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Preliminary measurements of heat transfer during condensation in microchannels

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ABSTRACT

A recent theory of condensation in microchannels employs only the Nusselt approximations and has no empirical input. The predictions are in good agreement with experimental data for fluids with relatively low surface tension, namely R134a and FC72. The theory demonstrates that surface tension plays a vital role for non-circular channels by causing condensate to be driven towards the corners leaving a thin film with consequently high heat-transfer coefficient along the sides. The paper reports provisional results of a new research programme aimed at providing very accurate data for fluids covering a wide range of properties, notably surface tension. New experimental results are reported here for steam where the surface tension is much higher than the fluids used hitherto. Preliminary results show the same trends as found with lower surface tension fluids but heat-transfer coefficients for steam are higher by factors of ten or more.

Nomenclature

G_v	steam mass flux in channel
m_c	total mass flow rate of coolant
$P_{v,in}$	vapor pressure at inlet
$Q_{c.low}$	heat-transfer rate to coolant in lower test section channel
$Q_{c.up}$	heat-transfer rate to coolant in upper test section channel
T_c	coolant temperature at inlet
$T_{s,in}$	vapor saturation temperature at inlet pressure
$T_{v,in}$	vapor temperature at inlet
$T_{v,out}$	vapor temperature at exit
x_o	vapor quality at exit
y	distance of thermocouple from top surface of test block
ΔP_v	pressure drop indicted by differential transducer
$\Delta T_{c.low}$	coolant temperature rise in lower test section channel
$\Delta T_{c.up}$	coolant temperature rise in upper test section channel
α	local channel surface heat-transfer coefficient
θ	Celsius temperature

INTRODUCTION

Air conditioning/refrigeration plant account for a significant proportion of electricity usage and consequently for fuel consumption and associated CO_2 emissions. Improvement in plant design and performance can make an important contribution to mitigation of these problems. Condensers employing multi-microchannel tubes (typical channel dimension ~ 1 mm) have been used successfully in automotive air conditioners for around 25 years and have proved both compact and effective. An overview of microchannel condensation has been given by Garimella (2005). Available experimental heat-transfer data for condensation in microchannels are widely scattered (see Su et al. (2009)). Vapor-side, heat-transfer coefficients have, with a few exceptions (Koyama et al. (2003a, 2003b), Cavallini et al. (2003, 2004, 2006), Matkovic et al. (2009), Derby et al. (2012) Kim et al. (2012), Kim and Mudawar (2012a, 2012b)) generally been inferred from overall measurements by subtraction of thermal resistances and/or using ‘‘Wilson plot’’ methods; such data have high uncertainty. Four heat-transfer correlations, based mainly on data for R134a (Wang et al. (2002), Koyama et al. (2003a,b), Cavallini et al. (2005), Bandhauer et al. (2006)), have been proposed. These agree quite well with each other when applied to R134a but wide differences (up to a factor of around four) are found when applied to ammonia (see Su et al. (2009)). A theory (Wang and Rose (2005, 2011)), based only on the approximations of Nusselt (1916) and having no empirical input, can only be compared with the correlations for R134a in an approximate manner (see Wang and Rose (2014)); the comparisons show generally satisfactory agreement. Accurate comparisons can be made with detailed direct measurements of Koyama et al. (2003a,b) for R134a and Kim et al. (2012), Kim and Mudawar (2012a, 2012b) for FC72 and remarkably good agreement found (Wang and Rose (2014)). All comparisons to date are for relatively low surface tension fluids. The present work was untaken in order to test the theory more stringently by using steam/water with widely different properties, notably surface tension.

APPARATUS

Figure 1 shows a flow diagram of the apparatus. The lower loop shows the circuit for the condensing fluid (in this case water/steam) and the upper loop shows the coolant (water) circuit. Steam from the

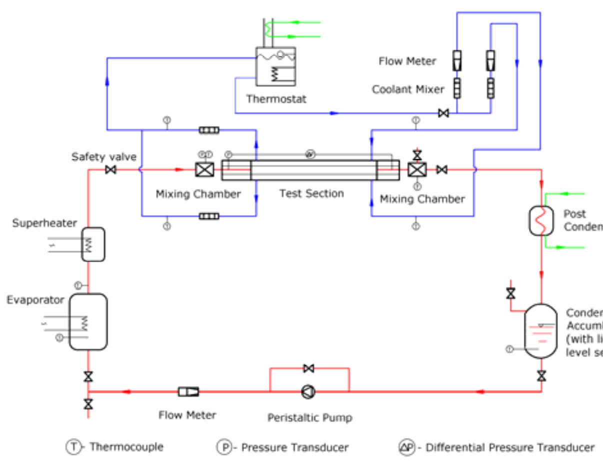

Fig. 1 Apparatus flow diagram

evaporator and superheater entered the test section inlet plenum via a mixing chamber. From the plenum the steam was distributed into 6 parallel channels (1.5 mm high x 1.0 mm wide) of the copper test section. The test section was cooled from above and below with water (see Fig. 2). As shown in Fig. 1 the condensate and excess vapour pass from the test section into an auxiliary condenser from where the condensate is returned via a pump and flow meter to the evaporator. Excess steam and condensate were vented from time to time at exit from the test section to minimize the possibility of non-condensing gas effects. Time was allowed to attain steady conditions after venting. Pressure and temperature were accurately measured in the inlet plenum. The temperature was also measured in the exit plenum and at entry to the evaporator. A differential pressure transducer was used to determine the pressure drop between positions 20 mm from the entry and exit, the channels being connected at the pressure tapping by a slot machined at right angles to the channels. The evaporator, superheater and test section were well insulated and a second determination of the vapor mass flow rate made from a steady flow energy balance between the boiler and test section inlets using the accurately measured power inputs. A small correction for thermal losses was made based on preliminary tests where the power input needed to just provide steam at the inlet plenum was measured (see Lee and Rose (1984)). The two measurements of flow rate agreed to within about 4%. The cooling water passed in counter flow over the upper and lower surfaces of the copper test block via flow meters and with mixers at inlet and exit. The coolant flow rates were set equal for the two streams.

The test section comprised a copper block of length 500 mm, width 40 mm and height 30 mm, made in two halves with lapped mating surfaces (see Fig. 2). Six parallel rectangular grooves (1.5 mm x 1.0 mm) were machined in the lower half of the test block flanked by O-ring grooves. Seven thermocouples were located in 0.6 mm diameter holes drilled to the centre in each of the block halves at 7 positions along the flow path making a total of 98 thermocouples in all.

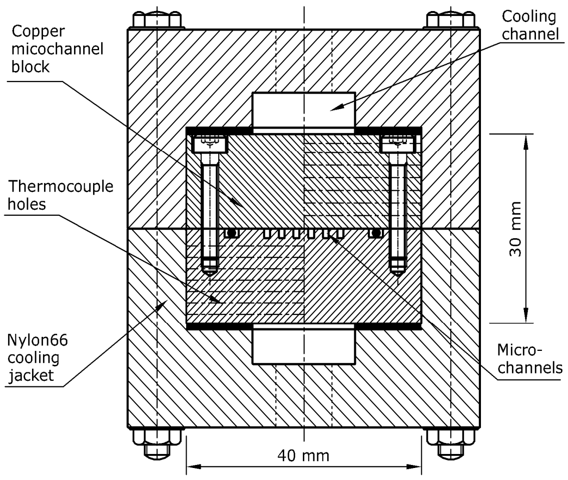

Fig. 2 Section though test section

PROCEDURE

All thermocouples were calibrated (±0.02 K) against a platinum resistance thermometer in a high-accuracy constant-temperature bath. The specifications of the instruments used are given below.

<u>Pressure transducers</u>: Honeywell Super TJE (absolute pressure) and FP-2000 (differential pressure); accuracies ±0.25 kPa and ±0.21 kPa, respectively; maximum working temperature of 120 °C.
<u>Peristaltic pump</u>: Watson-Marlow model 520U; accuracy of ±1.59 ml/min.
<u>Coolant thermostat</u>: Thermo Scientific HAAKE Phoenix II P2-B12 model; accuracy ±0.01 K.
<u>Flow meters</u>: for coolant are model Omega FL-10J-V; accuracy 0.25 l/min.

The superheater was not operated in the present preliminary tests and the vapour was saturated at the test section inlet in all cases. All tests were made with incomplete condensation in the test section. Condensate from the post condenser returned via the accumulator to the boiler. All tests were done under steady conditions when power inputs, temperatures, pressures and accumulator liquid level were constant. Measurements were recorded using a data-acquisition system (an Agilent 34580A frame unit and two Agilent 34922A modules with 70 channels each). Tests were done for a range of coolant flow rates and coolant inlet temperatures. The experimental parameters for the tests are given in table 1. The flow rate of steam to the test section given in table 1 was that determined by a steady flow energy balance with the measured power input to the boiler.

No.	G_v / kg m^{-2} s^{-1}	$P_{v,in}$ / kPa	$T_{v,in}$, ºC	$T_{s,in}$, ºC	$T_{v,out}$, ºC	T_c , ºC	m_c / g s^{-1}
1	163.7	156.3	112.5	112.6	99.2	28.58	35.3
2	163.7	161.7	113.6	113.6	99.2	28.58	19.9
3	163.7	164.7	114.1	114.2	99.2	28.49	13.0
4	163.7	167.3	114.6	114.7	99.2	38.16	13.0
5	163.7	172.4	115.5	115.6	99.3	47.85	12.9
6	163.7	175.9	116.2	116.2	99.3	58.35	12.9
7	163.7	176.5	116.3	116.3	99.4	68.47	19.5
8	163.7	192.5	119.0	119.0	99.8	67.86	8.0
9	150.6	147.8	110.9	110.9	98.8	28.74	35.3
10	150.6	149.7	111.3	111.3	99.0	28.71	27.6
11	150.6	157.6	112.8	112.8	99.3	28.62	13.1
12	150.6	160.9	113.6	113.5	99.3	28.68	13.1
13	150.6	165.1	114.3	114.2	99.6	38.29	13.0
14	150.6	168.8	114.9	114.9	99.7	47.77	13.0
15	150.6	169.9	115.2	115.1	99.8	56.91	12.9
16	150.6	174.4	116.0	115.9	100.0	68.48	12.8
17	148.0	176.1	116.3	116.2	100.6	78.26	12.8
18	126.7	142.1	109.6	109.7	99.3	24.75	13.1
19	126.7	143.9	110.0	110.1	99.3	32.88	13.0
20	126.7	146.0	110.5	110.5	99.4	38.17	13.0
21	126.7	149.7	111.2	111.3	99.6	47.80	13.0
22	126.7	153.9	112.1	112.1	99.7	58.14	12.9
23	126.7	157.7	112.9	112.9	99.9	68.40	12.8

Table 1 Parameters used in preliminary tests

RESULTS

The measured quantities are shown in table 2.

Differences between the heat transfer rates for the upper and lower channels are due to lack of precision with which the two flow rates could be set equal. The quality at exit was determined from a steady flow energy balance using the measured heat transfer rates to the coolant streams.

A typical set of observed temperature distributions through both halves of the test block at the seven positions along the flow path is shown in Fig. 3. The temperature and heat flux at the upper and lower surfaces of the channels was determined by inverse solution of the conduction equation (see Yu et al. (2014)). It is evident that closely similar results would have been obtained in this case by fitting straight lines. In this test run the channel surface temperature fell from about 105 °C at the most upstream position to about 94 °C at the most downstream position while the heat flux does not vary greatly.

No.	ΔP_v / kPa	$\Delta T_{c,up}$ / K	$\Delta T_{c,low}$ / K	$Q_{c,up}$ / kW	$Q_{c,low}$ / kW	x_o
1	54.22	17.45	16.57	1.288	1.224	0.25
2	60.03	31.21	35.69	1.296	1.483	0.17
3	63.15	46.21	42.97	1.259	1.171	0.28
4	65.66	41.43	35.98	1.126	0.978	0.38
5	70.73	35.58	29.39	0.964	0.797	0.48
6	74.07	30.47	25.17	0.823	0.680	0.56
7	74.44	20.44	15.12	0.838	0.620	0.57
8	89.01	26.27	23.32	0.442	0.392	0.76
9	47.03	14.10	14.61	1.042	1.079	0.32
10	48.61	19.20	17.34	1.107	1.000	0.32
11	56.41	37.94	33.77	1.035	0.922	0.37
12	58.49	30.80	28.14	0.842	0.769	0.48
13	61.64	28.46	24.82	0.775	0.676	0.54
14	65.26	25.93	22.27	0.704	0.605	0.58
15	66.18	22.85	17.94	0.618	0.486	0.65
16	70.11	21.31	15.65	0.574	0.421	0.69
17	69.82	16.83	15.03	0.452	0.403	0.73
18	39.69	33.70	35.57	0.921	0.972	0.27
19	41.39	32.53	32.25	0.887	0.879	0.32
20	44.50	31.25	27.75	0.851	0.756	0.38
21	47.43	28.40	22.92	0.771	0.622	0.47
22	51.07	25.00	18.83	0.676	0.509	0.55
23	54.22	21.12	14.69	0.569	0.396	0.63

Table 2 Measured quantities

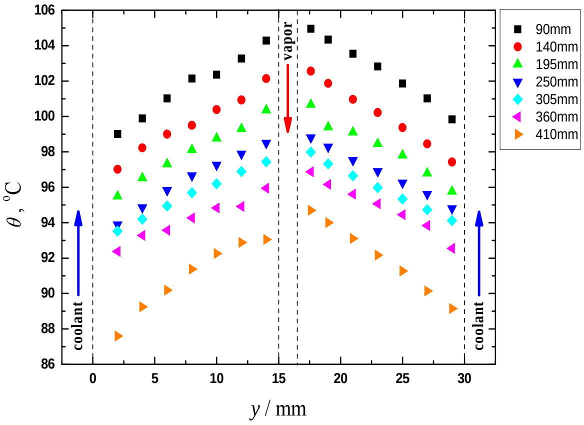

Fig. 3 Specimen temperature distributions in test block. Symbols denote distance from channel inlet (case 9). Positions of thermocouple junctions measured from upper surface (left side is upper half of test bock).

Figure 4 shows the distribution of heat-transfer coefficient along the channel walls (the Case No. is the same as the test No. listed in tables 1 and 2). As expected, the heat-transfer coefficient is very high near the inlet and falls in all cases to a fairly constant value of around 60 kW/m^2 K after a distance of around 200 mm along the channel.

Though the results for the different conditions do not vary greatly there is evidence that for the region of the channel near the inlet the heat-transfer coefficient is higher for high vapor mass flow rates.

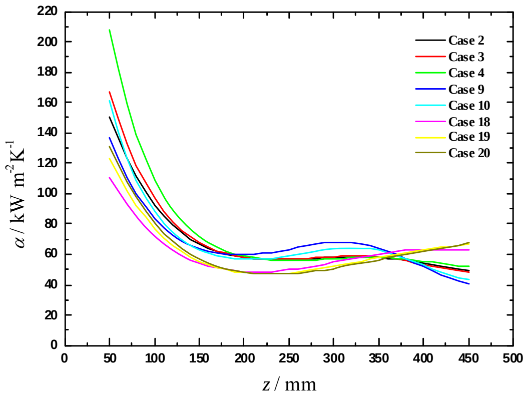

Fig. 4 Distribution of vapor-wall heat-transfer coefficient along the channel.

DISCUSSION

Only in two earlier investigations (Koyama and co-workers for R134a (Koyama et al. (2003a,b) and Kim and co-workers for FC72 (Kim et al. (2012), Kim and Mudawar (2012a, 2012b))) have detailed measurements been made in which the vapor at inlet was superheated or saturated as required for accurate comparison with theory. As indicated by Wang and Rose (2013), good agreement with these data was found. Detailed comparisons of results of the present investigation with theory will be made in due course when more extensive data have been obtained for steam as well as for FC72. However, it is noted here that the general trends of the results for steam shown in Fig. 4 are in line with those for the lower surface tension fluids but the heat-transfer coefficients are higher for the steam case by a factor of ten or more.

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