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**EHD ENHANCEMENT OF
BOILING/CONDENSATION HEAT TRANSFER
OF ALTERNATE REFRIGERANTS
AND FROST CONTROL**

Final Report
1993 - 1999

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EXECUTIVE SUMMARY

The Electrohydrodynamic (EHD) augmentation technique utilizes the effect of electrically-induced secondary motions to destabilize the thermal boundary layer near the heat transfer surface, leading to substantially higher heat transfer coefficients which, in turn, result in savings in system weight and volume while improving the system coefficient of performance. The EHD technique can also be used to control frost growth in refrigeration systems, so as to minimize defrost requirements, leading to substantial savings in energy consumption of typical refrigeration systems.

This is the Final Report on a project sponsored by the U.S. Department of Energy (US DOE), Grant No. DE-FG02-93CE23803 for the period of 1993 to 1996 and the amended three-year grant under Grant No. FG02-93CE23803-A004 covering the cumulative period of 1993 to 1999 for this project. For the six-year life of the project, the average annual funding by U.S. DOE was approximately \$80,000. The project was carried out in three distinct phases. Phase I focused on Applied Research, Exploratory Development, and Lab Scale technology demonstration units (TDU). In Phase I, the *Applied Research* phase, the scientific principles were investigated and the potential of the Electrohydrodynamics (EHD) enhancement technology for refrigeration and air conditioning applications was quantified. It was demonstrated that this approach has potential advantages over the existing, commercially available heat transfer enhancement technologies. In the *Exploratory Development* stage, Phase II, the concept of EHD heat exchanger enhancement as well as frost control by combined electric field and flow field was demonstrated via a number of benchmark tests. Effects of controlling parameters such as fluid flow rate, fluid temperature, fluid humidity, surface temperature, as well as electric field parameters such as magnitude of applied voltage, DC vs. variable frequency field were addressed. The *Advanced Technology Development* phase, Phase III, involved demonstration of the technology on laboratory-scale systems. The project was co-sponsored by a consortium of member companies in the area of refrigeration and heat exchanger design/manufacturing. The companies served as industrial partners to this project providing continuous technical feedback and working closely with us to ensure the operating conditions were representative of those encountered in practical

refrigeration/air conditioning systems of industrial significance. The focus of our work was in two broad areas: (1) EHD heat exchanger enhancement and performance improvement as related to use of alternative refrigerants and; (2) EHD control of frost in refrigeration systems, particularly those used in supermarket and transport refrigeration systems. Details of the research efforts have been provided in Quarterly Research progress reports sent to the DOE project managers, Mr. William Noel and Esher Kweller. The report is divided in two main parts. Part I reports on our findings for the heat transfer enhancement applications and Part II summarizes our work on frost control in refrigeration systems using the EHD technique. In each of the two parts conclusions and future recommended work are summarized. We have demonstrated that the EHD has a great potential in the next generation of smart heat exchangers and thermal management systems, with broad application in energy, refrigeration, power, and process industries. Up to several folds enhancement in the boiling and condensation heat transfer coefficients can be realized, all at negligible power consumptions. The typical targeted power consumption is much less than one percent of the energy exchange rate in the heat exchanger. The EHD control of frost experimental work also resulted in findings that are equally encouraging. EHD can reduce frost formation by more than 30% for operating conditions and refrigeration systems of practical significance.

As far as future work in the EHD arena, we strongly recommend additional research, particularly in three areas: (1) work that will lead to a better understanding of the phenomena to reduce voltage requirements can be reduced, ideally to conventional levels of 110 or 220 volts; (2) studies that will investigate any long term effects that EHD may have at the component or system levels, for example degradation of electrode materials or working fluids; (3) and finally commercialization efforts that will focus on technology transfers from the laboratory to the market place. Overall the results of this study clearly suggest researchers will peruse EHD in the future, as it remains a viable option to develop the next generation of smart thermal management systems.

EHD ENHANCEMENT OF BOILING/CONDENSATION HEAT TRANSFER OF ALTERNATE REFRIGERANTS

Final Report Part I

1993 - 1996

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NOMENCLATURE

A	= heat transfer surface area
e_1	= specific enthalpy of refrigerant at preheater inlet
e_2	= specific enthalpy of refrigerant at the exit of preheater
h	= heat transfer coefficient
h_0	= heat transfer coefficient (base case)
k	= thermal conductivity of the test section tube material
m_r	= refrigerant mass flow rate
NE	= No electrode present in the test-section tube
Nu	= Nusselt Number in the presence of electric field effect
Nu_0	= Nusselt Number at zero electric field conditions (base case)
P	= absolute test section pressure
Q	= heat transferred to the test section
Q_{ph}	= heat transferred to the refrigerant by the preheater
Q_e	= heat transferred to the refrigerant by the test-section heater (evaporator)
Re_L	= liquid Reynolds Number
T_w	= inner wall test section tube temperature
T_{wo}	= outer wall test section tube temperature
T_{sat}	= saturation temperature of the refrigerant
T_1	= refrigerant temperature at preheater inlet
X_2	= refrigerant quality at the exit of preheater
X_3	= refrigerant quality before entering the test section
X_4	= refrigerant quality at the exit of the test section
X_{avg}	= average test-section quality (average of X_3 and X_4)

IN-TUBE BOILING ENHANCEMENT OF R-134a USING SMOOTH AND MICROFINNED TUBES

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1. EXECUTIVE SUMMARY FOR PHASE I OF THE RESEARCH

The Electrohydrodynamic (EHD) augmentation technique utilizes the effect of electrically-induced secondary motions to destabilize the thermal boundary layer near the heat transfer surface, leading to heat transfer coefficients that are often an order of magnitude higher than those achievable by any of the conventional enhancement techniques.

A study of the in-tube flow and external boiling and condensation heat transfer enhancement of alternate, non-CFC, refrigerant/refrigerant mixtures utilizing the EHD technique was the subject of a three-year experimental investigation in a project funded by the U.S. Department of Energy. Detailed findings have been reported to the DOE project managers through Quarterly Progress Reports. The present Final Report represents a summary of the major findings in the project. It was demonstrated that the EHD technique is an effective heat transfer enhancement technology for applications in two-phase flow (in-tube and external boiling and condensation). It was found that under optimum electric field and heat transfer surface conditions the EHD-enhanced heat transfer coefficients are nearly an order of magnitude higher than their corresponding state of the art technologies. Moreover, it was found that EHD works better with enhanced tubes as compared to smooth tubes. For example it was found that microfin tubes respond more favorably to the EHD-enhanced boiling effect when compared to the smooth tubes. An increase of average heat transfer coefficient by seven fold was recorded by application of just 2 kV using a *helical* electrode. Similar order of enhancements were obtained with *cylindrical* electrode as well, but by employing much higher EHD voltage (≈ 15 kV). The power consumption in both cases were negligibly small, in the order of less than one Watt. Heat transfer experiments in the absence of electrodes were also performed and compared with data available in published literature, where possible.

2. WORK SCOPE

The objective of the present research is to quantify the applicability and limitations of the EHD technique for boiling and condensation heat transfer enhancement of alternate refrigerants/refrigerant mixtures. This will include experiments that will address two distinct issues: (1) determine the extent by which the EHD technique can enhance the heat transfer characteristics of alternate refrigerants/refrigerant mixtures, (2) determine any long term effects the EHD technique may have on the characteristics of the refrigerant/system, including aspects such as additional fouling effects, potential degradation of the refrigerant; reliability of the electrode system for continuous operation; and the economics of the technique in terms of added capital and operational cost. With these objectives in mind, the test rigs have been designed and fabricated such that both low and high pressure refrigerants can be tested. Of particular interest is to identify optimum conditions for maximum enhancement with minimum additional electrical power consumption. The overall goal of the project is to collect basic and application oriented data that will lead to development of EHD-enhanced, high-performance, advanced evaporators and condensers.

3. BASIC PRINCIPLES OF THE EHD TECHNIQUE FOR HEAT TRANSFER ENHANCEMENT APPLICATIONS

The electrohydrodynamic (EHD) technique utilizes the effect of electrically induced secondary motions, when a high-voltage, low current electric field is coupled with the flow field in a dielectric fluid medium. For boiling applications the net effect is a substantial increase in the boiling dynamics near the heat transfer surface, leading to heat transfer coefficients that are often an order of magnitude higher than those achieved with any of the conventional passive or active enhancement techniques.

The driving force behind electrically-induced secondary motions that are responsible for the EHD heat transfer enhancement is the electric (or EHD) body force. For a dielectric fluid of permittivity ϵ , density ρ , temperature T , under an applied electric field strength of $\vec{E}$, this body force can be expressed as [1]-[3] :

$$\vec{f}_e = \rho_c \vec{E} - \frac{1}{2} E^2 \nabla \epsilon + \frac{1}{2} \nabla [E^2 \rho \left(\frac{\partial \epsilon}{\partial \rho} \right)_T] \quad (1)$$

where ρ_c is the electric field space charge density in the fluid and ρ is the mass density. The first component of the right hand side of equation (1) is the *Coulomb force* resulting from the net free charges within the fluid. For the phase processes, the EHD current discharge is insignificant and thus, the first term does not contribute strongly to the EHD body force. However, the second term called the *dielectrophoretic* and the third term called the *electrostrictive force* contribute substantially to the EHD body force in such processes. The dielectrophoretic force is produced by the spatial change of dielectric permittivity ϵ and the electrostrictive force is associated with the variation of the dielectric in the presence of electric field. The dielectric constant κ , ($\epsilon = \kappa \epsilon_0$) can be approximately expressed in terms of the density of fluid ρ , using the Clausius-Mossotti law [4],

$$\frac{\kappa - 1}{\kappa + 2} = C \cdot \rho \quad (2)$$

where C is a constant which depends on the type of dielectric. Utilizing the above, it can be deduced that

$$\rho \frac{\partial \epsilon}{\partial \rho} = \frac{\epsilon_o}{3} (\kappa - 1) (\kappa + 2) \quad (3)$$

Substituting (3) in (1), yields the force exerted by the electric field in a dielectric fluid medium,

$$\bar{f}_e = \rho_c \bar{E} - \frac{\epsilon_o}{2} E^2 \nabla \kappa + \frac{\epsilon_o}{6} \nabla [E^2 (\kappa - 1) (\kappa + 2)] \quad (4)$$

Details of the EHD mechanism, its applicability and limitations can be found elsewhere (e.g. Ohadi [13] and Yabe [15]).

4. MAJOR ACCOMPLISHMENTS

Major accomplishments for this phase of the project include: (i) In-tube and external boiling base case experiments (no electrode present) utilizing microfin tubes; (ii) High voltage/electrode optimization studies; (iii) Parametric studies on EHD-enhanced heat transfer as a function of mass flux, quality, heat flux, electrode geometry, and the applied voltage; (iv) Studies on issues related to the EHD current stability; and (v) Comparison of the "no-electrode" base case data with those available in the literature.

5. EXPERIMENTAL APPARATUS

Overall description of the apparatus was given in our quarterly reports [5]-[7]. With some modifications in the test section and the heating method, the apparatus accommodated conducting

experiments with enhanced microfin tubes and various electrode designs. The basic test loops remained the same, except in the current experiments instead of a water heating loop for the test-section, resistive heating is employed. A constant heat flux condition associated with electrical heating helps in providing a better insight into the physics of EHD-enhanced boiling. The test-section length was also reduced from 1219 mm (4 ft) to 305 mm (1 ft) so that accuracy of measurements can be increased. In the modified test section more thermocouples per unit length were installed to obtain high precision in wall temperature measurement. The modified test rig is shown in Figs. 1-2.

A complete description of the experimental apparatus was given in [1]. A brief description of the test section apparatus used in the present experiments is given in the following.

5.I The Test Rig Design

The overall schematic diagram of the test apparatus is shown in Fig. 1. 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 two distinct flow loops : (1) the refrigerant flow loop which provides refrigerant flow through the test section and (2) a heat pump loop, which condenses and subcools the refrigerant entering the test section.

5.II The Test Refrigerant Loop

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 $20 \text{ kg/m}^2.\text{s}$ to $400 \text{ kg/m}^2.\text{s}$ (Liquid Reynolds number between 1,200 to 24,000) and the quality at which the refrigerant enters the test section can be controlled by the electric preheater. The test section is a horizontally mounted, Wolverine, Turbo-DX, microfin tube with 12.7 mm (0.5") OD. The heat flux required for boiling the refrigerant is provided by an electrical heater wrapped around the test-section tube. The test section instrumentation allows to operate up to 35 kW/m^2 of heat flux. The exit end of the test

Figure 1 Schematic of the experimental apparatus

section has a sight glass of almost the same inside diameter as the test section in order to observe the

Figure 2 Schematic of test-section tube showing thermocouple locations

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.

5. III The Test-Section Design

An internally finned, turbo-DX, copper tube manufactured by Wolverine was used in the present experiments. The nominal inside diameter of this tube is 11.78 mm (0.46") which is 1.7 mm (0.7") higher than the smooth tube previously used. The ridge height is 0.3 mm (0.012"). The test tube section, where electrical heating was applied was 495mm (19.5 in.) long. But, the wall temperature measurements were only done in the middle 305 mm (12 in.) test-tube. This was done to minimize the axial heat losses through the test section. As shown in the fig. 2. the electrode was placed only in this middle portion of the test-section and all calculation for heat transfer coefficients and enhancements were based on this test length. The location of thermocouples and exact dimensions of the test-section are indicated in fig.2. Wall temperatures are measured using copper-constantan thermocouples directly soldered on the outside of the tube. Temperatures are measured at four axial locations at equal intervals of 76 mm (3 in.). At each axial location, two thermocouples are placed, one at top and one at the bottom of the tube.

5.IV The Electrode Design

Two electrode geometries are examined in the present study. Figure 3 shows the schematic of the electrodes design used. The first electrode design is a simple coaxial cylindrical type of 3 mm (0.118 in) diameter. The second electrode has a helical configuration with 9.8 mm helical diameter and 0.5mm wire thickness. These electrodes are supported inside the test section by teflon spacers. Positive potential is applied to the electrode through a modified automotive spark plug whereas the tube is grounded.

5.V The Heat Pump loop

The objective is to study the effect of EHD for all qualities and heat fluxes (5-30 kW/m²). For this, a total of 0.2 - 3.0 kW of heating capacity of preheater and heater is needed, which effectively means that same range of cooling capacity is required from the heat pump loop.

Figure 3 Test section tube and electrode arrangement

However, it is virtually impossible for a heat pump loop to continuously operate in such a large range of cooling capacity without over heating the compressor. Specially, while operating at lower heating capacities, cooling of compressor becomes critical. So two bypass loops viz., hot gas bypass and liquid bypass loop, along with an accumulator were added to the system. Thus depending on the heat flux and inlet quality of operation, first mass of refrigerant, R-12, flowing in the heat pump loop is altered, then cold liquid bypass, then hot gas bypass and finally the expansion valve is controlled to control the system pressure.

5.VI The Pressure Controller

It was difficult to maintain test section pressure by controlling the expansion valve manually. Therefore, a stepper motor was installed on the expansion valve and controlled through the computer. The driver circuit for the computer controlled stepper motor was given in [1]. The driver source code program, written in QuickBasic, reads the actual pressure through A/D card, compares with the set pressure, sends proportional sequencing of controlling bits through the computer printer port to the driver circuit which depending on the controlling bits rotates the motor in clockwise (closes expansion valve) or counter-clockwise (opens expansion valve) after drawing current from a 10 V D.C. power supply. This controller is able to control the pressure within $\pm 0.2\%$ of the set pressure.

5.VII The Data Acquisition

The data acquisition system consists of a personal computer, two multiplexer board each of 32 channel, an A/D card and two multimeters. The complete data acquisition source code is written in QuickBasic, automatically monitors 24 copper-constantan thermocouples, 2 pressure transducers, one mass flowmeter, preheater and test-section heater power. Data reduction calculations pertaining to the heat transfer coefficient and quality are also performed using the same computer. The output of data reduction is dumped in user defined file name and hardcopy can be obtained whenever desired.

6. EXPERIMENTAL PROCEDURE

A typical experimental run began by first turning on the refrigerant pump and fixing the mass flux at the desired level. Next, the test-section heater was turned on and the heat flux was adjusted at a predetermined value. Then the heat pump loop was switched on and the mass flow rate of the refrigerant flowing in this loop, R-12, was adjusted such that the condensing capacity of heat pump loop became more or less equal to the heating capacity of test-section heater. The preheater was then switched on and set at a predetermined level such that a desired test-section quality was obtained. The system was then allowed to reach the steady state conditions (usually 2 hrs) before the data taking process began. Achieving "steady state" conditions was determined by requiring less than 1% fluctuations in system pressure, mass flux, inlet quality and temperatures in the test section loop. Once the steady state condition was reached the computer-assisted data acquisition system recorded measurements of temperature, pressure, and mass flow rates at various points around the experimental loop.

7. DATA REDUCTION

Calculation of the heat transfer coefficient is based on the following defining equation:

$$h = \frac{Q}{A (T_w - T_{sat})} \quad (5)$$

in which Q is the heat transferred to the test section, T_w is the tube wall temperature and is taken as the arithmetic average of eight thermocouples installed on the test-section wall. T_{sat} is the saturation temperature of the test refrigerant corresponding to the average test section pressure. Strictly speaking, T_w should be the average inner tube wall temperature. However, realizing the very high thermal conductivity of copper, it is often a well justified approximation to take it equal to the outside wall temperature.

The quality calculation for the refrigerant at different locations in the test-section is done by utilizing energy balance equations. The quality after the preheater (state 2, fig. 4) is calculated by using the following equations,

$$e_1 = f (T_1 , P) \quad (6)$$

$$e_2 = e_1 + \frac{Q_{ph}}{m_r} \quad (8)$$

$$X_2 = f (P , e_2) \quad (7)$$

Where e_1 and e_2 are the specific enthalpy of the refrigerant at the preheater inlet and exit, respectively. Having T_1 and P measured, the specific enthalpy can easily be obtained from the property tables. Q_{ph} is the heat transferred to the refrigerant by the preheater and m_r is the mass flow rate of the refrigerant. Here it is assumed that the pressure drop across the preheater is negligible. Similarly, the quality of the refrigerant at the test section entrance, X_3 (state 3, fig. 4) and exit, X_4 (state 4, fig. 4) can be obtained once Q_e , the heat input to the test section, is known. The average test-section quality, X_{avg} , is calculated by taking mean of X_3 and X_4 .

For a fixed operating condition such as heat flux, average test-section quality and refrigerant mass flux, two sets of heat transfer coefficients can be defined. The one is h_0 and corresponds to the

Figure 4 Schematic of test apparatus showing state points

case when the electrode is present in the test section tube but is not charged with the high voltage. The second one is h , when the high voltage is applied. Therefore, **heat transfer enhancement factor** due to the application of high voltage can be defined as the ratio of h and h_0 . Assuming a constant thermal conductivity of the refrigerant in the operating range, the heat transfer ratio with and without the EHD effect, (high voltage effect) reduces to a ratio of the corresponding Nusselt numbers

$$\frac{Nu}{Nu_0} = \frac{h}{h_0} \quad (9)$$

Where Nu and Nu_0 represent the Nusselt number for the high voltage and base case (zero electric field) conditions, respectively.

8. UNCERTAINTY ANALYSIS

All the uncertainty calculations were done based on root mean sum squares method. The maximum uncertainty in heat flux measurement was $\pm 4.6\%$ which occurred at the lowest heat flux (5 kW/m^2). The maximum uncertainty in heat transfer coefficient at zero electric field was $\pm 4.8\%$ and for the electric field was $\pm 10.3\%$. The maximum uncertainty in refrigerant quality calculation was found to be $\pm 4.7\%$. The uncertainty in heat transfer enhancement factor (Nusselt number ratio given by Eq. 5) was $\pm 11.4\%$.

9. RESULTS AND DISCUSSION

9.1 High Voltage Optimization Studies

Cylindrical Electrode. A coaxial cylindrical electrode was placed in the center of the microfin test-section tube using teflon spacers as shown in Fig.3. The gap between the heat transfer surface (grounded surface) and the electrode was approximately 4.3 mm. Figure 5 depicts the effect of varying the high voltage on heat transfer enhancement for different test-section quality. As seen there, increase in the applied high voltage increases the heat transfer coefficient due to higher electrical field strength associated with the high voltage. As the quality is increased from subcooled to 30%, the enhancement decreases from 11 to 2.4 times. This is because at high qualities the effect of EHD forces becomes less dominant compared to the already vigorous boiling dynamics associated with high heat flux at the surface of the tube. Figure 6 shows the corresponding EHD power consumptions. The currents are lower for higher quality. This is because the electrical conductivity of vapor refrigerant is lower than the liquid refrigerant. The currents are very low upto 7.5 kV, but it increases by 5 times as the voltage is increased to 15 kV. But for same increase in voltage, the enhancement almost remains unaltered for all the qualities, except for subcooled condition, where higher voltage helps in improving the heat transfer performance. Therefore, we can conclude that for low qualities (sub-cooled boiling) an electrode voltage of 15 kV will be beneficial and at higher qualities, an electrode voltage of 7.5-10 kV will be more effective.

Helical Electrode. The gap between the heat transfer surface and the electrode in this case was 1 mm. Therefore, the highest applied (sparkover) voltage was much less as compared to the cylindrical electrode. The effect of varying high voltage on heat transfer enhancement factor and EHD power is shown in Figs. 7 and 8, respectively. Increase in high voltage increases the heat transfer coefficient up to 1.5-2 kV, but further increase in the voltage reduces the heat transfer

Figure 5 Variation of heat transfer enhancement with EHD voltage for cylindrical electrode

Figure 6 Variation of EHD current with EHD voltage for cylindrical electrode

Figure 7 Variation of enhancement factor with EHD voltage for helical electrode

Figure 8 Variation of EHD power with high voltage for helical electrode

coefficient. This suggest that with helical electrode there lies an optimum high voltage range for better in-tube boiling performance. The power consumption curve also indicates that it is better to apply voltage less than 2 kV. As the voltage is increased from 2 to 2.5 kV the enhancement factor decreases in most cases while the power consumption almost doubles. Consequently, it was decided to apply 2 kV while doing the parametric study with the helical electrode.

9.II Parametric Study with Helical Electrode

Quality Variation. Experiments for R-134a were conducted at 575 kPa (83.4 psi) pressure. The saturation temperature corresponding to this pressure is 20.15°C (68.3 °F). This operating condition was selected so that the heat loss to the surroundings was kept minimum.

Figures 9-12 show comparisons of the average heat transfer coefficients with varying test-section quality at different heat flux levels and 50 kg/m².s ($Re_L \approx 3000$) mass flux. In these figures, three cases are presented : (i) no electrode in the flow field (NE); (ii) electrode present but not charged with the high voltage field; and (3) electrode present and charged with high voltage source (here 2 kV). With increase in quality heat transfer coefficient decreases for 0 kV and no-electrode case. The EHD effect is seen to substantially increase the heat transfer coefficients in all quality ranges. Figure 13 shows the heat transfer enhancement for varying quality. A number of observations can be made. Maximum enhancement of $\approx 700\%$ is obtained on application of 2 kV and this occurs at lowest heat flux. Increase in heat flux reduces the effect of EHD due to higher turbulence associated with high heat fluxes. Higher order of enhancements were obtained using microfin tube as compared to smooth tube. Recall that, with smooth tube the highest enhancement obtained was 2.2 fold, whereas, with microfin tubes it is around 7 fold. Also, with smooth tube, application of EHD voltage used to reduce the heat transfer coefficients at qualities more than 30%, due to liquid extraction phenomenon [4], whereas here it helps to improve heat transfer coefficients in all quality ranges. Due to the presence of sharp fins in the test tube, the highest electric field strength is near the heat transfer surface as compared to smooth tube where highest electric field was near the electrode. Therefore, no liquid extraction phenomenon occurs with enhanced tube.

Figure 9 Heat transfer coefficient versus average test section quality at 5 kW/m² and 50 kg/m².s

Figure 10 Heat transfer coefficient versus average test section quality at 10 kW/m² and 50 kg/m².s

Figure 11 Heat transfer coefficient versus average test section quality at 20 kW/m² and 50 kg/m².s

Figure 12 Heat transfer coefficient versus average test section quality at 30 kW/m² and 50 kg/m².s

Figure 13 Enhancement factor versus average test section quality

Figure 14 EHD Power versus average test section quality at different heat flux levels

Another interesting observation in Fig. 13 is that for a given heat flux, there is always a peak enhancement when the quality is varied. For example, at 5 kW/m^2 heat flux, the enhancement is maximum at 10% quality, whereas at 10 kW/m^2 , the enhancement is maximum at around 30% quality. This can be explained with the aid of EHD body force, Eq. (1). The second term, called the dielectrophoretic force is much stronger in two phase processes because it arises due to spatial change in the dielectric constant of refrigerant. At the liquid-vapor phase this force is very high because of large difference in κ of vapor and liquid refrigerant. Typically vapor refrigerant has $\kappa \approx 1$ and liquid refrigerant has $\kappa \approx 3$. When the net interface area between vapor and liquid in the test-section is highest as the quality is varied, this force will be largest. Therefore, this peak enhancements possibly occurs due to the flow regime such that it creates high vapor-liquid interface.

Attention is next drawn to Fig. 14, where variation of EHD power with quality is depicted. With increase in quality, the EHD power goes down due to lower electrical conductivity associated with vapor refrigerant. The power consumption in all cases is almost negligible. In terms of test-section heating capacity, the maximum power consumption is less than 0.125%. In fact, in most cases it is around 0.08% of the evaporator heating capacity.

Mass Flux Variation. Figures 15-17 show the variation of heat transfer coefficients with mass flux at 30% test-section quality and different heat flux levels. The mass flux was varied from 50 to $150 \text{ kg/m}^2\cdot\text{s}$ (3000 to 9000 of Re_L). It can be observed that the change in the heat transfer coefficient because of the addition of the electrode is not very significant. In fact, at higher mass flux levels, the presence of electrode decreases the heat transfer coefficient. Although the electrode promotes turbulence levels of the flow and slightly increases the heat transfer coefficient, the presence of the electrode decreases the hydraulic diameter of the test section tube which in effect acts like a flow straightener, attempting to laminarize the flow and thus reducing the heat transfer coefficients at higher mass flux levels. Application of EHD voltage of 2 kV further improves the heat transfer coefficients.

Figure 18 shows the heat transfer enhancement factor for varying mass flux levels. Increase in mass flux in general decreases the enhancement levels in higher heat fluxes. But at lower heat flux the enhancement factor first decreases, then reaches a minimum followed by an increase. Again

Figure 15 Variation of heat transfer coefficient with mass flux at 10 kW/m²

Figure 16 Variation of heat transfer coefficient with mass flux at 20 kW/m²

Figure 17 Variation of heat transfer coefficient with mass flux at 30 kW/m²

Figure 18 Heat transfer enhancement versus mass flux at various heat fluxes

this may be due to varied flow regimes associated with different mass fluxes and hence different dielectrophoretic force. The effect of diverse dielectrophoretic is more pronounced in the lower heat flux than higher heat fluxes due to its less dominance over the strong boiling dynamics associated with higher heat flux. It can be also be observed from the figure that EHD helps in improving the heat transfer coefficient, mostly in lower mass flux and heat flux levels.

Attention is next drawn to power consumption results in Fig. 19. Realizing the currents are so small that their measurement is subject to high uncertainty, no clear trend is observed with mass flux. However, it can be clearly stated that the power consumption is negligibly small. A heat flux of 10 kW/m^2 corresponds to approximately 120 W of test-section heat and the maximum power consumption is 0.11 W in this case. Therefore, maximum EHD power is less than 0.09% of the evaporator capacity.

9. III Effect of EHD Current on the Heat Transfer Enhancement Factor.

The first term in the EHD body force (Eq. 1) is due to free charge present in the refrigerant. The higher the EHD current, the higher will be the free charge and hence the EHD force. When high voltage is applied, initially the currents are high and with time it goes down. Figure 20 shows a typical behaviour of EHD current with time. Fixed voltage of 15 kV was applied across the cylindrical electrode - microfin tube arrangement. The electrode-tube arrangement here exactly behaves like a charged capacitor with some leakage current (EHD current) through the electrodes. The time-constant (time required for current to reach 36.8% of initial value) depends on the electrode geometry, operating conditions and the refrigerant dielectric permittivity. In this case, the time constant is approximately one hour. The most important observation, however, is that the peak, initial current is still very small (less than a milliampere).

The effect of EHD current on heat transfer enhancement factor is shown in Fig. 21. Seven sets of the same data were taken when the EHD current was changing in the system. The results show that, enhancement factor is affected marginally due to EHD currents. However, the enhancement remains unaffected after certain current (in this case, less than 0.10 mA). In all cases

Figure 19 Variation of EHD power with mass flux

Figure 20 Variation in EHD current with time

Figure 21 Effect of EHD current on enhancement factor

the currents are less than a milliampere. Therefore, it is concluded that for systems operating continuously for long time periods the EHD current will be negligible and the heat transfer enhancement magnitude will remain high. But for a system operating in an intermittent mode, the heat transfer enhancements will be slightly higher, but the currents will still not be very high.

9. IV Comparison of Results for Cylindrical and Helical Electrode.

The spacing between the cylindrical electrode and the heat transfer surface is around 4.3 mm and that between helical electrode and heat transfer surface is 1 mm. Therefore, the sparkover voltage in case of cylindrical electrode is much higher (≈ 20 kV) as that of helical electrode (≈ 3 kV). Note, that the EHD force depends on the electric field strength and not on the absolute value of the applied voltage. For a parallel plate geometry, electric field is roughly given as the ratio of voltage to the electrode spacing (V/d). For example, a voltage of 15 kV across a parallel plate spaced at 4.3 mm apart will have the same electric field strength in between the plates as compared to the applied voltage of 3.48 kV across a 1 mm apart plates. The two electrode geometries viz. cylindrical and helical, studied here simulate two different situations, one with a high electrode gap and the other with a small electrode gap. However, both are capable of producing the same electric field strength near the heat transfer surface. But the EHD-induced boiling dynamics might be different in these two geometries. Let us now examine this.

Figures 22 and 23 shows the variation of enhancement factor with the electrode voltage for the two geometries at two different qualities. The maximum enhancement more or less remains the same for two geometries. The optimum enhancement obtained for 25% quality is around 5.0 for both the electrodes while at 75% quality the enhancement is around 3.0. But in case of helical electrode same magnitude of enhancements are obtained by applying just 2 kV. Figures 24-25 show the corresponding power consumptions. In low quality zones, the power consumption with helical electrode is much lower than that of the cylindrical electrode while at high quality it is the other way around. Although at high quality, the power consumption with helical electrode is higher than the cylindrical electrode, the absolute value of power consumption is still low (less than a watt).

Figure 22 Comparison of enhancement factor for two electrode geometry used at 25% quality

Figure 23 Comparison of enhancement factor for two electrode geometry at 75% quality

Figure 24 Comparison of EHD power at 25% quality

Figure 25 Comparison of EHD power at 75% quality

Therefore, it may be concluded that a helical electrode is more suitable for EHD-enhanced in-tube boiling. Moreover, with helical electrode, since the power consumption and the applied voltage are low, the size of power supply will also be smaller.

10. AUTHENTICITY OF THE REPORTED DATA

10. I Repeatability of Data.

Figures 26-27 show the repeatability in the heat transfer coefficient at 0 kV and 2 kV respectively. As it can be seen, zero kV data repeats within the uncertainty of $\pm 4.8\%$. Also, the data at 2 kV applied EHD voltage, repeats within the uncertainty of $\pm 10.3\%$. Figure 28 shows the repeatability in the heat transfer enhancement factor. The fluctuation in the data is within $\pm 11.4\%$ which is the maximum uncertainty in the data. Realizing the complexity of the problem at hand, this indicates that the data are reasonably reproducible.

10. II Comparison of Base Case Results with the Published Data.

The results for "no-electrode" present in the flow-field was compared with the data already available in the literature. The only data which was available to us and was applicable to our geometry and test conditions, for comparison of in-tube boiling of R-134a in a microfin tube was those of Eckels [10]. However, the experiments conducted by Eckels was at an evaporator temperature of 1°C - 2°C , whereas, in our experiments it was around 20.15°C . Figure 29 shows the effect of evaporator temperature on boiling heat transfer coefficient. It can be seen that the heat transfer coefficient is not much affected by the saturation temperature. Thus, although the operating temperature is different from that of Eckels [10] and the comparison may not be one to one, but it is still reasonable. Figure 30 shows the comparison of some of the data which have similar operating conditions. Our results matches with those obtained by Eckels.

Figure 26 Verifying repeatability of heat transfer coefficient data at 0 kV

Figure 27 Verifying repeatability of heat transfer coefficient data at 2 kV

Figure 28 Verifying repeatability of enhancement factor with helical electrode

Figure 29 Effect of saturation temperature on boiling heat transfer coefficient

Figure 30 Comparison of heat transfer coefficient with Eckels [10]

10. III Effect of EHD Current on Wall Thermocouples.

The wall temperatures used for heat transfer coefficient calculations were measured directly from the thermocouples mounted on the test-section wall. The positive high voltage is applied to the electrode and the test-section tube wall is grounded. Therefore, the path of the EHD current is through the grounded tube wall. There is a possibility that the this current may influence the thermocouple readings. Certain experiments were conducted to examine this. Alternately, voltages of 2.5 kV and 0 kV were applied and the two wall thermocouple readings were monitored. Figure 30(a) shows how the EHD current changes with the high voltage. If the current has any influence on the readings, one may expect the wall temperatures to also fluctuate with the voltage. However, Fig. 30(b) shows that the temperature always goes up for both the thermocouples regardless of the applied high voltage. This rise in temperature is, in fact, due to the increase in system pressure. When temperature for TC#23 increased from 20.12°C to 20.65°C the pressure increased from 593.9 kPa to 601.9 kPa. This exactly corresponds to the increase in saturation temperature due to change in pressure. Same is the case with the other thermocouple. Therefore, it is concluded that *EHD current has no influence on the readings of the thermocouples mounted on the test section tube wall.*

11. FUTURE PLANS

The planned tasks and their schedule for the next quarter are as follows:

October 15, 94 - November. 30, 94

Modify the present test loop for conducting the condensation experiments.

December 1, 94 - Jan 10, 95

Conduct EHD-coupled, condensation experiments with enhanced tube. Optimize electrodes to realize higher heat traansfer rates.

Jan 11, 95 - Jan 20, 95

(a)

(b)

(c)

Figure 31 (a)-(c) Verification of the fact that EHD current does not influence thermocouple readings

Prepare quarterly progress report no. 6 and submit to the DOE and Consortium members.

12. CONCLUSIONS

The report represents the conclusion of phase-I of the project on EHD-enhanced in-tube boiling of alternate refrigerants, sponsored by the U.S. Department of Energy. In this phase, two alternate non-CFC refrigerants (HCFC-123 and HCFC-134a) were tested. It was successfully demonstrated that both refrigerants respond remarkably well to the EHD-effect. The experiments were conducted using smooth and enhanced microfin tubes. It was found that microfin tube performs much better than the smooth tube due to the presence of sharp fins which creates high electric field inhomogeneity near the heat transfer surface. While the maximum enhancement obtained using smooth tube and HCFC-134a was around two fold, the enhancement obtained with microfin tube ranged from 5 to 7 fold. It was also successfully demonstrated that the EHD-technique works in a wide range of qualities, mass flux and heat flux levels. However, best results are obtained in the subcooled to low quality region and at low mass and heat flux levels. High voltage and electrode optimization studies were also performed. These studies led to much improved enhancement levels with negligibly low EHD power consumptions. The power consumptions in all cases were less than 0.1% of the evaporator heat transfer capacity.

13. ACKNOWLEDGMENTS

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14. COMPLIANCE WITH AGREEMENT

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

15. PRINCIPAL INVESTIGATOR EFFORT

Dr. Michael Ohadi was the Principal Investigator and Director for this project.

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NON-BOILING WORK

SECTION A: EHD Enhancement of Pool Boiling Heat Transfer of R-134a over a Single Tube and Tube Bundle Utilizing Enhanced Tubes

SECTION B : Electrode Design Studies for EHD Air-Side Enhancement in Compact Heat Exchangers

SECTION C : Applicability of the EHD-Technique in Highly Compact Heat Exchangers

SECTION A

EHD ENHANCEMENT OF POOL BOILING HEAT TRANSFER OF R-134a IN SINGLE TUBE AND TUBE BUNDLE

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A.1 MAJOR ACCOMPLISHMENTS

The applicability of the EHD technique for heat transfer enhancement of R-134a in a tube bundle configuration was reported in the last quarterly progress report [1]. In the current reporting period, several new electrode and heat transfer surface configurations were designed and tested. Results of the heat transfer enhancement and electric power consumption data for the various tested configurations are presented in this report.

A.2 EXPERIMENTAL APPARATUS

Detailed description of the experimental setup was given in the previous report [1]. A brief description will be given here. The schematic diagram of the setup is shown in Figure 1. It consists of a high-pressure boiling chamber, an air-cooled refrigeration unit, pressure control system and the test section.

Modifications had been made in order to conduct the tube bundle experiments at higher heat flux levels. These included the modification of the loop to accommodate the cooling capacity. The water cooling loop was replaced with the refrigerant provided from a separate refrigeration unit directly into the condensing tube. In addition, two liquid by-pass lines were added in the refrigeration loop. These lines provided means for controlling the pressure of the boiling chamber by adjusting the refrigerant flow rate. One of the by-pass lines is controlled manually and the other one is controlled by the pressure control system which consists of stepper motor, A/D card, and computer. The amount of refrigerant flow rate adjustment is evaluated by the computer, based on the derivation of the boiling chamber pressure from the pre-set value. The computer signal will then be sent to the stepper motor utilizing the A/D card, so that appropriate adjustment in the refrigerant flow rate can be made.

Since the research also involves designing and testing of various electrode and heat transfer surface configurations, several new test sections were made. The various tested surface geometries and their specifications are summarized in Table 1.

Figure 1 Schematic Diagram of the Tube Bundle Setup.

Figure 2 Circular mesh electrode configuration

Figure 3 Rectangular mesh electrode configuration

PROPRIETARY DESIGN

For additional information on this electrode design/ characteristics, please contact
Dr. Ohadi at Tel: (301)405-5263 or Fax: (301)405-2025.

Figure 4 Internal Wire-Fin (IWF) electrode

Table 1. Tested electrode and heat transfer surface configurations.

Heat Transfer Tube	Electrode
40 fpi tube bundle diameter = 19mm length = 76mm	Circular external mesh diameter = 22.5mm mesh size = 6.35mm×6.35mm thickness = 1mm
40 fpi tube bundle diameter = 19mm length = 76mm	Rectangular external mesh dimensions = 24mm×26mm mesh size = 6.35mm×6.35mm thickness = 1mm
Wolverine's experiment single tube diameter = 13mm length = 50mm	Circular external mesh diameter = 22.5mm mesh size = 6.35mm×6.35mm thickness = 1mm
Wolverine's experiment single tube diameter = 13mm length = 50mm	Internal wire electrode thickness = 0.38mm
26 fpi single tube diameter = 19mm length = 76mm	Internal wire electrode thickness = 0.25mm
modified 40 fpi single tube diameter = 19mm length = 46mm	Circular external mesh diameter = 22.5mm mesh size = 6.35mm×6.35mm thickness = 1mm

A.3 EXPERIMENTAL PROCEDURE

An experimental run begins by switching on the air-cooled refrigeration unit that serves the cooling needs of the condenser loop. After a few minutes, the heater's power supply is turned on and the heat flux of the heaters is set to the desired value by utilizing the variable transformer. The pressure of the system is checked continuously by using the pressure transducer to make sure that it is below the designated pressure limit. In order to maintain the desired operating pressure, the flow rate of refrigerant through the liquid by-pass line is controlled manually by adjusting the valve position. Once the system pressure has stabilized within a reasonable tolerance (about ± 1 psi), the computer and the stepper motor were turned on. From there on, the computer would take care of controlling the pressure inside the boiling chamber.

Next, the high voltage supplier was switched on and set at the pre-determined level. Due to the absorption characteristic of the refrigerant, the system required about 4 hours to reach the steady state conditions. The system was considered to be steady when the fluctuation of the system pressure and the high voltage current were less than 1%. Once the system became steady, the wall temperature of the seven heater tubes, the temperature of the vapor refrigerant and the temperature of the liquid refrigerant were measured.

A.4 UNCERTAINTY ANALYSIS

Evaluation of experimental uncertainty is based on the fact that a differential change of a function can be expressed in terms of the differential change of the dependent variables and the partial derivatives in respect to the corresponding dependent variables. In other words,

$$dF(x_i) = \sum_i \frac{\partial F}{\partial x_i} dx_i \quad (1)$$

However, in terms of root mean sum squares,

$$dF = \sqrt{\sum_i \left(\frac{\partial F}{\partial x_i} dx_i \right)^2} \quad (2)$$

Therefore, in the experiments, the uncertainty of heat transfer (dQ) is found from the following equation,

$$dQ = \sqrt{\left(\frac{2V}{R} dV \right)^2 + \left(\frac{V^2}{R^2} dR \right)^2} \quad (3)$$

Similarly, the uncertainty of the heat transfer coefficient (dh) is found to be,

$$dh = \sqrt{\left(\frac{1}{A(T - T_c)} dQ\right)^2 + \left(\frac{-Q}{A(T - T_c)^2} dT\right)^2 + \left(\frac{Q}{A(T - T_c)^2} dT_c\right)^2} \quad (4)$$

In the experiments, the uncertainty of the temperature measurement is about 0.1°C. Therefore, dT and dT_c are both equal to 0.1 °C. For the heat input, the uncertainty of measuring the heater voltage (dV) and resistance (dR) are 0.1 V and 0.1 Ω, respectively. Table 2 summarizes the range of the uncertainty of the heat transfer coefficient for different electrode and heating tube configurations.

Table 2. Uncertainty of heat transfer coefficient in various configurations.

Heating Tube and Electrode Configuration	Range of uncertainty of h
40 fpi tube bundle/ Circular mesh electrode	±4.71% to ±35.36%
40 fpi tube bundle/ Rectangular mesh electrode	±5.13% to ±38.25%
Wolverine's experimental single tube/ Circular mesh electrode	±2.39% to ±5.50%
Wolverine's experimental single tube/ Between-fins electrode	±2.09% to ±3.97%
26 fpi single tube/ Between-fins electrode	±3.52% to ±19.92%
40 fpi single tube/ Circular mesh electrode	±5.26% to ±29.19%
modified 40 fpi single tube/ Circular mesh electrode	±6.74% to ±44.20%

A.5 DATA REDUCTION

In general, about 20 data are collected for each experimental run. These data are then reduced to the parameters of interest, such as the heat transfer coefficient, enhancement ratio, and the EHD power consumption ratio. The equations involved in the data reduction will now be discussed.

With the measurement of heater voltage and heater resistance, the heat transfer to the refrigerant is calculated as,

$$Q = \frac{V^2}{R} \quad (5)$$

The EHD power consumption is computed as,

$$Q_{EHD} = \phi I \quad (6)$$

where ϕ and I are the applied electric field potential and current, respectively.

The average wall temperature of the enhanced tubes is taken as the arithmetic mean of n measurement locations.

$$T_{wall} = \frac{\sum_{i=1}^n T_i}{n} \quad (7)$$

Similarly, the pool temperature is taken as the average of the measured refrigerant vapor and liquid pool temperatures.

$$T_{pool} = \frac{T_{vapor} + T_{liquid}}{2} \quad (8)$$

The heat transfer coefficient h is then calculated as,

$$h = \frac{Q}{A(T_{wall} - T_{pool})} \quad (9)$$

where A is the total effective heat transfer surface area of N tubes and it is found to be,

$$A = N \pi D L \quad (10)$$

The enhancement ratio is subsequently calculated as,

$$\text{Enhancement Ratio} = \frac{Nu_{EHD}}{Nu_0} = \frac{h_{EHD}}{h_0} \quad (11)$$

where subscript '0' represents the case when high voltage is not applied to the existing electrode on or around the heat transfer surface.

Finally, the ratio of the EHD power consumption to the total heat transferred to the refrigerant is calculated as,

$$\frac{Q_{EHD}}{Q_{Total}} = \frac{Q_{EHD}}{Q + Q_{EHD}}$$

A.6 RESULTS AND DISCUSSIONS

A.6.1 Tube Bundle Experiments

(i) Circular mesh electrode

The result from the last report [1] indicated that EHD-Enhanced boiling heat transfer enhancement decreases as the heat flux increases from 8000 W/m² to 12000 W/m². As discussed there, at higher heat flux levels, the boiling process is sufficiently rigorous and the additional activities induced by the high-voltage electric field become less significant. In order to quantify the effect of heat flux over a sufficiently large range, heat flux levels up to 20000 W/m² were tested in the present experiments. The results in Figure 5 clearly demonstrate the effect of heat flux on heat transfer enhancement. At a heat flux of 15000 W/m², the maximum enhancement factor becomes 2.2 (compare to 5.1 at 8000 W/m²). However, at a heat flux of 20000 W/m², the enhancement factor is about 1.5. Variation of the heat transfer coefficient as a function of applied EHD voltage and heat flux is shown in Figure 6. Notice that, particularly in the higher heat flux level, the heat transfer coefficient increases initially and begins to decline after a certain applied high voltage level. It is suggested that the electric field, in one hand, helps to increase the boiling activities around the heat transfer surface. On the other hand, it suppresses the movement of bubbles released from the heat transfer surface on their way to the pool free surface. Accordingly, there exists an optimum applied

voltage potential beyond which additional voltage will reduce the enhancements.

(ii) Rectangular mesh electrode

From the previous experiments [1], it was concluded that the mesh electrode performance was better than the straight wire electrode. That appears to the result from the higher electric field strength near the heating surface and the higher electric field non-uniformity along the longitudinal direction. In the present experiments, a rectangular mesh electrode was tested (see figure 3). The idea was to introduce an additional non-uniform electric field distribution along the angular direction. The results are summarized in figures 7, 8, and 9. Figure 7 shows the enhancement factors at various applied EHD voltage and heat flux levels. The maximum enhancement factor is found to be 5.5 at a heat flux of 8000 W/m^2 with an EHD voltage of 14kV. As expected, the enhancement factor decreases as the heat flux increases. In general, the trends are very similar to the one with the circular mesh electrode. The comparison of the two mesh electrodes is shown in figure 8 and 9 for the two heat flux levels which were tested (8000 kW/m^2 and 20000 kW/m^2). It is interesting to notice that the difference in the heat transfer coefficients for the two electrodes is insignificant. The scatter of data especially at the low heat flux level are considered within the range of experimental uncertainty ($\pm 5\%$ for the highest heat flux and as much as $\pm 38\%$ at the lowest heat flux condition).

A.6.2 Single Tube Experiments

(i) Wolverine's experimental single tube

A special design experimental tube was provided by Wolverine to test its applicability for EHD heat transfer enhancement. Two different kinds of electrodes were tested, circular mesh electrode and the IWF electrode (see figure 4). The heat transfer coefficients at various values of applied EHD voltages and heat flux levels are shown in figure 10 and 11. Results reveal that this tube does not response favorable to the influence of high voltage electric field. The maximum enhancement factors are found to be 1.4 and 1.3 for the circular mesh and the IWF electrodes, respectively. The corresponding EHD power consumption is 8.8% and 4.1% of the total heat transfer rate. By comparing figure 10 and 11, it can be seen that the heat transfer coefficients behave slightly

different. For the case with IWF electrode, the heat transfer coefficient increases linearly as the EHD voltage increases. It is in contrast with the case of using circular mesh electrode where the increment of the heat transfer coefficient decreases as the EHD voltage increases. It indicates that, in the presence of electric field, the IWF electrode provides less resistance for the vapor bubbles released from the boiling surface.

(ii) The 26 fpi Single Tube

Experiments were also done with the Wolverine 26 fpi tube in order to explore the applicability of the IWF electrode. Figure 12 shows the heat transfer coefficient at various heat flux levels as a function of the applied EHD voltage. The maximum enhancement of 1.65 is obtained at a heat flux of 5000 W/m^2 and an EHD voltage of 0.25 kV . For all three tested heat flux levels, the results indicate a substantial drop in heat transfer coefficients beyond the EHD voltage of 0.25 kV . In particular, for the heat flux level of 30000 W/m^2 , the heat transfer coefficient becomes even less than the base case value. This suggests that the gap between the fins may be too small to accommodate departure of the many bubbles generated and a substantial amount of traffic is created when electric field is applied. As expected, the EHD power consumption is extremely low. For the conditions in which maximum enhancement factor was achieved, the EHD power consumption is only about 0.009 W and it is less than 0.04% of the total heat transfer rate.

(iii) Modified Wieland's 40 fpi single tube

It is realized that the heat transfer surface structure plays an important role on the EHD enhancement of boiling heat transfer. Attempt has been made to design a suitable heating surface for the EHD enhancement applications. In the experiments, the 40 fpi tube is modified and tested with a circular mesh electrode. Figure 13 shows the enhancement factors at various heat flux and EHD voltage levels. The maximum enhancement factor of nearly four fold is obtained at a heat flux of 8000 W/m^2 ($\approx 2550 \text{ Btu/hr-ft}^2$). When compared with the results of the original 40 fpi tube, it is seen that the maximum enhancement factor there is 2.85 (see figure 14). Figures 15(a) to (c) show comparison of heat transfer coefficient between the two heat transfer surfaces at three different heat flux levels. For the base case condition (no electric field applied but the electrode is present), the heat

transfer coefficient of the modified 40 fpi tube is about 35% to 45% higher than the original one. However, in the presence of EHD, the modified 40 fpi tube performs up to 78% better than the original 40 fpi tube. At the maximum enhancement condition, the EHD power consumption of the modified and the original tube is 2.5% and 2.8% of the heat transfer rate, respectively.

A.7 EXPERIMENTAL RESULTS WITH THE IWF ELECTRODE AND R-123

In order to explore the effectiveness of IWF electrode, single tube pool boiling experiments were performed on Wolverine 19 fpi tube utilizing the IWF electrode with R-123. Figures 16 and 17 show the heat transfer coefficient at two different heat flux levels (8500 W/m^2 and 16000 W/m^2 under the influence of EHD. More than 5 fold enhancement is obtained at the heat flux of 8500 W/m^2 ($\approx 2695 \text{ Btu/hr-ft}^2$). When the heat flux increases to 16000 W/m^2 , a three folds enhancement is achieved. The maximum EHD power consumption is found to be 0.44% of the test section total heat transfer rate. When compared to the results of previous experiment utilizing circular external mesh electrode on the Wolverine 19 fpi tube with R-123, both electrode configurations yield a comparable degree of enhancement [2]. For the mesh electrode, about 4.5 fold enhancement was reported under the same conditions. However, the EHD power consumption of the IWF electrode is considerably lower than the one of the circular mesh electrode.

A.8 CONCLUSIONS

The tube bundle experiments confirm the earlier findings that the EHD effect is most significant at lower heat flux levels. The enhancement factor increases from 1.5 to 5.1 when the heat flux decreases from 20000 W/m^2 to 8000 W/m^2 . In addition, the experiments suggest that the performance of the rectangular and the circular mesh electrodes is comparable to each other. The EHD power consumptions for the two electrode geometries are also comparable. (in the range of 2% to 5% of the total test section heat transfer rate, depending on the magnitude of the applied voltage).

The IWF electrode yields enhancements that are comparable to those of the mesh electrode in both R-134a and R123. However, the power consumption is significantly reduced. For the 26 fpi single tube experiments with R-134a, a negligible EHD power consumption of 0.04% of the total heat transfer rate was obtained. And for the 19 fpi single tube experiments, the maximum EHD power consumption was 0.44% of the total heat transfer rate. Therefore, the IWF electrode can provide enhancements comparable to those of the mesh electrode with much less power consumptions when used with the 19 fpi and 26 fpi tubes tested here.

Preliminary results indicate the modified 40 fpi tube (axial grooves cut along the test section tube length) performs better than the original 40fpi tube, especially in the presence of the electric field. In the absence of the EHD effect, the modified tube has 35% to 45% higher heat transfer coefficients than the original one. When EHD is applied, the modified tube has up to 80% higher heat transfer coefficients when compared to the original 40 fpi tube.

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Figure 5 Enhancement factor of the tube bundle with circular mesh electrode.

Figure 6 Heat transfer coefficient of the tube bundle with circular mesh electrode.

Figure 7 Enhancement factor of the tube bundle with rectangular mesh electrode.

Figure 8 Comparison of heat transfer coefficients at 8000 W/m^2

Figure 9 Comparison of heat transfer coefficients at 20000 W/m^2 .

Figure 10 Heat transfer coefficient versus EHD voltage for the Wolverine's Experiment Tube with mesh electrode.

Figure 11 Heat transfer coefficient versus EHD voltage for the Wolverine's Experimental Tube with IWF electrode.

Figure 12 Heat transfer coefficient versus EHD voltage for the Wieland's 26 fpi tube with IWF electrode.

Figure 13 Enhancement factor of the modified 40 fpi tube at various EHD voltages.

Figure 14 Enhancement factor of the original 40 fpi tube at various EHD voltages.

(a)

(b)

(c)

Figure 15 Comparisons between the modified and the original 40 fpi tube.

Figure 16 Heat transfer coefficient of the Wolverine 19 fpi Tube with IWF electrode at 8500 W/m^2 .

Figure 17 Heat transfer coefficient of the Wolverine 19 fpi Tube with IWF electrode at 16000 W/m².

SECTION B

ELECTRODE DESIGN STUDIES FOR EHD AIR-SIDE ENHANCEMENT IN COMPACT HEAT EXCHANGERS

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B.1 PROJECT SUMMARY

Preliminary experiments were conducted to evaluate the applicability of the EHD technique for air-side heat transfer enhancement in highly compact heat exchangers. Three sets of experiments were performed to evaluate performance of potential electrode configurations: 1). Effect of air flow rate on IV curve/spark-over voltage, 2). Effect of very small electrode gap size (1-8 millimeters) on IV-curve, and 3). Effect of electrode geometry with very small gap size on IV-curve. The results show no dependence of air flow rate (Reynolds number between 440 to 5400) on IV-curve as predicted by Nelson and Shaughnessy [1986]. It was found that a stable corona discharge could be maintained for a gap size as small as 1.5 millimeters. The results suggest that for small electrode gap sizes, large electrode frontal surface areas are needed to achieve the desired result.

B.2 EXPERIMENTAL APPARATUS

Two experimental apparatus were constructed to test the EHD heat transfer enhancement applicability in compact heat exchangers. When applying EHD, a number of factors make compact heat exchangers different than other types. Two main factors include very narrow gap distance between the heat exchanger surfaces and the high air velocities in certain applications, including those used in the refrigeration and air conditioning industry. It is well established that the main component of air-side EHD heat transfer enhancement system is the ability to sustain a stable corona discharge current in the flow field near the heat transfer surface. One objective of the preliminary experiments conducted here was to find the limits for establishing a stable corona discharge, without field breakdown, over very small gap sizes in a narrow channel with applications to compact heat exchangers.

Two experimental apparatus were then built to: 1). Test the effect of air velocity on corona discharge (see Fig. 1) and 2). Test the effect of electrode spacing and geometry on the IV-curve characteristics (see Fig. 2). Test rig #1 (see Fig. 3) consists of two flat aluminum plates, 4 inches long and 0.1575 inch thick, insulated in a Delvrin body with the possibility to change the gap distance between the parallel plates. To create nonuniformity of the electric field a thin piece of rubber was glued to one plate and a 40 gauge constantan wire was placed on the center of the rubber. This wire was then positively charged and the other plate grounded, hence a nonuniform electric field was

created between the wire and the parallel plate.

Test rig #2, consisted of a metal stand with a screw-type vertical height adjustment control with a numerical counter for a movable arm. The moveable arm held a rubber stopper. An aluminum plate was placed on the base of the stand and used as the ground. Both the stand and the aluminum plate were grounded. A small hole (1/8") was drilled in a rubber stopper to both hold and insulate the positively charged test electrode.

The test electrodes were mounted on a 6" long, 1/8" diameter copper tube (see Fig. #4). Needle electrode was made by fitting a small needle (radius of curvature of approximately 0.01 inch) inside of the copper tubing and crimping the tubing closed. A steel ball 0.987 millimeter in diameter was soldered to the one end of the copper tubing to form the Ball-tip electrode. A commercial razor-blade (radius of curvature of approximately 0.001 inch) was soldered in a vertical orientation to form the Razor-Blade electrode. The test electrodes were then passed through the stopper and attached to the high voltage power supply.

B.3 EXPERIMENTAL PROCEDURE

A brief outline of the experimental procedures will be given in this section. For the air-flow rate experiments: First, an air flow rate was chosen (air velocities ranged between 180-2200 ft/min were used). Next, the IV-curve data were collected by sweeping through the high voltage from zero to spark-over voltage for various gap distances between the wire and the parallel plate heat transfer surface. The gap distances were varied between 2 to 8 millimeters.

Gap distance experiments were conducted as follows. First, the electrode was placed in the rubber stopper and the movable arm was lowered so that the electrode touched the aluminum plate and the counter reading was recorded. The arm was then raised to a known height above the aluminum plate, measured by a vernier caliper, and the counter reading was recorded. A distance per counter digit was then calculated (typically 0.25 millimeters per digit). Starting at a gap distance of 0.5 millimeters, the IV-curves were constructed for different gap size. The counter reading was increased in increments of one digit and IV-curve data were again taken in each case. This was repeated until an 8 millimeter gap was reached. These steps were repeated for the four electrode geometries tested.

B.4 RESULTS AND DISCUSSION

The results from the effect of air velocity on corona discharge show that for the range of air velocities tested, there was little effect on the IV-curves. This result agrees with the prediction given in the theoretical study by Nelson and Shaughnessy [1986] who state that the migration of ions from the positive electrode to the ground is not affected by the air flow velocities.

Gap distance testing of the possible electrode geometries provided some interesting results, suggesting encouraging signs for air-side EHD enhancement of heat transfer is possible in compact heat exchangers. Figure 5 shows the spark-over voltage versus gap distance for the four electrodes tested. This information is useful since large corona discharge currents inherently requires large voltages. Figure 6 shows the corona current as a function of gap size for the tested electrode geometries. It is clear that the Razor-Blade electrode provides both the smallest gap distance for stable corona discharge and the highest overall corona current. From this figure, this question arose, "Realizing the Razor-Blade electrode has the largest surface area, is this responsible for the high corona currents?". To address this question, the Razor-Blade electrode was modified by decreasing its frontal area by 1/3. In figs. 5 and 6 this modified electrode is called Small Razor-Blade electrode. The vast deference between the performance of the two Razor-Blade electrodes tested show that large frontal surface area electrodes are required to provided stable corona discharge for small gap distances in compact heat exchangers.

B.5 CONCLUSION

Preliminary results show that the EHD technique has a suitable potential for air-side heat transfer enhancement in compact heat exchangers. The results show a direct correspondence between frontal electrode surface area and the electrode performance. Increased electrode frontal area yields a decrease in the gap distance and an increase in the corona discharge current.

B.6 REFERENCES

- Nelson, D.A., Shaughnessy, E.J.; "Electric field effects on natural convection in enclosures"; J. Heat Transfer, V.108, pp.749-754, (1986).

Two Theoretical/Experimental Set-ups

Figure #1. Set-up to Test Effect of Air Velocity on I-V Curve

Figure #2. Set-up to Test Effect of Electrode Spacing and Electrode Geometry

Figure 3

B6

ITEM	PART NO	NAME	QTY/ASSEMBLY	FABRICATE	SOURCE
13			8		
12		NUT 3/8	8		
11		SS. THREADING ROD 3/8x24x12	4		
10		FITTING	2		
9	FVS. 01. 00. 09	GLASS	2		
8	FVS. 01. 00. 08	GASKET	1	--	
7	FVS. 01. 00. 07	GASKET	1	--	
6	FVS. 01. 00. 06	INSULATOR	2	--	
5	FVS. 01. 00. 05	TEFLOON RING	4	--	
4	FVS. 01. 00. 04	HOLDER PLATE	2	--	
3	FVS. 01. 00. 03	HEAT TRANSFER PLATE	2	--	
2	FVS. 01. 00. 02	FRAME	2	--	
1	FVS. 01. 00. 01	BODY	1	--	

UNIVERSITY OF MARYLAND
HEAT TRANSFER LAB
S. DESSIAI TOWN DATE 3.27.94
FLOW VISUALISATION SECTION
FILE NO. 22

Razor-Blade

Ball-tip

Small Razor-Blade

Needle

Figure 4 Various electrode configurations

Figure 5 Gap length versus sparkover voltage

Figure 6 Gap length versus corona current

SECTION C

APPLICABILITY OF THE EHD TECHNIQUE IN HIGHLY COMPACT HEAT EXCHANGERS

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C.1 PROJECT SUMMARY

The objective of the this project was to quantify the applicability/feasibility of the EHD augmentation technique for heat transfer enhancement in highly compact heat exchangers.

The experiments were designed to address three distinct issues: (i) determine the extent by which the EHD technique can be effective in geometries of significance to in highly compact heat exchangers (ii) determine any long term effects such as fouling potential, reliability of the electrode system for continuous operation, and economic consideration in terms of additional capital and operational costs (iii) quantify the economic feasibility .

C.2 MAJOR ACCOMPLISHMENTS

During the current reporting period perliminary experiments with R-134a as the test refrigerant with/without presence of the electric field were performed in the test apparatus that will be descibed in the following .

C.3 EXPERIMENTAL APPARATUS

A brief description of the experimental apparatus is given in the following . As shown in Fig.1(a) the setup consisted of three different loops:

C.2.1 The Boiling Loop

Boiling loop consisted of the preheater, heater, and the test section tube. The preheater part was a 21 in.(53.3 cm) microfin DX tube of ID = 0.44 in.(11.3 mm) and OD = 0.5 in.(12.7 mm). To be able to provide different inlet quality at the test section inlet, 18 in.(45.7 cm) of the tube was heated by using fiberglass insulated nichron wire. To decrease axial conduction effects, the preheater and heater were connected by a stainless steel pipe. The same length of pipe was used for the heater.

Figure 1 (a)-(b) Schematic diagram of the (a) test apparatus and (b) test section

For heater 16.5 in. (41.91 cm) is heated by fiberglass insulated nichron wire. The test section, as shown in fig.1(b), is 4.5 in. (11.43 cm) of the heated length. The boiling loop is designed in such a way that it can be tilted to any desired angle between horizontal to 90° (vertical) position. At the beginning of the test section pipe a spark plug was mounted for applying the electric field. The electric field was provided by a DC power supply (0 - 30 kV, 0 - 3 mA). The field was applied through a cylindrical electrode which was mounted in the pipe and the gap between electrode surface and the wall surface was adjusted by teflon insulator pinned around the electrode. After the heater, refrigerant enters a compact shell and tube condenser. The capacity of the condenser was controlled by a valve (tap) which could be adjusted manually as well as by a stepper motor. The flow rate adjustment helped in maintaining the desired pressure in the system. The cooled refrigerant after the condenser entered a magnetic driven gear (lubrication free) pump. The flow rate was changed by varying the voltage supply to the pump. Then refrigerant was pumped through the mass flow meter sensor (Coriolis type made by ABB K-Flow Inc.) equipped with a transmitter that indicated the mass

flow through a LCD screen. The refrigerant after passing through the control valve entered the preheater.

C.2.2 The Cooling Loops

Two cooling loops were used for this setup. One cooling loop was used to remove the heat from vapor refrigerant in the heater section. The cold water for condenser was provided by Neslab-HX-500 water chiller (12 kW capacity at $\Delta T=5^{\circ}\text{C}$), (manufactured by Neslab Instrument Inc., N.H). The capacity of the condenser could be varied by setting the chiller output water temperature and also by the water flow rate. The second cooling loop was a vapor compression which was used to subcool the refrigerant before entering the pump and this also helped in damping the fluctuations in the flow rate. The capacity of this cooling loop was approximately 2 KW and consisted of the basic components of a conventional vapor compression system. The electrical equipments consisted of a power supply (240 Volts - 1.7 kW) which provided electrical heating for the preheater and the heater. The power input to the heater or the preheater was through measurement of the voltage and current.

C.3 MEASUREMENTS

For calculation of the heat transfer the heat given to the test section and temperature difference between wall and liquid were needed. The wall temperatures were measured by four thermocouples located at five equally spaced axial locations. At each location thermocouples were installed circumferentially 90° apart. The soldering point of thermocouple to the tube wall was less than 1 mm in diameter. A temperature indicator (model Dp-41 manufactured by Omega Eng. Inc., CT) was used for temperature readings. The voltage and currents of heater and preheater were measured by a general purpose multimeter. In order to keep the pressure to the desired saturation pressure, the pressure was measured by a 0-100 psi pressure transducer (manufactured by Setra system, Inc., MA).

C.4 EXPERIMENTAL PROCEDURE

A typical experimental run started with first turning the refrigerant pump on and setting the mass flux or mass flow rate at a desired value. Next, the preheater or the heater, depending on the condition of the conducted specific test, were turned on and adjusted at the desired heat flux. Then the chiller was turned on and the water flow rate was controlled such that the cooling capacity of the cooling loop and subcooling loop became approximately equal to the heat given by the heater or the preheater. The system was allowed to reach the steady state conditions which normally took two hours before any data taking was attempted. The Steady state condition was determined when fluctuation in pressure, or the mass flux were less than 3%.

C.5 DATA REDUCTION

Heat transfer coefficient was calculated using the defining equation

$$h = \frac{Q}{A [T_{w,i} - T_{sat}]} \quad (1)$$

in which Q is the heat provided to the test section (by the heater), $T_{w,i}$ is the average inner wall test section tube temperature and T_{sat} is the saturation temperature of the refrigerant at a specified pressure. The quantity Q was measured by the relation

$$Q = \frac{V^2}{R} \quad (2)$$

in which V is the voltage and R is the resistance of the heaterwires. The inside tube wall temperature $T_{w,i}$ was calculated by the following equation assuming one dimensional heat conduction from outside

of the tube wall to the inside,

$$T_{w,i} = T_{w,o} - \frac{Q \ln\left(\frac{R_o}{R_i}\right)}{2\pi KL} \quad (3)$$

where $T_{w,o}$ is the average of four thermocouples at a given station. Since the ratio (R_o/R_i) is very small and the conductivity of the cooper is very high the $T_{w,i} - T_{w,o}$ is often a negligible value.

The inlet quality of the refrigerant to the test section, x_{ap} , was calculated by utilizing the energy balance equation on the test section and preheater as

$$e_{bp} = f(T_{bp}, p) \quad (4)$$

$$e_{ap} = e_{bp} + \frac{Q_{ph}}{m} \quad (5)$$

$$X_{ap} = f(p, e_{ap}) \quad (6)$$

where e_{bp} and e_{ap} are the specific enthalpy of refrigerant before and after preheater, respectively. e_{bp} can be found from property tables for the refrigerant and e_{ap} can be found from equation 5.

By assuming constant thermal conductivity, the heat transfer ratio with and without the EHD effect reduces to the corresponding Nusselt number ratio,

$$\frac{Nu}{Nu_o} = \frac{h}{h_o} \quad (7)$$

where h_o , Nu_o are heat transfer coefficients for the base case (Electrode present but no EHD effect

applied) and h and Nu are the heat transfer coefficient and Nusselt number in presence of high voltage, respectively.

The ratio of EHD power consumption to the total heat transferred to the test section can be calculated as,

$$\frac{Q_{EHD}}{Q_{Total}} = \frac{Q_{EHD}}{Q + Q_{EHD}} \quad (8)$$

C.6 UNCERTAINTY ANALYSIS

The uncertainty analysis is calculated based on the idea that a differential variation of a function of many variables can be calculated in terms of sum of the root mean square differential variations of its variables and partial derivatives with respect to the corresponding variables as

$$dF = \sqrt{\sum_{i=1}^n \left(\frac{\partial F}{\partial x_i} dx_i \right)^2} \quad (9)$$

since $Q = V^2/R$, the uncertainty of heat flux can be calculated by the following equation

$$dQ = \sqrt{\left(\frac{2V}{R} dV \right)^2 + \left(\frac{V^2}{R^2} dR \right)^2} \quad (10)$$

and similarly, the uncertainty for heat flux can be found by

$$\Delta(HF) = \sqrt{\left(\frac{dQ}{A} \right)^2 + \left(\frac{Q \Delta A}{A^2} \right)^2} \quad (11)$$

in which HF is the heat flux and Q is the heat rate provided to the test section. The average uncertainty of the tube wall temperature represents uncertainty in the average of 12 point temperature measurements. Therefore, the uncertainty can be calculated as

$$\Delta T_w = \frac{\Delta T_{\text{meas}}}{\sqrt{12}} \quad (12)$$

The uncertainty for heat transfer coefficient then can be found by the following equation

$$dh = \sqrt{\left(\frac{\Delta(HF)}{(T_w - T_{\text{sat}})} \right)^2 + \left(\frac{-(HF)}{(T_w - T_{\text{sat}})^2} \Delta T_w \right)^2 + \left(\frac{(HF)}{(T_w - T_{\text{sat}})^2} \Delta T_{\text{sat}} \right)^2} \quad (13)$$

In the experiments, the uncertainty of temperature indicator is 0.1 °C. So by substituting $\Delta T_{\text{meas}} = 0.1^\circ\text{C}$ in equation 11 that results in $\Delta T_w = 0.03^\circ\text{C}$. ΔT_{sat} can be calculated from variation of pressure which by some calculation resulted in as 0.06 °C. For the voltage and resistor the uncertainty of multimeter (dV and dR) is 0.1 V and 0.1 Ω, respectively. Therefore the maximum uncertainty of Q was found to be 0.51% which is for the case when the lowest heat flux (5000 W/m²) is applied. The uncertainty of heat transfer coefficient (h) was found to be in the range of 2.7% to 30% depending on the heat flux level. It should be noted that the highest uncertainty only occurs in the lowest temperature difference and also lowest heat flux case and minimum uncertainty occurs for the case in which both heat flux and temperature difference are high.

C.7 RESULTS

Preliminary results for flow boiling heat transfer enhancement of R-134a in highly compact configuration are encouraging. However, the data are preliminary subject to verification. Complete results should be available in the next project report.

EHD ENHANCEMENT OF PHASE CHANGE HEAT AND MASS TRANSFER IN REFRIGERATION EQUIPMENT

EHD Control of Frost in Refrigeration Equipment

Final Report Part II 1996 - 1999

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Nomenclature

AT	Air temperature [$^{\circ}\text{C}$]
AV	Applied voltage [kV]
BC	Base case
d	Hydraulic diameter [m]
D	Duration [hr]
EHD	Electrohydrodynamic
F	Frequency [Hz]
I	Electric current [A]
m_f	Mass of frost under an electric field [g]
m_o	Mass of frost not under an electric field [g]
p_g	Saturation pressure of vapor [Pa]
p_v	Partial pressure of vapor [Pa]
Q_{EHD}	EHD power consumption [W]
R	Percent mass reduction [%]
Re	Reynolds number, $\text{Re} = \rho v d / \mu$
RH	Relative humidity, $\text{RH} = p_v / p_g$
ST	Surface temperature [$^{\circ}\text{C}$]
V	Applied electric field potential [V]

Greek symbols

μ	Viscosity, Ns/m^2
v	Velocity, m/s
ρ	Density, kg/m^3

1. Executive Summary

The project's goal was to address the feasibility of frost control by the EHD technique for operating conditions and geometries of significance to refrigeration applications. The tasks were comprehensive and required a careful and systematic planning of the experiments throughout the project. The specific objectives of the proposed experimental investigation were:

- 1) Demonstrate by experiment the feasibility of the EHD technique for control of frost on a cold surface under operating conditions of direct significance to refrigeration applications.
- 2) Provide an experimental database that can quantify the role of various controlling parameters in frost control by using the EHD technique. Such parameters include electrode design/geometry, applied electric field voltage, heat transfer surface temperature, air temperature, air humidity, and duration of the experiment.
- 3) Where possible provide data to allow direct comparison of frost control performance (i.e., reduced frost mass accumulation) and the electrode electrical power consumption for the range of parameters of interest to refrigeration industry.

In order to achieve the aforementioned objectives, a series of experiments were conducted throughout the duration of the project. These experiments were selected, such that quantification of the effect of various parameters (such as surface orientation, electrode design, frosting surface temperature, air humidity content, air stream temperature, applied voltage magnitude, applied voltage frequency, and Reynolds number) could be performed. The parameters and their test ranges are listed in Table 1. The quantities identified in **bold** font represent the **base-case parameters**. The base case establishes a reference against which other parametric variations can be compared.

Table 1: Experimental parameters and their test range

PARAMETER	RANGE
Surface Orientation	Vertical orientation
Electrode Design	Two designs to be tested
Frosting Surface Temperature	-30°C
Relative Humidity	90%
Duration	1 hr, 2 hr , 3 hr, 4 hr
Air Temperature	0°C
Applied Voltage, DC	0kV , 5kV, 10kV, 15kV, 20kV, 25kV, 30kV, 35kV, 40kV

As stated in the project statement, experiments were conducted for several electrode designs, for the vertical orientation of the surface under base case parametric conditions (Table 2).

Table 2: Experiments to select electrode design

Test	Orientation	ST	RH	D	AT	AV	Electrode Design
1	Vertical	BC	BC	BC	BC	BC	Design 1
2	Vertical	BC	BC	BC	BC	BC	Design 2
Base Case (BC): Surface Temperature (ST), -30°C, Relative Humidity (RH), 85%, Duration (D), 2 hr, Air Temperature (AT), 0°C.							

In addition to the tasks outlined in the above work statement, the following additional tasks were accomplished to gain a better understanding of the phenomena:

- Frost growth visualization experiments using state of the art video and high-speed photography were performed.
- Software development for frost contour tracking, and measurement of frost crystal height as it is affected by the EHD field.

2. Introduction and Background

When humid air passes over a surface with a temperature below both the dew point temperature of the passing air and 0°C , frost forms on the plate's surface. Frost formation is a complicated transient process in which both heat and mass transfer occur simultaneously. In many engineering applications, frost is undesirable. For example, in refrigeration applications, frost contributes to the heat transfer resistance and reduces the rate of heat transfer. Frost may also reduce the flow cross-section and result in a lower mass flow rate of air and increased pressure drop in the system. Controlling and reducing both the frost mass and frost height on the heat transfer cold surface can significantly improve the system's performance.

In order to develop an effective mechanism to control frost growth, it is first necessary to understand how the frost crystals form. Therefore, this section of the report will present the fundamentals of frost crystal growth with and without an applied electric field. Next, a brief overview of the electrohydrodynamic (EHD) technique is applied to the present case to establish the relevant theoretical background.

2.1 The Frost Growth Process in the Absence of an Electric Field

Frost growth is affected by a large number of independent variables, including temperature of the frosting surface, air temperature, humidity of the air, location of frost on the frosting surface, airflow, and time duration. Frost growth has been modeled without electric fields using multiple governing equations, such as the energy equation, the ice-phase continuity equation, the gas diffusion equation, and thermodynamic relationships (Tao et al., 1993), which are coupled equations. There is no closed-form solution that relates the independent variables to frost mass accumulation.

A widely recognized study on the frost formation phenomenon was performed by Hayashi et al. (1977). The frost formation process was investigated by photographic observation and classified according to the structure of frost crystals. The process was divided into three periods: (1) *a crystal growth period*, (2) *a frost layer growth period*, and (3) *a frost layer full-growth period*. In the *crystal growth period*, a thin frost layer is formed on the cold surface, on which frost crystals appear and grow linearly in the direction perpendicular to the frosting surface. The layer is inhomogeneous. During the

frost layer growth period, the frost crystals generate branches and interact with other crystals, resulting in a meshed structure. In the *frost layer full-growth period*, alternate thickening and densification characterizes the growth of frost.

Figure 2.1 depicts the first two periods of frost formation as classified by Hayashi et al. (1977) for various environmental conditions. As explained above, during the crystal growth period the crystals grow linearly, as shown in the left column of figures. During the frost layer growth period, the crystals generate branches, as shown in the right column of figures. The environmental conditions corresponding to the crystal type are presented in Figure 2.2.

FROST FORMATION TYPE		CRYSTAL GROWTH PERIOD		FROST LAYER GROWTH PERIOD		
A	A-I
	A-II
B	B-I
	B-II
C	C-I
	C-II
	C-III

Figure 2.2 Environmental conditions corresponding with the crystal type

2.1.1 Frost Growth in Natural Convection

A number of experimental studies have been performed to characterize frost growth under natural convection. Tajima et al. (1972, 1973, 1974, and 1975), Kennedy et al. (1974), Tokura et al. (1983), and Cremers et al. (1982) are among those who have performed studies on frost under natural convection. Generally, it has been observed that on a vertical cold surface, frost grows more rapidly on the upper part of the surface than the lower surface. This is due to flow from the top to the bottom of the surface, which is caused by natural convection. The heat and mass transfer is higher at the top of the surface than at the bottom. Also, as air travels to the lower surface, it contains less moisture.

Tajima et al. (1972, 1973, 1974, and 1975) studied frost formation under natural convection. In a set of four experiments, a cold frosting surface was placed in different

orientations. In all experiments, growth patterns similar to those of Hayashi et al (1977) were observed.

Tokura et al. (1983) performed an experimental study that found dimensionless parameters effective in predicting frost density, thermal conductivity, and growth rates for frost on a cooled vertical plate under natural convection. These parameters are given in the following discussion. The experiments were run for up to nine hours at an ambient temperature of 20°C to 25°C, a frosting surface temperature of -6°C to -20°C, and relative humidity from 30 to 70%.

Tokura et al. (1983) expressed the heat flux, q , through the frost layer as a sum of convection, phase change, and radiation heat flux as

$$q = h(T_{\infty} - T_f) + \dot{m}h_{sg} + a\sigma(T_{\infty}^4 - T_f^4) \quad (2.1)$$

where h is the heat transfer coefficient, h_{sg} is the latent heat of sublimation, $\dot{m}$ is the rate of phase change, T_{∞} is the temperature of the ambient air, T_f is the temperature of the frost, and σ is the Stefan-Boltzmann constant. The coefficient a is a factor dependent on the emissivity of the frost and wall surfaces, and on the view factor between the surfaces. The effective thermal conductivity of the frost was defined as

$$k_f = \frac{q\delta_f}{T_f - T_c} \quad (2.2)$$

where T_c and δ_f are the temperature of the cold plate and the frost thickness, respectively. Using this relationship, it was found that the density of the frost layer affects the thermal conductivity more than other dependent parameters, including ambient temperature, humidity, and frosting surface temperature.

Tokura et al. (1983) then carried out a dimensional analysis to correlate thermal conductivity of frost to dependent variables. From dimensional analysis, the following variables were found:

$$\frac{k_f}{k_a} = F \left\{ \frac{h_m [\rho_v(\infty) - \rho_v(c)] h_{sg}}{h(T_{\infty} - T_c)}, \frac{h_m t}{x}, \frac{hx}{k_a} \right\} \quad (2.3)$$

In this equation, k_a , h_m , and ρ_v are the thermal conductivity of the air, the mass transfer coefficient, and the density of the vapor, respectively. The first term inside the brackets represents the ratio of latent heat of phase change to the sensible heat transferred

by convection. The second term is the ratio of the mass of water vapor transferred to the surface x^2 during time t to the mass of water contained in a volume x^3 . The final term is the Nusselt number.

Utilizing the dimensionless variables in Equation (3), a number of correlations between experimentation and the dimensionless variables were obtained. For a cold surface temperature less than -10°C , an ambient air temperature of 20°C , and a relative humidity greater than 35 percent, the following correlation was obtained:

$$\frac{\rho_f}{\rho_{ice}} = 0.001 \left[\frac{h_m t}{x} \frac{hx}{k_a} \right]^{1/2} \text{ for } \frac{h_m t}{x} \frac{hx}{k_a} > 5 \times 10^3. \quad (2.4)$$

This correlation measured experimental values within $\pm 30\%$, and is applicable to the three-dimensional growth period before melting occurs at the frost surface.

Detailed relationships between the frost thickness and the parameter Z were found, where Z is given as

$$Z = \left[\frac{\rho_v(\infty) - \rho_v(c)}{\rho_\infty} \right]^2 \left[\frac{h_m t}{x} \frac{hx}{k_a} \right]. \quad (2.5)$$

In this equation, ρ_∞ is the density of the ambient humid air. It was found that frost growth can be divided into two periods: 1.) a one-dimensional growth period (crystal growth period), where frost grows linearly with time, and 2.) a three-dimensional growth period (frost layer growth period), in which frost grows in proportion to the square root of time. For one-dimensional growth, the frost thickness correlation is

$$h\delta_f / k_a = 3.23 Z, \text{ for } Z < 0.11. \quad (2.6)$$

while for three-dimensional growth, the frost thickness correlation is

$$h\delta_f / k_a = 1.08 Z^{1/2}, \text{ for } Z > 0.11. \quad (2.7)$$

Cremers et al. (1982), White et al. (1981), and Schneider et al. (1978) also reported the square root of time dependence on the thickness of frost.

2.1.2 Frost Growth in Forced Convection

A large number of studies have been performed on frost formation under forced convection. Under forced convection, frost forms more rapidly than that of natural convection. Accumulation of frost often becomes so thick that it restricts airflow. Therefore, an understanding of frost formation under forced convection is important.

Experiments have determined the effect of plate temperature, air temperature, air humidity ratio, and Reynolds number on frost formation properties such as frost height, frost mass accumulation, frost deposition rate, and frost density. As with natural convection, frost formation does not occur uniformly over the whole surface, but varies with position. The frost layer grows faster upstream (i.e. leading edge of the plate) than downstream.

Schneider (1978) found frost thickness to be independent of variables that are usually significant in mass transfer, such as the surface orientation and Reynolds number. O'Neal and Tree (1992) found that frost height increases with decreasing plate temperature and increasing humidity. Air temperature was not found to be significant in the range tested (5 to 12°C). Frost height increased with Reynolds number until a critical Reynolds number was reached. Frost density was found to increase with increasing air temperature, humidity, and Reynolds number.

In a more recent study, Sahin (1994) found that frost height decreases as the temperature of the frosting surface increases, as the air temperature increases from 10 to 20°C, as the humidity ratio decreases, and as Reynolds number decreases. However, Schneider (1978) reported that the frost height was independent of Reynolds number. Frost height increases as the air humidity increases, since more water vapor diffuses into the frost layer. Reynolds number is not significant in frost height, since mass deposition of water vapor and frost density both increase with Reynolds number. These two properties offset one another, giving an unchanged frost height.

Other studies have found that an increase in Reynolds number leads to an increased or decreased frost thickness. Both arguments may be correct depending on the density or mass accumulation rate for the given case. One effect may be dominating the other.

Sahin (1994) also found the dependence of frost deposition rate on environmental conditions. The effect of plate temperature on deposition rate could not be found. However, mass deposition rate decreases with time, but and increases with increasing air humidity or Reynolds number. Frost density was found to increase with time, increasing Reynolds number, increasing frosting surface temperature, decreasing humidity, and increasing air temperature.

Schneider (1978) found a relationship between frost thickness and environmental parameters for the conditions tested on a cooled tube. The study took place with air velocity from 1.2 to 10 m/s, air temperatures from 5 to 15°C, relative humidity from 50 to 100%, and frosting surface temperature of -5 to -30°C. The following relationship was found:

$$\delta_f = 0.465 \left\{ \frac{k_f}{h_{sg} \rho_f} t [T(z = \delta_f) - T_c] \right\}^{1/2} \left(\frac{t}{1 \text{ hr}} \right)^{-0.03} \left[\frac{T(z = \delta_f) - T_c}{1 \text{ K}} \right]^{-0.01} \Pi^{0.25} F_t \quad (2.8)$$

where Π is defined as

$$\Pi = \frac{p_v - p'_F}{p' - p'_F} \quad (2.9)$$

and F_t is defined as

$$F_t = 1 + 0.052 \frac{T_\gamma - T_M}{T_M - T_c} \quad (2.10)$$

In these equations, p_v , p'_F , p' , $T_\square$, and T_M are the partial vapor pressure of the air, the pressure of the saturated vapor at the frost surface temperature, the vapor pressure of the saturated air, the air temperature, and the melting point temperature of the water-ice, respectively. Generally, it is seen that the frost thickness has a square root dependence on time, as Tokura et al. (1983) found for natural convection.

Mao et al. (1992) also found correlations for various frost properties for moist airflow over a flat plate. Correlations were found for dependent variables, including the mass of frost m_f , the frost height δ_f , the density of frost ρ_f , the thermal conductivity of frost k_f , the heat transfer coefficient of frost h_f , the overall Nusselt number Nu , the local Nusselt number Nu_x , the mass transfer coefficient of frost h_{mf} , and the Sherwood number Sh . Each variable was correlated in terms of the dimensionless variables X^* , W , T^* , Re , and Fo . X^* is defined as the dimensionless position ratio $X^* = x/d$, where d is the hydraulic diameter, and x is the distance from the leading edge. W is the humidity ratio of the inlet air. T^* is the dimensionless temperature ratio, which is defined as $T^* = (T - T_c)/(T_a + T_c)$. Re is the Reynolds number based on the inlet hydraulic diameter, and Fo is the Fourier number based on the thermal diffusivity of the air and the inlet hydraulic diameter. For a full listing of correlations, the reader is referred to Mao et al. (1992).

Correlations for the frost mass, frost thickness, and frost heat transfer coefficient over a horizontal cooled flat plate are given as follows:

$$m_f = 7.630 \times 10^{-4} X^{*-0.230} W^{1.318} T^{*0.083} \text{Re}^{1.064} \text{Fo}^{0.908} \quad (2.11)$$

$$\delta_f = 0.156 X^{*-0.098} W^{1.723} T^{*1.10} \text{Re}^{0.343} \text{Fo}^{0.655} \quad (2.12)$$

$$h_f = 0.506 X^{*-0.302} W^{0.297} T^{*0.725} \text{Re}^{0.717} \text{Fo}^{0.035} \quad (2.13)$$

2.2 Frost Growth under an Electric Field

When an electric field is applied to frost, the frost growth is affected. This modification is caused by a number of factors, including the alignment of solidifying water vapor molecules in the direction of the electric field, the difference in the dielectric constant at the interface of the ice and vapor phase, and the motion of the vapor molecules under the electric field.

Understanding how an electric field modifies frost growth is a very challenging task, and at present, it is mostly done experimentally. Knowing the basics of electrohydrodynamics (EHD) can improve our understanding of the phenomena and our ability to control the process. Without this fundamental knowledge of electrostatics and EHD, understanding the mechanisms governing frost growth under an electric field is very difficult.

Electrohydrodynamics combines two disciplines, namely electrodynamics and hydrodynamics. Typically, an electric field and a fluid field are coupled. In many cases, an electric field is applied to a dielectric fluid to enhance heat transfer. The electric field is applied between a charged electrode and a grounded electrode. A dielectric fluid typically has free ions and neutral molecules. In the presence of an electric field, the electric body force acting per unit volume can be expressed as follows:

$$f_e = \rho_c E - \frac{1}{2} E^2 \nabla \epsilon + \frac{1}{2} \nabla \left[E^2 \rho \left(\frac{\partial \epsilon}{\partial \rho} \right)_T \right] \quad (2.14)$$

The first term on the right side is the electrophoretic force that results from the net free charges within the fluid. The second term, known as the dielectrophoretic force, arises from the polarization of matter and produces the transnational motion of neutral

matter in a non-uniform electric field. The third term, called the electrostrictive force, represents a volume force whenever there is an inhomogeneous electric field within the dielectric.

The electrophoretic term is usually important in single-phase flows where the discharge of EHD current is mainly responsible for the generation of electrically induced secondary motions and the resulting enhancement mechanism. The second and third terms contribute substantially to the EHD body force in phase-change processes where the EHD body-force acts on the liquid-vapor interface. These terms are directly proportional to the square power of the applied electric field. In the second term, $\nabla\epsilon$ is a measure of the difference between the permittivity of the fluid and gas phases. Therefore, the higher the permittivity difference between the two phases, the greater the electric field force.

Electrophoresis is associated with the motion of the free charges in a uniform or non-uniform electric field. Dielectrophoresis, on the other hand, is the translation motion of neutral molecules caused by polarization in a non-uniform electric field. Any electric field exerts a force on charged particles. However, only non-uniform electric fields exert force on neutral molecules. In a uniform electric field, a charged particle is pulled toward an electrode of opposite polarity, whereas a neutral molecule is only polarized in a uniform electric field and produces no translational motion. The dipole may produce a torque if it is elongated in the field. The isotropic and spherical dipoles, however, will not produce any torque.

The electric field causes electrostriction, an induced electrical pressure on the frost crystals that results in longer and thinner molecules, and thus, longer and thinner crystals. These thin crystals are weaker than crystals not under the influence of an electric field and can be removed by their self-weight, airflow, or the electric body-force.

Frost is also modified due to the difference in the dielectric constant at the interface of two phases. When frost forms in air, there is an interface of two phases: the ice phase and the air-vapor phase. The large gradient in the dielectric constant between these phases at the interface causes an electric body force that acts upon the frost crystals.

Frost formation is also modified due to the changes in behavior of the vapor molecules under the influence of the electric fields. The neutral vapor molecules are

influenced by non-uniform electric fields and migrate toward either the positive or negative electrode, due to the polarization of the molecules and the unbalanced forces acting upon the molecules. Under an electric field, the frost crystal is charged. Therefore electric fields form around the crystal. At the sharp tip of the crystal, the electric field and electric field gradient is strong. Therefore, the molecules travel towards the tip of the frost crystals. Due to this enhanced vapor diffusion at the crystal tip, the tip growth velocity increases and diffusion brings more molecules to the high voltage tip.

The previous work done by others on control of frost by electric field is limited. Schaefer (1953) first observed modification of the growth of ice crystals under the influence of an electric field. He reported a rapid growth of ice in the form of whisker-like aggregates at high electric field gradients. Marshall and Gunn (1957) observed chaotic growth of many irregular branches on ice crystals under the influence of weak electric fields. Bartlett et al. (1963) observed that when an electric field is applied in excess of 500 V/cm, long thin needle-crystals grow rapidly, and that electric crystals frequently fracture spontaneously and travel in a horizontal path toward the electrode. Maybank and Barthakur (1967) observed that at electric field strengths of above 200 V/cm, the growth rate of ice crystals increased rapidly with electric field strength, and the crystals were more numerous, thinner, and more fragile compared to crystals not under the electric field.

Chuang and Velkoff (1971) reported a 200% increase in frost mass with a corona current of 200 μ A, and with a decrease in current, a decrease in the rate of frost deposition. The duration of their experiments, however, was only 5 minutes and did not detail the long-term effect of electric field on frost.

More recently, Munakata et al. (1993) performed experiments to control frost formation by an electric field. The electric field was applied to a flat-plate frosting surface by using a mesh electrode placed parallel to the surface. They found that frost mass accumulation decreased up to a certain voltage, and beyond that voltage, the mass accumulation increased. The maximum frost mass reduction at an applied voltage of 7.5 kV was 30%. Surface wettability was also observed not to affect the frost reduction by the electric field. Frost mass reduction was found to be larger for lower frosting surface temperatures.

Blanford et al. (1995) conducted experiments to determine the effect of electric field under forced convection. They reported a 20% decrease in frost formation at a corona discharge current of up to 20 μA , and a 220% increase in frost formation at a corona discharge current of 120 μA .

Libbrecht and Tanusheva (1999) studied frost growth under an electric field. The ice crystals were grown on a wire through which an electric potential was applied. Figure 2.3 (a) shows the growth of a dendrite at -15°C without an electric field. Figure 2.3 (b) and Figure 2.3 (c) show the growth at a potential of 2000 V. The frost tip velocity and sidebranch spacing increased as the tip advanced. Figure 2.3 (d) shows the smooth transition from normal growth to growth under an electric field. The tip velocities changed from 5 $\mu\text{m}/\text{sec}$ without a field, to 70 $\mu\text{m}/\text{sec}$ with an applied field.

Figure 2.3: Ice crystal dendrites grown from supersaturated air at -15°C
 (a) normal dendrite growth with no applied electric field; (b) growth at an applied field of 2000 V; (c) a continuation of (b); (d) a dendrite grown first without an electric field to a crystal grown under an electric field.

Without an electric field, surface tension normally stabilizes crystal growth. However when an electric field is applied, at a critical potential the growth becomes unstable and the tip velocity increases rapidly, forming an electric needle. Surface tension no longer stabilizes needle growth. In order for the growth to be stabilized, another mechanism must control the growth. Libbrecht and Tanusheva (1999) suggested that tip heating might stabilize electric needle growth. Tip heating is due to molecules

condensing on the frost crystal and heating up the crystal with their latent heat, thereby preventing overly rapid growth.

These findings were based on microscopic solvability theory, which has been successful in determining a mathematical solution for dendrite growth. Two physical processes, diffusion and surface tension, were used to model dendrite growth.

When a high frequency pulsating/AC field is applied, this behavior leads to continuous stretching and vibrations in the frost crystals, thus resulting in breakage of the crystals. If the frequency of the applied pulsating electric field is greater than the natural frequency of the respective crystals, then removal of the crystals from a frosting surface becomes very easy. This molecular behavior was revealed in the natural convection experiments performed in our laboratory. In addition, visualization studies indicated the presence of this behavior of frost under a pulsating electric field. To our knowledge, there is no other work reported in the open literature on frost growth under a high frequency pulsating/AC electric field. Only a few references were found in which electric fields with frequencies as high as 50Hz were tested.

Depending on the thermodynamic conditions employed (i.e. air and plate temperature, relative humidity), frost crystals undertake certain patterns and heights. Quantifying the effect of the electric field (i.e. proper voltage, electrode gap & frequency) in these conditions is a laborious task, but once done, can provide very useful means for better frost control in refrigeration and air conditioning equipment.

3. Experimental Apparatus

This section will present the experimental setup and instrumentation used throughout the two-year duration of the project to conduct experiments for frost growth control using an EHD active technique.

Two different experimental setups were designed and fabricated during this project. Experiments were initiated in the beginning of the project using a setup in which natural convection experiments were conducted with a vertical plate configuration. Very interesting and challenging problems were solved using this setup. However, this setup proved to provide good controllability only in natural convection experiments. Forced convection experiments proved to be troublesome and inconsistent using this setup. Therefore, a new closed-loop setup with an improved design and high controllability features was built and used to conduct the rest of the proposed experiments.

This section will present the construction of both setups and their components. All the important features and the accuracy of the instrumentation used will be presented as well.

All the previous experimental work that attempted to control frost growth by the use of electric fields was done with bare electrodes that are prone to sparking and current leakage in a condensing environment [i.e. Munakata et al. (1993), Blanford et al. (1995), and Mishra (1997)]. In actual refrigeration applications, this is undesirable. Therefore, this study utilized wire electrodes encased in Teflon insulation to allow for both frost control and prevention of sparking. Electric fields were applied between a cold flat plate (frosting surface) and a wire electrode encased in insulation and positioned parallel to the surface. Experiments were conducted under free, and forced convection conditions for different values of applied voltage, air temperature, plate temperature, and time duration. Experiments were also run for a bare wire mesh electrode, and the results were compared with those of the insulated electrode.

3.1 Natural Convection Experimental Setup

Experiments for natural convection frosting experiments were carried out in an experimental system consisting of a test chamber, a coolant loop, a data acquisition system, and a high voltage power supply.

A schematic diagram of the setup is shown in Figure 3.1. A brief description of each component will follow.

Figure 3.1 Schematic of the Natural Convection Experimental Setup

3.1.1 Test Chamber and Test Section

The test chamber consisted of an upright freezer chamber. Inside the chamber, a brass cooling plate was covered with insulation and attached to two aluminum plates. One of the plates served as the base-case (EHD-free) frost surface, whereas the other one was charged with the electric field. The aluminum plates were 114.3 mm long and 63.7 mm wide. A coolant loop ran the cooling fluid through the brass plate, which in turn, provided the required cooling to the test section aluminum plates. The temperature of the frosting plates was measured by a thermocouple. The aluminum plates acted as the frosting surface and were removable to allow for mass measurement. Applying a thermal-conductive paste between the brass and aluminum ensured an effective thermal contact between the plates.

Two electrodes were located parallel to the aluminum plates as shown in Figure 3.2. One electrode was attached to the high-voltage power supply, and the other was grounded. The arrangement of the electrodes allowed for symmetry in airflow during the experiments.

Figure 3.2 Top view of experimental test section

The electrodes and heat exchanger were located inside a Plexiglas duct in the freezer. At the bottom of the duct, a variable speed fan moved air through the test section for forced convection experiments. Pitot tubes connected to a differential pressure transducer measured the air velocity in the test section.

The air temperature and humidity were maintained inside the freezer with a data acquisition system (DAS). The air temperature was measured with a thermocouple, and the humidity was measured with a humidity sensor. A humidifier was located inside the freezer. The humidifier and freezer were turned on and off by the DAS to maintain the air conditions. The air and relative humidity were maintained within $\pm 1^\circ\text{C}$ and $\pm 5\%$, respectively.

3.1.2 Coolant Loop

The coolant loop consisted of a reservoir in which the coolant temperature was reduced to below the desired frosting surface temperature by a low-temperature chiller. A

circulating pump circulated the coolant, 200 proof ethyl alcohol (freezing temperature = -115°C), through the inside of the brass heat exchanger to maintain the cold plate frosting surface at the predetermined temperature. The loop was equipped with a temperature controller to maintain the coolant temperature at a pre-set value.

3.1.3 Data Acquisition System

The temperature and humidity of the air were maintained through the use of a Multiscan1200 data acquisition system (DAS). The air temperature was measured with a thermocouple, while a humidity sensor measured the humidity. These readings were provided to the DAS, which sent signals to turn on/off the cold air compressor of the test chamber (freezer) and humidifier in order to maintain the desired temperature and humidity of the air inside the experimental chamber. The temperature of the frosting surface was also measured by a thermocouple and recorded by the DAS. Likewise, the differential pressure was recorded by the DAS.

3.1.4 High Voltage System

The high voltage system consisted of a high voltage power supply with a range of 0 to 60 kV. The high voltage power supply was connected to the charged electrode, while the receiving electrode was grounded. The two electrodes were placed parallel to the frosting surface at a predetermined electrode gap. For some experiments, the high voltage was turned on/off at predetermined times. An interval timer relay was attached to the power supply that turned the high voltage on or off. A schematic of this relay is shown in Figure 3.3. Two potentiometer dials were adjusted to set the time to turn on and off the high voltage. Application of 12V to the interval timer started the relay. When ports 10 and 11 on the power supply were connected, the high voltage was supplied to the electrode.

Figure 3.3 Interval timer relay

3.1.5 Electrode Designs

A number of electrode designs were used in experimentation. Electrode designs were modified until the most suitable design for frost reduction was found. Previously, a flat metal plate was used in natural convection experiments as an electrode. However, the design was prone to current leakage and sparking and is not conducive to airflow movement in actual refrigeration applications. Therefore, an insulated wire electrode was used to control frost.

Electrode designs tested in experiments included: 1) a wire mesh electrode encased in Plexiglas, 2) a wire electrode encased in tubular insulation, 3) a bare (non-insulated) wire mesh electrode, and 4) an adjustable gap wire electrode encased in tubular insulation. The optimal electrode design used in experimentation was the wire electrode encased in tubular insulation.

The first electrode design tested in experimentation was composed of a wire mesh encased between two Plexiglas plates, as shown in Figure 3.4. With this electrode, the experimental results were inconsistent. Water condensed on the insulation, causing electrical charge build up, which reduced the electrical field to zero. Therefore, frost reduction experimentation with this electrode design was not pursued.

Figure 3.4 Wire mesh electrode encased in Plexiglas

The next design tested was a wire electrode encased in tubular Teflon insulation situated 20 mm from the frosting surface, as shown in Figure 3.5.

Figure 3.5 Insulated wire electrode

The wire diameter and the insulation diameter were 0.25 mm and 1.45 mm respectively. As with the previous electrode design, condensation formed on the electrode.

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Figure 3.6 Bare wire mesh electrode

Condensation forms on the electrode due to the fact that the electrode is at a cold temperature. Vapor molecules also migrate toward the electrode and condense due to the high electric field strength near the wire. Condensation caused grounding of the charged electrode. Therefore, a 1-watt heat source was installed on the high voltage insulated electrode to prevent condensation and grounding of the electrode. The heat source was observed to have a negligible effect on frost growth on the frosting surface.

Results with the insulated wire electrode were compared to results with a bare mesh electrode. This electrode is shown in Figure 3.6. The bare electrode was located 8 mm from the frosting surface. The wire diameter was 0.8 mm and the wires were spaced 6.4 mm apart. The final electrode used in experimentation was an insulated wire electrode with an adjustable electrode to frosting surface gap. The electrode design was the same as the previously discussed insulated wire electrode except for its ability to be adjusted. The side view of this electrode is shown in Figure 3.7.

Figure 3.7 Adjustable insulated wire electrode

3.1.6 Visualization Technique in the Natural Convection Setup

In order to take photographs of the frost crystals, a 35mm camera was used with a telephoto lens. A mirror was placed at a 45-degree angle above the freezer, to allow for ease of photography. The freezer was closed with a window that allows for viewing of frost during experiments. Just before taking each picture, the window was opened to allow for a clearer image. The window was then closed immediately after taking pictures. A fiber optic light source was used to illuminate the frost crystals during photography. A schematic of the photography setup is shown in Figure 3.8.

Figure 3.8 Schematic of photography setup

Videos of the frost growth were also taken with a camcorder. The lens opening of the camcorder was placed on the window above the frosting surface and focused on the frost. Videos allow for a more dynamic understanding of frost growth under electric field than photographs.

4. Experimental Procedure

The experimental procedure for frost mass measurements under both EHD and Base Case conditions is presented herein.

Step 1 – Setting the desired environmental conditions

- Before beginning the experimental runs, the aluminum plates were cleaned thoroughly with alcohol and then attached to the cooling heat exchangers with thermally conductive paste. The plates were grounded. All the equipment except the high voltage source was turned on until the desired environmental conditions (i.e., environmental chamber temperature, air relative humidity, and cold plate temperature) in the experimental chamber were reached. The surfaces of the aluminum plates were again cleaned with alcohol and completely dried with a clean, dry towel when the desired conditions were reached.

Step 2 – Running the experiment

Once all desired parameters reached steady state conditions, the high voltage power supply was turned on and the experimental run was begun. Continuous monitoring of the voltage and current was done throughout the duration of the experiment. At the end of each experiment, the high voltage was turned off and the aluminum plates (with their frost crystals) were removed from the test chamber and placed in separate sealed bags.

Three types of experiments were run with the insulated electrode. The plates were either removed immediately or 45 seconds after turning off the high voltage power supply. For the bare electrode, the plates were always removed immediately upon turning off the electric field. For the third type of experiment, the high voltage power supply was cycled on and off at predetermined times. The power supply was turned off at the end of the experiment, and the plates were removed 30 to 45 seconds after turning off the power supply.

Step 3 – Measurement

Each plate was immediately weighed with a high-precision scale with a resolution of 0.1 mg. The plates were then dried for 1 to 2 hours in a vacuum chamber to remove all water moisture and weighed again. The total frost mass accumulation during the experiment was determined from the difference in the two masses before and after

drying. The difference in frost accumulation on the plates with and without the high-voltage field were used to calculate the frost mass reduction as

$$R = 100 (1 - m_f / m_o) \quad (4.1)$$

In this equation, R is the percent mass reduction, and m_f and m_o are the frost mass with and without the applied electric field, respectively. The experiments were repeated 4 to 6 times to determine the repeatability of results and to obtain the average frost mass in both base case and EHD conditions. This was necessary due to the random nature of frost.

During the experiments, the applied electric field and current were recorded to determine the EHD power consumption, which is given as:

$$Q_{EHD} = V \cdot I \quad (4.2)$$

where V and I are the applied electric field potential and current, respectively.

5. Frost Height Measurement Technique

Software was developed in our laboratory to measure the frost height. The software is called "*Frost Track*" and it is a Windows based application. The Visual Java programming language was used to develop this software. The aim of this software is to digitally track the frost contour along the length of the test plate, and thus provide frost height distribution for the experiments conducted. By taking digital side pictures of the frosting plate and then loading the pictures into Frost Track's working environment, we obtain the frost image. Then, after noticing the color difference between the background pixels and the frost image pixels, the frost contour is drawn. All the information required to characterize the frost height is obtained internally in the software.

"Frost Track" has an attractive graphical interface, and it is a user-friendly Windows application. Options such as selecting contour color, thickness, or saving the frost height data along the contour were implemented in this version. Up to four contours can be loaded into this program, and thus frost height comparison between similar tests or different conditions can be achieved.

Once the side pictures of the frost are obtained in digital format (either by using a digital camera or by photo scanning), the steps involved in obtaining the frost contour and frost height values will be presented in the remaining part of this section. Thus, an overview of the software will be provided in the following.

5.1 Loading the frost picture

The first step in this process is loading the file into the Frost Track environment. This can be easily done through the standard Windows-based applications' "Open" menu. Once loaded, the picture will be displayed in the Frost Track work area, as shown in Figure 5.1. The orientation of the picture has to face upward at all times, this being a software design restriction. The image can be flipped in another program if this is not the case.

Figure 5.1 Loading the frost picture

5.2 Positioning/Repositioning the Base Line

The next step will be positioning of a reference base line, indicating the cold plate surface. Moving the base line in the desired position and then hitting the Enter tab can do this. By default, a frost contour curve will be calculated as shown in Figure 5.2(a). The line is not calculated 'smartly'; the first time it uses only default settings. You will almost certainly want to redo the line to better fit your image.

Figure 5.2 Frost contour

The repositioning of the base line can be done at any time through the help of a menu, which allows the possibility of recalculating the frost contour (Figure 5.3) as well. From this window, one can show or hide the image by using the checkbox. More importantly, the contour line can be tuned to your specifications by properly selecting the background color or replacing the baseline (see Figure 5.4). The background color can be

selected by clicking the mouse on the background area in the picture. The color will automatically be detected and selected by the Frost Track. Help is also provided through the Help button at all times.

Figure 5.3 Recalculating the base line

Figure 5.4 Tuning the frost contour

5.3 Graphical Display Options

Several graphical display options were included in this software, allowing a better manipulation and display of the frost contours obtained. The software allows loading up to four contours at the same time. For each of these contours, options regarding the display of the contours or the picture itself are available through the dialog box depicted in Figure 5.5.

Figure 5.5 Graphical display options

Changing the color or thickness of the frost contour lines, base lines, and the average line can be done in this software. These capabilities are included in a menu, which is shown in Figure 5.6.

Figure 5.6 Line Styles Window

5.4 Frost Height Data

The program calculates the values of the average, minimum, and maximum frost heights. These values can be displayed either on the frost contour by horizontal lines corresponding to the calculated points, or by displaying the calculated values in an info-box located under the frost picture (see Figure 5.2). Other calculated values displayed in this box are the median and the standard deviations of the frost height data.

The format of this info-box can be customized by selecting the values that should be displayed, the color of the box, and the displayed precision of the calculated values etc. These implemented features are shown in Figure 5.7

Figure 5.7 Info-Box Format

Another capability of this software is to collect all the frost height data points along the frost contour and save them in a text or tabular format. Thus, this data can be loaded into other programs (i.e. Excel), and frost contour can be plotted according to these points, as shown in Figure 5.8.

Figure 5.8 Saving frost data capabilities

5.5 Loading/Saving several pictures at the same time

As mentioned before, up to four contours can be loaded into the Frost Track working environment. Loading the pictures is done in a similar way as that presented before by answering 'yes' to the 'Keep other graphs?' question. Only one image can be loaded at a time. If an image is already open, its graph will remain but the image will be replaced. Alignment of the graphs is necessary when several pictures are loaded. Graphs can be vertically translated so that they all share a common base line. This common point will serve as the x-axis. By default, graphs will align to the first graph or to the most recently loaded image.

This can be changed with the 'Align' button, as shown in Figure 5.9.

Figure 5.9 Aligning multiple graphs

6. Experimental Results

The growth characteristics of frost under base-case conditions, and under an electric field are compared and discussed in this section.

Natural convection experiments for a vertical orientation of the cold plate will be presented in this section in both EHD and Base Case conditions. The outcome of this part is the selection of the best electrode configuration for best frost reduction for these conditions, as well as other important features, such as use of intermittent electric fields. Visualization of frost control using EHD will be also included.

6.1 Effect of EHD on Frost Grown on a Vertical Cold Plate

The effect of EHD on frost formation on a vertical frosting surface was tested. All the features of this setup, including the visualization technique were described in section 3 of this report. The characteristics provided by an insulated wire electrode and a bare wire mesh electrode under natural convection conditions are analyzed in this section. Frost growth photographs for the insulated and bare electrodes are presented. The growth characteristics are compared in each case and explained, based on experimental data and visualization. The effect of intermittent time on frost growth is also discussed. Forced convection experiments proved to be troublesome from the technical point of view in this setup. However, some preliminary forced convection results obtained with this setup will be also discussed.

6.1.1. Natural Convection

In all experiments under natural convection, it was observed that frost grows thicker at the top of the plate than at the bottom of the plate. Tajima et al. (1974) also observed this trend for a vertical flat plate under natural convection. This trend is due to the fact that natural convection causes air to travel from the top of the cold frosting surface to the bottom of the surface. When the air initially passes the top of the plate, the moisture in the air condenses and freezes on the plate. As the air continues down the plate, the air contains less moisture. Also, the heat and mass transfer coefficients are higher at the top of the plate than at the bottom.

In the presence of an applied electric field, a thin layer of frost formed on the frosting surface. This layer contributed to the overall mass of frost, which grew under an electric field. Detailed discussions are given in the following section. All the experimental results conducted with this setup are tabulated in Annex B.

6.1.2 Photographs of Frost Growth

Representative photographs of frost growth, with and without the electric field, for an insulated wire electrode and bare wire mesh electrode are presented in Figure 6.1. The photographs were taken from above the test section with the camera facing down. The photographs show the frost at the top of the frosting surface and are to the same scale. The line near the bottom of each photograph has been drawn to show the interface between frost and frosting surface. The left column of photographs shows frost growth with no applied electric field. The center column shows frost growth under the bare wire mesh electrode with an applied voltage of 8 kV, while the right column shows frost growth under an insulated wire electrode with an applied voltage of 35 kV. The frosting time increases from the top row to the bottom row (a - d) of the photographs from 0.5 to 2 hours.

The left column of photographs illustrates how the frost layer grows like a uniform forest of pine trees when not under an electric field. The frost gets thicker and taller with an increase of the frosting time. After two hours (Figure 6.1 d, left), the frost layer has a meshed structure. These photographs are consistent with the observations of Hayashi et al. (1977) for the frost layer growth period. It can be easily seen from the photographs that on average, frost not exposed to an electric field is thicker and taller than frost under an electric field, and that more frost grows without an electric field.

While a thick meshed structure characterized frost crystal growth without an electric field, the crystal growth is modified under an electric field. The center column of photographs (Figure 6.1 a-d, center) depicts frost growth under the bare wire mesh electrode. Under this electrode, long and thin frost crystals characterize the frost growth. The crystals do not grow uniformly over the frosting surface. There is more crystal growth in some locations. It can be seen in the photographs that the quantity of frost in

these locations increases with time. Frost crystals remain long and thin as time proceeds, while the frost layer density increases.

The right column of photographs shows frost growth under the electric field of the insulated wire electrode. Like the mesh electrode, the crystals initially grow longer and thinner in a non-uniform pattern (Figure 6.1 a, right). The crystals align themselves in the direction of the electric field. Crystals are pulled from the frosting surface toward the electrode by the electric field. These trends are consistent with the previous report by Munakata et al. (1993). Unlike the bare wire mesh case, the frost crystals become branched (Figure 6.1 b, right). Over the long term, however, the average crystal is shorter and thinner than those unexposed to an electric field (Figures 6.1 c-d). This feature is due to the fact that longer crystals are pulled from the frosting surface toward the electrode by the electric field. Over time, the insulated wire electrode grows a more uniform crystal than the bare mesh electrode (Figure 6.1 c-d).

Figure 6.1 Photographs of frost growth.

Left – no applied field, center – field applied by electrode, right – field applied by electrode #1; a) 30 minutes, b) 60 minutes, c) 90 minutes, d) 120 minutes

6.1.3 Insulated Wire Electrode

Electrode Charging

It is well known that charge accumulates in Teflon, which contains surface and bulk traps for electric charge. When Teflon is charged, its surface potential initially rises linearly as charge carriers are trapped at the surface. Eventually, the surface potential reaches a steady state. It has also been shown that in humid atmospheres, the surface potential of Teflon saturates at lower potentials than in dry atmospheres. The surface charge of Teflon also decays more rapidly in a humid atmosphere than in a dry atmosphere.

When frost grows under an electric field supplied by an insulated electrode, the electric field strength changes with time. The Teflon insulation around the wire electrode acts as a surface for electrical charge build-up, and the electric field diminishes with charge build-up. The amount of surface charge build-up is dependent on the humidity content of the air (Raposo et al., 1994). Eventually, the surface charge reaches a constant value, which is dependent upon humidity (Raposo et al., 1994).

This constant charge value is the surface charge saturation point. When the high voltage power supply is turned off, there is a transient period in which the electric field strength increases rapidly and reverses polarity.

Direct measurement of the electric field in the present experiments confirmed that the electric field strength and polarity changed during the experiments, as explained above. This finding is consistent with that reported in the literature (Giacometti et al., 1992).

During the field measurement, air temperature, relative humidity, and applied voltage were 0°C, 90%, and 35 kV, respectively. The electric field meter was placed on the frosting surface plate (ground electrode) with its sensor facing the positive electrode. A schematic of this configuration is shown in Figure 6.2.

Figure 6.2 Schematic of electric field meter measurement setup

In Figure 6.3, the high voltage power supply was turned on for 30 minutes and then turned off for ten minutes. This cycle was repeated for 80 minutes. The graphs are for two different runs of this case. All conditions were the same for the two runs. It can be seen that the electric field's behavior is not always consistent, due to variations in the charging of Teflon. Therefore, it can be expected that experimental results with intermittent timing intervals will not always be consistent. In Figure 6.4, the field was measured with the high voltage on/off cycles for intervals of both 10/5 minutes and 5/10 minutes. This was repeated for two hours.

When the power supply is initially turned on, the electric field strength is high. The field strength rapidly reduces and reaches a constant value. Upon reaching this constant value, Teflon reaches its charge saturation point. When the power supply is then turned off, the Teflon immediately discharges and the field strength magnitude increases with the opposite polarity. Again, the field strength rapidly reduces and reaches a constant value. When the power is turned back on, the field strength immediately reverses polarity and increases.

Figure 6.3 The electric field versus time; power supply on for 30 minutes and off for 10 minutes, two different experimental results; $T_a = 0^\circ\text{C}$, RH = 90%, $V = 35 \text{ kV}$

Figure 6.4 The electric field versus time; power supply cycles on/off for both 10/5 and 5/10 minutes; $T_a = 0^\circ\text{C}$, RH = 90%, $V = 35 \text{ kV}$

Continuous electric filed

When the high voltage power supply is turned off, a larger amount of frost crystals migrates from the frosting surface to the electrode than when the power supply is on. The crystals migrate vigorously during this transient period. The longer the high voltage power supply has been on before being turned off, the greater the crystal migration from the frosting surface to the electrode. This is due to the fact that more frost forms over more time and that the frost is highly polarized. Since more polarized frost is formed, more frost is pulled by the transient electric field. Visual observations indicate that the crystals migrate vigorously to the electrode for about 30 to 45 seconds after the electric field is turned off.

Figure 6.5 shows the normalized frost mass accumulation on the frosting surface versus the time when the high voltage was turned off. The crystals were weighed 45 seconds after the high voltage was turned off. Experiments were run at an applied voltage of 35 kV. The air temperature, plate coolant temperature, and relative humidity were maintained at 0°C, -30°C, and 90%, respectively. The ratio of the frost mass under a field to the frost mass not under a field was found to be independent of the time when the field was turned off for experimental runs of 1, 2, and 4 hours. Mass reduction of $30.8 \pm 5.0\%$ was measured in all three cases.

Figure 6.5 Normalized frost's mass accumulation vs. time for type 2 experiment;

$$T_a = 0^\circ\text{C}, T_p = -30^\circ\text{C}, \text{RH} = 90\%, V = 35 \text{ kV}$$

It is believed that when the high voltage power supply is turned off, the sudden change in the electric field causes the same ratio of crystals to be pulled from the frosting surface to the electrode, independent of time. According to Raposo et al. (1994), for 80% relative humidity, Teflon reaches a constant saturated surface charge level after charging for 30 minutes. As the humidity decreases, the time to reach a saturated surface charge level increases. Therefore, the electric field strength in experiments is constant after the Teflon reaches its saturated surface charge. Likewise, the electric body force that pulls frost from the frosting surface is constant after the saturation point is reached.

Figure 6.6 shows the total frost mass accumulation with and without an electric field for the same conditions presented in Figure 6.5. It is seen that the frost mass

accumulation increases linearly with time for the range of parameters tested with and without an electric field. This is expected since the normalized frost mass accumulation was shown to be constant, as indicated in Figure 6.5.

Figure 6.6 Frost mass accumulation versus time;
 $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, $\text{RH} = 90\%$, $V=35\text{kV}$

Since frost with and without an electric field increases linearly with time, mass diffuses into the frost layer at a constant rate. Frost is removed from the surface by the electric field at a constant rate, and the ratio of frost that is removed when the high voltage is turned off seems to be a constant.

In view of the fact that more frost is pulled from the frosting surface after the high voltage is turned off than while it is on, it is necessary to know how much frost is removed after the voltage is turned off. In Figure 6.7, the mass of frost on the frosting surface recorded immediately after turning off the high voltage is compared to the mass of frost on the surface when the crystals were allowed 45 seconds to migrate from the frosting surface to the electrode before being weighed. Experiments were run for two hours with the air temperature, plate coolant temperature, and relative humidity maintained at 0°C , -30°C , and 90% , respectively. For voltages less than 25kV , mass migration after turning off the electric field was not observed, while migration was observed for higher voltages. Therefore, for voltages of 25 kV or less, the same data was used for both types of experiments. For an applied voltage of 35kV , only 23.2% frost mass reduction was measured for experiments where mass of frost is weighed

immediately, as compared to 30.8% for experiments in which 45 seconds are allowed for crystals to migrate to the electrode before weighing. Therefore, 25% of the frost is pulled from the frosting surface by the electric field, while the rest is pulled from the surface after the high voltage is turned off during the transient period. Therefore, in order to improve frost mass reduction throughout the experiments where high humidities are employed, it is desirable to turn off the electric field from time to time.

Figure 6.7 Normalized frost mass accumulation versus voltage;

$$T_a = 0^\circ\text{C}, T_p = -30^\circ\text{C}, \text{RH} = 90\%$$

Intermittent electric field

Since the insulated electrode charges with time and crystals rapidly migrate from the frosting surface to the electrode when the power supply is turned off, experiments were run to determine the optimal timing to turn off and on the high voltage power supply. Table 6.1 lists the timing intervals tested and the percent of mass reduction of frost for experiments run for a total time of two hours.

The results in Table 6.1 show that more frost reduction occurred for cycles in which the power supply was turned off for one minute than for five minutes. This can be explained by the fact that the transient period after the power supply is turned off takes place in a very short time (less than a minute), as seen in Figure 6.4. Therefore, turning the power supply off for one minute is more desirable than turning the power supply off for five minutes. Therefore, frost is pulled off the electrode most rapidly during this time.

Table 6.1 Intermittent time results – two hours total time

Total Time (hour)	Cycle Time on/off (min)	Frost mass under field (g)	Frost mass not under field (g)	Percent mass reduction
2	29/1	0.54 ± 0.09	0.85 ± 0.09	36.1 ± 6.5
2	18/1	0.54 ± 0.04	0.81 ± 0.07	33.4 ± 3.6
2	10/1	0.57 ± 0.05	0.87 ± 0.07	34.7 ± 6.5
2	4.5/1	0.60 ± 0.03	0.88 ± 0.06	31.4 ± 3.8
2	25/5*	0.59 ± 0.12	0.86 ± 0.04	32.0 ± 11.9
2	15/5*	0.69 ± 0.14	0.91 ± 0.08	24.4 ± 10.7
2	10/5*	0.64 ± 0.16	0.86 ± 0.09	26.2 ± 15.6

*The cycle began with the power supply off

Since a cycle of one-minute time off is preferable to a cycle of five-minute time off, it is necessary to determine the optimal time cycle to leave the power supply on. Cycles were examined with one minute off, for periods of two hours. From these results, it can be seen that for the times tested for 10 to 29 minutes with the power supply on, the percent of frost mass reduction was largely independent of the amount of time the power supply was on. However, it was noted that the more often the power supply was cycled on and off, the less frost would accumulate on the frosting surface. When less frost was present on the frosting surface, the heat transfer coefficient was higher than when more frost was on the surface. Therefore, it is preferable to cycle over the shortest time interval that pulls off the most frost.

It was noticed that a five-minute-on cycle did not remove as much frost as the 10-29 minute cycles. Five minutes had only a 31.4% percent mass reduction as compared to 33.4% to 36.1% mass reductions. The five-minute cycle may not have been as effective as the ten-minute cycle due to the fact that over a short time interval, Teflon's surface charge may not have reached its saturation point. Reaching the surface charge saturation point allows for the maximal transient effect of pulling frost crystals from the frosting surface. Therefore, a ten-minute on cycle is preferable to a 5-minute on cycle because the surface charge saturation point has been met.

Looking at the results in Table 6.1, the optimal time interval may be a cycle of 10 minutes on and 1 minute off for the range of time intervals tested. However, this might be confusing if one only considers frost mass reduction. While the 29/1 minute cycle measured greater frost mass reduction than the 10/1minute cycle, the total amount of frost accumulation was less when the power was cycled more frequently. Therefore, the heat transfer coefficient increased for more frequent cycling. It is recommended that future experiments include measurement of the heat transfer coefficient to further clarify the optimal time ratio.

Next, the effect of total time on the 10/1 cycle was examined. Table 6.2 lists the time intervals tested and the frost mass reduction for this case.

Table 6.2 Intermittent time results

Total Time (hour)	Cycle Time on/off (min)	Frost mass under field (g)	Frost mass not under field (g)	Percent mass reduction
0.5	10/1	0.31 ± 0.05	0.35 ± 0.06	12.1 ± 19.6
1	10/1	0.39 ± 0.07	0.52 ± 0.11	24.6 ± 8.6
2	10/1	0.57 ± 0.05	0.87 ± 0.07	34.7 ± 6.5
4	10/1	0.99 ± 0.04	1.51 ± 0.12	34.4 ± 6.0

It can be seen from Figure 6.8 that normalized frost mass accumulation decreases time up to a total of about 2 hours. As time increased beyond two hours, the normalized frost mass accumulation or frost mass reduction was shown to be independent of time. This trend is consistent with the independence of normalized frost mass accumulation with time when the high voltage power supply was only turned off at the end of the experimental time.

Figure 6.8 Normalized frost mass accumulation versus total experimental time for on/off experiments for a cycle time of 10/1 minutes; $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V=35\text{kV}$

Figure 6.9 shows that the mass of frost both with and without an electric field increases linearly with time. The difference in the total mass accumulation under a field and not under a field increases with time. A similar linear relationship is seen in Figure 6.6.

Figure 6.9 Frost mass accumulation versus total experimental time for on/off experiments for cycle times of 10/1 minutes; $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V= 35 \text{ kV}$

Visualization of Frost Growth for Intermittent Applied Voltage

Video tapes were used to record the frost growth under an intermittent electric field with an on/off cycle time of 10/1 minutes for base-case environmental parameters. When the power supply was turned off for one minute, a number of interesting effects were seen for each cycle. At the moment when the power supply was turned off, a large number of crystals was pulled from the frosting surface toward the electrode. After this, the vapor in the air was seen to move vigorously toward the frosting surface, thereby enhancing diffusion into the frost layer. As a result, frost growth rapidly increased, but at the moment when the power supply was again turned on, these crystals instantly broke off of the frosting surface.

Figure 6.10 shows a series of 10 images extracted from the video described above. Images were taken at 10-second intervals for a period of 90 seconds, beginning 20 seconds before the power supply was turned off. All images are of the same locations and to the same scale. The video was taken from above the test section. The upper half of each image shows the frost surface and associated frost growth. The lower half of each image shows the electrode. The black image above the frost is the grounding clamp for the frost surface.

Figures 6.10a and b show images of frost before the power supply was turned off. Relatively little change in frost growth took place during this time period. However, some crystals were removed and grew during the 10-second time period.

Figure 6.10c shows frost just after the power supply was turned off. It can be seen that many crystals were removed from the surface. The large removal of crystals corresponds with the turning off of the power supply and the beginning of the transient period of frost growth and electric field strength.

Figures 6.10 (c – h) depict the rapid growth of frost while the power supply was turned off. The frost rapidly grew with time. The increase in frost growth during the 10 seconds between images *f* and *g* is particularly dramatic. The dramatic changes continue with time until the power supply was turned back on. The large changes in frost growth can be attributed to the transient electric field. The changing electric field caused greatly enhanced diffusion of the vapor molecules toward the frost crystal tips, thereby greatly

increasing the frost growth rate. It was observed visually that the vapor molecules moved rapidly during this period.

The power supply was turned on in Figure 6.10i. It can clearly be seen that the long crystals, which grew while the power supply was off, were removed. It was observed visually that the crystals were removed instantly when the power supply was turned on, and the polarity of the electric field reversed. The frost crystals were polarized and aligned in the direction of the electric field. When the field was suddenly reversed, an impulsive force instantaneously shocked the crystals and caused them to break.

Effect of Cold Surface Temperature and Air Temperature on Frost Growth

Another important parameter when considering frost growth is the frosting surface temperature. Figure 6.11 presents the effect of plate temperature on frost mass accumulation. Experiments were performed with an applied voltage of 35kV, an air temperature of 0°C, and a relative humidity of 90% for a two-hour duration. The coolant plate temperature was varied from -20°C to -30°C. Both under a field and not under a field, the total mass accumulation decreased as the plate temperature increased. As expected this trend was consistent with trends previously reported in the literature (Cremers et al., 1982). As the temperature of the plate drops, more vapors condense and turn to solid phase. It was also seen that less frost accumulated under an electric field than when no electric field was present for all plate temperatures.

The air temperature is another important variable in frost growth. Figure 6.12 presents the effect of air temperature on frost mass accumulation. Experiments were performed with an applied voltage of 35kV, a plate coolant temperature of -30°C, and a relative humidity of 90% for two hours. The air temperature was varied from -5°C to 5°C. The total mass accumulation increased as the air temperature increased for both with and without electric field conditions. As air temperature increased at a given relative humidity, the air contained more water vapor molecules. Therefore, there were more vapors available to condense on the frosting surface. It was also seen that for all air temperatures, less frost accumulated under an electric field than when an electric field was not present. Both Figure 6.11 and 6.12 confirm the fact that the greater the temperature difference between the air and the plate, the more frost will form (provided the plate is below the dew point of the temperature of the passing air and 0°C).

Figure 6.11 Mass accumulation versus plate temperature:

$T_a = 0^\circ\text{C}$, $\text{RH} = 90\%$, $V = 35 \text{ kV}$

Figure 6.11 Mass accumulation versus air temperature:

$T_p = -30^\circ\text{C}$, $\text{RH} = 90\%$, $V = 35 \text{ kV}$

6.1.4 Bare Mesh Electrode

For a comparison of the results for the insulated electrode, experiments with a bare wire mesh electrode were run. The experiments with the bare electrode were repeated six times. The air temperature, plate coolant temperature, relative humidity, and applied voltage were 0°C , -30°C , 90% , and 9 kV , respectively. Experiments were run for 2

hours. For this case, the average frost mass reduction with the electric field was $31.4 \pm 10.7\%$. Experimental repeatability was difficult due to current leakage. Unlike the case with the insulated electrode, frost crystals did not migrate from the frosting surface to the electrode when the high voltage power supply was turned off. The frost mass reduction measured was comparable to that of reported by Munakata et al. (1993) for a wire mesh electrode.

It is interesting to note that the frost mass reduction for both the insulated and bare electrode are around 30%. The similarity in frost reduction, despite different growth patterns, can be explained by comparing the average frost growth over the entire surface under both a bare and an insulated electrode. Overall, the frost is more uniform when using the insulated electrode than when using the bare electrode. The frost is also short when the insulated electrode is used, since long crystals have been pulled. When using the bare electrode, however, the frost is longer at some locations and shorter at others. When the frost mass accumulation is totaled over the entire surface, reductions are similar to those found when the mass of frost is not under a field.

6.1.5 Power Consumption

In order to justify the use of electric fields to reduce frost growth, it is important to know the power consumption of the electric field. For the experiments, the voltage and current supplied to the electrodes were recorded and the power consumption was tabulated. The power consumption results for various electrodes are summarized in Table 6.3. It can be seen that the power utilized is negligible due to the low current.

Table 6.3 EHD Power Consumption

Electrode	Potential (kV)	Current (μ A)	Power (mW)
Bare	9	2.3	20.7
Insulated	5	0.12	0.6
	10	0.15	1.5
	15	0.18	2.7
	20	0.18	3.6
	25	0.18	4.5
	30	0.18	5.4
	35	0.18	6.3
	40	0.18	7.2

In conclusion, from these experiments we can state that the insulated electrode had better performance characteristics than the bare electrode for frost reduction. It was found that increasing frost duration time reduced the electric field intensity due to the charge buildup on the insulated electrode. When the applied voltage was turned off, many crystals migrated from the frosting surface to the electrode. Therefore, an intermittent electric field had a desirable effect on frost reduction.

For natural convection, up to 36% frost mass reduction was obtained with an intermittent electric field as compared to 30% frost mass reduction without an intermittent field. With an intermittent field, less frost formed on the frosting surface since more crystals migrated from the frosting surface when the power supply was turned off. For forced convection experiments, as the Reynolds number increased, the effect of the electric field on modifying frost growth decreased.

7. Conclusions and recommendations for future work

An experimental setup was constructed and used to conduct natural convection experiments on a vertical cold plate. Some very interesting and important features on frost control using EHD were found and emphasized in this study. It was shown that an electric field has the potential to effectively reduce frost mass accumulation on a cold surface in operating conditions of practical significance to refrigeration applications.

Frost crystal growth is substantially affected under an applied electric field. The solidifying water vapor molecules align in the direction of the electric field. The electric field causes an induced electrical pressure, which results in longer and thinner molecules, and thus longer and thinner crystals. The large gradient in the dielectric constant between the vapor and ice phases also causes an electric body force that acts upon the frost crystals. Neutral vapor molecules are influenced by non-uniform electric fields and migrate toward the frost crystals, thereby enhancing vapor diffusion. This migration of molecules toward the frost crystal tip also contributes to long and thin frost crystal formation. The thin crystals are weaker than crystals formed under a no-electric-field condition and can be removed by self-weight, airflow, or the electric body-force. Visualization studies conducted in our laboratory indicated this behavior of frost under an applied electric field.

The electrode design is a crucial element in modifying the frost growth. An insulated electrode is preferable to a bare electrode, since an insulated electrode prevents sparking and current leakage. It was shown that an insulated electrode reduces frost accumulation more than a bare electrode. When frost grows under a DC electric field of an insulated electrode, the electric field strength changes with time. Certain insulation around the wire electrode acts as a surface for electrical charge build-up, and the electrical field diminishes with charge build-up. Eventually, the surface charge reaches a saturation point. When the high voltage power supply is turned off, there is a transient period in which the electric field strength increases rapidly and reverses polarity. During this period frost crystals are broken and frost is removed from the surface of the plate. When the high voltage power supply is turned off, a larger amount of frost crystals migrate from the frosting surface to the electrode.

In natural convection experiments for a vertical orientation of the cold plate, the ratio of the frost mass under the field to the frost mass in the absence of the field was found to be independent of the time when the field was turned off. This was tested for 1, 2, and 4 hours. Mass reduction of $30 \pm 5.0\%$ was measured in all three cases. When the high voltage power supply was turned off, the sudden change in electrical field caused the same ratio of crystals to be pulled from the frosting surface to the electrode, independent of time. Frost mass accumulation also increased linearly with frost duration time, with and without an electric field. Therefore, mass diffuses into the frost layer at a constant rate, frost is removed by the electric field at a constant rate, and the ratio of frost that is removed when the high voltage is turned off is a constant. Due to the fact that more frost crystals are removed from the frosting surface when the power supply is turned off, an intermittent field is preferable for reducing frost with an insulated electrode. This effect can be used when high humidity and high super-saturation degrees are involved in the experiments.

A software was developed to measure frost height and was used to quantify the frost height reduction when using a high frequency electric field. Visualization studies were conducted both by a high-speed camera and a digital camcorder to characterize the frost growth. It was shown that frost surface temperature and air temperature affects frost growth. The total mass accumulation decreases by increasing the plate temperature or decreasing the air temperature, both under the applied field and in the absence of the field. As the temperature of the plate drops, more water vapor condenses and turns to solid phase. As the air temperature increases at a given relative humidity, the air contains more water vapor molecules. Therefore, there is more water vapor available to condense on the frosting surface.

Overall, the use of an electric field is a promising method for frost growth control and reduction in a wide range of industrial application. The use of an intermittent DC electric field with an insulated electrode proved to be an effective method for reducing frost growth and should be explored further.

Due to the complex nature of frost formation, and the uncertainties associated with this type of research, on-line mass measurement is strongly recommended for future work. In addition, when less frost is present on the frosting surface, the heat transfer

coefficient is higher. Therefore, it is recommended that future experiments include measurement of the heat transfer coefficient.

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9. Attachments

Appendix A. Vertical cold plate – experimental results

Notation:

Type 1 Experiment – Experiments in which the plates were removed immediately

Type 2 Experiment – Experiments in which the plates were removed after 45 seconds

Type 3 Experiment – Experiments in which on/off electric fields were used

Table B.1 Natural convection – insulated electrode, Type 1 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 5\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	0.7839	0.8923	0.879
2	0.8426	0.936	0.900
3	0.8292	0.9448	0.878
4	0.7568	0.8673	0.873
5	0.7357	0.8546	0.861
average, $\bar{x}$	0.7896	0.8990	0.878
standard deviation, σ	0.0458	0.0403	0.014
relative uncertainty, %	11.6	9.0	3.3

Table B.2 Natural convection – insulated electrode, Type 1 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 10\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	0.7315	0.8844	0.827
2	0.7185	0.8677	0.828
3	0.7609	0.9534	0.798
4	0.6900	0.9253	0.746
5	0.8145	0.8707	0.935
6	0.8247	0.9267	0.890
average, $\bar{x}$	0.7567	0.9047	0.837
standard deviation, σ	0.0539	0.0353	0.067
relative uncertainty, %	14.2	7.8	16.0

Table B.3 Natural convection – insulated electrode, Type 1 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 15\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.8808	1.0471	0.841
2	0.7951	0.8976	0.886
3	0.8264	1.0369	0.797
4	0.7062	0.873	0.809
5	0.7947	0.8933	0.890
6	0.7764	0.8452	0.919
average, $\bar{x}$	0.7966	0.9322	0.857
standard deviation, σ	0.0576	0.0871	0.049
relative uncertainty, %	14.5	18.7	11.4

Table B.4 Natural convection – insulated electrode, Type 1 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 20\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.6912	0.9421	0.734
2	0.6862	0.9942	0.690
3	0.7764	0.9287	0.836
4	0.6523	0.9219	0.708
5	0.7007	0.8706	0.805
6	0.7017	0.9099	0.771
average, $\bar{x}$	0.7014	0.9279	0.757
standard deviation, σ	0.0409	0.0406	0.057
relative uncertainty, %	11.6673	8.7507	15.0

Table B.5 Natural convection – insulated electrode, Type 1 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 25\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.6624	0.9063	0.731
2	0.6568	0.8691	0.756
3	0.6137	0.7793	0.788
4	0.5943	0.8441	0.704
5	0.5950	0.8557	0.695
6	0.6964	0.9415	0.740
average, $\bar{x}$	0.6364	0.8660	0.736
standard deviation, σ	0.0417	0.0556	0.034
relative uncertainty, %	13.1038	12.8322	9.2

Table B.6 Natural convection – insulated electrode, Type 1 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 30\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.7246	0.8936	0.811
2	0.7898	0.9695	0.815
3	0.8103	0.9501	0.853
4	0.7085	0.8576	0.826
5	0.7452	0.8766	0.850
6	0.6628	0.9450	0.701
7	0.6879	0.9696	0.709
8	0.7225	0.8658	0.834
average, $\bar{x}$	0.7315	0.9160	0.800
standard deviation, σ	0.0494	0.0474	0.060
relative uncertainty, %	13.5042	10.3484	15.1

Table B.7 Natural convection – insulated electrode, Type 1 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.6141	0.8037	0.764
2	0.6234	0.8353	0.746
3	0.6498	0.8874	0.732
4	0.7163	0.8922	0.803
5	0.7378	0.9711	0.760
6	0.6887	0.8610	0.800
average, $\bar{x}$	0.6717	0.8751	0.768
standard deviation, σ	0.0506	0.0575	0.028
relative uncertainty, %	15.1	13.1	7.4

Table B.8 Natural convection – insulated electrode, Type 1 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 40\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.6671	0.8582	0.777
2	0.7982	0.9438	0.846
3	0.744	0.9328	0.798
4	0.6777	0.8592	0.789
5	0.6848	0.932	0.735
6	0.7183	0.8895	0.808
average, $\bar{x}$	0.7150	0.9026	0.792
standard deviation, σ	0.0497	0.0387	0.037
relative uncertainty, %	13.9	8.6	9.2

Table B.9 Natural convection – insulated electrode, Type 1 experiment, $T_a = 0^\circ\text{C}$, $T_p = -25^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.6957	0.8098	0.859
2	0.6093	0.8088	0.753
3	0.6638	0.8582	0.773
4	0.6138	0.8365	0.734
5	0.6346	0.7756	0.818
6	0.6873	0.9184	0.748
average, $\bar{x}$	0.6508	0.8346	0.781
standard deviation, σ	0.0371	0.0497	0.048
relative uncertainty, %	11.4	11.9	12.3

Table B.10 Natural convection – insulated electrode, Type 1 experiment, $T_a = 0^\circ\text{C}$, $T_p = -20^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.5751	0.7192	0.800
2	0.5226	0.8067	0.648
3	0.5542	0.8164	0.679
4	0.569	0.7635	0.745
5	0.5529	0.7813	0.708
6	0.5594	0.7775	0.719
average, $\bar{x}$	0.5555	0.7774	0.716
standard deviation, σ	0.0183	0.0346	0.053
relative uncertainty, %	6.6	8.9	14.8

Table B.11 Natural convection – insulated electrode, Type 1 experiment, $T_a = -5^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.4433	0.6179	0.717
2	0.4272	0.6126	0.697
3	0.54	0.6993	0.772
4	0.4515	0.6253	0.722
5	0.4755	0.5901	0.806
6	0.4439	0.6038	0.735
average, $\bar{x}$	0.4636	0.6248	0.742
standard deviation, σ	0.0406	0.0385	0.040
relative uncertainty, %	17.5	12.3	10.8

Table B.12 Natural convection – insulated electrode, Type 1 experiment, $T_a = 5^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.9427	1.142	0.825
2	0.8622	1.2445	0.693
3	0.9855	1.2482	0.790
4	0.9977	1.2468	0.800
5	1.0143	1.2701	0.799
6	0.903	1.2899	0.700
average, $\bar{x}$	0.9509	1.2403	0.768
standard deviation, σ	0.0594	0.0513	0.057
relative uncertainty, %	12.5	8.3	14.7

Table B.13 Natural convection – insulated electrode, Type 2 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	0.6711	0.9419	0.712
2	0.6416	0.9049	0.709
3	0.5189	0.7525	0.690
4	0.6088	0.9263	0.657
average, $\bar{x}$	0.6101	0.8814	0.692
standard deviation, σ	0.0659	0.0873	0.025
relative uncertainty, %	21.6	19.8	7.3

Table B.14 Natural convection – insulated electrode, Type 2 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 37.5\text{ kV}$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	0.5583	0.8655	0.645
2	0.605	0.9059	0.668
3	0.5872	0.8364	0.702
4	0.5634	0.8548	0.659
average, $\bar{x}$	0.5785	0.8657	0.669
standard deviation, σ	0.0217	0.0294	0.024
relative uncertainty, %	7.5	6.8	7.3

Table B.15 Natural convection – insulated electrode, Type 2 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, $t = 1\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	0.3353	0.5040	0.665
2	0.3888	0.6005	0.647
3	0.3753	0.5298	0.708
4	0.3926	0.5196	0.756
average, $\bar{x}$	0.3730	0.5385	0.694
standard deviation, σ	0.0262	0.0427	0.048
relative uncertainty, %	14.1	15.9	13.9

Table B.16 Natural convection – insulated electrode, Type 2 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, $t = 4\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	1.1207	1.6591	0.675
2	1.0607	1.5551	0.682
3	1.1111	1.6389	0.678
4	1.1084	1.5208	0.729
average, $\bar{x}$	1.1002	1.5935	0.691
standard deviation, σ	0.0269	0.0661	0.025
relative uncertainty, %	4.9	8.3	7.3

Table B.17 Natural convection – insulated electrode, Type 3 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, 25/5 min, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	0.5653	0.8463	0.668
2	0.65	0.8655	0.751
3	0.5315	0.8389	0.634
4	0.5361	0.8719	0.615
5	0.6448	0.882	0.731
average, $\bar{x}$	0.5855	0.8609	0.680
standard deviation, σ	0.0580	0.0179	0.060
relative uncertainty, %	19.8	4.2	17.5

Table B.18 Natural convection – insulated electrode, Type 3 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, 15/5 min, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	0.676	1.0168	0.665
2	0.6864	0.9101	0.754
3	0.5836	0.8626	0.677
4	0.692	0.8809	0.786
5	0.7852	0.9571	0.820
6	0.7084	0.9507	0.745
average, $\bar{x}$	0.6911	0.9123	0.756
standard deviation, σ	0.0720	0.0417	0.054
relative uncertainty, %	20.8	9.1	14.1

Table B.19 Natural convection – insulated electrode, Type 3 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, 10/5 min, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	0.5115	0.8156	0.627
2	0.6329	0.8554	0.740
3	0.6777	0.8294	0.817
4	0.7237	0.8998	0.804
5	0.6476	0.9217	0.703
average, $\bar{x}$	0.6387	0.8644	0.738
standard deviation, σ	0.0791	0.0453	0.078
relative uncertainty, %	24.8	10.5	21.1

Table B.20 Natural convection – insulated electrode, Type 3 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, 30/1 min, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	0.506	0.8571	0.590
2	0.5091	0.7815	0.651
3	0.6159	0.9088	0.678
4	0.5235	0.8349	0.627
5	0.5648	0.8724	0.647
average, $\bar{x}$	0.5439	0.8509	0.639
standard deviation, σ	0.0466	0.0472	0.033
relative uncertainty, %	17.1	11.1	10.2

Table B.21 Natural convection – insulated electrode, Type 3 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, 18/1 min, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	0.5347	0.7833	0.683
2	0.5289	0.7764	0.681
3	0.5148	0.7997	0.644
4	0.5698	0.8459	0.674
5	0.5555	0.855	0.650
average, $\bar{x}$	0.5407	0.8121	0.666
standard deviation, σ	0.0219	0.0362	0.018
relative uncertainty, %	8.1	8.9	5.5

Table B.22 Natural convection – insulated electrode, Type 3 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, 10/1 min, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	0.5469	0.8423	0.649
2	0.5567	0.9229	0.603
3	0.5746	0.8757	0.656
4	0.61	0.884	0.690
5	0.5652	0.8433	0.670
6	0.5452	0.8439	0.646
average, $\bar{x}$	0.5703	0.8740	0.653
standard deviation, σ	0.0247	0.0329	0.032
relative uncertainty, %	8.7	7.5	9.9

Table B.23 Natural convection – insulated electrode, Type 3 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, 4.5/1 min, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	0.5885	0.8498	0.693
2	0.6258	0.8913	0.702
3	0.5837	0.8686	0.672
4	0.593	0.8435	0.703
5	0.6088	0.922	0.660
average, $\bar{x}$	0.6000	0.8750	0.686
standard deviation, σ	0.0172	0.0322	0.019
relative uncertainty, %	5.7	7.4	5.5

Table B.24 Natural convection – insulated electrode, Type 3 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, 10/1 min, $t = 4\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f/m_o
1	0.9555	1.4802	0.646
2	0.9896	1.5171	0.652
3	0.995	1.5847	0.628
4	1.0072	1.4425	0.698
average, $\bar{x}$	0.9868	1.5061	0.656
standard deviation, σ	0.0221	0.0606	0.030
relative uncertainty, %	4.5	8.0	9.1

Table B.25 Natural convection – insulated electrode, Type 3 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, 10/1 min, $t = 1\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.4407	0.5838	0.755
2	0.3812	0.4662	0.818
3	0.3296	0.4431	0.744
4	0.3813	0.5098	0.748
5	0.3898	0.5787	0.674
6	0.39	0.5067	0.770
7	0.4258	0.5522	0.771
average, $\bar{x}$	0.3912	0.5201	0.754
standard deviation, σ	0.0357	0.0542	0.043
relative uncertainty, %	18.2	20.9	11.5

Table B.26 Natural convection – insulated electrode, Type 3 experiment, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%, $V = 35\text{ kV}$, 10/1 min, $t = 32\text{ min}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.3337	0.3902	0.855
2	0.2835	0.3651	0.776
3	0.3307	0.3485	0.949
4	0.2878	0.3678	0.782
5	0.3220	0.3127	1.030
6	0.2841	0.3226	0.881
average, $\bar{x}$	0.3070	0.3512	0.879
standard deviation, σ	0.0243	0.0293	0.098
relative uncertainty, %	15.8	16.7	22.3

Table B.27 Natural convection – bare electrode, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%,
 $V=9\text{kV}$, $t = 2 \text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.5132	0.8048	0.638
2	0.6376	0.9564	0.667
3	0.8117	1.0236	0.793
4	0.613	1.0476	0.585
5	0.7028	1.0115	0.695
6	0.7465	1.0132	0.737
average, $\bar{x}$	0.6708	0.9762	0.686
Standard deviation, σ	0.1057	0.0891	0.073
Relative uncertainty, %	31.5219	18.2630	21.4

Table B.28 Forced convection – insulated electrode, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%,
 $V=35 \text{ kV}$, $\text{Re} = 1800$, $t = 1 \text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.7511	0.9003	0.834
2	0.8769	0.9814	0.894
3	0.8714	1.0262	0.849
4	0.9344	1.0849	0.861
5	0.8462	0.9661	0.876
6	0.8376	1.0136	0.826
7	0.9267	1.1873	0.781
average, $\bar{x}$	0.8635	1.0228	0.846
Standard deviation, σ	0.0617	0.0921	0.037
Relative uncertainty, %	14.3	18.0	8.7

Table B.29 Forced convection – insulated electrode, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%,
 $V=35\text{ kV}$, $\text{Re} = 1400$, $t = 1\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	0.787	0.9743	0.808
2	0.8492	0.9783	0.868
3	0.9709	1.0066	0.965
4	1.2572	1.2123	1.037
5	0.9553	1.0926	0.874
6	0.971	1.028	0.945
7	1.0039	1.1057	0.908
8	1.3695	1.1365	1.205
9	1.422	1.2402	1.147
10	1.4604	1.419	1.029
average, $\bar{x}$	0.9651	1.0487	0.916
standard deviation, σ	0.2483	0.1399	0.127
relative uncertainty, %	51.5	26.7	27.6

Table B.30 Forced convection – insulated electrode, $T_a = 0^\circ\text{C}$, $T_p = -30^\circ\text{C}$, RH = 90%,
 $V=40\text{ kV}$, $\text{Re} = 11000$, $t = 2\text{ hr}$

Run number	mass with field, m_f (g)	mass without field, m_o (g)	m_f / m_o
1	3.9106	4.0606	0.963
2	3.3786	3.6493	0.926
3	4.0225	4.0002	1.006
average, $\bar{x}$	3.7706	3.9034	0.965
standard deviation, σ	0.3440	0.2221	0.040
relative uncertainty, %	18.2	11.4	8.3