University of Nevada, Reno

THERMAL MANAGEMENT OF ELECTRONICS EXPERIENCING LARGE POWER DISSIPATION TRANSIENTS

A thesis submitted in partial fulfillment of the
requirements for the degree of Master of Science in
Mechanical Engineering

By

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THE GRADUATE SCHOOL

We recommend that the thesis
prepared under our supervision by

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ABSTRACT

Recent developments with micro-porous surfaces have led to the introduction of very high heat flux boiling surfaces. For example, Wirtz and coworkers have introduced a structured porous surface that achieves in excess of 1.5MW/m^2 in saturated pool boiling of water at 0.2atm (T_{sat} = 60°C); and, this increases to 2.1MW/m^2 at 1.0atm (T_{sat} = 100°C). In this case, onset of nucleate boiling (ONB) occurs at approximately 2K superheat, and the high heat flux noted occurs at 9K superheat, so the boiling curve slope is very high, approximately 0.3MW/m^2 per Kelvin superheat. This suggests that a viable thermal management approach to systems that experience large transient heat loading would be to design the heat sink so that the heat sink-to-coolant heat transfer surface would "ride" its boiling curve. At baseline power dissipation levels, boiling would occur at near ONB; and, the heat transfer surface would climb the (very steep) boiling curve as the power dissipation level rises, so that there would be only a moderate electronics temperature excursion.

Vented (constant pressure), sealed and hybrid immersion cooled electronics systems are considered. A thermal response model, based on boiling curves having differing ONB and CHF points (boiling curve slopes) and boiling correlation calculates the thermal response of the system. Response time constants and maximum temperature excursions are correlated with boiling curve characteristics, and system geometric and thermo physical properties.

We have obtained the temperature response with respect to time were obtained with different system parameters and have been compared with each other

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NOMENCLATURE

A	Area [m^2]
c	Specific heat[kJ/kg*K]
C_f	Compression Factor
d	Diameter[mm]
D_h	Pore hydraulic diameter
g	Acceleration due to gravity[m/sec^2]
h	Heat Transfer coefficient[$watt/m^2K$]
K	Thermal Conductivity[watt/mK]
K_{Dmin}	Active Nucleation side density parameter
m	Mass[kg]
M	Mass parameter (equation 30.1)
n	Number of moles
Nu	Nusselt Number
P	Pressure[atm]
q''	Heat flux[$watt/m^2$]
R	Universal gas constant
Ra*	Modified Rayleigh number
T	Temperature[K]
u	Internal energy[kJ/kg]
U	Total Internal Energy[kJ]
v	Specific volume[m^3/kg]

V	Volume[m^3]
ΔT_{sat}	Wall super heat[k]

Greek symbols

α	Thermal diffusivity
β	Specific surface area[cm^2/cm^3]
δ	Laminate thickness[mm]
ε	porosity
ν	Kinematic viscosity[m^2/s]
ρ	Density[kg/m^3]
σ	Surface tension[N/m]

Subscripts

e	Electronics
f	fluid
g	gas
i	initial
s	settling
t	total

CHAPTER 1

INTRODUCTION

Ships have an advantage in that they carry sufficient power generation to make new high power density systems feasible. These systems, such as propulsion, pulse power applications, and sensor arrays, which dissipate high heat, will use electric power rather than chemical propellants, providing for a safer ship. A review of these systems shows that a wide range of power requirements varying form delivery of megawatts for seconds to terawatts for microseconds is required for these devices. The power requirements for the electric gun fall midway between these extremes. Modern navy ships are capable of developing this power by using conversion and storage capability to produce the electrical characteristics required by the loads. Therefore pulse power devices are not limited by available shipboard power, but by the thermal management of the systems [1].

All power system components will have thermal design issues when their performance is pushed to deliver higher power. For example, during shot fire, a great amount of heat is transferred to the gun structure. During operation, a gun may need to fire at a very high rate, leaving insufficient time between cycles to transfer heat input to the external environment by conventional cooling. In such cases the gun barrel can melt. To avoid such incidents higher heat flux thermal management is required [2].

High heat flux thermal management of onboard systems will soon become a bottleneck for these kinds of applications since traditional single phase heat transfer

applications will not be sufficient. Two-phase heat transfer promises to meet such stringent, transient applications. Of all the liquids available water at reduced pressures has highest latent heat capacity at temperatures targeted to electronics package cooling. But as pressure decreases the boiling performance also degrades. To increase boiling performance, enhanced surfaces have been developed [3]

It is well understood that boiling enhancement can be achieved by proper selection of boiling surface morphology. Boiling improvement on enhanced surfaces is commonly attributed to an increase in the number of active bubble nucleation sites, as well as increases in bubble departure frequency, and improved surface wettability. These effects all result in an increase to the time-averaged solid surface area that is in contact with rapidly moving thin film menisci of nucleating bubbles, and consequently by transient thin film evaporation. These enhancement mechanisms can be achieved by the creation of a large number of interconnected, reentrant vapor-containing cavities, and when appropriately selected, the enhanced surface will provide a substantial decrease in wall superheat, as well as provide improvement in CHF [4]. At issue in this work is how enhanced surfaces respond to transient power (heat transfer) inputs.

Penley and Wirtz [4] have conducted steady experiments on saturated sub-atmospheric pressure pool boiling. They have utilized fine filament screen-laminate enhanced surfaces as effective bubble nucleation sites. Results show that the boiling performance is a strong function of screen laminate geometry. They used dimensional analysis and multi-parameter regression to develop a heat transfer correlation that relates the boiling heat transfer coefficient to the lamination geometry and power (rate of heat)

input. In this work, the transient response of Penley and Wirtz's surface, subjected to a pulse heat input, is studied.

Literature on Transient Boiling, Duluc, Stutz, and Lallemand [5, 6] conducted experiments on transient pool boiling on both a thick copper block and a thin brass wire with a highly wetting fluid to investigate the influence of thermal inertia on transient boiling. The study shows that the thermal behavior of the system and the associated time scales are found to depend on temperature at boiling inception. For both samples, time evolution of the wall super heat has a similar shape: a temperature overshoot is observed before steady-state is achieved.

Auracher and Marquardt [7] conducted experiments on a copper heater with highly wetting fluid and fluids with large contact angles. Under transient conditions he described that the heating transients always yield higher heat flux than cooling transients at the same wall temperature.

Okuyama and Iida [8] conducted experiments on platinum wire in a pool of liquid Nitrogen. He has correlated the history of transient boiling heat transfer rate of liquid nitrogen corresponding to the bubble behavior during the transition stage to film boiling. In the case of a low heat generation rate, boiling transition occurs due to the coalescence of nucleate boiling bubbles. In the case of a high heat generation rate, a vapor sheath grows along the heater wire. Transition boiling occurs due to the settling of the vapor sheath around the wire and the heat transfer rate just after the boiling transition becomes much lower than that of stationary film boiling.
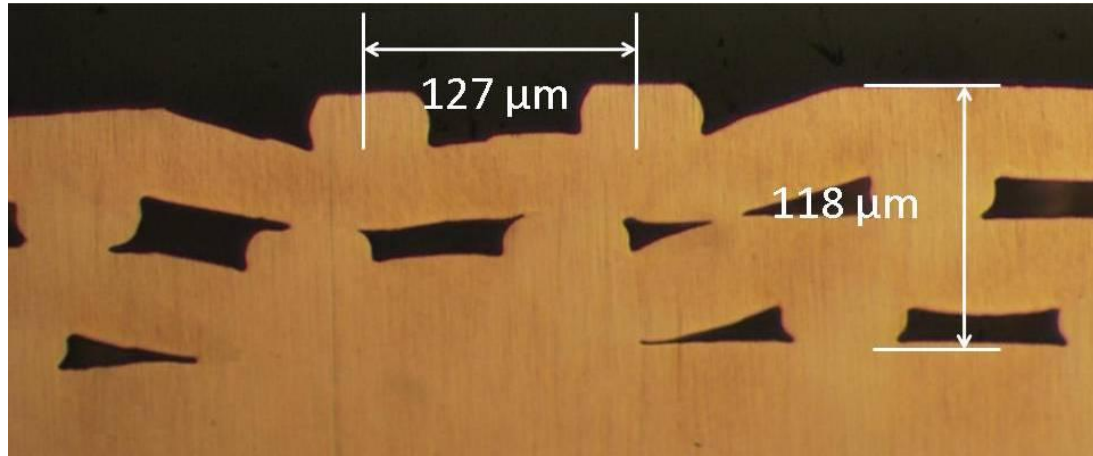
Figure 1 Cross-section of 200 Mesh 2 layer sample

Deev, Kutsenko, Lavrukhin and Kharitonov [9] conducted experiments on platinum wire immersed in saturated water at atmospheric pressure. In this study it was shown that the initial heat generation on a heater, strongly affects the process of transition from nucleate to film boiling under fast increase of heating power. It was shown that part of the heated surface occupying vapor before step power input strongly affects characteristics of transition to film boiling mode.

Sathyamurthi and Banerjee [10] have analyzed the transient surface temperature fluctuations during subcooled pool boiling on a large-size heater. They generated the boiling curve using surface temperature data obtained at each steady state condition. They have generated the phase plots for the original dataset and the noise reduced datasets for saturated and subcooled pool boiling. The phase plots for the original data shows a wool-ball shaped attractor. Thus, the attractor geometry is not clearly visible. In contrast, the noise-reduced attractors show well-defined structures that vary with wall-superheat levels.

15.0kV 12.4mm x150 SE(U) 3/23/10   300um
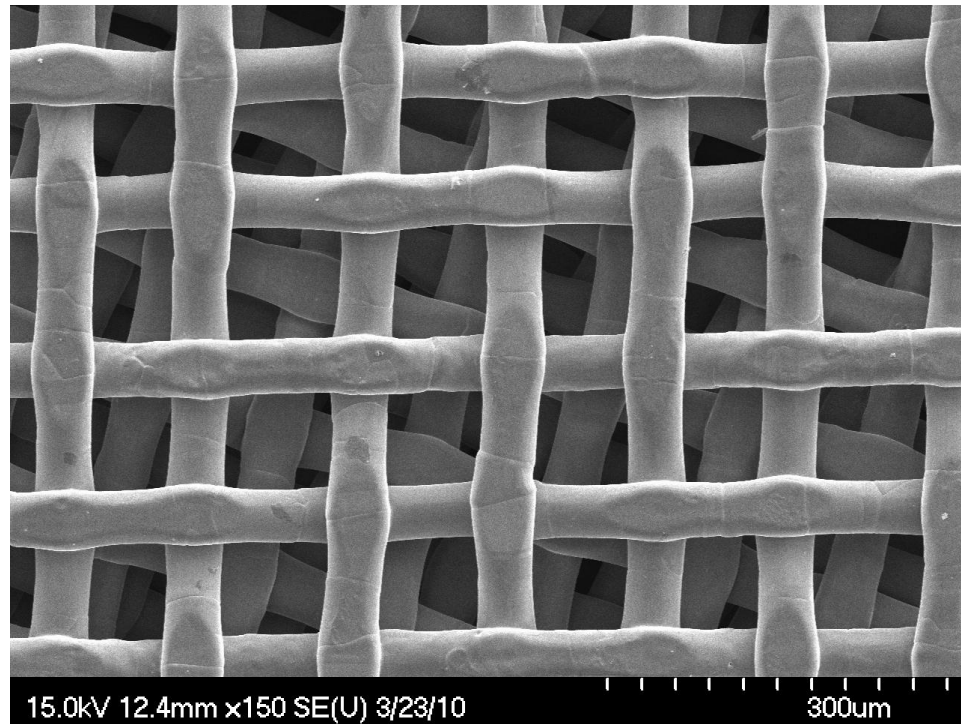
Figure 2 Plane view of 200 Mesh 2 layer sample

In present work we are simulating the response of temperature with time for Penley and Wirtz's enhanced boiling surface. A diffusion bonded sample is shown in Figs. 1 and 2. The surface is a 200 mesh surface. The cross-section of 200 mesh 2 layers and plane view of a 200 mesh 4 layers, diffusion bonded to copper foil is shown in Figs. 1 and 2 respectively.
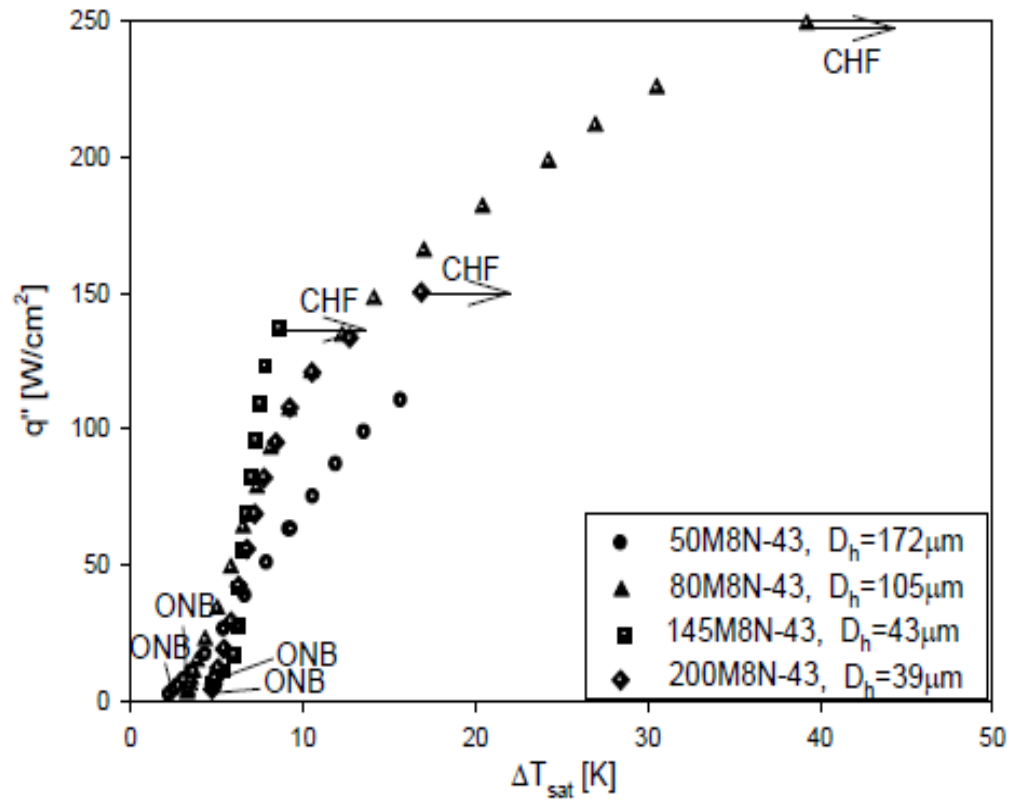
Figure 3: Boiling Performance on 0.2 atm of 8-layer screen laminates with Mesh Numbers

50, 80, 145 and 200

1.1 PENLEY/WIRTZ BOILING CURVE AND CORRELATION

The boiling curve is represented by heat flux on y axis and wall super heat on x axis. Reduction in pressure leads to reduced active nucleating site density and performance lag. However, screen laminate enhanced surfaces increases the active nucleation site density and significantly improve boiling at low pressure. The boiling performance as a function of size of screen laminate is shown in Fig 3. As evident in Fig 3, as the mesh number increases, the boiling performance increases, but their exists an
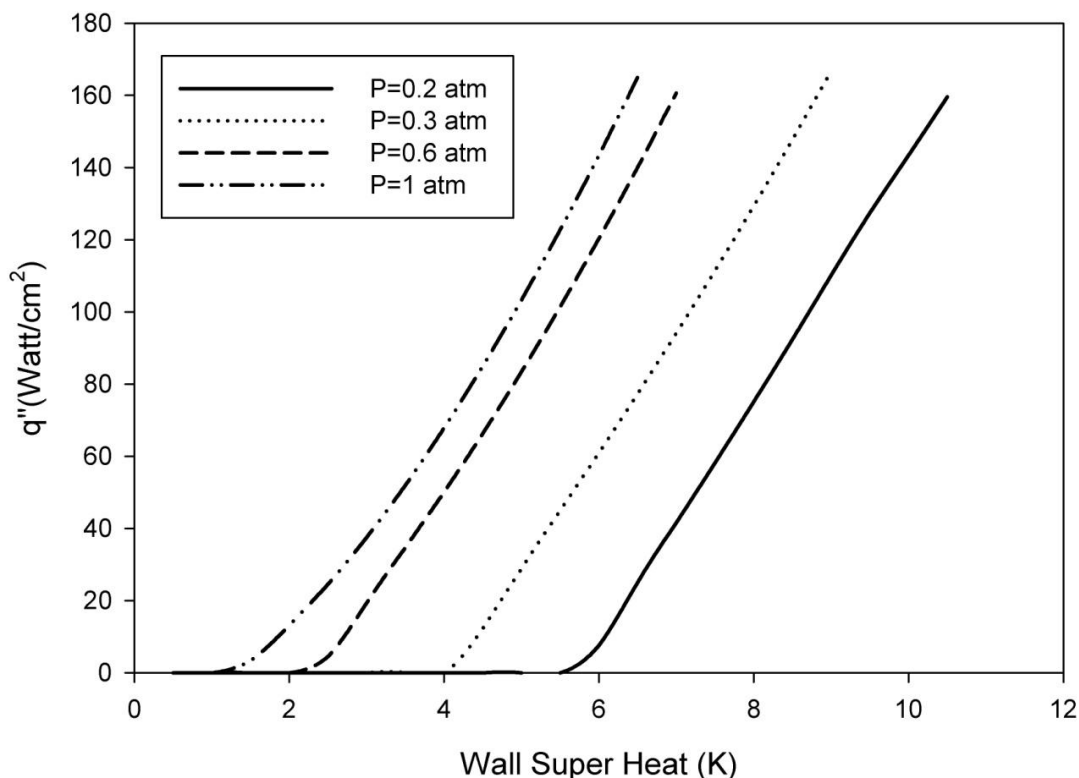
Figure 4: Boiling performances of 145 Mesh 8 layer data at various pressures

optimal mesh number where the boiling performance is maximum and as the Mesh number is increased further the boiling performance decreases.

The boiling curve under consideration for solving the temperature transients is obtained on 145 Mesh 8 layers surface in pool boiling condition, which is the best performer in the work done by Penley and Wirtz [4].

In both the closed system and hybrid system, the pressure of the system varies with respect to time. Hence the correlation for heat transfer and wall super heat of sub-atmospheric pool boiling from work of Penley and Wirtz [4] is considered. The correlation for sub atmospheric pool boiling is given by equation 1.1.

$$\left(\frac{q''.D_h}{\Delta T_{sat}k_f}\right) = 4559.\left(\frac{q''.D_h}{\varepsilon.h_{fg}\mu_f}\right).K_{D_{min}}{}^{0.52}.Ja_p^{.45}.e^{-.0025(\beta\delta-188)^2} \qquad (1.1)$$

Where, Ja_p is modified Jakob number, $\frac{q''.D_h}{\varepsilon.h_{fg}\mu_f}$ is a pore Reynolds number, $K_{D_{min}}$ is active nucleation site parameter, $\frac{q''.D_h}{\Delta T_{sat}k_f}$ is Nusselt number, $e^{-.0025(\beta\delta-188)^2}$ is a Gaussian form for $(\beta\delta)_{opt}$. The properties of water were taken from WaterSteamPro software.

The pressure effect on boiling performance on a surface is shown in Fig 4. In Fig 4 we can see that as pressure increases, the curve shifts to left which indicates increase in the boiling performance.

1.2 OBJECTIVE

The objective is to model the transient response of a convectively cooled device that is subjected to a step increase in power dissipation where cooling is via boiling heat transfer on a high-performance surface. In this project we are numerically simulating the transient temperature response of electronics using different pool conditions such as totally vented system, totally closed system and partially closed system.

Different pool conditions are considered because of their advantages. A closed system will reduce the amount of water vapor lost to the surroundings; whereas an open system will result in steady operating temperatures; and a hybrid (closed/vented) system is considered to take the advantage of both the open system in terms of stable temperatures and closed system in terms of the water vapor out of the system.

The simulation of transient cooling of a device is carried out using dimensional analysis to reduce the parameters in the equations. The conditions of different mass, heat generation rate of the device are also used to see the transient behavior of the system.

CHAPTER 2

MODELING OF SYSTEM

A numerical model for closed system, open system and hybrid system are developed. These models are based on two phase heat transfer coefficients described in the work done by Penley and Wirtz [4]. The transient response of temperature of electronics is calculated using energy balance equations on the control volumes. The analysis is based on lumped heat transfer analysis.

2.1 COOLING SYSTEM

The system shown Fig.5 is a general system. When heat is supplied to the boiling container, phase change occurs Figure 5 shows the schematic of a system. It will become a closed system when the outlet valve and inlet valve of the system is closed.

Figure 5 shows a system with two control volumes cv1 and cv2. The heat transfer between cv1 and cv2 is due to two phase heat transfer. There is an amount of heat lost to the surrounding, Q_n. The coolant tank consists of an inlet and outlet valve to supply and remove the liquid or vapor from it.
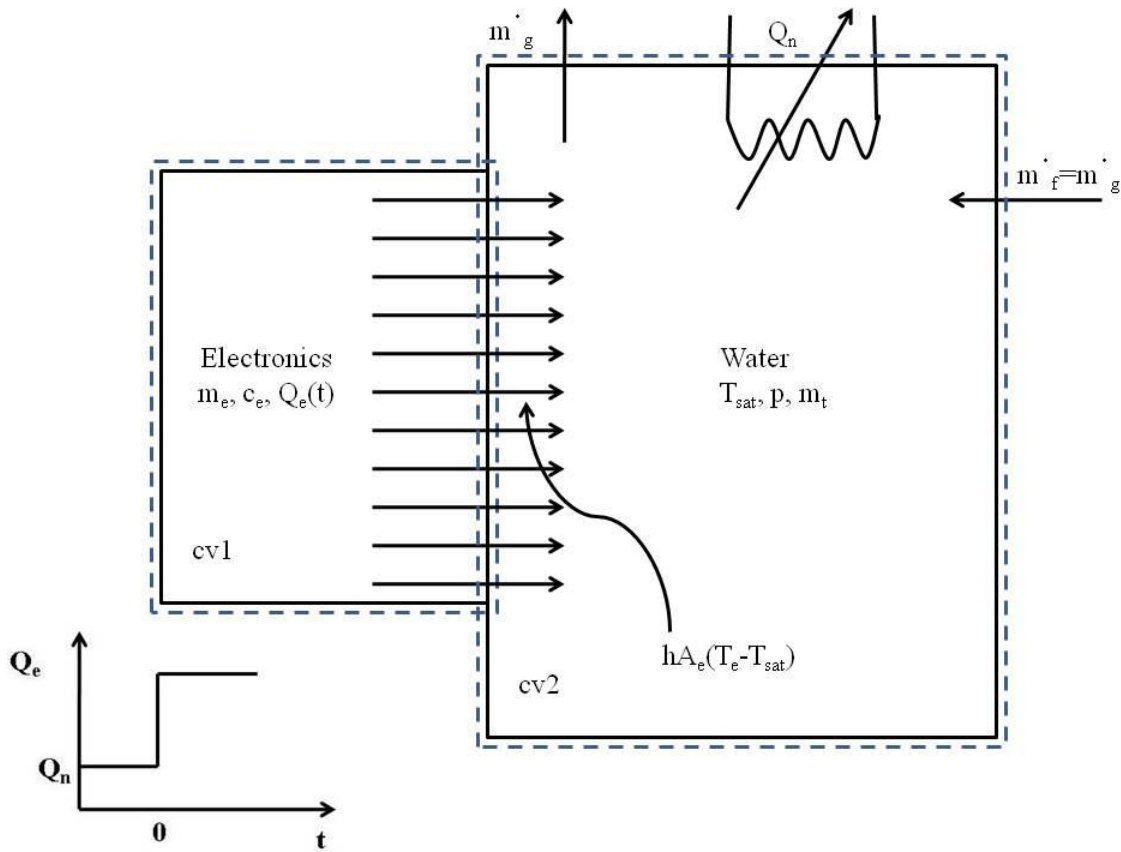
Figure 5: Schematic of system

Here, $m_e c_e$ is the thermal mass of electronics, $Q_e(t)$ is the heat generated by electronics as shown in the graph situated at left hand bottom corner of Fig. 5 , m_t is the total mass of steam and liquid inside the tank, T_{sat} is the saturation temperature of the liquid at pressure p, Q_n is heat lost to the surroundings, $\dot{m}_g$ and $\dot{m}_f$ are the mass of steam out of the system and mass of liquid into the system.

In this system the two phase heat transfer coefficient is a function of both the wall superheat and the pressure of the liquid container. As the pressure inside the container

changes the heat transferred from the electronics will be used in both making the bubbles and the pressure increase causes a change in saturation temperature of water.

2.1.1 CLOSED SYSTEM

In a closed system the total mass of fluid inside the boiling container is a constant. When heat is supplied to the boiling container, phase change occurs, and the vapor formed will not be vented out and hence the pressure inside the container increases. As pressure varies the two phase heat transfer coefficient will not be a constant for the given wall super heat with respect to time.

For closed system the mass out of the system and mass into the system are zero, i.e. $\dot{m}_g = \dot{m}_f = 0$

2.1.1.2 EQUATIONS FOR CLOSED SYSTEM

The advantage of the closed system is that, in defense application it will reduce the thermal energy dissipated to the surroundings and hence will reduce the chances of being seen on a thermal image.

In a closed system the total mass of fluid i.e m_t is constant and is equal to sum of both the mass of fluid in liquid state m_f and mass of liquid in vapor state m_g as per conservation of mass principle.

The equations to solve the transient heat transfer are derived using an energy balance on different control volumes. For this problem we choose two control volumes,

the first control volume (cv1, Fig 5) will be on the heat transfer surface, and the second control volume will be on the tank (cv2, Fig 5).

From conservation of energy principle, by applying an energy balance to cv1, we get the equation for temperature of electronics with respect to time as shown in Equation 1. The assumptions are that there is no contact resistance between the surface and electronics package hence we considered the surface to be a part of electronics package, and that the entire electronics package is at uniform temperature.

$$m_e c_e \frac{dT_e}{dt} = Q_e(t) - h(\Delta T_{sat}, P) A_e \Delta T_{sat} \tag{1}$$

where, $m_e c_e$ is the thermal mass of electronics, t is time, $\Delta T_{sat,i}$ is the wall superheat of electronics package, $Q_e(t)$ is the heat generated by electronics package, P is the pressure inside the system, $h(\Delta T_{sat}, P)$ is the two phase heat transfer coefficient on electronics derived from the boiling curve. It is a function of both wall super heat and pressure inside the system, A_e is the area of the electronics heat transfer surface. Equations 1 can further be reduced to Equation 2.

$$\frac{dT_e}{dt} = \frac{Q_e(t) - h(\Delta T_{sat}, P) A_e \Delta T_{sat}}{m_e c_e} \tag{2}$$

From conservation of energy principle, the energy balance on cv2 gives the internal energy and quality of steam which intern gives the pressure inside the system. The internal energy added to the control volume is given by Equation 3.

$$\frac{dU}{dt} = h(\Delta T_{sat}, P) A_e \Delta T_{sat} - Q_n \tag{3}$$

where U=m_tu, u is the internal energy, Q_n is the heat lost to the surrounding as shown in Fig 5.

From conservation of mass principle, in the closed system the mass of vapor formed is not vented out, so the total mass inside the system is constant ($m_t = m_g + m_f$). The quality of steam is the proportion of saturated steam in saturated water/steam mixture. Mathematically, quality is defined by the relationship as shown in Equation 4.

$$x = \frac{u - u_f}{u_{fg}} \tag{4}$$

Differentiation of Eq. 4 gives

$$\frac{dx}{dt} = \frac{u_{fg}\left(\frac{du}{dt} - \frac{du_f}{dt}\right) - (u - u_f)\frac{du_{fg}}{dt}}{u_{fg}^2} \tag{5}$$

where, x is the quality of steam, u is the internal energy supplied to fluid, $u_f(P)$ is the specific internal energy of water at that particular pressure, $u_{fg}(P)$ is the difference between the specific internal energy of water vapor and water at that particular pressure.

The pressure inside the system is the pressure exerted by vapor on liquid and on the wall. As water is almost an incompressible liquid, water acts as a wall in this case. The equation used to calculate the pressure inside the system is calculated by using Equation 6, which is obtained from the state law.

$$p = \frac{mRT_{sat}}{V} \tag{6}$$

Differentiation of Eq. 6 gives

$$\frac{dp}{dt} = \frac{V\left(T_{sat}\frac{dn}{dt}+m\frac{dT_{sat}}{dt}\right)-mT_{sat}\frac{dV}{dt}}{V^2} R \qquad (7)$$

Where m is the mass of vapor present in the system, $R(461.5 \text{ J/kgK})$ is gas constant for steam, T_{sat} is the saturation temperature at that particular pressure, V is the volume of steam at that particular pressure.

2.1.1.2 NON-DIMENSIONALIZING THE EQUATIONS

In order to remove the scaling effect from the equations, the Equations 2, 3, 5 and 7 are scaled using the following reference values

a) Time reference

$$t_r = \frac{m_e c_e}{h_i A_e} \qquad (8)$$

b) Temperature reference

$$T_r = T_{e,i} - T_{sat,i} = \frac{Q_n}{h_i A_e} \qquad (9)$$

c) Mass Reference

$$m_r = \frac{Q_n}{h_{fg}} \frac{m_e c_e}{h_i A_e} \qquad (10)$$

Where, h_i is the initial heat transfer coefficient of the system defined as in Equations 11.

$$h_i = \frac{Q_i}{T_r A_e} \qquad (11)$$

The Reference values are selected, based on the values fell out of non-dimensionalizing the equations 2, 3, 5, 7.

2.1.2.1 NON-DIMENSIONALIZING EQUATION 2

Equation 2 was derived by applying energy balance on electronics. Equation 2 is a differential equation of temperature with respect to time. We are using t_r, T_r, m_r to non-dimensionalize Equation 2

$$m_e c_e \frac{T_r}{t_r} \frac{d(\frac{T_e - T_{sat,i}}{T_r})}{d(\frac{t}{t_r})} = Q_e(t) - h(T_e - T_{sat}, P) A_e (T_e - T_{sat}) \quad (12)$$

The dimensionless temperature is given by the equation 13.

$$\bar{T}_e = \frac{T_e - T_i}{T_r} \quad (13)$$

where, T_i is the initial temperature of electronics,

$$\bar{t} = \frac{t}{t_r} \quad (14)$$

Input the dimensionless values from Equation 13 and 14 into Equation 12 we get

$$m_e c_e \frac{T_r}{t_r} \frac{d\bar{T}_e}{d\bar{t}} = Q_e(\bar{t}) - h(\bar{T}_e, P) A_e (T_e - T_{sat}) \quad (15)$$

Simplifying Equation 15 we get

$$\frac{d\bar{T}_e}{d\bar{t}} = \frac{Q_e(\bar{t})}{m_e c_e \frac{T_r}{t_r}} - \frac{h(\bar{T}_e, P) A_e (T_e - T_{sat})}{m_e c_e \frac{T_r}{t_r}} \quad (16)$$

Using values from Equation 13 and 14 we get

$$m_e c_e \frac{T_r}{t_r} = m_e c_e \frac{Q_n}{h_i A_e} \frac{1}{\frac{m_e c_e}{h_i A_e}} \quad (17)$$

Hence Equation 17 reduces to

$$m_e c_e \frac{T_r}{t_r} = Q_n \tag{18}$$

Input Equation 18 into Equation 16 we get

$$\frac{d\bar{T}_e}{d\bar{t}} = \frac{Q_e(\bar{t})}{Q_n} - \frac{h(\bar{T}_e,P)}{h_i}\frac{T_e - T_{sat}}{T_r} \tag{19}$$

In order to non-dimensionalize temperature from right hand side add and subtract T_i in temperature term. We get

$$\frac{d\bar{T}_e}{d\bar{t}} = \frac{Q_e(\bar{t})}{Q_n} - \frac{h(\bar{T}_e,P)}{h_i}\frac{(T_e - T_i + T_i - T_{sat})}{T_r} \tag{20}$$

We know that

$$T_i - T_{sat,i} = T_r \tag{21}$$

Put Equation 21 into Equation 20 we get

$$\frac{d\bar{T}_e}{d\bar{t}} = \frac{Q_e(\bar{t})}{Q_n} - \frac{1}{h_i}h(\bar{T}_e,P)\left(\frac{T_e - T_i}{T_r} + \frac{T_r}{T_r} + \frac{T_{sat,i} - T_{sat}}{T_r}\right) \tag{22}$$

The final equation is given by

$$\frac{d\bar{T}_e}{d\bar{t}} = \bar{Q}_e(\bar{t}) - \bar{h}(\bar{T}_e,P)(\bar{T}_e + 1 + \frac{T_{sat,i} - T_{sat}}{T_r}) \tag{23}$$

So the temperature of electronics is a function as given by

$$\bar{T}_e = \bar{T}_e(\bar{t}, \bar{Q}_e, \bar{h}(\bar{T}_e, P)) \tag{23.1}$$

But, the temperature in dimensional form is a function as given by

$$T_e = T_e(t, Q_e, h(T_e, P), m_e c_e, A_e) \tag{2.1}$$

It is clearly visible that the number of terms that the temperature of electronics dependent on has been reduced.

2.1.2.2 NON-DIMENSIONALIZING EQUATION 3

Equation 3 deals with the internal energy of the water inside the container. Internal energy is calculated to derive the dryness fraction of water at that particular time. The dryness fraction is in turn used to calculate the pressure inside the system which governs the two phase heat transfer coefficient.

$$\frac{u_{int}}{t_r}\frac{d(\frac{U}{u_{int}})}{d(\frac{t}{t_r})} = h(\Delta T_{sat},P)A_e\Delta T_{sat} - Q_n \tag{24}$$

We normalize Equation 3 using U_{int} (initial internal energy) and t_r to make internal energy and time as dimensionless terms

Equation 24 becomes

$$\frac{m_t u_{int}}{t_r}\frac{d\bar{u}}{d\bar{t}} = h(\Delta T_{sat},P)A_e\Delta T_{sat} - Q_n \tag{25}$$

Where, $\bar{U}$ is dimensionless internal energy. Now move $\frac{U_{int}}{t_r}$ to right hand side of the equation, the equation becomes.

$$\frac{d\bar{u}}{d\bar{t}} = \frac{t_r}{m_t u_{int}}(h(\Delta T_{sat},P)A_e\Delta T_{sat} - Q_n) \tag{26}$$

Use the value of t_r in Equation 26 we get

$$\frac{d\bar{u}}{d\bar{t}} = \frac{1}{m_t u_{int}}\frac{m_e c_e}{h_i A_e}(h(\Delta T_{sat},P)A_e\Delta T_{sat} - Q_n) \tag{27}$$

$$\frac{d\bar{u}}{d\bar{t}} = \frac{m_e c_e}{m_t u_{int}} \bar{h}(\Delta T_{sat}, P)\Delta T_{sat} - \frac{1}{m_t u_{int}} \frac{m_e c_e}{h_i A_e} Q_n \tag{28}$$

$$\frac{d\bar{u}}{d\bar{t}} = \frac{m_e c_e}{m_t u_{int}} \bar{h}(\bar{T}_e, P)(\bar{T}_e + 1 + \frac{T_{sat,i} - T_{sat}}{T_r})T_r - \frac{1}{m_t u_{int}} \frac{m_e c_e}{h_i A_e} Q_n \tag{29}$$

By rearranging the terms we get the non dimensional form of Equation 3

$$\frac{d\bar{u}}{d\bar{t}} = \frac{m_e c_e T_r}{m_t u_{int}} \bar{h}(\bar{T}_e, P)(\bar{T}_e + 1 + \frac{T_{sat,i} - T_{sat}}{T_r}) - \frac{t_r}{m_t u_{int}} Q_n \tag{30}$$

In open system $\frac{d\bar{u}}{d\bar{t}}$ will be zero. So the internal energy is a function as given by

$$\bar{u} = \bar{u}(\bar{t}, \frac{m_e c_e T_r}{m_t u_{int}}, \bar{h}(\bar{T}_e, P), \frac{Q_n}{m_t u_{int}}) \tag{30.1}$$

where, $\frac{m_e c_e T_r}{m_t u_{int}}$ is the mass parameter M. But, the internal energy in dimensional from is a function as given by

$$u = u(t, h(\Delta T_{sat}, P), A_e, Q_n, \Delta T_{sat,i}) \tag{3.1}$$

2.1.2.3 NON-DIMENSIONALIZING EQAUTION 5

Equation 5 is used to calculate the quality of steam inside the tank at a particular instance of time. The steam quality will give the pressure inside the system

$$\frac{dx}{d\bar{t}} = \frac{u_{fg}\left(\frac{du}{d\bar{t}} - \frac{du_f}{d\bar{t}}\right) - (u - u_f)\frac{du_{fg}}{d\bar{t}}}{u_{fg}^2} \tag{31}$$

But we get $\frac{d\bar{u}}{d\bar{t}}$ from Equation 30. In order to use the above function we have to non-dimensionalize the differential term.

$$\frac{dx}{d\bar{t}} = \frac{\bar{u}_{fg}\left(\frac{d\bar{u}}{d\bar{t}} - \frac{d\bar{u}_f}{d\bar{p}}\frac{d\bar{p}}{d\bar{t}}\right) - (\bar{u} - \bar{u}_f)\frac{d\bar{u}_{fg}}{d\bar{p}}\frac{d\bar{p}}{d\bar{t}}}{\bar{u}_{fg}^2} \tag{32}$$

2.1.2.4 NON-DIMENSIONALIZE EQUATION 6

Equation 6 is used to calculate the pressure inside the tank. The pressure is calculated with the help of quality of steam at that particular instance. The Equation 6 is non-dimensionalized by P_{int}, which is the initial pressure of the system at time equal to zero.

$$\frac{d\bar{p}}{d\bar{t}} = \frac{\bar{V}\left(\bar{T}_{sat}\frac{d\bar{m}}{d\bar{t}} + \bar{m}\frac{d\bar{T}_{sat}}{d\bar{p}}\frac{d\bar{p}}{d\bar{t}}\right) - \bar{m}\bar{T}_{sat}\frac{d\bar{V}}{d\bar{t}}}{\bar{V}^2} \tag{33}$$

And hence the pressure term with respect to time is calculated

$$\frac{d\bar{p}}{d\bar{t}} = \frac{\bar{V}\left(\bar{T}_{sat}\frac{d\bar{m}}{d\bar{t}}\right) - \bar{m}\bar{T}_{sat}\frac{d\bar{V}}{d\bar{t}}}{\bar{V}^2 - \bar{m}\bar{V}\frac{d\bar{T}_{sat}}{d\bar{p}}} \tag{34}$$

The pressure calculated from this equation is used in the rest of the equations which govern the two phase heat transfer coefficient, specific internal energy of fluid, vapor and latent heat of vaporization.

The final equations used for solving the transient solution for closed system are 23, 30, 32 and 34. The 8th order Runge-Kutta method is used for solving these equations.

As we can see, in each section the number of terms that the system is dependent has been reduced, it is easy to describe the effects of individual term on the temperature of electronics and properties of coolant tank with respect to time.

2.2 OPEN SYSTEM

In this case the pressure inside the boiling container is constant and hence for a given superheat, the two phase heat transfer coefficient will be a function of superheat only. The system in Fig. 5 becomes open system when the outlet valve and inlet valve are turned on.

The water in the pool is maintained at a constant temperature (T_{sat}), which is saturation temperature of water at that particular pressure, such that the heat supplied to the water is completely utilized in making vapor and not used for specific heating of the fluids. The vapor formed due to phase change is vented out of the system which is compensated by feed water supply into the system. The venting of the vapor formed ensures that the system is at a constant pressure.

The heat added to the pool is due to the heat dissipated by the electronics $Q_e(t)$ which is a function of time. The heat dissipated by electronics at time equal to zero is equal to Q_n, as soon as the time increases from zero the power dissipated will be equal to Q_e. The heat dissipated by electronics is used by electronics to increase its temperature and this temperature (the superheat) governs the two phase convective heat transfer coefficient.

The advantage of an open system is that the pressure inside the system is constant. Hence the properties of the liquid will be a constant (i.e. saturation temperature, specific internal energy of both liquid and vapor, latent heat of vaporization). By keeping the properties constant, the temperature response of the electronics will increase initially and after some time it will reach an equilibrium temperature.

2.2.1 EQUATIONS FOR OPEN SYSTEM

The open system is a special case of closed system, where the venting loop of the closed system is turned on. Hence the pressure inside the system will be equal to that of the surroundings. Therefore the properties of the fluid inside the tank will be constant with respect to time.

In open system the change in internal energy is given by

$$\frac{dU}{dt} = h(\Delta T_{sat}, P) A_e \Delta T_{sat} - Q_n - \dot{m}_g h_g + \dot{m}_f h_f \qquad (35)$$

where,

$$h(\Delta T_{sat}, P) A_e \Delta T_{sat} = Q_e \qquad (36)$$

and

$$Q_n + \dot{m}_g h_g - \dot{m}_f h_f = Q_e \qquad (37)$$

Put equations 36 and 37 in 35 we get

$$\frac{dU}{dt} = 0 \qquad (38)$$

2.3 HYBRID SYSTEM

The hybrid system is a combination of both closed and open system. It has both advantages of closed system and open system. In a closed system when boiling of liquid starts, the pressure inside the system increases rapidly. As the pressure increases the saturation temperature of liquid also increases and in the electronic cooling application

this may lead to failure of electronic components. For example as the pressure raises from 0.2 atm to 0.3 atm the saturation temperature increases from 60.69°C to 69.8°C

To avoid such rapid increase of pressure inside the system, a mechanism is placed such that whenever the pressure inside the boiling container reaches a set pressure, the pressure inside the system will remain constant i.e. the system will become an open system.

This system initially starts as a closed system and the initial pressure of the system is at 0.2atm. When boiling starts the pressure inside the liquid container starts increasing. When pressure reaches 0.22atm the system is set to become an open system. The reason is that due to the pressure increase the saturation temperature increases rapidly, we don't want to increase the specific heating of the fluids inside the tank and hence we have chosen a pressure of 0.22atm so that the saturation temperature change is at a minimal level.

2.3.1 EQUATIONS FOR HYBRID SYSTEM

The equations used for solving the hybrid system are same as equations of closed system. But when the pressure reaches a set value (0.22atm) the system becomes an open system and hence the change in pressure will become zero, so the two phase heat transfer coefficient will change only with respect to wall super heat. The hybrid system is a special case of closed system where in it starts as a closed system, whenever it reaches a maximum pressure allowed it converts to open system, and hence the conditions considered in open system to solve the equation are used.

CHAPTER 3

RESULTS FOR TRANSIENT TEMPERATURE RESPONSE

MODELS

The differential equations are solved using 8th order adaptive Runge-Kutta method of solving differential equations. This is an initial value problem. The pressure for open system was 0.2atm, where as in closed system and hybrid system the initial pressure for the system was 0.2atm. The maximum pressure that can be obtained in a hybrid system is about 0.22atm. The analysis, carried out using the code given in Appendix A, will give the temperature response of the system. The surface considered for modeling is a 145 mesh 8 layer surface, which is the best performer in the pool boiling experiments of Penley and Wirtz [4].

3.1 BENCH-MARK CALCULATION

A benchmark calculation is done to show the open system results will follow the trends of the wall super heat as described by Duluc, Stutz and Lallemand [5], who conducted the boiling of nitrogen on a copper block in transient state. In boiling liquid nitrogen the initial over shoot at ONB is very high as shown in Fig 6. The transient temperature responses of the flat surface which generate these curves have been given by Duluc et al. as shown in Fig 7. Hence the super heat will increase to a maximum temperature and then fall to a stable temperature. The basic values considered for the calculation are $m_e = 0.1 kg, c_e = 385 \frac{J}{KgK}, Q_n = 10w, A_e = 10cm^2$.
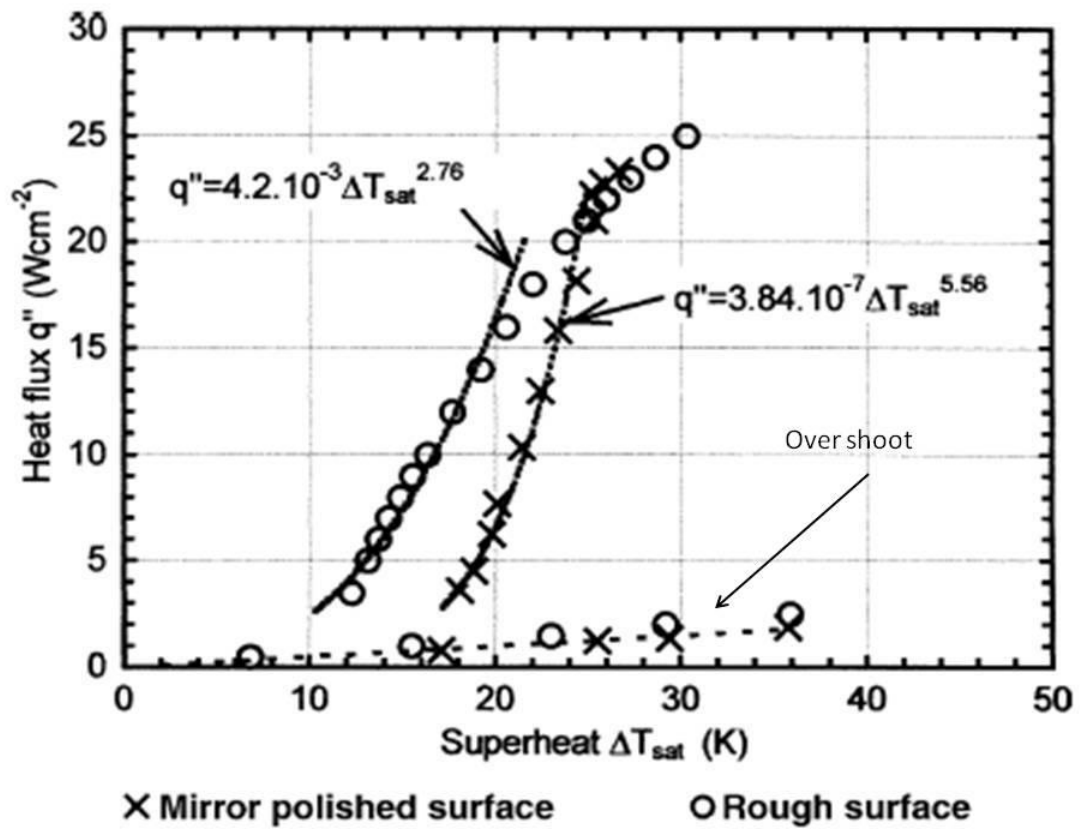
Figure 6: Steady-state boiling curve of liquid Nitrogen on thick flat sample from

Duluc et al. [5].
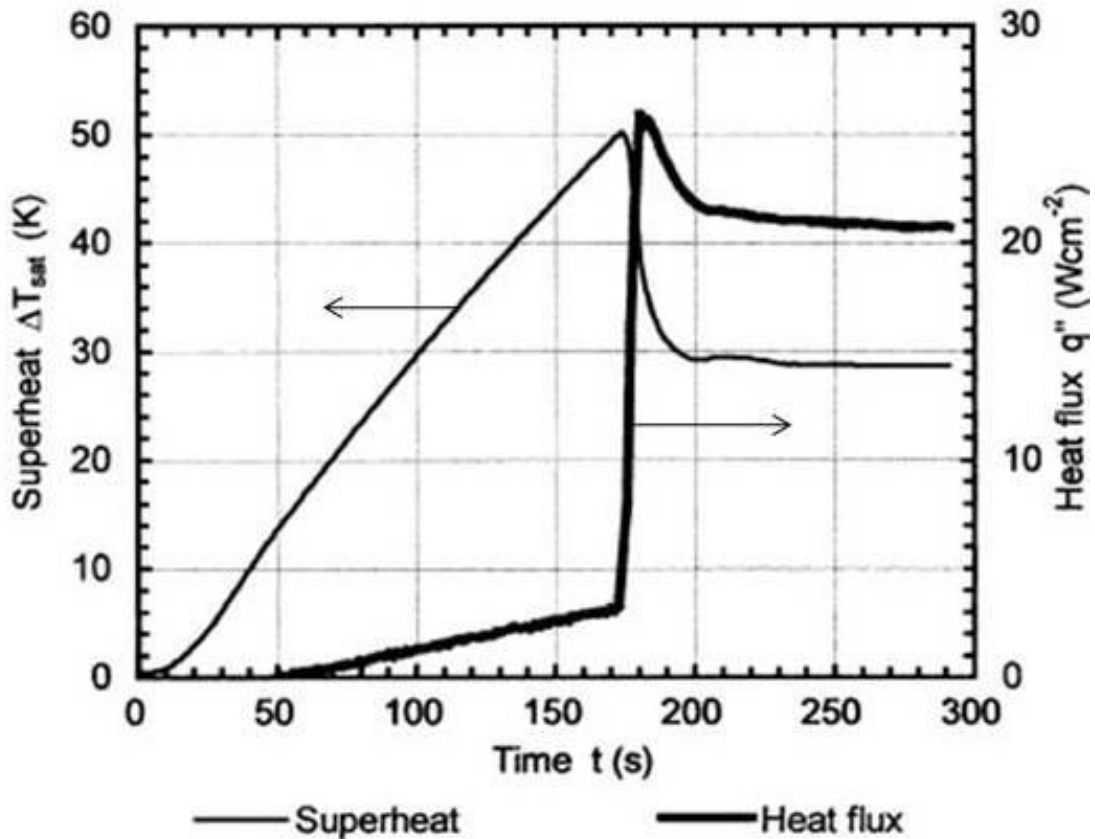
Figure 7: Transient superheat and heat flux versus time for stepwise heat generation of 80% CHF supplied to the cartridge heaters form Duluc et al. [5]

The Fig 7 show transient temperature result obtained by Duluc et al. [5]. As it is evident from Fig 7, that the temperature response shoots up before it settle downs, this is caused may be due to the overshoot issue at ONB.

Figure 8 shows the heat flux vs. superheat for 145 mesh 8 layer at 0.2 atm data. The overshoot in the data is very small so for the calculation purpose we have increased the wall superheat for the ONB.
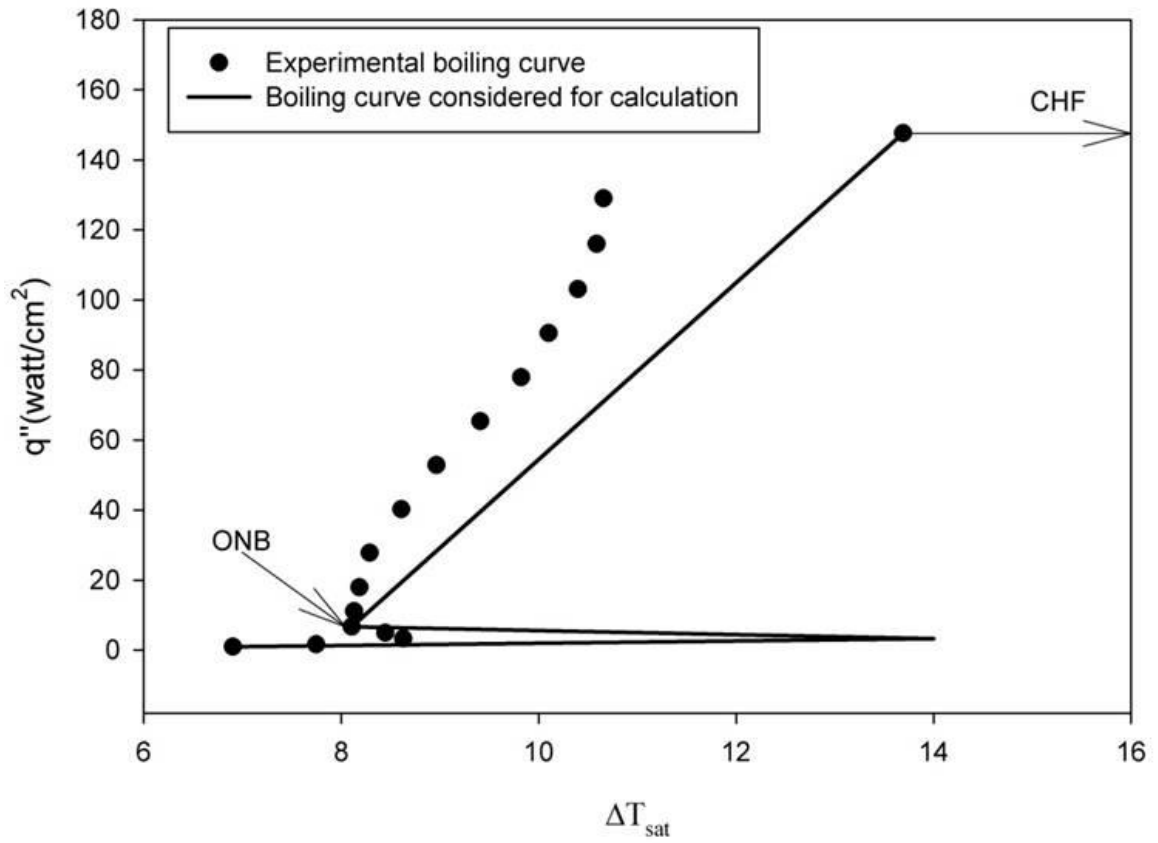
Figure 8: Experimental boiling curve and boiling curve considered for benchmark

calculation
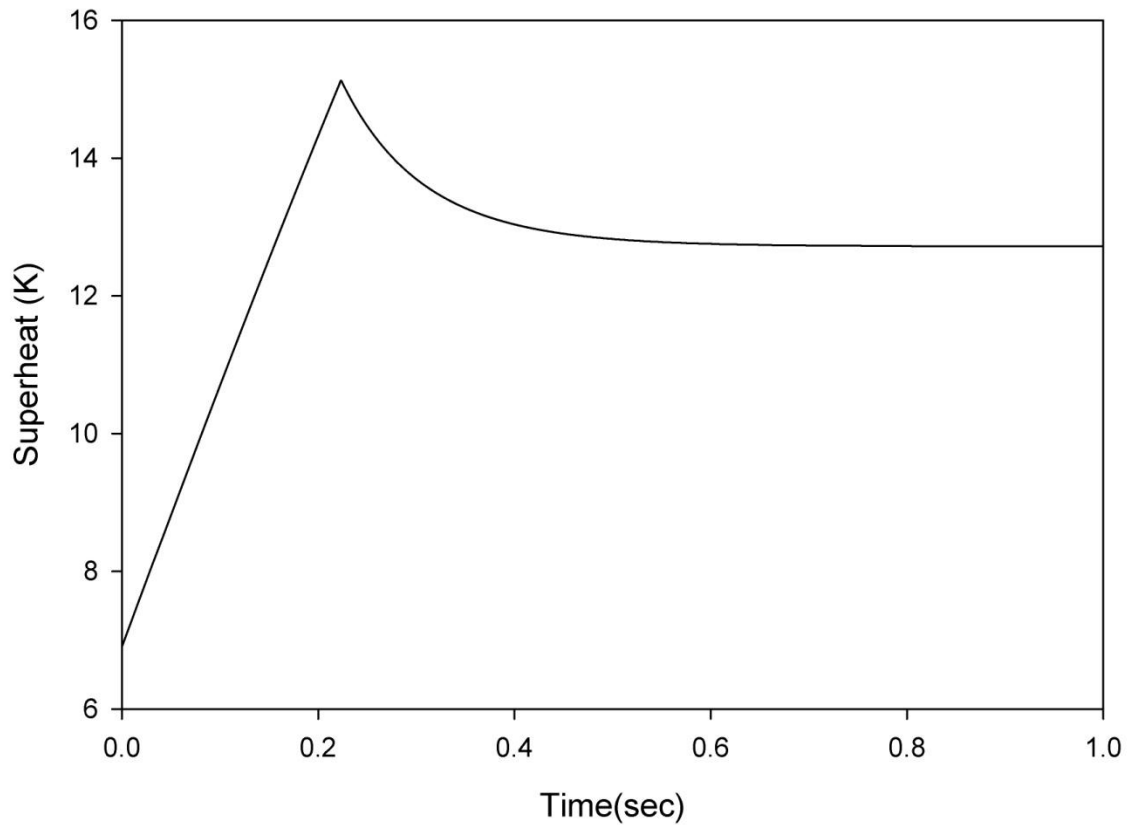
Figure 9 Benchmark Calculation

Figures 7 and 9 shows similar trends of temperature response. Figure 9 is the benchmark calculation done using water at 0.2 atm data to compare with the results obtained by Duluc et al. The observation in this is that, if we consider the overshoot in the boiling curve then there will be an overshoot in the temperature response for the system. So to avoid over shoot in the temperature response, the system must always operate above ONB.
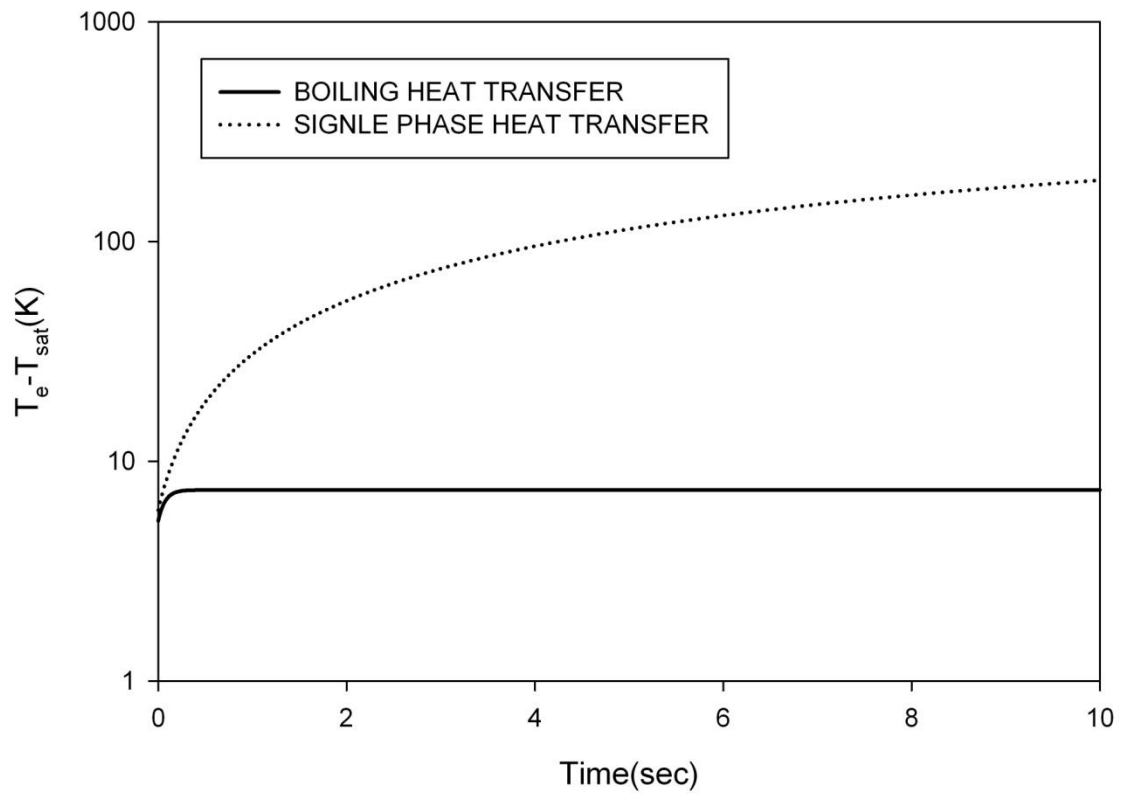
Figure 10 Temperature Response with time for open system boiling, vapor and liquid

phase heat transfer

3.2 COMPARASIONS BETWEEN BOILING OPEN SYSTEM, AND SINGLE PHASE HEAT TRANSFER

Boiling (open system), and liquid phase heat transfer (open system) was analyzed to compare the results. In single phase heat transfer we have considered the convection to be buoyancy driven convection. The liquid is kept in an open system and heated. The correlation used, is taken from [12] is as shown below

$$Nu = \frac{4}{3} 0.616 * (Ra^*)^{\frac{1}{5}} \tag{39}$$

where, Ra^* is modified Rayleigh number given by the equation $\frac{g\beta}{v\alpha k_f} q''L^4$, g is acceleration due to gravity, β is thermal expansion coefficient, ν is kinematic viscosity, α is thermal diffusivity, k_f is fluid thermal conductivity, q'' is heat flux($q'' = \frac{Q_e}{A_e}$), L is characteristic length($L = \sqrt{A_e}$), Nu is Nusselt number given by $\frac{hL}{k_f}$, h is heat transfer coefficient.

The heat transfer coefficient associated with liquid phase heat transfer is 2690 watt/m^{2}K. The temperature of the system is kept at saturation temperature of water at 0.2atm. The temperature response of the electronics with respect to time is given in Fig 10. The temperature obtained in liquid phase heat transfer is higher since the heat transfer coefficient in liquid phase is lesser than compared to other systems.

The main observation is that the temperature response obtained using single phase heat transfer is 60 folds higher than the temperature response obtained using two phase heat transfer.

3.3 THERMAL RESPONSE OF OPEN, CLOSED AND HYBRID SYSTEMS

For the calculation the values considered were same for all the systems as given in Table 1, 2 and 3. The analysis, carried out using the code given in Appendix A, will give the temperature response, mass of liquid evaporated for open system; temperature response, quality, pressure for closed system; and temperature response, and pressure for the hybrid system. The initial boundary condition for the problem is that the temperature of wall is equal to the ONB at 0.2 atm pressure, the quality of steam for closed system is 47.5%, the maximum pressure when the valves open in a hybrid system is 0.22atm

TABLES

Properties(symbols)	Values
Mass of Electronics(m_e)	0.1 kg
Specific heat of Electronics(c_e)	$385\frac{J}{KgK}$
Area of Electronics(A_e)	$10cm^2$
Nominal Heat Generation by Electronics(Q_n)	10 watt
Heat generation (Q_e)	$100*Q_n$
Saturation Pressure(P_{sat})	0.2atm
Saturation temperature(T_{Sat})	$60\ ^0c$
ONB($T_{ONB} - T_{Sat}$) [initial temperature]	$5.5\ ^0c$

Table 1 Values considered for open system

Properties(symbols)	Values
Mass of Electronics(m_e)	0.1 kg
Specific heat of Electronics(c_e)	$385\frac{J}{KgK}$
Area of Electronics(A_e)	$10cm^2$
Nominal Heat Generation by Electronics(Q_n)	10 watt
Heat generation (Q_e)	$100*Q_n$
Starting Pressure(P_i)	0.2atm
Saturation temperature at P_i (T_{Sat})	$60\,^0$c
ONB($T_{ONB} - T_{Sat}$) [initial temperature)	$5.5\,^0$c
Dryness fraction(x)	50%
Initial internal energy of system(U_{int})	616 J

Table 2 Values considered for closed system

Properties(symbols)	Values
Mass of Electronics(m_e)	0.1 kg
Specific heat of Electronics(c_e)	$385\frac{J}{KgK}$
Area of Electronics(A_e)	$10cm^2$
Nominal Heat Generation by Electronics(Q_n)	10 watt
Heat generation (Q_e)	$100*Q_n$
Starting Pressure(P_i)	0.2atm
Saturation temperature at P_i (T_{Sat})	$60\,^0$c
ONB($T_{ONB} - T_{Sat}$) [initial temperature)	$5.5\,^0$c
Dryness fraction(x)	50%
Initial internal energy of system(U_{int})	616 J
Pressure at which open system changes to closed system	0.22 atm

Table 3 Values considered for hybrid system

Figure 11 is a graph between the rise in temperature vs. time. The graph has the results for all the three systems. The graph was obtained for a pulse power as shown at the right side top corner of Figure 11. During the cool down phase the settling temperature for hybrid and closed systems are not the same as that of open system because of the energy that is given to the coolant is not vented out of the system.
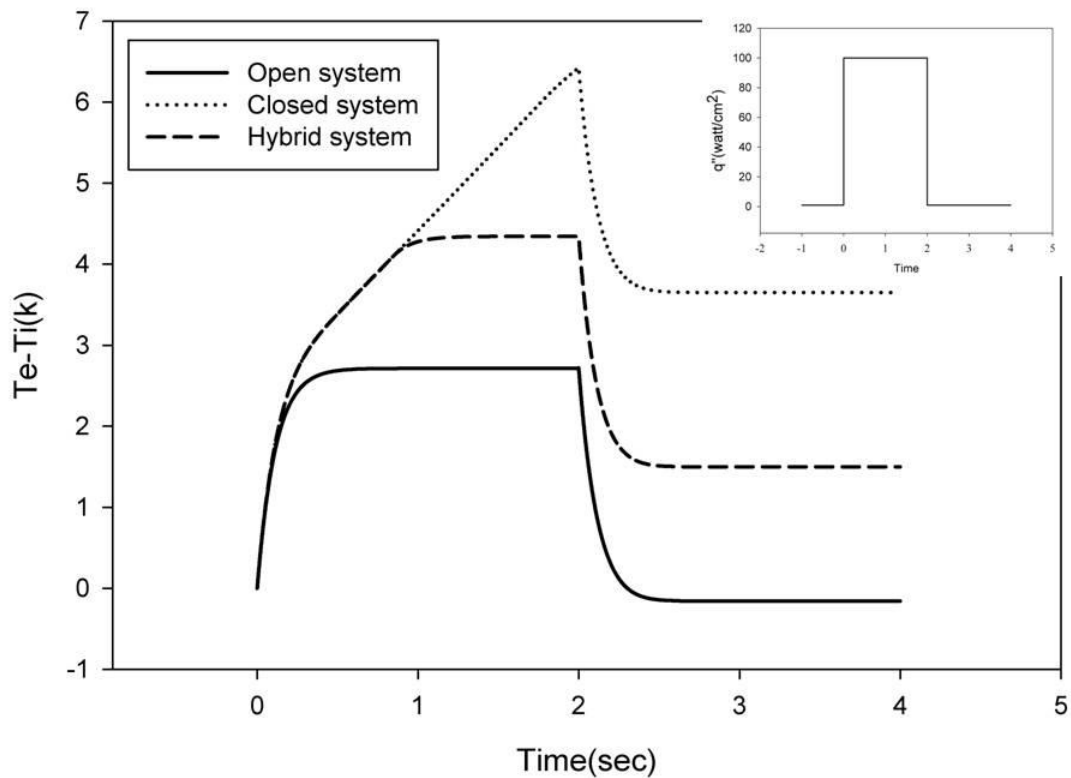
Figure 11: Temperature vs. time graph in Dimensional terms for all systems

3.3.1 OPEN SYSTEM

The temperature response of electronics package in an open system, when the heat generated by the electronics package changes from nominal value to a maximum value as shown in Fig 11. As the heat generated by electronics package changes, the temperature of electronics increases and after some time the temperature reaches a settling value as shown in the Figure. During the cool down phase the temperature reaches the initial temperature value, at which the system starts.
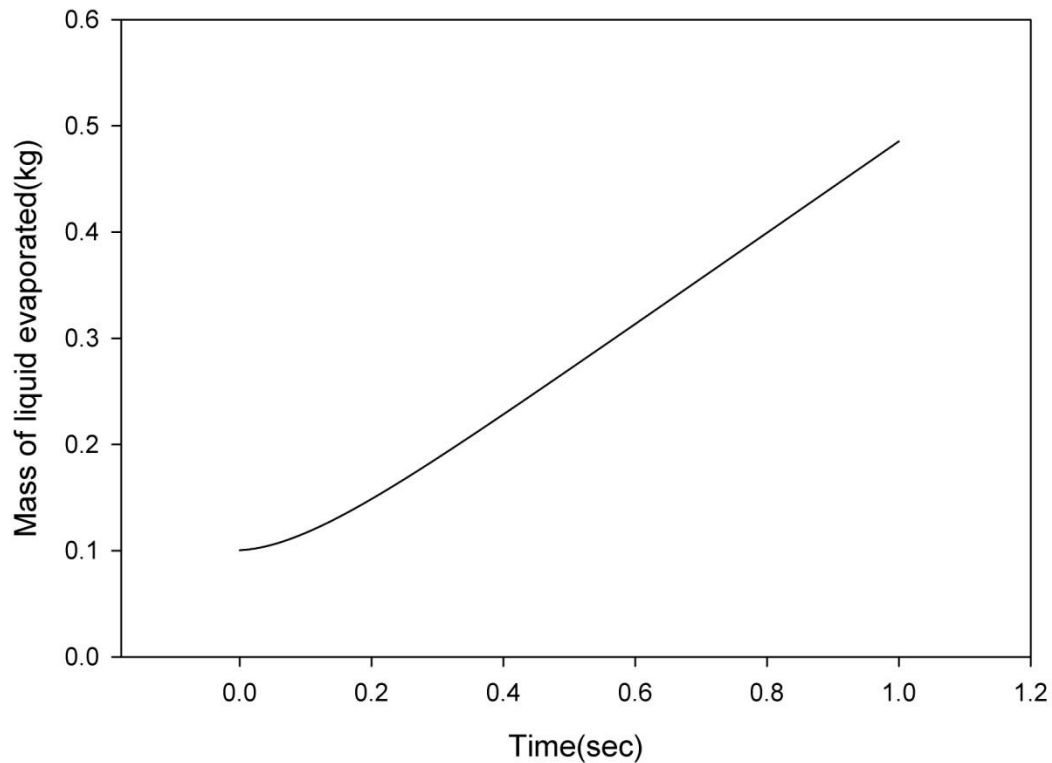
Figure 12 Mass of liquid evaporated vs. Time for open system

The mass of liquid evaporated with respect to time is shown in Fig 12. At the starting point the curve is not linear as the heat generated is used in specific heating of the electronics.

3.3.2 CLOSED SYSTEM

The temperature response of the electronics in the closed system, when the heat generation change from nominal value to a maximum value, is shown in Fig 11. As the heat dissipation rate changes the temperature of electronics increases rapidly, the temperature response is non linear as shown in the graph. This occurs due to the pressure increases inside the container, the heat dissipated by electronics will be used for specific
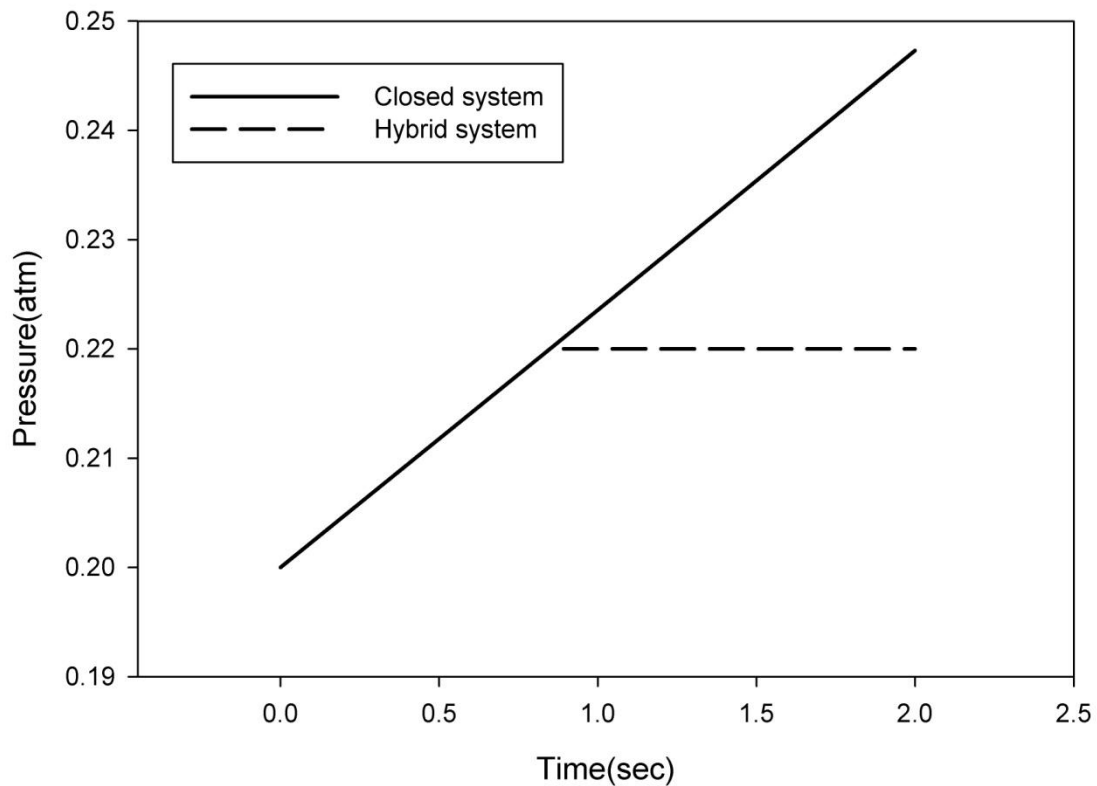
Figure 13 Pressure response with time for Closed and Hybrid systems

heating of liquid and hence the temperature of electronics increases. The pressure response inside the system is shown in Fig 13.

The Fig 14 shows that the temperature of electronics in closed system is always greater than that of the saturation temperature of liquid.

The closed and hybrid systems cool down to the high temperature because the energy given to the systems is not vented out as in the case of open system. To bring back the temperature of closed and hybrid system to the temperature that is obtained by open system, Q_n must have a higher value.
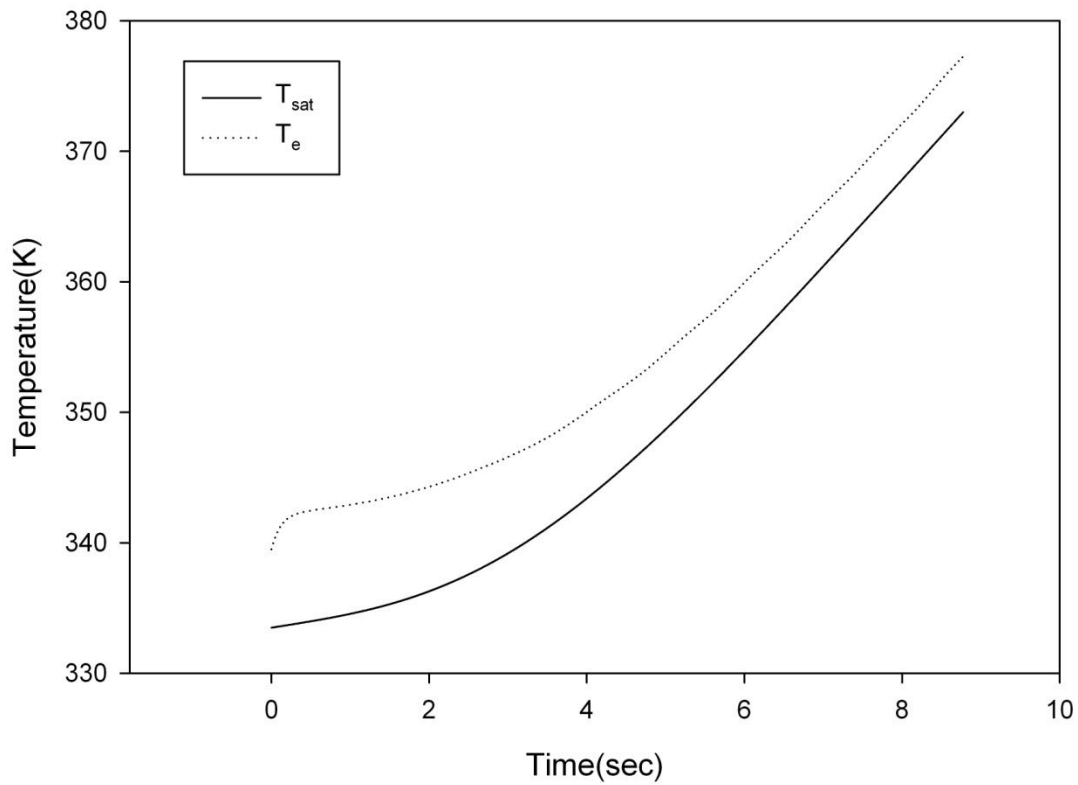
Figure 14 Variation of saturation temperature, temperature of electronics, for closed

system

3.3.3 HYBRID SYSTEM

The temperature response of electronics in hybrid system, when the heat

generation change from nominal value to maximum value is shown in Fig 11. As the

heat generation rate changes the temperature of electronics increases rapidly and after

some time the temperature of electronics increases at a different rate as shown in the

graph. When the pressure reaches the value of 0.22 bar the system converts from closed

system to open system. When this change occurs the pressure inside the system will be
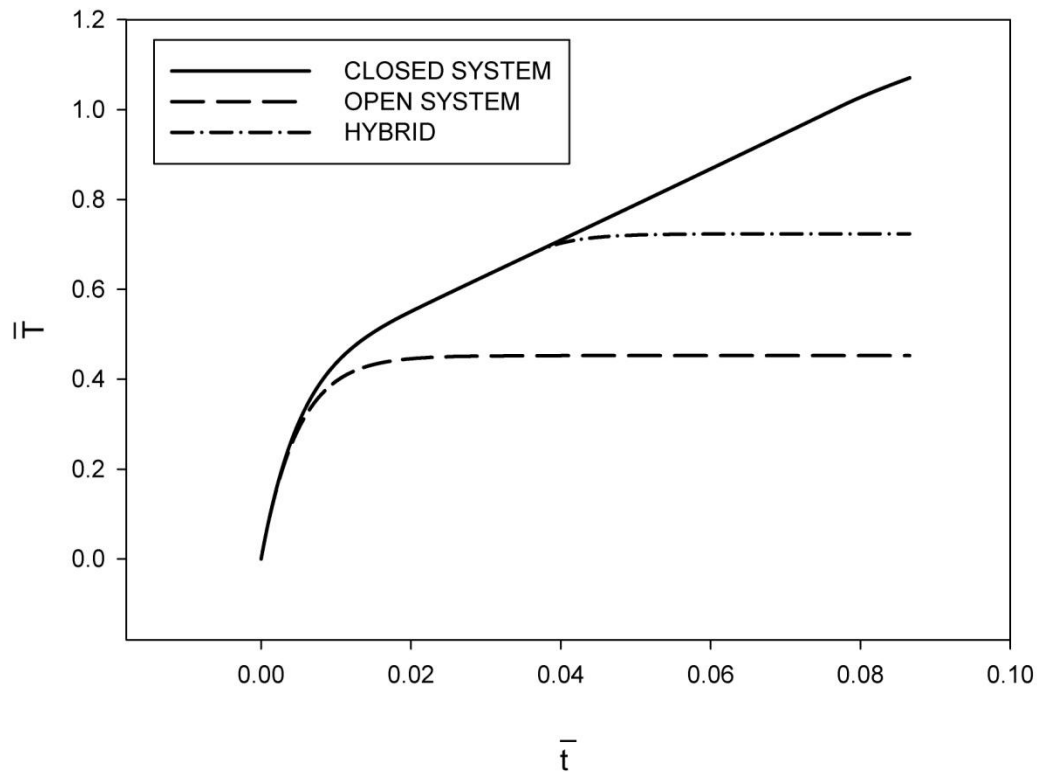
Figure 15 Temperature vs. Time, dimensionless plot

constant and hence the temperature will increase and reach a settling value. The pressure response inside the system is shown in Fig 13.

3.4 PARAMETRIC STUDY FOR OPEN, CLOSED AND HYBRID SYSTEM

The non-dimensional results for temperature response of open system, closed system and hybrid systems are shown in Fig 15. To do a parametric analysis we have considered a changed in m_e and $\bar{Q}_e$ which are shown in Table 4 and 5. The settling time is defined as the time taken to reach 90% of the settling temperature in open system and in closed system there is no settling time as the temperature keeps on increasing and the

settling temperature($T_{e,s}$) is the value at which the temperature of the system does not changes with time.

Property	I	II	III
Mass of electronics(m_e)(kg)	0.1	0.3	1

Table 4 Different Values of mass of electronics (m_e)considered to show the Effect of it on temperature of Electronics

Property	I	II	III
Heat Generation ($\bar{Q}_e$)	20	30	100

Table 5 Different Values of Heat Generation considered to show the Effect of it on temperature of Electronics

3.4.1 EFFECTS OF THERMAL MASS OF ELECTRONICS ($m_e c_e$) ON THE TEMPERATURE RESPONSE FOR VARIOUS SYSTEMS

In this section the effect of thermal mass of electronics ($m_e c_e$) on the temperature response is investigated. As the thermal mass of electronics ($m_e c_e$) changes the response time for the electronic temperature also changes. The thermal mass of electronics ($m_e c_e$) considered for this analysis were .1kg, .3kg, and 1kg, which are also given in table 4. The effects of thermal mass of electronics ($m_e c_e$) on temperature of electronics for different systems are shown below.
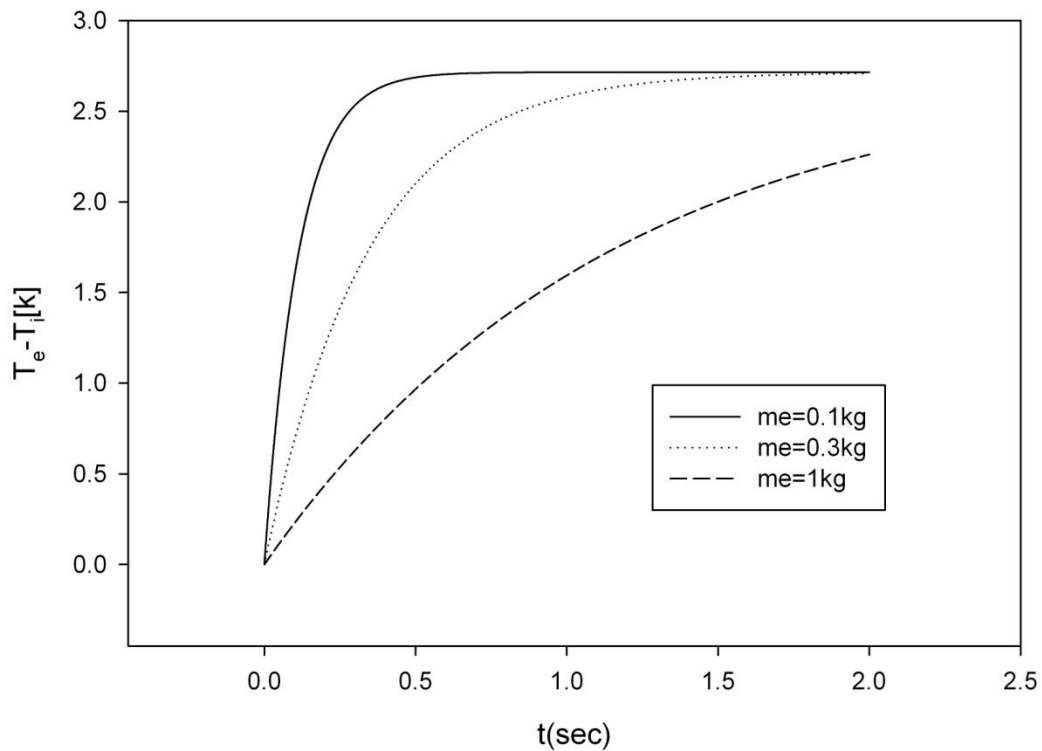
Figure 16 Temperature vs. time for open system for different values of thermal mass of

electronics ($m_e c_e$)

3.4.1.1 EFFECTS OF THERMAL MASS OF ELECTRONICS ($m_e c_e$) ON OPEN SYSTEM

In open system the change of thermal mass of electronics ($m_e c_e$) will change the response time to reach stable temperature. The stable temperature for all the three cases are same but the time taken to reach the stable temperature is different. The response time is different because the heat dissipated by electronics will be used in specific heating of the electronics, as the mass of the electronics changes, so does the thermal mass parameter. But the stable temperature is same irrespective of thermal mass of electronics ($m_e c_e$). Fig 16 shows the plot between temperature and time.
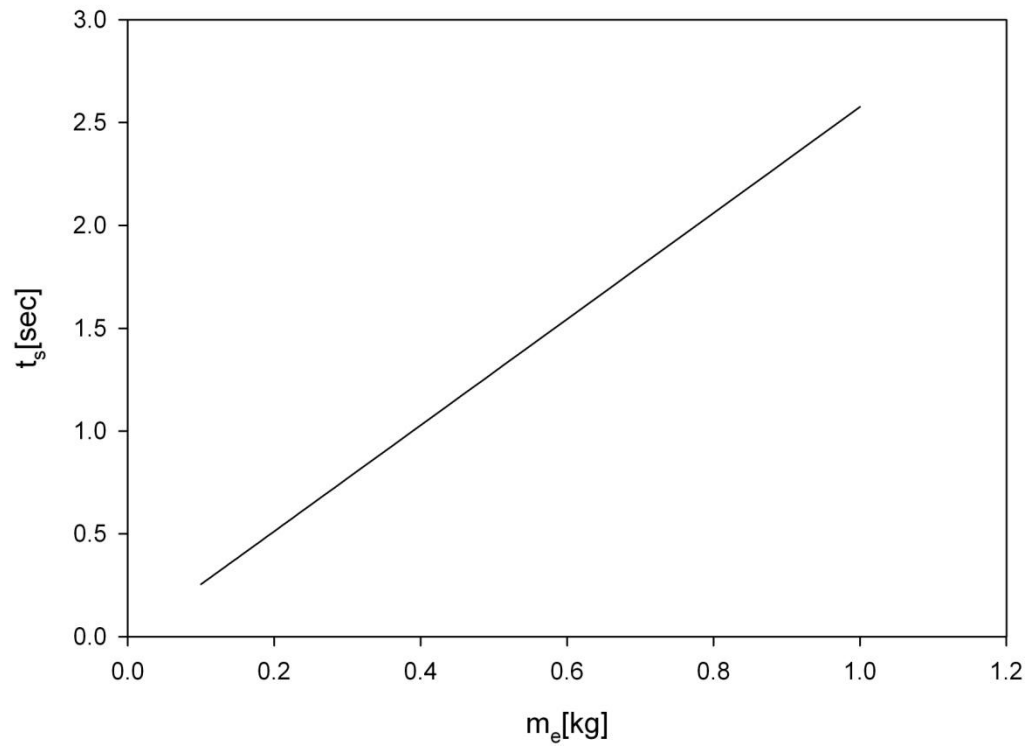
Figure 17 Settling time vs. thermal mass of electronics ($m_e c_e$) for open system

Figure 17 describes the settling time response with respect to the different mass of electronics for an open system. As the thermal mass of electronics ($m_e c_e$) increases the settling time of the electronics in the open system will also increases.
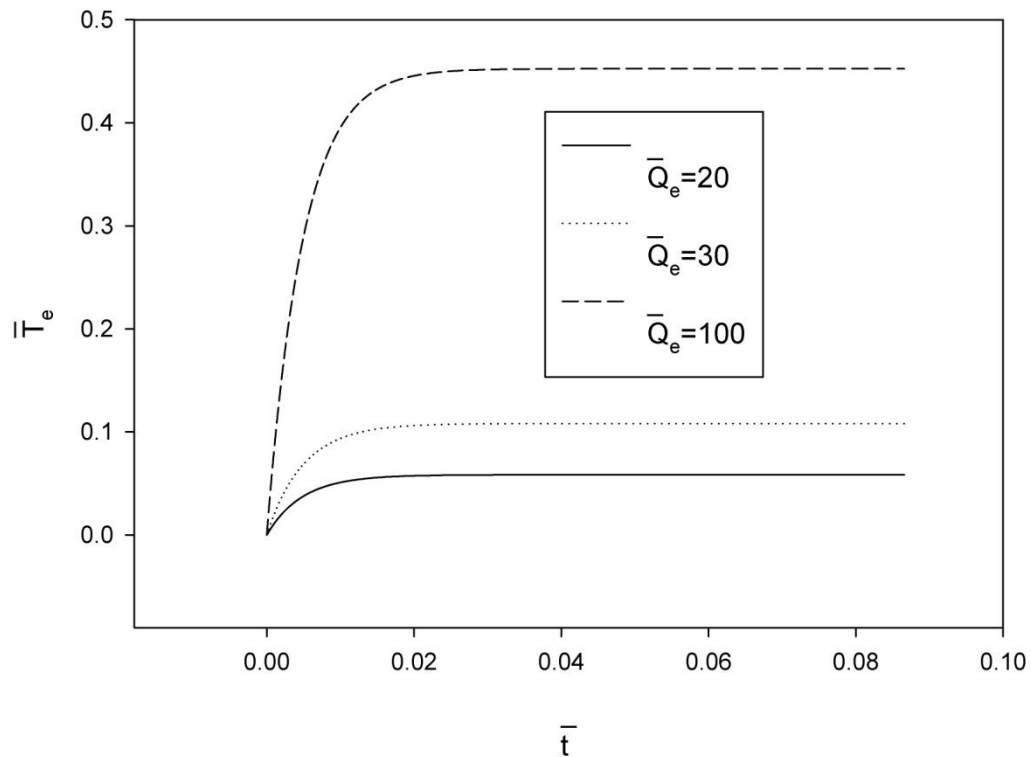
Figure 18 Dimensionless temperature vs. time for open system with different heat

generation

3.4.2 EFFECTS OF HEAT GENERATION ON THE TEMPERATURE RESPONSE FOR VARIOUS SYSTEMS

In this section the effect of heat generated on the temperature response is investigated. As the heat dissipation changes the electronics temperature response with time also changes. The different Non dimensional power $(\bar{Q}_e)$ considered for these analyses were 20, 30, 100, which are also given in table 5. The effects of Heat generation on temperature of electronics for different systems are shown below.
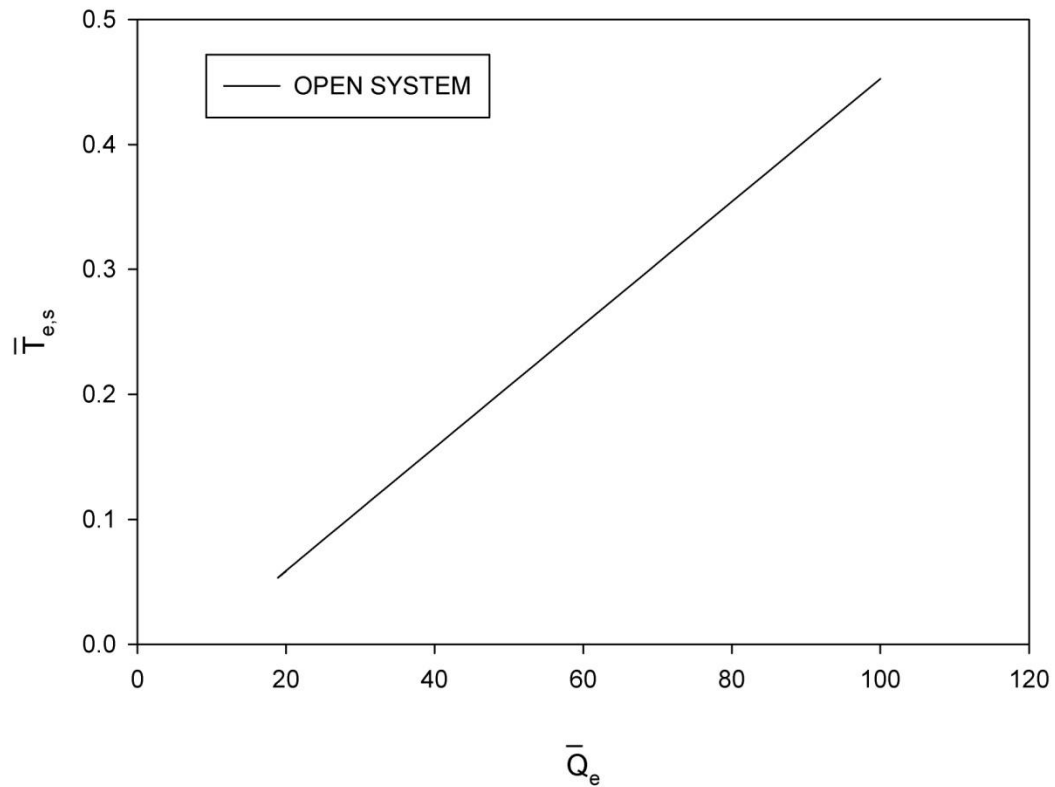
**Figure 19** Stable temperature vs. dimensional power generated by electronics for open system

3.4.2.1 EFFECTS OF HEAT GENERATION ON OPEN SYSTEM

In open system the change of heat generation will change the stable temperature. As the heat generation increases, the two phase heat transfer coefficient will be higher for higher heat generation rate and the wall super heat corresponding to the heat transfer coefficient is also higher for the boiling curves, and hence to the stable value for electronics with different heat generation rates are different. The graph is shown in Fig 18.
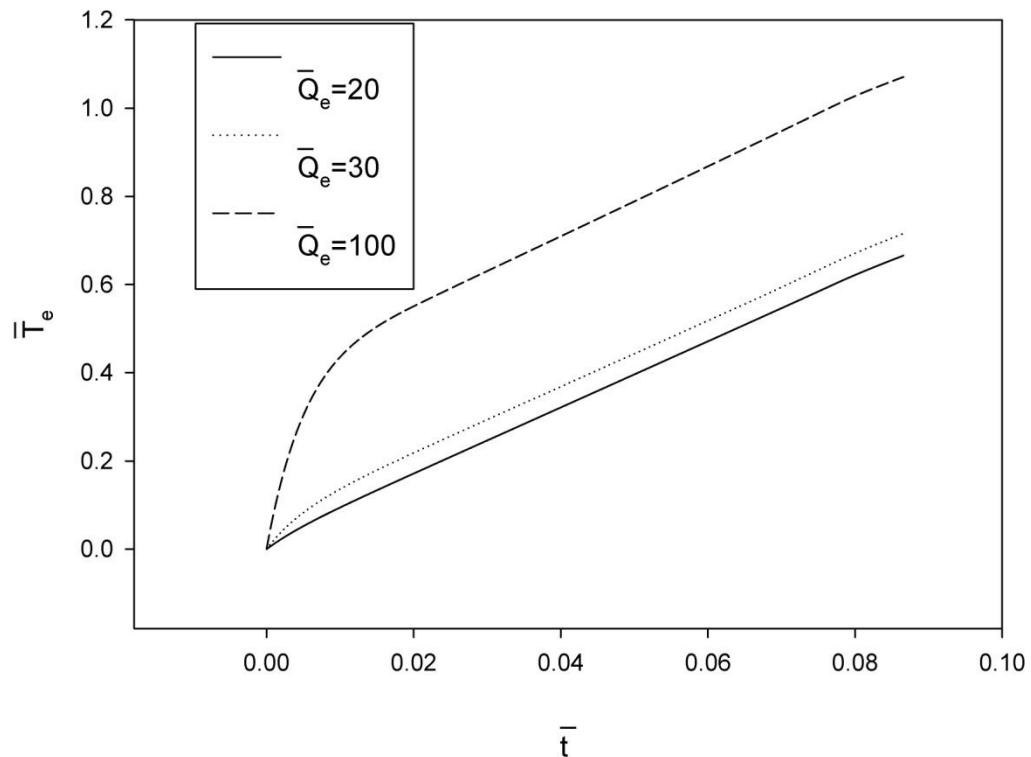
Figure 20 Dimensionless temperature vs. time for closed system with varying

heat generation

Figure 19 describes the settling temperature response with respect to the power

dissipated by electronics at various levels. As the power dissipation increases the settling

temperature of the electronics will also increase for an open system.

3.4.2.2 EFFECTS OF HEAT GENERATION ON CLOSED SYSTEM

In closed system the change of heat generation will change the temperature

response with time. As the heat generation increases, the two phase heat transfer

coefficient will be higher for higher heat generation rate and also the pressure increases

rapidly as the heat dissipated will be higher, this will change the wall super heat

corresponding to the heat transfer coefficient and pressure from the boiling curve at that

instance of time which is higher for higher heat generation rate. The graph is shown in Fig 20.

CHAPTER 4

CONCLUSIONS

All conditions were considered in solving the electronics dissipating high power transients. The main conclusions were found to be

- The system must operate over the ONB, because if the system is operated below the ONB, there will be unwanted temperature excursions.

- The single phase heat transfer and two phase heat transfer were compared, for same power output two phase heat transfer outperformed single phase heat transfer. The temperature obtained by single phase heat transfer is 60 folds that of two phase heat transfer

- For a 100 fold increase in the heat generation, the temperature change with these system is very low, indicating an effective cooling system

- During the cool down phase, different systems reaches to different temperatures because the final pressures inside the coolant container are different.

- For open system, the final temperatures obtained when the mass of electronics (m_e) is changed are same but the settling time is different.

- For closed system and hybrid, the final temperature obtained when the mass parameter changes are different.

The optimum pool condition is a hybrid system where initially it starts as a closed system but after reaching an optimum pressure, the pressure remains constant by opening

the system. The hybrid system has all the advantages of both the closed and open system.

The hybrid system will minimize the heat lost to surroundings as in the closed system and

also the temperature of electronics will reach a stable value as that of an open system.

CHAPTER 5

FUTURE WORK

As this work is based on simulation results, the future work based on this work is to construct an experimental setup where we can control the pressure inside the liquid tank. The pressure inside the tank is controlled by inlet and outlet valves. For hybrid system, a mechanism is setup such that whenever the pressure inside the system reaches an optimal pressure, the valves will be opened.

In this work we have studied the effects of the temperature when the heat dissipation by electronics increases in a step fashion. We haven't considered the effects of the temperature when the heat dissipation by electronics decreases in a step fashion and when the heat dissipation by electronics first increases and then decreases in a step fashion. The future work must also include the step increase and then step decease in heat dissipated by electronics

In this work we have considered only sub-atmospheric pool boiling. But, we haven't considered the condition when the pressure inside the system is greater than the atmospheric pressure. This condition is not applicable for electronics but can be used for many other applications.

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APPENDIX

MATHCAD PROGRAM

NOMENCLATURE FOR CODE

A   Area [m^2]

c   Specific heat[kJ/kg*K]

C_f Compression Factor

d   Diameter[mm]

D_h Pore hydraulic diameter

g   Acceleration due to gravity[m/sec^2]

h   Heat Transfer coefficient[watt/m^2K]

h_{cor} Dimension less heat transfer coefficient

K   Thermal Conductivity[watt/mK]

K_{Dmin} Active Nucleation side density parameter

m   Mass[kg]

M   Mass parameter (equation 30.1)

n   Number of moles

Nu   Nusselt Number

P   Pressure[atm]

q''   Heat flux[watt/m^2]

Q_c Nominal heat output[watt]

Q_e Dimension less power

R   Universal gas constant

Ra*   Modified Rayleigh number

T   Temperature[K]

u	Internal energy[kJ/kg]
U	Total Internal Energy[kJ]
v	Specific volume[m^3/kg]
V	Volume[m^3]
ΔT_{sat}	Wall super heat[k]

Greek symbols

α	Thermal diffusivity
β	Specific surface area[cm^2/cm^3]
δ	Laminate thickness[mm]
ε	porosity
ν	Kinematic viscosity[m^2/s]
ρ	Density[kg/m^3]
σ	Surface tension[N/m]

Subscripts

e	Electronics
f	fluid
g	gas
i	initial
s	settling
t	total

Reference:C:\Documents and Settings\bganja\Desktop\Pen Drive\mathcad 13 files\water and vapour properties.xmcd(R)

▼ Water properties

water properties

$$kJ := 1000J \qquad C \equiv K \qquad Ko := 273.16K \qquad kPa := 1000 \cdot Pa$$

Water Properties, All temps in Kelvin $\qquad Tsat_{ck} := 100K + Ko$

$$\sigma(T) := 235.8 \cdot \left(1 - \frac{T}{647.2K}\right)^{1.256} \cdot \left[1 - .625 \cdot \left(1 - \frac{T}{647.2K}\right)\right] \cdot \frac{dyne}{cm} \quad \sigma(65C + Ko) = 0.065 \cdot \frac{N}{m}$$

$$Tsat_{ck} = 373.16\,K$$

$$Tdat := \begin{pmatrix} 308.15 \\ 313.15 \\ 318.15 \\ 323.15 \\ 328.15 \\ 333.15 \\ 338.15 \\ 343.15 \\ 348.15 \\ 353.15 \\ 358.15 \\ 363.15 \\ 368.15 \\ 373.15 \end{pmatrix} K \quad hfgdat := \begin{pmatrix} 2418.6 \\ 2406.8 \\ 2394.8 \\ 2382.8 \\ 2370.7 \\ 2358.5 \\ 2346.2 \\ 2333.8 \\ 2321.4 \\ 2308.8 \\ 2296 \\ 2283.2 \\ 2270.2 \\ 2257 \end{pmatrix} \frac{kJ}{kg} \quad Prfdat := \begin{pmatrix} 5.88 \\ 3.79 \\ 2.69 \\ 2.0 \\ 1.75 \end{pmatrix}$$

$$\begin{pmatrix} 383.15 \\ 393.15 \\ 403.15 \\ 430 \\ 460 \\ 490 \\ 520 \end{pmatrix} \qquad \begin{pmatrix} 2230.2 \\ 2202.6 \\ 2174.2 \\ 2092.8 \\ 1990.4 \\ 1871.5 \\ 1731.3 \end{pmatrix}$$

$$g\beta o\nu 2dat := \begin{pmatrix} 366 \\ 1250 \\ 2980 \\ 6250 \\ 8500 \end{pmatrix} \cdot 10^{7} \frac{m^{-3}}{K} \qquad \alpha dat := \begin{pmatrix} 1.46 \\ 1.54 \\ 1.61 \\ 1.66 \\ 1.68 \end{pmatrix} \cdot 10^{-7} \frac{m^{2}}{sec}$$

$$Tdat2 := \begin{pmatrix} 300 \\ 313.15 \\ 320 \\ 333.15 \\ 340 \\ 353.15 \\ 360 \\ 373.15 \\ 400 \\ 430 \\ 460 \\ 520 \end{pmatrix} K \qquad cpfdat := \begin{pmatrix} 4177 \\ 4178 \\ 4176 \\ 4184 \\ 4187 \\ 4196 \\ 4204 \\ 4220 \\ 4240 \\ 4280 \\ 4450 \\ 4600 \end{pmatrix} \frac{J}{kg \cdot K} \qquad kfdat := \begin{pmatrix} 0.608 \\ 0.628 \\ 0.637 \\ 0.651 \\ 0.659 \\ 0.668 \\ 0.674 \\ 0.681 \\ 0.6857 \\ 0.6833 \\ 0.6713 \\ 0.6183 \end{pmatrix} \frac{W}{m \cdot K}$$

$$\text{hfg}(T) := \text{linterp}(\text{Tdat}, \text{hfgdat}, T) \qquad \text{hfg}\left(\text{Tsat}_{ck}\right) = 2.257 \times 10^6 \cdot \frac{J}{kg}$$

$$\text{cp}_f(T) := \text{linterp}(\text{Tdat2}, \text{cpfdat}, T) \qquad \text{cp}_f\left(\text{Tsat}_{ck}\right) = 4.22 \times 10^3 \frac{m^2}{K \cdot s^2}$$

$$\text{k}_f(T) := \text{linterp}(\text{Tdat2}, \text{kfdat}, T) \qquad \text{k}_f\left(\text{Tsat}_{ck}\right) = 0.681 \cdot \frac{W}{m \cdot K}$$

$$\mu_f(TK) := .00179 \text{Pa} \cdot \text{sec} \cdot e^{-1.74 - 5.306 \cdot \frac{273.16K}{TK} + 7.003 \cdot \left(\frac{273.16K}{TK}\right)^2} \qquad \mu_f\left(\text{Tsat}_{ck}\right) = 2.755 \times 10^{-4} \cdot \text{Pa} \cdot \text{sec}$$

Steam Properties

$$
\text{Tsatdat} := \begin{pmatrix} 308.15 \\ 313.15 \\ 318.15 \\ 323.15 \\ 328.15 \\ 333.15 \\ 338.15 \\ 343.15 \\ 348.15 \\ 353.15 \\ 358.15 \\ 363.15 \\ 368.15 \\ 373.15 \\ 383.15 \\ 393.15 \\ 403.15 \\ 430 \\ 460 \\ 490 \\ 520 \end{pmatrix} K
\qquad
\text{rhodatf} := \begin{pmatrix} 994.0357853 \\ 992.0634921 \\ 990.0990099 \\ 988.1422925 \\ 985.2216749 \\ 983.2841691 \\ 980.3921569 \\ 977.5171065 \\ 974.6588694 \\ 971.8172983 \\ 968.9922481 \\ 965.2509653 \\ 961.5384615 \\ 957.8544061 \\ 950.5703422 \\ 943.3962264 \\ 934.57944 \\ 910.3 \\ 879.4 \\ 844.3 \\ 803.8 \end{pmatrix} \frac{\text{kg}}{\text{m}^3}
\qquad
\text{Pvapdat} := \begin{pmatrix} 0.054508475 \\ 0.071506053 \\ 0.092910412 \\ 0.119612591 \\ 0.152639225 \\ 0.193123487 \\ 0.242615012 \\ 0.302082324 \\ 0.373656174 \\ 0.458983051 \\ 0.560096852 \\ 0.679225182 \\ 0.818886199 \\ 0.981113801 \\ 1.387893462 \\ 1.92251816 \\ 2.615981 \\ 5.63533 \\ 11.56674069 \\ 21.56427338 \\ 37.23661485 \end{pmatrix} \text{atm}
\qquad
\text{rhodatg} := \begin{pmatrix} 0.039651071 \\ 0.051229508 \\ 0.065530799 \\ 0.08312552 \\ 0.104504128 \\ 0.1303611 \\ 0.161368404 \\ 0.198333994 \\ 0.242072137 \\ 0.293513355 \\ 0.353606789 \\ 0.423549343 \\ 0.504540868 \\ 0.597728631 \\ 0.826446281 \\ 1.121201928 \\ 1.49588631 \\ 3.02 \\ 5.975 \\ 10.95 \\ 18.9 \end{pmatrix} \cdot \frac{\text{kg}}{\text{m}^3}
$$

$$\rho_f(T) := \text{linterp}(\text{Tsatdat}, \text{rhodatf}, T) \qquad \rho_f\!\left(\text{Tsat}_{ck}\right) = 957.847\,\frac{kg}{m^3}$$

$$\rho_g(T) := \text{linterp}(\text{Tsatdat}, \text{rhodatg}, T) \qquad \rho_g\!\left(\text{Tsat}_{ck}\right) = 0.598\,\frac{kg}{m^3}$$

$$\text{Psat}(T) := \text{linterp}(\text{Tsatdat}, \text{Pvapdat}, T) \qquad \text{Psat}(70C + Ko) = 3.062 \times 10^4\,Pa$$

$$\text{Tsat}(P) := \text{linterp}(\text{Pvapdat}, \text{Tsatdat}, P) \qquad \text{Tsat}\!\left(\text{Psat}\!\left(\text{Tsat}_{ck}\right)\right) - Ko = 100\,K$$

$$\text{Tsat}(1\text{atm}) = 373.614\,K$$

$$\text{Rvap} := .4615\,\frac{kJ}{kg\cdot K} \qquad \rho_g(P, T) := \frac{P}{\text{Rvap}\cdot T} \qquad \rho_g\!\left(\text{Psat}\!\left(\text{Tsat}_{ck}\right), \text{Tsat}_{ck}\right) = \blacksquare\,^{\blacksquare}$$

$$\text{Tsat}(99.538\text{kPa}) - 273.15K = 100.031 \quad 367 - 273 = 94$$

$$\text{Tsat}(20.1\text{kPa}) - 273.15K = 60.53\,K$$

🔺 Water properties

🔽 Penley correlation

Penley correlation

$$\text{ja}\rho(\text{dtsat}, \text{pressure}) := \frac{\text{dtsat}\cdot cp_f(\text{Tsat}(\text{pressure}))\cdot \rho_f(\text{Tsat}(\text{pressure}))}{\text{hfg}(\text{Tsat}(\text{pressure}))\cdot \rho_g(\text{Tsat}(\text{pressure}))} \qquad \text{ja}\rho(10K, .2\text{atm}) = 129.579$$

$$\text{dmin(dtsat, pressure)} := \cfrac{4\,\sigma(\text{Tsat(pressure)})}{\cfrac{\text{Psat(Tsat(pressure)+dtsat)}-\text{pressure}}{m}}$$

$$\text{db(pressure)} := 1.5\cdot10^{-4}\cdot\sqrt{\cfrac{\sigma(\text{Tsat(pressure)})}{g\cdot\left(\rho_f(\text{Tsat(pressure)})-\rho_g(\text{Tsat(pressure)})\right)}}\cdot\left(\cfrac{\text{Tsat(pressure)}\cdot cp_f(\text{Tsat(pressure)})\cdot\rho_f(\text{Tsat(pressure)})}{\text{hfg(Tsat(pressure))}\cdot\rho_g(\text{Tsat(pressure)})}\right)^{1.25}$$

$$\text{dtsatonb(tsat, pressure, dh)} := \text{Tsat}\left(\text{pressure}+\cfrac{4\cdot\sigma(\text{Tsat(pressure)})}{dh}\right)-\text{Tsat(pressure)}$$

$$\text{kdmin(dtsat, pressure, q'', M, N, kez, dcf, dh)} := (M\cdot\text{dmin(dtsat, pressure)})^2\cdot\min\left[N, \cfrac{kez}{q''\cdot dcf}\left[(\text{Tsat(pressure)}+\text{dtsat})-\text{Tsat}\left(\text{pressure}+\right.\right.\right.$$

$$\text{kdmin}\left[10K, 0.2\text{atm}, 100\,\frac{W}{cm^2}, \frac{145\cdot m}{in}, 8, (1-0.43)\cdot4\,\frac{W}{cm\cdot K}, \frac{0.42mm}{8}, .042mm\right] = 0.142$$

Guess q``

$$\text{dtsat} := 10K$$

$$q'' := 95.7\,\frac{W}{cm^2} \qquad\qquad Nu := 100\cdot\frac{W}{W}\frac{K}{K}\frac{cm}{cm}$$

$$\text{CTOL} := 10^{-3}$$

Given

$$\frac{q'' \cdot db(\text{pressure})}{\text{dtsat} \cdot k_f(\text{Tsat}(\text{pressure}))} = 4559 \left(\frac{q'' \cdot dh}{\varepsilon \cdot hfg(\text{Tsat}(\text{pressure})) \, \mu_f(\text{Tsat}(\text{pressure}))} \right)^{.91} \cdot \text{kdmin}(\text{dtsat}, \text{pressure}, q'', M, N, \text{kez}, \text{dcf}, dh)^{0.52} \cdot ja_{|}$$

$$\text{Sol}(\text{dtsat}, \text{pressure}, M, \text{kez}, \text{dcf}, dh, \varepsilon, \beta\delta, N) := \text{Find}(q'')$$

8L145

$$dh := .043\text{mm} \qquad M := \frac{145}{\text{in}} \cdot m \qquad \text{dcf} := \frac{.42\text{mm}}{8} \qquad M = 5.709 \times 10^3 \qquad \varepsilon := 0.43 \qquad \text{kez} := (1 - \varepsilon) \cdot 4 \frac{W}{\text{cm} \cdot K} \qquad \beta\delta := 17 \qquad N$$

$$\text{pressure} := 0.2\text{atm}$$

$$\text{Sol}(\text{dtsat}, \text{pressure}, M, \text{kez}, \text{dcf}, dh, \varepsilon, \beta\delta, N) = 143.424 \cdot \frac{\text{watt}}{\text{cm}^2}$$

$$\text{qpp} := \text{Sol}(\text{dtsat}, \text{pressure}, M, \text{kez}, \text{dcf}, dh, \varepsilon, \beta\delta, N)$$

$$\text{qpp} = 1.434 \times 10^6 \frac{\text{kg}}{\text{s}^3}$$

$$q(\text{dtsat}, \text{pressure}) := \text{Sol}(\text{dtsat}, \text{pressure}, M, \text{kez}, \text{dcf}, dh, \varepsilon, \beta\delta, N)$$

$$q(9K, 0.2\text{atm}) = 110.176 \cdot \frac{\text{watt}}{\text{cm}^2}$$

$$h(\text{dtsat}, \text{press}) := \frac{q(\text{dtsat}, \text{press})}{\text{dtsat}}$$

$$h(9K, 0.2\text{atm}) = 12.242 \cdot \frac{\text{watt}}{\text{cm}^2 K}$$

$$i := 0, 1 .. 85$$

$Ta :=$

	0
0	5
1	10
2	15
3	20
4	25
5	30
6	35
7	40
8	45
9	50

$T_i := Ta_i \cdot \dfrac{K}{10}$

$q(T_i, 0.217\text{atm}) =$

	0
0	0
1	0
2	0
3	0
4	0
5	0
6	0
7	0
8	0
9	0
10	0
11	18.955
12	36.236
13	53.585
14	71.35
15	...

$\cdot \dfrac{\text{watt}}{\text{cm}^2}$

▲ Penley correlation

Basic Variables

$m_e := (0.1)\text{kg}$ $c_e := 385 \dfrac{J}{\text{kg·K}}$ $h_{fg} := 2.3 \cdot 10^6 \dfrac{J}{\text{kg}}$ $Q_c := 10\text{watt}$

$A_e := 10\text{cm}^2$ $h_{cont} := \dfrac{Q_c}{A_e \cdot 6K}$ $h_i := 20314.43 \dfrac{\text{watt}}{\text{m}^2 K}$

Mass of Liquid and Mass of Vapor

$m_{liq} := 2.5 \times 10^{-4}\,\text{kg}$ $m_{vap} := .9 \cdot m_{liq}$

$$m_{vap} = 2.25 \times 10^{-4} \, kg$$

$$x := \frac{m_{vap}}{m_{liq} + m_{vap}}$$

$$x = 0.474$$

▼ Reference variables

Reference Variables

$$t_r := \frac{(m_e \cdot c_e)}{h_{cont} \cdot A_e} \qquad T_r := \frac{Q_c}{h_{cont} \cdot A_e} \qquad m_r := \frac{Q_c}{h_{fg}} \cdot \frac{(m_e \cdot c_e)}{h_{cont} \cdot A_e}$$

$$t_r = 23.1 \, s \qquad\qquad T_r = 6 \, K \qquad\qquad m_r = 1.004 \times 10^{-4} \, kg$$

▲ Reference variables

▼ Dimensionless power

Input Power

$$Q_{er} := 100$$

▲ Dimensionless power

▼ Sample calcualtion for heat transfer

sample calculation for heat transfer

$$q(8K, 0.2atm) = 75.109 \cdot \frac{watt}{cm^2}$$

$$h_{cor}(T_{sample}, P_{sample}) := \frac{q(T_{sample} \cdot T_r, P_{sample})}{T_{sample} \cdot h_{cont} \cdot T_r}$$

$$h_{cor}(1.1, .2atm) = 25.784$$

▲ Sample calcualtion for heat transfer

▼ Initial and maximum pressure value

Initial Pressure Value

$$p_{int} := .2atm$$

Maximum pressure setpoint

$$p_{max} := 0.22atm$$

▲ Initial and maximum pressure value

▼ Internal energy, total volume calculation

Internal Energy Calculation

$$\rho_g\left(\text{Tsat}\left(p_{int}\right)\right) = 0.135 \frac{kg}{m^3}$$

$$\rho_f\left(\text{Tsat}\left(p_{int}\right)\right) = 982.882 \frac{kg}{m^3}$$

$$m_{vap} = 2.25 \times 10^{-4} \, kg$$

$$vol_{liq} := \frac{m_{liq}}{\rho_f(\text{Tsat}(0.2\text{atm}))} \qquad vol_{gas} := \frac{m_{vap}}{\rho_g(\text{Tsat}(0.2\text{atm}))}$$

$$vol_{liq} = 2.544 \times 10^{-4} \, L \qquad vol_{gas} = 1.671 \, L$$

$$vol_{total} := vol_{liq} + vol_{gas}$$

$$m_t := m_{liq} + m_{vap}$$

$$m_t = 4.75 \times 10^{-4} \, kg$$

$$U_{int} := x \cdot m_t \cdot u_{vap}\left(p_{int}\right) + (1 - x) \cdot m_t \cdot u_{liq}\left(p_{int}\right) \qquad U_{int} = 615.88 \cdot J \qquad u_{int} := \frac{U_{int}}{m_t}$$

$$u_{int} = 1.297 \times 10^6 \cdot \frac{J}{kg}$$

$$vol_{total} = 1.671 \, L$$

▲ Internal energy, total volume calculation

▼ Change funtion of pressure and dryness fraction

change function of pressure and dryness for hybrid system

$$\text{press}(p,x) := \left| \begin{array}{l} \dfrac{\left[\dfrac{\text{vol}_{\text{total}} - (1-x)\dfrac{m_t}{\rho_f(\text{Tsat}(p))}}{\text{vol}_{\text{gas}}}\right]\cdot\left(\dfrac{\text{Tsat}(p)}{\text{Tsat}(p_{\text{int}})}\cdot\dfrac{m_t}{m_{\text{vap}}}\cdot 1.125\right) - \dfrac{m_t}{m_{\text{vap}}}\cdot x\cdot\dfrac{\text{Tsat}(p)}{\text{Tsat}(p_{\text{int}})}\cdot\dfrac{\left(1.125\dfrac{m_t}{\rho_f(\text{Tsat}(p))}\right)}{\text{vol}_{\text{gas}}}}{\left[\dfrac{\text{vol}_{\text{total}} - (1-x)\dfrac{m_t}{\rho_f(\text{Tsat}(p))}}{\text{vol}_{\text{gas}}}\right]^2 - \dfrac{m_t}{m_{\text{vap}}}\cdot x\cdot\dfrac{\text{vol}_{\text{total}} - (1-x)\dfrac{m_t}{\rho_f(\text{Tsat}(p))}}{\text{vol}_{\text{gas}}}\cdot dT\left(\dfrac{p}{p_{\text{int}}}\right)} \quad \text{if } p < p_{\text{max}} \\[30pt] 0 \quad \text{otherwise} \end{array} \right.$$

$$\text{dryness}(U,p) := \left| \begin{array}{l} \dfrac{\dfrac{\left(u_{\text{vap}}(p) - u_{\text{liq}}(p)\right)}{u_{\text{int}}}\cdot\left(\dfrac{U}{u_{\text{int}}} - dU_{\text{liq}}\left(\dfrac{p}{p_{\text{int}}}\right)\cdot 1.15\right) - \dfrac{\left(U - u_{\text{liq}}(p)\right)}{u_{\text{int}}}\cdot dU_{\text{fg}}\left(\dfrac{p}{p_{\text{int}}}\right)\cdot 1.15}{\left(\dfrac{u_{\text{vap}}(p)}{u_{\text{int}}} - \dfrac{u_{\text{liq}}(p)}{u_{\text{int}}}\right)^2} \quad \text{if } p < p_{\text{max}} \\[30pt] 0 \quad \text{otherwise} \end{array} \right.$$

▲ Change funtion of pressure and dryness fraction

▼ equations and initial conditions

equations and initial conditions

$$\text{init2} := \begin{pmatrix} 0 \\ 1 \\ x \\ 1 \end{pmatrix}$$

$$\text{de2}(t,y) := \begin{bmatrix} \dfrac{Q_{er} - h_{cor}\left[y_0 + \dfrac{\left(\text{Tsat}(0.2\text{atm}) + T_r - \text{Tsat}\left(y_3 \cdot P_{int}\right)\right)}{T_r}, y_3 \cdot P_{int}\right]\left[y_0 + \dfrac{\left(\text{Tsat}(0.2\text{atm}) + T_r - \text{Tsat}\left(y_3 \cdot P_{int}\right)\right)}{T_r}\right]}{} \\ \left[h_{cor}\left[y_0 + \dfrac{\left(\text{Tsat}(0.2\text{atm}) + T_r - \text{Tsat}\left(y_3 \cdot P_{int}\right)\right)}{T_r}, y_3 \cdot P_{int}\right]\left[y_0 + \dfrac{\left(\text{Tsat}(0.2\text{atm}) + T_r - \text{Tsat}\left(y_3 \cdot P_{int}\right)\right)}{T_r}\right] \cdot h_{cont} \cdot \dfrac{A_e \cdot T_r \cdot t_r}{m_t \cdot u_{int}} - \dfrac{Q_c \cdot t_r}{m_t \cdot u_{int}}\right] \\ \text{dryness}\left(y_1 \cdot u_{int}, y_3 \cdot P_{int}\right) \\ \text{press}\left(y_3 \cdot P_{int}, y_2\right) \end{bmatrix}$$

▲ equations and initial conditions

▼ Solution Matix

$$\text{solution3} := \text{Rkadapt}\left(\text{init2}, 0\,\frac{\text{sec}}{t_r}, 2\,\frac{\text{sec}}{t_r}, 2000, \text{de2}\right)$$

$$\text{s3}_{time} := \left(\text{solution3}^{\langle 0 \rangle}\right) \qquad\qquad \text{s3}_u := U_{int} \cdot \text{solution3}^{\langle 2 \rangle}$$

$$\text{s3}_{eletemp} := \left(\left(\text{solution3}^{\langle 1 \rangle}\right)\right)$$

$$\text{s3}_{press} := \left(\text{solution3}^{\langle 4 \rangle}\right)$$

$$s3_X := solution3^{\langle 3 \rangle}$$

▲ Solution Matix