EGR valve thermal behavior

Theoretical and experimental analysis

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In this work, the thermal behavior of a new Exhaust Gas Recirculation (EGR) valve, installed on a series Diesel engine, was examined to identify effective thermal loads on it, during its real operation. Both theoretical tools and experimental set-up were used to achieve feasible results. The two different theoretical approaches used were respectively at steady and unsteady operation. They were set-up to account for the complex thermal resistances network, due to different materials used and for the interaction of heat loads on components, due to their different thermal inertia and the characteristic operation of the valve, modelling both conduction and convection phenomena. Some tests on a engine bench have been carried out to validate theoretical models. An instrumented EGR valve was used, provided with thermocouples mounted on particular locations, inside and outside the valve. A good matching between theoretical and experimental results was found. Critical components were located in terms of reached thermal limits and a basis for improvement proposals was defined to reduce valve failure, due to thermal loads.

Introduction

Environmental issues, related to global climate changes and air quality, lead to the use of devices capable to reduce pollution, especially in automobile and truck engine development. As Diesel engines are the present most efficient power devices, their popularity is increasing very much and the overall interest about their efficiency improvement, as well as environmental compatibility, is becoming widely spread.
The current challenge is to minimize the pollutants by carefully driving the combustion processes really occurring in the combustion chamber, in order to deeply control thermodynamic cycles and gas and exhaust formation and distribution to reduce pollutant emissions [1]. NOx generation, in particular, can be lowered causing a charge dilution [2], and this result can be achieved forcing inert substances inside the combustion chamber in order to control the peak combustion temperature and make the NOx formation decrease.

The EGR valve is the device that brings to a reduction of pollutants, acting the above said way, and represents a strategic tool to fulfill the European Union standards for Diesel engines [3]. Thermal phenomena are of great concern in the correct development of the EGR, as the valve, during its operation, is exposed with its different elements to high and rapidly changing thermal loads [4-5].

The thermal analysis

Studying EGR temperature distribution in different working conditions allows to locate the

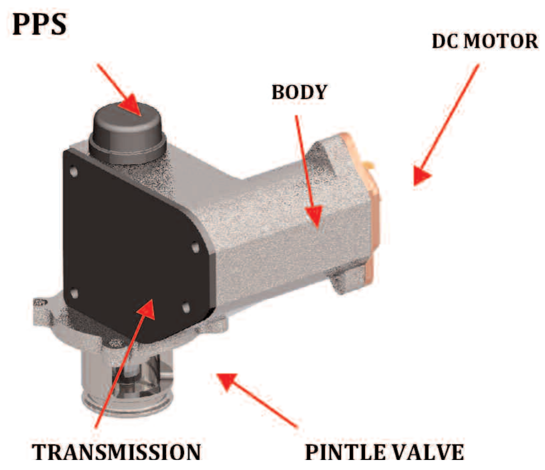

Fig. 1 – DC motor EGR valve: components overview and the gear transmission

most critical components and parts from the thermal point of view, leading to the proposition of possible design improvement proposals.
Both theoretical tools and experimental set-up have to be used to achieve feasible results. In this work a first analysis considered the EGR in the steady state operation with the hypothesis of a one-dimensional heat flow, i.e. that coming through the valve from the exhaust gases. A parametric model, based on a complex thermal resistance network, was developed to forecast the temperature gradient on the valve elements. On this basis, an estimation of the heat fluxes, involved in the complete phenomenon in the perpendicular direction, was also carried out.

In order to predict temperature gradients, the attention was focused on the heat power globally transferred along the exhaust flow direction, estimating heat losses in the direction normal to the valve axis.
In order to account for the transient thermal loads on the EGR valve, a second theoretical analysis has then been carried out. This considered an unsteady condition. An iterative procedure was developed to estimate the temperature level reached on the valve components, depending on the temperature EGR gas recirculated and on the flow conditions, considering both conduction and convection contributions.

Experimental tests on an EGR valve, mounted on a series internal combustion engine, have been then carried out to validate theoretical models. An instrumented EGR was used, at different engine controlled regimes, provided with thermo couples properly located inside and outside the valve.
The worst case from the thermal point of view was considered in the theoretical analysis and this meant positioning the EGR cooler downstream the valve.
On the other hand, as in the experimental set-up configuration, the EGR cooler was necessarily located upstream the valve, available data had to be compared with the relevant model and boundary conditions. The match between theoretical and experimental analyses has been reached extrapolating the real temperature information from the engine tests and using them to validate the theoretical model.

The DC electrical motor driven EGR valve

Fig. 1 shows the DC electrical motor driven EGR valve and its gear transmission to the valve shaft with its sealing plate (from now on named "pintle").

The main EGR valve components are:
- the Hall pintle position sensor (PPS), which detects the position of the valve pintle, using the Hall effect;
- the valve body, made of aluminum;
- the DC motor, which drives the gears and thus the moving of the pintle;
- the transmission, which involves three different gears and one rack and provides the conversion of rotation into translation of the valve pintle;
- the pintle valve, which controls the EGR flow rate.

Tab. 1 shows the components working upper temperature limits.
The main attention was focused on the temperature reached between the pintle shaft and the rack, since the pintle tip temperature is the most critical for the material safety and the correct operation of the valve.

PPS	150 [°C]
DC motor	150 [°C]
Rack*	130 [°C]

Tab. 1 – Temperature limits for the valve
**** The material used for the rack is plastic (UC-1006, PTFE, E=2GPa, ν=0.35, α=19μm/m°C)***

Theoretical Analysis: the steady state approach

When the EGR flow coming from the engine outlet reaches the valve, the temperature level can

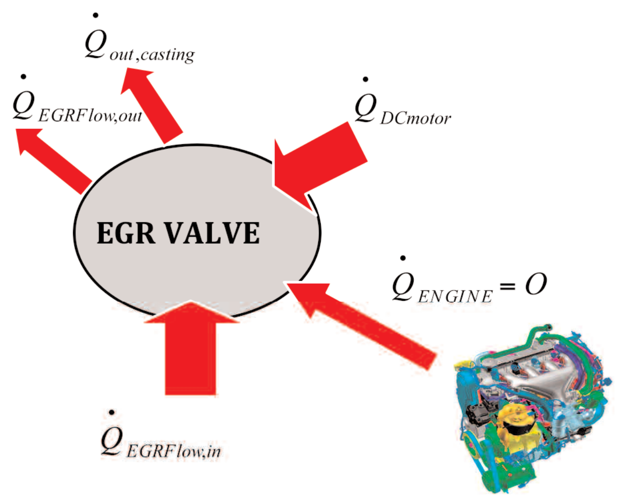

Fig. 2 – The thermal balance for the EGR valve in the working hypothesis

be very high, depending on the working conditions. In the worst thermal load case, the exhaust gases run through the valve placed before the cooler (high temperature and pressure), and not through the turbocharger. The temperature level of the exhaust flow can be estimated considering the mix between the compressed air and the flow exiting in the combustion chamber. The real intake conditions to the engine can be calculated, which determine temperature level on the EGR valve. The highest temperature can be estimated in about 500°C [5].

The heat contributions

In order to analyse the system from the thermal point of view, some preliminary hypotheses have been done. During the valve operation existing heat loads have been properly identified and they are depicted in Fig. 2. The heat stored by the EGR is the difference between the exhaust gas heat power entering it from the inlet and from the outlet, and the heat power generated by the DC motor. The influence of the cylinder block has been neglected compared to the others: no radiating or convective contribution to the system comes from the surrounding.

The thermal resistances network

In order to study the heat transfer phenomena on the valve, some simplifications have been made. The contact between the valve components has been supposed ideal, although this condition is hardly verified because of the material roughness. The temperature field for the one-dimensional geometry has been found using the analogy between electrical and thermal resistance networks. During its operation, the EGR is subject to the two following main perpendicular thermal flows:

- direction 1, which accounts for the heat coming from the exhaust gases;
- direction 2, which is due to the hot surrounding.

Fig. 3 shows the two directions and, for direction 1, the two heat paths, which can be considered with the thermal resistances representation.

In order to estimate the temperature distribution on the valve components, due to its operation, the most important thermal phenomenon is heat transfer in direction 1. The heat flow coming from the exhaust gases is mainly transferred by conduction.

Fig. 4 shows the thermal resistances along paths 1 and 2 in direction 1. The high temperatures found on the components (in particular between pintle tip and rack) suggest to consider thermal resistances in direction 2. This allows to take into account the influence of the convective heat transfer coming from the gases flowing around the pintle.

Using data and results from the first model, with the modifications due to the new schematization, the pintle temperature calculated was: T_{pintle}= 282°C.

Theoretical analysis: the unsteady state approach

In the unsteady conduction, temperature changes in space and time and thus temperature function has to take this in account. After a Biot number analysis (Bi≤0.1), single components of the EGR can be considered as a sub-element with negligible inside temperature gradients. Only changes in time had so to be accounted for.

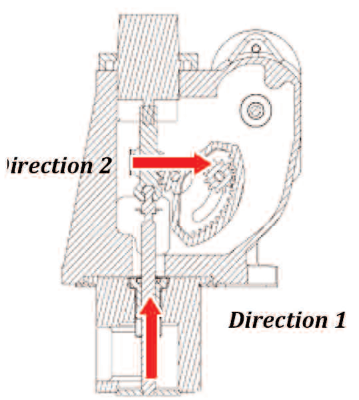

Fig. 3 – Heat flow main directions

i	$h^r_{,i}(t)$	$h^f_{,i}(t)$	$A^r_{,i}$	Nu(t)	$A^f_{,i}$
1	h_1	h_2	A_1	$0.228 Re^{0.731} Pr^{1/3}$	A_2
2	h_3	h_2, h_4	A_3		A_2,A_4
3	h_5	h_4, h_6	A_5		A_4,A_6
4	h_7	h_6	A_7	$Nu = f\,(Ra)$	A_6
$b_i(t)$			$[h^{av}_{,i}(t) A_{tot,i}]^{*1} / (\rho V_i c_p)$		
$T_i(t)$			from (1)		

*1 $h^{av}_{,i}(t)=\sum_{s,t}[h^r_{s,i}(t)A^r_{s,i}+h^f_{t,i}(t)A^f_{t,i}]/A_{tot,i}$

Tab. 2 – The model for the unsteady state approach

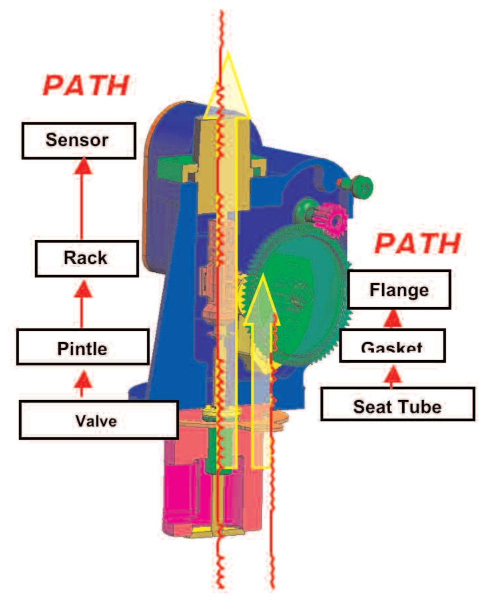

Fig. 4 – The thermal resistances network along direction 1

Each component or sub element of a more complex component was modeled with an expression similar to the following:

$$\frac{T(t)-T_\infty}{T_i-T_\infty}=e^{-bt} \tag{1}$$

where:

$$b=\frac{hA}{pVc_p} \tag{2}$$

A mesh of properly chosen sub-elements of each component was subject to opportune boundary and interface conditions. The pintle was divided into 5 sub-elements, as shown in Fig. 5, even it has been supposed sub-elements 4 and 5 are at the same temperature.

For each element, the problem consists in determining the correct *h* convective coefficient to be used in equation (1), which must consider the different flow conditions around the thermal conductive flow transferred to or from the adjacent parts. Getting the right interface condition means making an equivalence between heat transfer parameters in the Newton and Fourier laws. A fictitious convective thermal flow, which is due to the instant temperature difference between the undisturbed flow and the element, was calculated that accounted for the real conductive thermal flow, due to the instant temperature difference between adjacent sub-elements:

$$h_{fict}=\frac{\lambda}{d}\left(\frac{\Delta T_{cond}(t)}{\Delta T_{conv}(t)}\right)\left[\frac{W}{m^2K}\right] \tag{3}$$

where d is the geometrical distance between the centers of the two adjacent elements.

For each time step, the weighted average h coefficient for each pintle sub-element taking into account all contributions to the change of temperature. This allows to calculate the b coefficient with equation (2) and then the cor-

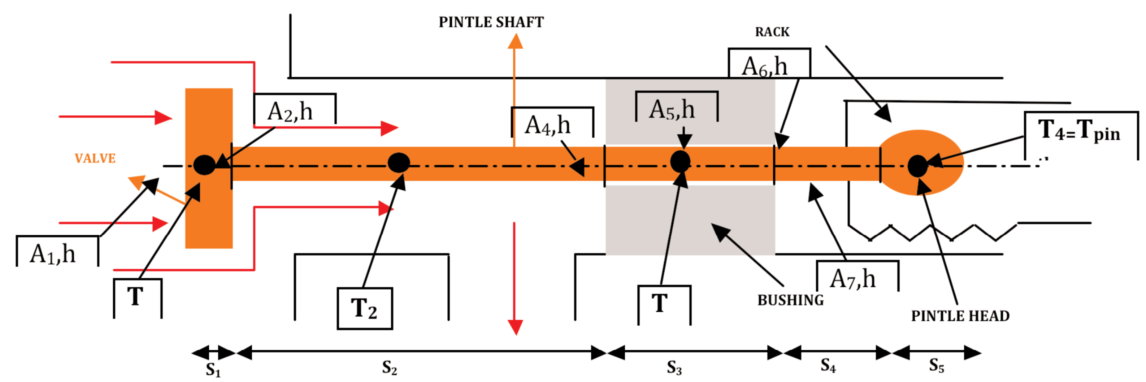

Fig. 5 – Pintle sub-elements schematization

TEST CONDITIONS	2800 [rpm]; 120 [Nm]
	1600 [rpm]; 40 [Nm]
	Coking and Soak Test

Tab. 3 – The engine tests

rect temperature distribution in time with (1) [6-7]. Referring to the symbols in Fig. 5, Tab. 2 shows the model schematization which has been used for each pintle sub element. The initial conditions are:

- T_0 for each sub element;
- T_{EGR} for the flow temperature around sub elements 1, 2 and 3;
- $T_{air,\,body}$ for the low temperature around sub element 4.

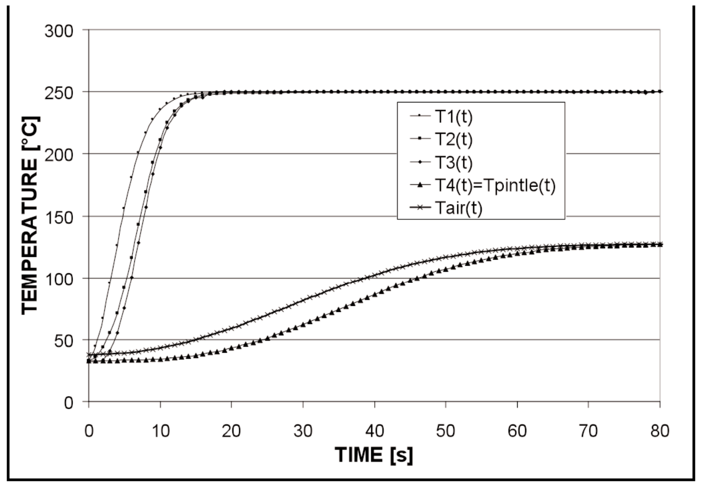

Fig. 6 – Test with $T_{EGR,in}$ = 250°C

Thus, considering the flow conditions and the heat transfer phenomena for each element, the temperature variation in time was evaluated. Fig. 6 and 7 represent temperature variation found when the EGR gas flow reaches the valve at 250°C and 500°C respectively.

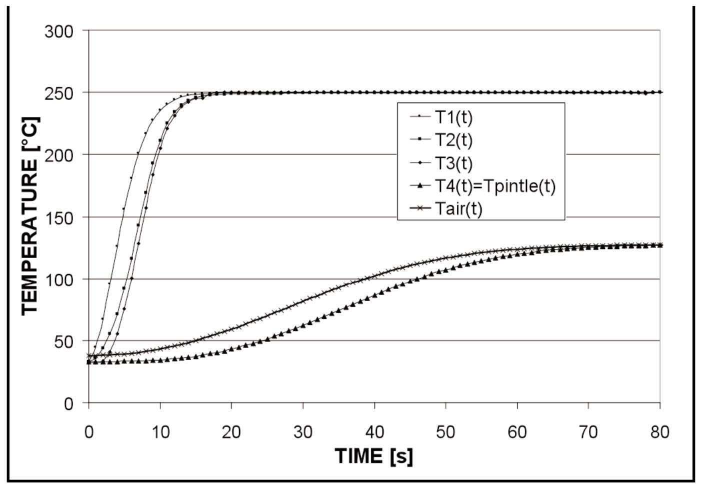

Fig. 7 – Test with $T_{EGR,in}$ = 500°C

Experimental set up and modus operandi

The experimental analysis on the thermal and endurance performance of the EGR valve has been developed carrying out the tests on a complete series engine bench. Tab. 3 shows a resume of the engine tests.

The tests were performed on a series Peugeot 1997 cm^3 displacement 4-cylinder Diesel engine, provided with a Siemens common rail system.
The EGR valve was a DC Motor driven valve equipped with 11 K-type thermo-couples (Ni-Cr and Ni-Al, [-270°C; 1350°C], 40.6 [µV/°C] at 25[°C], accuracy LT:2.2≈1.1[°C], HT:0.375≈0.75% [3 see Tab. 4]) monitoring the most critical locations inside and outside the valve body (LT: Low temperature range; HT: High temperature range).

Fig. 8 shows the thermo-couples location on the instrumented EGR valve. Two additional thermo-couples were located on the inlet and outlet tube as well.
The data acquisition during the engine test was done with a specific measuring system called Hydra, that kept real time data visible when running (useful to keep safe test elements) and recorded information for the post process activities [8].

An estimation of the measurement chain accuracy was done. This depended on the thermo couples used and on the measurement rate set up on the instrumentation. The set up was carried out using a fast acquisition rate. With K-thermocouples the accuracy was of ±1.47°C, as the measured range was of 20-350°C.

The engine tests

Fig. 9 shows the lay out used to perform the engine tests. The valve is located downstream

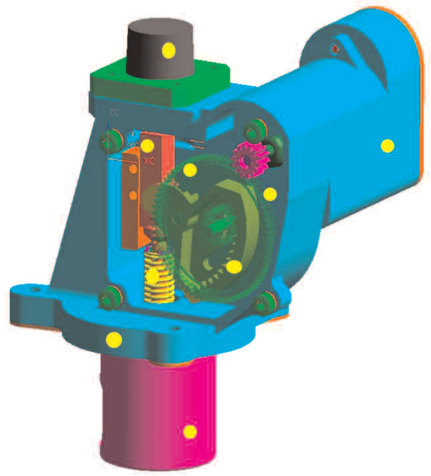

Fig. 8 – Thermo couples location on the instrumented EGR valve

the EGR cooler in the high pressure loop and the VGT was bypassed.

Two different kinds of tests have been carried out:
a) preliminary tests
b) fixed engine working point tests.

The a) tests were to check the response in terms of temperature reached in correspondence of different engine points; the b) tests were done to monitor real operation temperature and to explore the material resistance limits.
In order to check the thermal behavior of the EGR pintle, three specific tests have been carried out. Hereafter, the most important one for the pintle thermal behavior has been reported in Fig. 10. That shows the temperature distribution on the EGR components reached at the regime of 1.600 [rpm] - 40 [Nm] when the gas temperature was of 250°C. At the depicted engine working condition, the pintle temperature reached its steady value after about 2.000 seconds and this was due to the thermal inertia of the materials and to the heat transfer modes.

Point	$T_{EGR,in}$ [°C]	T_{pintle}[°C] experimental	T_{pintle}[°C] theoretical	Error [%]
A	250	134±2.57°C	144.5	7.3
B	125	67±2.57°C	77.4	13.4
C	500	N/A[3]	282	N/A

[3] This test was *not* carried out tested for experimental bench limitations

Tab. 4 – Comparison between the experimental data and the results from the steady state model

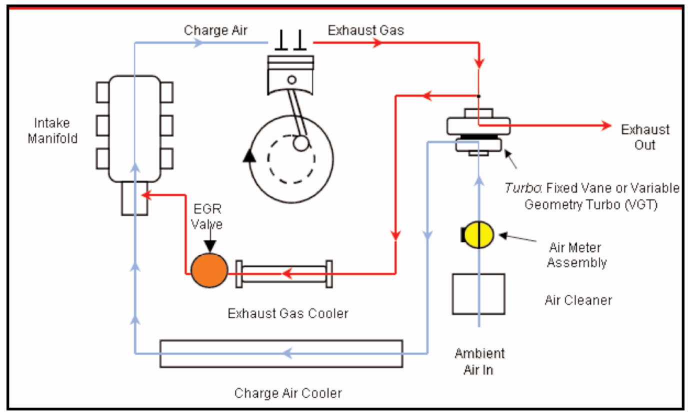

Fig. 9 – System lay out for the engine tests

The pintle temperature at the steady state remained below the maximum temperature to keep safe the rack (170°C). The temperature distribution on the valve was good to be compared with the data coming from the model application.

Experimental versus theoretical

After having reached the steady state regime, in order to validate the theoretical mo-dels, two engine working points have been used to validate results from the theoretical analysis and the experimental data, as the same boundary conditions were present. They were conditions in which the EGR gas temperature were respectively 250°C and 125°C.

Tab. 4 shows the results of the comparison on the pintle tip between the experimental

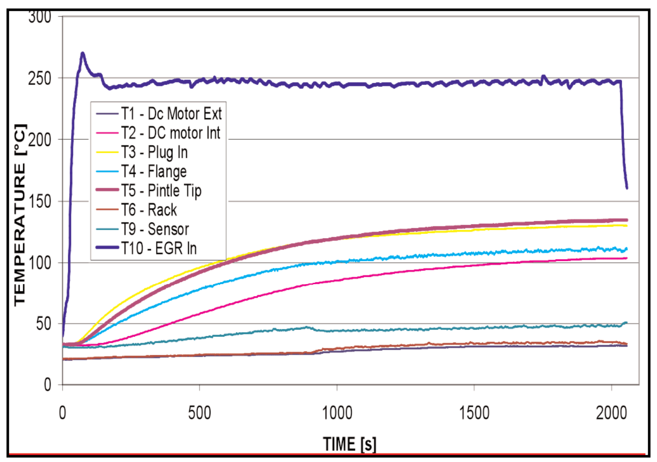

Fig. 10 – Temperature variation on the EGR components during the 1.600 [rpm] x 40 [Nm] test

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data and the theoretical model using the steady state approach.
In the same way, the comparison between the results from the unsteady state theoretical analysis has been done, as shown in Tab. 5. This comparison was made to verify if the parametric model can give a good approximation of the pintle temperature profile during the EGR operation.
The unsteady state analysis seems to forecast temperature values closer to the real ones than the steady state analysis. The percentage of error is actually lower than the previous.
A summary of the results reached with the two analyses done comparing with the experimental data is shown in Fig. 11.

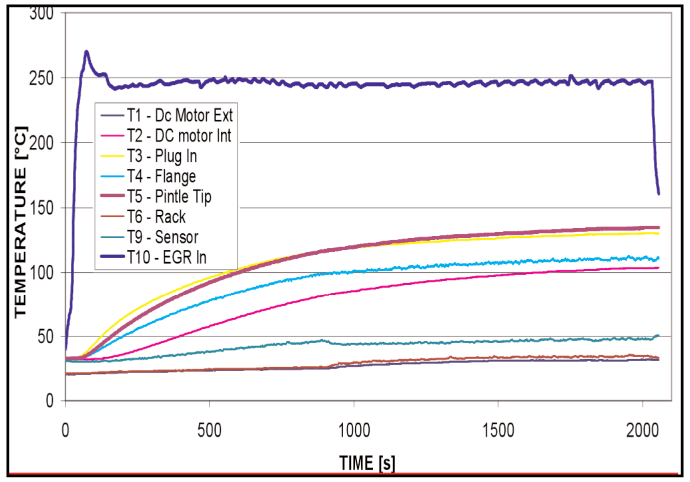

Fig. 11 – Comparison between the experimental, the steady and unsteady state models results

Tab. 5 – Comparison between the experimental data and the results from the unsteady state model

Point	$T_{EGR,in}$ [°C]	T_{pintle}[°C] experimental	T_{pintle}[°C] theoretical	Error [%]
A	250	134±2.57°C	127	-5.2
B	125	67±2.57°C	67.5	0.7
C	500	N/A	287	N/A

On the basis of the previous analysis, some solutions to reduce the temperature on the EGR components have been proposed. Tab. 6 shows the possible design propo-sals, regarding the EGR most critical components from the thermal point of view.
Furthermore, each design proposal must be evaluated regarding the constraints and the working conditions of the EGR valve employment. Of course, the feasibility will depend on the result of a careful cost-benefits analysis.

Tab. 6 – Design proposals for different EGR components

COMPONENT	DESIGN PROPOSALS
valve body	water cooling circuit
pintle	cross section reduction
	pintle finning
	pintle tip insulation
rack	new material
bushing	bushing cooling

Conclusions

In this work the thermal behavior of the EGR valve in a Diesel engine was examined. The identified thermal loads were sufficient to build a parametric model of the EGR valve and to get feasible temperature distributions in different working conditions, in steady and unsteady operation. The EGR critical points were explored with the valve installed in the hot side of the engine gas recirculation line.
The simplified model avoids making long experimental analyses and employing a complicated 3D geometry, which would give additional, but in this case dispensable information, lowering the design and test process. The engine tests validated the model, as a good match between experimental and theoretical was found.

Nomenclature

EGR	Exhaust Gas Recirculation
VGT	Variable Geometry Turbo
PPS	Pintle Position Sensor
DC	Direct Current
$\dot{Q}_{EGRFlow,in}$	Heat power entering from the EGR inlet
$\dot{Q}_{EGRFlow,out}$	Heat power entering from the EGR outlet
$\dot{Q}_{out,casting}$	Heat power spread from the valve casting
$\dot{Q}_{DCmotor}$	Heat power generated by the valve motor
$\dot{Q}_{ENGINE}$	Heat power coming from the engine
i	Pintle sub element index
$h^r_{,i}(t)$	Real convective coefficient
$h^f_{,i}(t)$	Fictitious convective coefficient
$h^{av}_{,i}(t)$	Average convective coefficient
T_o	Initial temperature
$T_{\infty,i}$	Flow temperature around the pintle sub element
T_{pintle}	Temperature between the valve pintle tip and the rack
h	convective heat transfer coefficient
A	element superficial area
V	total volume of the element
k	element thermal conductivity
ρ	density
λ	air thermal conductivity of the
ΔT_{cond}	temperature difference between two adjacent sub-elements
ΔT_{conv}	temperature difference between two adjacent sub-elements
d	distance between centres of two adjacent elements
$A^r_{,i}$	Pintle area involved in the real convective coefficient calculation
$A^f_{,i}$	Pintle area involved in the fictitious convective coefficient calculation
V_i	Pintle sub element volume
b_i	Pintle sub element Biot number
T_{EGR}	EGR flow temperature
$T_{air,\ body}$	Temperature of the air inside the EGR body

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