Air Cooled Condenser: A Future Need

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Abstract

In a Power plant efficiency of the plant is directly affected by the efficiency with which steam is condensed and cooled. Here it is discussed about the basic types of cooling systems utilized by utility power plants, and explain the reasons why it is advantageous to include a cooling tower in many dry cooling applications.

The shortage of water in remote areas where rich coal fields often exist prompts the use of direct dry cooled condensers for condensing steam from a power station steam turbine. A natural or a forced draft condenser system can be used to achieve direct dry cooling. Dry steam condensing systems offer major advantages compared to conventional wet cooling systems of no blow down, no make up water consumption, no visible plumb, reduced noise level and no special siting requirements. Ambient conditions such as temperature and wind speed have a major effect on the performance of such an ACC and in some instances can be detrimental to its reliability and availability. Due to the enormous size of the structure involved and the dynamics of the prevailing ambient conditions, the performance optimization for such a system is extremely difficult.

A system where a cooling tower is used in conjunction with an air cooled steam condenser is called a parallel condensing system. This type of system utilizes three traditional types of heat exchangers: a cooling tower, an air cooled steam condenser and a surface condenser. An optimized parallel condensing system reduces both investment costs and operational costs while using a minimum amount of water.

The Parallel Condensing System is the successful combination of two proven subsystems that operate in parallel to reduce plume and increase heat rejection capabilities.

Direct-type air-cooled condensers are designed both as single-row and multi-row systems. Dead-zone formation, an undesirable characteristic of multi-row condensers, is eliminated by adjusting the fin pitch of each row.

The Single Row Condenser (SRC) has been developed to improve performance and efficiency. The design features virtually 100% effective finned surface, while minimizing airside pressure drop. The large cross-section of the tube results in minimum inside pressure drop and therefore in high performance with a very low subcooling. It also allows a higher steam velocity in the secondary tubes without restricting the down flow of condensate, thereby allowing the owner to operate at lower backpressure at freezing conditions.

Two concepts for improving the heat transfer performance of the air-cooled condensers used in binary geothermal power plants are being developed and tested at the INEEL. These concepts involve (1) replacing the circular tubes with oval tubes and (2) adding strategically located vortex generators (winglets) in the fins. These concepts can be used individually or in unison. Depending on the various design parameters, the heat transfer coefficient can be enhanced by 25–35%, with a minimal increase in pressure drop.

The criteria as well as the methods and measurements of wind tunnel simulation on wind effects on air-cooled condensers in a power plant were discussed. The parameter of recirculation was suggested to describe the wind effects on the efficiency of the condenser. The result of practical project models shows that great wind effects of both wind speed and the angle of the incident flow on the efficiency of the condenser.

A PCS system is a synergy of established cooling system technologies and combines some positive features of dry and wet cooling systems; the water consumption is reduced compared to a 100 % wet system, the performance is improved compared to a 100 % dry system and the capital cost decreases as the proportion of wet in the PCS system is increased.

1. INTRODUCTION :

In the steam cycle of a power plant, low-pressure water condensed in the steam condenser is pumped to high pressure before it enters the boiler or Heat Recovery Steam Generator (HRSG) where superheated steam is produced. The superheated steam is sent to the steam turbine where the steam expands to low pressure providing the energy to drive a generator. This low-pressure steam has to be condensed in a condenser in order to complete the steam cycle.
The condensation of steam requires a cooling medium. Traditionally, this has been achieved using water from a river, a stream, a pond or seawater. The cold water is pumped through a heat exchanger and the warm water is discharged back to the water source. This is called ONCE THROUGH cooling system.
A once through system is an open loop system. The need to reduce the vast amount of water requires a closed loop system. Thus the WET COOLING system came into effect, and soon after the DRY COOLING and HYBRID COOLING systems.
Table 1.1 : Evolution of cooling systems used in power plants

Cooling System	Time period
Once Through	From 1930s
Wet Cooling	From 1950s
Dry	From 1970s
Hybrid	From 1980s
Parallel	From 1990s

1.1 SELECTION CRITERIA BETWEEN DRY COOLING SYSTEMS AND WET TOWERS:

Since a wet tower has a lower capital cost and has a better performance in hot weather, it will be the best choice if sufficient water is available at reasonable cost. But even if enough water is available, some other factors may play a role as well. At times of high humidity and cool air temperature, a wet cooling tower is likely to produce a plume which is a visible fog exiting the tower.
Dry cooling saves a lot of water but there is a price to pay for it; the capital cost is significantly greater and there may be plant limitations on the hottest days. Also the heat rate may be impacted on all but the coldest days.

1.2 PARALLEL CONDENSING SYSTEM (PAC):

Exhaust steam from the steam turbine is separated into two streams. One stream flows into a water cooled surface condenser while the other is directed to an air-cooled condenser.
Condensate from the surface condenser and the air-cooled condenser can be collected in a common hotwell. Water consumption is controlled by the distribution of the heat load between the two condensers.
The PAC System should not be confused with a "hybrid" cooling tower, which is used primarily to reduce visible plume from a wet cooling tower. A "hybrid" cooling tower has practical limits to the amount of heat that can be rejected in the dry section, since the latter is sized for plume abatement only. With the PAC System there is complete flexibility in the amount of heat rejected in the dry section.
The dry section of the PAC System employs direct condensation in contrast to most "hybrid" systems, which are indirect condensing systems, i.e. water is cooled through both the wet and dry sections and is then pumped through a common condenser. As a result, the dry section of the PAC System can efficiently reject a substantial amount of heat even on hot days, thereby reducing peak water usage. During cooler periods, the amount of heat rejected in the dry section can be increased up to 100% if so designed, thus further reducing the plant's water consumption.
An additional benefit of the PAC System is the reduction of plume. Plume can be reduced or eliminated entirely when danger of icing exists, simply by shutting off the wet section.

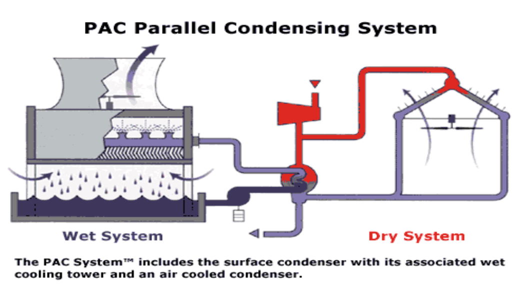


Figure 1.1: parallel condensing system

2. ADVANTAGES OF DRY COOLING SYSTEM OVER WET COOLING SYSTEM

2.1 WET COOLING SYSTEMS

The wet cooling tower system is based on the principle of evaporation. The heated cooling water coming out of the surface condenser is cooled as it flows through a cooling tower, where air is forced through the tower by either mechanical or natural draft.

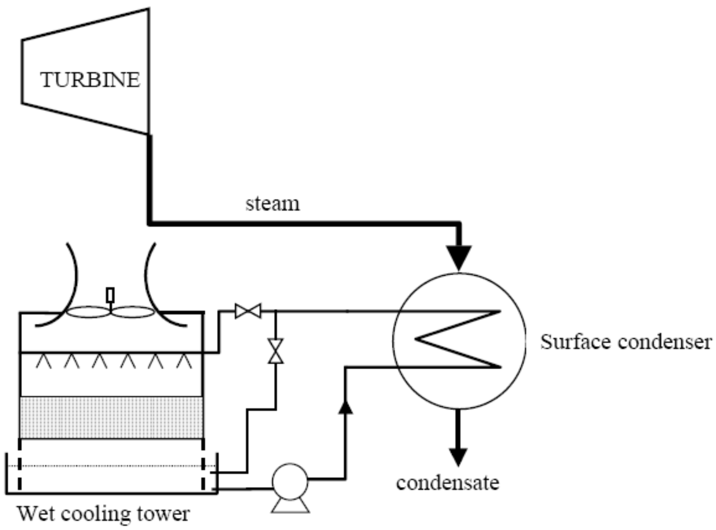


Figure 2.1 : Indirect cooling system with a wet cooling tower and surface condenser.

The steam from the steam turbine is condensed at the outside of the surface condenser tubes, using cold water coming from the cooling tower. Part of the cooling water is evaporated in the cooling tower, and a continuous source of fresh water (makeup water) is required to operate a wet cooling tower.
Makeup requirements for a cooling tower consists of the summation of evaporation loss, drift loss and blow-down.

2.1.1 Evaporation losses:

Evaporation losses can be estimated using the following equation:

$$m'_{evap} = 0.00095\, m'_{cool}\, (T_{hot} - T_{cold})$$

where m'_{cool} = cooling water flow rate at the tower inlet
T_{hot} = cooling water temperature at tower inlet in °F
T_{cold} = cooling water temperature at tower outlet °F

2.1.2 Drift:

Drift is entrained water in the tower discharge vapors. Drift loss is a function of the drift eliminator design, and a typical value is 0.005 % of the cooling water flow rate. New developments in eliminator design make it possible to reduce drift loss below 0.0005 %. Drift contains chemicals from circulating water.

2.1.3 Blow-down:

The amount of blow-down can be calculated according to the number of cycles of concentration required to limit scale formation. Cycles of concentration are the ratio of dissolved solids in the recirculation water to dissolved solids in the makeup water. Cycles of concentration involved with cooling tower applications typically range from three to ten cycles. The amount of blow-down can be estimated from the following equation:

$$m'_{blowdown} = \frac{m'_{evap}}{(\text{cycles} - 1)}$$

2.1.4 Typical water consumption examples:

As an example, a 600 MW coal fired plant operating at 70 % annual capacity factor typically would require between 5 x 106 m3 and 1 x 107 m3 of make-up water annually.
Fogging, icing of local roadways and drift that deposits water or minerals are some of the concerns regarding the plume. There are other environmental effects of cooling towers.
Sometimes because of the chemical content of the make-up water the blow-down cannot be discharged outside of the boundaries of the power plant. This is the case in power plants with "zero-discharge" requirements. But complete elimination of water consumption in the cooling system can only be achieved by using dry cooling systems, or air cooled condensers.

2.2 DRY COOLING SYSTEMS:

In a dry cooling system, heat is transferred from the process fluid, steam, to the cooling air via extended surfaces or fin tube bundles. The performance of dry cooling systems is primarily dependent on the ambient dry bulb temperature of the air. Since the ambient dry bulb temperature of the air is higher than the wet bulb temperature (wet bulb is the basis for a wet cooling tower design), dry cooling systems are less efficient. Although the capital cost of a dry cooling system is usually higher than that of a wet cooling system, the cost of providing suitable cooling water and other operational and equipment expenses may be such that the dry cooling system is more cost effective over the projected life of the power plant.

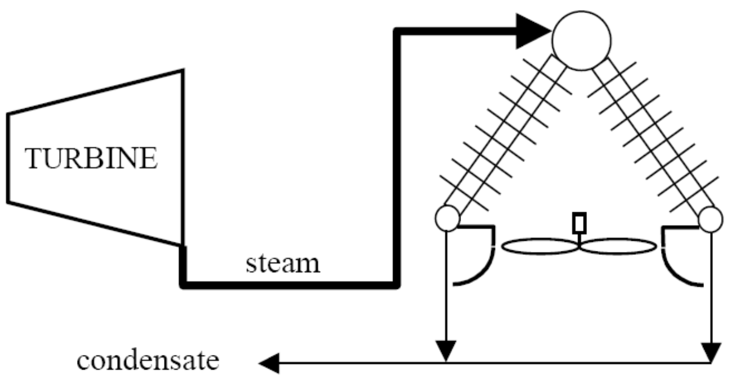


Figure 2.2 : Dry cooling system connected to steam turbine (direct system).

In dry cooling systems, the turbine exhaust is connected directly to the air cooled steam condenser (that is why it is called a direct system)

Table 2.1 : The advantages and disadvantages of dry cooling systems

ADVANTAGES OF DRY COOLING	DISADVANTAGES OF DRY COOLING
Can be located at fuel source	Large plot area required
No water required	Less efficient
No plume formation	Generates more noise
No impact on environment	
Less permitting required	

Recent studies indicate that on average, one third of the new power plants permitted in North America will require a dry cooling system. This is driven by the lack of water, concentration limit of particulate matter in cubic meter in air (annual arithmetic average not to exceed 50 micrograms per cubic meter of air), thermal limitations under state quality regulations to protect the population of shellfish, fish and wildlife in and on the body of water into which the discharge is made.
In some areas of the US, dry cooling will be the system of choice. In the state of Massachusetts for example, air cooled condensers are used in 70 % of the recently built power plants.

3. DRY COOLED STEAM CONDENSING SYSTEMS:

3.1 FROM CONCEPT TO SOLUTION:

Balcke-Dürr has many decades of experience with steam condensing systems utilized in power plants. Typical systems are shown in the accompanying diagrams:

- The direct system
- The indirect system (including secondary cooling water loop)
- The parallel dry/wet system

Balcke-Dürr's innovative concepts for solving the most challenging steam condensing problems are based on tubes with hot-dip galvanized steel fins of two types, elliptically shaped and flat. Both types feature
thermodynamic and hydraulic characteristics superior to those of circular fin tubes.
For clients desiring conventional tubes, Balcke-Dürr also supplies circular tubes with aluminum fins for both the direct and indirect systems.

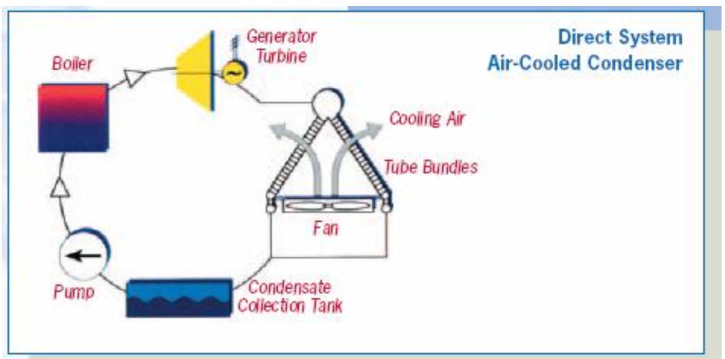


Figure 3.1 : Direct system air-cooled condenser

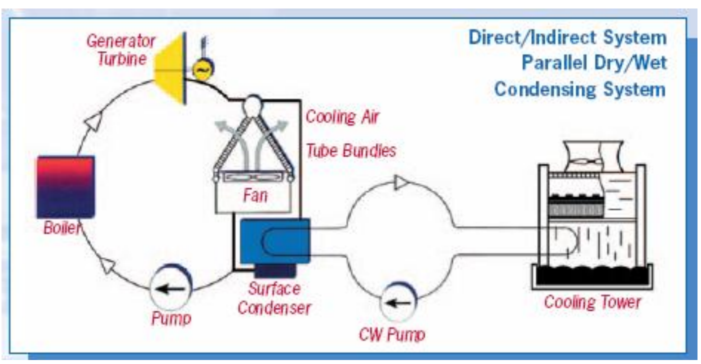


Figure 3.2 : Parallel condensing system

3.2 FIN TUBES FOR DRY COOLING:

Balcke-Dürr's innovative, globally-unique single row condenser tube with high-performance galvanized steel fins:

- No cold spots, no dead zone
- No flooding in reflux condenser
- Outstanding corrosion protection
- Excellent metallic welded fin bond
- Optimal heat transfer and power consumption
- Mechanically-strong fins
- Long service life
- Steel core tube
- Minimal fouling
- Suitable for cleaning with high pressure water up to 400 bar

3.2.1 Elliptical tube:

The advanced performer with wound-on steel fins, hot-dip galvanized:
- Outstanding corrosion protection
- Excellent metallic fin bond
- Optimal heat transfer and power consumption
- Mechanically-strong fins
- Long service life
- Steel core tube
- Minimal fouling
- Suitable for cleaning with high pressure water up to 400 bar

3.2.2 Circular tube:

Conventional with quality:
- Wound-on G-fin
- Wound-on L-fin
- Extruded fin
- Steel core tube
- Minimal fouling
- Suitable for cleaning with pressurized water or air

3.3 INNOVATION IN DIRECT DRY COOLING:

The Balcke-Dürr air-cooled condensers serve many types of industries and requirements, including:
- Power plants
- Chemical industry
- Waste incineration plants
- Steel industry
- All pressure levels with various fin tube systems
- Forced or induced draft convection

Direct-type air-cooled condensers are designed both as single-row and multi-row systems. Dead-zone formation, an undesirable characteristic of multi-row condensers, is eliminated by adjusting the fin pitch of each row and dimensioning the dephlegmator accordingly. Balcke-Dürr multi-row condensers benefit from this innovative, pragmatic engineering approach by being fully operational in cold climates. Confirming the design's merit are the many successfully functioning power stations, ranging from small waste incineration plants to large power station condensers up to 700 MW.

4. THE HAMON AIR COOLED CONDENSER:

4.1 SINGLE ROW CONDENSER DESCRIPTION:

The Hamon Air Cooled Condenser (ACC) consists of "A" roof-type streets of Single Row Condenser (**SRC**) tubes. A street can contain several modules, and each module is composed of an even number of bundles of finned tubes. An axial flow fan located in each module forces the cooling air through the fins.

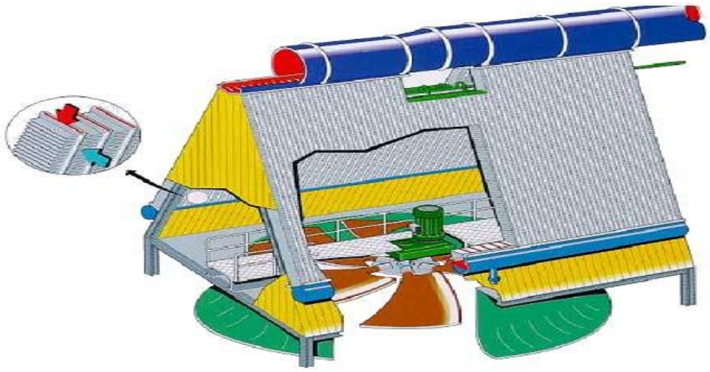


Figure 4.1 : Hamon Air Cooled Condenser

The steam flows through a large diameter duct to the condenser. The duct system branches into risers and steam distribution manifolds running along the top of each street. Steam enters from these distribution manifolds into the finned tubes at the top of the primary condensing bundles. The steam condenses partially during the downward flow in the primary tubes. Condensate and non-condensed steam is collected in large steam/condensate headers running along the bottom of the "A" roof. About 80% of the steam is condensed in these primary condensing bundles (steam and condensate flowing co-current from top to bottom).
The remaining steam (ca. 20%) enters the finned tubes of the secondary condensing bundles through their bottom connections with the steam/condensate headers. The steam condenses in counter-flow mode, i.e. remaining steam and non-condensable gases flow upwards while the condensate flows downward to the steam/condensate headers. In this way, the condensate is always heated by steam and sub-cooling is prevented.
The non-condensable gases accumulate near the top of the secondary condensing bundles and are drawn into the air take-off manifolds running along the top of these secondary bundles. These manifolds are connected to the vacuum system through air take-off lines in order to remove the non-condensable gases from the condenser.
The condensate collected in the steam/condensate headers drains under gravity to the main condensate tank and drains from the steam duct and from the turbine are pumped to the main condensate tank through the drain pot of the steam duct.

4.2 SINGLE ROW CONDENSERS:

The Single Row Condenser (SRC) has been developed to improve performance and efficiency. The Single Row Condenser Tube has been engineered to incorporate a number of characteristics favorable to the economical and trouble-free operation of the condenser. The design features virtually 100% effective finned surface, while minimizing airside pressure drop. The large cross-section of the tube results in minimum inside pressure drop and therefore in high performance with a very low sub-cooling. It also allows a higher steam velocity in the secondary tubes without restricting the down flow of condensate, thereby allowing the owner to operate at lower backpressure at freezing conditions.

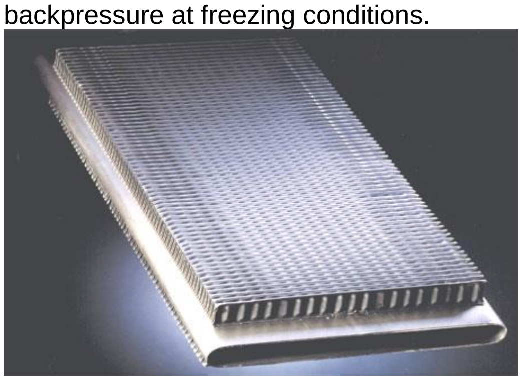


Figure 4.2 : Single Row Condenser

4.2.1 Single Row Condenser Advantages:

The Single Row Condenser tube has, beside its high performance and efficiency, a number of specific advantages over multi row tubes that deserve special attention.

4.2.1.1 *Corrosion Resistance:*
The fins and exterior of the SRC are aluminum, which offers an excellent resistance to corrosion. Extensive corrosion tests in research and development laboratories have proven the outstanding corrosion resistance properties of the SRC tube, indicating a life expectancy of the tubes in excess of the plant lifetime.

4.2.1.2 *Fin Damage Prevention:*
Since the fins are located between the tubes and recessed from the edges of the tube, mechanical damage during construction, maintenance or as a result of hail, is eliminated. In the unlikely event that an SRC tube does become damaged it is possible to replace a single tube at site.

4.2.1.3 *Low air side fouling and Cleanability:*
The design of the SRC tube ensures that it is both the least susceptible to fouling and most easily cleanable tube available today. The SRC is not subject to cleaning water pressure losses from above rows, turbulators or fin spacers that affect the cleanability of some of its multi-row counterparts. This ease of cleanability and reduced fouling probability result in a condenser that requires less frequent cleaning, less man hours to clean and less water to clean.

4.2.1.4 *Freeze Protection:*
The SRC tubes resistance to freezing is one it greatest assets.

Each tube in a SRC bundle has the same condensing capacity. This means that the primary cause of freezing in ACC's, "backflow", is eliminated (see Fig 4.3). Back flow occurs in multi-row condenser when steam from the outermost rows (row 2 - lowest condensing capacity) flows through the condensate header and into the inner most rows (row 1 - highest condensing capacity) in the opposite direction to the normal steam flow. These opposing flows condense and flood the small diameter (typically 1 inch) tubes, which then rapidly freezes, expands and can rupture the tubes.

In addition to this the SRC design has other features, which reduce the risk of freezing.
Due to the absence of risk of freezing, this single row design gives more flexibility in operation during extreme cold air temperatures.

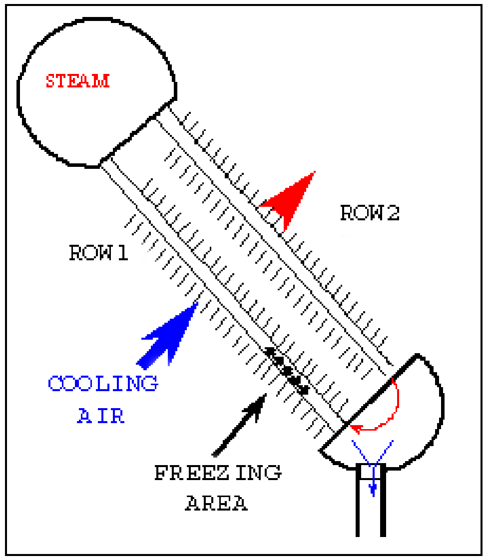


Figure 4.3 : Backflow in a multi-row bundle

The cross sectional area of a SRC tube is much greater than that of a multi-row tube (see Fig 4.4). The advantage of this is two fold. Firstly, the condensate can only partially fill the tube and is always in contact with steam, which is not yet condensed; this reduces sub-cooling and ensures that the condensate temperature does not approach freezing temperatures. Secondly, in the unlikely event that the condensate does start to freeze there is sufficient free space inside the tube that the ice can expand inside the tube and not cause it to rupture.

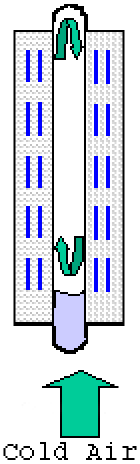


Figure 4.4 : Steam in contact with condensate in SRC tube

4.2.1.5 *Long Term Mechanical Integrity:*
The SRC design of bundles is made of separated finned tubes. The brazing quality of each individual tube can therefore be very easily controlled. Each finned tube can also freely expand without being restricted by its surrounding tubes in the bundle. Therefore, in all possible operating temperatures, tubes can for example freely move without transferring stresses to the neighboring tubes. In the contrary to other designs on the market, the SRC design is a stresses free concept of bundle. Only the absence of excessive internal stresses in bundles will guarantee a long term reliability of the equipment, regarding to the risks of fatigue failure and stress corrosion after several years.
This is validated by our extended references operating since many years in all climates and environmental conditions.

5. ADVANTAGES OF THE PARALLEL CONDENSING SYSTEM:

Parallel condensing systems, have been developed to save water, while avoiding the high cost of dry cooling systems and to ensure a relatively low steam turbine back pressure at high ambient conditions.
An excessive rise in steam turbine backpressure during periods of peak ambient temperatures and demand will result in a loss of efficiency of the steam turbine generator set. In such a case, the dry section of the system may be designed to reject the total heat load at a low ambient temperature while maintaining the turbine backpressure within specified limits at high ambient temperatures using the wet part of the system. One way of sizing the wet part of a PCS cooling system is to limit the quantity of make-up water according to the local water availability.
A PCS system is a synergy of established cooling system technologies and combines some positive features of dry and wet cooling systems; the water consumption is reduced compared to a 100 % wet system, the performance is improved compared to a 100 % dry system and the capital cost decreases as the proportion of wet in the PCS system is increased.

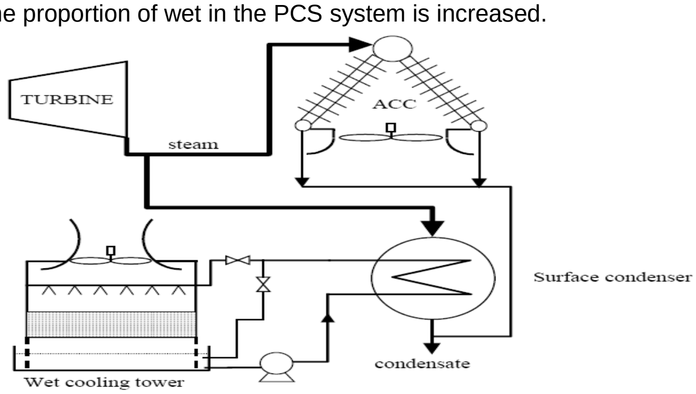


Figure 5.1 : Parallel condensing system (Dry/wet cooling system).

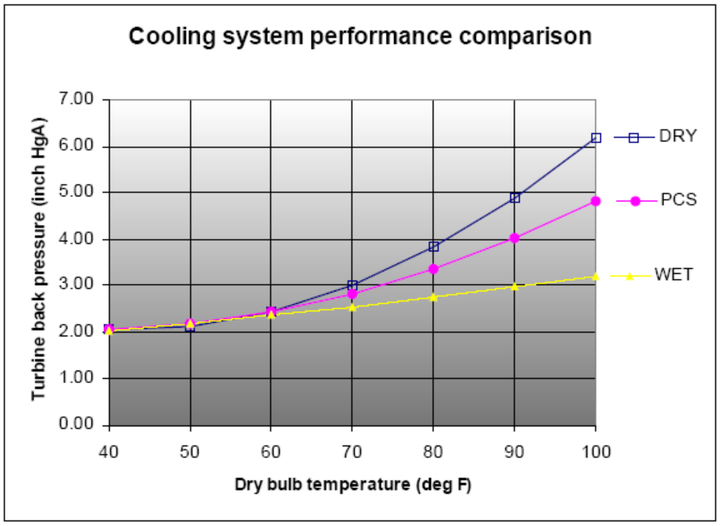


Figure 5.2 : Dry, PCS and wet cooling systems – comparison of the performance.

A typical cooling system performance as shown in the above figure the turbine back pressure is plotted as function of the dry bulb temperature.

The wet cooling system is able to maintain a much lower turbine back pressure at high ambient temperatures. The performance of the PCS system is in between the dry and wet cooling systems. The relative improvement of the PCS system with respect to the 100% dry cooling system is dependent on the amount of water that is used for wet cooling.

Table 5.1 : Design conditions for the air cooled condenser and PCS system.

DESIGN CONDITIONS	VALUE in SI units
ambient dry bulb temp	40.6 deg C
relative humidity	16%
ambient wet bulb temp	21.0 deg C
atmospheric pressure	946 mbar
required thermal duty	445.4 MW
turbine back pressure	< 270 mbar

A 100 % dry cooling system and a PCS system (using a small cooling tower) were designed for the design conditions as shown above.

The major requirement is to avoid a turbine trip (typical value is a turbine back pressure lower than 270 mbar) at the maximum ambient air temperature. In the following study it was decided to design the PCS system in such a way that the wet cooling tower should only operate on hot summer days (ambient dry bulb temperature above 32 deg C or 90 deg F).

In the parallel condensing system, the wet cooling tower can be shut down in spring, autumn and winter, because the dry portion of the cooling system is sufficient to handle the required thermal duty. In the graph below it can be noticed that the dry portion of the PCS system can handle the thermal duty up to an ambient temperature of about 32 degrees Celsius (90 deg F).

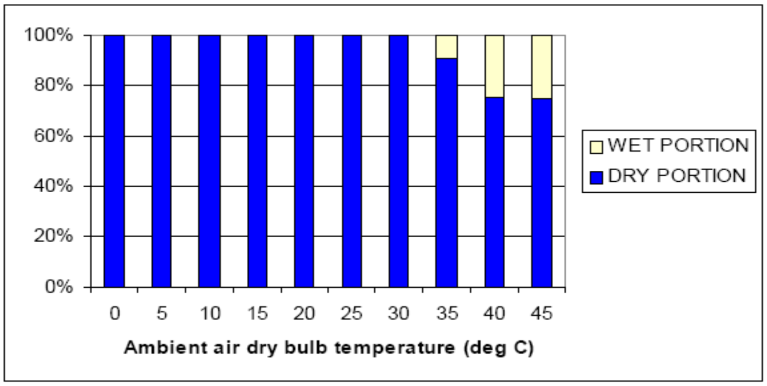


Figure 5.3 : Wet and dry portion of the thermal duty as function of ambient dry bulb temp.

As the ambient air temperature rises, a larger portion of the duty is handled by the wet cooling tower. At the maximum ambient dry bulb temperature, the wet cooling tower rejects about 25 % of the total thermal duty.

Assuming that the air cooled condenser cannot handle the thermal duty any more for ambient air temperatures exceeding 32 °C (89.6 °F), combined with the temperature distribution chart it is assumed that the wet cooling tower will be working for only about 30 days per year, which is a reasonable design for a PCS system.

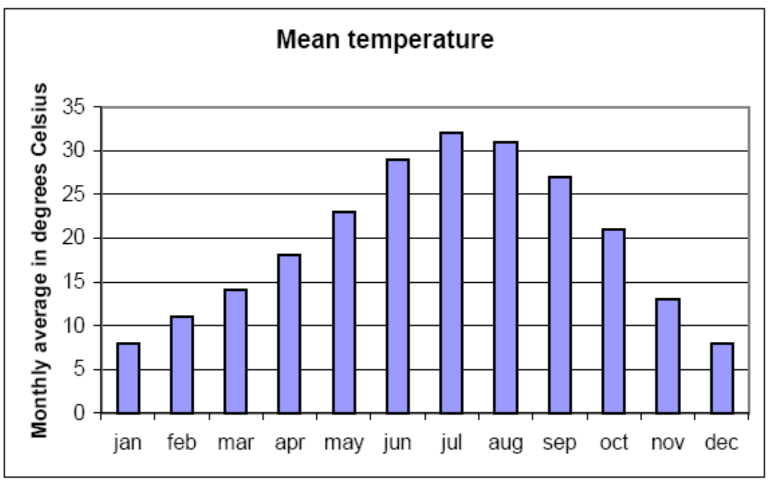


Figure 5.4 : Monthly average temperature for the PCS system design.

An estimation of the capital costs for the wet part of the PCS cooling system is based on the following breakdown, as shown in table 5.2 :

Table 5.2 : Capital cost breakdown for the **wet part** of the PCS cooling system.

ELEMENT	COST
Wet cooling tower	35 - 45 % of system cost
Installation/erection	included in base price
Surface steam condenser	35 - 45 % of system cost
Tower basin	3 - 6 % of system cost
Electricals and controls	Typically $25,000 per cell
Circulating water system	5 % of system cost
Water treatment/blowdown dischar	1 % of system cost

The capital cost of the air cooled condenser (reference value = 100 % dry cooling) includes the cost for installation and erection, and is estimated at about $31.2 million for a typical 500 MW combined cycle power plant.

Table 5.3 : Comparison between air cooled condenser and PCS system.

ITEM	ALL DRY SYSTEM	PCS SYSTEM
dry fraction of heat	100	73%
capital	100	79%
fan power	100	83%
plot	100	82%

As can be noticed from table 5.3 , the introduction of a small cooling tower (typically two cells) can reduce the capital cost by more than 20 % compared to a 100 % dry system (remark : a 100 % dry system refers to a cooling system where an air cooled condenser is responsible for one hundred percent of the total heat duty). Also the plot area and fan power consumption are more favorable for the PCS system. Operational costs are expected to be less for the PCS system in general.

6. GOLDENDALE ENERGY PROJECT- PROJECT DESCRIBTION:

The Goldendale Energy Project (GEP) is being developed to generate 248 megawatts (MW) of electricity for transmission to Goldendale Aluminum by the Klickitat PUD.

6.1 EQUIPMENT:

Equipment at GEP will include one combustion driving its associated electric generator; one heat recovery steam generator; one steam turbine driving its associated electric generator; a combination air-and-water steam condensing system; cooling towers. The steam turbine exhaust steam must be condensed through cooling to return to the steam cycle. Water is normally the primary source of cooling. Since water is scarce, a parallel condensing system was selected as the most effective means of meeting cooling needs while reducing water requirements. The parallel condensing system consists of a dry condenser and a wet condenser operating in parallel to provide the required heat dissipation over the range of ambient conditions. The dry condenser uses air to condense the steam. Water is not used when air temperatures are below 29 F. Water use gradually increases reaching maximum flow at 50 F and above. The wet condenser is, in effect, a topping condenser used only when air condensing is inadequate. This technology reduces the amount of water required. Most of the water used is evaporated in the wet cooling tower. The remainder is discharged to the wastewater system as cooling tower blowdown. The City of Goldendale will provide water and wastewater services to GEP.
The major sources of wastewater are cooling tower blowdown, boiler blowdown, demineralized water treatment system discharges including multimedia filter backwash and RO reject water. Where possible, all wastewater streams will be reused to minimize raw water use. Boiler blowdown and RO reject will be reused as cooling tower makeup, when water quality is suitable. Other wastewater streams will be discharged into the City of Goldendale wastewater system.

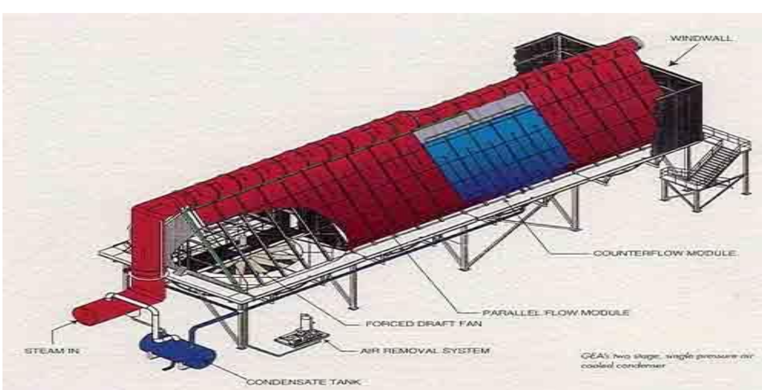


Figure 6.1 : The GEA air-cooled condenser

The GEA air-cooled condenser is comprised of finned tube bundles grouped together into modules and mounted in an A-frame configuration on a concrete or steel support structure. Vertical and horizontal configurations are also available.
GEA employs a two-stage, single-pressure condensing process to achieve efficient and reliable condensation. In this process, the steam is first ducted from the steam turbine to the air-cooled condenser, where it enters in parallel/concurrent flow from the top. The steam is only partly condensed in the parallel flow modules and the remaining steam is ducted to the lower headers of the counter-flow finned tube bundles (dephlegmator). The steam enters from the bottom and rises in the finned tubes to a point where condensation is completed. Non-condensables are drawn off above this point by vacuum equipment. The condensate drains to a condensate tank and is then piped back to the feedwater system to complete the cycle.

7. IMPROVING AIR-COOLED CONDENSER PERFORMANCE USING WINGLETS AND OVAL TUBES:

7.1 INTRODUCTION :

Two concepts for improving the heat transfer performance of the air-cooled condensers used in binary geothermal power plants are being developed and tested at the INEEL.
In a binary geothermal plant where there is not a sufficient supply of water for an evaporative cooling system, heat must be rejected to atmospheric air. This heat rejection is accomplished through the use of large air-cooled condenser units in which air is forced through several rows of long individually finned tubes by large fans.
The condenser tubes have fins on the outside surface in order to provide a large effective heat transfer surface area. Improving the air-side heat transfer coefficient is expected to result in smaller, more efficient heat exchangers and reduced plant cost.
INEEL researchers are investigating improving the condenser performance by incorporating one or both of the following two concepts. The first concept is to add properly sized and strategically located vortex generators/winglets on the fins. The second concept is to replace the circular tubes with oval tubes. Deployment of winglets on fin surfaces has been shown to enhance heat transfer through the generation of longitudinal vortices that produce localized thinning of thermal boundary layers.
The usage of oval tubes instead of circular tubes results in reduced form drag and increased tube-surface area for the same cross-sectional internal flow area. This strategy is not practical in all cases due to manufacturing considerations and the fact that circular tubes are inherently stronger and can therefore withstand much higher pressures with the same wall thickness.
By optimizing the shape and location of the winglets, the resulting vortices can minimize the size of the wake (stagnant flow) region behind a cylindrical tube and also improve the heat transfer downstream of the winglets.
Longitudinal vortices are generated naturally in fin-tube heat exchanger passages by the interaction of the flow velocity profile with the heat exchanger tube.
Vortices can also be generated if the flow is interrupted by vortex generators, small winglets placed in the flow path. The size, shape, and angle of attack of the vortex generators determine the specific characteristics of the vortices generated in the flow. These vortices lead to enhancement of heat transfer.
To take advantage of these phenomena and develop an acceptable practical design, the INEEL has been performing experimental and modeling research.

7.2 EXPERIMENTAL INVESTIGATION:

Beginning in 1999, the INEEL performed a series of laboratory-scale experiments to systematically evaluate the influence of vortex generators and oval tubes on heat transfer enhancement and changes in pressure drop. The single-tube heat transfer experiments were performed in a narrow rectangular flow channel designed to simulate a single passage of a fin-tube heat exchanger. A schematic of the flow loop is shown in Figure 7.1.

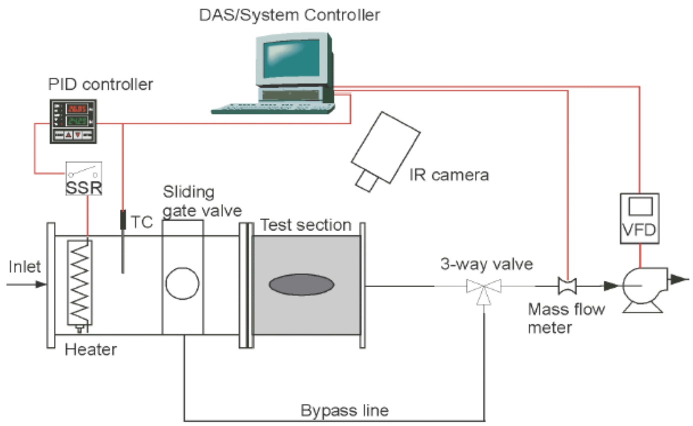


Figure 7.1 : Schematic of flow loop.

A transient heat transfer measurement technique was employed for obtaining detailed local heat transfer measurements on the model fin surface. Inlet air is heated to a desired setpoint temperature using an in-line feedback-controlled finned-element air heater (350 W). The heated air initially flows through a bypass line until the desired air temperature and flow rate is established. The air is then suddenly diverted through the test section by changing the position of a 3-way valve.

Local surface temperatures on the substrate increase at a rate that is dependent on the value of the local heat transfer coefficient. This transient localized heating is quantitatively recorded using an imaging infrared camera. Values of local heat transfer coefficients can then be determined from an inverse heat conduction analysis.

7.3 RESULTS:

Two local surface heat transfer coefficient contour plots obtained using the imaging infrared camera are presented in Figure 7.2. The addition of winglets yields a reduction in the size of the low-heat-transfer wake region and also provides localized heat transfer enhancement in the vicinity of the winglets. winglets. Peak local heat transfer coefficients in the vicinity of the winglets are similar to the peak values observed in the cylinder stagnation region.

Stagnation-region heat transfer coefficients are slightly higher for the winglet case compared to the no-winglet case.

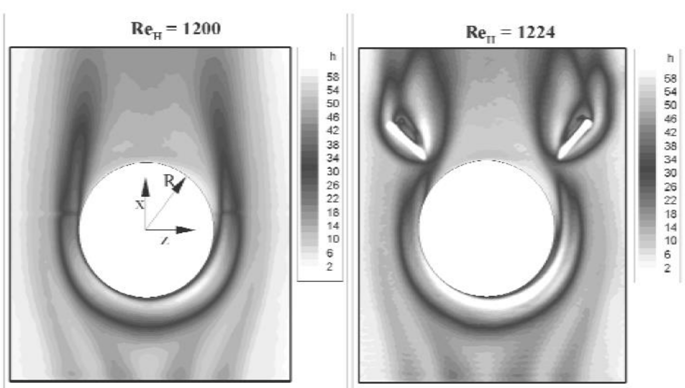


Figure 7.2 : Direct comparison of local heat transfer distributions for a circular cylinder with and without winglets.

A plot of the span-wise variation in local wake-region heat transfer coefficient at an axial location just downstream of the winglets is presented in Figure 7.3 for the same two data sets presented in Fig 7.2. The span-wise variation for the winglet case clearly shows a double peak associated with each winglet. A single peak associated with each horseshoe vortex is evident in the no-winglet curve.

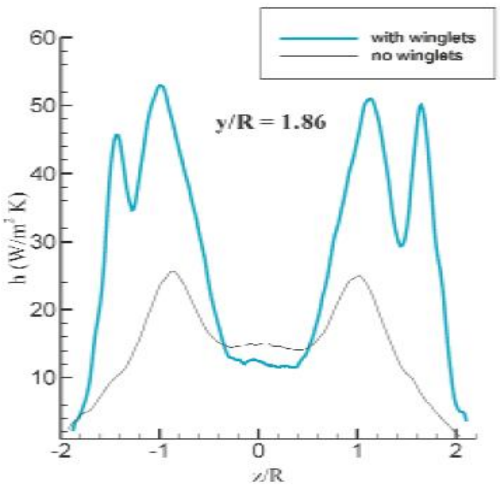


Figure 7.3 : Span-wise variation in local wake-region heat transfer coefficient, with and without winglets.

7.4 INFERENCE:

Laboratory-scale experiments have been conducted for measuring heat transfer coefficient corresponding to circular and oval tubes with and without vortex generators. All the data indicate that the addition of winglets increases the heat transfer coefficient by ~35% as compared to plain tubes.
Corresponding increase in friction factor is in the range 5–10% for Reynolds number, ReDh in the range 500–5000. Next, prototype-scale tube bundle tests will be performed. Meanwhile industrial collaboration for developing an economic manufacturing method is continuing.

8. WIND TUNNEL SIMULATION ON RE-CIRCULATION OF AIR COOLED CONDENSERS:

8.1 INTRODUCTION:

A project of an extension power station, located in northern China, planned to use GEA air-cooled condensers for a 2*200MW power plant. The GEA air-cooled condenser uses a space-saving A-frame design installed at grade level.
In an air-cooled condenser cell, exhaust turbine-steam flows inside the steel elliptical tubes; cooling air is drawn through the fins by a large fan, which is mounted underneath. The air takes the heat from the exhaust turbine-steam, which converts to condensate.
Due to the requirements of technological process of a power plant, air-cooled condensers platform usually sites behind the steam turbine room.

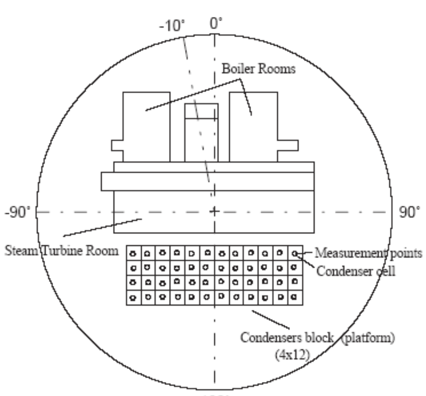


Fig. 8.1 : Schematic configurations of the proposed power plant together with the definition of the angles of incident flow, beta.

In order to better understand the characteristics as well as the mechanism of wind effects on the performance of air-cooled condensers of the project and to minimize the unfavorable wind effects, the phenomenon of re-circulation of air-cooled condensers as investigated by means of wind tunnel simulation. Total and distributions of re-circulation of hot air in the inlets of condensers platform were obtained and described.

8.2 EXPERIMENTAL APPARATUS AND DATA REDUCTION:

The measurements of concentration for the re-circulation were conducted in a boundary layer wind tunnel at Peking university, Beijing, China. The tunnel has a rectangular test section 3m wide, 2m high and 32m long. The wind speed may change from 0.3 to 10 m/s.
The model of the power plant, including the air-cooled condensers platform, the boiler rooms and the steam turbine room, were positioned on a turntable, which locates at the downstream of the test section.
The flow visualization experiments were conducted in another low speed wind tunnel at the same university. It has an open circular test section of 2.25m in diameter and 3.65m long.

8.3 RESULTS AND DISCUSSION:

It is expected that the arrangement and geometric configurations of boiler rooms and steam turbine room, wind directions and wind speed of oncoming flow have great effects on the results of re-circulation. Four model conditions with different heights of condensers platform, i.e. concrete circular cylindrical props and windbreak configuration were tested. The four model conditions are as listed in Table below.

Table 8.1 : Three models of model condition

Model	Conditions
1	Condensers platform height $H = 31\,\text{m}$
2	Condensers platform height $H = 34\,\text{m}$
3	Condensers platform height $H = 31\,\text{m}$, windbreak which facing to the steam turbine room raise 3 m
4	Condensers platform height $H = 31\,\text{m}$, mount an additional platform between the top of steam turbine room and the platform of condensers platform

It is obvious that the wind directions have great effects on the re-circulation. Figure 8.2 presents that the total re-circulation R_T(b) varies with the angle of incident flow beta.

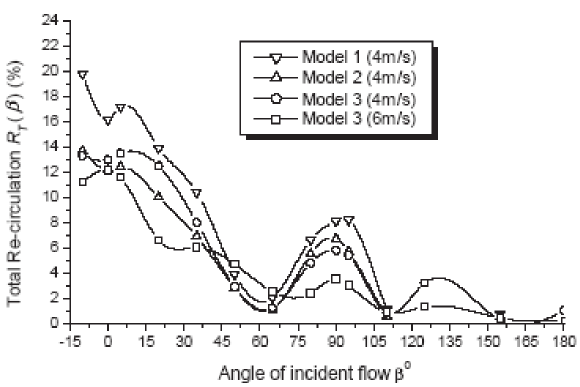


Figure 8.2 : Total re-circulation $R_T(\beta)$ varies with the angles of incident flow, β.

It is shown that as the wind blows normal to the boiler rooms or within +/- 10 degree, the most unfavorable effects of wind on condensers result. As the wind directions deviate from this region, the total re-circulations reduce quickly and reach the minimum value less than 3% at beta = 65 degree. However, as the wind blows normal to the gap between the steam turbine room and the block of condensers (beta = 90 degree), the values of total re-circulation increase again, which form the second peaks of re-circulation for the three individual model.

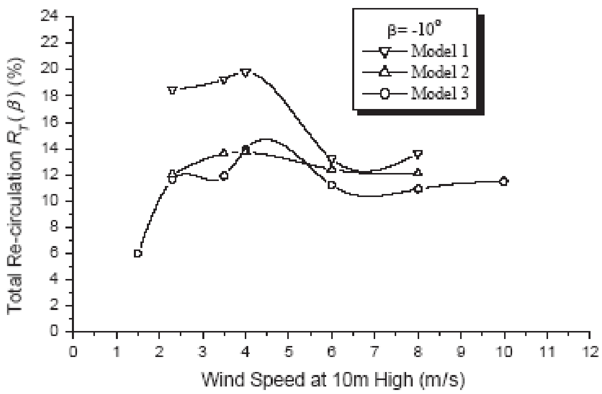


Figure 8.3 : Total re-circulation $R_T(\beta)$ ($\beta = -10^\circ$) varies with the wind speed for the three models

As the wind speed reduces to 1.5 m/s, the total re-circulation of both Models 2 and 3 reduces tremendously to only 6%. It is believed that if there is no wind, the re-circulation vanishes and it is confirmed at the beginning of the experiments. As the wind speed is between 6 and 10 m/s, the values of re-circulation change with the wind speed smoothly for all the three models.

8.4 <u>INFERENCE:</u>

It is concluded that at the most unfavorable wind direction, the most serious recirculation happens at the wind speed between 2 and 4 m/s. The heights of condensers have a strong effect on the re-circulation. As the wind speed exceeds to6 m/s, the re-circulations tends to a constant value.
By means of concentration measurements, characteristics of the performance of air-cooled condensers in a power plant were simulated in wind tunnel tests. The most important criteria must be met, especially the dynamic and thermal properties of the exhaust hot air from the condensers. Due to the interference of the neighboring buildings, such as the boiler rooms and the steam turbine room, the angles of incident flow have a great effect on the efficiency of air-cooled condensers. As the wind blows normal to or within +/-10 degree the boiler rooms, the most unfavorable effects of wind on condensers result. On the other hand, at the most unfavorable wind directions the most serious re-circulation takes place at the wind speed between 2 and 4 m/s. Combined with the information of local wind climate, this model condition should be avoided as much as possible for a power plant equipped with air-cooled condensers. There is a great advantage in reducing the unfavorable wind effect on the performance of condensers by raise the height of platform or the windbreak. Therefore, it is possible to have some steps to reduce the unfavorable effect of wind on the condensers by means of wind tunnel simulation. Wind tunnel simulation could play an important role in the design stage of a new or extension power plant with air-cooled condensers.

CHAPTER – 9

9. PHOTO GALARIES :

FIGURE 9.1
STATION: Mystic units 8 & 9
LOCATION: Everett, MA
PLANT GENERATION: 1600 MW combined cycle
START-UP: 2003

FIGURE 9.2
STATION: Sutter power plant
STATION: Sutter
LOCATION: Yuba City, CA
PLANT GENERATION: 500 MW combined cycle

START-UP: 2001

FIGURE 9.3
STATION: Fore River
LOCATION: Weymouth, MA
PLANT GENERATION: 800 MW combined cycle
START-UP: 2003

FIGURE 9.4
STATION: HSIN TAO
LOCATION: Hsinchu, Taiwan
PLANT GENERATION: 600 MW combined cycle
START-UP: 2001

10. SPECIFICATION OF NTPC NORTH KARANPURA STP (3*660 MW) ACC

1、ACC PERFORMANCE CURVE FOR VWO 3%MU (106.6mmHg) Ref HBD No.T0341R03

1.1 DESIGN PARAMETER

All Fans at 100% Speed

Total steam flow rate DG	1275.293	t/h
Average steam dryness xG	94.28	%
Back pressure	0.1449	ata
Site elevation	445[1)]	m
Air inlet temperature	38	℃
Wind speed 1m above ACC(max.)	5	m/s
Total consumed power of fans	17667[2)]	kW

To read performance at actual load:

- use steam flow D on horizontal axis

  where: $D = 100*(D_{actual}/DG) * (x_{actual} / xG)$

  D_{actual} is actual steam flow (kg/s)

  x_{actual} is actual steam dryness (kg/kg)

- use actual air temperature curve
- read actual turbine backpressure on vertical axis

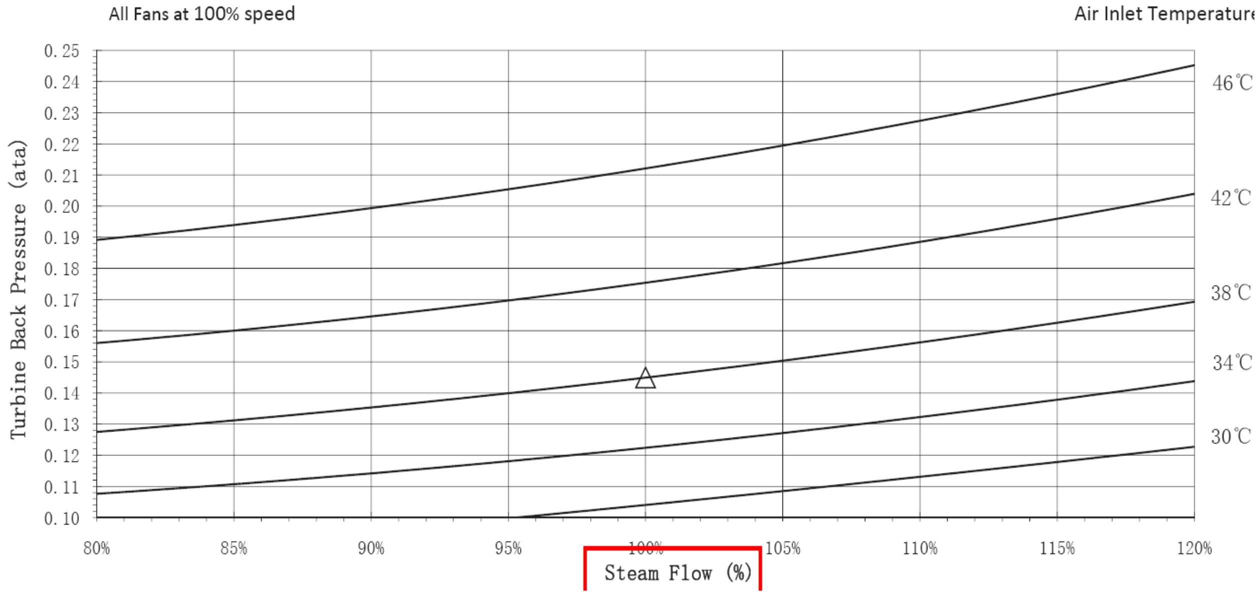
Note:

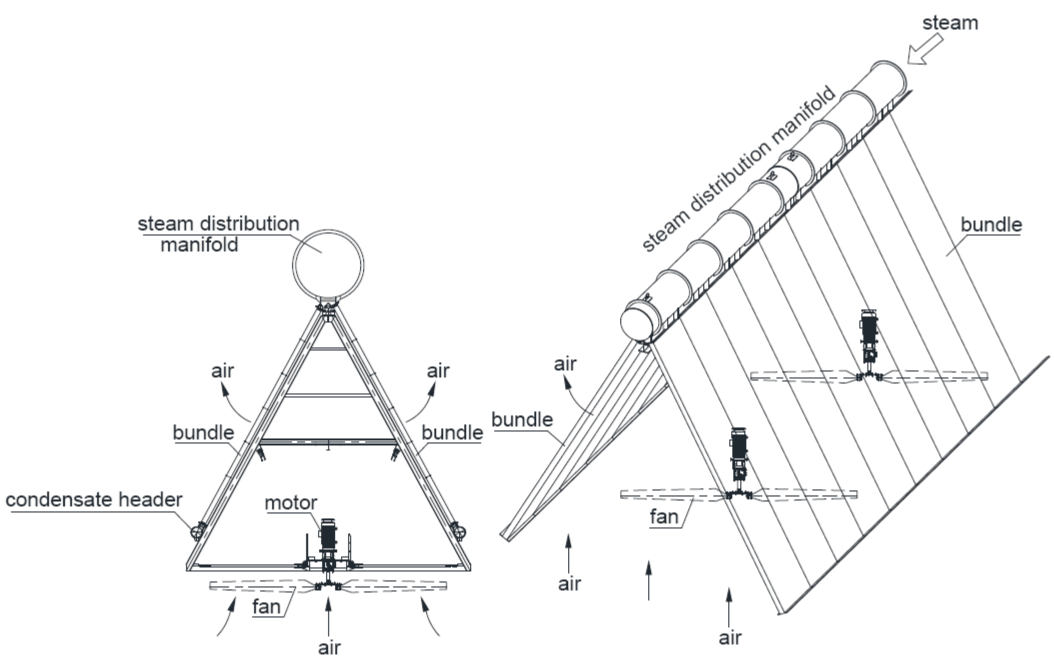
1）The atmospheric pressure 95.85kPa equal to 506m from MSL(445m+61m air inlet hight)

2）The power consumed shall be measured at the switchgear end.

3）Steam flow range for contractual acceptance test： 90%~110%(according to ASME PTC 30.1-2007)

1.2 ACC PERFORMANCE CURVE FOR VWO 3%MU(106.6mmHg) Ref HBD No.T0341R03





STRUCTURE OF AIR-COOLED CONDENSER OF NTPC NORTH KARANPURA

2、ACC PERFORMANCE CURVE FOR Guaranteed Auxiliary Power Condition (10 Load , 0% MU at 196.6mm Hg) Ref HBD No.T0302GR3

2.1 DESIGN PARAMETER

Fans run at combination of 50% speed and 100%speed

Total steam flow	1276.711	t/h
Average steam dryness	94.84	%
Back pressure	196(0.2665ata)	mmHg
Site elevation	445[1)]	m
Air inlet temperature	38	℃
Wind speed 1m above ACC(max.)	5	m/s
Total consumed power of fans	not exceed 5200	kW

Power consumption value to be indicated.

Note:

1）The atmospheric pressure 95.85kPa equal to 506m from MSL(445m+61m air inlet hight)

2）The power consumed shall be measured at the switchgear end.

11. CONCLUSION :

Because of restrictions on thermal discharges to natural bodies of water, almost all power generating plants or large industries requiring cooling will require closed cycle cooling systems. Evaporative or wet cooling systems (cooling towers) generally are the most economical choice for closed cycle cooling systems where an adequate supply of suitable water is available at reasonable cost to meet the make-up requirements of these systems. If only the plume is an issue the solution may be a wet/dry cooling system (hybrid cooling towers). But although these systems may save some water, the amount of make-up water is still significant and a plume will still be present under certain atmospheric conditions, and this may be unacceptable if the power plant is close to a major highway or airport.

If only a limited amount of water is available, or the water cost is too high, most power plants tend to go for a 100 % dry system without considering the PCS system. In some cases, a dry cooling system has been selected even if water is available at reasonable cost where political or environmental considerations prevail. But by selecting a parallel cooling system that is designed to use the available water for a cooling tower on hot summer days, the performance of air cooled condensers can be enhanced and significant savings on the capital and operational costs of the cooling system can be expected. Moreover, the wet cooling tower can be shut down most of the time (except on hot summer days), so the negative effects of a plume (fogging and icing in winter months) are not an issue.

Two concepts for improving the heat transfer performance of the air-cooled condensers used in binary geothermal power plants are being developed and tested at the INEEL. Laboratory-scale experiments have been conducted for measuring heat transfer coefficient corresponding to circular and oval tubes with and without vortex generators. All the data indicate that the addition of winglets increases the heat transfer coefficient by ~35% as compared to plain tubes.

In wind tunnel simulation on re-circulation of air-cooled condensers of a power plant it was found that height of platform of air cooled condenser and wind speed have an effect on performance of the condenser. It is recommended that in the initial stage of a new or an extension power plant, which is equipped with an air-cooled system, the wind tunnel simulation is necessary and helpful. Combined with the local wind climate data, a more reasonable, economic and safety schematic design of a power plant could be achieved.

12. VISION FOR FUTURE:

To restrict the environmental pollution of the plumes from a cooling tower , to protect the population of shellfish, fish and wildlife in and on the body of water into which the discharge is made from thermal pollution from cooling water and chemical pollution from blow down of cooling tower the parallel condensing system will be the system of choice for future power plants and retrofitting to existing power plants.

REFERENCES:

R1 : DRY COOL-03 ©2004 Marley Cooling Technologies, Inc. | Printed in USA/ 7401 W. 129 Street // Overland Park, KS USA 66213 // 800 462 7539 // info@marleyct.spx.com // **www.marleyct.com**

R2 : WHY EVERY AIR COOLED CONDENSER NEEDS A COOLING TOWER By Luc De Backer and William M. Wurtz, Presented at the 2003 Cooling Technology Institute Annual Conference/ San Antonio, Texas – February 10-13, 2003

R3 : GEA Power Cooling, Inc./143 Union Blvd., Ste. 400; Lakewood, CO 80228; tel: (303) 987-0123.

R4 : Klickitat County Economic Development/ Link to Source: mailto:Business@co.klickitat.wa.us ;

R5 : HAMON DRY COOLING, SPX Cooling Technologies Belgium S.A./N.V. , Rue Neerveld 107, 1200 Brussels, Belgium / Tel + 32 (0) 2 761 61 11 / Fax + 32 (0) 2 761 61 86;

R6 : Journal of Wind Engineering and Industrial Aerodynamics 93 (2005) 509–520 / Wind tunnel simulation on re-circulation of air-cooled condensers of a power plant / Zhifu Gua,_, Hui Lia, Wenhong Zhanga, Yan Lia, Jiye Pengb;

R7 : *Geothermal Resources Council Transactions,* Vol. 25, August 26-29,2001 / IMPROVING AIR-COOLED CONDENSER PERFORMANCE USING WINGLETS AND OVAL TUBES IN A GEOTHERMAL POWER PLANT; M. S. Sohal and J. E. O'Brien ; Idaho National Engineering and Environmental Laboratory ;
P. O. Box 1625 / Idaho Falls, ID 83415-3815 / *Sohalms@inel.gov;*