper and cordierite material. The first step was to use the copper pipes in the regenerator with heat conductivity $\lambda = 350$ W/(m^2 K). The temperature of the gas, which flows between the pipes to the DPF decreases along the pipes. The calculations were done for different temperature increments in DPF (100 – 600 K). The increment of the temperature outside and along the pipes is shown in Fig. 4. The difference between the beginning and end of the pipes is bigger for bigger ΔT_{DPF} and amounts almost 150 K. The increase of this temperature for ΔT_{DPF} =200 K is small (60 – 100 K).

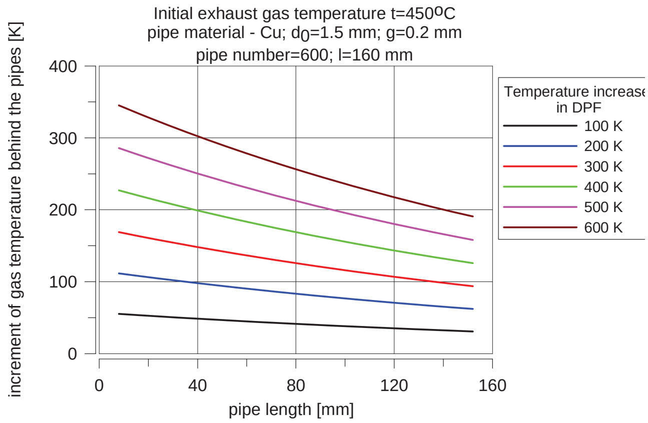

Fig. 4. Increase of the exhaust gas temperature after regeneration along the pipes at different temperature increase in DPF with initial temperature 450 °C (without radiation)

Inlet temperature in DPF is shown in Fig. 5 for initial exhaust gas temperature 450 °C, which is smaller at the pipe's outflow. This was calculated without radiation and for copper pipes with inner diameter d=1.5 mm and g=0.2 mm. The higher increment of the temperature in DPF, the bigger difference of temperature of the gas occurs inside the pipes at the pipes' inlet and the pipes' end (Fig. 6) for the same initial gas temperature 450°C.

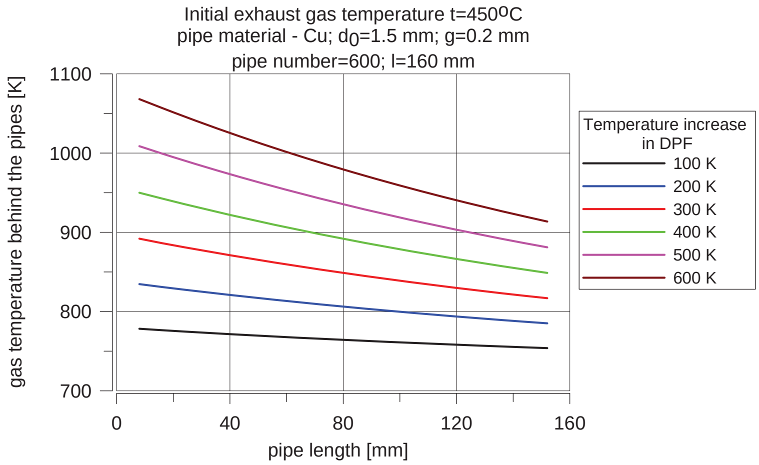

Fig. 5. Exhaust gas temperature after regeneration along the pipes at different temperature increase in DPF with initial temperature 450 ^{o}C (without radiation)

The wall temperature is needed for calculation of the radiation heat. The walls' temperature also decreases along the pipes, which is shown in Fig.7, and at the inlet of pipes is almost 150K smaller than the temperature of the gas inside the pipes for ΔT_{DPF} =600 K.

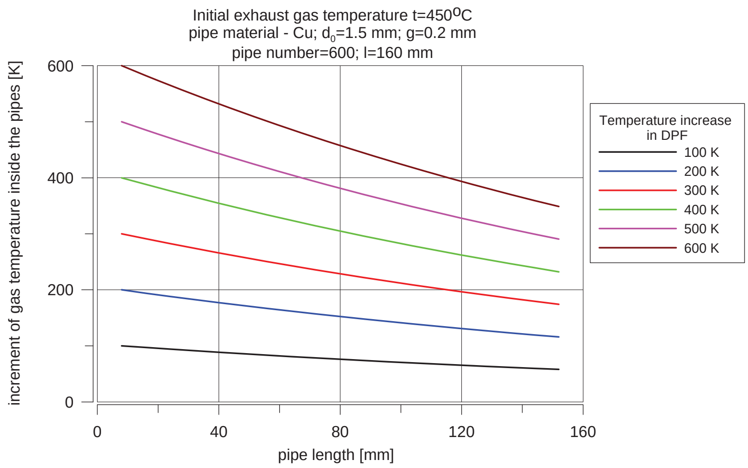

Fig. 6. Increase of the inner exhaust gas temperature along the copper pipes at different temperature increase in DPF (with initial temperature 450 ^{o}C) without radiation

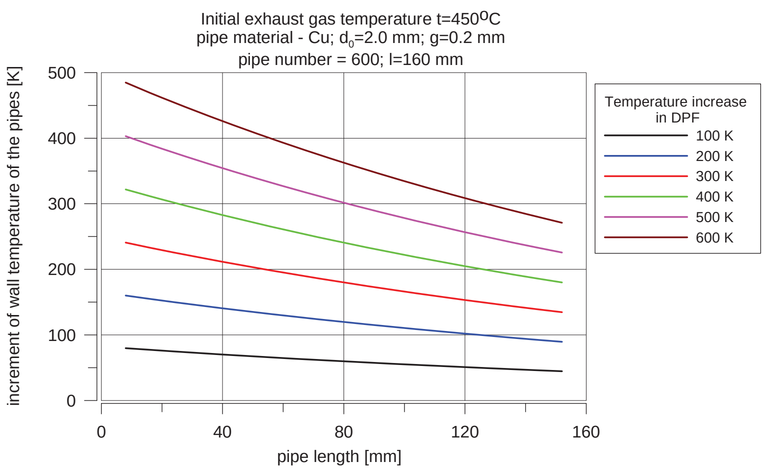

Fig. 7. Increment of the wall temperature at different temperature increment in DPF for copper pipes at initial gas temperature 450^{o}C without radiation

During soot combustion the heat is released, which is partly consumed by the heat exchanger for the increase of temperature of the inflowing gases. The following heats were calculated: heat power release, regeneration heat power and heat power losses (Fig.8) as a function of the temperature increase in the DPF. With the increase of ΔT_{DPF} the total heat powers also increases.

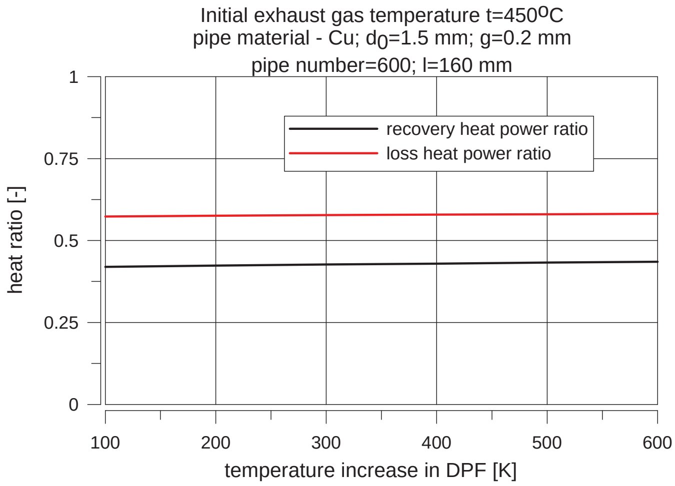

Fig. 8. Heat power of the regeneration without radiation (Cu pipes)

The linear dependence of the heat powers causes the independence of the temperature increment in DPF on the heat recovery ratio, which is about 40% of the heat release in DPF (Fig.9). The rest are heat losses – increment of enthalpy of the outflow gases from the heat exchanger.

In real conditions regeneration temperature should not exceed 1000°C because of possible damage of ceramic substrate and therefore the increment of temperature in DPF is not higher than 550 K.

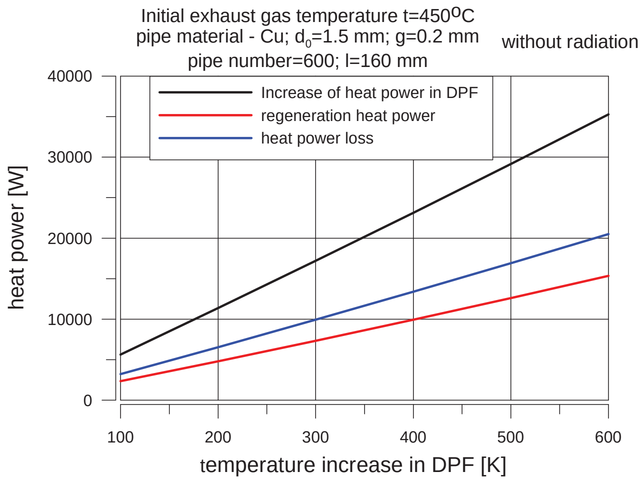

Fig. 9. Heat recovery at different increment of the temperature in DPF (without radiation)

5. Conclusions

The analyses carried out by one-dimensional model of the heat recovery systems (HRS) according to models A enabled to determine the possibility of use the heat from soot combustion for increase of enthalpy of inlet gases. The increase of inlet temperature can help the soot combustion during regeneration particularly for HCCI engines. On the basis of the presented analysis the following conclusions can be drawn:

1. Temperature in outlet channels of DPF after soot combustion is enough efficient for the temperature increase of the exhaust gases in the recovery system.

Above 40% of the total energy of the soot combustion can be taken over the exhaust gas

2. flowing into DPF in the heat exchanger in MFR with the heat recovery system A. The radiation in the heat exchanger system amounts only 30% of the total heat recovery.
3. The increase at the heat recovery above 20% requires an increment of the mean temperature outside the pipes more than 90^{o} C and can be achieved at the increment of the temperature in DPF after soot combustion on the level 200^{o}C in HRS-model A
4. With enlargement of heat exchange area (more pipes) one obtains over 20% heat recovery ratio at increment of gas temperature during soot combustion above 200 K. The higher heat recovery ratio can be obtained when temperature of outflow gases is higher.

6. Acknowledgment

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