TUBE BANKS TEST PROBLEMS

The non-proprietary tests used to validate INSTED® analysis of flow and heat transfer over tube banks are presented in this section. You may need to consult the original sources of the various test problems in order to assess the accuracy of INSTED® predictions in more detail. These sources are given, as are a few notes to aid you in your comparison exercise. Some diagnostic results reported in the sources are presented here and compared with INSTED® predictions. You will be expected to have simulated some of these test problems before you attempt to solve more realistic engineering problems.

Test Problem 1: Tube Banks

- **Problem Statement:**

Air conditioning systems often use "cooling coils" to reduce the temperature of air for human comfort. A "coil" consists of a bank of tubes placed in an air-conditioning duct such that the air flows normal to the tubes while a cooling medium (chilled water or refrigerant) flows through the tubes. The tube bank in this case consists of a staggered array of tubes with a diameter D = 1.59 cm. There are four rows of tubes with a longitudinal spacing S_L = 3.0 cm. Chilled water flowing through the tubes maintains the tube surface temperature at $T_s = 5^oC$. Atmospheric-pressure air is incident on the bank with freestream conditions: U_∞ = 5 m/s. $T_\infty = 30^oC$.

(a) Estimate the surface heat transfer coefficient.
(b) Find the pressure drop of the air across the tube banks.

- **Source:**

Alan J. Chapman. 1987. Fundamentals of Heat Transfer. Macmillan Publishing Co. New York. Page 353.

- **Comments**

- Use online unit conversion to convert units from ^oC to K.
- A staggered tube layout is used.
- Use thermophysical property values given in the source. (ρ = 1.20503 kg/m^3, μ = 0.17918 X 10^{-4} Ns/m^2, k = 0.02545 W/m K, Pr =0.714, Pr_{wall} =0.717)
-
- Diagnostic results from source are compared below with INSTED predictions. Below, the notations introduced earlier are used. V_{max} and ΔP are maximum velocity and pressure loss, respectively.

Variable	Chapman	INSTED®	Difference
V_{max} (m/s)	10.638	10.638	< 1%
Re	1.138 x 10^4	1.138 x 10^4	< 1%
Nu	71.66	72.53	< 1.2%
h (W/m^{2}K)	114.7	116.1	< 1.2%
ΔP (N/m^2)	90.1	93.96	< 4.2%

- Name of data file: CHAPMAN.353

Test Problem 2: Tube Banks

➢ **Problem Statement:**

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 34.3 and 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 70^{o}C, while the air upstream temperature and velocity are 15^{o}C and 6 m/s, respectively. Determine the air-side convection coefficient and the rate of heat transfer for the tube bundle. What is the air-side pressure drop?

➢ **Source:**

Frank P. Incropera & David P. Dewitt. 1990. Introduction to Heat Transfer. Page 395.

➢ **Comments**

- Use online unit conversion to convert units from ^{o}C to K.
- A staggered tube layout is used.
- Use thermophysical property values given in the source. (ρ = 1.217 kg/m^3, μ = 0.18036 X 10^{-4} Ns/m^2, k = 0.0253 W/m K, Pr =0.710, Pr_{wall} =0.705)
-
- Diagnostic results from source are compared below with INSTED predictions..

Variable	**Incrop**	**INSTED®**	**Difference**
V_{max}	12.6	12.6 m/s	< 1%
Re	13,943	13,948	< 1%
Nu	87.9	88.67	< 1%
h (W/m^{2}K)	135.6	136.77	< 1%
Log Mean ΔT (K)	49.6	53.94	< 9%
q' (kW/m)	19.4	21.3	< 10%
ΔP (N/m^2)	246	223.6	<10%

Explanation of differences in results

Reasons for the discrepancy are mostly due to differences in the model equations used and the manner of obtaining model constants. A continuous liner extrapolation is used in INSTED® whereas a step change (for example with Reynolds number) in model constants is allowed in the source. Note that differences are still within the tolerance for empirical relations.

♦ Name of data file: INCROP.395

Test Problem 3: Tube Banks

➢ **Problem Statement:**

Methane gas at 25°C is to be preheated in a heat exchanger consisting of a staggered arrangement of 4-cm-OD tubes, 5 rows deep, with a longitudinal spacing of 6 cm and a transverse spacing of 8 cm (see Fig 7.28 of source). Subatmospheric pressure steam is condensing inside the tubes, maintaining the tube wall temperature at 50°C. Determine (a) the average heat transfer coefficient for the tube bank and (b) the pressure drop through the tube bank. The methane flow velocity is 10 m/s upstream of the tube bank.

(c) Estimate the surface heat transfer coefficient.
(d) Find the pressure drop of the air across the tube banks.

➢ **Source:**
Frank Kreith & Mark S. Bohn. 1993. Fifth Edition. Principles of Heat Transfer. West Educational Publishing. Boston. Page 482.

➢ **Comments**
- Use online unit conversion to convert units from ^{o}C to K.
- A staggered tube layout is used.
- Use thermophysical property values given in the source. (ρ = 0.668 kg/m^3, μ = 0.1087 X 10^{-4} Ns/m^2, k = 0.0332 W/m K, Pr =0.73, Pr_{wall} =0.73)
-
- Diagnostic results from source are compared below with INSTED predictions.

Variable	**Krbohn**	**INSTED**	**Differences**
U_{max} (m/s)	20	20	< 1%
Re_D	49170	49162.80	< 1%
Nu_D	216	198.89	< 8%
h (W/m^{2}k)	165	165.08	< 1%
Δp (N/m^2)	167	174.35	< 5%
q (W/m)	-	2980.13	N/A
LMTD (K)	-	28.73	N/A
T_{outlet} (K)	-	295.65	N/A

♦ Name of data file: KRBOHN.482

Test Problem 4: Tube Banks

➢ **Problem Statement:**
Atmospheric air flows across a staggered tube bank consisting of 5 rows of tubes with 7 tubes each. The upstream velocity, U_∞, and temperature, T_∞, of the air are 5 m/s and 40°C, respectively. If the surface temperature of the tubes is maintained at 90°C, find (a) the average heat-transfer coefficient for the tube bank, (b) the convective heat transfer from the tube bank per meter of tube length, and (c) the pressure drop across the tube bank. So that we do not have to correct properties for temperature, assume that the changes in air temperature across the tube bank does not significantly change the air properties.

➢ **Source:**
Kirk D. Hagen. 1999. Heat Transfer with Applications. Prentice Hall. New Jersey. Page 275.

➢ **Comments**
- Base the air properties on the inlet temperature of 40°C (313 K)
- ρ = 1.1181 kg/m^3, c_p = 1007.5 J/kg•K, ν = 17.20 x 10^{-6}m^2/s
  k = 0.0273 W/m•K, Pr = 0.705
- The Prandtl number evaluated at the surface temperature of 90°C (363 K) is Pr_w = 0.697

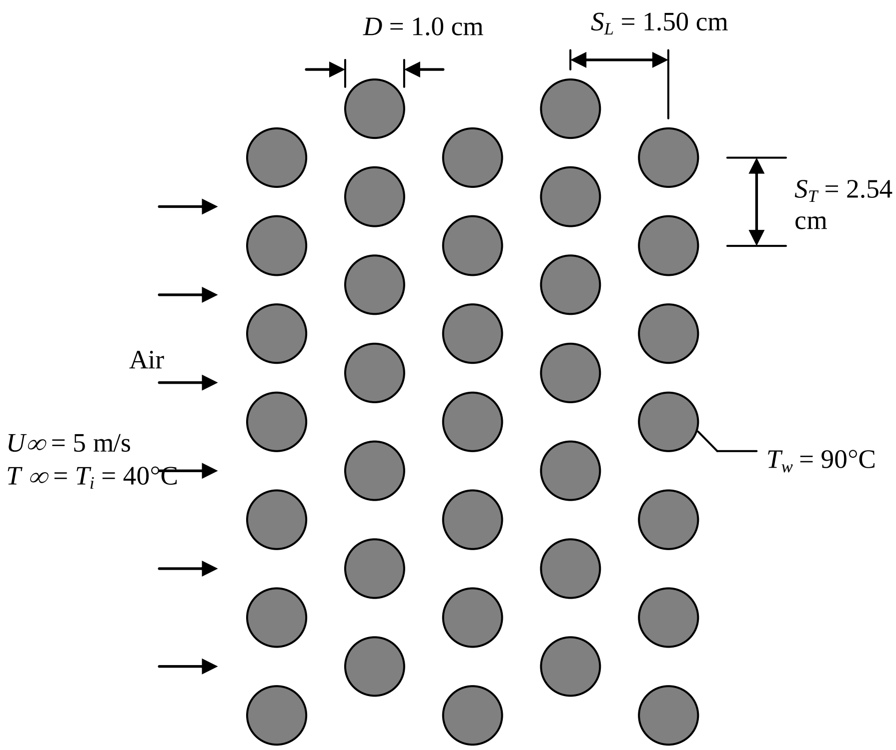
▪ Diagnostic results from source are compared below with INSTED predictions.

Variable	Hagen	INSTED®	Difference
U_{max} (m/s)	8.25	8.24	< 1%
Re_{Dmax}	4797	4826.8	< 1%
Nu_D	51.9	51.33	< 2%
Q/L (W/m)	7260	7141	< 2%
ΔP (N/m^2)	79.5	63.46	20.2%
h (W/m^2•K)	142	140.13	< 2%
T_o (°C)	46.8	47.14	< 1%
ΔT_m (K)	46.5	46.34	< 1%

Explanation of differences in results

Different equations are used for friction factor computation in Source and INSTED. Also, INSTED results are given for the cas e where the parameter χ =1, whereas the value in the source is 1.1. Use of 1.1 in INSTED gives a ΔP of 69.8 N/m^2. Note that χ is a measure of the deviation of the arrangement from a perfect square or equilateral triangle.

♦ Name of data file: HAGEN.275



Test Problem 5: Tube Banks

➢ **Problem Statement:**

Air at 1 atm and 10°C flows across a bank of tubes 15 rows high and 5 rows deep at a velocity of 7 m/s measured at a point in the flow before the air enters the tube bank. The surfaces of the tubes are maintained at 65°C. The diameter of the tubes is 1 in [2.54 cm]; they are arranged in an in-line manner so that the spacing in both the normal and parallel directions to the flow is 1.5 in [3.81 cm]. Calculate the total heat transfer per unit length for the tube bank and the exit air temperature.

➢ **Source:**

J.P. Holman. 1990. Heat Transfer. McGraw-Hill Publishing Company. New York. Page 312.

➢ **Comments**

- $T_{fi} = (T_w + T_\infty)2 = (65 + 10)/ 2 = 37.5°C = 310.5$ K [558.9°R]
- $\rho_f = p/RT = 1.0132 \times 10^5/(287)(310.5) = 1.137$ kg/m^3
- $\mu_f = 1.894 \times 10^{-5}$ kg/m • s
- $k_f = 0.027$ W/m • °C [0.0156 Btu/h • ft • °F]
- $c_p = 1007$ J/kg • °C [0.24 Btu/lb$_m$ • °F]
- Pr = 0.706
- Diagnostic results from source are compared below with INSTED predictions.

Variable	**Holman**	**INSTED®**	**Difference**
U_{max} (m/s)	21.0	21.0	< 1%
Re	32020.0	32020.9	< 1%
Nu_D	141.50	151.11	< 2%
H (W/m^2 • °C)	150.88	160.63	< 7%
$T_{\infty,2}$ (°C)	19.08	20.42	< 8%
Δp (W/m^2)	-	352.88	N/A

♦ Name of data file: HOLMAN.312

Explanation of differences in results

The model equations for the Nusselt number are different for the Source and INSTED.

Test Problem 6: Tube Banks

➢ **Problem Statement:**

A heat exchanger consists of 56 tubes of external diameter 16.4 mm and length of 0.5 m, mounted in a rectangular duct of height 0.2504 m and width 0.5 m. The tubes are mounted in a staggered array with 60° triangular pitch, with seven successive rows each consisting of eight tubes with a pitch of 31.3 mm. Corbels are attached to the walls to reduce bypass flow. The tubes contains hot water that maintains their surface temperature at 70°C and air flows across them at a rate of 0.914 kg/s. If the air enters the duct at 15°C, what is its exit temperature?

➢ **Source:**

G.F. Hewitt, G.L Shires, & T.R. Bott,. 1994. Process Heat Transfer. CRC Press, Inc. Boca Raton. Page 80.

- **Comments**
- Air 15°C
- Density $\rho = 1.217$ kg/m^3
- Viscosity $\eta = 1.80 \times 10^{-5}$ N • s/m^2 [(kg • m)/s^2]
- Specific Heat $c_p = 1007$ J/(kg • K)
- Conductivity $\lambda = 2.53 \times 10^2$ W/(m • K)
- With corbels treated as half-tubes, the number of transverse tubes is 8.5. The number of longitudinal tubes is 7. Note that the number of longitudinal tubes is 7 and only 56 tubes contribute to heat transfer. Remember to turn the radio button for corbels to the "ON" (filled) position.
- Specify flow rate as 0.914 kg/s
- Note that we have a triangular layout (not rotated), with Tube Layout Angle (TLA) of 30°.

- Diagnostic results from source are compared below with INSTED predictions.

Variable	Hewitt	INSTED®	Difference
V_{max} (m/s)	11.9	11.875	< 1%
Re	1.32×10^4	1.3124×10^4	< 1%
Nu	97.5	89.91	< 12%
α (W/m^2•K)	150.4	134.08	< 12%
ΔT (K)	49.0	49.79	< 2%
T_0 (K)	26.5	24.89	< 7%
Q (W)	21,200	20,397.8	< 4%
Δp (N/m^2)	300	199.91	N/A

Explanation for differences in results
The Nusselt number models and the constants in Source and INSTED are significantly different. The models in INSTED are probably more accurate. For Δp, the Source includes exit/entrance losses whereas INSTED calculations do not. The K-factors for an entrance depends on the geometry of the entrance. Therefore, no preset values are used in INSTED. The user calculates these losses in addition to other types of losses that might be encountered in conveying the fluid to the heat exchanger. Note that without the losses, the pressure drop calculated by the source is 211.1 N/m^2, to be compared with the value of 199.91 calculated by INSTED.

- Name of data file: Hewit.080

Test Problem 7: Tube Banks

- **Problem Statement:**

A heat exchanger, with staggered tubes, is used to heat atmospheric air entering at 5°C. The temperature of the surface of the tubes is maintained at 100°C with steam condensing inside the tubes. Other details of the heat exchanger are

Diameter of the tubes = 25 mm	$U_\infty = 15$ m/s
Number of columns = 20	Number of rows = 20
$S_L = S_T = 50$ mm	Length of tubes = $L = 3$ m

Determine

a) The exit temperature of air and the heat transfer rate.
b) The mass rate of condensation of steam assuming the steam condenses from saturated dry vapor state at 125 kPa.
c) The fan power required to maintain the air flow.

➢ **Source:**

N.V. Suryanarayana. 1995. Engineering Heat Transfer. West Publishing Company. Minnesota/St Paul. Page 434.

➢ **Comments**

- Assume an exit temperature of 60°C. The arithmetic mean of the inlet and exit temperature of the air is 32.5°C. From the software CC. the properties of air at 32.5°C and 101.3 kPa are
  ρ = 1.155 kg/m^3 $\quad$ c_p = 1007 J/kg°C
  μ = 1.877 x 10^{-5} N s/m^2 $\quad$ k = 0.0266 W/m°C
  Pr = 0.712 $\quad$ Pr_s(100°C) = 0.705
-
- Diagnostic results from source are compared below with INSTED predictions.

Variable	**Suryanarayana**	**INSTED®**	**Difference**
h (W/m^2 °C)	209.7	207.72	< 1%
Nu_d	205.6	195.23	< 6%
T_{be} (°C)	32.1	31.61	< 2%

Explanation for differences in results

The main source of the difference is the use of iterations in the Source. After the first iteration, the values in the Source and INSTED agree to within 1%. The values shown in this table for the Source are the converged values. The agreement with INSTED results are within the tolerance for this type of analysis.

♦ Name of data file: SURYA.434

Test Problem 8: Finned-Tube Banks

➢ **Problem Statement:**

A heat exchanger consists of four rows of eight steel tubes [thermal conductivity: 15 W/(m·K)] in equilateral triangular array (TLA= 30°) fitted with corbels or dummy half-tubes, to reduce bypass flow. The tubes have roll-formed rectangular cross-section fins and the following dimensions:

L=0.5 m (length of tube)
D_r = 1.64 x 10^{-2} m (diameter of bare tubes)
D_f = 2.46 x 10^{-2} m (diameter with fins)
h = 4.1 x 10^{-3} m (fin height)
w = 1.0 x 10^{-3} m (fin width)
s= 2.0 x 10^{-3} m (fin spacing)
p_1 =3.13 x 10^{-2} m (transverse pitch)
p_2 = 2.71 x 10^{-2} m (longitudinal pitch)

The tubes are heated on the inside by a condensing vapor that maintains a uniform tube wall surface temperature of 70°C (343 K) and cooled on the outside by cross-flow air initially at 15°C (288K) flowing at a rate of 0.914 kg/s. What is the total rate of heat transfer and pressure drop across the heat exchanger?

- **Source:**

Hewitt, G.F., Shires. G.L. and Bott, T.R. 1994. Process Heat Tranfser. CRC Press. pp.90

- **Fluid Properties**

For the purpose of this example, variation of fluid properties with temperature and pressure may be neglected and the following overall values may be used:

ρ= 1.217 kg/m^3 (density of fluid)
η = 1.80 x 10^{-5} (Ns)/m^2 (absolute viscosity of fluid)
c_p = 1007 J/(kg·K) (fluid specific heat at constant pressure)
λ = 2.53 x 10-2 W/ (m·K) (thermal conductivity of fluid)
Pr = 0.71 (Prandtl number of fluid).

- **Solution:**

Diagnostic results are compared in the table below.

Variable	**Hewitt**	**INSTED**	**Difference**
A_f (m^2) Fin surface area	2.542*	3.091	N/A
A_w (m^2) Area between fins	0.550	0.54957	< 1%
A (m^2) Total system area	3.092*	3.64089	N/A
V_{max} (m/s)	14.5	14.524	< 1%
Re	1.61 x 10^4	1.61049 × 10^4	< 1%
Nu	82.5	66.214	19.75%
Fin Efficiency	0.89	.91	< 3%
α (W/(m^2 · K))	127.3	102.147	19.75%
α' (W/(m^2 · K)) Effective	115.8	94.436	18.45%
ΔT (K)	45.6	45.8	< 1%
Q (W)	1.63 x 10^4	1.5755 × 10^4	< 4%
Δp (N/m^2)	414 (266.86)	- (270.03)	- (<2%)

*There seems to be some typographical error in the source with respect to the calculation of these areas. The difference in Nusselt number and heat transfer coefficients are due mostly on the empirical models used. The source uses the ESDU data where the models in INSTED are of a different type. The pressure calculation in the source includes entrance losses whereas it is not included in the INSTED solutions. The pressure values in parenthesis do not include entrance losses and compare very well. In fact, given the uncertainty in the empirical equations, the agreements are quite good.

Name of file: HEWITT.090

Test Problem 9: Finned-Tube Banks

- **Problem Statement:**

Air flows over a six-row finned tube bundle which has 10 inverse rows of tubes with a length of 2 m. The tubes have a base diameter of 2 cm, a pitch of 6 cm and have 200 radial fins per meter with a height of 1.5 cm and a constant thickness of 2 mm.The material of construction is a stainless steel which has a thermal conductivity of 30 W/mK. The air has a mass rate of flow of 20 kg/s and may be assumed to have following physical properties;

ρ =1.2 kg/m^3

C_{pb} = 1 kJ/kg K

η_b = 1.8 x 10^{-5} Ns/m^2

λ_b = 0.025 (W/m^2)/(K/m)

N_T=10, N_L=6, L=2, K_b=30 W/m.K, fin width (w) = 2mm, tube external diameter = 2 cm, pitch = 6 cm, fin height = 1.5 cm, fin spacing = ((1/200) - w)

Assuming negligible variation of physical properties, calculate the weighted heat transfer coefficient for the cases where the tubes are arranged in a square (in-line) and a triangular (staggered) arrangement respectively.

- **Source:**

Hewitt, G.F. 1998. Heat Exchanger Design Handbook. begell house, inc. Publishers. Page 2.5.3-25

- **Solution:**

Diagnostic comparison of results for the staggered case is presented in the following table.

Variable	**Kaunas**	**INSTED**	**Difference**
A_f (m^2) Fin surface area	86.71	86.71	< 1%
A_w (m^2) Area between fins	4.52	4.524	< 1%
V_{max} (m/s)	29.76	29.7619	< 1%
Re	39680	39682.5	< 1%
* Nu	79, 92.8, 161.49, 138.56	156.92	-97%, -69%, +3%, -14%
*Fin Efficiency	0.622, 0.638, 0.466, 0.498	.5755	+8%, +10%, -24%, -16%
*α (W/(m$^2\cdot$K))	98.75, 116.05, 201.86, 173.21	195.15	-98%, -69%, +4%, -13%
*α' (W/(m$^2\cdot$K)) Effective	63.27, 76.12, 99.40, 90.57	117.24	-84%, -54%, -18%, -30%

*The results using 4 methods to calculate the Nusselt number (Nu), fin efficiency, film coefficient (α) and effective film coefficient (α') are compared in the table. The four methods, in the order in which they are presented for these variables are: Schmidt method (In-line), PFR (In-line), Kaunas (Staggered), and PFR (staggered). Note that the INSTED results in the table are for the staggered case, so that a direct comparison with the Schmidt and the PFR methods, which use the in-line configuration, is not possible from this table. The various methods come with different levels of complexity. The Schmidt and PFR methods are rather very simple, allowing only for the ratio of heat transfer areas (other than the standard Reynolds number and Prandtl number. The Kaunas (Zukauskas and his co-workers at the

Lithuanian Energy Institute) and the models in INSTED are far more elaborate, as they include many other geometric factors. The effect of rounding off numbers in hand calculations should be acknowledged. During the comparison exercise, we have observed significant errors when hand calculations truncate numbers. Finally, the agreement between the various methods is reasonable; i.e., within the likely data scatter.

Name of file: KAUNAS.325

Test Problem 10: Finned-Tube Banks

- **Problem Statement:**

Suppose that the plain tubes described in the Example in Section A(h) are replaced by low fin tubes of 2.32, a fin outside diameter and with fins of height 1.6mm, with 630 fins per meter and with a fin thickness of 0.3 mm. Assuming a bulk temperature of 24C and constant physical properties, calculate the heat transfer coefficient for this revised bundle using both the ESDU and Rabas and Taborek correlations and compare the coefficient (and the multiple of the coefficient and the total cases. The thermal conductivity of the tube material is 30 $(W/m^2)/(K/m)$.

$\eta_b = 8.26 \times 10^{-5}$ Ns/m^2
$\lambda_b = 0.126$ $(W/m^2)/(K/m)$
$C_{pb} = 2.2$ kJ/kg K
$\rho = 730$ kg/m^3
Pr = 14.42

- **Source:**

Hewitt, G.F. 1998. Heat Exchanger Design Handbook. begell house, inc. Publishers. Page 2.5.3-27

- **Solution:**

Diagnostic comparison of results for the staggered case is presented in the following table.

Variable	Kaunas	INSTED	Difference
A_f (m^2) Fin surface area	18.07	17.84	< 2%
A_w (m^2) Area between fins	6.11	6.115	< 1%
V_{max} (m/s)	.1458	.1458	< 1%
Re	2577	2577.2	< 1%
* Nu	81.68, 72.76	59.40	+28%, +19%
*Fin Efficiency	0.856, 0.894	.928	-9%, -4%
*α $(W/(m^2 \cdot K))$	514.58, 458.39	374.24	+28%, +19%
*α' $(W/(m^2 \cdot K))$ Effective	457.9, 422.08	354.23	+23%, +17%

*The results using 2 methods to calculate the Nusselt number (Nu), fin efficiency, film coefficient (α) and effective film coefficient (α') are compared in the table. The two methods, in the order in which they are presented for these variables are: the ESDU methods and the correlation by Rabas and Taborek. The equations used by the various methods are fairly complicated, as are the INSTED models. The effect of rounding off numbers in hand calculations should be acknowledged. During the comparison exercise, we have observed significant errors when hand calculations truncate numbers. Finally, the agreement between the various methods is reasonable; i.e., within the likely data scatter.

Name of file: KAUNAS.327