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Research paper

Simple models for the heat exchange from exhaust gas to super- and sub-critical refrigerant R134a at high temperature differences

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HIGHLIGHTS

- Good agreement between experiments and simulation despite the models' simplicity.
- Significant influence of refrigerant fluid property changes transverse to the flow.
- Negligible influence of sub-critical changes in flow boiling mechanisms.
- Improbable strong delay of the incipient point of sub-cooled flow boiling.
- Highest heat flux density for complex exhaust gas cross flow tube configuration.

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ABSTRACT

The present paper aims at the simple modeling of the heat exchange of the exhaust gas to a super- and sub-critical fluid for heat recovery applications at high temperature differences and the analysis of their influence on the transferred heat flow. Thereto, a simple shell and tube exhaust gas heat exchanger prototype generating super-critical or saturated sub-critical refrigerant R134a was experimentally investigated. Simulation models based on heat transfer correlations from literature are compared to experiments, heat transfer coefficients quantified and some design improvements discussed. The experimental heat flows are reproduced quite well despite the models' simplicity. The influence of the fluid property changes transverse to the flow on the calculated heat flows is significant, while the influence of the sub-critical changes in flow boiling mechanisms is negligible.

1. Introduction

Lately, there has been quite some interest in the improvement of system efficiency by using exhaust gas waste heat of combustion engines in vehicles [1–17]. Some research work is dedicated to shell and tube exhaust gas heat exchangers in particular [18–22]. One special feature of the heat exchangers is the large temperature difference between the exhaust gases and the working medium of the waste heat recovery process, in particular in gasoline engines, which results in large temperature differences between the bulk and the wall. The impact of the resulting property changes is not being analyzed in the mentioned works and only exceptionally accounted for in terms of viscosity ratios [12]. The heat exchange to a slightly super- or sub-critical working medium exhibiting strong changes in fluid properties or flow boiling mechanisms in the operation pressure range is not discussed in any of the studies, however, is of interest for waste heat recovery applications near the critical pressure of a working medium [cf. e.g. [23–25]]. Fluid properties in the super-critical pressure range of up to 1.2 times the critical pressure usually show strong absolute changes, steep gradients around the pseudo-critical temperature T_{pc} (defined as the temperature at which the isobaric specific heat capacity reaches its maximum), a local maximum in the heat capacity (and possibly the heat conductivity) as well as inflection points. In the sub-critical pressure range (with sub-cooling of about 65 K to 90 K in the present case), ratios of enthalpy differences of sub-cooled to saturated flow boiling vary from 2 to 4 [cf. [26]].

Proceeding further to application, we find that heat transfer coefficients of different heat exchanger geometries for exhaust gas heat exchangers are rarely quantified [cf. [27]] or experimentally validated to derive good practice in design and geometry. The paper at hand therefore elaborates on these two issues; a simple way of modeling the heat exchange from exhaust gas to super-

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Nomenclature

Dimensionless numbers

Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number

Greek letters

α	convective heat tr. coeff. [$Wm^{-2}K^{-1}$]
Δ	difference [−]
η	dynamic viscosity [Pas]
λ	heat conductivity [$Wm^{-2}K^{-1}$]
ν	kinematic viscosity [m^2s^{-1}]
ρ	density [kgm^{-3}]

Latin letters

A	area [m^2]
c_p	isobaric specific heat capacity [$Jkg^{-1}K^{-1}$]
d	diameter [m]
h	specific enthalpy [Jkg^{-1}]
H	height [m]
K	correction factor [−]
k	overall heat tr. coeff. [$Wm^{-2}K^{-1}$]
L	length [m]
$\dot{M}$	mass flow rate [kgs^{-1}]
$\dot{m}$	mass flux [$kgs^{-1}m^{-2}$]
m	exponent [−]
n	number of tubes [−]
P	perimeter [m]
p	pressure [MPa]
$\dot{Q}$	heat flow [Jkg^{-1}]
$\dot{q}$	heat flux [Wm^{-2}]
R	specific gas constant [$Jkg^{-1}K^{-1}$]
T	absolute temperature [K]
V	volume [m^3kg^{-1}]
$\dot{V}$	volumetric flow rate [m^3s^{-1}]
v	specific volume [m^3kg^{-1}]
w	velocity [ms^{-1}]
W	width [m]
$\dot{x}$	vapor flow content [−]

Sub-/Superscripts/Abbreviations/Modified characters

$\mathscr{A}$	correction function
$\mathscr{B}$	dimensionless parameter
$\mathscr{K}$	curvature factor
a	average
C	condenser
c	transverse to the flow
cor	corrected
cr	critical
CW	cooling water
E	evaporation
Ex	exhaust gas
H	heater
h	hydraulic
i	inlet
j	running index
l	in flow direction
lam	laminar
o	outlet
pc	pseudo-critical
R	refrigerant
r	reference
S	spray flow
sb	sub-cooled flow boiling
sim	simulation
tr	transition
tp	two-phase
tu	turbulent
uncor	uncorrected
W	wall
WG	water glycol
0	characteristic
1	condenser, inlet
2	heater, outlet
$'$	vapor

and sub-critical refrigerant R134a at high temperature differences across the heat exchanger on the one hand and the quantification of achievable heat transfer coefficients for designing simple but power dense and efficient exhaust gas heat exchangers on the other hand. To this end, the test-rig of the experiments conducted with an exhaust gas heat exchanger is described first. Consecutively, two models are presented; super-critical convective heat exchange and sub-critical boiling are treated separately due to the different mechanisms. They take into account the particularities of both large temperature differences and the operation near or above the critical point. The experimental results are analyzed using the developed models based on adequate corrections of the respective Nusselt relations from literature with regard to large temperature differences and on equations specific to the refrigerant flow boiling heat transfer.

2. Test-rig and heat exchanger prototype

Fig. 1 depicts the schematic of the refrigerant circuit, the exhaust gas path and the cooling water circuit of the test-rig for the exhaust gas heat exchanger prototype under investigation.

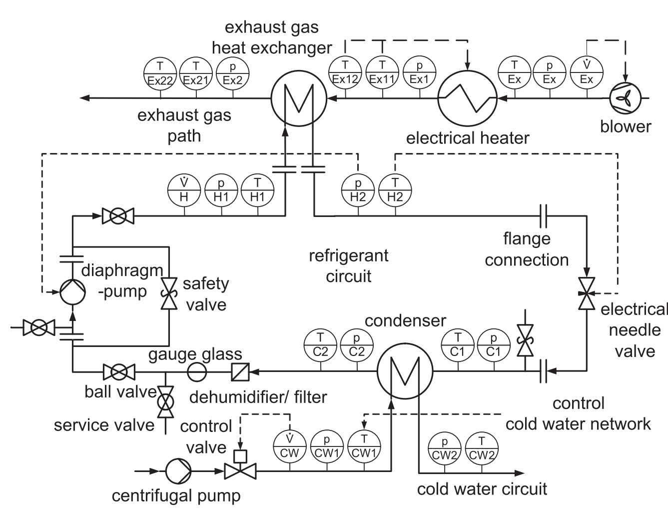

Fig. 1. Schematic of the test-rig for the exhaust gas heat exchanger.