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FAN PERFORMANCE IN AIR-COOLED STEAM CONDENSERS

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Abstract—It has in recent years been found that air-cooled steam condensors consisting of arrays of A-frame heat exchanger units, incorporating axial flow fans, have not been performing satisfactorily at a number of power plants. Subsequent tests have shown that the cooling air flow rate was inadequate in certain cases. In part this was shown to be due to an underestimation of the losses experienced by the air stream flowing through the A-frame unit. An analysis is presented in which the various losses are identified and a numerical example is given that clearly illustrates the significance of so-called "secondary losses" in a practical air-cooled condensor unit. It is shown that the sum of the "secondary losses" may be of the same order as that of the heat exchanger bundles under normal flow conditions.

NOMENCLATURE

A area, m^2
C_D drag coefficient
c_{pa} specific heat, J/kg K
d diameter, m
H height, m
K loss coefficient, $\Delta p/(\rho v^2/2)$
L length, m
m mass flow rate, kg/s
n number
P power, W
p pressure, N/m^2
Δp pressure differential, N/m^2
R gas constant for air, J/kgK
s tip clearance, m
T temperature, °C or K
V volume flow rate, m^3/s
v velocity, m/s
w width of fan unit, m
x distance, m

Greek letters

α_e kinetic energy coefficient
η efficiency
μ dynamic viscosity, kg/ms
ρ density, kg/m^3
σ area ratio
θ angle, °

Subscripts

a air
b bundle
c casing or contraction
d downstream
do downstream
F fan
F_l large fan
F_s fan static
fr frontal
h hub
he heat exchanger
i inlet
j jetting
l large fan

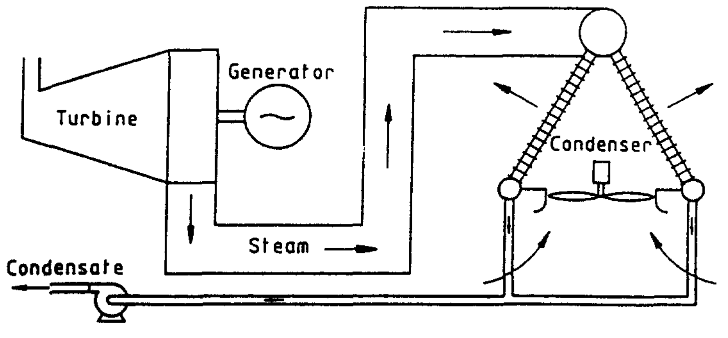

Fig. 1. Direct air-cooled condenser.

m mean
min minimum
o outlet
ob obstacle
pl plenum
r reference
s steam
t total
ts heat exchanger support
up upstream
w walkway
θ oblique or inclined.

INTRODUCTION

Air-cooled heat exchangers have found application in the chemical, process and other industries for many years. More recently an increasing number of air-cooled steam condensers have been constructed at power plants located in areas where cooling water is unavailable or costly. In the so-called "direct system", turbine exhaust steam flows directly through a large duct to finned tube heat exchanger bundles arranged in the form of A-frames, as shown in Fig. 1. Axial flow fans force ambient cooling air to flow across the tubes causing the steam to condense.

There is at present only one relatively large operational air-cooled (also referred to as dry-cooled) power plant owned by a utility in the U.S.A., i.e. the 330 MWe Wyodak plant [1]. Faced with limited siting options, non-utility power producers, however, increasingly rely on reduced water use and environmental effects from dry-cooling to avoid costly licensing and permitting delays. The steady growth of annually installed dry-cooled generation capacity in the U.S.A. is shown in Fig. 2 [2].

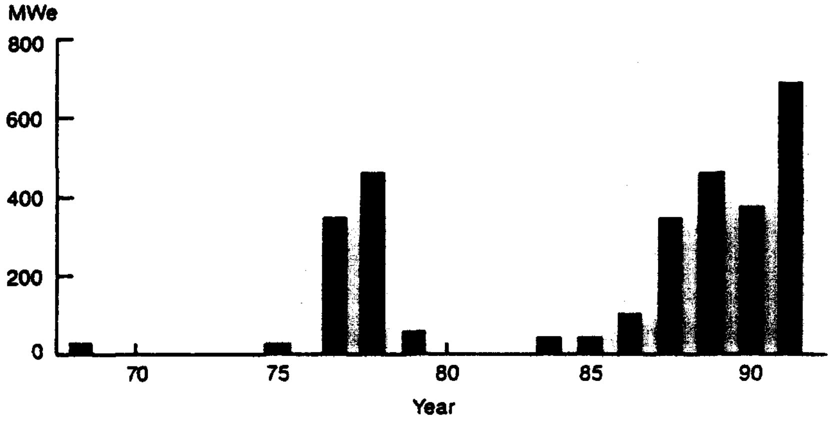

Fig. 2. Growth of annually installed dry-cooled generation capacity in the U.S.A.

Outside the U.S.A., dry-cooling has been used successfully for full-scale utility applications in many countries and especially in locations where water is scarce, such as the Middle East and South Africa, where the world's largest direct air-cooled condenser was recently completed at the Matimba power plant [3, 5]. This plant has an output of 6 × 665 MWe while almost 5500 MW heat is rejected by the air-cooled condenser.

Dwindling supplies of cooling water and a lack of adequate plant sites, rapidly rising water costs usually at well beyond inflation rates in most industrialized countries, environmental consideration and proliferating legislation all suggest that the need for air-cooled condensers will increase. In view of this trend, it is essential that reliable methods are applied in the design of such systems to ensure their ultimate effective performance. Unfortunately, it would appear that some smaller operational air-cooled condensers do not achieve specified performance requirements. To a large extent this is due to the fact that in addition to the primary air flow resistance or pressure drop through the finned tube heat exchanger bundle, not all secondary flow resistances are considered during the design stage, with the result that fans do not deliver the required flow rate. It is interesting to note that in the latest code for testing air-cooled heat exchangers, certain heat transfer equations are presented but information on the fan system and related flow losses is conspicuously absent [7]. It should be stressed that in any air-cooled heat exchanger design the fan system is as important as the heat exchanger.

FAN SYSTEM CHARACTERISTICS

The performance characteristics of axial flow fans are determined according to prescribed standards which are continuously being updated. Examples of well known standards are those of the British Standards Institution [8] and the Air Moving and Conditioning Association [9]. According to these standards, full details of test conditions and fan installation type are supplied with the fan performance characteristics. There are basically four standard fan installation types that are internationally recognised.

1. Installation type A: free inlet, free outlet.
2. Installation type B: free inlet, ducted outlet.
3. Installation type C: ducted inlet, free outlet.
4. Installation type D: ducted inlet, ducted outlet.

For the air-cooled condenser unit shown in Fig. 1, the installation type A is the most appropriate for evaluating fan performance. The performance characteristics of very large fans are usually obtained by applying the fan laws to test results obtained on smaller fan models. An example of performance curves for a large fan, obtained by testing a model fan in such an installation and scaling the data with the aid of the appropriate fan laws [8], is shown in Fig. 3.

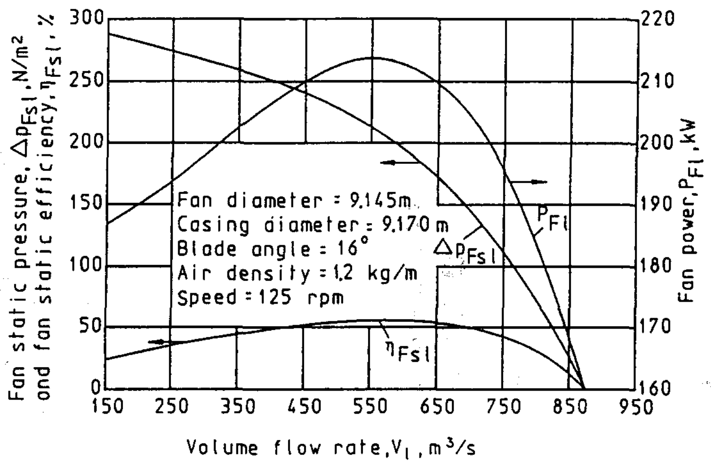

Fig. 3. Fan performance characteristics.

According to VDI standards [10], it is noted that if the Reynolds number based on the diameter of the test fan and that based on the diameter of the larger fan are different, the dependence of the frictional losses on this number should be considered. Due to boundary layer effects, the efficiency of the larger fan may be expected to be higher than that of the smaller test fan but no allowance for this scale effect should be made according to most standards unless otherwise agreed, demonstrated by adequate evidence, between manufacturer and purchaser. The VDI standard [10] gives the following conversion rule for tip clearance in fans of different diameters at a particular density:

$$s_{\mathrm{F1}} = s_{\mathrm{F}}(d_{\mathrm{F1}}/d_{\mathrm{F}})^{0.8}(\Delta p_{\mathrm{F}}/\Delta p_{\mathrm{F1}})^{0.1}$$

In most industrial applications a recommended difference between the fan casing diameter and the fan diameter (two times tip clearance) is of the order of one per cent of the fan diameter [11]. Venter and Kröger [12] find that over the practical operating range of a fan, i.e. in the region of maximum efficiency, both the fan static pressure and air volume flow rate through the fan decrease linearly with increasing tip clearance, as is shown in Fig. 4. Noise levels also generally increase with increasing tip clearance [13]. In view of the demand for high efficiency and quiet operation, small tip clearances are thus desirable.

Losses of mechanical energy or head losses (pressure drops) between any two sections of the system can be expressed in dimensionless form by a loss coefficient defined as

$$K = \frac{(p_1/\rho + \alpha_{e1} v_1^2/2) - (p_2/\rho + \alpha_{e_2} v_2^2/2)}{(v^2/2)}. \tag{1}$$

The interaction between the fan and the installation flow resistances (known as the system or installation effect) has been recognised and investigated in numerous studies. Venter and Kröger [14] conducted a survey of methods that predict the influence that upstream and downstream flow resistances (obstacles), such as fan support structures, screens and walkways, have on the performance of the fan. They conclude that for reasonably uniformly distributed resistances, the so-called bulk method [6] satisfactorily predicts effective pressure loss coefficients. The loss coefficients based on the velocity through the fan, for resistances created by obstacles located on the upstream or suction side, K_{up}, and the downstream or discharge side of the fan, K_{do}, are shown in Figs 5 and 6 respectively. These coefficients are a function of the projected area of the obstacle, A_{ob}, the

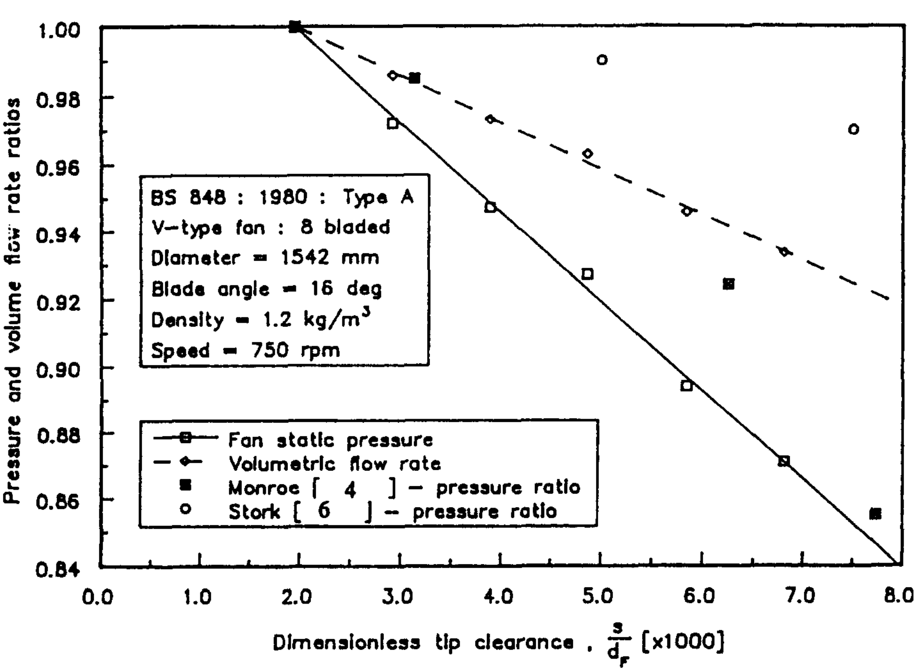

Fig. 4. Effect of tip clearance on fan performance.

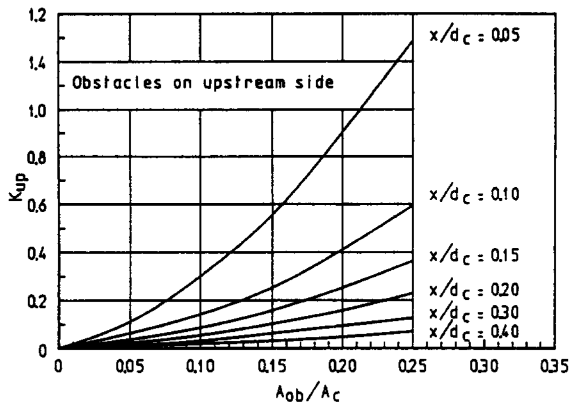

Fig. 5. Upstream loss coefficient due to obstacles.

obstacle distance x from the fan, the casing diameter d_c and the cross-sectional area A_c respectively.

In addition to the losses due to the obstacles close to the fan, losses are experienced in the plenum after the fan. For the A-frame configuration shown in Fig. 1, where the apex angle is approximately 60° and the area ratio of the fan cross-sectional to the total frontal heat exchanger area $A_F/A_{fr} \approx 0.3$, the effective plenum loss coefficient based on the mean velocity through the fan, is approximately equal to the kinetic energy coefficient, i.e. $K_{pl} \approx \alpha_{eF}$ as measured at the outlet of the fan when tested near its maximum efficiciency according to British Standard BS 848, test type A. For all practical purposes this means that the kinetic energy of the air leaving the fan under these test conditions is effectively lost in the plenum of the A-frame configuration. This was confirmed in a test conducted on a complete model of the A-frame fan unit attached to the outlet of a test type A installation. This is another reason why the type A fan test installation is recommended when performance characteristics are required for fans to be installed under A-frames.

Furthermore, losses are experienced by the air stream as it flows through the A-frame shown in Fig. 7. Other than the heat exchanger loss coefficient, K_{he}, significant inlet losses $(K_{i\theta} - K_e)$ due to the oblique approach of the air stream before the heat exchanger, jetting losses, K_{dj}, and losses due to a distorted outlet velocity distribution must be considered. Van Aarde and Kröger [15] present an empirical equation that takes into consideration these losses, i.e.

$$K_{\theta t} = K_{he} + \frac{2}{\sigma_{min}^2}\frac{(\rho_1 - \rho_2)}{(\rho_1 + \rho_2)} + \frac{2\rho_2}{(\rho_1 + \rho_2)}\left(\frac{1}{\sin\theta_m} - 1\right)\left[\left(\frac{1}{\sin\theta_m} - 1\right) + 2K_{ci}^{0.5}\right] + 2\rho_1(K_{dj} + K_0)/(\rho_1 - \rho_2), \quad (2)$$

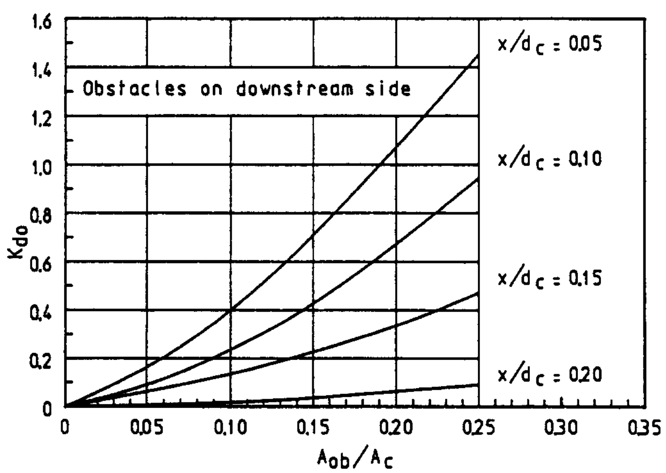

Fig. 6. Downstream loss coefficient due to obstacles.

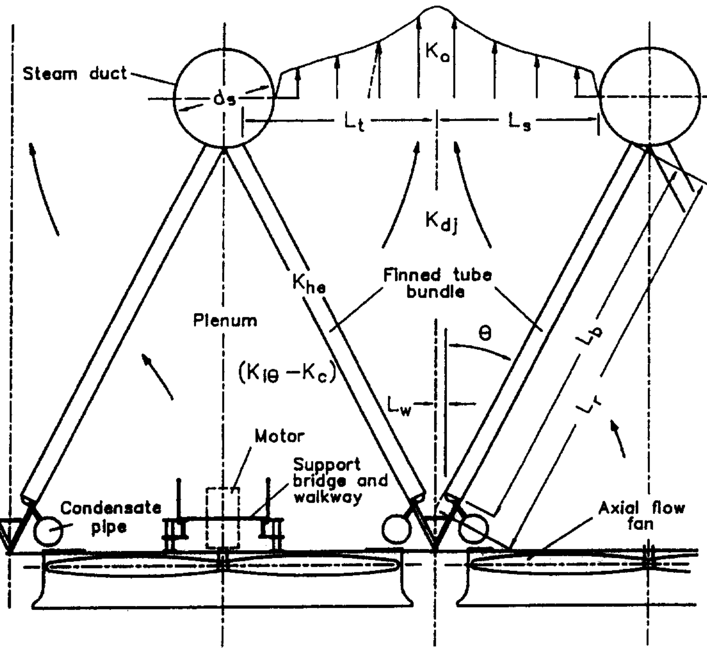

Fig. 7. Flow resistances through an array of A-frame heat exchangers.

where $\theta_m = 0.0019\theta^2 + 0.9133\theta - 3.1558$ and θ is the semi-apex angle in degrees.

The appropriate approximate inlet contraction loss coefficient based on the normal upstream velocity for turbulent flow between plates is given by Kays [16] as

$$K_{ci} = [(1 - 1/\sigma_c)/\sigma]^2, \tag{3}$$

where σ is the ratio of the flow area between the fins at their leading edge to the corresponding area immediately upstream of the fins. For plate fins, σ_c is given by

$$\sigma_c = 0.6144517 + 0.04566493\sigma - 0.336651\sigma^2 + 0.4082743\sigma^3 + 2.672041\sigma^4 - 5.963169\sigma^5 + 3.558944\sigma^6.$$

The jetting loss coefficient is expressed by the following relation:

$$K_{dj} = \left[\left\{-2.89188\left(\frac{L_w}{L_b}\right) + 2.93291\left(\frac{L_w}{L_b}\right)^2\right\}\left(\frac{L_b}{L_s}\right)\left(\frac{L_t}{L_s}\right)\left(\frac{28}{\theta}\right)^{0.4} + \left\{\exp(2.36987 + 5.8601\times 10^{-2}\theta - 3.3797\times 10^{-3}\theta^2)\left(\frac{L_s}{L_t}\right)\right\}^{0.5}\left(\frac{L_b}{L_r}\right)\right]^2, \tag{4}$$

where θ is in degrees.

The outlet loss coefficient is given by

$$K_0 = \left[\left\{-2.89188\left(\frac{L_w}{L_b}\right) + 2.93291\left(\frac{L_w}{L_b}\right)^2\right\}\left(\frac{L_s}{L_t}\right)^3 + 1.9874 - 3.02783\left(\frac{d_s}{2L_t}\right) + 2.0187\left(\frac{d_s}{2L_t}\right)^2\right]\left(\frac{L_b}{L_s}\right)^2. \tag{5}$$

These equations are valid for the $K_{he} \geqslant 30$, semi-apex angles of $20° \leqslant \theta \leqslant 35°$ and for $0 \leqslant d_s/(2L_t) \leqslant 0.17886$ and $0 \leqslant (L_w/L_b) \leqslant 0.09033$.

DRAFT EQUATION

Consider the example of a forced draft air-cooled steam condenser as shown schematically in Fig. 8. This unit may be one of many in an array of A-frames surrounded by a windwall. If the ambient air far from the heat exchanger is dry and the temperature gradient is according to the dry adiabatic lapse rate (DALR) 0.00975°C/m, the pressure difference between the ground at 1 and elevation 8 that coincides with the top of the heat exchanger is given by

$$(p_{a1} - p_{a7}) = p_{a1}[1 - (1 - 0.00975H_7/T_{a1})^{3.5}] \approx p_{a1}[1 - (1 - 0.00975H_6/T_{a1})^{3.5}] + p_{a6}[1 - \{1 - 0.00975(H_8 - H_6)/T_{a1}\}^{3.5}]. \quad (6)$$

In practice the DALR rarely occurs in the first 10 m above ground level and only becomes more meaningful for units where $H_3 > 10$ m.

The same pressure difference as given by equation (6) exists between 1 and the top of the heat exchanger at 7. By taking into consideration changes in elevation, consecutive flow losses and the presence of the fan, find

$$\begin{aligned} p_{a1} - p_{a7} = {} & p_{a1}[1 - (1 - 0.00975H_6/T_{a1})^{3.5}] + K_{ts}(m_a/A_2)^2/(2\rho_{a2}) \\ & + K_{up}(m_a/A_3)^2/(2\rho_{a3}) - \rho_{a3}P_F/m_a + K_{do}(m_a/A_4)^2/(2\rho_{a4}) \\ & + K_{pl}(m_a/A_c)^2/(2\rho_{a4}) + K_{\theta t}(m_a/A_{fr})^2/(2\rho_{a56}) \\ & + p_{a6}[1 - \{1 - 0.00975(H_7 - H_6)/T_{a6}\}^{3.5}], \end{aligned} \quad (7)$$

where $K_{\theta t}$ as given by equation (2) includes losses across the heat exchanger and kinetic energy losses at the outlet elevation 7. The loss coefficient through the heat exchanger supports can be expressed as follows:

$$K_{ts} = C_{Dts}L_{ts}d_{ts}n_{ts}/A_2, \quad (8)$$

where C_{Dts} is the drag coefficient of the support beam and L_{ts} and d_{ts} are its length and diameter respectively, while n_{ts} represents the number of beams and A_2 is the corresponding peripheral heat exchanger inlet area.

For this configuration the plenum loss coefficient $K_{pl} = \alpha_{eF}$ and it thus follows that

$$-\rho_{a3}P_F/m_a + K_{pl}(m_a/A_c)^2/(2\rho_{a4}) \approx -K_{Fs}(m_a/A_c)^2/(2\rho_{a3}), \quad (9)$$

where the fan static coefficient is defined as

$$K_{Fs} = 2\rho_{a3}\Delta p_{Fs}/(m_a/A_c)^2 = 2P_F\rho_{a3}^2A_c^2/m_a^3.$$

Substitute equation (6) into equation (7) and find with equation (9) the draft equation for the air-cooled condenser shown in Fig. 8.

$$\begin{aligned} & p_{a1}[\{1 - 0.00975(H_7 - H_6)/T_{a6}\}^{3.5} - \{1 - 0.00975(H_7 - H_6)/T_{a1}\}^{3.5}] \\ & = K_{ts}(m_a/A_2)^2/(2\rho_{a1}) + K_{up}(m_a/A_e)^2/(2\rho_{a3}) - K_{Fs}(m_a/A_c)^2/(2\rho_{a3}) \\ & \quad + K_{do}(m_a/A_e)^2/(2\rho_{a3}) + K_{\theta t}(m_a/A_{fr})^2/(2\rho_{a56}), \end{aligned} \quad (10)$$

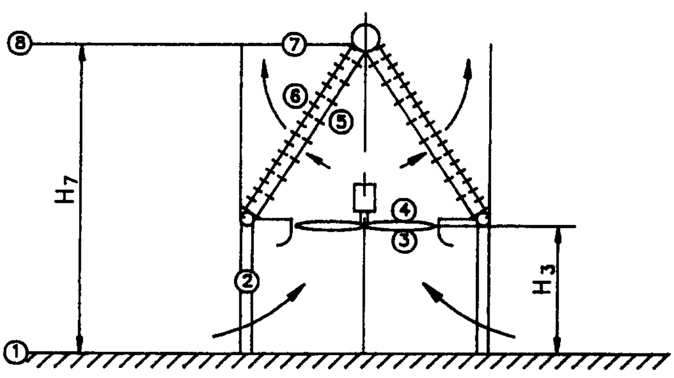

Fig. 8. Air-cooled condenser.

where $H_8 = H_7$ and it is assumed that $\rho_{a2} \approx \rho_{a1}$, $\rho_{a4} \approx \rho_{a3}$ and $\rho_{a7} \approx \rho_{a6}$. Furthermore $A_3 = A_4 = A_e = (A_c - A_h)$ where A_c and A_h are the fan casing and hub cross-sectional areas respectively.

The approximate air temperature at Section 3 is

$$T_{a3} \approx T_{a1} - 0.00975 H_3, \tag{11}$$

while the corresponding air density is

$$\rho_{a3} \approx p_{a1}/(RT_{a3}). \tag{12}$$

For given finned tube heat transfer characteristics, the air temperature T_{a6} after the heat exchanger can be determined. The corresponding air density is given by

$$\rho_{a6} \approx p_{a1}/(RT_{a6}) \tag{13}$$

and the harmonic mean density through the heat exchanger

$$\rho_{a56} = 2p_{a1}/[R(T_{a5} + T_{a6})]. \tag{14}$$

The mean temperature immediately upstream of the heat exchanger is approximately

$$T_{a5} = T_{a1} + P_F m_a / c_{pa1} - gH_5 / c_{pa1}. \tag{15}$$

Once the operating point of a particular fan in an air-cooled condenser unit has been determined, it is important to confirm that this point is at an air flow rate that is at, or preferably higher, than that where the maximum fan efficiency occurs. If this is not the case problems may be experienced at lower ambient air temperatures and during windy periods.

The relative significance of the different flow loss coefficients as given by equation (10), but all based on the frontal area of the heat exchanger, are best illustrated by means of a numerical example.

NUMERICAL EXAMPLE

To obtain some idea of the relative magnitudes of the different flow loss coefficients, consider a practical air-cooled condenser unit, as shown in Fig. 8.

This unit has a 9.145 m diameter fan with performance characteristics as shown in Fig. 3. The fan hub diameter $d_h = 1.4$ m. The fan drive unit is supported by a bridge structure, as shown schematically in Fig. 7. The loss coefficient of this structure, determined with the aid of Fig. 6 and based on the frontal area of the heat exchanger bundle, is found to be $K_{do} = 4.332$. Similarly the upstream loss coefficient, that takes into consideration the flow resistance through a protective screen and its support structure, is found to be $K_{up} = 3.292$, determined with the aid of Fig. 5.

The total effective frontal area of the heat exchanger (A-frame) located above the fan unit is $A_{fr} = 220.4\ \mathrm{m}^2$, having finned tubes of 9.5 m long. The apex angle of the A-frame is 60°.

The loss coefficient for normal flow through the heat exchanger is given by

$$K_{he} = 4177[m_a/(\mu_a A_{fr})]^{-0.43927}.$$

Furthermore $\sigma = 0.875$. The main steam header has an effective diameter of 1.25 m and supplies saturated steam at 60°C.

The fan platform is located 25 m above ground level. It is supported by steel beams with an effective loss coefficient of $K_{ts} = 1.426$ based on the frontal area of the heat exchanger. Walkways with $L_w = 0.2$ m are located between the A-frame and 10 m high windwalls.

The condenser unit operates under ambient conditions where the atmospheric pressure is 84600 N/m^3 and the corresponding air temperature is 15.6°C.

With the above specifications and specified condenser heat transfer characteristics, the draft equation can be solved. For an air outlet temperature $T_{a6} = 45.758$°C the resultant corresponding consecutive loss coefficients all based on the frontal area of the heat exchanger (followed by the corresponding pressure differentials in brackets) are listed at the top of the next page.

Condenser support loss	$K_{ts} = 1.426$	(5.540 N/m^2)
Upstream losses	$K_{up} = 3.292$	(12.792 N/m^2)
Downstream losses	$K_{do} = 4.332$	(16.834 N/m^2)
Heat exchanger inlet losses	$K_{hi} = 2.379$	(9.245 N/m^2)
Normal heat exchanger loss	$K_{he} = 22.683$	(88.143 N/m^2)
Jetting loss	$K_{dj} = 2.052$	(7.974 N/m^2)
Outlet kinetic energy loss	$K_o = \underline{8.090}$	(31.437 N/m^2)
Total loss	$K_t = \underline{44.254}$	(171.965 N/m^2)

The fan static coefficient at the condenser operating point is $K_{Fs} = 43.253$ (168.076 N/m^2), which is about one less than K_t. This difference is due to the fact that buoyancy effects assist the draft.

CONCLUSION

In the design of the fan system for any air-cooled heat exchanger and in particular in an air-cooled condenser where the heat exchanger bundles are arranged in the form of an A-frame above the fan, all possible secondary flow losses must be considered. It is important to note that for a typical practical condenser unit, the total loss may be about double the loss due to the heat exchanger alone, i.e. the sum of all "secondary losses" are of the same order as that of the heat exchanger. Higher losses require more fan power and result in correspondingly higher noise levels.

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