THERMAL-HYDRAULIC DESIGN OF PWT ACCELERATING STRUCTURES*

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Abstract

Thermal-hydraulic design of the disks and cooling rods of a Plane Wave Transformer (PWT) accelerating structure is presented. Experiments to measure the fluid-dynamic and water-cooling characteristics of an S-band PWT disk were performed. Based on these results and conservative assumptions on heat transfer coefficients, calculations using Mathcad models and the COSMOS/M code were made for an L-band PWT. Specially designed water circuits provide effective cooling of the structure.

INTRODUCTION

High-energy rf accelerator components are subject to wall-current induced Ohmic heating which may detune the structure due to excessive thermal expansion. Alternating thermal stresses may also cause component fatigue failure thereby shortening the life of the accelerator. Design of an rf accelerator must therefore provide for adequate removal of the rf-induced heat. The total amount of heat depends on the average rf power, which is equal to the product of peak power, pulse duration and rep rate, less the average beam power, which is equal to the product of beam current and beam voltage. The heat imparted to each component is affected by its material, its location and surface area, and to its relative contribution to the Q-factor. In some accelerators such as the L-band International Linear Collider (ILC) the amount of heat can be quite large due to the long rf pulse [1].

In this paper we consider the thermal-hydraulic design of a plane-wave-transformer (PWT) photoinjector that has been proposed as a possible polarized electron source for future linear colliders [2,3]. The multi-cell PWT rf cavity design consists of a number of iris-loaded, copper disks suspended by coolant-carrying pipes, all of which are inside a large metal cylinder. Removal of heat deposited to the disks and the rods is effected primarily by convection of coolant flowing through them.

PWT DISK COOLING CIRCUITS

For purposes of rf frequency stability and disk-to-disk field uniformity, it is necessary to maintain the disks at a fixed, constant temperature. If the coolant flow rate through each disk is constant and equal, the temperature *rise* of the coolant therein would be equal, provided all disks receive identical heat input. Fluid temperature and pressure at each disk entrance in the circuit network, however, differ due to heat transfer and pressure loss in pipes. The flow rate in each disk may be regulated by selection of a proper diameter for an orifice that connects the coolant supply pipe to disk internal channels (Fig. 1).

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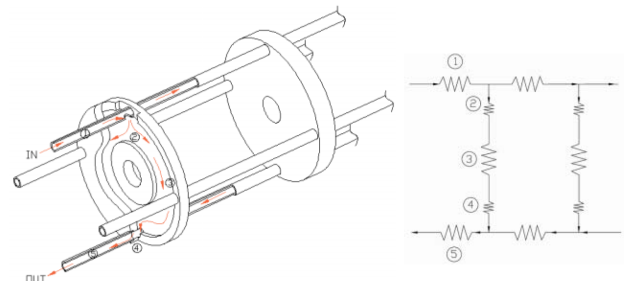

a) b)

Fig. 1 (a) Cutaway view of disk cooling channel, and (b) elements of a circuit model of flow through two disks.

The flow rates, pressure drops and temperatures at all nodes in a cooling circuit network are related by the continuity equation and energy conservation, analogous to Kirchoff's rules for an equivalent electric circuit.

SINGLE DISK EXPERIMENT

In order to reduce uncertainty in circuit modeling assumptions, we performed an experiment to measure the pressure drop and heat transfer rate in a single S-band PWT disk. Fig. 2 shows the hardware. Several disk inlet pipes were constructed, each with a different entry orifice size. The orifice was precisely located in the pipe transverse to the direction of coolant flow.

Fig. 2 PWT disk experimental setup.

Pressure gauges were placed at the inlet and outlet pipes near the disk. A micron filter was placed in the inlet pipe, and a flow meter at the outlet pipe. The measured disk flow rate *vs* pressure drop for various orifice sizes is shown in Fig. 3. The total pressure drop is:

$$H_{total} = H_1 + H_2 + H_3 + H_4 + H_5 , \qquad (1)$$

where H_1 and H_5 (subscripts denote numbers in Fig. 1) are pipe friction losses, H_2 is the pressure loss through the orifice at disk entrance, H_3 is the friction loss through the disk resulting from change of flow direction and wall friction, and H_4 is the expansion/contraction loss through the exit gap. For a small orifice, the dominant term in Eq. (1) is H_2. The measured pressure drop (psi) compares very well with a simple formula [4]:

$$H_2 = \frac{625 \cdot (2\omega/\rho)^2}{4 \cdot a^2} \quad . \tag{2}$$

In Eq. (2), a is the radius (inch) of the orifice, $\omega_p \equiv 2\,\omega$ is the flow rate (ft^3/s) through the orifice (into 2 channels inside the disk), and ρ is the coolant density (lb/ft^3).

The disk was immersed in a constant temperature (40°C) bath. Inlet and outlet water temperatures were measured with in-channel thermocouples, while other parameters were held constant. The PWT-disk temperature was held constant to ±1°C with a feedback control system. The feedback loop regulated a 1500-watt heater for the bath using the disk temperature, measured with another thermocouple, as control. A variable speed motor was used for stirring the bath in order to maintain a uniform temperature.

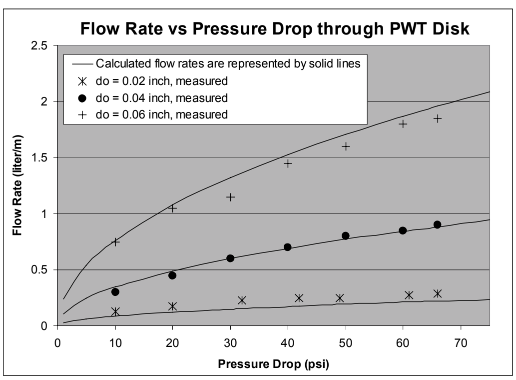

Fig. 3 Disk hydraulic measurements vs calculations.

Data were recorded in real time, and later analyzed with Microsoft Excel. The control system used a microcomputer to interface with an Analog Devices 4-channel, isolated thermocouple/conditioner, and a solid-state relay system that controlled the bath temperature.

The heat removed from the disk was measured by $\dot{Q} = C_p\,\omega_p\,\Delta T_p$, where $\dot{Q}$ is the heat rate, C_p is the specific heat, ΔT_p is the temperature difference between the inlet and outlet water, and ω_p is the measured flow rate in the pipe.

From the heat rate thus measured, the convection film coefficient can be calculated by $\dot{Q} = hA\,\Delta T$, where h is the film coefficient, A is the coolant/disk contact area and ΔT is the average temperature difference between the metal surface and the coolant. The integrated average temperature difference (as the coolant is heated in the disk) is calculated by applying a logarithmic-rise function to the coolant temperature, fitted to the inlet and outlet temperatures. A plot of the "measured" film coefficient is shown in Fig. 4 as a function of flow rate in the disk channel. The flow rate in each disk channel is $\omega = \omega_p/2$ since the inlet pipe flow is split into two identical channels inside each disk (Fig. 1a).

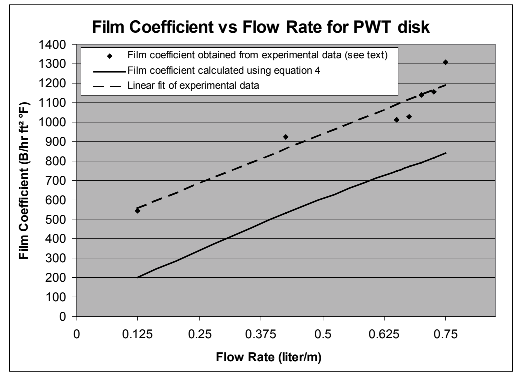

Fig. 4 Disk channel film coefficients, measured vs calculated.

Also shown in Fig. 4 are calculated film coefficients using a "canonical" formula [5] for flow inside a channel of hydraulic diameter D:

$$h = 0.023(k/D)(Re)^{.8}\,(Pr)^{.4} \tag{3}$$

where Re is Reynold's number, Pr is Prantl's number and k is the conductivity of the coolant.

The following conclusions can be drawn from this experiment: 1) The pressure drop through an orifice in the PWT disk is adequately represented by Eq. (2), whereas 2) the actual amount of heat removed by water in a PWT disk channel is more than that calculated using a film coefficient given by Eq. (3). In the next section we will evaluate the thermal-hydraulic properties of a cooling system for an L-band PWT based on these findings.

L-BAND PWT THERMAL-HYDRAULIC ANALYSIS

A feasibility study has been performed for an L-band PWT photoinjector as a possible polarized electron source for the ILC [3]. A preliminary rf design, with 8 copper disks suspended by 6 copper-plated cooling pipes inside a large stainless steel cylinder, requires two phase-synchronized, L-band klystrons to accelerate a low-emittance electron beam to 10 MeV. The average output power of a single klystron (10 MW peak power, 1.4 millisecond-long pulse and a rep-rate of 5 Hz) is 68.5 kW, while the expected average beam power is 0.53 kW. The 6 coolant-carrying pipes are configured into 3 separate circuits, supplying coolant to the internal channels of 3, 3 and 2 disks, in parallel (Fig. 5). The total heat load is 136 kW, of which 35 kW is deposited to 8 disks, 29 kW to 6 pipes, 8 kW to two copper endplates and 64 kW to the SS tank, based on their relative contributions to the loaded Q-factor of the PWT.

In order to assess the adequacy of the cooling circuits for the given heat load, a 3D finite-element disk model, including connecting pipes, was used to calculate the temperature profile of the disks and the pipes. Fluid temperatures and flow rates that were used as input to COSMOS/M models were obtained with Mathcad models. Film coefficients for these temperatures and flow rates were calculated for flows in pipes and disk channels, conservatively using Eq. (3).

MathCad simulations include a heat transfer analysis and a hydraulic analysis for each piping network circuit to obtain the temperature, flow rate and pressure drop at all nodes and at the initial inlet pipe. In the heat transfer analysis, all disk inlet flow rates are assumed to be ω_p (=2ω) ≡ ω_o = 20 lpm. Disks and pipe segments are represented by thermal lumped masses. Appropriate heat rates are applied to all nodes. As a first approximation, only convective heat transfer is included in the MathCad models. Each of the 3 circuits (Fig. 5) cools 2 or 3 disks and connecting pipes. The endplate is partially cooled by all 3 circuits, and additionally by a separate circuit that runs through the tank wall. The flow rate in each inlet pipe to the endplate is assumed to be the same as the disk.

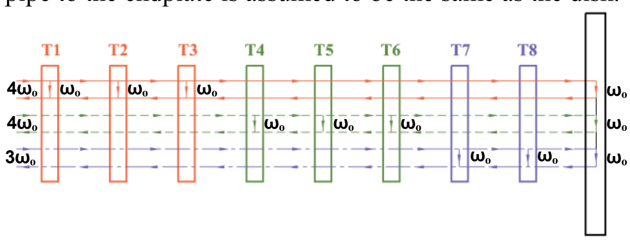

Fig. 5 Schematics of 3 parallel, disk/pipe cooling circuits. The first circuit, represented by solid line (red), cools the first 3 disks and the endplate. The second circuit, represented by dash line (green), cools the next 3 disks and the endplate. The third circuit, represented by dot-dash line (blue), cools the last 2 disks and the endplate.

The hydraulic analysis includes all pressure drops in each closed loop. Friction losses for pipe segments are calculated with Darcy's formula [6]. Pressure drops for orifices are calculated with Eq. (2). In addition, disk channel friction losses and expansion/contraction losses are included. All pressure drops, as well as disk orifice sizes, are solved by MathCad coupled equations once the ID of the pipes connected to the endplate is fixed (0.4").

Table 1: MathCad thermal-hydraulic analysis results for average disk temperature T (˚C), total flow rate Ω (lpm) and total pressure head H (psi) for each cooling circuit. (ω_o = 20 lpm , ambient temperature = 20˚C)

Cir-cuit	T1	T2	T3	T4	T5	T6	T7	T8	Ω	H
1	29.7	29.8	30.0	—	—	—	—	—	80	42
2	—	—	—	29.9	30.1	30.3	—	—	80	76
3	—	—	—	—	—	—	30.5	30.7	60	59

As shown in Table 1, a maximum disk temperature variation of only 1˚C resulted from the lumped-mass thermal models. The maximum temperature rise of any disk is about 10˚C. The maximum pressure head is 76 psi for a total flow rate of 80 lpm at the initial inlet pipe (Circuit #2).

The finite-element thermal model (Fig. 6) consists of a single half-disk, the internal cooling channel of which is connected to an inlet pipe via an orifice, and to an outlet pipe via an open gap; four other pipes that are used to cool other disks pass through it. Heat loads are applied to the outside surfaces of the disk and the pipes. Convection boundary conditions are imposed on all inside metal surfaces that are in contact with the coolant, using film coefficients calculated for conditions based on the MathCad results. A worst-case COSMOS/M 3D thermal analysis shows that the disk temperature is higher by as much as 7.5˚C than the MathCad results shown in Table 1. The difference is attributable to the finite copper conductivity of the disk, and effects of the connecting pipes. The effect on PWT frequency detuning due to a disk-to-disk temperature variation up to 10˚C (resulting in ≈.001" variation in disk diameter) is tolerable. The maximum average temperature of any pipe segment is 53.4˚C. Pipes with a small OD (≈0.5") can tolerate higher temperature without detuning the PWT structure.

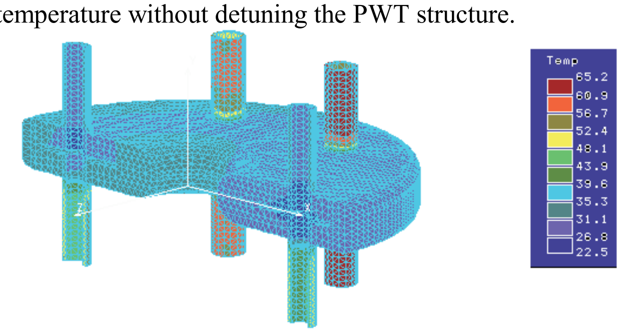

Fig. 6 COSMOS/M 3D finite-element thermal model of a single disk with connecting pipes.

A similar analysis of heat transfer from the PWT tank by 4 water circuits imbedded inside the tank shows that at a flow rate of 10 lpm and a pressure head of about 60 psi for each circuit, the temperature rise of the coolant between the inlet and outlet is less than 20˚C.

CONCLUSIONS

A thermal-hydraulic analysis of an L-band PWT photoinjector design under projected ILC operating conditions indicates that adequate cooling can be achieved. Future experimental tests should be performed with an L-band PWT disk to verify these results.

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