POWER HANDLING CAPABILITY OF WATER-COOLED BEAM STOPS*

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Abstract

Doubling the beam power on the RFQ1-1250 linear accelerator at Chalk River and designing a 40 kW beam diagnostic system for Tokamak de Varennes required a detailed investigation into the power handling capabilities of beam stops. Different techniques for augmentation of the critical heat flux on the cooling channel surface of beam stops are reviewed. In the case of a beam stop with twisted tape inserts, the swirl flow condition yields a higher critical heat flux than that of a straight axial flow. Although a critical heat flux in the order of 10 kW/cm^2 could be obtained at high flow velocities such as 45 m/s [1], such flows are not always practical in the design of beam stop cooling systems. At a water velocity of 4 m/s, the highest beam power density is estimated to be 1.4 kW/cm^2 for a beam stop design that uses double rows of cooling tubes. A similar design, where cooling channels are machined on a common copper block, would handle a power density up to 2.6 kW/cm^2. Some preliminary hydraulic test results, related to a third design where high flow turbulence is created by two rows of intersected-channels, are also reported.

Introduction

The higher beam power of the Chalk River RFQ1-1250 accelerator and the requirements of a beam diagnostics system for Tokamak de Varennes have led to a detailed investigation into different design concepts for beam stops and their power handling limits. The study was based on the flow conditions recommended in normal design practice for cooling systems, which include feeder pipes, pumps and heat exchangers, where the highest water velocity is usually limited to 4.6 m/s [2].

Different concepts for the enhancement of the power handling capabilities of beam stops are reviewed, analytical methods are given for the case of straight and swirl flow, and hydraulic test data are presented for the intersected-channels design concept.

The Critical Heat Flux and Techniques for its Augmentation

In boiling heat transfer, the critical heat flux (CHF) on the wall of the cooling channel is reached when the formation of the vapour phase is so rapid that the liquid phase is kept away from the wall. For a constant heat flux system with no means of controlling the wall temperature, the heat transfer through the vapour phase is so poor that it will cause an instantaneous increase in the wall temperature. This can result in temperatures that exceed the melting point of many materials, including copper, and can create a local melt of the channel.

The concepts for augmenting the CHF include:

i) higher operating pressure,
ii) higher flow turbulence,
iii) flow directed onto the heat transfer surfaces,
iv) swirl or spiralling flow, and
v) built-in conduction heat path to a cooler area.

An increase in the system pressure increases the coolant saturation temperature, delays the formation of the vapour phase and increases the CHF value.

Higher flow turbulence is effective in breaking and "washing" the vapour phase from the surface of the cooling channel. This is normally achieved by increasing the flow velocity, by using a rough cooling channel surface or by intersecting two flow directions.

By directing the flow onto the heat transfer surface, the coolant kinetic energy is used to effectively break-up the vapour film and remove it from the cooling surface.

In the case of a swirl flow, the higher spiralling flow velocity combined with centrifugal force is quite effective in "washing away" the vapour blanket and wetting the wall of the cooling channel. This flow condition can be generated by internal grooves, by a twisted tape insert inside the cooling channel, or, for a short heated length, can be adequately produced by a tangential inlet feeder to a circular cross-section cooling channel.

A built-in conduction heat path to a cooler area provides some control of the wall temperature in the high heat flux area of the cooling channel which prevents local dry-out conditions and allows operation with wall temperatures spanning the CHF region. This is equivalent to an increase of the effective area of the heat transfer surface and is normally obtained by using an internal fin in each cooling channel or by locating multiple rows of cooling channels in a solid block of material.

Some of the above concepts are under investigation at Chalk River to improve the design of beam stops.

Beam stop for Tokamak de Varennes

A simple copper block with a double row of cooling channels was chosen for the design of a 40 kW graphite-faced beam stop for neutral beam diagnostics on the Tokamak de Varennes.

Figure 1 shows a typical cross-section of the proposed design. The back row is in a cooler region, located further away from the surface which receives the beam power. It

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controls the channel wall temperature in the front row and prevents thermal run-away conditions when boiling exceeds the critical condition.

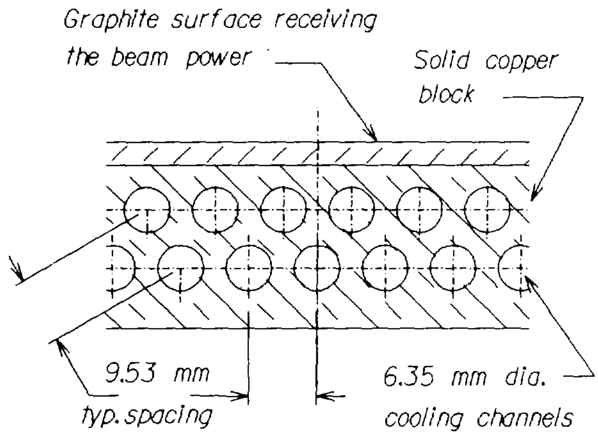

Fig. 1 Typical cross-section of the Tokamak de Varennes beam stop.

The thermodynamic quality of the coolant at the channel outlet end is $X_{outlet} = -0.18$ for a 4 m/s flow velocity, a 200 kPa system pressure and a 30°C water inlet temperature. This indicates that boiling is in the sub-cooled region (i.e., below bulk saturation of the coolant where X=0). The CHF in the cooling channel is estimated by:

$$CHF = CHF_{8mm}\ K_{dia}\ K_{hl}\ K_{fl} = 0.87\ \text{kW/cm}^2$$

where CHF_{8mm} is the standard compiled value for the case of a vertical upward flow in an 8 mm diameter tube, at the same pressure, flow velocity, and fluid quality condition [3]. K_{dia}, K_{hl} and K_{fl} correct for the actual cooling channel diameter, the heated length and the type of horizontal flow condition, respectively.

Since an azimuthal heat flux distribution will be present on the wall of the cooling channel, only half of the total cooling surface is considered effective in transferring the heat. The power handling capability of the present beam stop is estimated at 1.82 kW/cm^2 which fully satisfies the requirements for Tokamak de Varennes.

A similar design, where swirl flow is generated in the cooling channel, would be able to handle a heat flux in the order of 2.6 kW/cm^2 (see the CHF analysis for swirl flow in the next section).

RFQ1-1250 Beam Stop

Doubling the beam power on RFQ1-1250 to 90 kW with the new vanes required a closer look at the power handling capability of the existing beam stop. The original design used the swirl flow concept with cooling tubes arranged as shown in Fig. 2. Inside each tube, a full-length twisted tape insert was trapped by swaging down the tube diameter. A tape twist ratio, y, (inside diameters per 180° of twist) of 2.4 was used.

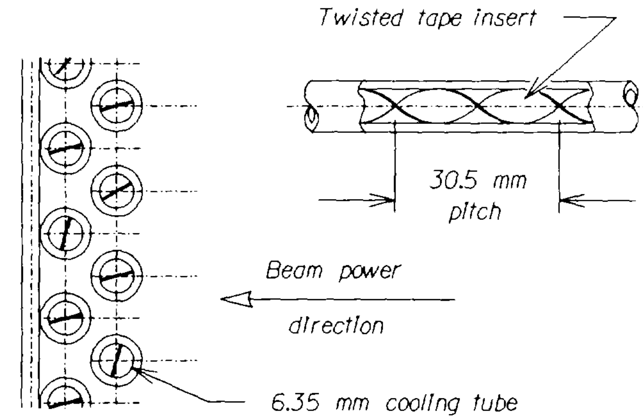

Fig. 2 Typical cross-section of RFQ1-1250's beam stop.

Although heat fluxes of the order of 10 kW/cm^2 have proven to be possible [1], these fluxes are usually attained at high flow velocity (45 m/s) on relatively shorter test sections (3.81 to 9.65 cm). Such conditions are not practical in the design of the beam stop and its cooling package. For an axial flow with a velocity of 4 m/s the estimated pressure drop, ΔP_a, is 11.7 kPa. The corresponding swirl flow frictional pressure drop, ΔP_s, is calculated by:

$$\Delta P_s = k_D\ k_{f,iso}\ k_{diab}\ \Delta P_a = 39.3\ \text{kPa}$$

where: $k_D = D/D_h$, the ratio between the tube inside diameter and the hydraulic diameter of the swirl flow configuration; $k_{f,iso} = 2.75\ y^{-0.406}$, the isothermal friction multiplier [4]; $k_{diab} = (\mu_b/\mu_w)^{-0.35(Dh/D)}$ corrects for the flow in the diabatic condition where a significant difference exists between the viscosity of the fluid near the wall, μ_w, and that of the bulk of the coolant, μ_b. The frictional pumping power per unit of the heated surface, A_h, is given by :

$$P_p = Q \times \Delta P_s / A_h$$

where Q is volumetric flow rate. Unless operated at high flow rate, the pumping power is normally low ($P_p \leq 10.7$ hp/m^2). In this case, the ratio between the CHF in swirl flow over the CHF in axial flow, based on the same pumping power, is equal to unity [5]. Although no net gain is shown, the CHF in the swirl flow at the same pumping power is equal to that of an axial flow with a higher flow rate. Using the same technique as discussed in the previous section to evaluate the CHF of the equivalent axial flow, the latter is estimated at 1.3 kW/cm^2. This limits the power handling capability of the design to about 1.4 kW/cm^2.

Since the present flow conditions fall within the range of the test data presented by Gambill [1] the direct correlation that he proposed can be used. The CHF estimated in this way is 1.2 kW/cm^2, which is in general agreement with the above analysis.

The Intersected-Channels Design Concept

Experiments are underway at Chalk River to investigate the intersected-channels cooling concept. In this scheme, the flow turbulence required in the main channel to produce a high critical heat flux, is generated by two intersecting flows. The test jig (Fig. 3) consists of 5 main channels, 305 mm long by 2.4 mm wide by 4.0 mm deep and spaced 2.4 mm apart, and 18 cross-channels, located at a 45 degree angle relative to the latter. The cross-sectional area of the cross channels was varied as a test parameter in this preliminary investigation where hydraulic data was collected in search of the optimum flow turbulence.

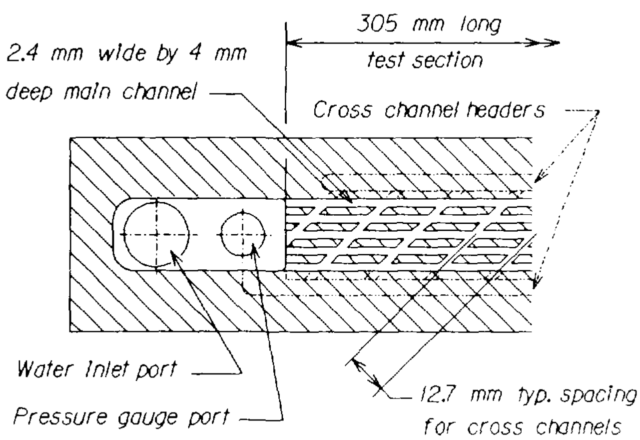

Fig. 3 Test jig for the intersected-channels concept.

Tests results shown in Fig. 4 indicate that increasing the flow area of the cross channel produces a higher cross flow and increases the flow turbulence in the main stream thereby resulting in a higher pressure drop.

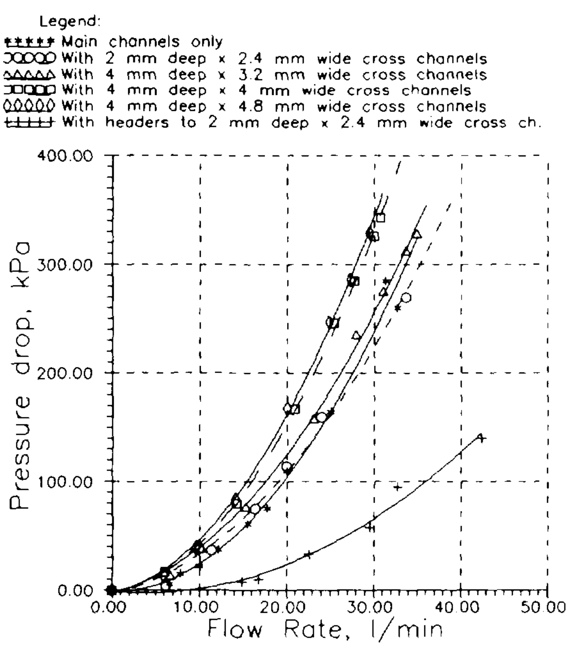

Fig. 4 Measured pressure drop as a function of the total flow rate for the intersected-channels design.

In the case of small cross channels (cross section ≤ 2.4 mm wide by 2.0 mm deep), the cross flow was insignificant in the test range. The recorded pressure drop for a given flow rate was almost identical to that of the case where only the main channels were machined on the test jig.

The optimum size of the cross channel for this particular test configuration is 4.8 mm wide by 4 mm deep. The corresponding pressure drop is about 47 percent higher than that of the undisturbed straight flow.

When headers are machined on the test jig to feed water into the cross channels, the arrangement introduces additional parallel flow circuits through the cross channel, which yield a higher total flow rate for a given pressure drop.

Tests are planned for a different flow configuration where a steeper angle will be introduced between the two flow directions. A prototype beam dump will be designed to evaluate the actual power handling capability of this design concept.

Conclusion

Several concepts for the design of beam stops have been analyzed using the flow conditions recommended in normal design practice (flow velocity = 4 m/s). The highest power handling capability is estimated at 2.6 kW/cm^2, obtained with a swirl flow inside a double row of cooling channels machined on a common copper block.

The concept of intersected-channels design is under investigation. Once the optimum flow configuration has been determined, a prototype beam stop will be built to evaluate its power handling limit.

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References

[1] W.R. Gambill, R.D. Bundy, and R.W. Wansbrough, "Heat Transfer, Burnout and Pressure Drop for Water in Swirl Flow Through Tubes With Internal Twisted Tapes", ORNL Report # 2911 (1960).

[2] ASHRAE Handbook, 1989 Fundamentals Volume, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (1989).

[3] D.C. Groeneveld, S.C. Cheng and T. Doan, "1986 AECL-UO Critical Heat Flux Lookup Table", Heat Transfer Engineering, 7, (1-2), 46-62 (1986).

[4] R.F. Lopina and A.E. Bergles, "Heat Transfer and Pressure Drop in Tape-Generated Swirl Flow of Single-Phase Water", J. Heat Transfer, **91**, 434 (1969).

[5] W.R. Gambill and R.D. Bundy, "An Evaluation of the Present Status of Swirl Flow Heat Transfer", ASME 62-ht-42 (1962).