SHELL-AND-TUBE TEST PROBLEMS

The problems that have been used to validate some of the capabilities in INSTED for the analysis of shell-and-tube heat exchanger are discussed in this chapter. You should 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. It is expected that you would have simulated some of these test problems before you attempt to solve more realistic engineering problems.

Test Problem 1

- **Problem Statement:**

In a facility where electricity is generated, condensed (distilled) water is to be cooled by means of a shell- and-tube heat exchanger. Distilled water enters the exchanger at 110^{o}F at a flow rate of 170,000 lbm/hr. Heat will be transferred to raw water (from a nearby lake) which is available at 65^{o}F and 150,000 lb/hr. Preliminary calculations indicate that it may be appropriate to use a heat exchanger that has a 17¼ in. inside diameter shell, and 3/4 in. OD, 18 BWG tubes that are 16 ft. long. The tubes are laid out on a 1 in. triangular pitch, and the tube fluid will make two passes. The exchanger contain baffles that are spaced 1 ft. apart. Analyze the proposed heat exchanger to determine its suitability.

- **Source:**

William S. Janna. Design of Thermal Systems. PWS-Kent Publishing. Boston. Page 251.

- **Comments**

- In INSTED, choose the 'outlet temperature' task.
- Hot fluid (distilled water) is in the shell. Raw water is in the tubes.
- In addition to the data above, it is also stated that the number of tubes is 196, the number of shell passes is 1, and the number of baffles is 15. From 2 tube passes and 196 total number of tubes, the total number of tube pass is 98.
- Specify triangular pitch, 2 tube passes.
- Mass flow rates of both tube and shell fluids are known.
- Use online conversion to convert all British units to SI.
- Use thermophysical properties given in source. Obtain absolute viscosity, μ, from the ρ and υ provided in the source.
- Overall U is now known apriori. Both the inner and outer film coefficients of the inner tube were obtained from internal flow analysis.
- To obtain the inner film coefficient of the inner tube, click the **Compute Tube Fim Coeff.** button on the 'Heat Transfer' dialog box.
- The heat transfer coefficient at the outer surface of the tubes is calculated as a function of the geometry of the system and the flow rate of the shell fluid. The friction factor calculation for the shell flow is calculated as 0.2774, the Nusselt number for shell flow is 126.8, and calculated film coefficient is 4427 W/m^{2}K.

- Note that the Kern's approach is used for shell-side film coefficient. More accurate results could be obtained with the Bell-Delaware and Stream Analysis methods but the extra data required for these methods are not available from the Source.
- The problem is solved without fouling factors. For the case with fouling factor, a fouling factor of 0.00015 m2K/W is assumed at both the inner and outer surfaces of tubes.
- Diagnostic results from source are compared below with INSTED predictions. Note that 's' and 't' below stands for shell and tube respectively.

➢ **Comparison of INSTED result with source**

Variable	**Janna (British)**	**Janna (SI)**	**INSTED (SI)**	**Difference**
Velocity (s)	2.123 ft/s	0.647074	0.64707 m/s	< 1%
Velocity (t)	2.943 ft/s	0.89696	0.896795 m/s	< 1%
Δp (s)	3.26 psi	22477.0	22487.70 N/m^2	< 1%
Δp (t)	1.4 psi	9652.0	9831.0 N/m^2	1.9 %
Friction (s)	0.277	0.277	0.277369	< 1%
Friction (t)	0.027	0.027	0.02793	3.4 %
Re(s)	1.769 x 10^4	1.769 x 10^4	1.77 x 10^4	< 1%
Re(t)	1.476 x 10^4	1.476 x 10^4	1.475 x 10^4	< 1%
h (s)	783.7 Btu/hft^{o}F	4449.8	4429.08 W/m^{2}K	< 1%
h (t)	689.2 Btu/hft^{o}F	3913.2	4057.48 W/m^{2}K	3.7 %
U	224.9 Btu/hft^{o}F	1277.0	1157.17 W/m^{2}K	9.4%
Temp. out (s)	89.8 ^{o}F	305.26	306.70 K	<1 %
Temp. out (t)	88 ^{o}F	304.98	302.56.7 K	< 1%

Explanation of differences in results

- Agreement between the two sets of results is apparent. The largest difference is observed in the heat transfer coefficient and the U value. The differences in the heat transfer coefficient results are mostly due to differences in the equation used for Nusselt number calculation. Dittus-Boelter is used in Janna, whereas INSTED uses Gnielinski-type equation. Use of Dittus-Boelter equation in INSTED gave Nusselt number and heat transfer coefficient results, which are within 1% of the values reported in Janna. For a computer-based approach, like INSTED, it seems appropriate to use a more accurate (and more complicated) empirical relation, such as the Gnielinki's formula, for the Nusselt number calculation. In fact, and as pointed out by Chapman (1990), the Dittus-Boelter equation should not be used in situations where temperature difference is very large. Log mean temperature difference in the present problem is approximately 12.96 K, which is modest and points to the close agreement between INSTED predictions and the results in Janna. Concerning the differences in the U value, INSTED includes thermal conductivity of the tube material.

- The F-factor calculated in Janna is actually for the case without fouling, although other analyses (calculated on page 278 of Janna) assume fouling. You might run INSTED for this problem with or without fouling factors. Results that are respectively within 1% of Janna's will be observed in either case.

- Name of data file:
  (a) JANNA.275

Test Problem 2

Problem Statement:

This is the same as test problem 1 above, except for the assumption that tube length is not known, and the task is to find the length of each tube in each pass. The output temperatures, which were calculated in test problem 1, are used as input data for the present test. (**Important Note**: This Task has been removed in the current version of INSTED. The user should therefore skip this test problem.)

Discussion of Result

The solution of test problem 2 using INSTED is a length of 4.52 m (sample screens are not shown for this test problem), compared to 4.877 m that was specified in test problem 1. The reason for this discrepancy (7.3% error) is now explained. For the 'length of tube' task, we assume the inner and the outer area of the tubes are equal in the calculation of U. On the other hand, when the length of tubes is known, U is based on the outer surface area of the tubes. The error caused by the assumption above increases with increasing thickness of tube. To be convinced, you could run INSTED, with the task of finding tube length, and assume that U is known (from test problem 1 above). A value of 1201 W/m^{2}K (not 1294, and not calculated with the internal flow module) should be specified for U as an input value to INSTED. The calculated length of tube will be approximately 4.87 m, which is closer to the specified length in test problem 1 and what one expects to find.

Test Problem 3

- **Problem Statement:**

A shell-and-tube heat exchanger has the following geometry:

Shell internal diameter	0.5398 m
Number of tubes	158
Tube outer diameter	2.54 cm
Tube inner diameter	2.0574 cm
Tube pitch (square)	3.175 cm
Baffle spacing	12.70 cm
Shell Length	4.8768 m
Tube-to-baffle diametral clearance	0.88 mm

Shell-to-baffle diametral clearance	5 mm
Bundle-to-shell diametral clearance	35 mm
Split backing ring floating head	assumed
Number of sealing strips per cross-flow row	1/5
Thickness of baffles	5 mm
Number of tube-side passes	4

(a) Use the Kern's method to calculate the shell-side heat transfer coefficient and pressure drop for the flow of a high hydrocarbon with the following specification (at bulk temperature):

Total mass rate of flow	5.5188 kg/s
Density	730 kg/m3
Thermal conductivity	0.1324 W/mK
Specific heat capacity	2,470 kJ/kgK
Viscosity	401 μNs/m^2

Assume no change in viscosity from the bulk to the wall.

(b) Use the Bell-Delaware method to calculate the shell-side heat transfer coefficient and pressure drop for the above problem and compare the results.

(c) Use the Stream Analysis method to calculate the shell-side heat transfer coefficient and pressure drop for the above problem and compare the results.

- **Source:**

Hewitt, G. F., Shires, G. L., Bott, T. R. Process Heat Transfer, CRC Press. 1998. Pages 273, 280, 289.

- **Comments**

- Select the 'Heat Flow Rate' button in the 'Problem Description' dialog box.
- Enter the appropriate data in the following dialog boxes - 'Tube Properties', 'Shell Properties', 'Shell and Tube Fluids', 'Heat Transfer Coeffs.'. Alternatively, you could load the data (for the above problem) from the sample data using the 'Project Admin' dialog box.
- Enter 'dummy' values for thermophysical properties of the tube fluid.
- Overall U is now known apriori. Both the inner and outer film coefficients of the inner tube were obtained from internal flow analysis.

- **Comparison of INSTED result with source**

<u>Kern's Method</u>

Variable	Hewitt	INSTED	Difference
Δp (Pa)	22,224	22,866.3	< 3%
Outer tube heat coeff. (W/m^{2}K)	977.94	971.444	< 1%

Bell-Delaware Method

Variable	Hewitt	INSTED	Difference
Δp (Pa)	4501	3439	< 23%
Outer tube heat coeff. (W/m^2K)	822	789.44	< 4%

Stream Analysis Method

Variable	Hewitt	INSTED	Difference
Δp (Pa)	3052	3688.93	< 21%
Outer tube heat coeff. (W/m^2K)	865	836.345	< 4%

Explanation of differences in results

- The widely-different pressure solutions obtained for different methods is apparent, as is the discrepancy between the source and INSTED for this variable. The heat transfer calculations agree well between the Source and INSTED. The Kern method is does not allow leakages. Therefore, the pressure results are unacceptably high. Overall, the agreement between the Source and INSTED is within the tolerance for empirical calculations.

♦ Name of data file:
(b) HEWITT.INP

Test Problem 4

➢ **Problem Statement:**

Determine heat transfer rate, outlet fluid temperatures and pressure drops on each fluid side for a TEMA E shell-and-tube heat exchanger with a fixed tubesheet and one shell and two tube passes. The tubes in the bundle are in 45° rotated square arrangement. The fluids are lubricating oil and sea water. Fouling factors for the oil and water sides are 1.76×10^{-4} m^2K/W and 8.81×10^{-5} m^2K/W, respectively. The geometric dimensions and operating properties are given in the following two tables. Assume mean fluid temperatures to be 63°C and 35°C for oil and water, respectively.

Shellside inside diameter, D_s = 0.336 m   Number of sealing strip pairs, N_{ss} = 1
Tubeside outside diameter, d_o = 19.0 mm Total number of tubes, N_t = 102
Tubeside inside diameter, d_i = 16.6 mm   Tube length, L = 4.3 m
Tube pitch, p_t = 25.0 mm   Width of bypass lane, w_p = 19.0 mm
Tube bundle layout = 45°   Number of tube passes, n_p = 2
Central baffle spacing, $L_{b,c}$ = 0.279 m Number of pass divider lanes, N_p = 2
Inlet baffle spacing, $L_{b,i}$ = 0.318 m   Diameter of the outer tube limit, D_{otl} = 0.321 m
Outlet baffle spacing, $L_{b,o}$ = 0.318 m   Tube-to-baffle hole diametral clearance, δ_{tb} = 0.794 mm

Baffle cut, ℓ_c = 86.7 mm   Shell-to-baffle diametral clearance, δ_{sb} = 2.946 mm

Tube material = admiralty (70% Cu, 30% Ni). Thermal conductivity k_w of tube wall is 111 W/mK.

(Baffle thickness = 0.00643 m, N_b = 14)

Operating Conditions

Oil flow rate, $\dot{m}_{oil} = \dot{m}_s = 36.3\,\text{kg/s}$ Water flow rate, $\dot{m}_{water} = \dot{m}_t = 18.1\,\text{kg/s}$

Oil inlet temperature, $T_{s,i}$ = 65.6°C   Water inlet temperature, $T_{t,i}$ = 32.2°C

Fluid	Density ρ_s, kg/m^3	Specific heat c_p, J/kg K	Dynamic viscosity, μ, Pa s	Thermal conductivity k, W/m^{2}K	Prandtl number Pr
Oil at 63°C	849	2094	64.6×10^{-3}	0.140	966
Sea water at 35°C	993	4187	0.723×10^{-3}	0.634	4.77

Use the Dittus-Boelter correlation, Eq. (7.77) in Table 7.6, for the tubeside heat transfer coefficient, and Table 7.12 for the shellside Nusselt number. Use McAdams correlation, Eq. (7.69) in Table 7.6, for the tubeside friction factor. For the shellside friction factor, use the following correlation: $f_{id} = 3.5(1.33 d_o / p_t)^b \, Re_s^{-0.476}$ where $b = 6.59/(1 + 0.14\, Re_s^{0.52})$.

Source: Dr. Ram Shah (Private Communication)

- **Comparison of INSTED result with source**

Variable	**Shah**	**INSTED**	**Remarks**
Velocity (s), m/s		1.304	"s" -> Shell, "t"-> tube
Velocity (t), m/s		1.651	
Δp (s), kPa	112* kPa	113.93* kPa	*Both are Bell-Delaware results
Δp (t), kPa	17.58 (15.62*)	15.62*	*Excludes minor losses
Friction (s)	-	1.043	
Friction (t)	0.02223	0.02223*	
Re(s)	326	325.63	
Re(t)	37643	37650	
h (s), W/m^{2}K	698.8	579.33	Base model equations for h differ
h (t), W/m^{2}K	7837	8343.38	
U, W/m^{2}K	536.1	462.72	Due differences in h(t), h(s)
T_{out} (s), K	333.15	334.19	
T_{out} (t), K	310.55	309.95	

Duty, kW	393.6	347.64*	* INSTED Result is consistent with $MC_p\Delta T$ for both shell and tube fluids
NTU	0.185	0.16	
Effectiveness	0.1555	0.1373	

- Name of data file:
  MAE522UB.inp

Test Problem 5

- **Problem Statement:**

Determine heat transfer rate, outlet fluid temperatures and pressure drops on each fluid side for a TEMA E shell-and-tube heat exchanger with one shell and one tube pass. The tubes in the bundle are in 45° rotated square arrangement. The cold fluid is R134a which will boil in cooling some proprietary fluid with given thermophysical properties. The geometric dimensions and operating properties are given in the following two tables. Assume inlet fluid temperatures to be 235K and 380.76K for R134a and fluid, respectively. The inlet pressure of R134a is 1 atm. As a result, the refrigerant enters into the heat exchanger in a completely liquid state.

Shellside inside diameter, D_s = 0.336 m   Number of sealing strip pairs, N_{ss} = 1
Tubeside outside diameter, d_o = 19.0 mm   Total number of tubes, N_t = 102
Tubeside inside diameter, d_i = 16.6 mm   Tube length, L = 4.3 m
Tube pitch, p_t = 25.0 mm   Width of bypass lane, w_p = 19.0 mm
Tube bundle layout = 45°   Number of tube passes, n_p = 1
Central baffle spacing, $L_{b,c}$ = 0.279 m   Number of pass divider lanes, N_p = 2
Inlet baffle spacing, $L_{b,i}$ = 0.318 m   Diameter of the outer tube limit, D_{otl} = 0.321 m
Outlet baffle spacing, $L_{b,o}$ = 0.318 m   Tube-to-baffle hole diametral clearance, δ_{tb} = 0.794 mm
Baffle cut, ℓ_c = 86.7 mm   Shell-to-baffle diametral clearance, δ_{sb} = 2.946 mm
Tube material = admiralty (70% Cu, 30% Ni). Thermal conductivity k_w of tube wall is 111 W/mK.

(Baffle thickness = 0.002643 m, N_b = 14)

Operating Conditions

R134a flow rate, $\dot{m} = \dot{m}_s = 20.3\,kg/s$ Water flow rate, $\dot{m}_{R134a} = \dot{m}_t = 8.0\,kg/s$

Fluid	Density ρ_s, kg/m^3	Specific heat c_p, J/kg K	Dynamic viscosity, μ, Pa s	Thermal conductivity k, W/m^{2}K
Proprietary	1326	1289	3.24×10^{-4}	0.0989

fluid at 380K				

Source: (Internal Diagnostic Data)

- **Sample INSTED results**

Variable	**INSTED**	**Remarks**
Δp (s), kPa	17195 kPa	
Δp (t), kPa	5004.4	
Friction (s)	0.49	
Friction (t)	0.595	
h (s), W/m^2K	1223.1	
h (t), W/m^2K	905.6	
U, W/m^2K	1399.9	
T_{out} (s), K	304.45	
T_{out} (t), K	292.05	
Duty, kW	2.03×10^6	
R134a Outlet Quality	100%	

The next figures show the temperature distribution for both the shell and tube fluids and quality through the exchanger.

- Name of data file:
  Tubeboil.inp