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THERMAL ANALYSIS AND LOSS IN SHUNT IMPEDANCE
OF THE LEP ACCELERATING CELL

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Summary

Basic formulae governing thermal flow in water-cooled solids are given. By calculating the heat dissipation at room temperature of the LEP accelerating cell with the program SUPERFISH and then using this as input to the thermal analysis program DOT, the surface temperatures and losses at full power have been estimated. Good agreement is obtained when these LEP results are scaled and compared with measurements made on similar accelerating structures in PEP and PETRA. The calculated decrease in shunt impedance of 4.2% with respect to the room temperature value has been substantiated by experimental measurements on a prototype LEP cavity.

Introduction

The required circumferential RF voltage of 400 MV for Phase 1 of LEP operation will be provided by 128 coupled cavity units consisting of accelerating cavities (with Q values of about 41000) and low-loss storage cavities (with Q values around 164000). The use of spherical storage cavities excited in the low-loss H_{011} mode enables a reduction of about 40% to be obtained in dissipated power. The effective shunt impedance of the coupled system is expected to be 40 $M\Omega/m$. The five-cell slot-coupled accelerating cavities operate in the π -mode at 352 MHz and are similar in their basic conception to those used in PETRA and PEP. The major part of the 16 MW of RF power will be dissipated in the walls of the cavities in the form of heat by electrical losses; only a small part of the RF power actually goes into the beams. This heat may prove to be detrimental to the performance of the RF system if it is not handled correctly. In designing the cavities, the cooling requirement should be determined not only by the total amount of heat which has to be carried away, but also by the increase of surface temperature that can be tolerated without creating an unacceptable loss in shunt impedance resulting from the temperature-dependent variation in electrical conductivity. This is especially important in big machines such as LEP Phase 1 where a 4% loss in shunt impedance has to be counterbalanced by about 1.0 MW more mains power. In a recent article¹ it was reported that in PETRA the shunt impedance is almost 19% lower than the assumed room temperature value and that more than 12% of this is attributed to a surface temperature which is considerably higher than the mean water temperature. In this paper the physical reasons behind such an overheating are discussed and a description of a thermal analysis of the LEP accelerating cell is given to illustrate one possible design approach to this sort of problem. The resulting change in shunt impedance, although less drastic than in the case of PETRA does however reduce the room temperature value by 4.2%.

Basic Heat Transfer Formulae

The heat dissipated within a few microns of the inside surface of the cavities (the skin depth for the LEP cavities is 4 microns) is transported by conduction throughout the cavity, by forced convection to the bulk of the cooling water, and by natural convection and radiation to the atmosphere unless the outer surface of the cavity is insulated. Steady state conduction and radiation present no difficulties since they can be expressed exactly by the mathematical laws of Fourier (1822) and Boltzmann (1884) respectively. Forced and

natural convection on the other hand involve the exchange of heat due to the complicated mixing motion of fluids for which there is no complete mathematical description. A differential equation describing the law of convective heat transfer therefore does not exist and this has necessarily led to the accumulation of vast amounts of empirical data covering an equally vast range of variables. This data is in general rationalized by expressing the empirical relationships in terms of dimensionless groups which establish a certain logical grouping of the variables. Most of the given dimensionless groups encountered in convective heat transfer analysis are given in reference 2, together with their physical significances. In the special case of heat transfer between a moving fluid and a solid surface the following simple relationship proposed by Newton in 1701 is used:

$$q = h(T_1 - T_2) \quad (1)$$

where q heat flow per unit area
 h surface heat transfer coefficient
 T_1 temperature of the wall
 T_2 usually the bulk fluid temperature.

The surface heat transfer coefficient h is usually found in the literature hidden in the dimensionless Nusselt number which for flow of water of thermal conductivity k in round tubes of diameter D is $N_{Nu} = hD/k$.

The value of h is influenced considerably by the velocity profile of the fluid close to the solid boundary; this in turn depends on whether the flow is laminar or turbulent. The flow regime is determined by the Reynolds number N_{Re} . For forced flow of water in round tubes and $N_{Re} < 2300$ laminar flow exists over the entire cross-section, whereas for $N_{Re} > 10000$ a well-established turbulent flow can be assumed. In order to obtain good cooling one normally chooses a turbulent flow for which the following empirical relationship² has proved successful:

$$N_{Nu} = 0.023 N_{Pr}^{0.4} N_{Re}^{0.8} \quad (2)$$

where N_{Nu} Nusselt number
 N_{Pr} Prandtl number
 N_{Re} Reynolds number

In this particular formulation all physical properties are evaluated at the bulk temperature.

During turbulent flow, despite its name, a very thin surface layer is created at the wall of the pipe in which the flow is laminar and in which the velocity increases rapidly. This layer has a thickness

$$\delta = 25.3 N_{Re}^{-0.75} d \quad (3)$$

where d pipe diameter.

Outside this layer the flow is turbulent with only small velocity gradients. Since high-velocity gradients correspond to high-temperature gradients, the surface temperature is considerably higher than the bulk temperature. (This must of course be the case for any efficient transfer of heat.)

For natural convection from horizontal cylinders (accelerating cavities) in air, the empirical relationship which best fits the available data is

$$N_{Nu} = 0.53 (N_{Gr} N_{Pr})^{0.25} \quad (4)$$

where N_{Gr} Grashof number
 $N_{Gr} N_{Pr}$ sometimes called the Rayleigh number

This relationship is valid for Rayleigh numbers between 10^3 and 10^9 .

Thermal Analysis of the LEP Accelerating Cell

The LEP cavity consists of five cylinders and six discs. The cylinders are cooled by water flowing in copper tubes brazed to their outer surfaces whereas the discs are cooled via a series of interconnecting borings which are drilled into the copper (see Fig. 1). The complete five-cell structure has 11 parallel cooling circuits, one for each disc and cylinder.

In order to obtain an upper limit on the loss in shunt impedance of the LEP accelerating cavity it was assumed in the thermal analysis that 25 kW of heat were dissipated in each of the five cells and that no losses to the atmosphere occurred by radiation or natural convection. (This corresponds to the condition where the thermal insulation used in the bake-out procedure is left permanently in place.)

The values of surface heat transfer coefficients used in the analysis were calculated from the above relationships and are given in the following table together with other relevant parameters.

	Units	Disc	Cylinder
Mass flow rate	kg s ⁻¹	0.230	0.105
Pipe diameter	cm	1.2	0.9
Velocity	ms ⁻¹	2.03	1.65
N_{Re}		27 682	16 875
Bulk water temperature	°C	28	28
N_{Pr}		6.0	6.0
h	w/cm ² °C	0.83	0.74

The thermal analysis was carried out using the axisymmetric finite-element program DOT⁵. This computer program allows point heat inputs to be applied at the mesh nodes and heat transfer coefficients to be associated with the faces of mesh elements. Using these boundary conditions the program calculates the steady state temperature distribution by solving the heat conduction equation

$$\nabla^2 T + \frac{Q}{k} = 0 \quad (5)$$

where Q heat generation rate
 k thermal conductivity
 T temperature

For symmetry reasons it is sufficient to consider a half-cell. This is shown in Fig. 1.

Fig 1 THE LEP ACCELERATING CELL

In order to use this axisymmetric program to model a truly three-dimensional problem one has to convert the non-axisymmetric arrangement of the cooling holes in the real disc into an axisymmetric one having equivalent surface heat transfer conditions for the computer model. This was done in the following way.

Using equation (1) the heat flow into the water from an annular area $2\pi\rho dp$ of the surface (see Fig. 1) for the model is

$$Q_m = 2\pi\rho dp q_m = h(\rho) 2\pi\rho dp (T_1 - T_2)_m \quad (6)$$

whereas for the real disc it is

$$Q_D = 2\pi\rho dp q_D = h_D 24\pi \frac{d}{2} dp (T_1 - T_2)_D \quad (7)$$

Assuming the heat flows and temperature differences to be equal for the model and the real disc, one obtains from (6), (7)

$$h(\rho) = \frac{7.2 \text{ cm}}{\rho} h_D \quad (8)$$

FIG. 2 DISSIPATION IN W/cm² AT ROOM TEMPERATURE

The dissipated power P_d was calculated with the program SUPERFISH⁶ from the value of the tangential magnetic field H at the surface S and a constant room-temperature value k of electrical conductivity as follows:

$$P_d = \frac{1}{k\delta} \int H^2 dS \quad (9)$$

where δ skin depth

The 25 kW is distributed as shown in Fig. 2.

The temperature distribution resulting from this analysis is given in Fig. 3.

The profile is rather uniform, the differences being not more than 8°C. There are no real 'hot points'. It is interesting to note that due to the good cooling in the discs the nose cones have moderate temperatures with a peak value of 40.6°C and that the hottest region at 44.4°C is the region between the cooling pipes on the cylinder.

A limited number of experimental measurements of temperature made on a LEP prototype cavity working under the same conditions are also shown in Fig. 3. The agreement between theory and practice is reasonably good.

FIG. 3. SURFACE TEMPERATURE DISTRIBUTION IN LEP ACCELERATING CELL

From the temperature distribution one can recalculate the wall losses to a good approximation from the tangential magnetic fields calculated with SUPERFISH using the following temperature-dependent relationship for the electrical conductivity.

$$\kappa = \kappa_0 [1 - 0.00391(T - T_0)], \quad (10)$$

$$\kappa_0 = 58 \times 10^6 \Omega^{-1}m^{-1} \quad \text{OFHC copper at } T_0 = 20^\circ\text{C}$$

The resulting losses are 4.2% higher than those at 20°C and therefore the shunt impedance of the LEP cavity will be 4.2% lower than the theoretical value at room temperature. These higher losses correspond to a mean surface temperature increase of 21°C of which 8.9°C are due to the mean water temperature increase; typically 8°C come from the surface layer of the cooling water and about 4°C occur in the copper.

In order to compare this result with measurements done at PETRA¹ one can scale the mean surface temperature

$$T_s = T_{in} + \frac{P^P}{P^L} \left[\frac{g^L}{g^P} 8.9^\circ\text{C} + \frac{h^L A^L}{h^P A^P} 8^\circ\text{C} + \frac{A^L}{A^P} 4.1^\circ\text{C} \right] = 45^\circ\text{C} + 9.7^\circ\text{C} + 11.6^\circ\text{C} + 6.6^\circ\text{C} = 72.9^\circ\text{C} \quad (11)$$

Here the superscripts P, L stand for PETRA and LEP and the different values are

		Units	LEP	PETRA
P	Dissipation per cell	kW	25	24
g	Mass flow	kg s ⁻¹	1.90	1.67
h	Mean heat transfer coefficient	W/cm ² °C	0.83	0.92
A	Mean cooling area	dm ²	2.05	1.22

This scaling, considered to be right within a few degrees, gives a mean surface temperature increase of 52.9°C compared to room temperature and accounts for only an 11% decrease in shunt impedance. The measured value of 18.9%, which would correspond to a surface temperature increase of 87°C, seems too high to be explained by the inaccuracy of the scaling and/or the simplification of an axisymmetrical calculation. It is possible that other effects, such as for instance losses on the tuners, cause this additional deterioration. When however the LEP cavity temperatures are scaled to PEP cavity conditions an excellent agreement is obtained with measured values of nose cone temperatures^{7,8}. In the case of LEP there is a 5.4°C temperature gradient in the copper between the nose cone and the cooling surface, and a further 6.5°C between this same cooling surface and the bulk water.

Scaling these values and taking a 45°C mean water temperature for PEP the nose cone temperature becomes

$$T_{nc} = 45^\circ\text{C} + \frac{P^P}{P^L} \left[\frac{k^L}{k^P} \cdot 5.4^\circ\text{C} + \frac{h^L A^L}{h^P A^P} \cdot 6.5^\circ\text{C} \right] = 45^\circ\text{C} + 65.3^\circ\text{C} + 26^\circ\text{C} = 136.3^\circ\text{C} \quad (12)$$

Now the superscript P stands for PEP. The values are

		Units	LEP	PEP
$\hat{P}$	Dissipation in nose cones	kW	1.35	9.00
k	Thermal conductivity	W/cm ² °C/cm	3.94	2.18
h	Heat transfer coefficient	W/cm ² °C	0.87	1.01
A	Cooling surface in nose cone region	cm ²	199	284

The values of dissipation in the nose cones were calculated from SUPERFISH. The measured PEP nose cone temperature was 139°C.

An experimental estimation of loss in shunt impedance on the LEP prototype accelerating cavity was made by G. Geschonke from loaded-Q measurements at 20°C and at the operating temperature of the cavity. Although the accuracy of the measurement does not enable a precise value to be quoted, the loss was found to be in the range 2-8%.

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