Heat Transfer Theory and Heat Pipe Development for Palm Oil Industry Application

Andrew Yap Kian Chung*

ABSTRACT

Heat could be transferred in three ways which are conduction, convection and radiation. The recent development to produce a faster heat transfer material was the heat pipe that combines the principles of both thermal conductivity and phase transition to efficiently manage the transfer of heat between two solid interfaces. The objective of this research work is to introduce the heat pipe phenomena using existing heat transfer theory and discuss the application in palm oil industry which may enable more biomass saving.

INTRODUCTION

Antoine Lavoisier (1743 – 1794) proposed the caloric theory in 1789 suggested that heat is the manifestation of motion at the molecular level. Benjamin Thompson (1753 – 1814) showed that heat can be generated and James P Joule (1818 – 1889) proved the sceptics of caloric theory in 1843 and put the theory to rest (Yunus and Michael, 1994).

Heat transfer process complies fundamental thermodynamic laws and the rate of heat transfer that may be constant or variable. Practically most cases are concerned with steady state heat transfer but there are cases of transient heat transfer. In general, there are three ways heat may be transferred which are conduction, convection and radiation. The heat transfer being from high temperature to lower temperature medium.

CONDUCTION

Conduction is the transfer of heat in physical contact. Fourier’s Law states that the rate of heat flow, $F$ through a single homogeneous solid is directly proportional to the area, $A$ of the section at right angle to the direction of heat flow and to the change of temperature with respect to the length of the path of the heat flow, $\frac{dt}{dx}$.

\[
F \propto A \frac{dt}{dx} \rightarrow Q = -\lambda A \frac{dt}{dx} \quad (1)
\]

where $\lambda$ is the thermal conductivity of the respective material. The value of thermal conductivity for most solids is approximately constant over a wide range of temperature. Thus, integrating equation above yield.
\[ F = \alpha \Delta t \]  

Newton's Law of Cooling states that the heat transfer from a solid surface at a temperature \( t_2 \) to a fluid of temperature \( t_1 \) is given by:

\[ F = \alpha A (t_2 - t_1) \]  

where \( \alpha \) is the heat transfer coefficient.

Consider a composite wall with \( n \) layers of material with thickness \( x_1, x_2, \ldots, x_n \) and thermal conductivity \( \lambda_1, \lambda_2, \ldots, \lambda_n \). Fluid A at temperature \( t_2 \) with heat transfer coefficient \( \alpha_a \) on one side of the composite wall and fluid B at temperature \( t_b \) with heat transfer coefficient \( \alpha_b \) on the other side of the composite wall. In steady state and heat transfer by conduction only, then the specific heat transfer rate, \( f \) at respective layer is shown in the equation below.

From fluid A to wall layer 1:

\[ f = \alpha_a (t_a - t_1) \]

From layer to layer within the wall:

\[ f = \frac{\lambda_y}{x_y} (t_{y+1} - t_y), \quad y = 1, 2, \ldots, (n - 1) \]

From wall layer \( n \) to fluid B:

\[ f = \frac{\lambda_n}{x_n} (t_n - t_b) = \alpha_n (t_n - t_b) \]

Combine all the equations above yield

\[ (t_1 - t_b) + (t_1 - t_2) + \ldots + (t_n - t_b) \]

\[ = \frac{1}{\sum_{y=1}^{n} \frac{x_y}{\lambda_y}} + \frac{1}{\sum_{y=1}^{n} \frac{x_y}{\lambda_y}} \]

\[ (t_2 - t_b) = \frac{1}{\sum_{y=1}^{n} \frac{x_y}{\lambda_y}} + \frac{1}{\sum_{y=1}^{n} \frac{x_y}{\lambda_y}} \]

\[ \text{CONVECTION} \]

Convection is the mode of energy transfer between a solid surface and the adjacent liquid or gas which is in motion involving effects combination of conduction and fluid motion. In order to apply Newton's Law of Cooling, value for the heat transfer coefficient need to be determined. Forced convection is concerned with the heat transfer between a solid surface and a forced moving fluid by external means such as fan, pump, etc. The thickness of the fluid film on the surface in term of fluid properties and fluid velocity is governed by the Reynolds number. Exact mathematical solution to such problem is difficult to obtain but approximate solution can be obtained by making suitable assumptions. Reynolds postulation expresses the heat transfer in term of the friction resistance to the flow.

\[ \text{Force per unit area:} \quad \sigma = \rho v \]

\[ \text{Heat transfer per unit area:} \quad f = mc \Delta t \rightarrow f = \frac{\rho c \Delta t}{v} \]

where \( c \) is the specific heat capacity of the fluid, \( m \) is the fluid mass flow rate, \( v \) is the bulk velocity of the fluid, \( \sigma \) is the shear stress in the fluid at the wall and \( \Delta t \) is the temperature difference between the surface and the bulk of the fluid. Simple Reynolds analogy can be applied for fluids with Prandtl, \( Pr \) number defined below approximately unity. For most gases, dry vapour and superheated vapour, \( Pr \) lies in between 0.65 to 1.2.

\[ Pr = \frac{\sigma \rho \eta}{\lambda} ; \eta \text{ is the fluid dynamic viscosity} \]

Natural convection is concerned with the heat transfer depends on the cubical expansion coefficient, \( \Phi \) due to the differences fluid density, \( \rho \) causing a natural circulation.

\[ \rho_1 = \rho_2 (1 + \Phi \Delta t) \rightarrow (\rho_1 - \rho_2) = \rho_2 \Phi \Delta t \]

The heat transfer depends on the fluid viscosity, fluid thermal conductivity and the velocity of the convection current dependent on the up-thrust determined as:

\[ (\rho_1 - \rho_2)g = \rho_2 \Phi \Delta t \]

where \( g \) is the local gravity acceleration. In many cases of natural convection, the heat
transfer coefficient could be estimated based on the Grashof number, \( Gr \) as defined:

\[
Gr = \frac{\Phi g l^3 \Delta \Theta}{v^2}; \quad l \text{ is characteristic length}
\]  

(10)

Typical values of \( \alpha \) is shown in Table 1.

**TABLE 1. TYPICAL HEAT TRANSFER COEFFICIENT VALUE**

<table>
<thead>
<tr>
<th>Substance</th>
<th>Heat transfer coefficient [W/(m²•K)]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Natural convection of gases</td>
<td>2 - 25</td>
</tr>
<tr>
<td>Natural convection of liquids</td>
<td>50 – 1 000</td>
</tr>
<tr>
<td>Forced convection of gases</td>
<td>25 - 250</td>
</tr>
<tr>
<td>Forced convection of liquids</td>
<td>50 – 20 000</td>
</tr>
<tr>
<td>Convection in boiling and condensation</td>
<td>2 500 – 100 000</td>
</tr>
</tbody>
</table>

**RADIATION**

The agitation of a substance molecules emit thermal radiation consists of electromagnetic waves propagate in straight lines at the speed of light without requiring any medium. The radiated energy per unit area per unit time is known as emissive power, \( E \). When the radiation strike a body, energy can be absorbed, reflected or transmitted according to the fractions of absorptivity, \( \beta \), reflectivity, \( \gamma \), and transmissivity, \( \tau \). Since the amount of radiation transmitted through the most solids and liquids encountered in engineering is negligible, then yield:

\[
\beta + \gamma = 1
\]

(11)

An ideal black body absorbs all the radiation. A grey body is an ideal surface assumed to have a constant emissivity, \( \varepsilon \) over all wavelengths and for all temperature. The wavelength for maximum emissive power, \( \phi_m \) is given by Wien’s Law shown below:

\[
\phi_m = \frac{2900}{t}; \quad t \text{ in K}; \quad \phi_m \text{ in } \mu\text{m}
\]

(12)

The Stefan-Boltzmann law states that the emissive power is proportional to the fourth power of absolute temperature as shown in the equation:

\[
f = E = 5.67 \times 10^{-8} \epsilon t^4 \text{ [Wm}^{-2}] ; \quad t \text{ in K}
\]

(13)

The radiation intensity variation is given by Lambert’s Cosine Law:

\[
I_o = I \cos \theta
\]

(14)

where \( I_o \) is the radiation intensity in any other direction at angle \( \theta \) to the normal and \( I \) is the normal radiation intensity (Eastop and McConkey, 1994).

**HEAT PIPE**

During the steam age, the general principle of heat pipes was commonly classified as two phase thermo-siphons. Thermo-siphons transfer heat by single-phase convection. RS Gaugler of General Motors suggested the modern concept for a capillary driven heat pipe in 1942 and later patented the idea (Gaugler, 1944). However, the technology at that period presented no clear need for such a device and it lay dormant for two decades. George Grover at Los Alamos National Laboratory demonstrated the benefits of employing capillary action in 1963 and subsequently published in the *Journal of Applied Physics* in 1964 (Grover et al., 1964).

RCA was the first corporation to undertake research and development of heat pipes funded by the US government for commercial applications from 1964 to 1966. NASA played a large role in heat pipe development during the late 1960s. Feldman, Eastman and Katzoff discussed applications of heat pipes to areas outside of government concern in 1967 and 1968. The rotational heat pipe concepts with its applications to turbine blade cooling and to cryogenic processes were introduced in 1969. In the meantime, many other applications have firmly established that heat pipes can solve many critical problems in heat transfer and temperature control. Sony was incorporating heat pipes into...
both forced convection and passive finned heat sinks cooling system for some of their electronic products since 1980. During the late 1990s, the number of US heat pipe patent applications increased three-fold due to the increasingly hot microcomputer. Today most development and production of heat pipes have moved from the US to Asia especially China. Modern CPU heat pipes are made from copper and use water as the working fluid.

The basic heat pipe is a closed container made of a high thermal conductivity material such as copper or aluminium which contains a mesh or sintered powder capillary wick structure to exert capillary pressure and a small amount of working fluid such as water, ethanol, acetone, sodium or mercury which is saturated in both the liquid and vapour phase at operating conditions as shown in Figure 1. Air is removed to eliminate diffusion during the bulk transfer of the vapour. Thus, the heat transfer rate is dependent on the vapour condensation to liquid at the cold end (Faghri, 1995).

Heat pipe contains no mechanical moving part and requires no maintenance. The heat transfer efficiency often expressed in terms of equivalent thermal conductivity is better than an equivalent cross-section of solid copper and even has demonstrated a heat flux of more than 230 MWm\(^{-2}\), nearly four times the heat flux at the surface of the sun (Danneskiold, 2000). However, non-condensing gases that diffuse through the wall or due to the working fluid impurities and chemical breakdown may eventually reduce the heat transfer efficiency especially during low vapour pressure at low temperature.

The heat transfer mechanism in heat pipe is a combination of both the thermal conductivity and phase transition principles as shown in Figure 2. When one end of the tube is heated the liquid turns to vapour absorbing the latent heat of vaporisation. The hot vapour flows to the colder end of the tube where it condenses to liquid and gives out the latent heat. The condensed liquid then flows back through the wick to the hot end of the tube. Wick may not be necessary if fluid surface tension can be overcome.

The heat pipe is partial vacuum thus the working fluid will take up latent heat and evaporate below its atmospheric boiling point. The latent heat of vaporisation is usually very large exceeding the sensible heat capacity. Pressure difference drives a rapid mass transfer approximately at the speed of sound in the absence of non-condensing gases to the condensing end where the excess vapour releases latent heat and condenses. The heat pipe will operate when the hot end is just slightly warmer than the working fluid melting point thus enabling considerable quantities of heat to be transported with a small temperature difference from one end to the other. The pressure drop is very small thus the heat pipe thermal cycle is essentially an isothermal process with minimum heat loss due to the Second Law of Thermodynamics (Patrick and Edward, 1979).

Active control of heat flux is carried out by manipulating the liquid volume of the evaporation section. Variable conductance heat pipes employ a large reservoir of inert immiscible gas attached to the condensing section. Varying the gas reservoir pressure changes the volume of gas charged into the condenser which in turn limits the area available for vapour condensation. Thus, a wider range of heat fluxes and temperature gradients can be accommodated with a single design.

HEAT PIPE DESIGN

The development of a practical heat pipe design requires the application of the theory in combination with a variety of considerations including physical, thermal and mechanical constraints; application require-
ments; materials properties; fabrication, processing and testing limitations; as well as reliability and safety as shown in Figure 3.

Liquid wicking capability in a body-force field is a ratio of surface tension forces, $\delta$ working against body forces proportional to the Wicking Height Factor, $H$ as shown in the equation which represents a basis of fluid comparison.

$$\max H \propto \frac{\delta}{\rho g} \quad (15)$$

It decreases with increasing temperature because the surface tension decreases faster than the liquid density. The relative merit
of the vapour phase can be described by the Kinematics Viscosity Ratio, $\omega$ which decreases with increasing temperature.

$$\min \frac{\omega}{\alpha, \frac{n_v \rho_v}{n_l \rho_l}}$$

(16)

Superior fluid properties with high liquid transport factor, high Wicking Height and low Kinematics Viscosity Ratio will require a smaller heat pipe diameter which will compensate for the pressure.

The homogeneous wire mesh wick design that is able to provide fine capillary sizes to achieve high static height but permeability factor is low. The composite wire mesh wick design avoids the compromise between fine and coarse capillaries but presents a priming reliability problem. Thus, alternate layers of coarse and fine wire mesh are used to disperse any inclusions so that the bubble entrapment problem and composite ratio could be minimised. The axially grooved design offers large open flow channels which are sensitive to gravity. Both the homogeneous and composite wicks are centrally located, spirally wound geometries. This arrangement removes the main wick from the wall thus provides optimum heat transfer at the evaporator and condenser sections.

**Application in Palm Oil Mill**

Boiler is the unit operation where heat pipe could find its application in palm oil mills via the incorporation of economisers. A palm oil mill boiler essentially consists of a fuel feeder, a furnace with water walls, vertically laid to form a wall and connected to top and bottom headers/drums known as steam and mud drums. Through the down comer tubes and riser tubes, the water circulation is established by means of thermosiphonic action. After the raw water treatment, the water is further softened in softener units and pumped into boiler feed water tanks where it is heated by process steam condensate. The water is then de-aerated either by vacuum produced by steam ejector pumps, jet type de-aerating contact heater or other methods. The feed water pump pumps the feed water into the boiler drum, its level being controlled by an automatic drum water level control system, so that the water level in the drum is maintained constant.

The heat of combustion of biogas is transferred by radiation and conduction to the water wall tubes and to the super-heater as well when it is located in that zone.

The advent of the water cooled furnace wall led to the integration of furnace, economiser, boiler, super-heater and air pre-heater into modern steam generators. High pressure steam generator requires more super-heating and reheating surface and less boiler surface. Figure 4 shows a schematic flow diagram of a common steam generator system. Feed water at 232.2°C to 260°C and high pressure enter the economiser and leaves it saturated or as a wet steam (two phase mixture). It then enters the steam drum at midpoint. The
steam drum water flows through insulated down-comer tubes which are situated outside the furnace to another header connected to risers. The water in the tubes receives heat from the combustion gases and boils further. The density differential between the water in the down-comer and that in the water tubes helps circulation. Steam is separated from the bubbling water in the drum and rises to the top and goes to the super-heater from which to the turbine inlet through a main stop valve. The bleed-off steam from the turbine is used to heat the feed water.

Atmospheric air from a forced draft fan is pre-heated by the flue gases after it leaves the economiser. From there it flows into the furnace, where it mixes with the fuel and burns to about 1649°C. The hot combustion gases (about 900°C) impart part of their energy to the water tubes and then the super-heater and economiser and leave the later at about 270°C. From there they pre-heat the incoming atmospheric air in the pre-heater, leaving it at about 149°C which is well above the dew point of SO₂ vapour in the gases to prevent condensation acids that would corrode metal components in its path. An induced draft fan draws the flue gases from the system and sends them up the stack.

The economiser is the heat exchanger that raises the temperature of the water leaving the high pressure feed water heater to the saturation temperature corresponding to the boiler pressure. This is done by gases leaving the last super-heater. These gases at high enough temperatures transfer heat to the super-heater and enter the economiser at 298°C. Part of their energy is used to heat the feed water. As the economiser is introduced before the feed water heating, the water temperature at economiser inlet would be low consequently reducing the outer tube temperature below the dew point of the flue gases. This cause condensation and corrosion because of the presence of SO₂ and SO₃ in the gases. The moisture also will promote the deposition of ash on its tube surface, thus fouling the outer tube.

Figure 4. Schematic flow diagram of a modern steam generator.
surfaces reducing heat transfer. Modern steam generators receive heated feed water and their economisers operate above the dew point of gases thus eliminating external corrosion and fouling. Economisers are generally placed between the last superheater and the air pre-heater. In some cases, a low temperature economiser is placed after the air pre-heater. Such an economiser is called a stack cooler and acts as a low pressure feed water heater except that the heating medium is the flue gas instead of steam bled from the turbine.

THE RANKINE CYCLE

William John M Rankine (1820-1872) devised a vapour-liquid reversible cycle which is most convenient to draw it on both the P-v and t-s diagrams with respect to the saturated liquid and vapour lines of the working fluid usually water. The original ideal Rankine cycle shown in Figure 5 is much simpler compared to the real modern power plant cycle. The curved lines to the left of the critical point are the saturated liquid lines and the regions to the left of these lines are the sub-cooled liquid regions. The curved lines to the right of the critical point are the saturated vapour lines and the regions to the right of these lines are the super-heat regions. The regions under the domes represent the two phase mixture regions.

The cycles start with adiabatic reversible expansion through a condensing turbine. The exhaust vapour usually in the two phase region then undergoes constant temperature and pressure heat rejection in the condenser. The saturated liquid at the condenser pressure is adiabatic reversible compressed by the pump to sub-cooled liquid at the steam generator pressure. The portion bringing the sub-cooled liquid to saturated liquid in the steam generator at constant pressure is called an economiser. The portion heating the saturated liquid to saturated vapour at constant pressure and temperature in the steam generator is called boiler. Super-heater heats the saturated vapour to produce super-heated steam. Based on a unit mass of vapour in the saturated cycle, the analysis yield:

\[
\begin{align*}
\text{Heat added} & \quad q_4 = h_1 - h_4 \\
\text{Turbine work} & \quad w_T = h_1 - h_2 \\
\text{Heat rejected} & \quad q_R = h_2 - h_3 \\
\text{Pump work} & \quad w_P = h_1 - h_3 = V_3(P_4 - P_3) \\
\text{Network} & \quad \dot{w}_n = \frac{\dot{w}_P}{\dot{m}} = \frac{(h_4 - h_2) - (h_2 - h_3)}{(h_4 - h_1)} \\
\text{Thermal efficiency} & \quad \psi = \frac{\dot{w}_n}{\dot{q}_A} 
\end{align*}
\]

External irreversibility is primarily the result of the temperature differences between the primary heat source and the working fluid; condensing working fluid and the heat sink fluid. Internal irreversibility is primarily the result of fluid friction, throttling, mixing in turbines, pump and pressure drops in heat exchangers, pipes, bends, valves, etc. In the thermodynamic analysis of cycle and power plants, the power output and the thermal efficiency which is the ratio of the network to the heat addition are of prime importance as an economic measure of heat rate.

Boiler horsepower, blhp, was originally used to indicate the size of the boiler. One blhp was defined as 0.9290304 m² of boiler heating surface but later was changed to the amount of heat required to evaporate 13.6078 kg hr⁻¹ of water at 37.7778°C to saturated steam at 482.633 kPa which is equal to 9782.1272 W. Later on this was changed to the amount of heat necessary to evaporate 15 6489 kg hr⁻¹ of saturated water to saturated steam at 101.325 kPa which is 9810.6576 W and is now rounded so that the modern value is 9812 W (El-Wakil, 1984).

DISCUSSION AND CONCLUSION

Heat pipes must be tuned to particular cooling conditions. The choice of pipe material, size and coolant all have an effect on the optimal heat pipes operating temperatures. If all of the working fluid in the heat pipe vaporise and the condensation process cease to occur; the heat pipe...
Figure 5. Rankine cycle with t-s and P-V chart for steam.
thermal conductivity is effectively reduced to the heat conduction properties of its solid metal casing alone. An overheated heat pipe generally will continue to conduct heat at around 1/80 of the original conductivity equivalent to copper heat conductivity.

If the working fluid does not undergo phase change below a certain temperature, the thermal conductivity will reduce to that of the solid metal casing. Thus, the desired operational temperature range is one of the key criteria for the working fluid selection. The lowest operational temperature limit should be a few degrees Celsius above the freezing point of the selected working fluid.

Most of the traditional heat pipe diameter is above 3 mm due to the material limitations and the heat pipe thermal conductivity efficiency is proportional to the diameter as shown in the Appendix.

**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Particular</th>
<th>Unit</th>
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<tr>
<td>$A$</td>
<td>Section area</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$E$</td>
<td>Emissive power</td>
<td>W m$^{-2}$</td>
</tr>
<tr>
<td>$F$</td>
<td>Heat transfer rate</td>
<td>W</td>
</tr>
<tr>
<td>$H$</td>
<td>Wicking height factor</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$I$</td>
<td>Radiation intensity</td>
<td>-</td>
</tr>
<tr>
<td>$P$</td>
<td>Pressure</td>
<td>kPa</td>
</tr>
<tr>
<td>$Q$</td>
<td>Heat</td>
<td>kJ</td>
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<tr>
<td>$V$</td>
<td>Specific volume</td>
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<tr>
<td>$W$</td>
<td>Work done</td>
<td>J</td>
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<td>$c$</td>
<td>Specific heat capacity</td>
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<td>$f$</td>
<td>Specific heat transfer rate</td>
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<td>$g$</td>
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<td>Length</td>
<td>m</td>
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<td>$m$</td>
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</tr>
<tr>
<td>$\Phi$</td>
<td>Cubical expansion coefficient</td>
<td>-</td>
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<tr>
<td>$\alpha$</td>
<td>Heat transfer coefficient</td>
<td>Wm$^{-2}$K$^{-1}$</td>
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**REFERENCES**


### Appendix

**TABLE A1. TYPICAL OPERATING CHARACTERISTICS OF HEAT PIPES**

<table>
<thead>
<tr>
<th>Temperature range (°C)</th>
<th>Heat pipe working fluid</th>
<th>Heat pipe vessel material</th>
<th>Measured axial heat flux [kW cm⁻²]</th>
<th>Measured surface heat flux [W cm⁻²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>-200 to -80</td>
<td>Liquid nitrogen</td>
<td>Stainless steel</td>
<td>0.067 @ -163°C</td>
<td>1.01 @ -163°C</td>
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<tr>
<td>-70 to +60</td>
<td>Liquid ammonia</td>
<td>Nickel, aluminium, Stainless steel</td>
<td>0.295</td>
<td>2.95</td>
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<tr>
<td>-45 to +120</td>
<td>Methanol</td>
<td>Copper, nickel, stainless steel</td>
<td>0.45 @ 100°C(x)</td>
<td>75.5 @ 100°C</td>
</tr>
<tr>
<td>+5 to +230</td>
<td>Water</td>
<td>Copper, nickel</td>
<td>0.67 @ 200°C</td>
<td>146 @ 170°C</td>
</tr>
<tr>
<td>+190 to +550</td>
<td>Mercury +0.02%</td>
<td>Stainless steel</td>
<td>25.1 @ 360°C*</td>
<td>181 @ 750°C</td>
</tr>
<tr>
<td>+400 to +800</td>
<td>Magnesium +0.001%</td>
<td>Nickel, stainless steel</td>
<td>5.6 @ 750°C</td>
<td>181 @ 750°C</td>
</tr>
<tr>
<td>+500 to +900</td>
<td>Potassium</td>
<td>Nickel, stainless steel</td>
<td>9.3 @ 850°C</td>
<td>224 @ 760°C</td>
</tr>
<tr>
<td>+900 to +1 500</td>
<td>Sodium</td>
<td>Niobium +1% zirconium</td>
<td>2.0 @ 1250°C</td>
<td>207 @ 1250°C</td>
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<tr>
<td>1 500 + 2 000</td>
<td>Lithium</td>
<td>Tantalum +5% tungsten</td>
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<td>413</td>
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</table>

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**Heat Pipe Spec. Inclination: 0°
Groove Type: No Modification**

**Feature Article**

**Palm Oil Engineering Bulletin No. 99**
Thermal Resistance vs. Heat Pipe Length
(Q = 10W, Radius = 3mm, Horizontal Orientation)

Thermal Resistance vs. Heat Pipe Length
(Q = 10W, Radius = 3mm, Horizontal Orientation)