Air Cooled Condensers

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An air cooled exchanger is used to cool fluids with ambient air. Several articles have been published describing in detail their application and economic analysis. This paper describes the general design of air cooled exchangers and presents a method of approximate sizing.

Arrangement and Mechanical Design

The basic components are one or more tube bundles served by one or more axial flow fans, fan drives, speed reducers, and an enclosure with supporting structure.

Air cooled exchangers are classed as forced draft when the tube bundle is located on the discharge side of the fan, and as induced draft when the tube bundle is located on the suction side of the fan. Coolers are normally manufactured in tube lengths from 2 M. to 15 M and in bay widths from 1 M. to 10 M. Use of longer tubes usually results in a less costly design compared to using shorter tubes.

Fan size range is from 1 M. to 10 M. diameter, however, 5 M. diameter is the maximum normally used. Fan drives may be electric motors, steam turbines, hydraulic motors, or 1´℃ engines. Usually a speed reducer, such as a V -belt drive or reduction gear box, is necessary to match the driver output speed to the relatively slow speed of the axial flow fan. Fan tip speeds are normally 70 meters per second. The general practice is to use V-belt drives up to about 30 bhp and gear drives at higher power. Individual drive size is usually limited to 50 hp.

Two fan bays are popular, since this provides a degree of safety factor against fan or driver failure and also a method of control by fan staging. Fan coverage is the ratio of the projected area of the fan to the face area of bundle or bundles served by the fan. Good practice is to keep this ratio above 0.40 whenever possible because higher ratios improve air distribution across the face of the tube bundle. Face area is the plan area of the heat transfer surface available to air flow at the face of the bundle or bundles.

The heat transfer device is the tube bundle which is an assembly of side frames, tube supports, headers and fintubes. Aluminium fins are normally used with the tubes to provide an extended surface on the air side, in order to compensate for the relatively low heat transfer coefficient of the air to the tube. Fin construction types are tension-wrapped, embedded, extruded or welded. The application of each type is a matter of agreement between manufacturers, contractors and users, depending on the temperatures and other conditions of the service.

Base tube diameters are 5/8" to 1 1/2 in. OD with fins from 1/2 in. to 1 in high, spaced from 7 to 11 fins per inch providing an extended finned surface of 12 to 25 times the outside surface of the base tubing. Tubes are usually arranged on triangular pitch with the fin tips of adjacent tubes touching or separated from each other by about 1/16 in. to 1/4 in. Matching of the tube bundle to the fan system and the heat transfer requirements usually results in the bundle having depth of 3 to 8 rows of fintubes, with 4 rows the most typical.

One inch OD tube is the most popular diameter and the most common fins are 1/2 in. or 5/8 in. high.

Air Side Control

Air-cooled exchangers are sized to operate at warm (summer) air temperatures. Seasonal variation of the air temperature can result in over cooling which is not a problem for Indian conditions. One way to control the amount of cooling is by varying the amount of air flowing through the tube bundle. This can be accomplished by using multiple motors, 2-speed motors, hydraulic variable speed drives, louvers on the face of the tube bundle, or automatically variable pitch fans.

Staging of fans or fan speeds may be adequate for systems which do not require precise control of process temperature or pressure. Louvers will provide a full range of air quantity control. They may be operated manually, or automatically operated by a pneumatic or electric motor controlled by a remote temperature or pressure sensor on the process stream. Louvers used with constant speed fans do not reduce fan power requirements with reduction in the load.

Auto variable pitch fans are normally provided with pneumatically operated blade pitch adjustment which may be controlled from a remote sensor. Blade pitch is adjusted to provide the required amount of air flow to maintain the process temperature or pressure at the cooler. The required blade angle decreases as ambient air temperature drops and this conserves fan power. Hydraulic variable speed drives reduce fan speed when less air flow is required and can also conserve fan power.

Thermal Design

The basic equation to be satisfied is the same as adopted for shell & tube exchangers, i.e., Q = UA DTm

Where Q is the hear transfer duty usually in Kcal/hr. and U is the overall heat transfer coefficient usually in Kcal/m2hoC and A is the effective heat transfer surface area in square meter and Tm is the mean temperature differential between the heat losing medium and air.

Normally Q is known, U and DTm are calculated, and the equation is solved for A. The ambient air temperature to be used will either be known from available plant data or can be selected from the summer dry bulb temperature data.

The design procedure starts with a step for approximating the air temperature rise. After the air outlet temperature has been determined, the corrected log mean temperature difference is calculated in the normal manner. The determination of the LMTD involves use of correction factors for the cross flow heat exchange which have been developed and available in most of the standard books on heat transfer.

The procedure for the thermal design of an air cooler consists of assuming a selection and then proving it to be correct. Typical overall heat transfer coefficients are used to approximate the heat transfer area required. The heat transfer area is converted to a bundle face area by correlating with the amount of extended surface available per square foot of bundle face area for two specific fintubes on two different tube pitches for 3,4,5 and 6 rows. Both the tube side and air side mass velocities are now determinable.

The tube side film coefficient for condensing turbine exhaust steam is calculated for standard condensing heat transfer conditions. Similarly, the air-side film coefficient based on outside extended surface, it is necessary to multiply the reciprocal of the tube-side coefficient and tube-side fouling factor by the ratio of the outside surface to inside surface. This results in an overall transfer rate based on extended surface, designed as Ux.

The basic equation will then yield a heat transfer area in extended surface, Ax and becomes:
Q = (Ux) (Ax) (DTm)
This method is used extensively by thermal design engineers. Fig.3 gives the typical overall heat transfer coefficients based on extended surface.
Fig.3 Typical overall heat transfer coefficients for air coolers (B.t.u/sq.foot/oF)

Service	One inch O.D.1/2 in by 9		fins/inch Fintube 5/8 in by 10 fins/inch	
1. Water & Water solutions	Ub	Ux	Ub	Ux
Engine jacket Water(rf = .001)	110	7.5	130	6.1
Process water (rf = .002)	95		110	5.2
-50 Ethyl glycol water (rf = .001)	5090	6.2	105	4.9
50-50 Ethyl. glycol water (rf = .002)	80	5.5	95	4.4

2. Hydrocarbon liquid coolers

Viscosity Cp	Ub	Ux	Ub	Ux
0.2	85	5.9	100	4.7
0.5	75	5.2	90	4.2
1.0	65	4.5	75	3.5
2.5	45	3.1	55	2.6
4.0	30	2.1	35	1.6
6.0	20	1.4	25	1.2
10.0	10	0.7	13	0.6

3. Hydrocarbon gas coolers

Temperature oF	Ub	Ux	Ub	Ux
50	30	2.1	35	1.6
100	35	2.4	40	1.9
300	45	3.1	55	2.6
500	55	3.8	65	3.0
750	65	4.5	75	3.5
1000	75	5.2	90	4.2

4. Air and flue gas coolers

Use one half of value given for hydrocarbon gas coolers.

5. Steam condensers

(Atmospheric pressure & above)

***	Ub	Ux	Ub	Ux
Pure steam (rf = .0005)	125	8.6	145	6.8
Steam with non-condensables	60	4.1	70	3.3

6. HC condensers

Pressure Psig	Ub	Ux	Ub	Ux
0o range	85	5.9	100	4.7
10o range	80	5.5	95	4.4
25o range	75	5.2	90	4.2
60o range	65	4.5	75	3.5
100o & over range	60	4.1	70	3.3

7. Other condensers

	Ub	Ux	Ub	Ux
Ammonia	110	7.6	130	6.1
Freon 12	65	4.5	75	3.5

Note: Ub is overall rate based on bare tube area and Ux is overall rate based on extended surface.
The minimum fan area is calculated using the bundle face area, number of fans, and a minimum fan coverage of 0.40. The calculated area is then converted to a diameter and rounded upto the next available fan size. The airside pressure drop is then calculated, and the fan total pressure duty is estimated. Finally, the fan horsepower is calculated assuming a fan efficiency of 70% , and driver horsepower is estimated by assuming a 92% efficiency for speed reducer.

Conclusion

Air cooled condensers are ideal choice in arid/semi arid locations and at places where there is shortage of fresh water supply. Since the steam will be condensed at higher pressures (for Indian conditions, the condensing pressure can be between 0.6 and 0.3 ata) the NPSH requirements of condensate extraction pumps will also be less severe. The only disadvantage is that the power developed by the turbine would be slightly less because of the higher exhaust pressure.