

Fouling of Exhaust Gas Recirculation Coolers



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A thesis submitted for the degree of
Master of Science in Mechanical Engineering to the
Faculty of Engineering, University of Porto

Porto, July 2017

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*“Das Gleiche läßt uns in Ruhe, aber der Widerspruch ist es, der uns
produktiv macht.”*

„Uniformity gives us serenity, but contradiction is what makes us
productive”

Johann Wolfgang von Goethe



Masterthesis

for Mrs. Sara Guerra, Porto, Portugal

CFD-Simulation of Fouling Effects in Diesel Engine EGR-Coolers

Fouling of Diesel engine exhaust-gas recirculation (EGR) coolers is an issue and requires adequate design of the cooler. Therefore, DEUTZ AG purchased comprehensive user-defined functions (UDFs) for the CFD-code ANSYS FLUENT from ATLANTING GmbH. With these UDFs it is possible to predict fouling tendencies of EGR coolers and in this way support the design process.

— Aim of this thesis is the examination and evaluation of these ATLANTING UDFs. First step shall be a literature study on this subject. After learning to work with the CFD-code ANSYS FLUENT, simulation results with the ATLANTING UDFs shall be compared with measured results from literature test cases. In order to better understand the fouling processes and the performance of the UDFs, single influences on EGR cooler fouling shall then be varied individually. All results shall be checked for physical plausibility. If possible, the final step shall be simulations of a real DEUTZ EGR cooler.

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Abstract

As the human species faces aggravating climate changes and the world's natural resources are being consumed at an incredibly fast rate, government policies struggle to find ecological and economical solutions that can prevent catastrophic outcomes.

In this scope, engine exhaust gas emissions are one of the biggest concerns in the current days, as the use of fossil fuels and non-renewable energy sources is taking a toll on our planet. Therefore, these legal policies are aimed to reduce the level of greenhouse gas emissions that contribute to the climate alterations.

The introduction of Exhaust Gas Recirculation (EGR) coolers in the functioning of diesel engines plays a very important role in reducing exhaust gas emissions and has for several decades been a theme of deep study. Contributing for the study of these components is DEUTZ AG, the company that manufactured the first motor engine, back in 1864.

This work focuses on a detailed study of the effect of fouling in Exhaust Gas Recirculation (EGR) coolers, its causes and the consequences that it has on the performance of these components. This analysis was achieved by the performance of simulations in the Computation Fluid Dynamics (CFD) Software *ANSYS Fluent*, with the contribution of C-routines provided by ATLANTING GmbH. These simulations allowed to conclude which factors have a greater impact in the functioning of these heat exchangers and also to assess the validity of the routines provided for this study.

This dissertation was carried out at DEUTZ AG in Cologne, Germany, within the framework of the Dissertation for the Master Degree in Mechanical Engineering at the Faculty of Engineering of the University of Porto (FEUP), Portugal, with a specialization in the field of Thermal Energy.

Resumo

A espécie humana tem vindo a enfrentar alterações climáticas cada vez mais agravantes e os recursos naturais do planeta têm vindo a ser consumidos a uma velocidade alarmante. Como tal, políticas governamentais têm sido desenvolvidas de modo a encontrar soluções ecológicas e económicas que possam prevenir resultados catastróficos.

Neste âmbito, as emissões de gases provenientes dos motores de combustão interna são uma das maiores preocupações atuais, uma vez que o uso de combustíveis fósseis e de recursos energéticos não renováveis tem um severo impacto no nosso planeta.

A introdução de EGR coolers no funcionamento dos motores a diesel desempenha um papel muito importante na redução de emissões de gases de combustão e tem vindo a ser tema de grande estudo nas últimas décadas. Um dos contribuidores para o estudo destes componentes é a DEUTZ AG, a primeira empresa a fabricar motores a combustão interna, em 1864.

Este trabalho foca-se no estudo detalhado do efeito do sujamento nos EGR coolers, as suas causas e as consequências que provoca no desempenho destes componentes. Esta análise foi conseguida através da realização de simulações com o software de CFD *ANSYS Fluent*, com a contribuição de rotinas em código C fornecidas pela empresa ATLANTING GmbH. Estas simulações permitiram concluir que fatores têm maior influência no funcionamento destes permutadores de calor e também discernir a validade das rotinas fornecidas para este estudo.

Esta dissertação foi realizada nas infraestruturas da DEUTZ AG em Colónia, na Alemanha, no âmbito da Dissertação para obtenção do grau de Mestre em Engenharia Mecânica, pela Faculdade de Engenharia da Universidade do Porto (FEUP), com uma especialização em Energia Térmica.

Acknowledgments

I would like to take an opportunity to express my gratitude to the following persons, who contributed for the achievement and completion of this dissertation:

First and foremost, to my parents and my brother, who made me the person I am today and always empowered and impelled me to pursue my dreams and ambitions; without whom I would not have had the chance to come abroad and fulfil my goals; for their unconditional support; for their joy to live and see me thrive.

To my close friends in Porto, whose support was remarkable, tireless and crucial, even miles apart; I am so glad to have you by my side.

To my *javalis*, the ones who walked alongside me for these last five years, which were, so far, the best ones of my life. To the memories I will never remember with people I will never forget. For all that we have shared, I am glad to have shared it with you; I wouldn't have had it with anyone else. It has been a hell of a ride!

To my German girls, for being the most welcoming and warm-hearted people I could have asked for; Köln acquired a special beauty thanks to you. To my Jule, for being so tireless and supportive; my everyday saviour.

I would like to express my utmost gratitude to Professor Dr. Carlos Pinho, whose guidance was the solid basis that allowed me to properly structure my work and do not lose sight of what is important; for his encouragement and dedication.

A special thanks to Dr. -Ing. Andreas Boemer and his team, from DEUTZ AG, for providing me with the chance to write my Master Thesis in an industrial context, and to Mr. Vedder from Atlanting GmbH; whose dedication and patience made this work possible and made me grow and overcome the obstacles I encountered.

A word of appreciation to Dr. -Ing. Peter Völk, for having the kindness to answer my many questions and for providing me with his thesis, which granted me a better foundation for the completion of my dissertation.

To the city of Porto, my hometown, for being a paradise by the ocean and the best place to have grown in; no matter how beautiful the world is, there is nothing like home.

Finally, a word of appreciation to the Faculty of Engineering, at the University of Porto (FEUP), which has been my second home throughout this entire journey, and has provided me with the chance to become an Engineer and to follow my dreams.

Obrigada!

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Nomenclature

Symbol	Designation	Units
A	Cooler cross-section area	m^2
C_c	Cunningham correction factor	
CFD	Computational Fluid Dynamics	
CHT	Conjugate heat transfer	
c_p	Specific heat capacity at constant pressure	$[\text{J}/(\text{kg K})]$
d_p	Particle diameter	$[\text{m}]$
D_p	Diffusion coefficient	
DPF	Diesel Particulate Filter	
EGR	Exhaust Gas Recirculation	
F_D	Drag force	$[\text{N}]$
F_L	Lift force	$[\text{N}]$
F_{th}	Thermophoretic force	$[\text{N}]$
F_V	Van der Waals force	$[\text{N}]$
F_W	Weight of the particle	$[\text{N}]$
HC	Hydrocarbon	
h_i	Convective heat transfer coefficient in the flow	$[\text{W}/(\text{m}^2 \text{K})]$
k	Thermal conductivity of a given material	$[\text{W}/(\text{m K})]$
Kn	Knudsen number	
K_{th}	Thermophoretic coefficient	
l	Molecular mean free path	$[\text{m}]$
L	Macroscopical physical length of interest	$[\text{m}]$
L_{cooler}	Cooler characteristic length	$[\text{m}]$
$\dot{m}$	Mass flow rate	$[\text{kg}/\text{s}]$
M	Molecular weight	$[\text{kg}/\text{kmol}]$
Nu	Nusselt number	
P	Perimeter	$[\text{m}]$
PM	Particulate Matter	
Pr	Mean flow Prandtl number	
$\dot{Q}$	Transferred heat flux	$[\text{W}]$
$Q(x)$	Cumulative particle distribution	
$q(x)$	Frequency particle distribution	
p_{vap}	Vapour pressure	$[\text{kPa}]$

Re	Reynolds number	
R_f	Thermal resistance of fouling layer	[m ² K/W]
r_p	Particle radius	[m]
Sc	Schmidt number	
St	Stokes number	
SFOC	Specific Fuel Oil Consumption	[g/kWh]
SST	Shear Stress Transport	
$T_{interface}$	Temperature at the interface of the deposit layer	[°C, K]
U	Overall heat transfer coefficient	[W/(m ² K)]
u	Particle velocity	[m/s]
V_D	Velocity of deposition due to eddy diffusion	[m/s]
V_{DP}	Velocity of deposition due to diffusiophoresis	[m/s]
V_t	Velocity of deposition due to impaction	[m/s]
V_{th}	Thermophoretic drift velocity	[m/s]
y_{gi}	Molar fraction at the interface of the deposit layer	
y^+	Dimensionless height of the first cell at the wall	

List of Greek Symbols

Symbol	Designation	Units
δ	Deposit layer thickness	[m]
ε	Soot deposition efficiency	[-,%]
ε_{cooler}	Effectiveness of the heat exchanger	[-,%]
$\varepsilon(d_p)$	Deposition efficiency for each particle diameter	[-,%]
ε_n	Deposition efficiency on the cooler length	[-,%]
$\lambda, \lambda_{fouling}$	Deposit layer thermal conductivity	[W/(m K)]
λ_p	Mean free path of gas molecules	[m]
λ_{tube}	EGR cooler tube thermal conductivity	[W/(m K)]
μ	Dynamic viscosity of the mean flow	[kg/(m s)]
ν	Gas kinematic viscosity	[m ² /s]
ρ	Density of the mean flow	[kg/m ³]
τ	Particle Relaxation time	[s]

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1. Introduction

Background

The development of the industry throughout the centuries has led to a stage where the engine industry is subdued to ever stricter gas emission laws, which lead the engine manufacturers to develop and enhance their engines' performances, where DEUTZ AG stands and is propelled to take action. Particularly in diesel engines, the requirements are high regarding the emission of nitrogen oxides and particulate matter.

The exhaust gas recirculation stands as a possibility to reduce NO_x emissions in order to abide the legal standards on its emissions. And to optimize the engine performance and increase its efficiency, the introduction of a cooling device becomes necessary, as the recirculated exhaust gas needs to be cooled before re-entering the cylinders. The increasing volume specific performance of modern engines requires a better cooling performance in increasingly smaller spaces (Morsch, 2016).

On the one hand, the cooling of the exhaust gas to be introduced in the engine leads to lower combustion temperatures, which decreases the amount of NO_x emissions. On the other hand, the high EGR rates will also lead to a higher production of soot components, so a compromise has to be met (Völk and Hörnig, 2011).

This device that cools down the exhaust gases – EGR cooler –, before they re-enter the engine, is a heat exchanger which commonly uses water as the cooling fluid. Due to the temperature gradient between the hot exhaust gases and the wall of the cooler, condensate forming species such as hydrocarbons and acids lead to the appearance of an adhesive isolative layer at the wall of the cooler, where particulate matter and ashes also get attached. This deposit layer reduces the efficiency of the EGR cooler throughout its lifetime and ought, therefore, to be studied (Hörnig, 2012).

The physical effective mechanisms that lead and contribute to the emerging of the isolative layer in this cooling device – the fouling effect - are studied in detail throughout this dissertation.

The study of these phenomena and the optimization of the performance of the EGR coolers has had a great contribution due to the Computational Fluid Dynamics (CFD) analysis. DEUTZ AG uses this type of software to develop its EGR coolers and improve the ultimate performance of its engines. In this ambit, this dissertation was developed at DEUTZ AG in Cologne, Germany, for the study of the fouling of EGR coolers through the help of the CFD software *ANSYS Fluent*.

Objectives

The primal objective of this dissertation was to analyse several C-routines provided by Atlanting GmbH and ascertain their validity and applicability at DEUTZ AG, as customized functions to associate with *Fluent* in a CFD analysis of the fouling of EGR coolers. This will be accomplished by testing the routines with several test cases from a detailed source for boundary conditions, having as a foundation a single basic geometry to validate the functions to be associated with the CFD simulation. Several more specific objectives can be outlined, for a better understanding of the conducting line of this thesis:

- Detailed study of the fouling phenomenon and the causes that lead to it or attenuate it along with the effects of its occurrence;
- Limitations and software features that improve the study of the particular case of heat exchangers;
- Characterization of the C-routines and their particular contribution for this work;
- Definition of a valid and simple geometry for simulation and validation of the routines in question for the assessment of fouling effects;
- Ascertaining all the boundary conditions necessary for the correct conduction of the simulations;
- Study of the parameters that play a big role in the final results, and how to faithfully replicate them, such as the particle distribution in the exhaust gas and empirical factors embedded in the code that alter the final results;

- Simulation of the test cases provided with several approaches regarding single variable variations in order to understand their behavior and influence on the evolution of the fouling effect.

The consequent results obtained from these simulations will be then compared to the experimental values from the sources provided, in order to estimate their accuracy and usability for the study of EGR coolers at DEUTZ AG.

Throughout the simulation phase the fluid used in the cooling process was always water; consequently, it will not be continually specified. The routines provided allow for the calculation of the final state of equilibrium, which represents the fouling effect under long term condition, that is, at an infinite length of time, where the time no longer plays any role. This is valid for the CHT and Fouling analysis, as later will be explained.

All the CFD simulations performed in the scope of this study were executed in the commercial Software *ANSYS Fluent, version 17.0*. The underlying objective of this thesis was to develop solid knowledge on CFD simulations and the behavioral analysis of fluid flows.

Thesis Outline

The outline of this thesis consists of nine main chapters: [1] introduction, [2] a brief review on the company where this dissertation was developed and the work achieved by it throughout the years and a literature review, [3] a thermodynamic analysis of the fouling effect, [4] an introduction to CFD and the parameters associated to it that are more relevant for this thesis, [5] a layout of the model used for the simulations and its prime features which allowed a correct replication of the values provided from the main sources, [6] the simulation parameters used and the ones that generated a greater influence in the final results, which were the biggest concern throughout this thesis, in order to accomplish viable results, [7] the display of the simulations performed and the respective achieved results, [8] the assessment of all the work done and final remarks, with an insight of what could the future work contemplate regarding the theme of this thesis.

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2. Literature Review

2.1. DEUTZ AG – Company Overview

DEUTZ AG takes the pride in being the first founded engine company, back in 1864, having developed the first combustion engine to be produced in a large scale: an atmospheric gas engine. Looking back on over 150 years of history, DEUTZ AG has produced cars, trucks, locomotives, tractors and, of, course, their engines (DEUTZ, 2017).

Nowadays it has specialized in the production of Diesel and Gas combustion engines for several fields of application, such as industrial and construction machinery, gensets, agricultural machinery, marine and automobile engines. Its engines' power output ranges from 12 to 500 kW, either with air, water or oil-cooling systems.

As the legislations regarding gas emissions become ever stricter, DEUTZ AG is put under pressure to develop better, more effective and more compact engines, emitting fewer emissions. For that to happen, the cooling system of the engines must be improved, and for modern combustion engines, smaller and more compact coolers with better performance are requested. This stands on the fact that a better cooling performance leads to a better usage of the heat produced by the engine (Morsch, 2016).

Therefore, one of the great investments DEUTZ AG undertakes is in the study and development of the coolers' performance with the aid of Computational Fluid Dynamics (CFD) software. In this line of thought, this work was a contribution for this process, as a study of the heat transfer and consequent fouling effect occurring in the EGR-Coolers for Diesel engines.

Although air used to be the main cooling fluid for combustion engines, today the state of the art are water-cooled engines. The advantages of the latter one are undeniable, as it provides a higher heat transfer coefficient and therefore, more heat can be dissipated from the exhaust gas; also, being beneficial when developing more compact coolers, as the water properties are much more suitable than air properties to endure the thermal strains occurring in this process.

2.2. EGR Coolers

2.2.1. European Legislation for Gas emissions

Diesel engine emissions are a major threat agent to the environment and human health, both in terms of global warming and carcinogenic action of these elements. This is especially critical concerning nitrogen oxides NO_x , CO_2 and particulate matter (PM). The common emissions from a Diesel engine are depicted in Table 2.1,

Exhaust gas component	At idle	At high load
Nitrogen oxide (NO_x)	50 – 100 ppm	600 – 2000 ppm
Hydrocarbon (HC)	50- 500 ppm	< 50 ppm
Carbon monoxide (CO)	100 – 450 ppm	< 300 ppm
Carbon dioxide (CO_2)	until 3.5 Vol.-%	about 12 Vol.-%
Water vapor (H_2O)	2 – 4 Vol.-%	until 11 Vol.-%
Oxygen (O_2)	18 Vol.-%	4 - 8 Vol.-%
Soot content - passenger vehicles	< 50 mg m^{-3}	50 – 120 mg m^{-3}

Table 2.1 - Main constituents of diesel exhaust gas (Hörnig, 2012, Völk and Hörnig, 2012)

Therefore, stricter limits regarding these emissions have been issued throughout the years, requiring the engine manufacturing industry a great effort to meet them, Table 2.2 (Walenbergh, 2003).

Year	Reference	CO	HC	NO _x	PM
2005	EURO IV	1.5	0.46	3.5	0.02
2008	EURO V	1.5	0.46	2.0	0.02
2013	EURO VI	1.5	0.13	0.4	0.01

Table 2.2 - EU Emission Standards for heavy-duty Diesel engines in g/kWh (Abd-Elhady *et al.*, 2011)

Comparing the current emission standard with EURO V, a 75% reduction in NO_x emissions has been demanded. In order to meet these strict demands, a technique to reduce NO_x emissions has since the 1970s been a subject of development: the Exhaust Gas Recirculation (EGR) (Jääskeläinen and Khair, 2016). This technique will be described in the following chapters.

Figure 2.1 shows the narrowing of the diesel emission limits in Europe, regarding a compromise between particulate matter and nitrogen oxides, for a high duty engine, as getting lower NO_x values means higher PM emissions.

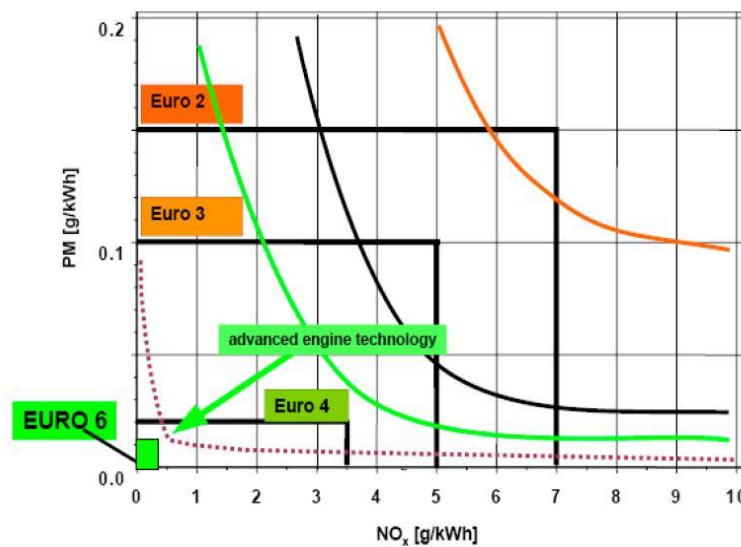


Figure 2.1 - Evolution of the Emission Standards in Europe, regarding PM and NO_x levels for Diesel HD engine (Johnson, 2008)

2.2.2. Technology Overview

As a highly populated and industrialized society, Europe faces increasingly stricter laws regarding gas emissions, as mentioned above. The most important gas pollutants of Diesel engines are nitrogen oxides (NO_x) and grime particles. In addition, most of the

NO_x produced by Diesel engines is thermal. Therefore, increasing combustion temperature will heighten the rate at which NO_x is produced. As a consequence, the temperature of the exhaust gases going into the cylinder must be reduced (Ghassemlou and Torkaman, 2016).

EGR is a widely-used technology to reduce NO_x emission levels. A portion of the inert exhaust gases is taken and mixed again with fresh air, forming a new mixture with an overall low oxygen concentration, decreasing the air-fuel ratio. This happens because EGR displaces oxygen and dilutes the intake air for the new upcoming cycle. It also increases the specific heat capacity of the mixture, leading to lower flame temperatures. However, the lower temperature and oxygen concentration caused by the EGR will lead to an increase in hydrocarbon and CO production, as they endorse slow combustion and partial burning (Kumar *et al.*, 2013, Lamas *et al.*, 2013, Priesching *et al.*, 2015).

For that matter and bearing in mind the products developed at DEUTZ AG, which are continually being studied and improved, EGR Coolers play a big role in meeting these emission limits. In order to achieve the European standards for gas emissions, the engine systems ought to be restructured and designed to reach optimal conditions of inlet temperature and pressure, time and amount of fuel injection, as well as quantity and timing of EGR input (Ghassemlou and Torkaman, 2016).

The main factors influencing NO_x emission levels are the concentrations of oxygen and nitrogen, alongside with local temperatures in the combustion process. Hereby the decrease in the combustion temperature will cause a remarkable reduction of NO_x levels (Lamas *et al.*, 2013).

Kumar *et al.* (2013) also concluded that even a small amount of EGR rate, about 15%, is enough to significantly reduce the NO_x emissions, without interfering with engine performance when it comes to thermal efficiency.

This thermal efficiency or just effectiveness can be easily determined, as it only depends on the temperature values. It is, therefore, defined as the ratio of the real heat transfer and the maximum possible heat transfer in a determined setup of working conditions (Sluder *et al.*, 2014),

$$\varepsilon_{cooler} = \frac{T_{gas,in} - T_{gas,out}}{T_{gas,in} - T_{coolant,in}} = \frac{\Delta T_{gas}}{\Delta T_{gas,max}} \quad (2.1)$$

Theoretically, the higher the amount of EGR used in a cycle, the steeper the NO_x reduction will be. However, in practice it is usually used no more than 50% of the EGR rate since particles in the exhaust gases will damage the engine. This can be seen in Figure 2.2, where NO_x lowers with increasing EGR rate.

The same does not happen to HC, CO and SFOC (Specific Fuel Oil Consumption), which keep continuously rising (Lamas *et al.*, 2013).

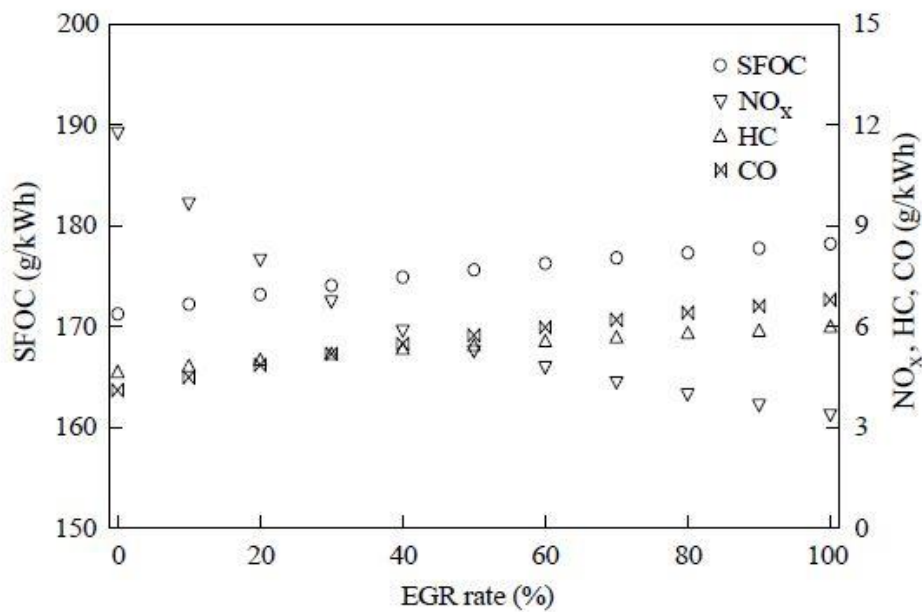


Figure 2.2 - SFOC, NO_x, HC and CO as a function of EGR rate (Lamas *et al.*, 2013)

In order to meet the NO_x emission requirements, the exhaust gases recirculated to the cylinders must undergo a heat transfer to lower their temperature. For this purpose, there is the EGR Cooler, which is going to be the focus of this dissertation, regarding its behavior and performance, as it will be later explained.

Another possible way to reduce NO_x is by delaying the injection timing. This change leads to lower peak pressures and, consequently, lower temperatures. It also means less burning time, therefore less NO_x is produced. It decreases the amount of fuel burnt before the peak pressure as well (Lamas *et al.*, 2013).

The emissions originated from the combustion process, which include hydrocarbons and soot, appear as a big problem for the performance of the EGR Cooler, as they can form a

layer on the cooler's walls, tampering with the heat transfer that is occurring (Priesching *et al.*, 2015).

On the other hand, it is noteworthy that the suspended carbon particles result from imperfect combustion and reduction of temperature (Ghassemlaglou and Torkaman, 2016).

There are two separate types of EGR in Diesel engines, Figure 2.3:

- Cool or high pressure EGR;
- Low pressure EGR.

The first, also known as short circuit EGR, is defined as a synthesis of high pressure exhaust gases and high-pressure air input. This method needs a variable Venturi with increased injection pressure and turbocharger usage. The exhaust gas is collected upstream of the turbocharger turbine, thus operating at boost pressures (Abd-Elhady *et al.*, 2011). The low pressure EGR is merely the synthesis of hot engine-out gases with air input before going through the turbocharger compressor (Ghassemlaglou and Torkaman, 2016). Therefore, this system works basically at atmospheric pressure. It has a great upside: it can take the EGR gas after the DPF (Diesel Particulate Filter), providing a recirculated flow free of particles and unburnt HC (Abd-Elhady *et al.*, 2011).

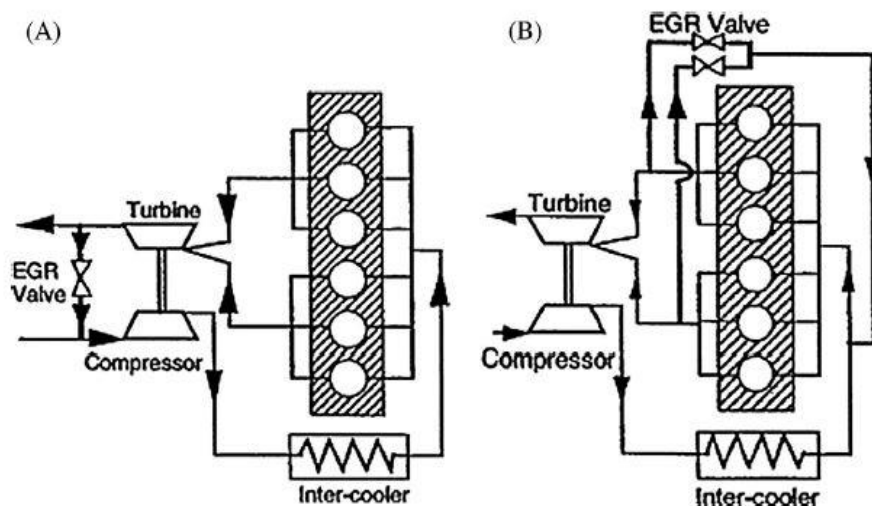


Figure 2.3 - (A) - High pressure EGR system; (B) - Low pressure EGR system (Ghassemlaglou and Torkaman, 2016)

It is more effective to use the cold EGR in order to achieve the NO_x emission reduction and to obtain minimum amount of increased specific fuel consumption. In addition, this type of EGR system obstructs any eventual sediment in the compressor

and intercooler. Besides, the reduction of input load temperature leads to a rise in combustion retardation and consequently induces a more homogenous synthesis between oxygen and the fuel steam (Ghassembaglou and Torkaman, 2016). Hence the use of EGR coolers.

2.3. Fouling

The EGR cooler's efficiency decreases during its lifetime. This is due to the fouling effect. Fouling can be defined as the deposition of particles or other components, originated from the exhaust gases of combustion, when they are recirculated through the EGR cooler (Priesching *et al.*, 2015). The driving force for the formation of EGR deposits is the temperature difference between the hot exhaust gases and the cold metal wall of the cooler, Figure 2.4. This causes particulate matter to deposit on the wall through thermophoresis and diffusion and the hydrocarbons to condense. This deposit forms a rather resistive layer that deteriorates the performance of the heat exchanger (Lance *et al.*, 2014, Sluder *et al.*, 2014). The formed deposit inhibits the heat exchange between the exhaust gas and the cooling fins, which is due to the addition thermal resistance this layer imposes. It can also cause a significant pressure drop in the cooler (Priesching *et al.*, 2015).

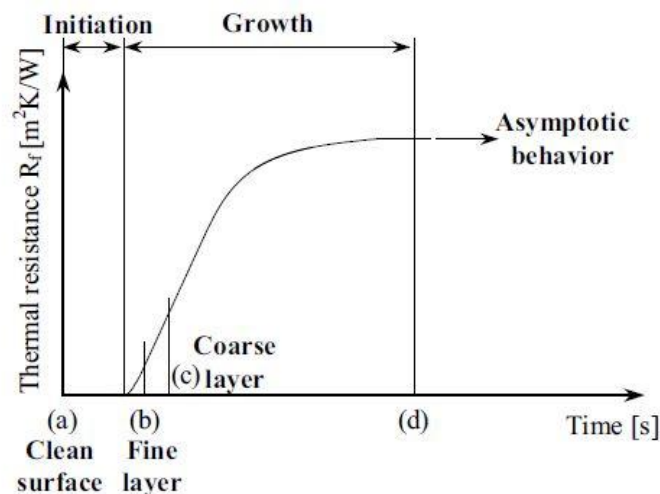


Figure 2.4 - Description of the thermal resistance evolution with fouling (Abd-Elhady *et al.*, 2011)

The fouling phenomenon is defined by a fast growth of the deposit at an initial stage and then it suffers a progressively slower growth, eventually reaching stable asymptotic conditions (Paz *et al.*, 2013).

The fouling to which the EGR coolers are subjected is so severe that it can reduce their thermal efficiency as much as 30% in a very short period of time. It is utterly important to underline that the deposit that is created in the cooler's surface is a mixture of particulate matter and unburned hydrocarbons, which are extremely hard to remove from the surface.

This layer is less thermally conductive than the cooler's metal, resulting in a lower heat transfer coefficient (Kumar *et al.*, 2013). Figure 2.5 displays a scheme of the fouling process with the phenomena involved and the particulates that play a role in it, as it will be later explained.

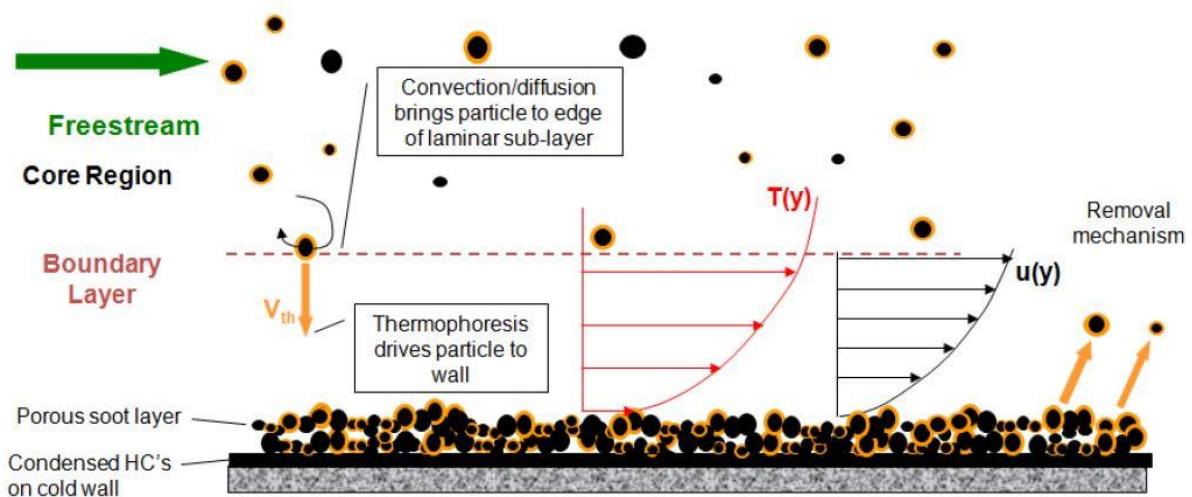


Figure 2.5 - Scheme of fouling process on the cooler wall (Ford, 2011)

The fouling has become even a greater problem throughout the years because the EGR coolers have improved: they have become smaller and more efficient, with a better performance and more economic due to packing. However, less space heightens the danger of plugging due to fouling (Kumar *et al.*, 2013). These authors refer that there is still too little knowledge about the thermal properties of the deposits formed and how they change with different engine conditions and with time. Therefore, there is still much more modelling to do in order to fully understand this phenomenon and to reduce its impacts.

Fouling occurs mainly due to the presence of (Sluder *et al.*, 2014):

- Soot;
- Condensable species (hydrocarbons and water vapor in the gas stream).

The soot deposits in EGR coolers are in fact particulate matter (PM) aggregates that form a rather porous deposit layer, with thermal conductivities of about 0.04 W/(m K) (could be compared to expanded polystyrene foam insulation) (Kumar *et al.*, 2013).

The soot particles (PM) are the most important component of the diesel exhaust gas concerning the effect of fouling, as they are a consequence of incomplete fuel burning inside the engine (Völk and Hörnig, 2012).

2.4. Fouling Mechanisms

2.4.1. Thermophoresis

Thermophoresis is a physical phenomenon in which, due to a temperature gradient, the suspended particles existing in a gas stream migrate in the direction of lower temperature. In the specific case of an EGR cooler, they migrate to the cooler's wall (Tsai *et al.*, 2004). This happens because the hot exhaust gas which is far from the wall has more energy content than the one near the wall; thermophoresis is by far the force that weighs the most on the particles (Atlanting, 2016).

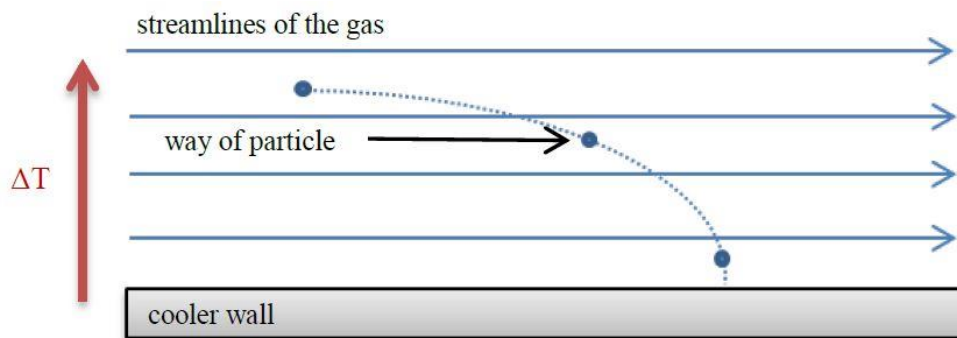


Figure 2.6 - Scheme of the thermophoresis effect (Priesching *et al.*, 2015)

This energy difference in the particles generates a force that moves the particles towards the cooling wall, and not along the streamlines, as it is shown in Figure 2.6 (Priesching *et al.*, 2015). After being attached to the wall, the particles remain so mainly due to Van der Waals forces (Han *et al.*, 2015). It is, by far, the dominant force for the deposition of EGR soot particles (Abarham *et al.*, 2010).

The thermophoretic force arises from the fact that hotter gas molecules display higher velocity, creating a net force toward the cooler area (Abarham *et al.*, 2009). The

transportation of particles is benefitted by a high temperature difference and low velocity of the flow of exhaust gas. There are two correlations commonly used to obtain the drift velocity due to thermophoresis (Abarham *et al.*, 2010, Warey *et al.*, 2012). They are the following:

Brock-Talbot-Correlation (Völk and Hörnig, 2011):

$$V_{th} = -K_{th} \frac{v}{T} \vec{\nabla} T = -K_{th} \frac{v}{T} \frac{\partial T}{\partial r} \quad (2.2)$$

being K_{th} the thermophoretic coefficient, defined as:

$$K_{th} = \frac{2C_s C_c}{(1 + 3C_m Kn)} \cdot \frac{\frac{k_g}{k_p} + C_t Kn}{1 + \frac{2k_g}{k_p} + 2C_t Kn} \quad (2.3)$$

and the respective Cunningham correction factor, C_c , described as follows:

$$C_c = 1 + Kn \left(A + B e^{\frac{-C}{Kn}} \right) \quad (2.4)$$

All the other coefficients A , B , C_m , C_s , C and C_t are empirical and respectively: 1.257, 0.4, 1.1, 1.14, 1.17 and 2.18.

Modified Cha-McCoy-Wood (MCMW) Correlation: instead of determining the thermophoretic velocity, this method introduces an equation to obtain the thermophoretic force (Völk and Hörnig, 2011):

$$F_{th} = 1.15 \frac{Kn}{4\sqrt{2}\alpha \left(1 + \frac{\pi_1}{2} Kn\right)} A = \pi r^2 \left[-\exp\left(-\frac{\alpha}{Kn}\right) \right] \left(\frac{4}{3\pi} \theta \pi_1 Kn \right)^{\frac{1}{2}} \frac{k_m}{d_m^2} \nabla T d_p^2 \quad (2.5)$$

where:

$$\pi_1 = 0.18 \frac{\frac{36}{\pi}}{(2 - S_n + S_l) \frac{4}{\pi} + S_n} ; \alpha = 0.22 \left[\frac{\frac{\pi}{6} \theta}{1 + \frac{\pi_1}{2} Kn} \right]^{\frac{1}{2}} \quad (2.6)$$

and

$$\theta = \frac{\epsilon_G - \epsilon_{tl}}{\epsilon_G + \epsilon_{tl}} \quad (2.7)$$

$$\epsilon_G = 15 \text{ (Graphite)} \quad (2.8)$$

$$\epsilon_{tl} = 1.00059 \text{ (Dry air)} \quad (2.9)$$

With the previous equations, the thermophoretic drift velocity can be obtained with:

$$V_{th} = \frac{F_{th}\tau}{m_p} = \frac{F_{th}\tau}{\rho_p \pi d_p^3 / 6} \quad (2.10)$$

knowing that τ is the particle relaxation time:

$$\tau = \left(\frac{\rho_p d_p^2 C_c}{18\mu} \right) \quad (2.11)$$

The determination of K_{th} is facilitated by the assumption of ideal spherical particles as a function of the **Knudsen number**, being this latter one defined as the dimensionless ratio between the molecular mean free path l and any macroscopic physical length of interest L for a gas, as follows:

$$Kn = l/L \quad (2.12)$$

The Knudsen number assesses the flow regime of a gas around a particle. Therefore, in the interval where $Kn \ll 1$ the gas behaves by the Navier-Stokes equations – a *free-molecular regime* –, whilst where $Kn \gg 1$ it obeys rarefied gas dynamics – a *continuum regime* (DeCarlo *et al.*, 2004). This means that this number helps to identify the formulation to be used to model the situation: either continuum mechanics or fluid mechanics (Trusler, 2011).

When the Knudsen number is very small, intermolecular collisions are dominant and the kinetic approximation becomes valid. On the other hand, when this number has a large value, the dominant effect are the collisions with the boundaries, as the molecules move almost freely (Bardos, 2011).

According to Abarham *et al.* (2010) and Völk and Hörnig (2011), when the particles are small under a $Kn = 2$, the Brock-Talbot equation deviates from the experimental values, which is the point where the particle diameter equals the mean free path of gas molecules λ_p , as according to Equation (2.13).

$$Kn = \frac{2\lambda_p}{d_p} \quad (2.13)$$

This can be corroborated by the following graphic:

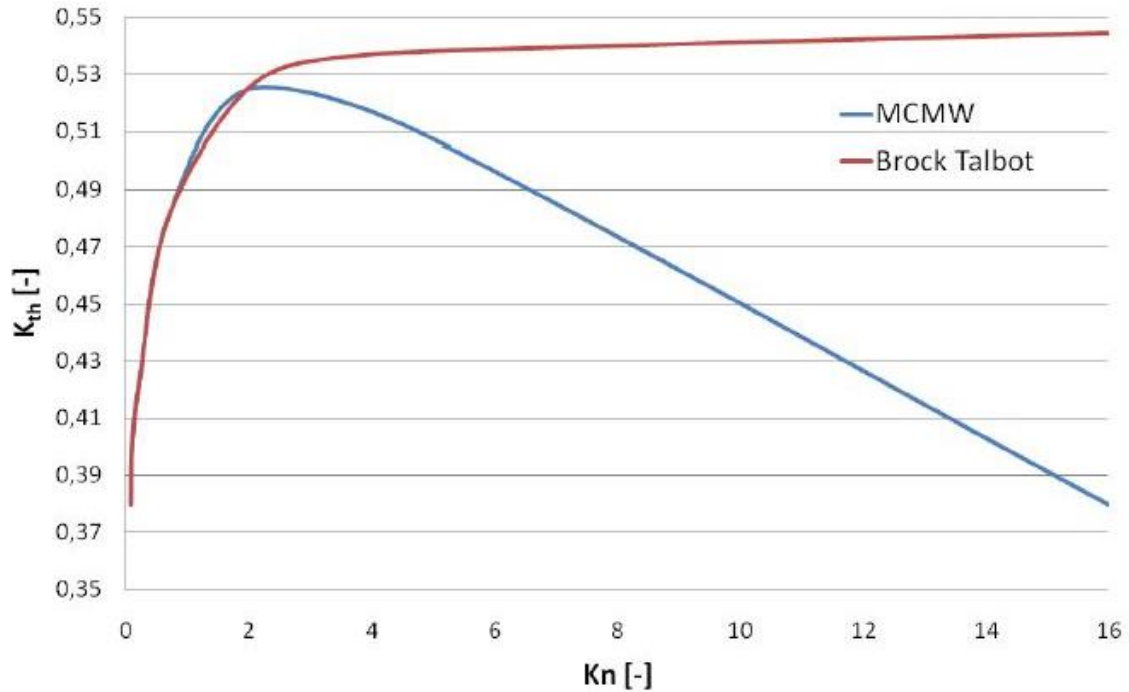


Figure 2.7 – Two correlations between the Knudsen number and thermophoretic coefficient (Völk and Hörnig, 2011)

This graphic shows the thermophoretic coefficient as a function of the Knudsen number, displayed for both mathematical models. From $Kn = 2$ onwards, the correlations evolve differently. Therefore, it is possible to determine the thermophoretic coefficient in the entire particle size spectrum, by combining these two methods.

When $Kn < 2$ the Brock-Talbot-Correlation is applicable, and when $Kn > 2$ the MCMW-Correlation is used (Völk and Hörnig, 2011).

As seen in Equation (2.13), the particle diameter has an influence in the thermophoretic deposition. The smaller the particle, the higher will be the Knudsen number, thus decreasing the absolute thermophoretic velocity. This validates the statement that smaller particles have more energy than bigger particles, allowing them to proceed with the flow instead of depositing on the wall.

2.4.2. Diffusiophoresis

Besides thermophoresis, diffusiophoresis also plays a big role regarding the EGR cooler fouling. They are analogous concepts since the diffusiophoresis is a phenomenon that takes place when a particle is placed inside a solution with different macroscopic concentration levels: the particle will migrate to the area of higher or lower concentration, depending on its charge and on the solution charge (Anderson and Prieve, 1984).

The migration of the particles is induced by the concentration gradient caused by the condensed components, which reduce the concentration near the wall (diffusion) (Atlanting, 2016). These condensed components emerge when the temperature of the cooler wall is lower than the dew point of the species involved (at its own partial pressure). The condensation can be either of water vapour, acid or organic if it involves hydrocarbon (Han *et al.*, 2015). During the movement towards the wall by the water vapour or HC unburnt molecules, they mix together with soot particles, diverting these towards the cooler wall as well. Therefore, it can be stated that the diffusion effect is enhanced by the diffusiophoresis (Völk and Hörnig, 2011, Völk and Hörnig, 2012).

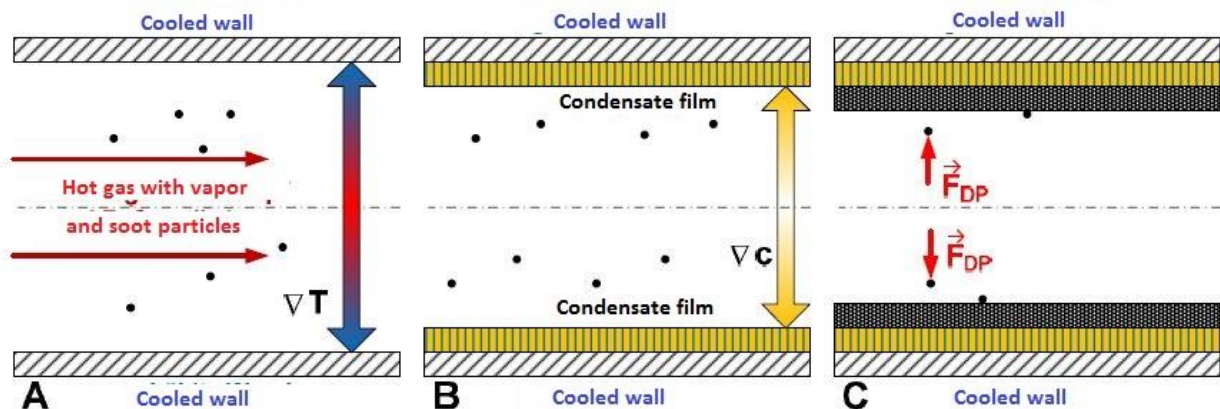


Figure 2.8 - Scheme of the diffusiophoresis caused by a concentration gradient (adapted) (Völk and Hörnig, 2012)

Conditions such as low coolant and gas temperatures enable diffusiophoresis to act. If high condensation rates are to occur, they exacerbate the effect of diffusiophoresis. In fact, when they are too high they can even cause as much particulate deposition as thermophoresis (Atlanting, 2016). This is evidenced by the Equation 2.14 (Atlanting, 2016), that describes the velocity of deposition due to diffusiophoresis:

$$V_{DP} = \frac{1}{x_{vapor} \cdot \sqrt{M_{vapor}} + x_{gas} \cdot \sqrt{M_{gas}}} \cdot \left(\frac{R \cdot T_{gas}}{\sqrt{M_{vapor}}} \cdot \frac{\dot{m}_{condensate}}{A_{tube,i,all}} \right) \quad (2.14)$$

However, it is stated that thermophoresis is more dominant than diffusiophoresis regarding a particle range of $r_p < 1 \mu m$, being r_p the radius of the particle present in the flow (Carstens and Martin, 1982).

2.4.3. Condensation

Water vapor condensation

A lower wall temperature will lead to water vapor condensation at the cooler wall on the gas side, once it allows the decrease of flow temperature below the water dew point. The concentration gradient it provokes enables the diffusion of soot particles towards the wall, in the areas of the cooler where the water vapor condensed. The current density of particles diverted to the wall is directly proportional to the current density of water vapor condensate (Völk and Hörnig, 2011).

HC and SO₂ condensation

The sulphur contained in the diesel exhaust gas is transformed into SO₂ during the combustion and, depending on the amount of sulphur in the diesel fuel, until 3% of this oxide may be transformed into SO₃. This compound will easily react with the water vapor present in the exhaust gas, creating the sulphuric acid H₂SO₄. When the temperature inside the cooler goes below the dew point of this acid, then condensation occurs. The unburnt hydrocarbons remaining from the combustion process, due to their high boiling point, are also a major agent contributing to the deposition of soot when they condense, increasing the Diffusiophoresis effect (Völk and Hörnig, 2011).

Besides, two types of condensation can be distinguished (Hörnig, 2012):

- Film condensation: happens mostly in clean surfaces. A cohesive liquid film is formed at the cooled wall;
- Drops condensation: single condensation drops with the ability to move are formed on the surface, and become bigger with progressing condensation process. They occur predominantly on dirty surfaces, which inhibit full moistening and thus, creation of a condensation film.

Through this mechanism a sticky layer emerges, in which the soot particles get embedded, thus promoting the rise of the fouled deposit. The condensation effect decreases with the rise in deposit thickness as the temperature at the wall – now upper layer of the deposit – rises, and the conditions to promote condensation no longer verify.

There are other mechanisms that are also responsible for fouling, such as: particulate sedimentation caused by gravity, solidification and spinning; electrostatic precipitation, interception and impaction (Völk and Hörnig, 2012, Kumar *et al.*, 2013).

2.4.4. Impaction

This phenomenon consists of the acting of the moment of inertia on the particles whenever the streamlines change their direction, for example, when hitting a surface. Its influence is higher on bigger particulates, which then do not have enough energy to move along with the flow (Atlanting, 2016). Therefore, both the velocity of the flow (turbulence is relevant) and the geometry of the EGR cooler have a large influence on the deposit by impaction (Völk and Hörnig, 2011). This effect is especially effective for particles bigger than 100 nm. Small particles have short relaxation times and, therefore, follow the flow instead of depositing.

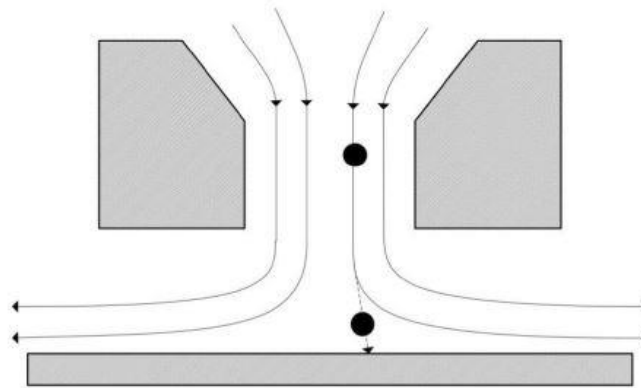


Figure 2.9 - Scheme of deposition due to impaction
(Atlanting, 2016)

In order to determine the contribution of the impaction effect on the deposition rate, the Stokes number St needs to be assessed, as it characterizes the behavior of particles suspended in a fluid flow (Völk and Hörnig, 2011):

$$St = \frac{\rho_p \cdot d_p^2 \cdot u_g \cdot C_c}{9 \cdot \mu \cdot d_c} \quad (2.15)$$

This number depends on the density and diameter of the particle, the velocity of the flow, the kinematic viscosity of air, the Cunningham correction factor C_C and the characteristic dimension of the flow deviation/obstacle. When $St > 1$, the impaction effect has a considerable impact on particle deposition (Völk and Hörnig, 2011).

The corresponding deposition velocity due to impaction is obtained by:

$$V_t = 4.5 \times 10^{-4} u^* \left(\frac{\tau}{(\nu/u^{*2})} \right)^2 \quad (2.16)$$

2.4.5. Diffusion

This is a rather weak deposition mechanism that can occur when the particles suffer a change in trajectory due to the Brownian movement. The diffusion is the transportation process for particles through the influence of a concentration difference.

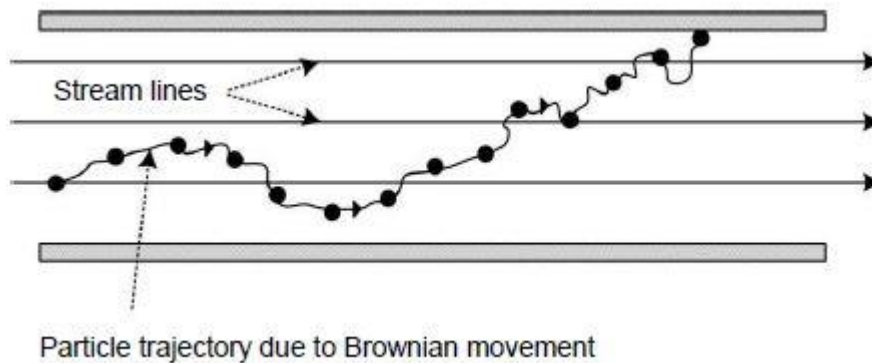


Figure 2.10 - Scheme of deposition through diffusion (Atlanting, 2016)

For a given gas flow rate, the particle diffusion is independent from the diameter of the tube: what is crucial is the time it takes for the particle to be transported to the cooler wall. In conclusion, with increasing flow rate, if the transportation time decreases, so does the deposition through diffusion (Völk and Hörnig, 2011).

Small particles, with diameter size of 100 nm and less, can be easily moved toward the wall by the effect of eddy diffusion. Its deposition velocity can be described by the following equation, for turbulent flows (Abarham *et al.*, 2010):

$$V_D = 0.057u^*Sc^{-\frac{2}{3}} \quad (2.17)$$

which is dependent on the particle Schmidt number Sc , defined as the ratio of gas kinematic viscosity to the particles diffusion coefficient:

$$Sc = \frac{\nu}{D_p} \quad (2.18)$$

This mechanism too is dependent from the particle's diameter, through the diffusion coefficient:

$$D_p = \frac{k_b \cdot T \cdot C_c}{3\pi \cdot \mu \cdot d_p} \quad (2.19)$$

Therefore, as smaller particles (10 - 50 nm) have lower Schmidt numbers, the higher diffusion velocities occur in this particle range, being this mechanism more effective in these particles rather than in bigger particles.

However, diffusion is a rather weak deposition mechanism, as the diffusion velocity for small particles (smaller than 50 nm) has one less order of magnitude than the thermophoretic deposition velocity, and it is not at all comparable with the thermophoretic velocity for bigger particles (Abarham *et al.*, 2010).

Even though these last phenomena can also occur, it can be stated that fouling is mainly caused by (Lance *et al.*, 2014):

- Thermophoresis (due to particulate matter);
- Diffusiophoresis (hydrocarbon condensation).

This can be corroborated by the following figure from Abarham et al. (2010), a comparison of the deposition velocities, Figure 2.11:

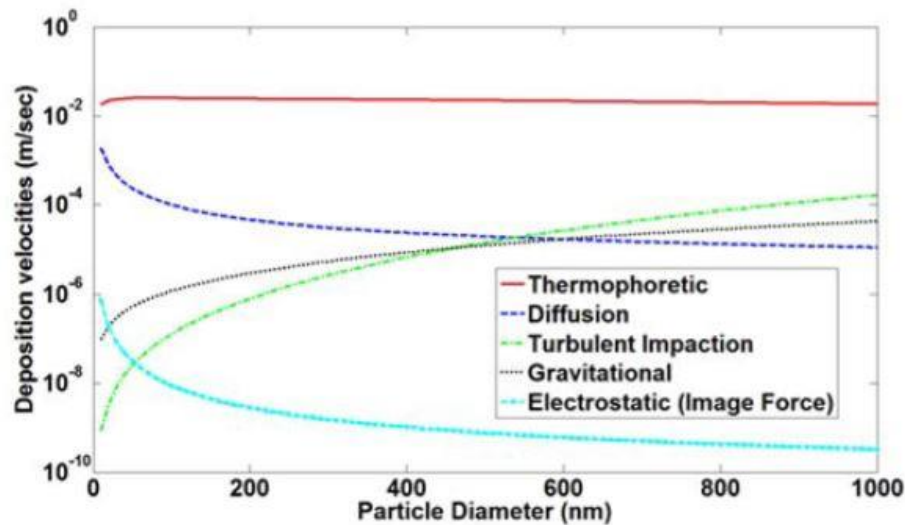


Figure 2.11 - Comparison of deposition velocities for submicron particles at 600K

2.5. Influence of Flow State

Besides the deposition mechanisms, also the conditions of the flow take a toll on the deposition efficiency, that is, whether the flow is laminar or turbulent. In laminar conditions, the mechanism that has bigger impact on particles smaller than 100 nm is the Brownian diffusion, contrarily to the impaction, that increases its influence with larger particle sizes. Therefore, there is a transition area where both effects have an impact, but none is more dominant than the other (Hörnig, 2012).

As for a turbulent state, there is a greater deposition of particles because the particle mass flow increases with increasing Reynolds number. Nonetheless, a higher velocity can also lead to a reduction of the deposition, since the particles take less time to flow through the cooler.

Bigger particles ($d_p > 100 \text{ nm}$) are also more easily deposited in turbulent conditions as they have more inertia, which can prevent them from following the flow and, thus, depositing on the cooler wall. Adding to this, also smaller particles ($d_p < 100 \text{ nm}$) have a higher deposition efficiency than in the laminar state, as the characteristic turbulent behavior may lead them directly to the wall (Hörnig, 2012).

2.6. Deposit Characteristics and Fouling Consequences

The stricter the emission regulations are, the higher are the needed flow rates and cooling stages. Therefore, the greater is the fouling effect and the deposition of soot (Kumar *et al.*, 2013). The soot particles are usually very small primary particles that will later evolve to fractal agglomerates, due to sticking and intertwining to each other. These agglomerates are very prompt for the adsorption of hydrocarbons (Han *et al.*, 2015). The deposit density is a great influence factor in the heat transfer process, because it affects both the deposit thermal conductivity and the deposit thickness. Usually in most conditions for EGR fouling, the non-volatile species (soot particles) occupy the most part of the space, whereas the condensed volatile species amount to just a small part of the total volume. Therefore, their influence on the densification effect is rather small, not displaying a significant change on the deposit properties (Sluder *et al.*, 2014).

Sluder *et al.* (2014) concluded with their study that the thermal resistance increases at a fairly constant rate with the deposit mass as this latter one develops. Since the thermal resistance displays a linear relation with the deposit thickness, it can be stated that the temperature at the interface between gas and deposit comes as a function of the deposit thermal conductivity, deposit thickness and the rate for heat transfer occurring in the deposit.

Fouling can lead to one of the following outcomes:

“Stabilization” of the Effectiveness

This phenomenon occurs when the deposit is mainly made up of soot, with minor deposition of HC; when the stable point is reached there will be no further degradation in effectiveness. To understand this concept, it is necessary to establish the definition of effectiveness: It is known as the temperature gradient undertaken by the gas from the inlet to the outlet of the cooler, divided by the maximum gradient of temperature the gas could have experienced.

Bearing in mind that Newton’s Law of Cooling (Equation 2.20) sets the relationship between inlet and outlet temperatures and the overall heat transfer coefficient U , it is known that the effectiveness is connected to this latter coefficient in a non-linear way. This relationship is a consequence of the exponential decay of temperature gradient

between gas and coolant. P is the inner perimeter and L is the length of the tube, c_p is the specific heat of the gas at constant pressure, and $\dot{m}$ is the mass flow rate. At last, ΔT_o and ΔT_i are the temperature differences between gas and coolant at the outlet and inlet, respectively (Sluder *et al.*, 2014).

$$\frac{\Delta T_o}{\Delta T_i} = \exp\left(-\frac{P \cdot L_{cooler}}{\dot{m} \cdot c_p} \cdot U\right) \quad (2.20)$$

Consequently, stabilization can happen even in the presence of a finite temperature gradient, that is, a thermophoretic driving force is still in place. However, it may seem to have stabilized but in truth it continues to rise, although at a lower rate. This is due to the reduced deposition rates that result in relatively small gains in thermal resistance with time (Sluder *et al.*, 2014).

With the increase in deposit thickness, the surface temperature increases due to the inability of the low thermal conductivity deposit to transfer the heat from the gas to the metal. This reduces the temperature difference between the gas and the deposit surface thus reducing the rate of thermophoretic deposition. Therefore, a higher deposit thermal conductivity results in a greater deposit thickness, for a given surface temperature rise (Lance *et al.*, 2014). When this above-mentioned thickness exceeds the available space in the cooler gas channel, a plug is formed.

Plugging

Plugging occurs when the deposit becomes too thick to allow the flow of the exhaust gas through the cooler; this happens when the deposit thermal resistance is lower due to the existence of large amounts of condensed species. In this case, the deposit will not experience a measurable growth in temperature as it gets thicker (Sluder *et al.*, 2014). The effect of plugging is much more frequent when the engine is subjected to a cold environment or to light loads. It is also more common in closely-spaced internal fins. In this phenomenon both the volatile portion of the deposit and the deposition method play a big role (Sluder *et al.*, 2014).

According to the experiments performed by Lance *et al.* (2014), in several coolers with different characteristics of the exhaust gases, they observed the following result on a

cooler whose flow had high fraction of volatile species and practically no soot, with already half-lifetime use.

The volatile species of interest for this phenomenon are:

- Unburned hydrocarbons;
- Water;
- Other species such as organic acids and anhydrides that are a result of partial combustion of fuel-related hydrocarbons.

The above-mentioned species are drawn to the cooler interface by diffusion instead of thermophoresis. As seen before about the diffusiophoresis, if the surface of the cooler can adsorb or condense a meaningful amount of the volatile species, then it will cause a concentration gradient which will lead more species to diffuse towards the interface. It can be stated that there is relation of proportion between the diffusion rate and the

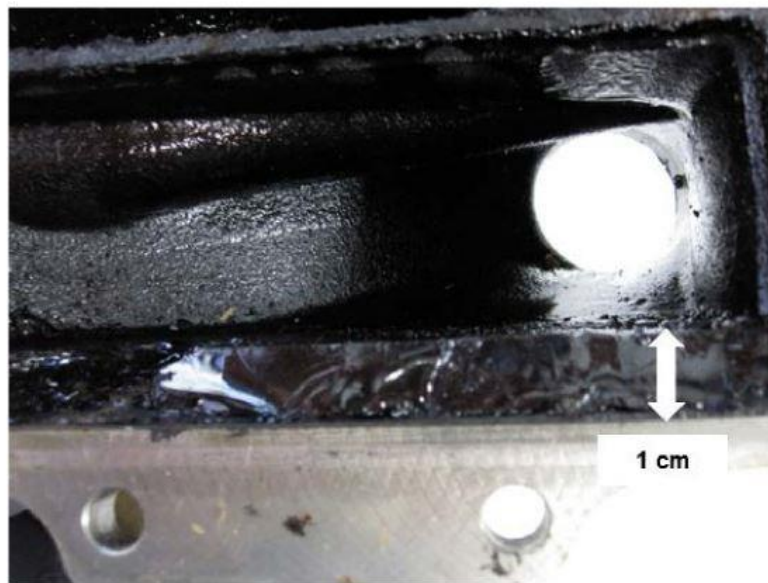


Figure 2.12 - Result of the plugging effect with a lacquer-like deposit for a half-lifetime used cooler (Lance *et al.*, 2014)

concentration gradient (Sluder *et al.*, 2014).

The thickening of the deposit will cause the diffusion to slow down, as it increases the distance of the porous layer, where the volatile particles must diffuse in order to reach the cooler cold wall. This will induce higher concentration of volatile species at the

deposit surface. However, it only happens as long as the deposit's temperature rises (Sluder *et al.*, 2014).

Also, according to Kumar *et al.* (2013), in the experiences they have conducted they observed that the deposited soot layer in single-tube devices is much larger when there is a high flow rate condition, that is, a turbulent flow. Also, the thickness of the deposit was much larger, restricting the flow area by up to 75% over a 4-diameter length.

2.7. Removal Mechanisms

Firstly, in order to understand what kind of causes can lead to removal of the deposit layer, it is essential to understand what forces the particles are subjected to, as displayed

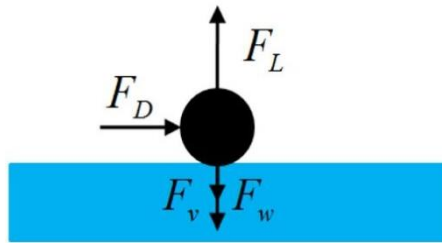


Figure 2.13 - Scheme of the forces acting on a particle sticking to the surface (Abarham *et al.*, 2010)

in Figure 2.13. The previous balance is composed by: drag force F_D , lift force F_L , weight of the particle F_W , and the Van der Waals force F_V . These forces are defined as:

$$F_D = 8\rho v^2 \left(\frac{u^* d_p}{v} \right)^2 = 8\rho v^2 Re_p^2 \quad (2.21)$$

$$F_L = 0,076 \rho v^2 \left(\frac{d_p u^*}{v} \right)^3 \quad (2.22)$$

$$F_W = \frac{\pi}{6} (\rho_p - \rho) g d_p^3 \quad (2.23)$$

$$F_V = \frac{A_H d_p}{12Z_0^2} \quad (2.24)$$

Regardless of any other technical assumptions, it is shown that particulate fouling can be reduced through (Kumar *et al.*, 2013):

- Increased gas velocity;
- Low temperature difference between inlet gas and coolant, that is, preventing excessive cooling of the exhaust gases, especially at the start of the operation;
- Complex cooler geometry that would rise the local area for heat transfer;
- Using DOC (Diesel Oxidation Catalyst) together with DPF (Diesel Particulate Filter).

According to Han et al. (2015), only the heavier hydrocarbons present in the fuel can be characterized by having dew points in a range relevant to EGR cooler fouling, which allows to conclude that only a small part of HC will contribute to the deposit formation. Nonetheless, when high levels of HC are in order, the result will be a very wet soot, which is not affected by any removal mechanisms, potentially leading to plugging.

The approach of a final state of equilibrium is related to the reduction of the cooler cross-section and, consequently, the increase in flow velocity, which causes a rise in the flow shear stress. As previously referred, higher turbulence in the narrower tube improves the heat transfer. Furthermore, the increase in heat transfer leads to a higher surface temperature, which reduces the condensation rates on the upper layer of the deposit. Therefore, with increasing thickness, fewer hydrocarbons can be deposited, inhibiting the fouling process (Völk, 2014).

If the emerging removal mechanisms are characterized by a mass removal rate proportional to the total deposit mass, then an exponential approach to the steady state value occurs, meaning that the removal rate matches the deposition rate (Abarham *et al.*, 2010). Here follow the main removal mechanisms for the deposition of particles:

2.7.1. Condensation

This phenomenon can occur whenever the following condition applies: the mole fraction of the component (water, hydrocarbons, sulphuric acid) contained in the inlet gas is of higher value than on the deposit layer, as shown in equation (2.25), being y_{gi} the mole fraction at the interface:

$$y_{gi} = \frac{P_{vap}}{P_{total}} \quad (2.25)$$

and the vapor pressure values can be determined through the Antoine coefficients (Abarham *et al.*, 2010):

$$\log_{10}P_{vap} = AA - \frac{BB}{T_{interface} + CC} \quad (2.26)$$

with the following values for the coefficients (Hörnig, 2012):

	<i>AA</i>	<i>BB</i>	<i>CC</i>
H ₂ O [kPa, °C]	7.07406	1657.5	227.02
H ₂ SO ₄ [atm, K]	16.259 ± 0.437	10156 ± 175	0
n-Dodecan [bar, K]	4.10549	1625.928	-92.839
n-Hexadecan [bar, K]	4.17312	1845.672	-117.054

Table 2.3 - Coefficients of the Antoine equation for the four condensate forming species

Water has the ability to permeate the porous layer and induce separation of the deposit from the cooler; it can create a wash-out effect. However this is only significant when the deposit layer does not contain an excessive amount of condensed hydrocarbons (Han *et al.*, 2015). It seems that water is more critical and effective in removing particles through flaking, rather than other organic solvents (Abarham *et al.*, 2010).

2.7.2. Gas flow shear stress

This is not a decisive mechanism for the removal of deposit, but it still may take a toll, nonetheless. The typical range for the velocity in EGR coolers is between 10 to 30 m/s. However, for particle removal to happen, the gas velocity would have to be around 42 m/s (and 0,03 kPa of shear stress), as shown in the studies held by Han *et al.* (2015). This suggests that removal would not happen under normal operating conditions.

In addition to this, Han et al. (2015) show that the particle size with the peak number for removal through shear stress is around 200 nm. As the normal diesel size particle distribution ranges from 10 to 300 nm, one can conclude that only the larger particles are prone to suffer removal by this mechanism, as demonstrated on Figure 2.14. The critical flow velocity – the main stream velocity above which rolling of the deposited particles will take place, avoiding the fouling effect – is a function of particle size and it decreases greatly with increasing particle size, being nearly too high to reach for the smaller particles (in the order of magnitude 10^{-8}) (Abd-Elhady *et al.*, 2011). As the shear stress is directly influenced by the density of the fluid, it is clear that this mechanism has a greater effect on liquids rather than on gases (Völk, 2014).

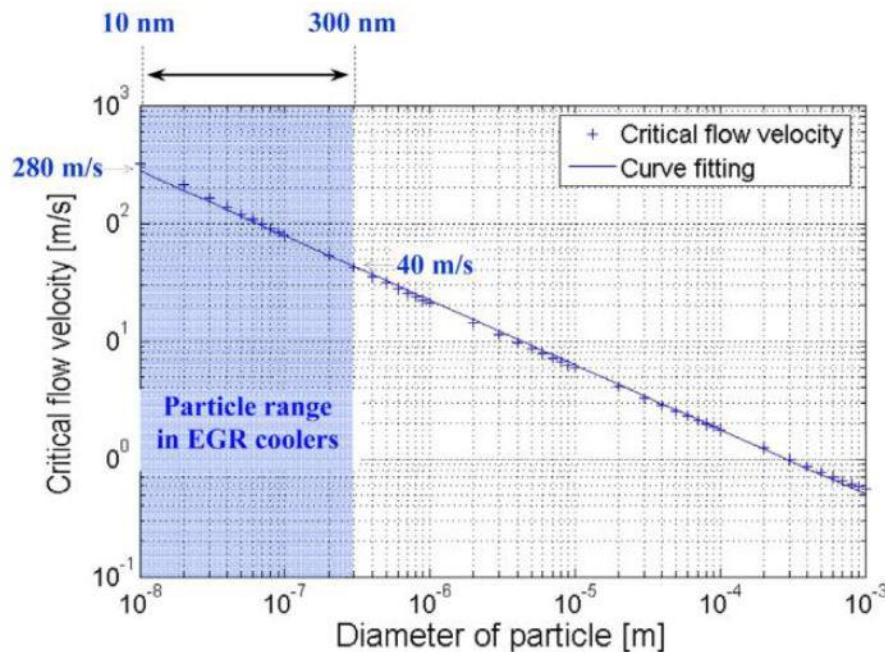


Figure 2.14 - Evolution of the critical flow velocity per particle diameter regarding removal by shear stress (Abd-Elhady *et al.*, 2011)

This can also be confirmed by the analysis performed by Abd-Elhady et al. (2011), which shows clearly that the fouling resistance decreases with the increase in Reynolds number, that is, with the increase in gas velocity (Figures 2.14 and 2.15).

2.7.3. Kinetic Energy

When the particles are attached to each other, it happens that Van der Waals force has a greater influence between them than thermal forces do. Therefore, particle removal can be achieved if their kinetic energy is higher than the Van der Waals force. This is possible for the particles situated at the gas-deposit interface, as they are subjected to a higher temperature and, consequently, their kinetic energy is greater than that of

particles nearer the wall of the cooler. If the temperature is high enough, it will allow the thermal force to overcome the Van der Waals force and remove the particle from the deposit. The interface temperature acts like a removing agent for the particles (Abarham *et al.*, 2010).

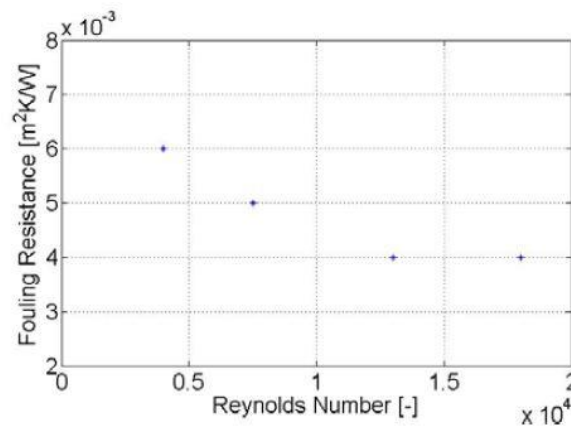


Figure 2.15 - Evolution of the fouling resistance of an EGR cooler as a function of gas velocity (through Re number)

2.7.4. Oxidation/Evaporation

Evaporation of the volatile components of the deposit may occur if the temperature is high enough to allow it. It may also provoke oxidation of soot particles, which will improve the cooler performance. Having an Oxidation Catalyst can help incite this effect (Abd-Elhady *et al.*, 2011, Sobh, 2011).

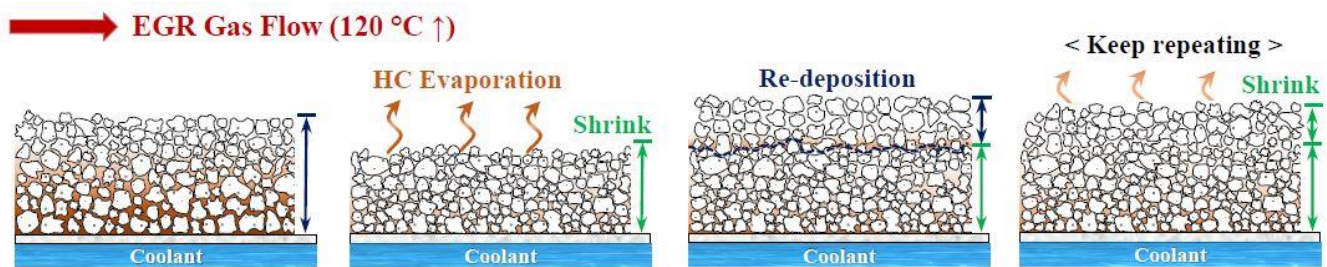


Figure 2.16: Scheme of the evolution of particle removal through evaporation (Han *et al.*, 2015)

3. Thermodynamic Analysis of Fouling

The heat flow can be determined either through the water cooling or the exhaust gas side, as is observed in the following equations (Völk and Hörnig, 2011):

$$\dot{Q}_{exhaust} = \dot{m}_{exhaust} \cdot c_{p,exhaust} (T_{exhaust,out} - T_{exhaust,in}) \quad (3.1)$$

$$\dot{Q}_{water} = \dot{m}_{water} \cdot c_{p,water} (T_{water,out} - T_{water,in}) \quad (3.2)$$

The transfer coefficient of the cooler to describe the fouling effect in the form of the deposition layer, on a cylindrical tube, as the one used on this dissertation (described in the next chapters), is obtained by:

$$kA = \frac{\dot{Q}}{\frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}} \quad (3.3)$$

being:

$$\Delta T_2 = T_{exhaust,out} - T_{water,in} \quad (3.4)$$

$$\Delta T_1 = T_{exhaust,in} - T_{water,out} \quad (3.5)$$

The stationary heat conduction through the fouled tube can also be described by the Péclet equation, taking into consideration the isolative layer (Völk and Hörnig, 2012):

$$\dot{Q} = \left(\frac{2\pi l \cdot (T_m - T_\infty)}{\frac{1}{h_i(r_i - s)} + \frac{1}{\lambda_{Fouling}} \ln\left(\frac{r_i}{r_i - s}\right) + \frac{1}{\lambda_{tube}} \ln\left(\frac{r_o}{r_i}\right) + \frac{1}{h_o r_o}} \right) \quad (3.6)$$

where T_m is the adiabatic mixing temperature, T_∞ is the ambient temperature and s is the thickness of a constant and homogeneous deposit layer built on the tube's inner wall. This Péclet equation for inner flow cylindrical tubes is valid only for stationary systems

and it sets the foundation of the modelling of fouling mechanisms. The mentioned temperatures should be measured at the centre of the flow, in order to reduce the influence of the temperature profile. The equation can be better understood by the scheme from on Figure 3.1, which depicts stationary heat conduction along a cross-section of the clean tube without deposition, in a configuration of concentric tubes, as later will be used in the performed simulations. Figure 3.1 and Equation (3.6) also show

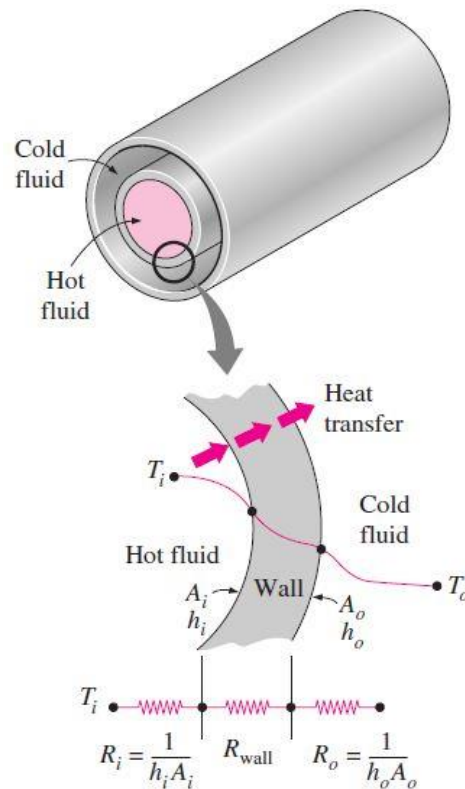


Figure 3.1 - Scheme of concentric tubes forming a heat exchanger and the associated thermal resistances (Çengel, 2003)

the thermal resistances that oppose the cooling of the flow: convection on the inside of the tube, conduction on the deposit layer, conduction on the tube wall and convection on the outside of the tube, in ambient conditions. The build-up of the isolative deposit layer leads to a deflection of the initial heat transfer coefficients, contributing for the following effects:

- Changing of the deposit surface smoothness;
- Changing of the Reynolds number due to the narrowing of the flow diameter;
- Appearance of condensation
- Rise of the temperature on the area of the tube where the deposit layer is thicker.

4. Computational Fluid Dynamics

4.1. Introduction – Fluent Approach

Computational Fluid Dynamics (CFD) is a computational tool that through numerical methods, using finite volume elements, allows to solve the governing equations for fluid flow and heat and mass transfer, obtaining a solution for each of those elements, defined as cells. *ANSYS Fluent* is the CFD solver software used in the work of this dissertation. It allows the performing of simulations either in serial or in parallel mode, using multiple CPUs simultaneously, in order to speed up the calculations. Mass transport equation and 3D momentum conservation equations are the foundation of CFD analysis (Yan, 2011). This software is extremely useful as it enables the study of a wide range of engineering flows, with all types of boundary conditions, dealing with compressibility of flows, viscosity, laminar and turbulent conditions, etc.

The CFD process is fundamentally divided into 3 steps:

- Pre-processing: the CFD model is set up along with the mesh and boundary conditions for each single operating condition
- Solving: the simulations are undertaken and the numerical solutions obtained, through a process of discretization of the variables;
- Post-processing: the obtained data is analysed through plots and pictures depicting contours, vectors, etc.

Besides the three fundamental Navier-Stokes equations, in order to assess the turbulent behavior, turbulence models require transport equations with the turbulence flow variables. Also, additional equations are necessary to describe and analyse multiphase models, but they are not referred as they will not be approached during this dissertation. The Navier-Stokes equations are very important for the study of fundamental dynamics, as they describe the flow of a Newtonian fluid quite accurately. They are, however, very

difficult to solve analytically, as they are non-linear and coupled equations; thus, the use of numerical methods. An approximate numerical solution can be obtained by the discretization process that approximates the differential equations by a set of algebraic equation. This approximation is applied in space or time to small domains, thus obtaining discrete solution values at every small domain considered (Ferziger and Peric, 2001).

4.1.1. Governing Equations

The set of equations that describe the full flow behavior – the Navier-Stokes equations – are now described:

Mass Conservation Equation

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{u}) = S_m \quad (4.1)$$

This equation is applicable for both compressible and incompressible flows and it is time dependent, being S_m the mass source term. It defines the mass variation inside a control volume, executing the balance between mass leaving and entering this last one (Oliveira, 2012).

Momentum Equation

The momentum conservation lies on the principles of the second Newton's law: the momentum variation in all directions is equal to the summation of all the forces acting in the same directions (Oliveira, 2012). The microscopic momentum fluxes across a surface are the forces due to pressure and stress that these surfaces are subjected to, from a molecular perspective (Ferziger and Peric, 2001). Equation (4.2) is the set of Navier-Stokes equations that describe the momentum conservation:

$$\frac{\partial}{\partial t} (\rho \vec{u}) + \nabla \cdot (\rho \vec{u} \otimes \vec{u}) = -\nabla p + \nabla \cdot (\bar{\bar{\tau}}) + \rho \vec{g} + \vec{F} \quad (4.2)$$

$$\bar{\bar{\tau}} = \mu \left[(\nabla \vec{u} + \nabla \vec{u}^T) - \frac{2}{3} \nabla \cdot \vec{u} \cdot \vec{I} \right] \quad (4.3)$$

being p the static pressure, $\bar{\bar{\tau}}$ the tensor that describes the molecular rate of transport of momentum (specified in equation X), $\rho \vec{g}$ the term corresponding to the gravitational body force, and $\vec{F}$ an external body force (Yan, 2011).

Energy Conservation Equation

The conservation of energy is based on the physical law that the rate of change of energy in one control volume equals the balance of energy transferred to and from the particle by heat and work (Henningson and Berggren, 2005). The following equation describes the energy transport along the fluid flow:

$$\frac{\partial}{\partial t}(\rho E) + \nabla \cdot (\vec{u}(\rho E + p)) = \nabla \cdot \left(k_{eff} \nabla T - \sum_j h_j \vec{j}_j + (\vec{\tau}_{eff} \cdot \vec{u}) \right) + S_h \quad (4.4)$$

in which k_{eff} stands for the effective conductivity (altered by the turbulence model), h_j corresponds the enthalpy of species j and $\vec{j}_j$ is the diffusion flux of species j . The term E corresponds to the internal energy of an element:

$$E = h - \frac{p}{\rho} + \frac{u^2}{2} \quad (4.5)$$

The three terms in brackets on the right hand side of equation (4.4) correspond to, respectively: energy transfer by conduction, species diffusion and viscous dissipation; the last term stands for all of the volumetric heat sources. (Yan, 2011)

This equation is essential for any problems having heat transfer or compressible flows in their scope.

4.2. Turbulence Analysis

The analysis of turbulent flow is extremely important as it is ruled by unpredictability (Sofialidis, 2013). It occurs at high values of Reynolds number, and is caused by the complex interaction between the inertial and viscous terms in the momentum equations (Bakker, 2006a). The main characteristics of a turbulent flow are (Oliveira, 2012):

- It is unsteady, irregular and chaotic due to random movements of the fluid;
- Turbulence can be described as a 3D system of diffusive transport of mass, momentum and energy, by means of vortices – they are rotational, that is, they have non-zero vorticity;

- It fluctuates in a broad scale of both time and space, which makes direct numerical simulation very difficult;
- In order to execute the modelling of the turbulence phenomenon, additional terms are required to the Navier-Stokes equations.

In turbulent flow, the energy is transferred from larger eddies (vortexes) to smaller ones, where the turbulent energy is converted into internal energy by viscous dissipation (Sofialidis, 2013). The large eddies interact with the mean fluid flow and extract energy from it by a process designated vortex stretching (Versteeg and Malalasekera, 2007).

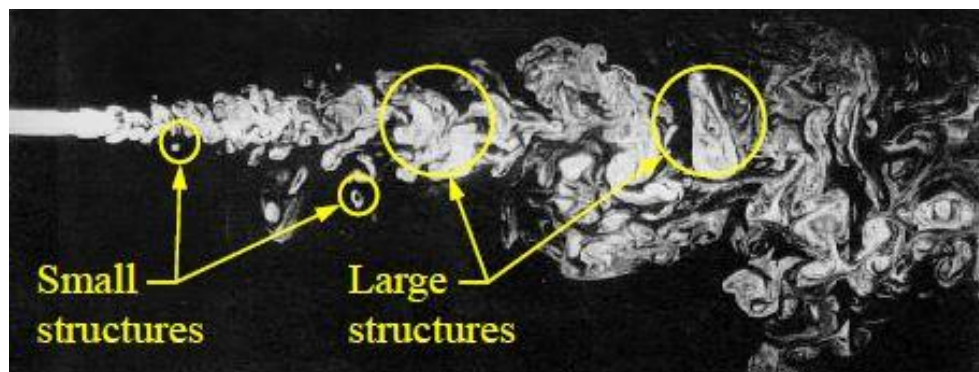


Figure 4.1 - Turbulent flow with highlight on the small and large eddies (Sofialidis, 2013)

One of the most important characteristics of a turbulent flow is its diffusivity: it enables rapid mixing and higher rates of momentum, energy and mass transfer – turbulence brings together fluids of different momentum concentrations. They also have a dissipative behavior: the kinetic energy is reduced due to the action of viscosity. The conversion of the turbulent energy to internal energy of the fluid is irreversible. Therefore, these flows “die out” fast when no energy is supplied to them (Ferziger and Peric, 2001).

Turbulence is a very important factor regarding heat and mass exchange transfer, as the mixing that takes place is much stronger and faster than in a laminar condition flow. This is due to the macroscopic level of the randomness in the turbulent flow, which makes its shear stress far larger than the one in laminar flow (Munson *et al.*, 2012).

In order to ensure a better and more precise simulation of the turbulent flow, an adequate turbulence model needs to be picked, which will influence greatly the final

results, as it regards different mesh definitions and computational effort. The goal is to develop equations that will be able to predict the time-averaged velocity, pressure and temperature fields without having to calculate the complete turbulent flow as a function of time. Nonetheless, one needs to keep in mind that the time-averaged characteristic of the flow pattern is a statistical property, as it is never possible to predict exactly how the flow is going to develop. The simplest way to start the modelling of turbulence is to introduce a turbulent viscosity, which is a proportionality relationship between the Reynolds stress components and the strain or deformation rate tensor (Henningson and Berggren, 2005).

The turbulent viscosity needs to be modelled and most of the numerical methods use to solve the Navier-Stokes equations use some kind of model to describe the turbulent viscosity (Henningson and Berggren, 2005). From the existing turbulence models available with *Fluent* software, the subsequent ones are the more adequate one to the flow studied in the scope of this dissertation:

- k - ε model
- k - ω model

The k - ε and k - ω turbulence models are two-equation models inserted in the group of *Reynolds Averaged Navier-Stokes* (RANS) turbulence models, which introduces extra variables to characterize the turbulence. This group of turbulence models is highly used as they supply good results with the advantage of needing very low computation effort. The k - ε and k - ω models are able to obtain precise results regarding stationary regime conditions, while simultaneously being the fastest in terms of calculation time (Oliveira, 2012).

4.2.1. k - ε model

This model is based on model transport equations for the turbulence kinetic energy k and the corresponding dissipation rate ε . There are three versions of the k - ε model: Standard, Realizable and RNG; they allow the determination of a turbulent length and time scale, by solving two different equations (ANSYS, 2016b). The main difference between the Standard and Realizable models lies on the fact that the first determines the k variable from exact equations and ε from empirical ones, and the latter derives them both from exact equations. The Realizable k - ε model allows a more precise result

regarding the separation of the boundary layer flows depicting a high swirling component, adverse pressure gradients and recirculation areas. It also predicts very accurately the spreading rate of both planar and round jets. It is commonly chosen over the Standard $k-\varepsilon$ model as this latter one, despite being robust, computationally economic and showing a reasonable precision of the results in most applications, it is only valid for fully developed turbulent flows and the influence of molecular viscosity is neglected (Oliveira, 2012).

The RNG $k-\varepsilon$ model was derived through a statistical technique designated renormalization group theory (RNG). This model was developed as an improvement of the standard model. It possesses an enhanced accuracy regarding rapidly strained flows, swirling flows; while the standard model is simply a model designed for high-Reynolds numbers, this model also accounts for the low-Reynolds effects.

All in all, the RNG $k-\varepsilon$ model and the Realizable $k-\varepsilon$ model are far more accurate and broad than the standard model. However, the realizable model has proven to be the one to deliver the best performance of all the $k-\varepsilon$ models for flows defined by separation and complex flow characteristics (ANSYS, 2016b).

4.2.2. $k-\omega$ model

In this model, the transport equation for the turbulent dissipation rate ε is replaced by an equation for the specific dissipation rate ω . These models show much better performance than $k-\varepsilon$ models regarding boundary layer flows, as they account more accurately for separation, transition, low- Re effects (Sofialidis, 2013). They are better at predicting equilibrium adverse pressure flows. There are three versions of this model: Standard, Baseline (BSL) and Shear Stress Transport (SST). All three of them have transport equations for k and ω . Despite being substantially more accurate than the $k-\varepsilon$ models near wall layers and with adverse pressure gradients, its standard model failed for flows with pressure induced separation and the ω -equation show a high sensitivity to the values of ω in the freestream outside the boundary layer. Hence the need to develop the BSL and SST versions (Menter *et al.*, 2003)

The main additional complexity with these two models lies on computing the distance from the wall, since there is a gradual change from the standard $k-\omega$ in the inner area of

the boundary layer to a high- Re number version of the k - ε model in the outer region of the boundary layer (ANSYS, 2016b).

The standard k - ω model incorporates modifications for low- Re number effects, compressibility and shear flow spreading, as it is based on the Wilcox k - ω model. The standard version is an empirical model with fundamentals on model transport equations for the turbulence kinetic energy k and the specific dissipation rate ω . This specific dissipation rate is defined as the ratio between ε and k (ANSYS, 2016b)

Concerning solely the SST version, it exhibits a modified turbulent viscosity formulation that accounts for the transport effects of the main turbulent shear stress, which does not happen in the BSL model. This model is a hybrid 2-equation model that combines the advantages from both k - ω and k - ε , since the first one shows much better performance than the k - ε when it comes to boundary layer flows, and due to the great sensitivity of the k - ω model to the free stream value of the specific dissipation rate ω , which does not occur in the other model. Accordingly, the SST model works as the k - ω model near the wall and as the k - ε in the freestream. Subsequently, this improved near-wall formulation has had a great contribution for the industrial heat transfer problems, as it has reduced the requirements for the near-wall grid resolution (Menter *et al.*, 2003).

All these features together make the SST model more accurate and reliable for a broader specter of flows that the standard and BSL models do not contemplate. Therefore, due to all of the above described, the turbulence model used in the simulations performed throughout this dissertation was the SST k - ω model.

4.3. Wall Treatment

The existence of walls in flows severely affects the behavior and prediction of turbulent flows, as they are the major source of vorticity and turbulence. Here is where large gradients of the variables occur and there is a greater transport of momentum and scalar quantities. The walls induce variations in pressure and velocity, mainly because the main velocity field is affected by the no-slip condition at the wall. In addition to that, close to the wall an effect of viscous damping is induced in the flow that reduces tangential velocity fluctuations. At the periphery of the near-wall region, the turbulent effects are heightened by the production of the turbulent kinetic energy (ANSYS, 2016b).

Therefore, a more rigorous study has to be undertaken at the wall, in order to permit a more successful prediction of the behavior of the flow in this region. In Figure 4.2, a scheme of the three different layers that can be distinguished in the near-wall region is displayed:

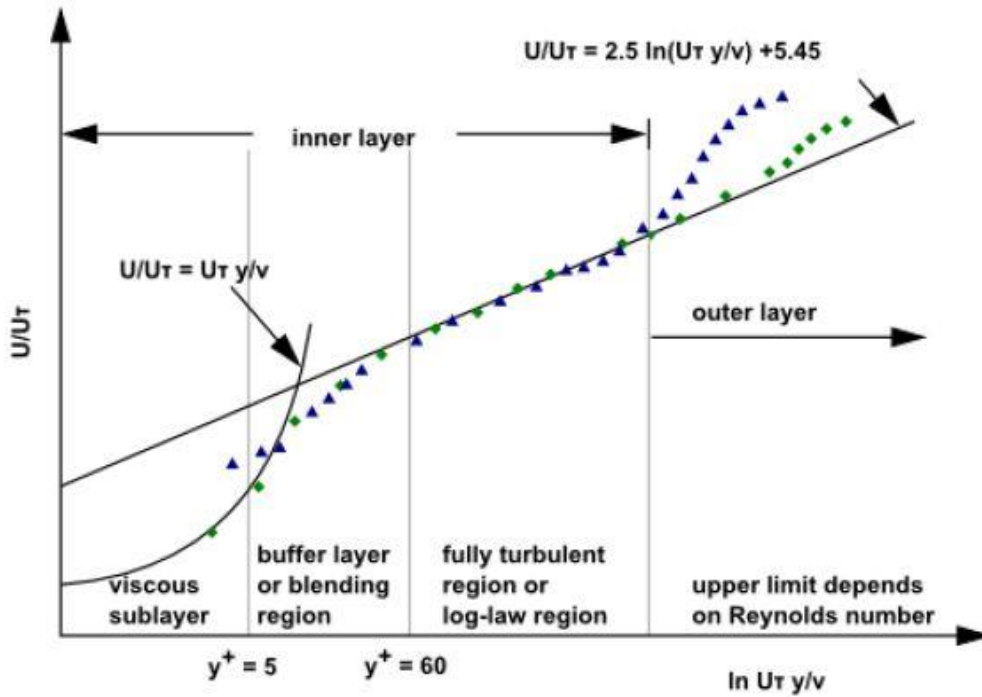


Figure 4.2 - Scheme of the three different layers composing the near-wall region (ANSYS, 2016b)

The inner layer, designated viscous sublayer, the flow is nearly laminar, where viscosity is the dominant factor regarding momentum, heat and mass transfer. The outer layer, on the other hand, is called the fully turbulent layer and the turbulence plays a major part in it. Here the influence of both molecular viscosity and turbulence is important (ANSYS, 2016b). As the behavior of the flow near the wall is extremely influential and highly difficult to predict, one needs to introduce a new parameter designated y^+ , which indicates the dimensionless height of the first cell at the wall. The value of parameter depends on the density ρ , the velocity u , and the dynamic viscosity of the fluid μ , as follows in equation (4.6) (Oliveira, 2012):

$$y^+ = \frac{\rho u y}{\mu} \quad (4.6)$$

It is the ratio between the turbulent and laminar influences within a cell and so it will highly influence the number of cells used for the modelling analysis of the near-wall

region and help choose the better wall treatment method, as values of $y^+ = 1$ require extremely refined meshes, and higher values will require far coarser meshes.

As one can see in Figure 4.2, in the viscous sublayer one ought to have $y^+ < 5$, since this is a region dominated by molecular viscous effects. In the outer layer, the value of the parameter should be $y^+ > 60$, and vary between 5 and 60 in the intermediate region, which is a transition area between laminar and turbulent conditions (Oliveira, 2012). There are two different approaches for the treatment of Wall Heat Transfer (Oliveira, 2012):

- Wall Function approach;
- Near-wall model approach.

In the Wall Function method, the viscous-affected region is not at all solved, but rather analysed by resorting to empirical formulations to assess the different flow variables in this zone, designated “wall functions”. These make a connection between the viscous-affected zone and the fully turbulent region, without the need to change the turbulence models to account for the presence of the wall. They go around the inability of the $k-\varepsilon$ model to predict a logarithmic velocity profile near the wall (ANSYS, 2016b).

The Near-wall model approach is the one recommended by the software, as it is more precise, particularly at low Re -numbers. By adapting the turbulence models to the viscous-affected region, it solves the governing equations for the whole mesh without the use of wall functions because it uses a very refined mesh closer to the wall. They have to be adapted because the high Reynolds numbers models may fail when predicting the behavior in the near-wall region.

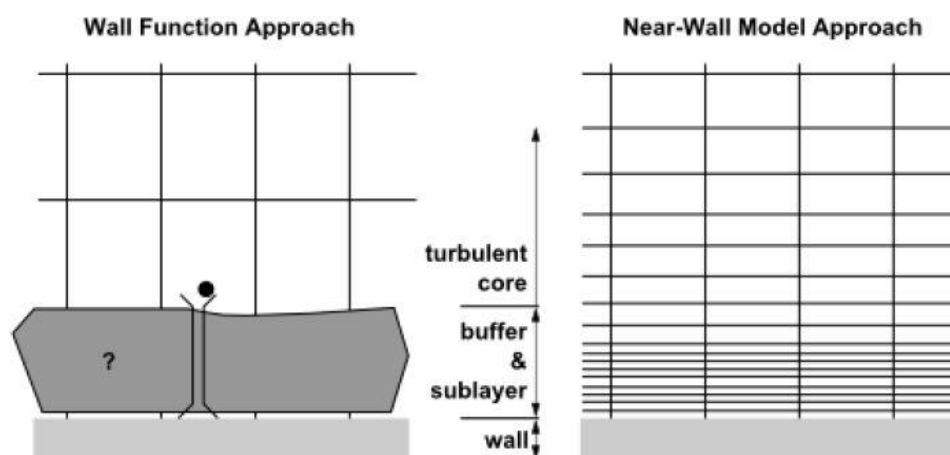


Figure 4.3 - Scheme of the two near-wall treatment methods (ANSYS, 2016b)

However, they require more computational effort, which in industrial problems is often not viable. These two different methods are both schematized in Figure 4.3.

To obtain numerical results of high quality in the boundary layer, this last one needs to have a sufficient overall resolution, which may even be more important than achieving the above described y^+ values. It is important to notice that, in order to have a more refined mesh, one only needs a refinement in the normal direction to the wall.

Values of y^+ close to 30 are more desirable for Wall Function method, while for the Near-wall approach values of $y^+ \approx 1$ are desired. Once a good refinement is in place, the $k-\omega$ model is well suited to model the boundary layer (Salim and Cheah, 2009).

4.4. Conjugate Heat Transfer

Fluent allows the computation and analysis of heat conduction through solids coupled with heat convection in fluid flows, an analysis called Conjugate Heat Transfer (ANSYS, 2006), provided that the mesh has a clear definition of the interface in the computational domain. This type of analysis is very useful in applications such as heat exchangers, cooling and heating systems (SimScale, 2016). It has also been applied in several engineering studies, such as heat transfer in Internal Combustion engines, aerothermal analysis and design of microelectronic devices (Xue, 2009). It is possible to establish coupled boundary conditions for wall zones that separate two cell zones, taking into account that both fluid and solid zones may contain heat sources (Bakker, 2006b).

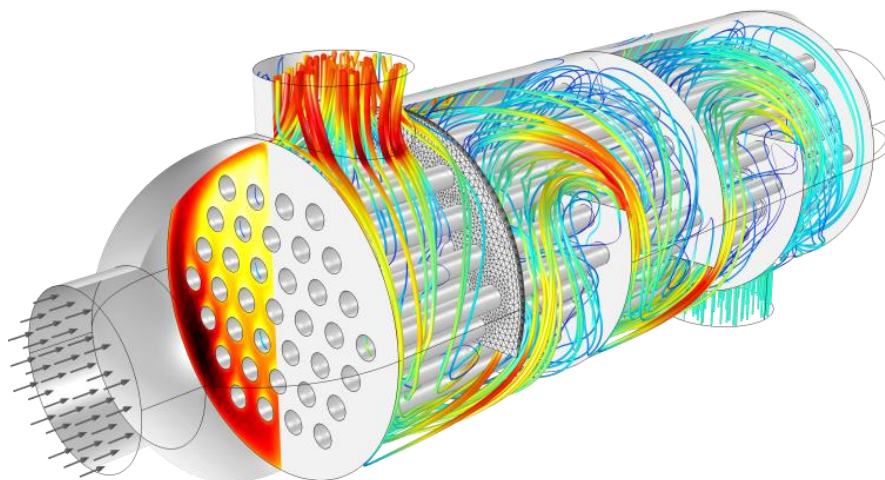


Figure 4.4: Example of CHT calculation (Huc, 2014)

In this kind of analysis, the Navier-Stokes equations are solved for the fluid domain, whereas as the heat conduction is solved in the solid domain; the temperature and heat transfer coefficients at the interface between the two domains will be determined by a local energy balance, at every cell. It is of extreme importance to define correctly the boundary conditions – existing heat sources and conditions of the flow, such as effects of buoyancy, natural or mechanic convection, laminar or turbulent flow, etc. – and make sure the meshing on the interface between the two domains is properly defined.

The interface boundary conditions require the same heat flux and temperature for both the fluid and the solid at all boundary points, on each grid node along the interface. The information on each neighboring node (from both sides) is saved and updated at every iterative step, for all the variables (temperature, pressure, etc) (Xue, 2009).

$$T_{fluid} = T_{solid} \quad (4.7)$$

$$k_{fluid} \frac{\partial T}{\partial n} \Big|_{fluid} = k_{solid} \frac{\partial T}{\partial n} \Big|_{solid} \quad (4.8)$$

However, regarding this dissertation, the interest lies only on the results for the fluid side, as the objective is to assess the fouling effect on the gas-side of the cooler.

4.5. CFD Coupling using UDF

User defined functions (UDF) are programmed routines in *C* language that allow to customize *Fluent*, enhancing its modelling capabilities. It permits the tailoring of the CFD for special applications, which otherwise would not be obtainable with standard *Fluent* interface. (Yan, 2011, ANSYS, 2016a)

Regarding this dissertation, several UDFs were supplied by Atlanting GmbH in order to allow the simulation of the EGR Cooler studied at DEUTZ AG, enabling the gathering of results concerning deposit thickness, heat fluxes, condensation rates, etc. inside the cooler. The UDFs are identified by a “.c” extension and a header file designated by “.h” that allows to direct and include the function in the compilation process.

The simulation process is divided into two stages:

- CHT analysis – basic *Fluent* analysis;

- Fouling analysis – running of the UDFs.

The UDFs are initially introduced (but not hooked) in *Fluent* software when all the boundary conditions are set for the CHT calculation. They should be compiled and not interpreted. Then the file is saved and again reopened, for the header file that associates the user defined memories names to load the renamed fouling routines. This way the fouling model will be able to run properly.

The simulation is then started until one obtains converged values (generally around 1000 iterations are necessary). This step is purely the steady state CHT analysis, without the fouling; that is, only the cooling performance of the (clean) cooler is analysed – that is, the heat transferred between gas and coolant -, along with the evolution of the exhaust gas flowing within (pressure drop, temperature difference, etc.), with basic *Fluent* settings. This stage is of extreme importance as the calculated flow, the turbulent behavior and temperature fields will be used as input information for the determination of the fouling analysis. However, in this initial simulation for CHT analysis, the option for layer conduction model must be already activated (at the gas side of the cooler wall) and with the limit value defined, as it is essential for the fouling model. If it is not initially activated, the solution will lose the values of the CHT analysis when starting the fouling model.

The value defined for the maximum layer thickness in the model depends on the cooler. It was actually fixed (but never reached) 1.3 mm, as a recommendation from the routines supplier, and it cannot be later changed in the fouling analysis. Also, the characteristics of the deposits need to be inserted as a new solid material, with the following values:

Property	Value
Density [kg/m ³]	17
Specific heat capacity [J/(kg K)]	100
Thermal Conductivity [W/(m K)]	10

Table 4.1 – Property values of pure soot for the deposits

However, the value that has a bigger influence is the thermal conductivity, which is chosen to make the effect on heat transfer negligible for the CHT analysis.

This stage of simulation is run with the turbulence model SST $k-\omega$ model. Residual convergence should be turned off and the actual convergence should be assessed based on the values obtained.

When starting the fouling analysis, three steps must be executed:

- Hooking the UDFs;
- Turning off the unnecessary equations;
- Changing the dependence of the deposits' thermal conductivity.

Thoroughly explained, after assessing the CHT results, it is necessary to hook the UDFs to *Fluent's* interface, i.e. to activate them. This simulation stage is only about evaluating the temperature evolution inside the cooler and in the exhaust gas due to the soot deposit layer, and the final state of equilibrium in the deposited layer. Consequently, in the menu Solving → Solution Controls → Equations the equations for **Flow** and **Turbulence** ought to be disabled, only remaining the **Energy** equation enabled to proceed with the heat transfer calculation. The flow no longer plays a role in the fouling results; the only goal is to determine the heat conduction. As in this stage the values of the layer will change according to the operating conditions and along the iteration process, different values will be obtained for its thermal conductivity. Therefore, this property of the deposits (solid material) ought to be changed in the *Fluent's* settings from constant to user-defined. From now on, this value will differ according to the boundary conditions introduced by the UDFs.

In the running of the C-routines, two conditions are assumed by them for deposit calculation:

- The walls are on fluid side and connected to solid;
- The layer thickness activated in the shell conduction model is mandatorily > 0 .

Depending on the local boundary conditions, the deposition and removal mechanisms are calculated until they come to a balance, i.e. until they reach a state of equilibrium and there is no longer a variation of the results. During this work, only a steady state solution was sought out, as the routines used provided only the results after reaching a steady state, that is, the values obtained are not time dependent. This is valid both for the Conjugate Heat Transfer (CHT) and the Fouling solutions.

4.6. Errors in CFD Calculations

In CFD analysis (and every other numerical analysis) it is important to keep in mind that the numerical results are always approximate, because errors may arise from several sources, and it is not possible to eliminate them all. Therefore, the difference between reality and computed results lies in simple reasons, such as the following:

- The differential equation may contain approximations, to simplify and make the calculations possible (considering inviscid flow, irrotational flow, etc.);
- Approximated numerical simulations are based on a discretization process;
- When solving the equations for each discretized domain, iteration processes are used, which also comes with errors as only running them for an infinite amount of time, would the exact solution be reached.

The visualization of the numerical solutions through the display of vectors, contours, etc. that describe the flows is a powerful tool and helps to better understand and have a 3D view of the flow. However, one needs to be very careful using these methods of post-processing the information, as many times the errors are not perceptible. The solution may look good but simultaneously be quantitatively erroneous, and not correspond to the actual values, boundary conditions, etc. (Ferziger and Peric, 2001). Residuals are the error magnitudes for the equations as the iterations proceed. They measure the local imbalance of a conserved variable in each control volume. They include the following governing equations:

- Navier-Stokes momentum equations;
- Continuity equations (for mass conservation);
- Energy equations (for heat transfer);
- Turbulence model equations.

The objective of running a very high number of iterations (usually > 1000) is to make the solution converge: that means the errors get smaller and smaller, the equation values change ever less.

Regarding heat transfer calculations usually a deep convergence is required: the scale residuals should be of the order of $1e-06$ and even lower (Bakker, 2006b).

5. Method – Model Validation

As main objective of this work, a few studies were used as source for validation of the routines supplied for this dissertation. Thence, a basic geometry is described in these documents, where several test bench experiments were performed according to it, obtaining values to be compared with the simulation analysis performed in this dissertation.

5.1. Geometry

Figure 5.1 depicts the geometry of the cooler used by Völk and Hörnig (2011; 2012) in their study and measurements. It has a double tube construction (concentric tubes) and the flows of the exhaust gas and the cooling fluid (water) run in opposite directions (Völk, 2014).

However, it must be stated that the scheme contains an incorrect value: the total length of the cooler cannot be 290 mm, as 225 plus 65 mm already performs the 290 mm. Therefore, it will be assumed that the outlet part of the cooler also measures 65 mm.

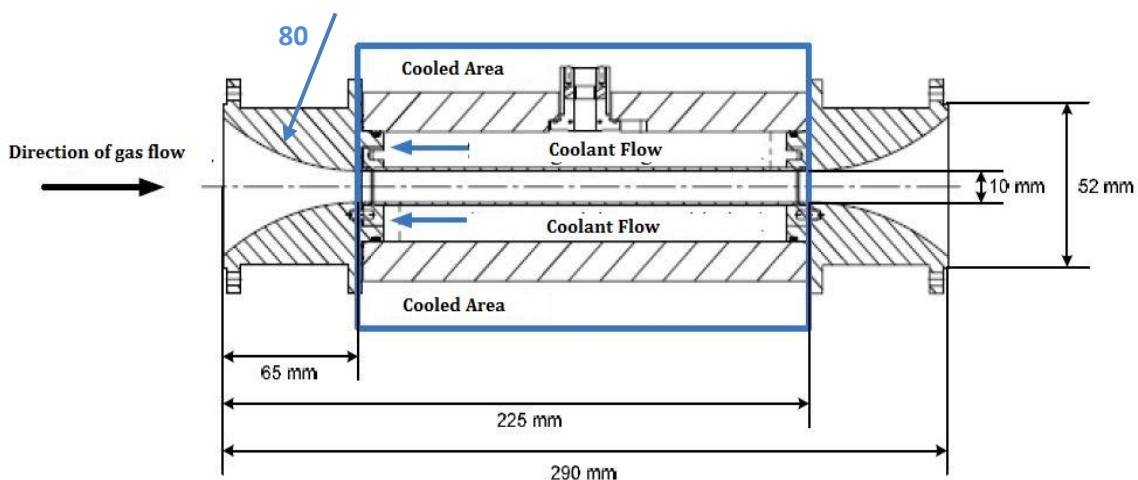


Figure 5.1: Layout of the cooler design for field tests used by Völk and Hörnig (2012)

The cooler tube is made of aluminium, with a length of 160 mm, outer diameter of 12mm and wall thickness of 1 mm (Hörnig, 2012). The control volume used for the simulations performed in this dissertation is defined by the blue rectangle, that is, specifically just the cooler tube, excluding the inlet and outlet funnels.

The entrance geometry plays a big role in the fouling occurrence, as sudden changes in geometry are very likely to induce fouling by impaction or interception of soot particles. Consequently, for legitimate values to be obtained with this geometry, the entrance was designed based on previous tests on EGR coolers, with the goal of minimizing the fouling effect (Hörnig *et al.*, 2011).

The equations of mass and energy conservation of the flow inside the cooler are simplified, as the following assumptions are made:

- Coolant temperature is constant and corresponds to the wall temperature (the water flow is not considered for this dissertation – and for that matter, nor in the sources used);
- Soot properties do not vary with deposit thickness or temperature.

As previously stated, the water-side flow was not considered in the study of Völk and Hörnig (2011, 2012). The water flow was setup in such conditions that the boundary condition defined at the outer wall of the inner tube was of constant temperature: the water flow was so high that the difference between the water temperature at the inlet and outlet is practically irrelevant, thus eliminating the need to study this flow.

With this geometry, two different test benches were set by Völk and Hörnig (2011) to perform the measurements. They are described in sections 5.6 and 5.7.

5.2. Mesh Characteristics and Influence

It is important to focus on the importance of the definition of the mesh used to perform the simulations, as it has a great impact on the final results. A coarse grid will cause poor spatial resolution of the gas phase results and, consequently, lead to a poor prediction of the interactions between gas and liquid phases (Xue, 2009). A refined mesh, with equal cell-wall distance along the entire radial cross-section of the tube was used, for a good accuracy of the simulations.

The mesh was developed and after the values of y^+ were tested to see if they fitted the recommended values for the near-wall treatment. One can only know the values of y^+ after running an initial simulation, since this value depends on the local fluid velocity and on the mesh; therefore, it is not possible to know it beforehand. The zoom on the mesh depicted in Figure 5.3 shows clearly the equal wall-cell distance, along the whole radial cross-section.

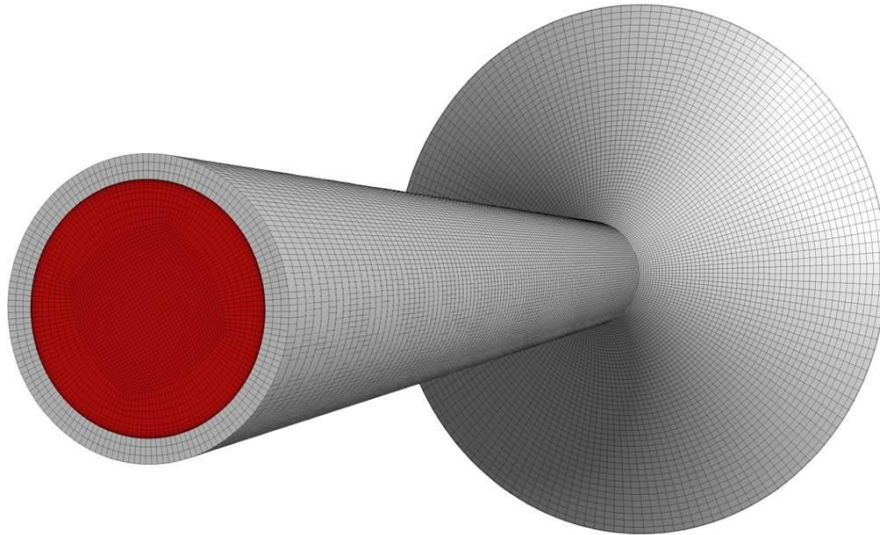


Figure 5.2 - Screenshot of the mesh used in Fluent for the chosen cooler geometry

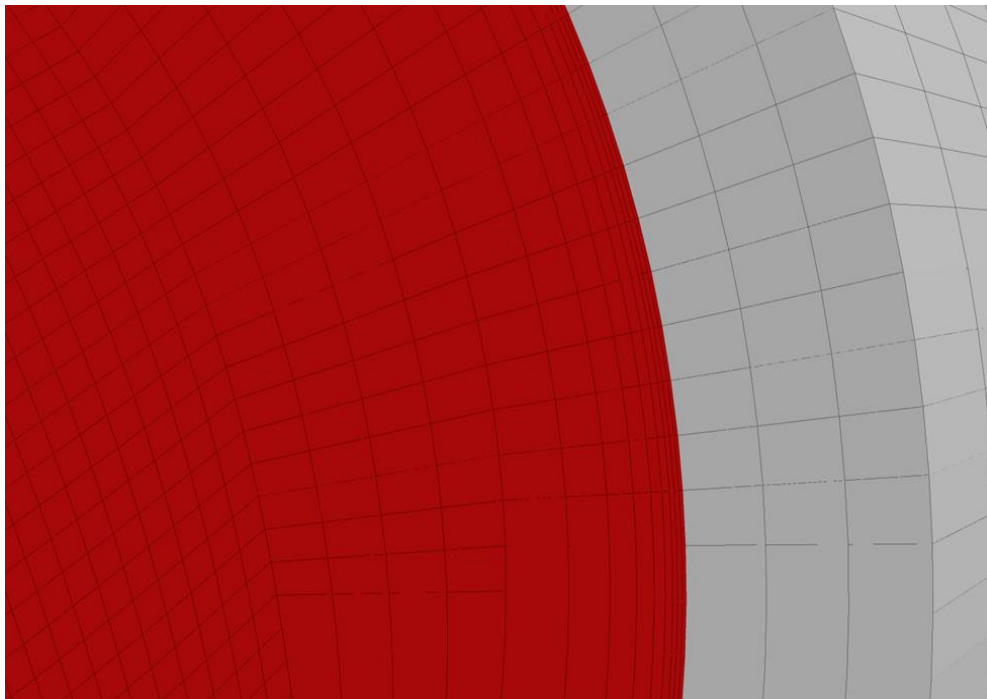


Figure 5.3 - Zoom-in on the final mesh used for the inside of the tube, 1/4 of the radial cross-section displayed

This mesh is extremely refined and uniform near the inner wall of the cooler, at the interface with the gas, where the deposit layer will set in, as it can be seen in Figure 5.3.

5.3. Exhaust Gas Composition

The simulations performed in this dissertation used several properties indicated by the supplier of the C-routines, Atlanting GmbH. The exhaust gas composition follows an ideal gas behavior. However, some properties such as the thermal conductivity, specific heat capacity and thermal conductivity are temperature dependent values, hence its introduction in *Fluent* as a piece-wise linear evolution, with the extreme points at 100 °C and 400 °C. The values presented in the next tables match the ones for ideal gas air, as mentioned. The properties of the cooler side are not important, even though one knows water is the fluid used. It is, nonetheless, not contemplated in the simulations in the scope of this thesis, as previously mentioned. The only needed parameter is the temperature on the water side, which will be considered constant temperature as a boundary condition.

Temperature [°C]	Thermal Conductivity [W/(m K)]	Specific heat capacity [J/(kg K)]	Dynamic Viscosity [kg/(m s)]
100	0.03139	1012	2.194E-05
400	0.04996	1069	3.335E-05

Table 5.1 - Description of the values adopted for the thermal conductivity, specific heat capacity and dynamic viscosity of the exhaust gas for a piece-wise linear evolution

5.4. Sulphur Concentration

As it can be seen on Table 5.2 the sulfur content is described in concentration of H₂SO₄. Therefore, a conversion needs to be done in order to have the value for the SO₂ content, as this is the one to be inserted as input in the routines.

During the combustion, most of the sulphur is transformed into SO₂, from which around 2% will be oxidized to SO₃. Then the emerging SO₃ quickly reacts with the water vapor existing in the exhaust gas, thus forming H₂SO₄. This can reinforce the particle deposition, through the condensation of the sulfuric acid, when the dew point of 112 °C – 120 °C is reached in the exhaust gas flow through the cooler (Völk, 2014).

According to Atlanting (2016), the conversion relationship between SO_2 and H_2SO_4 used in their code is:

$$H_2SO_{4_{content}} = SO_{2_{content}} \times 0.03 \quad (5.1)$$

Nowadays, sulphur plays an ever-smaller role in combustion and gas emissions. Ever since 2009, the European Union has set the sulphur level to a maximum of 10 ppm (as a mass ratio) in diesel fuel for passenger vehicles and light commercial vehicles. For non-road vehicles and machines the maximum value was set on 1000 ppm until the end of 2011, when the study used as source was produced (Völk and Hörnig, 2011); since then and until the present this value is also at a maximum of 10 ppm (Völk, 2014). The 1000 ppm content of sulphur corresponds to a 3.6 ppm amount of H_2SO_4 in the gas phase.

For the motor test bench, the value adopted by Völk and Hörnig (2011) was 18 ppm of sulphur in the diesel fuel, which coincides with 0.065 ppm of H_2SO_4 in the gas phase, according to the existing proportion.

In the following table are depicted the values for the conversions for both motor and model test bench used sulphur contents, according to the proportion stated and to the conversion relationship used by Atlanting (2016). These concentrations in ppm refer to a mass ratio of sulphur per mass of fuel.

Fuel	Exhaust gas	
Sulphur [ppm]	SO_2 [ppm]	H_2SO_4 [ppm]
1000	120	3.6
18	2.17	0.065
55.6	6.67	0.2
2777.8	333.3	10

Table 5.2 - Values of sulphur and respective conversions to SO_2 and H_2SO_4 used in both test benches

5.5. Soot Particles in Exhaust Gas

Soot is an extremely variable and changeable component. Its properties depend on several factors in the combustion cycle, such as: air-fuel ratio, oxygen content in the intake air, fuel composition, the mixture between air and fuel, loading charge, EGR rate and cylinder temperature (Hörnig, 2012). The particle size distribution is a critical factor for the growth rate and different constitutions of particulate fouling, as during the combustion not only coarse particles emerge but also aerosols (microscopic particles) (Abd-Elhady *et al.*, 2011); therefore, particle concentration distribution was a thoroughly studied and meticulously analysed topic throughout the production of this dissertation.

The graphic displayed in Figure 5.4 is a continuous lognormal distribution curve for the soot emitted both on the motor test bench, as on the model test bench. As a continuous distribution, the total area under each curve corresponds to the total number of particles emitted by each diesel source, according to the integral (Park *et al.*, 2014):

$$\int \frac{dn}{dd} dd = n \quad (5.2)$$

The unit depicted on the ordinate is a differential particle number concentration, that is: the particle number is normalized by range of particle diameter of the interval, which shows that the curve works in particle size (diameter) intervals. For this reason, we cannot determine the concentration of particle number for one specific particle diameter, but instead for a class of several diameters.

As the number of particles is the dependent variable it is described as a number distribution; if dN is the number of particles in a certain size interval from d_p to $d_p + dd_p$, then the number distribution function $n(d_p)$ can be described as (Kulkarni *et al.*, 2011):

$$dN = n(d_p)dd_p \quad (5.3)$$

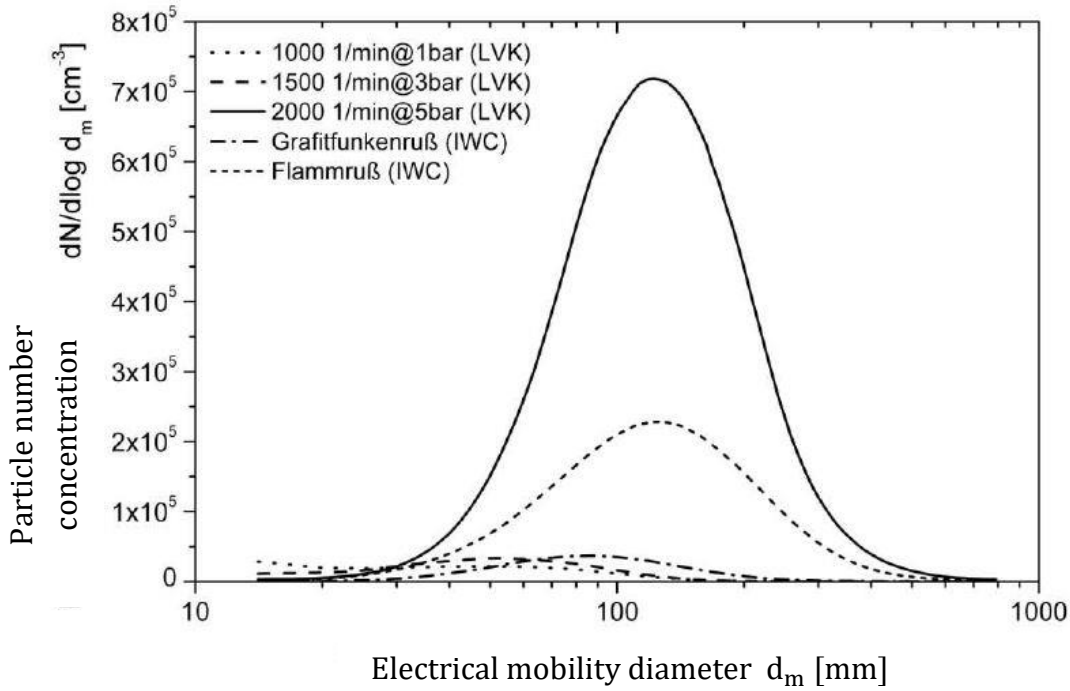


Figure 5.4 - Distribution of the concentration of the number of particles for the different types of soot emitted, per mobility diameter (Völk and Hörnig, 2011)

and since it typically ranges over several orders of magnitude, it is useful to describe the size interval as $d \log(d_p)$, and the size distribution will transform to:

$$dN = n(\log d_p) d \log(d_p) \quad (5.4)$$

In addition to this, the particle size distribution can be viewed in two different ways: as frequency of distribution of the number of particles, such as displayed in Figure 5.4 and the exemplary curve for the function $q_r(x)$ in Figure 5.5, or as the cumulative distribution, corresponding to the curve of $Q_r(x)$ (Stieß, 2008). They both contain the same information but the differential plot gives a better presentation of the distribution shape, and any effect of bias is not propagated throughout the entire size plot, as it happens with the cumulative plot. However, this last one provides a better estimate of the median diameter of the aerosol. The cumulative distribution is a percentage distribution of the variable displayed, which indicates always its value below a certain x value, between x_{min} and x_{max} . It can be described by the following equation:

$$Q_r(x_i) = \frac{\text{partial amount of variable } x_{min} \dots x_i}{\text{total amount of variable } x_{min} \dots x_{max}} \quad (5.5)$$

The curve of $Q_r(x)$ cannot decrease because the proportion of the quantity below a particle size x_b cannot be lower as the one below x_a , if $x_a < x_b$, since this curve

represents the cumulative values of the frequency distribution curve (Stieß, 2008). The following is also to be noted:

- for $x \leq x_{min}$, then $Q_r(x) = 0$;
- for $x > x_{max}$, then $Q_r(x) = 1$.

On the other hand, the frequency distribution shows the total amount of the variable inserted in one specific interval of sizes, a specific class, and can be defined by (Stieß, 2008):

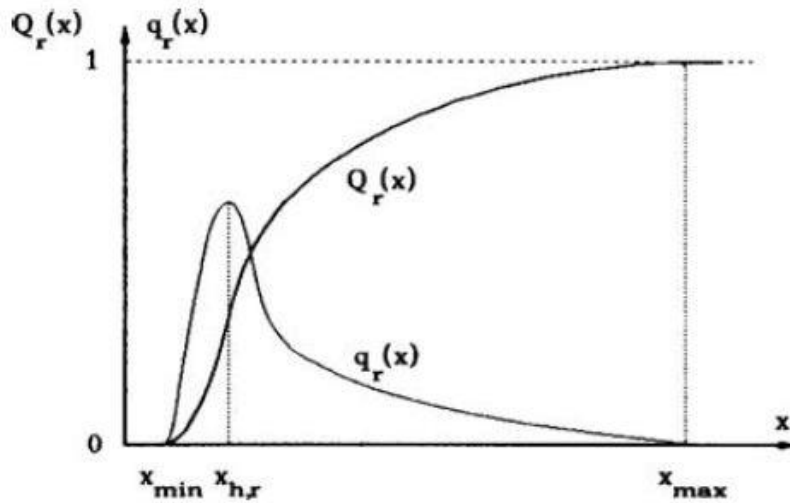


Figure 5.5 - Frequency and cumulative distribution of a normal distribution (Stieß, 2008)

$$q_r(x_i) = \frac{\text{amount of variable } x_{i-1} \dots x_i}{\text{total amount of variable} \cdot \text{width of the interval}} \quad (5.6)$$

The frequency distribution can be obtained from the cumulative distribution through the following relationship:

$$q_{r,i} = \frac{Q_r(x_i) - Q_r(x_{i-1})}{\Delta x_i} = \frac{\Delta Q_{r,i}}{\Delta x_i} \quad (5.7)$$

Thus, sustaining the differential:

$$q_r(x) = \frac{dQ_r(x)}{dx} \quad (5.8)$$

Finally, it is possible to conclude that:

$$\int_{x_{min}}^{x_{max}} q_r(x) dx = Q_r(x_{max}) = 1 \quad (5.9)$$

From the equations above the following ought to be highlighted:

- for $x \leq x_{min}$, then $q_r(x) = 0$;
- for $x \geq x_{max}$, then $q_r(x) = 0$;
- for $x = x_{IP}$, at the inflection point IP of the function $Q_r(x)$, then $q_r(x)$ has a maximum value.

This relationship described verifies for every type of particle variable depicted on the y-axis, such as: number, length, area, volume and mass. Particle number is the variable type addressed in this chapter and it is the only one of interest for the scope of this dissertation and simulations.

In order to assess the influence of the particle size distribution in the soot deposition along the cooler, the mass fractions of several particle classes were calculated. As indicated by the routine supplier, five different particle classes should be introduced as input parameters in the UDF code in order to perform the simulation. Therefore, the mass fractions were calculated both for the Motor test bench, as for the Model test bench, respectively: the curve for 2000 1/min@5bar and the *Flammruß* (flame soot) curve in Figure 5.4. However, the cumulative curve allows the acquirement of a better feeling of the distribution of the particles, reason why the curves of Figure 5.4 were integrated in order to obtain their respective cumulative displays, and regarding the particles mass. They provided a clearer view of the weight of the different particle diameters in the total soot mass and sped up the determination of the mass fractions for each source, and for the different particle size classes, whose influence will be analysed in further chapters. This was achieved with the use of 20 points in both the diesel and flame soot curves, allowing the obtaining of a smoother and more precise curve, than if using a smaller number of points. The curves in the following graphic were obtained by converting the distribution curve from Figure 5.4 in a finite number of zones (histogram view of the distribution): whose middle values correspond to the 20 points. Thus, 20 different columns were drawn, as the example sketched in Figure 5.6.

In every column, a medium diameter (corresponding to the chosen point) is defined as:

$$D_{med} = \frac{D_i + D_{i+1}}{2} \quad (5.10)$$

based on the extreme diameters of every column. The value of particle number concentration corresponding to that medium diameter is then read on the y-axis.

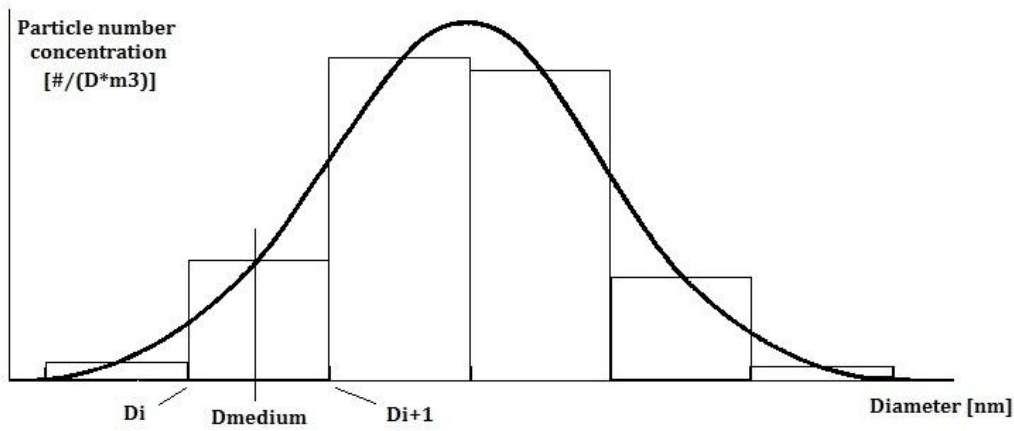


Figure 5.6 - Scheme of the calculation of the area of each of the classes chosen to obtain the cumulative mass particle distribution

Therefore, the area of one column/rectangle will be given by:

$$Area_{1column} = \left[\frac{\# \text{ particles}}{D \cdot \text{volume}} \right] \cdot [D_{i+1} - D_i] \quad (5.11)$$

and the approximate total area under the curve of particle number concentration is given by the sum of the areas of the columns, since the unit of the values read on the y-axis of the graphic are $[\# / (\text{diameter} \times \text{m}^3)]$. As it is a continuous distribution, the number concentration is proportional to the area of each column. Consequently, the mass of particles (per unit volume) associated with one specific medium diameter is given by:

$$m_{1class} = \frac{\# \text{ particles}}{D \cdot \text{volume}} \cdot [D_{i+1} - D_i] \cdot m_{1particle} \quad (5.12)$$

The mass of one diesel particle was obtained by the equation (5.12) recommended by Cambustion (2009) for size relationship with mass:

$$m_{1particle} [\mu g] = 2.20 \cdot 10^{-15} \cdot D_p^{2.65} [nm] \quad (5.13)$$

which was derived from the particle mass measurement with DMS spectrometers.

Finally, the mass fractions for one particular class with the respective medium diameter are given by:

$$mass\ fraction_{1class} = \frac{mass\ of\ one\ class}{mass\ of\ all\ classes} \quad (5.14)$$

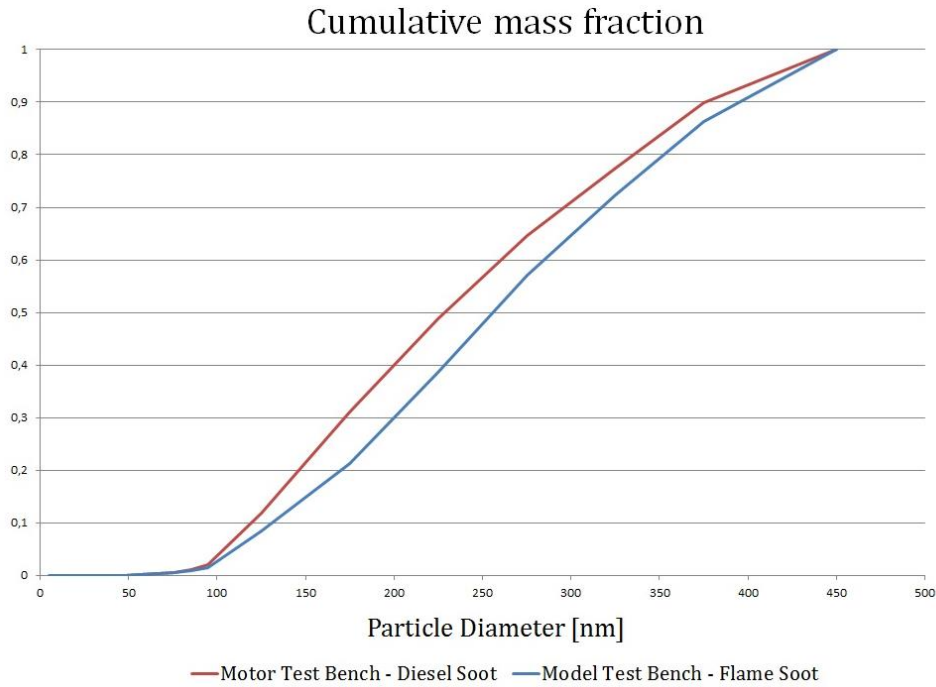


Figure 5.7 - Cumulative Mass Fraction Curves for Diesel and Flame Soot

Above in Figure 5.7, the two cumulative mass fractions curves for both test bench soot types are obtained, through the process explained. An approximation with a 3rd degree polynomial and a Fourier transform was pursued, in order to determine the mass fraction values for the diameter classes chosen for the model test bench. They are both fairly precise in adjusting the curve with the original mass fraction values, as one can see in Figure 5.8.

Hence, the 3rd degree polynomial curve was chosen for this effect. Through the use of the equation that defines this polynomial, the values for every extreme point of each of the five classes are introduced in the equation and the respective mass fractions are obtained.

To obtain the mass fractions for the five chosen medium diameters, one simply has to subtract the lower value of the class to the higher value of the class, for instance: to determine the mass fraction matching the diameter of 125 nm one needs to subtract the

mass fraction from 100 nm to the one from 150 nm. The final mass fractions for both model and motor test bench are displayed in Tables 5.3 and 5.4.

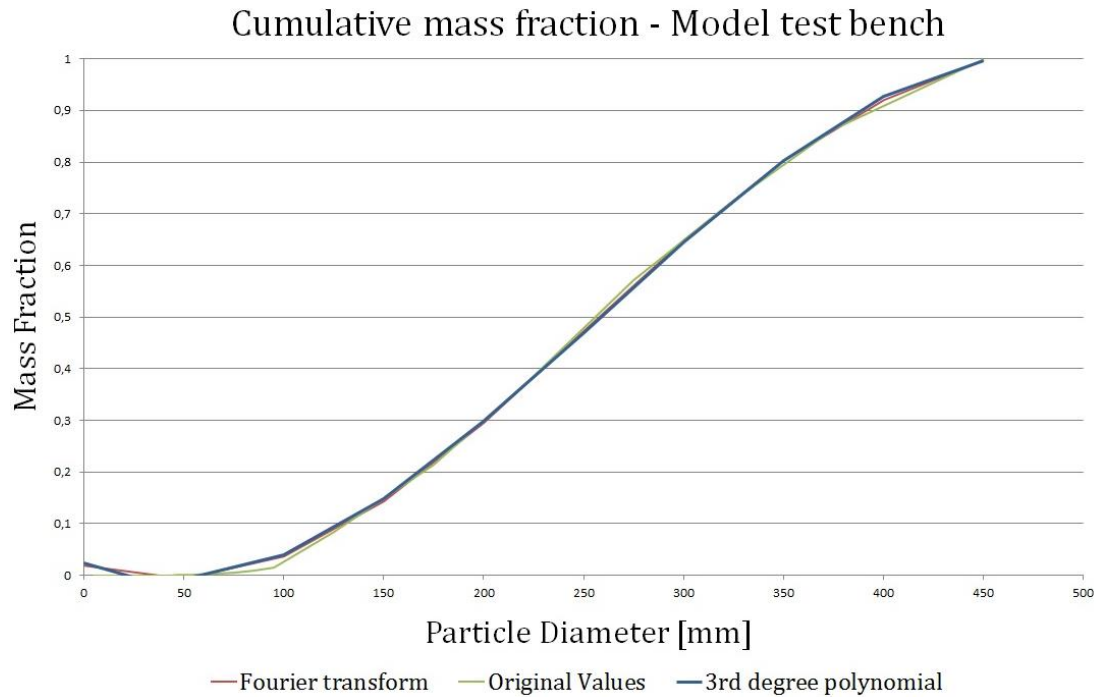


Figure 5.8 - Approximation curves to the cumulative mass fractions of the model test bench

Class Medium Diameter [nm]	Mass Fraction
50	0.04
125	0.11
175	0.15
275	0.51
400	0.19

Table 5.3 - Mass fractions associated to flame soot – model test bench - to be used as input for the UDFs

Regarding the motor test bench also two curves were used to approximate the original cumulative mass fractions curve. However, in this one, one can conclude from Figure 5.9 that the Fourier transform approximates better the original curve than the polynomial case does.

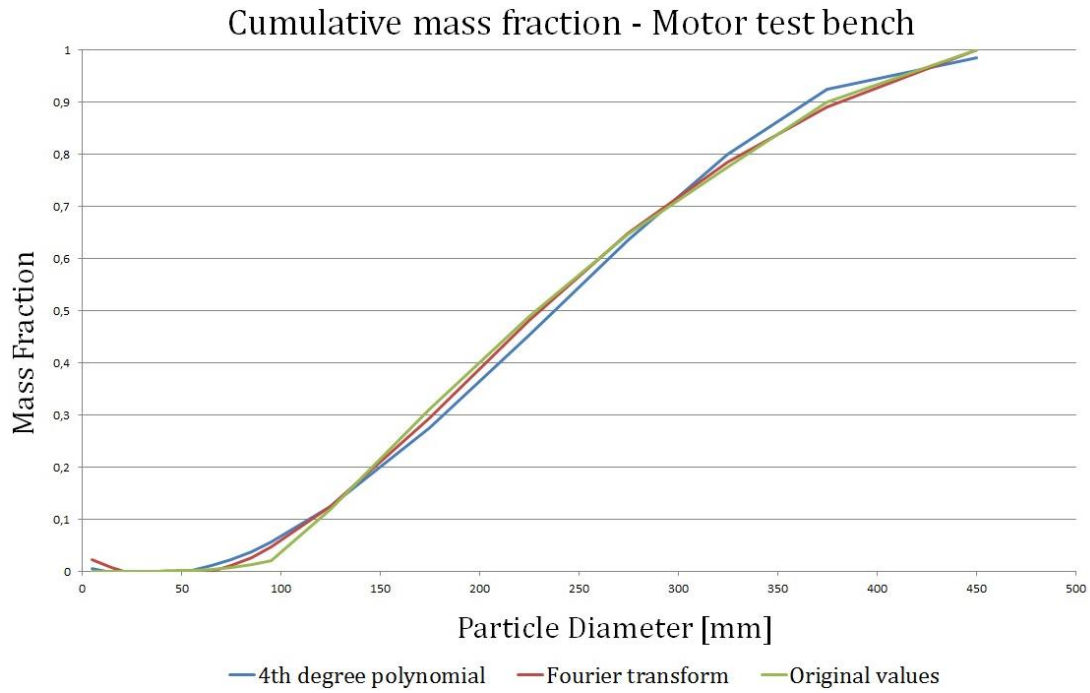


Figure 5.9 - Approximation curves to the cumulative mass fractions of the motor test bench

Class Medium Diameter [nm]	Mass Fraction
50	0.06
125	0.15
175	0.18
275	0.45
400	0.16

Table 5.4 - Mass fractions associated to diesel soot from motor test bench to be used as input for the UDF

5.6. Motor Test Bench

This test bench consisted on using the exhaust gas from a real diesel engine and linking it to the simple cooler geometry displayed above, where part of the exhaust gas can be recirculated. The exhaust gas provided for this test bench assessment was procured from a turbocharged BMW M57D30 Tü2 diesel engine, in three different load operating points: 1000 min^{-1} at 1 bar, 1500 min^{-1} at 3 bar and 2000 min^{-1} at 5 bar (mean effective pressure p_{me}). For the simulations performed about the motor test bench, however, only the highest load point was used, as it can be seen ahead in Figure 5.5., this is the point with highest particle concentrations; for this reason, Völk and Hörnig (2011) chose it to draw conclusions in their measurements.

The other two lower operating points generated very low deposition values (due to low particle concentration) in order to allow a valid analysis of the fouling effect. As the exhaust gas characteristics are fully defined for this engine, it is not possible to make singular changes in its composition, such as, for example, increasing the particle or HC-concentration, which would raise the propensity for soot deposition. For the chosen operating point, three different EGR-Rates were used to perform the measurements. Their values and corresponding exhaust gas data are displayed in Table 5.5:

EGR-Rate [kg/h]	4.5	9	15
Re [-]	5600	11000	16000
HC [ppm]	350-450	350-450	350-450
Soot mass flow [mg/h]	91.15	210.01	575.30

Table 5.5 - Exhaust gas data for the three different EGR-Rates used in model test bench by Völk and Hörnig (2011)

In this test bench, two different cases were picked from Völk and Hörnig (2011) to use as basis for comparison with the simulations performed, from here on described as Test Case 1 and Test Case 2. Their measurement results are described ahead in Chapter 7. The concentrations in ppm refer to a mass ratio.

	Test Case 1	Test Case 2
Variable Description	Value	Value
Exhaust gas mass flow [kg/s]	9	15
Exhaust gas inlet temperature [°C]	300	380
Soot mass flow [mg/h]	210.01	575.3
Cooling water temperature [°C]	20, 40, 60, 80	20, 80
HC concentration [ppm]	350 - 450	350 - 450
SO ₂ concentration [ppm]	2.17	2.17
H ₂ O concentration [kg/kg]	2% - 11%	2% - 11%

Table 5.6 - Description of the main input parameters for Test Cases 1 and 2 of motor test bench

5.7. Model Test Bench

In pursuance of isolating the different influences on the particle deposition in the motor test bench, a new test bench was developed, where the exhaust aerosol constitution was previously modelled, that is, it is fully customizable. Here follows a scheme of the setup used to produce this modelled exhaust gas.

The model aerosol used for the model test bench was obtained by first heating dry particulate free pressurized air up to a desired temperature, reaching values of 450 °C. Amounts of water, sulphuric acid and diesel fuel were added to the aerosol in values comparable to real diesel exhaust emissions. HC concentration levels were chosen based on previous measurements, which places them between 200 and 450 ppm – relatively high values. Thereafter, soot produced by a flame or diffusion burner is added to the heated gas (Hörnig *et al.*, 2011).

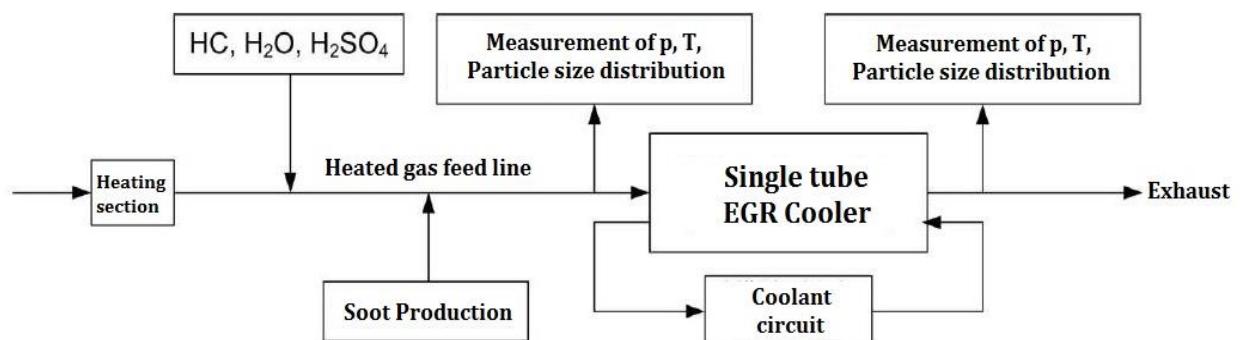


Figure 5.10 - Schematic setup of the model test bench (adapted) (Völk and Hörnig, 2011)

In this test bench, different exhaust gas compositions can be created through the variation in content of: water, adsorption of hydrocarbons and sulphur to the soot particles. The soot used for this test bench was polydisperse flame soot from a propane burner with $\dot{m}_{soot} = 45 \text{ mg/h}$, defined by Völk and Hörnig (2011, 2012). Their study also contemplated another type of soot; it had, however, too low particle concentration to allow significant deposition that could have a valid analysis, as it can be seen in Figure 5.4 by the name of *GfG-Ruß* (soot from Graphite radio-aerosol). Furthermore, the soot source chosen provides a rather high and constant mass flow, crucial for a good fouling analysis in this test bench. Also, it is important to notice that both modelled types of soot have a particle size distribution that is comparable the one of real diesel engine soot, used in motor test bench, as can be verified by the Figure 5.4.

The exhaust gas flow (EGR-Rate) used for this test bench varied from 5 to 15 kg/h but the one chosen for the simulations was $\dot{m}_{exhaust} = 5.5 \text{ kg/h}$, which verified for every operating point described in Table 5.7.

Working point	T_{in} [°C]	T_{water} [°C]	T_{out} [°C]	$T_{in} - T_{out}$ [°C]	c_{HC} [ppm]	c_{H_2O} [%]	$c_{H_2SO_4}$ [ppm]
E	300	20	240	60	0	0	0
F	150	20	110	40	430	2	0.2
G	150	80	130	20	430	2	0.2
I	150	20	110	40	430	2	10
J	400	20	320	80	230	0	0
K	400	20	320	80	430	0	0
L	400	20	330	70	0	2	0
M	400	20	290	110	0	5	0
N	400	20	310	90	430	2	0
O	400	20	320	80	430	2	10

Table 5.7 - Description of the several operating points used at the motor test bench by Völk and Hörnig (2011)

6. Simulation Parameters

Firstly, it will be made an approach to the study referred before, performed by Völk and Hörnig (2011, 2012), using basic *Fluent*. Simulations will be made based on some of their measurements and results obtained both for the motor and model test bench. The first simulations performed were carried out using the above-mentioned geometry (Chapter 5). The goal of this dissertation was to validate the routines with the described model and then study the behavior of a real cooler from DEUTZ AG.

In order to perform the simulations, it is necessary to determine all the inputs for the UDFs regarding the exhaust gas characteristics. According to the UDF code the following values in the exhaust gas ought to be pre-established and inserted in the respective parameter routine:

Parameter of Exhaust Gas	Units
Soot concentration	kg/kg
HC concentration	ppm
SO ₂ concentration	ppm
Water vapour concentration	kg/kg
Soot particle for five diameter classes	m
Soot particle mass fraction for five diameter classes	-
Soot particle density	kg/m ³
Soot deposit density	kg/m ³
Molar fraction of short chain HC	-
Mass fraction of SO ₂ converted to sulfuric acid	-

Table 6.1 - Parameters introduced in the UDFs

6.1. Features obtained by the UDFs

The routines used throughout this work at DEUTZ AG were provided by Atlanting GmbH and they made possible to perform the calculations necessary to do a fouling analysis on EGR coolers, as the *Fluent* software does not have such specific features. These routines control whether the steady state or the maximal possible deposit thickness has been reached. If the state of equilibrium is achieved, species concentrations do not vary anymore over the cooler length.

Regarding the exhaust gas composition, all the relevant species for the fouling effect were considered and contemplated in the User Defined Functions: particulate matter, H_2O , HC and H_2SO_4 . It is important to highlight that the condensation of these species depends on their respective vapour pressure curves, which are depicted in Figure 6.1. Also, the particle mass fractions have to be introduced in the code, as calculated for the 5 different size classes in section 5.5.

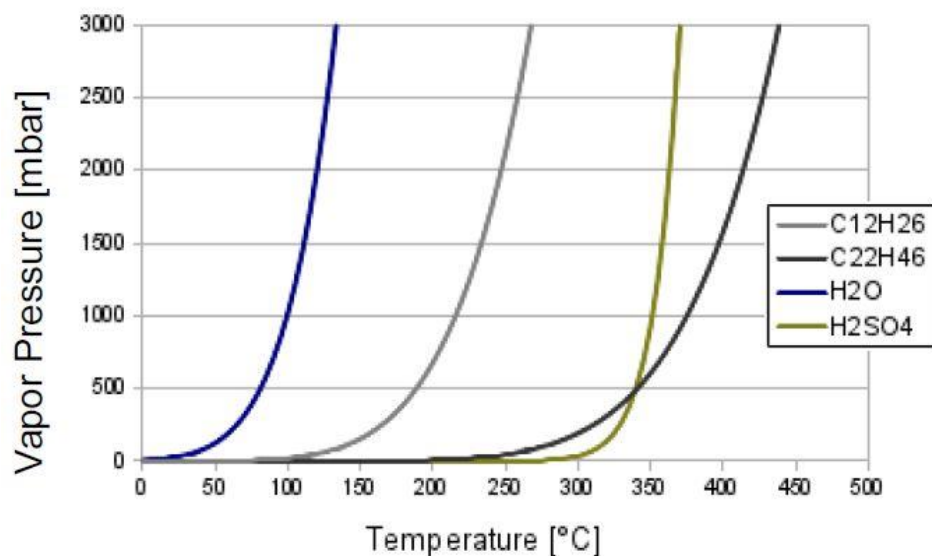


Figure 6.1 - Vapor pressure curves for every single species considered for the exhaust gas in the routines (Atlanting, 2016)

6.1.1. Deposit Thickness

The maximum allowed deposit layer thickness has to be demarcated when defining the setup for the CHT solution, and cannot be changed thereafter. The deposit thickness obtained by the measurements of Völk (2014) in his additional work showed almost no radial variation, that is, in a cross-section of the tube, the deposit thickness was somewhat constant, even though the cooler was positioned horizontally during the

experiments: this indicates a generally radial symmetric development of the deposit layer along the cooler, as long as high amounts of condensate do not emerge.

Therefore, deposit thickness ought to be determined by an Area-Weighted Average (in the Surface Integrals reports), in order to provide the averaged value of deposit thickness in the entire cooler. The local consistency of the deposit layer depends on the amount of soot and condensate that compose the layer.

Usually higher cooling temperatures lead to a faster attainment of a steady state, as the fouling is dry and creates, therefore, a thinner ultimate deposit thickness. If the number of adhesive components present in the exhaust gas is reduced, then so will the values of the correspondent maximum deposition mass and thickness, as the fouling will be less sticky (Völk and Hörnig, 2011).

6.1.2. Deposition Rates

The deposition rates concern only the soot deposition, that is, the deposition of the particles of the five different classes, having nothing to do with the species condensation. Obtaining these allows the direct determination of the deposition efficiency.

6.1.3. Deposition Efficiency

The deposition efficiency can be determined with *Fluent* and the routines, despite not being one of the features obtained by the UDFs. In order to do that it is necessary to firstly understand the concept of deposition efficiency: it describes how effective the deposition mechanisms are, which allows taking conclusions regarding the propensity for fouling to occur under the given circumstances in the EGR cooler. The equation (6.1) by Völk and Hörnig (2011) describes the deposition efficiency for each particle diameter through the entire cooler:

$$\varepsilon(dp) = \frac{c_{in}(dp) - c_{out}(dp)}{c_{in}(dp)} \quad (6.1)$$

On the other hand the equation (6.2) by Hörnig et al. (2011) describes the deposition efficiency of a determined particle size depending on the position in cooler length.

$$\varepsilon_n = \frac{N(x=0) - N(x)}{N(x=0)} \cdot 100\% \quad (6.2)$$

The deposition efficiency matches the percentage of the integral of the five different deposition rates (from the five diameter classes), in the total amount of soot mass flow in the initially clean cooler. That is, after the CHT calculations have been performed and the fouling simulation mode has been activated, with the hooking of the UDFs, the deposition rates can be determined (Atlanting, 2016).

$$\varepsilon = \frac{\text{Sum of the integrals of the five deposition rates}}{\text{Total soot mass flow}} \quad (6.3)$$

Because the routines provide the deposition rates in $\left[\frac{\text{kg}}{\text{m}^2 \cdot \text{s}}\right]$, each deposition rate needs to be read as the integral on the cooler inner surface (wall on the gas side), in order to obtain the deposition rate in $[\text{kg/s}]$. Then the deposition efficiency can be calculated as seen on Equation (6.3).

Regarding the Surface Integration available as a *Fluent* tool, one can read the values of parameters through different types of reports, such as: Area-weighted average, Integral, Mass-weighted average, etc. Depending on the parameter one wants to read, some reports are better suited than others. Regarding the case of the deposition rates, an Integral Surface Report was chosen, in detriment of the Area-Weighted Average. This choice is supported by the statement that the integral on a surface is computed by summing the product of the facet area and the selected field variable facet value. In simpler words, it means that each cell value for - in this case - the deposition rate is multiplied by the respective cell area, and then all values are summed, as expressed in the Equation (6.4) (ANSYS, 2016b):

$$\int \phi dA = \sum_{i=1}^n \phi_i |A_i| \quad (6.4)$$

Each facet is associated with a cell in the mesh. Therefore, the values of a certain variable on facets are the same as the cell values for the respective variable.

After a few iterations – here the simulation runs for about six iterations, as the first ones have high peak values, which are inconsistent and may be the result of numerical errors – the integral of the five deposition rates can be read as a Surface Integral Report in *Fluent*. It must be referred that node values should not be enabled when post-processing

any surfaces associated with shells (wall, shadow wall, etc.). The layers share nodes at the edges, and thus the averaging of such nodes is erroneous.

6.1.4. Condensation Rates

The condensation rates are one of the features possible to obtain directly from the routines. They contemplate the condensation rates for the following species: hydrocarbons of long and short-chain – respectively $C_{24}H_{46}$ and $C_{12}H_{26}$ -, sulphur (SO_2) and water. They are an important indicator for the speed of the deposit formation under the given operating conditions. The effect of diffusiophoresis calculated in the routines is neglected if the mass fraction of all condensate forming species is less than $1.0e-14$ kg/kg.

One important value that can help understand the impact of the different condensate forming species is their molecular weight. The values used in the routines are displayed on Table 6.2:

Species	H ₂ O	H ₂ SO ₄	C ₁₂ H ₂₆	C ₂₂ H ₄₆
Molecular weight [kg/kmol]	18	98	170	310

Table 6.2 - Values of the molecular weight of the condensate forming species used in the routines

6.1.5. Fouling Factor

This parameter indicates the thermal resistance due to the deposit layer. It is determined by:

$$Fouling\ factor = \frac{Deposit\ thickness}{Thermal\ conductivity} \left[\frac{(m^2 K)}{W} \right] \quad (6.5)$$

6.1.6. Thermal Conductivity

The reduction of the cooling performance due to the fouling effect is determined by the impact on local thermal conductivity. A lower thermal conductivity worsens the heat transfer, thus decreasing the ultimate performance of the cooler. This parameter allows to ascertain the composition of the deposit layer: when there is no condensation of species, the thermal conductivity of the layer is $0.1\ W/(m\ K)$, that is, it means that it is composed by pure soot – therefore, it is the minimal value possible to have. On the other

hand, when there are condensates present in the deposit layer, the value for the thermal conductivity rises above 0.1 W/(m K).

Condensate species	Thermal Conductivity [W/(m K)]
C ₁₂ H ₂₆	0.3200
C ₂₄ H ₄₆	0.3100
H ₂ SO ₄	0.2600
H ₂ O	0.5562

Table 6.3 - Thermal conductivity associated with each of the condensate species (Atlanting, 2017)

The typical values for thermal conductivity associated with each of the condensate forming species are depicted in Table 6.3 and should be taken in account in order to understand their impact on the final thermal conductivity of the deposit layer.

6.2. Influence of Particle Size

Mass fraction 100% diameter [nm]	Deposit thickness [mm]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conductivity [W/(m K)]
10	0.9036	290.74	314.69	0.273
50	0.8946		314.13	0.278
75	0.8551		313.27	0.279
100	0.7862		311.76	0.280
125	0.6925		309.64	0.280
150	0.5838		307.09	0.281
175	0.4705		304.26	0.281
200	0.3599		301.34	0.281
250	0.1616		295.55	0.283
300	0.0160		290.91	0.156
400	0		290.70	0.100

Table 6.4 - Study of the weight of the different particle sizes in the deposit thickness

To understand the weight and impact of the different soot particle sizes in the overall fouling effect regarding the model test bench, several simulations were performed, attributing 100% of the particles mass to one particular size class at a time.

As it can be seen in Table 6.4, the smaller particles have a much higher tendency to deposit than particles of bigger size. Under the present circumstances, particles from around 400 nm and above no longer deposit at the wall. The deposition forces depend, therefore, on the size of the particle. As stated in Section 2.4.5 the smaller particles are more easily driven towards the wall by eddy diffusion than the bigger ones. Also, as seen by Figure 2.14 from Section 2.7.2, when the velocity of the flow is so high that the particles at the wall are removed by gas shear stress, this critic velocity that determines if they are indeed removed or not, increases with the decrease of particle diameter. Therefore, it means that the bigger particles are much easier to remove. All these factors explain how the bigger particles may not have any influence in the deposition phenomenon – in this case, for the model used and in test case K, particles above 400 nm of diameter do not play anymore a role in the fouling effect.

In the model defined by Atlanting, the size of the particle on the wall will be determined as the average value of the deposited particles. There it can happen that, for sensitive/empiric conditions, the deposition stops when the particle is in average too big.

6.3. Influence of Layer Increment Parameter

The layer increment is a parameter portrayed in the routines which determines the increase in the build-up of deposit layer in each iteration step. The smaller the value, the higher the simulation accuracy will be, providing more precise results, however with higher computational effort and more time; small increments lead to a more extensive study and calculation of the deposit formation, which takes more time. The default value used is 0.01 mm. In order to assess the influence of this parameter, an analysis was conducted using the operating point K with three different layer increment values, for the five different particle sizes, until the thickness of 0.1 mm was reached. However, only P3 is displayed as the other particle classes follow the same tendency.

As observed in Figure 6.2, the layer increment value determines how fast the deposition rate reaches an almost constant value, which induces a linear deposit thickness growth.

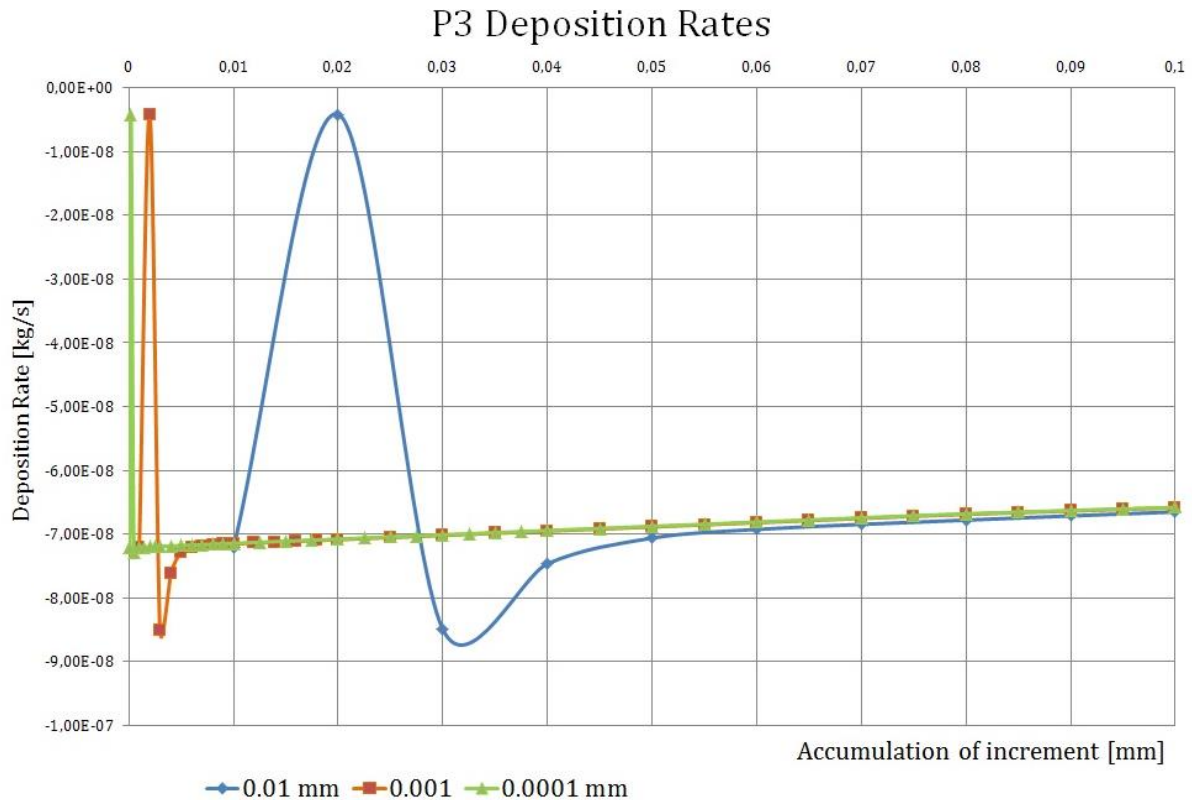


Figure 6.2 - Graphic displaying the evolution of deposition rate for particle class P3 with three different layer increments

The value cannot exactly be described as constant – as it is further ahead when the simulation reaches the fouling steady state. The deposition rate will permanently decrease until it reaches zero at the equilibrium state.

However, it can be considered to achieve a constant value in the first few iterations, which is the value sought for the deposition rate that will allow to determine the deposition efficiency. The correct deposition rate value is not exactly the one corresponding the first iteration because initially the result oscillates considerably, which might be indicative of numerical errors and may not match the reality, inducing errors in the determination of the deposition efficiency. Therefore, an empiric analysis was conducted to determine how many iterations would suffice to describe the initial deposition efficiency correctly. The deposition rate in the first few iterations assumes the highest value of deposition rate throughout the whole process, because it occurs when the cooler is still clean and the heat transfer process is starting to take place, thus having the highest level of thermophoretic effect and, consequently, a higher deposition rate.

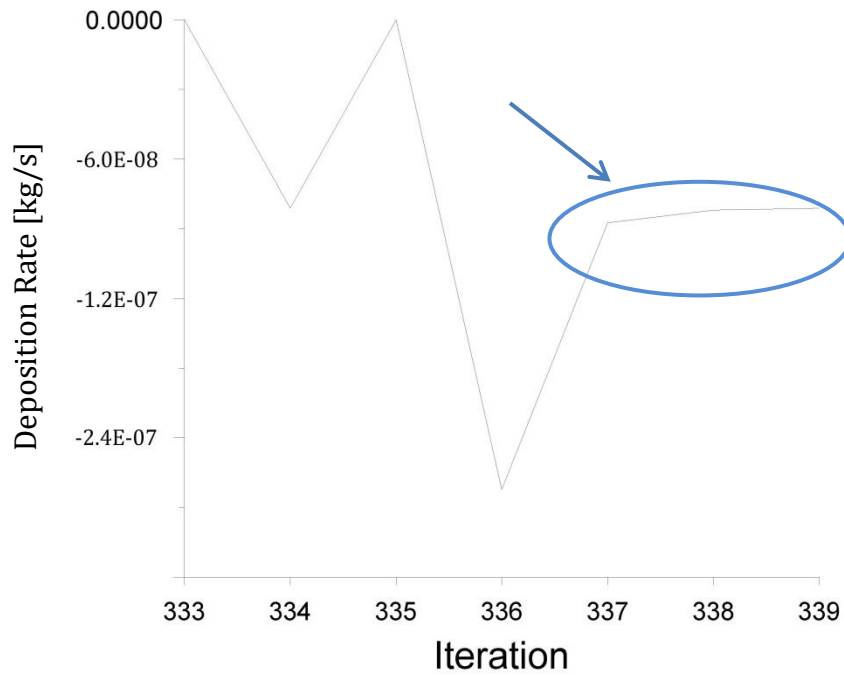


Figure 6.4 - Evolution of the deposition rate of particle class P3 in the first six iterations for operating point K

As it can be seen in Figures 6.3 and 6.4, where the point sought is highlighted in blue, it reaches a semi-constant value at first, after six iterations of inconsistent values. Then, as the deposition proceeds, the deposition rate will continuously decrease, until it reaches zero or, as it is probably the case depicted in Figure 6.4, the removal rates equal the deposition rates, and the final deposit thickness is obtained, as it no longer increases.

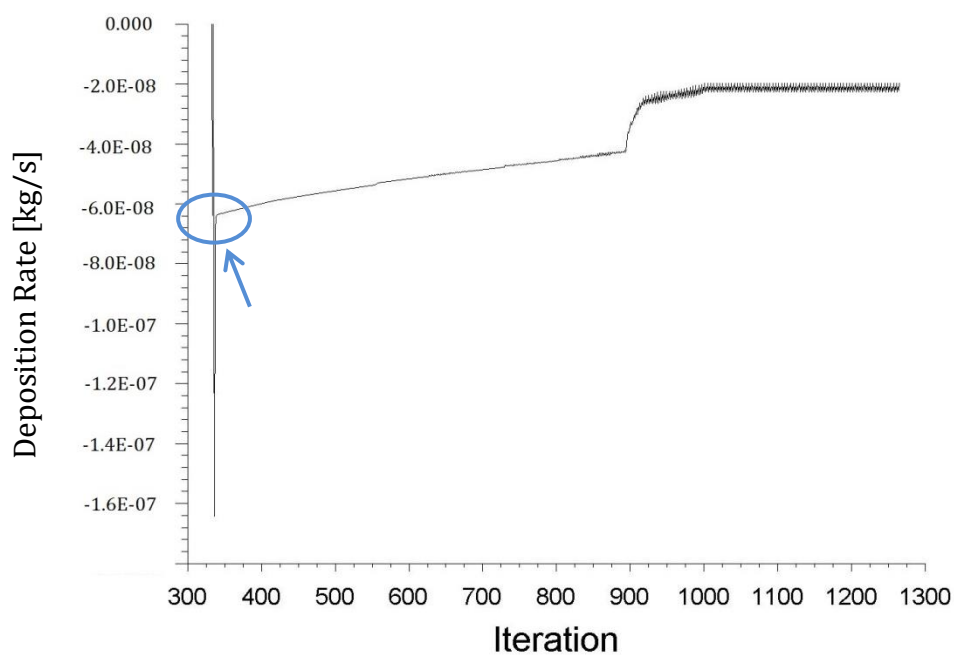


Figure 6.3 - Evolution of the deposition rate of particle class P3 for operating point K

Bearing this in mind, the increment used for the execution of the simulations will be 0.001 mm, suggested by the routine supplier, whom advises to choose a smaller increment value, so that almost no layer is built in each iteration, enabling the integration of the deposition rate after a few iterations (six as determined by the analysis of the graphics), once the value has slightly stabilized, thus determining the most correct deposition efficiency that is possible to obtain.

7. Simulations and Results

7.1. Simulations – Model Test Bench

The model test bench was the first to be studied. From the operating conditions for this test bench enunciated previously, five points were chosen to execute the simulations. As mentioned in section 5.7 the $\dot{m}_{exhaust}$ for this test bench is 5.5 kg/h.

Working point	T_{in} [°C]	T_{water} [°C]	T_{out} [°C]	$T_{in} - T_{out}$ [°C]	c_{HC} [ppm]	c_{H_2O} [%]	$c_{H_2SO_4}$ [ppm]
E	300	20	240	60	0	0	0
F	150	20	110	40	430	2	0.2
G	150	80	130	20	430	2	0.2
K	400	20	320	80	430	0	0
N	400	20	310	90	430	2	0

Table 7.1- Operating points from the model test bench chosen to perform simulations

From Völk and Hörnig (2011), the only result obtained from the measurements beside the outlet temperature is the deposition efficiency for every one of the points described above. Both the deposit thickness and the thermal conductivity are not provided in their study, but they will be determined in this work, so that one can get a perception of the relationship between these values and the deposition efficiency.

Working point	Deposition Efficiency [%]
E	13.6
F	33.3 – 44.4
G	22.2
K	17.8 – 22.2
N	15.6

Table 7.2 - Result values for the deposition efficiency obtained in the measurements by Völk and Hörnig (2011) for the five chosen operating points

The results obtained for the five different operating points are displayed in the table 7.3, along with the values obtained for deposition efficiency by Völk and Hörnig (2011), in order to allow an easier comparison:

	Simulation Results					Measurement Results from Völk and Hörnig (2011)	
Working point	Dep. Thick. [mm]	Deposition Efficiency [%]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conduct. [$\frac{W}{(m K)}$]	Deposition Efficiency [%]	T_{out} Fouling [°C]
E	0.1521	0.91	221.68	230.31	0.1000	13.6	240
F	0.3402	136.52	115.58	117.32	0.4969	33.3 – 44.4	110
G	0.2818	2.95	132.43	133.58	0.2802	22.2	130
K	0.1471	3.50	290.73	295.08	0.2830	17.8 – 22.2	320
N	0.1440	31.73	290.73	293.46	0.4609	15.6	310

Table 7.3 - Results obtained for the simulations of the five chosen operating points from the model test bench and the values of deposition efficiency and outlet temperature from Völk and Hörnig (2011)

When analysing Table 7.3, the deposition efficiencies obtained for these five operating points are very divergent, even so because the deposition efficiency of point F is above 100%, which apparently does not make sense. In order to understand what parameters have influence in the final result, a study will be made with an average point – point K – where every single parameter will be varied singularly, in order to see how it influences the outcome. However, some results can be easily explained: the point with highest

thermal conductivity – point F - shows the highest deposit thickness, which makes sense since a higher thermal conductivity enables a better heat transfer which, on its own, generates a greater thermophoretic effect that will lead to a greater deposition of soot and, therefore, to a thicker deposit. Accordingly, this same point is the one with the highest deposition efficiency. The operating point E is one extreme point characterized by pure soot, i.e. there are no condensate forming species, which indicates that the thermal conductivity of the deposit thickness matches the one of pure soot, 0.1 W/(m K). Nonetheless, the overall results are unsatisfying, as the values of the deposition efficiencies do not match the same order of magnitude as the ones provided by Völk and Hörnig (2011). In the next sections reasons for these discrepancies will be studied.

7.2. Results of Single Effect Variation

This section of Chapter 7 refers to a series of simulations that were performed for a chosen intermediate point: point K in the model test bench. Several variables were varied, one at a time, to assess the weight and influence of each one on the final results of the CFD analysis. For this analysis, no data for comparison is available in the sources. However, it allows one to have a perception of the influence of each of the variables in the final results. Such variables were:

- T_{water} ;
- $T_{exhaust}$;
- HC concentration;
- H₂O concentration;
- SO₂ concentration;
- $\dot{m}_{exhaust}$;
- $\dot{m}_{soot}$.

The original results for point K, without any variable variation are showed on Table 7.4:

Deposit Thickness [mm]	Deposition Efficiency [%]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conductivity [$\frac{W}{(m K)}$]
0.1471	3.500	290.73	295.08	0.2830

Table 7.4 - Original results obtained for the simulations of point K

7.2.1. Variation of water temperature

The first variation parameter consisted of changing the water temperature (constant value of temperature on the water-side of the cooler), and keeping the exhaust gas at 400°C and all the species concentration at zero. From Table 7.5 one can conclude that the tendency observed is that, with increasing water temperature, the deposit thickness progressively decreases, and so does the deposition efficiency. This is due to a lower temperature difference which reduces the thermophoretic effect.

T_{water} [°C]	Deposit Thickness [mm]	Deposition Efficiency [%]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conductivity $\left[\frac{W}{(m K)}\right]$
20	0.1471	3.500	290.73	295.08	0.2830
40	0.1410	3.001	298.08	301.82	0.2818
60	0.1293	2.654	305.06	308.04	0.2844
80	0.1275	2.314	311.56	314.24	0.2862

Table 7.5 - Results on the variation of the water-side temperature for test case K

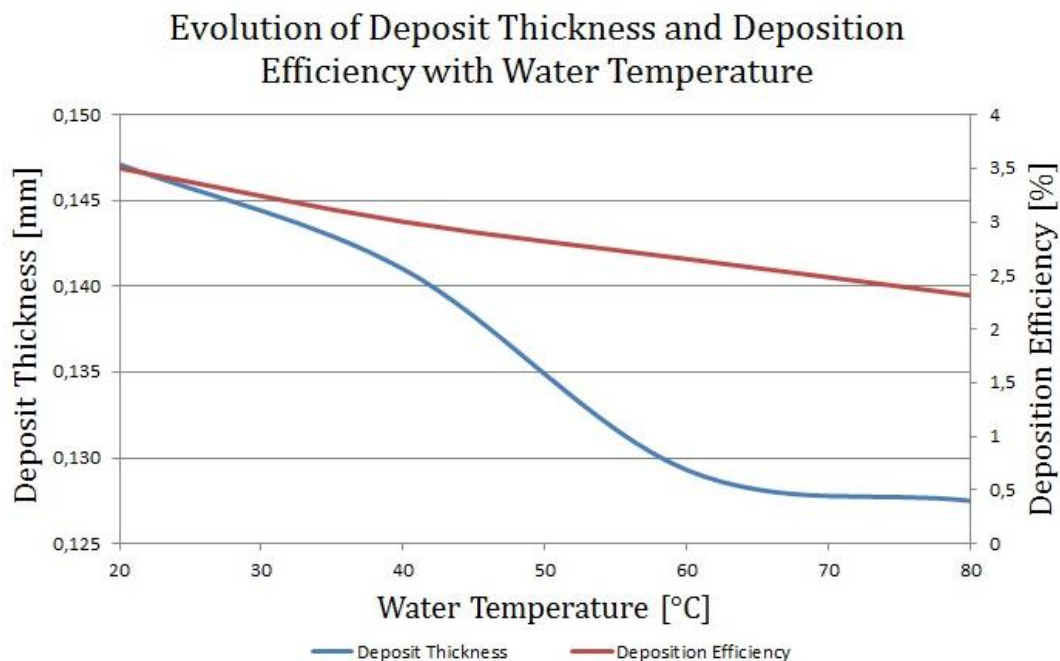


Figure 7.1 - Evolution of the deposit thickness and deposition efficiency with the variation of water temperature for test case K

On the other hand, the temperature at the outlet of the cooler progressively increases, which is coherent with the fact that a higher deposit thickness worsens the heat transfer, as the thermophoretic effect is endorsed by a higher temperature gradient. This evidenced by the Figure 7.1, which shows a graphic of the data from Table 7.5.

In addition, the variation of thermal conductivity is very small, as no change is made in the condensate or soot concentrations, which are the main factors to contribute to its variation.

7.2.1. Variation of exhaust gas temperature

Regarding the exhaust gas temperature, two different temperatures below and one above the 400°C from point K were used for the analysis of this parameter, in order to allow a perception of the evolution of the deposit building when the temperature gradient is higher and lower to the one of test case K. The results are shown in Table 7.6.

$T_{exhaust}$ [°C]	T_{water} [°C]	Deposit Thickness [mm]	Deposition Efficiency [%]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conductivity [$\frac{W}{(m K)}$]
200	20	0.2759	1.394	151.41	154.68	0.3000
	80	0.2518	0.749	169.20	170.92	0.3018
300	20	0.2081	2.336	221.61	225.89	0.2917
	80	0.1753	1.467	268.09	243.47	0.2944
400	20	0.1471	3.500	290.73	295.08	0.2830
	80	0.1275	2.314	311.56	314.24	0.2862
500	20	0.0937	4.774	358.19	361.74	0.2753
	80	0.0622	3.300	381.42	382.88	0.2759

Table 7.6 - First results on the variation of the exhaust gas temperature for test case K

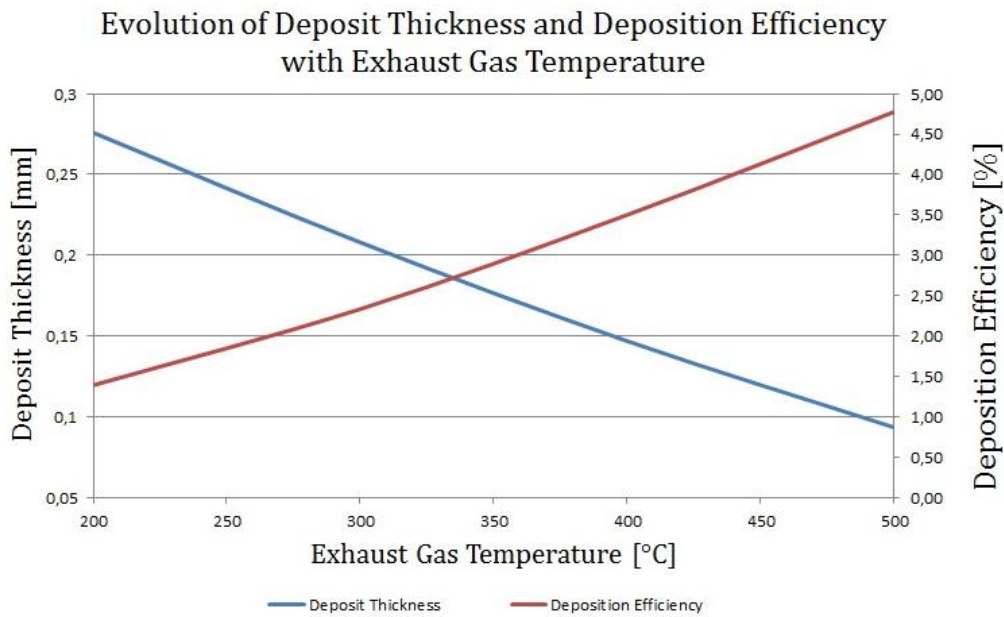


Figure 7.2 - Evolution of the deposit thickness and deposition efficiency with the variation of exhaust gas temperature at 20 °C of water temperature and constant mass flow for test case K

However, as it can be observed, the values of deposit thickness do not follow the tendency one could guess, that is: to increase with increasing exhaust gas temperature, which creates a greater temperature difference and, therefore, propitiates the deposition, even though it decreases the amount of condensates, as with a rise in temperature less condensation occurs. In fact, the opposite is observed. The evolution of the two parameters, deposit thickness and deposition efficiency is quite clearly portrayed in Figure 7.2, where one can see that the deposit thickness decreases with the rise in exhaust gas temperature, when it should increase instead. This can lie on the fact that, as the changes in temperature are so great, the density of the exhaust gas flow changes, thus changing the velocity of the flow, which will maximize the removal of particles by gas shear stress. This occurs because the input value introduced in *Fluent* is for constant mass flow rate ($\dot{m}_{exhaust} = 5.5 \text{ kg/h}$). Therefore, a different analysis was made, considering constant volume flow rate, instead of mass flow rate, to ensure constant velocity of the flow regardless of the temperature. The original values for point K were used as basis, so the volume flow rate at 400°C was determined:

T [°C]	ρ [kg/m ³]	$\dot{V}$ [kg/s]	$\dot{m}$ [kg/s]
400	0.5268	0.002904	0.00153

Table 7.7 - Values for the mass and volume flows and density for point K

Using the value of volume flow rate as constant, rather than the mass flow rate, the following mass flow values for the other temperatures are depicted in Table 7.8.

T [°C]	ρ [kg/m ³]	$\dot{m}$ [kg/s]
200	0.7487	0.002174
300	0.6184	0.001796
500	0.4589	0.001333
600	0.4066	0.001181

Table 7.8 - Density and mass flow values at constant volume flow from test case K at 400 °C

By the previous table one can verify that, the higher the temperature, the less dense becomes the flow, which has a direct influence on the mass flow rate (and, per correlation, on the velocity). Accordingly, the respective mass flows were introduced to *Fluent* and the results obtained are displayed in Table 7.9. With these adjustments, it is clearer the influence the exhaust gas temperature has on the soot layer deposition.

$T_{exhaust}$ [°C]	T_{water} [°C]	Deposit Thickness [mm]	Deposition Efficiency [%]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conductivity [$\frac{W}{(m K)}$]
200	20	0.0615	2.649	153.78	154.70	0.2923
	80	0.0634	1.249	170.59	171.16	0.2933
300	20	0.1147	3.008	223.30	226.00	0.2869
	80	0.0884	1.891	242.36	243.69	0.288
400	20	0.1471	3.500	290.73	295.08	0.2830
	80	0.1275	2.314	311.56	314.24	0.2862
500	20	0.1811	3.839	355.91	362.26	0.2796
	80	0.1515	2.668	378.23	382.16	0.2831
600	20	0.1860	4.188	421.50	428.45	0.2771
	80	0.1750	2.940	441.85	447.25	0.2808

Table 7.9 - Results of the mass flow change pursued regarding the exhaust gas temperature variation for test case K

The higher the temperature, the thicker the deposit layer is, as the deposition is endorsed by the thermophoresis effect, due to a greater temperature difference. However the rise in deposit thickness is not too steep because condensation is reduced with the increase in temperature, thus diminishing the amount of condensates in the deposit layer.

From Figure 7.3 it is possible to have a better perception of the evolution of the deposit thickness and deposition efficiency depicted in Table 7.9, for the cases related to 20 °C of water temperature. After performing the adjustments of the mass flow, it is possible to conclude that the density of the flow does take a big toll on the final results, because the two parameters now evolve in the same direction. With increasing exhaust gas temperature both deposit thickness and deposition efficiency increase, due to a greater impact of thermophoresis, as previously stated. However, as temperatures of 600 °C are reached, both variables begin to stagnate; they vary less with the same rise in temperature. This fact can also be seen in Table 7.8, as the density decrease is steeper from 300 °C to 500 °C than from 500 °C to 600 °C. This occurs because the air density (the assumption of air properties for exhaust gas is made, as previously stated) has a slower increase rate with the temperature rise.

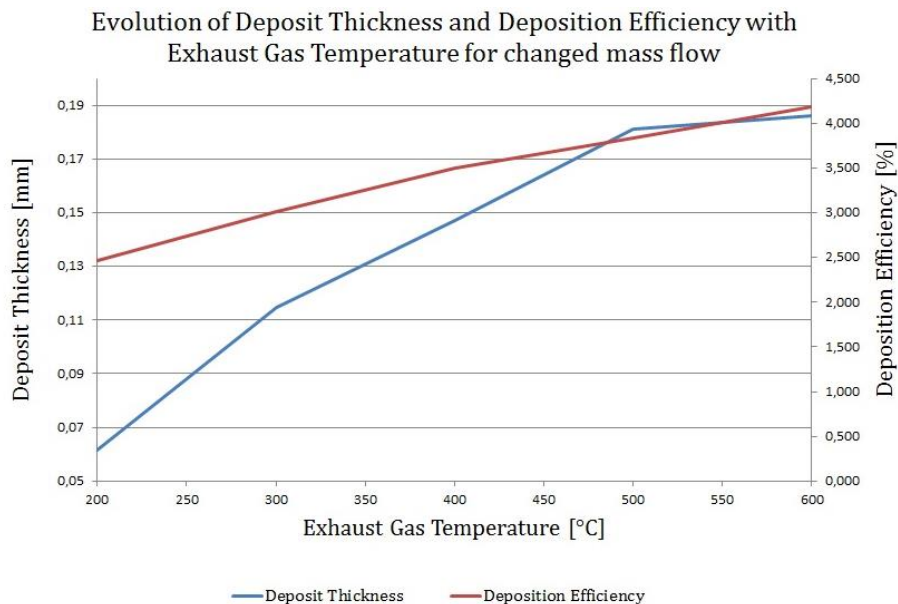


Figure 7.3 - Evolution of the deposit thickness and deposition efficiency with the variation of exhaust gas temperature at 20 °C of water temperature with respective mass flow variation for test case K

Now taking a closer look at Table 7.9, one can also observe that with increasing exhaust gas temperature, the deposition efficiency rises, being also in conformity with the water side temperature. This corroborates the theoretical background from Section 2.4.1 that states that, with increasing exhaust gas temperature – and, therefore, increasing temperature gradient between exhaust gas and the wall –, the thermophoretic effect is aggravated, thus increasing the amount of soot diverted to the wall which, by itself, means higher deposition efficiency.

Regarding the relation to the water side temperature, when the 20 °C version is used, the deposition efficiency is higher than the 80 °C version for the same exhaust gas temperature. For the deposit thickness, the same tendency between temperatures is observed, apart from the values for 200 °C, but as the values in question are very small, one may assume that it might be due to the presence of numerical errors. Regarding the values of the temperature at the outlet of the cooler, a thicker deposit decreases the heat transfer and, therefore, the temperature at the exit of the cooler should be higher with the increase in deposit thickness. This is true for every case. Nonetheless, one needs to pay attention to the different boundary conditions for the temperatures. Therefore, it is acceptable that the outlet temperature for 0.1471 mm of deposit is lower than for 0.1275 mm, because the first case is subjected to a temperature difference of 380 °C, while the latter to one of 320 °C. In addition, one also needs to mind the temperature without considering the fouling effect, that is, the temperature of the CHT analysis. The one for the fouling case should always be higher since the deposit is now taken into account, which indeed happens, as it can be observed; the temperature variations between CHT and fouling analysis are not too great, but the deposit thickness does not have very high values as well.

Finally, the performance of the cooler can be assessed by the evolution of the thermal conductivity which, even though at a low rate, continuously decreases with the rise in exhaust gas temperature. As stated in Section 2.6, with the increase in exhaust gas temperature, there is a rise in the deposit thickness due to the fostering of thermophoresis by a greater temperature difference. This rise in the deposit thickness makes the surface temperature increase due to the inability of the low thermal conductivity deposit to transfer the heat from the gas to the metal. The more deposit there is, the worst becomes the thermal conductivity due to the presence of more soot,

which ultimately prevents the heat transfer, leading to a stabilization of the deposit thickness - since the temperature at its surface comes near the temperature of flow, decreasing the thermophoresis. This corroborates the fact that a higher temperature difference of the fluids in question leads to a worse heat transfer performance, as the cooler is subjected to the fouling effect that damages the cooler.

7.2.2. Variation of HC concentration

The variation of HC concentration was performed at $T_{exhaust} = 400\text{ }^{\circ}\text{C}$, $T_{water} = 20\text{ }^{\circ}\text{C}$, and both zero concentrations of SO_2 and H_2O , just like originally in operating point K.

According to Table 7.10 and Figure 7.4 one can observe that the deposit thickness rises with the increase in HC concentration, which is the expected tendency, as the HC condensation contributes to the diffusiophoresis effect, bringing more particles towards the wall. The observed values for the deposit thickness are also in compliance with the values of thermal conductivity.

HC [ppm]	Deposit Thickness [mm]	Deposition Efficiency [%]	T_{out} CHT [$^{\circ}\text{C}$]	T_{out} Fouling [$^{\circ}\text{C}$]	Thermal Conductivity [W/ (m K)]
0	0.0930	1.424	290.74	298.09	0.1000
200	0.1394	3.314		295.30	0.2533
300	0.1431	3.405		295.15	0.2697
350	0.1444	3.445		295.10	0.2755
430	0.1471	3.500		295.08	0.2830
450	0.1461	3.520		295.03	0.2842
750	0.1490	3.738		294.93	0.2976
1000	0.1502	3.917		294.90	0.3023
2000	0.1508	4.638		294.83	0.3086

Table 7.10 - Results on the single variation of HC concentration for point K

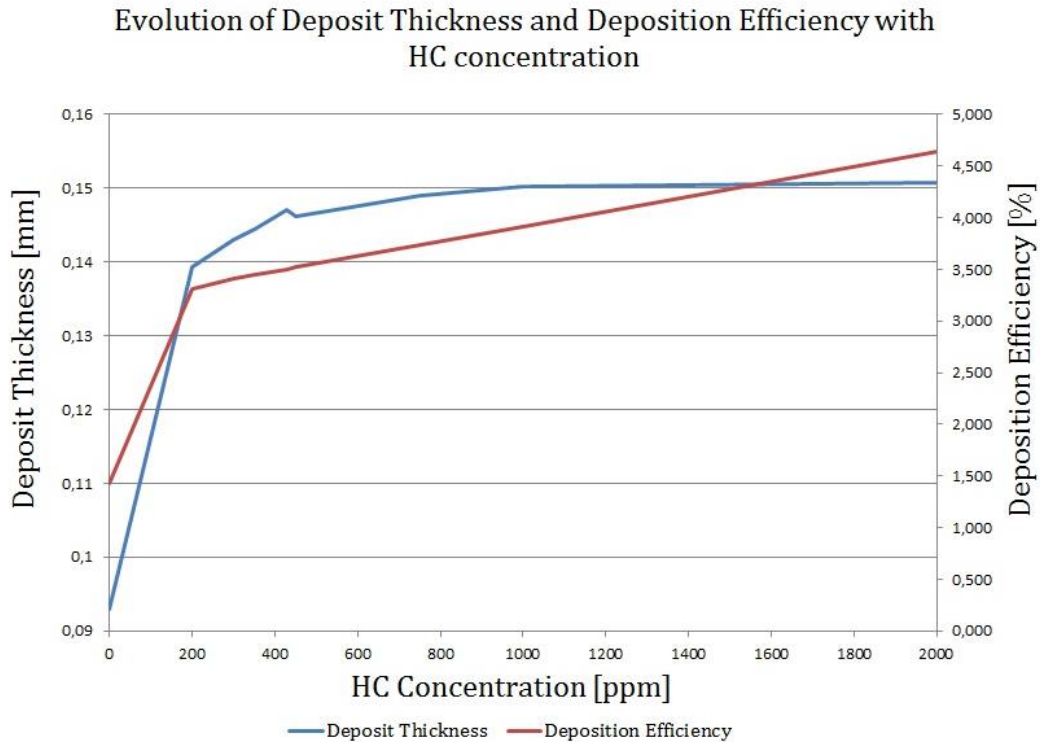


Figure 7.4 - Evolution of the deposit thickness and deposition efficiency with the variation of HC concentration for test case K

A bigger amount of HC condensates will lead to an increase in the overall thermal conductivity - due to the higher thermal conductivity inherent to the hydrocarbons -, which promotes a thicker deposit layer. It will also have an effect on the deposition efficiency: a higher amount of condensates brings more particles more easily to the wall, hence the increase in this value as well.

Through the analysis of Figure 7.4, which depicts the results from Table 7.10, it becomes clear that, from approximately 700 ppm of concentration, the HC no longer has a substantial influence on the deposition of soot; however, from around 300 ppm the change in deposit thickness is only slight. This effect has to do with the influence of the sticking effect, which is dependent on the amount of HC and SO₂ condensates existing in the exhaust gas. This will be further explained in Section 7.3, regarding the influence of the sticking factor of the HC and SO₂.

Concerning the temperature evolution, however, it does not seem to make sense that the temperature at the outlet of the cooler (in the fouling analysis) decreases. In order to understand that fact, the ratio between the thermal conductivity and the deposit thickness will be ascertained.

HC [ppm]	λ/δ
0	1.075
200	1.817
300	1.885
350	1.908
430	1.924
450	1.945
750	1.997
1000	2.013
2000	2.046

Table 7.11 - Ratio between the values of the thermal conductivity and deposit thickness for the HC different concentrations

When analysing the heat transfer equation through the cooler:

$$\dot{Q} = \dot{m}_{exhaust} \cdot c_{p_{exhaust}} \cdot (T_{in} - T_{out}) \quad (7.1)$$

And the respective heat being transferred to the deposit layer:

$$\dot{Q} = \lambda \cdot A \cdot \frac{(T_{interface} - T_{wall})}{\delta} \quad (7.2)$$

It is easy to comprehend the relation between the thermal conductivity λ and the deposit thickness δ . Therefore, it means that with a higher ratio, the temperature difference between the inlet and outlet of the cooler should be greater, which indeed happens – assuming that all the other variables remain constant. As the HC concentration rises, so does the ratio, as displayed in Table 7.11, and analogously the outlet temperature decreases – in Table 7.10 – increasing the overall temperature difference between inlet and outlet.

7.2.3. Variation of H_2O concentration

In Figure 7.5 it is possible to see that the water vapor condensation has a very big influence in the deposition of soot, as the value of the deposit thickness is always rising with the increase in H_2O concentration. This maximum concentration value of 11% was chosen for this study because Hörnig (2012) defines it as the common maximum value for the water vapor in diesel engines, as already mentioned in Section 2.2.1.

H_2O [-wt.%]	Deposit Thickness [mm]	Deposition Efficiency [%]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conductivity [W/(m K)]
0%	0.1471	3.500	290.73	295.08	0.2830
2%	0.1522	32.088		293.59	0.4493
4%	0.2011	138.365		293.92	0.5484
8%	0.2061	315.337		293.98	0.5503
11%	0.6709	408.411		301.29	0.5420

Table 7.12 - Results on the single variation of H_2O concentration for point K

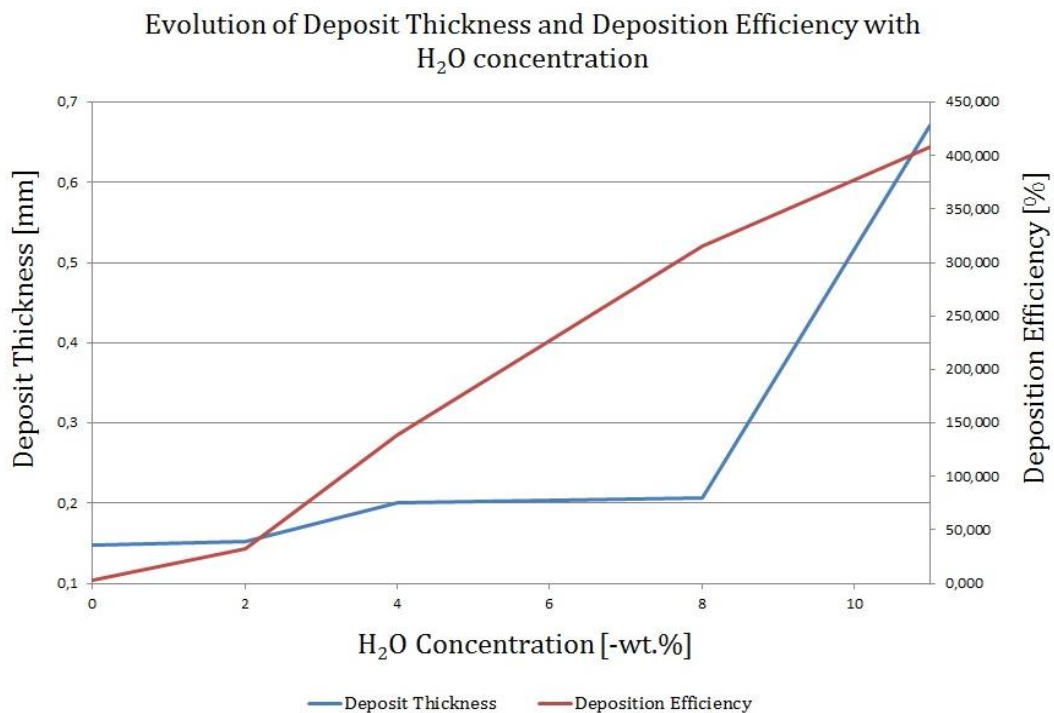


Figure 7.5 - Evolution of the deposit thickness and deposition efficiency with the variation of H_2O concentration for test case K

As it can be observed from the variation of water concentration in Table 7.12 and Figure 7.5, apparently senseless values of deposition efficiency were obtained, since they are higher than 100%, which would mean that the amount of soot suffering deposition would be higher than the actual rate of soot coming into the cooler – that is impossible, as it does not respect the law of mass conservation.

However, the results are valid since the routines assume that the initial soot concentration entering the cooler is constant throughout the whole simulation process, which means that it does not vary along the cooler: the soot mass concentration at the inlet and outlet of the cooler are the same and constant. This does not actually occur, since it does not follow the law of mass conservation, but it is an assumption made in code provided, therefore its effect must be considered. It must be stressed that this fact makes the values not totally realistic; therefore, one must keep it in mind and draw conclusions from it.

	Condensation Rates [kg/s]		
H ₂ O [-wt. %]	H ₂ O	HC – long chain (C ₂₄ H ₄₆)	HC – short chain (C ₁₂ H ₂₆)
0%	0	-2.282e-06	-1.217e-06
2%	-0.000273	-2.269e-06	-1.216e-06
4%	-0.001217	-2.252e-06	-1.199e-06
8%	-0.003162	-2.218e-06	-1.165e-06
11%	-0.004672	-2.193e-06	-1.140e-06

Table 7.13 - Condensation rates from the variations of water concentration values

In the particular case of this work, this fact is highly remarked due to the fact that the soot concentration at hand is very low and the deposition rates are very high in comparison to it. If the case were reversed, that is, if the concentration were higher and deposition rates very low, this fact would not be noticed, as the transformations throughout the cooler would have a very low impact on the flow, and the concentration at the outlet would be roughly the same as the one entering the cooler. This assumption

made by the supplier of the routines is valid as it performs the calculations solely for the equilibrium state. They fundamentally “remove” the time component and, therefore, the deposited particles obtained are not to be deposited “in every moment” but rather at an infinite length of time, where a steady state is reached.

This is particularly evidenced by Table 7.13, which displays the condensation rates corresponding to this water concentration variation – also after six iterations, just like the deposition rates of the particles, where it displays a higher value.

H ₂ O [-wt. %]	Deposit Thickness [mm]	Thermal Conduct. [$\frac{W}{(m K)}$]	Area [m ²]	Heat Transfer [W]	$T_{interface}$ [°C]
2%	0.1522	0.1522	0.004874	170.76	31.87
4%	0.2011	0.2011	0.004824	170.21	32.94
8%	0.2061	0.2061	0.004819	170.09	33.22
11%	0.6709	0.6709	0.004352	175.01	69.78

Table 7.14 - Results on the interface temperature between deposit layer and the flow, for variation of water concentration

From the observation of Table 7.13, one can conclude that the higher the water vapor concentration, the greater its condensation is, and the smaller are the condensation rates for the HC. In order to understand why this occurs, the temperature at the interface between the deposit layer and the flow was determined. The results for the heat flow that crosses the deposit layer was obtained with *Fluent* and then the matching temperature was procured from the Equation (7.2), assuming that the temperature at the wall is constant and equal to the water temperature. The results obtained are displayed in Table 7.14.

As the interface temperature increases with the rise in water concentration, one can understand why condensation of HC decreases: a higher temperature induces less condensation of this species at the interface. However, despite the influence from the HC, the water displays a much greater influence, as its condensation rate is of 3 magnitude orders higher. This has a great influence on the value of the deposition efficiency, as more condensed water will divert many more particles to the surface, that is, the diffusiophoresis effect is enhanced by a higher concentration gradient caused by

the change of phase of the water, as explained in Section 2.4.2. In addition, as seen in Section 6.1.5, water is the condensate species with the highest thermal conductivity of the four different species, which contributes even more for its great impact and it is visible in Table 7.14 by the steeply increasing value of the thermal conductivity with increasing water concentration. This fundamentals the higher deposit thickness, as a higher thermal conductivity benefits the heat transfer which, by itself, promotes the thermophoretic effect, hence the growth of the deposit layer. Accordingly, the thicker the layer is, the more it will affect the heat transfer, thus increasing the outlet temperature.

7.2.4. Variation of SO_2 concentration

In order to analyse the influence of the SO_2 content, the exhaust has necessarily to contain water, otherwise no sulphuric acid can be formed and, consequently, no condensation of the latter can occur (Hörnig 2012). With the original inlet temperature from test case K (400 °C) no condensation could occur.

$T_{exhaust}$ [°C]	H ₂ SO ₄ [ppm]	SO ₂ [ppm]
100	0	0
	0.065	2.17
	10	333.33
150	0	0
	0.065	2.17
	10	333.33
200	0	0
	0.065	2.17
	10	333.33

Table 7.15 - Display of the different variation attempts regarding the sulphur concentration

Therefore, a more exhaustive test was performed, with several concentrations for different temperatures, as seen in Table 7.15. As the sulfuric acid is characterized by a higher condensation temperature than water – supported by Figure 6.1 -, which at

approximately 1 bar corresponds to 350 °C, one would understand why no condensation was observed with such a high inlet temperature (400 °C). Thence, lower inlet temperatures were used to attempt to observe the effect of sulfuric acid condensation, as seen in Table 7.15. Nonetheless, all the other initial conditions from point K were maintained.

$T_{inlet} = 100\text{ }^{\circ}\text{C}$					
H_2SO_4 [ppm]	Deposit Thickness [mm]	Deposition Efficiency [%]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conduct. [$\frac{W}{(m\text{ }K)}$]
0	0.4423	31.370	79.44	80.61	0.5411
0.065	0.4361	110.222		80.63	0.5230
10	0.4358	135.853		80.64	0.5202

Table 7.16 - Results of the variation of sulfuric acid concentration on point K, with an inlet temperature of 100 °C

$T_{inlet} = 150\text{ }^{\circ}\text{C}$					
H_2SO_4 [ppm]	Deposit Thickness [mm]	Deposition Efficiency [%]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conduct. [$\frac{W}{(m\text{ }K)}$]
0	0.3435	31.275	115.77	117.39	0.3917
0.065	0.3338	109.793		117.41	0.5023
10	0.3331	135.437		117.42	0.4999

Table 7.17 - Results of the variation of sulfuric acid concentration on point K, with an inlet temperature of 150 °C

$T_{inlet} = 200\text{ }^{\circ}\text{C}$					
H_2SO_4 [ppm]	Deposit Thickness [mm]	Deposition Efficiency [%]	T_{out}^{CHT} [$^{\circ}\text{C}$]	T_{out}^{Fouling} [$^{\circ}\text{C}$]	Thermal Conduct. [$\frac{\text{W}}{\text{m K}}$]
0	0.2967	31.183	151.42	153.68	0.4823
0.065	0.2849	109.336		153.70	0.4581
10	0.2846	134.909		153.71	0.4600

Table 7.18 - Results of the variation of sulfuric acid concentration on point K, with an inlet temperature of 200 °C

When looking at Tables 7.16, 7.17 and 7.18 one can see that a significant variation in sulphur content leads to almost no variation of the final deposit thickness at the steady-state. This can be explained by the fact that the concentration of sulfur, compared to the concentration of water vapor existing in the exhaust gas is extremely small. However, the values of the deposition efficiency suffer a much greater discrepancy. In order to understand what caused this, an analysis of the forces in place was undertaken: the values of forces and mass flow rates of the several deposition mechanisms were obtained. The results are displayed in Tables 7.19 and 7.20. Regarding the forces acting on the cooler, one can observe that, even though the sulphur concentration changes, no change is visible in the forces. For the cases of 150 °C and 200 °C for the water temperatures, no significant variation of these values was seen, so they are not shown.

Forces [N] at $T_{inlet} = 100\text{ }^{\circ}\text{C}$				
H_2SO_4 [ppm]	Drag	Lift	Thermal	Sticking
0	5,00E-14	1,50E-14	1,22E-13	3,90E-13
0.065	5,00E-14	1,50E-14	1,22E-13	3,90E-13
10	5,00E-14	1,50E-14	1,22E-13	3,90E-13

Table 7.19 - Results of the forces acting in the cooler for the different sulphur concentrations at 100 °C of exhaust gas inlet temperature

Mass Flow Rates [kg/s] at $T_{inlet} = 100\text{ }^{\circ}\text{C}$				
H ₂ SO ₄ [ppm]	Thermophoresis	Diffusiophoresis	Impaction	Diffusion
0	7,00E-07	5,00E-05	2,00E-10	5,00E-08
0.065	7,00E-07	5,00E-05	2,00E-10	5,00E-08
10	6,00E-07	5,00E-05	2,00E-10	5,00E-08

Table 7.20 - Results of the mass flow rates from the deposition mechanisms in the cooler for the different sulphur concentrations at 100 °C of exhaust gas inlet temperature, for particle size P3

This leads to the conclusion – since no remarkable variation was seen in the parameters obtained in Tables 7.19 and 7.20 - that the impact of the sulfuric acid in the overall fouling effect is almost non-existent. It displays no impact whatsoever on the variation of the forces that cause the sticking of the particles to the wall. This relates to the sticking factor, which will be further ahead mentioned and discussed.

H ₂ SO ₄ [ppm]	Condensation Rates [kg/s]			
	H ₂ O	HC – long chain (C ₂₄ H ₄₆)	HC – short chain (C ₁₂ H ₂₆)	H ₂ SO ₄
0	-0.0002913	-2.336E-06	-1.277E-06	0.0
0.065	-0.0002912	-2.336E-06	-1.277E-06	-9.63E-10
10	-0.0002913	-2.336E-06	-1.277E-06	-1.48E-07

Table 7.21 - Condensation rates of the four different species for the variation of sulphur concentration at 100 °C with the presence of water

The condensation rates were also determined and analysed, to allow a perception of the relationship between the amounts of the different condensate forming species, Table 7.21. As one can conclude by observing the table, with variation of sulfuric acid concentration, only its condensation rate is altered. The other species remains the same, as expected. Furthermore, it is clear that the condensation rate of the sulfuric is the one that displays lower order values. This order difference is especially great when compared to the water vapor, which shows and proves how small the impact of the

sulphur in the overall deposition is. Regarding the other two temperature cases of 150 °C and 200 °C the condensation rates of HC and SO₂ slightly increase, whereas the one of water slightly decreases, as the exhaust temperature is increased - but keeping the same relationship in the orders of magnitude.

In Table 7.22 are depicted the results of the variation of sulphur concentration, but this time without the presence of water, in order to understand the difference between the deposition efficiency values. By taking a closer look at it, one can see that the deposition efficiency increases with rising sulphur concentration. The deposition efficiency provides a measurement for the velocity of deposition, but it does not necessarily mean that there will be more deposition and, therefore, a thicker layer in the state of equilibrium. In both results, with or without the presence of water vapor, one can see the influence of the sulphur in the deposition efficiency.

$T_{exhaust}$ [°C]	H ₂ SO ₄ [ppm]	Deposit Thickness [mm]	Deposition Efficiency [%]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conductivity [$\frac{W}{m K}$]
100	0	0.3928	0.633	79.44	81.21	0.3077
	0.065	0.3867	3.153		81.28	0.2903
	10	0.3857	3.216		81.28	0.2888
150	0	0.3184	0.991	115.77	118.28	0.3034
	0.065	0.3092	4.928		118.43	0.2768
	10	0.3086	4.972		118.43	0.2761
200	0	0.2946	1.358	151.42	154.97	0.2994
	0.065	0.2827	6.754		155.25	0.2643
	10	0.2825	6.792		155.25	0.2641

Table 7.22 - Results for the variation of sulphur concentration values, with null concentration of water vapor in the exhaust gas

However, this influence is magnified in the results from Tables 7.16 to 7.18 due to the condensation of water. An influence in the deposit thickness is then visible when the thermal conductivity obviously varies, which is also clear in both cases. A drop in the thermal conductivity explains a slimmer deposition rate, because it impedes the heat transfer, which will reduce the thermophoretic effect and, therefore, the soot deposition. As one knows beforehand that the sulphur is expected to have no influence in the deposition layer when the concentration of water is null, this is corroborated by the information in Table 7.22. The variation of deposit thickness is very small for every used exhaust gas temperature; therefore, the sulfuric acid plays almost no role in the deposition. And when it is in the presence of water vapor its amount is so small compared to the water, that its influence is almost non-existent. Furthermore, it is also important to take notice of the sticking factor associated with the condensation of the sulfuric acid, which explains why there is practically no influence of this species with the variation of its concentration. This parameter is further explained in Section 7.3.

7.2.5. Variation of the exhaust mass flow

In order to assess the variation of the exhaust mass flow input, the other values chosen for this study were values also used by Völk and Hörnig (2011) in other measurements, which are not depicted in this thesis. The other boundary conditions are the original values from test case K, apart from the $\dot{m}_{exhaust}$.

$\dot{m}_{exhaust}$ [kg/h]	Deposit Thickness [mm]	Deposition Efficiency [%]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conductivity [$\frac{W}{(m K)}$]	Heat transfer [kW]
2	0.7057	2.144	277.05	285.26	0.2931	262.87
4	0.3375	2.829	286.98	294.50	0.2871	483.27
5.5	0.1471	3.500	290.73	295.08	0.2830	642.45
7	0.0143	1.053	295.66	295.73	0.2064	780.78
9	0.0	0.000	298.19	298.19	-	979.51

Table 7.23 - Results on the single variation of exhaust mass flow for point K

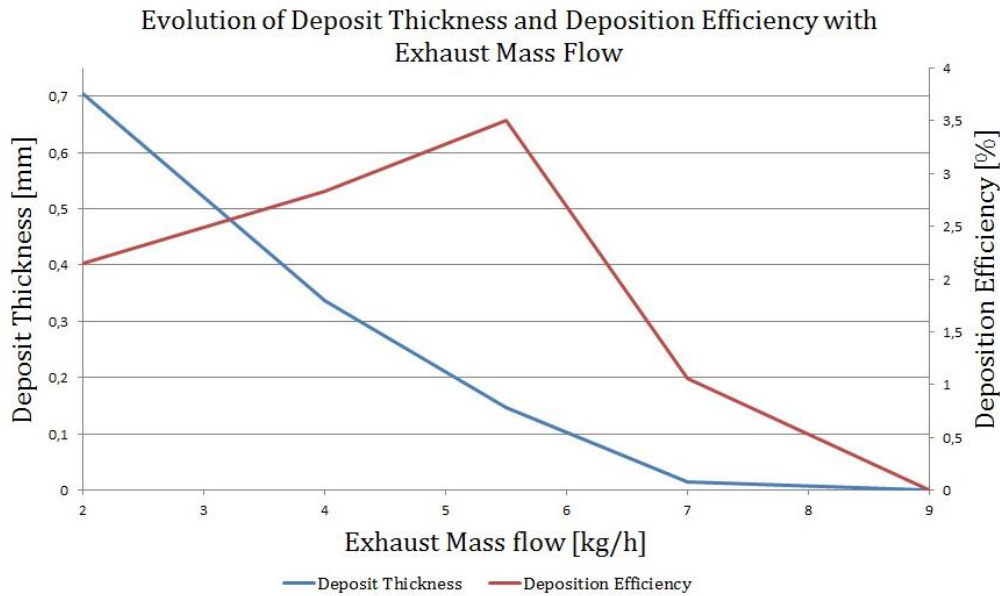


Figure 7.6 - Evolution of the deposit thickness and deposition efficiency with the variation of exhaust mass flow for test case K

As it can be observed in Table 7.23 and Figure 7.6, the higher the exhaust mass flow, the less deposit thickness is built, because a higher mass flow implies a higher flow velocity, which impedes the heat transfer since the particles have less time to flow across the cooler. Also, a greater velocity may cause higher removal forces, which in the case for 9 kg/h, they remove the entire deposit layer. As mentioned in Section 2.7.2, there is a critical velocity for every particle size above which the particle is removed from the wall. Looking at Figure 2.14 from the same section, one can see that in the particle range existing in the EGR coolers the critical flow velocities range from 40 to 280 m/s. In fact, the corresponding velocity for a 9 kg/h flow in the used geometry is around 60.4 m/s, which already falls within this interval, explaining why the removal of the deposit layer occurs. The lower the exhaust mass flow, the lower the velocity of the fluid, as long as there is no variation of the density of the flow. The lower the velocity, the more heat transfer occurs, thus favoring the deposition of soot by thermophoresis.

However, regarding the deposition efficiency, one would expect it to decrease with decreasing deposit thickness, as the deposition is promoted from the beginning of the deposition process, i.e. in the initial iterations. As explained in Section 2.5 when a turbulent state is present, it helps to mix the particles, which promotes the formation of the deposit layer, thus increasing the deposition efficiency. Nonetheless, when the velocity in this turbulent state is too high, it will stop promoting soot deposition and will begin to act as a removal mechanism, since the velocity has crossed the value for of

critical velocity defined in Section 2.7.2 – this is what happens between the cases for 5.5 kg/h and 9 kg/h in Table 7.23. As it can be seen, the deposition efficiency at 7 kg/h is already decreasing and by 9 kg/h it has reached zero.

Concerning the temperature distribution, when looking at the T_{out} of the CHT analysis, one could question why the temperature at the outlet is approximately 21 °C higher for the higher velocity case, comparing to the lower velocity case, even though the inlet temperature is equal for all of the cases, 400 °C. The answer lies in determining the amount of energy transferred in the CHT process. Through the thermal energy balance $Q = \dot{m} \cdot c_p \cdot \Delta T$, one can easily determine the energy transfer in every process, assuming the specific heat capacity of air at 400 °C. Even though the temperature is higher for the 9 kg/h case, the energy transfer is almost four times higher than in the 2 kg/h case. The increase in velocity diminishes the overall heat transfer, thus increasing the outlet temperature.

It can also be noted that the thermal conductivity decreases with increasing exhaust mass flow. As previously mentioned, the amount of soot and condensates are the main determining factors for the thermal conductivity of the final deposit layer. In this case, as the exhaust mass flow is increased, the amount of condensate species flowing in increases relatively to the soot amount, as the soot mass flow remains constant. Therefore, in theory, the thermal conductivity should increase due to a bigger amount of condensates present. However, due to the higher flow velocity, the temperatures in the deposit layer decrease, thus decreasing the level of condensation - even though the species concentration is higher.

$\dot{m}_{exhaust}$ [kg/h]	Condensation Rates [kg/s]	
	HC – long chain (C ₂₄ H ₄₆)	HC – short chain (C ₁₂ H ₂₆)
2	-2.525E-06	-1.365E-06
4	-2.333E-06	-1.238E-06
5.5	-2.282E-06	-1.217E-06
7	-2.277E-06	-1.233E-06

Table 7.24 - Condensation rates of the HC present in the flow, for exhaust mass flow variation of test case K

This is verified in Table 7.24, where one can clearly see that the HC condensation rates decrease – the only condensate forming species present in this section - as the flow increases, hence explaining the decreasing value of thermal conductivity.

Fundamentally, a higher exhaust mass flow prevents the creation of a deposit layer, which leads to the non-occurrence of the deterioration of the cooler's performance, but will also lead to higher outlet temperatures. When comparing the values for 2 kg/h and 9 kg/h one can see that the first has the highest deposit thickness but also the lowest temperature at the outlet (in CHT and fouling analysis, after the deterioration of the performance), while the latter has no deposit thickness and, however, has the highest outlet temperature.

7.2.6. Variation of the soot mass flow

$\dot{m}_{soot}$ [mg/h]	Deposit Thickness [mm]	Deposition Efficiency [%]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conductivity [$\frac{W}{(m K)}$]
20	0.1512	1.560	290.73	294.99	0.2976
45	0.1471	3.500		295.08	0.2830
90	0.1415	6.960		295.22	0.2619
150	0.1358	11.530		295.40	0.2407

Table 7.25 - Results on the single variation of soot mass flow for point K

The effect observed in the Table 7.25 and Figure 7.7 is that, with increasing soot mass flow, the deposition efficiency is clearly increased, whereas the deposit thickness slightly decreases. A higher soot mass flow means that, for the same amount of condensates, there is a bigger amount of soot. This will decrease the thermal conductivity because, as stated before in Section 6.1.6., the soot is characterized by a thermal conductivity of 0.1 W/(m K) whereas all the condensates possess higher values.

Therefore, with increasing soot mass flow, a thinner deposit layer will have the same isolative effect as a thicker one with less overall soot, as both the final density of the deposit layer and its thermal conductivity will be different.

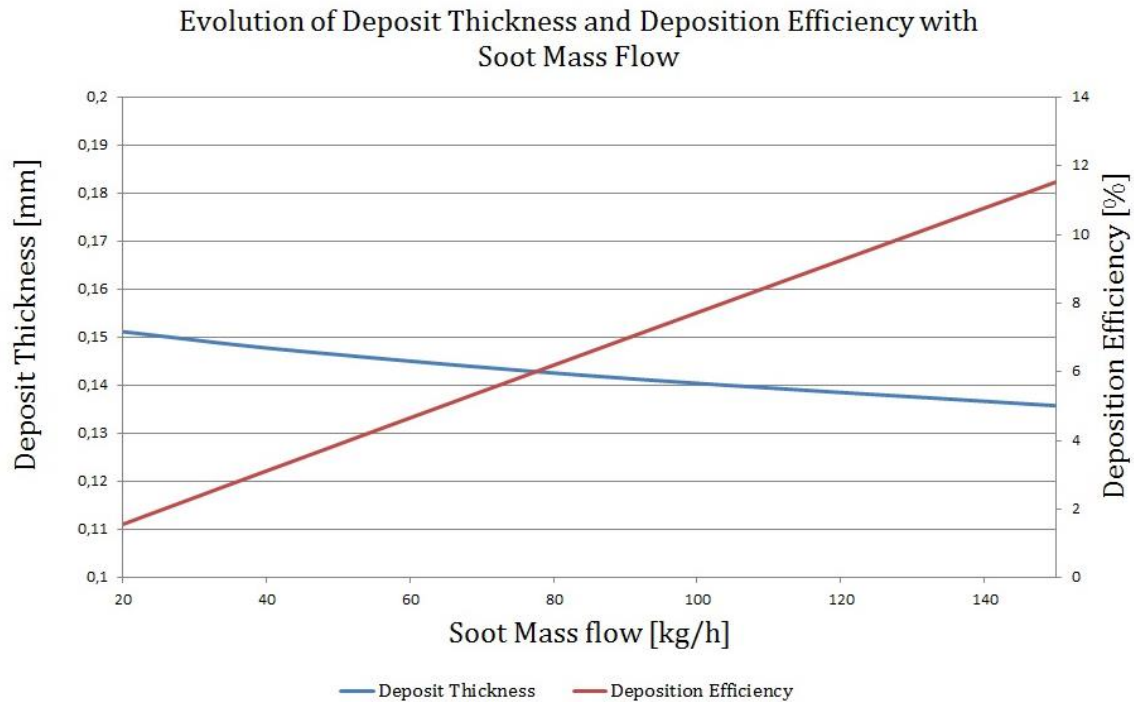


Figure 7.7 - Evolution of the deposit thickness and deposition efficiency with the variation of soot mass flow for test case K

The values obtained in this section are very satisfactory, as the thicker the deposit layer is, the higher will be the thermal conductivity, which matches the theoretical expected tendency. The more soot is present in the flow (compared to the existing condensates), the thinner will the deposit layer be at the final steady-state in the cooler.

When analysing the temperature, it is visible that there is almost no variation. This may lie on the fact that the flow boundary conditions are the same and the paired variation of the deposit thickness and thermal conductivity changes in a way that provides the same final temperature.

Finally, regarding the deposition efficiency, it is understandable that, with a higher amount of soot particles, with all the other boundary conditions remaining unchanged, the deposition efficiency will increase. Since there is a higher concentration of soot particles flowing at every given point through the cooler, more particles will be easily driven to the wall, as the deposition rate of each one of the particle classes will be incremented.

7.3. Sticking Factor

One crucial factor for the deposition of soot is the sticking factor. It is an empirical value used in the calculations of the routines which accounts for the impact of HC and H₂SO₄

(but not H₂O) condensation on the probability of soot to be deposited at the wall; it is yet only relevant when soot deposition occurs. After being obtained, this parameter is then multiplied by the sum of the deposition mass flow rates, thus influencing the overall deposit building.

In order to understand the different impact caused by these two condensate forming species, the sticking factor of each one of them was determined – after six iterations, for an analogy with the deposition rates -: simulations were carried out with different concentrations of SO₂ and zero concentration of HC, and vice-versa. The results are displayed in Tables 7.26 and 7.27:

SO ₂ [ppm]	Sticking factor [-]
0	1
0.05	1 – 3.22
0.1	1.59 – 4.11
0.5	3.80 - 5
2.17	5
333.33	5

Table 7.26 - Determined values for the sticking factor concerning solely SO₂ concentration, after six iterations, for test case K

Since the sticking factor depends on local soot deposition rate, it is not constant throughout the entire cooler; therefore, in Table 7.26 the range of sticking factors obtained for the SO₂ concentration is shown. One can conclude that the variation of sulphur does not play almost any influence in the results, as the only different noticed is between 0 and 2.17 ppm, which are extremely low quantities, especially when compared with the amount of water present, as seen before in section 7.3.5. As the value 5 is the maximum possible for the sticking factor of this species, one can understand why from 2.17 ppm onwards there is no variation of its influence: the maximum sticking effect it can provide to the deposit layer has already been reached. The maximum value of 5 has empirically been derived by the supplier of the UDFs.

HC [ppm]	Sticking factor [-]
0	1.000
200	2.270
300	2.289
350	2.290
430	2.298
450	2.293
750	2.296
1000	2.297
2000	2.298

Table 7.27 - Determined values for the sticking factor concerning solely HC concentration, after six iterations, for test case K

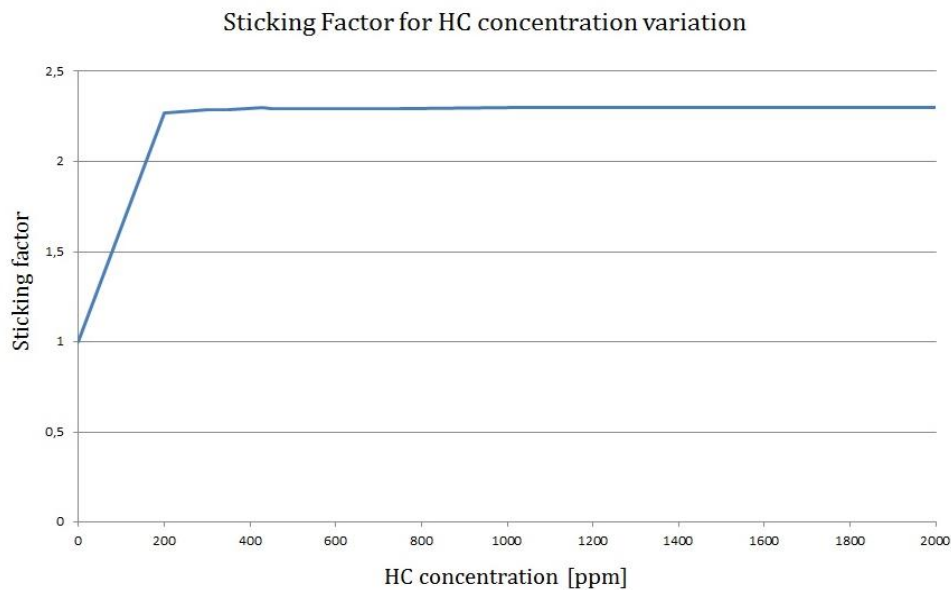


Figure 7.8 - Evolution of the sticking factor with the variation of HC concentration, for test case K, after 6 iterations

When comparing the values from Tables 7.10 and 7.27 it is possible to conclude that the sticking factor has an influence in the deposition due to the presence of HC. From around 300 ppm the value of the deposit thickness begins to change very scarcely and from around 700 ppm it has totally stagnated. Analogously, the same tendency is observed in Table 7.27 for the result of the sticking factor due to the presence of HC. Since the maximum value for the sticking factor from HC has been defined in the routines as 2.3, one can see that the values obtained have stagnated because it has reached its maximum. This is especially clear in Figure 7.8.

According to Table 6.2 it is clear why, even though the HC has got a lower sticking factor than the SO_2 , the impact of the first one in the deposit building is much greater as it possesses a much higher molecular weight, thus having a greater influence in the outcome of the fouling effect than the sulphur. Also, its thermal conductivity is slightly higher, as seen in Table 6.3, which may promote the growth of the deposit thickness, as it facilitates the heat transfer, thus increasing the effect of thermophoresis.

7.4. Simulations – Motor Test Bench

In this section, the two different test cases described in Section 5.6 are going to be pursued. The results of the measurements from Völk and Hörnig (2011) for the variation of T_{water} , Figures 7.9 and 7.10, depict the evolution of the deposit thickness along the cooler for these two test cases.

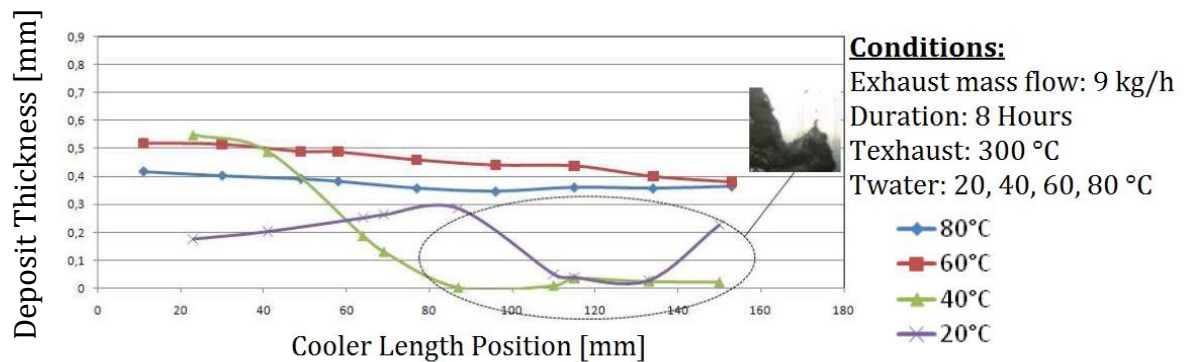


Figure 7.9 - Results of Deposit Thickness for Test Case 1 (adapted to English)

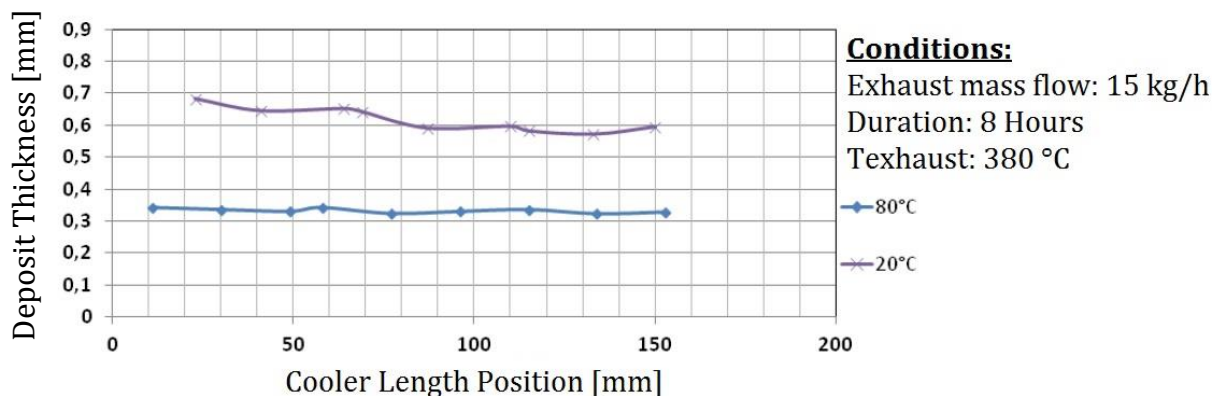


Figure 7.10 - Results of Deposit Thickness for Test Case 2 (adapted to English)

According to Völk and Hörnig (2011), in Test Case 1 of motor test bench, regarding the curves for 20°C and 40 °C it is not possible to obtain a full picture of the soot deposition by condensation throughout the entire cooler because of the deposit gaps seen in Figure 7.9. However, it allows to conclude that, in the final part of the cooler an effect of wash-out is observed, due to the condensation of water. This strong water condensation and the consequent areas of deposit removal in the cooler lead to an unsteady growth of the deposit layer, as it can be seen in Figure 7.9. In Figure 7.11 the measured mass deposition for each of the used water temperatures in Test Case 1 is displayed and the results for 40 °C, 60 °C and 80 °C follow the trend that, with lower water temperatures, higher condensation rates are observed which lead to an increased deposition growth.

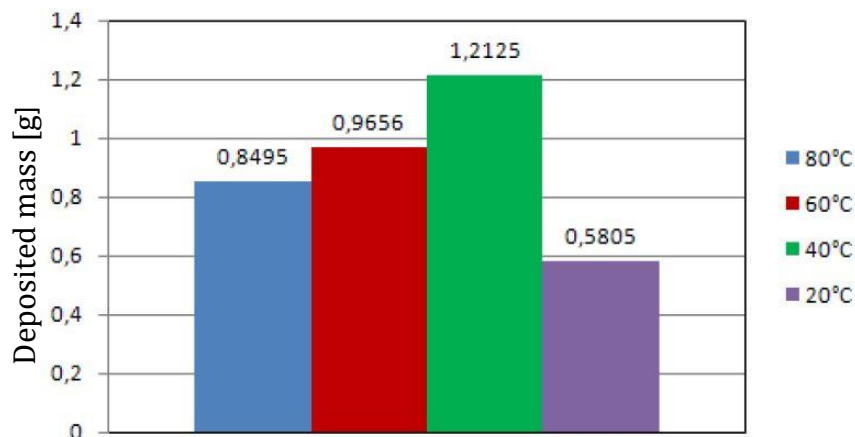


Figure 7.11 - Deposited mass for the four water temperature cases of Test Case 1 in motor test bench from Völk and Hörnig (2011)

The only case that does not verify this condition is the one of 20 °C, due to the fact that the water condensation is so strong that it continuously resists the building of the deposit layer (Völk, 2014). The case for 40 °C also experiences wash-out, reason why there are gaps in the deposit layer and it is lower than for 60 °C and 80 °C in some cases; however, as the condensate mass flow is lower than at 20 °C, it is not enough to significantly decrease the overall deposited mass, as it can be concluded by Figure 7.11.

The obtaining of an equilibrium state of the fouling process sets up at thinner deposit layers with increasing water temperature. This happens due to the smaller amount of condensates in the layer, which makes it drier and with a lower adhesive behavior. Analysing the results from Test Case 1 and Test Case 2 in their work, it is to conclude that the tendency for greater deposit thickness with decreasing cooling temperature is confirmed with the increase in exhaust mass flow rate.

Regarding the Test Case 2 results presented in Figure 7.10, the increased mass flow rate and the higher inlet temperature of 380 °C both contribute to the avoidance of condensation water films forming at the wall. Therefore, it is clearly observed that, even at water temperature of 20 °C, the wash-out effect does not take place; thus, increasing the deposit thickness at lower temperature, verifying the above described tendency. Furthermore, with the increase in moisture content in the layer – as it does not condense to cause wash-out -, the adhesive force is strong enough to promote the increase of the deposit layer, despite a greater gas shear stress caused by the higher flow rate (Völk, 2014). The results for the simulations performed on Test Case 1 are displayed in Tables 7.28 and 7.29, as follows:

	Simulation Results						Measurement Results	
T_{water} [°C]	Deposit Thick. [mm]	Dep. Eff. [%]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conduct. $[\frac{W}{(m K)}]$	Deposit Mass [g]	Deposit Thick. [mm]	Dep. Mass [g]
20	0.00861	3.799	226.79	226.84	0.1790	7.56E-04	0 – 0.30	0.5805
40	0.00328	0.369	233.14	233.16	0.1175	2.88E-04	0 – 0.55	1.2125
60	0.00205	0.268	239.88	239.83	0.1226	1.80E-04	0.45	0.9656
80	0.00183	0.300	244.84	244.84	0.2114	1.61E-04	0.40	0.8495

Table 7.28 - Results for Test Case 1 from motor test bench, with 350 ppm of HC and 2% of H₂O, and the respective results from Völk and Hörnig (2011)

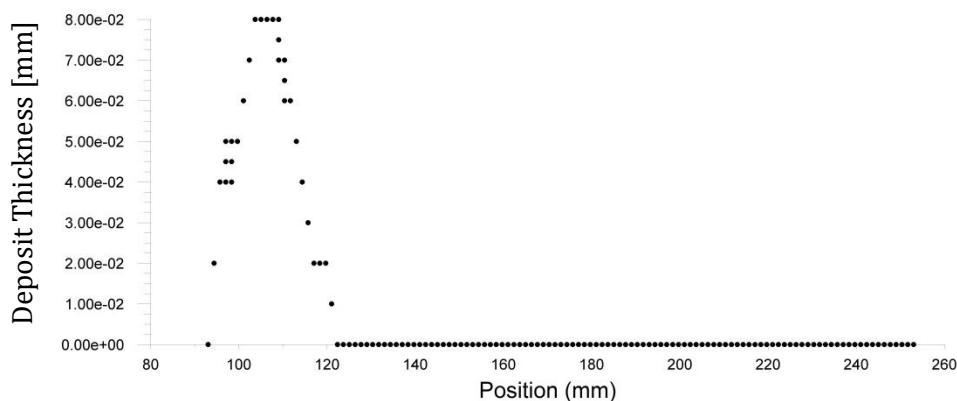


Figure 7.12 - Plot of the simulated evolution of deposit thickness throughout the entire cooler length for Test Case 1 at 20 °C of water temperature, with 350 ppm of HC and 2% of H₂O

For the simulation of test case 1, according to Table 5.6, there is an interval of the values for the input in HC and H₂O concentration. Therefore, for the simulations performed, two pairs of extreme values were adopted, and Table 7.28 displays the values for 350 ppm of HC and 2% of water, while Table 7.29 displays the values for 450 ppm of HC and 11% of water. In this section of the work accomplished by Völck and Hörnig (2011), regarding the measurements of the motor test bench, no information is mentioned on the contents of the water concentration. Therefore, as seen in Section 2.2.1, the interval of the possible water extreme concentrations is between 2% and 11%; the simulations performed in this dissertation took those values as the possible best and worst-case scenario, when it comes to water content damaging the performance of the cooler.

Figure 7.12 displays a plot of the evolution of deposit thickness throughout the cooler for the case of 20 °C from Table 7.28. It shows that the simulated evolution of the deposit thickness is not constant throughout the cooler which, despite not showing the same shape as the result from their measurements, is a plausible result. The greatest deposit thickness is depicted at the entrance of the cooler - the area where the thermophoretic force is at its highest -, because the temperature difference exhibits its maximum value.

From the geometry defined in Section 5.1, one knows that the entrance of the tube is 65 mm after the entrance in the cone. Knowing that the mesh defined in *Fluent* for the simulations has the zero (beginning of the cone) at 28 mm in the z-axis, it is clear that the tube starts at 93 mm, where the first point is depicted in Figure 7.12; and the end of the tube is at 253 mm. From about 120 mm no deposit thickness is visible, which is due to the high mass flow rate (which causes high shear stress) associated with a weakened effect of thermophoresis, as the temperature difference is not as great as at the entrance. Therefore, the removal forces surpass the deposition mechanisms. This occurs in both Figure 7.12 and 7.13. However, this effect is much weaker in the experiment.

	Simulation Results						Measurement Results	
T_{water} [°C]	Dep. Thick. [mm]	Dep. Eff. [%]	T_{out} CHT [°C]	T_{out} Fouling [°C]	Thermal Conduct. $[\frac{W}{(m K)}]$	Deposit Mass [g]	Deposit Thick. [mm]	Dep. Mass [g]
20	0.0169	31.151	226.79	226.90	0.3244	1.48E-03	0 – 0.30	0.5805
40	0.0144	20.271	233.14	233.21	0.2254	1.26E-03	0 – 0.55	1.2125
60	0.0023	0.4700	239.88	239.83	0.1159	2.02E-03	0.45	0.9656
80	0.0019	0.290	244.84	244.84	0.1251	1.67E-03	0.40	0.8495

Table 7.29 - Results for Test Case 1 from motor test bench, with 450 ppm of HC and 11% of H₂O

In Table 7.29 and Figure 7.13 the results for the highest extreme pair of HC and H₂O values is shown, and they both display the same tendencies as the case before, however with rather higher values, since the increased concentrations of HC and H₂O promote the

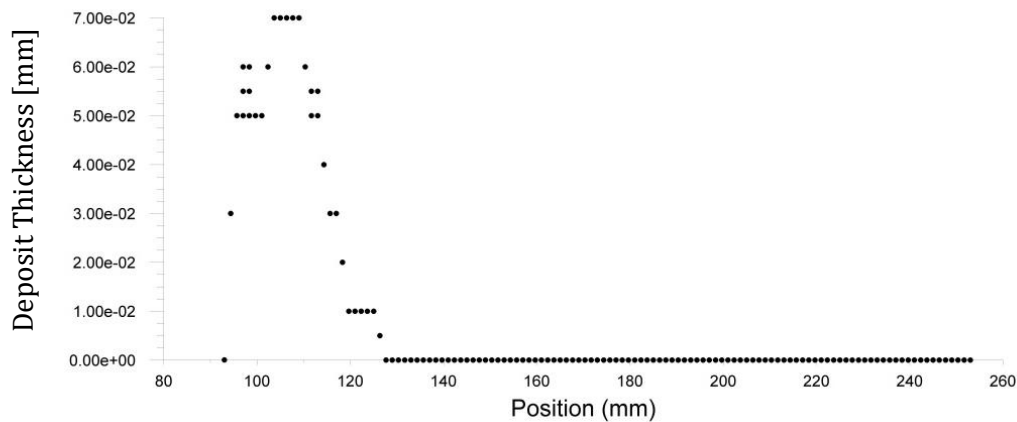


Figure 7.13 - Plot of the evolution of deposit thickness throughout the entire cooler length for test case 1 at 20 °C of water temperature, with 450 ppm of HC and 11% of H₂O

condensation of these species, increasing the impact of the soot deposition. The difference between Figures 7.12 and 7.13 is not great, but it can still be drawn. Despite the fact that the first case reaches a higher thickness value, the second curve is wider and the deposit layer is more spread through the length of the cooler than the first one. For this reason, the second case – matching the higher HC and H₂O concentration values – displays an overall higher deposit thickness.

In addition, one can see that in both cases the maximum deposit thickness is, respectively, 0.08 and 0.07 mm. Nonetheless, as the overall deposit thickness is determined as an Area-Weighted Average, the value is averaged through the entire tube area, and as in most of it there is no deposit, then one can easily understand why the overall values are smaller and, respectively: 0.00861 and 0.0169 mm.

When analysing the deposit thickness evolution with the water temperature (in both cases), a single tendency is perceived. With increasing water temperature, the deposit thickness increases due to the higher amount of condensates. However, contrarily to the results obtained by Völkl and Hörnig (2011), in the case for 20 °C, no wash-out effect is observed, because the routines do not contemplate this effect: a decrease of soot deposition due to condensation is not possible. Therefore, the deposit thickness does not decrease as it theoretically should.

In order for the deposited mass to be obtained, the approximate value of the deposit layer is determined – as the deposit thickness is an approximate value, it is not possible to acquire the exact value – and then multiplied by the density of the deposit layer. As previously mentioned, in the routines a constant value for this density is assumed and it is 17.5 kg/m³. As the deposit layers are extremely small for both results in Tables 7.28 and 7.29, it is understandable why the mass values are so small.

Regarding the results for the Test Case 2, the same approach was used. However, the results obtained for the extreme highest concentration pair were of zero deposit thickness and deposition efficiency, i.e. there is no occurrence of soot deposition. This extreme point is the one corresponding to 20 °C on the water side, with 450 ppm of HC and 11% of H₂O, where the deposition would be reinforced by a greater thermophoretic effect – grounded on a greater temperature difference - and higher condensation rates to promote diffusiophoresis - due to greater condensate forming species concentration. Therefore, since this is the point which would display stronger deposition efficiency, as it occurs in Test Case 1, and nothing occurs, then it is safe to conclude that for the other simulation points there is also no appearance of a soot deposit.

From the physical point of view, it is easy to comprehend why there is no formation of a deposit layer in the simulation of Test Case 2. With a tube geometry as small as the one used in this study and an exhaust mass flow of 15 kg/h the corresponding flowing

velocity is of approximately 369 km/h, i.e. about 102 m/s. Keeping in mind what is explained in Section 2.7.2, when the flow velocity increases and crosses the boundary of critical velocity, removal by gas shear stress starts to occur. The interval of critical velocities is dependent on the particle size and ranges from 40 m/s to 280 m/s. In this particular case, the velocity of 102 m/s falls in the middle of that range, which, allied to the fact that the particle distribution for this test bench displays a greater amount of particles with bigger size, allows to explain why there is no soot deposition occurring at all.

The visible difference between the deposit thickness results from the motor and model test bench are easily explained by the great difference in exhaust mass flow. As seen in Section 7.2.6, where this parameter was changed, one can see that by 9 kg/h – the mass flow existing in the motor test bench - in the model test bench, no emerging of a deposit layer was observed. However, despite the high value of exhaust mass flow in the present Test Case 1, due to a higher value of soot mass flow it is still possible to observe the appearance of the deposit layer, albeit being quite thin. Nonetheless, with a higher soot mass flow, it is no longer possible to observe the emerging of the deposit layer in Test Case 2, since the exhaust mass flow displays a value of 15 kg/h. The fact that it does emerge in the experiments conducted by Völk and Hörnig (2011) might either be caused by errors in their measurements or invalid model parameters.

Furthermore, the work accomplished by Völk (2014) shows a result for the lowest exhaust mass flow, even though the result is time-dependent.

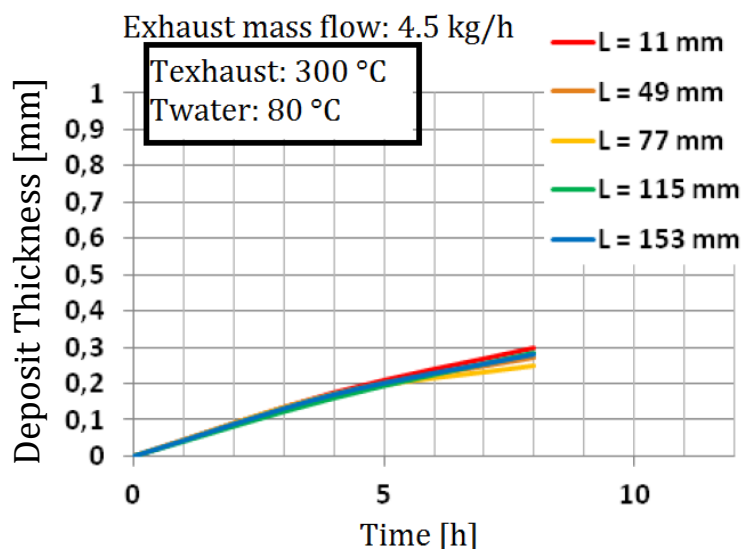


Figure 7.14 - Evolution of the deposit thickness for 4.5 kg/h mass flow, with exhaust temperature of 300 °C and water temperature of 80 °C

However, as it will be later mentioned in Section 7.4.2, when the measurements are endured for about 20 hours one can already consider the result to be in a state of equilibrium. Concretely in the case exhibited in Figure 7.14, the experiment was pursued for 8 hours and, carefully analysing the depicted curve, one can observe that from the value of 0.3 mm the deposit thickness growth is slowly starting to decrease. The results obtained for the simulations with the same boundary conditions are presented in Table 7.30. When comparing these results with the value from the curves in Figure 7.14, a resemblance is seen. That is, for an exhaust mass flow as low as 4.5 kg/h, the results obtained from the simulations with the user defined functions are very similar to the ones from the experiment measures: the values obtained were around 0.3 mm and through Figure 7.14 one can see that, even though the value has not yet stagnated, it is starting to decrease the curvature, thus placing the constant value not much further. This leads to the conclusion that the routines provided are valid and deliver fairly good results at low exhaust mass flow conditions.

	Deposit Thickness [mm]	Deposition Efficiency [%]	T_{out}^{CHT} [°C]	$T_{out}^{Fouling}$ [°C]	Thermal Conduct. [$\frac{W}{(m K)}$]
450 ppm HC; 11% H ₂ O	0.3143	7.349	239.42	244.43	0.2137
350 ppm HC; 2% H ₂ O	0.3097	6.578	239.42	244.48	0.2072

Table 7.30 - Simulation results for 4.5 kg/h mass flow, with exhaust temperature of 300 °C and water temperature of 80 °C

After arriving to this conclusion, one may question why, with a low exhaust mass flow, the results obtained by the simulations are similar to the values observed by Völk and Hörnig (2011), whereas with higher values for the exhaust mass flow, extremely lower values are obtained for the deposit that is built on the wall. In fact, the C-routines work with several empirical parameters that have an influence in the final results, and were not addressed in this scope of this work. A conclusion drawn from the analysis above and from the values obtained both for model and motor test bench is that, as with increasing exhaust mass flow the deposit built is always smaller, the removal mechanisms calculated in the routines are far stronger and have a greater impact in the final results. This is grounded in the fact that there are four empirical factors for the

calculation of these removal mechanisms and the adjustment of these factors may strongly change the final outcome.

7.4.1. Verification of Reynolds number

After performing the simulations for Test Cases 1 and 2, it is important to verify the Reynolds numbers observed in the simulations, in order to compare them with the values supplied by Völk and Hörnig (2011), also depicted in Table 5.5. The values obtained in the simulations are shown in Table 7.31,

$\dot{m}_{exhaust}$ [kg/h]	Density [kg/m ³]	Velocity [m/s]	Dynamic Viscosity [kg/(m s)]	Re
4.5	0.6167	26.91	2.953E-05	5620
9	0.6185	53.03	2.950E-05	11121
15	0.5460	99.52	3.240E-05	16765

Table 7.31 - Results of the values of Re obtained from the simulations

After analysing Table 7.31, it is possible to conclude that the Reynolds number values obtained are similar to those reported by Völk and Hörnig (2011) in their measurements, since the values expected were, respectively: 5600, 11000 and 16000. This corroborates the accuracy of the model defined in *Fluent* to perform the simulations in this dissertation.

7.4.2. Comparison with time-dependent results

Regarding the conditions for Test Case 1, Völk and Hörnig (2011) also performed studies along the time, most of them for as long as 20 hours, in order to observe the evolution of the following parameters: temperature at the wall [°C], cooling performance through transferred heat [W] and thermal conductivity [W/(m K)]. A 20-hour study is already long enough to be considered almost in state of equilibrium. In addition, most of the results obtained by them in this study are already constant before they reach the 20-hour limit. Therefore, these constant values will now be used to compare to the values obtained from the simulations in this dissertation for Test Case 1.

	Simulation Results 450 ppm HC; 11% H ₂ O		Simulation Results 350 ppm HC; 2% H ₂ O		Measurement Results Völk and Hörnig (2011)	
T_{water} [°C]	T_{wall} [°C]	Transferred heat [W]	T_{wall} [°C]	Transferred heat [W]	T_{wall} [°C]	Transferred heat [W]
20	20.94	187.87	21.0	187.28	38	230
40	41.20	170.87	40.71	171.62	58	150
60	60.20	153.72	60.52	153.52	72	150
80	80.42	140.39	80.42	140.91	93	180

Table 7.32 – Comparison of the simulated values and the measurement values from a 20-hour study for the temperature at the wall and the transferred heat for Test Case 1

Regarding the 60 °C and 80 °C cases in Table 7.32, no more occurrence of wash-out effects due to condensation can be observed, which makes the evolution of both parameters more constant and even throughout the time and the length of the cooler, as seen in Figure 7.9. The process of deposit growth falls with increasing water temperature and begins to show some parallelism to the wall of the cooler, due to the decrease of the adhesive behavior of the condensates in the deposit. The fewer amounts of condensates there is in the deposit, the more constant is the growth of the deposit. Also, the difference between the result at 40 °C and 60°C regarding the transferred heat is that the first reaches the level of 150 W only after 20 hours, while the latter is already in constant heat transfer from about 12 hours. The 40 °C case possesses a water film overlaid onto the deposit layer, which alters the overall heat transfer. The evolution of the interface temperature at 80 °C case behaves like that of a clean cooler, due to the low amount of condensates, which results in an extremely dry soot; the temperature at the wall matches a state of an homogenous deposit layer (Völk and Hörnig, 2011).

However, the values obtained by simulation are far from matching the temperatures shown in their measurements. But one cannot forget the disparity of the deposit thickness values seen above. Taking into consideration the low values obtained in the

simulations, the results displayed in Table 7.32 are correct and show the expected tendency, with the evolution in water temperature.

Thermal conductivity

The variation of thermal conductivity in the motor test bench will be analysed in 2 different perspectives: regarding a water temperature variation and an exhaust mass flow variation. As previously studied, the thermal conductivity increases with increasing amount of condensates in the deposit layer. Also, the minimal value it can exhibit is 0.1 W/(m K), corresponding to dry soot. The studies from Völk (2014) that provide these results had the duration of 8 hours.

	Simulation Results 450 ppm HC; 11% H ₂ O	Simulation Results 350 ppm HC; 2% H ₂ O	Measurement Results Völk and Hörnig (2011)
T_{water} [°C]	Thermal Conductivity [$\frac{W}{(m K)}$]		Thermal Conductivity [$\frac{W}{(m K)}$]
60	0.1159	0.1226	0.2900
80	0.1251	0.2114	0.2200

Table 7.33 - Comparison of the simulated values and the measurement values from an 8-hour study for the thermal conductivity for Test Case 1, with variation of water temperature between 60 °C and 80 °C

In order to avoid the creation of a condensate film that would increase the inhomogeneity of the deposit layer, Völk (2014) pursued the measurements only for 60 °C and 80 °C, in order to prevent the overstepping of the dew point.

The value of thermal conductivity for 80 °C is almost constant throughout the entire time evolution because it is the operating point with the lowest temperature difference; in addition, its condensation rates of hydrocarbons are far lower than the others and, due to the smaller temperature gradient, the deposit thickness almost does not change along the cooler (Völk and Hörnig, 2011). Therefore, the thermal conductivity is essentially constant. The decrease in water temperature provides an increase of the thermal conductivity, due to a lower temperature at the interface of the deposit layer caused by a better convection transfer coefficient, increasing the deposition of hydrocarbons. However, the values obtained in the simulations do not match this

tendency. The magnitude of the values is correct and similar to the ones reported by Völk (2014), but the evolution observed is not correct, as they decrease with decreasing water temperature. Nonetheless, this fact may be related to the concentrations introduced for the simulations, as the values are not exactly known, reason why an interval of possible values was sought. When carefully analysing the results for the case with 450 ppm of HC and 11% of H₂O in Table 7.33, one can conclude that the difference between the values is minimal and, therefore, may be considered correct, as the difference between the corresponding deposit thicknesses is also very small.

On the other hand, as it can be seen on Figure 7.15, the increase of exhaust mass flow also promotes the increase in thermal conductivity, since the condensate forming species concentrations flowing in are higher, thus increasing the amount of these components that can condensate at a certain instant. Furthermore, with the high thermal conductivity inherent to these components, the overall thermal conductivity of the layer will increase. In addition, with higher mass flows it is observed that the growth of thermal conductivity is faster, while by lower mass flow values it tends to stabilize faster, as it can be noticed in Figure 7.15.

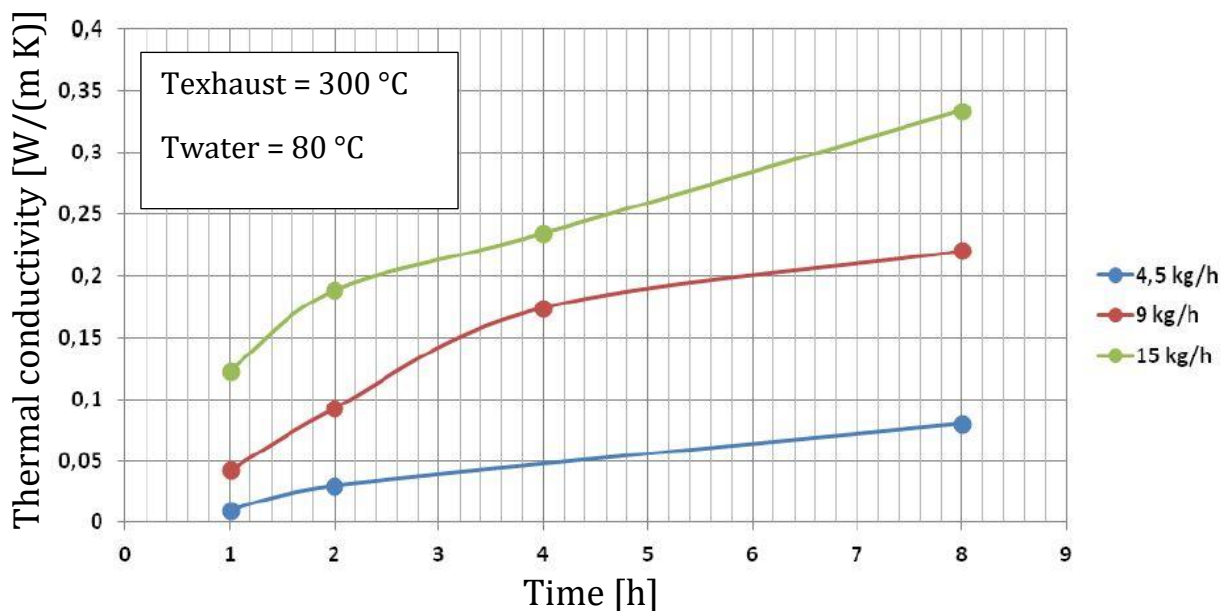


Figure 7.15 - Evolution of the thermal conductivity with the exhaust mass flow throughout the time in the measurements of Völk (2014)

In this ambit, the values obtained for this mass flow variation in the simulations are presented in Table 7.34. As one could see at the beginning of Chapter 7, the results for an exhaust mass flow of 15 kg/h were of no deposit formation; therefore, the thermal

conductivity of this one cannot be assessed. Regarding the cases of 4.5 and 9 kg/h, it can be observed that in both cases of species concentrations the tendency is the opposite of what was expected. However, this difference is emphasized in the case for higher species concentrations.

	450 ppm HC; 11% H ₂ O	350 ppm HC; 2% H ₂ O
$\dot{m}_{exhaust}$ [kg/h]	Thermal conductivity [W/(m K)]	
4.5	0.2137	0.2072
9	0.1251	0.2114
15	-	-

Table 7.34 - Results obtained from the simulations for the values of thermal conductivity on the variation of exhaust mass flow, for 80 °C on Test Case 1

Notwithstanding, it is sensible to make a deeper analysis of the values. In order to perform the simulations, the exhaust mass flow has to be directly inserted in *Fluent* while in the routines the ratio between soot mass flow and exhaust mass flow is used as input. In fact, as seen in Table 5.5, the soot mass flow for 9 kg/h is higher than for 4.5 kg/h, respectively the values: 210.01 mg/h and 91.15 mg/h. Accordingly, the ratio between soot and exhaust mass flow is higher in the 9 kg/h than in the 4.5 kg/h case. This analysis is made clearer in Table 7.35.

$\dot{m}_{exhaust}$ [kg/h]	$\dot{m}_{soot}$ [mg/h]	$\dot{m}_{soot}/\dot{m}_{exhaust}$
4.5	91.15	0.02026
9	210.01	0.02233

Table 7.35 - Analysis of the ratios between soot and exhaust mass flow

Finally, it can be concluded that, even though the tendency of thermal conductivity increase with increase in exhaust mass flow is wrong, the level of the values is correct and makes sense. In fact, the case of 9 kg/h ought to have higher value of deposit layer conductivity, as it possesses a higher amount of condensates flowing in. However, the amount of soot flow in comparison to the whole exhaust flowing in is higher in this case than at 4.5 kg/h. As soot is the component with lower inherent thermal conductivity, it is clear why the 9 kg/h case is characterized by lower overall thermal conductivity.

8. Conclusions and Future Work

8.1. Conclusions

In this dissertation, the topic of study was the numerical simulation of the fouling phenomenon occurring in the EGR coolers applied to diesel engines. The final goal was to validate a set of User Defined Functions that, when paired with *Fluent*, allow for the correct assessment of the fouling effect. This could be achieved by means of a detailed study of the boundary conditions that these cooling components are subjected to and the parameters that have an influence on its performance.

The routines provided have a great potential to describe the physical effects that take place in the EGR coolers. In the studies that supplied the measurements used as basis of comparison to the simulations performed, the following effects were reported as playing a big role in the fouling effect: thermophoresis, condensation, diffusiophoresis, impaction and diffusion. Through the simulations accomplished in the present dissertation, it can be concluded that thermophoresis and diffusiophoresis display the biggest influence in the final results. However, diffusion is also of great importance when turbulence is present, as it occurs when there is a high value of exhaust mass flow rate, which increases the flow velocity.

Regarding the model test bench, it is possible to conclude that, even though the final values of deposition efficiency and outlet temperature did not match the ones from Völk and Hörnig (2011), the expected tendency between the different operating points was observed. Notwithstanding, a deeper study was pursued regarding the operating point K, from the model test bench. A single-effect variation analysis was made, in order to understand the tendencies observed when just one parameter was changed and all the other boundary conditions remained unaltered. This analysis revealed itself of great

importance. The values did not match quantitatively to the ones expected, but the tendencies were correct and they were physically fundamented.

In addition, regarding the presence of water vapor, the great difference of values observed in the results when this component exists was thoroughly justified. The routines did not account for a variation of the concentration of soot in the exhaust gas throughout the cooler, thus considering there would be always the same amount of soot flowing inside the cooler at every length, from the inlet to the outlet. Knowing this, the results obtained were justified and plausible.

One very important factor depicted in the routines is the sticking factor, which is dependent on the condensation of HC and H_2SO_4 and it can, ultimately, have a great influence on the final values of the deposition rates. It described how sticky the deposit actually is, which will indicate how strongly it attaches to the wall.

The values obtained for the deposit thickness were far lower than the ones expected. Nonetheless, when the analysis to the lowest exhaust mass flow was attended – 4.5kg/h – and compared to the time-dependent value from Völk (2014), it allowed the conclusion that, for low exhaust mass flows, the solution obtained by the use of the routines approximates the value expected. However, when increasing the exhaust mass flow – firstly to 9 kg/h -, the values showed far more disparity and, ultimately – at 15 kg/h – no deposit layer was even formed. This leads to the acknowledgement that, in the simulations with the routines, the removal forces play a bigger role with the increase in exhaust mass flow. As the routines use four empirical factors for the calculation of these forces, it might be indicative that, when changing these empirical parameters, the final results will also change greatly.

It is important to remark that the routines did not account for the wash-out effect that also may occur due to condensation, which might also explain some of the disparity observed and some of the results obtained.

The physical justification for the effects and evolutions observed allowed the conclusion that the routines provide reasonable and plausible values, which allow a good analysis and interpretation of the fouling behavior inside the EGR coolers.

8.2. Future Work

Independently of the objectives reached in this work, there are many other paths that ought to be explored and researched in order to improve the knowledge on EGR coolers and on the exact functioning of the user defined functions provided by Atlanting GmbH.

In order to continue the study and development of the EGR coolers, a few indications are drawn, regarding the use of the routines. One of the suggestions for future work is the complete simulation of the cooler, including the flow on the water side, which could help obtain more real and plausible results.

Another important approach to be taken in the future is the thorough study of the empirical factors that have an influence in the deposition and removal mechanisms. As previously seen, there are several empirical factors on the calculation of both the deposition and removal mechanisms that may carry a great influence on the final outcome, thus having a repercussion in the concluding analysis of the fouling effect in EGR coolers.

Additionally, other sources of study should be used as comparison. During the production of this work there was not enough time to address to other possible studies that could help fundament the validity of the routines.

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