ADVANCED TECHNOLOGIES AND POWER INSTALLATIONS FOR THERMAL AND ELECTRIC ENERGY GENERATION

6.4. Application of air condensers in power industry

6.4.1. Analysis of application of air condensers in power industry

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Analyses of design and features of air condensers operation

For a long period of time air condensers (AC) were used only for steam turbines of low capacity – from 1 to 15 MW. An air condenser for a turbine of higher capacity (160 MW) was for the first time applied at Utrillas TPP, constructed in 1970 in a water-short region of Spain. The condenser was manufactured by GEA company (Germany). Technical decisions were developed and tested while designing, constructing and operating, which allowed further application of AC for turbines of even higher capacity and in more severe climate conditions. The above was confirmed by a long operational experience (since 1978) of the AC of the same manufacturer, which was installed with a turbine of 365 MW at Wyodak TPP in Wyoming (USA).

Air condensers are currently in operation in South Africa at six turbines of 665 MW at Matimba TPP and six turbines of 657 MW at Maiba TPP. Balcke Durr (Gemany) has supplied ACs for the turbines of 150 MW to Tuss TPP in Iran.

Some data on air condensers are given in [1] and presented in Tab. 2.28.

Table 6.28. **Technical characteristics of air condensers**

Characteristics	TPP		
	Utrillas	Wyodak	Matimba
Turbine capacity, MW	160	365	665
Design conditions:			
Exhaust steam flow rate, t/h	349,2	858	1588
Backpressure*, kPa	11,7	20,3	22
Ambient air temperature, °C	15	18,9	23
Condenser:			
Number of modules	40	69	48
Fan diameters , m	5,6	6,4	9,15
Fan rotation frequency, rev./min	220/115	180/190	—
Power consumption by the electric engine, kW	60/20	75/18	—
Share of power consumed by fans**, %	1,5	1,6	1,7
Condenser basement square, m^2	2900	5480	—

- *at wind speed to 5 m/s.
- ** at operation of all fans with full rotation frequency

Earlier designs of AC had modules with bundles of finned tubes, located vertically or with an angle to the vertical. Exhaust steam is supplied to the tubes from a collector, located at the top, but non-condensing gases (substantially air) as well as condensate are removed from a bottom collector (Fig. 6.15.). Steam and condensate flows are co-current, downward. Air draught, transversally washing the tubes outside, is forced (predominantly, by fans located below) or natural (cooling towers)

Steam-air temperature difference goes down with air movement in a tube bundle, due to air warming, consequently, a length of the tube section, where intensive steam condensation occurs, increases. This section happens to be the shortest at the cold air inlet and the longest at the warmed air exit. Tubes below the active condensation area are filled with a steam-air mixture with a low steam concentration, and a temperature of the condensate, flowing here, goes down approaching to a cooling air temperature. Under negative ambient air temperatures icing is possible in this area causing tube damage.

Some compensation of thermal load of tubes, located on the way of an air flow, can be obtained by changing the finning extent - installation of tubes with the largest space between the ribs at the air inlet and with the smallest space at the air exit. Besides, a design with two condensation stages is applied, aimed to prevention from condensate freezing during the cold weather.

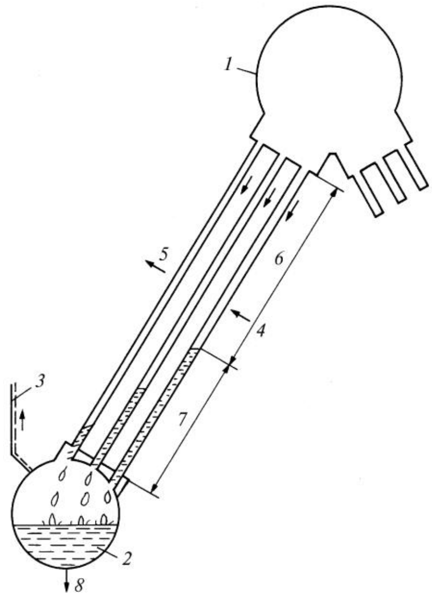

Fig. 6.15. Single-stage air condenser module
1 — Steam distribution collectors; *2* — condensate removal collector; *3* — to the ejector; *4* — cold air; *5* — warmed air; *6* — condensation area; *7* — area of steam-air mixture and condensate cooling; *8* — to a condenser collection tank

Two-stage air condenser GEA (Fig.6.16) has L-formed modules of two types, sequentially connected on the steam way.

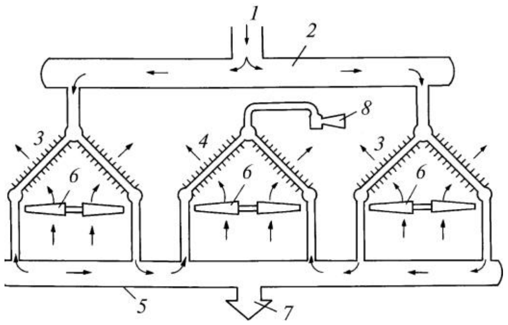

Fig. 6.16. Design of a two-stage GEA air condenser: *1* — exhaust steam; *2* — steam distribution collector; *3* — modules of the first stage of condensation; *4* — module of the second stage of condensation; *5* — collector of steam by-pass and condensate removal; *6* — forcing fan; *7* — to the condensate collection tank; *8* — ejector

Modules of the first stage are designed for condensation of approximately 85% of steam, entering the tubes. At co-crossing flow of steam and condensate in tubes, the condensate temperature remains positive under all operation modes. The remained non-condensed steam is bypassed through the below collector to the counter flow modules of the second stage. Here, the steam, entering the tubes from the bottom and moving towards the flowing down condensate to the bottom collector, warms it. Condensate flows from a bottom collector to the condensate collection tank and non-condensed gases are ejected from the top of this module.

Since all larger ACs are designed with the forced ventilation, their operation is controlled by change in a quantity of operating fans as well as in fan rotation frequency (drive electric engines are usually dual speed). Such air condensers are controlled, depending on a condenser load and an ambient air temperature and also based on condensate temperature data, received from gauges, installed after separate modules.

Fig. 6.17 presents another design of a double-stage condenser, manufactured by C-E Lummus (USA) and by Mitsubissi (Japan) under a license. The difference is that L-shaped modules have U-shaped finned tubes and each module has bundles of both the first and the second stages of condensation. Exhaust steam from a turbine comes to the horizontal steam line, located at the bottom. Branch distribution pipes are coming out from a collector upward at a certain angle to a vertical.

Steam from distribution pipes enters tubes of the first condensation stage and after that together with condensate gets to the down-take pipes, connected with the horizontal condensate removal line. Steam from down-take pipes of the first stage enters tubes of the second stage, having their down-take pipes, connected to the same condensate removal line. The latter has a hydraulic locking at its open end, providing a water level in the bottom part of each down-take pipe. This excludes a possibility of phase separation in the down-take pipe and appearance of steam in it through which under non-uniform pressure drop in working tubes the steam could enter working tubes from the bottom from an outlet section. Down-take pipes of the second condensation stage have bypasses on the top for non-condensing gas ejection. Until now such condensers are applied only at relatively small power plants. They have shown themselves sufficiently well in regions with cold climate conditions (according to the company data –under the temperature of –50°C) at variable operational modes.

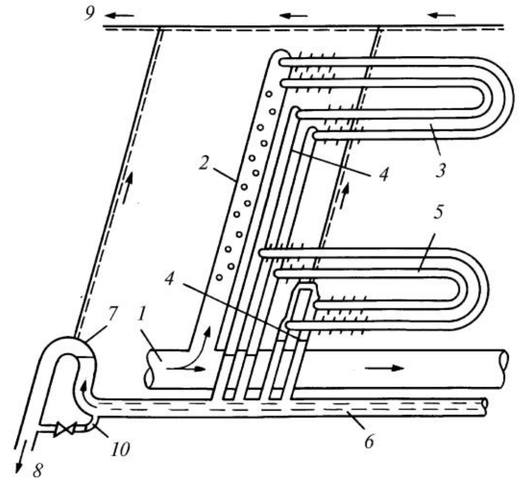

Fig. 6.17. Double-stage condenser module of "C-E Lummus" company: *1* — exhaust steam; *2* — steam distribution collector; *3* — tubes of the first condensation stage; *4* — down-take pipes; *5* — tubes of the second condensation stage; *6* —condensate line; *7* — hydraulic locking; *8* — pipeline to the condensate collection tank; *9* — pipeline to the ejector; *10* — bypass for dewatering

In large GEA air condensers tubes are made from carbon steel with set steel ribs with conic reinforcing rings. Tubes with ribs after appropriate preliminary treatment are immersed into the melted zinc alloy, forming a uniform covering of 60 mc thick. This covering, providing right connection of ribs with tubes, possesses sufficient thermal conductivity and protects finned tubes from corrosion, caused by cooling air. Such pipes are currently in operation. They have been serving without damages for 25-30 years and more.

Tube cross section is an ellipse or an oval with axis correlation 4:1 and more. Tubes are installed in bundles in a chessboard order. In case of condensate freezing under extreme climate conditions, tubes are distorted (expanded in the direction of the small cross section axis) without vivid thinning and damage of the wall due to a prolate cross section of such tubes. As a result there is no need in immediate shutdown of the condenser operation (or a module in case it has a stop valve at a steam inlet from a distribution collector)

Operational experience of the largest in the world air condenser, installed at a power plant Wyodack (USA), is of a great interest. The power plant is located in a water poor, but coal rich region. Air condenser became a dominating element in the construction of the whole unit of a 330 MW. Sizes of the air condenser, quantity of fans and hard operation conditions in summer and winter required a special approach at all stages of designing, development, manufacture, construction, test and operation. Particular attention was paid to operation tests. Information, collected during the operational period, showed positive technical and economic characteristic features of the air condenser and also their environmental safety. The condenser consists of 22 standard condenser groups. Each group consists of three modules. An A-shaped module has six cooling elements. The fan directs cooling air along the axis in a mechanical draft mode. Each cooling element consists of several rows of pipes with steel galvanized ribs. Besides, the mentioned AC includes modules with longer finned tubes with a large cross section. These modules are a part of a condensation system and intended for reduction of fan quantity and simplification of operation in winter period.

Such modules were called as C-modules or condensation modules. Approximately 75% of steam is condensed in a C-module. A so-called D-module or a dephlegmator is located in the center. D-module prevents condensate from supercooling, providing a counter flow of steam-air mixture and condensate. Non-condensing gases are removed from the system by ejectors. A module, consisting of two C-modules and one D-module, is called a C-D group. Modules of such type are in operation in the USA since 1969. These modules, as operation results showed, are more reliable and efficient compared to modules of other AC designs. Since a cooling air flow is the only controlled parameter to achieve the required vacuum, the design stipulates 18 combinations of fan speeds, providing a sufficiently wide regulation range. A certain fan switching mode was determined for each ambient air temperature, yielding the highest efficiency.

More complicated problems occur at operation of ACs under lower ambient air temperatures. Under negative temperatures of the cooling air, condensate freezing took place. However, ellipse-shaped tubes, according to the designers' opinion, prevented from decompression, but deformation of tubes was noticeable. Preparation of the condenser for a start-up does not exceed 30 minutes.

Prolate in the direction of air flow ellipse cross section of tubes leads to decrease in air pressure drop in a tube bundle and facilitates more efficient cleaning of the outer surface of tubes by blow-out. In cases when a heavily adherent dirty layer appears on tubes as a result of a mixture of atmospheric fallout with solids from air, a high-pressure (up to 25 MPa) water jet is used for its removal. This circumstance is one of the substantial arguments towards application of stronger steel ribs instead of aluminum, used in some cases. Hardness of square ribs is increased by distance elements, located at their outward edge.

It should be noted that test and operational data, presented in the work, arouse a certain discredit due to insufficiency of measurements and even accuracy of the measured parameters.

Summarizing the results, the authors of the work make a conclusion that air cooling not only at power plants, but also in industrial enterprises of USA proved it as economically efficient, especially in water-deficit regions, and also as environmentally safe. Application of ACs is quickly developing in the USA due to increase in demand in new capacities and also growing prices for obtaining and transportation of water.

Table 6.29. **Data on condenser operation at Wyodak TPP**

Characteristics	Year				
	1978	1979	1980	1981	1982
Operation hours	3750*	7324	8087	7809	7883
Availability factor, %	76	84	92,1	89,1	90,0
Ambient air temperature, °C					
minimum	-44,5	-29,5	-24,5	-27	-31
maximum	38,5	38,0	41,2	40	41

* since June.

Tab. 6.29 provides data on condenser operation at Wyodak TPP.

Since the base area of the AC is larger than the machinery hall area, ACs are located at a special steel supporting structure or above the machinery hall ceiling (at Utrillas TPP and some smaller TPPs) or in front of it. A sufficient space is left between the condenser platform and a machinery hall ceiling or a ground level for atmospheric air access to fans.

A choice of a dry cooling system and its sizes, as well as its main elements significantly depend on a number of local conditions. As an example, Tab. 6.30 provides results of calculations, performed by GEA for the power unit of 500 MW with steam parameters at a turbine inlet of 17 MPa and 538/538°C in application to the climate conditions of Germany (Essen). Four options are considered: AC and a circulation water supply system with a dry cooling tower (DCT) with a mechanical draft (fans) for both cases and the same two systems with a natural draft (reinforced concrete cooling towers). The following initial data were taken for estimation of all options: exhaust steam flow 1006 t/h, its parameters: pressure 11,7 kPa and temperature 49°C, ambient air temperature 9°C. A provision is made for steel pipes and ribs, treated with hot zinc.

The estimation results of four systems are provided in Fig. 6.18. Turbine characteristics (relationship between the efficiency and backpressure) depend on a specific load of the exhaust surface of the last stages (Fig. 6.19).

Common turbines with a specific load of the exhaust surface of 20 MW/m^2 are not applicable with ACs, as they allow operation without load decrease compared to a nominal value only within a backpressure range of 3…14 Pa.

Table 6.30. **Technical and economic characteristics of dry cooling systems for the unit of 500 MW (17 MPa, 538/538°C)**

Name	Draft			
	Mechanical		Natural	
	System			
	AC	DCT	AC	DCT
	Option number			
	I	II	III	IV
Initial data for estimations:				
Backpressure, kPa	11,7	11,7	11,7	11,7
Ambient air temperature, °C	9	9	9	9
Cooling water (intermediate circuit):				
Flow rate, t/h	—	33200	—	34600
Initial temperature, C		28,7		29,3
Final temperature, °C		46,0		46,0
Water pipeline diameter, m	—	2x2,3	—	2x1,4
Diameter of vacuum pipeline, m	2x6,3	—	—	2x6,0
Thermal exchange surface (from a steam side), m^2	62900	64400	79830	89130
Dimensions, m:				
height	27	30	155	155
basement diameter	—	100	130	121
length	85	—	—	—
width	72	—	—	—
Capital costs, FRG mln. marks	31,15	32,4	45,6	47,0
Maximal capacity of the unit at t_{amb} = 27°C, MW	456,8	417,9	440,0	410,2
Mode A:				
Generation (net), GW/year	3573,2	3734,3	3706,7	3682,3
Average specific heat consumption , kJ/(kW·h)	8985	9043	8637	8675
Electricity cost price, Pf. FRG/(kW·h)	5,327	5,504	5,234	5,370
Mode B:				
Generation (net), GW·h/year	1973,5	1960,2	1997,9	1990,6
Average specific heat consumption , kJ/(kW·h)	9043	9136	8683	8713
Electricity cost price, Pf. FRG/(kW·h)	6,520	6,727	6,257	6,672

A turbine is taken for calculations, modified by application of mechanically reinforced blades of shorter length for the last stages, and herewith by increase in specific load of the exhaust surface to 30 MW/m^2, yielding widening of the allowed backpressure range. At the accepted load of the exhaust surface the range changes from 5 to 25,5 kPa.

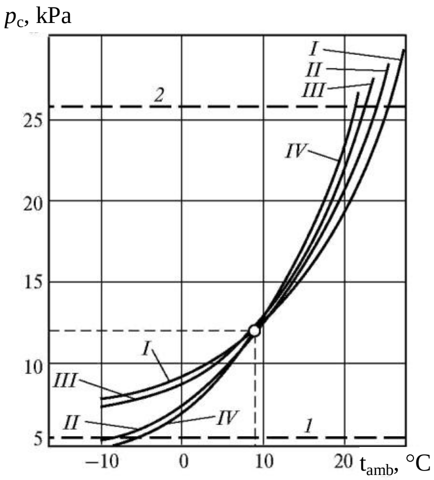

Fig. 6.18. Characteristics of condensing-cooling units of 500 MW:
1 and 2 — minimal and maximal backpressure of the turbine; *I*, *II*, *III*, *IV* — option numbers from Tab. 6.30

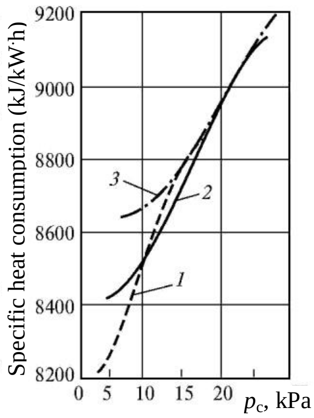

Fig. 6.19. Correlation between specific heat consumption (kJ/kW·h) and backpressure p_c at different specific loads of the exhaust surface of the last turbine stages: *1* — load of 20 MW/m^2; 2 — load of 30 MW/m^2; *3* — load of 40 kW/m^2

Economic characteristics are determined for two operational modes of the unit: base mode with annual use of capacity equal to 7512 hours (mode A) and midrange mode with a capacity use of 4000 hours (mode B). For modes A and B the following was, consequently, accepted: organic fuel cost 4,78 and 3,58 FRG marks/kJ, interest of capital, % - 15 and 20%, cost for replacement power energy 3 and 5 pf FRG/(kW·h) and replacement capacity 800 and 600 pf FRG/(kW·h). As it could be seen from Tab. 6.30 it was obtained for the given conditions that AC is in all cases more cost efficient than a circulation system with a dry cooling tower. Capital costs for a condensation-cooling unit with AC are by 4% (mechanical draft) and by 3% (natural draft) lower that for a dry cooling tower and electricity cost price is lower by 3,3 and 2,6%. However, it should be taken into account that results of comparison of different cooling systems may change if climate conditions differ from the taken for the above calculation and optimization of initial parameters is performed for each of the compared options.

Design of AC for Utrillas TPP determined that operation of the unit without its available power limit must be provided at the ambient air temperature t_{amb} to +30°C, but during the test it was reached at t_{amb} to +34°C and exhaust steam pressure 29 kPa, i.e. in this case operation of the unit with nominal capacity is possible within almost the whole ambient air temperature range. Contrary to that optimization calculations, performed for a unit of Wyodak TPP, led to a decision when increase in t_{amb} above the calculated value at operation of all fans with a nominal frequency results in decrease of available unit capacity.

According to the design data, the same also refers to the units of Matimba and Maiba TPPs (Fig. 6.20). Additional decrease in available capacity can be caused by influence on AC operation of wind with its speed of 5…6 m/s and higher as well as its unfavorable direction (according to tests at the AC model at Wyodak TPP it can reach 5…12% at clean condenser surface).

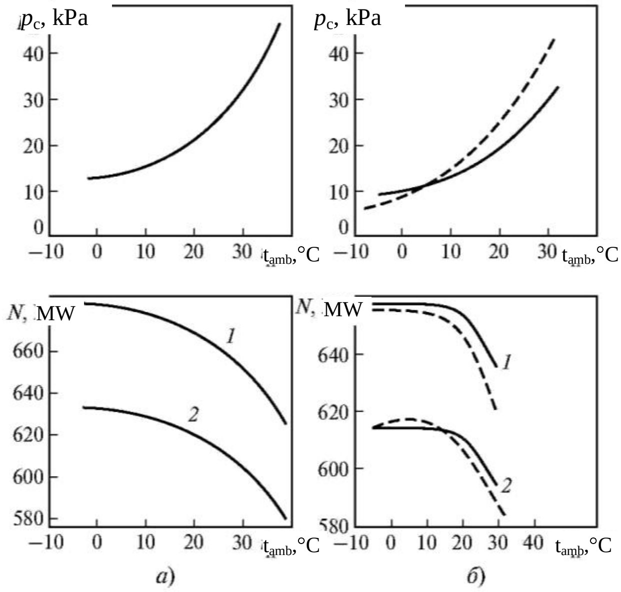

Fig. 6.20. Backpressure and available capacity *N* (MW) of a turbine (power unit) with air condenser, depending on ambient air temperature t_{amb}:
а —power unit of Matimba TPP; *б* — power unit of Mayba TPP; *1* — turbine capacity (gross); *2* — power capacity (net). Dashed lines — capacities at circulating system with a dry cooling tower

In order to balance a lack of power generation (under the terms of power system operation) in a hot period, in case of absence of a stand-by capacity and impossibility of its obtaining from another power system, the following means of exhaust pressure drop can be applied:

1) decrease in a temperature of atmospheric air, entering the condenser due to injection into it of the clarified and softened water, which evaporates before air entering the fan (or to the tube bundle).

2) condensation of a part of exhaust steam (for example, 20%) in an auxiliary water supply system, specially constructed for this purpose, equipped with a surface condenser, cooled with water and a traditional mechanical draft cooling tower of evaporation type.

Both methods require additional costs and water supply to cover evaporation losses.

Overwhelming majority of the published works presents only general designs of ACs without characteristics of various operational modes and other information on characteristic features of operation. Mainly, advertising information is presented in general publication. Thus, [2] informs that TPPs with ACs, located in USA, are situated in water-deficit regions close to fuel resources and in the northern regions.

However, for almost all ACs of condenser-dephlegmator type, provision of a reliable and efficient operation of the dephlegmator module under negative ambient air temperatures is a burning issue.

Another disadvantage of such units with air cooling is that steam, to be cooled before supply to the heat exchange surface, must be distributed in a given proportion between a dephlegmator heat exchange element, installed at a steam supply side and a dephlegmator element, operating at an air supply side of the heat exchange element, working in a condensing mode, that arouses difficulties at flow distribution, as a condensing unit must be operated at a changing load. In [3] a solution of this task is proposed by means of heating of cooling air for dephlegmator elements. Thus, efficiency of operation of heat exchange surfaces, connected by a dephlegmator, in relation to condensing surfaces remains, thus neither heat-transfer goes up, nor change in relation between dephlegmator and condensing surfaces takes place. In the authors' opinion it excludes an increase in heat exchange surface of the condensation unit and prevents from a risk of freezing of dephlegmator heat exchange elements even under operation at extremely low ambient air temperatures and at part-load unit operation. The unit scheme is provided in Fig. 6.21.

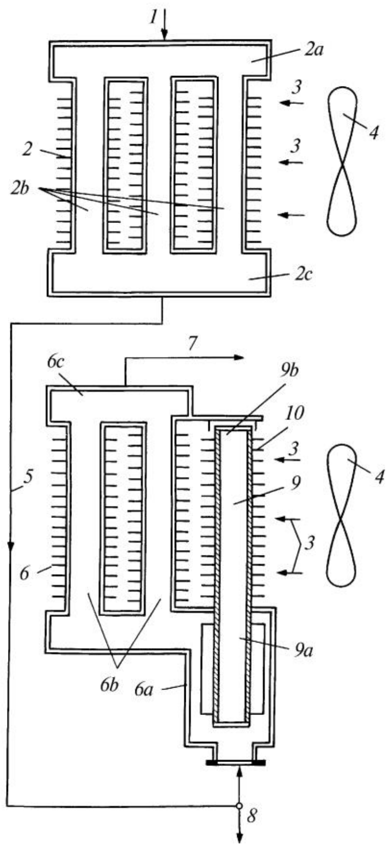

Fig. 6.21. Scheme of the unit of a condenser-dephlegmator type [3]

Steam *1*, enters the distribution chamber *2a* of the heat exchange element *2*, and is divided into many streams in tubes. Since a distribution chamber *2a* of a heat exchange element *2* is located higher than the collection chamber *2c*, a part of steam flow moves in finned tubes *2b* in the same direction as condensate, formed as a result of cooling. This refers to a condenser operating element – heat exchange element *2*.

The heat exchange element *2* operates at a surplus steam. Condensed steam is supplied to the heat exchange element *6* through the line *5* of the distribution chamber *6a*, located at the bottom of the heat exchange element *2*.

Heat exchange element 6 is provided with two lines of finned tubes *6b* of the directed flow *3*, where cooling air is pumped by a fan *4*; by that, the finned tubes are connected with each other by a collection chamber *6c*.

Since partial steam flows, directed through the finned tubes *6b* in this heat exchange element are counter-flow to the condensate, then here is referred to a dephlegmator connection element *6*.

Gases, in particular air and inert gases, collected in the collection chamber *6c* of the heat exchange element *6*, are removed, using an intake line *7*.

Condensate from both heat exchange elements *2* and *6* is discharged from the system through the condensate removal line *8*.

Before the first raw of the finned tubes *6b* at a cooling air side there is a raw of heating tubes *9* which are closed and filled with a heat carrier. Ammonia can be used as a heat carrier. Heat carrier, in heating tubes *9*, evaporates in a lower part of the heating tubes due to steam, entering the distribution chamber *6a*. This part of the heating tubes *9* is called an evaporation part *9a*. Since the upper part of the heating tubes *9* is under the flow of cold cooling air, heat carrier condensers in the upper part of the heating tubes *9*, called a condensing part *9b*. Steam, entering the distribution chamber *6a*, gives as much heat as needed for evaporation of the heat carrier in the evaporation parts *9a* of the heating tubes *9*. The evaporated heat carrier heats the tubes. Heat through the walls of tubes and lateral ribs *10* of condenser parts *9b*, is transferred to the cooling air. Therefore, the cooling air, entering the first raw of the finned tubes *6b* is thus heated, that temperature of walls of the first raw of tubes does not reach freezing temperature even under extremely low temperatures of the cooling air. This could be particular dangerous for the finned tubes *6b*, because at the upper end of the dephlegmatorly working finned tubes *6b*, a frost layer (freezing) may occur, if the wall temperature falls lower than the freezing temperature. Conditions are favorable for freezing because condensation stops at the upper part of the finned tubes *6b* due to high content of inert gases, so residue steam can cause tube wall icing.

In [4] main aspects are described that should be considered under determination of aerodynamic characteristics of ACs. Influence of distortion of an air flow speed in finned heat-exchangers is of particular attention because it can cause certain problems in operation that often result in main difficulties. It is noted that in order to increase an efficiency of ACs, the following factors should be considered in process of design: fan design, influence of natural convection on heat and aerodynamic characteristics, direction and strength of natural air flows.

Particular attention in [4] is paid to the accuracy of measurements of air flows. The authors notice that there are a lot of factors, studying which thermal and aerodynamic characteristics of ACs can be improved. These factors basically refer to understanding of air flow behavior at the inlet of AC, in rows of finned tubes and also at AC outlet.

It should be noted that to date in literature information about behavior of tube bundles, installed at different angles toward the cooling air flow, is very limited and dissipated. Most of it refers to bundles of smooth tubes with corridor and chessboard location.

The issue relating to a choice of rational design of ACs remains unsolved till now. At power plants, operating at a Rankin cycle, the most important parameters, used for determination of the optimal AC design are as follows:

- minimal front area
- minimal heat transfer surface;
- maximal cycle efficiency (net).

All these characteristics depend on a temperature in the condenser and cooling air speed. As all these parameters change, then naturally the designer should handle with optimal values, yielding achievement of maximum efficiency.

Stuart, et al [5] conducted optimization calculation of ACs. As a result, a temperature in the condenser was obtained correspondent to minimal front surface. Air velocity was also determined correspondent to a minimal heat transfer surface at a constant temperature in AC and constant useful capacity (determined as total capacity minus AC capacity of fans). In all cases the authors made an assumption that air temperature at AC outlet is equal to the saturation temperature in AC.

In [6] the results of optimization of unit costs of heat transfer surface are provided. However, for engineering calculations it is preferred to have general expressions for determination of both velocity of a cooling air and temperature in AC, providing optimal heat transfer surfaces, a front surface and net efficiency.

Referring to these tasks, a work [7] is of interest. In this work front surface, heat transfer surface and power plant efficiency are optimized, depending on the cooling air velocity and temperature in AC. The following assumptions are taken into the work:

1) Temperature of the cooling air at AC outlet is equal to saturation temperature in AC;

2) Tubes and their ribs form wide square cells in AC;

3) Ribs of tubes are very thin compared to the distance between them, so front surface of AC is the same as at the condensing section inlet;

4) Cooling air speed in AC is constant and equal to an average speed at the inlet of the condensing section;

5) Cooling air in AC is heated by the surface of ribs and the tube surface, having a saturation temperature.

Algebraic dependences were obtained, connecting independent parameters with a possible optimal design. The calculation results showed that high speeds of cooling air are recommended, although they are accompanied by increase in capacity, consumed by fans. Such conclusion arouses doubts in consistency of the carried out investigations and the results obtained. Besides, assumption on equality of the saturation temperature and air temperature at AC outlet can be accepted for an extreme case - indefinitely large heat exchange surface. In real conditions AC has a definite surface, that leads to a temperature difference between the saturation temperature and air temperature at AC outlet and can make δt = 5…7°C and more.

Assumption on temperature equity of a rib surface and a tube surface to the saturation temperature leads to a systematic error, as it does not consider thermal resistance of the tube wall and a temperature change along the rib length. All these leads to significant error at determination of heat transfer coefficient and, as a result, of condenser efficiency.

There are known the ACs, having bundles of tubes, installed vertically between two lattices, and a fan, installed along an apparatus axis [8].

One of the methods to eliminate condensate freezing at the outlet of the tube bundle is a method implemented in AC, presented in Fig. 6.22. Condenser [9] represents an inverted-V section with a mechanical draft air cooling, containing bundles of finned condenser tubes *1*. Tubes are connected at one end with an upper steam distribution chamber *2* and at the other end - with a lower chamber *3*.

In order to improve AC reliability under low cooling air temperatures, a counter flow of steam is provided in tubes, arranged due to installation of a condensate drain *4*, perpendicular to a tube axis. Steam, supplied from a steam distribution chamber 2, condensers in an upper part of the tubes and condensate is removed from a condensate drain and additional central tube *5* to a condenser collector *6*.

In an upper part of the additional central tube holes *7* are provided for steam air mixture exhaust, located lower than the condensate drain.

Such AC design yields reduction of condensate path along a cold inner wall of the upper tube part in the first tube row along the cooling air. Additional tubes consist of two parts of different diameters, installed one into another resulting in compensation of thermal extension.

Condensate collector *6* is equipped with a branch pipe *8* for steam-air mixture exhaust, connected with an air exhaust pipe *9* and branch pipe *10* for condensate removal to the condensate pipeline *11*. In the lower part of the branch pipeline *3* a branch pipe line *12* is installed for condensate discharge to the condensate collector *13*. Parts of pipes, located between the condensate drain and condensate collector, are executed with a growing length in a steam flow direction through collectors *3;* parts of tubes between an upper collector *2* and condensate drain *4* are executed with a growing length in each following raw along the cooling air (depthward the tube bundle).

Air condensation unit of Novomoskovsky manufacturing unit “Azot” is of a certain interest. It is used for condensation of the exhaust water steam after driving turbines of compressor, used for compression of natural gas. Air condensation unit is manufactured in 1972 by GEA (FRG).

Technical characteristics of air condensation units are presented in Fig. 6.31.

At Cherkassky chemical complex “Azot” an air condensation unit of “Lummus” firm is installed. Operation revealed the following problems:

- Freezing of condensate in horizontal tubes in winter and damage of tubes, mainly, in non-ribbed bends;
- Necessity of irrigation of heat transfer surface with condensate in a hot period for vacuum deepening;
- Insufficient operation of a reducing gear for the fan drive, it appeared to be a week unit of transmission;
- Necessity of heat transfer surface cleaning twice a year from depositions on ribs. Nature of depositions was not determined and is classified as dirt, washed with water.

At the same time corrosion at aluminum ribbing was not recorded.

In general, operational experience of the air condensation unit is estimated as positive.

- A particular problem of AC operation is control of its operational mode. It consists in maintaining of optimal pressure at the turbine exhaust in conditions of variable turbine loads and variable temperature of a cooling air, which depends on weather and season. Quality of control consists in achievement of sufficiently sensitive cooling air flow adjustment at maintaining maximum efficiency. It is achieved by the following methods:
- Change in angular inclination of fan blades;

- Change in power engine rotation frequency;
- Screening of the cooling surface of modules (jalousie);
- Fan shutdown;
- Water injection into air flow
- Cooling air temperature regulation by its circulation.

Table 6.31. **Technical characteristics of air condensation units 102-JC and 105-JC**

Name	Type of air condensation units	
	102-JC	105-JC
Heat accepted, kcal/h	$9{,}85 \cdot 10^6$	$27{,}1 \cdot 10^6$
Steam flow rate, kg/h	16650	52700
Initial steam temperature, °C	70,2	70,2
Final steam temperature, °C	67,0	67,0
Absolute operation pressure, 10^5 Pa	0,32	0,32
Pressure losses, 10^5 Pa:		
calculated	0,0173	0,0173
allowable	0,04	0,04
Surface with ribbing, m^2	12534	37480
Tubes cooling surface , m^2	986	2955
Number of tubes, pieces	1980	4752
Air temperature at the inlet, °C	28	28
Air temperature at the exhaust, °C	56	56
Air flow rate, m^3/h	1400000	3600000
Fans quantity, pieces	3	
Number of tube modules in a section, pieces	4	4
Number of tube rows in a module	3	3
Number of sections in a unit, pieces	3	6
Total capacity of fans, kW	120	258
Number of fan blades, pieces	6	6
Tube bundle air resistance, mm of water column	15,5	13,9
Calculated atmospheric pressure, mm of mercury column	730	730
Calculated value of relative air humidity, %	45	45
Weight of the unit, t	63	190

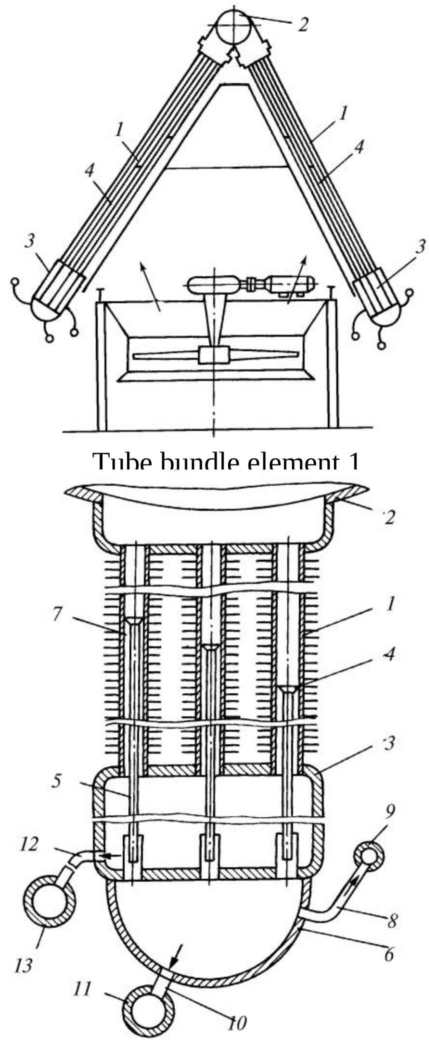

Fig. 6.22. Condenser design with an inverted-V section