PERFORMANCE OF PLATE HEAT EXCHANGERS UNDER SINGLE AND TWO-PHASE FLOW CONDITIONS - EXPERIMENTAL RESULTS

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SUMMARY This paper outlines the procedures and methods adopted and the results obtained from experimental research conducted to evaluate the performance of plate heat exchangers. The experiment was conducted in the Thermodynamics Laboratory of the School of Engineering at Auckland University. A Steam Generator in the Laboratory was used to supply hot fluid while the mains water supply was used as the cooling load. Tests were conducted at single and two-phase flow conditions. Measurement of terminal temperatures, pressure drops and flow rate on both the cold and hot fluid streams were made to evaluate the hydraulic and heat transfer performance of the exchangers. Correlations reported in literature for calculating pressure drop and heat transfer coefficient were evaluated and compared with the experimental results. The Lockhart-Martinelli method gave a good approximation for prediction of pressure drop for the condensing steam flow. Deviation of up to 15% between the measured and calculated pressure drop was obtained while the corresponding deviation for the heat transfer was about 20%.

1. Background

The plate heat exchanger (PHE), also known as plate and frame heat exchanger, was first developed in its present form in the early 1900. It was introduced in the 1930s to meet the hygienic demands of the dairy industry and it has been in use since this time. Due to lack of basic heat transfer and hydraulic design data and lack of understanding of the basic mechanisms involved, the application of the PHE has been restricted to certain fields of application. Until World War II the operating conditions were generally limited to pressures of up to 50 psia (3.5 bara) and temperatures below atmospheric boiling point. Rapid changes took place in the post war period. The development of continuous sterilisation of liquids which required temperatures as high as 150°C required rubber gaskets capable of withstanding these conditions. The need to retain high volumes of gases in liquids as in beer and mineral water production necessitated pressures as high as 20 bar. These two set the trend in further developments of PHE's. Currently rubber gaskets capable of withstanding temperatures of up to 170°C have been developed and the plates can operate under pressures of up to 20 bar applied as a differential load between the two sides of the plates.

The plate heat exchanger posses many unique characteristics of significance to various industries. The high heat transfer rate has the advantage of less heat transfer surface required for a specific duty and consequently less cost of material. Due to this feature the PHE is compact and light weight which has the advantage of less storage volume required and ease of transportation. High effectiveness and low temperature approach enable high recovery of heat from low grade heat sources. In most applications the PHE is capable of 90% heat recovery where as only 50% recuperation is economically feasible with tubular units (Raju and Chand, 1980).

One specific example is in the cooling system of power plants where large amounts of thermal energy is rejected to the surrounding. The use of efficient heat exchanger reduces the amount of fresh water to be pumped in order to absorb the heat. This advantage of PHE's has been recognised by the nuclear industry in the UK, where most of the nuclear power stations use PHE's in the secondary cooling circuit (Marriot, 1971).

The ability of plate heat exchangers to withstand fouling compared to other heat exchangers also makes them suitable for fouling fluids. Special design features which enable ease of opening also makes the PHE suitable for hygienic services demanding more frequent cleaning.

Flexibility of design is unique to the PHE both in the initial design and after installation. In the initial design, the basic size, total number and arrangement of the plates can be selected to meet the required duty. An existing PHE can very easily be extended or modified to suit an increased or reduced duty.

Limitations of plate heat exchangers include their low operating temperature and pressure (150°C and 20 bar). Operating with large volumes of low pressure vapour or gas is not economical. The upper limit for the number of plate packs is generally accepted as 400. Plate packs in excess of this limit will result in excessive frictional losses along the entry ports. Due to this reason the biggest PHE available is reported to have 650 m^2 of heat transfer surface.

Thus some of the trends in PHE development are towards larger capacity, higher operating temperatures and pressures, the ability to handle low pressure vapour (gas) and multi-phase fluids. The data available on the performance of PHE's for two-phase and steam applications are very limited. Therefore results of such experimental data will help in solving some of the uncertainties during application of the units under different situations, such as two-phase conditions.

2. Objectives and Scope of work

The primary objectives of this work were to investigate the heat transfer and hydraulic design features of a plate heat exchanger in single phase and two-phase flow conditions. Water, steam and two-phase (steam water mixture) were used as the test fluids. With these objectives in mind a laboratory scale heat exchanger test rig was developed and performance tests were conducted. In order to achieve this the following specific tasks were carried out.

- a comprehensive test programme was designed in order to investigate the performance of Plate Heat Exchangers.
- a laboratory scale test facility was built, instrumented and commissioned in order to carry out the performance test.
- performance tests were conducted at different loads and at varying fluid conditions and comparison between experimental and analytical results was made.

3. Rig Set up

A portable Heat Exchanger rig was built in a laboratory (Figure 1). The rig is composed of two type PO1 Alfa-Laval plate heat exchangers, whose plate dimensions and arrangement is given below. Heat is supplied to the rig in the form of steam, produced from a steam generator in the laboratory. The heat load is applied to the first heat exchanger, in the form of steam or two-phase fluid, and cooling water is passed through the second heat exchanger. Secondary water, circulating in a loop between the two heat exchangers, absorbs the heat at the hot end and dissipates it at the cold end. A circulation pump is used to circulate the secondary water between the two heat exchangers.

A mixing vessel, which was designed and fabricated in the University workshop, was used to produce two-phase fluid for testing. Inlet enthalpy of the hot fluid is regulated and adjusted to the required value by humidifying the incoming steam in the mixing vessel. A condensate trap is connected to the mixing vessel which is operated when dry steam is to be used for testing. Sight glass and non-return-valve assembly is used both at inlet and outlet to the mixing vessel for monitoring the flow in and out of the vessel.

Plate dimensions and arrangement

- Plate Characteristics:

Plate type	P 01 (Alfa-Laval)
Plate Effective Area, A	0.032m^2
Plate thickness, x	0.6 mm
Effective Plate width, w	0.1m
Effective Length of flow, L	0.32m
Channel Depth, b	2.4 mm
Corrugation Type	Chevron β =120°

- Material of Construction:

Plates	Stainless Steel 316
Gasket	Resin cured Butyl

- Flow Arrangement:

Total Number of Plates	16
Thermal Plates	14
Total Heat Transfer Area	0.45m^2
Number of Flow Channels	15
Number of Flow Passes	1

Measurement of flow, pressure drop and terminal temperatures at the heat exchangers for all fluid streams is made. The temperature and pressure sensors were connected to a data logger to record data during testing. All the temperature sensors were Resistance Temperature Detectors (RTD). Two types of pressure measuring devices were used. The Inverted U Tube manometer was used for measuring differential pressure while Pressure Transmitters with electrical signal output were used for both differential and static pressure measurement. Three types of flow measuring devices were used;

- Liquid Vortex flow meter with current signal output for measuring the loop flow.
- Annubars for measuring the flow of both the cooling water and the water injected to steam during two-phase flow tests.
- Rotameter for measuring the flow of condensate coming out of the hot heat exchanger.

No specific criterion existed for the choice of the flow meters, this was based on availability, cost, simplicity of construction and accuracy. The schematic of the experimental set up is given in figure 1.

3.1 Development of Test rig

One of the important consideration during the initial stage of the rig development was to make the rig as compact as possible and to avoid long piping. The rig was positioned at a reasonable proximity from the steam generator and the water supply line. This was critical as there was limited space in the Laboratory.

3.2 Equipment Testing

Testing of parts was important before installing them on the rig, especially those parts which operated under pressure. The steam-water mixing vessel was one such part which was designed and manufactured for operation at a pressure of up to 5 bara and a temperature of 150°C. After the vessel was manufactured a hydrostatic pressure test was made to ensure the safety of the vessel during the experiments. The first pressure test indicated leakage through flanges. The vessel was tested for a second time after the flanges were resurfaced and thicker gaskets were fitted. During the second test the vessel contained a pressure of 11 bara without any leak.

3.3 Commissioning

After calibrating all the instruments and installing them on to the test rig preliminary tests were carried out to check the functioning of parts. During the first such test steam leaked through pipe joints. The differential pressure from the Annubar on the steam line was fluctuating and the reading was not in agreement to what was expected. From mass and energy balance at the heat exchangers it was found that the steam generator produced unsaturated (wet) steam. Repeated tests gave a dryness of about 60%. It was later found that the steam generator was operating with a low supply of natural gas. The gas delivered to the steam generator is 16.5 m^3/hr which is about 33% lower than that needed to produce 250 kg/hr of saturated steam at 4.5 bar abs.

Since it was evident that two-phase fluid was flowing in the pipes, instead of dry steam, the Annubar could not be used for measuring the flow rate. This introduced some modifications to the original design for flow measurement of the hot fluid. A Rotameter was fitted to the outlet of the hot fluid leaving the heat exchanger, in the form of condensate, for measuring its flow.

3.4 Calibration and Error Analysis

The accuracy of the instruments for measuring temperature, pressure and flow rate was checked by conducting calibration tests for each instrument. The observed deviation between the actual and the indicated instrument reading is used to assess the accuracy of the measurement technique and to estimate overall error in the measurement procedure. The relative or percentage error (E) between the measurement instrument and that of a calibration instrument is calculated using the equation shown below (Tse, 1989):

$$E = \frac{\Delta T}{T} \times 100\ \%$$

Where AT is the difference between the two readings and T is the actual instrument reading.

The accuracy of measurement instruments used for the experiment, for a steady condition, is given below - note however that higher fluctuations have been observed due to variations in the cooling water pressure and due to oscillation of the Rotameter float.

Instrument	Accuracy
Temperature measurement-RTD	± 1.5%
Differential Pressure - Pressure Transducer	± 0.5%
Differential Pressure - U-Tube manometer	± 1%
Mass flow - Vortex flow meter	± 2.0%
Mass flow - Annubar	± 5%
Mass flow - Rotameter	± 5.0%

Since the heat transfer is given as a product of the fluid mass flow and the change in the fluid enthalpy (which is directly proportional to the temperature), the error in overall heat transfer calculation can be obtained from the error in the mass flow measurement and enthalpy (temperature) measurements.

The accuracy of the calculated overall heat transfer coefficient is given as +13 % for the hot fluid stream, ±3.5 % for the secondary fluid stream and ±15 % for the cold fluid stream.

4. Results and Discussion

4.1 Pressure Drop

The result of the experimental tests gave a power relationship between the fluid flow rate and the pressure drop (Figures 2- 4). Comparison of the measured pressure drop with calculated ones using pressure drop equations recommended for PHE's was made. In the calculation of the pressure drop the friction factor was determined from the Reynolds number according to the flow regime i.e. laminar, transition or turbulent. The measured flow rate was used to calculate the velocity and the Reynolds number.

An average deviation of about 15% between the actual (measured) pressure drop and the calculated pressure drop was obtained for the case of test with water. The highest of the deviations were obtained at the minimum flows. This was mainly due to the variation in the water flow caused by the fluctuation of the pressure in the mains water system. The deviation between the measured and calculated pressure drop for test with two-phase fluid was 7.3% while for steam a value of 9% was obtained. The high deviation of 15% for the liquid flow is mainly due to the inaccuracy of the liquid flow measurement (12%) as a result of fluctuations in the mains pressure.

Comparison of this test with previous works indicates that the deviations obtained are acceptable. Caicula and Rudy (1983) reported a deviation of 97% between measured and calculated pressure drop for test with water flow.

The measured pressure drop for *dry* ***steam*** at inlet was about twice that of wet steam (*66%* dryness) for the same mass flow. This indicates that a large portion of the pressure drop in a condensing steam takes place in the initial stage of condensation.

Curve fitting of the measured pressure drop plotted against mass flux of the test fluid gave a power relationship of the general form:

$\Delta P = a\, G^b$

Where:
ΔP = the pressure drop, Pa
G = the mass flux, kg/m^2s
a and b are constants.

The constants a and b for this test were found to be:

a = 1.301 b = 1.61.....for water flow
a = 27.77 b = 2.128....for steam flow
a = 8.417 b = 2.201...for two-phase flow

The above correlation can be used to estimate the pressure drop for a given flow rate per unit area. However since the pressure measurements for steam and two-phase flow tests were made at a fixed condition the above relationship may not be valid for other pressures, due to the dependence of the fluid density on the pressure. For the case of water the change of fluid density due to change in pressure or temperature is not significant and hence the relationship could be valid for other flow conditions.

4.2 Heat Transfer

Actual overall heat transfer coefficients were obtained by applying heat balance on both sides of the fluid streams. In theory the rate of heat transfer on both sides should be equal and therefore either of the two can be used to determine the overall heat transfer coefficient. However the experimental test indicated a discrepancy between the measured heat transfer rates on both sides of the heat exchanger. The major reason for this is heat loss from the heat exchanger surface. The other reason is the accuracy of the measurement of the temperatures and flows used in the heat balance equations. The inlet condition of the hot fluid (enthalpy) which was estimated from the measured pressure and temperature at the boiler is also very critical in the heat balance equation. For this test all measured heat transfer coefficients were based on the heat balance obtained from the secondary fluid stream. Both the flow and temperature measurements on the secondary fluid stream are fairly accurate.

Comparison of the measured values with those obtained using equations for calculating heat transfer coefficients in plate heat exchangers is made (Figures 5-7). The calculation of the overall heat transfer coefficient was based on the determination of the film coefficient of the fluids on either side of the heat exchanger. The determination of the film coefficient for a single phase fluid was made by calculating the fluid Nusselt number and applying the conventional relationship for determining the film coefficient. However for the case where the fluid property changed along the length of the heat exchanger, due to condensation of the vapour phase in the fluid stream, a two-phase film coefficient was estimated. For such a case the procedure reported'by Cooper **(1974)** was adopted.

Caciula and Rudy **(1983)** reported on the prediction of heat transfer and pressure drop performance of PHE's and compared the experimental result obtained from an operating heat exchanger to the values obtained from empirical correlations. This study also made a sensitivity analysis of the result by comparing the measured heat transfer coefficient and pressure drop with the calculated ones. The average percentage deviation between measure and calculated heat transfer coefficient ranged from 19.6% to 55.5%. This test was made for liquid flows on both sides of the exchanger.

In our experiment the percentage deviation for overall heat transfer coefficient ranged from 31% to 35% for liquid flow on both sides of the exchanger. The good agreement between the measured and calculated results suggests that overall heat transfer coefficients can be predicted with reasonable accuracy using correlations. Curve fitting of the measured data was made. Both power and logarithmic functions were the most appropriate to fit the test data. The power relationship gave a correlation for the overall heat transfer coefficient with the mass flux similar to that of pressure given above ($U = a\, G^b$). For which the coefficients are:

a = 1954.7 b = 0.1297....for water flow
a = 194.7 b = 1.272......for steam flow
a = 289.5 b = 0.878.....for two-phase flow

Unlike pressure drop the overall heat transfer coefficient is dependent on various parameter; of which mass flow, fluid properties like viscosity and density are the major ones. The correlation shown above gives an indication of the variation of the heat transfer coefficient with changes in the mass flow. Therefore it can not be used to predict the heat transfer coefficient for flows which are not similar to the test conditions.

5. Conclusions

The heat transfer and hydraulic performance of plate heat exchanger was determined experimentally. A portable heat exchanger test rig was constructed and instrumented in a Laboratory to conduct the experimental work. Tests were conducted using both single phase and two-phase fluids in counter current flow conditions. Comparison of the experimentally obtained results with those obtained by analytical methods was made. The general conclusion to be made from this work are:

The pressure drop plotted against the mass flow, follows a power relationship while the overall heat transfer coefficient follows a logarithmic relationship.

Prediction of pressure drop in a fully developed turbulent flow in PHE's can be made for single phase flows using the standard pressure drop equations used for PHE's. The error or deviation obtained between measured and calculated pressure drop for liquid flow ranged between 3% to 20% while error analysis indicates that the accuracy of the measured pressure drop is within 12%.

The Lockhart-Martinelli equation for predicting two-phase flow pressure drop gave a good estimation of the pressure drop in a condensing steam and two-phase flows. The absolute value of the deviation between measured and calculated pressure drop for condensing flow varied between 4% and 14%.

The measured pressure drop in a condensing type flow for dry steam, compared with wet steam (60% dryness) of equivalent mass flow, is about twice.

Prediction of overall heat transfer coefficient, using correlations, both for single phase and two-phase flow conditions is possible. The percentage deviation between measured and calculated overall heat transfer coefficient was between 20% to 55%.

Acknowledgements
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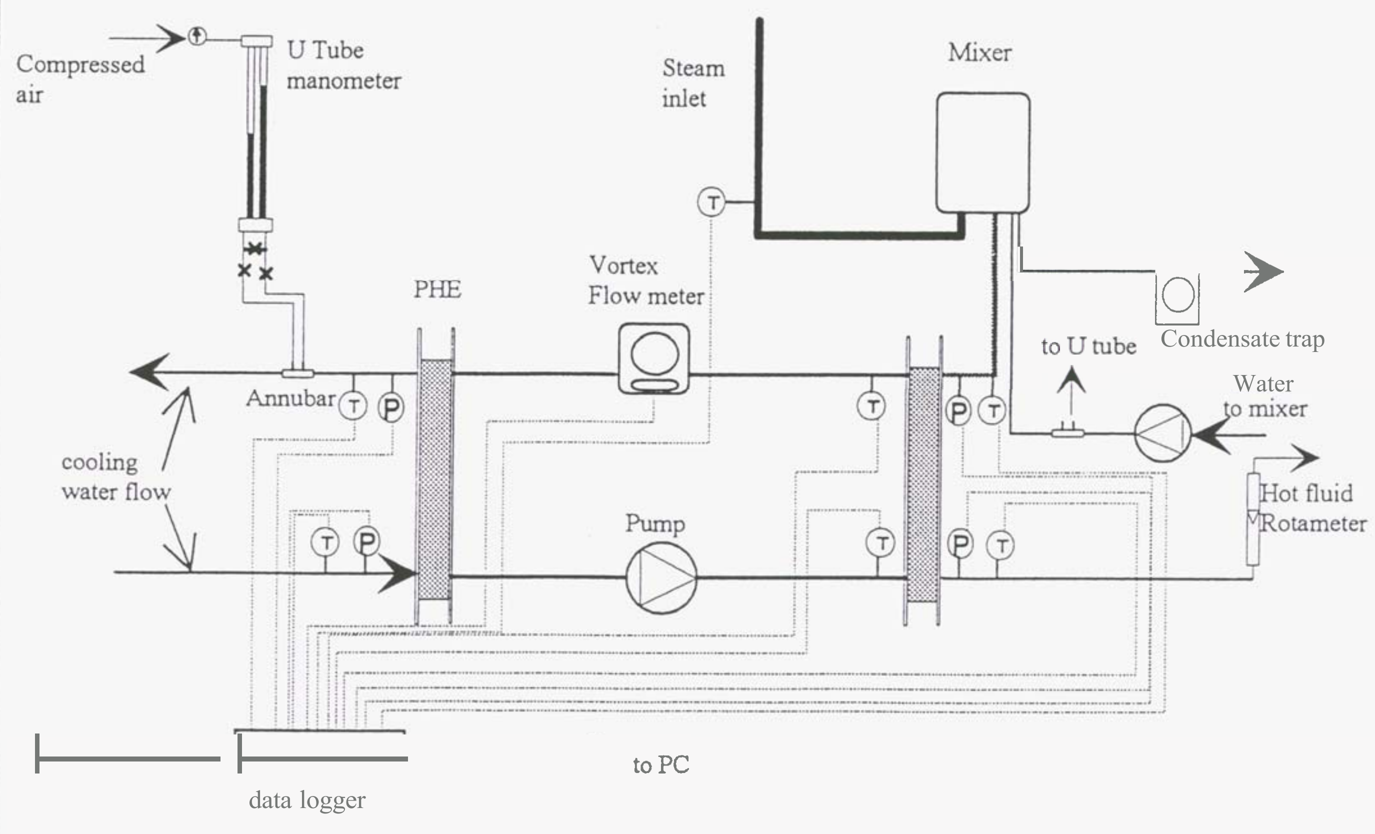

Figure 1 Schematic drawing of the Test rig

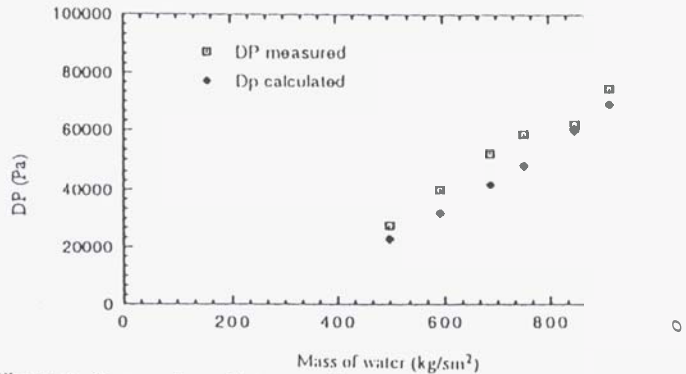

Figure 2 Comparison Between Measured and Calculated Pressure Drop as a Function of the Mass Flow of the Test Fluid (water).

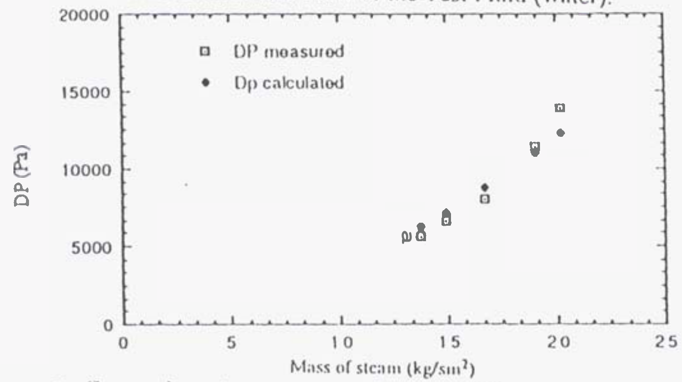

Figure 3 Comparison Between Measured and Calculated Pressure Drop as a function of the Mass Flow of the Test Fluid (steam).

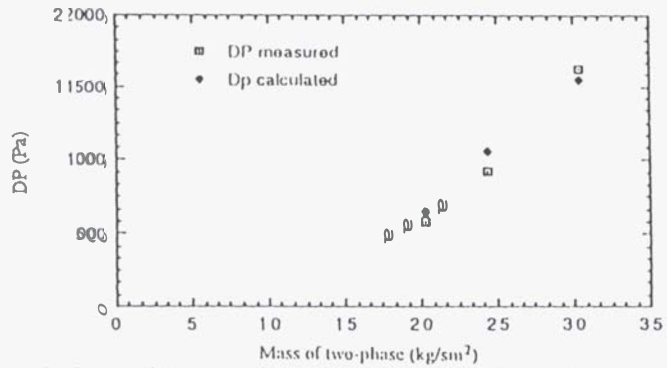

Figure 4 Comparison Between Measured and Calculated Pressure Drops as a Function of the Mass Flow of the Test Fluid (two-phase).

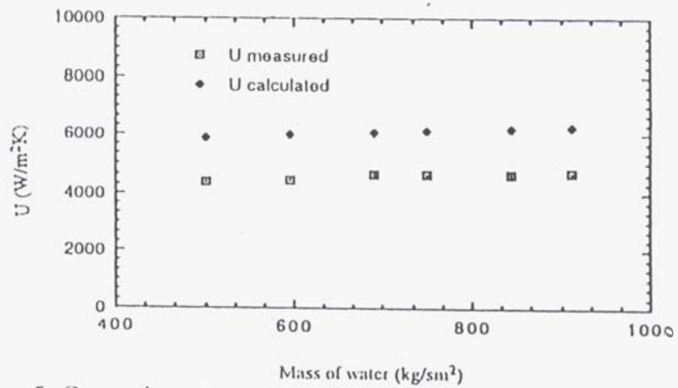

Figure 5 Comparison Between Measured and Calculated Overall Heat Transfer Coefficient as a Function of the Mass Flow of the Test Fluid (water).

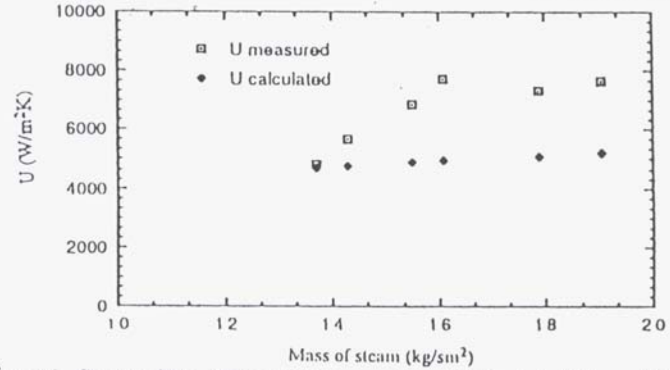

Figure6 Comparison Between Measured and Calculated Overall Heat Transfer Coefficient as a Function of the Mass Flow of the Test Fluid (steam).

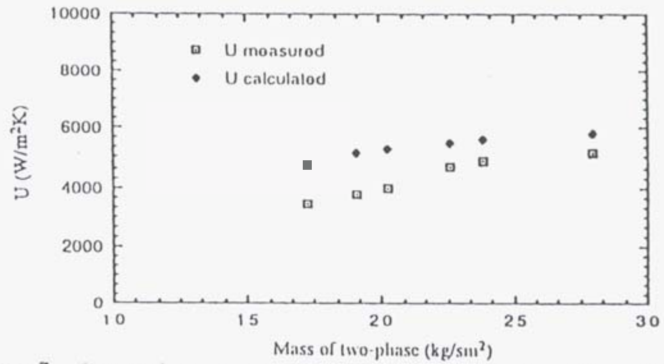

Figure 7 Comparison Between Measured and Calculated Overall Heat Transfer Coefficient as a function of the Mass flow of the Test fluid (two-phase).