AIR-COOLED HEAT EXCHANGERS AND COOLING TOWERS

THERMAL-FLOW PERFORMANCE EVALUATION AND DESIGN

VOLUME I

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CONTENTS

Volume I

Preface

The objective of these volumes is to provide modern analytical and empirical tools for evaluation of the thermal-flow performance or design of air-cooled heat exchangers and cooling towers. People who can make use of this information include students, design engineers, manufacturers, contractors, planners, plant managers, and end users. They may be in the fields of air-conditioning, refrigeration, mining, processing, chemicals, petroleum, power generation, and many other industries. They will be able to prepare improved specifications and evaluate bids more critically with respect to thermal performance of new cooling systems. Possible improvements through retrofits of existing cooling units can be determined, and impacts of plant operations and environmental influences can be predicted. Reasons for poor performance will be better understood, and where necessary, a plant can be optimized to achieve the lowest life cycle cost.

The format and presentation of the subject matter has evolved from courses offered at universities and from industry-based research, development, and consultation over many years. Volume I consists of chapters 1 through 5; Volume II consists of chapters 6 through 10. The Table of Contents for the companion volumes is listed in Appendix D in each volume.

An attempt is made to maintain a meaningful compromise between empirical, analytical, and numerical methods of analysis to achieve a satisfactory solution without introducing an unnecessary degree of complexity or cost. In some cases, sophisticated numerical methods have to be used in order to obtain sufficient insight into aspects of a particular problem. Such programs and the related infrastructure and manpower may be expensive, so it is desirable to stress analytical or empirical methods where meaningful.

The reader is introduced systematically to the literature, theory, and practice relevant to the performance evaluation and design of industrial cooling systems. Problems of increasing complexity are presented. Many of the procedures and examples presented are not only of academic interest, but are applicable to actual systems, and have been tested in practice. The design engineer is supplied with an extensive and up-to-date source of information.

In order to be an informed planner or client, it is important to understand the factors that influence the type and thermal-flow design of any cooling system in order to prepare clear and detailed specifications. A lack of insight and poor specifications often lead to serious misunderstandings between suppliers and clients and can result in significant increases in the ultimate cost of the plant.

The merits of a particular cooling system should be evaluated critically. Often this can only be done in a holistic and interdisciplinary approach. This approach takes into consideration the entire cycle or plant and its environment when an optimization exercise is performed.

For those interested in further reading, an extensive list of references is included at the end of every chapter.

In view of the iterative nature of solving most heat exchanger performance evaluation or thermal-flow design problems, computers are essential tools. An exception is simple first approximations.

In the numerical examples, values are often given to a large number of decimal places. These numbers are usually from the computer output and do not necessarily imply a corresponding degree of accuracy. However, increasingly improved designs are essential to reduce system costs in the design of large systems or where mass production is involved. In view of the increasing competition, access to computers and more reliable design information will lead to more refined and sophisticated designs. With a better understanding of the performance characteristics of a cooling system, control can be improved in different operating conditions. The worked problems not only show how to apply various equations but each problem forms a part of the learning process and introduces important additional information. The problems gradually lead up to more extensive and complex evaluations.

I am grateful to many friends and colleagues in both the academic and industrial worlds who directly or indirectly contributed to this work. However, this text would not have been written without the support and patience of my family and the valuable input of my graduate students.

List of Symbols

A	Area, m^2
ATD	Air travel distance, m
a	Coefficient; constant; length, m; surface area per unit volume, m^{-1}
B	Breadth, m
b	Exponent; constant; length, m; defined by Equation 3.3.4
C	Coefficient; heat capacity rate mc_p, W/K; C_{min}/C_{max}; cost
c	Concentration, kg/m^3
c_p	Specific heat at constant pressure, J/kgK
c_v	Specific heat at constant volume, J/kgK
D	Diffusion coefficient, m^2/s
$DALR$	Dry adiabatic lapse rate, K/m
d	Diameter, m
d_e	Equivalent or hydraulic diameter, m
E	Elastic modulus, N/m^2; energy, J
Ey	Characteristic pressure drop parameter, m^{-2}
e	Effectiveness
F	Force, N; fan; correction factor
f	Friction factor
G	Mass velocity, kg/sm^2
g	Gravitational acceleration, m/s^2; gap, m
H	Height, m
h	Heat transfer coefficient, W/m^2K
h_D	Mass transfer coefficient defined by Equation 4.1.3, m/s
h_d	Mass transfer coefficient defined by Equation 4.1.13, kg/m^2s
I	Insolation; Bessel function
i	Enthalpy, J/kg
i_{fg}	Latent heat, J/kg
J	Bessel function
K	Loss coefficient; incremental pressure drop number
k	Thermal conductivity, W/mK
L	Length, m
L_{hy}	Hydraulic entry length, $(x/d_e Re)$
M	Molecular weight, kg/mole; torque, Nm; mass, kg
m	Mass flow rate, kg/s
N	Revolutions per minute, $minute^{-1}$; NTU

NTU	Number of transfer units, $UA/C_{\min}$
Ny	Characteristic heat transfer parameter, m^{-1}
n	Number; exponent
P	Pitch, m; power, W
P_e	Perimeter, m
p	Pressure, N/m^2
p_{cr}	Critical pressure, N/m^2
Q	Heat transfer rate, W
q	Heat flux, W/m^2
R	Gas constant, J/kgK; thermal resistance, m^{2}K/W
Ry	Characteristic flow parameter, m^{-1}
r	Radius, m; recirculation factor defined by Equation 8.4.1
s	Blade tip clearance, m
s_t	Yield or ultimate stress, N/m^2
T	Temperature, °C or K
Tu	Turbulence intensity
t	Thickness, m
U	Overall heat transfer coefficient, W/m^{2}K
u	Internal energy, J/kg
V	Volume flow rate, m^3/s; molecular volume; volume, m^3
v	Velocity, m/s
W	Work, J; width, m
w	Humidity ratio, kg water vapor/kg dry air
X	Mole fraction
x	Co-ordinate; elevation, m; distance, m; quality
Y	Defined by Equation 5.2.4
y	Co-ordinate
z	Co-ordinate; elevation, m; exponent

Greek Symbols

α	Thermal diffusivity, k/rc_p; thermal expansion coefficient; void fraction
α_e	Kinetic energy coefficient defined by Equation 1.4.5
α_m	Momentum velocity distribution correction factor defined by Equation 1.4.25
α_Q	Defined by Equation 9.2.9
β	Volume coefficient of expansion, K^{-1}; porosity

Γ	Flow rate per unit length, kg/sm
γ	c_p/c_v; as defined by Equation 3.4.39
Δ	Differential
δ	Boundary layer thickness, m; condensate film thickness, m
ε	Surface roughness, m; expansibility factor
η	Efficiency; degree of separation
θ	Angle, °; temperature differential, K; potential temperature, °C
κ	Von Karman constant
λ	Eigenvalue; defined by Equation 2.7.4; defined by Equation 4.4.19
μ	Dynamic viscosity, kg/ms
ν	Kinematic viscosity, m^2/s; Poisson's ratio
ξ	Temperature lapse rate, K/m
ρ	Density, kg/m^3
σ	Area ratio; surface tension, N/m
τ	Shear stress, N/m^2; time, s
φ	Potential function; angle, °; defined by Equation 3.2.21 or Equation 3.3.13; relative humidity defined by Equation 4.1.21; expansion factor defined by Equation 5.2.3; dimensionless temperature difference
ψ	Defined by Equation 2.7.5

Dimensionless Groups

Eu	Euler number, $\Delta p/(\rho v^2)$
Fr	Froude number, $v^2/(dg)$
Fr_D	Densimetric Froude number, $\rho v^2/(\Delta\rho dg)$
Gr	Grashof number, $g\rho^2 L^3 \beta\Delta T/\mu^2$ for a plate or $g\rho^2 d^3 \beta\Delta T/\mu^2$ for a tube
Gz	Graetz number, $RePrd/L$ for a tube
Ku	Kutateladze number, $i_{fg}/(c_p\Delta T)$
j	Colburn j-factor, $StPr^{0.67}$
Le	Lewis number, $k/(\rho c_p D)$ or Sc/Pr
Le_f	Lewis factor, $h/(c_p h_d)$
Me	Merkel number, $h_d a_{fi} L_{fi}/G_w$
Nu	Nusselt number, hL/k for a plate or hd/k for a tube
Oh	Ohnesorge number, $\mu/(\rho d\sigma)^{0.5}$

Pe Péclet number, $RePr$
Pr Prandtl number, $\mu c_p/k$
Re Reynolds number, $\rho vL/\mu$ for a plate or $\rho vd/\mu$ for a tube
Sc Schmidt number, $\mu/(\rho D)$
Sh Sherwood number, $h_D L/D$ for a plate or $h_D d/D$ for a tube
St Stanton number, $h/(\rho v c_p)$ or $Nu/(RePr)$

Subscripts

a Air or based on air-side area
abs Absolute
ac Adiabatic cooling
acc Acceleration
al Aluminum
amm Ammonia
av Mixture of dry air and water vapor
app Apparent; approach
b Base; bundle; bend; boundary layer
c Concentration; convection heat transfer; combining header; casing; contraction; cold; critical; condensate
cd Conservative design
cf Counterflow
cp Constant properties
cr Critical
ct Cooling tower
ctc Cooling tower contraction
cte Cooling tower expansion
cu Copper
cv Control volume
D Darcy; drag; drop; diffusion
d Diameter; diagonal; drop; dynamic; dividing header; dry section; diffusion; mass transfer
do Downstream
db Drybulb
de Drift or drop eliminator
ds Steam duct

dif	Diffuser
e	Energy; expansion; effective; equivalent; evaporative
F	Fan
F/dif	Fan/diffuser
Fhe	Fan to heat exchanger distance
f	Fin; friction; fluid; factor
fi	Fill
fr	Frontal; face
fs	Fill support
g	Gas; ground
gen	Generator
H	Height
h	Hot; header; hub
he	Heat exchanger
i	Inlet; inside
isen	Isentropic
id	Ideal
$i\ell$	Inlet louver
iso	Isothermal
j	jet; junction
ℓ	Laminar; longitudinal; liquid; lateral; large
ℓm	Logarithmic mean
m	Mean; momentum; model; mass transfer; mixture
max	Maximum
min	Minimum
mo	Monin-Obukhov
n	Nozzle; normal
na	Noise attenuator
nu	Non-uniform
o	Outlet; outside; initial; oil; original
ob	Obstacle
P	Poppe
p	Constant pressure; production; plate; process fluid; passes; plume
$p\ell$	Plenum chamber
q	Constant heat flux
r	Root; row; radial co-ordinate; refrigerant; reference; recirculation; ratio
re	Effective root

rec	Recovery
red	Reducer
rz	Rain zone
s	Screen; steam; static; saturation; shell; support; superficial; steel; soil; scaling; spray
sc	Settling chamber; surface condenser
si	Inlet shroud
sp	Spray
ss	Supersaturated
T	Constant temperature; temperature; T-junction; test
ΔT	Constant temperature difference
t	Total; tube; tape; transversal; turbulent; transition; terminal; blade tip; fin tip
tp	Two-phase
tr	Tube row
ts	Tube cross section; tower support
tus	Wind tunnel upstream cross section
ud	Upstream and downstream
up	Upstream
v	Vapor
vc	Vena contracta
w	Water; wall; wind; walkway; wet section
wb	Wetbulb
wd	Water distribution system
x	Co-ordinate; quality
y	Co-ordinate
z	Co-ordinate; zinc
θ	Inclined; yawed
π	At 180°
∞	Infinite; free stream

1

Air-Cooled Heat Exchangers and Cooling Towers

1.0 Introduction

In any power generating or refrigeration cycle, heat has to be discharged. This is also true in many chemical and process plant cycles, internal combustion engines, computers, and electronic systems. The efficiency of a modern automobile engine is such that most of the energy contained in the fuel is rejected through the exhaust and the radiator. In a fossil-fired power plant with an efficiency of about 40%, more than 40% of the heat input has to be rejected through the cooling system. Even more heat has to be rejected in less efficient nuclear power plants. Considerably less heat is rejected in a modern combined cycle power plant.

In the past, the hydrosphere has been the commonly used heat sink at industrial plants. The simplest and cheapest cooling method was to direct water from a river, dam, or ocean to a plant heat exchanger and to return it, heated, to its source. In industrialized countries, the permissible rise in temperature of such cooling water is often limited and restricts the use of natural water for once-through cooling.

For practical reasons, other configurations (Fig. 1.2.4) may be preferred for particular applications. The rectangular arrangement (Fig. 1.2.4a) is very compact and finds application in closed circuit cooling plants, while the vertical arrangement (Fig. 1.2.4b) is suitable for smaller plants. The V-configuration (Fig. 1.2.4c) is often used with counterflow condensers.

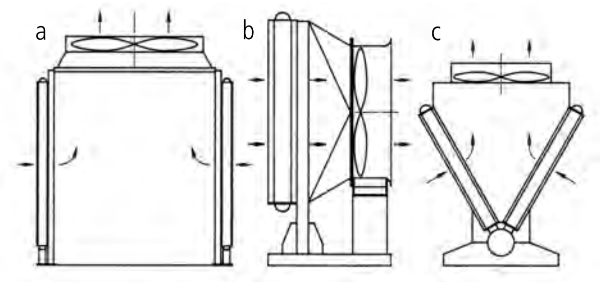

Fig. 1.2.4 Air-Cooled Heat Exchanger Configurations (a) Rectangular (b) Vertical (c) V-configuration

An example of an air-cooled refrigerant condenser is shown in Figure 1.2.5.

Air-cooled heat exchangers, usually referred to as radiators, find application in vehicles ranging from passenger cars to military vehicles, power generating sets, etc. (U.S. Army Command *Engineering Design Handbook*).

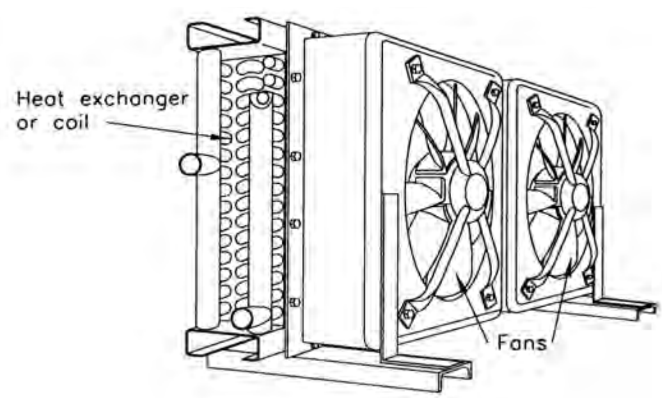

Fig. 1.2.5 Air-Cooled Refrigerant Condenser

the same as from a wet-cooling tower and has the same duty. Of course, the secondary circulation loop through a heat exchanger with its additional resistance to heat transfer is eliminated. If a bank of finned tubes is used in an evaporative cooler, it may be possible to reduce the annual water consumption by operating the unit in the dry mode during winter months when the ambient temperature is low.

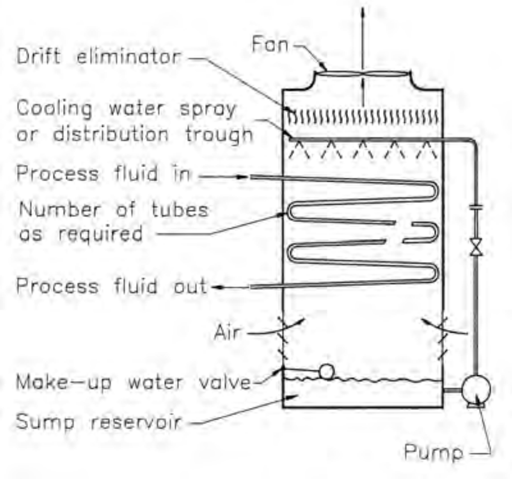

Fig. 1.3.1 Closed Circuit Evaporative Cooler

There are other ways of combining dry- and wet-cooling in a single heat rejection system according to Von Cleve, Nowosad, and Mitchell. These include *deluge enhancement*, combinations of dry- and wet-cooling units, and *adiabatic cooling*, precooling the entering air by humidification.

In the case of the former, the performance of a dry-cooled system is enhanced during periods of high ambient temperature and/or high cooling demand by deluging the air side of the heat transfer surface with water. The air flowing over the water causes evaporation and lowers the air/water interface temperature. The resultant increase in temperature difference between the internal hot fluid and the external deluge film increases the rate of heat transfer. The rate of heat transfer can be increased by a factor of up to five by deluging the air-side surface of the heat exchangers compared to a dry-cooled system at equivalent temperature and air-side pressure-drop conditions. A concept that incorporates this form of cooling

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Using Equation 2.2.2, the pressure drop between the inlet and the outlet of the duct is

$$\Delta p = 4 f_{app} \left(\frac{L}{d_e} \right) \frac{\rho v^2}{2} = 4 \text{x} 0.02269 \left(\frac{0.2}{0.006542} \right) \left(\frac{1.2134 \text{x} 3.015^2}{2} \right) = 15.3 \ N/m^2$$

Turbulent flow

With fully developed turbulent flow in ducts, the friction loss depends on flow conditions as characterized by the Reynolds number and on the nature of the duct wall surface. The quantity, ε, having the dimension of length is introduced as a measure of the surface roughness. From dimensional analysis, it follows that the friction factor is a function of the Reynolds number and the relative roughness *ε/d*. The graphical representation of this relationship is known as the Moody diagram and is presented in Figure 2.2.3 according to Moody.

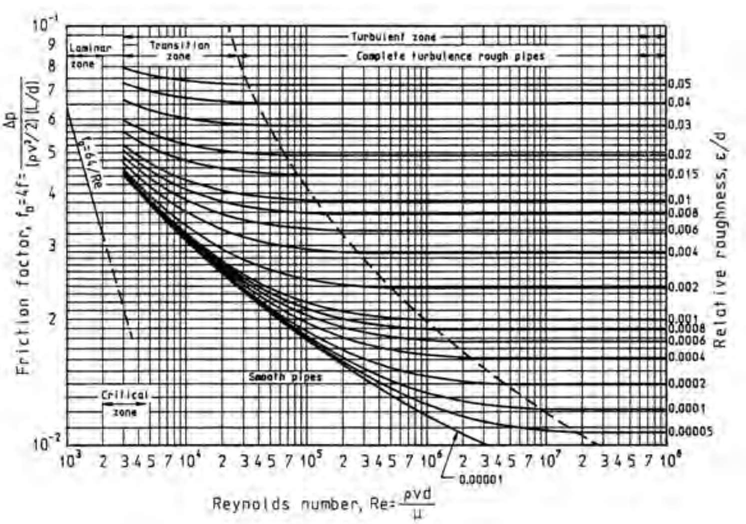

Fig. 2.2.3 Friction Factors for Pipe Flow