

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

Friedrich Frass

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Dipl.-Ing. Dr. Friedrich Frass
Institute for Thermodynamics and Energy
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Principles of Finned-Tube Heat Exchanger Design for Enhanced Heat Transfer

by

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Institute for Thermodynamics and Energy Conversion
Vienna University of Technology

Vienna, October 2007

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. Jandjsek, 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.

Vienna, October 2007

Friedrich Frass

Contents

1	Introduction	3
2	Fundamentals of heat transfer	3
2.1	Design of finned tubes	3
2.2	Fin efficiency	5
2.2.1	Plain geometry	6
2.2.2	Finned tubes	9
2.3	Special consideration in the calculation of heat transfer	12
3	Equations for the external heat transfer coefficient	14
3.1	Staggered tube arrangements	14
3.1.1	Overview of equations	14
3.1.2	Equations for a single tube row	22
3.1.3	Influence of geometrical dimensions of the finned tube and of bundle geometry	24
3.1.4	Evaluation of different calculation formulas	32
3.2	In-line tube arrangements	39
3.2.1	Enumeration of equations	39
3.2.2	Evaluation of the influence of fin parameters with in-line tube arrangement	42
3.2.3	Proposal for an enhanced calculation formula	49
3.3	Selection method for finned tubes	50
3.4	Substitution of fluid properties	54
3.5	Heat exchanger with a small number of consecutive tube rows	56
3.5.1	Reduction methods for staggered tube arrangements as presented in tables and diagrams	56
3.5.2	Calculations according to measurements on staggered finned tube bundles with less than 8 tube rows	57
3.5.3	Heat exchanger with small number of consecutive tube rows in in-line arrangement	60
3.6	Serrated fins	61
3.7	Geometrical arrangement of tubes in a bundle	63

3.8	Summary of heat transfer	69
4	Finned tube bundles with continuous fins	71
4.1	Finned tube bundles with continuous smooth fins and circular tubes	72
4.2	Finned tube bundles with continuous wavy fins and circular tubes	74
4.3	Finned tube bundles with non-circular tubes and continuous smooth fins	77
4.4	Finned tube bundles with flat tubes and continuous wavy fins . .	82
5	Pressure drop	86
5.1	Fundamentals for the determination of pressure drop at finned tubes	86
5.2	Problems with test result evaluation	86
5.3	Evaluation of pressure drop for staggered finned tube bundles . .	90
5.3.1	Equations for pressure drop in staggered finned tube bundles	91
5.3.2	Discussion of cited pressure drop equations	96
5.3.3	Recommendation for a calculation to predict pressure drop at staggered finned tube bundles in cross-flow	107
5.4	Calculation of pressure drop for finned tubes arranged in line . .	110
5.4.1	Presentation of equations	110
5.4.2	Discussion of pressure drop equations for in-line tube bundle arrangements	114
6	Conclusion and recommendations	123
7	Appendix: Test facility for heat transfer measurements	124

List of Figures

1	Finned tube with annular fins	3
2	Finned tube with spiral fins	4
3	Finned tubes with spiral fins mounted by pressure	4
4	Fins with t-shaped fin base	5
5	Fins with l-shaped fin base	6
6	Definition of fin efficiency	7
7	Heat conduction through the finned tube	13
8	Free-flow cross-section and free-flow cross-section within the out- line of the finned tube	20
9	Influence of tube diameter on heat transfer with unmodified fin geometry and fin pitches	24
10	Influence of tube diameter on heat transfer with unmodified fin geometry and adapted transverse pitch (staggered arrangement) .	26
11	Influence of tube diameter on heat transfer with unmodified fin geometry and transverse pitch (staggered arrangement) at constant Reynolds number	26
12	Influence of tube diameter on heat transfer with unmodified Reynolds number and fin geometry and adapted transverse pitch (staggered arrangement)	27
13	Influence of fin pitch on heat transfer (staggered arrangement) . .	27
14	Influence of fin height on heat transfer (staggered arrangement) .	28
15	Influence of fin height on heat transfer at adapted transverse pitch (staggered arrangement)	29
16	Influence of fin thickness on heat transfer (staggered arrangement)	29
17	Influence of gas velocity on heat transfer (staggered arrangement)	30
18	Influence of transverse pitch on heat transfer (staggered arrange- ment)	31
19	Influence of longitudinal pitch on heat transfer (staggered arrange- ment)	32
20	Influence of triangular pitch on heat transfer	32
21	Experimental results by Mirkovics showing the characteristic di- ameter d_{Mi}	33

VIII

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

22	Heat transfer measurements by Mirkovics and ITE on staggered finned tube arrangements evaluated with Mirkovics' formulas . . .	35
23	Heat transfer measurements by Mirkovics and ITE on staggered finned tube arrangements evaluated with formula (85)	36
24	Heat transfer measurements by Mirkovics and ITE on staggered finned tube arrangements evaluated using formula (86)	37
25	Heat transfer measurements by Mirkovics and ITE on staggered finned tube arrangements evaluated using formula (91)	38
26	In-line finned tube arrangement	40
27	Influence of tube diameter the heat transfer with constant Reynolds number (in-line arrangement)	42
28	Influence of tube diameter on heat transfer at constant gas velocity (in-line arrangement)	43
29	Influence of tube diameter on heat transfer at constant Reynolds number and with adapted tube pitches (in-line arrangement) . . .	43
30	Influence of tube diameter on heat transfer at constant gas velocity and with adapted tube pitches (in-line arrangement)	44
31	Influence of fin pitch on heat transfer (in-line arrangement)	44
32	Influence of fin thickness on heat transfer (in-line arrangement) .	45
33	Optimum fin thickness with respect to heat transfer for St35.8 fins (in-line arrangement)	46
34	Optimum fin thickness with respect to heat transfer for Austenite fins (in-line arrangement)	46
35	Influence of fin height on heat transfer (in-line arrangement) . . .	47
36	Influence of fin height on heat transfer at adapted transverse pitch (in-line arrangement)	48
37	Influence of fin height on heat transfer at adapted tube pitches (in-line arrangement)	48
38	Influence of transverse pitch on heat transfer (in-line arrangement)	48
39	Influence of longitudinal pitch on heat transfer based on measurements by ITE (in-line arrangement) (tube diameter 38 mm, 150 fins per m, 16x1 mm, transverse pitch 80 mm)	49
40	Influence of the Reynolds number on heat transfer at in-line arrangement	50
41	Results of the proposed equation (100) in comparison with available equations for heat transfer at in-line finned tube bundles . .	50

42	Flow displacement in dependence of relative transverse pitch $a = (t_q - (d_A + 2h))/t_q$	54
43	Reduction coefficient for heat transfer with a small number of consecutive tube rows (staggered arrangement)	55
44	Heat transfer with 8, 6, 4 and 2 consecutive tube rows with d_{Ch} as characteristic dimension (staggered arrangement)	56
45	Heat transfer with 8, 6, 4 and 2 consecutive tube rows with d_A as characteristic dimension (staggered arrangement)	58
46	Heat transfer with 8, 6, 4 and 2 consecutive tube rows with d_{Mi} as characteristic dimension (staggered arrangement)	58
47	Averages for heat transfer with 8, 6, 4 and 2 consecutive tube rows with d_A as characteristic dimension (staggered arrangement)	59
48	Reduction coefficient K_z for heat transfer with 8, 4, 2 and 1 consecutive tube rows (staggered arrangement)	60
49	Heat transfer with small number of consecutive tube rows and in-line arrangement	60
50	Reduction coefficient for heat transfer with a small number of consecutive tube rows in in-line and staggered arrangement	61
51	Sectional view of a finned tube with serrated fins	62
52	Comparison of Nusselt numbers for finned tubes with serrated and annular fins	62
53	Comparison of pressure drop coefficients for finned tubes with serrated and annular fins	63
54	Staggered finned tube arrangement with semi-tubes on the channel wall	64
55	Staggered finned tube arrangement	65
56	Partly staggered finned tube arrangement	65
57	Transition from an in-line to a staggered finned tube arrangement	66
58	Heat transfer measurements of Stasiulevicius in dependence of the angle α according to figure (57)	67
59	Heat transfer measurements by ITE in dependence of the angle α according to figure (57). Tube diameter 31.8 mm	67
60	Heat transfer measurements by ITE in dependence of the angle α according to figure (57). Tube diameter 38 mm	68
61	Heat transfer calculated according to ESCOA in dependence of the angle α according to figure (57)	68

62	Schematic representation of the experimental setup by Stenin Kuntysh [36]	69
63	Experimental results of Stenin Kuntysh [36]	69
64	Circular tubes with continuous fins	71
65	Wavy fins: corrugated or wavy formed	74
66	Comparison of Nusselt numbers for circular tubes with continuous smooth or wavy fins	75
67	Comparison of pressure drop coefficients at circular tubes with continuous smooth or wavy fins	76
68	Comparison of performance numbers according to equation (132) for circular tubes with continuous smooth or wavy fins	76
69	Flat tubes with continuous fins	77
70	Flat tubes with differing profiles according to Geiser [30]	81
71	Pressure drop coefficients of flat tubes with continuous fins	84
72	Nusselt-numbers at flat tubes with continuous fins	85
73	Performance numbers according to equation (132) for flat tubes with continuous fins	85
74	Exponent n according to equation (138), tube diameter 38 mm, 150 fins per m (16 x 1 mm), $t_q=85\text{mm}$	88
75	Pressure drop measurements with and without heat transfer. Tube diameter 38 mm , fins 16 x 1 mm, $t_q=85$ mm. Eq.(7) in the figure is identical with equation (143)	89
76	Correlation of pressure drop measurements at different gas temperatures. Tube diameter 31.8 mm, 200 fins per m, 15 x 1 mm. Eq.(7) in the figure is identical with equation (143)	90
77	Pressure drop coefficient in dependence of fin height according to VDI 431 [38]. Tube diameter 38 mm, 150 fins per m, $t_q = 85\text{mm}$, $t_l = 75\text{mm}$, staggered arrangement	95
78	Influence of tube diameter upon the pressure drop coefficient at constant velocity in the narrowest cross-section (staggered arrangement)	97
79	Influence of tube diameter upon the pressure drop coefficient at constant Reynolds number (staggered arrangement)	98
80	Influence of tube diameter upon the pressure drop coefficient at constant velocity in the narrowest cross-section and varied transverse pitch (staggered arrangement)	99

81	Influence of tube diameter upon the pressure drop coefficient at constant Reynolds number and varied traverse pitch (staggered arrangement)	99
82	Influence of fin thickness upon the pressure drop coefficient (staggered arrangement)	100
83	Influence of fin thickness upon the pressure drop coefficient according to measurements by Mirkovics	101
84	Influence of fin pitch upon the pressure drop coefficient (staggered arrangement)	101
85	Influence of fin height upon the pressure drop coefficient (staggered arrangement)	102
86	Influence of fin height and varied traverse pitch upon the pressure drop coefficient (staggered arrangement)	103
87	Influence of transverse pitch upon the pressure drop coefficient (staggered arrangement)	103
88	Influence of longitudinal pitch upon the pressure drop coefficient (staggered arrangement)	104
89	Influence of magnitude of triangular pitch upon the pressure drop coefficient	105
90	Influence of fin pitch upon the pressure drop coefficient: comparison of measured values and calculation. Tube diameter $d_A=38$ mm, fins $16 \times 1 \text{ mm}$	105
91	Influence of longitudinal pitch upon the pressure drop coefficient. Comparison of measured values and calculation for $d_A=31.8$ mm .	106
92	Influence of longitudinal pitch upon the pressure drop coefficient. Comparison of measured values and calculation for $d_A=38$ mm . .	107
93	Influence of the Reynolds number upon the pressure drop coefficient (staggered arrangement)	108
94	Comparison of the pressure drop coefficient calculated according to equation (171) with the results of equations from the literature	110
95	Influence of tube diameter upon the pressure drop coefficient with constant velocity in the narrowest cross-section (in-line arrangement)	113
96	Influence of tube diameter upon the pressure drop coefficient with constant Reynolds number (in-line arrangement)	114

XII

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

97	Influence of tube diameter upon the pressure drop coefficient with constant velocity in the narrowest cross-section and adapted transverse and longitudinal pitch (in-line arrangement)	115
98	Influence of tube diameter upon the pressure drop coefficient with constant Reynolds number and at adapted transverse and longitudinal pitch (in-line arrangement)	115
99	Influence of fin thickness upon the pressure drop coefficient (in-line arrangement)	116
100	Influence of fin pitch upon the pressure drop coefficient (in-line arrangement)	116
101	Influence of fin height upon the pressure drop coefficient (in-line arrangement)	117
102	Influence of fin height upon the pressure drop coefficient at adapted transverse and longitudinal pitch (in-line arrangement)	118
103	Influence of fin height upon the pressure drop coefficient with adapted transverse pitch (in-line arrangement)	118
104	Influence of the Reynolds number upon the pressure drop coefficient (in-line arrangement)	119
105	Influence of transverse pitch upon the pressure drop coefficient (in-line arrangement)	119
106	Influence of longitudinal pitch upon the pressure drop coefficient (in-line arrangement)	120
107	Influence of the number of consecutive tube rows upon the pressure drop coefficient (in-line arrangement)	121
108	Constant of equation (191) calculated by measurement values of the pressure drop coefficient in dependence of the number of tube rows (in-line arrangement)	122
109	Pressure drop coefficient values according to equation (191) in comparison with the values for in-line arrangements from the literature	122
110	Layout and design of the test facility	126

List of Tables

1	Constants in the formula of Brandt	18
2	Function $E(n_R)$	57
3	C_1 and C_2 for few tube rows according to [19]	74
4	Flat tubes with differing profiles according to Geiser [30]	80
5	Circumference and surface of profile flat tubes	82

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, \dots, C_7	[-]	constant
$C1 \dots C6$	[-]	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_{of}
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 ² K]	heat transfer coefficient

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
<i>ESCOA</i>	Extended Surface Corporation of America
<i>HEDH</i>	Heat Exchanger Design Handbook
<i>FDBR</i>	Fachverband Dampfkessel-, Behaelter- und Rohrleitungsbau

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

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.