

**A PARAMETRIC STUDY OF A
THREE-PHASE, SPRAY-COLUMN DIRECT CONTACT HEAT EXCHANGER**

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ABSTRACT

A numerical model of a three-phase, direct-contact, spray-column heat exchanger has been developed. This model has been used to investigate a number of characteristics of these devices, such as temperature and holdup distributions through the column. Little has been given in the literature before about quantitative variations of performance as a function of the key independent variables, and information on these aspects is presented here. Although the results presented are for a specific geometry (0.61 m diameter, 3 m active column height, evaporating pentane in 85°C water), the variations shown can give insights generally into the factors affecting performance in these devices. In virtually all cases examined here, extremely good comparisons are shown between predictions and measurements. Conclusions are drawn about the applicability of the model and the important effects demonstrated.

NOMENCLATURE

A Cross-sectional area of the column
 C_D Drag coefficient
 C_P Specific heat at constant pressure
 D_p Diameter of droplet
 \bar{D} Averaged diameter, $(D^2 + D_0^2)/(2DD_0)$
 g Acceleration due to gravity
 h Heat transfer coefficient
 h Enthalpy
 k Thermal conductivity
 m Mass flow rate
 M Molecular weight
 n_b Number of drops per unit volume
 P Pressure
 Pr Prandtl number
 Q Heat transfer
 R Droplet radius
 Re Reynolds number based on droplet diameter
 T Temperature
 U Velocity (if without subscript, difference value)

V Volume
 x Quality
 z Vertical distance along column from bottom
 β Vapor half opening angle
 η Ratio: heat transfer to the bubble to heat transfer from the continuous phase
 ϕ Holdup ratio
 μ Viscosity
 ρ Density
 Subscripts and superscripts
 c Continuous fluid
 cd Between continuous and dispersed fluids
 d Dispersed fluid
 l Liquid
 $loss$ Loss
 p Pentane
 o Initial
 v Vapor
 w Water

INTRODUCTION

Three-phase direct contact heat exchangers have been proposed for a variety of applications including power generation and water desalination. One of the major problems still present in this technical area is a lack of comprehensive design tools. Since only limited experiments have been performed, there is also a lack of broadly applicable data. This is in contrast to the large amount of experimental and theoretical work on physical behavior of conventional closed heat exchangers.

Although there has been a large number of analytical studies of heat transfer to a single bubble reported in the literature, there have been relatively few that have addressed the overall performance of three-phase spray columns. Typical of the previous work is that of Moxhtarzadeh and El-Shirbini [1]. They have analyzed droplet growth when the droplet is rising and evaporating in an immiscible liquid. They assumed the droplet was traveling through an infinite, stationary medium of constant temperature. Results given by them included parametric curves of the effects of key variables.

More recently Battya, Raghavan and Seetharamu [2] have performed a similar type of analysis. Again the assumptions of infinite, isothermal continuous phase have been invoked, but they included a downward velocity for this fluid.

Little analytical insight about the overall performance of complex systems has been given. Only the work of Jacobs and Golafshani [3], which addresses liquid/liquid systems, comes to mind. Designers are without good numerical models to describe three-phase spray columns.

The work reported here is an attempt to develop a calculational procedure to describe the flow and heat transfer of a direct contact spray column, and to use this model to investigate the effects of different variables on the total heat transfer of the column. A better understanding of parametric effects of the physical parameters on heat transfer in the column will hopefully help to design the next generation of direct contact heat exchangers.

MODEL DEVELOPMENT

The development of the governing equations for the numerical model starts with the continuity equations. Using the assumption that the mass flow rates for both continuous and dispersed phases remain constant (this assumes, for example, that there is none of either the dispersed or continuous phase that evaporates or goes into solution in the other), the continuity equations can be written as;

$$m_d = \rho_d A \phi U_d \quad (1)$$

$$m_c = \rho_c A (1-\phi) U_c \quad (2)$$

where m_d , ρ_d and U_d are the mass flow rate, density and velocity of the dispersed fluid, respectively. A subscript c refers to continuous fluid. The holdup ratio, ϕ , is the volume of the dispersed fluid per unit volume of the total fluid.

A momentum equation for the total flow can be written as

$$\frac{d}{dz} [\rho_d U_d^2 \phi] + \frac{d}{dz} [\rho_c U_c^2 (1-\phi)] = -\frac{dP}{dz} - [\rho_c (1-\phi) + \rho_d \phi] g \quad (3)$$

where z is the vertical coordinate measured from the bottom of the column. The variables P and g represent local pressure and gravitational acceleration, respectively. All of the other variables are the same as used in the continuity equations. Substituting the continuity equations (Equations 1 and 2) into Equation (3), the momentum equation becomes

$$\frac{dP}{dz} = \frac{m_d}{A} \frac{dU_d}{dz} - \frac{m_c}{A} \frac{dU_c}{dz} - [\rho_c (1-\phi) + \rho_d \phi] g \quad (4)$$

Note that the third term on the right-hand side of the equation shows the hydrostatic pressure difference. The first two terms in the right-hand side indicate the effect of velocity changes on the pressure change.

Similarly the energy equations governing one-dimensional steady-state flow for the dispersed and continuous fluids can be written as;

$$\frac{d}{dz} [\rho_d \phi U_d h_d] = \frac{Q_d}{V} \quad (5)$$

$$\frac{d}{dz} [\rho_c (1-\phi) U_c h_c] = -\frac{\eta Q_d}{V} \quad (6)$$

In these equations Q_d indicates the heat transferred to the dispersed liquid and η defines a heat loss coefficient. For the case of heat loss to the surroundings, η will be greater than unity. It is assumed that the heat loss originates from the continuous phase. Substituting the continuity equations into Equations (5) and (6) results in the following relationships.

$$\frac{dh_d}{dz} = \frac{A}{m_d} \frac{Q_d}{V} \quad (7)$$

$$\frac{dh_c}{dz} = -\frac{A}{m_c} \frac{\eta Q_d}{V} \quad (8)$$

The heat transfer from the bubble, Q_d , is calculated from the following equation [4] along with the area of the droplets, and the difference of the mean temperatures of the dispersed and continuous phases at a given height:

$$h = k_1 / (2R) (Pr_c Re_c)^{1/3} \cdot (0.466 (\pi - \beta + 0.5 \sin 2\beta)^{2/3} + (5 k_v / R) \cdot (1 - 0.466 (\pi - \beta + 0.5 \sin 2\beta)^{2/3}) \quad (9)$$

This equation applies to the heat transferred ultimately to the vaporizing fluid. The development of the equation assumes that the droplet is spherical and the droplet surface is rigid. An additional term from that given in [4] accounts for the heat transfer directly to the vapor is based upon a transient conduction formulation [5]. The properties of quantities denoted with the c subscript are for the continuous phase, while the diameter of the droplet is used in the Reynolds number.

Empirical relations are used to solve for the velocity and droplet diameter. One of the relationships for the velocity difference (between the dispersed and continuous phases) variation was given by Raina and Grover [6] in the following form:

$$U = \frac{1.1547 \left[\left(1 - \frac{\rho_d}{\rho_c} \left(\frac{D_0}{D} \right)^3 \right) \left(\frac{D g}{C_d} \right) \right]^{1/2} D^{5/4} \left(\frac{1}{T_c} \right)}{\left[\frac{T_c^2 + T_d^2}{2 T_c T_d} \right] \left[\frac{C_p \mu_c}{k_c} \right]^{1/3}} \quad (10)$$

The variable C_d in Equation (10) is the drag coefficient. To calculate the drag coefficient, an experimentally determined curve-fitting of

Data for a solid sphere is combined with a correlation for Stokes formulation for a drag force [7]. The resultant equation is given by

$$C_D = \left[\frac{24}{Re} + \frac{6}{1 + Re^{1/2}} + 0.4 \right] \left[\frac{1 + \frac{2\mu_c}{3\mu_d}}{1 + \frac{\mu_c}{\mu_d}} \right] \quad (11)$$

The last relation needed is derived from the continuity equation. Here the mass flow rate of the dispersed phase at any location can be found from the mass flow rate at the bottom of the column. The radius of the droplets is given as

$$R = R_o \left[\frac{n_{do} \rho_{do} U_{do}}{n_d \rho_d U_d} \right]^{1/3} \quad (12)$$

where o indicates the initial conditions and n_b is the number of droplets per unit volume for a given distance. n_b is given by

$$n_b = \frac{\phi}{4 \pi R^3/3} \quad (13)$$

If there is no coalescence or breakup of the bubbles, n_b is constant everywhere in the column. Therefore Equation (12) reduces to:

$$R = R_o \left[\frac{\rho_{do} U_{do}}{\rho_d U_d} \right]^{1/3} \quad (14)$$

With these additional relationships, the appropriate number of equations is obtained to solve the system to examine effects in a three-phase spray-column heat exchanger.

The computational order progresses in the following manner. First the velocity of the dispersed phase is calculated. Then the holdup ratio and the continuous phase velocity are found from the continuity equation. Then the droplet radius is found. Initial derivatives of the velocities are assumed to be zero and new values are calculated from the stepwise change. For example, the dispersed phase velocity variation is found from:

$$\frac{\Delta U}{\Delta z} = \frac{U_{d,i} - U_{d,i-1}}{\Delta z} \quad (15)$$

The momentum and energy equations are solved by using

$$P_i = P_{i-1} + \Delta z \left[\frac{dP}{dz} \right]_i \quad (16)$$

$$h_{c,i} = h_{c,i-1} + \Delta z \left[\frac{dh_c}{dz} \right]_i \quad (17)$$

$$h_{d,i} = h_{d,i-1} + \Delta z \left[\frac{dh_d}{dz} \right]_i \quad (18)$$

Functional values of pressure and enthalpies are calculated from these equations. In the boiling region the quality of the dispersed phase is calculated from the following equation

$$x_i = x_{i-1} + \Delta z \left[\frac{dx}{dz} \right]_i \quad (19)$$

The actual enthalpy of the point is found from the thermodynamic relations of the saturation state. All thermodynamic properties are calculated by using subroutines developed at the University. Transport properties such as surface tension and conductive heat transfer coefficients are supplied to the model as temperature dependent curvefits from published values. Viscosities are taken as temperature and pressure dependent curvefits. Transport properties of pentane, butane and isobutane are built into the model. The properties of additional fluids can also be easily added to the existing list of the fluids.

More details about the model development are given in the paper by Coban and Boehm [5]. Comparisons are given there between computed temperature profiles and experimentally-determined counterparts [8]. Excellent agreement is shown both in overall temperature magnitudes as well as the locations where phase change begins.

EXPERIMENTS

Experiments were performed in a 0.61 m diameter tower that measured 6.1 m high and was made entirely of conventional pipe. In all tests reported here an active tower height of approximately 3 m was used. Water entered the tower through an inlet nozzle at about the middle of the column and flowed downward, exiting through the bottom. Commercial grade n-pentane was used as a dispersed fluid. Pentane entered the column from the bottom of the column through a multiple showerhead arrangement. Pentane flowed upward in the column in droplet form while vaporizing, and the pentane vapor exited at the top of the column. Design of the showerhead was such that initial pentane droplet diameters of approximately 3-4 mm were produced. Operating pressures of approximately 2-3 atmospheres were normally used in the column. Both the pentane and water loops to the direct contact heat exchanger were completely closed other than in the column where they came in contact.

Orifice plates were used to measure the flow rates of the hot water and the liquid pentane. Pressure transducers were connected in parallel with manometers to indicate pressure drops. Temperature measurements were accomplished with the use of shielded Chromel-Alumel thermocouples. Twenty four thermocouples were used to determine temperatures throughout the system, including the inlet and outlet stream temperatures as well as the bulk temperatures at various locations within the column.

Data analysis for the heat transfer calculation used the following energy balance equation

$$m_{v,i}^1 h_{v,i}^1 + m_{p,i}^1 h_{p,i}^1 = m_{p,o}^v h_{p,o}^v + m_{v,o}^1 h_{v,o}^1 + (m_{v,i}^1 - m_{v,o}^1) h_{v,o}^1 + Q_{loss} \quad (20)$$

Subscripts i and o represent inlet and outlet conditions, and w and p stand for pentane and

water, respectively. Superscripts l and v represent the liquid and vapor cases. Q_{loss} is the heat loss from the vessel. In order to determine the water mass flow rate, it is assumed that partial pressures of water and pentane vapor are proportional to mole flow rates of the phases. The partial pressure of the pentane at the top of the column was inferred by assuming that the pentane existed at a saturated condition at the exit temperature. Thus

$$\frac{P_{vt}(T_{po})}{P_{pt}} = \frac{(m_{vl}^1 - m_{vo}^1) / M_v}{m_{po}^v / M_p} \quad (21)$$

Here M_v and M_p are the molecular weights of water and pentane, respectively.

More information on the experimental work is available in the paper by Goodwin, et al. [8].

RESULTS AND DISCUSSION

The influence of various parameters on total heat transfer during the process of direct contact evaporation has been investigated. It is necessary to ascertain how well the heat transfer model developed here describes the actual performance of such a device. Steady-state data are available for the temperature profiles from the data taken as the experimental part of this project [8]. Experimental heat transfer was calculated from this data by the equations given previously. A numerical model is developed, and the effect of several physical parameters on the total heat transfer is investigated. All values shown refer to the same basic experimental situation, that of a 0.61 m diameter, 3 m high spray column using water at 85°C to evaporate pentane.

In Figure 1 relations between initial dispersed phase temperature and total heat transfer are given for two different initial pressures. As is seen from the figure, the total heat transfer decreases when the temperature increases, and it increases when the pressure increases. Available experimental data [8] are also shown on this plot. The data predicted by the model are nearly the same as those from the experimental situation.

Figure 2 shows a limited number of calculations used to infer the relation between total heat transfer and mass flow rate of the water. The data indicate an increasing dependence of total heat transfer on the water mass flow rate.

Figures 3 and 4 show the effects of mass flow rates of the dispersed phase, initial pentane temperature and total heat transfer relations. Note that all parameters are the same between these two figures except that the continuous phase mass flow rate is greater in Figure 4. The same kind functional dependence is observed. Total heat transfer decreases linearly with an increasing initial dispersed phase temperature. A nonlinearly increasing relation with dispersed phase mass flow rate is also seen. For these cases the agreement between the experimental data and the model prediction is very good.

The effect of the initial temperature of

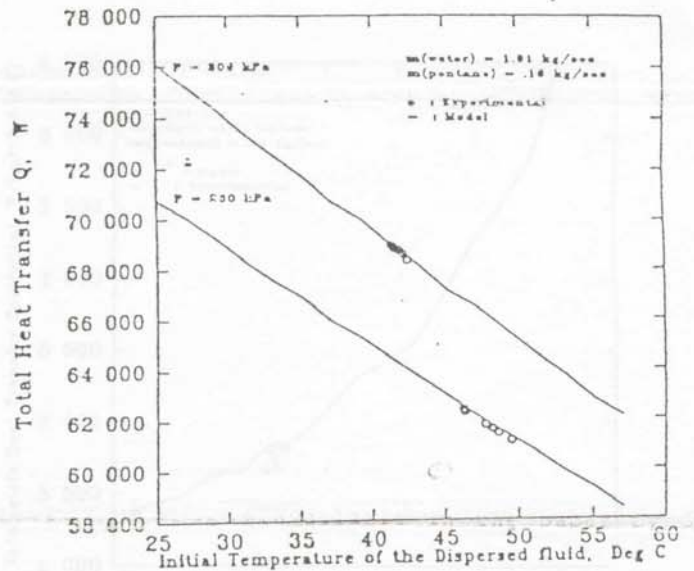


Figure 1. Effects of initial dispersed phase temperature on the total heat transfer accomplished, for two vessel pressures.

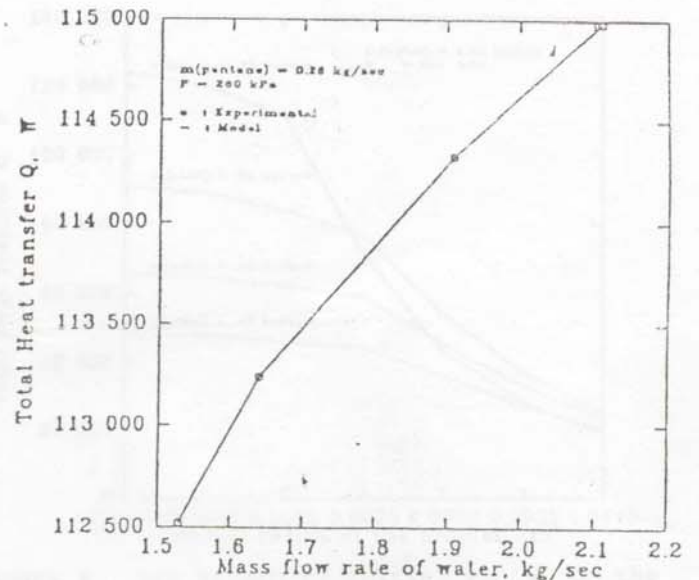


Figure 2. Variation of total heat transfer with continuous phase mass flow rate for specific values for pentane mass flow and vessel pressure.

the dispersed phase on the total heat transfer coefficient is shown in Figure 5. Disjointed variations in slopes are a result of variations in the way the thermodynamic properties were evaluated. The volumetric heat transfer coefficient is defined here as the total heat transfer per unit volume divided by the logarithmic mean temperature difference (LMTD). The LMTD is used here because of the previous application of this very simple concept to this admittedly complex situation. An exponential increase in the heat transfer coefficient is observed by increasing the inlet dispersed fluid temperature. Considering a decreasing total heat transfer for the same range of data leads to the conclusion that the volumetric

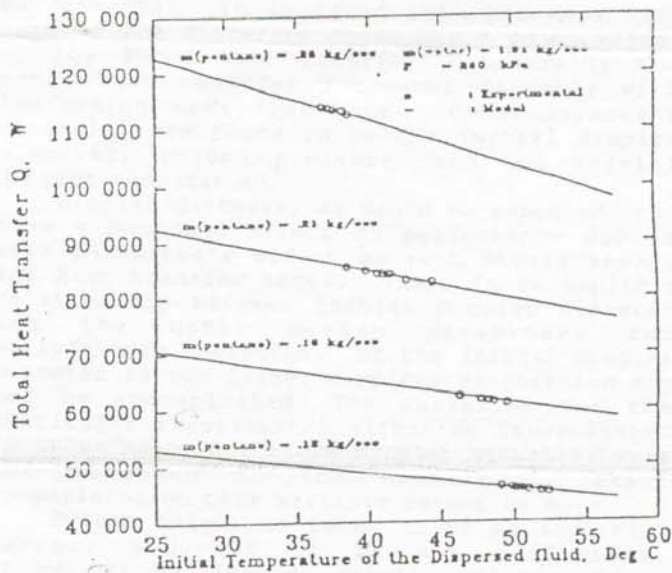


Figure 3. The influence of dispersed phase (pentane) mass flow and initial temperature on total heat transfer accomplished, for a continuous phase flow rate of 1.92 kg/sec.

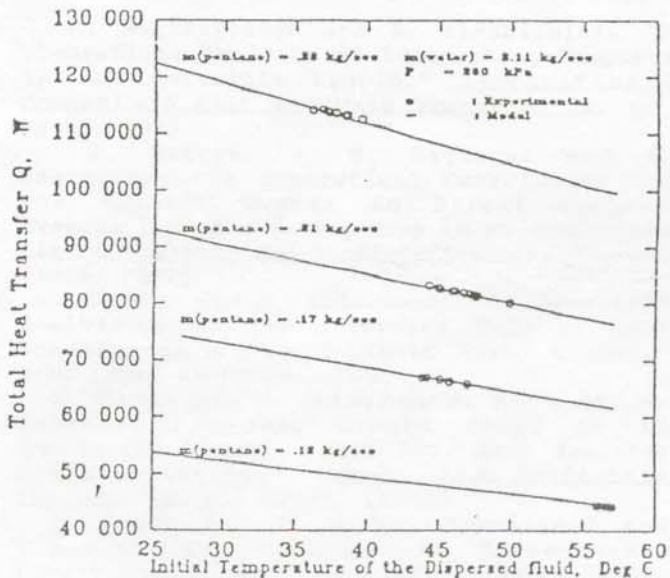


Figure 4. The influence of dispersed phase (pentane) mass flow and initial temperature on total heat transfer accomplished, for a continuous phase flow rate of 2.11 kg/sec. Comparing this figure to Figure 3 demonstrates the effects of the continuous phase mass flow.

heat transfer coefficient will not always reflect the general variations of the total heat transfer.

The effect of initial radius of the droplet on total heat transfer is shown in Figure 6. It is clearly seen from this figure that the total heat transfer is inversely proportional to the initial droplet radius. The sudden change in the slope of the curves is caused by incomplete boiling that occurs for some critical initial droplet radius for the given column height. This figure also shows that a larger dispersed liquid mass flow rate causes a larger total heat transfer. This was an

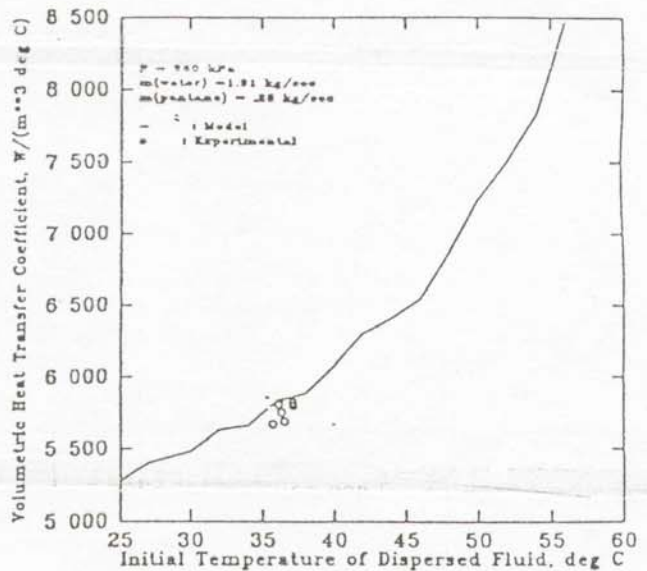


Figure 5. The variation of volumetric heat transfer coefficient with the initial temperature of the dispersed phase (pentane).

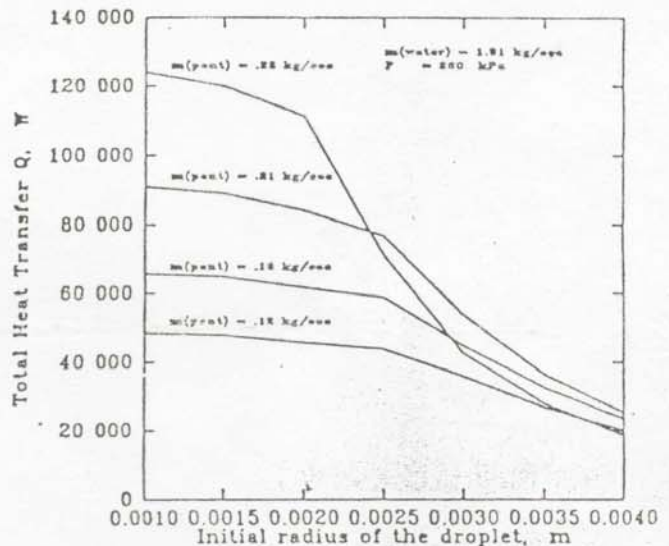


Figure 6. The effect of initial radius of the droplet on the total heat transfer for several pentane mass flow rates.

expected outcome. Comparisons to experiments are not possible here because droplet diameters were not measured in the vessel. However, the numerical model is assumed to be validated at this point.

CONCLUSIONS

A one-dimensional steady-state model of a three-phase spray-column counterflow direct contact heat exchanger has been developed by considering total flow inside of the column. The numerical model was successfully applied to solve for the local variation of heat transfer and fluid motion. This model is validated by showing comparisons to experimental results found from operation of a large column. Effects of several physical variables on the

heat transfer are investigated and compared to experiments. It is found that the mass flow rate of the dispersed phase has a major effect on the local heat transfer. Generally the total heat transfer increases directly with increasing mass flow rate. Other important variables are found to be the initial droplet diameter, column pressure, and the initial droplet temperature.

Droplet diameter, as would be expected, can have a profound effect on performance due to this parameter's effect on both stored energy and heat transfer rates. There is an implicit relationship between initial droplet diameter and the other design parameters for satisfactory operation. If the initial droplet diameter is too large, complete evaporation may not be accomplished. The variation for the particular experimental situation investigated is shown here, but since droplet diameters were not measured in the experiment, exact comparisons on this variable cannot be made.

Column height is found to be an important factor; and, if it is not sufficient, incomplete boiling can result. If the column height is sufficient to complete the boiling, additional column height has only a small effect on the total heat transfer.

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ACKNOWLEDGEMENT

The partial support by the Solar Energy Research Institute and Utah Power & Light Company of the experimental work reported here is gratefully acknowledged.

