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OPTIMISATION OF ABB'S WIND TURBINE GENERATOR PERFORMANCE BY INSTALLING A HEAT PIPE HEAT EXCHANGER

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1. ABSTRACT

The cooling system of ABB Low Voltage Motors' wind turbine electrical generator utilises a fan that drags the ambient air through a counter flow unmixed stream heat exchanger, to extract heat from hot air circulating in a closed loop around the rotor and the stator by internal fan.

As an optimisation to the system, a secondary cooling unit is to be added: A heat pipe heat exchanger will be used to remove at least 1 kW from the circulating hot stream, and drive it to a heat sink placed in the outlet of cooling air duct. This paper focuses on the design procedure of the heat pipe evaporator and heat sink at certain ambient temperature.

Key words: ABB, Heat Pipe, Wind Turbine Generator, optimisation of cooling unit.

2. INTRODUCTION

An 850 kW wind turbine generator dissipates 22.3 kW heat as a lost energy. The generator is equipped with a cooling unit: internal closed cycle, in which air is circulating through holes in the rotor and is dragged and distributed around the stator using internal centrifugal fan. The air returns back and enters the holes again. The rotor emits 8.3 kW of energy while the stator emits 14.0 kW. The air stream provides cooling effect by extracting heat from both the rotor and the stator. In order to repeat the cycle, the heat absorbed should be transported to a colder air stream. The cold air stream is the ambient air being dragged by an external fan through holes at the circumference of the generator closure, where the heat transport occurs through a radiator shaped surface, as shown in Figure1.

Increasing the cooling effect by post-cooling the internal air stream before it enters the rotor holes again is the principal idea of the project. Installing a heat pipe heat exchanger with its evaporator section located in the grey shaded area, figure 1, and the condenser, heat sink, in the face of the cold air stream coming out of the duct, will post-cool the air stream before it enters the rotor holes.

Installing a new cooling unit or enlarging the original one are not valid solutions because of the space limitations. The internal circulating air should be kept closed protecting it from dust and moisture. Heat in this situation, needs to be rejected from the outside of the sealed enclosure.

We can summarize the reasons of choosing a heat pipe thermal solution for this thermal design problem in the following points:

1. A heat pipe will transport the heat to a location where it can be effectively dissipated by natural or forced convection.
2. The heat pipe provides a thermal path through the enclosure wall, while the internal air cycle is kept close.
3. There will be no need for extra cooling fan that would consume extra power; since the original cooling fan used to drag cooling air for the primary cooling unit is the one to be used in the cooling of the heat sink by forced convection.

2.1 Problem Specifications:

Based on previously measured data, the problem specifications were taken as follows:

- The hot air stream is leaving the counter flow heat exchanger at flow rate equal to 0.15 m^3/s, and at a temperature ranging between 80 and 90^oC. (Average of 85^oC will be considered).

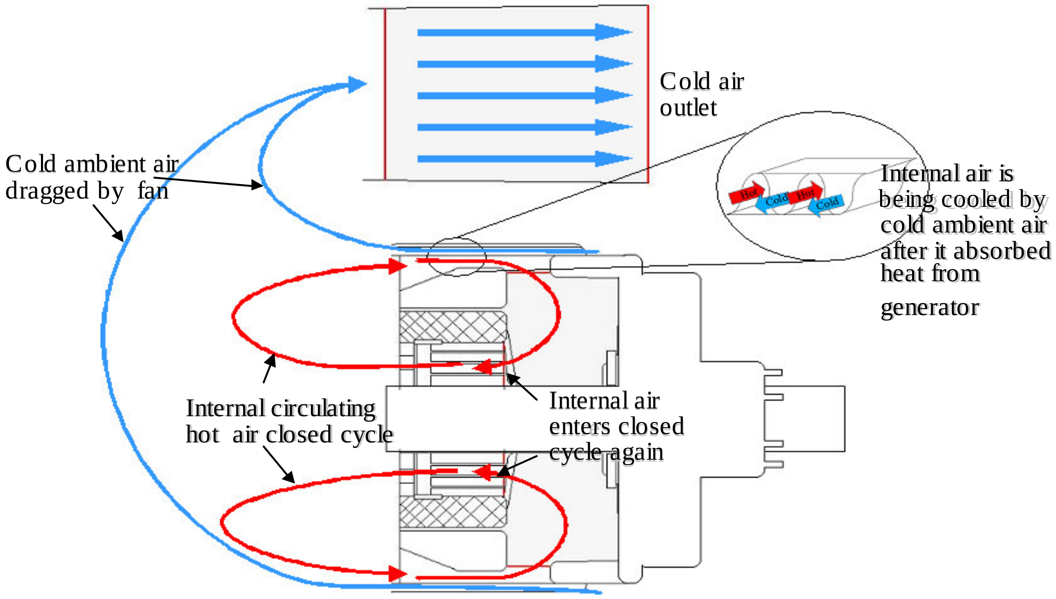

Figure 1, cross section of the generator showing the mechanism of the original cooling unit: hot air circulating in the closed cycle is being cooled by cold ambient air dragged by external fan.

- The cooling air stream is flowing at the duct at a rate of 2.1 m^3/s, and a temperature 34.2 ^{o}C.
- The hot air stream is being heated further, because it passes by the rotor winding which is at a temperature of 122^{o}C.
- The pressure everywhere is close to the atmospheric pressure and it will be assumed as that.

In addition, the rotor is rotating at 1500 rpm causing a remarkable tangential air velocity component of 20m/s at average, at the inlet to the inner cooling holes. They are 24 holes each with 25 mm diameter. See figure 2 for further clarification. The requirement was to remove, at least, 1kW of heat from the hot air stream in the internal air cycle. The design proposed theoretically removes more than 1.5kW.

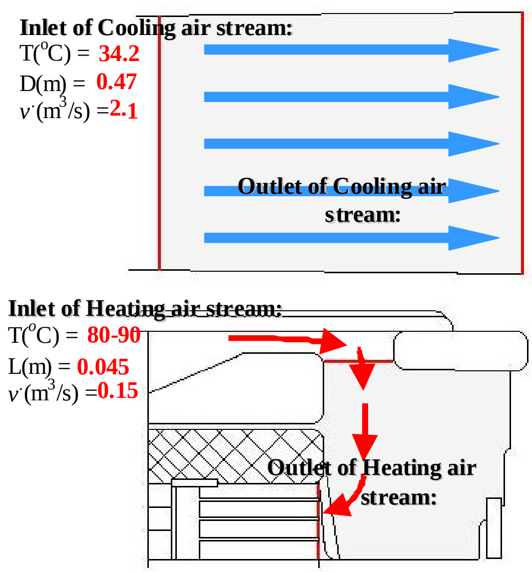

Figure 2, summary of problem specifications

2.2 Design Considerations:

A heat pipe consists of: evaporator, adiabatic link and a heat sink or a condenser. From the manufacturer viewpoint, it is a container that contains a working fluid. In addition to the porous wick structure, which is beyond the scope of this research since the heat pipe mechanism in our case will be gravity aided (thermosyphone heat pipe). Additional information about heat pipes is found in References 2, 3, 9, and 12.

2.2.1 Container: The container should isolate the working fluid from the outside environment. This implies that the container has to be leak-proof, maintain the pressure drop across its walls and its material should be non-porous to prevent the diffusion of vapor. The container should also enable heat transfer to take place from and into the working fluid. This means that the container material should have high thermal conductivity. The container material should be compatible with both the working fluid and external environment. A material with good fabrication properties including weldability, machineability and ductility, is preferable (ref 8).

2.2.2 Working Fluid: The vapor temperature at the operating pressure of the working fluid should be less than the temperature of the hot stream flowing across the evaporator section, and higher than the temperature of the cold stream flowing across the condenser. Within the approximate temperature band, several possible working fluids may exist, and a variety of characteristics must be examined, including:

a. Compatibility of the working fluid with the container wall material.
b. The thermal stability of the working fluid.
c. High latent heat of vaporization is desirable in order to transfer large amounts of heat with minimum fluid flow.
d. The thermal conductivity of the working fluid should preferably be high in order to minimize the

radial temperature gradient and to reduce the possibility of nucleate boiling at container wall.

e. Low values of vapor and liquid viscosities to minimize the resistance to fluid flow.

f. Acceptable freezing point in comparison to the operating temperature range.

The properties mentioned above are those significant for the case gravity-aided operations and don't involve factors affecting the operation of heat pipes acting against the gravity. (Example: High surface tension.)

A particular working fluid can only be functional at certain temperature ranges. Also, the particular working fluid needs a compatible vessel material to prevent corrosion or chemical reaction between the fluid and the vessel. See table 1 for the selection of the suitable working fluid at a given range of temperature, and the selection of the compatible vessel material compatible with the heat pipe working fluid (ref 6).

Temperature Range	Working Fluid	Vessel Material
-70 to 60	Liquid Ammonia	Nickel, Aluminum, Stainless Steel
-45 to 120	Methanol	Copper, Nickel, Stainless Steel
5 to 230	Water	Copper, Nickel

Table 1, selection of working fluid, and the suitable vessel material, for a given range of temperature

3.3 Limitations to Heat Pipe operation

The most important heat pipe design parameter is the amount of power the heat pipe is capable of transferring. The maximum heat transport capability of the heat pipe is governed by several limiting factors, which must be kept into account when designing a heat pipe. There are five primary heat pipe heat transport limitations, which are a function of the heat pipe operating temperature, viscosity, sonic, capillary pumping, entrainment or flooding, and boiling.

As far as a heat pipe is operating below the recommended operating temperature, higher viscous forces may develop hindering the vapor flow to the heat sink. The operating temperature should therefore be raised to match the recommended range.

If the evaporator is extracting too much power relative to the operating temperature, which is the case at start-up when the heat pipe is cold, the exiting vapor would reach sonic velocity resulting in a constant heat pipe transport power and large temperature gradients. As the heat pipe warms up, the temperature gradient will be reduced.

Another consequence of working at low temperature ranges, or above the designed power input, is the entrainment or flooding due to the fact that the high velocity vapor will prevent the condensate from returning back to the evaporator. Increasing the operating temperature and having large enough heat pipe diameter to allow for the cross flow of vapor and condensate inside the pipe would account for this limitation.

High radial heat flux through the evaporator wall will initiate film boiling. This would result in heat pipe dry-out and large thermal resistances. Using a higher thermal conductivity working fluid and vessel materials, and redesign the heat pipe wall structure to spread the heat load would aid in solving the problem (ref 4).

3. DESIGN PROCEDURES:

3.1 Materials Selection:

Based on the heat pipe working fluid selection criteria and the operating range of temperature, Methanol was fluid of choice. It would provide a temperature potential capable of driving the required amount of heat because of its low value of boiling point T_{sat}. Since methanol freezes at a very low temperature, -97C, it is useful in gravity-aided, pool boiling applications where water heat pipes would be subject to freezing. Table 2 shows the methanol properties of direct effect on heat pipe operation. It is highly recommended to check a safety data sheet or a hazard sheet that provides information about safety about dealing with methanol. Evaporator tubes will be made of copper. Heat sink, will be made of aluminium.

Boiling Temp. T_{sat}(°C)	Heat of vaporization h_{fg}(kJ/kg)	Liquid density ρ_l(kg/m^3)	Vapor Density ρ_v(kg/m^3)
64.5	1100	792	1.1
Thermal Capacity Cp(J/kg.°C)	Thermal Conductivity k(J/kg.°C)	Dynamic Viscosity μ(kg/m.s)	Surface Tension σ(N/m)
2510	0.2	0.000817	0.023

Table2, methanol properties of direct effect on the operation of the heat pipe.

3.2 Evaporator design

We will assume that the air temperature T_{air} is the average between the inlet and the outlet streams temperatures. To obtain T_{air} we first need to evaluate the temperature at the outlet of the air stream.

The design assumes that the evaporator should absorb 1500 W from the hot air stream. The rotor is at a temperature of 122^{o}C, which causes heat transfer from the rotor surface subjected to the air stream. The heat transfer coefficient h is assumed to be equal to 36 W/m^2.^{o}C, the outlet air temperature is found from the relation:

$$T_{air\ out} = T_{air\ in} + \frac{Q_{rotor-}\ Q_{cooling}}{\rho_{air} u^{\cdot} Cp_{air}} \qquad (1)$$

where $u^{\cdot}$ is the air velocity,

$Q_{rotor} = A_{rotor}.h.(T_{rotor}\text{-}T_{air})$ where $A_{rotor} = 0.085$ m^2, A_{rotor} is the global rotor area subjected to air flow causing a convection heat transfer component.

and

$$T_{air} = (T_{air\ in} + T_{air\ out}) / 2 \qquad (2)$$

Manipulating (1) and (2) gives:

$$T_{air\ out} = \frac{T_{air\ in} + \dfrac{A_{rotor}\ h\ (T_{rotor}\ \text{-}T_{air}\ ) - Q_{evaporator}}{\rho_{air} u^{\cdot} Cp_{air}}}{1 + \dfrac{A_{rotorr}\ h_o}{2\rho_{air} u^{\cdot} Cp_{air}}} \qquad (3)$$

Heat transferred to the evaporator by radiation will not be considered. Evaluate $T_{air\ out}$ from (3), using approximated values for the air properties, and use it to find T_{air} from (2). Using the air properties at T_{air}, $T_{air\ out}$ is computed again from (2) Repeat, until no significant change occurs on T_{air}. T_{air} is found to be $\approx 81^oC$. This is an approximate value since it is based on approximate value for the heat transfer coefficient h.

This discussion assumes steady state heat transfer, where the temperature of air remains at T_{air}, and it does not cover the period during which the air temperature rises to this value.

The evaporator wall temperature T_{wall} can be approximated to be equal to the boiling temperature of the working fluid T_{sat}, a result of the huge value of the boiling heat transfer coefficient, reducing the thermal resistance between the working fluid and the wall surface to negligible value. The thermal resistance across the wall was also neglected because of the thin wall made of high conductive material. Thus the temperatures on the inner and the outer surfaces of the wall are assumed to be equal, and have a value of T_{sat}.

To find more accurate value for the temperature difference between the cooling liquid and air we'll use the concept of the logarithmic mean temperature difference $LMTD$, replacing T_{wall} by T_{sat} based on the assumptions stated above, where:

$$LMTD = T_{air\ out} = \frac{\Delta T_{in} - \Delta T_{out}}{\ln \dfrac{\Delta T_{in}}{\Delta T_{out}}} \qquad (4)$$

where

$\Delta T_{in} = T_{air\ in} - T_{sat}$

$\Delta T_{out} = T_{air\ out} - T_{sat}$

Evaporation will not start, until the temperature of the working fluid rises from its initial value T_o to the saturation temperature T_{sat} . The time until this happens Δt is estimated from the heat transfer relationship as follows:

$$\Delta t = m\ C_p\ (\ T_{sat} - T_o)/\ Q^{\cdot}_{evaporator} \qquad (5)$$

An evaporator design that that can provide maximum heat transfer area within the constrain of space limitation, see figure 3, is a bundle of 18 tubes, each with 22mm diameter, that go around the end winding space of the generator in the shape of a ring of diameter 475mm.

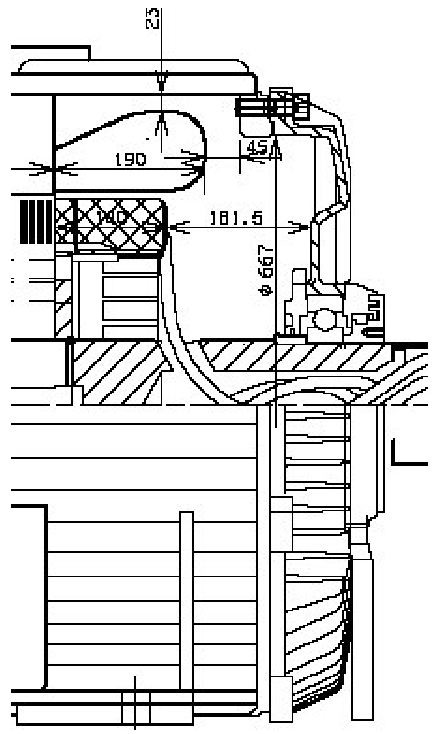

Figure 3, dimensions of space where evaporator will be placed.

In a case of a tube bundle, an average value for the heat transfer coefficient for each tube will be determined and used to calculate the rate of heat flow to the bundle. The heat transfer coefficient associated with a tube is determined according to its position in the tube bank. The coefficient for a tube in the first row is approximately equal to that for a single tube in cross flow, whereas larger heat transfer coefficients are associated with tubes of the of the inner rows. The tubes of the first few rows act as a turbulence grid, which increases the heat transfer coefficient for tubes in the following rows. This would result in a value of the average heat transfer coefficient that is larger than that in the case of air flowing across a single pipe (ref7).

The average heat transfer coefficient for airflow across tube bundles h can be obtained from the modified Grimison correlation (ref 7)

$$Nu_D = 1.13 C_1 C_2 \, Re^{m}_{max} \, Pr^{\,1/3} \qquad (6)$$

This correlation assumes that the bundle at least consists of 10 rows, the maximum Reynolds number Re_{max} value is between 2000 and 40,000 and the minimum value for Pr of the fluid passing across the bundle is 0.7

All air properties are evaluated at the film temperature T_{film}, which is average between the air temperatures, evaluated to be 81°C in previous analysis, and the temperature of the tube wall, which can be approximated as the saturation temperature T_{sat} of the working fluid as mentioned previously. T_{film} = 72.5 °C.

The characteristic dimensions of the tube bundle will be set as follows: The diameter D of each tube will be 22 mm. The torus diameter D_T will be 475 mm. Also, the number of tubes N will be 18, distributed in a staggered configuration in 6 rows (N_L=6) such that the transverse pitch S_T will be taken to be 37.5 mm while the Longitudinal pitch S_L will be taken to be 25 mm. The constants in equation (6) can be obtained from (ref 7), were the nearest values of the constants are chosen in a rough approximation to be C_1 = 0.478 *and* m = 0.565

C_2 is a correction factor for a tube bundle with N_L <10. It can be taken to be 0.95 (ref 7).

The design procedure followed in (ref7) assumes that the velocity of the air, before inserting the tube bundle is constant. The air velocity at the leading edge, first row of tubes, is the same everywhere in the configuration. However, in our system, the air velocity is increasing as the air is flowing radially from around the stator and through the space, where the heat pipe is going to be installed, to the holes through the rotor. Flowing across the tubes, the minimum velocity will be at the leading edge, while a maximum will be attained at the last row. A reasonable approximation is to make the calculations based on the velocity of the air when it reaches the centre of the configuration, in other words, a distance equal to the torus radius r_c from the axis of the generator.

The maximum velocity at the centre of the configuration $V_{c\ max}$, based on the chosen configuration, is obtained from the following relationship

$$V_{c\,\max} = \frac{u^{\cdot}\, S_T}{2\pi r_c\,(0.1315)(S_T - D)} \qquad (7)$$

to be equal to 1.85 m/s. $Re_{c\ max}$ can be obtained from the relation

$Re_{c\ max} = V_{c\ max} D/\nu_{air}$

$Re_{c\ max}$ = 1980, notice the error arising from using equation (7) since $Re_{c\ max}$ is not laying in the range of validity.

$h = Nu_D\ D/k_{air}$, h can be evaluated to be 45 W/m^2.°C. This value can be improved by using it in the analysis described previously to obtain T_{air}, and thus, more accurate values for the air properties can be got.

The temperature of the air at the outlet of the evaporator tube bank $T_{air\ out}$ can now be evaluated using equation (3) to be 76.2°C and used to evaluate the log mean temperature difference *LMTD*, equation (4), to give a value of 15.7 °C.

The rate of heat transfer to the tube bundle is evaluated as follows

$$Q_{evaporator} = N\pi^2 D D_T\, h(LMTD) \qquad (8)$$

which gives about 1308W convection power.

The weigh of the liquid inside the evaporator m, can be computed to be 6.68kg. The weight of the empty tubes is equal to about 0.002ρ kg, where ρ is the density of the material to be used in manufacturing the evaporator. The time required for the operation of the heat pipe to begin can be estimated using equation (5) to be 8.76 minutes. Methanol vapor will be formed on the inner walls of the evaporator tubes at a rate equal to $Q_{evaporator}/h_{fg}$ = 1.30 g/sec or 4.65 kg/hr

The pressure drop associated with flow across a tube bank should be recognised, as the power required to move the fluid across the bank could be a major operating expense. This power is directly proportional to the pressure drop, which may be expressed as

$$\Delta p = N_L \chi (\rho_{air} V^2_{max}/2) f \qquad (9)$$

where f is the friction factor and χ is the correction factor. Values of f and χ are obtained from ref7] for the case of staggered arrangement of tubes in the form of an equilateral triangle. The analysis will give a pressure drop ΔP of no more than 5 Pa, which is less than 1% of the total pressure drop in the internal air flow.

3.3 Condenser design:

The objective now is to remove the heat absorbed by the evaporator and transferred by the adiabatic section to the condenser. It is important to state our design assumptions that will essentially simplify the problem: 1. heat transfer inside the duct was assumed to be totally due to forced convection of air driven by the duct fan and affect of natural convection was disregarded. Effect of radiation was disregarded too due to low emissivity of wall surface subjected to flowing air. 2. Heat transfer to the ambient environment is assumed to be due to natural convection and radiation. 3. Steady operating conditions exist. 4. Condensation of working fluid on the surface of the heat sink will provide a uniform surface heat flux, 5. The surface of the heat sink is smooth. 6. Air is an ideal gas.

The evaporated methanol will pass through seven separated 22 mm circular tubes that represent the surface of the heat sink. The tubes will be arranged in a staggered configuration in which the perpendicular distance between two following tubes is 30 mm. Four tubes will have a diameter of 440 mm while three tubes will have a 420 mm diameter. The configuration will be constrained by a length of 200 mm due to the existence of 11 radial wings that are used to direct the air, as shown in figure 4. Notice how the wings can work as extended surfaces that will improve the convection inside the duct. This research will spot the light on the rate of heat removed by the tube bundle, the heat removed by forced convection of air driven by the fan, and will not consider the other modes of heat transfer to the outside of the duct.

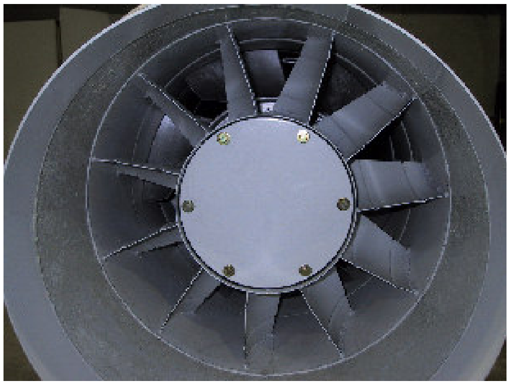

Figure 4, duct with internal air directing wings

A value for the air temperature inside the duct T_{air} is needed for the design. The air properties are function of this unknown temperature but since it is predicted that the rise in temperature due to the heat rejected by the heat sink will be so small, because of the relatively high value air velocity, the air properties will be taken at temperature of air leaving the fan, $T_{air\ in}$ = 34.2 ^{o}C. An average value for the heat transfer coefficient h needs to be evaluated**.** The previous process used to calculate the heat absorbed by the evaporator will be used again. The air velocity, required to evaluate Re, is not the same all the way. It has a maximum value of 14.8m/s before the air faces a sudden expansion then a gradual decrease in the cross sectional area of the duct until it reaches a velocity of 12.1 m/s at the duct outlet. A simplified profile for the velocity distribution inside the duct is shown in figure 5.

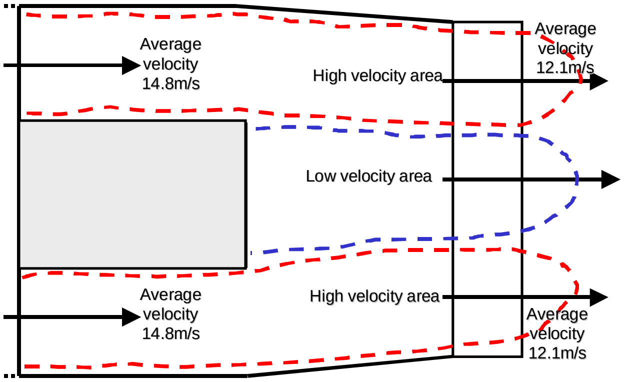

Figure 5, simple velocity profile for air inside the duct

The velocity that is going to be used in the computation is the average between the velocities at the inlet and the outlet of our system. The average value of the velocity is evaluated to be 13.4 m/s which gives Re = 13050 satisfying the condition of the validity of equation (6). h is evaluated to be 155.5W/m^2.^{o}C.

The corresponding air properties are υ = 0. 206 x 10^{-4} m^2/s, k= 0.03W/m.^{o}C and Pr = 0.706. Equation 6, and using the appropriate constants from [7], well lead to a rough approximation for the value of h equal to 164 W/m^2.C. T_{sat}-T_{air}= 64.5-34.5= 30^{o}C, the total heat sink surface area is 0.651 m^2 and the rate of heat transfer, provided by the effort of the condenser tubes only, is approximately 3200W. The final design drawing of the heat pipe is found in the conclusion.

4. DESIGN SUMMARY:

Two-dimensional design drawings, of the evaporator and the condenser are shown in this section, figures 6. Dimensions are provided in millimetres.

The total heat transfer to the evaporator tubes is equal to 1308W. The total amount of heat that can be emitted by the heat sink is equal to 3200W.

When monitoring heat pipe performance, the key parameter is the temperature difference between the surfaces of the evaporator and the condenser. It can be measured by attaching thermocouples to both parts and reading the temperatures measurements periodically. A similar evaporator-to-condenser temperature difference ΔT_{hp} should be measured at the beginning of the system life and throughout the duration of the test, providing equal heat input rates. If ΔT_{hp} is increasing as the test proceeds, this is then typically due to the non-condensable gas NCG collecting in the condenser region of the heat pipe. The NCG is swept there by flowing vapour. The region that contains the NCG is not an active part of the heat transfer mechanism; consequently, it shows up as a cold region. As time moves on and more gas collects this region, the heat pipe, as determined by the attached thermocouple will be colder at that region (ref 6).

Ethanol, with boiling temperature 78 ^{o}C can be used as an alternative working fluid, if extreme conditions were assumed and the ambient temperature was taken to be about 40 ^{o}C. Ignoring unlinearities that are hard to be figured out in measurements, the rise in the ambient temperature will correspond to an equal rise in the temperature of each point inside the generator. This includes the temperature of air surrounding both the evaporator and the condenser. The will provide a temperature differences in the two parts, capable of driving the operation of the heat pipe.

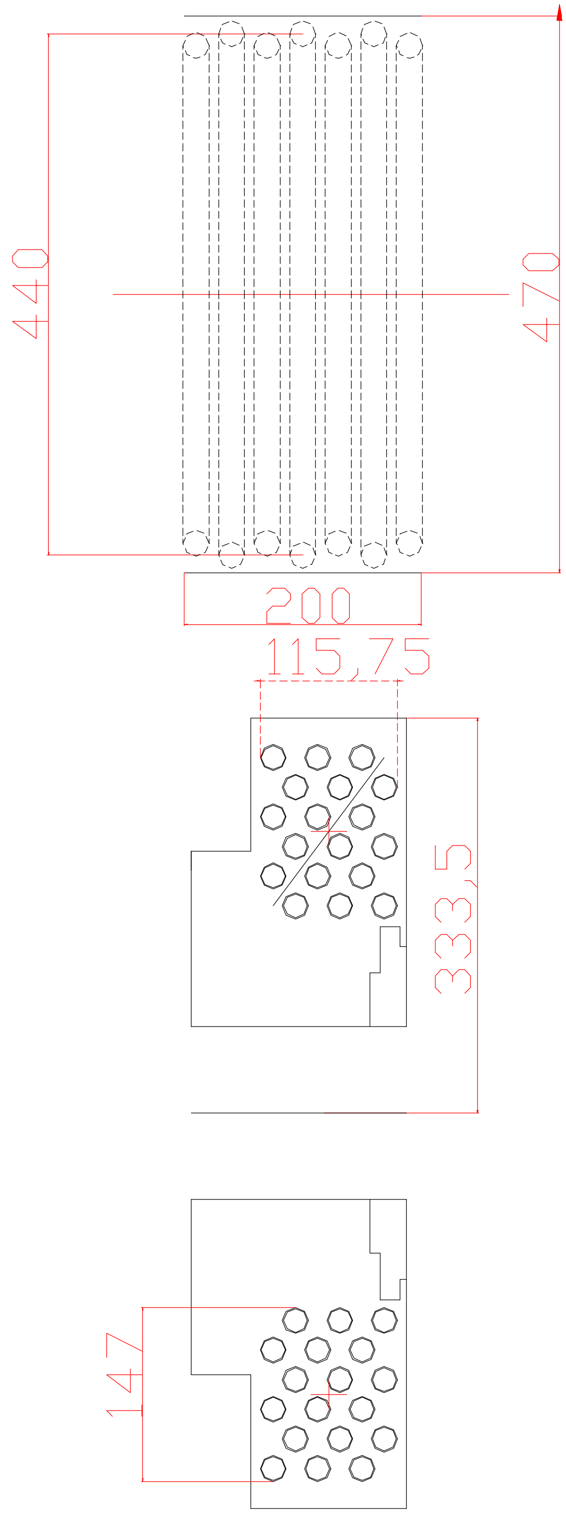

Figure 6, undetailed full heat pipe drawing

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