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EHD ENHANCEMENT OF POOL AND IN-TUBE BOILING OF ALTERNATE REFRIGERANTS

Final Report

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NOMENCLATURE

h = heat transfer coefficient
Q = heat transferred to the test section
T_{wi} = inside test tube wall temperature
T_{wo} = outside test tube wall temperature
T_{sat} = saturation temperature of test refrigerant
A = heat transfer surface area
m_w = mass flow rate of hot water
c_w = specific heat of water
δx = test tube wall thickness
k = thermal conductivity of test section tube material
Nu = Nusselt Number in the presence of electric field
Nu_0 = Nusselt Number at zero electric field (base case)
h_0 = heat transfer coefficient (base case)
P = test section pressure (in-tube boiling)
T_0 = refrigerant temperature before preheater (in-tube boiling)
e_0 = specific enthalpy of refrigerant before preheater (in-tube boiling)
e_1 = specific enthalpy of refrigerant after preheater (in-tube boiling)
Q_{ph} = heat transferred to refrigerant by preheater (in-tube boiling)
Q_{ts} = heat transferred to refrigerant by hot water (in-tube boiling)
m_r = refrigerant mass flow rate (in-tube boiling)
X_1 = refrigerant quality after preheater and before test section (in-tube boiling)

1. ABSTRACT

The Electrohydrodynamic (EHD) is an active heat transfer augmentation technique which utilizes the effect of secondary motions generated through the application of an electrostatic potential to a dielectric fluid. The net result is better momentum and heat transfer between the fluid and the heat transfer wall through destabilization of the thermal boundary layer and better mixing of the fluid adjacent to the heat transfer surface. EHD enhancement of refrigerant /refrigerant oil mixtures heat transfer using the Electrohydrodynamic (EHD) technique is the subject of a three-year experimental investigation in a project funded by the U. S. Department of Energy, effective June 1, 1993. For the interim period between November 1992 and June 1993 when the DOE funds became available, the AirConditioning and Refrigeration Technology Institute (ARTI) provided partial funding for our EHD research program with the aim of accomplishing three major tasks: (1) conduct a comprehensive search of the literature on EHD-enhanced, in-tube and external boiling heat transfer enhancement of alternate refrigerants; (2) Design, fabricate, and instrument an in-tube, EHD-enhanced boiling/ condensation test rig and perform preliminary testing of the setup; (3) conduct experiments and document new findings on EHD-enhanced external boiling of alternate refrigerants/refrigerant mixtures in an existing pool boiling test rig apparatus. Description of the takes performed and discussion of the results are documented in this report.

2. WORK SCOPE

The work reported here was a six-month effort in which the major task was to design, fabricate, and perform preliminary testing of a test rig suitable for testing of EHD-enhanced in-tube boiling/condensation heat transfer of alternate refrigerants/refrigerant oil mixtures. The EHD technique has demonstrated significant potential for heat transfer enhancement of two-phase flows including the boiling/condensation of CFCs and alternate refrigerants. A detailed review of the literature is documented in this report. As cited there, the previous work on in-tube, EHD-enhanced boiling/condensation is limited to a study in Japan for a 97% R-123 and 3% R-134a mixture [Yabe, 1991] for flow boiling in a straight-tube . The aim of the current research was to design and fabricate a forced convection boiling/condensation test rig such that both low and high pressure alternate refrigerants can be tested. The objective was to obtain both average and local heat transfer coefficients during evaporation of a pure refrigerant or a mixture of refrigerants and lubricating oil. Before finalizing the test apparatus, the suggested design was faxed to the industrial advisory members to seek their feedback and comments. Almost all the suggestions received from the individual members have been reflected in the final design presented here. The work scope in this project also included conducting additional experiments and documenting new findings on pool boiling enhancement of alternate refrigerants in an existing external boiling test rig capable of testing both low and high pressure refrigerants.

3. MAJOR ACCOMPLISHMENTS

The three major planned tasks for the project were all accomplished. These were: (1) perform a comprehensive literature search on in-tube boiling of refrigerants/refrigerant oil mixtures; (2) design, fabrication and preliminary testing of the in-tube, EHD enhanced boiling/condensation test rig; and (3) perform additional experiments and document the new findings on external boiling of alternate refrigerants/refrigerant oil mixtures. Description of these tasks will now follow.

4. LITERATURE SURVEY

To evaluate the potential of the various augmentation techniques for heat transfer enhancement of alternate refrigerants and for comparison of the results with our base case experiments (zero electric field condition) it was necessary to perform a careful review of the earlier research work. Accordingly, a thorough search of the relevant literature was performed and the results are documented in a tabular form in Appendices A and B for external and in-tube boiling, respectively. The pool boiling search was limited to the EHD-enhanced earlier studies. On the other hand, for the in-tube boiling because the earlier EHD-enhanced work was limited to one paper, the non-EHD papers of relevance to the current project were included in our search.

5. THE EXTERNAL BOILING TEST RIG

5.I The Design Features

The overall schematic design of the test rig is shown in Figs. 1 and 2. A horizontal shell and-tube heat exchanger was used as the EHD enhanced evaporator test section. A 1.87 m (74 in) long stainless steel shell of .2 m (8 in) inside diameter designed to withstand pressures up to 20 bar (300 psi) is used as the test section so that both high and low pressure refrigerants could be tested. While designing the test rig, two objectives were kept in mind. Firstly, the system was to be made as simple as possible by having minimum number of joints and connections so as to avoid the leakage problem at high pressures and, secondly, to make the experimental conditions as close as possible to that of a practical heat exchanger. Therefore, the condenser section was put inside the shell and hot water was used for heating of the test section.

The experimental apparatus consisted of two main sub-loops: a hot water loop which provided heating of the refrigerant and a cold water loop which provided condensation of the refrigerant. The hot water loop included a turbine flowmeter, a pump, an in-tube heater and the "lo-finned" (19 fins/in), .019 m (.75 in) diameter copper test section tube. The in-tube heater, employing a 3.175 mm (1/8 in) diameter and 914 mm (36 in) long stainless steel wire as one pole and a .019 m (.75 in) copper tube as the other pole, provided the heating energy directly to the water. This design reduced the required mass flow rate of the water circulating in the loop and hence the time constant of the hot water loop. This also helped to reduce the thermal losses to the surroundings. The hot water loop was thermally well-insulated so that electrical power input

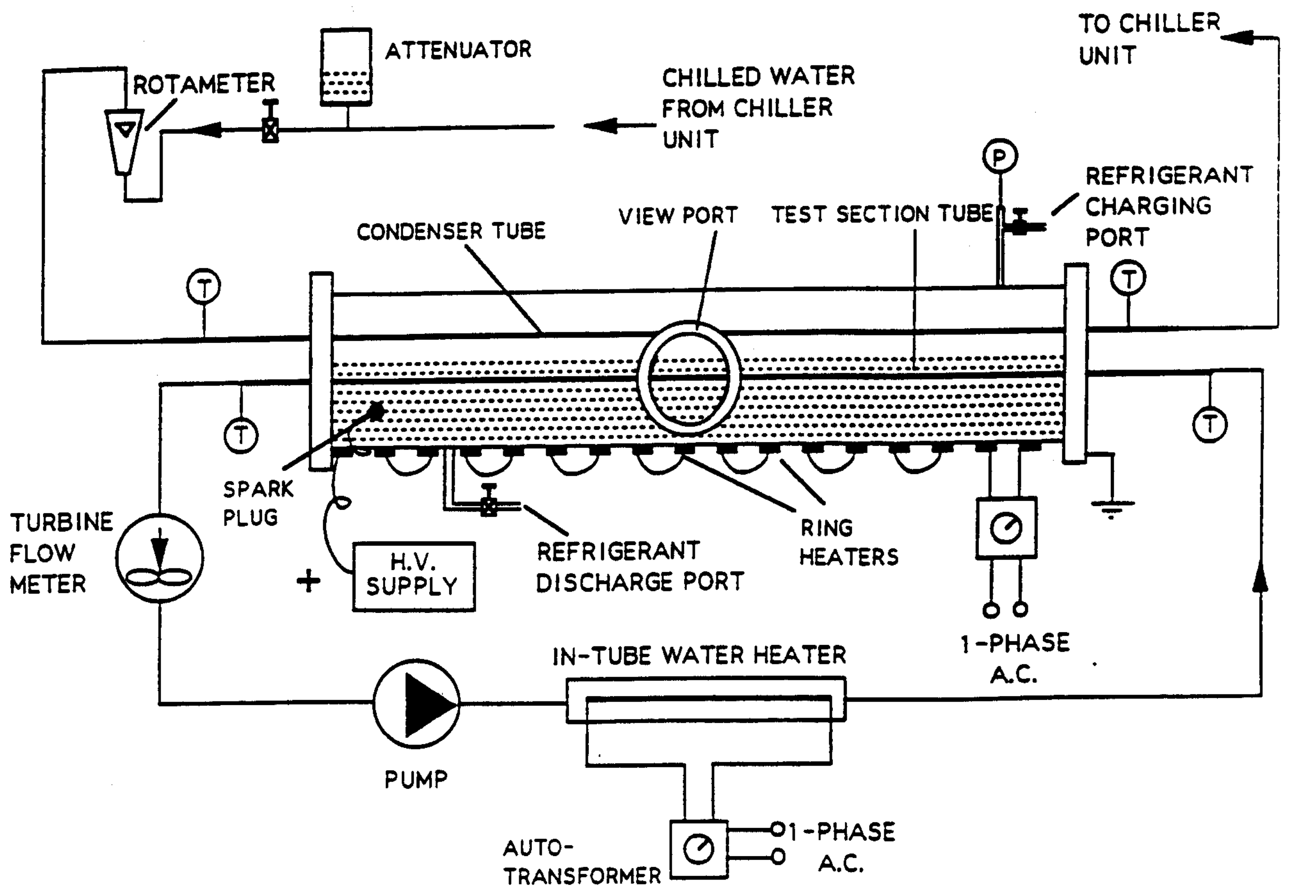

Fig. 1 Schematic of the external boiling test rig

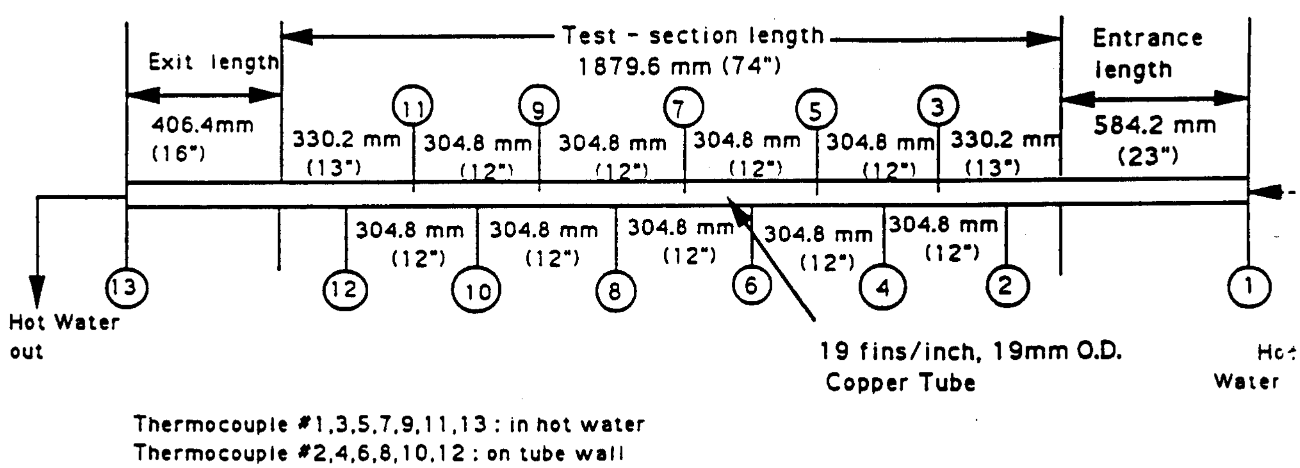

Fig. 2 Layout of thermocouples for the external boiling test section

in the test section.

The cold water loop consisted of a pressure attenuator, a rotameter and an externally enhanced, .019 m (.75 in) diameter copper tube which served as the condenser for the system. Based on earlier studies[1-6], it was determined that the wire mesh electrode was an optimum configuration to use. Accordingly, a stainless wire mesh in the arrangement shown in Fig. 3 was used As seen there, the wire mesh had four mesh grids per linear inch with the wire diameter of 0.8 mm (0.03 in). The wire mesh electrodes were supported concentrically with respect to the experimental tube axis by teflon insulating rings placed at intervals of 0.175 m (7 in). While installing the electrodes on the tube, proper care was taken to ensure a gap of 3 mm (0.12 in) existed between the test section and the electrodes. Positive high voltage was applied to the electrodes through a modified automobile spark plug whereas the shell and tube were grounded.

The temperature measurements were all made using gauge 30 , type "T" thermocouples. The wall temperature of the tube was measured at six axial locations and, to avoid the interaction of thermocouple wires with the high voltage electrodes, all the thermocouple leads were routed through the inside of the copper tube. As shown in Fig. 2, seven thermocouples were placed axially to quantify the variation of hot water temperature as it traveled along the tube. To facilitate degassing, and to ensure that the entire pool was at the saturation temperature, the bottom half of the shell was heated with half ring strip heaters. The pool temperature was measured using two thermocouples immersed in the liquid refrigerant.

There were four main independent parameters that were controlled in the apparatus. These were: the test section heat flux determined by the setting of the variable transformer for in-tube heater, the applied electric field potential which was controlled manually on the high voltage supply; pressure of the pool which was regulated by the cold water flow rate in the condenser, and finally the hot water flow rate which was controlled by adjusting the rotational speed of the pump.

Execution of an experimental run began by turning on the shell ring heaters, the in-tube heater, the hot water pump and the cooling water loop. The high voltage supply was then turned on and adjusted to slightly less than the spark-over voltage. The hot water heater was next set at the highest heat flux by adjusting the voltage applied to the heater. Pressure was maintained in the shell by controlling the cooling water flow rate. A "steady state" condition was defined as when, for the given heater input, all thermocouple signals in the hot water loop along the test section remained unchanged for about 30 minutes. Because of the presence of hysteresis, it was decided to collect the data only in the decreasing heat flux order, for both with and without high voltage, so as to maintain consistency in the collected data.

5.II Data Reduction

The average heat transfer coefficient was calculated using the defining equation, Eq. (1):

$$h = \frac{Q}{A(T_w - T_{sat})} \tag{1}$$

where Q is the rate of heat transfer to the test section tube, determined by the heater current and voltage reading to the in-tube heater. T_w is the average tube wall temperature determined by

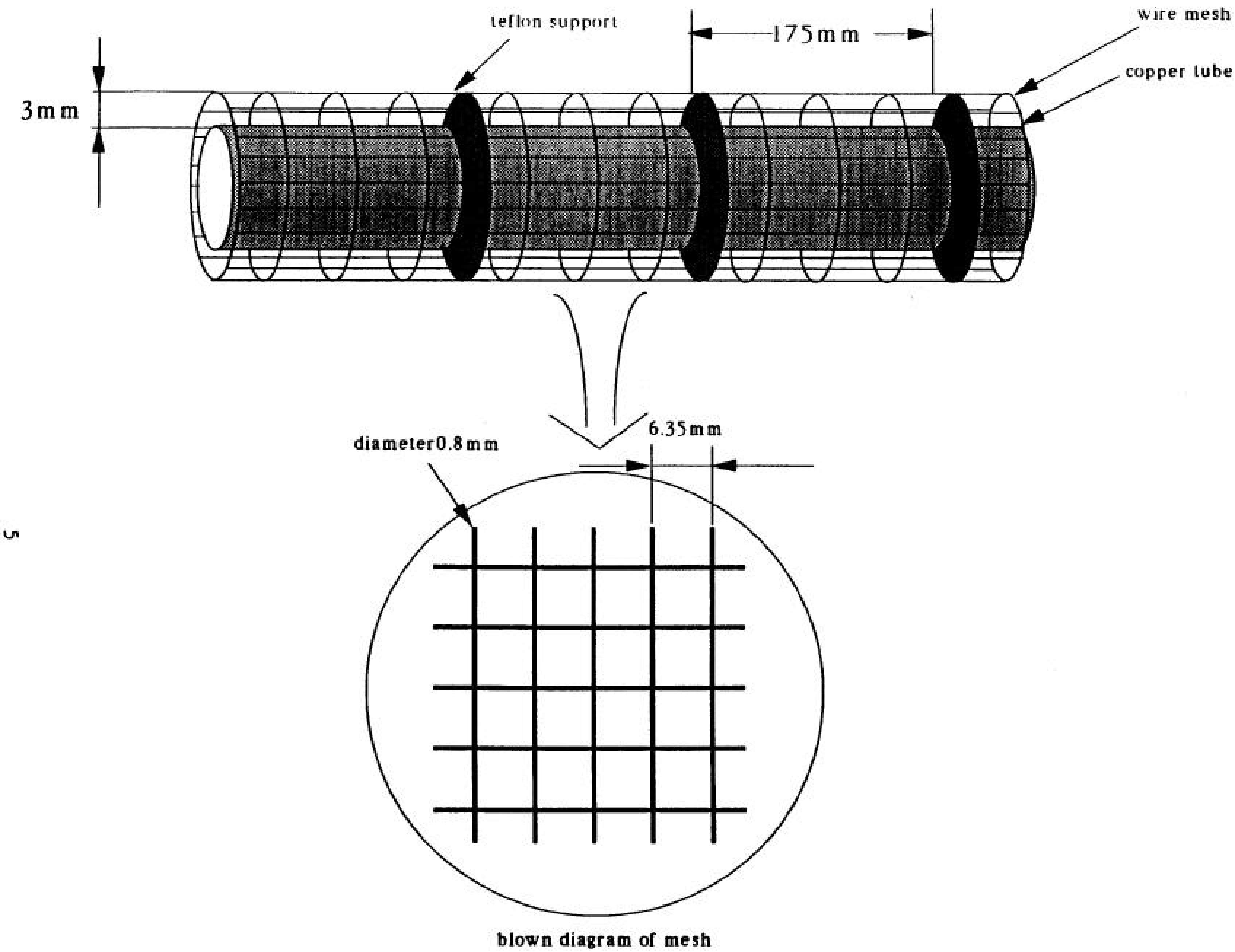

Figure 3: Details of wire mesh electrode surrounding the hot water tube

taking the arithmetic mean of the six thermocouples installed on the test section wall. T_{sat} is the saturation temperature which was taken equal to the liquid pool temperature. Experiments were conducted to obtain the heat transfer coefficient for R-134a and R-123 at 0 kV (the base case) as well as for applied potentials of 5, 10, 15, and 20 kV to parametrize the effect of the applied field potential. Brief presentation of selected results will be given in the following.

Assuming a constant thermal conductivity, the Nusselt ratio reduces to the corresponding ratio of the heat transfer coefficients for the present experiments.

$$\frac{Nu}{Nu_0} = \frac{h}{h_0} \tag{2}$$

5.III Results

The external boiling experiments included tests with both refrigerant R-123 and R-134a. All tests were performed on 19 fins/in manufactured by Wolverine Inc., Decatur, Alabama. For experiments with R-123 the effect of lubricating oil was also studied. The results so far obtained for R-134a do not include the effect of lubricating oil, but this will be addressed in the experiments planned for near future.

Attention is first brought to Figs. 4 and 5 where EHD heat transfer enhancement of R-123 using a mesh type electrode is shown. These results were obtained with a smaller test section that used resistive-heating as the source of heating. Details of this test section are given in [1]. From Figs. 4 and 5 it is seen that with the presence of the EHD effect over eight-fold enhancement in heat transfer with the mesh type electrode can be obtained. Although the results are not shown here, with the straight-wire electrode close to seven-fold enhancements were obtained for the same operating conditions. As expected, the lubricant oil causes a degradation in the magnitude of the heat transfer enhancements. However, at 2% oil concentration still enhancements of over seven-fold for the mesh electrode and over five-fold for the straight wire electrode are achieved.

EHD-enhanced boiling of R-134a on a 19 fins/in tube ("Turbo-A", manufactured by Wolverine Tube, Inc.) similar to that of R-123 is shown in Figs. 6 to 8 for pressures of 590, 680, and 790 kPa, respectively. As seen there, highest enhancements are obtained at the lowest heat flux and lowest operating pressure. For the range of parameters experimented here, the highest enhancement was observed at the pressure of 590 kPa and a super heat of 1.5 °C and field potential of 20 kV. For these conditions the value of h, the heat transfer coefficient, is over 9 times higher than that of a zero field case -- therefore, an enhancement of over 800% for h.

Effects of various parameters in the trends observed in Figs. 4 - 8 can be explained as follows. When the pressure is low, the electric force can agitate the flow more effectively due to the fact that at high pressures, molecules have less freedom to move around and, therefore, electric charges will have a less pronounced effect on promoting the boiling dynamics. With respect to the effect of heat flux, at a higher heat flux the boiling dynamics is high enough and the EHD induced agitation effects will have a less pronounced role. As for the effect of applied electric potential a higher field magnitude implies stronger electrical body forces and, therefore, higher EHD-induced effects in promoting the bubble break-up and increasing the bubble departure speeds which collectively lead to higher heat transfer rates.

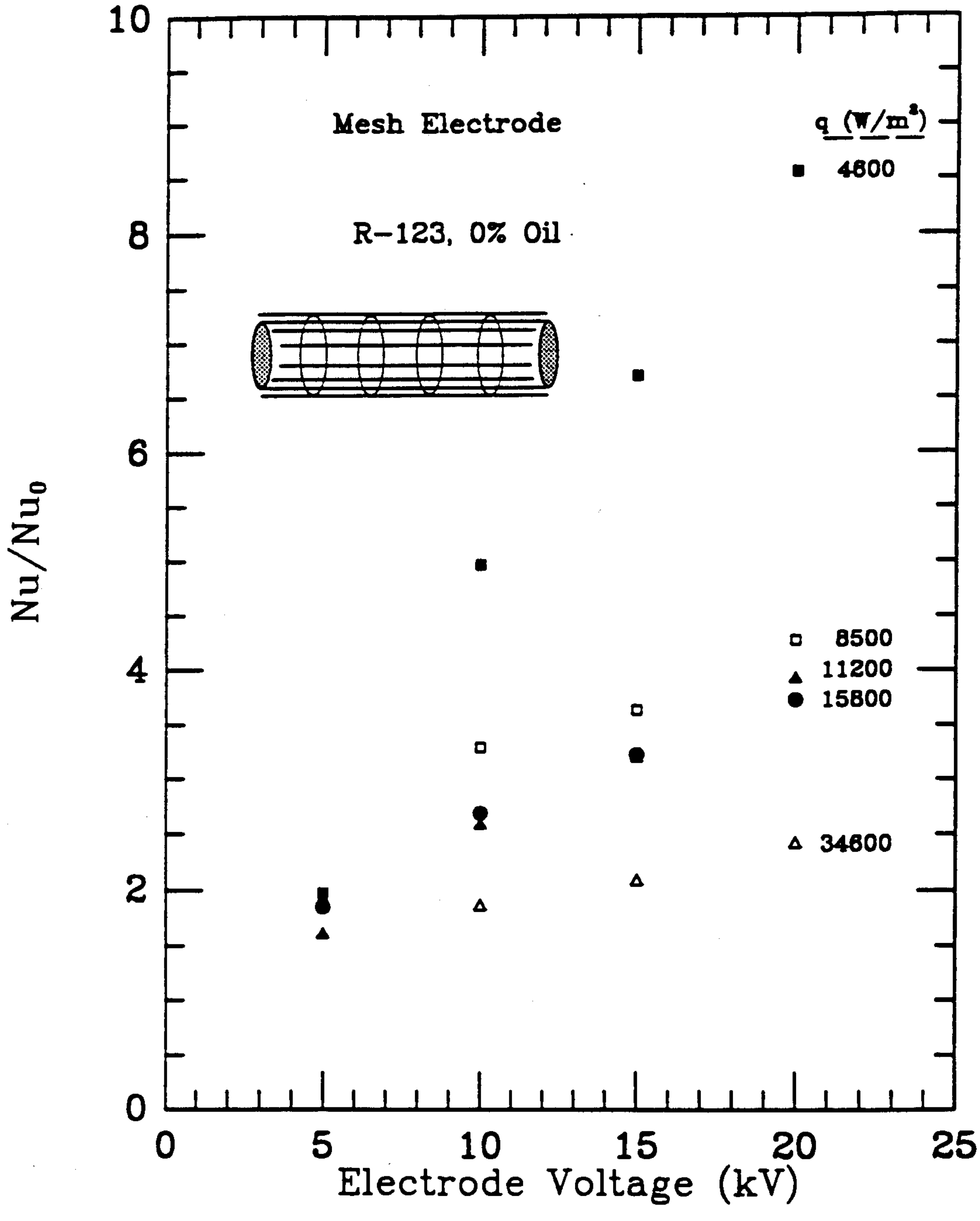

Fig. 4 EHD-enhanced external boiling of R-123 with wire mesh electrode at 0% oil

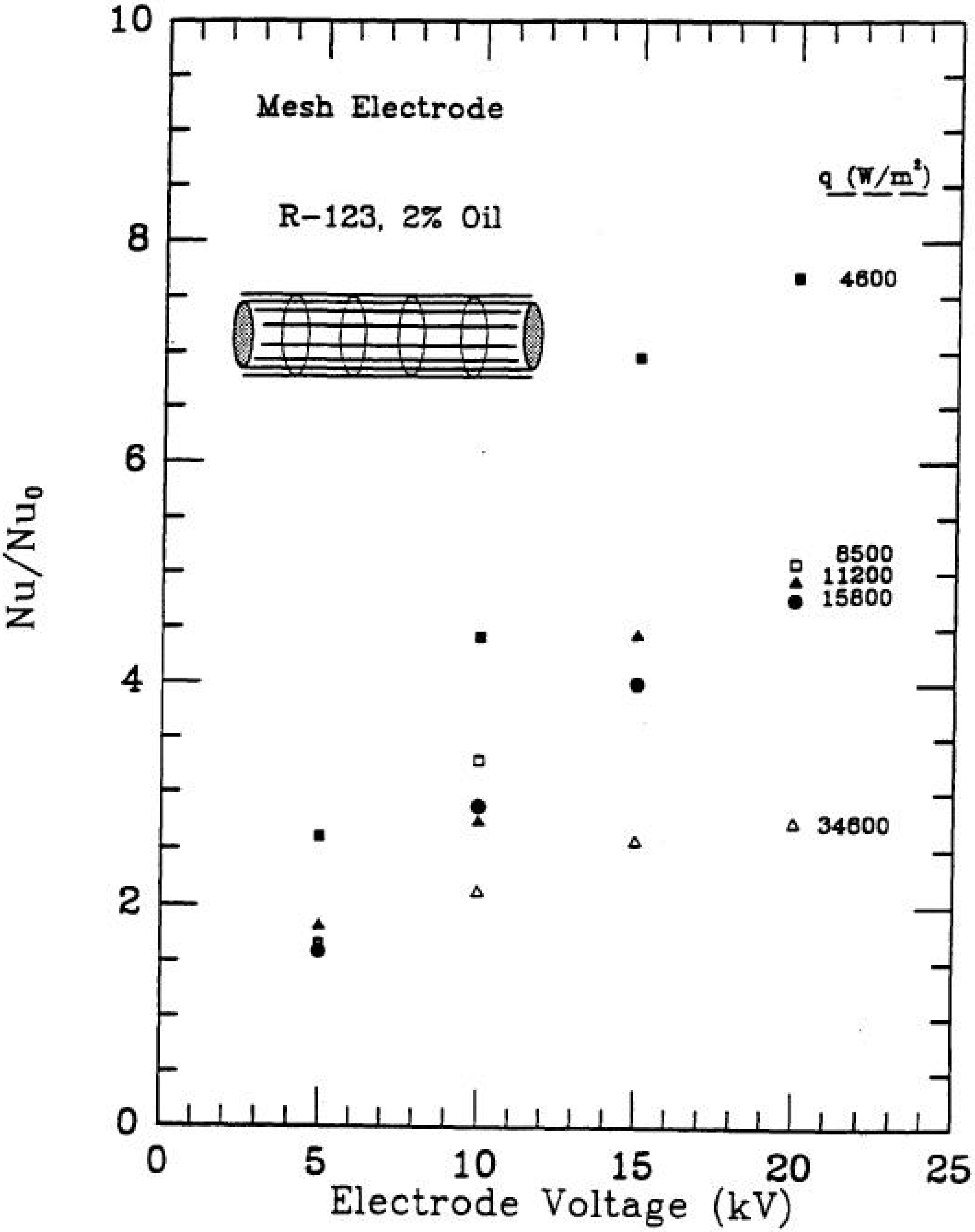

Fig. 5 EHD-enhanced external boiling of R-123 with wire mesh electrode at 2% oil

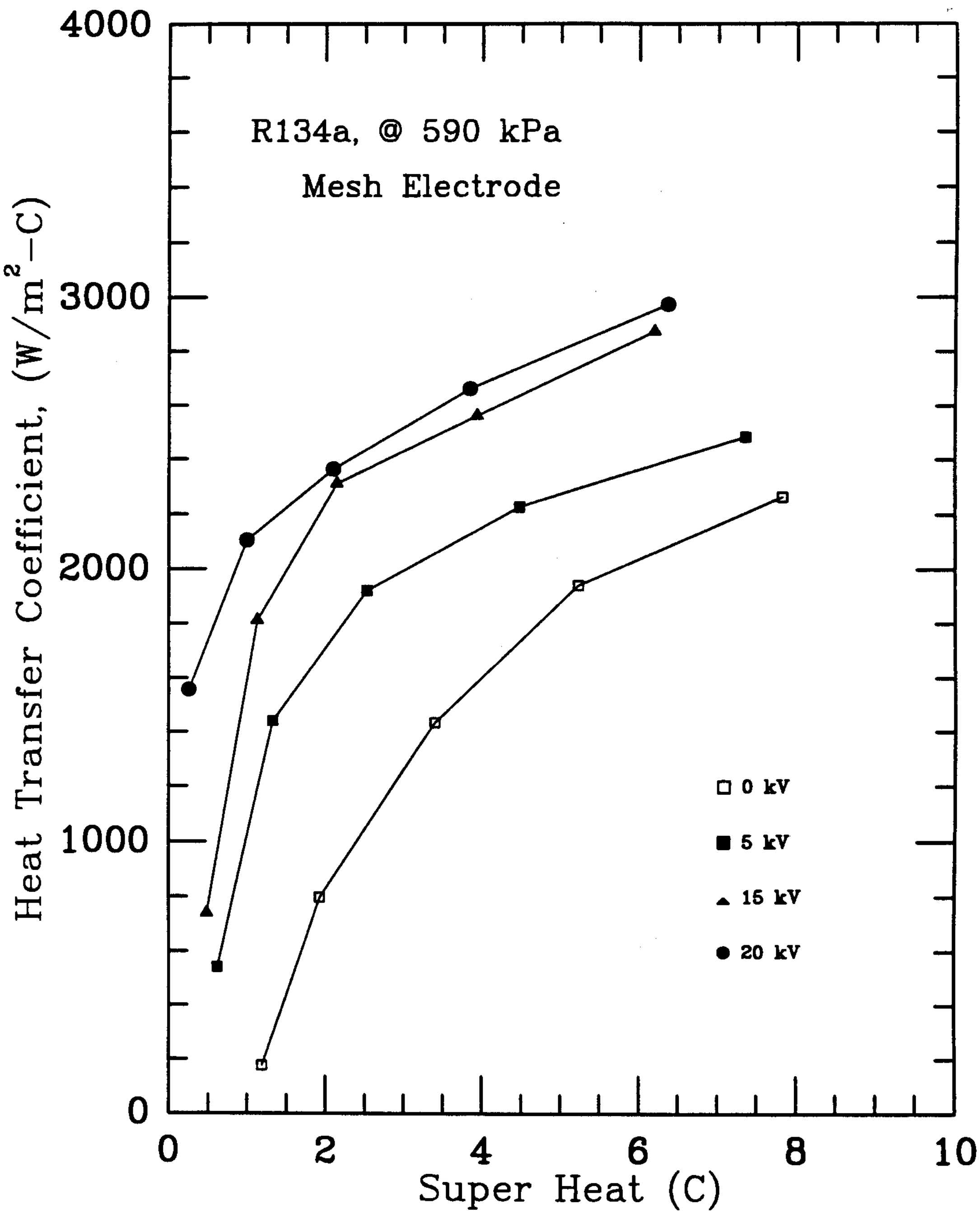

Fig. 6 EHD-enhanced external boiling of R-134a at P=590 kPa and T_{sat}=20.85°C

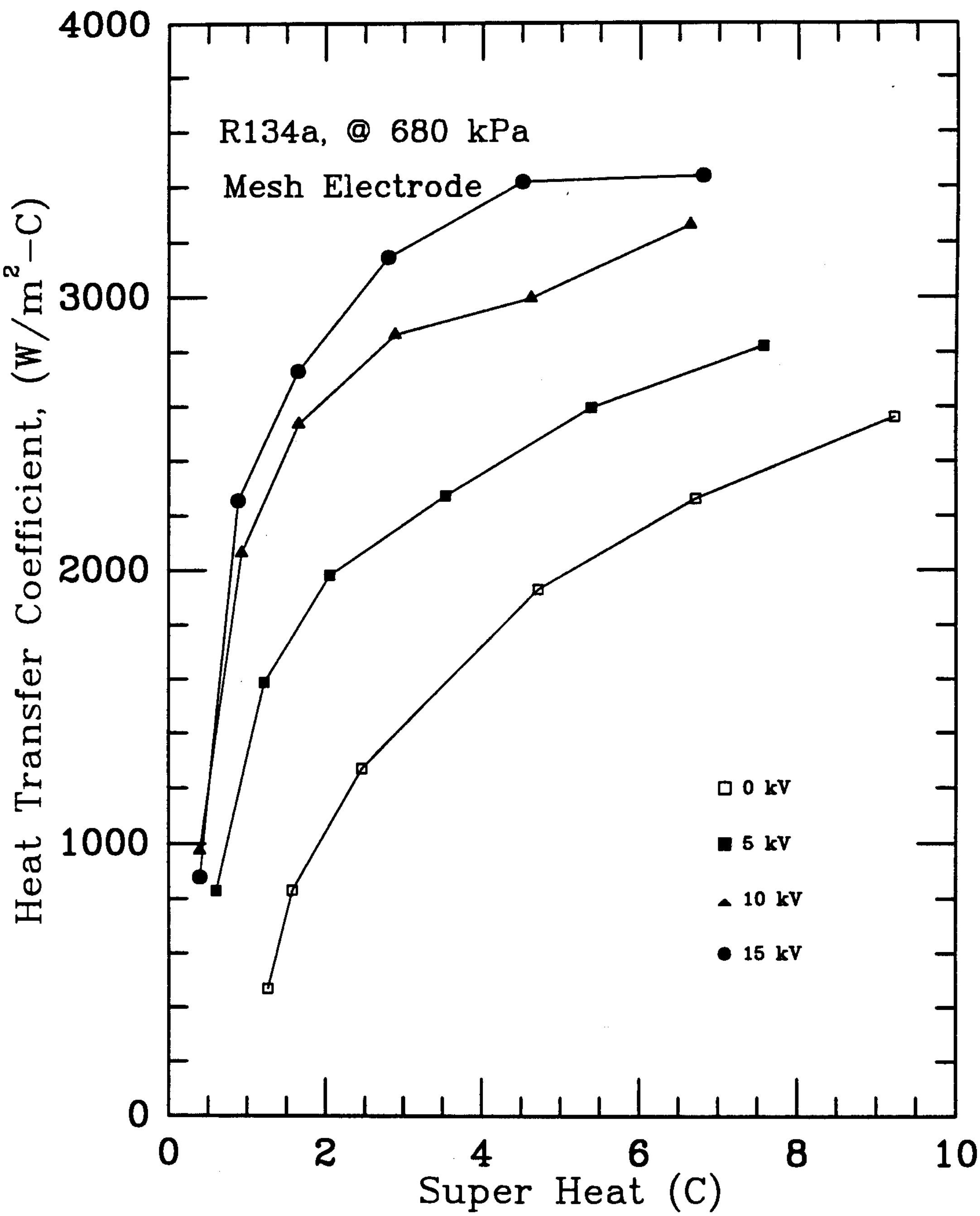

Fig. 7 EHD-enhanced external boiling of R-134a at P=680 kPa and $T_{sat.}$=25.85°C

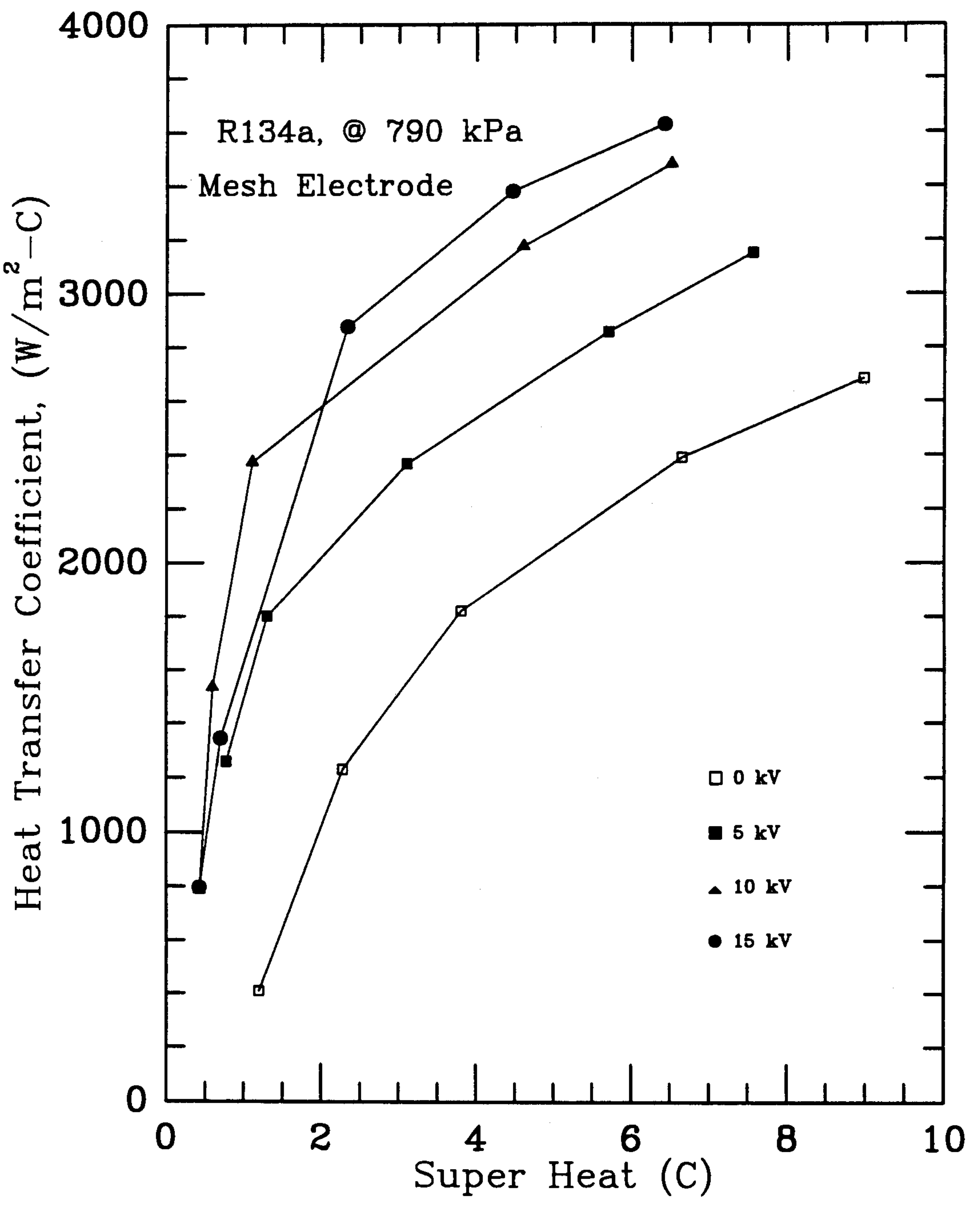

Fig. 8 EHD-enhanced external boiling of R-134a at P=790 kPa and T_{sat}=30.85°C

For R-134a additional experiments are needed to confirm the trends of the preliminary data reported here. Also, effect of lubricating oil and experiments with refrigerant mixtures are in the future planned work.

5.IV Technical Problems Encountered

A number of technical problems were encountered during the course of fabricating and testing of the set up. The leakage of high pressure fluid from the system was taken care of by charging the system first with nitrogen gas and doing a meticulous search until all fluid leakages were removed. Also fabrication and proper installation of the electrodes was a critical job, since short circuits could disrupt the process of testing. Different wires were examined in the process of optimizing the electrodes. R-134a has shown a complicated electrical behavior. The current for a given voltage is variable during a course of time for different pressures or heat fluxes. More data needs to be obtained to be able to assess the electrical behavior of R-134a in terms of current-voltage relation. Realizing that R-134a is a new refrigerant, little information is known on its heat transfer performance. This limited comparison of the data with the base case zero electric field condition.

5.V Future Work

The data reported here for R-134a are preliminary. More experiments are needed to verify the trends observed and the physics involved. Long term effects and design/operational aspects of using R-134a in the presence of EHD field will have to be investigated as well. This will include examining issues such as electrode life, fouling, refrigerant degradation and material compatibility. These and additional experiments will be performed utilizing the funds available through an industrial consortium of sponsoring members.

5.VI References

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6. THE IN-TUBE BOILING TEST RIG

6.I The Test Rig Design

The overall schematic diagram of the test apparatus is shown in Fig. 9. The test section was designed to withstand up to 500 psi pressure so that both low and high pressure refrigerants can be tested. The setup consists of three distinct flow loops : (1) the refrigerant flow loop which provides refrigerant flow through the test section; (2) a hot water loop which provides heating energy to the test section; and (3) a heat pump loop, which condenses and subcools the refrigerant entering the test section.

The refrigerant loop includes a hermetic oil-free pump, a drier filter, a precision coriolis mass flowmeter, an electrical preheater, the test section, a condenser unit and an accumulator. The refrigerant flow rate can be controlled by the pump in the range of 20 $kg/m^2.s$ to 600 $kg/m^2.s$ and the quality at which the refrigerant enters the test section can be controlled by the electric preheater. Presently, the system pressure is controlled manually by regulating the expansion valve of heat pump loop. However, efforts are in progress to install a stepper motor with feedback control loop to perform automatic control of the pressure in the test section.

The test section is a horizontally mounted tube-in-tube heat exchanger. The inner tube of 9.4 mm (0.37 in) inside diameter and 1.22 m (4 ft) long carries the test refrigerant while the outer tube of 19 mm (0.75 in) inside diameter carries the hot water. The test tube wall is 1.65 mm (0.065 in) thick so that it can withstand high pressures (up to 500 psi). To minimize the axial heat losses through the test section, a stainless steel tube, grade SA304, is used. The outer tube is made of plexiglass. The test section instrumentation allows to operate up to 35 kW/m^2 of heat flux. The exit end of the test section has a sight glass of almost the same inside diameter as the test section in order to observe the flow pattern. Swagelok connections are used on both ends of the test section so as to facilitate quick and easy replacements of the electrodes and the test section. The schematic of the test section is shown in the Fig. 10.

The initial electrode design to be used in the experiments is a simple coaxial cylindrical type of 3 mm (0.118 in) diameter. This electrode is supported inside the test section by teflon spacers at intervals of 300 mm (1 ft). Positive potential is applied to the electrode through a modified automotive spark plug whereas the tube is grounded. Future experiments will include study of the effect of electrode geometry and orientation on the nature and magnitude of heat transfer enhancement rates.

Also shown in Fig. 10 is the location of thermocouples in the test rig. Wall temperatures are measured using copper-constantan thermocouples directly welded on the outside of the tube. Temperatures are measured at 4 axial locations at equal intervals of 300 mm (11.8 in). At each axial location, thermocouples are placed at 4 circumferential positions at an interval of 90 degrees from the bottom point of the tube.

Heat to the test section is supplied by hot water which in turn is heated using an electrical resistance rod heater. To reduce the time constant, which is proportional to the total volume of hot water in the loop (in the present setup it is just 750 ml), the heater is placed axially inside the hot water tube (See Fig. 9). This also helps us to quantify heat input to the water more accurately. The entire hot water loop is thermally insulated. To ensure thorough mixing of water

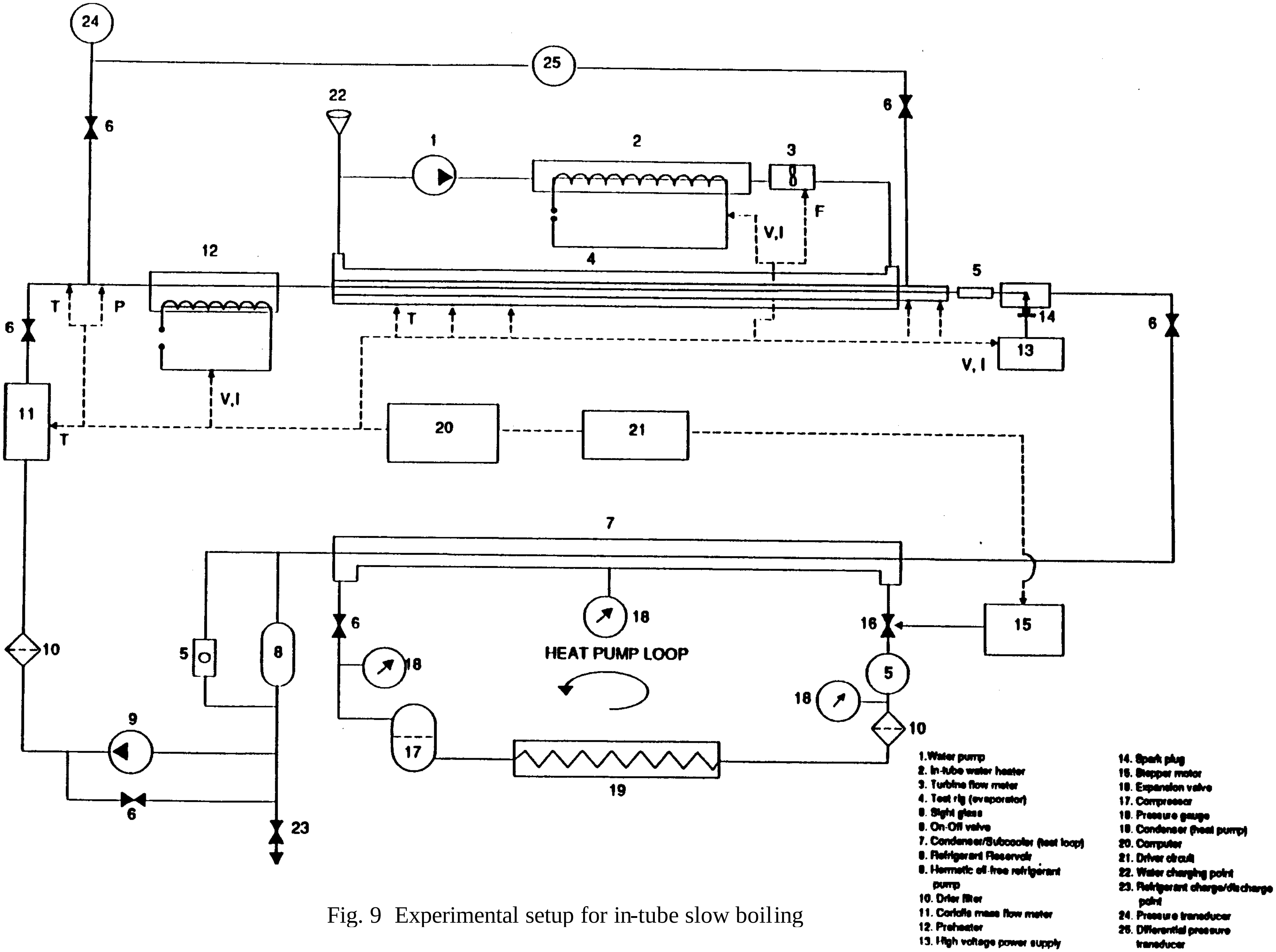

Fig. 9 Experimental setup for in-tube slow boiling

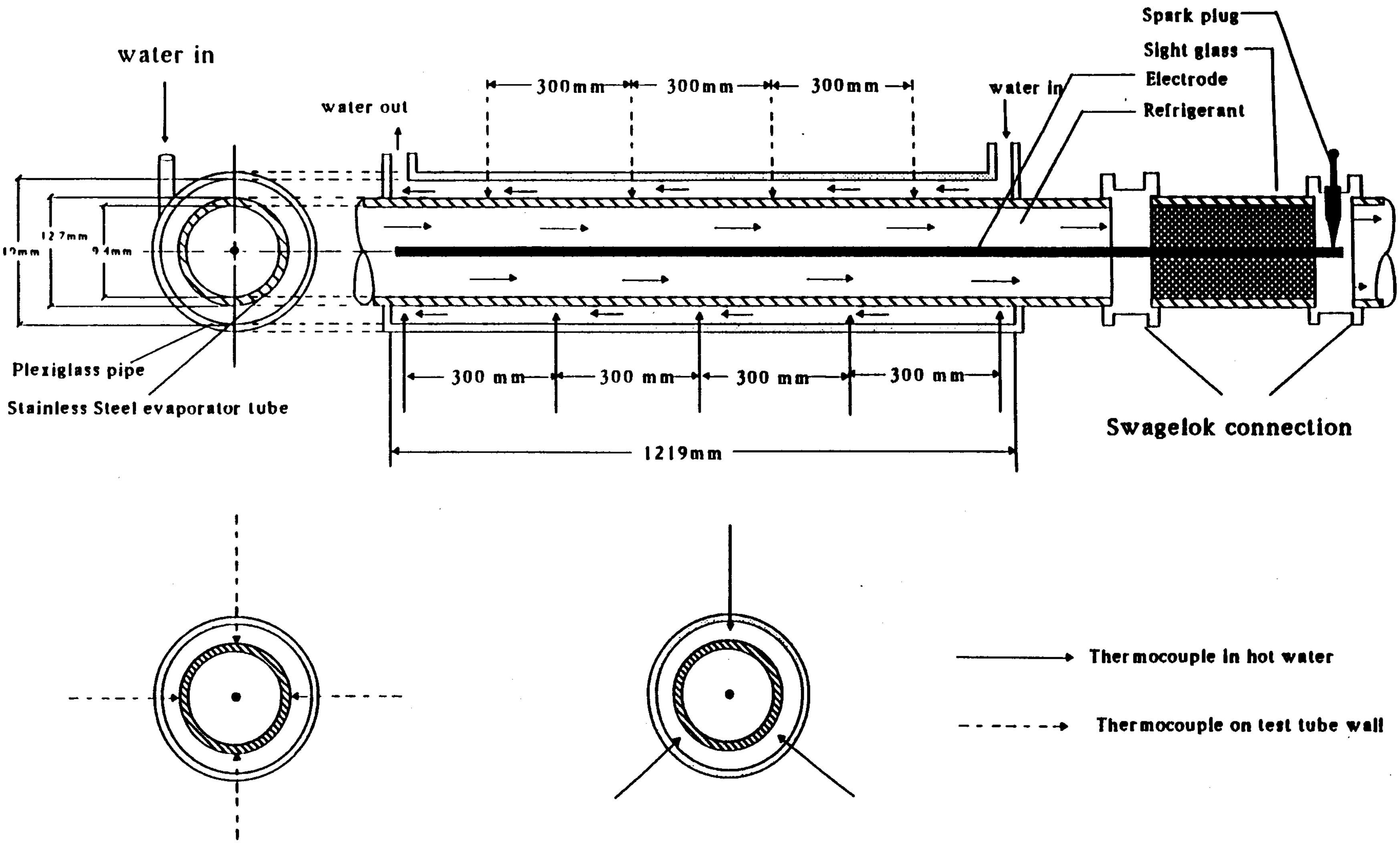

Fig. 10 Layout of lest rig showing thermocouple locations

inside the test section, water enters and leaves the test section tangentially. Heater power is controlled using a variable transformer. Thermocouples are placed at 5 axial locations in the water side to measure hot water temperature and at each axial location 3 thermocouples are placed circumferentially at an interval of 120 degrees from the top point of the tube. Locations of the thermocouples are shown in Fig. 10.

Evaporating pressure at the inlet and exit is measured using a pressure transducer and a differential pressure transducer, so that accurate local pressure and pressure drop measurements can be performed. The inlet and exit of the test section in the refrigeration loop is equipped with a thermocouple to provide comparison with the saturation temperature inferred from the pressure measurement.

There are three main independent variables that can be controlled in the current test section design: (i) heat flux, determined by the hot water heater power; (ii) refrigerant flow rate, adjusted manually by changing input voltage to the hermetic oil-free pump and (iii) the inlet vapor quality, controlled by regulating heat input in the preheater.

A typical experimental run begins by first turning on the refrigerant pump and fixing the mass flux at the desired value. Next, the heat pump loop is switched on and is followed by the hot water pump. The hot water heater is then turned on and set at the required heat flux. The refrigerant always enters the preheater in a subcooled condition. Hence to fix the quality before the refrigerant enters the test section, the preheater is then switched on and is set at the proper level. A "steady state" condition for the test rig is defined as when all the temperature and pressure signals remain unchanged for 15 minutes.

6.II Data Reduction

Data acquisition is performed with a personal computer, a multiplexer and computer-interfaced multimeters. All the temperatures, pressure, flow rate, heater power, preheater power, EHD voltage and current signals are acquired by the computer. Data reduction calculations pertaining to the heat transfer coefficient, quality, the mass flow rate, etc. are also performed by the same computer. The heat transfer coefficient calculation is based on the following defining equation:

$$h = \frac{Q}{A\,(T_{wi} - T_{sat})} \qquad (3)$$

in which Q is the energy transferred to the test section, T_{wi} is the average inside wall temperature calculated using outside wall temperature and T_{sat} refers to the saturation temperature at average test section pressure. The quantity Q is taken to be equal to the hot water heater input when the average heat transfer coefficient over the length of tube is desired. However, when the local heat transfer coefficient is of interest, Q is obtained from the energy balance on the water side between any two axial positions:

$$Q = m_w \, c_w \, (T_{w,1} - T_{w,2}) \tag{4}$$

Here, $T_{w,1}$ and $T_{w,2}$ represent the average temperature of water side thermocouples at any two consecutive axial locations. T_{wi}, the inside tube wall temperature is calculated by assuming that heat is conducted only radially from outside to the inside of tube as:

$$T_{wi} = T_{wo} - \frac{Q\delta x}{kA} \tag{5}$$

where, T_{wo} is the average of the 4 circumferential wall thermocouples at an axial position, δx is the test tube wall thickness, k is the thermal conductivity of the tube material and A is the inside surface area of the tube.

Quality of the refrigerant coming out of the test section will be calculated by performing an energy balance, 1st law analysis, on the test section (see Fig. 11). The quality at which the refrigerant is entering the test section is obtained by using the following equations:

$$e_0 = f\,(\,T_0\,,\,P\,) \tag{6}$$

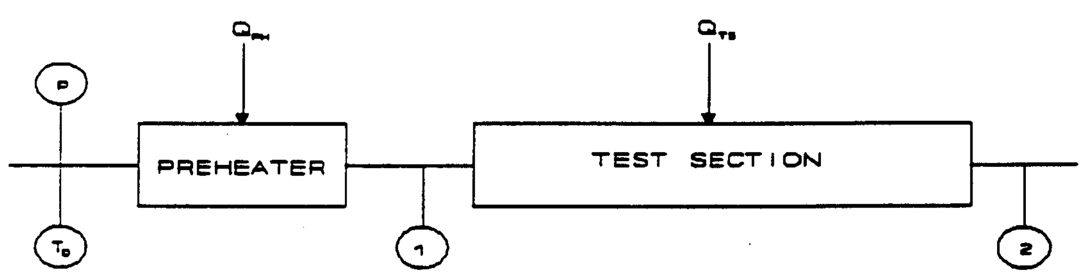

Fig.11

$$e_1 = e_0 + \frac{Q_{PH}}{m_r} \quad (7)$$

$$X_1 = f(P, e_1) \quad (8)$$

Here it is assumed that the pressure drop across the preheater is negligible. Similarly, quality at which refrigerants are coming out of the test section can be obtained once Q_{TS}, heat input to the test section, is known.

While determining the average heat transfer coefficients, the heat input to the test section can also be quantified based on the hot water mass flow rate and the temperature drop. But uncertainty in the heat input measurement using this method was found to be in the range of ±5.6% to ±28.2% (depending on the heat flux rate) whereas that of the electrical power input to the in-tube heater is between ±0.6% and ±1.53%. Uncertainty is much lower in the later case mainly because accuracy in measurement of voltage and current (electrical heater power input) is much better as compared to the corresponding temperature measurement. The maximum uncertainty in the measurement of heat transfer coefficient is ±11.34% whereas for quality is ±16.87% and both occurs at lowest heat flux.

6.III The Preliminary Results

To verify the test section design and the experimental procedure, the low pressure R-123 refrigerant was used for preliminary runs. Fig. 12 shows the variation of EHD discharge current with time for the freshly charged refrigerant at fixed pressure, heat flux, mass velocity and applied EHD voltage. Initially, EHD current increases and reaches a maximum value of around 0.25 mA and then starts decreasing slowly as time passes by. This happens because for the freshly charged refrigerant more current is required initially for the polarization of charges to take place in the refrigerant. It should be also noted that for this particular case EHD power consumption is less than one Watt.

Fig. 13 shows the variation of EHD current with mass velocity of refrigerant. It can be seen that EHD current increases with an increase in mass velocity. This is because as the mass velocity is increased the quality of the refrigerant in the test section decreases for a given heat flux which leads to less vapor flow. Since conductivity of liquid R-123 is more than vapor, the current is high for higher mass velocities.

6.IV Technical Problems Encountered

The following problems wee encountered during the fabrication and testing of the in-tube boiling apparatus:

1. Initially, when the setup was started, it was difficult to maintain a steady mass flux of the refrigerant. This happened because the refrigerant pump being new, took some time to reach its operating characteristics curve. Now we are able to maintain mass flux within ±2% of the

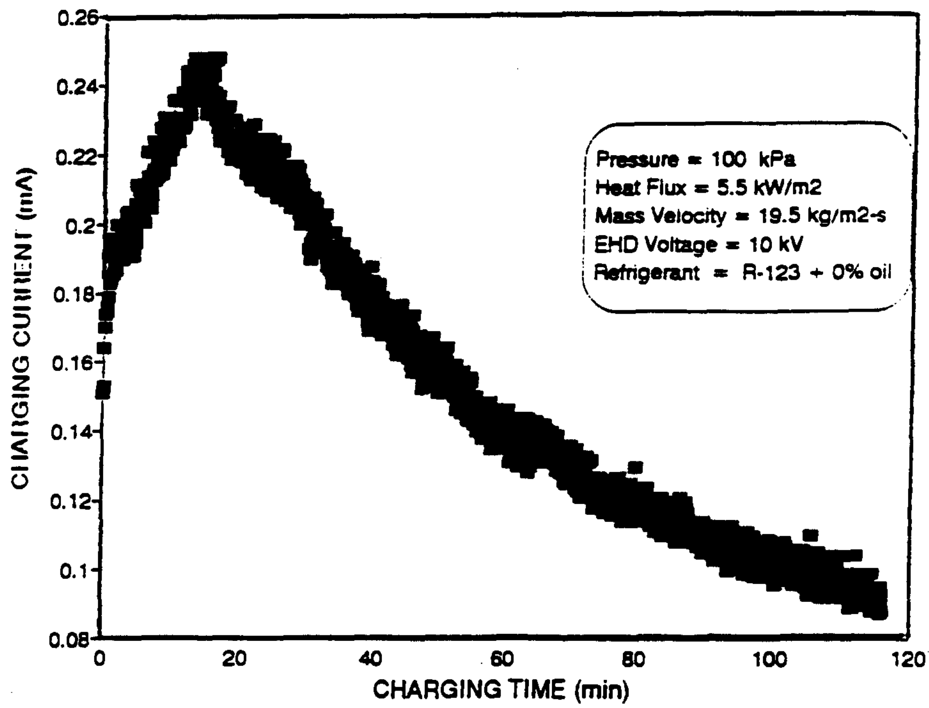

Fig. 12 Variation of EHD current for the in-tube boiling test section

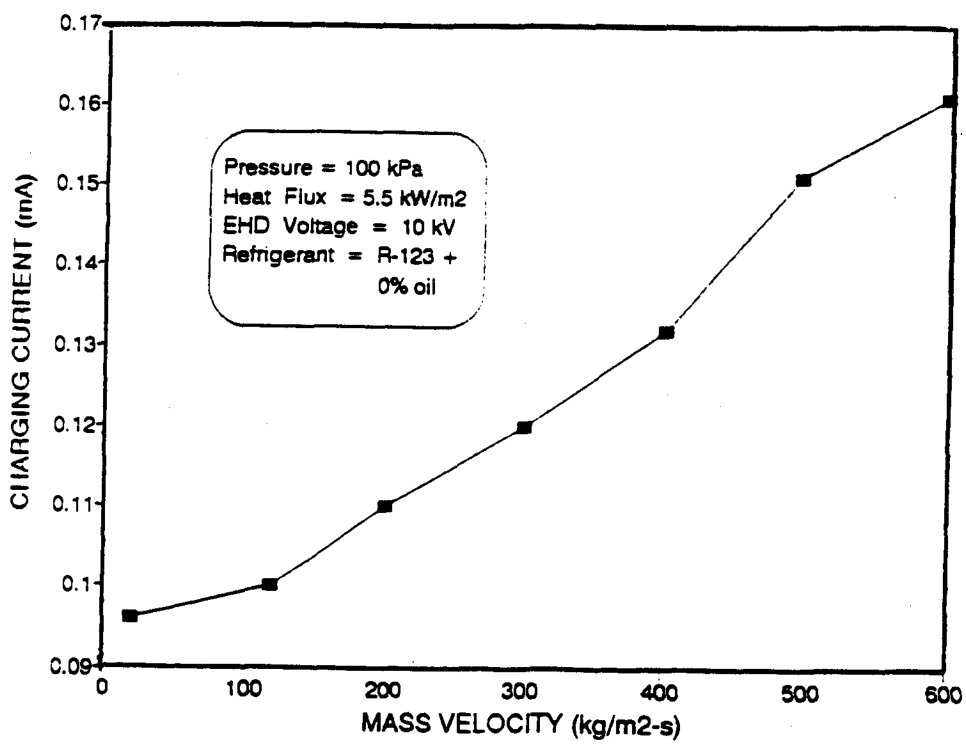

Fig. 13 Mass velocity vs. the EHD current for the in-tube boiling test section

flow.

2. Problems were encountered with calibration of the multiplexer boards used for data acquisition of the temperature signals. The Resistance Temperature Detector (RTD) present on the multiplexer board for cold junction compensation did not give a consistent value. Accordingly, it has been decided to use ice bath as our reference junction instead of RTD. Since thermocouples are very susceptible to noise pickup, each signal is acquired 17 times and an arithmetical average is taken after rejecting the lowest and highest data. Also a moving average is taken to reduce the influence of noise in the data.

3. It was difficult to maintain test section pressure by controlling the expansion valve manually. Therefore, we are in the process of making a stepper motor feedback control to take care of this problem.

4. It took some effort to find all the leaks in the system and come up with a totally leak-free test loop apparatus.

6.V Future Plans

As planned, the ARTI sponsored test rigs described here will be used to conduct EHD-enhanced in-tube flow boiling experiments on alternate refrigerants/ refrigerant oil mixtures. The U.S. department of energy will support these efforts, effective June 1,1993. Concurrent to the flow boiling experiments, EHD-enhanced pool boiling alternate refrigerants on selected tube geometries will also be studied. The funds provided by the DOE have been augmented by contributions from four industrial sponsoring members. The sponsoring members will each have one representative in the five-member advisory board (four industrial member plus DOE project manager). Collective suggestions of the members will determine the direction of the research and the tasks to be performed in the current project. Following is the tentative time schedule for the planned tasks in the next quarter for the in-tube boiling experiments.

June 1 - July 15	Conduct Wilson Plot experiments using R-123 as the testing fluid.
July 15 - Aug 15	Conduct EHD-coupled experiments using R-123 to test the apparatus.
Aug 15 - Aug 31	Perform preliminary EHD experiments with R-134a.
Sept l - Sept 15	First Quarterly Report preparation for the U.S. DOE.

7. SUMMARY

All of the major tasks planned in the current project were accomplished. A comprehensive search of the literature on EHD boiling heat transfer enhancement of alternate refrigerants was conducted and the tabulated results are included in Appendices A and B. Design, fabricating, and testing of the in-tube, EHD-enhanced boiling test rig was completed. Preliminary data on the "I-V" characteristics of the electrode system were presented in this report. The experiments on in-tube boiling of R-123 have just begun, using a single conventional wire-type electrode. Similar experiments are planned for R-134a. Future experiments will include testing of refrigerant mixtures.

The data collected on external boiling of R-123 and R-134a suggest an excellent feasibility of the EHD technique for boiling heat transfer enhancement of these two refrigerants. Up to nine-fold heat transfer enhancements for R-134a and seven-fold for R-123 have been obtained Based on the pool boiling results, we expect that R-134a will respond favorably to the EHD enhancement in the flow boiling regime as well.

Upon completion of this ARTI-sponsored effort, as of June 1, 1993 the in-tube and external boiling projects will be funded jointly by grants from the U.S. Department of Energy and a consortium of industrial members. An advisory committee (composed of representatives of the industrial sponsors) and the DOE Project manager will provide feedback and direction to this research.

8. ACKNOWLEDGMENTS

This research was conducted under ARTI Contract No. 655-51700. The Program Manager for this project was Glenn C. Hourahan.

9. COMPLIANCE WITH AGREEMENT

No modifications or deviations from the technical performance of work as described in the contract agreement was necessary during this reporting period.

10. PRINCIPAL INVESTIGATOR EFFORT

Dr. Michael Ohadi was the Principal Investigator and director for this project. He devoted a total of 190 hours to this project toward its completion.

Appendix A

Tabulated Literature Survey for
EHD-Enhanced External Boiling

Pool Boiling
Literature Survey

Source	Liquid Used	Test Section	Enhancement Technique	Test Condition	Remarks
1. Allen, P. [A1]	R114	finned tube	EHD	copper wire mesh cylinder max voltage: 30 kV DC, AC	max enhancement is 10 fold
2. Asch, V. [A2]	R113	2 circular plates	EHD	circular plate DC, AC	max enhancement is 5 fold
3. Blachowicz, R. [B1]	benzene	vertical glass tube	EHD	disc type max voltage: 20 kV DC	max enhancement is 2 fold
4. Cooper, P.[C1]	R114	finned tube within a concentric shell	EHD	copper wire mesh cylinder max voltage: 30 kV DC, AC	max enhancement is up to 10 fold
5. Hahne, E. [H1]	R11	copper	fins 19fpi and 26 fpi fin height of 1.52mm	**	h for finned tubes is smaller than palin tubes at q < 1 Kw/m K in nucleate boiling regime, h from single finned tubes is greater than plain tubes.
6. Johnson, R.[J1]	good dielectrics (not a specific one)	**	EHD	**	an analytical study for a boiling flat surface.

Source	Liquid Used	Test Section	Enhancement Technique	Test Condition	Remarks
7. Kawahira, H. [K1]	R11 2%wt oil	smooth OD 22.4 mm	EHD wire insulators placed equidistant axially 75mm apart. negative DC positive DC AC	P: near atmosphere T: 25°C hf: 2.8 -1.7 applied volt: 30kV	the stronger the electric field, the smaller the no. of bubbles. h is 2-3 times without application of electric field. deterioration due to oil contamination was not observed.
8. Lovenguth, R. [L1]	R113, R21, chloroform, & carbon tetrachloride	a glass container a platinum wire heater	EHD	electrically conducting Pyrex tube max voltage: 15 kV nonuniform DC	max enhancement is 3 fold
9. Markels, M. [M2]	deionized water	concentric cylinders	EHD	tube max voltage: 5 kV, 60 Hz AC	max enhancement is 2 fold
10. Markels, M. [M3]	isopropyl alcohol	horizontal tube	EHD	aluminum tank max voltage: 10 kV, 60 Hz DC, AC	max enhancement is 4 fold
11. Memory, S.B. [M1]	R114 0,3,10 wt% oil	smooth copper OD 15.91 mm SS cartridge heater OD 6.35 mm	26fpi GEWA-K 26fpi GEWA-T 26fpi GEWA-YX Turbo-B Thermoexcel-E High Flux	hf:max 100 T:2.2°C	in natural convective region, oil has no effect on h. for pure R114, incipient hf for structured surfaces is less than that of smooth and finned tubes.
12. Ogata, J. [O1]	R11 ethanol 2%wt	copper cartridge heater	EHD electrode wires OD 22.4 mm 3 mm away from heat transfer surface 7 mm separation between each wire.	P: 105kPa hf: 5.8 T: 25°C positive DC	enhancement ratio was 8.5 times that without electric field. generated bubbles pushed against the injection plate and moved on it before migrating through electric field.

Source	Liquid Used	Test Section	Enhancement Technique	Test Condition	Remarks
13. Ogata, J. [O2]	silicon oil & ethylalcohol mix	transparent conductive horizontal parallel plates	EHD	parallel glass plates max voltage: 20 kV DC	break up of bubbles were investigated.
14. Ohadi, M.M. [O3]	R123	wire-cylinder	EHD	tube-wire max voltage: 25 kV DC	max enhancement is 5 fold
15. Olinger, J. [O4]	distilled, deionized water	flate surface	EHD	flat grid electrode DC, AC	
16. Rutkowski, J. [R1]	nitrogen	cylinder	EHD	coaxial & parallel cylinder max voltage: 50 Hz AC	max enhancement is 2 fold
17. Takano, K. [T2]	water, ethanol, R113, R10, & cyclohexane	horizontal plate electrode over a liquid surface	EHD	stainless steel & brass circular discs max voltage: 30 kV DC	
18. Thome, J. [T1]	95% pentane/ 5% tetradecene 20% n-pentane 20% n-heptane 20% cyclohexane 35% p-xylene 5% 1-tetradecene	copper Diam 12.7 mm cartridge heater	GEWA-TX 18.8fpi	P:2.07 bar 6.9 bar hf: 2- 200	GEWA-TX had boiling performance of 4-10 times that of ordinary smooth tube.
19. Uemera, M. [U1]	R113	a flat plate under atmospheric pressure	EHD	stainless steel wire mesh max voltage: 35 kV DC	max enhancement is 14 fold nucleate and film boiling investigated.

Source	Liquid Used	Test Section	Enhancement Technique	Test Condition	Remarks
20. Wanniarchchi, A.S. [W3]	R114	smooth copper OD 15.9 mm 457.2 mm long porous coated copper nickel heater max power 100W	Porous coating	hf: .5-95 T:-2.2°C 6.7°C	with porous coating, h is 10 times that of smooth tube. nucleate boiling occurred at lower hf and superheats compared to smooth tube. presence of oil reduced h by 35% in porous coated tube.
21. Webb, R.L.[W1]	R22,R11, R12, R123,R134a	165 mm long copper OD19.05 mm ID9.53 mm cartridge heater 500 W	standard fin TU-B fin W-SE fin 26-40 fpi 2 internal axial grooves	T:27°C hf:2-70	h for R123 and R134a is approximately equal to R11 and R12 respectively at 26fpi.
22. Webb, R.L. [W2]	R22,R11, R12, R123,R134a	copper OD 17.5-19.1 mm ID 9.53 mm cartridge heater 500 W	1024fins/m integral fin GEWA TX-19 GEWA SE Turbo-B all have specially made 9.53mm inside bore	T: 4.44 C° and 26.7°C hf: 3-90	h at given hf increases w/ sat T for all tube geometries h for R123 and R134a are within 10% of values for R11 and R12.
23. Yabe, A. [Y1]	Furonsorubu AE (96% w.w. R113 & 4% w.w. ethanol)	horizontal plate	EHD	ring electrode max voltage: 25 kV DC	max enhancement is 100 fold for convective heat transfer and 2 fold for boiling
24. Zharzholianni, A. [Z1]	acetone (polar) benzene n-pentane	cylindrical & planar	EHD	plate electrode max voltage: 35 kV DC	max enhancement for: acetone is 2.14 fold benzene is 1.6 fold n-pentane is 1.2 fold

Nomenclature

h: heat transfer coefficient
hf or q: heat flux [kW/m^2 K]
P: Pressure
T: Temperature
SS: Stainless Steel
OD: Outer diameter
ID: Inner diameter
fpi: fins per inch

Pool Boiling
Literature Survey

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Appendix B

Tabulated Literature Survey for In-Tube Boiling

Forced Convection Boiling (In-Tube)
Literature Survey 1986-1992

Source	Liquid Used	Tube used	Enhancement Technique	Test condition	Remarks
1. Damiandis, C. [D3]	R114	smooth ID 17.9mm OD 19.1mm lo fin brass Thermo-excel ID 14.97mm OD 19.05mm	lo fin fin height .5mm fin pitch .88mm EHD	**	application of 10 kV eliminated hysteresis. keeping bubbles attached to surfaces incresed turbulence and mixing. enhancement ratio decreased with increasing heat flux. EHD produced 2-3 times more heat transfer than the lo-fin tubes.
2. Eckels, S.J. [E1]	R134a R12	smooth 3.67m long ID 8mm OD 9.25mm	none	mf:125-400 hf:-- x:10-90% P: 0.35-0.49MPa T:5-15 ^{0}C	for single phase flow, h for R134a is 33% higher than R22. for evaporation , h for R134a is higher than R22 by 35-45%.
3. Eckels, S.J. [E2]	R12 R134a	smooth 3.67m long ID 8mm OD 9.52mm microfin 3.67m long ID max 8.72mm OD 9.52mm	microfin 17°spiral angle 60 fins fin height .2mm	mf: 125-400 x:5-13% avg T: 5-15°C	for R134a, as mass flux increased three times the heat transfer coefficient increased by 50 % for microfin and 100% for smooth tube. penalty factor decreased as mass flux increased.
4. Ha, Samchul [H5]	R12 oil 3GS 0-5%	copper 1.2m long ID 8 mm OD 9.5mm	microfin 60 fins .18mm height 18°spiral angle	mf:25-100 hf:5-10 P: 0.32MPa x:10-60%	suppression of foaming and a highly oil-rich film around the perimeter of the tube. liquid film vaporization inhibited by oil concentration
5. Hambraeus, K. [H4]	R134a synthetic oil[EXP-0275] 0-2.5mass%	smooth copper 1m long ID 12mm	none	mf:60-240 hf: 2-10 x:25% P:2.3 bars for oil-free mixture T:20 -5°C	R134a showed a higher heat transfer coefficient than R22 for 0.5 mass %oil content. Best heat transfer coefficient was at 2 and 4 kW/m^2

Source	Liquid Used	Tube used	Enhancement Technique	Test condition	Remarks
6. Jensen, M.K. [J5]	R113	SS304 total heated and unheated length 1.727m ID 8.10mm OD 9.53mm (vertical tube)	3 types of twist tape tubes tape twist ratios of 3.94, 8.94, 13.92. tape widths were on average 7.85mm	mf:120-1600kg/m^2s hf:0-50 x:0-61% P:276,551,827 kPa	as heat flux increases tape twist ratio has less effect on heat transfer coefficient.
7. Jung, D.S. et al. [J1]	R22 and 0, 21,60,89% R22 in R22/R152a mixture	SS 4m long ID 0.9cm OD 0.95cm.	none	mf=250-720kg/m^2s hf=10,17,26,36,45kW/m^2 x=up to 95% outlet P=330kPa for R12 360kPa for R152a	no circumferential wall temp variation observed.
8. Jung, D.S. et al. [J2]	23,47,77% R22 in R22/R114 mixture	SS 4m long inner ID .9cm OD .95cm	none	mf:250-720 hf:10,17,26,36,45 x: no greater than 90% P:400kPa for R22 260kPa for R114	suppression of nucleate boiling leads to temperature reduction of h as x increases.
9. Kedzierski, M.A. [K8]	R11, R123 .55% alk. in R123a .2% alk. in R123a .0005 mass fraction lubricant	quartz tube ID 9mm OD 12mm	five micron polish	q: 15-30 Re: 0-9500	for Reynold's number below 9500, h of R123 was 22% greater than R11. addition of lubricant caused more sites to become active in generating bubbles.
10. Khanpara, J.C. [K11]	R113	copper 1.21m long inner ID 8.78mm OD 9.5mm outer ID 19.1mm	9 types of microfins fin heights .1-.19mm 60-70 fins spiral angle of 8^{o}-25^{o}	mf:197-594 hf:10.72-53.65 x:15-85% P:312-351 kPa T:76.4°C Re: 5,000-11,000	experiment showed enhancement factors of 1.3-2.0 pressure drop increased at most by factor of 1.8 round peaked microfin with 70 fins had the best performance.

Source	Liquid Used	Tube used	Enhancement Technique	Test condition	Remarks
11. Murata, K. et al. [M1]	R11, R114 and 0,25,50,75 % mass fraction R11 in R11/R114	smooth copper 734mm long inner ID 10.3mm OD 12.7mm	none	mf:100,200,300 q:10,20,30 x:10-90% P:2 bars	binary mixture h << pure refrigerant h in the boiling dominant region.
12. Reid, R.S. et al. [R2]	R113	copper ID 8.50mm OD 9.525mm —— High fin 30° copper ID 8.712mm OD 9.525mm —— Microfin 17.5° SS ID 10.92mm OD 12.7mm —— Twisted tape 5 other variations of microfin and high fin also used	High fin 30° Microfin 17.5° Twisted tape	High fin mf:398-409 hf:19.994-27.518 x:0-70% P:337kPa Microfin mf:234-601 hf:15.094-39.638 x:0-65% P:332-340kPa Low Fin mf:233-237 hf:12.528-21.491 x:0-70% P:337kPa	twisted tape has enhancement factor of 1.5 finned tubes has enhancement factor of 1.1-2.8 microfin tubes have better performance ratio than high fin. performance ratio of 1.1 for low fin and .5 for high fin
13. Ross, H.D. et al. [R4]	R13B1 and R152a 0.18%wt-0.833wt% R13B1 in R13B1/R152a	2 apparatus tested 1.SS 2.7m long inner ID .9 cm OD .95 cm 2.SS .6m long inner ID .9cm OD .95cm	none	mf:150-1200 hf:10-95 x:0-100% P:1.2-7bars $1/X_{tt}$= .3-35 Re=3000-50,000 Pr=3-4	circumferential variation in h makes modelling difficult. suppression of nucleate boiling done by lowering pressure.
14. Sami, S. M. et al.[S2]	R22, R12 40, 60, 80% R22 in R22/R114 70, 80, 90% R22 in R22/R152a 1% oil	inner and outer copper inner ID .0212m OD .0226m outer ID .0508m	double fluted inner tube (4 flutes)	mfr:50-90 x: 25% Re: $9.8 \times 10^3 - 2.2 \times 10^4$ P: 180-600kPa	increase in R114 decelerates nucleate boiling

Source	Liquid Used	Tube used	Enhancement Technique	Test condition	Remarks
15. Sami, S. M. et al.[S3]	R22/R114 1% oil	copper 1.2 m long inner ID 17.5 mm OD 28.6 mm outer ID 32.3mm	double fluted inner tube	mf:180-290 hf:7-24 x:10-60% P:570kPa for R22 517kPa for 80%R22/R114 457kPa for 60%R22/R114 417kPa for 40%R22/R114	most influential fluid properties are thermal conductivity and heat of evaporation
16. Sami, S. M. et al.[S5]	R22 & R152a 40, 60, 80%R22 in R22/R114 70,80,90%R22 in R22/R152a	inner and outer copper 3.5m long inner ID .0212m OD .0226m outer ID .0508m	double fluted inner tube (4 flutes)	mfr:50-90 x: 25% P: 180-600kPa Re: $9.8 \times 10^3 - 2.2 \times 10^4$	With a higher concentration of R152a there is a slight increase in h.
17. Sami, S. M. et al. [S8]	R12 R134a R22	equivalent diameter .024m 1.2 m long	double fluted inner tube	mf:181,242,288 hf:7-24 x:10-60% P:365kPa for R-12,R134a 570kPa for R22	at high qualities, two-phase boiling heat transfer coefficient depends on mass flow rate experiments showed enhancement factors of 3.0-3.3 enhancement more significant in R134a than in R12
18. Sami, S. M. et al.[S10]	R22 40,60,80% R22 in R22/R114	equivalent diameter .024m 1.2 m long	double fluted	mf:181,242, 288 hf:7-24 x:10-60% P:570kPa for R22	heat transfer enhancement factor depends on mixture concentration
19. Schlager, L. M. et al. [S7]	R22	copper 3.7m long inner ID 11.7mm OD 12.7mm	microfin 60-70 fins height of .15mm-.30mm angles 15-25 degree	mf:75-300 hf:100-300 x:15-85% P:0.5-0.6 MPa	experiments showed enhancement factors of 1.5 to 2.2 and penalty factors of 1.2-1.35
20. Schlager, L.M. [S14]	R22 naphthenic base mineral oil 0-5%	2 tests 1. smooth ID 8.0mm OD 9.52mm 2. microfin ID 8.72mm OD 9.52mm	microfins height: 0.2mm 18° spiral angle	mf:25-400 hf: not given x:15-85% P:0.5-0.6MPa T: 0-6°C	small quantities of oil improve heat transfer. enhancement factor of smooth tube was 1.36 compared to 1.11 at small oil quantities. increasing mass flux, diminishes oil enhancement.

Source	Liquid Used	Tube used	Enhancement Technique	Test condition	Remarks
21. Schlager, L.M. [S17]	R22 150 SUS naphthenic mineral oil 0-5%	3.67 m long ID 9.5mm	Lo-fin	mf:125-400 x:15-85% T:3°C	only oil concentrations of 1.5% or lower lead to heat transfer augmentation w/lo-fin tubes. for smooth tube, enhancement factor is 1.36 for 2.5 % oil concentration.
22. Takamatsu, H. et al. [T1]	R22, R114 25,50,75% bulk molar fraction of R22 in R22/R114	inner copper 4.8m long ID 8.32mm OD 9.52mm outer: polycarbonate resin ID 16mm	internal spiral grooves with depth of .55mm 60 grooves lead angle 30 degrees	mf:77-347 q:.9-3.1 kW x:16-29% P:0.06-.7MPa	q and h increase until dryout point
23. Wattelet, J. M. [W4]	R134a R12	smooth copper 2.43m long ID 10.21mm	none	mf:100, 300, 500 hf:5-30 x:20-60% P: 350 kPa for R134a 363kPa for R12 T:5°C	R134a had 25% more heat transfer coefficient than R12 in annular flow regime. h did not show dependence on heat flux except at low qualities(x=20%) and high heat flux(20-30 kW/m).
24. Yabe, A. [Y8]	R123 R134a	inner: Cupronickel 3.75 m long ID 10 mm outer : acrylic resin	EHD	mf:33-66	with EHD, heat transfer improves by 3 times. EHD is effective in low mass flux region where pressure drop is very small. pressure loss depended on total amount of heat transferred. pressure loss does not depend on the strength of the electric field.
25. Yoshida, S. et al. [Y1]	R22 Suniso oil 0,1,3,6% mass fraction	2 tests 1.SS 4m long ID 10.6mm OD 13.8mm 2.SS 3m long ID 15.4mm OD 19mm	none	mf:100-300 hf:5-30 P: 0.59MPa	at low mass velocity (<100kg/m^{2}s) oil content of 3 and 6% improves h.

Nomenclature

mfr:mass flow rate [g/s]
G or mf:mass flux [kg/m^{2}s]
q or hf: heat flux [kW/m^2]
x: quality
P: pressure of inlet unless stated otherwise
T: saturation temperature
h:heat transfer coefficient

$$\text{heat enhancement factor} = \frac{\text{heat transfer coefficient (augmented)}}{\text{heat transfer coefficient (smooth tube)}}$$

$$\text{performance ratio} = \frac{\text{heat enhancement factor}}{\text{penalty factor}}$$

$$\text{penalty factor} = \frac{\text{pressure drop (augmented tube)}}{\text{pressure drop(smooth tube)}}$$

SS: Stainless steel
ID:inner diameter
OD:outer diameter

Forced Convection Boiling (In-Tube)
Literature Survey

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search conducted under the following headings:

Flow Boiling
Forced Convection Boiling
Refrigerant
EHD
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Index used were:

NTIS 1985-1989 and 1990-1993
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Sources Used in Literature Section

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1.	ASHRAE Journal	TP 490.A75	TH 7201.A22
2.	International Journal of Multiphase Flow	TA357.I57	TA357.I57
3.	International Journal of Heat and Mass Transfer	QC320.I55	QC320.I55
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