The Design of Photon Absorbers for the ESRF Vacuum Chamber

G. Marot , R. Bennett
EUROPEAN SYNCHROTRON RADIATION FACILITY
BP 220 F-38043 GRENOBLE Cedex

1.INTRODUCTION

The total synchroton radiation power emitted by the E.S.R.F. Storage Ring dipoles reaches 1 MW under extreme conditions .

This power will be dissipated in two types of absorbers (ref 1) :

- low power absorbers , subjected to power densities less than 40 W/mm^2 ,
- high power absorbers (crotches) , with densities reaching 400 W/mm^2 .

This paper presents the methodology used to design the low power absorbers : in section 2 , design principles and rules are described . An analytical model of the absorber thermal behavior is developed in section 3 :This model is necessary to study the influence of main parameters , in order to optimize the design . A comparaison with a numerical model for validation is given in section 4 .

2.DESIGN PRINCIPLES.

2.1.GEOMETRY

The shape of the low power absorbers is illustrated fig 1 : an inner OHFC-copper slab , absorbing the power , is brazed on the vacuum chamber wall (316LN Stainless Steel) . In order to evacuate the heat , the outer face of the chamber wall is cooled by a water flow . Three water cooling devices are studied in this paper :

- tube cooling (fig 1.a) : a copper slab, with circular ducts for water flow , is brazed on the outer face of the wall .
- film cooling (fig 1.b) : water flows in a flat channel , without any additional copper .
- fins cooling (fig 1.c) , expected to improve the efficiency of the configuration 1.b .

These types of absorbers have two main advantages :

- water is outside the vacuum chamber ,
- no vacuum joint is needed .

Nevertheless , heat is tranfered from inner copper to water flow through a Stainless Steel strip:the low thermal conductivity of such a material (17 W/m.C) limits the design to low power densities .

2.2.DESIGN RULES.

Several physical phenomenas limit the performance of the absorbers :

- Degassing limits the maximum temperature to 350 °C .
- boiling of the cooling water occurs for water/metal interface temperature higher than the boiling limit T_b.Vapour can then block the channels and stop the water flow .T_b depends on pressure : 100 °C at 1 bar , 150 °C at 5 bar .
- restrained thermal expansion leads to thermal stresses; care must be taken to avoid : plastic deformation , brazing failure and fatigue .
- a maximum thickness of 30 mm in normal incidence is required for a few absorbers , for photon beam position monitoring .

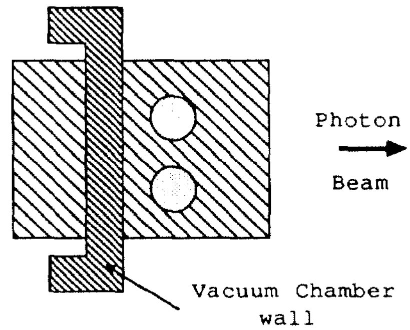

Fig 1a : Tube Cooling

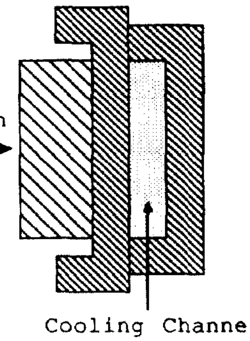

Fig 1b : Film Cooling

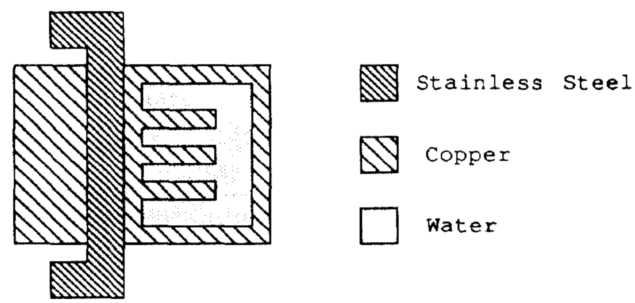

Fig 1c : Fins Cooling

3.ANALYTICAL MODEL OF ABSORBERS THERMAL BEHAVIOR

An analytical model has been prefered to numerical models for 3 reasons :

- the geometry is simple enough ;
- calculations are very fast : it is easy to find the optimum value of a parameter ;
- the total water flow rate for the storage ring is limited : one absorber cannot be dimensioned independantly from the others . Analytical models are well adapted to a global analysis of all the absorbers , with imposed local (for each absorber) or overall (such as total flow rate) conditions .

3.1.ANALYTICAL MODEL.

3.1.1.ASSUMPTIONS.

- Steady state ;

- Uniform beam power over a rectangular impact $l_x . l_z$ (l_z in the vertical direction , l_x in the horizontal direction) ;
- because to $l_x >> l_z$, conduction in the x-direction is ignored . The temperature field is then purely bidimensionnal : T(y,z) ;
- natural cooling , at least 100 time smaller than water cooling , is neglected ;
- the brazing joints are perfect , and thin enough to avoid temperature drop .
- constant thermal properties (thermal conductivity(W/m.C) : copper : 386 ; Stainless Steel : 17).

Calculations are divided into 6 elementary parts : inner copper , stainless steel wall , tube cooling , film cooling and fins cooling. The global model is then evaluated .

3.1.2.INNER COPPER.

Geometry and notations are given in fig 2 .

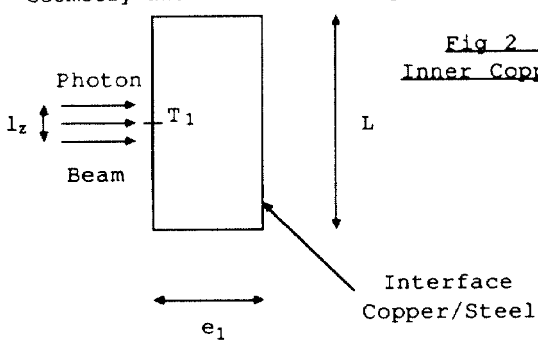

Fig 2 : Inner Copper

The action of the other elements of the absorber is condensed in a transfer coefficient h_c on the Copper/Steel interface .

The 2-D analytical solution is obtained by a separated variables method (ref 2). The maximum temperature T_1(at centre of beam impact) is expressed by :

$$T_1^* = f \left(Bi , \frac{L}{e_1}, \frac{l_z}{e_1} \right)$$

with :

$$T_1^* = \frac{k_c (T_1 - T_a)}{P_1} \qquad Bi = \frac{h_c e_1}{k_c} \text{, Biot number}$$

(k_c is the copper thermal conductivity,T_a is the water temperature,P_1 is the beam power per unit l_x (W/m)).

*influence of absorber height L : due to lateral (in z-direction) conduction in the copper , T_1 decreases as L increases .For L higher than L_{max} , this effect becomes negligible , and T_1 remains at its minimum value T_{1min}.It is useful to know L_{max} : it gives the lowest temperatures for the minimum absorber height . With practical values of Bi , L_{max} is 10 to 20 times higher than the copper thickness e_1 .

*influence of thickness e_1 : for small height L,lateral conduction is small , and T_1 increases with e (this is a classical result in 1-D conduction , corresponding to $L=l_z$).At the other extreme , for $L > L_{max}$, lateral conduction is important , and increases with e: therefore , T_1 decreases with a thickness increase .

3.1.3.VACUUM CHAMBER WALL .

Heat conduction in the chamber wall is almost purely monodimensional . Temperature gradient from the inner face (mean temperature:T_2),to the outer face (mean temperature:T_3) is simply expressed by :

$$\frac{(T_2 - T_3)}{P_1} = \frac{e_2}{k_{ss}\, l_z}$$

(e_2:vacuum chamber wall thickness;k_{ss}:Stainless Steel thermal conductivity).

The wall should be , of course , as thin as possible : nevertheless , it seems to be difficult to decrease its thickness , even locally , under 1 mm .

3.1.4. TUBE COOLING.

Conduction in the outer copper , and convection between copper and water are successively calculated .

-conduction :

The solution is extrapolated from an infinite number of tubes (ref 3), to a finite number of tubes : N (fig 1a).Calling T_3 the mean temperature of the interface steel/copper , and T_w the mean temperature of tube walls :

$$\frac{(T_3 - T_w)}{P_1} = k_c \frac{\mathrm{Log} \left(\frac{2l}{\pi \varnothing} \sinh \left(\frac{2\pi x}{l} \right) \right)}{2\pi N}$$

($\varnothing$: tube diameter ; l : distance between tubes axis ; x : distance between tubes axis and copper/steel interface).

It can be then deduced that :

*the tubes must be as closed as possible to the steel;

*the efficiency increases with the number of tubes . Nevertheless , this influence is not linear : for typical geometries , using 3 tubes rather than 2 tubes increases the efficiency of 30 to 40 % ; the efficiency is again 10 to 20 % higher by using a fourth tube , and the influence of additional tubes quickly becomes negligible .

-convection : The exchange coefficient between copper and water , h_a , is calculated from Dittus-Boelter empirical correlation , valid for turbulent flow (ref 3):

$$Nu = .02\, Re^{.8}\, Pr^{.4}$$

Nu , Re and Pr are the classical Nusselt , Reynolds , and Prandtl numbers , calculated with the hydraulic diameter $\varnothing$.

Therefore :

$$\frac{(T_w - T_a)}{P_1} = \frac{\varnothing^{.8}}{\pi N^{.2} h_0(T_f)\, Q^{.8}} \qquad T_f = \frac{T_w + T_a}{2}$$

The influence of the number of tubes on convection efficiency is almost negligible : it is the result of a competition between the increase of cooling area with N , and the decrease of transfer coefficient (for a given flow rate) .

3.1.5.FILM COOLING .

In this case , $T_3 = T_w$ .Exchange coefficient h_a is estimated from tables given in ref 3 , for flow between parallel plates . Energy conservation gives here :

$$\frac{T_w - T_a}{P_1} = \frac{L^{.2}\, e_3}{h_1(T_f)\, Q^{.8}}$$

The cooling efficiency decreases with a film thickness increase , due to water speed decrease for a given flow rate.

3.1.6.FINS COOLING .

A typical design is given fig 3 .

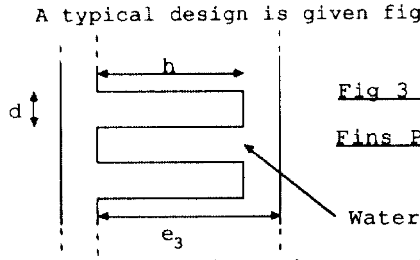

Fig 3 :

Fins Parameters

- convection : in such a complex geometry , exchange coefficient h_a is not easy to evaluate; locally , the flow is film like : thus , the correlation of previous section has been used , with the hydraulic diameter :

$$D_h = 4\,\frac{A_f}{p_w}$$

A_f is water flow area in a cross section , p_w is the wet perimeter :

$A_f = e_3.L - N.h.d \qquad p_w = 2(e_3 + L) + 2.N.h$

- conduction : conduction between copper/steel interface and water is derived from the fin efficiency given in ref 3 .

$$\frac{T_3 - T_a}{P_1} = \frac{1}{h_a\, p\, \left(1 + \frac{d}{p}\left(\frac{th(m) + n}{n\,(n\, th(m) + 1)} - 1\right)\right)}$$

where :

$$n = \sqrt{\frac{h_a\, d}{2\, k_c}} \qquad m = \frac{2\,h}{d}\sqrt{\frac{h_a\, d}{2\, k_c}}$$

(the temperature drop between the copper/steel interface and the fins bases is small , and is not taken into account in the above formula) .

For a given flow rate,the cooling efficiency results from a competition between 2 phenomena :

- h_a decreases with fin length increase ;
- for a given h_a , the fins efficiency increases with fin length h.

A computation of the previous formula , for typical values of parameters , shows a optimum fin length h from 1 mm to 2 mm .

3.1.7. GLOBAL THERMAL MODEL .

From sections 3.1.3. to 3.1.6. , the exchange coefficient h_c defined in section 3.1.2. is deduced (h_c represents the global effect on the back face of the inner copper):

$$h_c = \frac{L\, P_1}{T_2 - T_a}$$

T_1 is then calcultated with formula of section 3.1.2. . T_w (useful for the boiling limit) is estimated by the same method .

A few approximations have been made in the analytical model . Their validity must be checked by a numerical model .

4.FIRST RESULTS . NUMERICAL VALIDATION .

In order to validate the approximations of the analytical model , the Finite Elements Software MODULEF has been used (ref 5) in the case of tube cooling , with the following parameters :

- photon beam : $P_1 = 15$ W/mm ; $l_z = 1.25$ mm .
- inner copper : $e_1 = 2$ mm ; L = 10 mm ;
- vacuum chamber wall : $e_2 = 1$ mm ;
- cooling tubes : ø = 2 mm ; number N = 2 ; x = 2 mm ; Q = .15 m^3/h ; $T_a = 20$ °C .

Results are given in the following table:

	analytical	numerical
T_1 (°C)	204	202
T_w (°C)	73	68-76

The difference between both models is quite acceptable .

From this case , an increase of absorber height L from 10 mm to 30 mm leads to lower temperatures : $T_1 = 166$ °C . It is still possible to reduce T_1 by increasing inner copper thickness e_1 from 2 mm to 4 mm : $T_1 = 159$ °C . Greater thickness does not lead to significant improvement .In any case , T_w stays unchanged .

Using a 1 mm thick film channel instead of 2 tubes improves this situation : $T_1 = 135$ °C , $T_w = 57$ °C , but higher flow rates are necessary to avoid laminar flow : Q = .3 m^3/h .

Adding 15 fins (h = 2 mm , d = 1 mm) and increasing the flow rate to .6 m^3/h gives : $T_1 = 112$ °C

5.CONCLUSIONS .

An analytical model of E.S.R.F. low power absorbers has been developed . A numerical validation and first results are presented .

Similar work is being done for high power absorbers (crotches) , including mechanical behavior .

These models will be used as practical tools for design .

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