AIR-COOLED HEAT EXCHANGERS AND COOLING TOWERS

THERMAL-FLOW PERFORMANCE EVALUATION AND DESIGN

VOLUME II

By Detlev G. Kröger

PennWell®

CONTENTS

Volume II

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
cp	Specific heat at constant pressure, J/kgK
cv	Specific heat at constant volume, J/kgK
D	Diffusion coefficient, m^2/s
$DALR$	Dry adiabatic lapse rate, K/m
d	Diameter, m
de	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
hD	Mass transfer coefficient defined by Equation 4.1.3, m/s
hd	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
Pe	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
st	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, $k/\rho c_p$; 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, $g\rho^2L^3\beta\Delta T/\mu^2$ for a plate or $g\rho^2d^3\beta\Delta T/\mu^2$ 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, $k/(\rho c_pD)$ or Sc/Pr
Lef	Lewis factor, $h/(c_ph_d)$
Me	Merkel number, $h_da_{fi}L_{fi}/G_w$
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
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

6

Fans

6.0 Introduction

Different types of fans find application in air-cooled heat exchangers and evaporative coolers. These fans include axial flow, centrifugal flow, mixed flow, and crossflow. When selecting a fan for a particular application, the factors usually considered are:

- cost
- performance (stability of operation, ease of control, power consumption, flow range)
- mechanical arrangement (convenience of installation)
- self cleaning blade properties
- noise emission characteristics

The effective operation of fans in a system may be influenced by various structural and aerodynamic factors. Numerous books on the subject have been published including those by Berry, Jorgensen, Eck, Daly, Wallis, Osborne, and Bleier. Guides for the selection of appropriate fans in specific applications are also available, such as *A Guide to Fan Selection and Performance,* Examples of the other standardized test airways are shown in Figure 6.1.2. The installation type or types selected for testing a particular fan depends on the intended application of the fan. Although fans are sometimes installed in duct systems similar to one of the standard installation types, this is not always the case. If the fan system geometry deviates considerably from one of the standard installations, performance tests should be conducted on the system or a model.

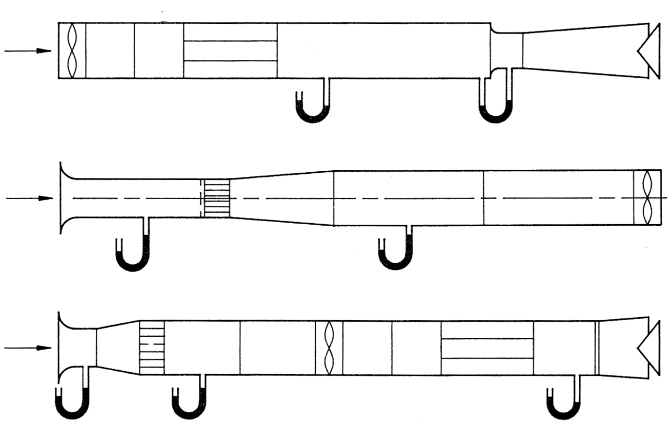

Fig. 6.1.2 Standardized Airways (a) Free Inlet, Ducted Outlet, Type B Installation, (b) Ducted Inlet, Free Outlet, Type C Installation, (c) Ducted Inlet, Ducted Outlet, Type D Installation

6.2 Presentation of Data and Results

Since it is usually impossible to carry out fan tests at exactly the speed or density specified, conversion rules, also known as *fan laws*, are used to determine the fan performance at the specified rotational speed, *N,* and the fan inlet density, ρ, i.e., authors such as Daly, Deeprose and Smith, Roslyng, and Zaleki. Studies by Cory and Coward are relevant to the same range of fan applications. The effect of fan plenum chambers on the fan performance is presented by Lambert et al, Stone and Wen, Russell and Berryman, and Meyer and Kröger. In view of the complexity of the flow in some systems, it may be prudent to perform model tests on such systems.

Upstream and downstream obstacles

Venter and Kröger conducted experiments on the V-type fan described in the previous sections to determine the influence on performance of flow resistances located immediately upstream and downstream. The resistances included support structures, screens, and walkways. For a uniformly distributed resistance, their conclusions showed the "bulk method" described by Ventilatoren Stork Hengelo satisfactorily predicted effective pressure loss coefficients. Figures 6.4.1 and 6.4.2 show the loss coefficients based on the velocity through the fan for resistances created by obstacles located on the upstream or suction side and the downstream or discharge side of the fan. These coefficients are a function of the projected area of the obstacle, A_{ob}, and the distance, x, from the fan.

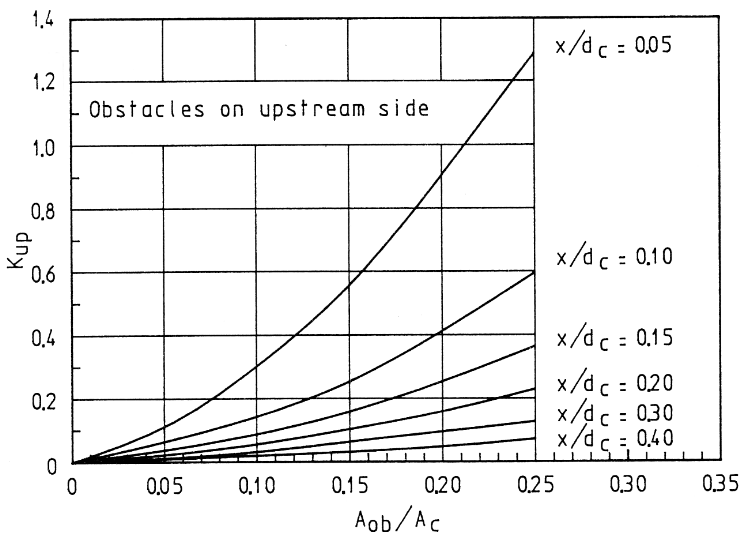

Fig. 6.4.1 Upstream Loss Coefficient Due to Obstacles

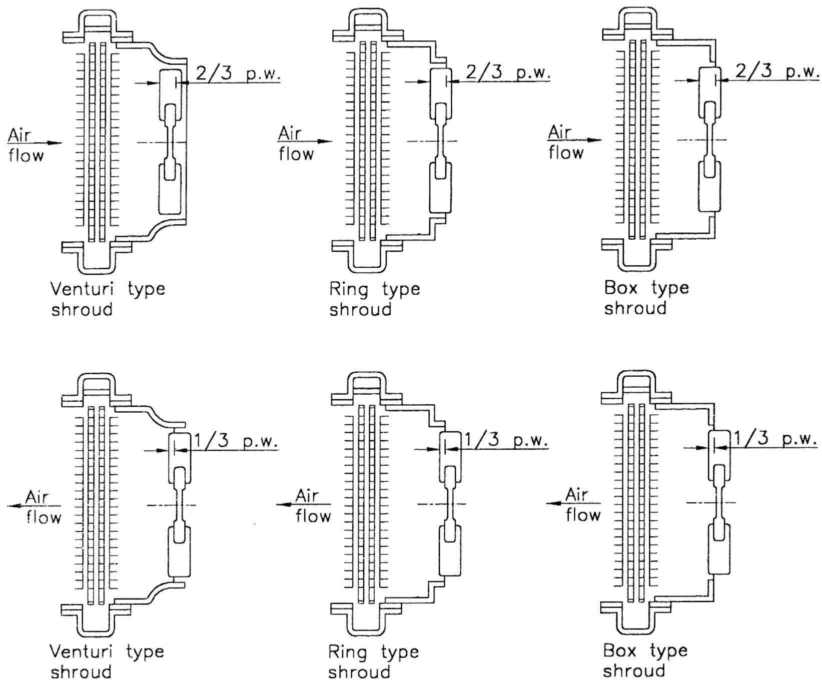

Fig. 6.4.8 Fan Shroud Types and Recommended Fan Positions

Diffuser

By installing a diffuser downstream of an axial flow fan, the air velocity and corresponding kinetic energy is reduced, and its static pressure is increased. The result is a net decrease in the power absorbed by the fan. Published diffuser efficiency data indicate peak values of 0.9 (Fig. 2.3.9).

The included angle of a practical diffuser lies in the range $12^{\circ} \leq 2\theta \leq 17^{\circ}$. An example of such a diffuser is shown in Figure 6.4.9. Depending on the cost structure, other geometries may be found to be more effective.

For a hyperbolic tower with a cylindrical outlet the loss coefficient is given by

$$K_{to} = \Delta p_{a56}/(\rho_{a5} v_{a5}^2/2) = 2\rho_{a5}\Delta p_{a56}/(m_a/A_5)^2$$

$$= -0.28 Fr_D^{-1} + 0.04 Fr_D^{-1.5} \qquad (7.1.9)$$

where

$Fr_D = (m_a/A_5)^2/[\rho_{a5}(\rho_{a6} - \rho_{a5}) g d_5]$.

This equation is valid for

$$0.5 \leq d_5/d_3 \leq 0.85 \text{ and } 5 \leq K_{he} \leq 40$$

Using Equation 1.4.2, the temperature at the tower outlet can be approximated by

$$T_{a5} = T_{a4} + [(v_{a4}^2 - v_{a5}^2)/2 + g(H_4 - H_5)]/c_{pa} \approx T_{a4} + g(H_4 - H_5)/c_{pa}$$

$$= T_{a4} - g(\gamma - 1)\ (H_5 - H_4)/(\gamma R) = T_{a4} - 0.00975\ (H_5 - H_4) \qquad (7.1.10)$$

since

$$(v_{a4}^2 - v_{a5}^2) << g(H_4 - H_5)$$

for natural draft cooling towers. From the perfect gas relation it follows that for

$$p_{a5} \approx p_{a6}$$

the density at the outlet of the tower is

$$\rho_{a5} = p_{a6}/[R\ \{T_{a4} - 0.00975\ (\mathrm{H}_5 - \mathrm{H}_4)\}] \qquad (7.1.11)$$

- the water distribution changes in the radial direction (according to Gösi) or is non-uniform
- plugging or other disturbances are present (according to Eldredge)

The following approach is similar to that applied to the dry-cooling tower in the previous section. A procedure is presented showing how the performance of a counterflow wet-cooling tower with a uniform thickness fill can be determined to a high degree of accuracy.

Neglecting the evaporation loss, which is 1–3% of the water mass flow rate, an energy balance for the wet-cooling tower shown in Figure 7.2.1. yields

$$Q = m_a\,(i_{ma5} - i_{ma1}) = m_w\,c_{pwm}\,(T_{wi} - T_{wo}) \tag{7.2.1}$$

where

m_a = the dry air flow rate through the tower

m_w = water mass flow rate through the tower

i_{ma5} = the outlet enthalpy of the air flowing through the tower

i_{ma1} = inlet enthalpy of the air flowing through the tower

The specific heat of the water is evaluated at the mean water temperature. Equation 7.2.1 is analogous to Equation 7.1.2. However, in addition to Equation 7.2.1, a relation must be satisfied that expresses the heat transfer rate in terms of an overall transfer coefficient analogous to Equation 7.1.3 but is applicable to the wet-cooling tower.

Energy is transferred in three regions of the cooling tower shown in Figure 7.2.1, i.e., rain, fill, and spray zone.

In the rain zone of this particular tower, the transfer coefficient from Equation 4.6.12 can be expressed approximately as

In large, natural-draft, counterflow cooling towers, the fill is sometimes installed in annular layers, which increase in thickness with increasing radius (Fig. 7.2.2). Cooling air entering at the lower edge of the shell is not exposed to much of the rain zone and will tend to flow through a greater fill thickness or height than the air entering near ground level. The condition of the air leaving the fill should be more uniform than in the case of constant fill thickness. According to Lowe and Christie, other arrangements of packings are found in practice. Variations in fill density, spray nozzle pitch, or water flow rate through the fill may be considered to improve performance. Numerical methods are required to analyze the flow through such configurations.

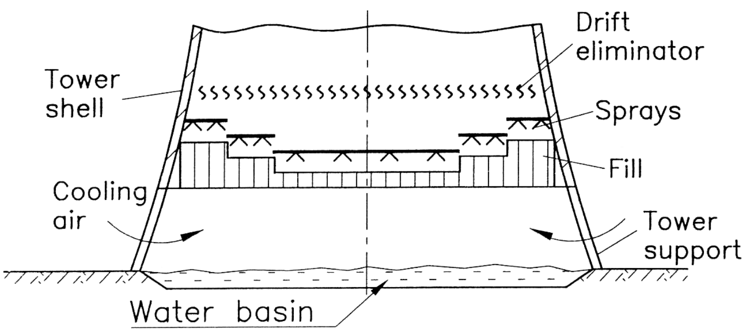

Fig. 7.2.2 Annular Variable Thickness Fill Installation

Another tower inlet configuration found in water-recovery, counterflow cooling towers is shown in Figure 7.2.3 from both Vogelsang and Hackeschmidt and Goldwirt et al.. The fill is of uniform thickness. Droplets leaving the fill are collected in plastic troughs located immediately below the fill before being returned to the condenser. This reduces the required water pumping power. In the absence of a rain zone, noise levels are significantly reduced, and the conditions of the air leaving the fill are relatively uniform.

This correction factor is valid in these ranges:

- $7.5 \leq d_i/H_i \leq 20$
- $3 \leq d_d \leq 6\text{mm}$
- $1 \leq G_w \leq 3\ \text{kg/m}^2\text{s}$
- $1.2 \leq G_a \leq 3.6\ \text{kg/m}^2\text{s}$
- $80 \leq d_i \leq 120\ \text{m}$
- $5 \leq K_{fi} \leq 25$

where $G_a = \rho_a v_i$ and $G_w = \rho_w v_w$.

Example 7.3.1

Determine the heat rejection rate of a natural-draft, hyperbolic, concrete, dry-cooling tower shown in Figure 7.1.1, if m_w = 4390 kg/s and hot water enters at T_{wi} = 61.45 ˚C. The diameter of the upper section of the tower is constant.

Ambient conditions:

- Air temperature at ground level: T_{a1} = 15.6 ˚C (288.75 K)
- Wetbulb temperature at ground level: T_{wb} = 0˚C (essentially dry air)
- Atmospheric pressure at ground level: p_{a1} = 84600 N/m^2
- Ambient temperature gradient: dT_a/dz = -0.00975 K/m from ground level

Cooling tower specifications:

- Tower height: H_5 = 120.0 m
- Tower inlet height: H_3 = 13.67 m
- Tower inlet diameter: d_3 = 82.958 m
- Tower outlet diameter (throat): d_5 = 58.0 m
- Number of tower supports: n_{ts} = 60
- Length of tower support: L_{ts} = 15.78 m