

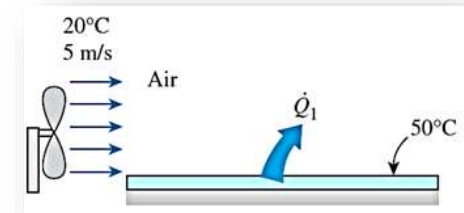
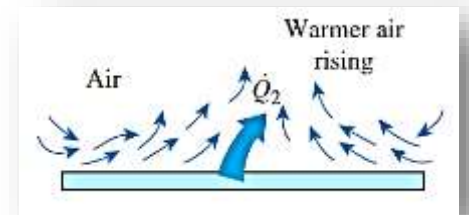
Convection Heat Transfer and Applications in Electronics Equipment

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AP/Mech

CONVECTIVE HEAT TRANSFER

- HT through Fluid in the Presence of Bulk Fluid Motion
- Natural and Forced
- Fluid Motion Caused by Natural Means (by buoyancy)
- Cooling Down hot Coffee without a Fan ...
- Fluid Forced to Flow by External Means (by external Force)



Governed by Newtons Law of Cooling ... $Q = hAdT$

COMPONENTS of CONVECTION

Conduction + Advection

- Flow Properties Like v , ρ , μ , γ , and so on
- Thermal & Velocity Boundary Layers, and
- Flow Patterns like Laminar and Turbulent Flows

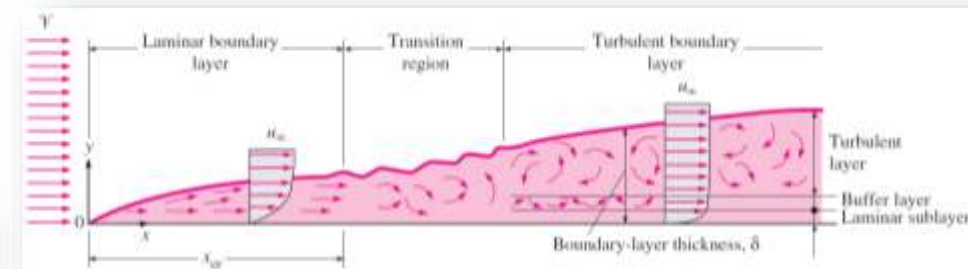
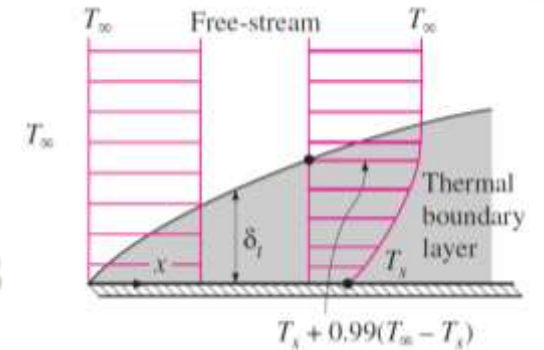
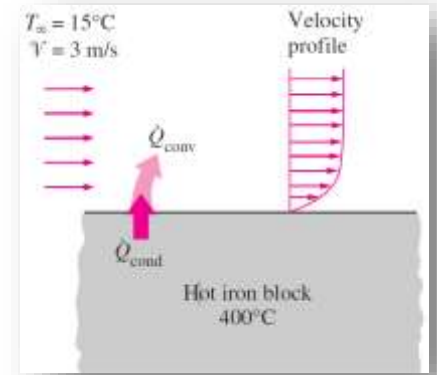
- Reynolds Number
- Prandtl Number and,
- Nusselt numbers

$$Re = \frac{\rho v D}{\mu}$$

$$Pr = \frac{\mu C_p}{k}$$

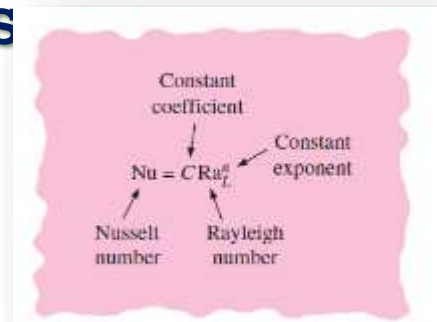
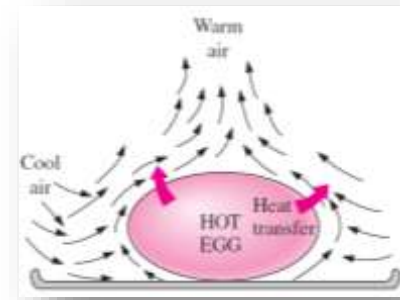
$$Nu = \frac{h L_c}{k}$$

$$Q = h A d T$$



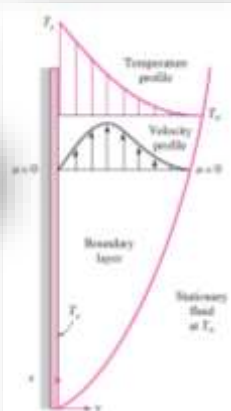
HEAT TRANSFER COEFFICIENTS

- Quantitative Characteristic between Wall & Fluid
- Depends on Thermal & Hydrodynamic Characteristics
- Hydrodynamic and Thermal Boundary Conditions
- Nusselt Correlations for General Estimation
- Enthalpy Difference for Varying Heat Capacity
- Overall HTC for Liquid to Liquid HT

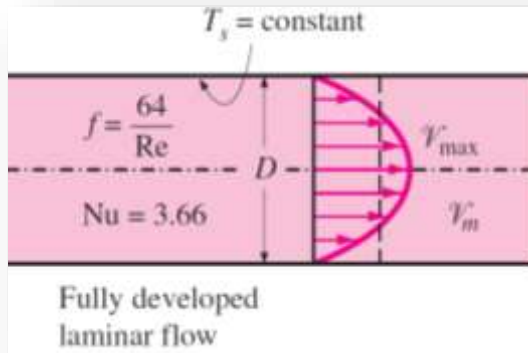


$$Nu = \frac{hL_c}{k} = C(Gr_L Pr)^n = C Ra_L^n$$

$$Ra_L = Gr_L Pr = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} Pr$$

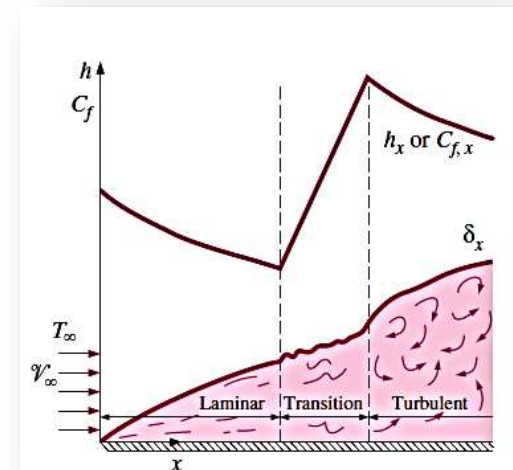
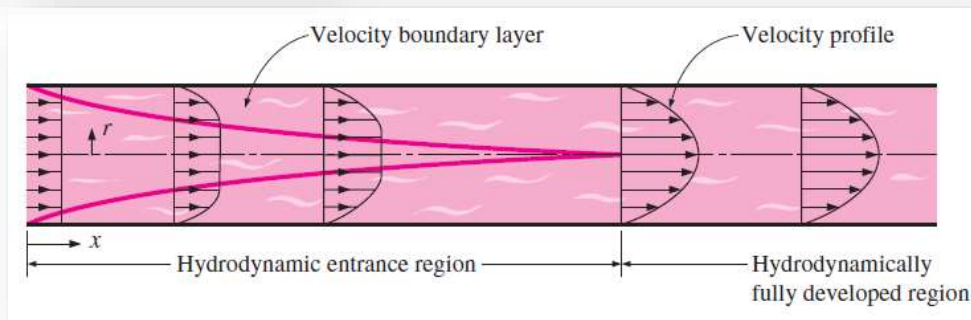
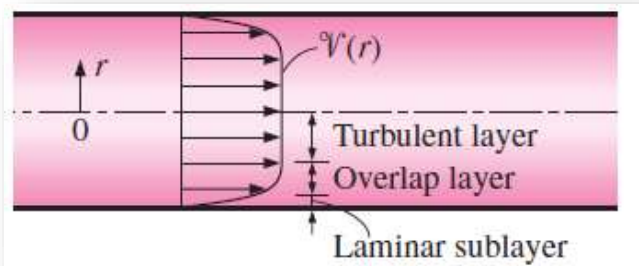


CONVECTIVE HEAT TRANSFER - ESTIMATION



Laminar:
$$Nu_x = \frac{h_x x}{k} = 0.332 Re_x^{0.5} Pr^{1/3} \quad Pr > 0.60$$

Turbulent:
$$Nu_x = \frac{h_x x}{k} = 0.0296 Re_x^{0.8} Pr^{1/3} \quad \begin{matrix} 0.6 \leq Pr \leq 60 \\ 5 \times 10^5 \leq Re_x \leq 10^7 \end{matrix}$$



CONVECTIVE HEAT TRANSFER - ESTIMATION

HEAT AND MASS TRANSFER DATA BOOK 33

PROPERTY VALUES OF GASES AT ONE ATMOSPHERIC PRESSURE (use $\beta = 1/T$, T in K)

Fig given h , ν , μ and Pr may be taken as not sensitive to pressure. But α , κ , ρ should be corrected for pressure, by calculating the value of p at the pressure.

| Temperature t °C | Density ρ kg/m ³ | Absolute Viscosity μ Ns/m ² | Kinematic Viscosity ν m ² /s | Thermal Diffusivity α m ² /s | Prandtl Number Pr | Specific Heat c_p J/kgK | Thermal Conductivity k W/mK |
|--------------------------|--|---|--|---|-------------------------|------------------------------------|--|
| DRY AIR | | | | | | | |
| -50 | 1.594 | 14.81 × 10 ⁻⁶ | 9.23 × 10 ⁻⁶ | 22.84 × 10 ⁻⁶ | 0.728 | 1013 | 0.02033 |
| -40 | 1.515 | 15.26 × 10 ⁻⁶ | 10.06 × 10 ⁻⁶ | 23.75 × 10 ⁻⁶ | 0.728 | 1013 | 0.02117 |
| -30 | 1.453 | 15.89 × 10 ⁻⁶ | 10.90 × 10 ⁻⁶ | 24.81 × 10 ⁻⁶ | 0.733 | 1013 | 0.02186 |
| -20 | 1.395 | 16.16 × 10 ⁻⁶ | 11.61 × 10 ⁻⁶ | 25.94 × 10 ⁻⁶ | 0.733 | 1009 | 0.02279 |
| -10 | 1.343 | 16.67 × 10 ⁻⁶ | 12.40 × 10 ⁻⁶ | 27.44 × 10 ⁻⁶ | 0.733 | 1009 | 0.02361 |
| 0 | 1.293 | 17.38 × 10 ⁻⁶ | 13.39 × 10 ⁻⁶ | 28.90 × 10 ⁻⁶ | 0.757 | 1005 | 0.02442 |
| 10 | 1.247 | 17.85 × 10 ⁻⁶ | 14.30 × 10 ⁻⁶ | 30.06 × 10 ⁻⁶ | 0.756 | 1005 | 0.02512 |
| 20 | 1.205 | 18.34 × 10 ⁻⁶ | 15.19 × 10 ⁻⁶ | 31.41 × 10 ⁻⁶ | 0.759 | 1005 | 0.02580 |
| 30 | 1.165 | 18.83 × 10 ⁻⁶ | 16.10 × 10 ⁻⁶ | 32.86 × 10 ⁻⁶ | 0.751 | 1005 | 0.02675 |
| 40 | 1.128 | 19.32 × 10 ⁻⁶ | 17.09 × 10 ⁻⁶ | 34.39 × 10 ⁻⁶ | 0.699 | 1005 | 0.02736 |
| 50 | 1.093 | 19.81 × 10 ⁻⁶ | 17.95 × 10 ⁻⁶ | 35.72 × 10 ⁻⁶ | 0.696 | 1005 | 0.02826 |
| 60 | 1.060 | 20.30 × 10 ⁻⁶ | 19.07 × 10 ⁻⁶ | 37.19 × 10 ⁻⁶ | 0.696 | 1005 | 0.02896 |
| 70 | 1.029 | 20.59 × 10 ⁻⁶ | 20.02 × 10 ⁻⁶ | 38.53 × 10 ⁻⁶ | 0.694 | 1000 | 0.02960 |
| 80 | 1.000 | 21.08 × 10 ⁻⁶ | 21.09 × 10 ⁻⁶ | 39.19 × 10 ⁻⁶ | 0.692 | 1000 | 0.03047 |
| 90 | 0.972 | 21.45 × 10 ⁻⁶ | 22.15 × 10 ⁻⁶ | 40.69 × 10 ⁻⁶ | 0.692 | 1000 | 0.03138 |

Laminar:

$$Nu_x = \frac{h_x x}{k} = 0.332 Re_x^{0.5} Pr^{1/3} \quad Pr > 0.60$$

Turbulent:

$$Nu_x = \frac{h_x x}{k} = 0.0296 Re_x^{0.8} Pr^{1/3} \quad \begin{matrix} 0.6 \leq Pr \leq 60 \\ 5 \times 10^5 \leq Re_x \leq 10^7 \end{matrix}$$

1. Flat Plate

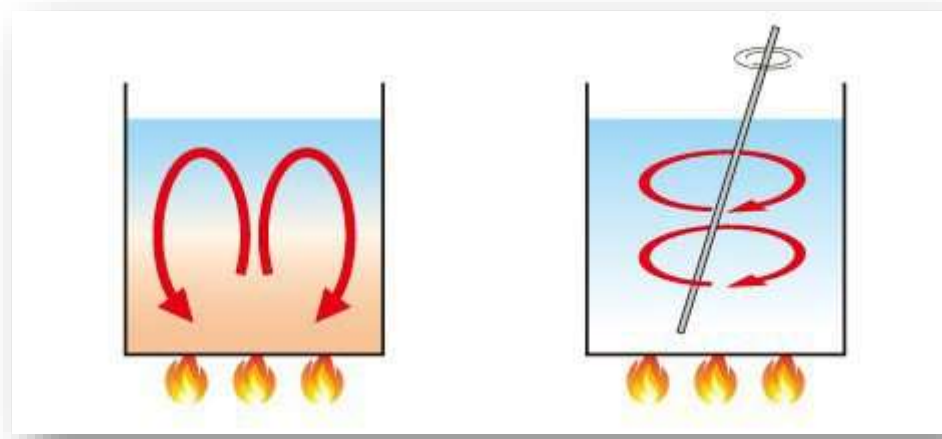
| Flow conditions | Correlation and Validity | Notations |
|---|---|--|
| 1.0 LAMINAR FLOW ($Re_x \leq 5 \times 10^5$) Boundary Layer Thickness | $\delta_x = 5 \sqrt{Re_x^{-1/2}}$ $\delta_{99} = \delta_x Pr^{-1/3}$ $\delta_x = \delta_{99}/2$ $\delta_x = \delta_{99}/4$ | T_f = Film temperature = $(T_w + T_\infty)/2$ T_∞ = Free stream fluid temperature T_w = Plate surface temperature Re_x = Reynolds number at a distance x from leading edge Re_{x_1} = Reynolds number at location x_1 |
| Friction Factor | $C_{f,x} = 0.664 Re_x^{-1/2}$ $C_{f,L} = 1.328 Re_L^{-1/2}$ | δ_{99} = Hydrodynamic boundary layer thickness at a distance x from leading edge δ_{99} = Thermal boundary layer thickness at a distance x from leading edge δ_x = Displacement thickness at x δ_{99} = Momentum thickness at x $C_{f,x}$ = Local friction coefficient defined by $\tau_w/(\rho u_\infty^2/2)$ (Fanning friction factor) |
| 1.1 Constant Wall Temperature | $Nu_x = 0.332 Re_x^{1/2} Pr^{1/3}$ $0.6 \leq Pr \leq 100$ | τ_w = wall shear stress at x , N/m ² $C_{f,x}$ = Average friction coefficient upto the distance L from leading edge |
| 1.1.1 If heating starts from a distance x_1 from leading edge | $Nu_x = 0.332 Re_x^{1/2} Pr^{1/3} \left[1 - (x_1/x)^{1/4} \right]^{-1/4}$ | |
| 1.1.2 For liquid metals (low Prandtl numbers) and for | $Nu_x = 0.565 Re_x^{1/2} Pr^{1/4}$ $0.01 \leq Pr \leq 10$ | |

HEAT AND MASS TRANSFER DATA BOOK 113

| Flow conditions | Correlation and Validity | Notations |
|--|--|--|
| 1.2.1 For liquid metals, or alloys Constant heat flux | $Nu_{x,L} = \frac{0.62 Re_L^{1/2} Pr^{1/4} [1 + 0.4 Pr^{1/4} + 0.06 Pr^{1/2}]}{[1 + 0.03 Re_L^{1/4}]^{1/4}}$ Valid for: $Pr \geq 0.1$ and $Pr \leq 100$ and for $Re_x, Pr \geq 100$ | $Nu_{x,L}$ = Average Nusselt Number upto length L Nu_x = Nusselt Number at location x δ_x = Stanton Number |
| 1.2 Average values for constant heat flux or constant wall temperature | $\overline{Nu_L} = 2.85 Re_L^{1/2} Pr^{1/4}$ $(Re_L Pr)^{1/2} = C_{f,L}/2$ | $\frac{Nu}{Re Pr}$ $C_{f,L}$ = Average friction coefficient |
| 1.4 FLAT PLATE TURBULENT FLOW $5 \times 10^5 \leq Re_x \leq 10^7$ | $\delta_x = 0.37 x Re_x^{-1/2}$ $\delta_{99} = \delta_x$ | x = distance from leading edge δ_{99} = hydrodynamic boundary layer thickness at x δ_x = thermal boundary layer thickness at x |
| 1.4.1 Fully Turbulent from leading edge | $\delta_x = 0.37 x Re_x^{-1/2}$ $\delta_{99} = 0.37 x Re_x^{-1/2}$ $Nu_x = 0.0296 Re_x^{1/4} Pr^{1/3}$ | δ_x = displacement thickness at x δ_{99} = momentum thickness at x |

CONVECTIVE VS NATURAL HEAT TRANSFER

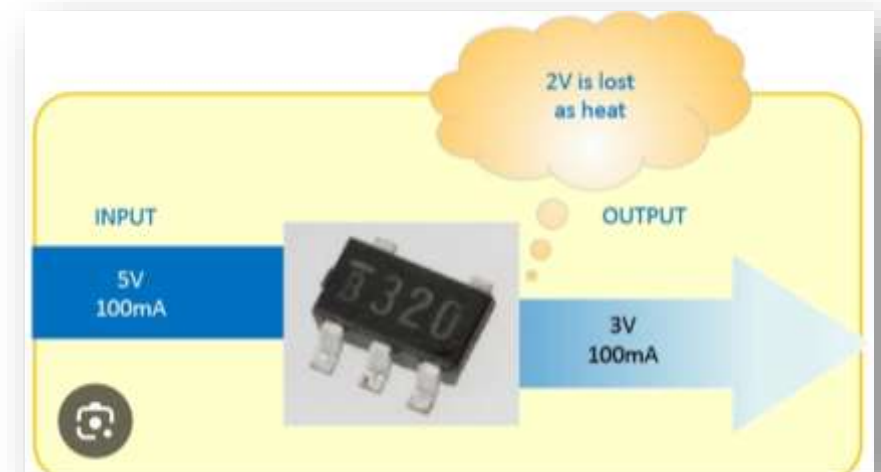
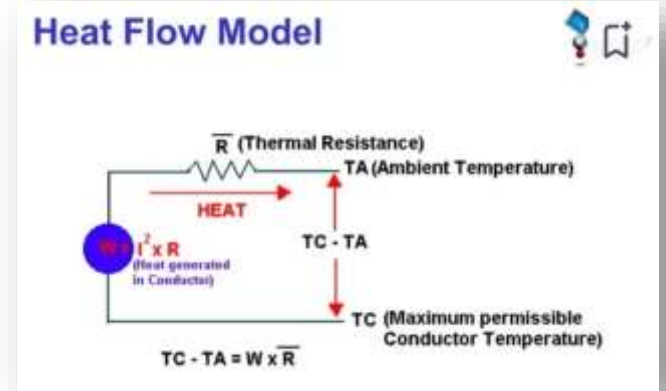
- Motion within Fluid – Density Difference
- Motion Caused by External Agent like **ower**



CONVECTION HT IN ELECTRONICS

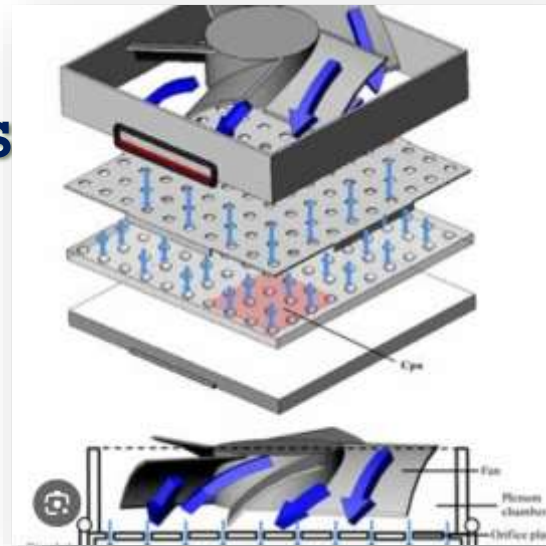
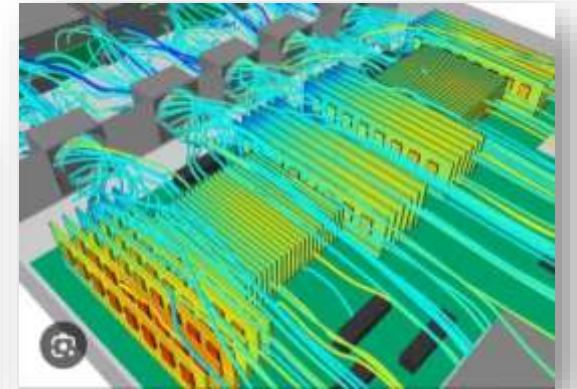
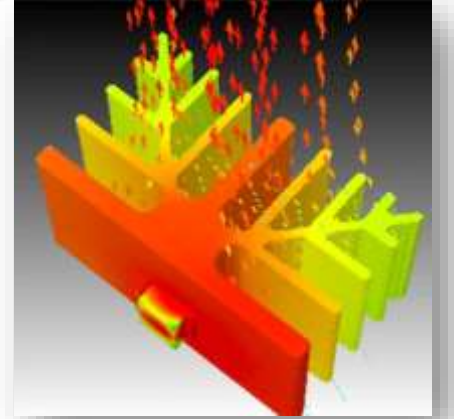


- Need of Cooling Electronic Components
- Flow of Electron generates Heat
- Evacuation Needed to Maintain below BDT
- Increased Area Enhances Better HT
- Cooling Inevitable



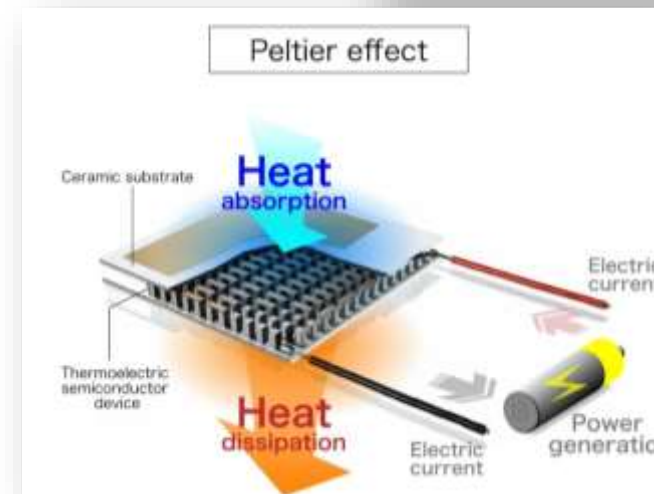
FORCED HT in COMPONENT COOLING

- Evacuation of Heat - Continuous
- Failing to Evacuate Breaks Down Component
- Stagnation Possible in Natural Convection
- Mostly Turbulent Pattern Exists
- Fans Adopted for Cooling



FORCED HT Efficient with FINS

- Increased Area Obtained through Fins
- Enhances Better Evacuation Rate
- Heat Transfer Area Exposes Well
- Miniaturization Balanced by Fins
- Great Scope for Cooling



RECAP . . .

- Heat Transfer By Convection – Property Oriented
- Natural and Forced Convection – Flow Pattern Dependent
- Miniaturization Makes Cooling Inevitable
- Cooling Module Design – Never Ending
- Fins – Vital in Cooling Modules
- Complexity Minimized with Numerical Modelling