HEAT PIPE BASED AIR PREHEATERS FOR THERMAL POWER PLANTS

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ABSTRACT

Air Preheaters are used in modern thermal power plants to preheat the secondary/ primary air required for combustion of fuel in the boiler using the energy available in exhaust flue gases. The poor performance of Air Preheaters in the modern power plants is one of the main reasons for higher Unit Heat rate and is responsible for deterioration in boiler efficiency. Heat Pipe heat exchangers are the most promising alternative to the existing tubular and rotary (Ljungstrom) Air Preheaters and have the potential to eliminate many of the problems associated with them. Simulation of Heat Pipe Air Preheater has been carried out using different combinations of wicks and working fluids to arrive at the design of a Heat Pipe based Primary Air Preheater for 500 MW Coal Fired Thermal Power Plant. The different types of wicks studied are wrapped screen mesh, rectangular groove and screen covered rectangular groove. It has been concluded that screen covered rectangular grooved Heat Pipes can be an economical and efficient design for heat transfer in such Air Preheaters. A comparison of various suitable working fluids for the same heat duty and for screen covered rectangular groove wick has been done. Water or Toluene has been used as a working fluid inside the Heat Pipe, for low temperature operating range (30- 200 ^{o}C) & Naphthalene or Dowtherm for high temperature (200-350).

KEYWORDS

Air Preheater, Heat Pipe, Wick structure, working fluid, Thermal power plants

INTRODUCTION

In an Air Preheater of a thermal power plant, flue gases coming from the boiler are used for preheating the secondary/ primary air required for combustion of fuel in the boiler. Apart from the need to preheat air to dry coal and improve combustion efficiency, the benefit from preheating combustion air can be shown to be a 1% improvement in boiler efficiency, for each 22°C rise in the combustion air temperature [1]. By utilizing the Air Preheater the temperature of the combustion product can be reduced from about 370-450^{0}C to about 120-135^{0}C [2]. In an Air Preheater, the minimum flue gas temperature is controlled by avoidance of fouling of Air Preheater and cold end corrosion etc. Rotary Regenerative type Ljungstrom Air Preheaters are normally used in most modern power plants. Leakage from the combustion air side of the exchanger into the flue gas side is the biggest problem with this design. Although seals are provided around the rotor the leakage occurs across the radial seals, in the clearance between the rotor and the metal case and also by entrainment from the basket gas passages as the baskets rotate from the air side into the flue gas side. When new, the air leakage may be as low as 6% (Secondary Air Preheater) to 13% (Primary Air Preheater) of the incoming flue gas flow, depending on Air Preheater size and air-to-gas pressure differentials. As the seals wear and the Air Preheater ages, the leakages often increase up to 12% (low pressure Secondary Air Preheater) to 21% or even 24 % (high pressure Primary Air Preheater) [1]. Shah and Giovannelli [3] have reported a study where Heat Pipe heat exchangers have been rated best for high temperature (250^{0}C – 400^{0}C) heat recovery applications.

The Heat Pipe is a relatively new concept in heat exchangers, which can be used for utility Air Preheaters. The Heat Pipe Air Preheater consists of a bank of Heat Pipes. A single Heat Pipe as shown in Fig. 1 can be divided in to three parts: evaporator, adiabatic section and condenser. The Heat Pipe tubes are partially filled with a heat transfer working fluid. The Heat Pipe tube is sealed under high vacuum to ensure that only gas inside the tube is the working fluid vapor. Passing hot flue gases over the evaporator causes the working fluid to boil and the vapors to flow to the cold end of the tube. Cold air flowing over the condenser in counter flow direction condenses the vapors releasing latent heat that heats the air. Inside a Heat Pipe, boiling and condensing heat transfer mechanisms transfers heat. For these mechanisms, heat transfer can proceed at extremely high rates as compared to conduction and/or convection. Since the Heat Pipes are mounted at a slight angle from horizontal the condensed liquid flows back by gravity to the evaporator end of the pipe to repeat the cycle. Wall grooves or wicks are used inside the Heat Pipe tubes to improve wall wetting and heat transfer. As the heat transfer temperature range in the Air Preheater is approx. 27 ^{o}C to 370 ^{o}C, a single fluid cannot be used in all the Heat Pipes along the flow direction [4]. Water and Toluene, normally, has been used as a working fluid inside the Heat pipe for low temperature range (30°C to 200°C). Dowtherm and Naphthalene can be used for higher temperature range [4].

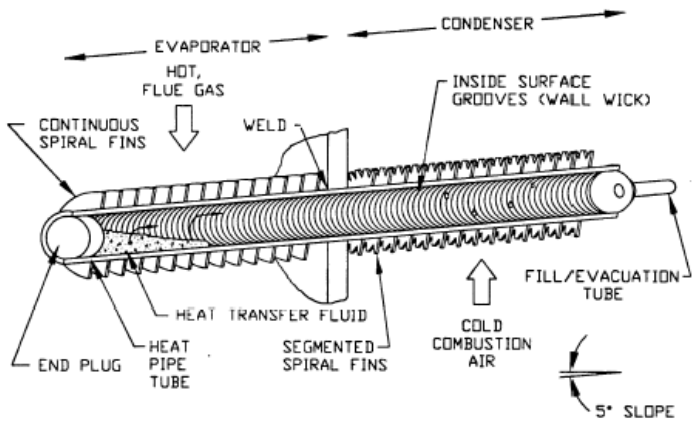

Fig. 1 Heat Pipe Air Preheater [5]

A Heat Pipe Air Preheater consists of two ducts with a common wall. Individual Heat Pipe tubes extend through the common wall across both ducts. Hot flue gases flow through one duct while cold combustion air flows through other duct. The tubes are usually seal welded or gasket fitted at the common wall to prevent air leakage between the flue gas and air sections. The ends of the tubes are free to expand or contract within the duct casing. Providing fins can extend the individual tube surfaces and compact units can be designed.

Heat transfer in a Heat Pipe Air Preheater can be calculated on the basis of thermal resistance circuit and the design is checked for critical heat transport limitations. Simulation of Heat Pipe based Air Preheaters has been carried out for various wick structures viz. wrapped screen mesh, rectangular groove and screen covered rectangular groove.

MODELLING OF A HEAT PIPE AIR PREHEATER

The thermal resistance circuit of a Heat Pipe is shown in Fig. 2. The heat carried by the Heat Pipe has to cross the evaporator wall, pass through the wick structure at both evaporator and condenser section, through the vapor core and finally across the condenser wall. The overall thermal resistance R_T of a Heat Pipe to transfer heat from flue gas to combustion air is given as:

$$R_T = R_{c,a} + R_{s,a} + R_{p,a} + R_{w,a} + R_v + R_{c,g} + R_{s,g} + R_{p,g} + R_{w,g} \tag{1}$$

The overall thermal conductance of the heat exchanger is given by $(UA)_{HP} = \dfrac{1}{R_T}$.

Evaluation method of these resistances is described in Shah & Giovannelli [3]. The overall heat transfer rate of the Heat Pipe Air Preheater can be evaluated as:

$$Q_{HP} = (UA)_{HX}\,(t_g - t_a) \tag{2}$$

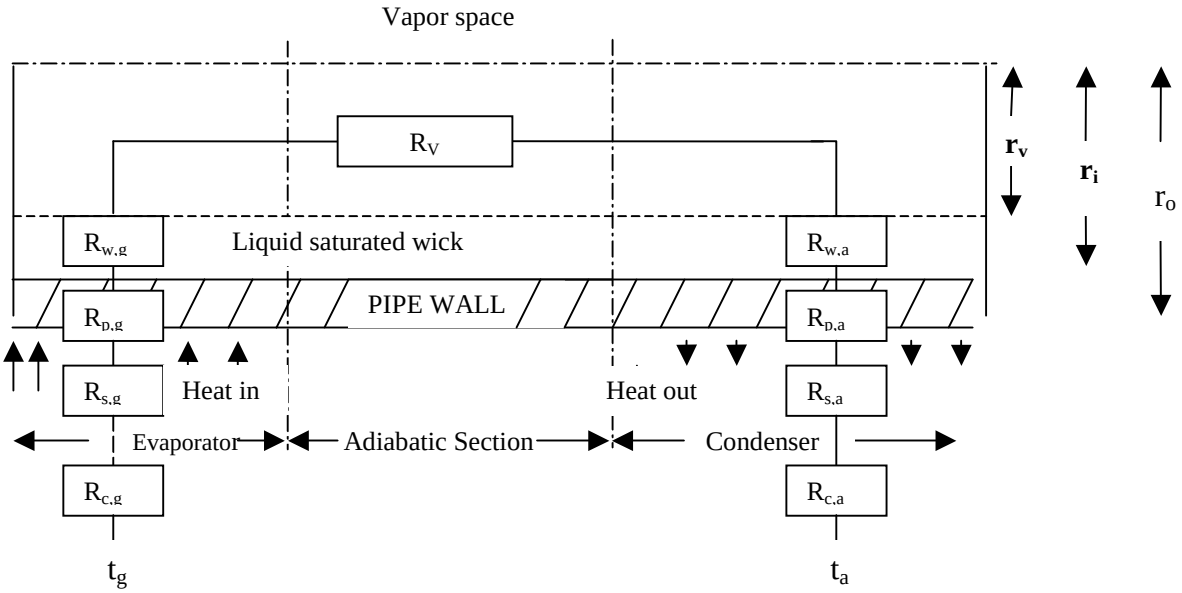

Fig. 2 Thermal Circuit for a Heat Pipe Heat exchanger

Correlations for Heat Transfer and Pressure drop

For plain plate finned tube having staggered array, "Briggs and Young" proposed following correlation based on regression analysis [3].

$$j = 0.134\,\mathrm{Re}_d^{-0.319}\left(\frac{s}{l_f}\right)^{0.2}\left(\frac{s}{\delta F}\right)^{0.1134} \tag{3}$$

Where $s = \frac{1}{N_f} - \delta F$ and j is Colburn factor.

For friction factor f, the "Robinson & Briggs" proposed [3] the following correlation.

$$f = 9.465\,\mathrm{Re}_d^{-0.316}\left(\frac{X_t}{d_o}\right)^{-0.927}\left(\frac{X_t}{X_d}\right)^{0.515}\left(\frac{D_h}{X_l}\right) \tag{4}$$

Valid for $2{,}000 \le \mathrm{Re}_d \le 50{,}0000$.

Heat transport limitations

The maximum heat transfer rate that a Heat Pipe can transfer is limited either by the breakdown of continuous circulation of the working fluid or by limits on the maximum circulation. There are five limits associated with the Heat Pipe. These limits are Viscous limits and Sonic limits due to vapor flow, the Wicking limits due to liquid flow, the Entrainment limits due to interaction of liquid and vapor flows, and the Boiling limit due to nucleate boiling in the evaporator rather than evaporation taking place at the wick vapor interface. The governing equations used for the calculations of the above heat transport are as follows

Wicking limits can be calculated as [3]:

$$q_{w,\max} = \frac{\left[\frac{2\sigma}{r_e} - \Delta P_{g,\parallel} - \Delta P_{g,\perp}\right]}{(F_v + F_l)L_{eff}} \tag{5}$$

The gravitational pressure drop/rise parallel ($\Delta P_{g,\parallel}$) to and perpendicular ($\Delta P_{g,\perp}$) to Heat Pipe to be calculated as follows:

$$\Delta P_{g,\parallel} = \rho_l g L \sin\Phi$$
$$\Delta P_{g,\perp} = \rho_l g d_i \cos\Phi \tag{6}$$

The friction coefficient for liquid and vapor flow F_l , F_v are calculated as:

$$F_l = \frac{\mu_l}{KA_w \rho_l h_{fg}} \tag{7}$$

Here K is wick permeability. Analytically permeability for rectangular groove wick and screen covered rectangular groove can be determined [4] as:

$$K = \frac{D_h^2 \varphi}{2(f\,\mathrm{Re}_{l,h})} \tag{8}$$

D_h is wick hydraulic radius. For rectangular groove $D_h = \frac{4d_g w}{2d_g + w}$

$f\,\mathrm{Re}_{l,h}$, Calculated as per correlation [4]

$$f\,\mathrm{Re}_{l,h} = 24(1 - 1.3553\tau + 1.9467\tau^2 - 1.7012\tau^3 + 0.9564\tau^4 - 0.2537\tau^5) \tag{9}$$

Where $\tau = \frac{w}{d_g}$.

For screen wrapped wick of different sizes experimentally determined values of K are taken from Faghri [4] & Shah & Giovannelli [3].The frictional coefficient of vapor flow F_v depends upon whether the flow is laminar or turbulent and compressible or incompressible. The values of F_v can be calculated using the correlations given in Shah & Giovannelli [3] & Dunn & Reay [6].

Calculating Viscous limit as:

$$q_{v,\max} = \frac{d_v^2 h_{fg} \rho_v P_v A_v}{64 \mu_v L_{eff}} \tag{10}$$

Calculating Sonic limit as:

$$q_{s,\max} = A_v \rho_t h_{fg} \sqrt{\gamma_v R_g T_t} \left[\frac{(1+\gamma_v)}{2}\right]^{(1+\gamma_v)/2(1-\gamma_v)} \tag{11}$$

Where T_t & ρ_t are stagnant temperature and pressure respectively.

Calculating Entrainment limit as:

$$q_{e,\max} = A_v h_{fg} \sqrt{\frac{\sigma \rho_v}{Z}} \tag{12}$$

Where Z is a characteristic dimension depends upon wick type. For wrapped screen mesh Z is wire spacing and for axial groove wick Z=2w.

Calculating Boiling limit as:

$$q_{b,\max} = \frac{2\pi L_{1g} K_e T_{wv,e} N_t \left(\frac{2\sigma}{r_n} - P_c\right)}{h_{fg} \rho_v \ln\left(\frac{d_i}{d_v}\right)} \tag{13}$$

$$\text{Where } P_c = \frac{2\sigma}{r_e} \tag{14}$$

Physical properties of fluid

The heat transfer rate and fluid pumping power are dependent upon the fluid thermo physical properties. Hence it is essential that the fluid properties to be calculated accurately at each section corresponding to the temperature. Fluid properties required for analysis are density, specific heat, viscosity, thermal conductivity and Prandtl number. Correlations [7] has been used to calculate thermal conductivity (K) & viscosity (μ) of flue gas as follows, depending upon the volumetric constituent of flue gas.

$$K = \frac{1}{2}\left(\sum \frac{\alpha_i p_i K_i}{P} + 1/\sum \frac{\alpha_i p_i}{K_i P}\right) \tag{15}$$

Where K_i & p_i are the thermal conductivity & partial pressure of individual constituent of mixture, and α_i are correction factors to account for molecular kinetic theory of gases.

$$\mu = 1/\sum \frac{\beta_i p_i}{\mu_i P}) \tag{16}$$

Where, μ_i is viscosity of individual constituent of mixture, and β_i are correction factors. The curve fit equations of thermo physical properties of constituent of gases and other working fluids taken from Yaws [7].

Simulation model

In the simulation model the flow arrangement considered is counter flow direction. The Air Preheater is sub divided in "n" small section. The flue gas temperature drop in all the section is assumed to be constant and is equal to $(t_{go}-t_{gi})/n$. The calculation start from "n^{th}" part of the Air Preheater where the flue gas inlet temperature is equal to $t_{go}+(t_{go}-t_{gi})/n$ and outlet temperature is $t_{go.}$ Air inlet temperature is known for the nth section. Hence surface area required for transfer of heat in the nth section and air outlet temperature of the section can be calculated using the design methodology [3]. Assuming the Heat Pipe length in the section, the Air Preheater depth and width and number of Heat Pipe can be calculated based on allowable pressure drop consideration of the section. The method is repeated by the process of successive substitution till the first section.

The dimensions (width, height & depth) of all the sub sections of Air Preheater obtained as above may be different, which would cause difficulties in fabrication of such a design. To overcome this, the program has been run by fixing the same suitable frontal dimensions (Length of Heat Pipe and number of Heat Pipe in a row) for all the sections. The flue gas outlet temperature and air inlet temperature of each row is again calculated sequentially. The process is repeated till flue gas outlet temperature and air outlet temperature after Air Preheater is equal to desired value.

RESULTS & DISCUSSION

The developed design code is used to design Heat Pipe Air Preheater. The configuration is selected on the basis of previous work done [5] in this area. The Heat Pipe tubes are 2 inch outer diameter, 0.095 inch tube thickness, the fin tip diameter is 1.75 times the tube diameter and the transverse tube pitch is 1.875 times the tube diameter. Because of the erosive nature of the flue gas containing fly ash, fin density has been kept low at the flue gas side. The selected configuration of a Heat Pipe Air Preheater is shown in Table 1.

SN	Description	Dimensions in mm
1	Heat Pipe tube outer diameter	50.8
2	Heat Pipe tube inner diameter	45.974
3	Fin tip diameter of a circular fin	88.9
4	Transverse tube pitch	95.25
5	Longitudinal tube pitch	82.49
6	Fin thickness air side	1.0
7	Fin thickness gas side	1.5
8	Fin height (both air and gas side)	19.05
9	Fin density (gas side)	125 fin per meter
10	Fin density (air side)	275 fin per meter

Table 1 Heat Pipe Air Preheater design details

The fins are of plain circular type and the Heat Pipes are arranged on staggered triangular pitch. The fin and tube material selected is carbon steel. The flow parameters given in Table 2 are taken from a typical Primary Air Preheater of 500 MW Thermal Power Plant.

S.N.	Parameter	Value
1	Air flow rate (kg/s)	77.5
2	Gas flow rate (kg/s)	94.5
3	Air inlet temperature (^oC)	27
4	Gas inlet temperature (^oC)	370
5	Gas outlet temperature (^oC)	140-135
6	Heat duty (MW)	24
7	Gas inlet pressure (Pa)	-9800.0
8	Air inlet pressure (Pa)	98000.0
9	Allowable air side pressure drop (Pa)	960
10	Allowable gas side pressure drop (Pa)	780

Table 2 Heat Pipe Air Preheater flow parameter

The design study of Heat Pipe heat exchanger has been carried out, by selecting three types of wicks wrapped screen mesh, rectangular groove and screen covered rectangular groove. Based on the standard criteria for selection of working fluids (Faghri [4], Dunn & Reay [6] and Peterson [9]), four working fluids can be used in the given temperature range viz. Water, Toluene, Dowtherm and Naphthalene. But the temperature range is too large and thus for a given design two working fluids have been proposed in different section of Air Preheater. For low temperature section (85-230 ^oC) Water & Toluene has been used and for high temperature section (235-350 ^oC) Dowtherm & Naphthalene has been used.

Comparison of different wick structures

A comparison of the feasible design obtained for the three selected wicks at the same heat duty is shown in Table 3. Due to the poor wicking limit of the wrapped screen mesh Heat Pipes at lower inclination, a safe design has been achieved at vertical inclination. However, in case of other two wick types used the required wicking action can be achieved with lower inclination. With wrapped screen mesh the conductivity of liquid saturated wick for water is from 1.2 to 1.35 W/m ^oC and for Toluene, Naphthalene and Dowtherm is 0.12 to 0.3 W/m ^oC. Due to the poor conductivity the overall heat transfer coefficient is less and the surface area required is very high. With the rectangular groove wick, the thermal conductivity of liquid saturated wick is quite higher, than the wrapped screen. Hence, overall heat transfer coefficients are higher. But, in this case the restriction comes from entrainment limit and boiling limit. As the depth of the wick decreases the boiling limits increases and as width of the groove decreases, the entrainment limit increases. The safe design achieved with very small width (0.8 mm) of the groove. Which may increases the cost of manufacturing.

SN	Parameters	Wrapped screen mesh	Rectangular groove	Screen covered rectangular wick
1	Total length of tube required for Heat Pipe in the Air Preheater. (m)	81,004	36,142	27,443
2	Length of Heat Pipe (m)	7.5	4.3	9.5
3	Air side pressure drop (Pa)	698	874	893
4	Gas side pressure drop (Pa)	598	716	726
5	Working fluid	Water & Dowtherm	Water & Dowtherm	Water & Dowtherm
6	Inclination of Heat Pipe with horizontal	90°	6°	5°

Table 3 Comparison of Heat Pipes with different wick

Further the thermal conductivity of liquid saturated wick in evaporator section is quite less than in condenser section of Rectangular groove wick. The entrainment limit and thermal conductivity of liquid saturated wick can be improved by the use of screen covered rectangular wick. Thus, the sizes

of groove can also be made higher than that of rectangular groove wick.

It is concluded from Table 3 that screen covered rectangular groove take only 33.88 % of Heat Pipe required by wrapped screen mesh and 75.93 % of Heat Pipe required by Rectangular groove Heat Pipe. Hence, it can be a best feasible option for a Heat Pipe Air Preheater.

Comparison of Working Fluids

A comparison between the working fluid for the same heat duty and for screen covered rectangular wick has been carried out. The comparison has been done on the basis of Wicking, Entrainment and Boiling heat transfer limitations because the other two (Sonic and Viscous limits) are of the order of 100 times higher than the operating range.

The variation of Heat Pipe limitations with working fluid temperature for Toluene is shown in Fig. 3. The Boiling limit of Toluene at high temperature is low and there is very low margin with required heat to be transferred. The Entrainment and Wicking limits are low at lower temperature of working fluid.

The variation of Heat Pipe limitations with working fluid temperature for Water is shown in Fig. 4. Here, all the heat transfer limitations are well above the required heat transfer rate. Hence, Water can be a better alternative than Toluene.

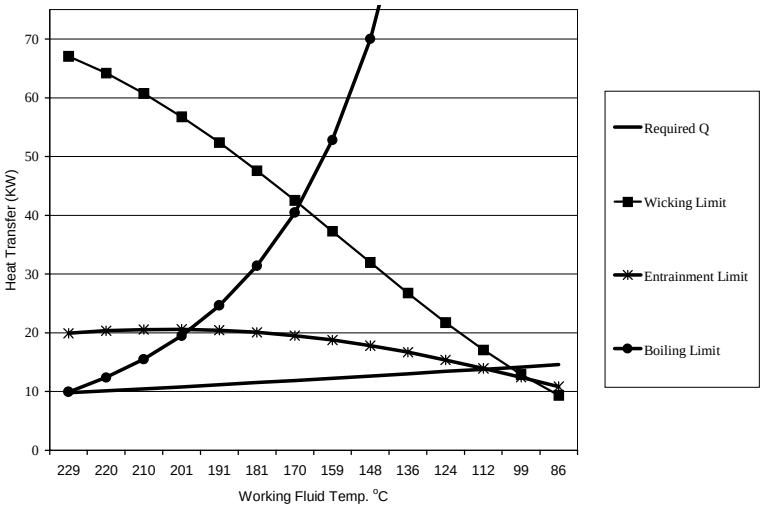

Fig. 3 Heat Transfer limitations for Toluene

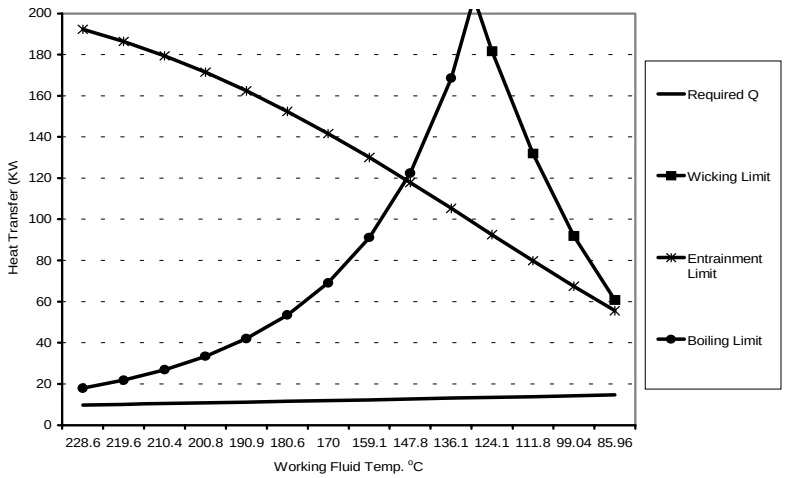

Fig. 4 Heat Transfer limitations for Water

The variation of Heat Pipe limitations with working fluid temperature for Naphthalene and Dowtherm are shown in Fig. 5 & 6 respectively. Both the fluids can work well in the operating temperature range of 235 to 346 ^{o}C. However, the entrainment limit margin for Dowtherm at lower temperature is low. This can be improved by using finer screen mesh.

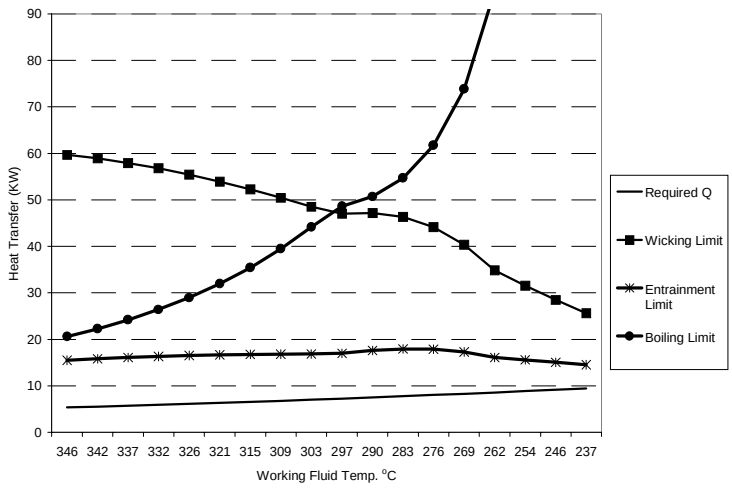

Fig. 5 Heat Transfer limitations for Naphthalene

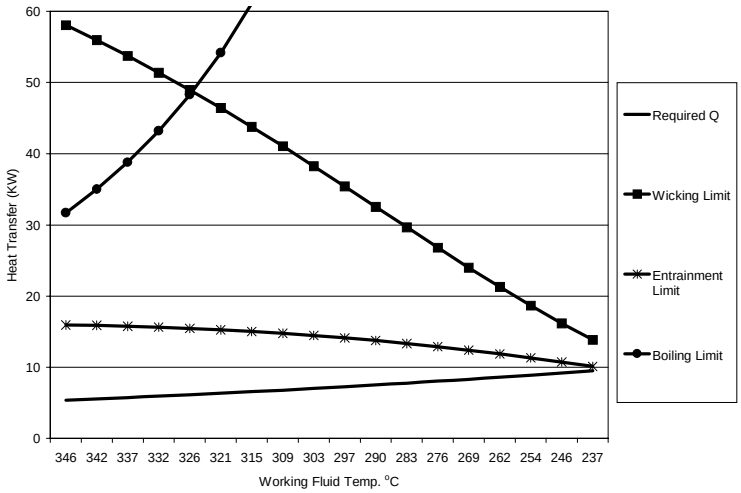

Fig. 6 Heat Transfer limitations for Dowtherm

CONCLUSIONS

In this study, an alternate design of Air Preheater has been proposed by using Heat Pipes. Designs of Heat Pipe Air Preheaters have been obtained by using different types of wick structures which are wrapped screen mesh, rectangular groove wick and screen covered rectangular wick and four different working fluids.

Screen covered rectangular wick has been found to be the best option in terms of overall size of the Air Preheater. A comparison between the working fluid for the same heat duty and for screen covered rectangular wick has also been carried out. As the Heat Pipe Sonic and Viscous limitations for all the above four working fluids are well above the requirement, the comparison has been done on the basis of Wicking, Entrainment and Boiling heat transfer limitations. Use of Water as a working fluid for low temperature section and Naphthalene for high temperature section has been proposed.

The Design of Heat Pipe obtained for Primary Air Preheater of typical 500 MW Thermal Power Plant and the heat duty of 24 MW is 10 m length (5.5 m in evaporator section and 4.5 m in condenser section) of Heat Pipe arranged in 32 rows and each rows containing 80 Heat Pipes using water and Naphthalene as working fluid.

Nomenclature

A	Total heat transfer area (m^2)
A_v	Cross sectional area of vapor space (m^2)
d_e	Fin tip diameter of a circular fin (m).
d_i	Heat Pipe inside tube diameter (m).
d_o	Heat Pipe outside tube diameter (m).
dv	Heat Pipe vapor core diameter (m).
Dh	Hydraulic diameter.
d_g	Groove depth of rectangular groove wick.
F_l	Frictional coefficient for liquid flow.
F_v	Frictional coefficient for vapor flow.
f	Friction factor.
g	Gravitational acceleration (m/s^2).
h_{fg}	Latent heat of vaporization of the working fluid.
j	Colburn modulus (dimensionless).
K	Thermal conductivity (W/m ^oC)/ Wick permeability.
L	Length of Heat Pipe (m).
l_f	Fin Length (m).
N_f	Number of Fin per meter length.
P	Fluid static Pressure (Pa).
P	Partial pressure (Pa).
ΔP	Pressure drop (Pa).
q	Heat transfer rate (W).
R	Thermal resistance (W/k).
Re	Reynolds number (dimensionless).
r_e	Effective capillary radius.
s	Fin Pitch.
T	Temperature (^oK).
t	Temperature (^oC).
U	Over all heat transfer coefficient. (W/m^2 K).
w	Groove width (m).
X_l	Longitudinal tube pitch (m).
X_t	Transverse tube pitch (m).
X_d	Diagonal tube pitch (m).
Z	Characteristic dimension of wick.

GREEK SYMBOLS

γ_V	Vapor specific heat ratio,
ρ	Density of fluid (Kg/ m^3).
μ	Viscosity (Pa. s).
φ	Porosity (dimensionless).
σ	Surface tension (N/m).
δf	Fin Thickness.
Φ	Heat Pipe inclination angle.
τ	Ratio of groove width and depth.

SUBSCRIPTS

a	air or cold fluid.
b	boiling
c	convective
e	evaporator, entrainment
f	Fin
HP	Heat Pipe
g	gas or hot fluid
l	liquid
max	maximum
o	outlet, outer
s	scale, sonic
v	vapor, viscous
w	wall or wick

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