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Control Simulation of Heat Transfer in Rectangular Microchannel

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ABSTRACT

Modeling of heat transfer in a rectangular microchannel and its coolant temperature control simulations are presented. Microchannel heat exchanging systems are widely used for thermal management of electronic devices. Therefore the control of temperature of coolant fluid with respect to the operating temperature limit of electronic device has greater significance. Heat transfer through rectangular microchannel is modeled using conventional theory of heat transfer and correlations. Present heat transfer model take care the variation of ambient temperature to which the electronic chip is exposed. Control simulations for thermal management of a rectangular microchannel has been carried out using PI, PID and cascade control and their effectiveness are compared.

Keywords: Microchannel heat exchangers, thermal management of electronics, PI, PID and Cascade control.

I. INTRODUCTION

Modern technology developments and microminiaturization associated with systems in microelectronics, aerospace, nuclear power plants, laser diodes, avionics, and hybrid vehicle power electronics etc. calls for the requirements to transfer or handle large amount of heat in smaller volumes. Miniaturization of integrated circuits leads to the development of thermal design of more powerful cooling systems. Heat sources with very small surface areas like high power semiconductor laser diode arrays needs high heat flux heat exchanger for their cooling. Power electronics for electric traction systems in hybrid electric vehicles are also using heat exchangers for handling high heat flux incurring in their operation.

Effective operation of microchannel heat exchangers involves optimum control of the fluid outlet temperature, mass flow rates and maximization of heat transfer from the chip to the coolant. Established methods for process modeling and dynamics of physiochemical systems using the numerical tools and commercial software are given by Wayne Bequette [1]. In the recent past, many researchers have attempted to work out novel methods for overcoming challenges for advanced cooling applications [2]. Conventional electronics cooling devices [3] have become obsolete for the new generation defense and computing electronic devices due to the requirement of managing high heat flux.

Modeling, analysis and control strategies of heat exchangers for these advanced heat exchangers have open out bright opportunities to take up challenging research tasks. Lee and Mudawar [4] and Zhang [5] worked on cooling strategies of high heat flux electronic devices. Apart from conventional heat exchangers, refrigerator based cooling systems [6] were also proposed for such applications. Accurate control of temperature heat exchanging fluids with phase changing process was the key aspect in all such applications. Recent attempts are reported [7-9] for the modeling and control of heat exchanger applications involving phase change. Apart from the challenges due to phase changing process, new attempts have been made for realistic modeling of heat exchanging devices [10, 11]. These control methods are quite useful in other process control applications also.

The dynamics of heat exchanging device attached to an electronic chip, such as very-large-scale integrated chips (VLSIC), has to take care the variation of coolant temperature with respect to variations in its inlet mass flow rate, chip temperature, energy content etc. In order to obtain optimum performance of heat exchanging device, control of dynamics of heat exchanger is to be modeled. As the performance of heat exchanger depends on the variation of this parameter with space and time, it is required to model the exchanger appropriately for implementing the control schemes. Present study elucidates heat transfer model of a rectangular micro channel using water as the coolant in channels and is exposed to still ambient on its flat surface. Advantage of such a heat transfer model is that the developed model take care the variations in ambient temperature to which the chip is likely to be exposed.

II. STRUCTURE OF THE COOLING SYSTEM

Cooling System

Cooling system consist of a closed loop heat exchanging system circulating coolant through rectangular microchannel heat exchanging passages as shown in Fig. 1. A pump circulates the coolant through the chip with a size of 56 × 56 mm. One side of the chip is integrated with an array of microchannels, through which the coolant is divided in equal amounts to facilitate heat transfer. A similar microchannel was experimentally studied by Jia [12]

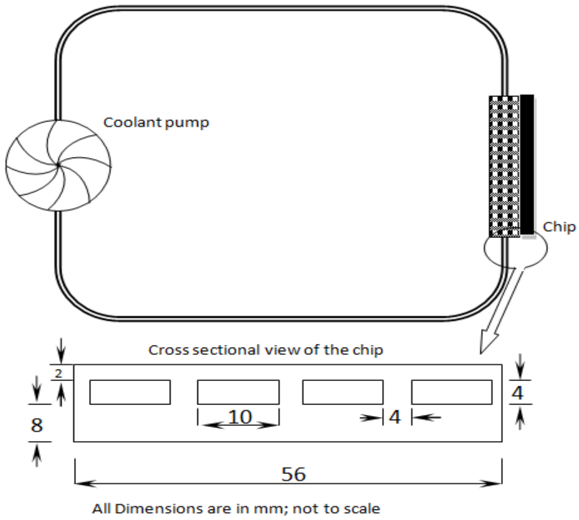

Fig 1. Structure of the cooling system

Microchannel heat exchanger

Microchannel heat exchanger consist of an array of four channels passing through a rectangular metal block that can be attached to the chip to be cooled. Each rectangular microchannel is having a size of 10 × 4 mm and are spaced at 4 mm apart as shown in Fig. 1. Heat generated in chip is transferred to the coolant passing through an array of rectangular microchannels. Flat face of metal block containing rectangular microchannels is also exposed to atmosphere and thus a part of the heat is lost directly to the atmosphere.

III. HEAT TRANSFER MODEL AND ANALYSIS

One dimensional heat transfer from the chip to the coolant and the atmosphere is modeled using fundamental heat transfer theory. A schematic, showing the direction of heat flow and resistances to be negotiated, is given in Fig. 2. Heat generated in chip (Q_{chip}) is transferred to the coolant and is subjected to a conduction resistance offered by the chip material (R_4) and a convection resistance (R_3) due to coolant flow in rectangular microchannel. Also heat transfer from/to the atmosphere (depending on ambient air temperature T_∞) is also subjected to a similar convection resistance (R_3) due to coolant flow in rectangular microchannel, a conduction resistance offered by the channel block material (R_2) and natural convection resistance by the ambient air (R_1). Heat transfer from the lateral surfaces can be neglected.

Energy balance

Basic heat transfer model for heat carried away by the coolant (Q_{fa}) can be obtained by energy balance.

$$Q_a + Q_{chip} = Q_{fa} \tag{1}$$

The heat transfer from ambient (Q_a) can be expressed as

$$Q_a = \frac{T_\infty - T_{fa}}{R_1 + R_2 + R_3} \qquad (2)$$

Where T_∞ and T_{fa} are respectively ambient air temperature and coolant fluid temperature.

The heat transfer from the chip (Q_{chip}) can be expressed as

$$Q_{chip} = \frac{T_6 - T_{fa}}{R_4 + R_5} \qquad (3)$$

Where T_6 is the desired temperature to be maintained at the surface of the chip.

The heat carried away by the coolant passing through the microchannel is given by

$$Q_{fa} = N\dot{m}c_c \;\; T_4 - T_1 \qquad (4)$$

Where N is the number of channels in the array, c_c is the specific heat of coolant, and $\dot{m}$ represents the mass flow rate of coolant passing through the microchannel, expressed as

$$\dot{m} = \rho_c A_{mc} V_c \qquad (5)$$

ρ_c is the density of coolant, A_{mc} is the cross-sectional area of the microchannel, and V_c is the velocity of coolant flowing through the microchannel.

Evaluation of conduction and convection resistances

One face of the block containing rectangular microchannel array is exposed to ambient air. Natural convection resistance offered by the ambient air can be found as

$$R_1 = \frac{1}{h_a \; A_s} \qquad (6)$$

A_s is the surface area of the face of the block exposed to ambient.

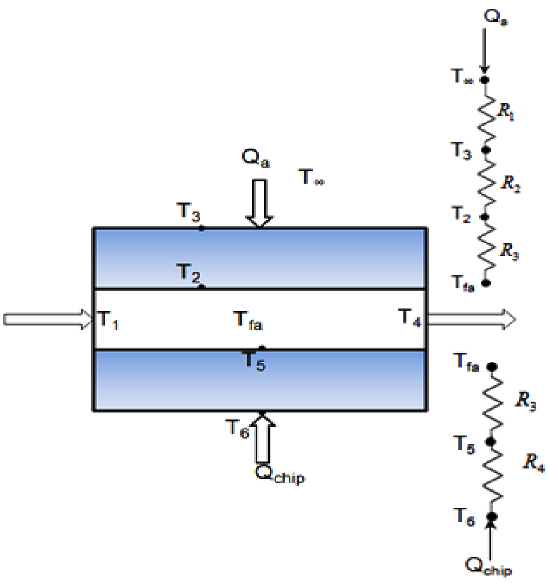

Fig 2. Heat transfer model

The heat transfer coefficient in natural convection h_a is evaluated as follows.

$$h_a = \frac{Nu_a k_a}{x} \tag{7}$$

k_a is the thermal conductivity of air and x is the height of vertical face which equal to the length of the coolant channel (L). Nusselt number correlation applicable for vertical wall with constant temperature [13] is

$$Nu_a = 0.68 + \frac{0.67 \;\; \mathrm{Gr\,Pr}^{\;0.25}}{\left\{1+\left[\dfrac{0.437}{\mathrm{Pr}}\right]^{0.5625}\right\}^{0.444}} \tag{8}$$

One dimensional conduction resistance in the block between outer face exposed to ambient and coolant channel can be expressed as

$$R_2 = \frac{\delta}{k_b A_s} \tag{9}$$

Where δ is the spacing between outer face and the channel and k_b is the thermal conductivity of the material of the block.

Convection resistance offered by the coolant flow in rectangular microchannel can be obtained as follows

$$R_3 = \frac{1}{N h_c L P} \tag{10}$$

Here P is the perimeter of the rectangular microchannel and heat transfer coefficient for internal forced convection is found as

$$h_c = \frac{Nu k_c}{D_h} \tag{11}$$

Here k_c is the thermal conductivity of coolant and D_h is the hydraulic diameter of the rectangular microchannel. Nusselt number for thermally developing laminar flow region on rectangular channel [14] is given by

$$Nu = 8.24 + 8.68 \;\; 10^3 x^{*} \;\;^{-0.506} \;\; e^{-41x^{*}}$$

$$x^{*} = \left(\frac{x/D_h}{\mathrm{Re\,Pr}}\right) \tag{12}$$

One dimensional conduction resistance in the block between chip face and coolant channel can be expressed as

$$R_2 = \frac{\delta'}{k_b A_s} \tag{13}$$

Where δ' is the spacing between chip face and the channel.

Analysis of flow rate dependence on chip surface temperature

Temperature developed on chip surface depends on its given power rating, ambient temperature and coolant flow rate. Thermo-physical parameters used for flow rate dependence study is given in Table I. The chip surface temperature for different coolant flow rate was evaluated based on the heat transfer model elaborated before with an extreme ambient temperature (60^{o}C).

Table 1.Parameters used for flow rate dependence study

Chip power rating (W)	Thermal properties		
	Medium	***Conductivity (W/m K)***	***Specific heat (kJ/kg K)***
60	Chip material	10.4	745
	Coolant	0.613	4.179

Flow rate dependence study was conducted by providing very small amount of flow (corresponding to a coolant flow velocity of 0.1m/s through each rectangular channel) initially. Chip surface temperature was found to drop drastically with increase in coolant flow velocity up to about 1.5 m/s. Thereafter coolant flow rate have no effect in providing further drop in chip surface temperature. Variation of chip surface temperature with coolant flow velocity through rectangular microchannel is plotted in Fig. 3.

Modeling of internal forced convection by the coolant fluid plays a vital role in the estimation of chip surface temperature. Correlation for thermally developing laminar flow region on rectangular channel, as given in Eqn. 12, has been chosen for the present study. Due to the non-availability of wide operational range for coolant flow rate and the possibility of variation in ambient temperature, design of control strategy for this application becomes a challenging task.

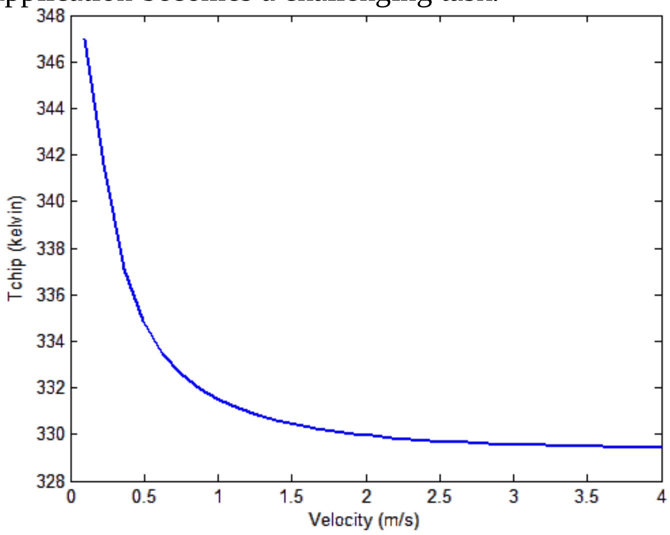

Fig 3. Flow rate dependence on chip temperature

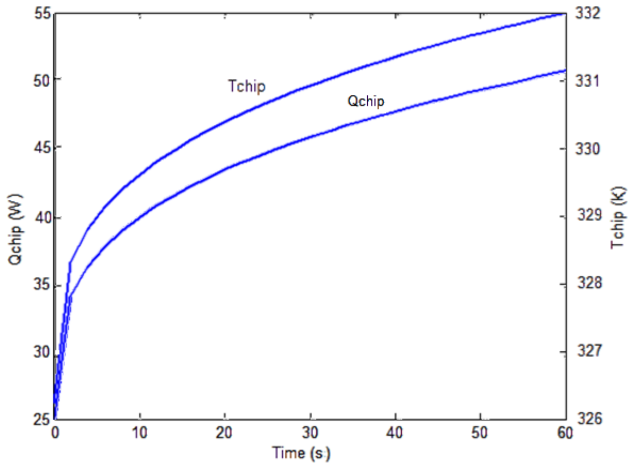

Fig 4. Given power variation for simulating initial transient and corresponding chip surface temperature development

Chip surface temperature development during its initial transient operation

Transient thermal analysis of the system with a given inlet coolant temperature (303 K) was performed in order to assess the variation of chip surface temperature with development of chip power. Given power variation for simulating initial transient and the corresponding chip surface temperature development is plotted in Fig. 4. A linear dependence between chip surface temperature and chip power has been observed for a given coolant temperature.

Analysis of coolant inlet temperature dependence on chip surface temperature

An independent analysis has been conducted, with the heat transfer model elaborated before, in order to study the influence of coolant inlet temperature on chip surface temperature. A linear dependence between coolant inlet temperature and chip surface temperature was observed for a given power rating of the chip (55 W).

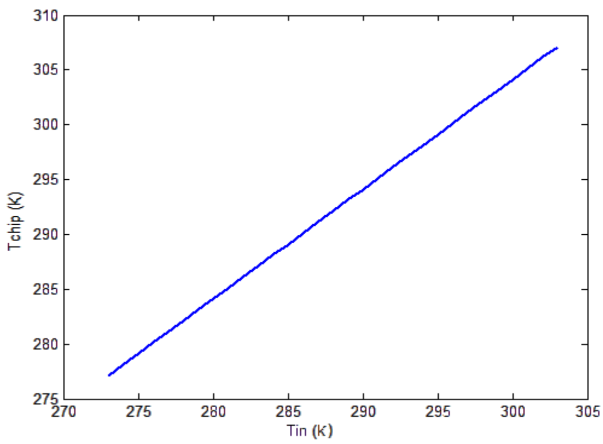

Fig 5. Chip surface temperature variation with coolant inlet temperature

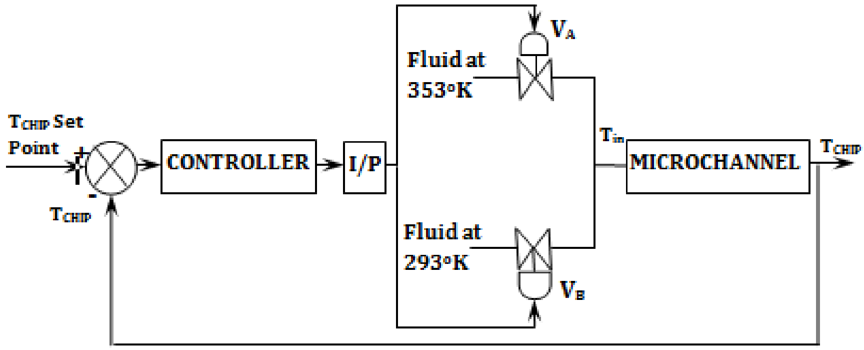

Fig 6. Flow diagram for PI and PID based control system

IV. CONTROL SYSTEM DESIGN AND SIMULATION

Configuration and components

Initially a PI and PID based control system has been proposed to regulate the heat transfer from the chip to the coolant as shown in Fig 6. The cooling system consists of controller & valve arrangement to obtain required coolant temperature. The controller executes the control algorithm by comparing the chip temperature with the desired set-point value and gives necessary control signal to the final control element (valve), through an actuator. The actuator converts the control signal to pressure signal to operate the valve. The valve then actuates in accordance with the control signal from the controller. A fixed flow rate requirement of input coolant is achieved by using dual acting linear valve. The dynamics of valve is approximated to be linear, by neglecting any delay in its response. Dual acting linear valve is given with input from coolant sources at two different temperatures. Valve regulates hot and cold water in the required proportion by keeping the net flow rate as constant.

The responses of system with conventional PID and PI have considerable oscillations about the set-point, which is undesirable. Oscillations are often generated due to the variations of coolant temperature. The coolant temperature oscillations can be reduced by using a cascade controller. In cascade controller, the secondary loop consists of proportional controller, actuator and valve. The whole system forms the primary loop, controlled by a PID controller as shown in Fig. 7. The output of the PID controller forms the set-point to secondary controller.

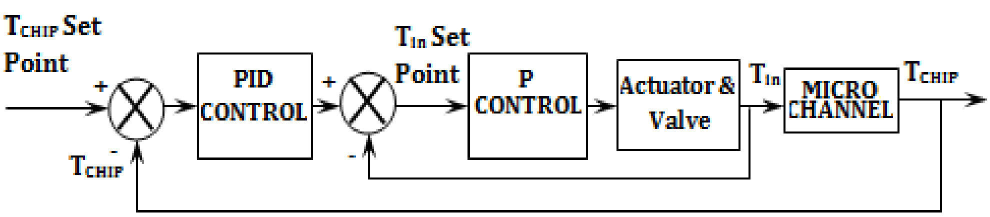

Fig 7. Flow diagram for cascade control system

Table 2. Gain Values

Controller	Gain		
	K_P	K_I (s^{-1})	K_D (s^{-1})
PID	0.7002	1.001	4.004
PI	0.52515	0.6006	
Cascade Controller			
Primary controller-PID	1.935	1.001	4.004
Secondary controller-P	0.52515		

Controller Design

Design of present classical PID and PI controller is aimed at maintaining a constant desired chip temperature (309 K) by varying the coolant temperature. Control signal of PID and PI controller are respectively given by

$$c(t) = K_p \left[e\ t\ + K_I \int_0^t e\ t\ dt + K_D \frac{de}{dt} \right] + c_o \tag{14}$$

$$c(t) = K_p \left[e\ t\ + K_I \int_0^t e\ t\ dt \right] + c_o \tag{15}$$

Where K_P, K_I and K_D are proportional, integral and derivative gain respectively, e(t) is the error between chip temperature and set point, c_o is the initial controller output.The K_P, K_I and K_D values are tuned using Ziegler- Nichols Method. The values obtained after tuning the same are given in Table 2. The proportional controller of cascade control system is initially tuned using Ziegler- Nichols Method to achieve a faster response. Then primary controller is also tuned along with the secondary loop to obtain the desired response.

Control System Simulation

Heat exchange system has been independently controlled using PI, PID and cascade control systems and compared their performance. The performance of the controllers is evaluated based on time integral performance criteria Integral of the Square Error (ISE), Integral of the Absolute value of the Error (IAE) and Integral of the Time-weighted Absolute Error (ITAE). A comparison of time integral performance criteria is given in Table III. IAE is found to be the smallest value among all three criteria for all the three controllers analysed in the present study; therefore they are best suited to suppress small errors in the present system. Cascade controller is found to have the minimum of ISE, IAE and ITAE, hence this is ideally suited for the present thermal system based on the time integral performance criteria.

Table 3. Comparison of Time Integral performance criteria

Control Scheme	Criteria		
	ISE	**IAE**	**ITAE**
PID	526.9981	51.6440	171.7025
PI	479.4752	59.9835	307.3276
Cascade	13.1311	7.6635	32.5460

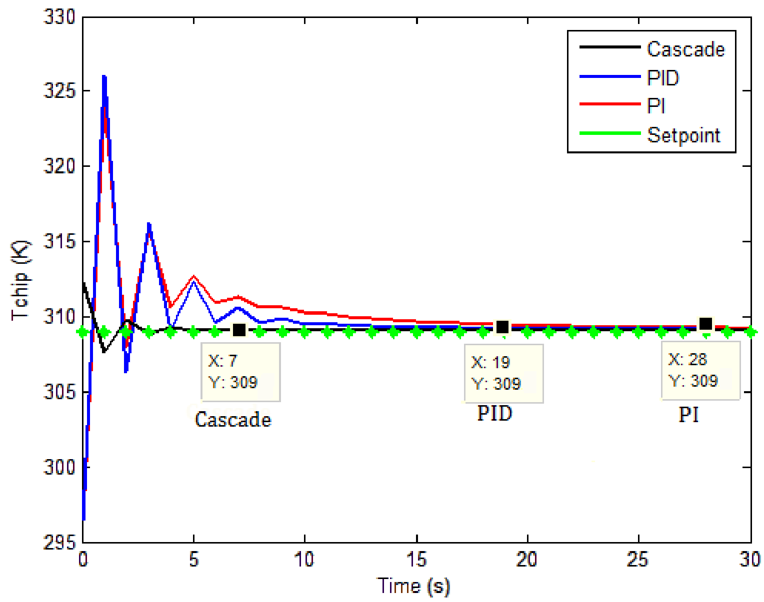

Fig 8. Comparison of control system performance

Comparison of control system performance is given in Fig. 8. Cascade controller is found to settle early (7s) with minimum overshoot. This is due to the control of coolant temperature using secondary controller. The control of coolant temperature minimises the oscillation of chip temperature about the setpoint. The PID and PI controller settles by 19s and 28s respectively, with considerable overshoots and oscillations about the setpoint.

V. CONCLUSION

A heat exchanging system for electronic chip with rectangular microchannel has been modeled. PI, PID and cascade control schemes are implemented and simulated to study their performance. Following are the major conclusions derived from the present study.

- Heat transfer model accounts for the variation of ambient temperature to which the electronic chip is exposed. Therefore cooling system can efficiently work in wide range of ambient temperatures.
- Chip surface temperature was found dropping only for coolant flow velocity up to about 1.5 m/s. Unlike conventional heat exchangers, this restriction imposes challenges in control of chip surface temperature using coolant flow.
- A cascade controller used in the present rectangular microchannel heat exchange system is found to provide robust performance than PI and PID controllers. This is due to the reduction of coolant temperature oscillation before controlling the chip temperature.

Present study can be extended by making use of a phase changing coolant for enhancing heat transfer.

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