Heat transfer in specularly reflecting tubes

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Heat exchange inside a specularly reflecting tube is analysed. Expressions are obtained for heat transfer between cross-sections, between incremental wall annuli, between finite wall annuli, and combinations of the above. The expressions are related to the angle factor for opposed discs, but are infinite summations; they are easily evaluated, requiring some 20 emissivity terms. The basic disc-to-disc expression also represents the fraction of radiation leaving a disc that is still propagating at some distance along the tube, including reflection. Special case results are obtained for the radiant loss from double and single ended holes. For the practical application considered, it was found that conduction and radiation could be treated separately, permitting evaluation of a radiation loss factor for a specularly reflecting tube between two heat reservoirs

Keywords: *heat transfer, tube, specular*

The problem analysed here is the radiant exchange of heat in a specularly reflecting tube between two black body reservoirs, and from point to point of the tube itself. This was motivated by a specific engineering application, in minimising heat losses from a satellite attitude control thruster (a thermal storage resistojet), where a small heat exchanger, 5 mm diameter and 15 mm long, is to be maintained at high temperature, eg 1000°C or more, on a very small power input, eg 5 W. The heat-exchanger nozzle unit is fed with occasional pulses of gas through a feed pipe of 2–3 mm diameter and 50–70 mm long. When all other heat losses are closely controlled, the radiant heat flux along the feed pipe becomes significant. The outside of the heat-exchanger nozzle unit and feed pipe are provided with extremely high performance thermal insulation taking advantage of the vacuum of space. The analysis here therefore neglects radiation from the outside surface of the tube. Consideration is, however, given to conduction, and to its interaction with the radiation. End conditions are such that the use of black body end planes for the model tube is probably a good approximation.

The condition of specular reflection is of interest as a possible improvement on a diffusely reflecting surface, and because some associated test equipment had been so equipped.

Real problems in radiative heat transfer are usually concerned with diffusely reflecting high emissivity surfaces. The methods which have been developed to handle such problems usually rely on an assumption of diffuse reflection. Those that do not assume this have usually been valid only for low reflectivities[1-9]. A well finished metal surface, however, is of rather low emissivity and exhibits dominantly specular reflection[10] and, therefore, traditional techniques are not applicable. The following analysis is unusual in that it makes the specific assumption of specular reflection, and is valid for all values of emissivity.

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Direct tube radiation

A circumscribing sphere of radius R contains diametrically opposed blackbody discs of radius r (Fig 1). If the internal surface of the sphere is a Lambertian emitter, the direct radiation between any two areas is proportional to the product of the areas. This is uniform self-irradiation, and is known as Sumpner's theorem[11]. Specifically:

$$P = WS_1S_2/S_s$$

where $S_1 = S_2 = 2\pi R(R - a/2)$, the area of a spherical cap, $S_s = 4\pi R^2$ and $W = \sigma T^4$. This initially surprising result is easily confirmed, and is due to an exact compensation of angle and distance.

Now, because we are postulating black body surfaces, the radiant interchange between the two discs must be equal to that between the two spherical caps behind them (Walsh's theorem[12]). Therefore, between the two discs:

$$P = W[2\pi R(R - a/2)]^2/(4\pi R^2)$$
$$= \pi W(R - a/2)^2$$
$$= \pi W(R^2 - Ra + a^2/4)$$

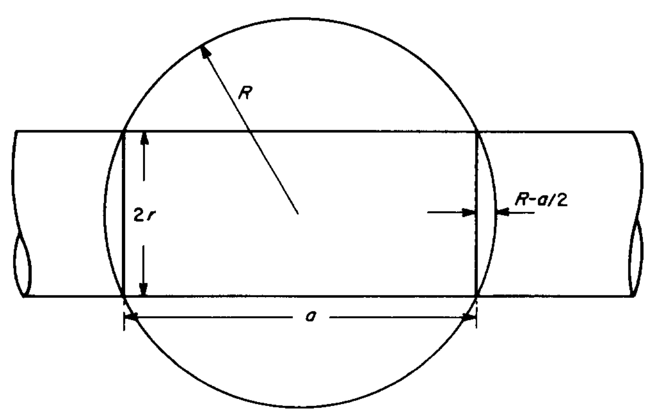

Fig 1 Cylinder sections and spherical caps

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Using:

$$R^2 = r^2 + a^2/4$$

$$P = \pi W[r^2 + \tfrac{1}{2}a^2 - a(r^2 + \tfrac{1}{4}a^2)^{1/2}]$$

The proportion of the radiation from one surface which is directly intercepted by a second surface is known as the angle (or view) factor, which we denote H, ie:

$$P = WS_1H_{12}$$

In this case:

$$H = P/(\pi r^2 W)$$

$$= 1 + \tfrac{1}{2}(a/r)^2 - (a/r)(1 + \tfrac{1}{4}(a/r)^2)^{1/2}$$

or, writing $z = a/r$, ie z is the non-dimensionalised axial position:

$$H = 1 + \tfrac{1}{2}z^2 - (z^2 + \tfrac{1}{4}z^4)^{1/2}$$

This has the approximate form of an exponential decay with characteristic length 1.0. It is true for the two discs in any non-obstructing and non-reflecting environment, and is a known expression[3].

Now any radiation passing an imaginary disc at a (Fig 2) would be absorbed by blackbody surfaces of the disc at $a + \delta a$ and the annulus between. Therefore the angle factor for the annulus is equal to the difference between the angle factors of the two discs (Bartlett's theorem[13]). Denoting the angle factor from a disc to an incremental co-cylindrical annulus by I:

$$I = -\delta H$$

$$= -\frac{dH}{dz}\delta z$$

$$= -H'\delta z$$

It is worth noting, in passing, that:

$$I = -H'\frac{\delta S}{2\pi r}$$

where δS is an incremental area. Because of the symmetry of the disc and tube, this last expression is true for any incremental area at z, not just an annulus.

The angle factor from incremental annulus to disc, J, may be derived using the reciprocity theorem:

$$S_1H_{12} = S_2H_{21}$$

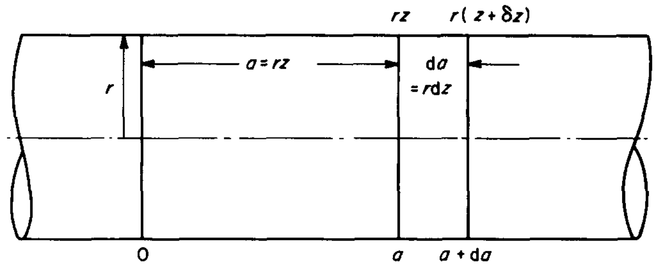

Fig 2 Dimensions for disc-annulus transfer

ie:

$$\delta SJ = \pi r^2 I$$

$$2\pi r\delta aJ = \pi r^2 I$$

$$\therefore J = \tfrac{1}{2}\frac{r}{\delta a}I$$

$$= \tfrac{1}{2}\frac{r}{\delta a}(-H'\delta z)$$

$$J = -\tfrac{1}{2}H'$$

and black body heat transfer:

$$P = WS_1H_{12}$$

$$\delta P = W2\pi r\delta a(-\tfrac{1}{2}H')$$

$$\delta P = -\pi r^2 WH'\delta z$$

The angle factor K between two incremental annuli (Fig 3) is given by the difference between two annulus-to-disc values:

$$\delta P = 2\pi r\delta a_1 WK_{12}$$

$$\delta P = 2\pi r\delta a_1 W[J(z) - J(z + \delta z)]$$

$$K_{12} = -\frac{dJ}{dz}\delta z_2$$

$$= \tfrac{1}{2}H''\delta z_2$$

Also, by symmetry:

$$K_{12} = \tfrac{1}{2}H''\delta z_1$$

Notation

a	Disc spacing, m
H	Angle factor
I	Disc to annulus angle factor
J	Annulus to disc angle factor
K	Annulus to annulus angle factor
n	Number of reflections
P	Radiant power, W
r	Cylinder radius, m
R	Sphere radius, m
S	Surface area, m^2
U	Reference power ($\pi r^2\sigma T_1^4$), W
W	σT^4, W/m^2
y	Non-dimensional annulus length
z	Non-dimensional axial position
ε	Emissivity
ρ	Reflectivity

Subscripts

AB	Range of z, eg P_{AB}
b	Of base, eg ε_b
B	From base, eg P_B
d	Of disc, eg ε_d
e	Effective, referred to exit area, eg ε_e
m	Tube annulus count, eg $_mP_s$
n	Number of reflections, eg P_n
s	Specular, eg H_s, or of sphere, eg S_s
z	Limiting axial position, eg P_z
1	Of area or disc 1, eg S_1
12	From area 1 to area 2, eg H_{12}

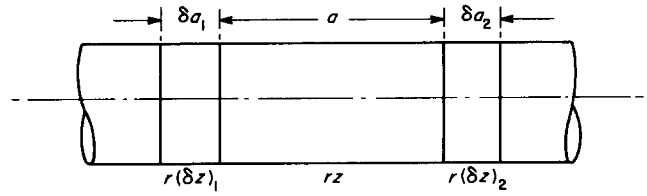

Fig 3 Dimensions for incremental annuli

The black body heat transfer is:

$$\delta P = 2\pi r r \delta z_1 W \tfrac{1}{2} H'' \delta z_2$$
$$= \pi r^2 W H'' \delta z_1 \delta z_2$$

Writing:

$$U = \pi r^2 W = \pi r^2 \sigma T_1^4$$

the disc to disc radiation is:

$$P = W S_1 H_{12}$$
$$= \pi r^2 W H = U H$$

The disc to annulus radiation is:

$$\delta P = W S_1 H_{12}$$
$$= \pi r^2 W(-H'\delta z)$$
$$= -U H' \delta z$$

The annulus to disc radiation is:

$$\delta P = W 2\pi r \delta a_1 (-\tfrac{1}{2} H')$$
$$= W \pi r^2 2 \delta z_1 (-\tfrac{1}{2} H')$$
$$= -U H' \delta z_1$$

The annulus to annulus radiation is:

$$\delta P = W 2\pi r \delta a_1 \tfrac{1}{2} H'' \delta z_2$$
$$= W \pi r^2 2 \delta z_1 \tfrac{1}{2} H'' \delta z_2$$
$$= U H'' \delta z_1 \delta z_2$$

Summarising:

$$H(z) = 1 + \tfrac{1}{2} z^2 - (z^2 + \tfrac{1}{4} z^4)^{1/2}$$
$$H'(z) = z - (z^2 + 2)(z^2 + 4)^{-1/2}$$
$$H''(z) = 1 - (z^3 + 6z)(z^2 + 4)^{-3/2}$$

and we have the following black body heat transfer equations:

$$U = \pi r^2 \sigma T_1^4$$
$$I = -H' \delta z$$
$$J = -\tfrac{1}{2} H' \delta z$$
$$K = \tfrac{1}{2} H'' \delta z_2$$

Disc to disc $P = UH$

Disc to annulus, Annulus to disc $\Big\}$ $\delta P = -UH'\delta z$

Annulus to annulus $\delta P = UH''\delta z_1 \delta z_2$

These equations have been published in rather different form before, but here are in a form convenient for the next task.

Specular reflection

It is now possible to derive equations describing the transmission of radiant heat within a tube including the effects of specular reflection. Consider a specularly reflecting tube of radius r, with an incremental annulus $\delta a = r\delta z$ acting as a Lambertian source, of emissivity ε at temperature T(Fig 4). The energy passing a plane at distance $a = rz$ from the source, after n reflections is δP_n:

$$\delta P_0 = -\varepsilon U H'(z) \delta z$$

Now consider any ray leaving δa. Where it strikes the tube wall it makes a specular (symmetrical) reflection, and therefore between each successive reflection it will proceed an unchanging distance both axially and circumferentially. Therefore, any ray making its first reflection at a distance greater than $a/2$ from the source annulus will pass the plane at a without further reflection. Therefore, for rays making a single reflection, the power is:

$$\delta P_1 = -\varepsilon U[H'(z/2) - H'(z)]\delta z \times \rho$$

where ρ $(= 1 - \varepsilon)$ is the reflectivity, causing a power loss. By the same arguments, applied to rays making more reflections;

$$\delta P_n = -\varepsilon U \rho^n [H'(z/(n+1)) - H'(z/n)]$$

The total power is:

$$\delta P = \delta P_0 + \delta P_1 + \delta P_2 + \ldots$$
$$= -\varepsilon U H'(z)\delta z - \varepsilon U \rho [H'(z/2) - H'(z)]\delta z - \ldots$$
$$= -\varepsilon U \delta z \{H'(z) + \rho[H'(z/2) - H'(z)] + \ldots + \rho^n [H'(z/(n+1)) - H'(z/n)] + \ldots\}$$
$$= -\varepsilon U \delta z (1 - \rho)\{H'(z) + \rho H'(z/2) + \ldots + \rho^n H'(z/(n+1)) + \ldots\}$$
$$= -\varepsilon U \delta z \sum_{n=1}^{\infty} (1 - \rho)\rho^{n-1} H'(z/n)$$
$$= -U \delta z \sum_{n=1}^{\infty} \varepsilon^2 (1 - \varepsilon)^{n-1} H'(z/n)$$

Therefore the total transfer from the annulus, including specular reflection, to a black body disc is:

$$\delta P = -U H_s' \delta z$$

where:

$$H_s'(z, \varepsilon) = \sum_{n=1}^{\infty} \varepsilon^2 (1 - \varepsilon)^{n-1} H'(z/n)$$

Now, if the absorbing disc in the above analysis becomes a grey body heat source at emissivity ε_d, we may write that

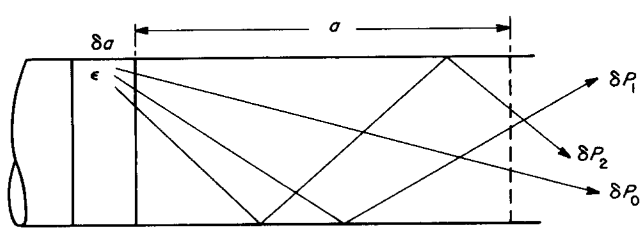

Fig 4 Power contributions for various numbers of reflections

the heat transferred from the disc to the annulus is, making use of the reciprocity theorem:

$$\delta P = -\varepsilon_d U H_s' \delta z$$

$$\frac{dP}{dz} = -\varepsilon_d U H_s'$$

The heat adsorbed into the tube between $z=A$ and $z=B$ is then

$$P_{AB} = \int_{z=A}^{B} -\varepsilon_d U H_s' dz$$

$$= -\int_{A}^{B} \left\{ \varepsilon_d U \sum_{n=1}^{\infty} \varepsilon^2 (1-\varepsilon)^{n-1} H'(z/n) \right\} dz$$

$$= -\varepsilon_d U \sum_{1}^{\infty} \left\{ \varepsilon^2 (1-\varepsilon)^{n-1} \int_{A}^{B} H'(z/n) n d(z/n) \right\}$$

The above interchange of summation and integration has a physical meaning. With the summation under the integral, we are first finding the total heat transfer, by any number of reflections, to an element, and then integrating over all elements. With the integral under the summation, we integrate the heat transfer for each number of reflections separately, and then sum them. Either is physically acceptable.

$$P_{AB} = -\varepsilon_d U \sum_{1}^{\infty} \{ \varepsilon^2 (1-\varepsilon)^{n-1} n [H(z/n)]_A^B \}$$

$$= \varepsilon_d U \sum_{1}^{\infty} \{ \varepsilon^2 (1-\varepsilon)^{n-1} n [H(z/n)]_B^A \}$$

In the particular case of $A=0$ and $B=z$:

$$P_{0z} = \varepsilon_d U \sum_{1}^{\infty} \varepsilon^2 (1-\varepsilon)^{n-1} n [H(0) - H(z/n)]$$

$$= \varepsilon_d U \sum_{1}^{\infty} \varepsilon^2 (1-\varepsilon)^{n-1} n [1 - H(z/n)]$$

$$= \varepsilon_d U \left\{ \sum_{1}^{\infty} \varepsilon^2 (1-\varepsilon)^{n-1} n - \sum_{1}^{\infty} \varepsilon^2 (1-\varepsilon)^{n-1} n H(z/n) \right\}$$

$$= \varepsilon_d U \left\{ 1 - \sum_{1}^{\infty} \varepsilon^2 (1-\varepsilon)^{n-1} n H(z/n) \right\}$$

giving:

$$P_{0z} = \varepsilon_d U [1 - H_s\}$$

where:

$$H_s = \sum_{n=1}^{\infty} \varepsilon^2 (1-\varepsilon)^{n-1} n H(z/n)$$

and the previously defined $H_s' = dH_s/dz$, correctly.

Also for the particular case of $A=z$ and $B=\infty$:

$$P_z = \varepsilon_d U H_s$$

The total emission from the source disc is $\varepsilon_d U$. The function $H_s(z, \varepsilon)$ represents the proportion of radiation propagating further than rz from the source disc, including specular reflection. Also, if we imagine a blackbody disc at rz, $\varepsilon_d U H_s$ is the radiation flux from one disc to the other, including reflection.

$U = \pi r^2 \sigma T^4$ will be evaluated for the source disc. The value of ε used for the walls should be that one appropriate to the source disc temperature.

In summary, so far we have established that the radiation flux from a disc to a co-cylindrical annulus with tubular specular reflection is given by:

$$\delta P = -\varepsilon_d U H_s' \delta z$$

or:

$$\frac{dP}{dz} = -\varepsilon_d U H_s'$$

where:

$$U = \pi r^2 \sigma T^4$$

and:

$$H_s' = \sum_{n=1}^{\infty} \varepsilon^2 (1-\varepsilon)^{n-1} H'(z/n)$$

The flux from one disc to a second, black body, one (ie the radiation passing further than rz) is:

$$P = \varepsilon_d U H_s$$

where:

$$H_s = \sum_{n=1}^{\infty} \varepsilon^2 (1-\varepsilon)^{n-1} n H(z/n)$$

H_s is fully analogous with H.

Annulus to annulus

In order to evaluate heat transfer from the walls of the tube, it is necessary to establish heat transfer rates between finite sized annuli at arbitrary spacing. Energy from annulus dz passing disc 1 including specular reflection (Fig 5) is:

$$\delta P = -U H_s' \delta z$$

$$= -U \sum_{n=1}^{\infty} \varepsilon^2 (1-\varepsilon)^{n-1} H'(z/n)$$

The total power from annulus ry_1 through disc 1 is:

$$P_{S1} = -U \int_{z_1}^{z_1+y_1} \sum_{1}^{\infty} \varepsilon^2 (1-\varepsilon)^{n-1} H'(z/n) dz$$

so:

$$P_{S1} = \varepsilon^2 U \sum_{1}^{\infty} n (1-\varepsilon)^{n-1} [H(z_1/n) - H((z_1+y_1)/n)]$$

also:

$$P_{S2} = \varepsilon^2 U \sum_{1}^{\infty} n (1-\varepsilon)^{n-1} [H((z_1+y_2)/n) - H((z_1+y_1+y_2)/n]$$

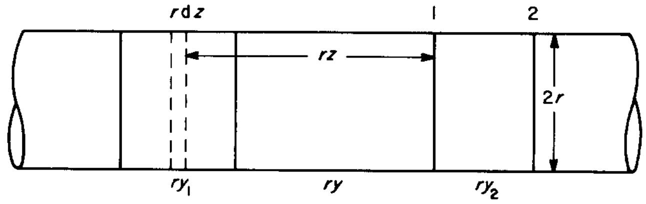

Fig 5 Dimensions for finite annuli

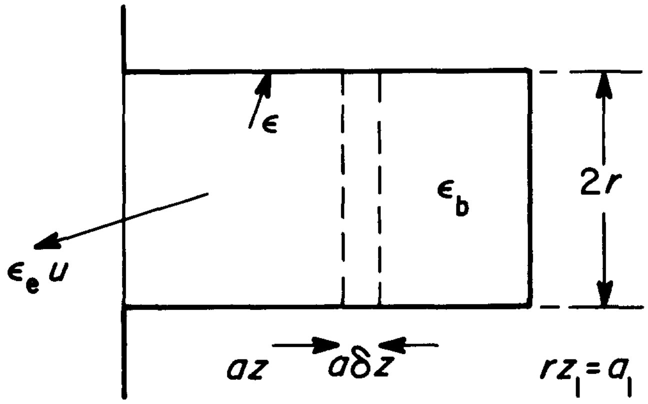

Fig 6 Circular cylindrical hole

The heat transfer to annulus ry_2 is $P_{S1}-P_{S2}=P_S$ where:

$$P_S=\varepsilon^2U\sum_{n=1}^{\infty} n(1-\varepsilon)^{n-1}[H(z_1/n)-H((z_1+y_1)/n)$$
$$-H((z_1+y_2)/n)+H((z_1+y_1+y_2)/n]$$
$$P_S=U[H_S(z_1/n)-H_S((z_1+y_1)/n)$$
$$-H_S((z_1+y_2)/n)+H_S((z_1+y_1+y_2)/n)$$

In the case when $y_1=y_2=y$ and $z_1=(m-1)y$ (m integer), the tube is divided into a number of equal size segments:

$$_{m}P_{s}=U\left[H_S\left(\frac{m-1}{n}y\right)-2H_S\left(\frac{m}{n}y\right)+H_S\left(\frac{m+1}{n}y\right)\right]$$

This form of the equation is ideal for numerical (ie computer) evaluation of the heat transfer.

Where the radiation is passing along a non-isothermal tube it is necessary to consider the effect of variation of reflectivity with temperature. Fortunately, the reflectivity of a surface at temperature T_2 to radiation whose spectral distribution is characterised by T_1 is effectively a function of T_1 only. Therefore the radiation from an element at T_1 is distributed along the tube according to the function $H_s(z,\varepsilon)$ where $\varepsilon=\varepsilon(T_1)$. The rate of heat transfer from any particular element may therefore be found without reference to any temperature other than that of the source, a very important simplification.

Radiation loss from an isothermal hole

The source is an isothermal cylindrical hole, specularly reflecting, with emissivity ε on the wall, and ε_b on the base (Fig 6). If P_{W1} is the radiation from the wall which escapes without reflection from the base, P_{W2} the radiation from the wall which escapes after reflection from the base, and P_B the radiation from the base which escapes, with or without reflection:

$$\delta P_{W1}=-UH_s'\mathrm{d}z$$

$$P_{W1}=\int_0^{a/r}-UH_s'\mathrm{d}z$$
$$=-U[H]_0^{a/r}$$
$$=U(H_s(0)-H_s(z))$$
$$\therefore\quad P_{W1}=U(1-H_s(z_1,\varepsilon))$$

$$P_{W2}=\rho_B\int_{a/r}^{2a/r}-UH_s'\mathrm{d}z$$
$$=\rho_B U[H_s]_{2a/r}^{a/r}$$
$$\therefore\quad P_{W2}=\rho_B U(H_s(z_1,\varepsilon)-H_s(2z_1,\varepsilon))$$
$$P_B=\varepsilon_B UH_s(z_1,\varepsilon)$$
$$P=P_{W1}+P_{W2}+P_B$$
$$=U[(1-H_s(z_1,\varepsilon))+\rho_B U(H_s(z_1,\varepsilon)$$
$$-H_s(2z_1,\varepsilon))+\varepsilon_B UH_s(z_1,\varepsilon)]$$
$$=U[(1-H_s(z_1,\varepsilon))+\rho_B H_s(z_1,\varepsilon)$$
$$-\rho_B H_s(2z_1,\varepsilon)+\varepsilon_B H_s(z_1,\varepsilon)]$$
$$P=U(1-\rho_B H_s(2z,\varepsilon))$$
$$\varepsilon_e=1-(1-\varepsilon_B)H_s(2z,\varepsilon)$$

where ε_e is the effective emissivity on the exit plane area. Two cases of special interest are:

1. $\varepsilon_B=1$: $\varepsilon_e=1$, independent of ε (as expected)
2. $\varepsilon_B=\varepsilon$: $\varepsilon_e=1-(1-\varepsilon)H_s(2z_1,\varepsilon)$

Radiation loss from a double-ended isothermal hole

The loss from each end (Fig 7) is:

$$P_1=\int_0^{z}-UH_s'\mathrm{d}z$$
$$=-U[H_s]_0^z$$
$$=U[H_s(0)-H_s(z)]$$
$$=U(1-H_s(z))$$

The total loss is $P=2P_1$:

$$P=2U(1-H_s(z))$$
$$\varepsilon_e=1-H_s(z)$$

based on a total exit area of $2\pi r^2$.

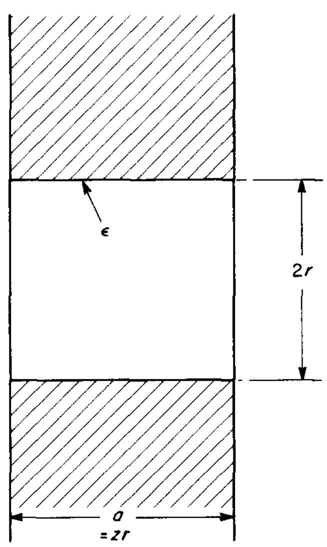

Fig 7 Circular cylindrical perforation

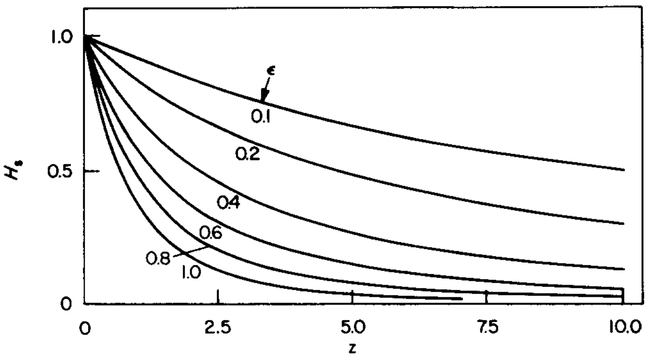

Fig 8 Specular view factor H_s *versus axial distance*

Evaluation of H functions

In order to utilise the equations developed above, it is necessary to evaluate the H functions. $H(z)$, $H'(z)$ and $H''(z)$ are found by direct substitution. H_s is the sum of an infinite series which is easily evaluated by computer (Fig 8).

Applications

The equations for emissivity of isothermal holes and evaluation of H_s have already been presented, permitting immediate use. However, the main application of the above theory to the thermal storage resistojet is to evaluate the radiation loss in the non-isothermal feed pipe. This may be modelled by a tube between two heat reservoirs. Some assumption must be made regarding the tube-reservoir interface, and this has been taken to be equivalent to a blackbody disc at the reservoir temperature. If the tube is now divided into a number of equal sized elements, the equations presented above may be used to compute the radiant heat exchange between the elements, and between the elements and ends, and also directly from end to end, under some assumption of temperature distribution. The final heat flux for each element may then be used to give a new estimate of the temperature distribution, iteration continuing until specified tolerances on element heat flux balance and uniformity of total heat flux are established. Conduction and external radiation may be included as required. The metal thermal conductivity has been included as a function of temperature which may be changed for different materials. For rhenium, $k=0.04$ W/mmK has been used.

For realistic thermal storage resistojet values, when a complete calculation including conduction and internal radiation is performed, the result is an almost identical loss to that obtained by calculating radiation and conduction independently. This is very useful because a radiation loss factor (loss/$\pi r^2\sigma(T_1^4-T_2^4)$) may be evaluated as a function of ε and z without reference to associated conduction. The results are presented graphically in Fig 9. This makes a useful estimate of the radiation loss available without repeated computing.

It is notable in these figures that a lower emissivity gives a higher loss. For radiation exchange from an element of the tube, the emitted power is proportional to ε, and the axial distance travelled before absorption, is, broadly, inversely proportional to ε, so the effective transfer would be insensitive to ε. However, with a black body source, as at the ends, the low emissivity allows many reflections before absorption, and hence a high heat loss.

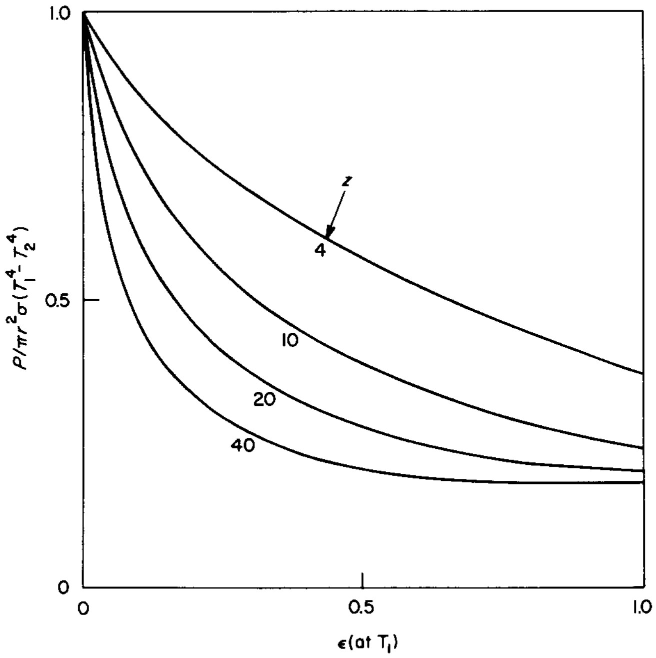

Fig 9 Power transfer versus emissivity

Conclusions

The radiant heat interchange between cylinder cross sections, between wall annuli, and between a cross section and an annulus, for the case of a tube with internal specular reflection, may be expressed in terms of a specular angle (view) factor H_s and its derivatives:

$$H_s=\sum_{n=1}^{\infty}\varepsilon^2(1-\varepsilon)^{n-1}n\left[1+\tfrac{1}{2}\left(\frac{z}{n}\right)^2-\left(\left(\frac{z}{n}\right)^2+\tfrac{1}{4}\left(\frac{z}{n}\right)^4\right)^{1/2}\right]$$

where the term in square brackets corresponds to the angle factor for direct radiation only. This is easily computer summed, requiring about $20/\varepsilon$ terms, regardless of z.

Expressions are easily derived for related heat transfer problems with cylindrical specular reflection, including a general formulation of heat balance for a complete tube.

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BOOK REVIEW

Gas Turbine Combustion

A. H. Lefebvre

This book should be compulsory reading for those interested in combustion processes. It is directed particularly at gas-turbine combustion and brings together a wealth of personal knowledge and experience within a framework which is sensible and helpful to the reader. The approach to the various aspects of the subject is a mixture of phenomenological and empirical.

Of the 11 chapters, two describe basic principles of combustors and combustion, two are concerned with the aerodynamic characteristics of the flow in diffusers and combustors, three with the combustion topics of efficiency, stability and injection, one with heat transfer, two with fuels and their injection and one with pollution. Consistent with the author's main interests, approximately half of the book is devoted to the chapters dealing with aerodynamics, heat transfer, fuels and their injection. Since the basis for the book has been provided by a lecture course given and developed at the Cranfield Institute of Technology and elsewhere over a period of years, it is not surprising that the material is based largely on papers and reports which are more than 10 years old. An important exception is the material on emissions which, necessarily, is more recent.

The lack of emphasis on more recent material means that development arising from numerical methods and optical diagnostic techniques tend to be ignored. It can be argued, and the author would probably subscribe to this view, that these developments have so far done little to improve design methods. It is a little odd, nevertheless, to end the chapter on aerodynamics with the advice that the major aerodynamic problem is one of stability, and that the cure may lie in the compressor.

The text is clear and displays an enthusiasm for the subject. As a result, reading is a pleasure and gleaning new information an easy matter. There are few books on combustion and very few which deal with gas-turbine combustion. It is a pleasure to recommend this one to all with an interest in the subject.

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Pool boiling performance of finned surfaces in R-113
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Natural convection in an inclined enclosure containing internal energy sources and cooled from below
S. Acharya

Numerical simulation of transient response of heat transfer from a hot-wire anemometer transducer
K. J. Bullock, M. A. Ledwich and J. C. S. Lai

A technique for obtaining approximate solutions for a class of integral equations which arise in radiative heat transfer
B. C. Choi and S. W. Churchill

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B. R. Clayton and Y. S. M. Morsi

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Prediction of turbulent source flow between stationary and rotating discs
C. R. Truman and D. F. Jankowski

These articles, listed in alphabetical order of first-named author, will appear in forthcoming issues of the *International Journal of Heat and Fluid Flow*.