#### AutoTherm Heat Transfer Tools – Validation and Tutorials

List of Heat Transfer Tools:

- 01: Heat Conduction Composite Systems (Walls/Cylinders/Spheres)
- 02: K effective Parallel/Series
- 03: K effective Vias
- 04: Conduction with Uniform Heat Generation (Walls/Cylinders/Spheres)
- 05: Fin Efficiency/Effectiveness
- 06: Heat Sink Analysis
- 07: Multidimensional Heat Transfer in Common Configurations
- 08: Lumped Capacity (Constant Ambient Temperature/ varying Ambient Temperature)
- 09: Transient Conduction (Walls, Cylinders, Spheres)
- 10: Semi-Infinite Solids
- 11: Contact of Two Semi-Infinite Solids
- 12: Flat Plate in Parallel Flow
- 13: Flow Over 3D Bodies
- 14: Flow Over Tube Banks
- 15: Internal Flow Heat Transfer
- 16: Natural Convection over Bodies
- 17: Natural Convection Vertical Channels
- 18: Heat Exchanger Epsilon/NTU Calculator
- 19: Heat Exchanger Performance Analysis Tool
- 20: Radiation View factor Calculator

# HT-01: Heat Conduction Composite Systems (Walls/Cylinders/Spheres)

This tool provides an automated interactive panel to study one-dimensional heat transfer in layered composite structures (walls, cylinder and spheres).

Required Input:

#### 1. Initialization

- Choose geometry type (plane wall, cylinder, or sphere)
- Provide the number of layers to be modeled in the composite structure.Click Initialize. This will result in modifications to input/output panels to
- accommodate the number of layers and geometry type

#### 2. Boundary Conditions

- Choose boundary condition type (convection, fixed temperature, or specified heat flux) for the "*Right*" (higher coordinate value) and "*Left*" (lower coordinate value).
- For convection BC, enter value for heat transfer coefficient and ambient temperature.
- For Fixed Temperature enter the value for the temperature.
- For Specified Heat Flux, enter heat flux value.
- Choose any unit for each from options provided in Unit Combo Boxes.

Input O Plane	Wall	O Cylinder	○ Sphere
No. of Layers	3	(	Initialize

Right BC (x=0)			
O Convection	h	10	W/m2.K $\sim$
-	mb	20	C ~
O Fixed Temp	Ts		C ~
			W/2
O Heat FLux	q"s		W/m2 ~
Right BC (x=L)			
-	h	40	W/m2.K ~
<ul> <li>Convection</li> </ul>		40	W/m2.K ~
<ul> <li>Convection</li> </ul>			
<ul> <li>Convection</li> </ul>			
• Convection	amb Ts		C ~
• Convection	amb		C ~

X0	0.0000	cm	~	L	1.00	m	~
				w	1.00	m	~

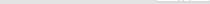
#### 3. Geometric Parameters

*Wall*: Minimum x value (Default is zero); Wall Size (Default 1m x 1m) *Cylinder*: Inner Radius; Cylinder Length *Sphere*: Inner Radius

#### 4. Thermal Contact Resistance

Interfacial contact resistance may be applied between any layers. Film resistance may also be applied at inner/outer surfaces.

Thermal Contact Resistance			
Layer_1/Layer_2 v	Value	0.095	C.in2/W ~
			Annha



#### 5. Layer Details

Provide thermal conductivity and thickness for each layer.

- Choose unit for each can be set at the top of this panel.
- Enter values in the spreadsheet.

	Layer Name	Therm. Cond.	Thickness
	Layer_1	0.78	4
	Layer_2	0.026	8
•	Layer_3	0.78	4

#### Results:

#### 1. Layers

In this section results for each layer is presented in a spreadsheet format. These include

- Temperature at boundaries and interfaces.
- Thermal resistance in each layer.

These results are presented using units above the spreadsheet.

#### 2. Overall Results

These results consist of heat flux, total rate of heat transfer, total thermal impedance, and overall heat transfer coefficient.

	X Left	T Left	Rth	X Right	T Right
Þ	0.000	14.23	0.005	4.000	13.93
	4.000	13.93	0.385	14.000	-8.26
	14.000	-8.26	0.005	18.000	-8.56

Results		
q"tot	57.7065	W/m2 🗸
qtot	69.2478	W ~
R"tot	0.5199	C.m2/W $\sim$
U	1.9236	W/m2.K v

#### **Example – Composite Wall:**

A 0.8 m by 1.5 m glass window (thermal conductivity of 0.78 W/m.K) with a thickness of 8 mm. Assume the interior of the room is maintained at 20 °C with natural convection heat transfer coefficient of 10 W/m<sup>2</sup>.K, and the external ambient at -10 °C with heat transfer coefficient of 40 W/m<sup>2</sup>.K (which include thermal radiation effects). Determine the steady rate of heat transfer through this window and the temperature of its inner surface for the following two conditions:

- a) Single pane with a thickness of 8 mm.
- b) Double-pane consisting of two 4-mm thick glass layers separated by 10-mm thick stagnant airspace.

# Solution:

- a) Single pane
  - Open "Heat Conduction Composite Systems" Panel:

Heat Trasfer Thermodynami	cs Numerical Methods Help	
General Concepts		
Heat Conduction	▶ Steady-State - 1D ▶	Background
Convective Heat Transfer	<ul> <li>Steady-State - 2D/3D</li> </ul>	Multi-Layer Conduction
Thermal Radiation	Transient Conduction	Tool: Composite Wall
Heat Exchangers		Effective Thermal Conductivity Concept
Advanced Tools	ated Eng	Tool: K Effective Parallel/Series
		Tool: K Effective Vias
	Ductocianala	Conduction with Heat Generation
students and	l Professionals	Tool: Wall - Uniform Heat Generation
		Fins Extended Surfaces
e place		Tool: Fin Efficiency/Effectiveness/Area
uirces equist	ions and tool	Tools: Heat Sink Analysis

Select *Plane Wall* (default) and set number of layers to "1" (for Case-a) and click *Initialize*.

Input O Plane	Wall	O Cylinder	O Sphere
No. of Layers	1		Initialize

Set Width and Length to 0.8 m and 1.5 m

X0 0.0000	 L	1.5	m	~
	w	0.8	m	~

Set values for Left BC (h=10 W/m<sup>2</sup>.K, T<sub>amb</sub>=20 °C), and Right BC (h=40 W/m<sup>2</sup>.K, T<sub>amb</sub>=-10 °C)

Left BC (X=0)		
O Convection h	10	W/m2.K ∨
Tamb	20	C ~
○ Fixed Temp Ts		C ~
O Heat FLux 9"	s	W/m2 ~
Right BC (X=L)		
rugin DC (X=L)		
Convection h	40	W/m2.K $\vee$
h	40 -10	W/m2.K ~ C ~
O Convection h		

Change Layers thickness unit to mm; and for Layer\_1 enter Therm. Cond=0.78 and Thickness=8
 Click Update to solve. The finished form with results is shown below:

• Plane Wall O Cylind	er 🔿 Sphere	x0 0.0000 cm × 1 1.00 m ×	
lo. of Layers 1	Initialize		
ight BC (x=0) Convection h Tamb 20 Fixed Temp Ts Heat FLux q"s ight BC (x=L) Convection h Tamb 40 -10 Fixed Temp Ts Fixed Temp Ts Heat FLux q"s	W/m2.K ~ C ~ C ~ W/m2 ~ W/m2.K ~ C ~ C ~ W/m2 ~	W     1.00     m       Thermal Contact Resistance       Xmin     Value     0.0000       C.m2/W ~       Apply   Layers       Layer Name     Therm. Cond.       Thickness       Layer_1     0.78	$T_{N} $ $T$
sults	][	Layers	R <sub>1</sub> R <sub>N</sub>
"tot 221.8009	W/m2 v	X mm V T C V Rth C.m2/W V	
qtot 221.8009	W ~	X Left         T Left         Rth         X Right         T Right           0.000         -2.18         0.010         8.000         -4.45	
"tot 0.1353	C.m2/W ~		Update
U 7.3934	W/m2.K v		

The heat flux is 221.8 W/m<sup>2</sup>. The total rate of heat transfer through the window is 266.2 W. The surface temperature at the inside wall is -2.19 °C, as shown under  $T_{Left}$  above.

- b) Double-pane
- Change No. of Layers to **3** and click Initialize.
- Now, enter thermal conductivity and thickness for all layers, as shown below.

Layers	Thichness mm	~ k	W/m.K ~
	Layer Name	Therm. Cond.	Thickness
	Layer_1	0.78	4
	Layer_2	0.026	10
.1	Layer_3	0.78	4

Click *Update* to solve. The finished form with results is shown below:

Heat Conduction - Composit Systems		X
Input Plane Wall Cylinder Sphere	x0 0.0000 cm → L 1.5 m →	
No. of Layers 3 Initialize	X0 0.0000 cm V L 1.5 m V 0.8 m V	
Right BC (x=0)		
O Convection h 10 W/m2.K ∨	Thermal Contact Resistance	
Tamb 20 C 🗸	Xmin Value 0.0000 C.m2/W V	
○ Fixed Temp Ts C	Apply	$\uparrow \uparrow $
O Heat FLux q"s ₩/m2 ∨	Layers Thichness mm v k W/m.K v	
Right BC (x=L)	Layer Name Therm. Cond. Thickness	
O Convection h 40 ₩/m2.K ∨	Layer_1 0.78 4	$h_1, T_{\infty,1}$
Tamb C	Layer_2 0.026 10	
○ Fixed Temp Ts	▶ Layer_3 0.78 4	
		R <sub>cnv1</sub> Mr. R <sub>cnv2</sub>
O Heat FLux q"s ₩/m2 ∨		

		] [							К1		κ <sub>N</sub>
Results			Layers								
q"tot	57.7065	W/m2 🗸		X mm	<u> </u>		Rth C.m2/	w ~			
				X Left	T Left	Rth	X Right	T Right			
qtot	69.2478	W ~	•	0.000	14.23	0.005	4.000	13.93			
R"tot	0.5199	C.m2/W ~		4.000	13.93	0.385	14.000	-8.26		Update	
N LOL	0.5188	C.m2/W ~		14.000	-8.26	0.005	18.000	-8.56			
U	1.9236	W/m2.K v									
					-						Show Tutorial

For this case, the heat flux is 57.7 W/m<sup>2</sup>, and the rate of heat transfer through the window is 69.24 W. The surface temperature at the inside wall is 14.23 °C, as shown under  $T_{Left}$  in the first row above. Note, considerably better performance with the double-pane window.

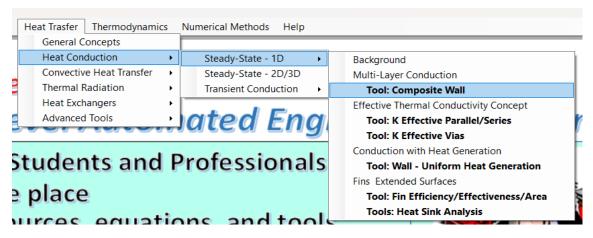
<<End-of-Tutorial>>

#### HT-01b:

**Example – Composite Cylinder:** 

Steam at 320 °C flows in a 5-cm diameter cast iron pipe (k=80 W/m.K) with wall thickness of 0.25 cm. The pipe is covered with 3-cm-thick glass wool insulation (k=0.05 W/m.K). Assuming internal heat transfer coefficient of 60 W/m<sup>2</sup>°C and external ambient at 5 °C with heat transfer coefficient of 18 W/m<sup>2</sup>°C, determine the rate of heat loss from the pipe per unit length and temperature drops across the pipe shell and the insulation.

# Solution:



Open "Heat Conduction Composite Systems" Panel:

Select *Cylinder* and set number of layers to "2" and click *Initialize*.



Set values for *Left BC* (*h*=**60** *W*/*m*<sup>2</sup>.*K*, *T*<sub>amb</sub>=**320** °C), and *Right BC* (*h*=**18** *W*/*m*<sup>2</sup>.*K*, *T*<sub>amb</sub>=**5** °C)

Left BC (X=0)				
<ul> <li>Convection</li> </ul>	h	60	W/m2.K ~	
	Tamb	320	C ~	
O Fixed Temp	Ts		c ~	
⊖ Heat FLux	q"s		W/m2 v	
Right BC (X=L)				
<ul> <li>Convection</li> </ul>	h	18	W/m2.K ∨	
	Tamb	5	C ~	
O Fixed Temp	Ts		C ~	
⊖ Heat FLux	q"s		W/m2 v	
Set r0	)=2.5	cm and len	gth, L=1 n	n (default)
-0 25		(m)	V I 10	

r0 2.5 cm ~ L 1.00 m ~

Change *Layers thickness* unit to **mm**; and for *Layer\_1* enter *Therm. Cond=0.78* and *Thickness=8* Now, enter thermal conductivity and thickness for all layers, as shown below:

Layers				
-	71.1		JALL N	]

	Thichness cm	k	W/m.K ~
	Layer Name	Therm. Cond.	Thickness
	Layer_1	80	0.25
.1	Layer_2	0.05	3

Click *Update* to solve. The finished form with results is shown below

Heat Conduction - Composit Systems		- 0 X
Input     Plane Wall     Cylinder     Sphere       No. of Layers     2     Initialize       Left BC (X=0)     Initialize	r0 2.5 cm ~ L 1.00 m ~	h <sub>2</sub> / T <sub>amb2</sub>
Left BC (x=0)         h         60         W/m2.K ∨           Tamb         320         C         ∨           Fixed Temp         Ts         C         ∨           Heat FLux         q <sup>n</sup> s         W/m2 ∨         V/m2 ∨	Thermal Contact Resistance       rmin     Value       0.0000     C.m2/W v       Apply       Layers       Thichness     cm       k     W/m.K	
Right BC (X=L)         h         18         W/m2.K ~           Convection         h         5         C ~           Fixed Temp         Ts         C ~         C           Heat FLux         q <sup>45</sup> W/m2 ~         W/m2 ~	Layer Name     Therm. Cond.     Thickness       Layer_1     80     0.25       Layer_2     0.05     3	$r_0$ $r_N$ $r_N$ $L$ $h_1/T_{amb1}$
Results	Layers X cm v T C v Rth C.m2/W v	
qr         120.7861         W →           R"tot         2.6079         C.m2/W →           R"tot         C.m2/W →         C.m2/W →           U         0.3834         W/m2.K →	r Left         T Left         Rth         r Right         T Right           ▶         2.500         307.18         0.000         2.750         307.16           2.750         307.16         2.348         5.750         23.57	Update

The rate of heat transfer from the pipe is 120.8 W. The temperature drop across the shell is 307.18 - 307.16 = .02 °C, and the temperature drop across the insulation is 307.16 - 23.17 = 283.59 °C.

<<End-of-Tutorial>>

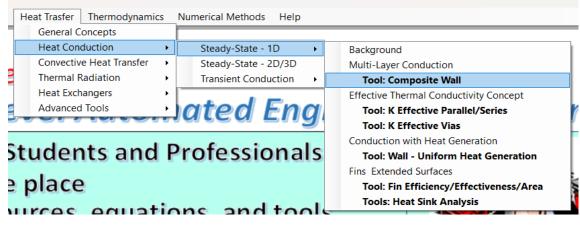
# HT-01c:

# **Example – Composite Sphere:**

A stainless-steel spherical tank (k=15 W/m.K), with 3 m internal diameter of and 2 cm thickness, is used to store ice water at 0 °C. If the environment is at 22 °C, and assuming 80 W/m<sup>2</sup>°C and 16 W/m<sup>2</sup>°C for internal and external heat transfer coefficients, determine the rate of transfer into the container.

# Solution:

# Open "Heat Conduction Composite Systems" Panel:



Select *Sphere* and set number of layers to "1" and click *Initialize*.

#### Initialize

Set values for *Left BC* (*h*=**80** *W*/*m*<sup>2</sup>.*K*, *T*<sub>*amb*</sub>=**0** °*C*), and *Right BC* (*h*=**16** *W*/*m*<sup>2</sup>.*K*, *T*<sub>*amb*</sub>=**22** °*C*)

Right BC (r=0)			
<ul> <li>Convection</li> </ul>	h	80	W/m2.K ∨
	Tamb	0	C ~
O Fixed Temp	Ts		<b>C</b> ~
O Heat FLux	q"s		W/m2 v
Right BC (r=R)			
Right BC (r=R) Convection	h	16	W/m2.K ~
<ul> <li>Convection</li> </ul>		16 22	W/m2.K ~ C ~
<ul> <li>Convection</li> </ul>			
<ul> <li>Convection</li> </ul>			
• Convection	Tamb		C ~

Set r0=1.5 cm.

r0	1.5	m	~

Keep Layers thickness unit to cm; and for Layer\_1 enter Therm. Cond=15 and Thickness=3

Layers	Thichness cm	~	k W/m.K v
	Layer Name	Therm. Cond.	Thickness
•	Layer_1	15	2

#### Click *Update* to solve. The finished form with results is shown below

Heat Conduction - Composit Systems		– o x
Input Plane Wall O Cylinder O Sphere No. of Layers 1 Initialize	r0 <u>1.5 m v</u>	
Right BC (r=0) Tamb80 W/m2.K $\checkmark$ C $\bigcirc$ Convection 	Thermal Contact Resistance         rmin       Value       0.0000       C.m2/W v         Apply         Layers       Thichness       K       W/m.K v         Layer Name       Therm. Cond.       Thickness         Layer_1       15       2	$\begin{array}{c} h_2 / T_{amb2} \\ \hline \\ r_0 \\ h_1 / T_{amb1} \\ k_1 \\ \hline \\ k_N \end{array}$
Results         W            qr         -8329.1342         W            R"tot         0.0026         C.m2/W             R"tot         C.m2/W              U         378.5970         W/m2.K	Layers         X         m         T         C         Rth         C.m2/W ∨           r         r         Left         Rth         r         Right         T         T         Right         Right         T         Right         Right<	Update
		Show Tutorial

The rate of heat transfer into the pipe is 8329.1W (negative sign indicated heat transfer in -r direction, or into the sphere).

#### <<End-of-Tutorial>>

#### HT-02: K effective Parallel/Series

This tool provides can be used to determine "effective thermal conductivity" for composite layers (such as printed circuit boards), where a number of "thin" planes with relatively high thermal conductivity and embedded in a background material. Two values K<sub>eff</sub> are provided for "inplane" (parallel) and thru-plane (normal). These values are commonly used to simplify numerical modeling of complex geometries consisting of multiple layers.

**Required Input:** 

#### 1. Initialization

- Choose the number of "Groups". A group consist of a number of layers with uniform thickness. In PCBs these layers are commonly made of copper, with thickness provided in oz-cu (0.00137 in) or mils (0.001 in).
- Click Initialize. This will result in creation of rows for each group to define layers.

Layers			
	No. of Groups	(	Initialize
		W/m.K v	oz-cu 🗸

#### 2. Definition of Layers

- Enter values for
  - Number of layers for each group.
  - Thermal conductivity for these layers (Choose unit from the Combo-

	Nu. of Layers	Therm. Cond.	Thickness
	2	390	2
•	3	390	1

- Box above).
- Thickness for each layer (Choose unit from the Combo-Box above).

#### 3. Background (Dielectric) Definition

Enter thermal conductivity and total thickness for the background (board).

#### Results:

Values for in-plane and normal equivalent thermal conductivities are presented in user-selected units.

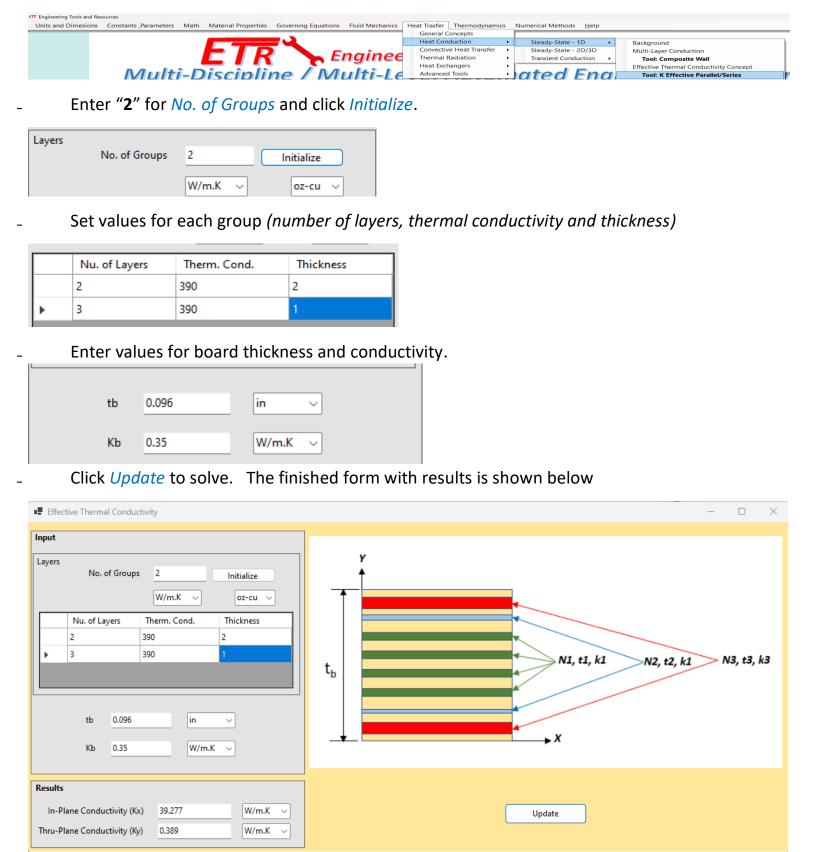
#### **Example:**

Consider a 0.096-inch-thick PCB, with two 2-oz-cu layers and three 1-oz-cu layers. Determine inplane and thru-plane effective thermal conductivities for this board. Assume the board is made pf FR4 (k=0.35 W/m.K) and copper conductivity is 390 W/m.K.

#### Solution:

tb	0.096	in	~
КЬ	0.35	W/m.K	~

39.277	W/m.K v
0.389	W/m.K v



The in-plane conductivity is 39.28 W/m.K and thru-plane conductivity is 0.39 W/m.K (dominated by board conductivity for series resistance).

<<End-of-Tutorial>>

# HT-03: K effective Vias

**Example:** Thermal vias are placed underneath an electronic component to enhance heat transfer through the PCB into a heat sink below. Determine the effective thermal conductivity through the PCB assuming

- Via region: 0.65 in x 0.4 in
- Via inside diameter: 0.3 mm
- Barrel wall thickness = 1 oz-copper
- Via spacing 1 mm.

# Solution:

#### Open "Effective Conductivity - Vias" Panel:



Choose *Via Spacing* option and enter the following parameters:

```
Via Inner Diameter = 0.3 mm
```

```
Via Wall Thickness = 1 oz-cu
```

```
Via Conductivity (Kv) = 395 W/m.K (Default)
```

Filler Conductivity (Kf) = 0.03 W/m.K (Default)

```
Back Conductivity (Kb) = 0.35 W/m.K (Default)
```

#### *L* = 0.65 in

#### *W* = 0.4 in

Effective Conductivity -     Input O Number of					- 0 X
<ul> <li>Via Spacing</li> <li>Vias Inner Diameter</li> <li>Vias Wall Thickness</li> <li>Vias Conductivity (Kv)</li> <li>Filler Conductivity (Kf)</li> <li>Back Conductivity (Kb)</li> <li>L</li> <li>W</li> </ul> Results	9     1       0.3     1       395	mm v mm v oz-cu v W/m.K v W/m.K v in v W/m.K v	w		k <sub>v</sub> , D <sub>v</sub> – Via k <sub>f</sub> , D <sub>f</sub> – Filler K <sub>b</sub> - Background
			(	Update	

Click *Update* to solve. The finished form with results is shown below

The calculated thru-board conductivity is 14.107 W/m.K. Note that this analysis assumes unfilled vias (air conductivity of 0.03 W/m.K is used by default).

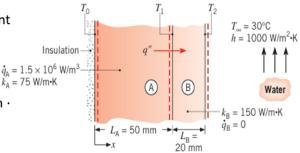
<<End-of-Tutorial>>

# HT04: Conduction with Uniform Heat Generation (Walls/Cylinders/Spheres)

#### Example: Composite wall with Heat Generation: [Source: Bergman-Lavine Example 3.7]

A plane wall of material A has uniform heat generation 1.50 MW/m<sup>3</sup>,  $k_A = 75$  W/m · K, and thickness  $L_A = 50$  mm. The outer surface of the wall (larger x value) is cooled by a water stream with a temperature of 30 °C and heat transfer coefficient of 1000 W/m2.K. Determine the inner wall temperature for the following three scenarios.

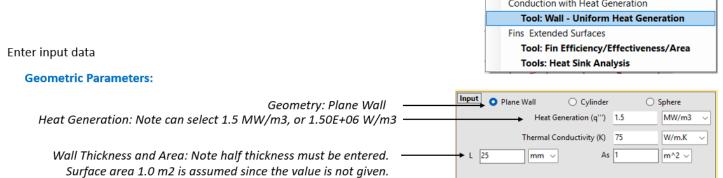
- a) The inner surface is exposed to air at 30  $^\circ C$  with heat transfer coefficient of 250 W/m2.K
- b) The Inner surface is insulated.
- c) The inner surface is insulated, and the outer surface is attached to  $k_A^{\mu}$  another wall with material B that has no generation with  $k_B = 150 \text{ W/m} \cdot \text{K}$  and thickness  $L_B = 20 \text{ mm}$ .



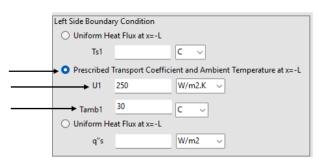
# Solution:

a) Open "Solids with Uniform Heat Generation" Panel:

	General Concepts	- F			
Thermal Radiation  Transient Conduction Transient C	Heat Conduction	•	Steady-State - 1D	•	Background
Heat Exchangers	Convective Heat Transfer	•	Steady-State - 2D/3D	•	Multi-Layer Conduction
Elective memory concerns	Thermal Radiation	•	Transient Conduction	•	Tool: Composite Wall
	Heat Exchangers	· · [			Effective Thermal Conductivity Concept
Tool: K Effective Parallel/Series	Advanced Tools	L			Tool: K Effective Parallel/Series
Tool: K Effective Parallel/Series	Advanced Tools	•			

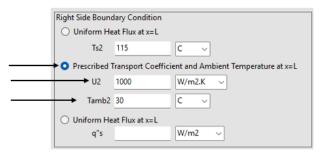


Select Convective BC. Enter the value for convection coefficient on the left side. Enter the value for the ambient temperature on the left side.



#### Right side (x=L) Boundary Conditions:

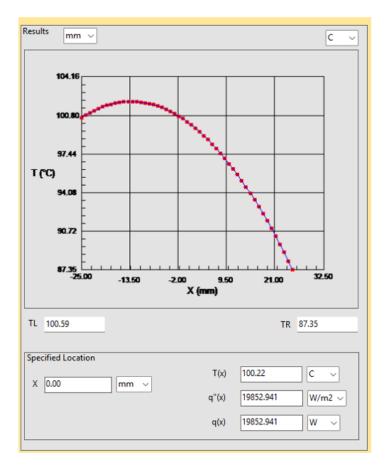
Select Convective BC. Enter the value for convection coefficient on the right side. Enter the value for the ambient temperature on the right side.



#### Results

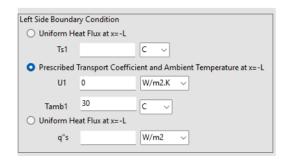
Click Update to solve.

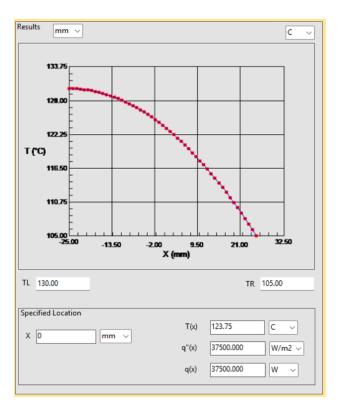
- **1**. Plot of temperature distribution through the slab is provided.
- 2. Left-side (x=-L) and Right-Side (x=L) temperatures are shown.
- 3. The user can select the units for temperature and length for the plot are from unit boxes at the top.
- 4. Specified Location section allows the user to access local values.
- 5. Heat flux and heat rate values for this problem are identical because surface area is set to 1 m<sup>2</sup>.



#### b) The Inner surface is insulated

The Only change to be made is to make the left side insulated. This can be done by assigning a value of zero for the overall hea transfer coefficient U1.





#### Results

Click Update to solve.

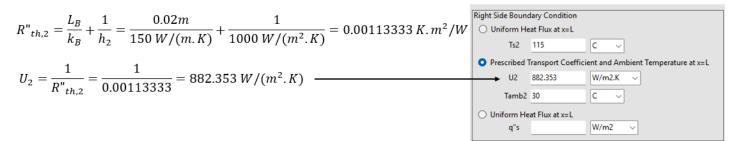
Plot of temperature distribution through the slab indicates insulated BC on the left face and updated temperature and heat flux values. c) The inner surface is insulated, and the outer surface is attached to another wall with material B that has no generation with  $k_B = 150$  W/m

 $\cdot$  K and thickness L<sub>B</sub> = 20 mm.

The Only change to be made is to the right-side Boundary by adjusting U2 to account for the additional conductive layer of material B.

This can be done by calculating the total thermal resistance on the right side of the heat generating section by using

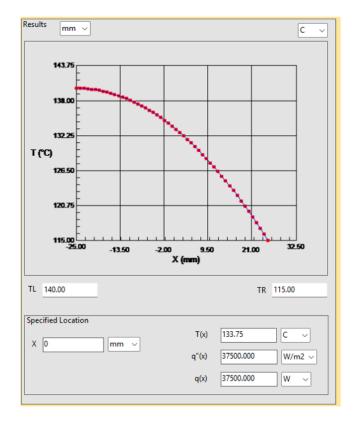
- The composite wall tool (this method will be demonstrated on another example using a composite cylinder.
- Calculating the thermal resistance.



#### Results

Click Update to solve.

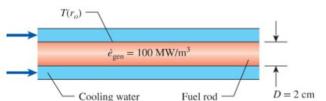
Plot of temperature distribution through the slab indicates insulated BC on the left face and updated temperature and heat flux values.



#### Example: Water-Cooled Fuel Rod [Source: Cengel-Ghagar Problem 2.104]

A cylindrical fuel rod (k = 30 W/m.K) 2 cm in diameter is encased in a concentric tube and cooled by water. The fuel rod generates heat uniformly at a rate of 100 MW/m3, and the average temperature of the cooling water is 75C with a convection heat transfer coefficient of 2500 W/m2.K. The operating pressure of the cooling water is such that the surface temperature of the fuel rod must be kept below 200°C to prevent the cooling water from reaching the critical heat flux (CHF). The critical heat flux is a thermal limit at which a boiling crisis can occur that causes overheating on the fuel rod surface and leads to damage. Determine the variation of temperature in the fuel rod and the temperature of the fuel rod surface. Is the surface of the fuel rod adequately cooled?

#### Solution:



#### Open "Heat Generation in a Solid" Panel

2500 W/m<sup>2</sup>·K, 75°C

General Concepts				
leat Conduction		Steady-State - 1D		Background
Convective Heat Transfer	•	Steady-State - 2D/3D	•	Multi-Layer Conduction
hermal Radiation	•	Transient Conduction		Tool: Composite Wall
Heat Exchangers	1 m		_	Effective Thermal Conductivity Concept
Advanced Tools				Tool: K Effective Parallel/Series
				Tool: K Effective Vias
				Conduction with Heat Generation
			Γ	Tool: Heat Generation in a Solid
			_	Fins Extended Surfaces

Tool: Fin Efficiency/Effectiveness/Area

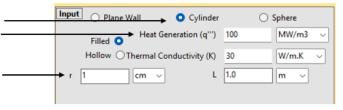
Tools: Heat Sink Analysis

Enter input data

#### **Geometric Parameters:**

Geometry: Cylinder and select "Filled" option Heat Generation: Enter 100 and change unit to MW/m3 Enter thermal conductivity 30 W/m.K

Radius: Enter 1 cm (diameter is given as 2 cm). Length 1.0 m is assumed since the value is not given.



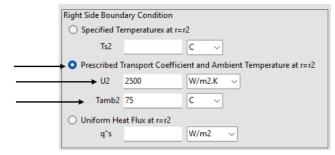
#### Left side (r=r1) Boundary Conditions:

This panel is disabled since we have a solid (filled) cylinder.

Left Side Boundar	y Condition
O Uniform He	at Flux at r=r1
Ts1	C ~
O Prescribed T	ransport Coefficient and Ambient Temperature at r=r1
U1	W/m2.K 🗸
Tamb1	c v
<ul> <li>Uniform He</li> </ul>	at Flux at r=r1
q"s	W/m2 🗸

#### Right side (r=r2) Boundary Conditions:

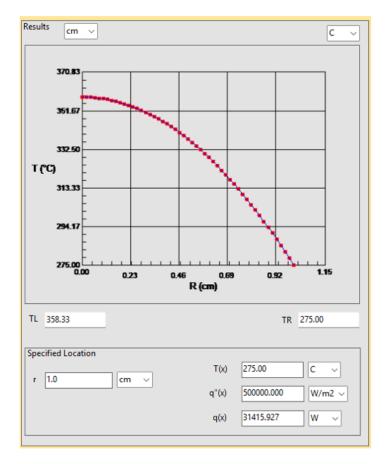
Select Convective BC. Enter the value for convection coefficient on the right side. Enter the value for the ambient temperature on the right side.



#### Results

Click Update to solve.

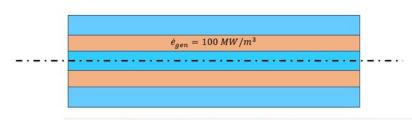
- 1. Plot of temperature distribution through the slab is provided.
- Left-side (centerline) and Right-Side (r=1 cm) temperatures are shown.
- 3. The user can select the units for temperature and length for the plot are from unit boxes at the top.
- 4. The surface temperature of 275 °C is 75°C higher than the temperature necessary to prevent the cooling water from reaching the CHF. So, the fuel rod is not adequately cooled.
- 5. The center of the rod is at 358.3 °C.
- 6. Specified Location section allows the user to access local values. Here r=1 cm points to the rod surface.
- 7. The surface heat flux is 500,000 W/m2. And the total heat is calculated to be 31,415.9 W for a 1-mete-long section of the rod.



#### Example: Water-Cooled Fuel Rod [Source: Cengel-Ghagar Problem 2.104 - Modified]

Repeat the above example with added inner cooling channel of 2 cm, cooling water at 75 °C with a convection heat transfer coefficient of 2500 W/m<sup>2</sup>.K. Assume the fuel rod has an inner radius of 1 cm and the outer radius determined to keep the total volume (and therefore, heat dissipation) unchanged. Assume the external surface of the rod is exposed to an identical convective environment.

#### Solution:



#### Open "Heat Generation in a Solid" Panel:

I:	Heat Iraster	Thermodynamic	S	Numerical Methods Help			_
	General Cond			Charle Chate 1D			
		e Heat Transfer	• •	Steady-State - 1D Steady-State - 2D/3D Transient Conduction	• •	Background Multi-Layer Conduction Tool: Composite Wall	
19	Heat Exch Advanced		•	Doguina DI	~	Effective Thermal Conductivity Concept Tool: K Effective Parallel/Series Tool: K Effective Vias	

Conduction with Heat Generation Tool: Heat Generation in a Solid

**Tools: Heat Sink Analysis** 

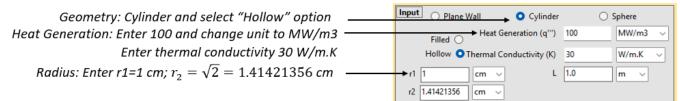
Tool: Fin Efficiency/Effectiveness/Area

Fins Extended Surfaces

To determine the outer diameter of the hollow fuel rod, we set the volume equal to the volume of the solid rod.

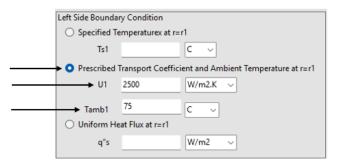
Enter input data

#### **Geometric Parameters:**



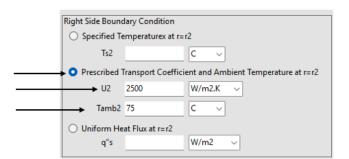
#### Left side (r=r1) Boundary Conditions:

Select Convective BC. Enter the value for convection coefficient on the inner surface. Enter the value for the ambient temperature on the inner surface.



Right side (r=r2) Boundary Conditions:

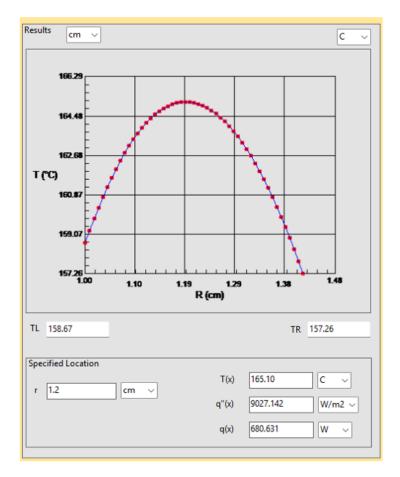
Select Convective BC. Enter the value for convection coefficient on the outer surface. Enter the value for the ambient temperature on the outer surface.



#### Results

Click Update to solve.

- 1. Plot of temperature distribution through the slab is provided.
- 2. Left-side (centerline) and Right-Side (r=1 cm) temperatures are shown.
- 3. The user can select the units for temperature and length for the plot are from unit boxes at the top.
- 4. The surface temperature of 158.57 °C and 157.26 °C are well below 200 °C. So, the fuel rod is adequately cooled.
- 5. Specified Location section allows the user to access local values. Here r=1.2 cm points to the max temperature in the rod.
- 6. The temperature at this location is 165.10 °C, heat flux is 9,027.9 W/m2. And the total heat is calculated to be 680.6 W for a 1-mete-long section of the rod.

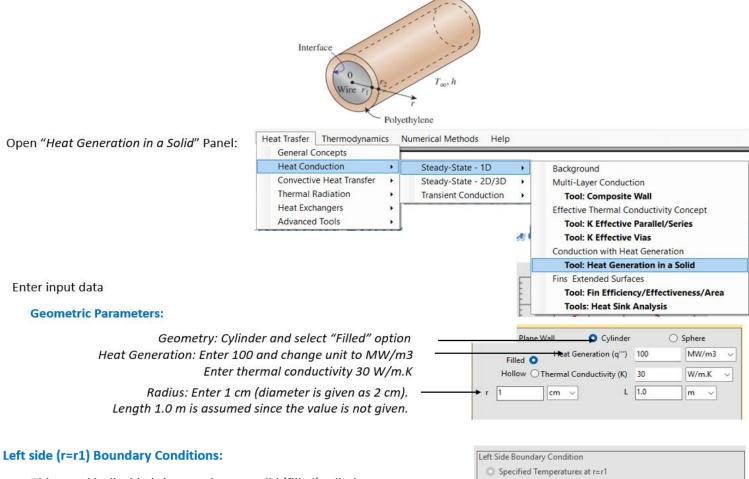


# Example: Electrical Resistance wire with Insulation [Source: Cengel-Ghagar Problem 2.105 -

#### Modified]

A long electrical resistance wire of radius r1 = 0.2 cm has a thermal conductivity  $k_w$ = 15 W/m.K. Heat is generated uniformly in the wire as a result of resistance heating at a constant rate of 1.2 W/cm<sup>3</sup>. The wire is covered with polyethylene insulation with a thickness of 0.5 cm and thermal conductivity of  $k_{ins}$  = 0.4 W/m.K. The outer surface of the insulation is subjected to convection and radiation with the surroundings at 20 °C. The combined convection and radiation heat transfer coefficients is 7 W/m<sup>2</sup>.K. Determine the temperature at the interface of the wire and the insulation and the temperature at the center of the wire. The ASTM D1351 standard specifies that thermoplastic polyethylene insulation is suitable for use on electrical wire with operation at temperatures up to 75C. Under these conditions, does the polyethylene insulation for the wire meet the ASTM D1351 standard?

#### Solution:



This panel is disabled since we have a solid (filled) cylinder.

Left Side Boundary Condition	
Specified Temperaturex at r=r1	
Ts1 C	·
O Prescribed Transport Coefficient and	Ambient Temperature at r=r1
U1 W/m2	K v
	-
<ul> <li>Uniform Heat Flux at r=r1</li> </ul>	
q"s W/m2	~

#### Right side (r=r2) Boundary Conditions:

The external surface of the wire is exposed to the ambient air through an intervening Polyethylene layer. Therefore, the transport coefficient  $U_2$  must account for both convective and conductive effects. The overall heat transfer coefficient can be found by first calculating the total thermal resistance for the combined system and then determining the UA and U2:

$$R_{th,2} = \frac{ln\left(\frac{r_2}{r_1}\right)}{2\pi Lk} + \frac{1}{(2\pi r_2 L)h_2} \implies UA = \frac{1}{R_{th,2}} \implies U_2 = \frac{UA}{(2\pi r_1 L)}$$

Alternatively, the Composite Solid tool may be used to evaluate  $U_2$  automatically.

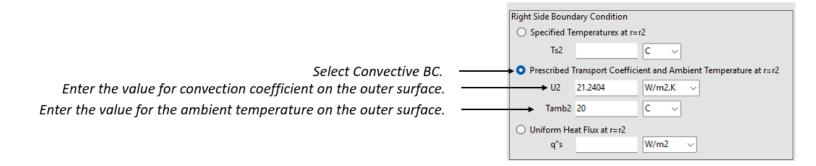
- Select Cylinder option.
- Set No. of Layer to "1"
- Click Initialize to set the form.
- Set r0 to 0.2 cm
- In the Layers table set the thermal conductivity to 0.4 W/m.K and thickness to 0.5 cm.
- In the left BC panel: set Ts to any temperature (30 °C).
   Note that the value of temperature does not matter since we are only interested in *R<sub>th</sub>* calculations.
- In the right BC panel: set h to 7 W/m<sup>2</sup>.K and  $T_{amb}$  to 30  $^{\circ}\mathrm{C}.$
- Click Update to solve.
- The UA value is calculated to be 0.2669 C/W
- The U\_Rmin value of 21.24 W/m<sup>2</sup>.K is the value we need for the analysis; it is the U value using the radius

nput				
O Plane Wall	<ul> <li>Cylinder</li> </ul>	O Sphere Initialize	r0 0.2 cm v L 1.00	m
Left BC (r=0)				
<ul> <li>Convection h Tamb</li> </ul>		W/m2.K ∨ C ∨	Thermal Contact Resistance	C.m2/W ~
• Fixed Temp Ts	30	C ~		Apply
⊖ Heat FLux q"s	i	W/m2 ~	Layers Thichness cm V k W/m.K	~
Right BC (r=R)			Layer Name Therm. Cond. Thickr	ness
Convection h	7	W/m2.K ~	▶ Layer_1 0.4 0.5	
Tamb	20	C ~		
○ Fixed Temp Ts		C ~		
O Heat FLux 9"s	i	W/m2 ~		
esults			Layers	
			X cm ~ T C ~ Rth C.m2/	w ~
qr 2.6691		W ~	r Left T Left Rth r Right	T Right
R"tot 3.7465		C/W ~	▶ 0.200 30.00 0.498 0.700	28.67
UA 0.2669		W/C ~		
U Rmin 21,2404		W/m2.K ~		



U_Rmax 6.0687 W/m2.K ~	U_Rmin 21.2404	W/m2.K ~	
	U_Rmax 6.0687	W/m2.K ~	

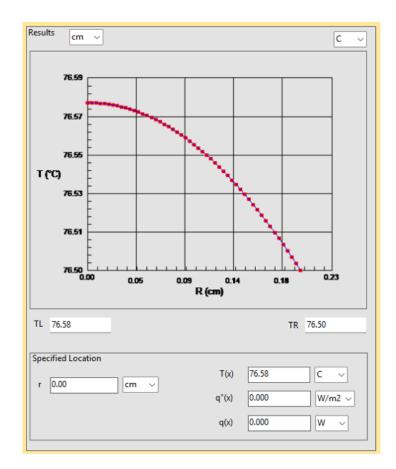
Now, we get back to "Heat Generation in a Solid" and enter the calculated vale of U2 in the Right-Side BC Panel.



#### Results

Click Update to solve.

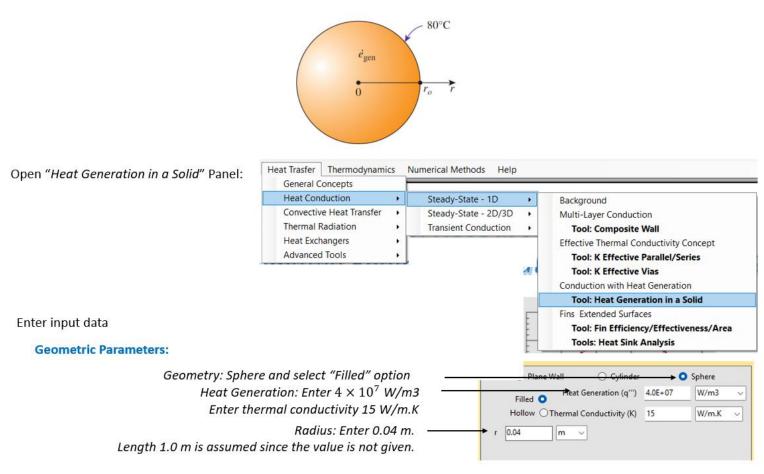
- 1. Plot of temperature distribution through the slab is provided.
- 2. Left-side (centerline) and Right-Side (r=1 cm) temperatures are shown.
- The user can select the units for temperature and length for the plot are from unit boxes at the top.
- 4. The surface temperature of 76.5 °C is 1.5 °C higher than the specification of the ASTM D1351 standard for polyethylene insulation.
- 5. There is very little temperature rise in the wire, as the center temperature is less than 0.1 °C higher than the interface temperature.



#### **Example:**

Consider a homogeneous spherical piece of radioactive material of radius r<sub>o</sub> = 0.04 m that is generating heat at a constant rate of  $e_{gen} = 4 \times 10^7 \text{ W/m}^3$ . The heat generated is dissipated to the environment steadily. The outer surface of the sphere is maintained at a uniform temperature of 80C, and the thermal conductivity of the sphere is k = 15 W/m.K. Assuming steady one-dimensional heat transfer, determine the temperature at the center of the sphere.

# Solution:

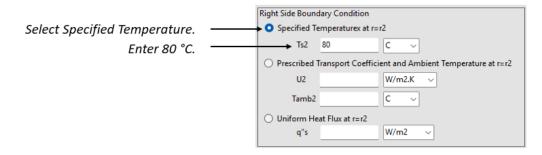


#### Left side (r=r1) Boundary Conditions:

This panel is disabled since we have a solid (filled) cylinder.

Left Side Boundar	y Condition						
Specified Temperaturex at r=r1							
Ts1	C v						
O Prescribed T	ransport Coefficient and Ambient Temperature at r=r1						
U1	W/m2.K 🗸						
Tamb1	C v						
O Uniform He	at Flux at r=r1						
q"s	W/m2 🗸						

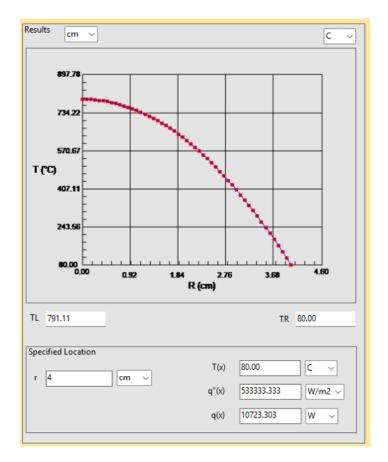
#### Right side (r=r2) Boundary Conditions:



#### Results

Click Update to solve.

- 1. Plot of temperature distribution through the slab is provided.
- Left-side (center) and Right-Side (r=4 cm) temperatures are shown.
- 3. The user can select the units for temperature and length for the plot are from unit boxes at the top.
- 4. The temperature at the center of the sphere is calculated to be 791.1 °C.
- 5. The surface heat flux is 533,333.3 W/m2 and the total heat is calculated to be 10,723.3 W.

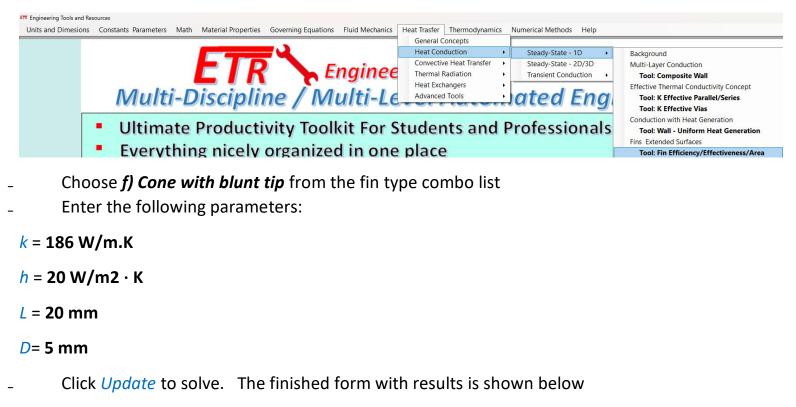


# HT-05: Fin Efficiency/Effectiveness

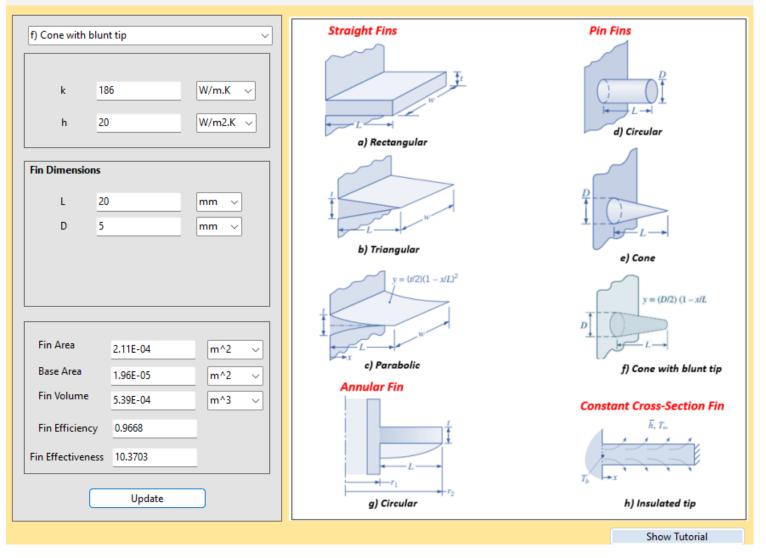
**Example:** A heat sink utilizes aluminum 2024-T6 (k =  $186 \text{ W/m} \cdot \text{K}$ ) pin fins of parabolic profile with blunt tips. Each fin has a length of 20 mm and a base diameter of 5 mm. If the fins are in an environment with heat transfer coefficient of 20 W/m2  $\cdot$  K, determine the efficiency and effectiveness for each fin.

# Solution:

Open "Fin Efficiency" Panel:







The fin efficiency is 96.67% and the effectiveness is calculated to be 10.37.

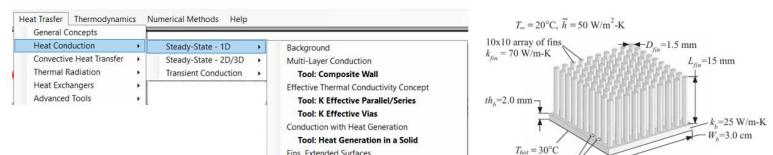
# HT-06: Heat Sink Analysis

# Example – THERMOELECTRIC HEAT SINK [Source: Nellis & Klein Example 1.6-2]

Heat rejection from a thermoelectric cooling device is accomplished using a 10 × 10 array of 1.5 mm diameter pin fins that are 15 mm long. The fins are attached to a square base plate that is 3 cm on each side and 2 mm thick, as shown below. The conductivity of the fin material is 70W/m-K and the thermal conductivity of the base material is 25W/m-K. There is a contact resistance of  $1\times10-4$  m<sup>2</sup>-K/W at the interface between the base of the fins and the base plate. The hot end of the thermoelectric cooler is at 30°C and the surrounding air temperature is 20°C. The average heat transfer coefficient between the air and the surface of the heat sink is h = 50 W/m2-K.

- a) What is the total thermal resistance between the hot end of the thermoelectric cooler and the air?
- b) What is the rate of heat rejection that can be accomplished under these conditions?

# Solution:





// - $R_c''=1 \times 10^{-4} \frac{\text{m}^2 - \text{K}}{\text{W}}$ 

			d) Circular	~	
	Heat Sink Input				
	<ul> <li>Rectangular Heat Sink</li> </ul>	O Circular Heat Sink	k 70	W/m.K v	
	Base Plate Length (L) 3	cm ~	h 50	W/m2.K ~	
	Base Plate Width (w) 3	cm v			
	Base Plate Thickness (t) 2	mm ~	Fin Dimensions		4-The Fin Tool will Automatical
lant Ciale la sut	Base Pltate Conductivity (kb) 25	W/m.K v	L 15	mm v	close, and the results will be entered into the Fin Details
leat Sink Input	Contact base-source (R"c,b) 0.00	C.m2/W ~	D 1.5	mm v	section.
	Number of Fins (Nfin) 100				
	Contact Base-Fin (R"c,fin) 0.0001	C.m2/W ~			Fin Details Fin surface area 7.07E-05 m
					Fin base area 1.77E-06 m
	Avg Heat Tran Coeff (h) 50	W/m2.K ~			Fin Efficiency 0.8780
	Ambient Temp. (Tamb) 20	C ~	Fin Area	m^2 ~	Use Fin T
	Fin Details Fin surface area	m^2 ~	Base Area	m^2 ~	<u>†</u>
	Fin base area		Fin Volume	m^3 ~	
	Fin Efficiency		Fin Efficiency		
		Use Fin Tool	Fin Effectiveness		
				Update	
	2-Click "Use Fin Tool" to	open the Fin Tool	3-Choose circular fi	and enter fin's	
	and get fin properties.	open ine i m iooi	thermal and geor	-	
			Click on Update		
frmHeatFink     Heat Sink Input     Rectangular Heat Si				Ì	×
Heat Sink Input Rectangular Heat S Base Plate Length (L) Base Plate Width (w)	3 cm v 3 cm v				
Heat Sink Input <ul> <li>Rectangular Heat S</li> <li>Base Plate Length (L)</li> </ul>	3 cm ~ 3 cm ~				
Heat Sink Input Rectangular Heat S Base Plate Length (L) Base Plate Width (w)	3 cm ~ 3 cm ~ 2 mm ~				
Heat Sink Input <ul> <li>Rectangular Heat S</li> <li>Base Plate Length (L)</li> <li>Base Plate Width (w)</li> <li>Base Plate Thickness (t)</li> </ul>	3     cm v       3     cm v       2     mm v       25     W/m.K v				
Heat Sink Input Rectangular Heat S Base Plate Length (L) Base Plate Width (w) Base Plate Thickness (t) Base Plate Conductivity (kb)	3     cm ~       3     cm ~       2     mm ~       25     W/m.K ~       0.00     C.m2/W ~				
Heat Sink Input Rectangular Heat S Base Plate Length (L) Base Plate Width (w) Base Plate Thickness (t) Base Plate Conductivity (kb) Contact base-source (R"c,b)	3     cm v       3     cm v       2     mm v       25     W/m.K v       0.00     C.m2/W v       100				
Heat Sink Input Rectangular Heat S Base Plate Length (L) Base Plate Width (w) Base Plate Thickness (t) Base Plate Conductivity (kb) Contact base-source (R"c,b) Number of Fins (Nfin) Contact Base-Fin (R"c,fin)	3     cm ~       3     cm ~       2     mm ~       25     W/m.K ~       0.00     C.m2/W ~       100		Redin Rem	$R_{fin} = \frac{1}{\eta_{fin}hA_{sfin}}  (0)$	thermal resistance – single fin)
Heat Sink Input Rectangular Heat S Base Plate Length (L) Base Plate Width (w) Base Plate Thickness (t) Base Plate Conductivity (kb) Contact base-source (R°c,b) Number of Fins (Nfin) Contact Base-Fin (R°c,fin) Avg Heat Tran Coeff (h)	3     cm v       3     cm v       2     mm v       25     W/m.K v       0.00     C.m2/W v       100     0.0001       0.0001     C.m2/W v       50     W/m2.K v		Restin Nfin Mfin Mfin	$R_{fin} = \frac{1}{\eta_{fin}hA_{xfin}} \qquad (t)$ $R_{unf(nnsd)} = \frac{1}{h(A_b - N_{tri})}$	thermal resistance – single fin)
Heat Sink Input Rectangular Heat S Base Plate Length (L) Base Plate Width (w) Base Plate Thickness (t) Base Plate Conductivity (kb) Contact base-source (R"c,b) Number of Fins (Nfin) Contact Base-Fin (R"c,fin) Avg Heat Tran Coeff (h) Ambient Temp. (Tamb)	3     cm ~       3     cm ~       2     mm ~       25     W/m.K ~       0.00     C.m2/W ~       100	Reb Rec	Restin Refin Nytin Nytin	$R_{fin} = \frac{1}{\eta_{fin}\bar{h}A_{s,fin}} \qquad (t)$ $R_{unfinned} = \frac{1}{\bar{h}(A_b - N_{fi})}$	$\frac{1}{\frac{1}{\frac{1}{\frac{1}{\frac{1}{\frac{1}{\frac{1}{\frac{1}$
Heat Sink Input Rectangular Heat S Base Plate Length (L) Base Plate Width (w) Base Plate Thickness (t) Base Plate Conductivity (kb) Contact base-source (R°c,b) Number of Fins (Nfin) Contact Base-Fin (R°c,fin) Avg Heat Tran Coeff (h)	3     cm ~       3     cm ~       2     mm ~       25     W/mK ~       0.00     C.m2/W ~       100     0.0001       0.0001     C.m2/W ~       50     W/m2.K ~       20     C ~		md b	$R_{fin} = \frac{1}{\eta_{fin} \hbar A_{s,fin}} $ (t) $R_{unfinned} = \frac{1}{\hbar (A_b - N_{fi})}$ $R_{cond,b} = \frac{t_b}{k_b A_b} $ (t)	$\frac{1}{10000000000000000000000000000000000$
Heat Sink Input  Rectangular Heat S Base Plate Length (L) Base Plate Width (w) Base Plate Thickness (t) Base Plate Conductivity (kb) Contact base-source (R"c,b) Number of Fins (Nfin) Contact Base-Fin (R"c,fin) Avg Heat Tran Coeff (h) Ambient Temp. (Tamb) Fin Details Fin surface area	3     cm ~       3     cm ~       2     mm ~       25     W/mK ~       0.00     C.m2/W ~       100     0.0001       0.0001     C.m2/W ~       50     W/m2.K ~       20     C ~	$R_{c,b}$ $R_{cc}$ $T_{hot}$	md b	$R_{fin} = \frac{\eta_{fin} \bar{h} A_{sfin}}{\eta_{fin} \bar{h} A_{sfin}} \qquad (t)$ $R_{unfinned} = \frac{1}{\bar{h} (A_b - N_{fi})}$ $R_{cond,b} = \frac{t_b}{k_b A_b} \qquad (t)$ $R_{c,b}, R_{c,fin} \qquad (t)$	$\frac{1}{n^{A}c_{d}(n)}$
Heat Sink Input  Rectangular Heat S Base Plate Length (L) Base Plate Width (w) Base Plate Thickness (t) Base Plate Conductivity (kb) Contact base-source (R°,c,b) Number of Fins (Nfin) Contact Base-Fin (R°,c,fin) Avg Heat Tran Coeff (h) Ambient Temp. (Tamb) Fin Details Fin surface area Fin base area	3     cm ∨       3     cm ∨       2     mm ∨       25     W/m.K ∨       0.00     C.m2/W ∨       100     0.0001       0.0001     C.m2/W ∨       50     W/m2.K ∨       20     C ∨       7.07E-05     m^2 ∨       1.77E-06     m^2 ∨	R <sub>c,b</sub> R <sub>cc</sub>	md b	$\begin{aligned} R_{fin} &= \frac{1}{\eta_{fin} \overline{h} A_{s,fin}} & (t) \\ R_{unfinned} &= \frac{1}{\overline{h} (A_b - N_{fi})} \\ R_{cond,b} &= \frac{t_b}{k_b A_b} & (t) \\ R_{c,b}, R_{c,fin} & (c) \\ A_{s,fin} &\equiv Fin \ total \ surface \end{aligned}$	$\frac{1}{1+$
Heat Sink Input  Rectangular Heat S Base Plate Length (L) Base Plate Width (w) Base Plate Thickness (t) Base Plate Conductivity (kb) Contact base-source (R°,c,b) Number of Fins (Nfin) Contact Base-Fin (R°,c,fin) Avg Heat Tran Coeff (h) Ambient Temp. (Tamb) Fin Details Fin surface area Fin base area	3     cm ∨       3     cm ∨       2     mm ∨       25     W/m.K ∨       0.00     C.m2/W ∨       100     0.0001       0.0001     C.m2/W ∨       50     W/m2.K ∨       20     C ∨       7.07E-05     m^2 ∨       0.8780     0.8780	$R_{c,b}$ $R_{cc}$ $T_{hot}$	nd,b	$R_{fin} = \frac{1}{\eta_{fin} \overline{h} A_{xfin}}  (t)$ $R_{unfinned} = \frac{1}{\overline{h} (A_b - N_{fi})}$ $R_{cond,b} = \frac{t_b}{k_b A_b}  (t)$ $R_{c,b}, R_{c,fin}  (c)$ $A_{x,fin} \equiv Fin \text{ total surface}$ $A_{c,fin} \equiv Fin \text{ cross-section}$	$\frac{1}{1+$
Heat Sink Input  Rectangular Heat S Base Plate Length (L) Base Plate Width (w) Base Plate Thickness (t) Base Plate Conductivity (kb) Contact base-source (R°,c,b) Number of Fins (Nfin) Contact Base-Fin (R°,c,fin) Avg Heat Tran Coeff (h) Ambient Temp. (Tamb) Fin Details Fin surface area Fin base area	3     cm ∨       3     cm ∨       2     mm ∨       25     W/m.K ∨       0.00     C.m2/W ∨       100     0.0001       0.0001     C.m2/W ∨       50     W/m2.K ∨       20     C ∨       7.07E-05     m^2 ∨       1.77E-06     m^2 ∨	$R_{c,b}$ $R_{cc}$ $T_{hot}$		$\begin{aligned} R_{fin} &= \frac{1}{\eta_{fin} \overline{h} A_{s,fin}} & (t) \\ R_{unfinned} &= \frac{1}{\overline{h} (A_b - N_{fi})} \\ R_{cond,b} &= \frac{t_b}{k_b A_b} & (t) \\ R_{c,b}, R_{c,fin} & (c) \\ A_{s,fin} &\equiv Fin \ total \ surface \end{aligned}$	$\frac{1}{1+$
Heat Sink Input  Rectangular Heat S Base Plate Length (L) Base Plate Width (w) Base Plate Thickness (t) Base Plate Conductivity (kb) Contact base-source (R°,c,b) Number of Fins (Nfin) Contact Base-Fin (R°,c,fin) Avg Heat Tran Coeff (h) Ambient Temp. (Tamb) Fin Details Fin surface area Fin base area	3     cm ∨       3     cm ∨       2     mm ∨       25     W/m.K ∨       0.00     C.m2/W ∨       100     0.0001       0.0001     C.m2/W ∨       50     W/m2.K ∨       20     C ∨       7.07E-05     m^2 ∨       0.8780     0.8780	$R_{c,b}$ $R_{cc}$ $T_{hot}$		$R_{fin} = \frac{1}{\eta_{fin} \overline{h} A_{xfin}}  (t)$ $R_{unfinned} = \frac{1}{\overline{h} (A_b - N_{fi})}$ $R_{cond,b} = \frac{t_b}{k_b A_b}  (t)$ $R_{c,b}, R_{c,fin}  (c)$ $A_{x,fin} \equiv Fin \text{ total surface}$ $A_{c,fin} \equiv Fin \text{ cross-section}$	$\frac{1}{1+$
Heat Sink Input  Rectangular Heat S Base Plate Length (L) Base Plate Width (w) Base Plate Thickness (t) Base Plate Conductivity (kb) Contact base-source (R°c,b) Number of Fins (Nfin) Contact Base-Fin (R°c,fin) Avg Heat Tran Coeff (h) Ambient Temp. (Tamb) Fin Details Fin surface area Fin Efficiency	3     cm ∨       3     cm ∨       2     mm ∨       25     W/m.K ∨       0.00     C.m2/W ∨       100     0.0001       0.0001     C.m2/W ∨       50     W/m2.K ∨       20     C ∨       7.07E-05     m^2 ∨       0.8780     Use Fin Tool	$R_{c,b} = R_{cc}$ $T_{hot} \bullet \bigcirc $	nd,b	$R_{fin} = \frac{1}{\eta_{fin} \overline{h} A_{xfin}}  (t)$ $R_{unfinned} = \frac{1}{\overline{h} (A_b - N_{fi})}$ $R_{cond,b} = \frac{t_b}{k_b A_b}  (t)$ $R_{c,b}, R_{c,fin}  (c)$ $A_{x,fin} \equiv Fin \text{ total surface}$ $A_{c,fin} \equiv Fin \text{ cross-section}$	$\frac{1}{1+$
Heat Sink Input  Rectangular Heat S Base Plate Length (L) Base Plate Length (L) Base Plate Thickness (t) Base Plate Conductivity (kb) Contact base-source (R"c,b) Number of Fins (Nfin) Contact Base-Fin (R"c,fin) Avg Heat Tran Coeff (h) Ambient Temp. (Tamb) Fin Details Fin surface area Fin Efficiency Results	3     cm ∨       3     cm ∨       2     mm ∨       25     W/m.K ∨       0.00     C.m2/W ∨       100     0.0001       0.0001     C.m2/W ∨       50     W/m2.K ∨       20     C ∨       7.07E-05     m^2 ∨       0.8780     Use Fin Tool		nd,b	$R_{fin} = \frac{\eta_{fin} \bar{h} A_{sfin}}{\eta_{fin} \bar{h} A_{sfin}} \qquad (t)$ $R_{unfinned} = \frac{1}{\bar{h} (A_b - N_{fi})}$ $R_{cond,b} = \frac{t_b}{k_b A_b} \qquad (t)$ $R_{c,b}, R_{c,fin} \qquad (c)$ $A_{s,fin} \equiv Fin \ total \ surface$ $A_{c,fin} \equiv Fin \ cross-section$ $A_b \equiv Baseplate \ Area$	$\frac{1}{n^{A}c_{d}(n)}$
Heat Sink Input  Rectangular Heat S Base Plate Length (L) Base Plate Width (w) Base Plate Thickness (t) Base Plate Conductivity (kb) Contact Base-source (R"c,b) Number of Fins (Nfin) Contact Base-Fin (R"c,fin) Avg Heat Tran Coeff (h) Ambient Temp. (Tamb) Fin Details Fin surface area Fin Efficiency  Results Total Thermal Resistance (I)	3     cm ∨       3     cm ∨       2     mm ∨       25     W/m.K ∨       0.00     C.m2/W ∨       100     0.0001       0.0001     C.m2/W ∨       50     W/m2.K ∨       20     C ∨       7.07E-05     m^2 ∨       0.8780     Use Fin Tool	Source Temperature (The	$\begin{array}{c} \text{nd,b} \\ & \\ & \\ & \\ & \\ & \\ & \\ & \\ & \\ & \\ $	$R_{fin} = \frac{\eta_{fin} \bar{h} A_{sfin}}{\eta_{fin} \bar{h} A_{sfin}} \qquad (t)$ $R_{unfinned} = \frac{1}{\bar{h} (A_b - N_{fi})}$ $R_{cond,b} = \frac{t_b}{k_b A_b} \qquad (t)$ $R_{c,b}, R_{c,fin} \qquad (c)$ $A_{s,fin} \equiv Fin \ total \ surface$ $A_{c,fin} \equiv Fin \ cross-section$ $A_b \equiv Baseplate \ Area$	$\frac{1}{n^{A}c_{d}(n)}$

Note: The above results are identical to those in Nellis-Klein as obtained using EES program.

# HT-07: Multidimensional Heat Transfer in Common Configurations

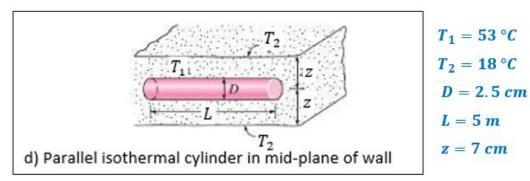
**Example:** Hot water at 53 °C flows through a 5-m long section of a thin-walled hot water pipe at an average velocity of 5 m/s. The pipe passes through the center of a 14-cm thick wall filled with fiberglass (k =  $0.035 \text{ W/m} \cdot \text{K}$ ) insulation. Assuming the surfaces of the wall are at 18 °C, determine a) the rate of heat transfer from the pipe to the air in the rooms and b) the temperature drop of the hot water as it flows through the pipe are to be determined.

#### Solution:

# Open "Common Configuration/Shape Factor" Panel:

		Numerical Methods <u>H</u> elp	
1	General Concepts		
	Heat Conduction	Steady-State - 1D	
	Convective Heat Transfer	Steady-State - 2D/3D  Background	
4	Thermal Radiation	Transient Conduction   Common Configurations	
1	Heat Exchangers	Tool: 2D Common Configurations	
	Advanced Tools		

Choose configuration *f*) from the Shape Factors combo list



Enter input in given units:

Dimensions		
D	2.5	cm 🗸
L	5	m ~
z	7	cm 🗸
Thermal Input	t	
k	0.035	W/m.K v
		W/III.K V
T1	53	C ~

a) Click *Update* to the rate of heat transfer from the pipe to the air in the rooms.

Results		
S	15.993	m ~
Rth	1.786	K/W ~
Q12	19.592	W ~
	Update	

The results (shown above) indicate the heat loss from the pipe is 19.6 W.

b) to calculate the temperature drop, we will perform an energy balance on the pipe:

$$\dot{Q} = \dot{m}c_p \Delta T$$

$$\Delta T = \frac{\dot{Q}}{\dot{m}c_p} = \frac{\dot{Q}}{\rho \dot{V}c_p} = \frac{\dot{Q}}{\rho VA_c c_p} = \frac{\dot{Q}}{(1000 \text{ kg/m}^3)(0.4 \text{ m/s})} \left[\frac{\pi (0.025 \text{ m})^2}{4}\right] (4180 \text{ J/kg. °C}) = 0.024^{\circ}\text{C}$$

<<End-of-Tutorial>>

# HT-08: Lumped Capacity (Constant Ambient Temperature/ Varying Ambient Temperature)

# Example – Thermal Response of Thermocouple [Source: Bergman-Lavine Example 5.1]

A thermocouple junction, which may be approximated as a sphere, is to be used for temperature measurement in a gas stream. The convection coefficient between the junction surface and the gas is h = 400 W/m2.K, and the junction thermophysical properties are k = 20 W/m.K, c = 400 J/kg.K, and  $\rho = 8500 \text{ kg/m3}$ . Assuming the junction diameter of the thermocouple to be 0.8 mm and initial temperature,  $T_i=25$  °C.

- a) temperature versus time for eight seconds.
- b) Determine the junction temperature at time t=1 s.
- c) Determine how long it will take for the junction to reach 198 °C.

# Solution:

	Thermodynamics Numerical Me	thods <u>H</u> elp		Leads
General Concepts			$T_{\infty} = 200^{\circ}\text{C}$	$\left  \right  / 2$
Heat Conduction	Steady-State - 1D 🔹	1	$\tilde{h} = 400 \text{ W/m}^2 \cdot \text{K}$	Thermocouple $k = 20 \text{ W/m} \text{-K}$ junction $c = 400 \text{ J/kg} \text{-K}$
Convective Heat Transfer	Steady-State - 2D/3D 🔸		$\rightarrow$	$T_i = 25^{\circ}\text{C} \qquad \int \rho = 8500 \text{ kg/m}^3$
Thermal Radiation	Transient Conduction	Lumped Capacitance Method	Gas stream	
Heat Exchangers		Tool: Lumped Capacitance	Gas stream	
Advanced Tools		Spatial Variation		
		Tool: Spatial Variation		

Enter input data

1. Click on "Choose 3D Shape" button to open the 3D shape panel area and volume calculation.

Geometric Parameters Choose 3D Shape	Input		Geometric Param	Choose 3D S	hape
Area         m^2 ~           Volume         m^3 ~	f-Solid Sphere v 2 0.4 mm v		(3)	2.0106e-06 m^2 2.6808e-10 m^3	~
<ol><li>Choose option "f" for Solid Sphere and enter the radius.</li></ol>		I	Material Library A	4 Aluminum_Pure	~
<ol> <li>Click update to automatically enter these parameters into "Lumped Capacity Form".</li> </ol>	4. Enter solid thermal prop	perties.	User		ı/m3 ∨ kg.K ∨

W/m.K

k 20

5) In "Ambient Conditions" section:

- Enter values for heat transfer coefficient and initial temperature.
- Keep Constant *T<sub>inf</sub>* option "on" and enter the ambient temperature

Ambient Conditions							
Heat Trans. Coeff.	400	W/m2.K 🗸					
Initial Temp	25	C ~					
<ul> <li>Constant Tinf</li> </ul>	200	C ~					
○ Time Depende	nt with Heat Gener	ration					

s

Update

 $\sim$ 

Plot Temp vs time

8

End Time

6) In "Plot Temp vs Time" section:

• Enter 8 second for "End Time" and click on Update.

#### Part b):

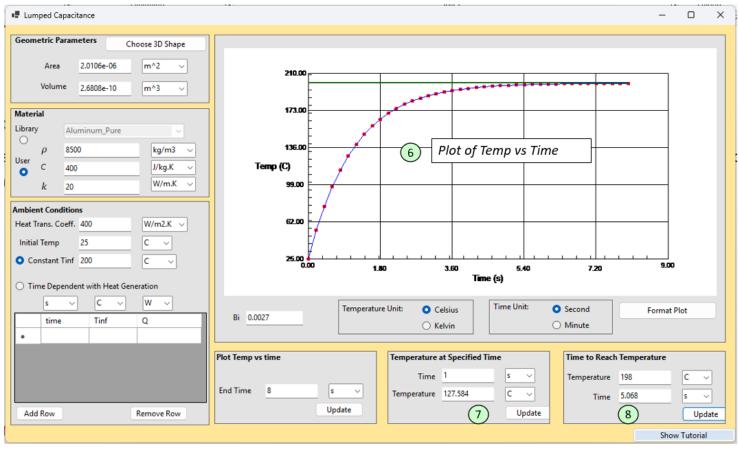
7) To obtain *temperature at 1 second* enter 1 s for time and click update.

#### Part c):

8) To obtain *Time to reach 198 °C* enter 198 °C for temperature and click update.

Temperature at Specified Time							
Time	1	s ~					
Temperature		<mark>C ~</mark>					
		Update					
Time to Reach	Temperature						
Temperature	198	<b>C</b> ~					
Time		5 V					
		Update					

#### Finished form is shown below.

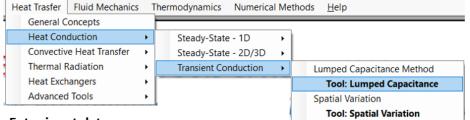


*Note*: the value of Bi = 0.0027 << 0.1, indicates lumped capacitance is assumption is valid.

# **Example – Thermal Response of Thermocouple in a Thermal Chamber**

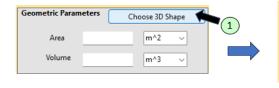
Consider a 5-inch solid metallic ball made of Aluminum\_6061-T6 at initially at 85 °C in a thermal chamber with a heat transfer coefficient h=250 W/mK. Plot the sphere temperature versus time with varying chamber temperature profile shown below.

Time (min)	Temperature (°C)
0	85
10	-50
30	-50
40	85
60	85

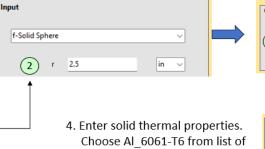


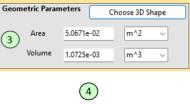
#### Enter input data

1. Click on "Choose 3D Shape" button to open the 3D shape panel area and volume calculation.



- 2. Choose option "f" for Solid Sphere and enter the radius.
- 3. Click update to automatically enter these parameters into "Lumped Capacity Form".





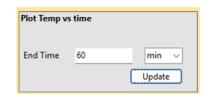
materials in the library.

Mater	rial		
Libran	у	Aluminum_606	1_Temper-T6 🗸 🗸 🗸
-	ρ	2710	kg/m3 v
User	С	1256	J/kg.K 🗸
Ŭ	k	167	W/m.K ~

5) In "Ambient Conditions" section:

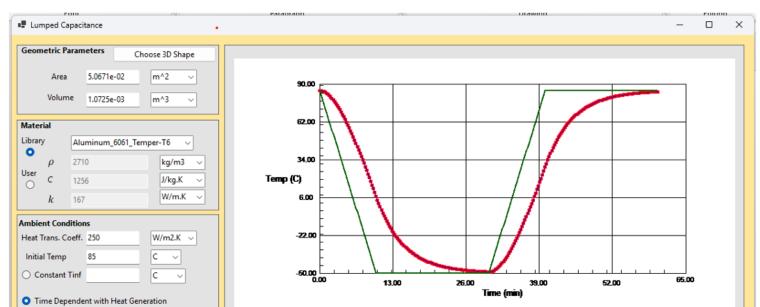
- Enter values for heat transfer coefficient and initial temperature.
- Change T<sub>inf</sub> option to "Time Dependent" and enter values for time and chamber temperature from profile table above. Make sure to change the time unit to minutes.

Ambient Conditions								
Heat Trans. Coeff.	250	W/m2.K 🗸						
Initial Temp	85	C ~						
O Constant Tinf		C ~						
• Time Dependent with Heat Generation								
min 🗸	<b>C</b> ~	W ~						
time	Tinf	Q						
0	85							
10	-50							
30	-50							
40	85							
Add Row Remove Row								



6) In "Plot Temp vs Time" section:

- Enter 60 minutes for "End Time"
- Change Time unit in the plot area to minutes
- click on Update.



Finished form is shown below.

	min ~	C ~	W ~		Temperatur	e Unit: O Celsius	Time Unit: (	Second	Format Plot
	time	Tinf	Q	Bi 0.0317		Kelvin	→ (	Minute	
	0	85				0 1121111			
	10	-50		Plot Temp, vs time		Temperature at Specified	I Time	Time to Reach	Temperature
	30	-50				Time	s ~		
_	40	85		End Time 60	min ~	Temperature	° ⊂ ⊂	Temperature Time	C ~
Add	Row		Remove Row		Update		Update		Update
									Show Tutorial

Note: the value of Bi = 0.037 << 0.1, indicates lumped capacitance is assumption is valid.

#### HT-09: Transient Conduction (Walls, Cylinders, Spheres)

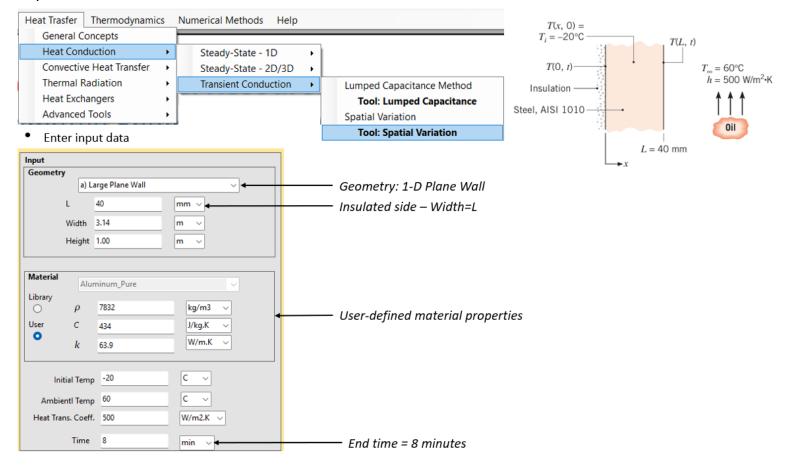
#### Example – Plate: Heating of a Steel Pipeline [Source: Bergman-Lavine Example 7.7]

Consider a 1-m diameter steel pipeline (AISI 1010) with a wall thickness of 40 mm. The pipe is heavily insulated on the outside, and is initially at a uniform temperature of -20°C. With the initiation of flow, hot oil at 60°C is pumped through the pipe, creating a convective condition corresponding to h = 500 W/m2  $\cdot$  K at the inner surface of the pipe.

- a. Determine the temperature of the exterior pipe surface covered by the insulation at t = 8 min?
- b. How much energy per meter of pipe length has been transferred from the oil to the pipe at t = 8 min?

# Solution:

#### • Open "Transient Conduction" Tool



# Click Update and Solve

Transient Conduction  $\times$ \_ Input Geometry 2La) Large Plane Wall 40 mm 🗸 Width 3.14 m  $\sim$ 0  $r_0$ 0 7 L Height 1.00 m  $\sim$ Material Aluminum\_Pure b) Long Cylinder c) Sphere Library a) Large Plane Wall 7832 ρ kg/m3  $\sim$ User С J/kg.K 434 0 W/m.K  $\sim$ k 63.9 С  $\sim$ -20 Initial Temp 2H2HC  $\sim$ 60 Ambientl Temp Heat Trans, Coeff. 500 W/m2.K  $\sim$ Time 8 min /2W 2WResults Center Temp 43.13 0.313 C d) Long Rectangular Bar e) Rectangular Bar f) Cylinder Bi  $\sim$ Fo 5.640 Total Energy Gain -5.457E+07 J  $\sim$ Internal Temp x/L Update 1.0



Comments:

Temp

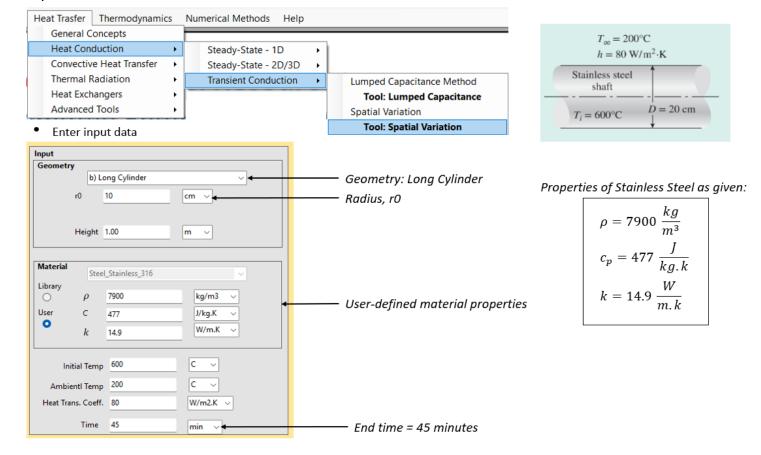
45.46

- 1. The Biot number, *Bi*=0.313 and the Fourier number Fo=5.64
- 2. The "*Center Temperature*" refers to the temperature at the center of a wall with thickness 2L. Here, it indicates the temperature at the insulated side of the wall.
- 3. The "Total Energy Gain" of -5.457E+07 J is the energy gained by a wall of thickness 2L during the period of 8 minutes. Here, since we the wall has an insulated side (L), the total energy gain will be half of this value; Q<sub>tot</sub>=-2.73E+07 J per meter of the pipe. The negative sign implies heat transfer into the pipe from the oil
- 4. Temperature at x/L=1 (45.46 °C) represents the temperature on the surface that is exposed to the oil.

# Example – Long Cylinder: Cooling of a Long Stainless-Steel Cylindrical Shaft [Source: Cengel-Ghagar Example 4.4]

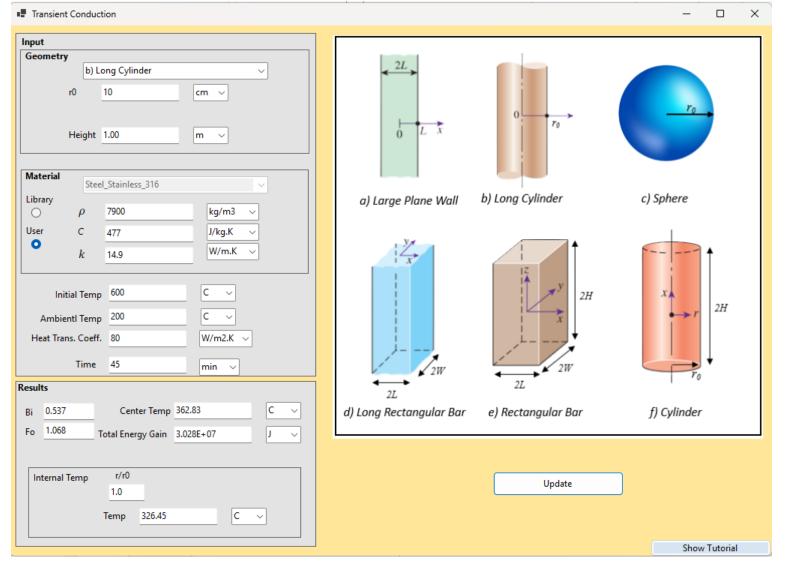
A long 20-cm diameter cylindrical shaft made of stainless-steel 304 is out of an oven at a uniform temperature of 600 °C. The shaft is then allowed to cool in an environmental chamber at 200 °C and a heat transfer coefficient of 80 W/m<sup>2</sup>.K. Determine a) the temperature at the center of the shaft after 45 minutes and b) total amount of heat transferred from the shaft to the air during the cooling process per unit length.

#### Solution:



#### Open "Transient Conduction" Tool

#### Click Update and Solve



Comments:

- 1. The Biot number, *Bi*=0.537and the Fourier number Fo=1.068
- 2. The "Center Temperature" refers to the temperature at the center of the cylinder is 362.83 °C.
- 3. The "Total Energy Gain"; Q<sub>tot</sub>=3.028E+07 J per meter of the pipe.
- 4. The surface temperature (r/r0=1) is calculated to be 326.45.

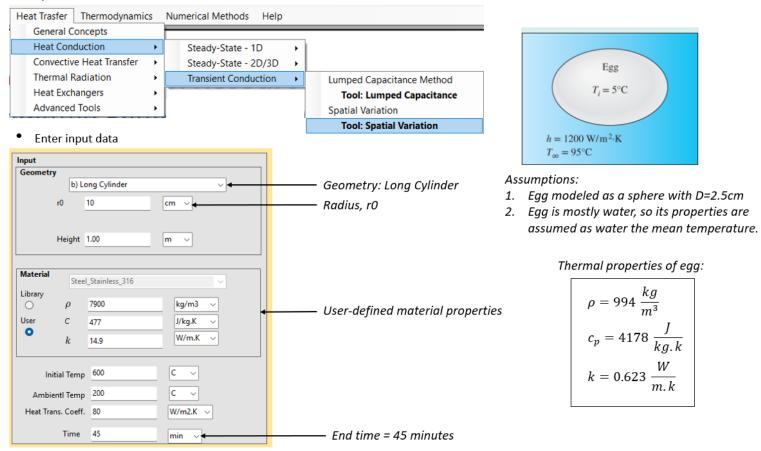
#### **Example – Sphere: Boiling of Egg**

An egg may be modeled as a spherical object of 5-cm diameter ( $\rho = 1000 \text{ kg/m3}$ ,  $cp = 4150 \text{ J/kg.} \circ C$ , k = 0.627 W/m.K). Determine the temperature at the center of an egg after 14 minutes in boiling

water at 95 °C. Assume the egg is initially at 5 °C and the convection heat transfer coefficient of the boiling water to be 1200  $W/m^2$ .K.

#### Solution:

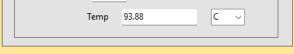
#### • Open "Transient Conduction" Tool



#### Click Update and Solve

Transient Conduction  $\times$ Input Geometry 2Lc) Sphere  $\sim$ r0 2.5  $\sim$ cm  $r_0$ 0 X L Material Aluminum\_Pure Library a) Large Plane Wall b) Long Cylinder c) Sphere 994 kg/m3 ρ  $\sim$ User С J/kg.K 4178 0 W/m.K  $\sim$ k 0.623 С  $\sim$ Initial Temp 5 2H2HС Ambientl Temp 95  $\sim$ Heat Trans. Coeff. 1200 W/m2.K 🗸 Time 14 2W min 2W $\sim$ 2LResults 27. 48.154 Center Temp 71.71 C e) Rectangular Bar Bi  $\sim$ d) Long Rectangular Bar f) Cylinder Fo 0.202 Total Energy Gain -2.244E+04 r/r0 Internal Temp Update 1.0

Show Tutorial



Comments:

- 1. The Biot number, *Bi*=48.15 and the Fourier number Fo=0.202
- 2. The "Center Temperature" is 71.7 °C.
- 3. The surface temperature of the egg is calculated to be 93.9 °C.
- 4. The "Total Energy Gain";  $Q_{tot}$ =2.244E+04 J into the egg.
- 5. The surface temperature (r/r0=1) is calculated to be 93.88.

#### Example – Short Cylinder: Cooling of a Brass Bar

A short 10-cm diameter brass cylinder with the height H = 12 cm is initially at a uniform temperature  $T_i = 120$ C. The cylinder is now placed in atmospheric air at 25°C, where heat transfer takes place by Convection, with a heat transfer coefficient of h = 60 W/m2.K. Calculate the temperature at (a) the

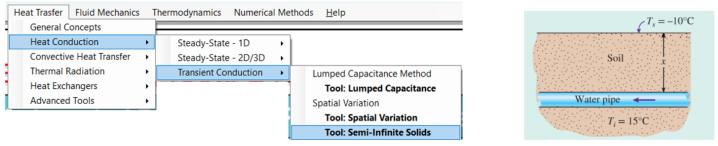
center of the cylinder and (b) the center of the top surface of the cylinder 15 min after the start of the cooling.

# Solution:

# HT-10: Semi-Infinite Solids

#### **Example – Burial Depth of Pipes to Avoid Freezing**

The ground at a particular location is covered with snowpack at -10°C for a continuous period of three months (90 × 24 = 2160 *hours*), and the average soil properties at that location are k = 0.4 W/m.K,  $\rho$ =8333.3 kg/m<sup>3</sup>, and cp =320 J/kg.K. Assuming an initial uniform temperature of 15 °C for the ground, determine a water pipe will freeze at the burial depth of 80 cm during the 3 month period. Repeat the simulation using the depth of 500 cm and notice.



#### Enter input data

Mate	rial		
Librar	у	Aluminum_Pure	~
0	ρ	8333.3	kg/m3 v
User	С	320	J/kg.K 🗸
	k	0.4	W/m.K v

- 1. Select "User" material type.
- 2. Enter density, specific heat and thermal conductivity.

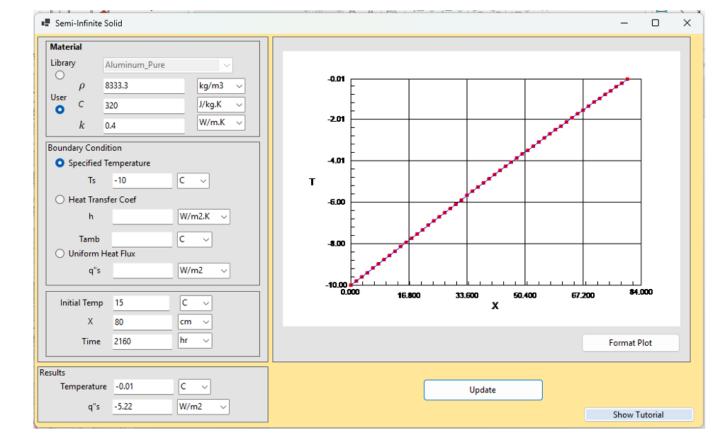
Boundary Condition						
<ul> <li>Specified Temperature</li> </ul>						
Ts -10	C ~					
○ Heat Transfer Coef						
h	W/m2.K 🗸					
Tamb	C ~					
O Uniform Heat Flux						
q"s	W/m2 ~					

3. Choose "Specified Temperature" BC and enter the given value.

W	ater pipe	←	
	$T_i =$	15°C	
Initial Temp	15		

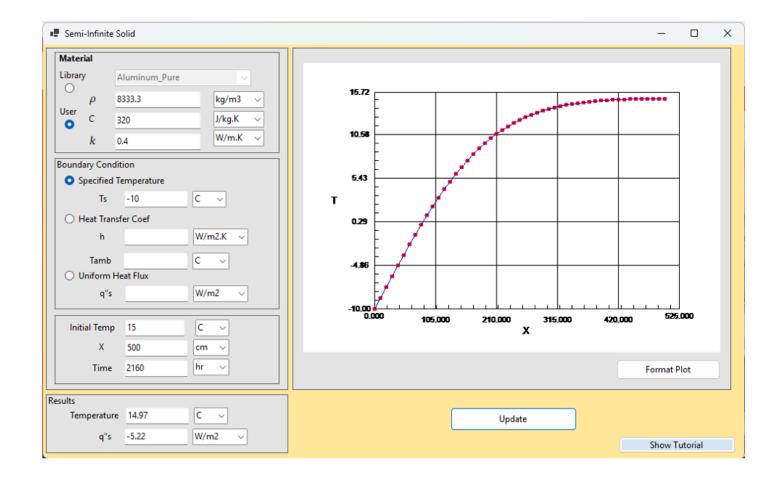
Initial Temp	15	C ~
x	80	cm 🗸
Time	2160	hr 🗸

 Enter values for Initial Temperature, Depth into the soil, and time.



Finished form is shown below.

Note: 1. The value of 80 cm is the minimum burial depth of pipes, since the pipe temperature reaches 0 °C after 3 months.
2. As evident from the plot, the at this depth the variation is nearly linear. If you rerun with x=500 cm, we can see the decay.



# **Example – Surface temperature rise of Heated Block**

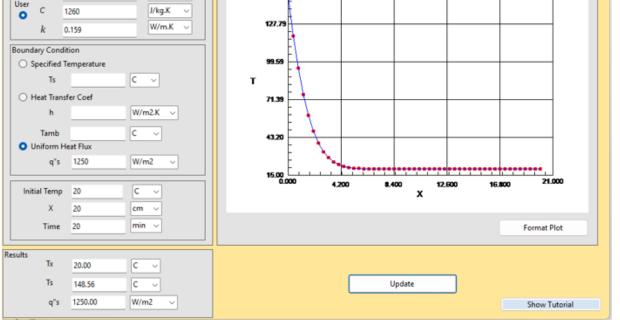
A thick, black-painted wood block ( $k = 0.159 \frac{W}{m.K}$ ,  $\rho = 721 \frac{m^3}{kg}$ ,  $c_p = 1260 \frac{J}{kg.K}$ ) at 20 °C is subjected to constant solar heat flux of 1250 W/m2.

- 1) Determine the exposed surface temperature of the block after 20 minutes. Plot results to 20 cm into the block.
- 2) Repeat for block made of pure aluminum, and Plot results to 100 cm into the block.

Convective Heat Transfer  Thermal Radiation Heat Exchangers Advanced Tools	ics Numerical Methods Help istate - 1D • istate - 2D/3D • t Conduction • Lumped Capacitance Method Tool: Lumped Capacitance Spatial Variation Tool: Spatial Variation Tool: Semi-Infinite Solids	$\dot{q}_i = 1250 \text{ W/m}^2$ Wood block $T_i = 20^{\circ}\text{C}$
Enter input data Material Library Aluminum Pure p 721 User C 1260 k 0.159 W/m.K V 1. Select "User" material type. 2. Enter density, specific heat and thermal conductivity.	Boundary Condition         Specified Temperature         Ts       C ~         Heat Transfer Coef         h       W/m2.K ~         Tamb       C ~         O Uniform Heat Flux         q"s       1250         W/m2 ~       S. Choose "Uniform Heat Flux" BC and enter the given value.	Initial Temp       20       C         X       0       cm         Time       20       min         4. Enter values for Initial       Temperature, Depth into the soil         (0 for surface), and time.

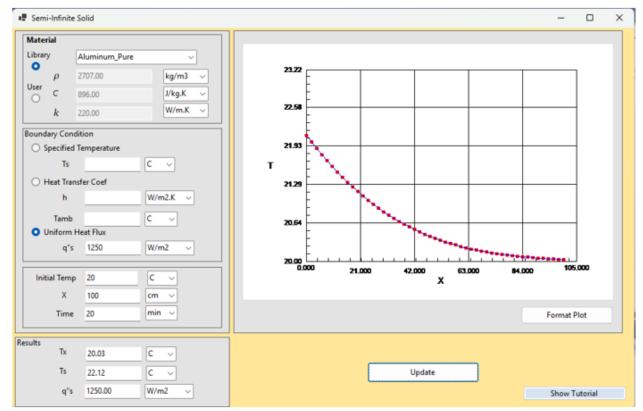
Finished form is shown below.

🖷 Semi-Infinite	Solid				-		×
Material Library							
	Aluminum_Pure	✓ kg/m3 ✓	155.99			1	



Wood Block: 1. The surface temperature is 148.6 °C.

2. The plot shows thermal penetration within the wooden block to the depth of 4.2 cm.



Aluminum: 1. The surface temperature is 22.12 °C.
 2. The plot shows thermal penetration within the wooden block to the depth of 100 cm.

# Example – Compliance of ASME Codes for Bolts Exposed to Cryogenic Fluid

A series of long stainless-steel bolts (ASTM A437 B4B) are fastened into a thick metal plate. The metal plate has a thermal conductivity of 16.3 W/m.K, a specific heat of 500 J/kg .K, and a density of 8 g/cm3. The upper surface of the plate is occasionally exposed to cryogenic fluid at -70°C with a convection heat transfer coefficient of 300 W/m2.K. The bolts are fastened into the metal plate from the bottom surface, and the distance measured from the plate's upper surface to the bolt tips is L = 1 cm. The ASME Code for Process Piping limits the minimum suitable temperature for ASTM A437 B4B stainless steel bolt to -30 °C. If the initial temperature of the plate is 10°C and the plate's upper surface is exposed to the cryogenic fluid for 30 minutes, would the bolts fastened in the plate still comply with the ASME code?

# HT-11: Contact of Two Semi-Infinite Solids

# HT-12: Flat Plate in Parallel Flow

# Example – Flow of Hot Oil Over a Flat Plate

Engine oil at 60°C flows over the upper surface of a 5-m-long flat plate whose temperature is 20°C with a velocity of 2 m/s. Determine the total drag force and the rate of heat transfer per unit width of the entire plate.

# HT-13: Flow Over 3D Bodies

# **Example – Heating of Horizontal Cylinder in Cross Flow**

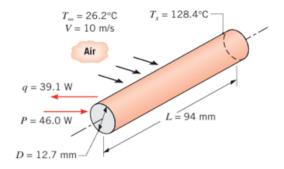
Experiments have been conducted using a metallic cylinder 12.7 mm in diameter and 94 mm long. The cylinder is heated internally by an electrical heater and is subjected to a cross flow of air in a low-speed wind tunnel. Under a specific set of operating conditions for which the upstream air velocity and temperature were maintained at V = 10 m/s and 26.2°C, respectively, the heater power dissipation was measured to be P = 46 W, while the average cylinder surface temperature was determined to be Ts = 128.4°C. Assuming that 15% of the power dissipation is lost through the

cumulative effects of surface radiation and conduction through the endpieces:

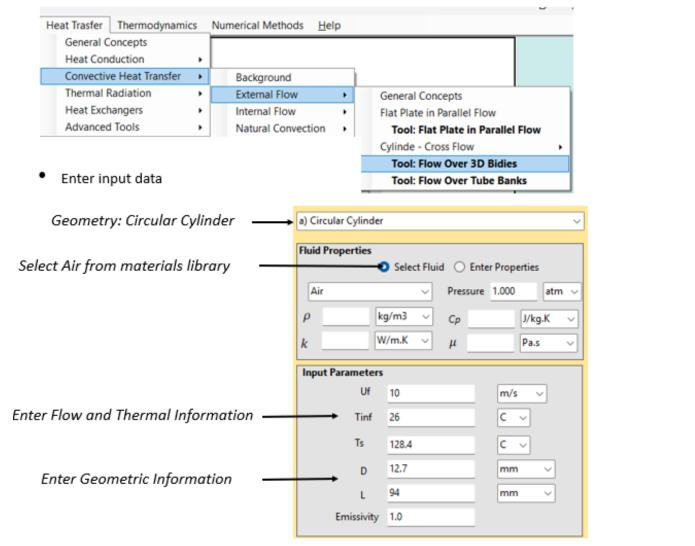
- a) Determine the convection heat transfer coefficient from the experimental observations.
- b) Compare the experimental result with the convection coefficient computed from the correlation.

a) The convection coefficient can be evaluated from the experimental data, assuming 85% of the total heat dissipation dissipated from the surface of the rod:

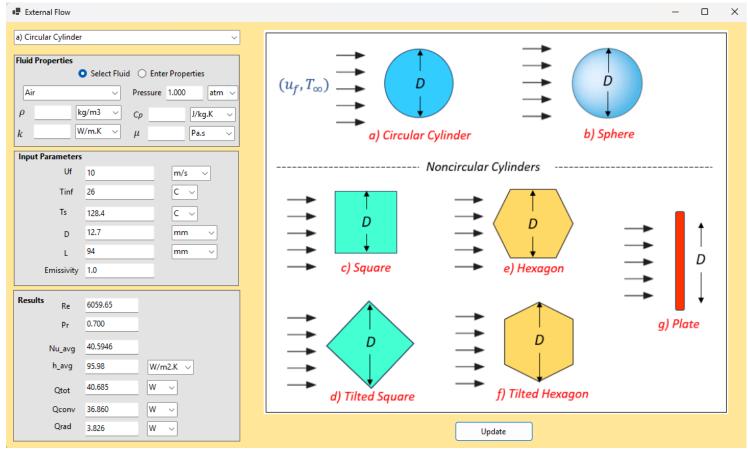
$$\bar{h} = \frac{q}{A(T_s - T_{\infty})}$$
$$\bar{h} = \frac{0.85 \times 46}{\pi (0.0127)(128.4 - 26.2)} = 102 \frac{W}{m^2 K}$$



b) To obtain the heat transfer coefficient using the correlation, open "Flow Over 3D Bodies Tool"



#### Click Update and Solve



Comments:

- 1. The Reynolds number, *Re*<sub>D</sub>=6059.
- 2. The Nusselt number, Nu=40.6.
- The average heat transfer coefficient is evaluated to be 95.6 W/m<sup>2</sup>.K. This is very close (about 6% difference) to the experimentally determined value of 102 W/m<sup>2</sup>.K.
- 4. The theoretically calculated rate of heat transfer from the rod is 40.69 W, which is very close to

the experimentally measured value of 39.1 W.

#### Solution:

#### **HT-14: Flow Over Tube Banks**

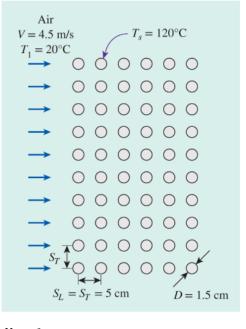
The following three examples illustrate how to solve heat transfer problems involving flow across tube b both in-line and staggered configurations.

#### Example – Preheating Water in a Bank Tube [Source: Cengel-Ghagar Example 7.8]

In an industrial facility, air is to be preheated before entering a furnace by geothermal water at 120 °C flowing through the tubes of a tube bank located in a duct. Air enters the duct at 20 °C and 1 atm with a mean velocity of 4.5 m/s and flows over the tubes in normal direction. The outer diameter of the tubes is 1.5 cm, and the tubes are arranged in-line with longitudinal and transverse pitches of  $S_L = S_T = 5$  cm. There are 6 rows in the flow direction with 10 tubes in each row. Determine the rate of heat transfer per unit length of the tubes and the pressure drop across the tube bank.

#### Heat Trasfer Thermodynamics Numerical Methods Help General Concepts Heat Conduction Convective Heat Transfer ٠ Background Thermal Radiation External Flow General Concepts Heat Exchangers Internal Flow Flat Plate in Parallel Flow Natural Convection Advanced Tools **Tool: Flat Plate in Parallel Flow** Cylinde - Cross Flow Enter input data Tool: Flow Over 3D Bidies Tool: Flow Over Tube Banks Fluid Properti Select Fluid O Enter Properties Pressure 1.000 Air atm ρi kg/m3 kg/m3 ρ J/kg.K Ср W/m.K Pa.s k μ Pr Prs Input Par ٧ 4.5 m/s $\sim$ C v Ti 20 Ts C 120 $\sim$ 1.5 cm D 5 cm ST cm SL 1.0 L m No of Columns - Longitudinal Dir, NL 6 No of Rows - Transverse Dir, NT 10

Open "Flow over Tube Banks" Tool



 $N_L = 6$  $N_T = 10$ L = 1 m

#### • Click Update and Solve

#### Flow Across Tube Banks $\times$ 🗿 In-line 🛛 🔘 Staggered **Fluid Properties** Select Fluid O Enter Properties $S_L$ Pressure 1.000 Air atm 🗸 $(V,T_{\infty})$ hoi 1.19336 kg/m3 1.170883 kg/m3 ρ $\sim$ Cp 1006.96 J/kg.K $S_T$ 0.02614 W/m.K Pa.s 1.8358E-0! k μ $\sim$ Pr 0.708 Prs 0.691 Input Parameters ۷ 4.5 m/s $\sim$ $A_1 = S_T L$ С Ti a) Inline Arrangement 20 $\sim$ $A_T = (S_T - D)L$ Ts 120 С $A_D = (S_D - D)L$ $S_I$ $S_D$ 1.5 cm D 5 ST cm $(V,T_{\infty})$ 5 cm SL $S_T$ 1.0 m L $\sim$ No of Columns - Longitudinal Dir, NL 6 No of Rows - Transverse Dir, NT 10 $A_1$ Results 6150 Re 55.2343 Nu\_avg W/m2.K h\_avg 96.2 $\sim$ b) Staggered Arrangement Te С 29.6 $\sim$ Q 25888.1 W $\sim$ Update Pa 145.166 Dp/xf $\sim$ Show Tutorial

Comments:

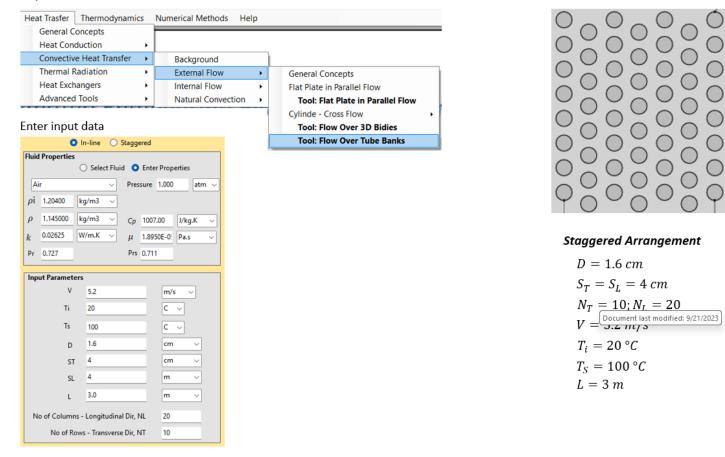
- 5. The Reynolds number, Re=6150 and it is based on  $V_{max}$
- 6. Average heat transfer coefficient is 96.2 W/m.K
- 7. Air exit temperature is 29.6 °C
- 8. Heat transfer into air is 25888.1 W (25.8 kW)
- Pressure drop across the bank can be calculated to be 23.2 Pa, by multiplying *Dp/xf* value of 145.66 Pa by the product of the friction factor and the correction factor (roughly 0.16 and 1.0 from the chart).
- 10.Note that, following the solution process, values fluid thermal properties are displayed corresponding to the latest updated film temperature. The user can use fixed properties by selecting "Enter Properties" option.
- $11.\rho i$  (1.193 shown in the first density box) represents the value of density at the inlet (evaluated using inlet temperature). This value is used to determine the mass flow rate through the bank.

# Example – Air Heating by Steam Tubes [Source: Cengel-Ghagar Problem 7.112]

Air is to be heated by passing it over a bank of 3-m-long tubes inside which steam is condensing at 100 °C. Air approaches the tube bank in the normal direction at 20 °C and 1 atm with a mean velocity of 5.2 m/s. The outer diameter of the tubes is 1.6 cm, and the tubes are staggered with longitudinal and transverse pitches of S = ST = 4 cm. There are 20 rows in the flow direction with 10 tubes in each row. Determine the rate of heat transfer.

# Solution:

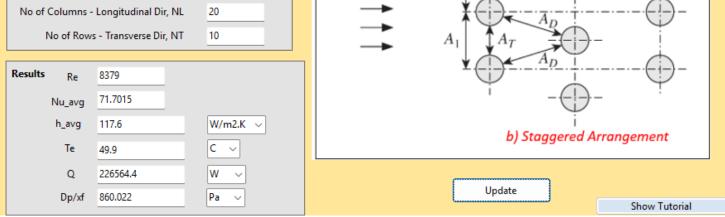
• Open "Flow over Tube Banks" Tool



Note that, for this example, user properties (fixed properties) are used instead of fluid selection.

Click Update and Solve

Flow Across Tube Banks	- 🗆 X
💿 In-line i Staggered	
Fluid Properties         Select Fluid O Enter Properties         Air       Pressure       1.000       atm         ρi       1.20400       kg/m3       Cp       1007.00       J/kg.K       V         ρ       1.145000       kg/m3       Cp       1007.00       J/kg.K       V         k       0.02625       W/m.K       μ       1.8950E-0!       Pa.s       V         Pr       0.727       Prs       0.711	$(V, T_{\infty})$ $S_{T}$ $A_{1}$ $A_{T}$ $S_{L}$ $C$
Input Parameters	$\rightarrow$ $A_1 \downarrow \downarrow A_T$
V <u>5.2</u> m/s ~	
Ti 20 C ~	$A_1 = S_T L$ a) Inline Arrangement
Ts 100 C ~	$ \begin{array}{c} A_1 = S_T L \\ A_T = (S_T - D)L \\ A_D = (S_D - D)L \end{array} \qquad \qquad \textbf{a) Inline Arrangement} \end{array} $
p 1.6 cm ~	
ST 4 cm ~	$(V, T_{\infty})$
SL <u>4</u> m ~	
L 3.0 m ~	$s_T$ $- \bigcirc \bullet$ $D$



#### Comments:

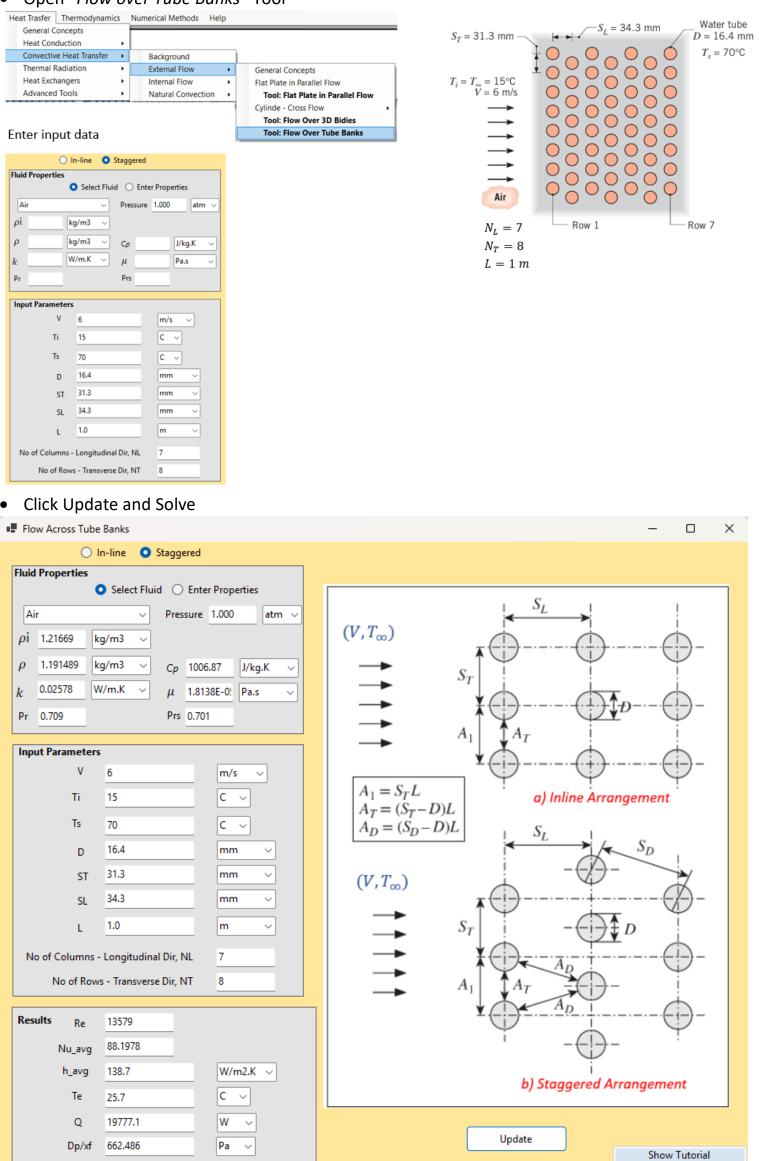
- 1. The Reynolds number, Re=8379 is based on  $V_{max}$
- 2. Average heat transfer coefficient is 117.6 W/m.K
- 3. Air exit temperature is 49.9 °C
- 4. Heat transfer into air is 226564.4 W (22.7 kW)

 Pressure drop across the bank can be calculated to be 283.8 Pa, by multiplying *Dp/xf* value of 860.022 Pa by the product of the friction factor and the correction factor (roughly 0.33 and 1.0 from the chart).

#### **Example 7.7– Space Heating Using Pressurized Water [Source: Bergman-Lavine Example 7.7]**

Pressurized water is often available at elevated temperatures and may be used for space heating or industrial process applications. In such cases it is customary to use a tube bundle in which the water is passed through the tubes, while air is passed in cross flow over the tubes. Consider a staggered arrangement for which the tube outside diameter is 16.4 mm and the longitudinal and transverse pitches are  $S_L = 34.3$  mm and  $S_T = 31.3$  mm. There are seven rows of tubes in the airflow direction and eight tubes per row. Under typical operating conditions the cylinder surface temperature is at 70 °C, while the air upstream temperature and velocity are 15 °C and 6 m/s, respectively. Determine the air-side convection coefficient and the rate of heat transfer for the tube bundle.

#### Solution:



#### Open "Flow over Tube Banks" Tool

Comments:

- 1. The Reynolds number, *Re*, is 13,579 and it is based on  $V_{max}$
- 2. Average heat transfer coefficient is 138.7 W/m.K
- 3. Air exit temperature is 25.7 °C
- 4. Heat transfer into air is 19,777 W (19.8 kW)
- 5. Pressure-drop across the bank can be calculated to be 241.15 Pa, by multiplying *Dp/xf* value of 662.49 Pa by the product of the friction factor and the correction factor (roughly 0.332 and 1.04 from the chart).
- Note that, following the solution process, values fluid thermal properties are displayed 6. corresponding to the latest updated film temperature.

# **HT-15: Internal Flow Heat Transfer**

# Example – Developing laminar Flow of Oil in a Pipeline Through a Lake [Source: Cengel-Ghagar Example 8.3]

Consider the flow of oil at 20 °C in a 30-cm-diameter pipeline at an average velocity of 2 m/s. A 200m-long section of the horizontal pipeline passes through icy waters of a lake at 0C. Measurements indicate that the surface temperature of the pipe is very nearly 0 °C. Disregarding the thermal resistance of the pipe material, determine (a) the temperature of the oil when the pipe leaves the lake, (b) the rate of heat transfer from the oil, and (e) the pumping power required to overcome the pressure losses and to maintain the flow of the oil in the pipe.

# Solution:

Heat Trasfer       Thermodynamics       Numerical Methods       Help         General Concepts       Heat Conduction       Heat Conduction       Heat Convective Heat Transfer       Background         Convective Heat Transfer       Background       External Flow       Heat Exchangers       Natural Flow         Advanced Tools       Natural Convection       Natural Convection       Heat Convection	Hydrodynamics Concepts Thermal Concepts _Energy Balance Convection Correlations in Circular Pipes Convection Correlations in Non-Circular Tubes Tool: Internal Flow Heat Transfer	20°C Oil Dilla Dilla Dilla Dilla Dilla Dilla Dilla Dilla Dilla Dilla Dill
Enter input data		$\rho = 888.1 \text{ kg/m}^3$ $\nu = 9.429 \times 10^{-4} \text{ m}^2/\text{s}$
<ul> <li>Select "Enter Properties" and enter — values.</li> </ul>	Fluid Properties         O Enter Properties           Oil         ~           Pressure         1.000         atm ~           ρ         888.1         kg/m3 ~         Cp         1880         J/kg.K ~           k         0.145         W/m.K ~         μ         0.8373         Pa.s ~	$k = 0.145 \text{ W/m} \cdot \text{K}$ $c_p = 1880 \text{ J/ kg} \cdot \text{K}$ $Pr = 10,863$
<ul> <li>Keep "Circular" cross-section.</li> <li>Enter given values for diameter and tube length in given units.</li> </ul>	Geometric Information Circular Circular D 0.3 m L 200 m V	
• Select "Uin" to enter inlet velocity. —	Inlet Flow Information       ○ mdot       ○ Vdot       ○ Uin         2         m/s	
• Enter the Inlet Temperat	Ure. Uniform	n 20 C ~

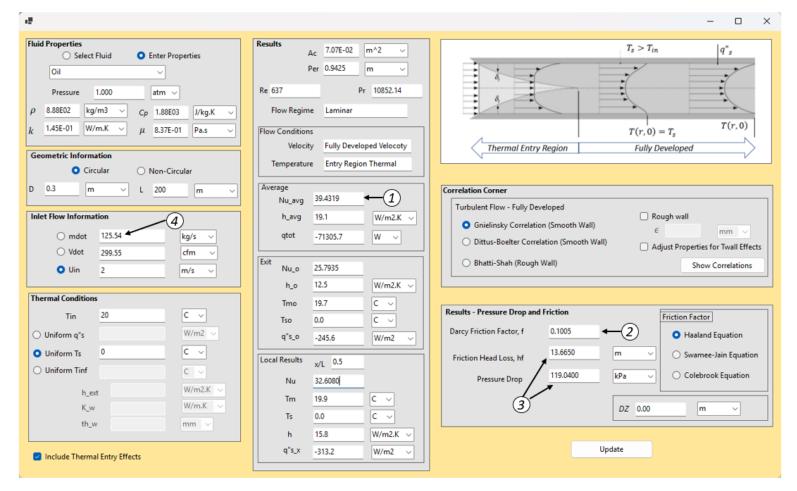
Select "Uniform Surface Temperature" option and Enter the value provided.

O Uniform Tinf	C 🗸
h_ext	W/m2.K 🗸
K_w	W/m.K 🗸
th_w	mm 🗸

Make sure Thermal Entry Effects are included.

Include Thermal Entry Effects

Click Update to Solve



#### Comments:

- It is evident from the Nusselt number value (being much larger than the fully-developed value of 3.66) that the thermal profile is not developed and we are operating in the thermal entry region. This is typical of "high Prandtl number" fluids (here Pr = 10852).
- 2. The friction coefficient is calculated to be 0.01005. This value corresponds to laminar flow in circular pipes  $\left(f = \frac{64}{Re}\right)$ .
- 3. The head loss  $(h_f)$  and corresponding  $\Delta p$  are calculated to be 13.66 m and 119.04 kPa
- 4. The mass flow rate is 125.54 kg/s. The required pumping power can be obtained from:

$$\dot{W}_P = \frac{\dot{m}\Delta p}{\rho} = \frac{(125.54)(119.04)}{(888.1)} = 16.8 \, kW$$

#### Example – Developing Laminar Flow with Uniform Heat Flux – [Source: Lienhard Example 7.2]

A fully developed flow of air at 27C moves at 2 m/s in a 1 cm I.D. pipe. An electric resistance heater surrounds the last 20 cm of the pipe and supplies a constant heat flux to bring the air out at Tb = 40C. What power input is needed to do this? What will be the wall temperature at the exit?

# Example – Low Prandtl Number Flow – Liquid Mercury Flow in a Pipe [Source: Cengel-Ghagar Problem 8.125]

Liquid mercury flows at 0.6 kg/s through a 5-cm diameter tube, with inlet mean temperature of 100 °C. surface temperature is kept constant at 250 °C.

- a) Determine the outlet mean temperature at x=50 cm.
- b) Determine the rate of heat transfer to mercury for this length of pipe.

# Example – Prescribed External Temperature and Convection Coefficient – Hot Air in Cold Ambient Environment [Source: Bergman-Lavine Example 8.6]

Hot air, with an inlet temperature of  $103^{\circ}$ C, flows with a mass rate of 0.050 kg/s through an uninsulated sheet metal duct of diameter D = 0.15 m and L = 5 m, which is in the crawlspace of a

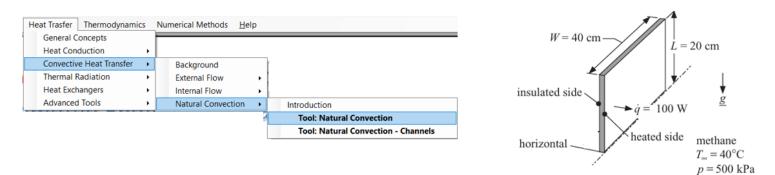
house. The heat transfer coefficient between the duct outer surface and the ambient air at  $T_{\infty} = 0^{\circ}C$  is known to be  $h_{ext} = 6 \text{ W}/(\text{m}^2 \cdot \text{K})$ .

- 1) Calculate the rate of heat loss (W) from the duct over the length L.
- 2) Determine the heat flux and the duct surface temperature at x = L.

#### **HT-16: Natural Convection over Bodies**

#### Example – Vertical Flat Plate – [Source: Nellis and Klein Example 6.2-1 Modified]

A rectangular plate heater is placed in the ullage space of a fuel tank on a military aircraft (as shown below). One side of the heater is insulated and the other is heated. The heater is oriented vertically with respect to gravity and achieves a nearly uniform temperature. The length of the heater is L = 20 cm, and the width is W = 40 cm. The plate is exposed to fuel that has properties consistent with methane at  $T_{\infty}$  = 40 °C. Assuming heater power of 100 W, determine the surface temperature of the heater for a) Fuel at atmospheric pressure, b) Fuel at p =500 kPa.



# **HT-17: Natural Convection Vertical Channels**

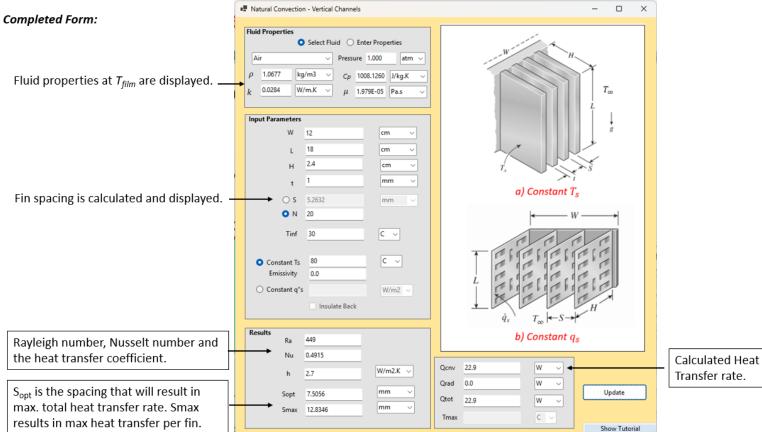
# Example – Heat Sink Fin Spacing – [Source: Cengel-Ghagar Example 9.3]

A 12-cm-wide and 18-cm-high vertical hot surface in 30 °C air is to be cooled by a heat sink with equally spaced fins of rectangular profile. The fins are 1 mm thick and 18 cm long in the vertical direction and have a height of 2.4 cm from the base. Assuming the base temperature of 80 °C.

- a) Determine the rate of heat transfer by natural convection from the heat sink using 20 fins.
- b) Determine the rate of heat transfer by natural convection from the heat sink using optimal fin spacing.
- c) Determine the rate of heat transfer from the heat sink using optimal fin spacing and including thermal radiation (emissivity=0.25).

#### Solution:

a)		
Heat Trasfer Thermodynamics Numerical Methods H General Concepts Heat Conduction • Convective Heat Transfer • Thermal Radiation • Heat Exchangers • Advanced Tools • Enter input data	;	$H = 2.4 \text{ cm}$ $L = 0.18 \text{ m}$ $T_{2} = 80^{\circ}\text{C}$ $t = 1 \text{ mm} \rightarrow   \leftarrow \rightarrow   S   \leftarrow$
Enter Fluid Properties: • Select "Air" from Fluid Lib	Fluid Properties         Orary.         Air       Pressure $\rho$ kg/m3 $\checkmark$ $C\rho$ k       W/m.K $\checkmark$ $\mu$	
Enter Input:         • Enter the Width of Heat Sink         • Enter the Vertical Height of         • Enter the Height from Base         • Enter the Fin Thickness         • Enter Number of Fins         • Enter Ambient Temperature         • Enter Sink Temperature         • Enter $\epsilon = 0$ to Neglect Radiation	Input Parameters         →       W       12       cm       ~         →       L       18       cm       ~         →       H       2.4       cm       ~         →       t       1       mm       ~         →       t       1       mm       ~         →       t       1       cm       ~         →       N       20       ~       ~         →       Tinf       30       C       ~         →       Emissivity       0.0       C       ~         ↓       Emissivity       0.0       W/m2       ~         ↓       Insulate Back	<ol> <li>Note:</li> <li>The used can choose to provide number of fins or fin spacing.</li> <li>The user may select specified sink temperature or constant heat flux.</li> </ol>
Click "Update" to Solve.		



#### b)

To use optimal fin spacing:

- 1. Change input *from Number of Fins* to *Spacing*.
- 2. Copy/Paste the value of the Optimal Spacing from the results to input.

Leave all other parameters unchanged.

#### Click "Update" to Solve.

- Heat transfer coefficient is increased.
- Heat rate is increased from 22.9 W to 29.9 W.

Note: If you repeat the solution using  $S_{max}$  for the spacing, the heat transfer coefficient will increase to 6 W/m.k, but the total rate of heat transfer will decrease to 23.3 W due to reduced number of fins.

#### Results 1301 Ra 1.3066 Nu Qcnv 29.9 W W/m2.K v 4.9 h w Qrad 0.0 $\sim$ mm 7.5056 Sopt Qtot 29.9 W $\sim$ 12.8346 mm 0 Smax Tmax С Results 6506 Ra Nu 2.7110 Qcnv 23.3 w W/m2.K 🗸 6.0 h Qrad 0.0 w $\sim$ 7.5056 Sopt Qtot 23.3 W $\sim$ mm $\sim$ Smax 12.8346 Tmax

80

0.25

Constant Ts

Emissivity

mm

7.5056

**O** S

N 20

#### c)

To Include thermal radiation, enter a non-zero value for emissivity:

#### Click "Update" to Solve.

 This will result in an additional 5.9 W heat loss due to radiation, bringing the total heat transfer rate to ambient to 35.8 W.

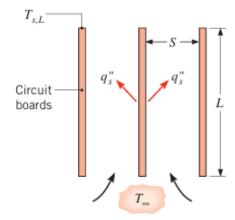
Results	Ra	1301				
	Nu h	1.3066 4.9	W/m2.K v	Qcnv Qrad	29.9 5.9	W ~ W ~
	Sopt Smax	7.5056 12.8346	mm ~ mm ~	Qtot Tmax	35.8	W V C V

С

Note: The effective radiation surface area is much smaller that the convection surface area due to close spacing between the fins. Therefore, addition of fins do not significantly enhance the radiation heat transfer here.

# Example – Natural Convection Cooling of Vertical PCBs – [Source: Bergman-Lavine Problem 9.59]

A vertical array of circuit boards is immersed in quiescent ambient air at  $T_{\infty} = 17^{\circ}$ C. Although the components protrude from their substrates, it is reasonable, as a first approximation, to assume flat plates with uniform surface heat flux q"<sub>s</sub>. Consider a 60 cm wide heat sink with boards of height and length H = L = 40 cm with 1.5 mm thickness and spacing S = 25 mm. Assuming each board is populated on both sides and dissipates 88 W, determine maximum board temperature and total heat dissipation of this unit.



#### HT-18: Heat Exchanger Epsilon/NTU Calculator

#### **Example – Heat Exchanger Effectiveness-NTU Relationships**

Complete the following table using the automated Epsilon-NTU Tool in AutoTherm.

Heat Exchanger Flow Arrangement	Capacity Ratio, Cr	NTU	Epsilon
Parallel Flow	0.5	2.5	
Counter Flow	0.5	2.5	
Shell-Tube 2 Tube Passes; Single Shell Pass	0.5	2.5	
Shell-Tube 2 Tube Passes; 2 Shell Pass	0.5	2.5	
Cross-Flow Both Fluids Unmixed	0.5	2.5	
Cross-Flow Both Fluids Unmixed	0.75		0.65
Parallel Flow	0.75		0.65

# Solution:

Heat Trasfer	Thermodynamics		Numerical Methods Help
General Concepts			
Heat Conduction		•	
Convectiv	Convective Heat Transfer		
Thermal R	adiation	+	
Heat Exch	angers	•	Fundamental Concepts
Advanced	Tools	•	Analysis Using Epsilon-NTU
			Tool: Epsilon-NTU
			Tool: HX Performance

Tool utilizes a very simple user input panel shown below:

		Input
•	Select heat exchanger flow arrangement.	d) Shell & Tube: n Shell pass and 2n, 4n, tube pass 🗸
•	Enter the capacity ratio.	Cr Cr
•	Enter Number of shell passes (only for shell-tube).——	No. of Shell passes
•	Enter NTU to calculate Effectiveness (performance mode), or epsilon to get NTU (design mode).	NTU Given     O Epsilon Given

Heat Exchanger Flow Arrangement	Capacity Ratio, Cr	NTU	Epsilon	
Parallel Flow	0.5	2.5	0.651	
Counter Flow	0.5	2.5	0.8328	
Shell-Tube 2 Tube Passes; Single Shell Pass	0.5	2.5	0.7237	
Shell-Tube 2 Tube Passes; 2 Shell Pass	0.5	2.5	0.802	
Cross-Flow Both Fluids Unmixed	0.5	2.5	0.7911	
Cross-Flow Both Fluids Unmixed	0.75	2.1165	0.65	
Parallel Flow	0.75	No Solution	0.65	

Note: Epsilon-NTU relations always provide value for effectiveness for all positive NTU's. However, as seen from the last row, not every positive epsilon will result in a valid NTU.

# HT-19: Heat Exchanger Performance Analysis Tool Example – Heat Exchanger Performance – [Source: Nellis and Klein Example 8.3-1]

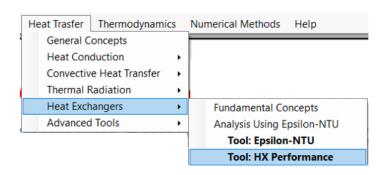
The cross-flow heat exchanger with both fluids unmixed is used to heat air with hot water. Water enters the heat exchanger tubing with a mass flow rate, of 0.03 kg/s and 60 °C. Air at 20 °C and atmospheric pressure is blown across the heat exchanger with a volumetric flowrate of 0.06 m<sup>3</sup>/s. The conductance of this heat exchanger has been calculated to be 58.4 W/K, based on the compact heat exchanger correlations. Determine the outlet temperatures of the water and air and the heat transfer rate using the  $\varepsilon$ -NTU method.

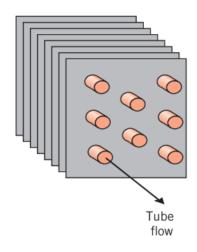
# Solution:

Heat Trasfer Thermodynami		Numerical Methods Help
General Concepts		
Heat Conduction		
Convective Heat Transfer		
Thermal Radiation	•	
Heat Exchangers	•	Fundamental Concepts
Advanced Tools	•	Analysis Using Epsilon-NTU
		Tool: Epsilon-NTU
		Tool: HX Performance

Tool utilizes a very simple user input panel shown below:

- Select heat exchanger flow arrangement. d) Shell & Tube: n Shell pass and 2n, 4n, ... tube pass
- Enter the capacity ratio.
- Enter Number of shell passes (only for shell-tube).-No. of Shell passes
- NTU Given Enter NTU to calculate Effectiveness (performance Epsilon Given mode), or epsilon to get NTU (design mode).





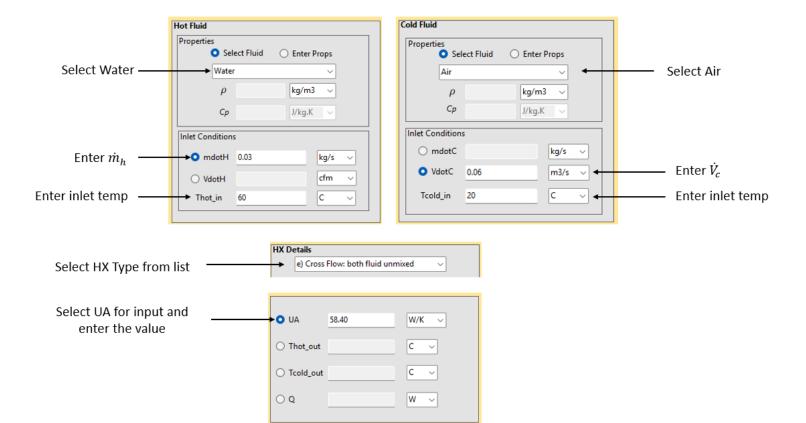
#### Enter input data

Hot Side



Input

Cr



#### Click "Solve".

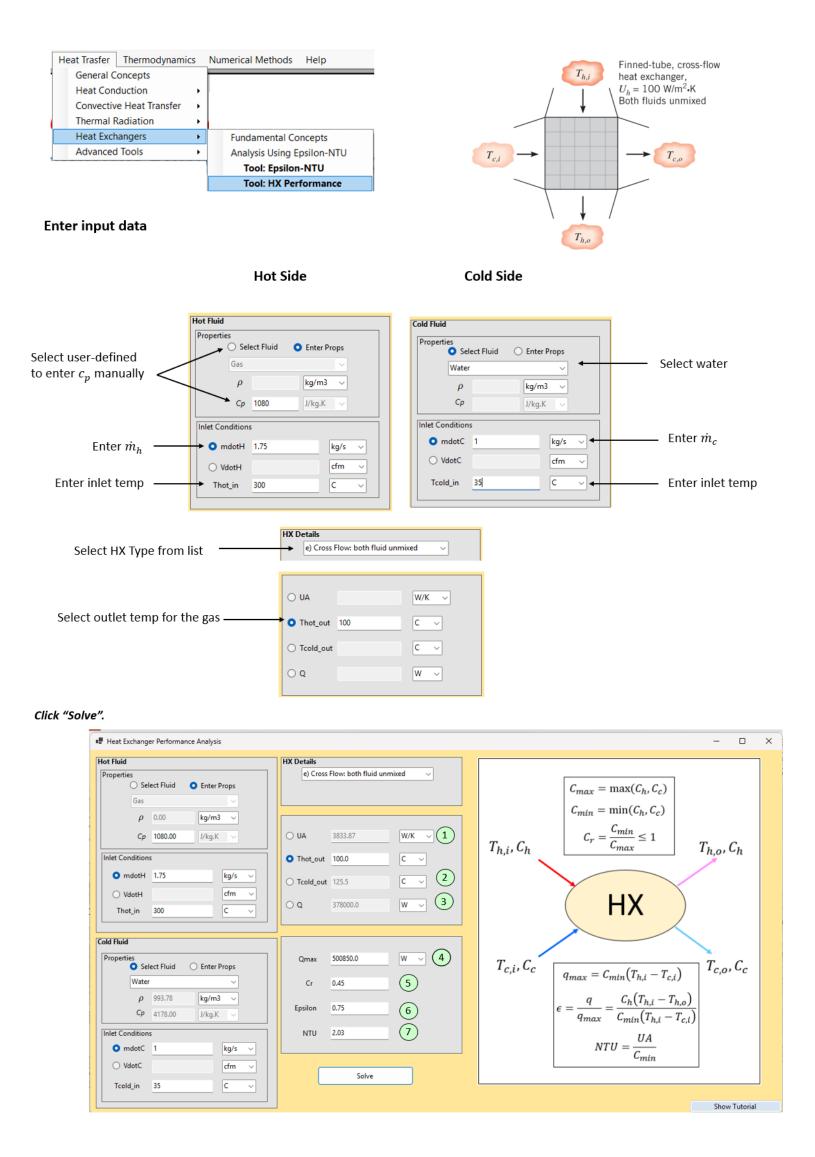
Heat Exchanger Performance Analysis – 🗆 🗙 Hot Fluid HX Details e) Cross Flow: both fluid unmixed Properties  $\sim$ Select Fluid O Enter Props  $C_{max} = \max(C_h, C_c)$ Water  $\sim$  $C_{min} = \min(C_h, C_c)$ ρ 979.50 ka/m3 ~  $C_r = \frac{C_{min}}{C_{max}} \le 1$ Cp 4187.96 J/kg.K 🗸 O UA 58.40 W/K ~  $T_{h,i}, C_h$  $T_{h,o}, C_h$ Inlet Conditions O Thot out 49.1 C ~ (1) O mdotH 0.03 kg/s  $\sim$ c ~ 2 O Tcold\_out 38.9 cfm ⊖ VdotH  $\sim$ HΧ W ~ 3 0 Q 1365.2 Thot\_in 60 С Cold Fluid Properties O Select Fluid Qmax 2883.7 w ~ (4)  $T_{c,i}, C_c$  $T_{c,o}, C_c$ O Enter Props  $q_{max} = C_{min} \big( T_{h,i} - T_{c,i} \big)$ Air 0.57 (5) Cr  $\frac{q}{q_{max}} = \frac{C_h (T_{h,i} - T_{h,o})}{C_{min} (T_{h,i} - T_{c,i})}$ ρ 1.19 kg/m3  $\epsilon = -$ 0.47 Epsilon 6 Cp 1006.86 J/kg.K  $\overline{7}$ Inlet Conditions NTU 0.81  $NTU = \frac{UA}{C_{min}}$ ⊖ mdotC kg/s v O VdotC 0.06 m3/s ~ Solve С Tcold\_in 20  $\sim$ Show Tutorial

- 1. The outlet temperature of water is 49.1 °C.
- 2. The outlet temperature of air is 38.9 °C.
- 3. The total rate of heat transfer between water and air is 1365.2 W.
- 4. The maximum heat transfer rate  $(Q_{max})$  is 2883.7 W.
- 5. The capacity ratio is calculated to be 0.57.
- 6. The heat exchanger effectiveness is calculated to be 0.47.
- 7. The number of transfer units (NTU) is 0.81.

#### Example – Heat Exchanger Design – [Source: Bergman-Lavine Problem 9.59]

Hot exhaust gases ( $c_p$ =1080 J/kg.K), which enter a finned-tube, cross-flow heat exchanger with a flow rate of 1.75 kg/s at 300°C and leave at 100°C, are used to heat pressurized water at a flow rate of 1 kg/s from 35 °C. The overall heat transfer coefficient based on the gas-side surface area is  $U_h$  = 100 W/m2.K. Determine the required gas-side surface area  $A_h$  and outlet temperature of water.

#### Solution:



1. The UA is calculated to be 3833.37 W/K.

Since U is given in the problem statement to be 100 W/m2.K, heat transfer area may be determined from:

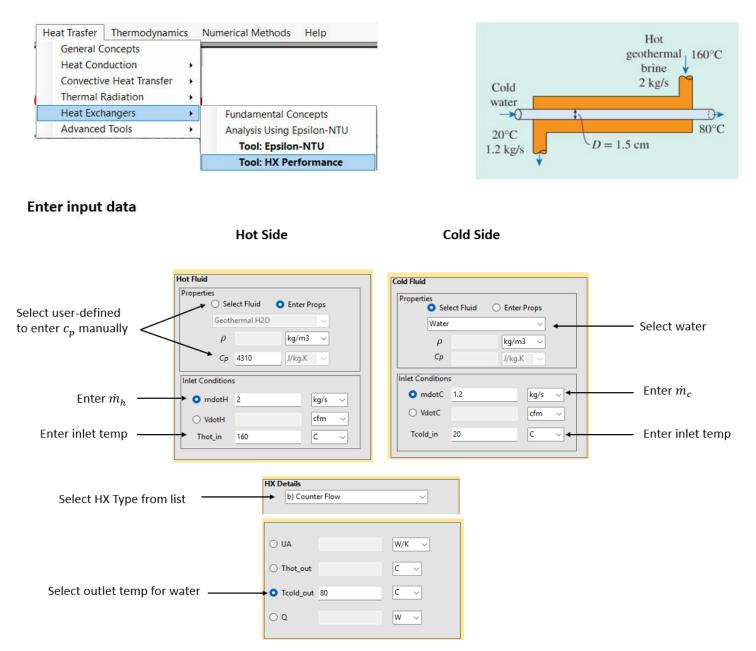
$$A_h = \frac{UA}{U_h} = \frac{3833}{100} \to A_h = 38.33 \ m^2$$

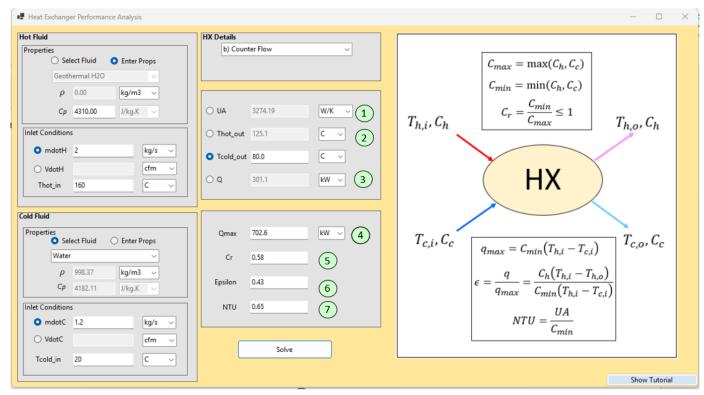
- 2. The outlet temperature of water is 125.5 °C.
- 3. The total rate of heat transfer between hot gas and water is 378 kW.
- 4. The maximum heat transfer rate  $(Q_{max})$  is 500850 W.
- 5. The capacity ratio is calculated to be 0.45.
- 6. The heat exchanger effectiveness is calculated to be 0.75.
- 7. The number of transfer units (NTU) is 2.03.

# Example – Heat Exchanger Design – [Source: Cengel-Ghagar Example 11.8]

A counterflow double-pipe heat exchanger is to heat water from 20 C to 80 C at a rate of 1.2 kg/s. The heating is to be accomplished by geothermal water available at 160 C at a mass flow rate of 2 kg/s (assume  $c_p$ =4310 J/kg.K). The inner tube is thin-walled and has a diameter of 1.5 cm. The overall heat transfer coefficient of the heat exchanger is 640 W/m2. K, determine the length of the heat exchanger required to achieve the desired heating.

# Solution:





1. The UA is calculated to be 3274.2 W/K. Since U is given in the problem statement to be 640 W/m2.K, The length can be found:

$$A_s = \pi DL = \frac{UA}{U_h} \implies L = \frac{UA}{(U)(\pi D)} = \frac{(3574.2)}{(640)(\pi)(0.015)} \implies L = 108.6 m$$

- 2. The outlet temperature of geothermal brine is 125.1 °C.
- 3. The total rate of heat transfer between hot gas and water is 301.1 kW.
- 4. The maximum heat transfer rate ( $Q_{max}$ ) is 702.6 kW.
- 5. The capacity ratio is calculated to be 0.58.
- 6. The heat exchanger effectiveness is calculated to be 0.43.
- 7. The number of transfer units (NTU) is 0.65.

HT-20: Radiation View factor Calculator