CONCENTRIC EXCHANGER TEST PROBLEMS

Introduction
The tests used to validate INSTED analysis of concentric exchanger module are presented here. 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

- **Problem Statement:**

A counterflow, concentric tube heat exchanger is used to cool the lubricating oil for a large industrial gas turbine engine. The flow rate of cooling water through the inner tube (D_i = 25mm) is 0.2 kg/s, while the flow rate of oil through the outer annulus (D_0 = 45mm) is 0.1 kg/s. The oil and water enter at temperatures of 100 and 30^{o}C, respectively. How long must the tube be made if the outlet temperature of oil is to be 60^{o}C.

- **Source:**

Frank P. Incropera and David P. Dewitt. 1990. Introduction to Heat Transfer. John Wiley & Sons. Page 611.

- **Comments**

- In INSTED, choose the 'length of tube' task.
- Hot fluid (oil) is in annulus.
- Arrangement is counterflow.
- Annulus fluid outlet temperature is known.
- Mass flow rates of both tube and annulus fluids are known.
- Use online conversion to convert all British units to SI.
- Use thermophysical properties given in source. Obtain density of water. Density of unused engine oil, and Prandtl number of unused engine oil from Thaerocomp's database. The latter are not provided in the source.
- Inner diameter of the inner tube is specified as 0.025m. The outer diameter of inner tube is not provided. A 'dummy' value of 0.025001m is used since INSTED requires this data. The appropriate wall thickness would have been used, if pipe dimension had been obtained from Thaerocomp's database.
- 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,
- To obtain the outer film coefficient of the inner tube,
- There is no fouling factor for either surface.
- Diagnostic results from source are compared below with INSTED predictions. Note that 'a' and 't' below stand for annulus and tube respectively.

- **Comparison of INSTED result with source**

Variable	Incropera	INSTED	Difference
Re(a)	14, 050	14, 050	< 1%
Re(t)	56.0	55.97	< 1%
Nu(a)	5.56	5.642	1.3 %

Variable	Chapman	INSTED	Difference
Nu(t)	90	92.918	3.2 %
h (a)	38.4	38.93 W/m^2K	1.2 %
h (t)	2250	2322.97 W/m^2K	3.2 %
U	37.8	38.24 W/m^2K	1.2 %
Length	66.5	65.70 m	1.2 %
Temp. out (T)	313.35	313.35 K	< 1%

Explanation of differences in results

- The differences in the 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 the present code, 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 in the present problem is approximately 43.20 K, which seems to high for the use of the Dittus-Boelter equation. The differences for the annulus fluid are quite small.
- The friction factor is excluded in the source but included in the INSTED procedure.
- Note: Because the flow in the annulus is laminar, one is tempted to use a modified Sieder and Tate equation to calculate Nusselt number. However, this is not of much help in te present problem because the length of pipe, which is required in the Sieder and Tate model, is not known. Other approaches are used in INSTED to circumvent this situation.

♦ Name of data file:
  (a) INCROP2.611 (hydraulic diameter used for annulus)
  (b) INCROP1.611 (equivalent diameter used for annulus)

Test Problem 2

➢ **Problem Statement:**

Water at a temperature of 195°F and a mass flow araate of 5000 lbm/hr is to be used to heat ethylene glycol. The ethylene glycol is available at 85°F with a mass flow rate of 12,000 lbm/hr. A double pipe heat exchanger consisting of a 1¼- standard type M copper tubing inside of 2-standard type M copper tubing is to be used. The exchanger is 6 ft. long. Determine the outlet temperature of both fluids using counterflow and again using parallel flow.

Source:

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

➢ **Comments**

- In INSTED, choose the 'outlet temperature' task.
- Cold fluid (ethylene glycol) is in annulus.
- Parallel exchanger will be discussed, although both the parallel and counterflow were analyzed with INSTED, with very good agreements with the results in the source. The data files for both cases are included in the distribution disks.
- Mass flow rates of both tube and annulus 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,
- To obtain the outer film coefficient of the inner tube,
- There is no fouling factor for either surface.
- Diagnostic results from source are compared below with INSTED predictions. Note that 'a' and 't' below stand for annulus and tube respectively.

➢ **Comparison of INSTED result with source**

Variable	**Janna (British)**	**Janna (SI)**	**INSTED (SI)**	**Difference**
Velocity (a)	4.20 ft/s	1.2802	1.281	< 1%
Velocity (t)	2.48 ft/s	0.756	0.7574	< 1%
Re(a)	1.07×10^4	1.07×10^4	1.088×10^4	1.7 %
Re(t)	5.20×10^4	5.20×10^4	5.5207×10^4	< 1%
Nu(a)	185	185	174	1.3 %
Nu(t)	190	190	233.2	3.2 %
h (a)	214 Btu/hft^{o}F	1215	1140 W/m^{2}K	1.2 %
h (t)	664 Btu/hft^{o}F	3770	4628 W/m^{2}K	3.2 %
U	159 Btu/hft^{o}F	902.7	903.4 W/m^{2}K	1.2 %
Temp. out (a)	89.8 ^{o}F	305.26	305.3 K	1.2 %
Temp. out (t)	188 ^{o}F	359.8	359.7 K	< 1%

Explanation of differences in results

- The differences in the 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 the present code, 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 in the present problem is approximately 57.71 K, which seems to high for the use of the Dittus-Boelter equation. The differences for the annulus fluid are quite small.
- The friction factor is excluded in the source but included in the INSTED procedure.
- It seems suprising that, inspite of the differences in Nusselt number and film transfer coefficient values, the final temperature values, and the overall U value are quite close. The reason is due to the annulus coefficient, which is controlling, and has values that are reasonably close in the source and INSTED.

♦ Name of data file:
(c) JANNA.251 (parallel)
(d) JANNA.252 (counterflow)

Test Problem 3

- **Problem Statement:**

Benzene is used in a process for the manufacture of detergent. A double pipe heat exchanger must be sized to exchange heat between benzene (ρ = 54.3 lbm/ft3. C_p = 0.425 BTU/lbm.^{o}R, μ = 50 cp, k_f = 0.091 BTU/hr.ft.^{o}R and Pr = 1.78) and water. The benzene flow rate is 10,000 lbm/hr and it is to be heated from 75^{o}F to 125^{o}F. The water is available at 200oF. Select an appropriate heat exchanger and determine the required water flow rate.

- **Source:**

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

- **Comments**

- In INSTED, choose the 'length of tube' task.
- Hot fluid (water) is in the annulus.
- Results are presented for counterflow. The data file for this case is included in the distribution disks.
- Annulus fluid outlet temperature is known.
- Note that only fluid mass flow rate is known, initially. However, an input value of 4.31 lbm/s is specified for the annulus fluid based on a preliminary analysis in Janna. This just goes to remind us that a computer code cannot do all the thinking work.
- 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.
- Inner diameter of the inner tube is specified as 0.025m. The outer diameter of inner tube is not provided. A 'dummy' value of 0.025001m is used since INSTED requires this data. The appropriate wall thickness would have been used, if pipe dimension had been obtained from Thaerocomp's database.
- 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,
- To obtain the outer film coefficient of the inner tube,
- A fouling factor of 2 x 10^{-4} m^2K/W is assumed for both inner and outer surface of inner tube.
- Diagnostic results from source are compared below with INSTED predictions. Note that 'a' and 't' below stand for annulus and tube respectively.

SHELL AND TUBE TEST PROBLEMS

Introduction

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. Because the shell and tube heat exchanger module is intended for realistic systems, it will not be suitable for many textbook examples. Test problems 1 and 2 are from the text 'Design of Thermal Systems', by William Janna. This book is one of the few texts that treat the shell and tube heat exchanger problem from a realistic engineering viewpoint and provides realistic examples. The second example is obtained from a realistic engineering problem while the final one is from the text 'Process Heat Transfer' by Hewitt G. F., Shires, G. L. and Bott, T. R. 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

- **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 100^{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 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 tube 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.

- 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	0.64704	0.64704 m/s	< 1%
Velocity (t)	2.943 ft/s	0.89696	0.89696 m/s	< 1%
Δp (s)	3.26 psi	22477.0	22477.0 N/m^2	< 1%
Δp (t)	1.4 psi	9652.0	9652.0 N/m^2	1.9 %
Friction (s)	0.277	0.277	0.2774	< 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%
Nu(s)	127.4	127.4	126.8	< 1%
Nu(t)	108.5	108.5	112.3	3.5 %
h (s)	783.7 Btu/hft^{o}F	4449.8	4427 W/m^{2}K	< 1%
h (t)	689.2 Btu/hft^{o}F	3913.2	4052 W/m^{2}K	3.5 %
U	224.9 Btu/hft^{o}F	1277.0	1201 W/m^{2}K	< 1%
Temp. out (s)	89.8 ^{o}F	305.26	305.3 K	1.2 %
Temp. out (t)	188 ^{o}F	359.8	359.7 K	< 1%

Explanation of differences in results

- Agreement between the two sets of results is apparent. The largest difference is observed in the Nusselt number for tube flow. The differences in the 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 the present code, 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 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..
- Outlet temperature results are not compared. 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:
(e) JANNA.275

Test Problem 2

- **Problem Statement:**

Recommend a suitable shell-and-tube heat exchanger configuration for the following service. The information given in the answer should be complete enough either for preliminary pricing and plant layout or for entering into the Delaware method.

Design a shell and tube heat exchanger to heat 49,800 bpd (597,000 lb/hr) of 34^o API Midcontinent crude from 125^{o}F to 180^{o}F, using 13,200 bpd (152,000 lb/hr) of 28^o API gas oil at 410^{o}F, cooling it to 220^{o}F. Pressure drop is limited to 15 psi on each side.

The properties of the fluids are:

	Hot Fluid (315^{o}F)	Cold Fluid (150^{o}F)
Density (lbm/ft^3)	49.3	51.2
Specific Heat (Btu/lbm^{o}F)	0.58	0.51
Viscosity (bulk) (lbm/fth)	2.90	7.0
Viscosity (wall) (lbm/fth)	7.5 (200^{o}F)	4.4 (200^{o}F)
Thermal Conductivity (Btu/hr ft^{o}F)	0.061	0.071

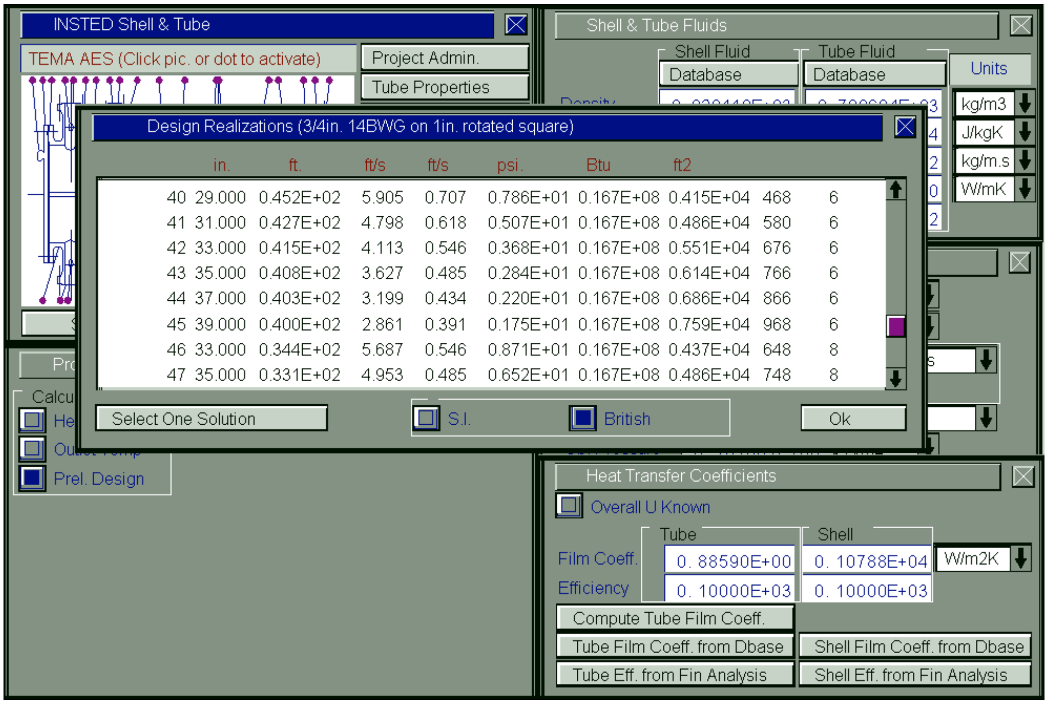
- The fouling factor for both crude and gas oil will be taken as 0.002 hrft^{2o}F/Btu for each stream.
- In addition, the following are also to be taken into consideration:
- TEMA standards apply. Preferably place the hot fluid in the tubes since the hotter fluid is more likely to foul.
- The data for this sample problem is also located in the file X:\insted30\shtube\samples\prelimd.inp (where X is the drive in which you chose to install the demo).

- **Source:**

Bell, K. J., Section 3.1.4. of Heat Exchanger Design Handbook, 1998

- **Comments**

- Select the 'Prel Design' 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.
- Click on the 'Design Dialog' button to open up the associated dialog box. Provide the bounds (min, max) on velocity, shell diameter and tube length that should be enforced. Also enter the maximum allowed flow rate and pressure. (It is advised to be initially generous with the constraints in order not to reject all design possibilities at the first design attempt.) Click **Ok** on this dialog when you are done.
- Click 'Compute' on the Main dialog box. Various design realizations are listed in a table.
- Use the mouse to select the design you prefer most. Click the 'Select One Solution' button. This design is accepted. You are now ready to do a detailed rating calculation. (Note that clicking Ok without selecting one of the design options rejects all the realizations. You may then alter some values or relax/tighten some constraints and try again.)



- **Comparison of INSTED result with source**

For 3/4 in. 14 BWG tubes with 6 passes, the following design realizations were extracted for comparison with the result in the source.

Shell Diameter (in.)	**Shell Length (ft.)**			**No. of Tubes**	**Inner Tube heat coeff. (W/m^2K)**	**Outer Tube heat coeff. (W/m^2K)**
	Bell	**INSTED**	**Difference**			
39	13.5	40.0	1%	968	196	497
37	15	40.3	1%	866	233	512
35	16.5	40.8	1%	766	256	528
33	19	41.5	1.9 %	676	294	545
31	21	42.7	1%	580	347	554
29	25	45.2	3.4 %	468	434	586
27	29	46.2	1%	420	486	609

Comparison of Result for Shell diameter = 31 in.

Variable	**Bell**	**INSTED**	**Difference**
Velocity (t)	4.70 ft/s	4.798 ft/s	<1%
Δp (t)	9.73 psi	5.07 psi	1.9 %
Friction (t)	0.027	0.02793	3.4 %
Re(t)	14000	14285.3	
Area	3880 ft^2	4861.308 ft^2	
Q	1.675 X 10^7 Btu/hr	1.67 X 10^7 Btu/hr	<1%

Explanation of differences in results

- The calculations in the source are based on a fixed inner tube heat transfer coefficient; with a value of 1100 W/m^2K. The calculations used in INSTED for the various shell diameters is based on an internal flow analysis of the heat loss in the inner walls of the tubes. This calculation is dependent on several parameters including the flow rate within the tubes. The flow rate within the tubes will depend on the number of tubes since the total flow rate remains constant. Consequently, an approach using a case-dependent calculation of the heat flow rate will provide a more realistic result. The value of the computer inner tube heat transfer coefficient is presented in colmn 6 of the first table. The table show a value of heat coefficient ranging between 196 and 486 W/m^2K
- The outer tube heat transfer coefficient used in the source is also constant while INSTED uses a Kern's method to compute the shell side heat transfer coefficient for each case.

♦ Name of data file:
(f) PRELIMD.INP

Test Problem 3

➢ **Problem Statement:**

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

Shell iinternal 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**

Kern's Method

Variable	Hewitt	INSTED	Difference
Δp (Pa)	22,224	22,866.3	
Outer tube heat coeff. (W/m^2K)	977.94	971.444	
Reynolds number	25,224	25,232.7	

Bell-Delaware Method

Variable	Hewitt	INSTED	Difference
Δp (Pa)	4501	3439	
Outer tube heat coeff. (W/m^2K)	769	789.44	
Reynolds number	20,970	21,036	

Stream Analysis Method

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

Explanation of differences in results

- A close comparison with the source can be observed.

♦ Name of data file:
(g) HEWITT.INP

PLATE-FIN HEAT EXCHANGER TEST PROBLEMS

Introduction
The problems that have been used to validate some of the capabilities in INSTED for the analysis of plate-fin 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. You will be expected to have simulated some of these test problems before you attempt to solve more realistic engineering problems.

Test Problem 1

- **Problem Statement:**

A cross flow plate-fin heat exchanger is designed for the gas turbine cycle illustrated in the figure below. It uses heat form the turbine exhaust gases at 460oC to preheat air leaving the compressor at 300oC. The flow rates of gas and air are respectively, 25.0 and 25.4 kg/s, and the dimensions of the heat exchanger are as hown in the figure. What are the temperatures of air and gas leaving the heat exchanger and what are the corresponding pressure drops for the two streams?

- **Source:**

Hewitt, G. F., Shires, G. L., Bott, T. R. Process Heat Transfer, CRC Press. 1998. Pages 321-323.

- **Comments**

- In INSTED, choose the 'outlet temperature' task.
- The hot gas is the hot fluid while air is the cold fluid.
- Enter the thermophysical properties for both the hot and cold fluids as given above in the 'Plate-Fin Fluids' dialog box.
- Enter the fin properties including the pitch, height thickness and fin type in the 'Fin Properties' dialog box.

- **Comparison of INSTED result with source**

In the table below, 'c' refers to cold fluid property or result while 'h' refers to hot fluid result.

Variable	**Hewitt**	**INSTED**	**Difference**
Velocity (h)	33.90	33.935	< 1%
Velocity (c) (m/s)	7.42	7.422	< 1%
Δp (h) (Pa)	6845	5295	
Δp (c) (Pa)	4337	3762	
Friction factor (h)	0.0122	0.010563	< 1%
Friction factor (c)	0.0164	0.012512	
Re(h)	1590	1589.1	< 1%
Re(c)	3130	3128.1	< 1%
Heat trasfer coeff. (h)	129	128.01	< 1%
Heat trasfer coeff. (c)	209	219.70	
U	239	240.04	< 1%
Q (W)	3.10×10^6	3.1364×10^6	
Mean Temp. Diff. (K)	27	26.885	
Temp. out (h) (K)	618	612.42	
Temp. out (c) (K)	690	695.83	

Explanation of differences in results

- Agreement between the two sets of results is apparent. The largest difference is observed in the pressure drop is due to the difference in the method of calculating the friction factor.
- Differences in the heat transfer coefficient results from the fact that INSTEAD considers the heat transfer due to conduction in the plate material. This small contribution is neglected in the problem as solved in the source. This difference also impacts the values obtained for *Q*, *U* and the outlet temperatures.

- ♦ Name of data file:
  (h) SAMP1.INP