BOILING OF TWO ZEOTROPE MIXTURES AND R-502 INSIDE A PLAIN HORIZONTAL TUBE

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ABSTRACT

Intube flow boiling experiments for two new zeotropic refrigerant mixtures and R-502 have been carried out at three saturation temperatures over a wide range of local vapor qualities. The zeotropic mixtures tested were HP80 and HP62. Local flow boiling coefficients for HP62 are slightly larger than those for R-502 under the same test conditions while those for HP80 are slightly smaller. The heat transfer performances of the three fluids are generally quite accurately predicted by an existing flow boiling correlation modified here to include a nucleate boiling mixture equation. However, in the post dryout regime at high vapor qualities the measured coefficients are much lower than those predicted.

INTRODUCTION

A two-phase flow and heat transfer test program was undertaken to obtain flow boiling test data for two new replacements for R-502. Flow boiling of zeotropic refrigerant mixtures has recently become an important research topic while work on binary and multicomponent hydrocarbon mixtures and aqueous mixtures has already been underway for several decades. For a comprehensive review of forced convective boiling refer to Collier and Thome [1] while for a complete review of mixture boiling fundamentals see Thome and Shock [2]. Enhanced mixture boiling has been reviewed in Thome [3].

TEST FACILITIES AND PROCEDURES

Figure 1 depicts a simplified flow diagram of the test facility. The two double-pipe sections mounted in series have refrigerant flowing inside the inner tube with heat supplied by hot water flowing counter-currently in the annulus. The refrigerant is circulated by a stainless steel pump with a magnetically driven rotor which operates without any lubricating oil, eliminating any possibility of oil entering the test loop. A speed control on the pump is used to modify the refrigerant flow rate. The chilled water-glycol solution circulated through the condenser is cooled with a separate refrigeration system. The hot water is circulated by a stainless steel pump to eliminate corrosion and rust.

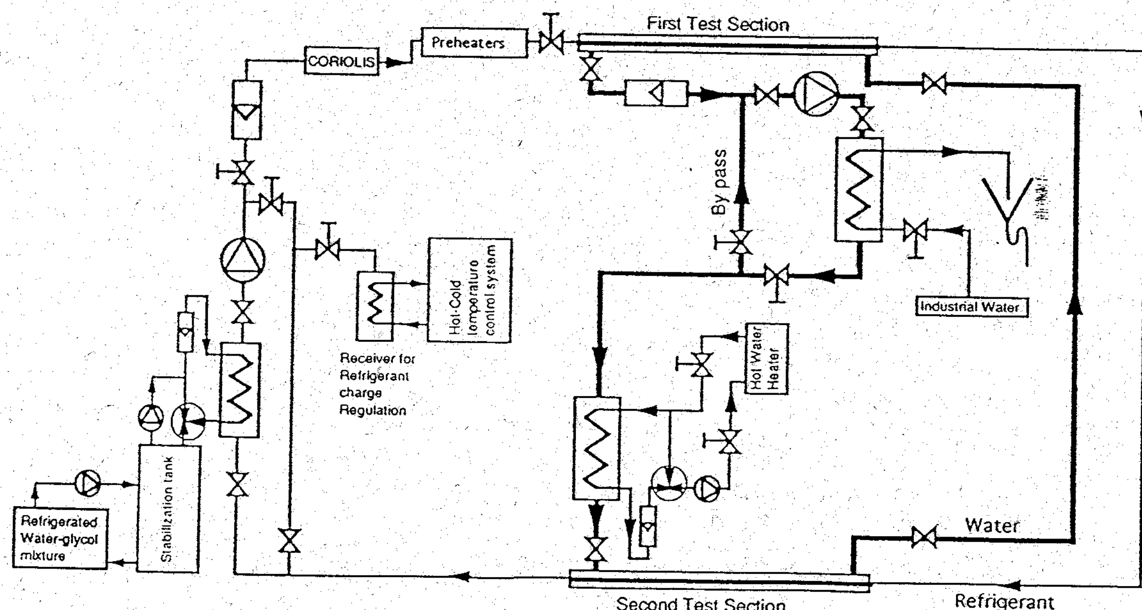

Figure 1. Layout of LENI test facility flow circuits.

A data acquisition system together with a PC computer acquires, processes and stores the data. The thermocouples are calibrated using the double precision method available in the acquisition system to obtain the highest accuracies possible, that is 0.03°C. After leaving the rig off all night, all temperatures are within a maximum deviation of 0.10°C and most are within 0.05°C of the average. The refrigerant flow rate is measured to 0.2% by a Coriolis meter and the water flow rate to within 1.0% by a float meter. The energy balances between the water and refrigerant flows agree to within 2% maximum error. The experimental heat transfer coefficients are estimated to have a maximum error of about 7.5% but usually less, depending on the test conditions.

The two test sections are plain, drawn copper tubing of 12.00 mm bore with 1.00 mm tube walls. Each section is 3.013 m long for heat transfer measurements. The outer tubes of the double-pipe test sections are a novel design. They are precision machined in two halves from PVC Hard, which is a very hard and durable plastic material that also insulates the test section. Its internal diameter is thus very precise, much more than that of standard pipe.

An O-ring is compressed between the two halves to form the seal. O-ring sealed screws are inserted through the outer PVC tube to center the copper test sections.

Figure 2 shows the test section measurement layout diagram. At the refrigerant inlet, thermocouple #600 is located in the refrigerant flow stream (the tube is well insulated from this point up to the location of the four water-side thermocouples #614, #613, #501 and #615 with a tightly-fitted teflon sleeve) similar to the refrigerant thermocouples #601, #602 and #603. At these locations the absolute pressures of the refrigerant and the differential pressure drops across the two test sections are measured. On the water-side, four thermocouples are installed at each measurement location in the annulus at 0°, 60°, 120° and 180° from the top. There are four measurement locations on both the top and bottom test sections and thus there are six test zones to obtain evaporation coefficients over narrow changes of vapor quality (essentially local values) during tests.

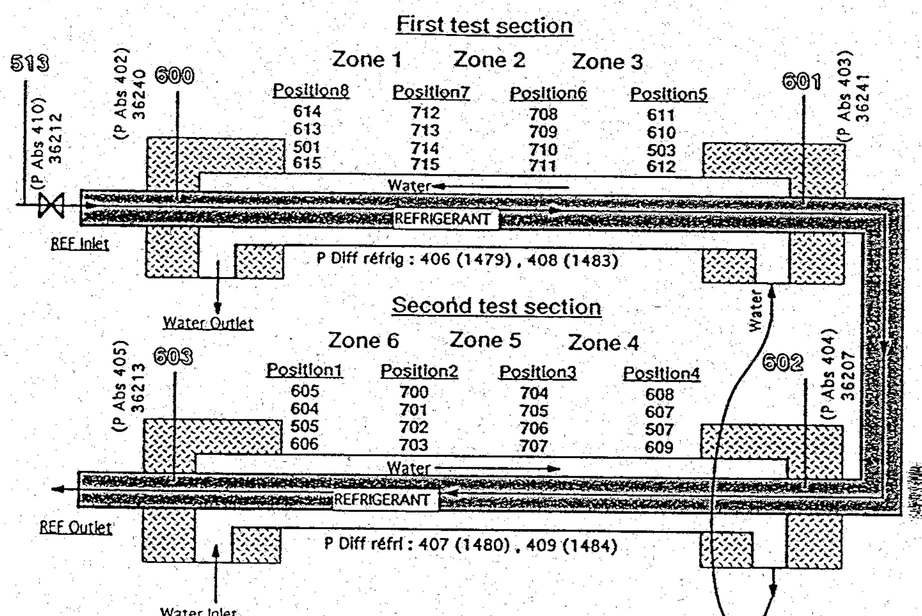

Figure 2. Measurement locations in the test sections.

MODIFIED WILSON PLOTS AND TEST FLUIDS

Traditional modified Wilson plot methods are based on the Dittus-Boelter type of turbulent flow heat transfer correlation. However, this equation cannot model the transition flow regime of the water in the annulus, i.e. $2300 < Re_D < 10{,}000$. A new approach was thus developed using the Gnielinski [4] heat transfer correlation applicable for the range $2300 < Re_D < 1{,}000{,}000$.

HP80 and HP62 are ternary "near-azeotropic" mixtures. Their boiling ranges (dew point temperature minus bubble point temperature) are 1.01°C and 0.45°C, respectively.

HP80 has a composition of 60% R-125/2% R-290/38% R-22 (by wt.) with the designation R-402A while HP62 is composed of 44% R-125/52% R-143a/4% R-134a (by wt.) and was recently designated as R-404A. R-502 is a binary azeotropic mixture (48.8% R-22/51.2% R-115) with no boiling range. R-502 properties were obtained with ASHRAE methods; those for HP80 and HP62 were obtained from DuPont tables or using REFPROP [5].

DEFINITION OF MIXTURE BOILING COEFFICIENT

For a mixture, the heat transfer coefficient is defined using the bubble point temperature corresponding to the local bulk liquid concentration at the local pressure as

$$\alpha = q/(T_w - T_{bub}) \tag{1}$$

During the evaporation of a mixture, the composition of the more volatile component in the vapor is larger than in the liquid. Consequently, the local bubble point temperature increases as the composition of the more volatile component in the liquid phase decreases during evaporation along a heated tube. The change in enthalpy H of the mixture during evaporation along the tube is:

$$dH = h_{LV}\, dx + (1-x)\, dT_{bub}\, (c_p)_L + x\, dT_{bub}\, (c_p)_V \tag{2}$$

whose terms are comprised of (i) the latent heat of vaporization of the amount vaporized dx, (ii) the sensible heating of the liquid phase (1-x) and (iii) the sensible heating of the vapor phase x. The values of h_{LV}, $(c_p)_L$ and $(c_p)_V$ are a function of the local liquid and vapor compositions. The heat release curves were determined using REFPROP with <u>user-supplied</u> binary interaction parameters. Using the mean bubble point temperature for the zone, the boiling heat transfer coefficient was calculated from the water-side coefficient, the heat duty and the log mean temperature difference (LMTD) for the zone and the thermal resistance of the copper tube.

PREDICTION OF FLOW BOILING HEAT TRANSFER TO MIXTURES

The present test data were compared to two pure fluid correlations of Gungor and Winterton [6,7] modified to include the Thome [8] mixture boiling correction factor:

$$F_c = \left[1 + (\alpha_I/q)\, \Delta T_{bp} \left[1 - \exp\left(\frac{-\, b_o\, q}{\rho_L\, h_{LV}\, \beta_L}\right)\right]\right]^{-1} \tag{3}$$

where α_I is the ideal heat transfer coefficient calculated with the Gungor-Winterton correlation without mixture effects but with the mixture physical properties, ΔT_{bp} is the boiling range, b_o is a scaling factor (equal to 1.0), q is the local heat flux, ρ_L is the liquid density of the mixture, h_{LV} is the latent heat of the mixture and β_L is the liquid phase mass transfer coefficient that is assumed to have a constant value of 0.0003 m/s. Eq. (3) is an analytical expression and has been shown to work for pool boiling of aqueous, hydrocarbon, cryogenic and refrigerant mixtures; it is accurate up to boiling ranges of 30°C, beyond which other factors become important. F_c approaches 1.0 at very low heat fluxes and decreases in value with increasing heat flux. Eq. (3) can be inserted into any pure fluid flow boiling correlation.

The Gungor-Winterton [7] correlation is given as:

$$\alpha = E\,\alpha_L \tag{4}$$

where E now includes F_c applied to the Boiling number Bo (which models the nucleate boiling contribution) such that

$$E = 1 + 3000\,(Bo\,F_c)^{0.86} + 1.12\,[x/(1-x)]^{0.75}\,(\rho_L/\rho_V)^{0.41} \tag{5}$$

For HP80 and HP62, the boiling ranges are very small and F_c ranges from 0.98 to 0.99, i.e. almost no adverse mixture mass diffusion effect. The Gungor-Winterton [6] correlation is similarly modified such that

$$\alpha = E_o\,\alpha_L + S\,\alpha_{pool}\,F_c \tag{6}$$

EXPERIMENTAL RESULTS

Figure 3 depicts boiling data at three mass velocities in the heat flux range of 8000-10000 W/m^2 for HP80, HP62 and R-502. A distinct maximum in heat transfer coefficient at a vapor quality of about 8% is observed at the highest mass velocity while a more moderate one occurs at about 15% at the medium mass velocity. The heat transfer coefficient is seen to decrease monotonically with increasing vapor quality at the lowest mass flow rate. The maxima may be caused by a change in flow pattern or the onset of nucleate boiling. A moderate increase in the heat transfer coefficient is observed with mass velocity as expected. The boiling coefficients trail off to lower values at high vapor qualities.

Figure 4 depicts a direct comparison of experimental data for HP80 and HP62 to R-502 at a fixed inlet dew point temperature of 2.4-2.6°C and also shows their confrontation with the Gungor and Winterton correlations [GW86 and GW87, respectively]. Above, the heat transfer coefficients are seen to be very similar over the vapor quality range from 10-90%. HP80 tends to have a thermal performance slightly below that of R-502 while that of HP62 is almost the same. The GW87 correlation including the Thome mixture boiling correction factor F_c accurately predicts the HP80, HP62 and R-502 data. The correlation follows the variation with vapor quality rather well but underpredicts at low vapor qualities and overpredicts at qualities approaching 90%. The GW86 correlation including F_c significantly overpredicts the HP80 and HP62 data while modelling the R-502 data accurately. The average coefficients for the experimental data in Figure 4 are 1229, 1384 and 1413 W/m^2 K for HP80, HP62 and R-502, respectively; thus the average values for HP80 and HP62 are -13% and -2% relative to R-502, respectively.

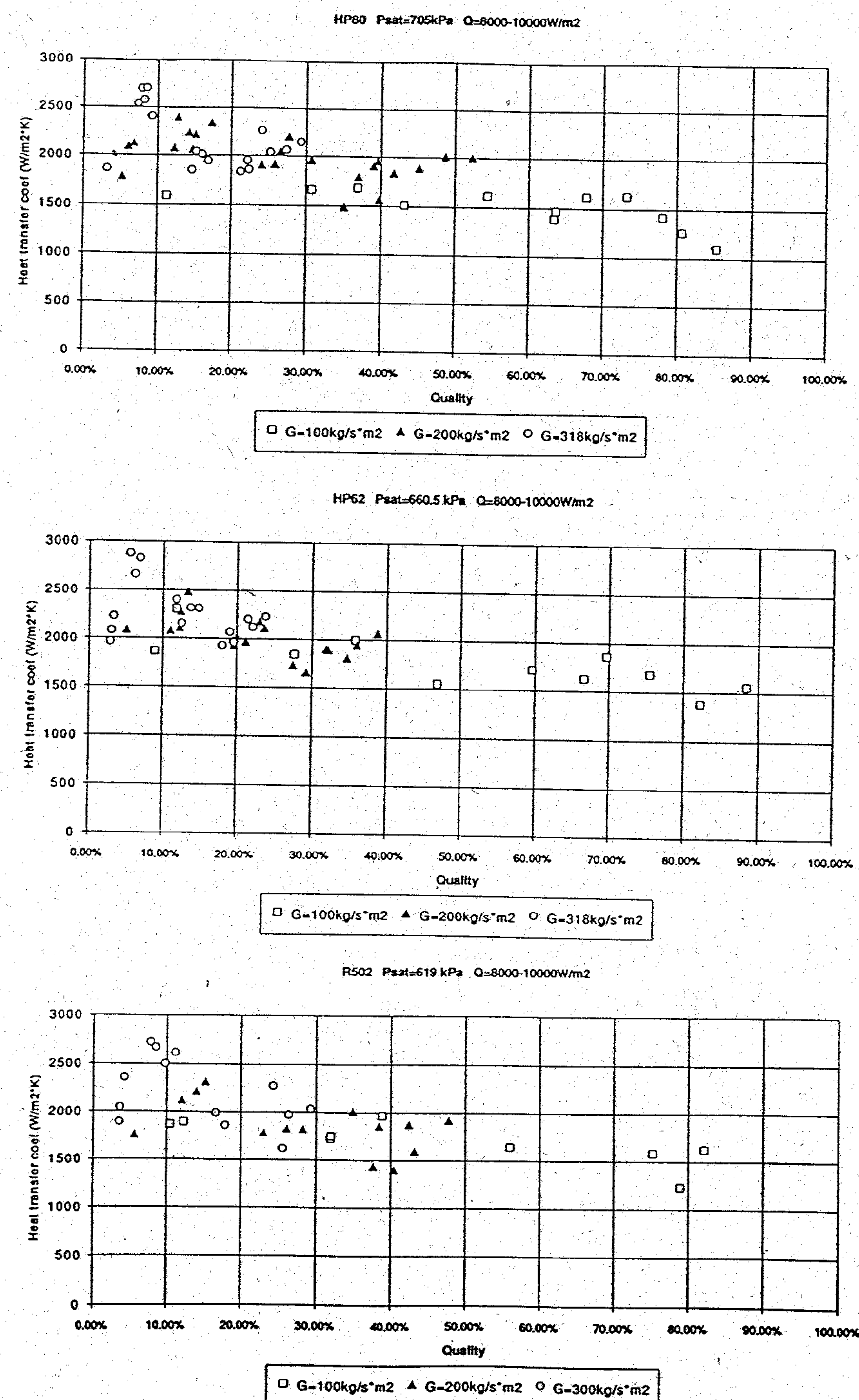

Figure 3. Test results at three mass velocities.

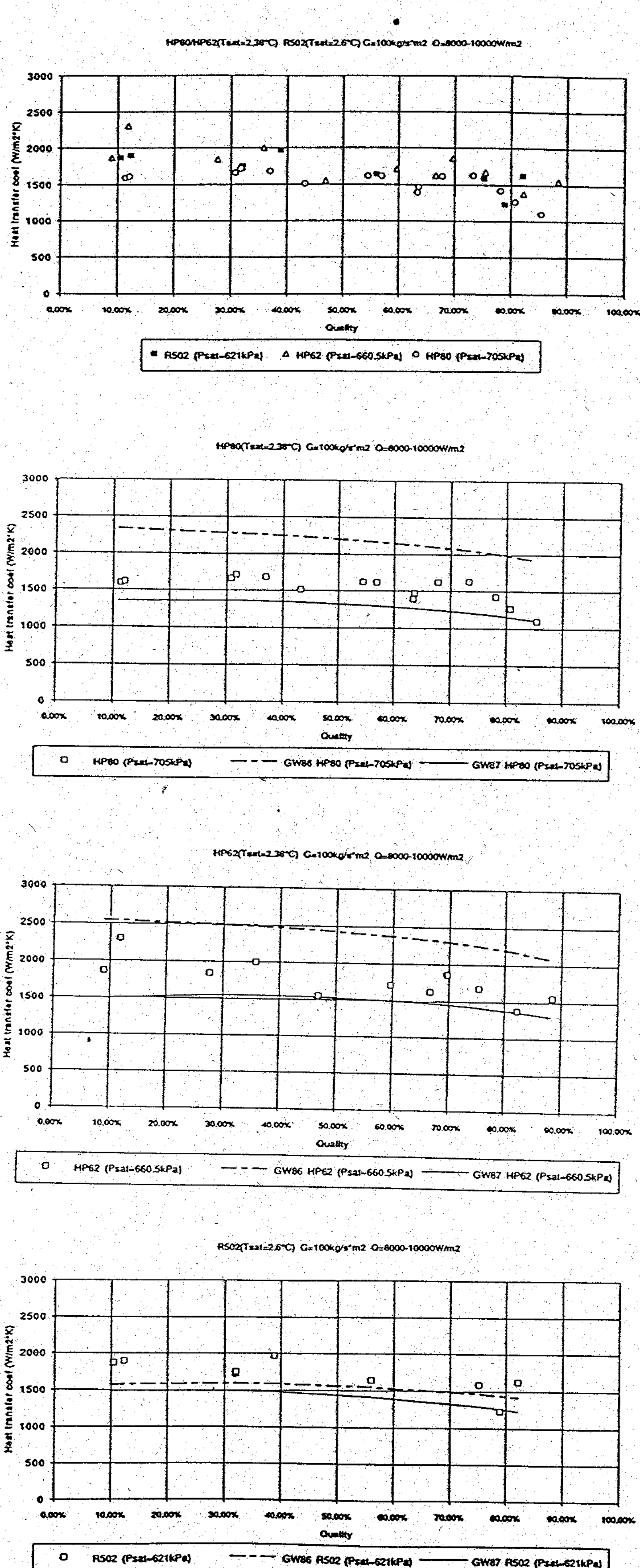

Figure 4. Comparison of performances at low mass flux.

Figure 5 shows a similar comparison at the same heat flux range for mass velocities of 300-318 kg/m^2 s. The GW87 correlation including F_c accurately predicts the HP80 data except for the maximum while the GW86 correlation predicts the data in the maximum well and overpredicts the rest of the data. The comparisons to the HP62 and R-502 data depict the same trends as HP80. The average coefficients for the experimental data in Figure 5 are 2168, 2275 and 2200 W/m^2 K for HP80, HP62 and R-502, respectively; the average values for HP80 and HP62 are -1% and +3% relative to the value for R-502, respectively, for these test conditions.

Table 1 shows the comparison of all the test data to the two Gungor-Winterton correlations corrected for mixture boiling. The GW87 method is consistently more accurate than the GW86 method for HP80, HP62 and R-502, respectively. For all 1011 data points the standard deviation for GW87 is 21.19% while for GW86 it is 27.77%. Work is continuing in this area and the data will also be compared to other correlations.

Table 1. Comparison to Gungor-Winterton correlations.

Fluid	T_{bub} (°C)	G (kg/m^2s)	q (W/m^2)	x (%)	Data Points	% Standard Deviation GW86	GW87
HP80:	-1.3	317	4503-20,836	2-55	67	28.35	21.76
	2.4	102	3253-16,765	4-90	72	34.35	21.38
	2.4	200	3278-22,897	2-88	108	46.52	37.65
	2.4	318	3414-22,179	1-61	94	24.63	19.30
	10.2	303	2506-28,597	3-84	102	25.63	20.93
HP62:	-1.3	320	5323-21,777	2-55	65	26.37	20.97
	2.4	102	3418-10,345	4-88	59	26.82	20.00
	2.4	200	4483-21,104	3-91	82	19.31	13.62
	2.4	318	4577-30,551	2-77	102	20.24	14.98
	10.2	300	4372-30,573	2-89	93	19.89	13.95
R-502:	2.6	100	4252-11,161	7-94	36	23.27	19.41
	2.5	200	4349-23,197	3-99	49	37.73	29.82
	2.4	300	4257-27,720	2-80	82	27.64	21.59
Total (Mean Error All Points):					1011	27.77	21.19

CONCLUSIONS

The boiling coefficients for HP62 are slightly larger than those for R-502 while those for HP80 are slightly smaller. The heat transfer performances of the three fluids are accurately predicted by an existing flow boiling correlation modified here to include a nucleate boiling mixture equation.

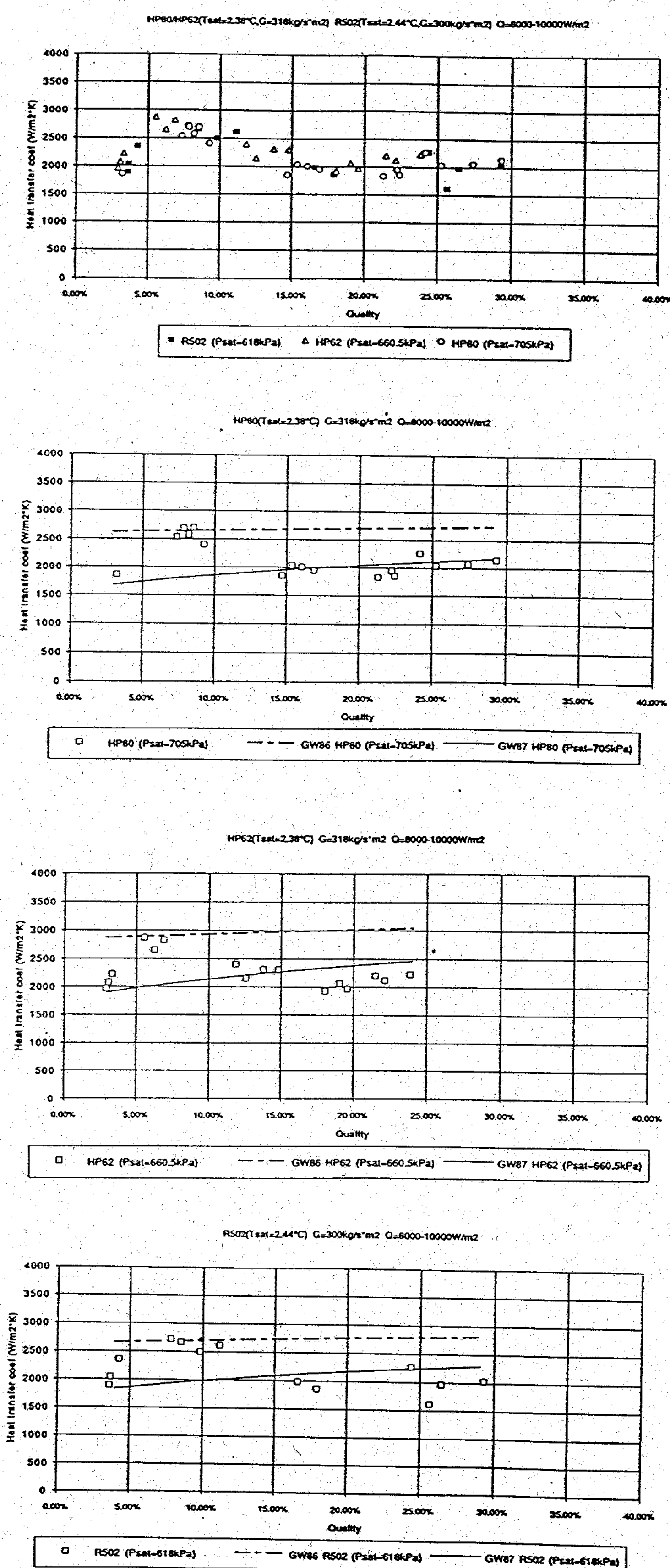

Figure 5. Comparison of performances at high mass flux.

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NOMENCLATURE

$(c_p)_L$	Liquid specific heat of mixture (J/kg K)
$(c_p)_V$	Vapor specific heat of mixture (J/kg K)
E	Two-phase convection multiplier in Eq. (8)
E_o	Two-phase convection multiplier in Eq. (13)
F_c	Mixture boiling correction factor
G	Mass velocity (kg/m^2 s)
H	Enthalpy (kJ/kg)
h_{LV}	Latent heat of vaporization (J/kg)
q	Heat flux (W/m^2)
S	Boiling suppression factor
ΔT_{bp}	Boiling range of mixture (= $T_{dew}-T_{bub}$) (°C)
T_{bub}	Bubble point temperature (°C)
T_{dew}	Dew point temperature (°C)
T_{sat}	Saturation temperature (°C)
T_w	Wall temperature (°C)
x	Vapor quality
α	Local boiling heat transfer coefficient (W/m^2 K)
α_L	Liquid heat transfer coefficient (W/m^2 K)
β_L	Liquid phase mass transfer coefficient (m/s)

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