4.4. Preliminary Design Procedures

4.4.1. Principles of the Design Process

The essential feature of most design problems involving air coolers is that a certain thermal change must be made on the process stream, using air which can only undergo limited temperature and pressure changes.

The air-side heat transfer process is usually controlling in the heat removal process, and the limited pressure drop possible with the air restricts the values of velocities (and, therefore, heat transfer coefficients) to a very narrow range.

The first problem of design is to select the general features of the heat exchanger configuration. The second problem is to calculate whether or not the configuration selected will transfer the required amount of heat within the pressure drop limitations.

There are basically three possible outcomes to the second problem:

1. The configuration selected will transfer the heat, using (but not exceeding) the available pressure drops. This then represents the desired design.

2. The configuration will not transfer the heat specified unless the air-side pressure drop is exceeded. In this case, it is necessary to select a different (larger) design and return to the heat transfer and pressure drop calculations, until the first outcome above is achieved.

3. The configuration will transfer the heat specified, but does not use the pressure drop available. In this case, the exchanger is too large and can be reduced in size, saving money, until the first outcome is realized.

The above is a gross over-simplification of the design problem, but it illustrates the essentials of the process and the criteria by which success is measured.

This section will concentrate upon the first problem – the selection of a design whose major features are fairly close to the final design. In fact, for many purposes, such as plant capital cost estimates or preliminary plant layout, the first-cut design may be sufficient.

4.4.2. Selection of Preliminary Design Parameters

1. *Selection of Tube.* Finned surface tubes are mainly of interest when the fluid on the fin side has a much lower coefficient than on the tube-side. To a very rough approximation, the tube should be chosen so that

$$h_i A_I \approx h_o A_o \tag{4.16}$$

or in terms of the area ratios available,

$$\frac{Ao}{Ai} \approx \frac{h_i}{h_o} \tag{4.17}$$

Since, as we have seen, ho for typical air-cooler applications is about 10 Btu/hr ft^2°F, we may make a rough tube selection by choosing one with an A_o/A_i ratio approximately one-tenth the numerical value of hi when the latter is given in Btu/hr ft^2°F.

Thus, if experience had taught that hi was about 100 Btu/hr ft^2°F (typical of cooling a medium weight organic liquid in the lower turbulent flow regime), then one would consider choosing a tube with $A_o/A_i \approx 10$, of which there are many available.

However, many streams to be cooled (such as water or condensing steam) give an $h_i \geq 1000$ Btu/hr ft^2°F, and there simply are no tubes available that have the corresponding area ratios. In those cases, one simply selects one of the higher ratio tube configurations, taking into account material, availability, prices, and other less tangible considerations.

Process stream considerations do affect the choice of tube also. For low flow rates, and for liquids generally, the smaller diameter tubes are generally preferable so that tube-side velocities can be kept up to ensure that the flow is in the turbulent regime and that fouling is minimized. Turbulent flow is generally preferred, partly because of the better heat transfer coefficient, but also to reduce the possibility of a flow maldistribution among the tubes.

For high tube-side flow rates, or for gases and condensing vapors, or where tube-side pressure drop is limited, larger diameter tubes are generally preferable. No absolute rules can be written - each case must be considered on its own merits in terms of operability and cost.

2. *Selection of a Tube Layout*. It has been emphasized above that staggered tube layouts - usually equilateral triangular, less commonly other triangular or rotated square - must be used to minimize bypassing in tube banks. Within this limit, however, and for a given tube, there is the question of what pitch to use. There must be some minimum clearance - on the order of 3/16 to ¼ inch - between fin tips to prevent fin-to-fin impact (with noise and mechanical damage resulting) during operation. Larger clearances are possible and perhaps desirable, since pressure drop decreases more rapidly than heat transfer coefficient as the pitch increases and the velocity decreases. Inevitably, however, the result is to increase the size and cost of the exchanger. Therefore, the usual practice is to put the tubes as close together as possible.

3. *Selection of Design Air Temperatures*. The selection of the inlet and exit air temperatures are both matters of concern to the designer, though the considerations in their respective selection are quite different.

The inlet air temperature at any given moment at a given location is set by nature, but there is usually a substantial range of air temperatures over the course of a year or even a day. For most plant locations, the data are available to plot the percentage of hours in the year when the air temperature will exceed a given value. The designer (or usually, the process engineer) must then decide which temperature to choose in terms of the fraction of time that the heat exchanger will be nominally under-designed and incapable of handling the design heat load at the process stream temperatures specified.

Thus, the designer may elect the 5 percent level – that air temperature that is exceeded only 5 percent of the hours of the year. In principle, this means the exchanger will fail to meet demand 5 percent of the year and will be over-designed the other 95 percent. In practice, it is not nearly that simple. Uncertainties at many other points in the design process, including heat transfer coefficients, process stream conditions, changes in the operation of the plant, fouling transients, etc., plus the provision of process control and flexibility in other parts of the process greatly affect even the criterion of what constitutes failure to meet process requirements.

Therefore the selection of an inlet temperature loses much of the central importance that has been assigned to it in some past discussions, and the major concern is to select one near but below the maximum air temperature likely to be encountered. Then the exchanger is designed with an eye upon flexibility to application and operation in the particular circumstances of the given problem.

It should be noted that problems are as likely to arise from the fact that the air temperature is colder than the design value most of the time. Thus the process stream may be overcooled, leading to freeze-up or, in a reflux condenser, overloading a column or its reboiler. These problems should be anticipated by the process engineer, rather than the exchanger designer, but someone needs to make provision for controlling the air flow rate or other variables to avoid the worst consequences.

More often than not, choice of the design inlet air temperature fixes the minimum process fluid exit temperature. This is because it is not generally economically justified to cool the tube-side fluid to a temperature lower than about 20°F hotter than the inlet air; stated another way, the temperature approach at the cold end is generally chosen to be at least 20°F. However, approaches as low as 10°F have been specified, leading to a substantial increase in area. Larger approaches can be used, of course, if there is no need to cool the process fluid so far.

The exit air temperature must also be chosen by the designer, but the considerations limiting this choice are set by the process rather than the climate. That is, the exit air temperature must always be less than the inlet temperature of the process stream. If there are several tube-side passes and if the other assumptions underlying the validity and the logarithmic mean temperature difference derivation are satisfied, then in theory one can obtain a workable design for any exit air temperature less than the inlet process fluid temperature. In practice, this is never pushed to the limit, first, because there are too many things that can cause the theoretical model to be violated, and second, because it simply leads to an excessively large and costly exchanger.

The approach temperature limit sometimes occurs between the exit air and the inlet process fluid. In that case, the approach is rarely less than 20'°F and more commonly 40'°F.

Where only one or two tube-side passes are involved, it is important to calculate F, the MTD correction factor, for the temperatures chosen and make sure that it has a good value ($F > 0.8$) and that it is not on or near the steep portion of the curve. If these conditions are not satisfied, it is necessary to back off on the temperatures specified and pay the penalty in process efficiency.

If the assumptions underlying the F-LMTD derivation are not satisfied, then a much more careful analysis of the approach temperature selection and the MTD evaluation are required.

4.4.3. Fundamental Limitations Controlling Air-Cooled Heat Exchanger Design

1. *The Nature of the Problem*. There are two or three aspects of air-cooled heat exchanger design which are in some sense in competition. On the one hand, there is the limited capacity of the air to absorb heat; this we may term the "thermodynamic limitation." On the other hand, there is the limited rate at which heat can be transferred to the air; this we may term the "rate limitation."

The thermodynamic limitation calls for moving very large quantities of air across the exchanger, accepting the maximum possible change in the air temperature. Given the low pressure drops acceptable, however, this large air flow must be accommodated by using a large face area and shallow depth in the exchanger.

The "rate limitation" requires that the temperature difference between the streams be kept as great as possible (which implies small temperature changes in the air) and that the velocities be kept as high as possible, again within the fan capabilities.

Both limitations have in common the desire to use high air velocities to overcome them, and the mutual limit upon this we could call the "pumping limitation."

In every problem, these three limitations must be balanced, but the point of balance is very dependent upon the particular features of each problem. In this section, we present the common design standards that have evolved and deduce from them a procedure for selecting a preliminary design.

2. *The "Pumping Limitation. "* The fans in use on air cooled heat exchangers give a maximum practical pressure drop of one inch of water (5.20 lb_f/ft^2 = 0.0361 lb_f/in^2). However, the usual design range is from 0.3 inches of water to 0.7 inches, with 0.5 inches being a convenient design target. This converts very approximately into two somewhat more convenient design quantities.

The first is in terms of the mass velocity passing through the minimum flow area, defined in terms already used a, $\rho_{air}V_{max}$,

Typical values of $\rho_{air}V_{max}$ as a function of the number of rows of tubes are shown in Table 4. 1.

The second design quantity commonly cited is the face velocity, the average air velocity approaching the face of the tube bank, which as a function of the number of rows is given in Table 4.2.

Table 4.1
Typical Mass Velocities for Air-Cooler Design

n, No. of Rows of Tubes	$\rho_{air}V_{max}$ $lb_m/hr\ ft^2$
3	5000-6000
4	5000
5	4500
6	4000
8	3500

Table 4.2
Typical Face Velocities for Air-Cooler Design

n, No. of Rows of Tubes	V_{face} ft/min
3	900
4	800
5	700
6	600
8	500

These are, of course, only representative values, and the comparability of these values between the two tables is only approximate. They are, however, very useful for preliminary estimates.

3. The *"Thermodynamic Limitation. "* The thermodynamic limitation is nothing more than a heat balance and thus exists in all heat exchangers. But it is particularly critical in air-cooled exchangers because of the low mass rate at which air may be blown across the tube bank. Thus, if the total duty of the exchanger is Q Btu/hr, and if the air inlet and exit temperatures are t_i and t_o, respectively, then the mass flow rate of air required is

$$w_{air} = \frac{Q}{c_{p,air}(t_o - t_i)} \qquad (4.18)$$

For all practical purposes, the value of $c_{p,air}$ may be taken as constant at 0.24 Btu/lb°F.

The mass flow rate of air may be related to the face velocity V_{face} and face area A_{face} by

$$A_{face} = \frac{w_{air}}{\rho_{air} V_{face}} \tag{4.19}$$

where ρ_{air} is evaluated at air inlet temperature. Care must be taken of course to keep the units consistent.

But as noted in the discussion of the pumping limitation, there is a fairly narrow range of values of V_{face} that can be provided in an air-cooled exchanger, and this is inversely related to the number of tube rows. By combining Eqs. (4.18) and (4.19).

$$\frac{(A_{face})_T}{Q} = \frac{1}{c_{p,air}(t_o - t_i)\rho_{air} V_{face}} \tag{4.20}$$

where the notation $(A_{face})_T$ indicates that this is the face area required by the "thermodynamic limitation". From Table 4.2 we may find typical design values for V_{face} as a function of n from the pumping limitation. Substituting these values into Eq. (4.20), noting the ft/min units used on V_{face}, allows us to calculate the face area required per unit of heat to be transferred as a function of n given the values of t_i and t_o.

Example: Assume that 100,000 lb/hr of water is to be cooled from 180°F to 120°F, using air available at 90°F (ρ_{air} = 0.0737 lb_m/ft^3). If the air is heated to 140°F, what face area is required on a thermodynamic basis as a function of the number of tube rows, using the values from Table 4.2:

Solution: From Eq. (4.20),

$$\frac{(A_{face})_T}{Q} = \frac{1}{\left(0.24\frac{Btu}{lb_m{}^\circ F}\right)(140-90)^\circ F\left(0.0737\frac{lb_m}{ft^3}\right)\left(V_{face}\frac{ft}{\min}\right)\left(60\frac{\min}{hr}\right)} = \left[\frac{0.01885}{V_{face\,ft/\min}}\right]\frac{ft^2}{Btu/hr}$$

Since Q = (100,000 lb_m/hr)(1 Btu/(lb_m°F))(180 – 120) °F = 6×10^6 Btu/hr, we may also calculate the face area required, the results are:

n	V_{face}, ft/min	$(A_{face})_T$ ft² / Q Btu/hr	$(A_{face})_T$ ft²
3	900	2.09×10^{-5}	126
4	800	2.36×10^{-5}	141
5	700	2.69×10^{-5}	162
6	600	3.14×10^{-5}	189
8	500	3.77×10^{-5}	226

Alternatively, had we specified a tube and tube layout we could have carried out a similar set of calculations based upon Table 4. 1. This would probably be a better criterion to use for actual design, but it is not quite so convenient as the procedure given.

Before developing the implications of the thermodynamic limitation further, let us take a look at the heat transfer rate limitation.

4. *The "Rate Limitation.* " The basic equation for the rate of heat transfer is:

$$A_o = \frac{Q}{U_o(MTD)} \tag{4.21}$$

where A_o and U_o must be on the same area basis, usually the total outside area of the finned tube. If reasonable estimates of U_o and MTD can be made quickly, the A_o is easily found. This heat transfer area can be put on a comparable basis with the "Thermodynamic Limitation" if it is translated into the face area required by the equation:

$$\left(A_{face}\right)_{HT} = \frac{A_o}{n\ A^*_{HT}} \tag{4.22}$$

where n is the number of tube rows, and A^*_{HT} is the finned tube heat transfer area per square foot of face area and per row. A^*_{HT} must be determined for each choice of tube and layout; for the case calculated in the previous example,

$$A_{fin}\ /\ ft\ of\ length = \frac{\pi}{4}\left[(1.625\,\text{in.})^2 - (0.75\ in.)^2\right]\left(\frac{2\ sides\ /\ fin}{144\ in.^2\ /\ ft^2}\right)\left(9\ \frac{fins}{in.}\right)\left(12\ \frac{in.}{ft}\right) = 2.45\ ft^2\ /\ ft$$

$$A_{root}\ /\ ft\ of\ length = \pi\left(\frac{0.75\ in.}{12\ in./\ ft}\right)\left[1\,ft - \left(9\ \frac{fins}{in.}\right)\left(12\ \frac{in.}{ft}\right)(0.019\ in.)\left(\frac{1\,ft}{12in.}\right)\right] = 0.163\ ft^2\ /\ ft$$

Total heat transfer area per foot of tube = 2.61 ft^2

With a 1.875 in. pitch, each foot of width of tube bank has $\frac{12\ in.}{1.875\ in.} = 6.40$ tubes per row, so

A^*_{HT} = (6.40 tubes/ ft-row)(2.61 ft^2/ft) = 16.70 ft^2/ft^2 of face, per row of tubes

By combining Eqs. (4.21) and (4.22) we may obtain the equation for the face area required by the rate limitation:

$$\frac{\left(A_{face}\right)_{HT}}{Q} = \frac{1}{n\ A^*_{HT} U_o (MTD)} \tag{4.23}$$

A necessary condition of a final design is that the left hand sides of Eqs. (4.20) and (4.23) be equal. It is immediately observable that, other things being equal, $\frac{(A_{face})_T}{Q}$ increases with increasing number of rows of tubes n, whereas $\frac{(A_{face})_{HT}}{Q}$ decreases with n increasing. Therefore, there is some cross-over point for any given design problem, and if this point can be approximately located by rapid calculations, the main features of the preliminary configuration can be set. In order to do that, it is necessary to estimate values of U_o and MTD.

As noted previously, the overall heat transfer coefficient is largely controlled in air-cooled exchangers by the air side coefficient. In the example given, the air-side coefficient was found to be 10.9 Btu/hr ft^2°F at a face velocity of 600 ft/min and a maximum mass velocity of (1150 ft/min)(60 min/hr)(0.0765 lb$_m$/ft) = 5280 lb$_m$/ft^2 hr. Corresponding values of h. would be 14.4 Btu/hr ft^2 °F at a face velocity of 900 ft/min and 8.3 Btu/hr ft^2 °F at a face velocity of 400 ft/min.

These values may be compared to typical values for the other terms given in Eq. (4.5) as follows:

		Btu/hr ft^2°F
h_i, Based on inside tube area	Viscous liquid	50
	High pressure gas	75
	Medium liquids	150
	Light liquids	250
	Water	1200
	Condensing organic vapors	300
	Condensing steam	2000
$1/R_{fi}$, Based on inside tube area	Heavy fouling	100
	Moderate fouling	500
	Light fouling	2000
$\frac{k_w}{\Delta x_w}$	(Stainless steel liner, based on liner area)	2000
$\frac{k_w}{\Delta x_w}$	(Aluminum tube, based on root area)	15,000
$1/R_c$	(Maximum contact resistance, based on contact area)	300
$1/R_{fin}$	(Maximum resistance for aluminum fins, based on fin area)	125

From these values, we can see that the air-side resistance can vary from almost 100 percent of the total resistance (for, e.g., condensing steam) to about half of the total (for a viscous liquid, assuming that the tube has been chosen following the $h_oA_o \approx h_iA_i$ criterion). Thus, for preliminary calculations, it is a reasonable approximation to estimate U_o = 10 Btu/hr ft^2°F, based on total outside finned tube area, shading this to values as low as 5 Btu/hr ft^2°F where low air velocities and/or low intube heat transfer coefficients are involved. Alternatively, for high tube-side coefficients and high air velocities (corresponding to shallow tube banks), the overall coefficient may be estimated as high as 12 Btu/hr ft^2°F, based on total outside tube area.

To carry out the rate calculations, it is also necessary to have an estimate of the mean temperature difference. For preliminary estimation purposes, in cases where very close temperature approaches are not contemplated, it is often sufficient to use the arithmetic mean temperature difference:

$$\text{AMTD} = \tfrac{1}{2}\,[(T_i - t_o) + (T_o - t_i])] \qquad (4.24)$$

The AMTD is always equal to or greater than the LMTD, the difference depending upon the ratio $(T_i - t_o)/(To - t_i)$. When this ratio is close to unity, AMTD ≈ LMTD; as the ratio departs further from unity, the discrepancy between AMTD and LMTD becomes greater. Additionally, for one or two tube-side passes, the configuration correction factor F must be used to convert the LMTD to the MTD. For all practical air-cooler designs,*$1.0 \le F \le 0.8$, so a value of F = 0.9 is a good estimate.

*If F < 0.8 for a given problem, using Figs.. (4.13) or (4.14), it is probably necessary to change design temperatures or number of tube-side passes to ensure a good design.

Example. Continue using the previous example, cooling water from 180°F to 120°F using air available at 90°F. Assume also that Wolverine H/R Trufln with d_r = 3/4 in. and d_o = 1 5/8 in., on a 1 7/8 in. equilateral triangular pitch is used; thus A^*_{HT} = 16.70 ft^2/ft^2 of face area per row.

If we start by estimating an exit air temperature of 140°F as before, the AMTD is quickly found to be:

$$\text{AMTD} = \tfrac{1}{2}\ [(180 - 140) + (120 - 90)] = 35°\text{F}$$

The LMTD is 34.8°F. We may also wish to check F at this point:

$$P = \frac{140 - 90}{180 - 90} = 0.556$$

$$R = \frac{180 - 120}{140 - 90} = 1.20$$

and from Fig. 4.13 or 4.14 we get for F a value of about 0.82 if there is one tube-side pass and about 0.91 for two tube-side passes.

We may now set up a table for $\left(\frac{A_{face}}{Q}\right)_{HT}$ similar to that for $\left(\frac{A_{face}}{Q}\right)_T$ in the previous section:

Comparison of this table with the previous one indicates that the two closely correspond at the point of an air cooler with six rows of tubes and a face area of 190 ft^2 or in round numbers, a unit 20 feet long and 10 feet wide. The expectation that the fan requirements are probably within design range is due to the fact that we have used typical face velocities. The total heat transfer area (including fins) required is (6 rows)(10 feet wide)$\left(\frac{12\ in./\ ft}{1.875\ in.\ pitch}\right)$(20 feet long)(2.61 ft^2/ft of tube) = 20,000 ft^2, or a total of 7680 ft of tube.

The next step is to verify the heat transfer coefficients and pressure drop in each side, and verify that two tube rows/pass are necessary and sufficient to maintain good tube-side velocity. From the air-side pressure drop calculation, a fan and driver specification may be obtained. There will almost certainly be some modifications in the approximate design obtained here, but this gives the designer a good place to start.

n	No. of Tube Side Passes	F	MTD °F	V_{face} ft/min	U_o	$(A_{face})_{HT}$ ft^2 / Q Btu/hr	$(A_{face})_{HT}$ ft^2
3	1	0.82	28.5	900	12	5.84×10^{-5}	350
4	2	0.91	31.7	800	11	4.29×10^{-5}	258
5	2	0.91	31.7	700	10	3.78×10^{-5}	227
6	3	1.00	34.8	600	9	3.19×10^{-5}	191
8	4	1.00	34.8	500	8	2.69×10^{-5}	161