JOURNAL OF THERMOPHYSICS AND HEAT TRANSFER
Vol. 20, No. 1, January–March 2006

# Actively Pumped Two-Phase Loop for Spray Cooling

Lanchao Lin*
*Universal Energy Systems, Inc., Dayton, Ohio 45432-1894*
and
Rengasamy Ponnappan† and Kirk Yerkes‡
*U.S. Air Force Research Laboratory, Wright–Patterson Air Force Base, Ohio 45433-7251*

**A new closed two-phase loop that combines with a large area spray cooling unit for the cooling of high heat-flux power sources is developed. The fluid circulation is sustained by a magnetic gear pump operating with an ejector unit. The motive flow of the ejector shares the pumping liquid flow with the multinozzle spray. With the assistance of the ejector, the maximum spray pressure drop across the nozzle can be enhanced by at least 0.56 bar at critical heat fluxes (CHF). This increases CHF of the spray cooling by up to 16%. More importantly, the use of the ejector prevents the uncondensed vapor from entering the magnetic gear pump and stabilizes the circulation of the two-phase flow. During the experiment, a multinozzle assembly with 48 miniature nozzles is employed. The target spray cooling area is 19.3 cm$^2$. FC-72 and water are used as the working fluid. The present design concept can be applied to cooling systems operating in the aerospace environment.**

## Nomenclature

| | | |
|---|---|---|
| $A$ | = | cross-sectional area of flow, m$^2$ |
| $a$ | = | diffuser area ratio, $A_t/A_d$ |
| $b$ | = | jet nozzle area ratio, $A_n/A_t$ |
| $c$ | = | $(A_t - A_n)/A_n$ |
| $D$ | = | diameter, m |
| $K_{\mathrm{en}}$ | = | throat entry friction loss coefficient |
| $K_n$ | = | ejector nozzle friction loss coefficient |
| $K_{\mathrm{td}}$ | = | throat and diffuser friction loss coefficient |
| $M$ | = | liquid flow ratio, $Q_2/Q_1$ |
| $M_L$ | = | cavitation-limited liquid flow ratio |
| $N$ | = | pressure ratio, $(p_d - p_s)/(p_i - p_d)$ |
| $p$ | = | pressure, N/m$^2$ |
| $p_v$ | = | vapor pressure, N/m$^2$ |
| $Q_1$ | = | primary liquid flow rate, gpm |
| $Q_2$ | = | secondary liquid flow rate, gpm |
| $q''$ | = | heat flux, W/cm$^2$ |
| $q''_c$ | = | critical heat flux, W/cm$^2$ |
| $T_{\mathrm{sat}}$ | = | spray saturation temperature, °C |
| $T_w$ | = | cooling surface temperature, °C |
| $t$ | = | time, s |
| $V$ | = | fluid velocity, m/s |
| $Z$ | = | jet dynamic pressure, N/m$^2$ |
| $\rho$ | = | liquid density, kg/m$^3$ |
| $\sigma$ | = | cavitation coefficient |
| $\Delta p$ | = | pressure drop across the spray nozzle, N/m$^2$ |

*Subscripts*

| | | |
|---|---|---|
| $i, s, n$ | = | locations (see Fig. 3) |
| $o, t, d$ | = | locations (see Fig. 3) |

## I. Introduction

TO dissipate heat from current and future high-power directed energy devices for space-based applications, advanced thermal management approaches should be adaptable to high heat-flux acquisition and high heat rate transport in extreme space environment. The directed energy device power levels and repetitive fire rates are dependent on how well the large amount of waste heat can be managed in the system and effectively removed from the system. Evaporative spray cooling has been exhibited to be an effective method of removing high heat fluxes (greater than 500 W/cm$^2$ using water as working fluid) from surfaces with low superheat at a low coolant flow rate. The high thermal performance of the spray cooling has been verified for varied cooling surface areas up to 20 cm$^2$ at pressure drops greater than 1.03 bar (Refs. 1–3). The use of spray-cooling technology for the cooling of the high power sources will make it possible to significantly increase its pulsed peak power and repetitive fire rate. In comparison, a single-phase cooling system using microchannel flow with subcooled nucleate boiling[4] would require seven times more mass flow rate with higher pressure drop for achieving the same heat flux as the spray-cooling experiment. However, to be effective in the space environment, the problem of the pump malfunction should be solved. It has been found that the vapor entering a mechanical pump used in a closed-loop spray-cooling system reduces pumping pressure head or even makes the pump unable to generate any pressure head. This type of pump malfunction is associated with vapor quality at the inlet of the pump. The vapor flowing towards the pump could result from insufficient subcooling of the fluid from the condenser, an insufficient fluid fill amount, and unsteady two-phase flow. One solution to this problem is to apply an ejector connecting with a bypass line. The motive liquid flow of the ejector merges with the liquid flow from the condenser. The use of the ejector prevents the uncondensed vapor from entering the mechanical pump and stabilizes the circulation of the two-phase flow. An alternative solution to the circulation of two-phase flow in the closed loop was the use of a gas compressor to pressurize the vapor that could be used to atomize the spray.[5] The present paper deals with the pumping enhancement caused by the use of the ejector. The thermal testing result of the large area spray cooling is presented at an enhanced spray pressure drop.

## II. Spray-Cooling System with an Enhanced Pumping Capability

Figure 1 shows the schematic of the experimental setup with a mechanically pumped two-phase loop coupled with the spray-cooling unit. The closed loop consists of a preheater, a spray chamber

Received 13 December 2004; presented as Paper 2005-0381 at the AIAA 43rd Aerospace Sciences Meeting and Exhibit, Reno, NV, 10–13 January 2005; revision received 19 April 2005; accepted for publication 5 April 2005. 

*Research Scientist, 4401 Dayton-Xenia Road; currently Senior Research Engineer, University of Dayton Research Institute, Dayton, Ohio 45469-0179. Member AIAA.

†Senior Mechanical Engineer, Electrochemistry and Thermal Sciences Branch (AFRL/PRPS), Propulsion Directorate. Associate Fellow AIAA.

‡Senior Research Engineer, Electrochemistry and Thermal Sciences Branch (AFRL/PRPS), Propulsion Directorate. Senior Member AIAA.

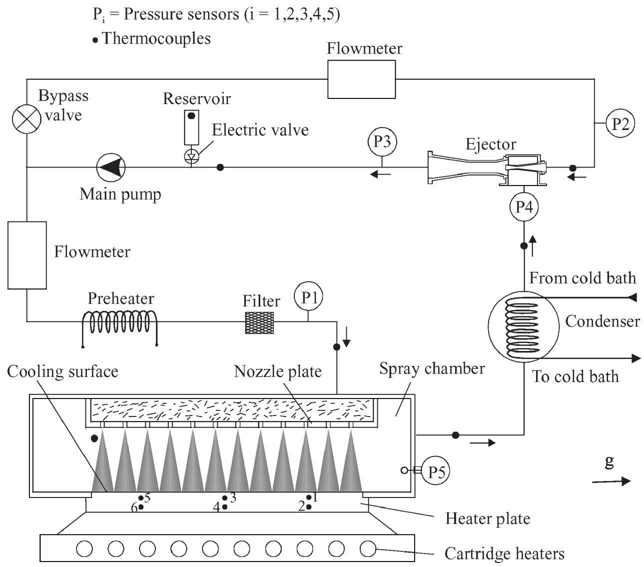


**Fig. 1 Schematic of the test setup with an ejector bypass line.**

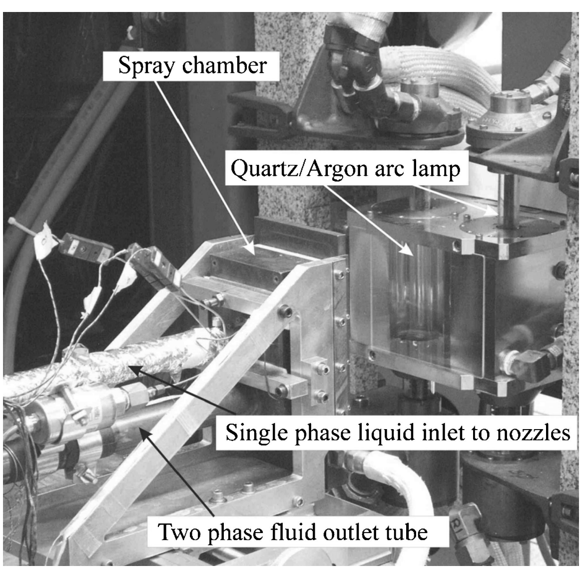


**Fig. 2 Test article facing the Vortek plasma heater with two high-power quartz lamps.**

housing a multinozzle assembly with 48 miniature nozzles (forming a $4 \times 12$ nozzle array), a heater assembly with a cooling surface area of $2.54 \times 7.60$ cm$^2$, a coaxial coil-type condenser, an ejector unit with the motive flow from the bypass loop, a magnetic gear pump, a liquid reservoir for liquid charge, and a filter. A cold bath is used to supply the cooling water to and from the condenser. The inlet port of the magnetic gear pump is connected with an extension of the discharge port of the ejector. Between the pump and ejector, there is a section of transparent tube for observation of the flow towards the pump. FC-72 and water are used as the working fluid. The closed-loop system is evacuated before filled with the working fluid. The two-phase fluid from the spray chamber flows into the condenser where the vapor condenses. The motive flow of the ejector drives the subcooled liquid from the condenser, and they mix in the ejector. The merging liquid flow is then pressurized through the magnetic gear pump. The liquid from the pump is divided into the flow for the spray cooling and the flow as the motive liquid for the ejector. For the cooling of high heat fluxes greater than 400 W/cm$^2$, water is used as the working fluid, and a Vortek plasma heater is used as the heat source (other than the cartridge heater for testing with FC-72). Figure 2 shows the test article facing the Vortek plasma heater with two high-power quartz lamps. The spray-cooling surface is vertically placed.

The individual nozzle configuration for the present test is similar to the previous one.[6] The spray nozzle discharge orifice diameter, the distance between two nozzles, and the spray distance are 0.25, 6.35, and 10 mm, respectively. The overall spray chamber space dimensions are 26.2 mm (high), 118 mm (long), and 72.0 mm (wide). The spray droplets partially vaporize upon reaching the cooling surface, removing heat mainly through phase change. In the spray chamber, there is a side channel surrounding the spray nozzles to effectively remove the two-phase fluid away from the cooling surface. The two-phase fluid in the side channel is driven to the outlet channel by the spray momentum-induced pressure drop and also because of the suction effect created by the ejector pump. A multinozzle spray performance test using FC-72 has shown that at a pressure drop between 1.72 to 1.86 bar the spray array just inscribes the rectangular cooling surface area (19.3 cm$^2$) at the spray distance of 10 mm. This spray impact pattern on the cooling surface has been considered being desirable for the spray thermal performance.[7] Overlapped or sparser spray cones can result in a lower critical heat fluxes (CHF) of the spray cooling.

The liquid flow rates in the spray loop and the bypass line are measured using two turbine flow meters. All pressures are measured using pressure sensors. The spray pressure drop, $\Delta p = p_1 - p_5$, is controlled by the power input to the pump and the bypass valve. The spray chamber pressure $p_5$ corresponds to the spray saturation temperature $T_{\mathrm{sat}}$. The fluid temperatures in the spray-cooling system are measured using type-T probe thermocouples. The liquid temperature at the inlet of spray chamber is regulated by adjusting the cold bath temperature and input power to the preheater. The ejector performance is characterized with three pressure sensors at each port of the ejector and two flowmeters in the spray loop and the bypass line. During the test, the pressure drop across the spray nozzles is varied from 1.03 to 2.28 bar for water and to 3.80 bar for FC-72.

Six thermocouples are embedded in 0.58-mm holes drilled along two planes in the heater plate, forming three pairs of thermocouples for input power measurement. The distance between two thermocouples in each pair is 5.33 mm. The distance between the cooling surface and the upper plane of the thermocouple locations is 2.85 mm. The thermocouple bead diameter is 0.3 mm. The heat rate is calculated using the average temperature difference between the two thermocouple location planes, the interface area of the heater plate with the heat source and the thermocouple pair distance. The average temperature on the cooling surface is estimated through the extrapolation of the thermocouple readings across the two planes. During the test, the input power is varied from 200 W up to the amount relating with CHF. All measured quantities reflected by voltage signals are collected in a data-acquisition/switch unit and converted into digital signals that are transferred to a PC for monitoring and recording.

The data-acquisition unit and the thermocouples are calibrated, and the system accuracy is found to be within $\pm 0.2^\circ$C over the range of interest. The accuracy of the distances between two thermocouples in each pair and between the cooling surface and its closer thermocouple locations in the heater plate is $\pm 0.2$ mm. The uncertainty of the heat flux is 12% at $q'' = 10$ W/cm$^2$, which is the smallest experimental value used. The uncertainty of the hot surface temperature $T_w$ is estimated within $3.1^\circ$C at $q'' = 500$ W/cm$^2$, which is the upper limit in the present test. The effect of the temperature gradient across the thermocouple beads in the heater plate is not considered in the uncertainty analysis. The accuracies of the pressure sensors reading are $8.6 \times 10^{-3}$ bar. The uncertainty of $T_{\mathrm{sat}}$ is estimated within $0.3^\circ$C. The turbine flowmeter is calibrated. The uncertainty of the flow rate is 3% of reading.

## III. Ejector Flow Model

The liquid-jet pump transfers energy from the primary (motive) liquid to the secondary (suction) fluid (the same liquid in the present case). The ejector is used to generate the suction effect at the end of the condenser, and this enhances the pumping capability of the spray loop. The primary drawback is efficiency. Both friction losses and unavoidable mixing losses are incurred. Nevertheless, careful

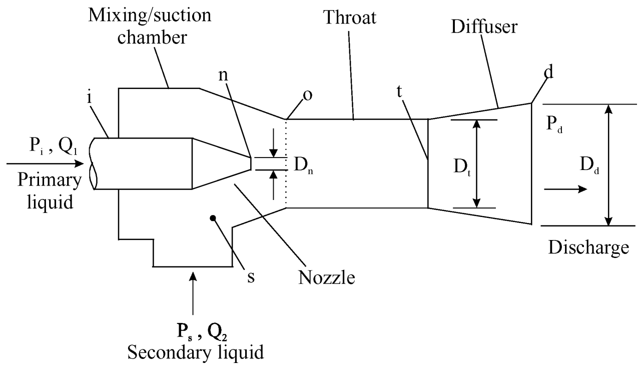


**Fig. 3 Liquid-jet ejector configuration.**

design can produce pumping effect with efficiencies on the order of 30 to 40%. A typical liquid-jet ejector is shown in Fig. 3.

The liquid-jet ejector model is based on one-dimensional flow analyses.[8] The ejector performance can be characterized by the pressure ratio $N$, which is defined as

$$N = (p_d - p_s)/(p_i - p_d) \tag{1}$$

and the liquid flow ratio $M$, which is defined as

$$M = Q_2/Q_1 \tag{2}$$

Using the one-dimensional model and assuming the same liquid density for the primary flow and secondary flow, the pressure ratio can be calculated by

$$N = \left[2b + M^2\left(\frac{2b^2}{1-b} - \frac{1+K_{\text{en}}}{c^2}\right) - b^2\left(1 + K_{\text{td}} + a^2\right) \times (1+M)^2\right] \Big/ \left[1 + K_n - 2b - m^2\left(\frac{2b^2}{1-b} - \frac{1+K_{\text{en}}}{c^2}\right) + b^2\left(1 + k_{\text{td}} + a^2\right)(1+M)^2\right] \tag{3}$$

The recommended values for the friction loss coefficients are $K_n = 0.05$, $K_{\text{td}} = 0.2$, and $K_{\text{en}} = 0$ (Refs. 8 and 9). The value of $K_{\text{td}}$ can range from 0.17 to 0.4.

The ejector can encounter cavitation that occurs in the mixing throat. In the case of cavitation, the throat inlet pressure is reduced to the vapor pressure $p_v$ of the secondary liquid. Any further drop in the backpressure has no effect on the liquid flow ratio that reaches the cavitation-limited liquid flow ratio $M_L$. The value of $M_L$ can be predicted as follows[10].

$$M_L = c\sqrt{(p_s - p_v)/\sigma Z} \tag{4}$$

where the cavitation coefficient $\sigma$ is recommended to be 1.35 and the jet dynamic pressure $Z$ is given by

$$Z = \rho_1 V_n^2 \big/ 2 \tag{5}$$

The occurrence of cavitation will cause large departure of the measured pressure ratio $N$ from the calculated curve.

## IV. Results and Discussion

### A. Ejector Performance

A positive result of maintaining a sound fluid circulation by using the ejector unit is obtained. For appropriately adjusted values of $M$, no vapor entering the magnetic gear pump is observed at varied heat rates up to CHF and at different nozzle pressure drops up to 3.80 bar for FC-72 and to 2.28 bar for water (at CHF). On the contrary, in the previous spray-cooling tests without the ejector unit the vapor could occasionally enter the pump along with the circulating liquid at spray pressure drops above 1.03 bar for water. The comparison shows that using the ejector in the closed two-phase loop enhances the capability of maintaining the two-phase fluid circulation.

The ejector pressure characteristics $N$ in relation with the liquid flow ratio $M$ for different values of $K_{\text{td}}$ are calculated, and the results are presented in Fig. 4. Also presented in this figure are the experimental data points reflecting $N$. During the test with FC-72 as the working fluid, $M$ is widely varied to find the relation between $N$ and $M$. With water as the working fluid, however, $M$ is adjusted only between 1.05 and 1.12. The liquid flow ratios of all of the data points in Fig. 4 are below the corresponding values of $M_L$. The data point for water is slightly higher than the predicted value with $K_{\text{td}} = 0.20$. Most of the data points for FC-72 are lower than the predicted curve with $K_{\text{td}} = 0.20$ and fall in the region between $K_{\text{td}} = 0.20$ and 0.40. This means that in most cases with FC-72 the ejector efficiency is lower than the recommended one (with $K_{\text{td}} = 0.20$).

Figure 5 shows a phenomenon of unstable flow during the test at a high liquid flow ratio of $M = 1.5$ for FC-72. Both spray flow rate and bypass flow rate oscillate at $M = 1.5$, but they are stabilized at $M = 1.25$. More acquired experimental data points support the fact that to prevent the uncondensed vapor from entering the magnetic gear pump the liquid flow ratio needs to be adjusted between $M = 0.7$ and 1.25.

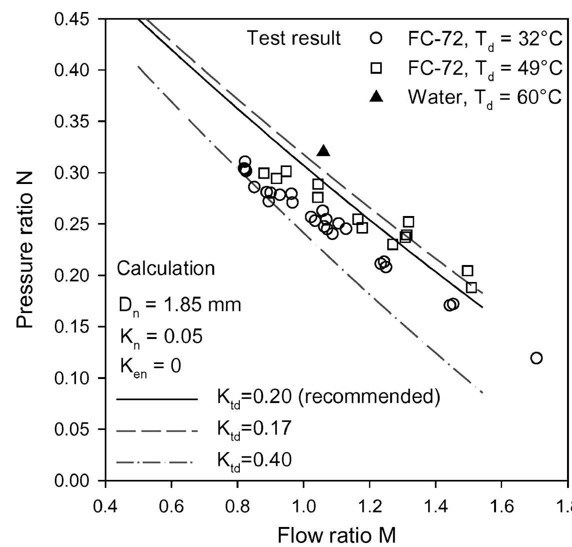


**Fig. 4 Ejector pressure characteristic *N* as the function of the flow ratio *M*.**

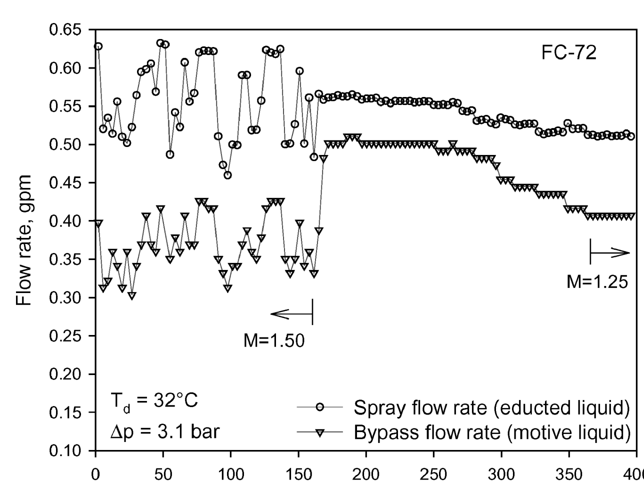


**Fig. 5 Flow rate oscillating during the test at a high flow ratio of *M* = 1.5 and being stabilized at *M* = 1.25.**

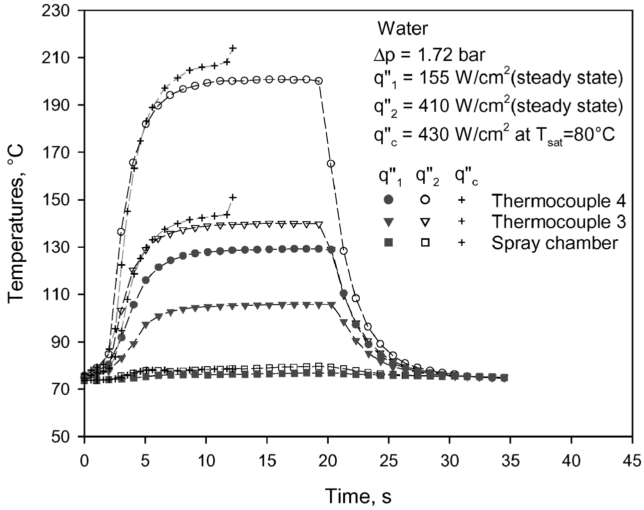


**Fig. 6 Response of thermocouples in the heater plate and spray chamber to a pulsed heat load at $\Delta p = 1.72$ bar.**

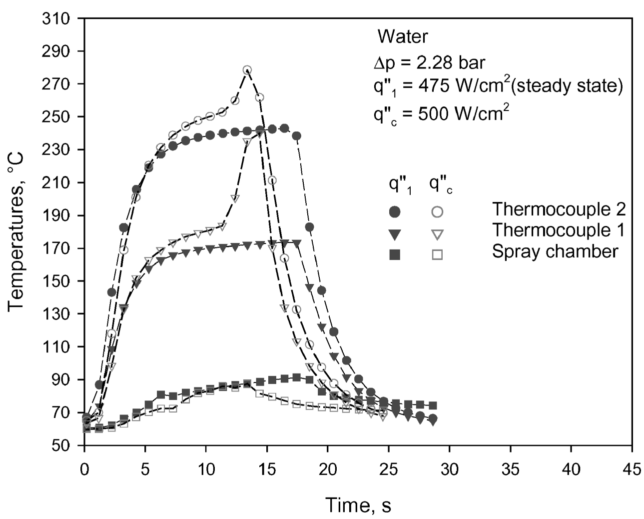


**Fig. 7 Response of thermocouples in the heater plate and spray chamber to a pulsed heat load at $\Delta p = 2.28$ bar.**

### B. Thermal Response of the Test Article

If the ejector is not used, the maximum nozzle pressure drop that the loop system with water can reach is 1.72 bar. In this case, the thermal response characteristics of the spray cooling with water to the pulsed high flux radiant heat are shown in Fig. 6. The spray saturation temperature is 80°C. The locations of the thermocouples 3 and 4 are shown in Fig. 1. During the transient process, the change in the spray chamber temperature is relatively small for the varied heat-flux levels of 155, 410, and 430 W/cm$^2$. The steady state is achieved in about 8 s after energizing the plasma heater. The critical heat flux $q_c''$ is 430 W/cm$^2$.

With the assistance of the ejector, the maximum spray pressure drop across the nozzle can be enhanced up to 2.28 bar for water. At the enhanced nozzle pressure drop, the thermal response of the thermocouples in the heater plate and spray chamber is presented in Fig. 7. In this case, CHF of the spray cooling reaches 500 W/cm$^2$ increasing by 16% compared to the case of 1.72 bar. More importantly, the use of the ejector prevents the uncondensed vapor from entering the magnetic gear pump and stabilizes the circulation of the two-phase flow. At heat fluxes above 400 W/cm$^2$ and below CHF, the heat-transfer coefficient $q''/(T_w - T_{\mathrm{sat}})$ is higher than 110 kW/m$^2$K.

## V. Conclusions

1) The use of the ejector prevents the uncondensed vapor from entering the magnetic gear pump, enhances the pumping capability, and stabilizes the circulation of the two-phase flow.

2) To effectively prevent the vapor from coming into the pump, the liquid flow ratio should be controlled within the range from $M = 0.7$ to 1.25.

3) With the assistance of the ejector, the maximum spray pressure drop across the nozzle can be enhanced by at least 0.56 bar at CHF. This increases CHF of the spray cooling by up to 16%.

4) The spray-cooling system with a large cooling area of 19.3 cm$^2$ reaches CHF of 500 W/cm$^2$ for water as the working fluid.

5) The present design concept can be applied to cooling systems operating in the aerospace environment.

## Acknowledgments

This research was supported by the Propulsion Directorate of the U.S. Air Force Research Laboratory (AFRL), Wright–Patterson Air Force Base, Ohio. The test was performed at the Thermal Laboratory of the Propulsion Directorate and the Vortek Radiant Heating Facility of the Air Vehicles Directorate. The authors wish to thank Roger P. Carr [Universal Energy Systems (UES)], Sean Coghlan (AFRL), and Larry Kretz (AFRL) for their efforts in setting the data-acquisition system and coupling the large area spray-cooling system with the plasma heater. The authors also wish to thank John E. Tennant (UES) for making the spray-cooling systems.

## References


[1] Lin, L., Ponnappan, R., Yerkes, K., and Hager, B., "Large Area Spray Cooling," AIAA Paper 2004-1340, Jan. 2004.

[2] Lin, L., Ponnappan, R., Yerkes, K., Coghlan, S., and Kretz, L., "Spray Cooling of High Power Sources," 17th Solid State and Diode Laser Technology Review, Directed Energy Professional Society, Albuquerque, NM, June 2004.

[3] Rini, D. P., Chow, L., and Bass, M., "Lightweight Cooling Systems for Solid State Lasers," 16th Solid State and Diode Laser Technology Review, Directed Energy Professional Society, Albuquerque, NM, May 2003.

[4] Vrable, D. L., and Donovan, B. D., "Thermal Management for High Power Microwave Sources," 1st International Energy Conversion Engineering Conference, 2003; also AIAA Paper 2003-5957, Aug. 2003.

[5] Gruen, A., "New Technique Cools High-Power Solid-State Lasers," *The MDA Update*, Spring 2004, p. 10.

[6] Lin, L., and Ponnappan, R., "Heat Transfer Characteristics of Spray Cooling in a Closed Loop," *International Journal of Heat and Mass Transfer*, Vol. 46, No. 20, 2003, pp. 3737–3746.

[7] Mudawar, I., and Estes, K. A., "Optimizing and Predicting CHF in Spray Cooling of a Square Surface," *Journal of Heat Transfer*, Vol. 118, No. 3, 1996, pp. 672–679.

[8] Gunningham, R. G., "Liquid Jet Pump Modeling: Effects of Axial Dimensions on Theory-Experiment Agreement," *Proceedings of the 2nd Symposium on Jet Pumps and Ejectors, BHRA Fluid Engineering*, Cranfield, Bedford, England, 1975.

[9] Gunningham, R. G., "Liquid Jet Pumps for Two-Phase Flows," *Journal of Fluids Engineering*, Vol. 117, No. 2, 1995, pp. 309–316.

[10] Gunningham, R. G., Hansen, A. G., and Na, T. Y., "Jet Pump Cavitation," *Journal of Basic Engineering*, Vol. 92, Sept. 1970, pp. 483–494.