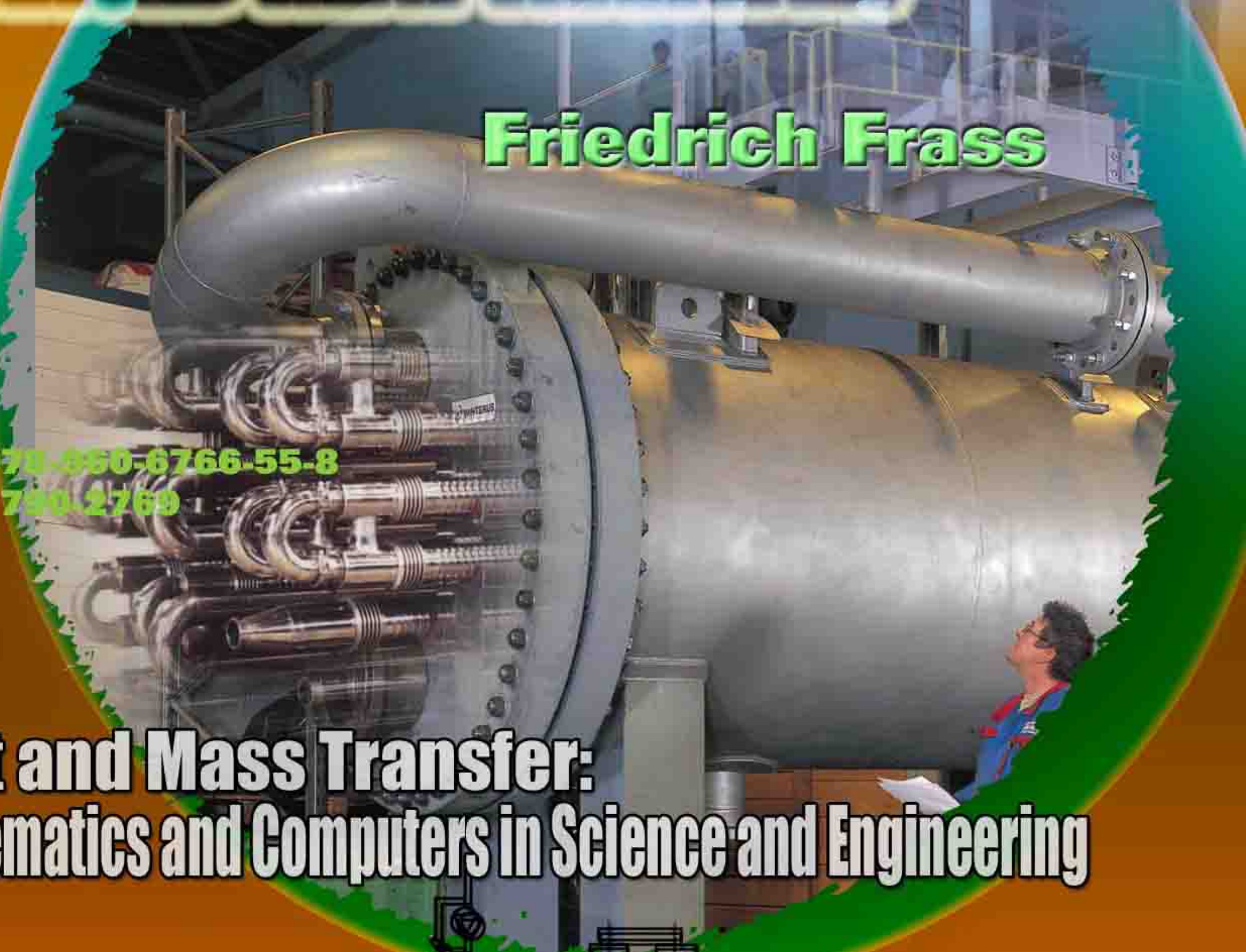
PRINCIPLES OF FINNED-TUBE HEAT EXCHANGER DESIGN FOR ENHANCED HEAT TRANSFER

Friedrich Frass



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Institute for Thermodynamics and Energy Conversion Vienna University of Technology

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Principles of Finned-Tube Heat Exchanger Design for Enhanced Heat Transfer

by

Dipl.-Ing. Dr. Friedrich Frass

Translated and Edited by
Dipl.-Ing. René Hofmann
A.o. Univ. Prof. Dipl.-Ing. Dr. Karl Ponweiser

Institute for Thermodynamics and Energy Conversion
Vienna University of Technology

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Preface

The present work was carried out at the Institute for Thermodynamics and Energy Conversion of the Vienna University of Technology in the course of several years during my activities as a scientific researcher. This work is based on measurements done on the experimental facility for heat transfer, described in the appendix, as well as on accompanying studies of the literature and reports about measurements taken using other methods.

My most grateful thanks go to o. Univ. Prof. Dr. W. Linzer for providing the impulse for this research and for the support during realization.

Many thanks to the Simmering Graz Pauker AG, as well as their successor company Austrian Energy and Environment, for allocating resources during the construction of the test facility and for providing, together with Energie und Verfahrenstechnik (EVT), the finned tubes.

Furthermore, I would like to thank our colleagues at the laboratory of the institute, M. Effenberg, H. Haidenwolf, W. Jandejsek, M. Schneider as well as R. Steininger, for the construction and assembly of the experimental facility in the lab and for altering the assembly many times in order to be able to examine other finned tube arrangements.

I also thank my colleagues at the Institute who gave me advice, particularly during the implementation of data collection and analysis.

The efforts of many individuals helped contribute to the development of this book. I would especially like to take this opportunity to thank Dipl.-Ing. René Hofmann whose encouragement and priceless assistance proved invaluable to the success of this work.

Finally I would like to thank A.o. Univ. Prof. Dr. Karl Ponweiser providing the impulse for doing further research on the experimental facility for optimization of heat transfer enhancement.

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Contents

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ISBN: 978-960-6766-55-8 ISSN: 1790-2769 Published by WSEAS Press www.wseas.org

List of Figures

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List of Tables

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List of Symbols

Symbol	Unit	Physical dimension
A	[m^2]	surface area of the fin
a	[m]	small axis of the flat tube
A_f	[-]	fractional free flow cross-section with flat tubes
A_{Ftot}	[m^2]	total surface area with flat tubes
a_{Ko}	[m]	surface area of the fin top per m tube
A_R	[m]	surface area of the smooth tube
A_{Ri}	[m]	surface area of the fins per m tube
a_{Ri}	[m]	surface area of the fin side per m tube
A_{Ro}	[m]	surface area of the bare tube per m tube
a_{Ro}	[m]	surface area of the bare tube per m tube
A_{tot}	[m]	total surface area per m finned tube
A_{0f}	[m]	proportional free flow cross-section
a_w	[m]	shorter dimension of the rectangular fin
b	[m]	large axis of the flat tube
b_w	[m]	longer dimension of the rectangular fin
C	[-]	common constant
C_1,C_2...,C_7	[-]	constant
$C1...C6$	[-]	coefficient according to ESCOA
D	[m]	outside diameter of fins
d_A	[m]	outside diameter of tube
d_E	[m]	diameter equivalent to area
d_e	[m]	characteristic diameter according to HEDH
d_i	[m]	inside diameter of tube
d_h	[m]	hydraulic diameter
d_q	[m]	equivalent diameter according to FDBR
d'	[m]	equivalent diameter according to HEDH
$E1, E2, E3$	[-]	constant according to FDBR
e_l	[-]	dimensionless longitudinal pitch
e_q	[-]	dimensionless transverse pitch
Eu	[-]	Euler number
f_f	[-]	Fanning friction factor
f_N	[-]	factor according to Brandt to account for a small number of consecutive tube rows in cross-flow
h	[m]	fin height
h'	[m]	equivalent fin height
h_{red}	[m]	reduced fin height
h_x	[m]	fin height as a coordinate
K_{An}	[-]	arrangement factor according to Brandt

Symbol	Unit	Physical dimension
Kf_t	[-]	factor for bundle geometry
Ku	[-]	universal characteristic number for heat transfer
K_z	[-]	factor to account for a small number of consecutive tube rows in cross-flow
l'	[m]	characteristic dimension
l_k	[m]	characteristic dimension according to Mirkovics
m	[m^{-1}]	parameter for fin efficiency
m^*	[kg $m^{-2}s^{-1}$]	mass velocity
n	[-]	exponent
n_A	[-]	arrangement factor for smooth tube bundles
n_R	[-]	number of consecutive tube rows in cross-flow
Nu	[-]	Nusselt number
Pr	[-]	Prandtl number
Pr_L	[-]	Prandtl number of air
R	[m]	radius above fins
r	[m]	radius
r_A	[m]	radius of the basic tube
R_b	[-]	quotient according to Nir
Re	[-]	Reynolds number
s	[m]	half fin thickness as a function
s_R	[m]	fin thickness
St	[-]	Stanton number
s_W	[m]	head width of hexagonal fins
$s_W^{'}$	[m]	smaller head width of hexagonal fins
$s_W^{''}$	[m]	larger head width of hexagonal fins
t_d	[m]	diagonal pitch
t_l	[m]	longitudinal pitch
t_q	[m]	transverse pitch
t_R	[m]	fin pitch
U	[m]	circumference
V	[m^3]	volume
W	[-]	A_{tot}/A_{0f}
w_E	[m/s]	gas velocity in the narrowest cross-section
w_m	[m/s]	mean gas velocity
w_R	[m/s]	effective gas velocity
w_0	[m/s]	gas velocity in the empty channel
y'	[-]	variable
z	[-]	variable
z_q	[-]	factor for transverse pitch according to Wehle
α	[W/m^2K]	heat transfer coefficient

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Symbol	Unit	Physical dimension
α_i	[W/m^{2}K]	inside heat transfer coefficient of the bare tube
α_0	[W/m^{2}K]	real heat transfer coefficient
Δp	[N/m^2]	pressure drop
η	[kg/m.s]	dynamic viscosity
ϑ	[C]	temperature
λ	[W/mK]	thermal conductivity
ν	[m^2/s]	kinematic viscosity
ξ	[-]	pressure drop coefficient
ρ	[kg/m^3]	density
φ	[-]	factor
ψ	[-]	porosity

Index	Denoation
Bm	mean boundary layer
F	fluid
gm	gas mean
$g1$	gas inlet
$g2$	gas outlet
m	mean
RF	fin base
Ri	fin
Ro	tube
Wa	wall
wm	water mean

Abbreviation	Denoation
$ESCOA$	Extended Surface Corporation of America
$HEDH$	Heat Exchanger Design Handbook
$FDBR$	Fachverband Dampfkessel-, Behaelter- und Rohrleitungsbau

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Abstract

In designing and constructing heat exchangers with transverse finned tubes in cross-flow, it is necessary to know correlations for calculating heat transfer and pressure drop. In addition to the common use of the Reynolds and Nusselt groups of dimensionless numbers, heat conduction in the fins also has to be accounted for in calculating heat transfer. A reduction coefficient termed "fin efficiency" is therefore introduced, by which the actual heat transfer coefficient is multiplied in order to get the apparent heat transfer coefficient. "Fin efficiency" is computed according to the laws of heat conduction under the assumption that the actual heat transfer coefficient is uniformly distributed across the fin surface.

Introducing geometrical constants for the fins, that is fin height, fin pitch, and fin thickness, into the equations for heat transfer and pressure drop makes these equations more bulky than the one for bare tube heat exchangers. Moreover, there is no self-evident characteristic dimension for a finned tube, as is the case with tube diameter for bare tubes, therefore many different proposals for the characteristic dimensions exist, which are in turn needed for setting the Reynolds and Nusselt dimensionless number groups. Some authors even use different characteristic dimensions for the Reynolds number and for the calculation of heat transfer and pressure loss.

Due to the complex geometry of finned tube designs, equations for heat transfer and pressure loss are derived mostly from experiments. When using for design purposes the equations obtained, a thorough knowledge of the condition of the tested finned tubes is necessary, i.e. of the materials and shape of fins, tubes and mode of attachment. For steam boilers and high pressure heat exchangers in the process industry, spiral finned tubes are commonly used today; here a ribbon of steel is wound spirally around a boiler tube and welded to it. For these finned tubes, coefficients of heat transfer and pressure loss are higher than for tubes with circumferential fins. Finned tubes are mostly arranged in bundles, which may be arranged staggered or in line. The later coefficients of heat transfer are in fact approximately only two thirds compared to staggered arrays. Therefore, many more staggered finned tube bundles have been tested. The equations for heat transfer in finned tube bundles give the results for a certain number of rows in longitudinal direction. For a smaller number of rows in staggered bundles, heat transfer is lower, while for in-line bundles it is higher.

With air coolers and heaters, tube bundles often have continuous fins, which may be easier to manufacture as long as fin pitch and the tube diameter are small. The equations for heat transfer and pressure loss are somewhat different for such tube bundles with continuous fins as compared to serrated finned tubes. In order to achieve a very small air-side pressure loss, extended tubes of various shapes may be used in the place of circular tubes, when fluid pressure in the tubes permits

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non-circular tubes. In some cases, corrugated or wavy fins are used, whereas corrugated fins increase heat transfer and wavy fins have a better ratio of heat transfer to pressure loss.

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