Chapter 4

Computer Aided Design of Shell and Tube Heat Exchangers (Incorporating Most Recent Developments)

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1. Introduction

Heat exchangers are practically omnipresent in all process industries, power plants, heat recovery units and the like. The feedstock is to be preheated or the product solution (product gas mixture) is to be cooled down to a specific temperature and for these, heat exchangers become invariable. As a result, efficient and economical design and operation of heat exchangers becomes a fundamental parameter that is crucial to the overall economy of the industry.

Among the industrial heat exchangers, exchangers of shell and tube configuration are one of the most popular ones, particularly for large capacity installations. These exchangers are composed of a tube bundle (consisting of 50 – 1000 or more tubes) enclosed within a large diameter shell. The tubes are held at both ends by drilling them into two tubesheets (fixed tubesheet construction). The *effective length* of each tube (L_e) is the length of the tube between the two tubesheets (those portions of the tubes that are drilled into the tubesheets are excluded). Popular values of L_e used in industrial exchangers are 2.5 m, 3.0 m, 3.5 m, 5.0 m and 6.0 m. Of these, L_e = 5.0 m, 6.0 m are most popular. Tubes are usually either 19 mm OD or 25.4 mm OD and the tube pitch (p_T), which is the center to center distance between adjacent tubes, is commonly maintained at 1.25 to 1.5 times the tube OD (see tube count **Tables 3A** to **3F**).

Table 3A: Tube Count Tables (Database – 3)

19 mm OD tubes on 25.4 mm triangular pitch											
Shell ID mm	**Fixed tubesheet construction (T.E.M.A. L or M) Number of passes**				**Floating head construction (T.E.M.A. P or S) Number of passes**				**U – tube construction (T.E.M.A. U) Number of passes**		
	1	**2**	**4**	**6**	**1**	**2**	**4**	**6**	**2**	**4**	**6**
203.2	42	40	26	24	31	26	16	12	32	24	24
254.0	73	66	52	44	56	48	42	40	52	48	40
304.8	109	102	88	80	88	78	62	68	84	76	74
336.5	136	128	112	102	121	106	94	88	110	100	98
387.3	183	172	146	148	159	148	132	132	152	140	136
438.1	237	228	208	192	208	198	182	180	206	188	182
488.9	295	282	258	248	258	250	228	220	266	248	234
539.7	361	346	318	320	320	314	290	276	330	316	296
590.8	438	416	382	372	400	384	352	336	400	384	356
635.0	507	486	448	440	450	442	400	392	472	440	424
685.8	592	574	536	516	543	530	488	468	554	528	502
736.6	692	668	632	604	645	618	574	556	648	616	588
787.4	796	774	732	708	741	716	666	648	744	716	688
838.2	909	886	836	812	843	826	760	740	852	816	788
889.0	1023	1002	942	920	950	930	878	856	974	932	908
939.8	1155	1124	1058	1032	1070	1052	992	968	1092	1056	1008
990.6	1277	1254	1194	1164	1209	1184	1122	1096	1224	1180	1146
1066.8	1503	1466	1404	1372	1409	1378	1314	1296	1434	1388	1350
1143.0	1726	1690	1622	1588	1635	1608	1536	1504	1652	1604	1560
1219.2	1964	1936	1870	1828	1887	1842	1768	1740	1894	1844	1794
1371.6	2519	2466	2380	2352	2399	2366	2270	2244	2426	2368	2326
1524.0	3095	3058	2954	2928	2981	2940	2832	2800	3006	2944	2884

Table 3B: Tube count tables (Database – 3)

19 mm OD tubes on 25.4 mm square / rotated square pitch							
Shell ID mm	**Floating head construction (T.E.M.A. P or S) Number of passes**				**U – tube construction (T.E.M.A. U) Number of passes**		
	1	**2**	**4**	**6**	**2**	**4**	**6**
203.2	28	26	16	12	28	24	12
254.0	52	48	44	24	52	44	32
304.8	80	76	66	56	78	72	70
336.5	104	90	70	80	96	92	90
387.3	136	128	128	114	136	132	120
438.1	181	174	154	160	176	176	160
488.9	222	220	204	198	224	224	224
539.7	289	272	262	260	284	280	274
590.8	345	332	310	308	348	336	328
635.0	398	386	366	344	408	392	378
685.8	477	456	432	424	480	468	460
736.6	554	532	510	496	562	548	530
787.4	637	624	588	576	648	636	620
838.2	730	712	682	668	748	728	718
889.0	828	812	780	760	848	820	816
939.8	937	918	882	872	952	932	918
990.6	1048	1028	996	972	1056	1044	1020
1066.8	1224	1200	1170	1140	1224	1224	1212
1143.0	1421	1394	1350	1336	1436	1408	1398
1219.2	1628	1598	1548	1536	1640	1628	1602
1371.6	2096	2048	2010	1992	2108	2084	2068
1524.0	2585	2552	2512	2476	2614	2584	2558

Table –3C: Tube count tables (Database – 3)

25.4 mm OD tubes on 31.75 mm square / rotated square pitch							
Shell ID mm	**Floating head construction (T.E.M.A. P or S) Number of passes**				**U – tube construction (T.E.M.A. U) Number of passes**		
	1	**2**	**4**	**6**	**2**	**4**	**6**
203.2	17	12	8	12	14	8	6
254.0	30	30	16	18	30	24	12
304.8	52	48	42	24	44	40	32
336.5	61	56	52	50	60	48	44
387.3	85	78	62	64	80	72	74
438.1	108	108	104	96	104	100	100
488.9	144	136	130	114	132	132	120
539.7	173	166	154	156	172	168	148
590.8	217	208	194	192	212	204	198
635.0	252	240	230	212	244	240	230
685.8	296	280	270	260	290	284	274
736.6	345	336	310	314	340	336	328
787.4	402	390	366	368	400	384	372
838.2	461	452	432	420	456	444	440
889.0	520	514	494	484	518	504	502
939.8	588	572	562	548	584	576	566
990.6	661	640	624	620	664	644	640
1066.8	776	756	738	724	764	748	750
1143.0	900	882	862	844	902	880	862
1219.2	1029	1016	984	972	1028	1008	1004
1371.6	1310	1296	1268	1256	1320	1296	1284
1524.0	1641	1624	1598	1576	1634	1616	1614

Table –3D: Tube count tables (Database – 3)

25.4 mm OD tubes on 31.75 mm triangular pitch											
Shell ID mm	**Fixed tubesheet construction (T.E.M.A. L or M) Number of passes**				**Floating head construction (T.E.M.A. P or S) Number of passes**				**U – tube construction (T.E.M.A. U) Number of passes**		
	1	**2**	**4**	**6**	**1**	**2**	**4**	**6**	**2**	**4**	**6**
203.2	27	26	8	12	18	14	8	12	14	12	6
254.0	42	40	34	24	33	28	16	18	28	24	24
304.8	64	66	52	44	51	48	42	44	52	40	40
336.5	81	74	62	56	73	68	52	44	64	56	52
387.3	106	106	88	92	93	90	78	76	90	80	78
438.1	147	134	124	114	126	122	112	102	122	112	102
488.9	183	176	150	152	159	152	132	136	152	140	136
539.7	226	220	204	186	202	192	182	172	196	180	176
590.8	268	262	236	228	249	238	216	212	242	224	216
635.0	316	302	274	272	291	278	250	240	286	264	246
685.8	375	360	336	324	345	330	298	288	340	320	300
736.6	430	416	390	380	400	388	356	348	400	380	352
787.4	495	482	452	448	459	450	414	400	456	436	414
838.2	579	554	520	504	526	514	484	464	526	504	486
889.0	645	622	586	576	596	584	548	536	596	572	548
939.8	729	712	662	648	672	668	626	608	668	636	614
990.6	808	792	744	732	756	736	704	692	748	728	700
1066.8	947	918	874	868	890	878	834	808	890	856	830
1143.0	1095	1068	1022	1000	1035	1008	966	948	1028	992	972
1219.2	1241	1220	1176	1148	1181	1162	1118	1092	1180	1136	1100
1371.6	1577	1572	1510	1480	1520	1492	1436	1416	1508	1468	1442
1524.0	1964	1940	1882	1832	1884	1858	1800	1764	1886	1840	1794

Table 3E: Tube count tables (Database – 3)

15.875 mm OD tubes on 20.637 mm square / rotated square pitch							
Shell ID mm	Floating head construction (T.E.M.A. P or S) Number of passes				U – tube construction (T.E.M.A. U) Number of passes		
	1	2	4	6	2	4	6
203.2	55	48	34	24	52	40	32
254.0	88	78	62	56	90	80	74
304.8	140	138	112	100	140	128	108
336.5	178	172	146	136	180	164	148
387.3	245	232	208	192	246	232	216
438.1	320	308	274	260	330	312	292
488.9	405	392	352	336	420	388	368
539.7	502	484	442	424	510	488	460
590.8	610	584	536	508	626	596	562
635.0	700	676	618	600	728	692	644
685.8	843	812	742	716	856	816	780
736.6	970	942	868	840	998	956	920
787.4	1127	1096	1014	984	1148	1108	1060
838.2	1288	1250	1172	1148	1318	1268	1222
889.0	1479	1438	1330	1308	1492	1436	1388
939.8	1647	1604	1520	1480	1684	1620	1568
990.6	1840	1794	1700	1664	1882	1816	1754
1066.8	2157	2112	2004	1968	2196	2136	2068
1143.0	2511	2458	2326	2288	2530	2464	2402
1219.2	2856	2808	2686	2656	2908	2832	2764
1371.6	3656	3600	3462	3404	3712	3624	3556
1524.0	4538	4472	4310	4256	4608	4508	4426

Table 3F: Tube count tables (Database – 3)

19 mm OD tubes on 23.8 mm triangular pitch											
Shell ID mm	**Fixed tubesheet construction (T.E.M.A. L or M) Number of passes**				**Floating head construction (T.E.M.A. P or S) Number of passes**				**U – tube construction (T.E.M.A. U) Number of passes**		
	1	**2**	**4**	**6**	**1**	**2**	**4**	**6**	**2**	**4**	**6**
203.2	64	48	34	24	34	32	16	18	32	24	24
254.0	85	72	52	50	60	62	52	44	64	52	52
304.8	122	114	94	96	109	98	78	68	98	88	78
336.5	151	142	124	112	126	120	106	100	126	116	108
387.3	204	192	166	168	183	168	146	136	180	160	148
438.1	264	254	228	220	237	228	202	192	238	224	204
488.9	332	326	290	280	297	286	258	248	298	280	262
539.7	417	396	364	348	372	356	324	316	370	352	334
590.8	495	478	430	420	450	430	392	376	456	428	408
635.0	579	554	512	488	518	498	456	444	534	500	474
685.8	676	648	602	584	618	602	548	532	628	600	570
736.6	785	762	704	688	729	708	650	624	736	696	668
787.4	909	878	814	792	843	812	744	732	846	812	780
838.2	1035	1002	944	920	962	934	868	840	978	928	904
889.0	1164	1132	1062	1036	1090	1064	990	972	1100	1060	1008
939.8	1304	1270	1200	1168	1233	1196	1132	1100	1238	1200	1152
990.6	1460	1422	1338	1320	1365	1346	1266	1244	1390	1336	1290
1066.8	1703	1664	1578	1552	1611	1580	1498	1464	1632	1568	1524
1143.0	1960	1918	1830	1800	1875	1834	1736	1708	1882	1820	1770
1219.2	2242	2196	2106	2060	2132	2100	1998	1964	2152	2092	2044
1371.6	2861	2804	2682	2660	2730	2684	2574	2536	2748	2680	2628
1524.0	3527	3476	3360	3300	3395	3346	3228	3196	3420	3340	3286

1.1. Tube Sheet Layout

Tubes may be laid on the tubesheet using a square pitch arrangement, in which the tubes are aligned in line (**Figure 1**) or using a rotated square or triangular layout, in which cases a staggered tube arrangement is employed (Figures 2, 3). When the arrangement of tubes is staggered, the flow of shellside fluid (which flows over the tubes) becomes more tortuous, there shall be more intimate contacting between fluid elements and consequently, the shellside heat transfer coefficient gets enhanced. It has been observed that the magnitude of shellside heat transfer coefficient (h_0) attained with a triangular pitch layout is often 1.25 to 1.30 times that obtained with a square pitch layout. However, increased totuosity of flow path causes increased resistance to flow of shellside fluid and this would demand higher pumping power requirement and higher operating cost.

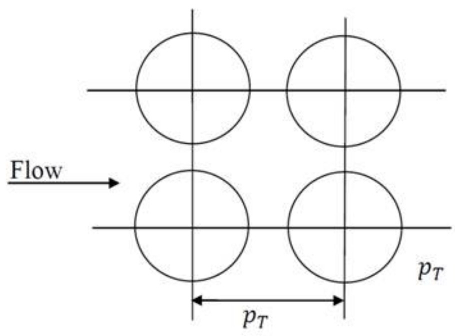

Figure 1: Square pitch layout

A *square pitch arrangement*, therefore, though provides the lowest shellside heat transfer coefficient among the three, causes the lowest pressure drop as well and thereby brings down the operating cost. In an arrangement like this, the tubes remain easily accessible for external cleaning. Thus, if large scale fouling is anticipated on the outer surface of tubes, a square pitch arrangement is to be preferred. The number of tubes that can be accommodated within a given shell diameter is, however, lower in this case as compared to a triangular pitch layout (see tube count tables 3A to 3F in the Appendix). From Figure [1], it can be seen that for a layout like this,

$$p_n = p_P = p_T \tag{1}$$

Where

$p_n =$ tube pitch normal to flow

$p_P =$ tube pitch parallel to flow

In a *rotated square layout* **(Figure 2)**, the number of tubes within a shell diameter does not differ much from that in a square pitch layout, but since the tubes are staggered, the layout provides larger shellside heat transfer coefficient. The shellside pressure drop shall nevertheless be higher (demanding higher pumping cost) and the tubes shall be less accessible for external cleaning. For this type of layout, p_P and p_n shall be equal in magnitude, but not equal to p_T. From **Figure (2),** based on simple geometry,

$$p_n = p_P = (p_T / \sqrt{2}) = (0.707\, p_T) \tag{2}$$

A *triangular pitch arrangement* **(Figure 3),** as stated earlier, contributes the largest shellside heat transfer coefficient. This layout also accommodates the largest number of tubes within a given shell diameter and thus provides large heat transfer surface to the exchanger. On the other hand, the shellside pressure drop shall be of larger magnitude and this leads to increased operating cost. The accessibility of tubes for external cleaning when fouled shall also be lower. From **figure (3)**, it can be easily deduced that for a triangular pitch layout like this, p_P and p_n are related to p_T as

$$p_P = 0.866\, p_T \tag{3}$$

$$p_n = 0.5\, p_T \tag{4}$$

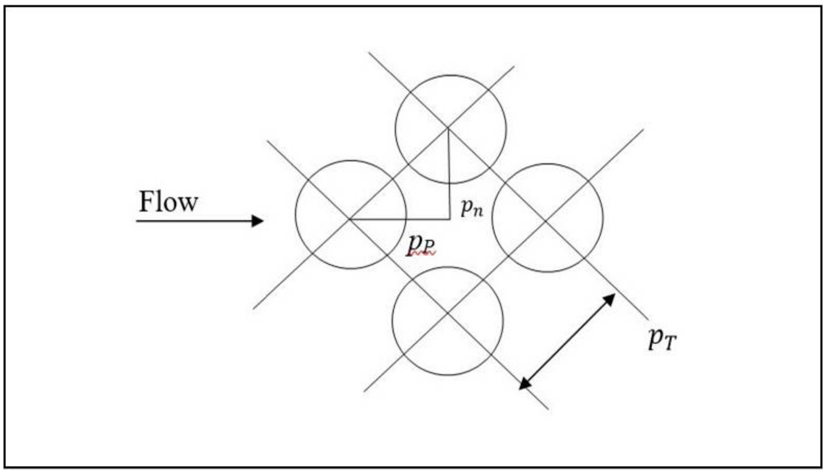

Figure 2: Rotated squre pitch layout

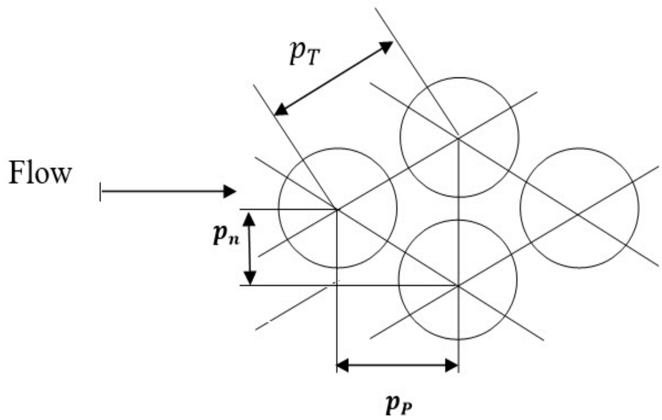

Figure 3: Triangular pitch layout

The tube sheet layout must be thus selected keeping all the above pros and cons in mind.

1.2. Tube sheet Construction

Fixed tubesheet construction **(Figure 4)** is the simplest and cheapest mode of construction for these heat exchangers. Obviously, it is the first choice of the manufacturers. However, this type of construction becomes unreliable when the temperature difference handled by the exchanger is too large. At high working temperatures, the tubes tend to expand and this could lead to fracture (or cracking) of tubes. In such cases, an exchanger with a floating head at one end and a fixed tubesheet at the other end could be used **(Figure 5)**. In such a *pull through floating head construction*, the tubes are free to expand and this differential expansion between the shell and the tube bundle shall not cause damage to the exchanger. The tube bundle is removable for inspection, repair and replacement. However, this type of construction is much more expensive than the conventional fixed tubesheet construction and is therefore recommended only when large scale differential expansion of tubes is anticipated or when frequent mechanical cleaning of tube surfaces due to fouling is imperative.

A still further alternative is to use the *U – tube construction*. In U – tube exchangers, each tube is bent in the shape of the English letter U (U – shaped tube) and these U – tubes are enclosed in the shell. The tubes are supported only at one end using a fixed tube sheet, the U – ends of the tubes remain free or floating. However, the total number of tubes that can be accommodated within a given shell diameter shall be less in this case since tubes cannot be bent to form a sharp U (they tend to crack).

1.3. Baffles and Baffle Pitch

Baffles are installed in the shells of practically all shell and tube heat exchangers. These are mostly circular plates with a number of holes punched (drilled) on them, through which the tubes pass. But, each baffle does not occupy the entire cross – section of the shell. 25% cut segmental baffles are most popular, which are circular discs with 25 per cent of surface being chopped off. The height of the baffle thus becomes three – fourth (75 %) of the shell diameter. The distance between the bottom tip of the baffle and the shell wall is called the *baffle cut* (B_C). For 25% cut segmental baffles, $B_C = (D_S)/4$.

Baffles are seldom welded to the shell wall. They are held in position by means of tie rods and spacers. The spacing between two adjacent baffles is called the *baffle spacing or baffle pitch* (B_S) and is an important design parameter. If L_e is the effective length of each tube, then the number of baffles (N_b) shall be

$$N_b = (L_e / B_S) - 1 \tag{5}$$

This is based on the assumption that a uniform baffle spacing or baffle pitch has been used through the length of the exchanger. Often, a larger baffle spacing may have to be used at the inlet and also at the outlet to accommodate inlet and outlet shellside nozzles. If B_{Si} is the baffle spacing employed at the inlet and B_{So} that at the outlet, then

$$N_b = [(L_e - B_{Si} - B_{So})/B_S] + 1 \qquad (6)$$

No doubt, it is most preferable to use a uniform baffle spacing throughout, as far as practicable. The shellside fluid flows over the tubes, between two baffles. This flow space between two adjacent baffles is called the *crossflow section*. The shellside fluid thus flows up or down each crossflow section and thereby moves from one end of the exchanger to the other see (**Figure 4**). The smaller the baffle pitch (B_S) used (and thereby the larger the number of baffles used), the smaller will be the flow area between baffles and the larger the flow velocity of shellside fluid. Consequently, the shellside Reynolds number (Re_S) shall be of higher magnitude and this enhances the shellside heat transfer coefficient (h_o).

It is due to the presence of baffles the shellside fluid tends to execute more and more crossflow (between baffles) and the heat transfer coefficient in crossflow is much higher than that in countercurrent flow or co-current flow (parallel flow).

Baffles also act as support plates for tubes and help in minimizing tube vibrations. Tubes tend to vibrate when shellside fluid flows over them. If these vibrations are of large amplitude, then the tubes tend to undergo fracture or fatigue failure. By supporting tubes between baffles, chances of such fatigue failure of tubes are minimized. To note that the maximum unsupported length of each tube is equal to the baffle pitch (B_S).

However, it is not all smiles with respect to the use of baffles. When the baffle pitch or baffle spacing chosen is small (or the number of baffles installed is large), the flow velocity of shellside fluid increases (as stated earlier) and this leads to increase in the shellside pressure drop as well. It is to be noted that the shellside pressure drop is proportional to the square of the shellside fluid velocity (see equations discussed subsequently in this Chapter) and consequently, a small increase in flow velocity could cause a substantial increase in the pressure drop penalty. A large increase in the shellside pressure drop means large pumping power requirement and increased operating cost.

The baffle spacing (B_S) and the number of baffles to be installed must be, therefore, judiciously chosen. The number of baffles must be sufficiently large (the baffle spacing sufficiently small) so as to maintain the shellside heat transfer coefficient sufficiently large and also to ensure adequate support to tubes, but it should also be not too large such that the shellside pressure drop penalty does not exceed the maximum permissible limit. TEMA (Tubular Exchangers Manufacturers Association) specifies the following criterion for the selection of baffle spacing / baffle pitch in a shell and tube heat exchanger:

$$(D_S/5) \leq B_S \leq (D_S) \tag{7}$$

In other words, the baffle spacing must be so chosen that it never falls below 20% (one – fifth) of the shell diameter (D_S) and should also never exceed the shell diameter itself. Thus

$$B_S \text{ (min)} = (D_S/5) \tag{8}$$

$$B_S \text{ (max)} = (D_S) \tag{9}$$

During the design of the exchanger, it is common practice to choose the minimum baffle spacing at the outset and subsequently increase it if the shellside pressure drop is found to exceed the maximum permissible limit.

There are occasions where tubes are avoided in the baffle window. This is called the *no – tubes – in – baffle window* construction. The *baffle window* is the space between two *alternate* baffles and adjacent to the shell wall see **(Figures 4 and 5)**. It is therefore obvious that the maximum unsupported length of each tube in the baffle window is ($2B_S$) and not one baffle pitch as is the case with other tubes in the crossflow section. Consequently, the tubes in the baffle window tend vibrate at larger amplitudes when the shellside fluid flows over them and the chances of fatigue failure of these tubes become larger. However, by avoiding tubes in the baffle window, the effective number of tubes in the exchanger gets reduced, thereby bringing down the heat transfer surface available. The *no – tubes – in – baffle window* construction must be therefore used only when the shellside mass velocity is too large and large scale tube vibrations are anticipated in the baffle window.

1.4. Multipass Construction

Most of the industrial shell and tube heat exchangers employ multipass construction. For example, a 1 – 2 exchanger (in which the number of shellside passes = n_S = 1 and the number of tubeside passes = n_t = 2) is what is sketched in **Figure (4).** This uses one pass partition at one end of the exchanger. The tubeside fluid enters at this end, flows through all the tubes above the pass partition (there could be 50 to 500 or more tubes in this section) and after reaching the other end of the exchanger, flows back through the remaining tubes below the pass partition and is discharged from the end – 1 itself. Since the fluid traverses the length of the exchanger twice, the number of tubeside passes becomes equal to 2 (n_t = 2). The shellside fluid, on the other hand, enters at one end of the exchanger, flows over tubes in each crossflow section and is discharged from the other end, thereby constituting only one pass (n_S = 1).

In a similar way, a 2 – 4 heat exchanger (n_S = 2, n_t = 4) is what is sketched in figure (5). There are three tubeside pass partitions in the exchanger, two at one end (where the tubeside fluid enters) and one at the other end. The tubeside fluid is thus made to traverse the length of the exchanger four times (each time through one – fourth of the total number of tubes), thereby executing four tubeside passes ($n_t = 4$). On the shellside, there is one longitudinal pass partition (along the axis of the shell) which forces the shellside fluid to execute two shellside passes ($n_S = 2$).

Multipass constructions provide higher heat transfer coefficients and thereby help in attaining improved heat transfer effectiveness for the exchanger. However, such exchangers are more expensive to fabricate, install and maintain. Both the tubeside and shellside pressure drop penalties shall be higher. There shall be additional pressure drop due to flow reversal.

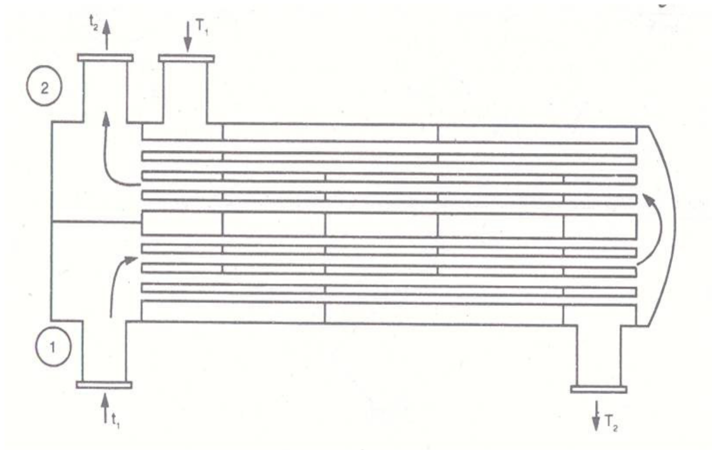

Figure 4: Schematic of a 1 – 2 shell and tube heat exchanger (with fixed tube sheets)

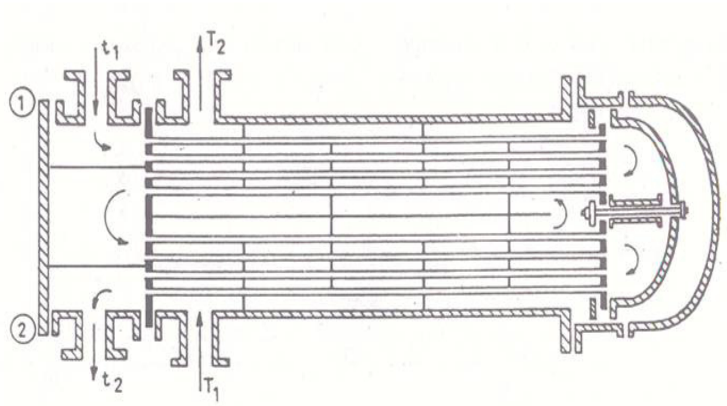

Figure 5: Schematic of a 2 – 4 shell and tube heat exchanger (with floating head construction)

A construction with larger number of passes must be therefore employed only at high capacities, when the amount of fluid to be handled (the amount of fluid being heated or cooled) is large. It is usual practice to start with an exchanger with one shellside pass ($n_S = 1$) and two or more tubeside passes ($n_t = 2, 4, 6$ etc) and if it is found unsuitable for the purpose, then go for a 2 – 4 construction or an exchanger with two shellside passes ($n_S = 2$) and four or more tubeside passes ($n_t = 4, 8, 12$ etc). During the estimation of the required heat transfer surface (discussed later under the CAD package in Section – 2), we do get signals regarding the suitability of the pass arrangement chosen. For example, during the computation of the heat transfer surface using F_T method, if the computed value of F_T factor happens to be negative or indeterminate (logarithm of a negative quantity appears in the expression), then it means that the chosen pass arrangement is non-operable and an alternate pass arrangement is to be selected. In the same way, under ϵ - NTU Method, if the computed value of $NTU\ (max)$ is found to be negative or indeterminate, then again it means that the pass arrangement considered is unsuitable.

In high capacity installations, it is also common practice to use exchangers in series or in parallel.

1.5. Selection of Tube Side and Shell Side Fluids

Among the cold and hot fluids, the question of which one is to be placed on the tubeside and which one on the shellside is mostly dictated by economic considerations. A few thumb rules could be useful here. For example, the more corrosive or more fouling fluid is recommended to be used on the tubeside, since cleaning and replacement of the large diameter shell shall be more laborious and expensive. When the fluids are pumped at high pressure (mainly in the case of gases), the high pressure fluid be used on the tubeside to avoid an expensive, thick-walled, high pressure shell.

When there is large difference between the flow rates of the two fluids, the larger stream be placed on the tubeside and the smaller stream on the shellside. This is because fully developed turbulent flow can be achieved on the shellside at much lower Reynolds number (at $Re_S \geq 3000$), while Reynolds numbers exceeding 10000 are required on the tubeside for maintaining fully developed turbulent flow. However, in such cases, special care should be taken to ensure that the pressure drop penalty on the tubeside is well within the maximum permissible limit prescribed.

The CAD package discussed may very well be re-executed considering both alternatives and based on the results, the choice could be made.

2. CAD Preliminaries

Design of heat exchangers involves, broadly speaking, two steps:

(a) Estimation of the heat transfer surface requirement of the exchanger,

(b) Estimation of the pressure drop penalty in each fluid stream (in the cold fluid stream and in the hot fluid stream).

For a well – designed heat exchanger, the heat transfer surface requirement must be reasonably low. In other words, the exchanger must be able to perform the duty (must heat the cold fluid to the specified temperature at the specified rate or cool the hot fluid to the desired temperature at the desired rate) with a reasonably low heat transfer surface requirement.

By pressure drop penalty, we mean the pressure difference driving force required for pumping the cold fluid / hot fluid at the required flow rate through the exchanger. The operating cost (pumping cost of fluids) of the exchanger is thus decided by the pressure drop penalty and for an economic operation of the exchanger, this penalty must be reasonably low.

On occasions, the above two conditions could contradict against each other and we would have to make a compromise between the two. For example, to restrict the operating cost (to maintain the pressure drop penalty in both fluid streams below the maximum permissible limit), we may have to accommodate a larger heat transfer surface. Conversely, to retain the heat transfer surface requirement of the exchanger at a reasonably low value, a larger pressure drop penalty and thereby a larger operating cost may have to be tolerated.

Heat exchanger problems could be a sizing problem or a rating problem. In a *sizing problem*, we design a heat exchanger for a specific duty, while in a *rating problem*, the heat exchanger is available and we estimate whether the available heat exchanger is suitable for performing the given duty. The design procedures are similar, though the sizing problem demands an iterative (trial and error) procedure, while in a rating problem, the computations are relatively straightforward.

2.1. Cad Package for Sizing Problem

Let us first consider a sizing problem. As stated above, here we design a shell and tube heat exchanger for a specific purpose, such as for heating a cold fluid from temperature t_1 to temperature t_2 at the rate of $\dot{m}$ kg / hr using a hot fluid flowing at $\dot{m}_S$ kg/hr or vice versa. The step by step procedure is described below. This entire procedure has also been illustrated in all details in the CAD flow sheet of this section **(Figures 7A to 7p).**

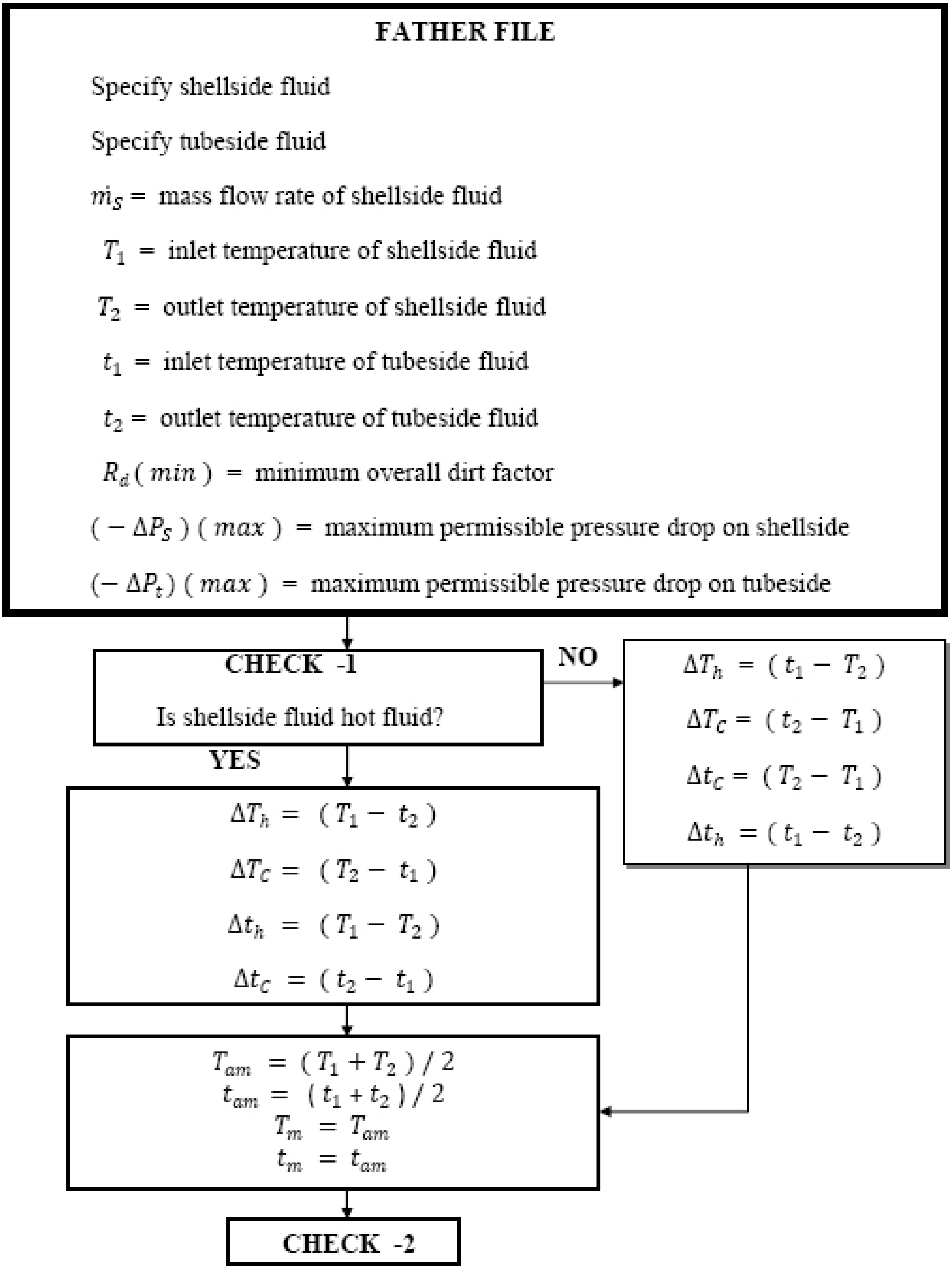

Figure 7A: Computer Aided Design of Shell and Tube Heat Exchangers (Sizing Problem)

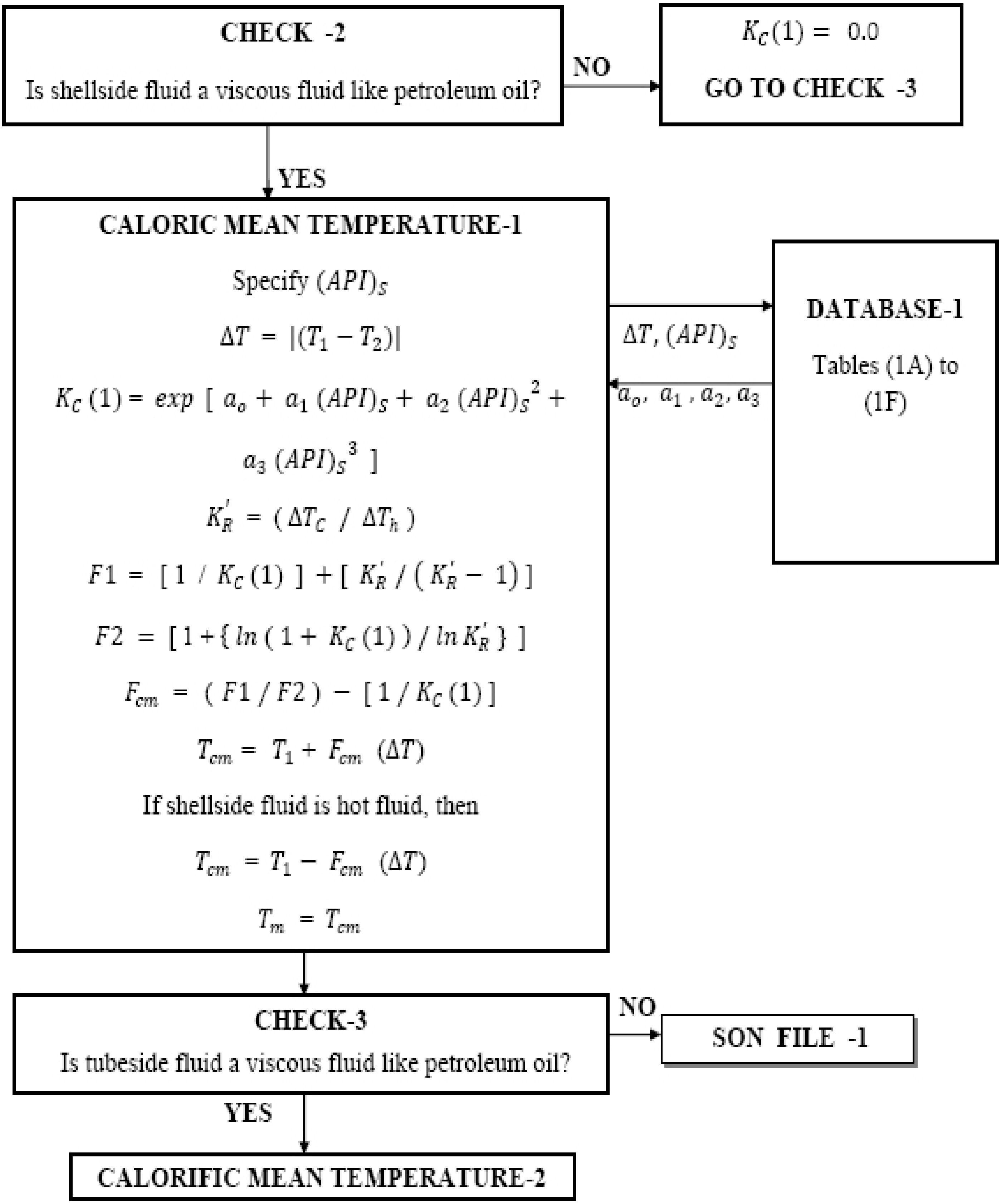

Figure 7B: CAD of Shell and Tube Heat Exchangers (Sizing Problem)-continued

CALORIC MEAN TEMPERATURE-2

Specify $(API)_t$

$$\Delta t = |(t_1 - t_2)|$$

$$\Delta T = |(T_1 - T_2)|$$

$$K_C(2) = exp\ [a_o + a_1\ (API)_t + a_2\ (API)_t^{\ 2} + a_3\ (API)_t^{\ 3}\ ]$$

K_C = Larger of $K_C(1)$ and $K_C(2)$

$$K'_R = (\Delta T_C / \Delta T_h)$$

$$F1 = [\ 1\ /\ K_C\ ] + [\ K'_R / (K'_R - 1)]$$

$$F2 = [\ 1 + \{\ ln(1 + K_C) / ln\ K'_R\}]$$

$$F_{cm} = (\ F1 / F2\ ) - [\ 1\ /\ K_C\ ]$$

$$t_{cm} = t_1 + F_{cm}\ (\Delta t)$$

$$T_{cm} = T_1 + F_{cm}\ (\Delta T)$$

If tubeside fluid is hot fluid, then

$$t_{cm} = t_1 - F_{cm}\ \ (\Delta t)$$

If shellside fluid is hot fluid, then

$$T_{cm} = T_1 - Fcm\ \ (\Delta T)$$

$$t_m = t_{cm},\ T_m = T_{cm}$$

If $K_C(1) = 0.0,\ T_m = T_{am}$

$\Delta t, (API)_t$ →

DATABASE-1

Tables (1A) to (1F)

← a_o , a_1 , a_2, a_3

SON FILE - 1

Specify the correlation constants for property value estimation:

$d_1,\ d_2,\ d_3$: correlation constants for density of tubeside fluid

$q_1,\ q_2,\ q_3$: correlation constants for specific heat of tubeside fluid

$k_1,\ k_2,\ k_3$: correlation constants for thermal conductivity of tubeside fluid

$m_1,\ m_2$: correlation constants for viscosity of tubeside fluid

SON FILE – 1 (CONTINUED)

Figure 7C: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

SON FILE – 1 (CONTINUED)

d_4, d_5, d_6 : correlation constants for density of shellside fluid

q_4, q_5, q_6 : correlation constants for specific heat of shellside fluid

k_4, k_5, k_6 : correlation constants for thermal conductivity of shellside fluid

m_3, m_4 : correlation constants for viscosity of shellside fluid

PROPERTY VALUES

$$\rho_f = d_1 + d_2 (d_3 - t_m)$$

$$C_P = q_1 + q_2 (q_3 - t_m)$$

$$k_f = k_1 + k_2 (k_3 - t_m)$$

$$\mu_f = exp [- m_1 - (m_2 / t_m)]$$

$$\rho_{fS} = d_4 + d_5 (d_6 - T_m)$$

$$C_{PS} = q_4 + q_5 (q_6 - T_m)$$

$$k_{fS} = k_4 + k_5 (k_6 - T_m)$$

$$\mu_{fS} = exp [- m_3 - (m_4 / T_m)]$$

HEAT BALANCE

$$\Delta T = |(T_1 - T_2)|$$

$$\Delta t = |(t_1 - t_2)|$$

$$Q = \dot{m}_S C_{PS} \Delta T$$

$$\dot{m} = Q / (C_P \Delta t)$$

Select overall design heat transfer coefficient U_D from specified range given in Database – 2 .

Let $U_i = U_D$

HEAT TRANSFER AREA

Figure 7D: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

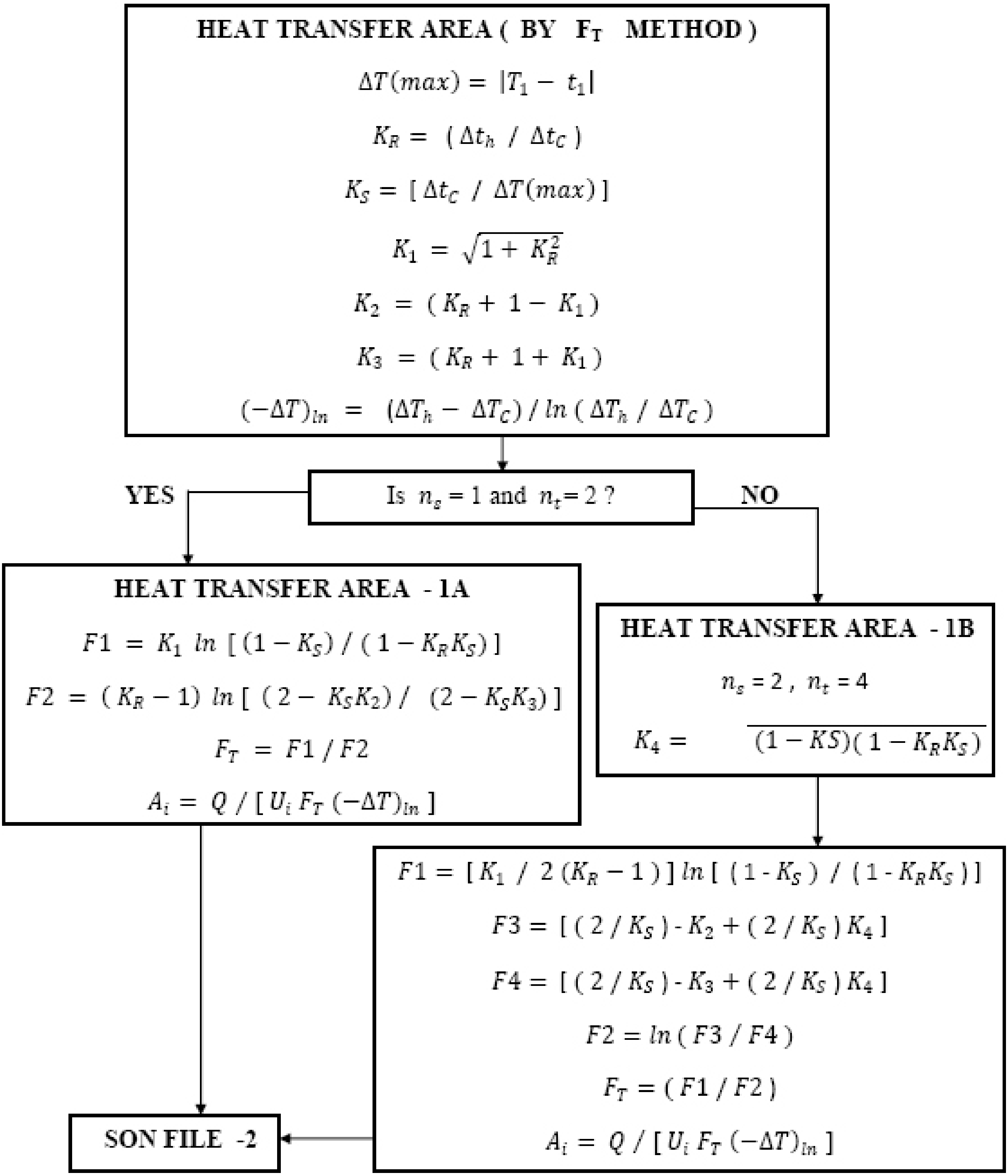

Figure 7E: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

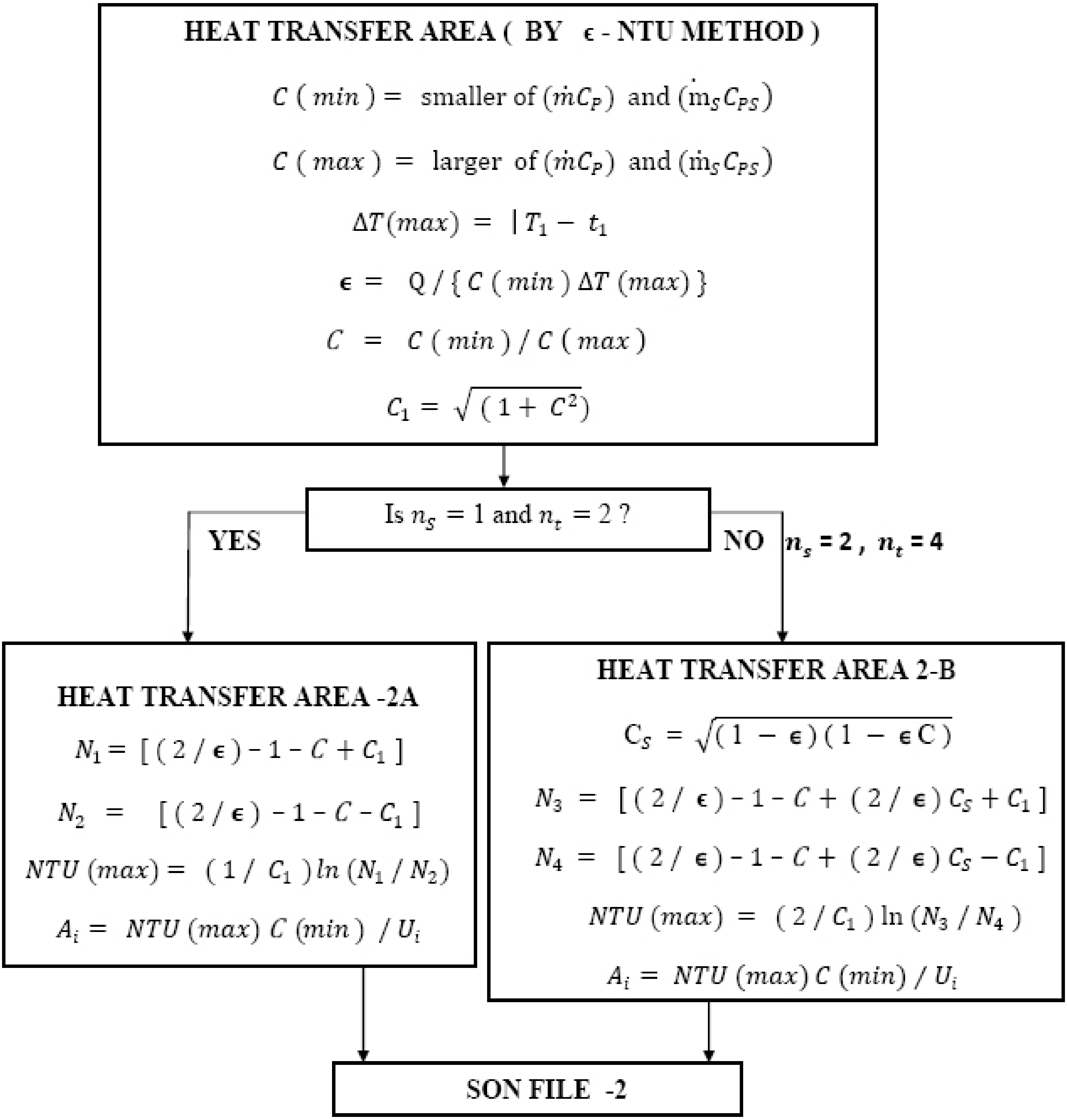

Figure 7F: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

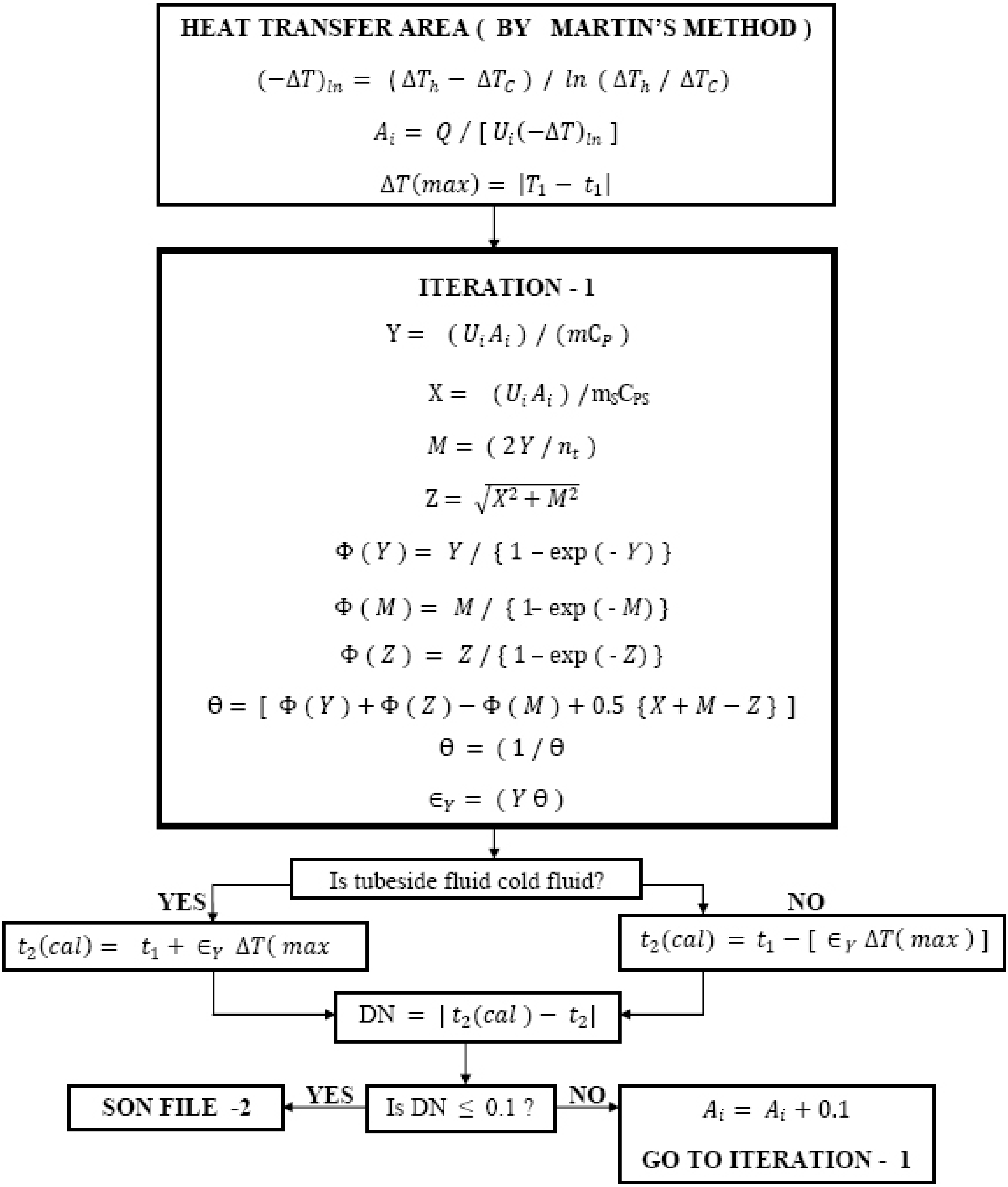

Figure 7G: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

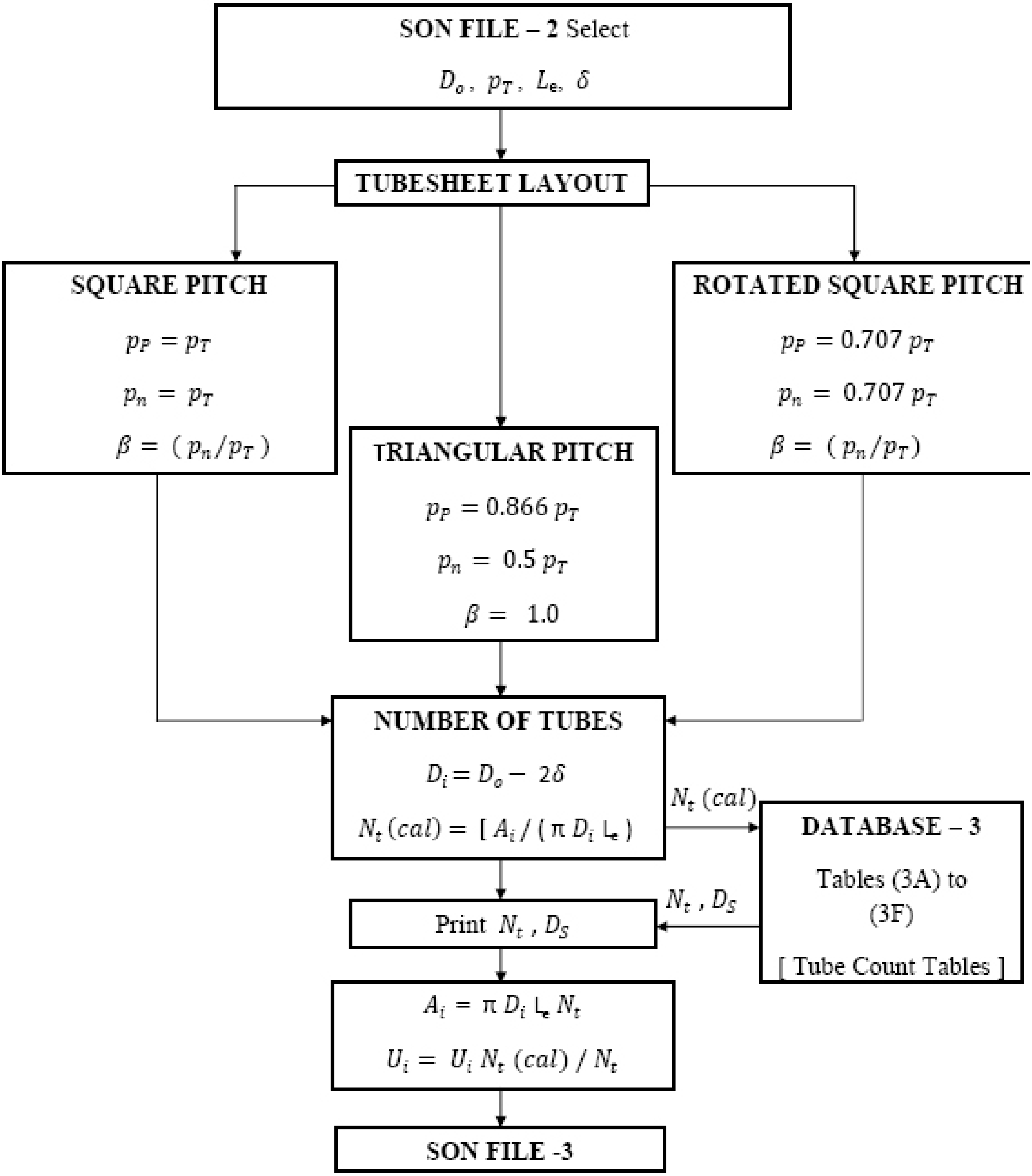

Figure 7H: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

SON FILE – 3

Select , B_S. For example, $B_S = B_S\ (\text{min}) = (D_S / 5)$

Specify , B_{Si} , B_{So}

Select Baffle cut (B_C)

For 25% cut segmental baffles , $B_C = (D_S / 4)$

Specify N_{SS} (Number of sealing strips installed per cross flow section)

↓

TUBESIDE COEFFICIENT

$$a_t = (\pi D_i^2 / 4)(N_t / n_t)$$

$$G_t = (\dot{m} / a_t)$$

$$Re_t = (D_i G_t) / \mu_f$$

$$Pr = (\mu_f C_P) / k_f$$

↓

Let $\varphi = (\mu_f / \mu_{fw})^{0.14} = 1.0$

↓

CHECK - 4

Figure 7I: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

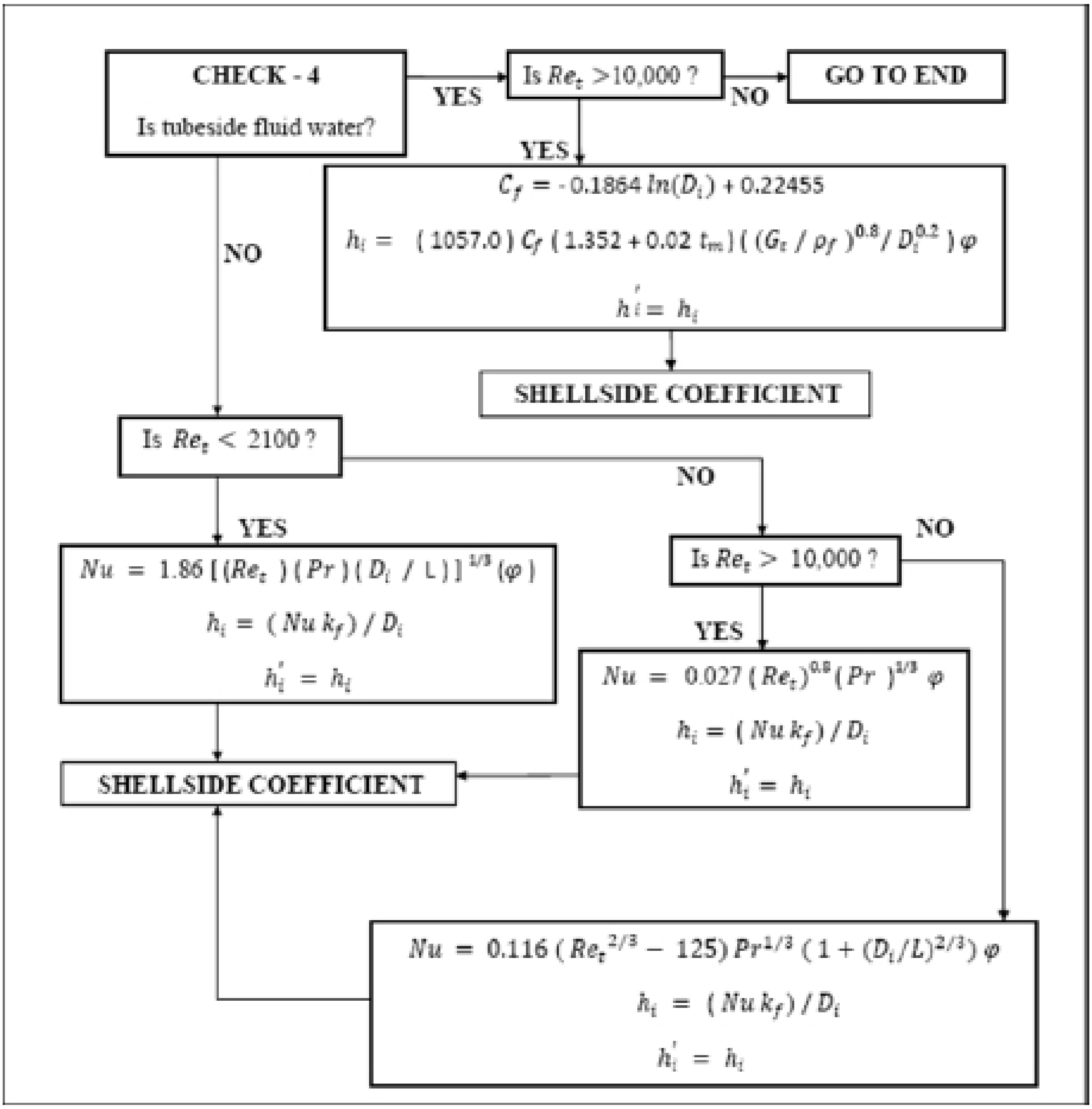

Figure 7J: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

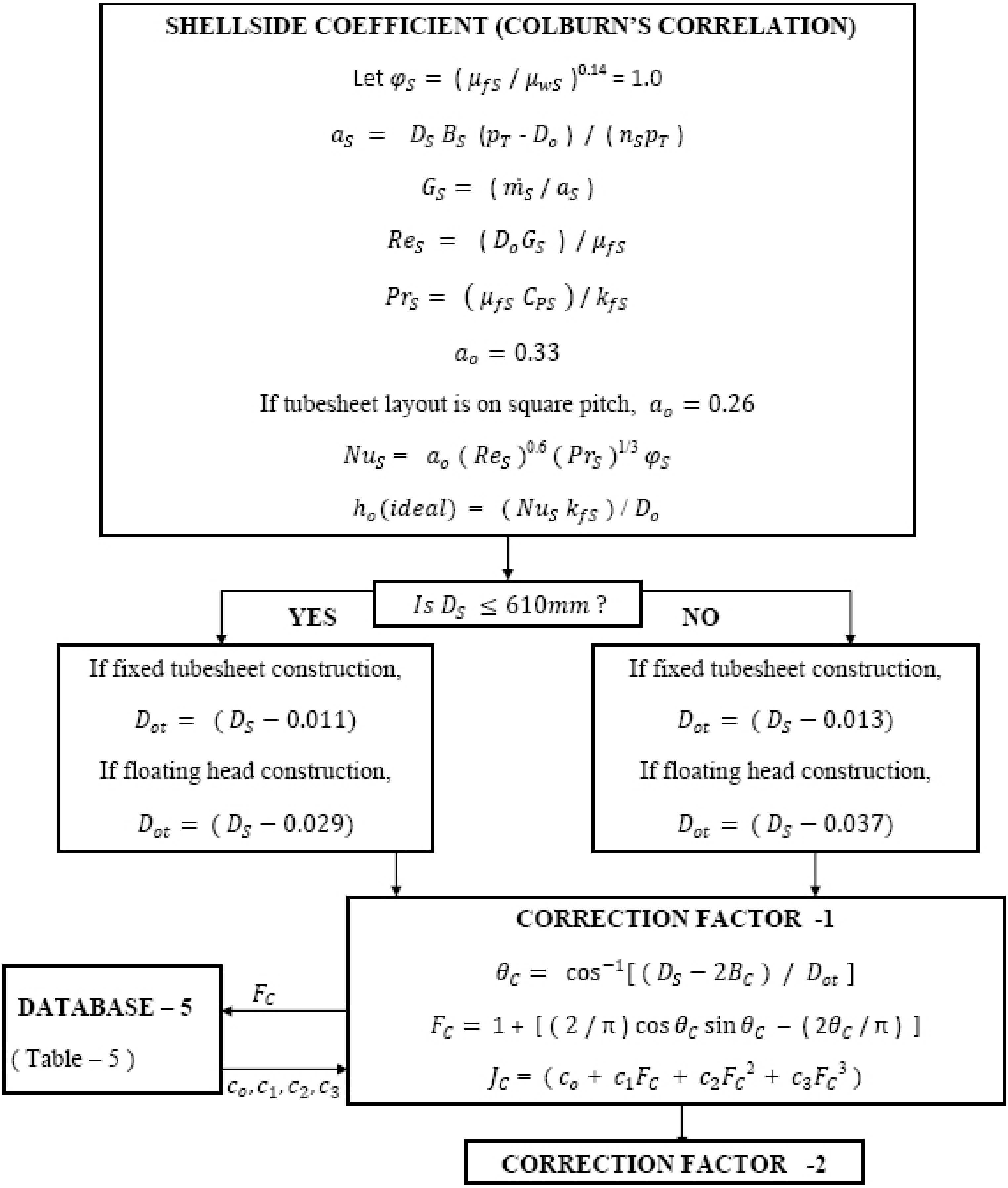

Figure 7K: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

CORRECTION FACTOR – 2

If $(2 B_S) < 910$ mm, $\delta_{tb} = 0.8$ mm

If $(2 B_S) > 910$ mm, $\delta_{tb} = 0.4$ mm

$$A_{tb} = \pi D_o \, \delta_{tb} \, N_t \, (1 + F_C) \; / 2$$

$$A_{Sb} = (\pi \, D_S \, \delta_{Sb} \; / 2) \, [1 - (\theta \, / \, 2\pi)]$$

$$A_m = B_S \, [D_S - D_{ot} + \{ (D_{ot} - D_o) (p_T - D_o) / \beta \, p_T \}]$$

$$S_r = [(A_{tb} + A_{Sb}) / A_m$$

$$S_S = [A_{Sb} / (A_{tb} + A_{Sb})]$$

$$J_l = [a_o + a_1 S_r + a_2 S_r^{\,2} + a_3 S_r^{\,3}]$$

D_S → **DATABASE -6** (Table – 6) → δ_{Sb}

S_S, S_r → **DATABASE -7** (Table – 7) For, $S_{S1} \leq S_S \leq S_{S2}$ → a_o , a_1, a_2, a_3

CORRECTION FACTOR – 3

$$F_{bP} = (D_S - D_{ot}) B_S \; / A_m$$

$$G_m = (\dot{m}_S \; / A_m)$$

$$Re_{Sm} = (D_o G_m \, / \, \mu_{fS})$$

$$N_C = (D_S - 2B_C) \, / \, p_P$$

$$J_b = [exp (- m_1 F_{bP})]$$

Re_{Sm}, (N_{SS} / N_C) → **DATABASE – 8** (Table – 8) → m_1

CORRECTION FACTOR – 4

$$N_b = [(L_e - B_{Si} - B_{So}) \; / \; B_S] + 1$$

$$F1 = (N_b - 1) + (B_{Si} / B_S)^{0.4} + (B_{So} / B_S)^{0.4}$$

$$F2 = (N_b - 1) + (B_{Si} / B_S) + (B_{So} / B_S)$$

$$J_S = (F1 / F2)$$

$$h_o = h_o(ideal) \, (J_C \, J_l \, J_b \, J_S)$$

$$h_o' = h_o$$

CHECK -5

Figure 7L: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

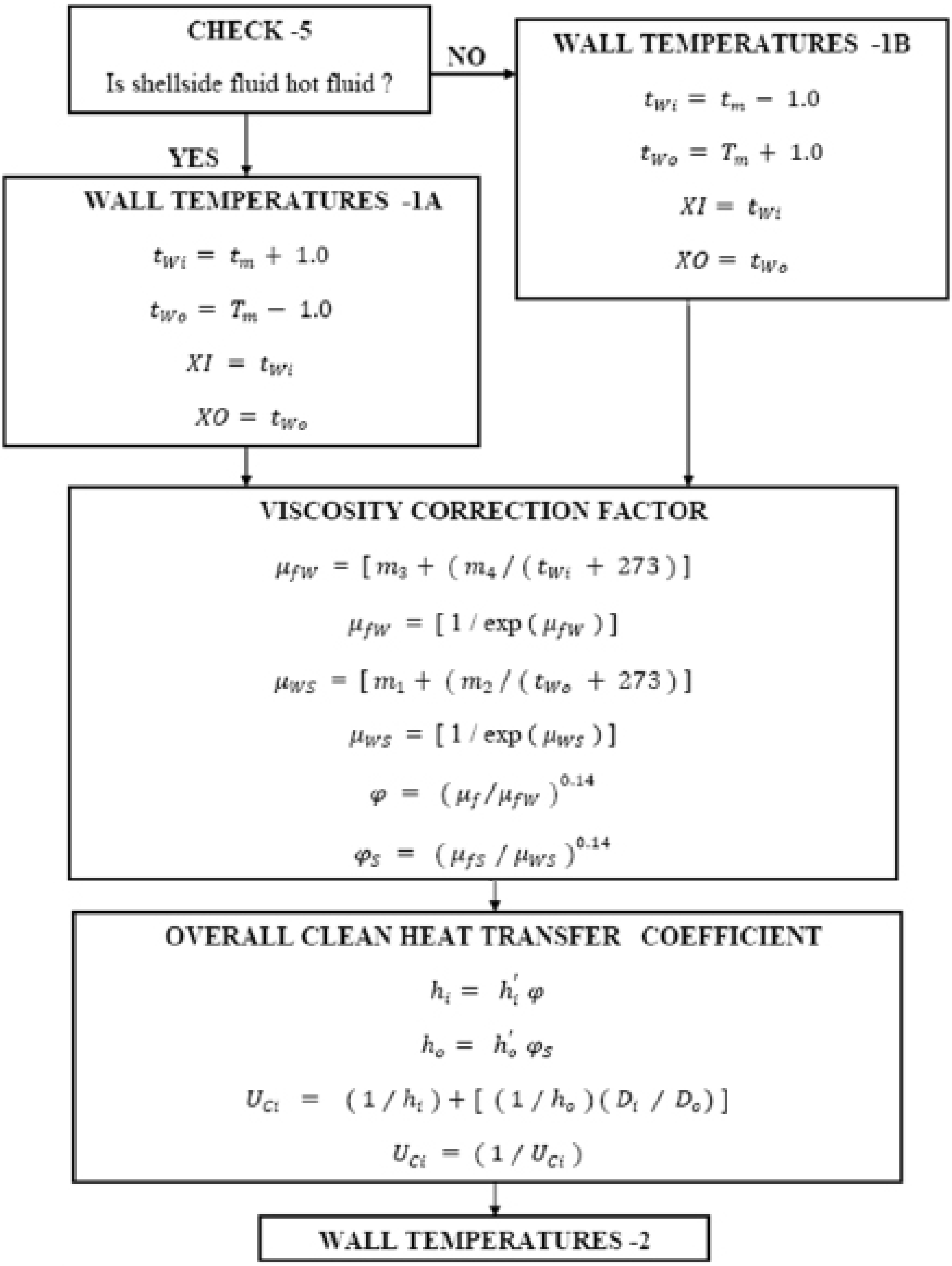

Figure 7M: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

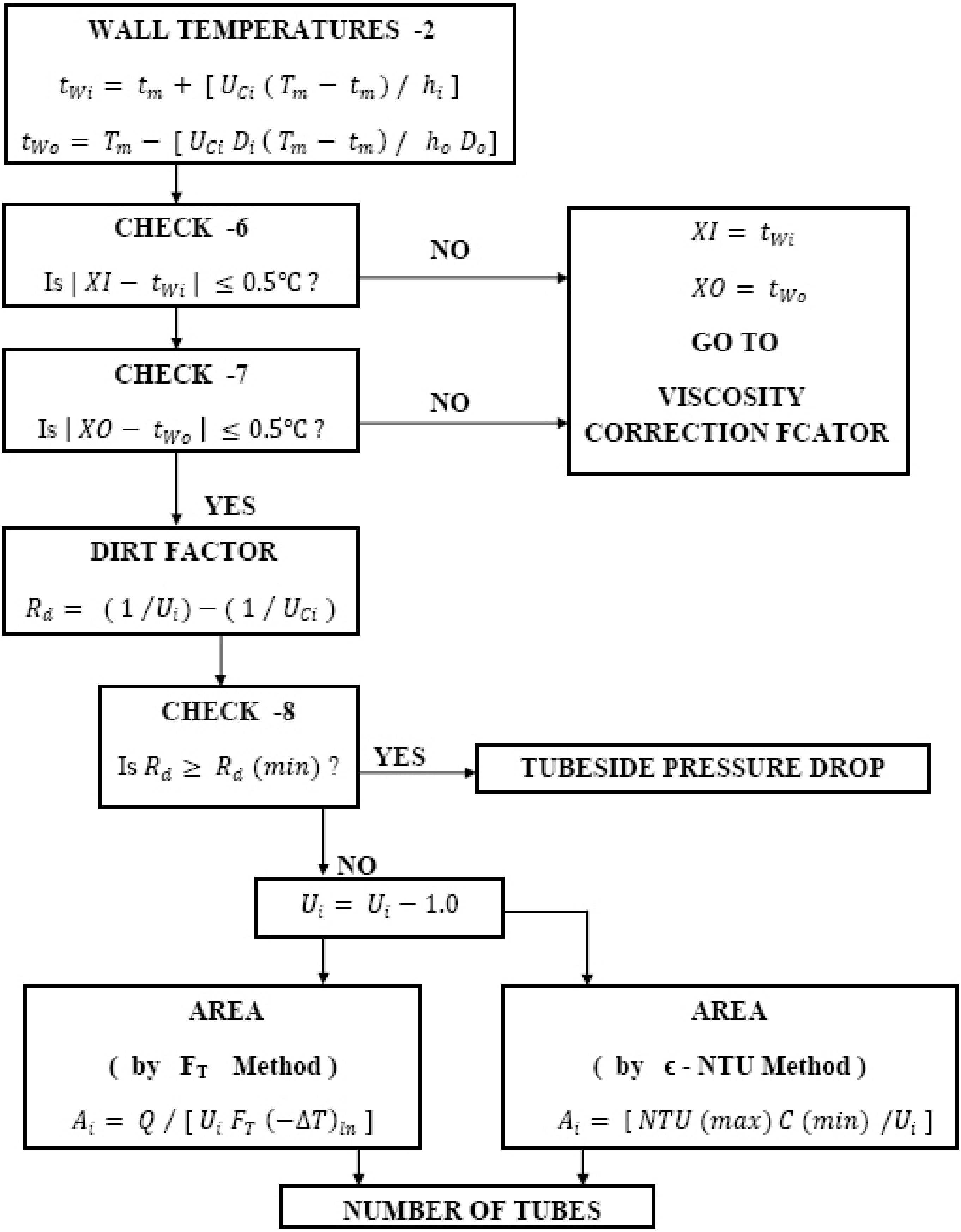

Figure 7N: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

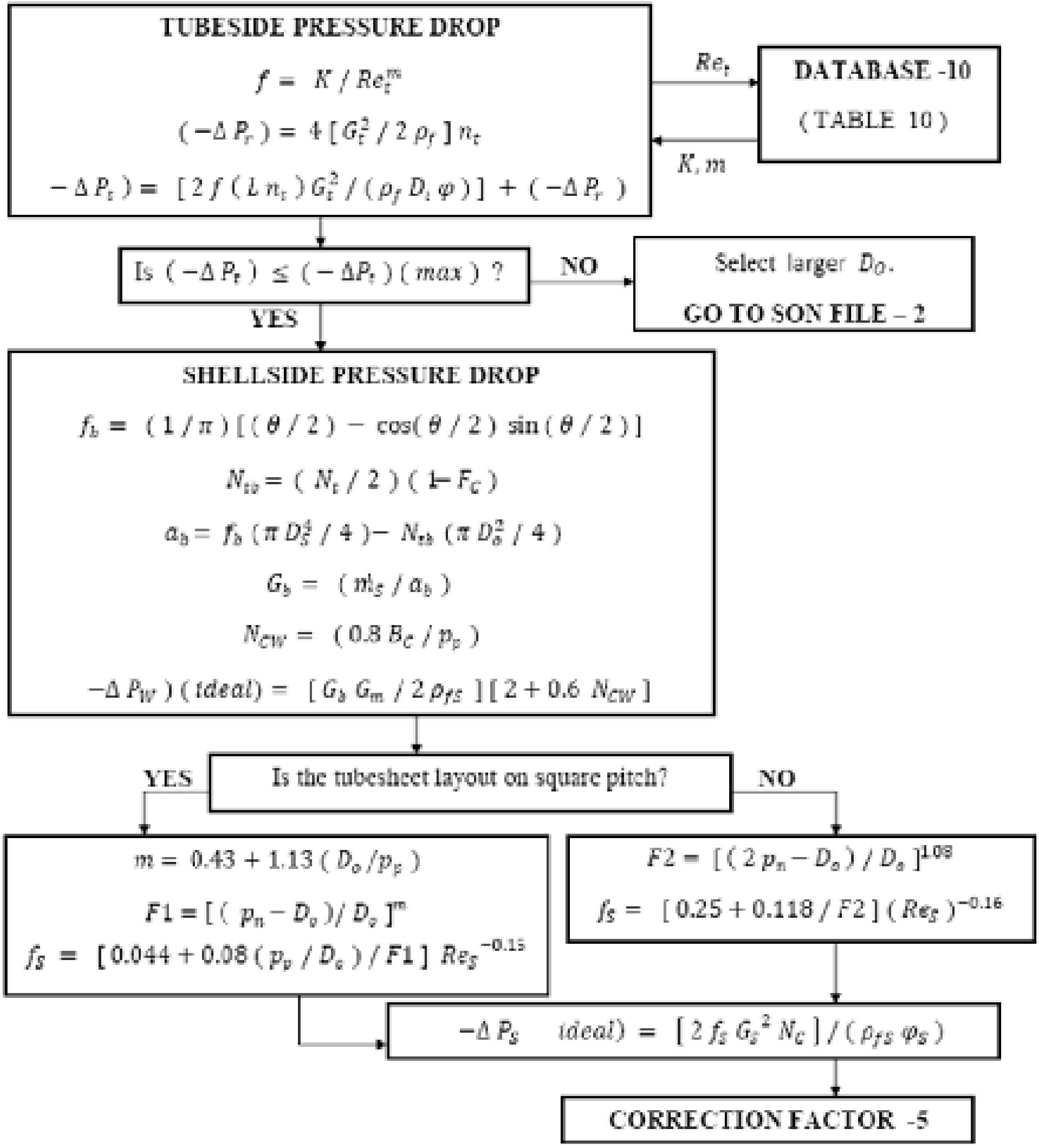

Figure 7O: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

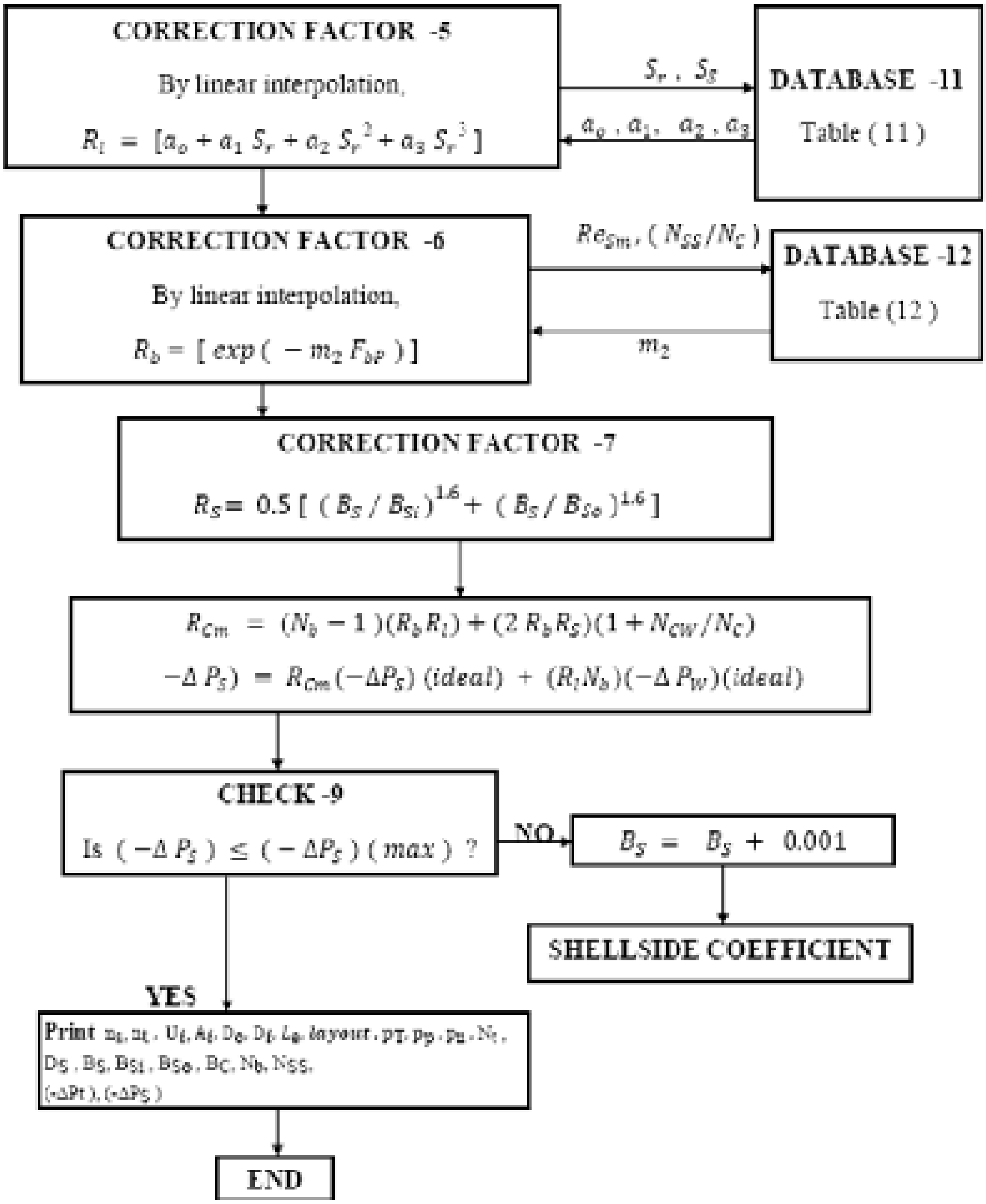

Figure 7P: CAD of shell and Tube Heat Exchangers (Sizing Problem)- countinued

Step 1: specification of father file parameters:

The *father file* is the memory file of the computer in which we enlist the initial problem specifications such as the mass flow rate of shellside fluid and that of tubeside fluid ($\dot{m}_S$, $\dot{m}$),the terminal temperatures of heat exchanger (t_1, t_2, T_1, T_2).

In most cases, one among these parameters could be unknown. For example, let the problem specify $\dot{m}_S$ (mass flow rate of shellside fluid), inlet and outlet temperatures of shellside fluid (T_1, T_2) and also inlet and outlet temperatures of tubeside fluid (t_1, t_2).The mass flow rate of the tubeside fluid ($\dot{m}$) is unknown. This is then evaluated from the overall heat balance shown in step – 3.

It is also required to specify the maximum permissible pressure drop on shellside, ($-\Delta P_S$) (max) and that on tubeside, ($-\Delta P_t$) (max) and also the minimum overall dirt factor prescribed, $R_d(min)$.

Step 2: Estimation of property values of process fluids

Since the physical and transport properties of the fluids (density, viscosity, thermal conductivity) are functions of temperature, they are specified at the mean fluid temperatures (t_m or T_m). Here, t_m is the mean temperature of tubeside fluid and T_m is the mean temperature of the shellside fluid.

If the fluid is a low viscous liquid such as water or aqueous solution, then its property values may be specified at the arithmetic mean temperature (t_{am}, T_{am}). Thus,

$$t_{am} = (t_1 + t_2)/2 \quad (10)$$

$$T_{am} = (T_1 + T_2)/2 \quad (11)$$

In the case of viscous liquids such as petroleum oils, the property values are better specified at caloric mean temperature (t_{cm} or T_{cm}) rather than at the arithmetic mean temperature. The caloric mean temperature is to be computed as given below:

$$t_{cm} = t_1 \pm F_{Cm}|t_1 - t_2| \quad (12)$$

$$T_{cm} = T_1 \pm F_{Cm}|T_1 - T_2| \quad (13)$$

The plus sign is to be used for cold fluid and minus sign for hot fluid.

F_{Cm} = caloric fraction = ($F1/F2$) – ($1/K_C$) (14)

Where,

$$F1 = [1/K_C] + [K_R'/(K_R' - 1)] \quad (15)$$

$$F2 = [1 + \{ln(1 + K_C)/ ln K_R'\}] \quad (16)$$

$$K_R' = (\Delta T_C / \Delta T_h) \quad (17)$$

(ΔT_C) = temperature difference at the cold end of the heat transfer surface / heat exchanger.

(ΔT_h) = temperature difference at the hot end of the heat transfer surface / heat exchanger.

Let the tubeside fluid be the cold fluid and shellside fluid be the hot fluid. Then, from figures (6a) and (6b), **end -1** is the cold end (where the cold fluid enters and the hot fluid leaves) and the **end -2** is the hot end (where, hot fluid enters and the cold fluid leaves).

Accordingly,

$$\Delta T_c = (T_2 - t_1) \quad (18)$$

$$\Delta T_h = (T_1 - t_2) \quad (19)$$

This situation will be reversed if the tubeside fluid is the hot fluid. In such a case, **end-1** shall be the hot end and **end-2** will be the cold end. And,

$$\Delta T_h = (t_1 - T_2) \quad (20)$$

$$\Delta T_c = (t_2 - T_1) \quad (21)$$

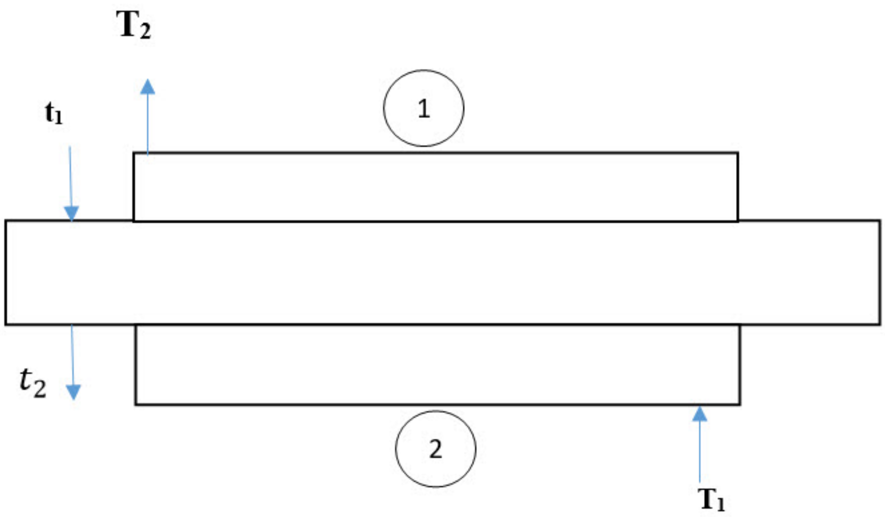

Figure 6 (a): 1-2 Exchanger, indicating the two ends of heat transfer surface.

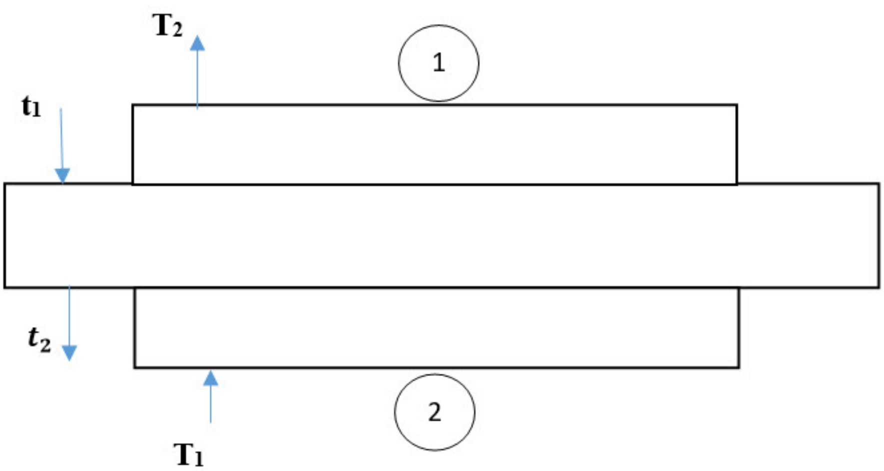

Figure 6 (b): 2-4 Exchanger, indicating the two ends of heat transfer surface.

The value of the parameter K_C depends on the API gravity of the petroleum oil and the temperature difference of the fluid. It may be computed from the correlation given below:
$$ln\,K_C = a_0 + a_1(API) + a_2\,(API)^2 + a_3\,(API)^3 \qquad (23)$$

The values of the correlation coefficients a_0, a_1, a_2, a_3 depend upon the API gravity of the oil and the temperature difference (Δt_C or Δt_h) and are listed in Tables (1A) to (1F). Here,

Δt_C = temperature difference of cold fluid

Δt_h = temperature difference of hot fluid

For example, if the tubeside fluid is the cold fluid, then

$$\Delta t_C = (t_2 - t_1) \qquad (24)$$

$$\Delta t_h = (T_1 - T_2) \qquad (25)$$

For intermediate values of Δt_C or Δt_h, the value of K_C could be estimated by linear interpolation. It is also important to keep in mind that if both the fluids are viscous fluids like petroleum oils, then the value of K_C is to be computed separately for each of them and the larger value is to be used for the computation of caloric fraction, F_{Cm} **(from equation 14)**. For example, let

K_C (1) = the value of K_C for tubeside fluid

K_C (2) = the value of K_C for shellside fluid

Then K_C = larger of K_C (1) and K_C (2)

This has been illustrated in the CAD flow sheet **(Figures 7B to 7C)** under ***Caloric Mean Temperature.***

Table 1A

Table 1A: Values of Correlation Constants for Computation of K_C – factor (Equation – 23) [Database -1]

Coefficients	Δt_c or Δt_h = 278°C				R^2 value
	API = 20 – 29	API = 29 – 39	API = 39 – 50	API = 50 – 70	
a_o	6.3085	–0.47	2.09	0.606	0.996
a_1	–1.818	0.04	–0.036	a.12 x 10^{-3}	
a_2	– 2.73 x 10^{-3}	–6.42 x 10^{-4}	7.36 x 10^{-6}	–3.2 x 10^{-5}	
a_3	9.445 X 10^{-5}	2 x 10^{-5}	9.8 x 10^{-7}	–1.92 x10^{-6}	

Table 1B

Coefficients	Δt_c or Δt_h = 222°C				R^2 value
	API = 13 –28.5	API =28.5 - 41.8	API = 41.8 - 55.5	API = 55.5-70	
a_o	2.43	6.594	87.06	– 5.8768	0.991
a_1	– 0.025	5.743 x10^{-3}	– 45.672	0.0337	
a_2	– 3.858 x10^{-4}	7.482 x10^{-6}	0.879	1.486 x 10^{-4}	
a_3	– 1.552 x10^{-5}	– 1.0289 x10^{-5}	– 4.915 x 10^{-6}	1.2856 x 10^{-5}	

Table 1C

Coefficients	Δt_c or Δt_h = 167°C				R^2 value
	API= 13 - 26	API = 26 - 38.5	API = 38.5 - 53	API = 53 - 68.2	
a_o	2.077	3.074856	4.0074	– 1.8865	0.991
a_1	– 0.0393	– 0.20098	– 0.0799	7.2938 x 10^{-3}	
a_2	2.674 x 10^{-3}	0.00377	– 9.643 x 10^{-6}	4.51 x 10^{-6}	
a_3	– 1.18 x 10^{-4}	– 0.00003	8.7598 x 10^{-8}	2.647 x 10^{-6}	

Table 1D

Coefficients	Δt_c or Δt_h = 111°C				R² value
	API = 10 –23	API = 23.7 –32	API = 33 – 42	API = 43 –55.5	
a_o	0.917844	28.1986	– 9.4515	– 294.69	0.991
a_1	– 0.0435737	– 3.14375	0.620646	17.6774	
a_2	0.00225156	0.116005	– 0.0142829	– 0.353604	
a_3	0.00023358	– 0.0014481	0.000093020	– 0.00234255	

Table 1E

Coefficients	Δt_c or Δt_h = 56°C			R² value
	API = 10 – 20	API = 21 – 30	API = 31 – 48	
a_o	– 1.48379	4.99609	13.1336	0.996
a_1	0.502394	– 0.5937	– 1.07437	
a_2	– 0.039699	0.0217476	0.0264278	
a_3	0.00086013	– 0.00030225	– 0.000223917	

Table 1F

Coefficients	Δt_c or Δt_h = 28°C				R² value
	API = 10 – 14	API = 15 – 20	API = 21 – 30	API = 31 – 35	
a_o	– 16.2581	– 9.60895	– 14.3198	1.43606	0.990
a_1	4.54308	1.55559	1.77582	– 0.194326	
a_2	– 0.40621	– 0.0829894	– 0.0761433	0.00416506	
a_3	0.0116306	0.00133767	0.00103175	– 0.0000396	

In many cases, correlations are available for the estimation of property values at any specified temperature. As for example, let the tubeside fluid be water and its property values are being specified at the arithmetic mean temperature t_{am}, while the shellside fluid is a viscous petroleum oil and its property values are being specified at the caloric mean temperature, T_{Cm}. In other words, $t_m = t_{am}$ and $T_m = T_{Cm}$. Now, for the tubeside fluid,

$$C_p = [\, q_1 + q_2\,(q_3 - t_{am}\,)\,] \tag{26}$$

$$\rho_f = [d_1 + d_2(d_3 - t_{am})] \quad (27)$$

$$k_f = [k_1 + k_2(k_3 - t_{am})] \quad (28)$$

$$ln\,\mu_f = [m_1 + (m_2 / t_{am})] \quad (29)$$

Similarly, for the shellside fluid,

$$C_{PS} = [q_4 + q_5(q_6 - T_{Cm})] \quad (30)$$

$$\rho_{fS} = [d_4 + d_5(d_6 - T_{Cm})] \quad (31)$$

$$k_{fS} = [k_4 + k_5(k_6 - T_{Cm})] \quad (32)$$

$$ln\,\mu_{fS} = [m_3 + (m_4 / T_{Cm})] \quad (33)$$

Where q_1 to q_6, d_1 to d_6, k_1 to k_6 and m_1 to m_4 are correlation constants.

Step 3: Overall heat balance:

As stated earlier, out of the six parameters such as the two flow rates ($\dot{m}, \dot{m}_S$) and the four terminal temperatures (t_1, t_2, T_1, T_2), one of them could be unknown. This is evaluated from the

following overall heat balance:

$$Q = \dot{m}C_P|t_2 - t_1| = \dot{m}_S C_{PS}|T_1 - T_2| \quad (34)$$

This step also computes the magnitude of all overall rate of heat transfer, Q.

Step 4: Initial choice of overall design heat transfer coefficient (U_D)

The recommended range of values of overall design heat transfer coefficient (U_D) for different process fluids are given by TEMA and these are listed in Table (2). The value of U_D is to be selected based on this table. The maximum value of U_D is first selected (since this would correspond to minimum heat transfer surface requirement for the exchanger) and the value of U_D is subsequently decreased if the computed value of overall dirt factor (R_d) is found to be below the minimum prescribed value, such as, $R_d(min)$.

For example, let the specified range of U_D for the process fluids at hand be 425 – 850 $W/(m^2.K)$. Then, the computations are started by assuming U_D = 850 $W/(m^2.K)$. This value of U_D is decreased subsequently and computations repeated if R_d(computed) is found to be less than $R_d(min)$.

The selected value of U_D may be specified as U_0 or U_i. In the CAD flowsheet, it has been specified as U_i. It must be kept in mind that since ($U_i A_i$) = ($U_0 A_0$), the final results shall remain the same in spite of whether U_D has been selected as U_i or U_0.

Step 5: Computation of heat transfer surface (A_o, A_i)

For the computation of the required heat transfer surface of the exchanger, there are three alternate methods, such as

(a) F_T Method

(b) ϵ - NTU Method

(c) Martin's Method

All the above three methods are based on the heat balance equations written separately for each pass of the exchanger and then clubbed together. Accordingly, each of the above methods should predict the same value of the heat transfer surface (A_o or A_i). The choice of the method, therefore, lies on the convenience of the user. All of the above three methods are illustrated in the CAD flowsheet.

Table 2: Recommended Values of Overall Design Heat Transfer Coefficient (U_D) [Database – 2]

Hot fluid	Cold fluid	$U_D, W / (m^2 K)$
Water, methanol, ammonia, aqueous solutions	Water	1420 – 2840
Water	Brine	570 – 1135
Aqueous solutions	Aqueous solutions	1420 – 2840
Light organics (liquids with viscosities less than 0.5 cp like benzene, toluene, acetone, ethanol, MEK, gasoline, naphtha)	Light organics	227 – 425
Light organics	Water	425 – 850
Medium organics (liquids with viscosities 0.5 to 1.0 cp like kerosene, light gas oil)	Medium organics	113 – 340
Medium organics	Water	284 – 710
Heavy organics (liquids with viscosities more than 1.0 cp like lube oils, fuel oils, reduced crude oils, tars, asphalts)	Heavy organics	57 – 227
Heavy organics	Light organics	170 – 340
Heavy organics	Water	30 – 425
Light organics	Heavy organics	56 – 227
Steam	Aqueous solutions ($<$ 2 cp)	1135 – 3975

Steam	Aqueous solutions ($>$ 2 cp)	570 – 2840
Steam	Light organics	570 – 1135
Steam	Medium organics	284 – 570
Steam	Heavy organics	34 – 340
Steam	Water, methanol, ammonia	1135 – 3975
Steam	Gases	30 – 284
Gases	Water	12 – 284

Source: TEMA standards

(a) F_T Method

This method utilises a correction factor F_T, such that,

$$Q = U_i A_i F_T(-\Delta T)_{ln} = U_0 A_0 F_T(-\Delta T)_{ln} \quad (35)$$

where, $(-\Delta T)_{ln}$ = logarithmic mean temperature difference

$$= (\Delta T_h - \Delta T_C) \,/\, ln\,(\Delta T_h / \Delta T_C) \quad (36)$$

ΔT_h = temperature difference at the hot end of the heat transfer surface / heat exchanger (defined earlier)

ΔT_C = temperature difference at cold end of the heat transfer surface / heat exchanger (defined earlier)

The correction factor F_T is to be computed as per the equations given below. For a 1-2 heat exchanger (or for an exchanger with n_S = 1 and n_t = 2, 4, 6 etc),

$$K_1 = \sqrt{(1 + K_R^2)} \quad (37)$$

$$K_2 = (K_R + 1 - K_1) \quad (38)$$

$$K_3 = (K_R + 1 + K_1) \quad (39)$$

$$F1 = K_1\, ln\,[(1 - K_S)/(1 - K_R K_S)] \quad (40)$$

$$F2 = (K_R - 1)\, ln[(2 - K_S K_2)/(2 - K_S K_3)] \quad (41)$$

$$F_T = F1 / F2 \quad (42)$$

In the above equations,

$$K_S = \Delta t_C / \Delta T(max) \quad (43)$$

$$K_R = (\Delta t_h / \Delta t_C) \qquad (44)$$

$$\Delta T(max) = \text{ maximum temperature difference } = |T_1 - t_1| \qquad (45)$$

It is obvious that the maximum temperature difference in the case of any exchanger shall be the difference between the inlet temperature of the hot fluid (highest temperature) and the inlet temperature of the cold fluid (lowest temperature). *It must be noted that the parameter* K_R *defined above is different from* $K_R^{'}$ *defined in equation (17) and used for the computation of caloric mean temperature.*

For a 2-4 heat exchanger (or for an exchanger with n_S = 2 and n_t = 4, 8, 12 etc),

$$K_1 = \sqrt{(1 + K_R^2)} \qquad (46)$$

$$K_2 = (K_R + 1 - K_1) \qquad (47)$$

$$(K_R + 1 + K_1) \qquad (48)$$

$$K_4 = \sqrt{(1 - K_S)(1 - K_R K_S)} \qquad (49)$$

$$F1 = [K_1 / \{2 (K_R - 1)\}] \ln [(1 - K_S) / (1 - K_R K_S)] \qquad (50)$$

$$F3 = [(2 / K_S) - K_2 + (2 / K_S) K_4] \qquad (51)$$

$$F4 = [(2 / K_S) - K_3 + (2 / K_S) K_4] \qquad (52)$$

$$F2 = ln(F3 / F4) \qquad (53)$$

$$F_T = (F1 / F2) \qquad (54)$$

Once the value of F_T has been computed, then the heat transfer surface required (A_o or A_i) can be estimated from equation (35).

As stated earlier, it is better to choose a 1 – 2 exchanger ($n_S = 1$, $n_t = 2$) at the outset. If the computed value of F_T factor turns out to be negative or indeterminate, then it means that such an exchanger is unsuitable and we have to proceed to design an exchanger of larger number of passes (such as a 2 – 4 exchanger).

b) ϵ - NTU Method

Here, we define two parameters such as heat exchanger effectiveness (ϵ) and number of transfer units, $NTU\ (max)$. These are defined as given below:

$$\epsilon = Q / [C\ (min) \Delta T\ (max)] \qquad (55)$$

$$NTU\ (max) = [\ U_o A_o\ /C\ (min)] \tag{56}$$

$$= [\ U_i A_i\ /C\ (min)] \tag{57}$$

where,

$$C\ (\min) = \text{ smaller of } (\dot{m}C_P) \text{ and } (\dot{m}_S C_{PS}) \tag{58}$$

The number of transfer units, $NTU\ (max)$, can be computed as described below.

For a 1-2 heat exchanger (or for an exchanger with n_S = 1 and n_t = 2, 4, 6 etc),

$$N_1 = [\ (\ 2/\epsilon\ ) - 1 - C + C_1\ ] \tag{59}$$

$$N_2 = [\ (\ 2/\epsilon\ ) - 1 - C - C_1\ ] \tag{60}$$

$$NTU\ (max) = (\ 1\ /\ C_1\ )\ ln\ (N_1/\ N_2) \tag{61}$$

where

$$C = C\ (\ min\ )\ /\ C\ (\ max\ ) \tag{62}$$

$$C_1 = \sqrt{(\ 1 + C^2)} \tag{63}$$

$$C\ (max) = \text{ larger of } (\dot{m}C_P) \text{ and } (\dot{m}_S C_{PS}) \tag{64}$$

For a 2-4 heat exchanger (or for an exchanger with n_S = 2 and n_t = 4, 8, 12 etc),

$$C_S = \sqrt{(\ 1 - \epsilon\ )\ (\ 1 - \epsilon\ C\ )} \tag{65}$$

$$C_S = \sqrt{(\ 1 - \epsilon\ )\ (\ 1 - \epsilon\ C\ )} \tag{65}$$

$$N_3 = [\ (\ 2\ /\ \epsilon) - 1 - C + (\ 2\ /\ \epsilon\ )\ C_S + C_1\ ] \tag{66}$$

$$N_4 = [\ (\ 2/\ \epsilon) - 1 - C + (\ 2\ /\ \epsilon\ )\ C_S - C_1\ ] \tag{67}$$

$$NTU\ (max) = (\ 2\ /\ C_1\ )\ \ln\ (N_3\ /\ N_4\ ) \tag{68}$$

Once the value of $NTU\ (max)$ has been computed, then the heat transfer surface required (A_o or A_i) can be estimated from equation (56) or (57). As stated under F_T – method, in this case also, if by considering a 1 – 2 exchanger, the computed value of $NTU\ (max)$ is seen to be negative or indeterminate, then it indicates that the selected exchanger is inadequate and we have to go for a 2 – 4 exchanger.

C) Martin's Method

The method proposed by Martin involves a trial and error procedure. A value of A_o or A_i

is to be assumed at the outset and subsequently verified. The procedure is outlined below:

1. Assume a value of A_o or A_i.

For example, let

$$A_o = Q / [U_o(-\Delta T)_{ln}] \qquad (69)$$

or,

$$A_i = Q / [U_i(-\Delta T)_{ln}] \qquad (70)$$

2. Compute parameters X, Y, Z as

$$X = (U_i A_i) / (\dot{m}_S C_{Ps}) \qquad (71)$$

$$Y = (U_i A_i) / (\dot{m} C_P) \qquad (72)$$

$$M = (2 Y / n_t) \qquad (73)$$

$$Z = \sqrt{X^2 + M^2} \qquad (74)$$

3. Compute $\Phi (Y)$, $\Phi (M)$ and $\Phi (Z)$ as

$$\Phi (Y) = Y / [1 - \exp (- Y)] \qquad (75)$$

$$\Phi (M) = M / [1 - \exp (- M)] \qquad (76)$$

$$\Phi (Z) = Z / [1 - \exp (- Z)] \qquad (77)$$

4. Compute the parameter Θ as

$$(1 / \Theta) = \Phi (Y) + \Phi (Z) - \Phi (M) + 0.5 [X + M - Z] \qquad (78)$$

5. Compute ϵ_Y as

$$\epsilon_Y = (Y \Theta) \qquad (79)$$

6. Compute exit temperature (t_2) of tubeside fluid as

$$t_2(cal) = t_1 \pm \epsilon_Y \Delta T(max) \qquad (80)$$

The plus sign is to be used if the tubeside fluid is cold fluid and the minus sign if the tubeside fluid is hot fluid.

7. If the above – computed value of t_2 agrees with the value of t_2 specified in the problem within 1°C, then print A_o or A_i. Otherwise, increase A_o or A_i (for example, $A_i = A_i + 0.1\, m^2$)

and repeat the computations starting from Step 2.

Step 6: Computation of number of tubes required

Select exchanger specifications such as OD of tubes (D_o), tube wall thickness (δ) , tube pitch (p_T) and the tubesheet layout (square pitch / rotated square pitch / triangular pitch). Select also the effective length of each tube (L_e). Now, compute the number of tubes required as,

$$N_t\,(cal) = [\, A_i \,/\, (\pi\, D_i\, L_e)\,] \tag{81}$$

$$= [\, A_o \,/\, (\pi\, D_o\, L_e)\,] \tag{82}$$

The above calculated value of N_t is to be rounded off to the nearest higher standard value with reference to the standard tube count tables (tables 3A to 3F which constitute **database – 3**). The value of A_o(or A_i)) and U_o(or U_i) are to be recomputed based on the above chosen value of N_t . The internal dameter of shell (D_S) is also retrieved from the tube count table (**database– 3**). This has been clearly illustrated in the CAD flowsheet. Select also the baffle spacing (B_S). It is common practice to start the computations by choosing

$$B_S = B_S\,(min) = (\, D_S \,/\, 5\,) \tag{83}$$

This would provide the largest magnitude of shellside heat transfer coefficient (h_o). However, the value B_S would have to be increased subsequently if pressure drop considerations demand so. This is discussed in one of the subsequent steps (Step – 14).

If the baffle spacing at the inlet (B_{Si}) and that at the outlet (B_{So}) are to be chosen different from B_S, then the values of B_{Si} and B_{So} are also to be specified. As stated earlier, larger baffle spacing is often required at the shell inlet as well as at the shell outlet in order to accommodate the shell inlet nozzle and shell outlet nozzle. No doubt, it is always desirable to employ a uniform baffle spacing such that

$$B_{Si} = B_{So} = B_S \tag{84}$$

Once the tubesheet layout has been chosen, it is also necessary to specify the tube pitch parallel to flow (p_p) and that normal to flow (p_n), based on equations (1) to (4).

Step 7: Computation of Tubeside heat transfer coefficient (h_i)

The tubeside heat transfer coefficient (h_i) depends on the tubeside Reynolds number (Re_t) and the Prandtl Number of tubeside fluid (Pr) and these are defined below:

$$Re_t = [\, D_i\, G_t \,/\, \mu_f\,] \tag{85}$$

where

$G_t =$ mass velocity of tubeside fluid

$$= (\dot{m} / a_t) \quad (86)$$

$a_t =$ tubeside flow area

$$= \pi(D_i^2 / 4)(N_t / n_t) \quad (87)$$

$n_t =$ number of tubeside passes

$$Pr = (C_p \mu_f / k_f) \quad (88)$$

In most industrial shell and tube heat exchangers, the tubeside fluid is usually made to execute fully developed turbulent flow (Re_t >10,000) so as to maintain the tubeside heat transfer coefficient at a high magnitude. In such cases, when Re_t is greater than 10000, the value of tubeside heat transfer coefficient could be computed from the Dittus – Boelter equation (modified by Sieder and Tate) and this is reproduced below. This correlation is valid for a Prandtl number range of $0.7 \leq Pr \leq 16700$:

$$Nu = 0.027 (Re_t)^{0.8} (Pr)^{0.33} \varphi \quad (89)$$

where

$Nu =$ tubeside Nusselt number

$$= (h_i D_i / k_f) \quad (90)$$

$\varphi =$ viscosity correction factor

$$= (\mu_f / \mu_{fw})^{0.14} \quad (91)$$

$\mu_{fw} =$ viscosity of tubeside fluid at the inside wall temperature (t_{wi}) of tubes

The above equation is applicable for the flow of all Newtonian fluids ***except water***. If the process fluid is water, then the value of tubeside heat transfer coefficient should be estimated from the *dimensional* correlation reported by Perry [1] and subsequently modified by Narayanan and Bhattacharya [2,3]. This correlation is based on the graphical data reported by Eagle and Ferguson [4]:

$$h_i = (1057.0)\, C_f\, (1.352 + 0.02\, t_m)\, [(G_t / \rho_f)^{0.8} / D_i^{0.2}]\, (\varphi) \quad (92)$$

where $$C_f = -0.1864\ ln(D_i) + 0.22455 \quad (93)$$

It must be kept in mind that the above correlation is *dimensional* in nature and all parameters involved must be expressed in their corresponding SI units. This equation is also valid for only fully developed turbulent flow of water through straight, cylindrical tubes (Re_t >10,000). However, in most commercial heat exchangers, the velocity of cooling water through tubes is maintained at more than 1.8 m/s (to minimize *precipitation fouling*) and consequently, the flow regime shall be in the fully developed turbulent zone.

Precipitation fouling is caused by the dissolved salts present in water such as sulfates, silicates and hydroxides of calcium and magnesium which are called *inverse solubility salts*, since the solubility of these salts decreases with increase in temperature. At high temperatures therefore, these salts precipitate out and deposit on the heat transfer surfaces causing fouling or scaling. The deposited scale being a poor conductor of heat offers additional resistance to heat transfer and thus brings down the performance of the exchanger. At high fluid velocities, the deposited dirt could get re-entrained into the flowing fluid stream and this helps in impeding precipitation fouling. Also, the exit temperature of cooling water should not necessarily be permitted to increase beyond 50C, since scaling occurs predominantly at high temperatures.

It is not yet fully understood why the Dittus – Boelter equation (equation – 89) is not valid for water, though it is applicable to all other Newtonian fluids. A possible reason is that the properties of water (density, thermal conductivity) exhibit unusual (often, anomalous) temperature dependence [1,2,3].

At the outset, the value of viscosity correction factor (φ) may be taken equal to unity and the value of h_i be computed from any of the correlations given above. This value of h_i(i.e, the value h_i at $\varphi = 1.0$) is denoted as h_i' in the CAD flowsheet. The incorporation of φ and the estimation of corrected value of h_i is discussed in one of the subsequent steps. It may also be noted that this correction factor (φ) is not a detrimental parameter. For water and many aqueous solutions, this factor may be taken more or less equal to unity.

Step 8: Computation of Shellside Heat Transfer Coefficient, h_o $(ideal)$

As stated earlier, the flow of shellside fluid is, in fact, tortuous. It flows over the tube bundle in the section between the baffles, thereby executing crossflow. But, as it flows from one crossflow section to another, it executes countercurrent or co-current flow (depending on the flow direction of tubeside fluid). The shellside fluid, thus, executes partly crossflow, partly countercurrent flow and partly co-current or parallel flow. All of the experimental correlations reported in literature are those which consider *true crossflow* or in other words, that consider *ideal crossflow section*. Accordingly, the shellside heat transfer coefficient predicted by these correlations is h_0 (ideal). Correction factors are to be, therefore, incorporated to take care of supplementary effects and thereby to estimate the actual value of shellside heat transfer coefficient (h_0).

For the estimation of h_0 (ideal), one of the reliable correlations is that proposed by Colburn [5]. This correlation is given below:

$$Nu_S = a_o (Re_S)^{0.6} (Pr_S)^{1/3} \varphi_S \quad (94)$$

where Nu_S = shellside Nusselt number

$$= h_0 \text{ (ideal) } D_0 / k_{fs} \quad (95)$$

Re_S = shellside Reynolds number

$$= (D_o G_S / \mu_{fs}) \quad (96)$$

G_S = mass velocity of the shellside fluid based on flow area, a_S

$$= (\dot{m}_S / a_S) \quad (97)$$

a_S = minimum free flow area between baffles at the shell axis

$$= D_S B_S (p_T - D_o) / (n_S p_T) \quad (98)$$

a_o = 0.33, for staggered tubes (for tubes that are in triangular pitch or rotated square pitch arrangement)

= 0.26, for tubes in line (for tubes that are in square pitch arrangement)

Pr_S = Prandtl number of shellside fluid

$$= (C_{PS} \mu_{fs}) / k_{fs} \quad (99)$$

φ_S = viscosity correction factor for shellside fluid

$$= (\mu_{fs} / \mu_{WS})^{0.14} \quad (99a)$$

μ_{WS} = viscosity of shellside fluid at outer wall temperature (t_{wo}) of tubes

The above correlation is valid for $2000 \leq Re_S \leq 32{,}000$. An alternate correlation for the estimation of $h_o(ideal)$ has been proposed by Donohue [6]. This correlation uses a modified shellside Reynolds number that is based on the geometric average of the mass velocity of shellside fluid in the crossflow section (G_S) and that in the baffle window (G_b). Thus

Re_S' = shellside Reynolds number

$$= (D_o G_e) / \mu_{fs} \quad (100)$$

where

$$G_e = \sqrt{G_S G_b} \tag{101}$$

G_b = mass velocity of shellside fluid in baffle window

$$= (\dot{m}_S \,/\, a_b) \tag{102}$$

a_b = free area for flow of shellside fluid in the baffle window (discussed subsequently in Step – 14 under pressure drop computations)

Donohue's correlation often predicts much lower value of h_0 (ideal) as compared to that predicted by Colburn's correlation. Though the approach used by Donohue is more renovated, dubiousness does exist over the accuracy of employing a geometric average of G_S and G_b. These two mass velocities are not always of comparable magnitude.

Alternate correlations have been proposed by McAdams [7] and also by Kern [8]. Kern has defined an equivalent diameter for the shell and has used the same in the correlation. However, the flow area used for defining the equivalent diameter is the free area (free space) between tubes. Since shellside fluid does flow over the tubes (over the tube bundle), the approach of Kern cannot be treated as fully accurate.

After comparing the different experimental correlations available, ***it is recommended that for the usual case of shellside Reynolds number (Re_S) exceeding 3000, Colburn's correlation (equation – 94) be used for computing h_0(ideal).*** No doubt, it is to be multiplied by the appropriately defined correction factors (discussed subsequently) to obtain the actual magnitude of shellside heat transfer coefficient, h_0.

Step 9: Estimation of correction factors and actual shellside heat transfer coefficient (h_o)

As stated above, the value of shellside heat transfer coefficient computed from Colburn's correlation is that for ideal crossflow section, $h_o(ideal)$. In an industrial heat exchanger however, supplementary effects come into play such as baffle configuration effect, baffle leakage effect, bundle bypassing effect and that due to unequal baffle spacing. Correction factors are to be incorporated to account for each of these effects. Thus,

$$h_o = h_o(ideal) (J_C \, J_l \, J_b \, J_S) \tag{103}$$

where

J_C = correction factor that accounts for baffle configuration effect,

J_l = correction factor that accounts for shell to baffle leakage and. tube tobaffle leakage,

J_b = correction factor that accounts for bundle by passing effect and J_S = correction factor that accounts for unequal baffle spacing.

Elaborate graphical data have been reported by Bell [9] for the computation of these correction factors. Bell's graphical data have been fitted into analytical correlations by Narayanan and Bhattacharya [2]. These are discussed below :

The correction factor J_C is to take care of the fact that a portion of shellside fluid that flows through the baffle window executes more or less countercurrent flow or co-current flow, rather than crossflow. Since heat transfer coefficient is highest in crossflow, this tends to bring down the overall magnitude of the shellside heat transfer coefficient. If tubes are avoided in the baffle window (no – tubes – in – baffle window construction), then J_C = 1.0. The correlation developed by Narayanan and Bhattacharya [2] for the estimation of this correction factor is as follows:

$$J_c = c_o + c_1(F_c) + c_2(F_c)^2 + c_3(F_c)^3 \quad (104)$$

where

F_C = fraction of total tubes in crossflow

$$= 1 + (2/\pi)\cos\theta_C \sin\theta_C - (2\,\theta_C/\pi) \quad (105)$$

$$\text{Cos}\,\theta_C = [(D_S - 2\,B_C)/D_{ot}] \quad (106)$$

B_C is the baffle cut and as stated earlier, for 25% cut segmental baffles that are popularly used, $B_C = (D_S/4)$. In the above equation (105), θ_C must be expressed ***in radians***.

D_{ot} is called the outer tube limit and it depends on the type of exchanger construction and the shell ID. It is to be kept in mind that in a shell and tube heat exchanger, tubes are laid in the shell within D_{ot} and not within the entire cross – section of the shell. The values of D_{ot} specified by TEMA are listed in Table (4) which constitutes **Database – 4**. It can be seen from table (4) that for pipe shells (lower diameter shells), D_{ot} is around 11 mm less than the shell diameter when a fixed tubesheet construction is used, while it is 29 mm less than the shell diameter for a floating head construction. Similarly, in the case large diameter plate shells, D_{ot} is 13 mm less than the shell diameter in fixed tube sheet exchangers, whereas in floating head exchangers, it is 37 mm less than the shell diameter.

Table 4: Values of Outer Tube Limit (D_{ot}) recommended by TEMA Standards [Database – 4]

	Shell diameter (D_S), mm	D_{ot}, mm	
		Fixed tubesheet	Internal floating head with split backing ring
Plate shells	$>$637 mm	D_S – 13 mm	D_S – 37 mm
Pipe shells	$\leq$ 610 mm	D_S – 11 mm	D_S – 29 mm

The values of correlation constants c_o, c_1, c_2, c_3 are listed in Table (5) which constitutes **Database – 5.**

Table 5: Values of Correlation Constants for Computation of Correction Factor J_c (equation –104)

Correlation constant	$F_c \leq 0.6$	$0.6 \leq F_c \leq 0.8$	$0.8 \leq F_c \leq 0.9$	$0.8 \leq F_c \leq 1.0$
c_o	0.531428	0.6406	- 2.1616	557.71946
c_1	0.7737	0.588	7.37824	- 1793.534
c_2	0.0	0.0	- 4.11426	1925.5329
c_3	0.0	0.0	0.0	- 688.7156

J_l is the correction factor to account for the leakage of shellside fluid through the shell-to-baffle clearances and the tube-to-baffle clearances. Its value varies from 0.7 to 0.8. A portion of the shellside fluid flows through the shell to baffle clearances and also through the tube to baffle clearances. These are called the *leakage streams*. Due to these leakage streams, the fraction of shellside fluid executing crossflow gets reduced and this penalizes the shellside heat transfer coefficient. As specified by TEMA, the tube to baffle clearance (δ_{tb}) ranges from 0.4 to 0.8 mm and the shell to baffle clearance (δ_{sb}) varies from 2.54 mm for small diameter pipe shells to as high as 10.8 mm for large diameter plate shells. The values of δ_{sb} and δ_{tb} as specified by TEMA are listed in Table (6) which constitutes **Database – 6.** The correlation developed by Narayanan and Bhattacharya [2] for the estimation of this correction factor is given below:

$$J_l = a_o + a_1(S_r) + a_2(S_r)^2 + a_3(S_r)^3 \qquad (107)$$

where

$$S_r = (A_{tb} + A_{sb}) / A_m \qquad (108)$$

$$S_s = A_{sb} / (A_{tb} + A_{sb}) \qquad (109)$$

A_{tb} = tube to baffle leakage area

$$= [\,\pi D_o\, \delta_{tb}\, N_t\,(1 + F_C)\,]\,/\,2 \tag{110}$$

A_{sb} = shell to baffle leakage area

$$= (\pi D_s \delta_{sb}/2)[1-(\theta/2\pi)] \tag{111}$$

δ_{tb} = tube to baffle clearance

δ_{Sb} = shell to baffle clearance

θ = baffle cut angle (in ***radians***)

$$= 2\,Cos^{-1}[\,(D_S - 2\,B_C)/\,D_S\,] \tag{112}$$

As stated above, for 25 % cut segmental baffles, B_C = (D_S / 4) and therefore, θ = 120^0 or ($2\pi/3$) radians.

A_m = crossflow area at or near center line

$$= B_S\,[\,D_S - D_{ot} + \{(D_{ot} - D_o)\,(p_T - D_o)\,/\beta p_T\}] \tag{113}$$

If tubes are arranged on the tubesheet on a triangular pitch layout, then β = 1.0 and if they are laid on a square or rotated square layout, then $\beta = (p_n/p_T)$.

The values of correlation constants a_o, a_1, a_2, a_3 are listed in Table (7) which constitutes **Database – 7.**

Table 6: [Database 6] Recommended Values of Tube to Baffle Clearance (δ_{tb}) and Shell to Baffle Clearance (δ_{sb})

δ_{tb} = 0.8 mm, if maximum unsupported tube length (usually $2B_S$) $\leq$ 910 mm, δ_{tb} = 0.4 mm, if ($2B_S$) $>$ 910 mm.

	Shell diameter, mm	δ_{Sb} mm
Pipe shells	203.2 – 336.5	2.54
	355.6 – 439.1	3.175
	457.2 – 590.8	3.81
Plate shells	609.6 – 990.6	7.62
	1016.0 – 1371.6	8.89
	Above 1397.0	10.80

Table 7: [Database– 7] Values of Correlation Constants for Computation of Correction Factor J_l (equation – 107)

	S_S	a_0	a_1	a_2	a_3
$S_r \leq 0.2$	0.0	0.997	– 2.54167	15.239	– 36.276
	0.25	1.0	– 3.0845	17.2089	– 38.6776
	0.50	0.9957	– 3.804	22.045	– 50.586
	0.75	0.9952	– 4.0808	21.764	– 47.946
	1.0	0.9916	– 5.0	29.0	– 66.532
$0.2 < S_r \leq 0.7$	0.0	0.8975	– 0.4375	0.0	0.0
	0.25	0.87	– 0.55	0.0	0.0
	0.50	0.8525	– 0.6625	0.0	0.0
	0.75	0.825	– 0.775	0.0	0.0
	1.0	0.7925	– 0.8375	0.0	0.0

The correction factor J_b has been incorporated to take care of the *bundle bypassing effect*. That portion of the shellside fluid which flows through the clearance between the outermost tube and the shell wall has a tendency to flow adjacent to the shell wall and thereby bypass the tube bundle (it does not flow over the tube bundle). In the case of fixed tube sheet exchangers, the clearance between the outermost tube and the shell wall is usually maintained small and hence, this effect is not predominant and the value of J_b shall be quite high (around 0.9). However, in floating head exchangers, J_b values as low as 0.7 have been reported. One of the means of minimizing the bundle bypassing effect is to install sealing strips, which are typically longitudinal strips of metal installed between the outside of the tube bundle and the shell and fastened to the baffles. These strips force back the bypass stream into the main crossflow stream and thereby reduce the bypassing effect and improve the heat transfer coefficient. It must be, however, kept in mind that sealing strips are cumbersome to install and maintain. The correlation developed by Narayanan and Bhattacharya [2] for the estimation of this correction factor is given below:

$$J_b = [\, exp\,(-m_1 F_{bP})\,] \quad (114)$$

where

$$F_{bP} = (D_S - D_{ot})\ B_S \ / A_m \quad (115)$$

The value of correlation constant m_1 depends on the values of Re_{Sm} and the ratio (N_{SS} / N_C) and can be retrieved from Table (8) which constitutes **Database – 8**. Here,

Re_{Sm} = modified shellside Reynolds number based on G_m

$$= (D_o G_m / \mu_{fs}) \quad (116)$$

where

G_m = shellside mass velocity based on A_m

$= (\dot{m}_S \ / A_m)$ (117)

N_{SS} = number of sealing strips installed per cross flow section and N_C = number of tube rows crossed during flow through one crossflow section.

$= (D_S - 2B_C) / p_P$ (118)

p_P = tube pitch parallel to flow

It is thus clear from Table (8) that when the ratio ($N_{SS} \ / \ N_C$) is equal to 0.5 or more, J_b shall be equal to 1.0 and the bypassing effect shall be absent.

The correction factor J_S is to take care of the effect of unequal baffle spacing on the shellside heat transfer coefficient. It has been explained earlier that due to the presence of nozzles, a larger baffle spacing is often required to be used at the inlet and at the outlet of the exchanger (B_{Si}, B_{So}). If $B_{Si} = B_{So} = B_S$ (which is most preferable), then

$J_S = 1.0$ (119)

The value of J_S thus obviously depends on the $(B_{Si} \ / \ B_S)$ and (B_{So}/B_S) ratios and the number of baffles used and can be estimated as follows:

$J_S = (F1 / F2)$ (120)

where

$F1 = (N_b - 1) + (B_{Si} \ / \ B_S)^{0.4} + (B_{So}/B_S)^{0.4}$ (121)

$F2 = (N_b - 1) + (B_{Si} \ / \ B_S) + (B_{So}/B_S)$ (122)

N_b = number of baffles

$= [(L_e - B_{Si} - B_{So}) / B_S] + 1$ (123)

Table 8: Values of Correlation Constant for Computation of Correction Factor J_b (Equation – 114)[Database – 8]

$N_{SS} \ / \ N_C$	$Re_{Sm} \geq 100$	$Re_{Sm} < 100$
	m_1	m_1
0.0	1.2344	1.3433
0.05	0.6704	0.72975
0.10	0.5095	0.5811
0.167	0.37895	0.4324
0.30	0.1777	0.2055
0.5 and above	0.0	0.0

Once all the four correction factors have been evaluated, then the value of h_0 can be computed from equation (103). To start with, viscosity correction factor φ_S may be assumed equal to 1.0. Accordingly, the above – computed value of h_0 be designated as $h_0^{'}$.

Step 10: Viscosity correction

As stated earlier, the viscosity correction factor (φ, φ_S) is not a controlling parameter. For many systems, its value is very close to 1.0. To compute this correction factor, we need to estimate the tube surface temperature (t_{Wi}, t_{Wo}). However, it must be kept in mind that tube surface temperature varies from one end to the other end of the exchanger and it is also a parameter that is difficult to record experimentally. But since, as stated earlier, φ or φ_S is not a highly influencing parameter, we need to determine only an order of magnitude of t_{Wi} and t_{Wo} . In the design computations therefore, approximate estimation of inner surface temperature of tubes (t_{Wi}) and outer surface temperature of tubes (t_{Wo}) is performed from the following ***approximate*** heat balance equations:

$$h_0 A_0 |T_m - t_{Wo}| = h_i A_i |t_m - t_{Wi}| \tag{124}$$

$$= U_{Ci} A_i |T_m - t_m| \tag{125}$$

$$= U_{C0} A_0 |T_m - t_m| \tag{126}$$

where

U_{Ci} , U_{Co} = overall ***clean*** heat transfer coefficient (value of U when fouling coefficients or dirt factors are excluded) based on A_i and based on A_o respectively

It is important to note that the above equation is approximate (since it does not accurately define the temperature difference driving force) and should be used for the approximate estimation of t_{Wi} and t_{Wo} only. Now, from the above equations,

$$t_{Wi} = t_m \pm [\, U_{Ci}(T_m - t_m) \,/\, h_i \,] \tag{127}$$

$$t_{Wo} = T_m \pm [\, U_{Ci} D_i \,(T_m - t_m) \,/\, h_o D_o] \tag{128}$$

The plus sign is to be used for the cold fluid and the minus sign for the hot fluid.

The computation of viscosity correction factor φ (or φ_S) based on the above equations involves a trial and error procedure, which is summarized below:

1. Assume the values of t_{Wi} and t_{Wo}. For example, if the tubeside fluid is cold fluid then,

$$t_{Wi} = t_m + 1.0 \tag{129}$$

Similarly, if the shellside fluid is hot fluid, then

$$t_{Wo} = T_m - 1.0 \quad (130)$$

2. Put $XI = t_{Wi}$ (131)

and $XO = t_{Wo}$. (132)

3. Compute μ_{fw} (viscosity of tubeside fluid at t_{Wi}) and μ_{WS} (viscosity of shellside fluid at t_{Wo}) from the available property value correlations and then, estimate φ and φ_S as,

$$\varphi = (\mu_f/\mu_{fW})^{0.14} \quad (133)$$

$$\varphi_S = (\mu_{fS}/\mu_{WS})^{0.14} \quad (134)$$

4. Compute the corrected values of tubeside heat transfer coefficient (h_i) and that of shellside heat transfer coefficient (h_0) as

$$h_i = h_i'\varphi \quad (135)$$

$$h_o = h_o'\varphi_S \quad (136)$$

5. Compute the ***clean*** overall heat transfer coefficient U_{Ci} (or U_{Co}) as

$$1/U_{Ci} = (1/h_i) + (1/h_o)(D_i/D_o) \quad (137)$$

$$1/U_{Co} = (1/h_o) + (1/h_i)(D_o/D_i) \quad (138)$$

6. Re-compute the tube surface temperatures (t_{Wi} and t_{Wo}) from equations (127 and 128), using the above computed value of overall ***clean*** heat transfer coefficient, U_{Ci} or U_{Co}.

7. Compute the deviations as

$$DWI = |XI - t_{Wi}| \quad (139)$$

$$DWO = |XO - t_{Wo}| \quad (140)$$

8. If either DWI or DWO has been found to exceed 0.5°C , then repeat the computations starting from Step – 2. Otherwise, print the values of h_i and h_o.

Usually, the scheme shall converge within two to three iterations.

Step 11: Computation of overall dirt factor

Compute the overall dirt factor (R_d) as

$$R_d = (1/U_i) - (1/U_{Ci}) \quad (141)$$

$$= (1/U_o) - (1/U_{Co}) \quad (142)$$

If the above-computed value of R_d falls below $R_d\ (min)$, then proceed to Step – 12 for re-computation of the heat transfer surface. Otherwise, proceed to Step – 13 for pressure drop computations.. The recommended values of minimum dirt factor specified by TEMA are listed in Table (8) which constitutes **database – 9.**

Step 12: Re-computation of heat transfer surface

Since the computed value of overall dirt factor has been found to be less than the minimum required value of $R_d\ (min)$, the value of overall heat transfer coefficient U_i (or U_o) is to be decreased and the computations repeated as outlined below :

(i) Put $U_i = U_i - 1.0$ (143)

or

$$U_o = U_o - 1.0 \quad (144)$$

(ii) Re-compute the heat transfer surface as $A_i = Q / [\, U_i F_T (-\Delta T)_{ln} \,]$ (145)

or

$$A_i = [NTU\ (max)\, C\ (min) /\ U_i\,] \quad (146)$$

(iii) Repeat the computations starting from **Step – 6.**

The procedure is to be continued until the computed value of overall dirt factor (R_d) exceeds $R_d\ (min)$.

Table – 9: Minimum Recommended Values of Dirt Factor, $R_d(min)$ [Database – 9]

Process fluid	Dirt factor, $R_d(min),\ (m^2.K)/W$
Fuel oil	0.0009
Machine oil, transformer oil	0.00018
Quenching oil	0.0007
Vegetable oils	0.0006
Organic liquids	0.0002
Refrigerating liquids	0.0002
Brine (cooling)	0.0002
Organic vapours	0.0001
Steam (non-oil bearing)	0.0
Alcohol vapours	0.0
Steam, exhaust (oil bearing)	0.0002
Air	0.0004
Coke oven gas, manufactured gas	0.002

Source: TEMA Standards

Note : In the case of water, with temperature of water ≤ 52 C and water velocity ≥ 1.2 m/s, the recommended value of $R_d\ (min)$ is 0.0001 $(m^2.K)\ /\ W$ for sea water, distilled water and treated boiler feed water, while it is 0.0002 $(m^2.K)\ /\ W$ for brackish water. clean river water and treated make-up water used in cooling towers. At the same water velocity and temperature, $R_d\ (min)$ specified for hard water (over 15 grains / gal) is 0.0006 $(m^2.K)\ /\ W$ and that for muddy or silty river water is 0.0004 $(m^2.K)\ /\ W$.

Step 13: Computation of tubeside pressure drop

Heat exchanger calculations are incomplete, unless the pressure drop in either stream is evaluated and ascertained that neither of them (pressure drop in the tubeside fluid or that in the shellside fluid) exceeds the maximum permissible limit. To note that the operating cost of the exchanger is decided by the magnitude of pressure drop in the two streams. The tubeside pressure drop includes frictional pressure drop (due to *skin friction* between the tube wall and the fluid layer) which is predicted by the modified form of Fanning's equation (corrected for non-isothermal flow) and the additional pressure drop due to flow reversal (by virtue of multipass construction). Thus

$$(-\Delta P_t) = \left[2 f\,(L_e\, n_t) G_t^2 / \left(\rho_f\, D_i\, \varphi\right)\right] + (-\Delta P_r) \qquad (147)$$

where

$(-\Delta P_r)$ = additional pressure drop due to flow reversal = four velocity heads per pass (observed experimentally)

$$= 4\,[\,G_t^2\,/\,(2\,\rho_f)\,]n_t = 2\,(\,G_t^2 n_t\,)\,/\,\rho_f \qquad (148)$$

n_t = number of tubeside passes f = tube side friction factor (for non-isothermal flow)

$$= K\,/\,Re_t^m \qquad (149)$$

where K and m are empirical constants. The values of these constants are listed in table (10) which constitutes **database – 10**.

Table – 10: Friction factor in Non – isothermal flow Values of Correlation Constants K and m (Equation – 149) [Database – 10]

Re_t	Smooth tubes		Commercial pipes	
	K	m	K	m
≤ 1000	18.0	1.0	18.0	1.0
1000 to 10^5	0.12	0.272	0.105	0.243
10^5 to 10^6	0.087	0.2413	0.0423	0.164

The tubes used in shell and heat exchangers are relatively smooth. Accordingly, in the present case, the values of correlation constants (K and m) are to be retrieved from column – 2 on smooth tubes. Double pipe heat exchangers employ industrial pipes which have a given degree of roughness on their inner surface. In the case of those exchangers therefore, the values of K and m are to be read from column – 3 on commercial pipes. It is also important to keep in mind that the conventional friction factor versus Reynolds number plots (Moody's plots) are not applicable here since those plots are for isothermal flow. The above correlation (149) is based on the graphical data reported by Sieder and Tate and reproduced by Kern [8].

If the above computed value of tubeside pressure drop happens to exceed the maximum permissible value, then computations are to be repeated after selecting a larger tube diameter, starting from **Step – 6**.

Step 14: Computation of shellside pressure drop

The shellside pressure drop is more difficult to estimate accurately. This is because, as discussed earlier, the flow of shellside fluid through the exchanger is too much tortuous, it executes both crossflow and countercurrent flow as well as parallel flow. For flow over a submerged object, the *form drag* comes into play, which is of higher magnitude than skin friction. Since the shellside fluid flows over the tube bundle, the frictional resistance includes form drag and it is more cumbersome to quantify.

For a reasonably reliable estimate of shellside pressure drop therefore, we first estimate the pressure drop for flow through ideal crossflow section, $(-\Delta P_S)\ (ideal)$ and that for flow through ideal baffle window section, $(-\Delta P_W)(ideal)$. The actual value of shellside ptressure drop is then computed by incorporating the correction factors, R_b , R_l and R_S which are similar to the correction factors, J_b , J_l , J_S used for the estimation of shellside heat transfer coefficient. Thus

$$(-\Delta P_S) = R_{Cm}(-\Delta P_S)(ideal) + (R_l N_b)(-\Delta P_W)(ideal) \quad (150)$$

where

R_{Cm} = combined correction factor

$$= (N_b - 1)(R_b R_l) + (2 R_b R_S)(1 + N_{CW}/N_C) \quad (151)$$

$(-\Delta P_S)(ideal)$ = pressure drop in an ideal crossflow section

$$= \left[2 f_S G_S{}^2 N_C\right] / (\rho_{fs}\, \varphi_S) \quad (152)$$

f_S = shellside friction factor $(-\Delta P_W)(ideal)$ = pressure drop in an ideal baffle window section

$$= [\, G_b\, G_m \,/\, 2\, \rho_{fs} \,]\, [\, 2 + 0.6\; N_{cw} \,] \qquad (153)$$

G_b = mass velocity of shellside fluid in baffle window

$$= (\, \dot{m}_S \,/\, a_b \,) \qquad (154)$$

a_b = free area for flow of shellside fluid in the baffle window

$$= f_b\, (\, \pi\, D_S^2 \,/\, 4\,) - N_{tb}\, (\, \pi\, D_0^2 \,/\, 4\,) \qquad (155)$$

f_b = fraction of the shell cross-sectional area occupied by the baffle. window

$$=(\, 1 \,/\, \pi\,)\, [\, (\, \theta \,/\, 2\,) - \cos(\,\theta/2\,)\sin(\,\theta/2\,)\,] \qquad (156)$$

θ = baffle cut angle (in ***radians***)

$$=2\, \cos^{-1}[\, (D_S - 2\, B_C)/\, D_S\,] \qquad (157)$$

N_{tb} = number of tubes in baffle window

$$= (\, N_t \,/\, 2\,)\, (\, 1 - F_C\,) \qquad (158)$$

F_C = fraction of total tubes in crossflow (defined earlier in equation – 105)

N_{CW} = number of effective crossflow rows in each baffle window

$$= (\, 0.8\, B_C \,/\, p_P\,) \qquad (159)$$

R_b, R_l, R_S = correction factor to account for bundle by passing effect, baffle leakages and unequal baffle spacing respectively on shellside pressure drop.

Bell and coworkers [9] have reported extensive graphical data for the estimation of these correction factors as well and Narayanan and Bhattacharya [2] have converted them into analytical correlations through rigorous regression analysis. The correlations developed by them are reproduced below:

$$R_l = \alpha_o + \alpha_1 (S_r) + \alpha_2 (S_r)^2 + \alpha_3 (S_r)^3 \qquad (160)$$

The values of correlation constants $\alpha_0, \alpha_1, \alpha_2, \alpha_3$ are listed in Table (11) which constitutes **Database – 11.**

Table 11: [Database – 11] Values of Correlation Constants for Computation of Correction Factor R_l (equation – 160)

	S_S	α_0	α_1	α_2	α_3
$S_r \leq 0.2$	0.0	0.995	– 4.94	26.952	– 58.77
	0.25	0.9947	– 6.651	40.5936	– 95.67
	0.50	0.9985	– 7.3934	37.7854	– 75.146
	0.75	0.993	– 9.3936	56.934	– 132.37
	1.0	0.995	– 11.256	71.358	– 170.295
$0.2 < S_r \leq 0.7$	0.0	0.7267	– 0.5737	0.0	0.0
	0.25	0.66	– 0.71	0.0	0.0
	0.50	0.5933	– 0.8476	0.0	0.0
	0.75	0.5133	– 0.9506	0.0	0.0
	1.0	0.4667	– 1.1476	0.0	0.0

The parameters S_r and S_S have been defined earlier (see equations – 108, 109). The correction factor R_l which takes care of the effect of tube to baffle and shell to baffle leakages on shellside pressure drop is thus analogous to factor J_l defined earlier under computation of shellside heat transfer coefficient. In a similar way, the correction factor R_b is similar to J_b and it takes care of the effect of bundle bypassing effect on shellside pressure drop. It may be computed from

$$R_b = [\, exp\,(-m_2 F_{bP})\,] \tag{161}$$

The value of correlation constant m_2 depends on the modified shellside Reynolds number, Re_{Sm} and the (N_{SS} / N_C) ratio and can be retrieved from Table (12). This table constitutes **Database – 12**. The dimensionless parameter F_{bP} has been defined earlier in equation (115). As evident from table (12), when the number of sealing strips installed is large such that the ratio (N_{SS} / N_C) is equal to or more than 0.5, $R_b = 1.0$.

Table 12: Values of Correlation Constant for Computation of Correction Factor R_b (equation – 161)[Database – 12]

N_{SS} / N_C	$Re_{Sm} \geq 100$ m_2	$Re_{Sm} < 100$ m_2
0.0	3.7041	4.3524
0.05	2.0245	2.4183
0.10	1.5270	1.8522
0.167	1.1684	1.30898
0.30	0.5944	0.72975
0.5 and above	0.0	0.0

The correction factor, R_S has been incorporated to account for the effect of unequal baffle spacing on shellside pressure drop. Evidently, its magnitude shall depend on the (B_{Si} / B_S) ratio and the (B_{So}/B_S) ratio, as shown below:

$$R_S = 0.5\,[(B_S/B_{Si})^{1.6} + (B_S/B_{So})^{1.6}] \tag{162}$$

The shellside friction factor f_S is a non-linear function of shellside Reynolds number, Re_S. It is also a function of the tubesheet layout chosen and the tube pitch (p_p, p_n). A reasonably satisfactory estimate of f_S can be obtained from the correlation proposed by Grimson [10]. It is given below:

For staggered tubes,

$$f_S = [\,0.25 + 0.118 / F2\,]\,(Re_S)^{-0.16} \tag{163}$$

where

$$\mathbf{F2} = [\,(2\,p_n - D_o) / D_o\,]^{1.08} \tag{164}$$

For tubes in line,

$$f_S = [\,0.044 + 0.08\,(p_p / D_o) / F1\,]\,Re_S^{-0.15} \tag{165}$$

where

$$F1 = [(p_n - D_o)/ D_o\,]^m \tag{166}$$

$$m = 0.43 + 1.13\,(D_o/p_p) \tag{167}$$

Grimson's correlation is valid within the Reynolds number range of 2000 $\leq Re_S \leq$ 40,000.

If the above computed value of shellside pressure drop happens to exceed the maximum permissible value, then a larger value of baffle spacing (B_S) is to be chosen and computations repeated starting from **Step – 8**.

Step 15: Print Results

The entire procedure described above has been illustrated in all details in the CAD flowsheet given in Figures – 7A to 7P.

It is needless to comment that the CAD package presented could very well be re-executed with different pass arrangements and with different choices of son file parameters and the most satisfactory design could be located from the results, keeping the heat transfer surface requirement, fabrication cost and the pressure drop penalties (both on tubeside as well as on the shellside) in mind. This, in fact, forms the inherent flexibility of all types of CAD (software) packages.

We shall illustrate a numerical example here to demonstrate the applicability of the above-described CAD package. The package is executed with the following Father File parameters:

Shellside fluid : Petroleum Oil (hot fluid), Tubeside fluid : Water (cold fluid)

$\dot{m}_S$ = mass flow rate of shellside fluid = 36300 kg / hr

Inlet temperature of shellside fluid = T_1 = 160°C

Outlet temperature of shellside fluid = T_2 = 45°C

Inlet temperature of tubeside fluid = t_1 = 20°C

Outlet temperature of tubeside fluid = t_2 = 42°C

R_d *(min)* = 0.0005 ($m^2. K$)/ W

$(-\Delta P_t)(max) = (-\Delta P_S)\,(max) = 60\ kPa$

The results obtained are,

Type of exchanger recommended: 1 – 2 heat exchanger

Mass flow rate of water = $\dot{m}$ = 104198.4 kg / hr

$U_D = U_i$ = 471.5 $W/ (m^2.K)$ [finalized by trial, from the prescribed range of 284 – 710 $W/ (m^2.K)$]

Heat transfer surface required = A_i = 111.08 m^2

Heat exchanger specifications : D_o = 19 mm, D_i = 17 mm, L_e = 5.0 m, tubesheet layout = triangular pitch (p_T = 25.4 mm, p_p = 22 mm, p_n = 12.7 mm).

Construction : Fixed tubesheet

Total number of tubes = N_t = 416

Shell ID = D_S = 590.8 mm

Baffle spacing = B_S = B_{Si} = B_{So} = 196.93 mm (finalized by trial)

Baffle cut = B_C = 147.7 mm (25 % cut segmental baffles)

Number of baffles = N_b = 44

Number sealing strips per crossflow section = N_{SS} = 2

Tubeside heat transfer coefficient = h_i = 3228.46 $W / (m^2. K)$

Shellside heat transfer coefficient = h_o = 700.92 $W / (m^2. K)$

R_d *(computed)* = 0.000532 $(m^2. K)/ W$

Tubeside pressure drop = $(-\Delta P_t)$ = 6.0 kPa

Shellside pressure drop = $(-\Delta P_s)$ = 9.0 kPa

2.2. CAD Package for Rating Problem

As explained earlier, in a rating problem, a heat exchanger of known specifications is available and we have to determine whether this exchanger is suitable for a specific purpose (for performing the specified duty). The overall design procedure is very similar to that involved in the sizing problem, except that we do not have to resort to any trial and error (iterative) computations here. The step by step procedure is summarized below:

Step 1: Specification of father file parameters

As discussed in Step – 1 of the sizing problem, in the father file, five among the six parameters such as the mass flow rate of shellside fluid ($\dot{m}_S$) and that of the tubeside fluid ($\dot{m}$), the four terminal temperatures (t_1, t_2, T_1, T_2) are specified. The sixth unknown parameter is then estimated from the heat balance shown in Step – 3. For example, let the unknown parameter be the mass flow rate of the tube side fluid ($\dot{m}$). This is then evaluated from the overall heat balance as shown in step – 3.

Being a rating problem, the heat exchanger specifications are available and these are also to be listed in the father file, such as number of tubeside passes (n_t), number of shelleside passes (n_S), Inner and outer outer diameter of tubes (D_i , D_o), Tubesheet layout (Triangular / Square / Rotated Square Pitch), Tube pitch (p_T , p_P ., p_n), Effective length of each tube (L_e), Number of tubes (N_t), Shell Diameter (D_S), Baffle spacing (B_S , B_{Si} , B_{So}), Baffle cut (B_C), Number of sealing strips installed per crossflow section (N_{SS}), Also to be specified are the maximum permissible pressure drop on the shellside, $(-\Delta P_S)\ (max)$ and that on the tubeside, $(-\Delta P_t)\ (max)$ and the minimum overall dirt factor prescribed, $R_d(min)$.

Step 2: Estimation of property values of process fluids

Estimate the property values of the tubeside fluid (C_P , ρ_f , μ_f , k_f) and those of the shellside fluid (C_{PS} , ρ_{fs} , μ_{fs} , k_{fs}) at the mean temperature t_m and T_m respectively, as discussed in Step – 2 of sizing problem.

Step 3: Overall heat balance

Determine the unknown parameter (here, $\dot{m}$) from the overall heat balance equation, as shown in Step – 3 of sizing problem. Compute also the overall rate of heat transfer (Q).

Step 4: Computation of tubeside heat transfer coefficient ($\boldsymbol{h_i}$)

Assuming the viscosity correction factor (φ) to be equal to unity, compute the tubeside heat transfer coefficient from available correlations such as from equation (89) or (92), as described in Step – 7 of sizing problem and denote it as h_i'.

Step 5: Computation of shellside heat transfer coefficient, $\boldsymbol{h_o}$

Assuming the shellside viscosity correction factor (φ_S) to be equal to unity, compute the shellside heat transfer coefficient as discussed in Steps – 8 and 9 of sizing problem and denote it as h_0' . The value of $h_o(ideal)$ is to be computed first from Colburn's correlation (equation – 94) and thereafter the correction factors ($J_C\, J_l\, J_b\, J_S$) incorporated to obtain the value of h_0' .

Step 6: Viscosity correction

Perform the viscosity correction as described in Step – 10 of sizing problem and estimate the actual value of tubeside heat transfer coefficient (h_i) and that of shellside heat transfer coefficient (h_o).

Step 7: Computation of overall heat transfer coefficient ($\boldsymbol{U_i}$ or $\boldsymbol{U_o}$)

Compute the overall heat transfer coefficient (U_i or U_o) as

$$(1 / U_i) = (1 / h_i) + (1 / h_o)(D_i / D_o) + R_d\,(\text{min}) \quad (168)$$

$$= (1 / U_{Ci}) + R_d\,(\text{min}) \quad (169)$$

$$(1 / U_o) = (1 / h_o) + (1 / h_i)(D_o / D_i) + R_d\,(\text{min}) \quad (170)$$

$$= (1 / U_{Co}) + R_d\,(\text{min}) \quad (171)$$

Step 8: Computation of Heat Transfer Surface

Compute the heat transfer surface required (A_i or A_o) using any of the three methods such as the F_T Method, ϵ - NTU Method or Martin's Method as described in Step – 5 of the sizing problem.

Step 9: Computation of required tube length

Compute the effective tube length required, $L_e(req)$ as given below

$$L_e(req) = A_i / (\pi D_i N_t) \tag{172}$$

$$= A_o / (\pi D_o N_t) \tag{173}$$

If the above-computed value of $L_e(req)$ exceeds the value of L_e specified in the father file (Step – 1), then print ***" the exchanger is not suitable for the purpose with respect to heat transfer surface requirement"***. Otherwise, proceed to Step – 10 for the computation of tubeside pressure drop.

Step 10: Computation of tubeside pressure drop

Compute the tubeside pressure drop $(-\Delta P_t)$, as discussed in Step – 13 of the sizing problem. If this value of $(-\Delta P_t)$ exceeds the maximum permissible value, $(-\Delta P_t)$ *(max)*, specified in the father file (Step – 1), then print, ***" the exchanger is not suitable for performing the given duty"***. Otherwise, proceed to Step – 11 for the computation of shellside pressure drop.

Step 11: Computation of shellside pressure drop

Compute the shellside pressure drop $(-\Delta P_S)$, as described in Step – 14 of the sizing problem. If this value of $(-\Delta P_S)$ is found to exceed the maximum permissible value, $(-\Delta P_S)(max)$, specified in the father file, then print, ***" the exchanger is not suitable for performing the given duty"***. Otherwise, proceed to Step – 12.

Step 12: Print: The given exchanger is suitable for performing the specified duty.

The readers are encouraged to prepare the detailed CAD flowsheet for the rating problem by themselves, as an interesting exercise.

3. Improved Design of Shell and Tube Heat Exchangers

Since shell and tube heat exchangers are quite popular in all process industries and power plants, attempts have been made by many authors to propose improved design of these exchangers. However, needless to comment, in many cases, though the heat transfer coefficient (and thereby the heat transfer efficiency of the exchanger) gets enhanced, there is simultaneous increase in the pressure drop penalty (and thereby in the operating cost) and consequently, the net benefit of employing the proposed design becomes marginal. In alternate cases, the modified design demands complex and expensive construction or expensive accessories. Examples are flow interception using corona discharge (expensive accessories), admitting process fluids through multiple jets (too high operating cost), insertion of twisted tapes inside tubes (too cumbersome when a tube bundle composed of 500 – 1000 tubes are used and fouling fluids are handled, net benefit marginal), installation of fins on tube surfaces (high manufacturing cost, simultaneous increase in pressure drop penalty tends to compensate higher heat transfer coefficient attained unless used for gases) etc.

A novel approach in this connection is the use of variable area construction for shell and tube heat exchangers [11, 12]. A Variable Area Exchanger (VAE) employs a bundle of diverging – converging tubes (periodically constricted tubes) instead of straight, cylindrical tubes, as shown schematically in figures (8) and (8A). Each tube is composed of a number of segments, each segment being made up of two frustums of cones joined base to base. .The tube diameter or cross – sectional area thus varies continuously along the length of the tube. If D_2 is the maximum diameter of each segment, D_1 the minimum diameter and L_S the segment length, then from simple geometry, the angle of divergence / convergence (θ) is predicted by

$$\tan(\theta) = (D_2 - D_1)/L_S \tag{174}$$

The optimum value of θ reported is 5^0 [11,13] or $\tan(\theta) = (1/12)$. The geometry of each tube thus deviates from straight, cylindrical geometry by only 5^0. If n is the total number of segments per tube, then the total effective length (L_e) of each tube shall be

$$L_e = (n\, L_S) \tag{175}$$

The specific advantages of using such a construction are,

1.They provide substantially large heat transfer coefficient (350 to 400% higher than, or 3.5 to 4.00 times, that in a conventional heat exchanger of same heat transfer surface per unit length) within a large range of flow rates (both in laminar flow and in turbulent flow), both under constant wall temperature conditions as well as constant wall heat flux conditions.

2. The simultaneous increase in pressure drop penalty has been, however, observed to be relatively negligible (only by 15 to 20 % or 1.15 to 1.2 times).

3. The performance efficiency of these exchangers is thus significantly high, but they do not demand any large scale increase in the operating cost. This has been found to be true while handling Newtonian fluids (water, aqueous solutions, petroleum oils) as well as while handling Non – Newtonian fluids such as suspensions and polymer solutions [14].

4. Since the shellside heat transfer coefficient in a variable area exchanger is substantially large, the shell of the exchanger need not have to be baffled. No doubt, a minimum number of baffles may still be installed, keeping the baffle spacing (B_S) at the maximum permissible value $(B_S = B_S\,(\max) = D_S)$, to act as support plates for tubes.. In the case of tubesheet layout, it is recommended that the tube hole diameter be kept equal to (or slightly more than) D_2

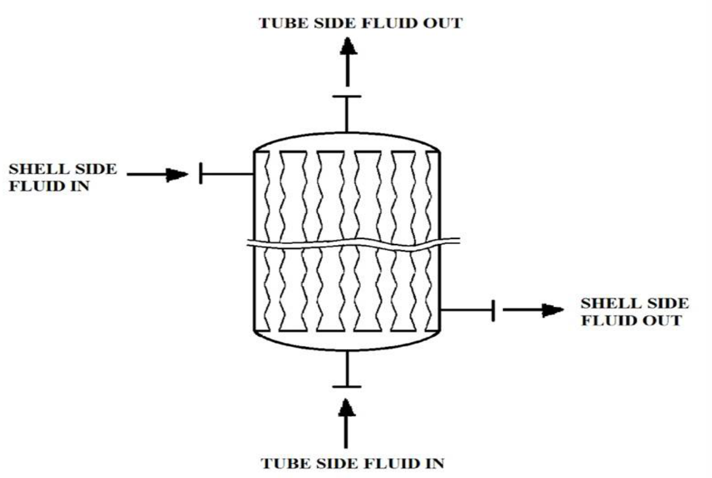

Figure 8: Schematic of Variable Area Shell and Tube Heat Exchanger (showing 1 – 1 construction)

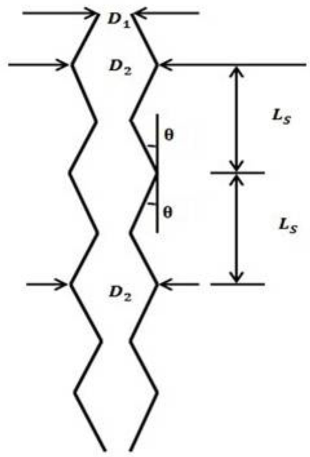

Figure 8A: Schematic of Diverging – Converging Geometry

(the maximum diameter of diverging - converging tube) so as to retain the flexibility of the construction.

5. An additional interesting feature of these exchangers is that they exhibit lower tendency to precipitation fouling. The tortuous wall geometry of the d-c (diverging – converging) tube induces a degree of turbulence into the flow field and this tends to dislodge the deposited dirt from the tube surface and gets it re-entrained into the flowing fluid. If fouling does occur, then cleaning of the tube surface could be accomplished using high pressure liquid jets or by using chemical solvents (chemical cleaning). Mechanical cleaning, no doubt, shall be relatively more troublesome in the present case.

6. Supplementary effects such as bundle bypassing and baffle leakages that tend to diminish the magnitude of shellside heat transfer coefficient shall not be significant in these exchangers. For example, the bundle bypassing effect would not be significant in the proposed design due to the fact that this bypass stream also tends to execute a tortuous flow owing to the diverging – converging nature of the tube wall geometry. The baffle leakage effects (leakage of shellside fluid through shell to baffle and tube to baffle clearances) will also not be predominant since

the shell is to be fitted with minimum number of baffles

The performance characteristics of variable area exchangers have been studied both mathematically as well as experimentally [11 – 14]. Rigorous mathematical models (software packages) have been developed which have been duly verified by comparing with extensive experimental data compiled both on laboratory scale and pilot plant scale.

This construction has been successful not only for the improved design of shell and tube heat exchangers, but also for the design of evaporators / condensers, solar flat plate collectors, solar parabolic trough concentrators (in which the absorber tube is made of variable area design) and also in the case of mass transfer equipment such as gas – liquid absorbers, membrane separation units and column reactors [12].

One of the major reasons for the attractive augmentation characteristics exhibited by these exchangers stems from the fact that the tortuous wall geometry of the d-c (diverging – converging) tube induces additional turbulence into the fluid stream and this increases the intimacy of contacting between the fluid elements. This is substantiated by the fact that the velocity profile in a d-c tube even at low Reynolds numbers ($Re \leq 1500$) has been observed to be flat within the central core of the tube and the velocity is seen to fall sharply to zero at the wall. Such flat velocity profile is obtained in straight cylindrical tubes only in fully developed turbulent flow (at $Re \geq 10000$). It is also to be kept in mind that the onset turbulence in a d-c tube occurs at a much lower Re.

Due to the improved radial mixing of fluid elements, the formation of any stagnant liquid film or thermal layer at the wall of d-c tube is either absent or even if formed, its thickness is quite low. This is evidenced by the nature of velocity and temperature profiles in these systems which exhibit a boundary layer character. The velocity of the fluid falls sharply to zero at the tube wall and the fluid temperature rises sharply in the close vicinity of the heated wall. Such destruction of stagnant layer at the wall reduces the resistance to momentum and heat transport and the transfer coefficient gets enhanced.

Due to the diverging – converging wall geometry of the tube, the flow direction of the fluid varies along the length of the tube (in the converging section, the fluid flows towards the tube axis, while in the diverging section, it flows towards the wall) and the average velocity of the fluid also varies from section to section. This could be causing a type of pressure recovery, like that in a venturi tube. This also helps in providing heat / mass transfer enhancement without the expense of much additional pressure drop.

The thermal penetration distance from the heated wall into the fluid bulk is much larger in the tubes of this geometry as is evident from the enhancement provided. This is in contrast to the assumption usually involved with straight cylindrical tubes (while developing heat transfer correlations) that the heat penetrates chiefly within a thin annular layer at the wall within which the velocity distribution may even be assumed linear.

The fabrication cost of these exchangers shall be, no doubt, higher. However, this increased initial investment could necessarily be recovered within 1 – 2 years since the exchanger operates with enhanced performance efficiency with relatively little increase in the operating cost.

This design has been successfully adapted to quite a few industries. More large scale industrial utilization of this design must be anticipated keeping in mind the attractive benefits / features of this construction.

4. Nomenclature

a_b = free area for flow of shellside fluid in the baffle window, *m^2*

a_S = minimum free flow area between baffles at the shell axis, *m^2*

a_t = tubeside flow area, *m^2*

A_i = inner heat transfer surface (inside surface area of tubes), *m^2*

A_m = crossflow area at or near center line, *m^2*

A_o = outer heat transfer surface (outer surface area of tubes), *m^2*

A_{Sb} = shell to baffle leakage area, *m^2*

A_{tb} = tube to baffle leakage area, *m^2*

B_S = baffle spacing (baffle pitch), *m*

B_{Si} = baffle spacing (baffle pitch) at shell inlet, *m*

B_{So} = baffle spacing (baffle pitch) at shell outlet, *m*

$C\,(\min)$ = smaller of $(\dot{m}C_P)$ and $(\dot{m}_S C_{PS})$, *W/K*

$C\,(\max)$ = larger of $(\dot{m}C_P)$ and $(\dot{m}_S C_{PS})$, *W/K*

C = $C\,(min)\,/\,C\,(max)$, dimensionless

C_P = specific heat of tubeside fluid, *J/(kg.K)*

C_{PS} = specific heat of shellside fluid, *J/(kg.K)*

D_i = inside diameter (ID) of tubes, *m*

D_o = outer diameter (OD) of tubes, *m*

D_{ot} = outer tube limit, *m*

D_S = inside diameter (ID) of shell, *m*

f = tubeside friction factor (for non-isothermal flow), dimensionless

f_S = shellside friction factor (for non-isothermal flow), dimensionless

F_{bP} = bundle bypass coefficient (equation – 115), dimensionless

F_C = fraction of total tubes in crossflow

F_{Cm} = caloric fraction, dimensionless

F_T = correction factor to LMTD for multipass construction, dimensionless

G_b = mass velocity of shellside fluid in baffle window, *kg/(m².s)*

G_e = geometric average mass velocity of shellside fluid (equation – 101), *kg/(m².s)*

G_S = mass velocity of shellside fluid based on flow area a_S , *kg/(m².s)*

G_m = mass velocity of shellside fluid based on flow area A_m, *kg/(m².s)*

G_t = mass velocity of tubeside fluid, *kg/(m².s)*

h_i = tubeside heat transfer coefficient, *W/(m².K)*

h'_i = value h_i when viscosity correction factor ($\varphi) = 1.0$, *W/(m².K)*

h_o = shellside heat transfer coefficient, *W/(m².K)*

h'_o = value h_o when viscosity correction factor ($\varphi_S) = 1.0$, *W/(m².K)*

h_o *(ideal)* = shellside heat transfer coefficient for an ideal crossflow section, *W/(m².K)*

J_C = correction factor that accounts for baffle configuration effect, dimensionless

J_l = correction factor to account for shell to baffle and tube to baffle leakages, dimensionless

J_b = correction factor that accounts for bundle bypassing effect, dimensioless

J_S = correction factor that accounts for unequal baffle spacing, dimensionless

k_f = thermal conductivity of tubeside fluid, *W/(m.K)*

k_{fS} = thermal conductivity of shellside fluid, *W/(m.K)*

K_C = parameter defined in equation (23), dimensionless

K_R = parameter defined in equation (44), dimensionless

K_R' = parameter defined in equation (17), dimensionless

K_S = parameter defined in equation (43), dimensionless

L_e = effective length of each tube, *m*

$\dot{m}$ = mass flow rate of tubeside fluid, *kg/s*

$\dot{m}_S$ = mass flow rate of shellside fluid, *kg/s*

n_S = number of shellside passes

n_t = number of tubeside passes

N_b = number of baffles used

N_C = number of tube rows crossed during flow through one crossflow section

N_{CW} = number of effective crossflow rows in each baffle window

N_{SS} = number of sealing strips installed per crossflow section

N_t = total number of tubes

$NTU\,(max)$ = number of transfer units (maximum)

Nu = tubeside Nusselt number, dimensionless

Nu_S = shellside Nusselt number, dimensionless

p_n = tube pitch normal to flow, *m*

p_P = tube pitch parallel to flow, *m*

p_T = tube pitch (overall), *m*

Pr = Prandtl number of tubeside fluid, dimensionless

Pr_S = Prandtl number of shellside fluid, dimensionless

Q = overall rate of heat transfer, W

R_b = correction factor for bundle bypassing effect on shellside pressure drop, dimensionless

R_l = correction factor for baffle leakages on shellside pressure drop, dimensionless

R_S = correction factor for unequal baffle spacing on shellside pressure drop, dimensionless

R_d = overall dirt factor, *(m²·K)/W*

$R_d(min)$ = minimum required value of overall dirt factor, *(m²·K)/W*

Re_S = shellside Reynolds number, dimensionless

Re'_S = shellside Reynolds number defined by Donohue (equation – 100), dimensionless

Re_{Sm} = modified shellside Reynolds number (equation – 116), dimensionless

Re_t = tubeside Reynolds number, dimensionless

S_r = dimensionless parameter defined in equation (108)

S_s = dimensionless parameter defined in equation (109)

t_1 = inlet temperature of tubeside fluid, K

t_2 = outlet temperature of tubeside fluid, K

T_1 = inlet temperature of shellside fluid, K

T_2 = outlet temperature of shellside fluid, K

t_{am} = arithmetic average temperature of tubeside fluid, K

T_{am} = arithmetic average temperature of shellside fluid, K

t_{cm} = caloric mean temperature of tubeside fluid, K

T_{cm} = caloric mean temperature of shellside fluid, K

t_m = mean temperature of tubeside fluid, K

T_m = mean temperature of shellside fluid, K

t_{Wi} = inner surface temperature of tubes, K

t_{Wo} = outer surface temperature of tubes, K

U_{Ci} = *clean* overall heat transfer coefficient based on A_i, *W/(m*^{2}*.K)*

U_{Co} = *clean* overall heat transfer coefficient based on A_o, *W/(m*^{2}*.K)*

U_D = overall design heat transfer coefficient, *W/(m*^{2}*.K)*

U_i = overall heat transfer coefficient based on A_i, *W/(m*^{2}*.K)*

U_o = overall heat transfer coefficient based on A_o, *W/(m*^{2}*.K)*

Greek Letters

δ_{tb} = tube to baffle clearance, *m*

δ_{Sb} = shell to baffle clearance, *m*

$(-\Delta P_t)$ = tubeside pressure drop, N/m^2

$(-\Delta P_r)$ = additional pressure drop due to flow reversal, N/m^2

$(-\Delta P_S)$ = shellside pressure drop, N/m^2

$(-\Delta P_S)(ideal)$ = pressure drop in ideal crossflow section, N/m^2

$(-\Delta P_W)(ideal)$ = pressure drop in ideal baffle window, N/m^2

Δt_C = temperature difference of cold fluid, K

Δt_h = temperature difference of hot fluid, K

$\Delta T(max)$ = maximum temperature difference, K

$(-\Delta T)_{ln}$ = logarithmic mean temperature difference, K

(ΔT_C) = temperature difference at the cold end of the heat transfer surface / heat exchanger

(ΔT_h) = temperature difference at the hot end of the heat transfer surface / heat exchanger

ϵ = heat exchanger effectiveness, dimensionless

θ = baffle cut angle (in ***radians***)

μ_f = viscosity of tubeside fluid at temperature t_m, *kg/(m.s)*

μ_{fS} = viscosity of shellside fluid at temperature T_m, *kg/(m.s)*

μ_{fw} = viscosity of tubeside fluid at the inside wall temperature (t_{Wi}) of tubes, *kg/(m.s)*

μ_{ws} = viscosity of shellside fluid at outer wall temperature (t_{wo}) of tubes, *kg/(m.s)*

ρ_f = density of tubeside fluid, *kg / m³*

ρ_{fs} = density of shellside fluid, *kg / m³*

φ = viscosity correction factor for tubeside fluid, dimensionless

φ_s = viscosity correction factor for shellside fluid, dimensionless

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