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Heat pipes – performance of two-phase closed thermosyphons

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HEAT PIPES – PERFORMANCE OF TWO-PHASE CLOSED THERMOSYPHONS

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HEAT PIPES – PERFORMANCE OF TWO-PHASE CLOSED THERMOSYPHONS

1. NOTATION AND UNITS

The *SI* system of units is used throughout.

		<i>SI</i>
A	cross-sectional area, $\pi D^2/4$	m^2
Bo	Bond number, defined in Equation (5.4)	
C	constant in Equations (4.1) and (4.2)	
c_p	specific heat capacity	J/kg K
D	internal diameter	m
D_o	external diameter	m
f_1	a function of Bond Number, obtained from Figure 1	
f_2	a function of K_p , obtained from Equation (5.6)	
f_3	a function of β , obtained from Figure 2	
F	liquid fill (fraction of evaporator covered by static pool), $V_l/(Al_e)$	
g	local gravitational acceleration*	m/s^2
h	heat transfer coefficient, referred to inside diameter	$W/(m^2K)$
H	depth	m
K_p	dimensionless pressure parameter, defined in Equation (5.5)	
L	specific latent heat of vaporisation	J/kg
l	length	m
p_a	atmospheric pressure [†]	Pa (<i>i.e.</i> N/m ²)
p_p	pressure at bottom of pool	Pa (<i>i.e.</i> N/m ²)
p_v	vapour pressure	Pa (<i>i.e.</i> N/m ²)
$\dot{Q}$	rate of heat transfer	W
$\dot{Q}_{max}$	maximum rate of heat transfer	W

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Re_f	liquid-film Reynolds number in adiabatic section between condenser and evaporator, $4\dot{Q}/(L\mu_l\pi D)$	
r	radius	m
S	surface area, $\pi D l$	m ²
T	temperature	K
T_s	saturation temperature	K
ΔT	effective overall temperature difference ($T_{so} - T_{si} - \Delta T_h$)	K
ΔT_h	mean temperature difference due to hydrostatic head (Section 4.2)	K
V	volume	m ³
V_l	volume of liquid in unheated pipe	m ³
z	thermal resistance, $1/(hS)$	K/W
β	inclination of pipe to horizontal	deg
λ	thermal conductivity	W/(m K)
ε	porosity, (volume of pores)/(volume of porous material)	
μ	dynamic viscosity	N s/m ² (i.e. kg/(s m))
ρ	density	kg/m ³
σ	surface tension	N/m
Φ_2	Figure of Merit for condensation, $(L\lambda_l^3 \rho_l^2/\mu_l)^{0.25}$ (from Reference 9)	kg/(K ^{0.75} s ^{2.5})
Φ_3	Figure of Merit for nucleate boiling (Equation (4.5))	

Subscripts

a	refers to adiabatic section
c	refers to condenser
e	refers to evaporator
f	refers to liquid film
h	refers to hydrostatic head
l	refers to liquid
m	mean

<i>o</i>	refers to external dimension
<i>p</i>	refers to nucleate boiling in the pool
<i>si</i>	refers to heat sink
<i>so</i>	refers to heat source
<i>v</i>	refers to vapour
<i>w</i>	refers to wick
<i>x</i>	refers to container wall

* Standard value at sea level = 9.81 m/s^2 .

† Standard value at sea level = 101 325 Pa.

2. INTRODUCTION

2.1 Scope

This Data Item, which is one of a series concerned with heat pipes, provides methods of estimating the steady-state performance of a two-phase closed thermosyphon, *i.e.* a device that performs the same function as a heat pipe, but which does not rely on capillary forces to circulate the liquid; it places the condenser higher than the evaporator in order to use gravity to drive the liquid from the condenser to the evaporator. A greater rate of flow of liquid is possible if it is physically feasible to use gravity, instead of capillary forces, to produce this flow. Furthermore the cost of purchasing and installing wicks into the pipes is avoided.

The use of thermosyphons should be considered whenever it is possible to place the source of the heat at or below the level of the sink of the heat. Applications include the recovery of waste heat from the hot exhaust gas leaving an industrial process or furnace (Reference 10); the heat sink may be a steam generator (References 10 and 12), or a preheater for the incoming air or process fluid (Reference 7). They have been considered for the prevention of ice formation on navigation buoys (Derivation 18).

Typical thermosyphons covered by this Data Item are shown in Sketches 2.1 and 2.2. The devices dealt with have:—

- (i) circular tubes with uniform cross-section,
- (ii) a single-component working fluid and no non-condensable gas,
- (iii) either no wick, or a simple wick or insert in the evaporator section to improve liquid contact with the wall,
- (iv) angle of inclination to the horizontal 5° to 90° , with the evaporator at the lower end.

The gravity-assisted heat pipe, *i.e.* where capillary and gravity forces combine to drive the liquid from the condenser to the evaporator, is a special case not covered by this Data Item; this device is dealt with in Reference 8.

A procedure for calculating the performance of an existing thermosyphon is given in Section 6 and a flow chart is provided in Figure 3; this may also be used to check a design. The procedure is useful for design purposes, although no method for optimising the design has been included because many optimising parameters exist and it is difficult to generalise about their relative importance. The details of the calculational procedure depend on prescribed boundary conditions, as discussed in Section 3.3.

The performance calculations consist of two parts.

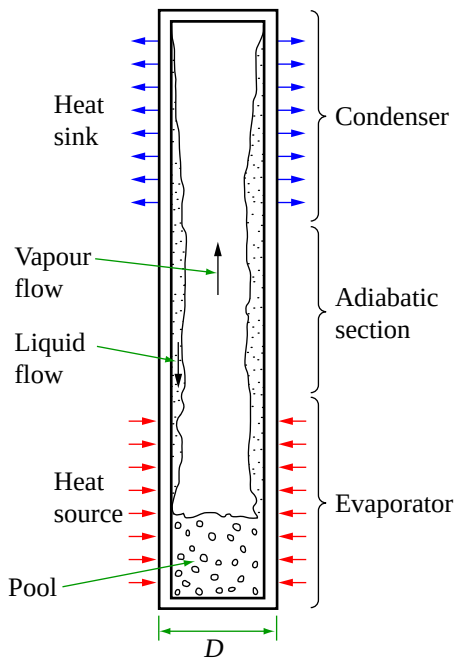
- (1) Any unknown temperatures are calculated, as required; if the rate of heat transfer, $\dot{Q}$, is not given, it must be determined by trial-and-error to satisfy the prescribed temperatures. These calculations are described in Sections 3 and 4.
- (2) The rate of heat transfer is compared with various calculated limits, to ensure that they are not exceeded, as described in Section 5.

An example (Section 7) illustrates the performance calculations.

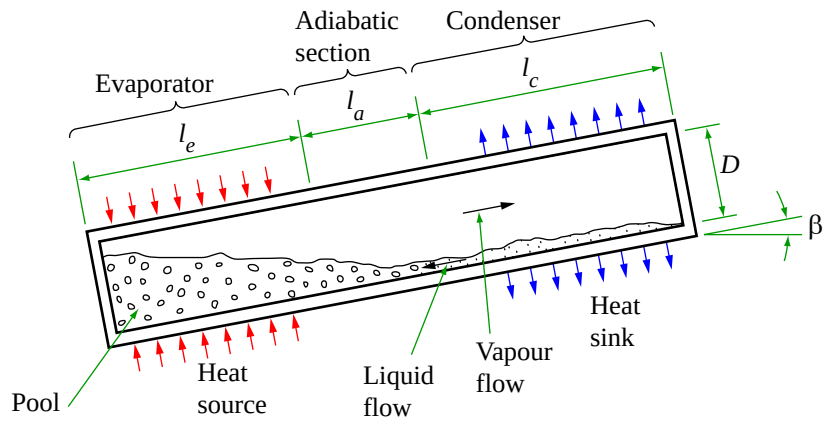
2.2 Steady-state Performance

Thermosyphons are generally either vertical or inclined at about 5° to the horizontal. Sketch 2.1 illustrates a typical two-phase closed thermosyphon operating vertically; Sketch 2.2 shows a similar item operating almost horizontally. Each comprises an evaporator, an adiabatic section (which may be very short) and a condenser, the evaporator being at a lower level than the condenser. It is important to distribute the film of liquid evenly over the part of the surface of the evaporator not covered by the pool. Thus a vertical item must be installed as nearly as possible in the vertical position, otherwise the condensate will be unevenly distributed around the periphery where it enters the evaporator. When this is not possible, *e.g.* in a transport application, and with inclined items, the distribution of liquid over the heated surface must be encouraged, either by installing a wick, or by cutting spiral grooves inside the pipe, in the evaporator section; alternatively, excess liquid may be used (see Section 2.3).

The advantages of the vertical over the almost horizontal item are that the precautions to ensure good wetting are less important, and the maximum heat flux is higher (see Section 5.5). Almost horizontal pipes are used when it is more convenient to place the heat source beside, rather than below, the heat sink, and when the heat transfer rate is well below the maximum. Intermediate angles may be used; these need the precautions to ensure good wetting, and if the inclination to the horizontal is less than 50° there may be a reduction in the maximum heat flux (see Section 5.5).



Sketch 2.1 Vertical thermosyphon



Sketch 2.2 Inclined thermosyphon

2.3 Liquid Fill

Insufficient liquid fill leads to dryout (see Section 5.3). Excess liquid fill should be avoided, because any liquid carried up into the condenser reduces the performance by rendering the covered area of the condenser virtually useless. Also, the liquids used are often expensive.

The liquid fill, denoted by F , is defined as the ratio of the volume of the liquid in an unheated pipe (V_l) to the volume of the evaporator; hence

$$F = \frac{V_l}{Al_e} \quad (2.1)$$

where $A (= \pi D^2/4)$ is the internal cross-sectional area of the pipe and l_e is the length of the evaporator section.

It is recommended in Derivation 19, p. 311, that, for a vertical pipe, F should be more than 50% and also that

$$V_l \geq 0.001D(l_e + l_a + l_c) \quad (2.2)$$

where l_e , l_a and l_c are the lengths of the evaporator, the adiabatic, and the condenser sections respectively; this is sufficient to provide a mean film thickness of more than 0.3 mm over the total length.

On the basis of the information available in the Derivations, it is recommended here that, for a pipe without a wick, Equation (2.2) should be satisfied and the liquid fill (F in Equation (2.1)) should be in the range 40% to 60% for vertical pipes and 60% to 80% for inclined pipes. There is some reduction (5% to 10%) in the maximum heat flux at the upper values of the liquid fill (Derivations 21, 22 and 26).

When a wick is fitted in the evaporator section, the only requirement is that the minimum liquid fill should be

$$V_l = 0.001D(l_a + l_c) + A_w l_e \varepsilon \quad (2.3)$$

where A_w is the cross-sectional area of the wick and ε is its porosity (see Reference 8).

3. OVERALL THERMAL RESISTANCE

When operating below its maximum overall rate of heat transfer, $\dot{Q}_{max}$, the performance of a two-phase closed thermosyphon can be characterised by the overall thermal resistance*, z . The actual overall rate of heat transfer, $\dot{Q}$, and the effective temperature difference between the heat source and the heat sink ($\Delta T = T_{so} - T_{si} - \Delta T_h$), are then related by

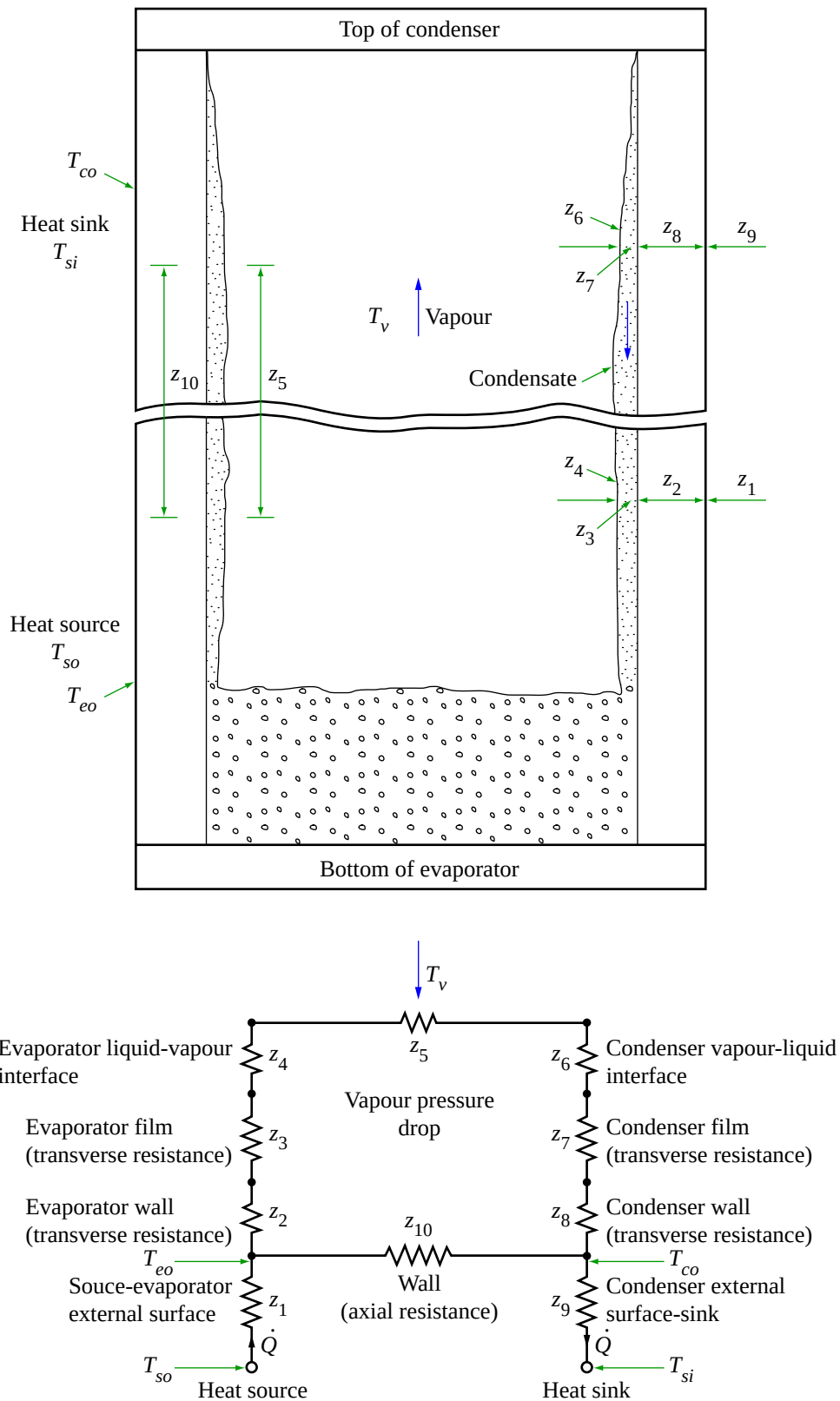
$$\dot{Q} = \Delta T / z. \quad (3.1)$$

3.1 Network of Resistances

Section 4.2 deals with the estimation of ΔT_h , the mean temperature difference due to hydrostatic head.

The overall thermal resistance can be represented by the idealised network of thermal resistances z_1 to z_{10} as shown in Sketch 3.1 and explained in this Section. The subscripts are consistent with other Data Items in the series.

* In this Data Item, thermal resistance is as defined by Equation (3.1), and has units of K/W.



Sketch 3.1 Thermal resistances and their locations

The thermal resistances arise as follows.

z_1 and z_9 The thermal resistances between the heat source and the evaporator external surface and between the condenser external surface and the heat sink respectively. In some cases the temperature of one or both of the evaporator and condenser external surfaces may be known or the heat flux at one of these surfaces may be prescribed. In these cases one or both of z_1 and z_9 need not be calculated. These resistances are given by

$$z_1 = 1/(h_{eo}S_{eo}) \text{ and } z_9 = 1/(h_{co}S_{co}). \quad (3.2)$$

If the heat source is a condensing vapour, z_1 can be calculated from the normal correlations for condensation (References 1 and 2). The same applies to z_9 if the heat sink is a boiling liquid. If heat is removed at the sink by natural convection and radiation, Reference 6 may be useful.

For single-phase forced convection in flow across a bank of plain tubes the heat transfer coefficients, h_e and h_c , can be calculated by the method given in Reference 5. If external transverse fins are attached to the pipe, Reference 4 may be consulted for the estimation of their efficiency and of the heat transfer coefficient in flow across banks of tubes with transverse fins (correlations or curves of the heat transfer coefficient can often be obtained from the manufacturers of the finned surfaces).

z_2 and z_8 The thermal resistances across the thickness of the container wall in the evaporator and the condenser respectively. For a tube of wall thickness t_x these resistances are given by Equations (3.3) (Reference 8, p.23):

$$z_2 = \frac{\ln(D_o/D)}{2\pi l_e \lambda_x} \text{ and } z_8 = \frac{\ln(D_o/D)}{2\pi l_c \lambda_x}. \quad (3.3)$$

z_3 and z_7 The internal thermal resistances of the boiling and condensing fluid in the thermosyphon. This is a complicated function of the fluid properties, the dimensions of the thermosyphon and the rate of heat transfer. Methods of estimating z_3 and z_7 are given in Section 4.1 for vertical tubes; the possible extension to inclined tubes is discussed in Section 4.3.

z_4 and z_6 The thermal resistance that occurs at the vapour-liquid interface in the evaporator and the condenser respectively. These are always neglected, being exceedingly small.

z_5 The effective thermal resistance due to the pressure drop of the vapour as it flows from the evaporator to the condenser. As a result of this pressure drop, there is a drop in the saturation temperature (which controls the rate of condensation). Thus z_5 is the drop in saturation temperature between the evaporator and the condenser divided by the rate of heat transfer ($\dot{Q}$). When operating well below the maximum heat transfer (see Section 5), z_5 is small compared with z_3 and z_7 . A method of estimating z_5 from the pressure drop of the vapour is given in Reference 8, page 23.

z_{10} The axial thermal resistance of the wall of the container. As well as conducting heat through its thickness, the wall conducts heat axially. This resistance is given approximately by

$$z_{10} = (0.5l_e + l_a + 0.5l_c)/(A_x \lambda_x) \quad (3.4)$$

where A_x is the cross-sectional area of the wall of the container and λ_x its thermal conductivity.

For the normal mode of operation (with the evaporator below the condenser) axial conduction in the container contributes very slightly to the performance. When heat is flowing in the reverse direction, the small amount of axial conduction is the only mode of heat transfer; thus the thermosyphon may be used to act as a “thermal diode”.

A practical criterion for negligible (less than 5%) axial conduction is

$$z_{10}/(z_2 + z_3 + z_5 + z_7 + z_8) > 20. \quad (3.5)$$

If Equation (3.5) is satisfied, the overall thermal resistance is approximated by

$$z = z_1 + z_2 + z_3 + z_5 + z_7 + z_8 + z_9. \quad (3.6)$$

Otherwise,

$$z = z_1 + [(z_2 + z_3 + z_5 + z_7 + z_8)^{-1} + 1/z_{10}]^{-1} + z_9. \quad (3.7)$$

With reverse heat flow in a thermal diode, there is no internal heat transfer and the overall thermal resistance is

$$z = z_1 + z_9 + z_{10}. \quad (3.8)$$

3.2 Internal Resistance

The term “internal resistance” is applied to the sum of the resistances inside the pipe, namely $z_3 + z_4 + z_5 + z_6 + z_7$. Since z_4 and z_6 are always negligible and z_5 is usually negligible (see Section 5), it is assumed in the following Sections that the internal resistance is $z_3 + z_7$.

3.3 Boundary Conditions

Four common types of boundary condition are considered, the symbols for the temperatures being as shown on Sketch 3.1.

- (1) The rate of heat transfer ($\dot{Q}$) is specified and it is required to estimate the temperature difference across the thermosyphon ($T_{eo} - T_{co}$), to ensure that this is small compared with the overall temperature difference. One temperature must also be given, so that the temperature of the vapour may be calculated.
- (2) The temperatures of the condenser external surface (T_{co}) and of the heat sink (T_{si}) are specified; it is required to estimate the rate of heat transfer ($\dot{Q}$) and the temperature required at the heat source (T_{so}). This applies to heating a liquid when there is a limit to the temperature of the heating surface.
- (3) The temperatures of the evaporator external surface (T_{eo}) and of the heat source (T_{so}) are specified; it is required to estimate the rate of heat transfer ($\dot{Q}$) and the temperature required at the heat sink (T_{si}). This applies to cooling a gas or liquid when the temperature of the cooling surface