An open oscillatory heat pipe steam-powered boat

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Abstract An open oscillatory heat pipe is a very simple device that is capable of doing work. Although as an engine its thermal efficiency is relatively poor, as a heat transfer device it can transfer a relatively large quantity of heat against gravity, without a wicking material or any moving parts. The thermal and fluid dynamic equations are given whereby the device may be mathematically modelled. Computer-generated results for thrust, pressure and temperature as a function of time are given and discussed. Comparing the theoretical with experimentally obtained thrust curves, it is concluded that the mathematical model reflects the physical behaviour of the device reasonably well. Limitations of the present model are discussed and it is recommended that the device should find application in heat transfer rather than as an engine.

Keywords oscillatory heat pipe; pulsating heat pipe; open oscillatory heat pipe; putt-putt boat; thermodynamic cycle; fluid dynamics; non-linear mechanics; chaos theory

Notation

A	area, m^2
c_p	specific heat at constant pressure, J/kg °C
c_v	specific heat at constant volume, J/kg °C
C_f	coefficient of friction
d	diameter, m
g	gravitational constant, 9.81 m/s^2
g	force (due to gravity), N
h	heat transfer coefficient, W/m^2 °C
i	specific enthalpy, J/kg
i_{fg}	latent heat of vaporisation, J/kg
k	thermal conductivity, W/m °C
L	length, m
L_c	capillary length (see equation 1), m
m	mass, kg
$\dot{m}$	mass flow rate, kg/s
$\dot{m}''$	mass flux, kg/sm^2
P	pressure, Pa
$\dot{q}$	heat transfer rate, W
$\dot{q}''$	heat flux, W/m^2
R	specific gas constant, J/kgK
R	radius of curvature, m
R	thrust, N

Re Reynolds number, $Re = \rho vd/\mu$
T temperature, °C or K
t time, s
U overall heat transfer coefficient, W/m^2 °C
V volume, m^3
v velocity, m/s
x distance, m

Greek letters
Δ difference
δ thickness, m
θ contact angle, °
μ dynamic viscosity, kg/ms
ρ density, kg/m^3
σ suface tension, N/m
$\hat{\sigma}$ accommodation coefficient
τ shear stress, N/m^2

Subscripts
a adiabatic
c condenser, condensation, cold
e exit, environment, evaporator
f friction, film
h hot
i inlet
ℓ liquid
L leading
p plug, pipe
T trailing
v vapour, velocity

Introduction and object

At Stellenbosch University, as part of their design course, the second-year mechanical engineering students are given a group project entailing an element of ingenuity, creativity and competition. In one such annual project, groups of about four students each competed to see who could come up with a boat that would go the furthest given a small amount of fuel. One of the more successful designs consisted simply of a bent piece of copper tube and a spirit lamp, as the engine, mounted on a float. As a toy boat, such a design has been known for many years. The theory upon which the engine works is explained qualitatively in a number of sources but no quantitative explanation was found.

A very simple do-it-yourself kit has been developed from which a working model of a boat can easily and quickly be constructed, not so much to expound the suitability and merits of various hull construction details but rather as a simple and

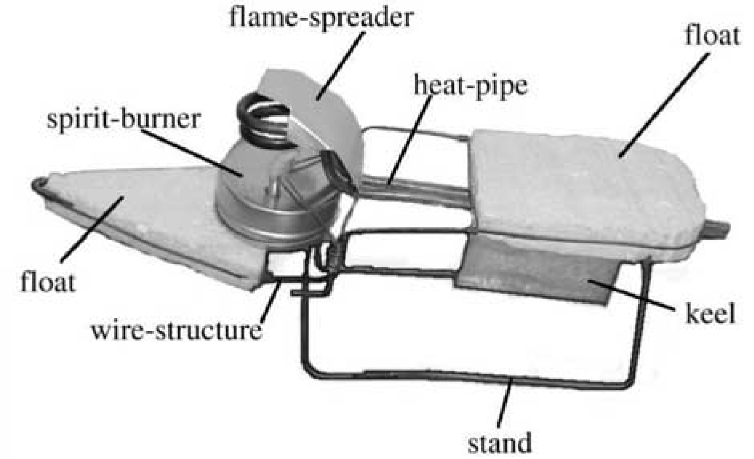

Fig. 1 *Open oscillatory heat pipe steam (oohps) boat.*

hopefully interesting example of a working thermodynamic engine. Fundamental thermodynamic and fluid dynamic principles may be then used to explain its functioning. A built-up model is shown in Fig. 1 and it can be seen that it is relatively easy to construct, with a minimum of sophistication, by even the youngest of mechanical engineers.

The theory on which the open oscillatory heat pipe steam engine works is explained in this article. The approach has been not to treat the engine as a peculiarity in its own right but rather to treat it as an open oscillatory or pulsating heat pipe, namely as a particular device in the relatively new field of heat pipe science and technology. General background theory is given, including some historical facts, before a mathematical model is presented. Computer-generated solutions are given, compared with experimentally obtained results, discussed and conclusions drawn.

Background theory

The concept of a passive two-phase heat transfer device capable of transferring large quantities of heat with a minimal temperature drop was first introduced by Gaugler in 1942 [1]. This device received little attention until 1964, when Grover and his colleagues at Los Alamos National Laboratories published the results of an independent investigation and first applied the term *heat pipe* [2]. Since that time, heat pipes have been employed in many applications, ranging from temperature control of the permafrost layer under the Alaska pipeline to the thermal control of optical surfaces in spacecraft [3].

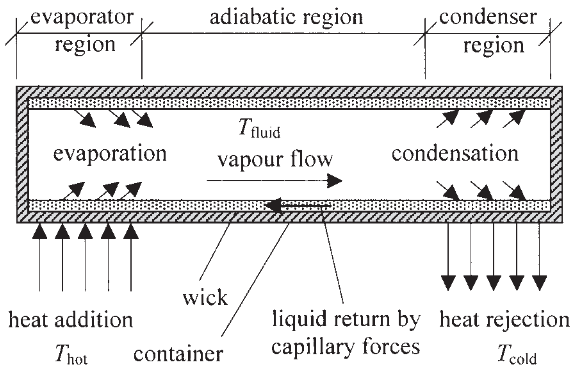

Fig. 2 *Typical heat pipe construction and operation.*

A heat pipe consists typically of a sealed container with a wicking material. The container is evacuated and filled with just enough liquid to fully saturate the wick. As illustrated in Fig. 2, a heat pipe consists of three distinct regions: an evaporator or heat-addition region of the container, a condenser or heat-rejection region, and an adiabatic or isothermal region. When the evaporator region is exposed to a high temperature, heat is added and the working fluid in the wicking structure is heated until it evaporates. The high temperature and the corresponding high pressure in this region cause the vapour to flow to the cooler condenser region, where the vapour condenses, giving up its latent heat of vaporisation. The capillary forces in the wicking structure then pump the liquid back to the evaporator. The wick structure thus ensures that the heat pipe can transfer heat if the heat source is below the cooled end (*bottom heat mode*) or if it is above the cooled end (*top heat mode*).

If the heated region is below the cooled region (bottom heat mode) the condensate is able to return to the evaporator under the influence of gravity. Under this bottom heat mode operating condition, the wicking structure may be dispensed with and the heat pipe is then often called a wickless or gravity-assisted heat pipe or probably most commonly a two-phase closed *thermosyphon*.

An oscillatory or pulsating heat pipe consists of a long pipe with a relatively small diameter and bent so as to meander back and forth between the heated and cooled ends, as shown in Fig. 3. It is typically filled with a liquid to vapour volume ratio of about 50%, resulting in a large number of liquid plugs and vapour bubbles. The great advantage of such a device is that, although it has no wick structure, it is able to, unlike a thermosyphon, transfer heat from the heated region to the cooled region independent of gravity. When heat is being transferred, the liquid plugs (see Fig. 4a) may be observed in a glass heat pipe to oscillate back, forth and around the bends in a haphazard and seeming random fashion. It is important, though, that there is a relatively large number of turns, say about 20 or more.

If the diameter is too large, the liquid will stratify as shown in Fig. 4b, and the pulsating heat pipe will revert to a thermosyphon and be able to transfer heat only

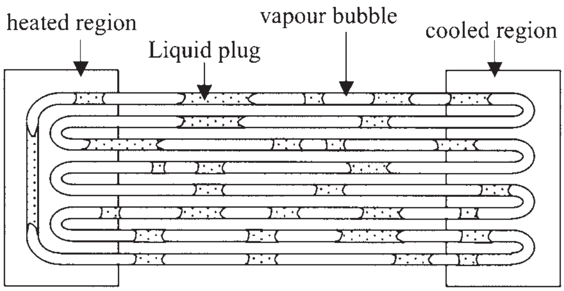

Fig. 3 *An example of a closed oscillatory heat pipe.*

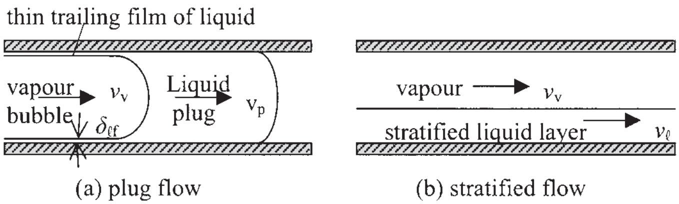

Fig. 4 *Two types of flow in a pipe: (a) plug plow, and (b) stratified flow.*

in the bottom heat mode. To ensure that the liquid will form plugs it has been empirically found that the diameter must be between 0.7 and 1.8 of the capillary length [4], which itself is given by:

$$L_c = \sqrt{\sigma / (\rho_\ell - \rho_v) g} \tag{1}$$

The capillary length is a grouping of fluid properties and gravity. It appears in the relationship for the shape of a free liquid surface meeting a vertical plane wall [5]. For water at 20 °C, a suitable diameter using equation (1) would be between 2 and 5 mm.

If the heat pipe container is not sealed but open to the atmosphere it is called an open heat pipe. A rather dramatic example of such a heat pipe is a geyser, where a burst of steam-driven hot water is propelled into the sky, as shown in Fig. 5. The hot surrounding rock heats the water collecting in the earth's fissures. This causes the water to boil and form steam beneath the surface of the earth. The steam is hindered from escaping by colder water on the surface, builds up pressure and ultimately blasts the water upwards in order to escape. The escaping steam actually drags the water along with it on its rapid motion through the earth's fissures. (A famous geyser is Old Faithful in Wyoming's Yellowstone National Park in the USA. Every 60 minutes or so Old Faithful sends a spray of steam-driven hot water some 30–45 m into the sky.)

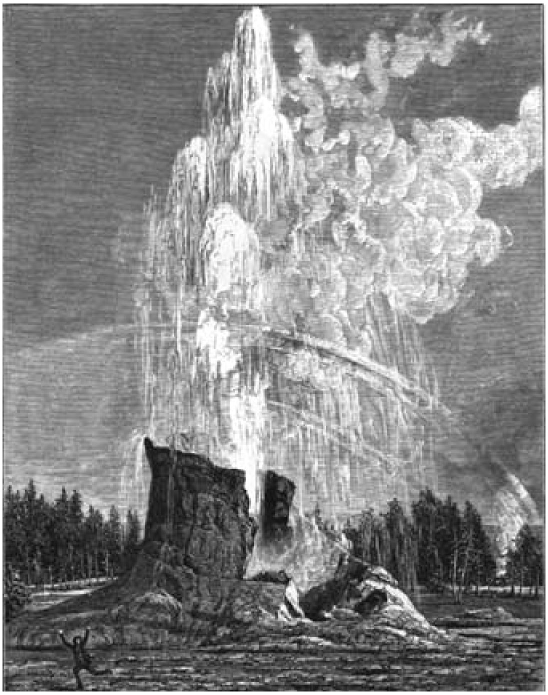

Fig. 5 *A geothermal geyser.*

A less dramatic but somewhat more controlled example of an open heat pipe is the one used to power a putt-putt or pop-pop boat. A putt-putt boat is a toy boat and originated in an 1891 British patent for water pulse engines by an inventor named Thomas Piot [6]. It is also known by other names, in Germany as toc-toc, and elsewhere as put-put, phut-phut and pouet-pouet. In this case the heat pipe is as shown in Fig. 6 and is quite capable of propelling a small boat. The pipe is filled with water and the closed end is heated with a flame. High-pressure steam formed in the heated portion pushes a plug of liquid out of the pipe. As the plug moves out, the hot steam is exposed to the cooled region, where condensation takes place, the pressure decreases and the liquid plug is pushed back into the pipe by the atmospheric air pressure. The propulsion process is essentially silent, and for this reason most toy boat engines have the heated end replaced with a flat box-type structure. As the pressure fluctuates in the box, the flat side buckles back and forth, resulting in a characteristic 'click-clicking' or 'putt-putt' sound.

If the liquid–vapour interface of the plug in the pipe is idealised as a portion of a sphere, as shown in Fig. 7, the pressure difference across the plug that causes the plug to move in a horizontal pipe is given by:

$$P_T - P_L = 2\sigma\left(1/R_T - 1/R_L\right) \quad (2)$$

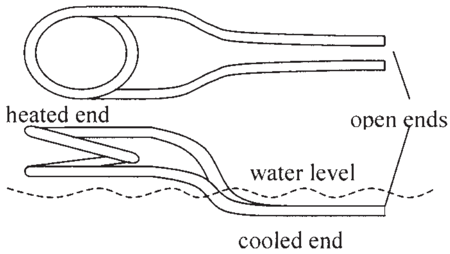

Fig. 6 *Pipe bent to form an open.*

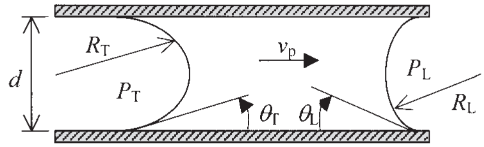

Fig. 7 *Liquid plug moving in a pipe.*

where, if the radius of the pipe is small compared with the capillary length, the radius of curvature R is approximately uniform and equal to $d/(2\cos\theta)$ [5]. The advancing contact angle for water on copper is $\theta_L = 84°$ and $\theta_T = 33°$ for the receding or trailing contact angle [7].

As the liquid plug moves, it actually leaves a thin film on the inside wall of the tube, as shown in Fig. 4a. The thickness of the trailing film was experimentally determined as being in the order of 0.2 mm for water at 27 °C flowing under the influence of gravity in plastic tubes of diameters 3.7 to 6.0 mm and lengths of about 250 mm.

Analysis of phase change in basic texts usually assumes thermodynamic equilibrium at the liquid–vapour interface. The high heat transfer rate associated with evaporation and condensation implies that the assumption of thermodynamic equilibrium is invalid. By considering the liquid–vapour interface at the molecular level and making use of the kinetic theory of gasses, mass transfer flux across a vapour–liquid interface may be derived in terms of pressure and temperature as [8]:

$$\dot{m}'' = \frac{2\hat{\sigma}}{(2-\hat{\sigma})}\sqrt{\frac{1}{2\pi R}}\left(\frac{P_v}{\sqrt{T_v}} - \frac{P_\ell}{\sqrt{T_\ell}}\right) \tag{3}$$

where $\hat{\sigma}$ is termed the *accommodation coefficient* and has reported values of between 0.02 and 0.04 for water. The basic problem in using equation (3) is that it is applicable only if we know the surface area across which the evaporation is taking place. Within the heat pipe it is not certain if in fact a uniform film is maintained on the pipe wall. For film boiling, depending on the wall heat flux, one might expect convection, nucleate (film) boiling and nucleate film boiling with significant entrainment. As the film becomes thinner it may be observed in a plastic tube that the film breaks up and forms droplets on the pipe wall. On inclined portions of the wall, a film may also be observed to drain dry as a result of gravity. These detailed phenomena are also discussed in the literature and a *minimum wetting ratio* is defined as being the minimum flow rate at which permanent dry patches are formed on a heated surface [9]. Further complexities of this film boiling phenomenon are also illustrated in the literature [10].

The mass flux may also be accounted for by the rate at which heat is transferred to/from the liquid–vapour interface to allow for the vaporisation/condensation to take place. Such a mass transfer equation would then be related to a heat transfer equation, of the type:

$$\dot{m}'' = \dot{q}''/i_{fg} = U\left(T_{h,c} - T_v\right)/i_{fg} \tag{4}$$

where U is an appropriate heat transfer coefficient. For equilibrium conditions, the rate at which evaporation/condensation can take place is limited to the rate at which heat is transferred to the liquid–vapour interface. The heat transfer coefficient for a 70 °C laminar uniform water film of 50 μm is often considered to be $h_{film} \approx k_\ell/\delta_{film} = 0.643/50 \times 10^{-6} = 12\,860$ W/m^2 °C. For natural convection from a 4.7 mm diameter copper tube at 70 °C surrounded by water at 20 °C, the heat transfer coefficient is in the order of 1200 W/m^2 °C. The overall heat transfer coefficient would then be in the order of $U = (1/12\,860 + 1/1200)^{-1} = 1079$ W/m^2 °C.

Mathematical model

The open oscillatory heat pipe shown in Fig. 6 consists essentially of a long pipe symmetrically bent about its middle. The coiled portion above the water level may be heated (by a spirit lamp for instance) and is called the evaporator. The portions in the cooling water are called the condensers. Between the evaporator and the condensers there are two lengths of pipe, which, and although not insulated, may be called the adiabatic lengths. If one of the symmetrical halves of the bent pipe is straightened out it can be represented as shown in Fig. 8. L_e is the length of the

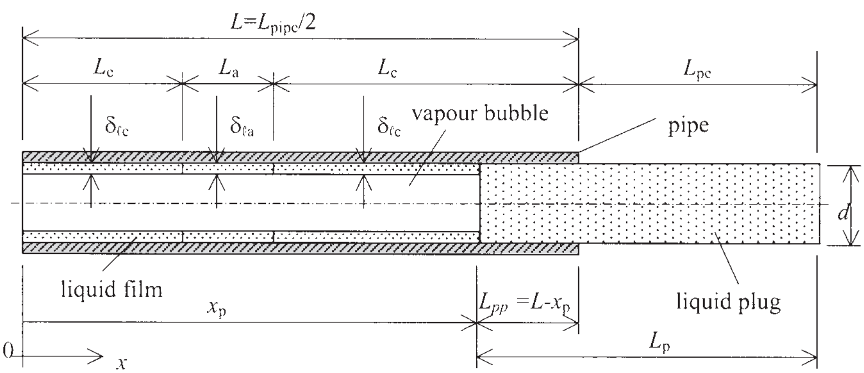

Fig. 8 *Model of the heat pipe showing the important dimensions.*

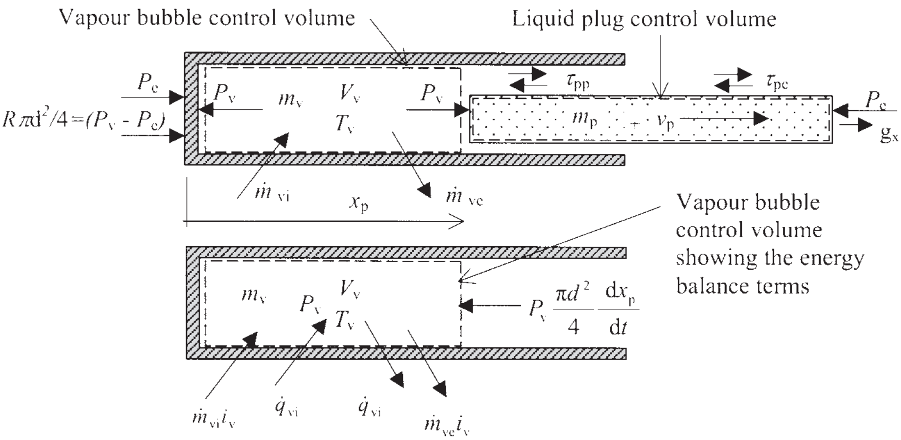

Fig. 9 *One-dimensional vapour bubble and liquid plug control volumes.*

symmetrical half that is heated (the evaporator), L_c is the length that is cooled by the water (the condenser) and L_a is a relatively adiabatic section between the evaporator and the condenser. The open oscillatory heat pipe is modelled by making use of three types of one-dimensional control volumes: a vapour bubble, an axisymmetrical thin layer of liquid film and a liquid plug. The liquid plug is shown projecting out of the pipe and into the cooling water; the portion in the pipe is shown as L_{pp} and the portion in the cooling-water as L_{pe}.

The mathematical model applies the equations of change to the bubble, the film and the plug. Conservation of mass is applied to the vapour bubble, liquid film and the liquid plug, conservation of energy only to the vapour bubble and conservation of momentum only to the liquid plug. In the mathematical formulation, a number of assumptions are made and are explained as appropriate in the following analysis.

Conservation of mass

Consider the vapour bubble control volume shown in Fig. 9. Vapor enters the portion of the bubble exposed to the heated area of the wall, $A_{vi} = \pi d L_{vi}$, of the evaporator according to:

$$\dot{m}_{vi} = \left(U_i / i_{fg}\right) A_{vi} \left(T_h - T_v\right) \tag{5}$$

Vapour leaves the bubble in the portion exposed to the cooled wall, $A_{ve} = \pi d L_{ve}$, of the condensing section as:

$$\dot{m}_{ve} = \left(U_e / i_{fg}\right) A_{ve} \left(T_v - T_c\right) \tag{6}$$

The important assumption made here is that the rate of evaporation/condensation depends on the heat transfer rate to the liquid–vapour interface (see equation 4). No mass transfer in the axial direction across the liquid plug and vapour bubble is assumed to take place.

The continuity equation for the vapour bubble is:

$$\frac{\mathrm{d}m_{\mathrm{v}}}{\mathrm{d}t} = \dot{m}_{\mathrm{vi}} - \dot{m}_{\mathrm{ve}} \tag{7}$$

and for the liquid film in the evaporator:

$$\frac{\mathrm{d}m_{\ell\mathrm{f}}}{\mathrm{d}t} = -\dot{m}_{\ell\mathrm{f}} \tag{8}$$

where $\dot{m}_{\ell\mathrm{f}} = \dot{m}_{\mathrm{vi}}$

The effect of the thin liquid film in the adiabatic and condenser portions of the pipe is neglected. As the liquid plug moves back into the pipe it takes up liquid and as the liquid plug moves out it leaves behind it a new but thin film of liquid. The liquid film in the evaporator cannot be neglected, however, as it plays an important role. Provided there is liquid in the evaporator, evaporation can take place, mass is added to the vapour bubble and a relatively high vapour bubble pressure is maintained. Under these conditions the liquid plug oscillates back and forth in the condenser region. However, as soon as the liquid dries up in the evaporator, the bubble pressure decreases sharply, the liquid plug moves into the evaporator and as it moves out it leaves behind a thin liquid film. In this way the evaporator is charged with liquid.

The mass of the liquid plug is given in terms of the bubble length and geometry as:

$$m_{\mathrm{p}} = \rho_{\ell}\pi d^2 L_{\mathrm{p}}/4 = \rho_{\ell}\pi d^2 \left(L/2 - x_{\mathrm{p}} + L_{pe}\right)/4 \tag{9}$$

The extra length of the plug that extends into the rest of the liquid is an attempt to take into account that, as the liquid leaves the open end of the pipe, it displaces cooling water and hence its actual length is longer than its length within the pipe.

Conservation of energy

Ignoring potential and kinetic energy, the conservation of energy requires that the change in internal energy of the vapour bubble depends on an enthalpy difference, the convective heat transfer rate and the work done:

$$\frac{\mathrm{d}U_{\mathrm{v}}}{\mathrm{d}t} = \dot{m}_{\mathrm{vi}}i_{\mathrm{vi}} - \dot{m}_{\mathrm{ve}}i_{\mathrm{ve}} + \dot{q}_{\mathrm{vi}} - \dot{q}_{\mathrm{ve}} - P_{\mathrm{v}}\frac{\pi d^2}{4}\frac{\mathrm{d}x_{\mathrm{p}}}{\mathrm{d}t} \tag{10}$$

The change in internal energy may be given in terms of the specific volume, as $\mathrm{d}U_{\mathrm{v}}/\mathrm{d}t = m_{\mathrm{v}}c_{\mathrm{vv}}\mathrm{d}T_{\mathrm{v}}/\mathrm{d}t$, and the enthalpy as $i_{\mathrm{v}} \approx 2500 + c_{\mathrm{p}}T_{\mathrm{v}}$. The heat transfer due to convection from the heated wall to the vapour bubble may be given as $\dot{q}_{\mathrm{vi}} = h_{\mathrm{vi}}A_{\mathrm{vi}}(T_{\mathrm{h}} - T_{\mathrm{v}})$ and $\dot{q}_{\mathrm{ve}} = h_{\mathrm{ve}}A_{\mathrm{ve}}(T_{\mathrm{v}} - T_{\mathrm{c}})$. The vapour may be assumed to be an ideal gas with equation of state $PV = m\mathrm{R}(T + 273.15)$. The convection heat transfer coefficient can be estimated using basic heat transfer theory [8] to be about $10\,\mathrm{W/m^2\,°C}$. This is some two orders of magnitude less than the boiling/condensation overall heat transfer coefficient value of about $1000\,\mathrm{W/m^2\,°C}$ as determined by equation (4).

Conservation of momentum

Referring to the liquid plug control volume in Fig. 9, the equation of motion for the liquid plug is given as:

$$m_p \frac{dv_v}{dt} = (P_v - P_e)\frac{\pi d^2}{4} - \pi d L_{pp}\tau_{pp} - \pi d L_{pe}\tau_{pe} - \pi d\sigma\left(\cos\theta_{\ell vT,L} - \cos\theta_{\ell\ell L,T}\right) + g_x \quad (11)$$

Note that the surface tension terms are not shown in Fig. 9. The surface tension terms take into account that the leading and trailing contact angles a moving liquid plug, $\theta_{\ell v}$, may not be the same. The term taking the end of the liquid plug in contact with the cooling liquid is not really applicable and hence is taken as $\theta_{\ell\ell} = 90°$. The shear stress between the liquid plug and the pipe wall is assumed to be:

$$\tau_{pp} = C_{fpp}\,\rho_\ell v_p^2/2 \quad (12)$$

and similarly for the portion of the plug in contact with the surrounding liquid as:

$$\tau_{pe} = C_{fpe}\,\rho_\ell v_p^2/2 \quad (13)$$

The coefficient of friction may be approximated by:

$$C_f = 0.078Re^{-0.25} \text{ for } Re > 1180 \text{ or } C_f = 16/Re \text{ if } Re < 1180 \quad (14)$$

where $Re = \rho_\ell v_p d/\mu_\ell$.

Numerical solution

An explicit finite difference scheme is used to solve the equations of change; the basic procedure follows.

Depending on the position of the plug relative to the closed end (symmetrical centre of the vapour bubble), x_p, the length of the vapour bubble exposed to the heated L_{vi} and the cooled L_{ve} portions of the pipe, and the length of the plug in contact with the pipe wall, $L_{pp} = L - x_p$ and the environment, L_{pe}, are determined. The volume of the vapour, $V_v = x_p\pi d^2/4$ and the mass of the plug, $m_p = \rho_\ell\pi(L_{pp} + L_{pe})d^2/4$, are determined.

The liquid film in the evaporator is further divided into a number of smaller liquid film control volumes and the mass of liquid in each of the control volumes is accounted for. Liquid is lost by evaporation but gained by the trail of thickness $\delta_{\ell f}$ left by the liquid plug as it moves out of the evaporator.

Assuming appropriate values (see comment relating to equation 4) for the overall heat transfer coefficients between the heated pipe wall and the vapour, U_i, and the cooling environment, U_e, the vapour mass flow rates into and exiting the control volume are given, respectively, by:

$$\dot{m}_{vi} = U_i\pi d L_{vi}\,(T_h - T_v)/i_{fg} \quad (15)$$

$$\dot{m}_{ve} = U_e\pi d L_{ve}\,(T_h - T_c)/i_{fg} \quad (16)$$

Depending on the Reynolds number, the coefficient of friction between the liquid plug and the pipe wall is determined as $C_{fpp} = 0.078Re^{-0.25}$ if $Re = \rho_\ell v_p d/\mu_\ell > 1180$, otherwise $C_{fpp} = 16/Re$. It was assumed that the coefficients of friction between the length of the liquid plug and the surrounding liquid are the same, $C_{fpe} = C_{fpp}$. The shear stresses acting between the plug and the pipe and the plug and the surrounding are determined from $\tau_{pp} = C_{fp}\rho_\ell v_p^2/2$ and $\tau_{pe} = C_{fe}\rho_\ell v_P^2/2$. The sum of the forces acting on the plug is given by:

$$\sum F_p = (P_v - P_e)\frac{\pi d^2}{4} - \pi d L_{pp}\tau_{pp} - \pi d L_{pe}\tau_{pe} - \pi d\sigma\left(\cos\theta_{\ell vT,L} - \cos\theta_{\ell\ell L,T}\right) + g_x \quad (17)$$

The *new* values at time step $t + \Delta t$ are determined from the known *old* values at time t by

$$m_v^{new} = m_v + (\dot{m}_{vi} - \dot{m}_{ve})\Delta t \quad (18)$$

$$m_{\ell f}^{new} = m_{\ell f} - \dot{m}_{vi}\Delta t \quad (19)$$

$$T_v^{new} = T_v + \frac{(\dot{m}_{vi} - \dot{m}_{ve})\left(2500 + c_{pv}T_v\right) + \dot{q}_{vi} - \dot{q}_{ve} - \left(P_v \pi d^2 \Delta x_p/4\right)}{m_v c_{vv}}\Delta t \quad (20)$$

$$P_v^{new} = \frac{m_v R_v (T_v + 273.15)}{V_v} \quad (21)$$

$$v_p^{new} = v_p + \frac{\sum F_p}{m_p}\Delta t \quad (22)$$

$$x_p^{new} = x_p + v_p\Delta t \quad (23)$$

$$\Delta x_p^{new} = x_p^{new} - x_p \quad (24)$$

The old values are replaced by the new values and the procedure repeated for the next time step, and so on.

Theoretical results

A computer program was used to solve the above sequence of equations. For the basic values given in Table 1, a number of interesting analyses may be undertaken.

Fig. 10 shows a graph of the liquid plug position, pressure of the bubble and the amount of water in the liquid film on the wall in the heated section as a function of time. It is seen that the motion of the liquid plug is periodic at about 8 Hz and oscillates back and forth in the adiabatic and cooled lengths. As the liquid plug oscillates the liquid film in the heated section evaporates and its mass decreases from a maximum of about 10 mg to zero. Each time that the water is completely evaporated the vapour bubble pressure decreases and the liquid plug is forced back into the heated section at intervals of about 1 to 1.3 seconds by the external environment pressure. In this way water is supplied periodically to the heated section by the trailing end of the liquid plug as it moves out of the heated section.

Fig. 11 gives the thrust as a function of time. It is seen that the amplitude of the thrust is a maximum of about 0.8 N when the water is replenished in the heated section and the amplitude then decreases evenly at 8 Hz until the water dries out and

TABLE 1 *Base-case values for the open oscillatory heat pipe calculations*

Variable	Value
L_e	0.15 m
L_a	0.05 m
L_c	0.3 m
d	0.00334 m
L_{pe}	$15d$
A	$\pi d^2/4$
T_h	200 °C
$T_c = T_e$	20 °C
U_i	1000 W/m^2 °C
U_a	0
U_e	600 W/m^2 °C
h_{vi}	0 W/m^2 °C
h_{va}	0 W/m^2 °C
h_{ve}	0 W/m^2 °C
R_v	461 J/kgK
c_{pv}	1900 J/kg °C
c_{vv}	$c_{pv} - R_v$
i_{fg}	2.34 MJ/kg
μ	$\mu_{@(Th+Tc)/2}$ kg/ms
ρ_p	1000 kg/m^3
$\delta_{\ell f}$	0.000040 m
$\theta_{\ell vT}$	60 °
$\theta_{\ell vL}$	80 °
Initial conditions	
x_0	0.15 m
v_0	0 m/s
P_{v0}	100 000 Pa
T_{v0}	20 °C
and hence	
V_{v0}	$x_0 A$
m_{v0}	$(P_{v0}V_{v0})/(R_v(T_{v0} + 273.15))$

is subsequently replenished in cycles of about 1–1.5 seconds. The average amplitude of the thrust curve is about 0.5 N. Note that this thrust is reckoned for only half of the symmetrically shaped heat pipe. As such, these amplitude values, when doubled, compare very favourably in magnitude, period and frequency with the experimentally determined thrust curves given in Fig. 19 (see below). Fig. 12 also shows the thrust as a function of time but on a larger time-scale. On the same graph the average thrust of each cycle is indicated as a solid dot and the average thrust for the 3–4 s time period is indicated as a straight line of about 0.023 N. This average value is only some 6% of the amplitude of the positive thrust pulses of 0.4 N. This small percentage is also indicative of a relatively small net amount of work done during a cycle. In Fig. 13 it is seen that although the work done by the vapour and

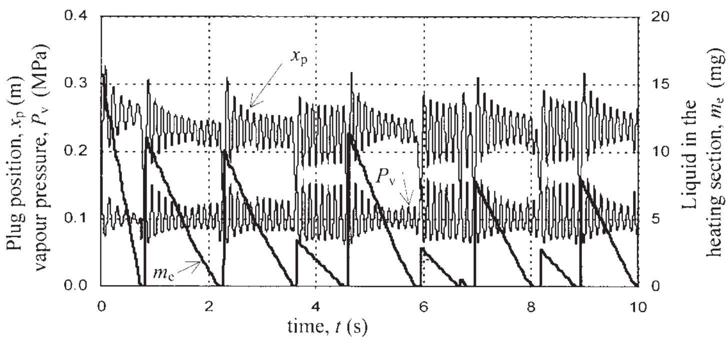

Fig. 10 *Theoretically determined plug positions, vapour pressure and amount of liquid in the heating section as a function of time.*

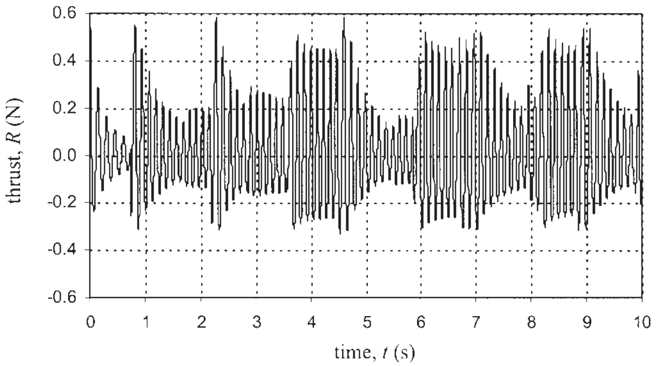

Fig. 11 *Theoretically determined thrust as a function of time.*

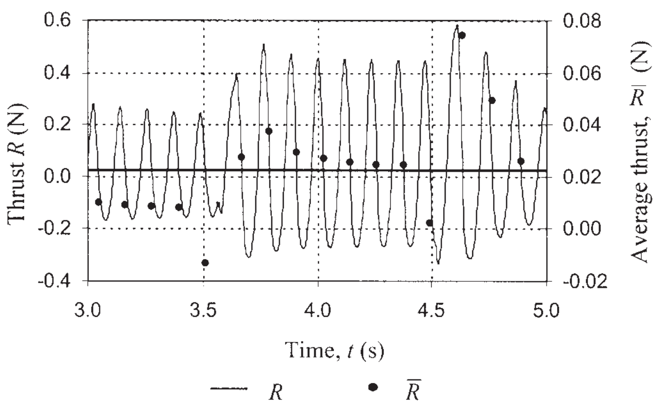

Fig. 12 *Theoretically determined thrust and average thrust as a function of time.*

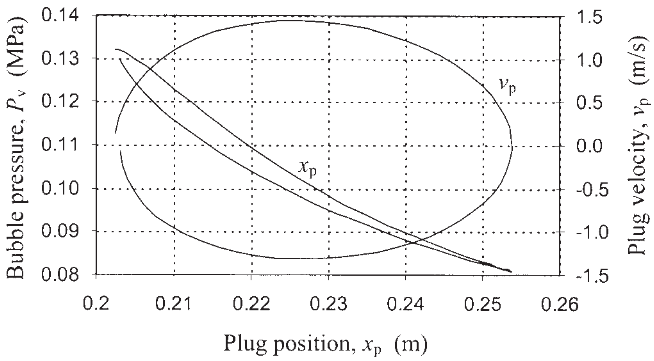

Fig. 13 *Theoretically determined bubble pressure and plug velocity as a function of the plug position.*

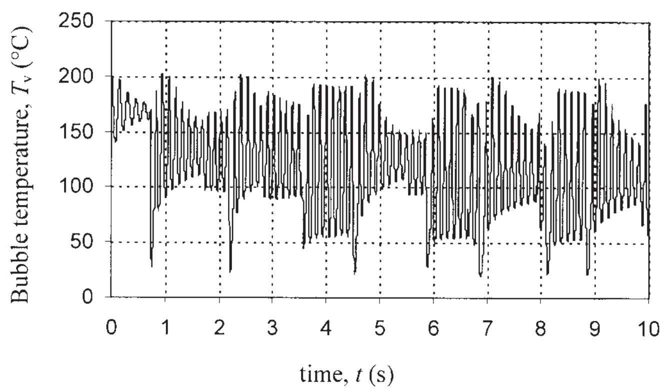

Fig. 14 *Theoretically determined temperature of the vapour bubble as a function of time.*

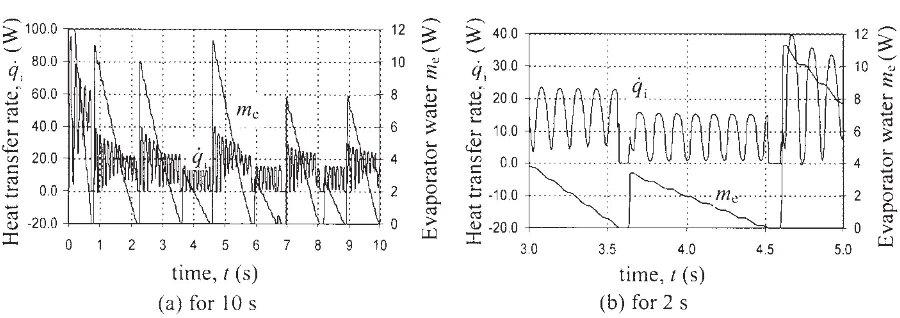

Fig. 15 *Heat transferred into the evaporator as a function of time, (a) for a period of 10 s and (b) for an expanded scale of the period 3 to 5 s shown in (a).*

the work done on it are both relatively large, the net work done during the cycle is small.

The temperature of the vapor is shown as a function of time in Fig. 14. It is seen that the temperature varies from about 100 °C to 200 °C as the liquid plug moves back and forth. However, each time the water is replenished in the heated section, the temperature decreases to a minimum of about 25 °C.

Fig. 15 shows the heat transfer rate into the heated region. It is seen that, at a time of about 4 s, the amplitude is about 15 W. The heat input during the cycle time of about 0.125 s is thus about 1 J. The net work done on the liquid plug during the cycle shown in Fig. 13 is about 0.014×10^{-6} J. The thermal efficiency (defined as the net work out divided by the gross heat input) under these conditions is thus very small, about 1.4×10^{-6}%. It is thus possible to show that although as an engine the open oscillatory heat pipe is extremely inefficient, as a heat transfer device it is relatively efficient.

Varying the heat transfer rate, the amount of liquid deposited by the trailing liquid plug, the length of liquid plug in the surrounding liquid (L_{pe}), and the temperature of the heated section all influence the frequency of water being replenished in the heated section. However, for the same pipe length, although the magnitude of the amplitude the oscillations differs, the frequency of the oscillations is not significantly altered. It was found that the frequency of the oscillations is essentially only a function of the length of the pipe, as follows: pipe lengths of 0.8L, 0.9L, 1.0L and 1.2L give oscillation frequencies of, respectively, 10, 9, 8 and 7 Hz.

The open oscillatory heat pipe, in common with non-linear systems, exhibits interesting chaotic behavioural characteristics [11]. Fig. 16 gives a phase diagram of the plug velocity as a function of its displacement. In Fig. 16b only the first 1.2 s of the motion is plotted to show how the plug position tends to seemingly be *attracted* towards a position of about 0.26 m. Very small changes in base values often resulted in dramatic changes in the stability of the system. Fig. 17 shows that for hot temperatures, T_h, of less than 123 °C and above 228 °C the oscillatory behaviour eventually dies out. For temperatures varying between these two values the oscillations do not die out and, although there are similar characteristics, distinct differences are seen in the graphs for different temperatures.

Experimental set-up and results

A piece of copper tube (3.34 mm inside diameter, 0.4 mm wall thickness and 1.0 m long) was symmetrically fashioned as shown in Fig. 6. It was mounted to a float and tested in a water bath, as shown in Fig. 18. The float was hinged to a calibrated strain-gauge-type thrust sensor with a relatively high response time. A similar type of sensor was used to sense the pulse rate of the water as it was expelled from the open ends of the tube.

A typical thrust curve as a function of time but for different time scales is given in Fig. 19. The frequency of the thrust pulses may be determined from Fig. 19c as 9.8 Hz. Experiments using two separate pulse sensors indicated that the pulses from the open ends were in phase with each other. This indicated that the heat pipe could

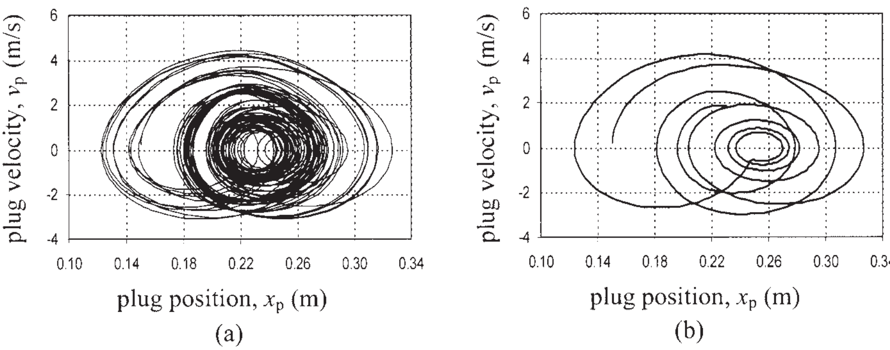

Fig. 16 *Plug velocity as a function plug position for different time periods.*

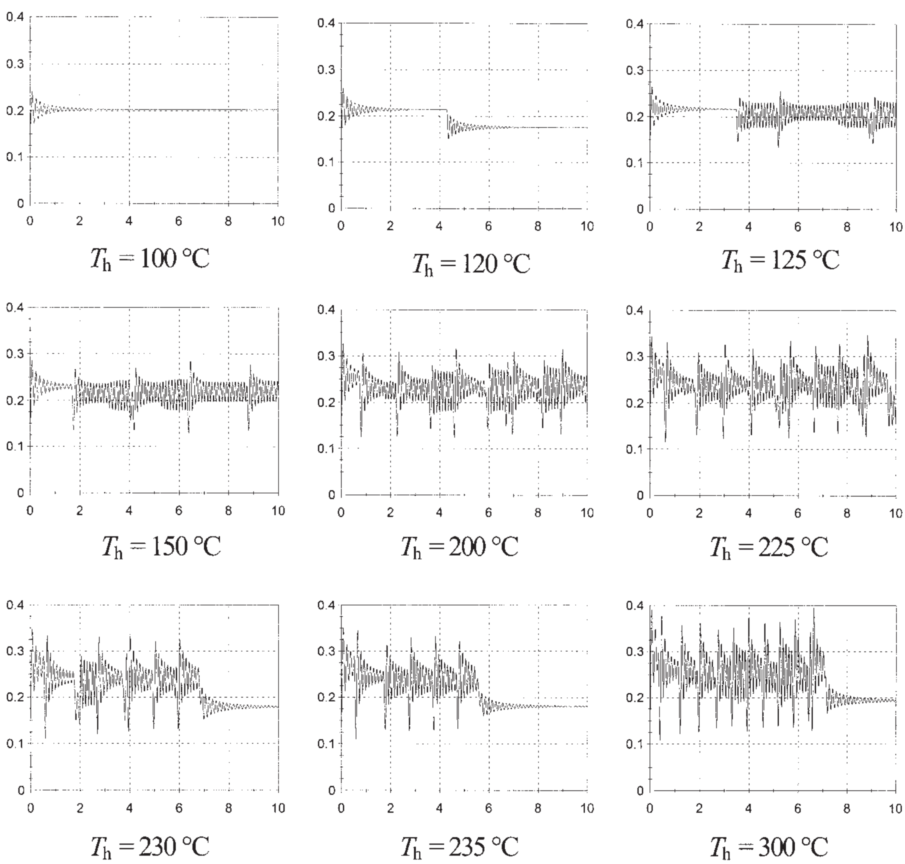

Fig. 17 *Position of the plug as a function of time for different hot temperatures, T_h.*

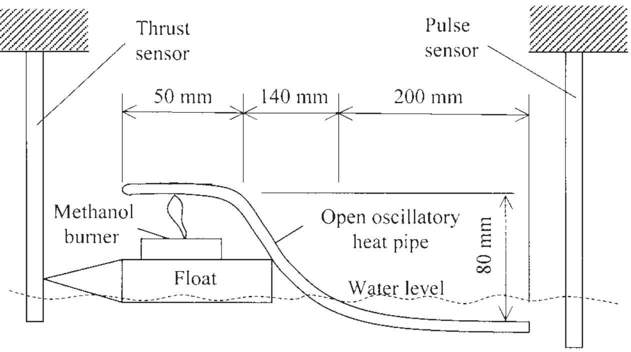

Fig. 18 *Experimental set-up.*

be assumed to operate symmetrically about the centre or the pipe length. The boat was disconnected from its securing support and a force of typically 0.0027 N was needed to restrain it from moving forward.

Discussion and conclusions

The theoretically determined results and in particular the theoretical thrust curve shown in Fig. 11 corresponds remarkably well with the experimentally determined thrust curve of Fig. 19. From this we conclude that the mathematical model presented above reflects reasonably accurately the complex non-linear behaviour of an open oscillatory heat pipe.

The work done by the open oscillatory heat pipe is very small compared with the heat supplied and as a result it has a practically zero thermal efficiency. From this it is concluded that practical applications for open oscillatory heat pipe technology be sought in heat transfer applications rather than as a prime mover or working engine.

Due to the relatively complex nature of the thermo-fluid processes taking place in the open oscillatory heat pipe, a number of issues have, as yet, not been satisfactorily accounted for. These include the mass transfer rates, shape of the liquid free surface, and heat transfer by convection. Attempts to include heat transfer by convection from the heated wall of liquid film were not really successful. In Fig. 18 it can be seen that the heated end is above the open ends; the effect of gravity on this height was not taken into account in the computer program of the model. Mass transfer rates for evaporation are given by equations (3) and (4). Equation (3), however, could not be satisfactorily used in the modelling process. Equation (3) gave mass fluxes of some 40 000 times larger than would normally be expected. It is doubtful whether a uniform film exists in the heating section and hence film theory is possibly inappropriate in the heating section. Ways to determine appropriate heat transfer correlations are thus deemed necessary.

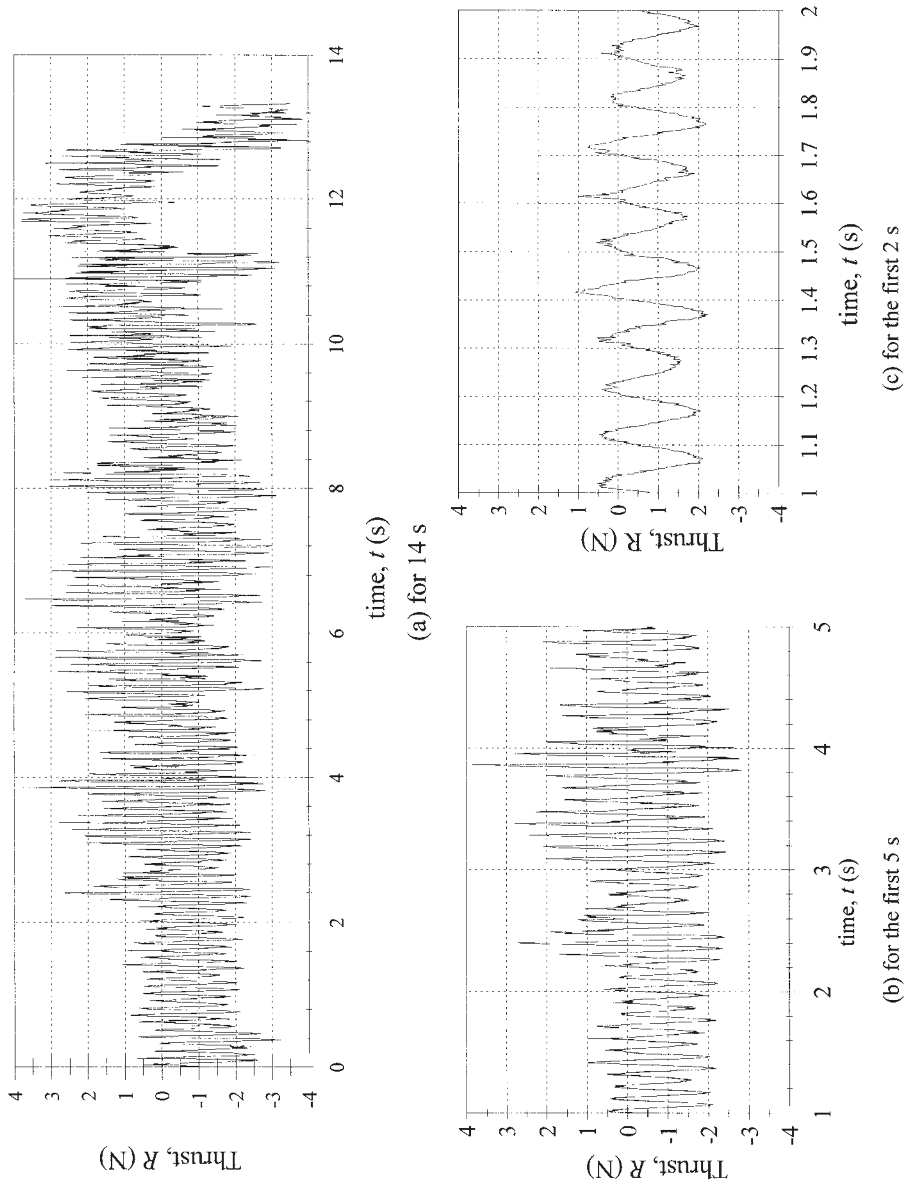

Fig. 19 *An experimentally measured thrust–time curve is shown in (a). Shorter time periods but on larger scales are shown in (b) and (c).*

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