Computers and Fluids 167 (2018) 158–165

Contents lists available at ScienceDirect

Computers and Fluids

journal homepage: www.elsevier.com/locate/compfluid

Development and verification of an in-flow water condensation model for 3D-CFD simulations of humid air streams mixing

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ARTICLE INFO

Article history:
Received 2 November 2017
Accepted 27 February 2018
Available online 6 March 2018

Keywords:
Turbocharger
Computational fluid dynamics
Long route EGR
Compressor damage

ABSTRACT

Bulk flow condensation caused by the mixing of air streams at different temperatures and humidities is a thermodynamic process that requires strong assumptions to be calculated with low computational effort. The applicability of a model that correctly predicts this phenomenon has grown recently due in part to the deployment of the Long Route Exhaust Gas Recirculation emission reduction technique in combustion engines and the damage to the turbocharger caused by the condensation produced when the intake air is mixed with the combustion gases. This work is addressed to expose a condensation model that is implemented in a commercial 3D-CFD code and is then verified, checking whether the implemented physical equations are behaving as intended. Finally, a practical application is made, showing the potential of model to predict water condensation in a LR-EGR T-joint.

1. Introduction

Due to the constant increase of emission regulations restrictions and the competitiveness in automotive engines, techniques that seek better efficiency combined with low emissions are being developed constantly. This non trivial combination of goals also requires improving or keeping the mechanical power.

Emissions reduction technologies used nowadays are either in-cylinder control or aftertreatment (A/T) control [1,2]. In-cylinder control systems reduce the generation of emissions within the source, during the combustion process. For instance, controlling the injection scheme [3], the maximum in-cylinder pressure, with thermal management [4] or using recirculating exhaust gases (EGR). On the other side, A/T systems reduce pollutant emissions along the exhaust line [5], such as diesel particulate filters (DPFs), selective catalytic reduction (SCR) or diesel oxidation catalysts (DOC) [6]. This work is focused on one of these systems, specifically on the exhaust gas recirculation (EGR). This consists in reintroducing part of the exhaust gases resulting from the combustion process back again in the cylinders mixed with fresh air, which causes a decrease in the maximum temperature achieved during the combustion process so that the Nitrogen oxides (NOx) formation is reduced while the mean pressure in the cylinders is kept almost the same. EGR most common application is CI engines, nevertheless it is beneficial in turbocharged SI engines too [7,8].

As turbochargers are indispensable in diesel engines, two possible paths of recirculation are available. Nowadays, the most common configuration consists in connecting both outlet and inlet manifolds, resulting in the so-called high-pressure or short route EGR (SR-EGR). Using this connection, a cooler and a valve to control the mass flow rate are usually employed. This system has been very popular during the last decades due to its simplicity. However, as it will be explained below, it introduces some drawbacks which are partially avoided by using the long route EGR (LR-EGR), which consists in extracting part of the low-pressure gases downstream the turbine and DPF and reintroducing them before the compressor.

The main practical difference between these two EGR configurations resides in the mass flow going through the turbine and the compressor. As the SR-EGR valve opens, the flow passing the turbocharger is reduced, shifting the working point towards generally lower efficiency values and reducing the compressor surge margin, hence requiring a selection of turbomachinery that withstands good performance over a wide range of operating conditions. In addition, lowering the turbine mass flow implies reducing its power, worsening its transient response as stated by Desantes et al. [9]. Conversely, with LR-EGR the turbine mass flow rate is not directly affected by the valve opening. From the point of view of air charge, the SR-EGR loop leads to an increase in the intake manifold temperature, which has a negative effect on BSFC (see Ladommatos et al. [10] and Desantes et al. [9]). Finally, a relevant aspect

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https://doi.org/10.1016/j.compfluid.2018.02.032

List of Symbols

c_p	isobaric specific heat capacity	$J \cdot kg^{-1} \cdot K^{-1}$
D	diffusion coefficient	$m^2 \cdot s^{-1}$
f	additonal terms	
h	enthalpy	$J \cdot kg^{-1} \cdot K^{-1}$
L	latent heat	$J \cdot kg^{-1}$
MW	molecular weight	$kg \cdot mol^{-1}$
$\dot{m}$	mass flow rate	$kg \cdot s^{-1}$
p	pressure	bar
RH	relative humidity	%
S	source term	
t	time	s
T	temperature	K
$\vec{u}$	velocity	$m \cdot s^{-1}$
w	specific humidity	$g_{H_2O} \cdot kg_{air}^{-1}$
x	coordinate in dominant direction	m
Y	mass fraction	
α	thermal diffusivity	$m^2 \cdot s^{-1}$
Δt	characteristic time	s
ϵ	relative difference	%
ν	kinematic viscosity	$m^2 \cdot s^{-1}$
φ	generic variable	
ρ	density	$kg \cdot m^{-3}$

Sub- and Superscripts

0	stagnation variable
air	dry air
egr	egr flow
ener.	energy
H_2O	liquid water
i	x,y,z cartesian components
in	intake flow
m	mass
mix	mixing flow
mom.	momentum
sat	saturation
t	thermal
v	viscous
vap	vapor component

List of abbreviations

0D	zero dimensional
3D	three dimensional
A/T	After treatment
BSFC	Break specific fuel consumption
CFD	Computational fluid dynamics
CI	Compression ignited
DOC	Diesel oxidation catalyst
DPF	Diesel particulate filter
LR-EGR	Long route exhaust gas recirculation
SR-EGR	Short route exhaust gas recirculation
NOx	Mono-Nitrogen oxides
RANS	Reynolds-averaged Navier-Stokes
SCR	Selective catalytic reduction
SI	Spark ignited

of the engine performance is the cylinder to cylinder EGR distribution. SR-EGR system may not be able to uniformly distribute EGR gas between the cylinders because the EGR outlet duct discharges over the inlet manifold, and the flow may not have enough time to be mixed with fresh air. This phenomenon is responsible for possible dis-uniformities of charge among cylinders [11]. On the opposite hand, LR-EGR mixes both flows far upstream the cylinders, letting a perfect distribution of the EGR gas between them.

Hence, the LR-EGR may look like a better solution although it is not without drawbacks [9,12]. Firstly, it presents a higher design complexity since inlet and outlet connections are relatively far from each other, meaning longer ducts within the underhood. In addition, the volume of the whole ducting implies a transient buffer of exhaust gas mass flow, which may decrease EGR rate responsiveness. The addition of the EGR discharge upstream the compressor produces a geometry modification that may impact on the compressor performance, as studied previously by some authors [13,14]. Finally, due to the high water content of the EGR flow, condensation may arise provided that the necessary conditions are met. This may be a problem for the LR-EGR, since water droplets may damage the compressor wheel. For the SR-EGR, since the exhaust gases are reintroduced downstream the compressor, condensation is less critical. In fact it may even be a feasible technique for on-board regeneration of the cooler efficiency [15,16].

Condensation when using LR-EGR may be produced in the EGR cooler as well. Despite the lower acidity due to suspended contaminants than in SR-EGR [17,18], condensation may still cause damage to the compressor wheel. LR-EGR condensation may also be produced within the flow when EGR stream is mixed with cold ambient fresh air. A combination of both ambient and engine/EGR operating point conditions may lead the mixing process to reach saturated conditions and the water to condensate. For example, ambient temperatures below +5 to +10 °C are susceptible to condensate EGR within the bulk of the mixing region for moderate EGR rates.

If condensation occurs, water droplets are driven towards the compressor wheel which is normally rotating at very high speed, implying elevated relative momentum and impact damage [19]. Dramatic consequences are observed at the leading edges of the compressor wheel if droplets impact the blades for long enough, as noticed by Serrano et al. [20]. Surface coating is normally applied to the wheel, partially protecting against this erosion, partially protecting against this erosion, nevertheless it is not infallible, as observed by Koka et al. [21]. Once the water penetrates the compressor inducer, due to its inertia, it may not follow the air streamlines but get attached to the wheel walls, forming rivers and leaving depositions [20]. The load of the compressor wheel may also change, according to Surendran et al. [22].

The complexity of the condensation within the flow has to be remarked. Wall condensation in the interior of tubes has been deeply studied by many authors [23,24], and wall condensation in free streams by others [25–27]. However, bulk flow condensation has not been researched extensively because there are less applications in which this process is important and in addition, a combination of both 3D fluid dynamic and psychrometric models are needed, since the complex turbulent flow structures upstream the compressor [14] and the mixing process control condensation rate [28]. Condensation will be produced at constant pressure and as a consequence of the mixing of two different flows at different psychrometric conditions.

It is worth noting that a different cause that also produces condensation in the bulk flow has been studied by Moses et al. [29] and later Grübel et al. [30], where the water appears due to the decrease of temperature caused by the acceleration of the flow in a de Laval nozzle. Due to the high velocities found in these sort of nozzles, small errors predicting the appearance of condensation imply greater errors in its location, therefore complex models that accurately predict the appearance and growth of water droplets within the fluid are required. The model developed in this work would not be suitable for solving this type of condensation.

The CFD code used for this work is CD-adapco STAR-CCM+® [31] which allows the creation of user field functions, enabling the implementation of a custom condensation model, as will be explained afterwards.