Research News

High Heat Flux Cooling by Liquid Jet-Array Modules

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Liquid jet impingement has a demonstrated capacity for high heat flux cooling. Here, we describe the use of a water jet array in removing heat fluxes of 1 to 17 MW/m^2 over an area of 10 cm^2. The jet array size may be increased for thermal management of larger areas. Cooling in this system is by single-phase convection. Metal film resistance heaters have been developed for testing purposes.

1 Introduction

Heat removal at high power density is a continually challenging problem. Traditionally, the highest heat fluxes have been removed by flow boiling systems, for which critical heat flux is the usual limiting factor. Indeed, the highest steady-state heat fluxes obtained have generally been CHF values for tube flows of water, with maximum critical fluxes in the range of 100 to 350 MW/m^2. These heat loads can generally be supported over only a small length of the tube.

Recently, jet impingement systems have been investigated as alternatives to tube flow systems for the highest heat fluxes. Liu and Lienhard [1] used high-velocity water jets to cool a small heated region, with fluxes ranging between 50 and 400 MW/m^2. They confined the heating to the high-pressure stagnation region of a 1.9 mm diameter jet. Jet velocities ranged from 50 to 134 m/s, and associated stagnation pressures were between 12 and 89 atm. Their heat fluxes included the highest steady-state fluxes reported to date. The fluxes obtained were generally limited by the heat source used or by the ability of the heat transfer surface to support large thermal stresses, and the data showed no evidence of critical heat flux limitations.

In either of the above situations, the heat loads were supported over only a small area, but in many circumstances it is desirable to carry large fluxes over larger areas. In this note, we report a cooling module designed to carry heat fluxes of up to 20 MW/m^2 over areas of 10 cm^2 or more [2,3]. The module uses an array of impinging liquid jets to convectively cool the rear of a faceplate whose forward surface is subjected to an imposed heat load.

Experimental studies of very high heat fluxes can often be limited by the availability of a suitable heat source, particularly when it is desired to impose a fixed heat flux upon a solid surface. Such solid surfaces usually represent the main surface of a cooling system and separate the cooling fluid from the environment that delivers the heat load. In practice, these large heat fluxes arise when plasmas, lasers, or X-rays interact with the cooling surfaces. In the case of plasma fusion systems, fluxes have design levels near 5 to 30 MW/m^2 over areas of several square meters, and the transferred heat is used to drive power conversion cycles [4]. In the case of synchrotron X-ray monochrometers, the fluxes may exceed 90 MW/m^2 over areas of square millimeters and represent undesired heat loads that must be removed to maintain system performance [5].

Large carbon dioxide lasers, electron beams, plasma arcs, and linear accelerators have all been used to deliver heat fluxes in this range for experimental testing [1,6,7], but such methods are either very expensive or produce spatially nonuniform heat loads (generally Gaussian distributed) that may not represent actual operating conditions. An alternative method is to form an electrical resistance heater atop the test surface [8,9]. In the present work, we have deposited a thin metal film onto the test surface using thermal spraying; this film serves as a resistance heating element that can provide a uniform and easily controlled flux. A thin dielectric layer is sprayed first, so as to isolate the heating element from the metallic faceplate surface. High current, low voltage DC power is used to drive the heater. The direct deposition of the films onto the test surface, as opposed to a mechanically attached heater, helps to minimize interfacial contact resistances which can cause large temperature differences at these high flux levels.

2 Cooling Module Configuration

The cooling module is shown in Fig. 1. The module consists of an externally heated faceplate and two stainless steel manifolds connected through an array of 14 tube nozzles. The lower manifold is at a higher pressure than the upper manifold, and liquid driven through the tube nozzles impinges on the faceplate that caps the upper manifold, cooling it.

The high-pressure manifold is tapered, to limit streamwise variations in dynamic pressure as liquid exits the manifold; this manifold receives water at 2 MPa (300 psi) from a 5 cm ID pipe. The tube nozzles are 5 cm long and are threaded into a 6 mm thick stainless steel plate separating the manifolds. The nozzles have an inner diameter of 2.78 mm and an outer diameter of 6.35 mm. The 14 nozzles have a center-to-center separation of 10 mm on a hexagonal planform. The outlets of the nozzles, in

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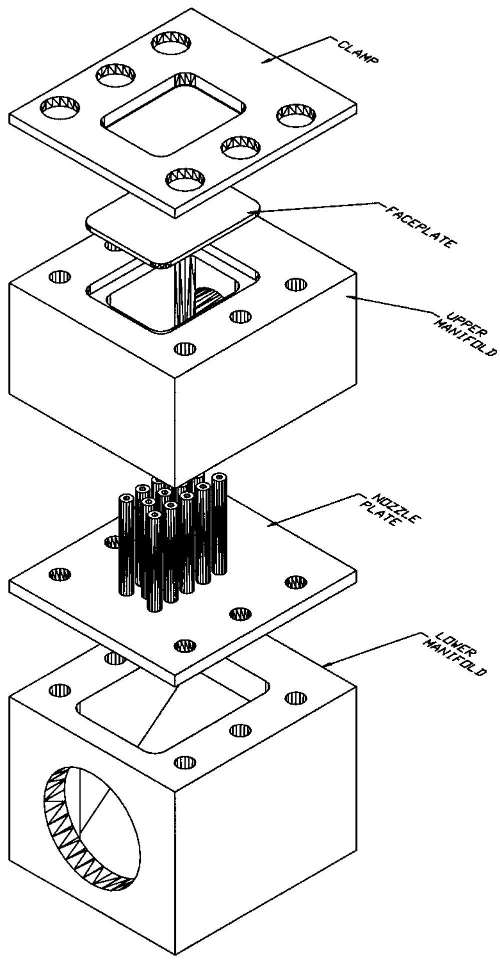

Figure 1. Exploded view of cooling module.

the low-pressure manifold, are located roughly two inner diameters (5.9 mm) behind the faceplate; the impinging jets so produced are fully submerged. The nozzle inlets are contoured to reduce head loss. After impinging on the faceplate, liquid exits the upper manifold through a 2.5 cm ID pipe.

The faceplate is a separate piece attached to the upper manifold. Various faceplate materials can be used in this system; our tests have been done primarily with dispersion-strengthened copper plates (alloy C15715) ranging from 2.5 to 4.0 mm in thickness. This material has both a high thermal conductivity (365 W/m · K) and a high yield strength (about 400 MPa); in addition, this alloy exhibits little softening at high temperature, in contrast to precipitation-hardened copper alloys. The molybdenum alloy TZM has also been used.

The faceplate has a heated area of 2.8 cm by 5.1 cm (14 cm^2). The faceplate itself is somewhat larger (5.1 cm by 6.6 cm), to allow support at its outer edge by the upper manifold. The plate sits in a groove machined into the outer manifold, with a silicone O-ring beneath it and a clamping frame pressed down on the upper edge; this configuration facilitates lateral expansion of the plate during heating and helps to limit thermal stresses.

The flow loop used to test these modules is driven by a 22.4 kW pump, rated at 2.24 MPa and 6.3 L/s. During operation, 98 % of the system pressure drop occurs through the module, mainly in the nozzles. This pressure drop is related to the jet speed as $\Delta p \approx 1.47\ (\rho u_j^2/2)$.The pressure drop is controlled by two flow-regulating valves in the loop, giving typical speeds of 40 to 55 m/s. Backpressure in the module can be adjusted to raise liquid saturation temperatures; for the present experiments, the water saturation temperature was 157 °C.

The bulk temperature of water entering the module averaged 35 °C. The volume flow rate through the module was approximately 4.3 L/s. With a design heat flux of 20 MW/m^2, the bulk temperature increase of the liquid passing through the module is less than 2 °C; however, much higher liquid temperatures obviously occur within the thermal boundary layer on the faceplate.

The objective in selecting the dimensions and liquid speed of the array was to divide the thermal resistance evenly between the faceplate and the liquid. Estimates of the mechanical and thermal stress limits of the faceplate showed that it must be 2 to 3 mm thick in order to avoid yielding during operation if it is made of dispersion-strengthened copper [2,3,10]. To balance the solid and liquid thermal resistances, $\overline{h}$ must be in the range of 200 kW/m^2K. At a flux of 20 MW/m^2, the upper surface of the plate will be about 250 °C hotter than the bulk liquid, while the rear surface of the plate will be roughly 100 °C above the bulk water temperature. An additional large temperature rise occurs through the heating films deposited atop the module faceplate.

The design value of $\overline{h}$ was used with existing correlations for jet impingement [11,12], extrapolated to the present high Reynolds numbers, to set the dimensions of the jet array. Smaller diameters raise $\overline{h}$, but jet diameters of less than 1 to 2 mm become impractical owing to manufacturing considerations and the potential to clog the nozzles. Increasing jet speed raises $\overline{h}$ as well. However, whereas $\overline{h}$ rises as $u_j^{0.6}$, liquid flow rate rises as u_j, the supply pressure rises as u_j^2, and the pumping power rises as u_j^3. The cost of pumps seems to be more nearly proportional to pumping power, so that a value of $u_j \approx 50$ m/s was the highest speed deemed cost-effective for this project. Having set the jet speed, nozzles of 2 to 3 mm diameter and spacings of about 10 mm provide the desired range of $\overline{h}$ while requiring no special fabrication.

3 Experiments on the Modules

As previously noted, the heating elements are directly deposited onto the module faceplate in a two-layer arrangement. In all cases, an insulating alumina layer 77 to 100 μm thick was deposited by high-velocity oxygen fuel spraying (HVOF). Heater films of Ni80-Cr20 were produced either by vacuum plasma spraying (VPS) or by HVOF and ranged from 75 to 280 μm in thickness [8]. Electrical power was provided by a 72 kW DC motor-generator set rated 24 V at 3000 A. Current was delivered to the heating element through a pair of copper electrodes that are attached to the edges of the heating element.

The heat flux generated by the film was calculated by dividing the measured electrical power by the measured area of the film between the electrodes, correcting for small heat losses to the surrounds. The measurement uncertainty in the heat flux calculation (at a 95 % confidence level) was 10 %. Spatial variation in the flux and temperature, owing to film nonuniformities and heat conduction in the faceplate, are discussed in [8] and [13].

For all heaters tested, it was possible to increase the electrical current to such a high level that the heaters failed. In every case, the failed heaters had cracks running through the heater film across the entire width of the film. The insulating film was usually intact, and the faceplate was undamaged. Available evidence suggests that failure of the bond between the heater and insulator films was the root cause of these failures. Failure of the film-to-film bond would most likely result from thermal stresses that increase as the temperatures rise: the temperature changes are substantially different for each layer, and the thermal expansion coefficients of the materials also differ.

The temperature of the heater film was measured by a set of K-type thermocouples cemented to the heater film. The thermocouples were always located within the central region of the heating element, in which heat flow to the coolant is one-dimensional and which is free of edge effects. The thermocouples were in contact with the metal surface of the heater, to minimize fin conduction errors associated with the thermocouple leads. The thermocouples had an estimated uncertainty of ± 2 °C at a temperature of 240 °C.

4 Test Results

The heat transfer coefficient of the jet array was measured in several experiments by using a thermocouple implanted in the faceplate to determine the plate temperature at the liquid surface (Fig. 2). A jet array speed of 46.5 m/s produces a heat transfer coefficient measured to be 220,000 W/m^2 ± 20 % at the stagnation points and which varies with position by less than this uncertainty [13]; numerical studies of the conduction through the faceplate show that the effect of these variations on the spatial variation in temperature at the opposite surface of the faceplate amounts to a few degrees at most.

The readings from all thermocouples were averaged and were used to compute the difference between the bulk liquid temperature and the average surface temperature of the heater film, ΔT. This temperature difference is plotted against the heat flux, q, in Fig. 3. Heat fluxes removed range as high as 17 MW/m^2. The maximum flux shown in each case is just below the flux at which the heat films failed. In each case, the faceplate temperature on the liquid side is below the saturation temperature, so that heat removal is by single-phase convection.

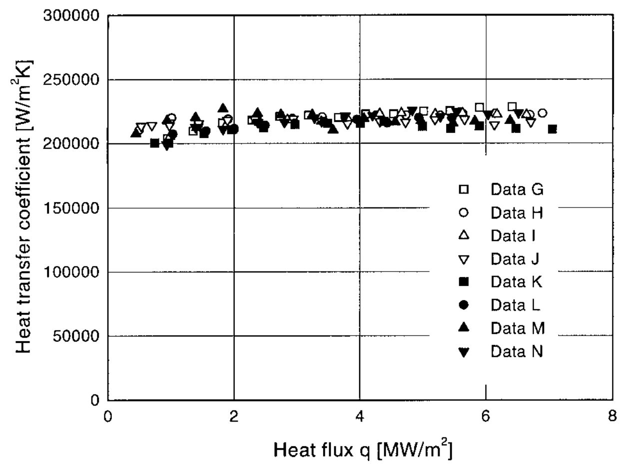

Figure 2. Heat transfer coefficient measured for the jet array as a function of heat flux in several experiments with u_j = 46.5 m/s. The uncertainty of individual data points is between ± 17 and ± 30 % for fluxes above 2 MW/m^2.

The slope of the curves in Fig. 3 equals the thermal resistance between the heater film surface and the coolant, and it can be used to estimate the thermal resistance of the pair of films. On a one-dimensional basis, deducting the thermal resistance of the faceplate, t_f/k_f, and the boundary layer, $1/\bar{h}$, from the total thermal resistance, $R_{t,total} = \Delta T/q$, the film resistance is:

$$R_{t,films} = R_{t,total} - \frac{t_f}{k_f} - \frac{1}{\bar{h}} \tag{1}$$

The films' combined resistance ranges from 10 to 27 m^{2}K/MW [8]; the 2σ uncertainty of the film resistance varies from case to case, but has a typical value of ± 25 %. These film resistances are less than half those of the best films produced by the air-plasma-spray process [9]. The temperature drop through the films amounts to about half of the temperature rise. For example, for the highest heat flux in data set B in Fig. 3, the copper plate's maximum temperature is only 265 °C.

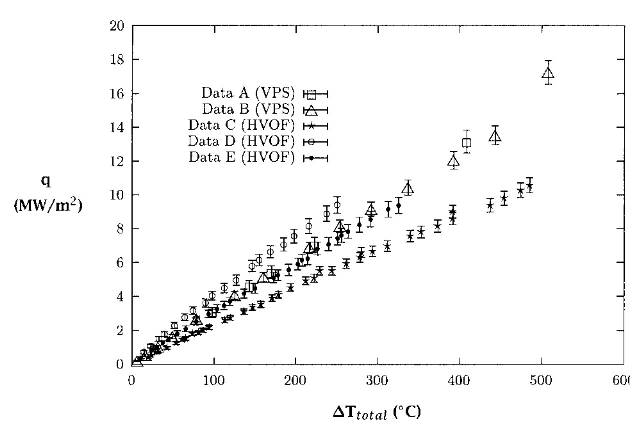

Figure 3. Difference between average surface temperature and coolant bulk temperature as a function of heat flux for several thicknesses of heating element on a 100 μm insulating film.

5 Summary

Jet-array cooling systems are capable of removing very high heat fluxes over significant surface areas. This note has described the performance of a jet-array module for this purpose. Heat transfer coefficients of 220,000 W/m^{2}K have been obtained and heat fluxes of up to 17 MW/m^2 have been removed by single-phase convection. Test heaters for this systems have been fabricated using thermal spraying processes. The design of any such system must take also account of the thermal stress present in the primary heat transfer surface.

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