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Condensation heat transfer on single horizontal smooth and finned tubes and tube bundles for R134a and propane

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ABSTRACT

For the refrigerants 1,1,1,2-tetrafluoroethane (R134a) and propane (R290), the condensation heat transfer was investigated on coated and uncoated horizontal smooth, standard finned and high performance tubes by experiments and computational fluid dynamics (CFD)-simulations. Measurements were carried out at a saturation temperature of 37 °C varying the heat flux between 4 kW m^{-2} and 102 kW m^{-2}. In comparison to the Nußelt theory, enhancement factors of 12.8 to 30.2 were experimentally found for single standard finned and single high performance tubes. For both refrigerants, the high performance tubes showed a larger bundle effect than the standard finned tubes, although the latter show a lower heat transfer performance. In general, experimental results for coated tubes show a slightly lower heat transfer performance and no improvements of the condensate drainage could be observed for the modification. CFD-simulations were used to predict the condensation heat transfer on single tubes for the first time. Our experimental data and CFD-simulations were compared with analytical models available in the literature.

1. Introduction

Shell and tube condensers which are used, e.g., in refrigeration, air conditioning, and petroleum industry, are objects of continuous improvement and optimization. In order to increase their efficiency, finned tubes are used, which can be classified into standard finned or high performance tubes. The latter have additional structures on the fin flank leading to a significant increase in the condensation heat transfer coefficient (HTC) on single tube. For the design of tube bundles with standard finned or high performance tubes, the performance of the single tube and the rate of condensate must be taken into account. The drained condensate from upper tubes leads to an additional mass flow rate on the lower tubes in the bundle. Thus, the thermal resistance of the condensate layer increases and the condensation HTC on lower tubes is reduced in comparison to upper ones. This phenomenon, which reduces the performance and efficiency of shell and tube condensers significantly, is known as bundle effect.

Many efforts can be found in literature, which are focused on the study of the condensation HTC on the outside of horizontal single tubes and tube bundles with synthetic refrigerants [1–10]. Due to the extensive safety precautions, which must be fulfilled in connection with flammable natural refrigerants, only few results can be found for their condensation HTC [11]. Within this work, the condensation HTC was studied for propane (R290) and 1,1,1,2-tetrafluoroethane (R134a) on smooth, finned and high performance tubes. With a new developed apparatus, the condensation HTC on single tubes and tube bundles should be experimentally investigated. Furthermore, for the first time to the best of our knowledge, CFD-simulations should be performed to calculate theoretically the condensation HTC outside smooth tubes and low finned tubes with trapezoidal fins. Here, the validity of the developed CFD-model should be verified by comparing the theoretical results with the experimental data. Additionally, the effect of surface modifications, e.g., in form of ion implantation and plasma polymer coating, on the heat transfer performance should be tested. The individual tasks mentioned above should contribute to a fundamental understanding of the bundle effect. In the following, the apparatus and the data evaluation will be presented as well as the investigated tube characteristics. Subsequently, the basics of the CFD-simulations for modeling the condensation HTC

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Nomenclature

A	area
a	normalized effective tube surface area per tube length
C	adjustable parameter
c_p	specific heat capacity at constant pressure
D	diameter
g	gravity
h	fin height
Δh_{LV}	heat of vaporization
k	overall heat transfer coefficient
L	length
$\dot{m}$	mass flow rate
n	number of tubes
p	pitch
$\dot{Q}$	heat flow
$\dot{q}$	heat flux
s	tube wall thickness
T	temperature
$\dot{V}$	volume flow rate

Greek symbols

Θ	condensation retention angle
α	heat transfer coefficient
λ	thermal conductivity
μ	viscosity
ξ	pressure loss coefficient
ρ	density

Subscripts

c	condensate
Cond	condensation
CW	cooling water
dyn	dynamic
f	fin
Film	film
in	inside
int	interface
Inund	inundation
j	running number
L	liquid
LMTD	logarithmic mean temperature difference
m	mean
n	tube row number
out	outside
R	root
Sat	saturated vapor
static	static
T	tube
V	vapor
VE	volume element
W	wall

Abbreviations

CFD	computational fluid dynamics
fpi	fins per inch
HTC	heat transfer coefficient
Pr	Prandtl number
R134a	1,1,1,2-tetrafluoroethane
R290	propane
Re	Reynolds number
UDF	user defined function
VOF	volume of fluid

will be introduced. Finally, the experimental and theoretical data are discussed in comparison with literature including analytical models.

2. Experiment

2.1. Apparatus

Measurements for the condensation HTC for R134a and propane are performed with the apparatus shown in Fig. 1. The main components are the condenser (1) and the evaporator (2), which are connected on the liquid and vapour phase. The refrigerant natural circulation is presented in Fig. 1 with dashed lines. The condenser and the evaporator are designed for a maximum operating pressure of 40 bars and a maximum temperature of 100 °C. For visual observations of the condensation flow mode, the condenser is equipped with three optical accesses with a diameter of 100 mm. By using hot water inside the evaporator tubes (9, 10) the refrigerant is evaporated and flows through three vapor tubes (5) and a demister unit (3) into the condenser. Below the vapor inlets of the condenser, a baffle plate is installed to distribute the vapor refrigerant uniformly and to avoid vapor shear effects on the test tubes. The vapor refrigerant is liquefied on the outside of the water cooled smooth, standard finned or high performance tubes. The condensate flows back (8) into the evaporator due to the hydrostatic pressure difference between the condenser and the evaporator. The eight condenser tubes with an effective length of 1.46 m are arranged into two columns. The two columns are connected horizontally with a U-bend to reach a total tube length of 2.92 m for one row. The cooling water in- and outlets are alternatively arranged. Thereby, a uniform distribution of condensate over both tube columns is ensured. With the apparatus the condensation HTC on single tubes or on tube bundles can be investigated.

For the experimental simulation of a tube bundle, the test facility was equipped with a liquid overfeed unit, which is represented by solid lines in Fig. 1. For this purpose, a storage tank for the refrigerant (12) is connected with the liquid (11) and vapor phase (22) of the evaporator. Furthermore, the storage tank is connected with an oil-free refrigerant pump (13), which is used to feed the liquid refrigerant first through a coaxial heat exchanger (15) for the temperature control of the refrigerant close to saturation conditions. The refrigerant flows through a subsequent variable area flow meter (16) into the overfeed tubes, which are installed directly above the two condensation tube columns. The required overfeed flow rate is controlled coarsely with the help of the bypass of the pump, and is adjusted fine by using a needle valve (14).

During operation, the refrigerant pump adds a considerable amount of heat to the system. Most of the refrigerant is circulated by the bypass (18) back into the storage tank. However, a plate heat exchanger (19) is installed to control the temperature of the refrigerant in the bypass with the help of a chiller (20). After the variable area flow meter a distribution unit (17) feeds the two overfeed tubes with refrigerant from both sides. The temperature of the overfeed is measured close to the entrance of the overfeed tubes. Before entering the condenser, the volume flow of the overfeed refrigerant is adjusted by four needle valves to achieve a uniform distribution over the whole tube length.