

August 2021

ICS 49.140

English version

Space Engineering - Thermal design handbook - Part 13: Fluid Loops

Ingénierie spatiale - Manuel de conception thermique -
Partie 13 : Boucles fluides

Raumfahrttechnik - Handbuch für thermisches Design -
Teil 13: Flüssigkeitskreisläufe

This Technical Report was approved by CEN on 28 June 2021. It has been drawn up by the Technical Committee CEN/CLC/JTC 5.

CEN and CENELEC members are the national standards bodies and national electrotechnical committees of Austria, Belgium, Bulgaria, Croatia, Cyprus, Czech Republic, Denmark, Estonia, Finland, France, Germany, Greece, Hungary, Iceland, Ireland, Italy, Latvia, Lithuania, Luxembourg, Malta, Netherlands, Norway, Poland, Portugal, Republic of North Macedonia, Romania, Serbia, Slovakia, Slovenia, Spain, Sweden, Switzerland, Turkey and United Kingdom.



**CEN-CENELEC Management Centre:
Rue de la Science 23, B-1040 Brussels**

Table of contents

European Foreword	29
1 Scope	30
2 References	31
3 Terms, definitions and symbols	32
3.1 Terms and definitions	32
3.2 Abbreviated terms	32
3.3 Symbols	34
4 General introduction	46
4.1 Fluid loops	47
4.2 Comparison between fluid loops and alternative systems	48
4.2.1 Passive thermal insulations	48
4.2.2 Thermoelectric devices	48
4.2.3 Phase change materials (pcm)	49
4.2.4 Heat pipes	50
4.2.5 Short-term discharge systems	50
5 Analysis of a fluid loop	52
5.1 General	52
5.2 Thermal performance	53
5.3 Power requirements	56
6 Thermal analysis	58
6.1 General	58
6.2 Analytical background	58
6.2.1 Heat transfer coefficient	58
6.2.2 Dimensionless groups	60
6.2.3 Simplifying assumptions	61
6.2.4 Temperature-dependence of fluid properties	61
6.2.5 Laminar versus turbulent fluid flow	63
6.2.6 Heat transfer to internal flows	63
6.2.7 Heat transfer to external flows	65

6.3	Thermal performance data.....	67
6.3.1	Heat transfer to internal flow	67
6.3.2	Heat transfer to external flows.....	83
7	Frictional analysis	92
7.1	General.....	92
7.2	Analytical background.....	92
7.2.1	Introduction	92
7.2.2	Fully developed flow in straight pipes.....	93
7.2.3	Temperature-dependence of fluid properties.....	97
7.2.4	Several definitions of pressure loss coefficient.....	98
7.2.5	Entrance effects	100
7.2.6	Interferences and networks	101
7.2.7	Flow chart	102
7.3	Pressure loss data	105
7.3.1	Straight pipes	105
7.3.2	Bends.....	106
7.3.3	Sudden changes of area	113
7.3.4	Orifices and diaphragms	116
7.3.5	Screens.....	119
7.3.6	Valves	120
7.3.7	Tube banks	121
7.3.8	Branching of tubes	124
8	Combined thermal and frictional analysis.....	125
8.1	General.....	125
8.2	Analogies between momentum and heat transfer	125
8.2.1	The Reynolds analogy	125
8.2.2	The Prandtl analogy	128
8.2.3	The Von Karman analogy.....	129
8.2.4	Other analogies.....	129
9	Heat transfer enhancement	130
9.1	General.....	130
9.1.1	Basic augmentation mechanisms.....	131
9.1.2	Criterion for the evaluation of the several techniques	132
9.1.3	Index of the compiled data	133
9.1.4	Validity of the empirical correlations	133
9.2	Single-phase forced convection data	136

10 Working fluids	170
10.1 General	170
10.2 Cooling effectiveness of a fluid	170
10.2.1 Simplified fluid loop configuration	172
10.2.2 Thermal performance of the simplified loop	172
10.2.3 Power requirements of the simplified loop	173
10.2.4 Several examples	173
10.3 Properties of liquid coolants	178
10.4 Properties of dry air	212
11 Heat exchangers	214
11.1 General	214
11.2 Basic analysis	217
11.2.1 Introduction	217
11.2.2 Analytical background	218
11.2.3 Exchanger performance	221
11.3 Exchanging surface geometries	236
11.3.1 Tubular surfaces	237
11.3.2 Plate-fin surfaces	240
11.3.3 Finned tubes	246
11.3.4 Matrix surfaces	248
11.4 Deviations from basic analysis	249
11.4.1 Introduction	249
11.4.2 Longitudinal heat conduction	250
11.4.3 Flow maldistribution	253
11.5 Manufacturing defects	263
11.5.1 Introduction	263
11.5.2 Variations of the flow passages	263
11.5.3 Fin leading edge imperfections	267
11.5.4 Brazing	267
11.6 In service degradation	271
11.6.1 Introduction	271
11.6.2 Fouling	271
11.7 Existing systems	274
12 Pumps	283
12.1 General	283
12.2 Specified speed	287
12.3 Net suction energy	289

12.4	Requirements for spaceborne pumps	290
12.5	Commercially available pumps	291
12.6	European pump manufacturers.....	297
13	System optimization.....	298
13.1	General.....	298
13.2	Basic analysis	298
13.2.1	Interface heat exchanger.....	299
13.2.2	Supply and return plumbing	300
13.2.3	Radiator	301
13.3	Special examples.....	301
13.3.1	Constraints based on source temperature.....	302
13.3.2	Constraints imposed by the integration	305
14	Two-phase flow.....	309
14.1	General.....	309
14.2	Pressure loss.....	311
14.2.1	Lockhart-martinelli correlation	311
14.2.2	Improvements upon martinelli correlation.....	316
14.3	Annular flow.....	317
14.3.1	Ideal annular flow model	318
14.3.2	Annular flow with entrainment model.....	327
14.4	Condensation in ducts	341
14.4.1	Condensing flow model.....	341
14.4.2	Variation of the vapor quality along the duct in the stratified model	347
14.4.3	Limits of validity of the stratified model.....	349
14.4.4	Annular flow model.....	350
14.4.5	Variation of the vapor quality along the duct in the annular model.....	354
15	Two-phase thermal transport systems	357
15.1	General.....	357
15.1.1	Evolution of thermal transport systems.....	357
15.1.2	Two-phase loop general layout	358
15.1.3	About the nomenclature of this clause.....	361
15.2	Tms trade-off study.....	361
15.2.1	TMS study baseline.....	364
15.2.2	TMS design concepts.....	364
15.2.3	Evaluation of tms concepts	367
15.3	Design for orbital average load	370

15.3.1	Phase change capacitor performance	370
15.4	Off-design operation	376
15.4.1	Temperature control	378
15.4.2	Instrumentation requirements	381
15.5	Radiator-loop interaction	382
15.5.1	Boosting radiator temperature with a heat pump	383
15.5.2	Thermal-storage assisted radiator	388
15.5.3	Steerable radiators	391
15.5.4	Radiators coupling	402
15.6	Capillary pumped loop (cpl) technology	404
15.6.1	Advantages of cpl systems	408
15.6.2	CPL performance constraints	408
15.6.3	CPL basic system concept	408
15.7	Components	411
15.7.1	Pumping systems	411
15.7.2	Mounting plates	414
15.7.3	Vapour quality sensors	416
15.7.4	Fluid disconnects	420
16	Control technology	422
16.1	Basic definitions	422
16.2	General description of control systems	423
16.2.1	Introduction	423
16.2.2	Closed-loop control systems	424
16.2.3	Open-loop control system	424
16.2.4	Adaptative control systems	425
16.2.5	Learning control system	426
16.2.6	Trade-off of open- and closed-loop control systems	426
16.3	Basic control actions	431
16.3.1	Introduction	431
16.3.2	Two-position or on-off control action	432
16.3.3	Proportional control action (p controller)	433
16.3.4	Integral control action (i controller)	434
16.3.5	Proportional-integral control action (pi controller)	435
16.3.6	Proportional-derivative control action (pd controller)	436
16.3.7	Proportional-integral-derivative control action (pid controller)	437
16.3.8	Summary	438
16.4	Implementation techniques of control laws	439

16.4.1	Introduction	439
16.4.2	Devices characterization	441
16.4.3	Analog-controller implementation techniques	445
16.4.4	Summary	456
16.5	Hardware description	458
16.5.1	Introduction	458
16.5.2	Controllers	460
16.5.3	Sensors	465
16.5.4	Actuators. Control valves	468
16.6	Control software	469
16.7	Existing systems	472
16.7.1	Space radiator system	472
Bibliography.....		476

Figures

Figure 5-1:	Schematic representation of the fluid loop.....	52
Figure 6-1:	Nusselt numbers, Nu , for fully developed laminar flow through straight pipes of several cross-sectional shapes. Nu_q is the Nusselt number for constant heat transfer rate along the duct, and Nu_T that for constant wall temperature along the duct. From Kays & London (1964) [102].	69
Figure 6-2:	Nusselt numbers, Nu , vs. ratio, a/b , of short side to long side for fully developed laminar flow through straight pipes of rectangular cross section. From Kays & London (1964) [102].	70
Figure 6-3:	Nusselt numbers, Nu , vs. ratio of inner to outer diameter, r_1/r_2 , for fully developed laminar flow in concentric- circular-tube annuli. Constant heat transfer rate. From Kays & London (1964) [102].	70
Figure 6-4:	Influence of coefficients, Z , vs. ratio of inner to outer diameter, r_1/r_2 , for fully developed laminar flow in concentric-circular-tube annuli. Constant heat transfer rate. From Kays & London (1964) [102].	71
Figure 6-5:	Nusselt number, Nu , vs. Dean number, K , for fully developed laminar flow in curved pipe of circular cross section. Constant heat transfer rate. Results are shown for different Prandtl numbers, Pr . Calculated by the compiler after Mori & Nakayama (1965) [128].	71
Figure 6-6:	Thermal entry length Nusselt numbers, Nu , vs. non-dimensional axial distance, x^+ , for laminar flow through straight pipes. Constant wall temperature. Calculated by the compiler after Kays (1966) [101].	72
Figure 6-7:	Thermal entry length Nusselt number, Nu_x , vs. non-dimensional axial distance, x^+ , for laminar flow through straight pipes. Constant heat transfer rate. Also shown the influence coefficient, Z , for laminar flow between parallel plates with one side insulated. Calculated by the compiler after Kays (1966) [101].	72
Figure 6-8:	Thermal entry length Nusselt numbers, Nu_x , and influence coefficients, Z , vs. dimensionless axial distance, x^+ , for laminar flow in concentric-circular-	

tube annuli. Constant heat transfer rate. Calculated by the compiler after Kays (1966) [101].	73
Figure 6-9: Thermal entry length Nusselt number, Nu_x , vs. non dimensional distance along the coil centerline, x^+ , for laminar flow through a coil. The results are given for two values of the ratio, r/R , between the cross-sectional radius and the coil radius. Constant wall temperature. Calculated by the compiler after Kubair & Kuloor (1966) [111].	73
Figure 6-10: Nusselt numbers, Nu , vs. non-dimensional axial distance, x^+ , for the combined hydrodynamical and thermal entry length. Laminar flow through straight pipes of circular cross section. Constant wall temperature. $Pr = 0.7$. Replotted by the compiler after ESDU 68006 (1968) [48].	74
Figure 6-11: Local Nusselt number, Nu_x , vs. non-dimensional axial distance, x^+ , for the combined hydrodynamical and thermal entry length. Laminar flow through straight pipes of circular cross section. Constant heat transfer rate. Results are shown for different Prandtl numbers, Pr . Calculated by the compiler after Heaton et al. (1964) [82].	74
Figure 6-12: Local Nusselt number, Nu_x , and influence coefficient, Z , vs. dimensionless axial distance, x^+ , for the combined hydrodynamical and thermal entry length. Laminar flow between parallel plates, one of them insulated. Constant heat transfer rate. Results are shown for different Prandtl numbers, Pr . Calculated by the compiler after Heaton et al. (1964) [82].	75
Figure 6-13: Local Nusselt number, Nu_x , vs. Reynolds number, Re , for fully developed transitional flow through cylindrical ducts of circular cross section. Constant wall temperature. Gas Flow ($Pr \approx 0.7$). From ESDU 68006 (1968) [48].	75
Figure 6-14: Nusselt number, Nu , vs. Reynolds number, Re , for fully developed turbulent flow through cylindrical ducts. Constant heat transfer rate. Results are shown for different Prandtl numbers, Pr . Calculated by the compiler after Petukhov & Roizen (1975) [143].	76
Figure 6-15: Ratio of Nusselt number at constant heat transfer rate, Nu_q , to Nusselt number at uniform wall temperature, Nu_T , vs. Reynolds number, Re , for fully developed turbulent flow through a straight pipe of circular cross section. Results are shown for different Prandtl numbers, Pr . From Sleicher & Tribus (1957) [167].	76
Figure 6-16: Nusselt number, Nu , vs. Reynolds number, Re , for fully developed turbulent flow between parallel plates, one of them insulated. Constant heat transfer rate. Results are shown for different Prandtl numbers, Pr . Calculated by the compiler after Kays (1966) [101].	77
Figure 6-17: Influence coefficient, Z , vs. Reynolds number, Re , for fully developed turbulent flow between parallel plates. Constant heat transfer rate. Results are shown for different Prandtl numbers, Pr . Calculated by the compiler after Kays (1966) [101].	77
Figure 6-18: Nusselt number, Nu_{11} , and influence coefficient, Z_1 , vs. Reynolds number, Re , for fully developed turbulent flow in concentric-circular-tube annuli. $r_1/r_2 = 0,2$. Constant heat transfer rate. Results are shown for different Prandtl numbers, Pr . Calculated by the compiler after Kays (1966) [101].	78
Figure 6-19: Nusselt number, Nu_{22} , and influence coefficient, Z_2 , vs. Reynolds number, Re , for fully developed turbulent flow in concentric-circular-tube annuli. $r_1/r_2 = 0,2$. Constant heat transfer rate. Results are shown for different Prandtl numbers, Pr . Calculated by the compiler after Kays (1966) [101].	78

Figure 6-20: Nusselt number, Nu_{11} , and influence coefficient, Z_1 , vs. Reynolds number, Re , for fully developed turbulent flow in concentric-circular-tube annuli. $r_1/r_2 = 0,5$. Constant heat transfer rate. Results are shown for different Prandtl numbers, Pr . Calculated by the compiler after Kays (1966) [101].	79
Figure 6-21: Nusselt number, Nu_{22} , and influence coefficient, Z_2 , vs. Reynolds number, Re , for fully developed turbulent flow in concentric-circular-tube annuli. $r_1/r_2 = 0,5$. Constant heat transfer rate. Results are shown for different Prandtl numbers, Pr . Calculated by the compiler after Kays (1966) [101].	79
Figure 6-22: Nusselt number times Prandtl number to the minus 0.4 power, $NuPr^{0.4}$, vs. Reynolds number, Re , for fully developed turbulent flow in helically coiled tubes. The results are given for two values of the ratio, r/R , between the cross-sectional radius and the coil radius. Constant heat transfer rate. Calculated by the compiler after an experimental correlation obtained by Seban & McLaughlin (1963) [162] from data for water.	80
Figure 6-23: Thermal entry length Nusselt numbers, Nu , vs. non-dimensional axial distance, x/D , for fully developed turbulent flow through a straight pipe of circular cross section. Constant wall temperature. $Pr = 0.01$. Results are shown for different Reynolds numbers, Re . Calculated by the compiler after Kays (1966) [101].	80
Figure 6-24: Thermal entry length Nusselt numbers, Nu , vs. non-dimensional axial distance, x/D , for fully developed turbulent flow through a straight pipe of circular cross section. Constant wall temperature. $Pr = 0.7$. Results are shown for different Reynolds numbers, Re . Calculated by the compiler after Kays (1966) [101].	81
Figure 6-25: Ratio of thermal entry length Nusselt number, Nu_x , to Nusselt number for fully developed turbulent flow, Nu , vs. non-dimensional axial distance, x/D . Straight pipe of circular cross section. Constant heat transfer rate. $Pr = 0.01$. Results are shown for different Reynolds numbers, Re . Calculated by the compiler after Kays (1966) [101].	81
Figure 6-26: Ratio of thermal entry length Nusselt number, Nu_x , to Nusselt number for fully developed turbulent flow, Nu , vs. non-dimensional axial distance, x/D . Straight pipe of circular cross section. Constant heat transfer rate. $Re = 10^5$. Results are shown for different Prandtl numbers, Pr . Calculated by the compiler after Kays (1966) [101].	82
Figure 6-27: Ratio of thermal entry length Nusselt number, Nu_x , to Nusselt number for fully developed turbulent flow, Nu , vs. non-dimensional axial distance, x/D_E . Parallel plates at distance $2D_E$, one of them insulated. Constant heat transfer rate. Also shown the influence coefficient, Z . Results are shown for three different Prandtl numbers, Pr , and two Reynolds numbers, Re . Calculated by the compiler after Kays (1966) [101].	82
Figure 6-28: Nusselt number, Nu , vs. Reynolds number, Re . Flow of a fluid having constant physical properties over a constant temperature circular cylinder whose axis is normal to the incoming flow. From ESDU 69004 (1969) [50].	84
Figure 6-29: Effect of variable fluid properties, (a) and (b), and of inclination angle, (c), on the Nusselt number corresponding to the flow of a fluid over a constant temperature cylinder. Nu_b (Nu_{90°) can be deduced from Figure 6-28. From ESDU 69004 (1969) [50].	85
Figure 6-30: Guide for the selection of the curves given in Figure 6-31 and Figure 6-32 concerning in-line tube banks of different relative pitches. From ESDU 73031 (1973) [57].	86

Figure 6-31: Reference Nusselt number, Nu_r , for $Pr_b = 1$, as a function of Reynolds number, Re . In-line tube banks. See Figure 6-30 for the meaning of the numbers which appear on the curves. From ESDU 73031 (1973) [57].	87
Figure 6-32: Reference Nusselt number, Nu_r , for $Pr_b = 1$, as a function of Reynolds number, Re . In-line tube banks. See Figure 6-30 for the meaning of the numbers which appear on the curves. From ESDU 73031 (1973) [57].	88
Figure 6-33: Reference Nusselt number, Nu_r , for $Pr_b = 1$, as a function of Reynolds number, Re . In-line tube banks. Staggered tube banks. From ESDU 73031 (1973) [57].	89
Figure 6-34: Effect of the Prandtl number, Pr_b , on the reference Nusselt number, Nu_r , for both in-line and staggered tube banks. From ESDU 73031 (1973) [57].	89
Figure 6-35: The factor F_1 to account for variable fluid properties. From ESDU 73031 (1973) [57].	90
Figure 6-36: The factor F_2 accounting for abnormal number of rows vs. that number, N . From ESDU 73031 (1973) [57].	90
Figure 6-37: The factor F_3 accounting for the effect of yaw vs. the inclination angle, θ . From ESDU 73031 (1973) [57].	90
Figure 6-38: The factor F_4 for estimating the Nusselt number of the n -th row. From ESDU 73031 (1973) [57].	91
Figure 7-1: Friction characteristics associated with four types of roughness geometry. Notice that the equivalent roughness is different in every case. From Reynolds (1974).	96
Figure 7-2: Friction factor, λ_c , as a function of Reynolds number, Re , for different values of the relative roughness, e/D : Cylindrical tubes of circular cross section. From Idel'cik (1969) [97].	105
Figure 7-3: Correction factor, K , to be used when the cross section of the duct is not circular. Laminar flow. $K = 1$ for turbulent flow through hydraulically smooth ducts. From ESDU 66027 (1966) [46].	105
Figure 7-4: Boundary between short and long circular arc bends. From ESDU 67040 (1967) [47].	106
Figure 7-5: Boundaries between laminar, transitional and turbulent flows in long circular arc bends. From ESDU 67040 (1967) [47].	106
Figure 7-6: Pressure loss coefficient per unit bend angle, c_K/θ , as a function of the dimensionless radius of curvature of bend centerline, R/D , for different values of Reynolds number, Re . Either circular or square cross section. From ESDU 67040 (1967) [47].	107
Figure 7-7: Pressure loss coefficient, c_K , as a function of the dimensionless radius of bend centerline, R/D , for different values of Reynolds number, Re . Laminar flow through short circular arc bends. From ESDU 67040 (1967) [47].	108
Figure 7-8: Pressure loss coefficient, c_K , as a function of the dimensionless radius of bend centerline, R/D , for different values of bend angle, θ . Turbulent flow through short circular arc bends. Either circular or square cross section. From ESDU 67040 (1967) [47].	109
Figure 7-9: Pressure loss coefficient, c_K , for short circular arc bends, having a short downstream tangent of length, L_d , as a function of L_d/D , for different values of the dimensionless radius of bend centerline, R/D . Turbulent flow. Either circular or square cross section. From ESDU 67040 (1967) [47].	110

Figure 7-10: The factor α_1 to account for the aspect-ratio of the bend cross section. From ESDU 67040 (1967) [47].	110
Figure 7-11: The factor α_2 to account for the bend angle. From ESDU 67040 (1967) [47].	111
Figure 7-12: Pressure loss coefficient, c_K , for single mitre bends, as a function of bend angle, θ , for different values of the dimensionless length, L_d/D , of the downstream tube. Turbulent flow. Either circular or square cross section. From ESDU 67040 (1967) [47].	111
Figure 7-13: Factor β , which account for the interaction between two 90° -circular arc bends-, as a function of the dimensionless distance between both bends, L_d/D . From ESDU 68035 (1968) [49].	112
Figure 7-14: Factor β , which account for the interaction between two mitre bends, as a function of the dimensionless distance between both bends, L_d/D . From ESDU 68035 (1968) [49].	113
Figure 7-15: Total-pressure loss coefficient, c_{Kt} , as a function of Reynolds number, Re_{D1} , for different values of the area ratio, ψ . Enlargement with a duct downstream $4D_2$ long. Uniform incoming flow at low Reynolds number. From ESDU 72011 (1972) [54].	113
Figure 7-16: Different velocity profiles upstream of a sudden enlargement. From ESDU 72011 (1972) [54].	114
Figure 7-17: Total-pressure loss coefficient, c_{Kt} , as a function of area ratio, ψ . Enlargement with a duct downstream $4D_2$ long. Numbers on curves indicate the velocity profile in Figure 7-22 for which the curve applies. From ESDU 72011 (1972) [54].	114
Figure 7-18: Static-pressure loss coefficient, $-c_{Ks}$, as a function of area ratio, ψ . Enlargement with a duct downstream $4D_2$ long. Numbers on curves indicate the velocity profile in Figure 7-22 for which the curve applies. From ESDU (1972) [54].	115
Figure 7-19: Total-pressure loss coefficient, c_{Kt} , as a function of Reynolds number, Re_{D2} , for different values of the area ratio, ψ . The pressure loss coefficient is expressed in terms of the dynamic pressure at clause 6. From Idel'cik (1969) [97].	115
Figure 7-20: Reference values of the pressure loss coefficient, c_K , as a function of the ratio, ϕ , of the area available for fluid flow to the total area of the duct cross section. Perforated plates and orifices. From ESDU 72010 (1972) [53].	116
Figure 7-21: The factor α_3 to account for the effect of plate thickness when $t/d < 0,8$. c_{K0} is given in Figure 7-19. From ESDU 72010 (1972) [53].	117
Figure 7-22: The factor α_4 to account for the effect of plate thickness when $t/d \geq 0,8$. $c_{K0,8}$ is given in Figure 7-19. From ESDU 72010 (1972) [53].	118
Figure 7-23: Comparison between the pressure loss coefficients, c_K , in the intermediate region calculated by assuming either of the two extreme cases, fully- separated or reattached orifice flow. From ESDU 72010 (1972) [53].	119
Figure 7-24: Reference pressure loss coefficient, c_{Kr} , as a function of porosity, ϕ . Round-wire gauzes. From ESDU 72009 (1972) [52].	119
Figure 7-25: Factor α_5 to account for low Reynolds number effects in round-wire gauzes. Reynolds number based on the wire diameter. From ESDU 72009 (1972) [52].	120

Figure 7-26: Reference pressure loss coefficient, c_{K_r} , as a function of Reynolds number, Re , for diaphragm and butterfly valves fully open. Prepared by the compiler after ESDU 69022 (1969) [51].	120
Figure 7-27: Factor α_6 , which accounts for the partial opening of the valve, as a function of the degree of valve opening, δ . δ is defined as the ratio of valve control travel from closed position to total valve control travel. From ESDU 69022 (1969) [51].	121
Figure 7-28: Graphics for estimating the pressure loss coefficient, c_K , for in-line tube banks of several relative pitches, s_l , s_t , and yaw angles, θ . The influence of the heat exchange on the pressure loss is taken into account through the tube bank inlet and exit temperatures, T_i and T_o , respectively. From Idel'cik [97].	122
Figure 7-29: Graphics for estimating the pressure loss coefficient, c_K , for staggered tube banks of several relative pitches, s_l , s_t , and yaw angles, θ . The influence of the heat exchange on the pressure loss is taken into account through the tube bank inlet and exit temperatures, T_i and T_o , respectively. From Idel'cik (1969) [97].	123
Figure 7-30: Pressure loss coefficient, c_K , as a function of the ratio of lateral to total mass flow rates in branching tubes. The mixed confluence-branching case is not considered. From Idel'cik (1969) [97].	124
Figure 8-1: The ratio $2St/f$, for turbulent flow in constant wall temperature cylindrical tubes, as calculated by use of several expressions, vs. the Reynolds number, Re . E: Correlation of experimental results. From Goldstein (1950) [73]. R: Reynolds Analogy. P: Prandtl Analogy. K: von Kármán Analogy. Calculated by the compiler.	127
Figure 9-1: Constant power heat transfer ratio, $(h_a-h_o)_P$, vs. Reynolds number based on non-augmentative conditions, Re_o . From Bergles (1969) [8].	136
Figure 9-2: Roughness function $u_e^+(e^+)$ for Nikuradse's sand roughness. (1) Hydraulically smooth. (2) $u_e^+ = 8,48$, completely rough. From Schlichting (1960) [157].	139
Figure 9-3: Constant power heat transfer ratio, $(h_a-h_o)_P$, vs. Reynolds number based on non-augmentative conditions, Re_o . Curves A to D are from Bergles (1969) [8], curves E and F have been calculated by the compiler after Webb, Eckert & Goldstein (1971) [186].	139
Figure 9-4: Roughness function $u_e^+(e^+, b/e)$ for repeated-rib roughness. From Webb et al. (1971) [186].	142
Figure 9-5: Flow pattern near the wall for different values of b/e .	142
Figure 9-6: Constant power heat transfer ratio, $(h_a-h_o)_P$, vs. Reynolds number based on non-augmentative conditions, Re_o . Curves A, B, C are from Bergles (1969) [8], curves D to G have been calculated by the compiler after Sheriff & Gumley (1966) [166].	143
Figure 9-7: Roughness function, $u_e^+(e^+, b/e)$, for wire coil roughness. Plotted by the compiler after Sheriff & Gumley (1966) [166].	146
Figure 9-8: Velocity and Temperature distributions across the annulus.	147
Figure 9-9: Constant power heat transfer ratio, $(h_a/h_o)_P$, vs. Reynolds number based on non-augmentative conditions, Re_o . From Carnavos (1974) [19].	149

Figure 9-10: Constant power heat transfer ratio, $(h_a/h_o)_P$, vs. Reynolds number based on non-augmentative conditions, Re_o . From Carnavos (1974) [19].....	151
Figure 9-11: Constant power heat transfer ratio, $(h_a/h_o)_P$, vs. Reynolds number based on non-augmentative conditions, Re_o . From Carnavos (1974) [19].....	153
Figure 9-12: Constant power heat transfer ratio, $(h_a/h_o)_P$, vs. Reynolds number based on non-augmentative conditions, Re_o . From Bergles (1969) [8].....	156
Figure 9-13: Constant power heat transfer ratio, $(h_a/h_o)_P$, vs. Reynolds number based on non-augmentative conditions, Re_o . Calculated by the compiler after Hong & Bergles (1976) [91].	157
Figure 9-14: Constant power heat transfer ratio, $(h_a/h_o)_P$, vs. Reynolds number based on non-augmentative conditions, Re_o . Curves A to I are from Bergles (1969) [8], curves J to M have been calculated by the compiler after Thorsen & Landis (1968) [178].	160
Figure 9-15: Isothermal Nusselt number, Nu_{ab} , divided by the ratio of friction factors, Γ , vs. the Reynolds number, Re_T , for different values of the Prandtl number, Pr . Calculated by the compiler after Thorsen & Landis (1968) [178].	163
Figure 9-16: Constant power heat transfer ratio, $(h_a/h_o)_P$, vs. Reynolds number based on non-augmentative conditions, Re_o . From Bergles (1969) [8].....	164
Figure 9-17: Constant power heat transfer ratio, $(h_a/h_o)_P$, vs. Reynolds number based on non-augmentative conditions, Re_o . Curves A, B, C from Bergles (1969) [8], curves D, E, F from Bergles, Lee & Mikic (1969) [9].	167
Figure 10-1: Product of cooling effectiveness, F , of several fluids times the equivalent length of the loop, L_E , as functions of the difference between the heat source and the inlet temperature, $T_s - T_i$, for the following reference values: Inner diameter of the duct, $D = 10^{-2}$ m. Diabatic length of the duct, $L = 1$ m. Heat flux, $q = 250$ W.m ⁻² for Air, Carbon Dioxide, Carbon Tetrachloride, Hydrogen and Nitrogen, $q = 1000$ W.m ⁻² for Ethylene Glycol, Flutec PP50 and Water. Calculated by the compiler.	171
Figure 10-2: Schematic representation of the fluid loop considered for estimating the fluid cooling effectiveness.	172
Figure 10-3: Graphical method allowing for values of heat flux, q , and inner diameter of the duct, D , different from those used in Figure 10-1.....	174
Figure 10-4: Graphic for estimating the product of the fluid cooling effectiveness, F , times the equivalent length of the loop, L_E , as a function of the difference between the heat source and the inlet fluid temperature, $T_s - T_i$. Fluid: Air. Reference values: $D = 10^{-2}$ m, $L = 1$ m, $q = 250$ W.m ⁻² . Values FL_E for different D and q , yet $L = 1$, can be calculated graphically as is indicated in the text. Prepared by the compiler.....	175
Figure 10-5: Graphic for estimating the product of the fluid cooling effectiveness, F , times the equivalent length of the loop, L_E , as a function of the difference between the heat source and the inlet fluid temperature, $T_s - T_i$. Fluid: Ethylene Glycol. Reference values: $D = 10^{-2}$ m, $L = 1$ m, $q = 1000$ W.m ⁻² . Values FL_E for different D and q , yet $L = 1$, can be calculated graphically as is indicated in the text. Prepared by the compiler.	176
Figure 10-6: Graphic for estimating the product of the fluid cooling effectiveness, F , times the equivalent length of the loop, L_E , as a function of the difference between the heat source and the inlet fluid temperature, $T_s - T_i$. Fluid: Flutec PP50. Reference values: $D = 10^{-2}$ m, $L = 1$ m, $q = 1000$ W.m ⁻² .	

Values FL_E for different D and q , yet $L = 1$, can be calculated graphically as is indicated in the text. Prepared by the compiler.	177
Figure 10-7: Graphic for estimating the product of the fluid cooling effectiveness, F , times the equivalent length of the loop, L_E , as a function of the difference between the heat source and the inlet fluid temperature, $T_S - T_i$. Fluid: Water. Reference values: $D = 10^{-2}$ m, $L = 1$ m, $q = 1000$ W.m ⁻² . Values FL_E for different D and q , yet $L = 1$, can be calculated graphically as is indicated in the text. Prepared by the compiler.	178
Figure 10-8: Vapor pressure, p_{sat} , of Water vs. temperature, T . From Vargaftik (1975) [183].	186
Figure 10-9: Density, ρ , of Water vs. temperature, T . From Vargaftik (1975) [183].	186
Figure 10-10: Specific heat, c_p , of Water vs. temperature, T . From Vargaftik (1975) [183].	186
Figure 10-11: Thermal conductivity, k , of Water vs. temperature, T . From Vargaftik (1975) [183].	187
Figure 10-12: Dynamic viscosity, μ , of Water vs. temperature, T . From Vargaftik (1975) [183].	187
Figure 10-13: Vapor pressure, p_{sat} , of Carbon Tetrachloride vs. temperature, T . From Vargaftik (1975) [183].	187
Figure 10-14: Density, ρ , of Carbon Tetrachloride vs. temperature, T . From Vargaftik (1975) [183].	188
Figure 10-15: Specific heat, c_p , of Carbon Tetrachloride vs. temperature, T . From Vargaftik (1975) [183].	188
Figure 10-16: Thermal conductivity, k , of Carbon Tetrachloride vs. temperature, T . From Vargaftik (1975) [183].	188
Figure 10-17: Dynamic viscosity, μ , of Carbon Tetrachloride vs. temperature, T . From Vargaftik (1975) [183].	189
Figure 10-18: Vapor pressure, p_{sat} , of Coolanol 15, 25, 35 and 45 vs. temperature, T . From Filippi & Guerra (1977) [64].	189
Figure 10-19: Density, ρ , of Coolanol 15, 25, 35 and 45 vs. temperature, T . From Filippi & Guerra (1977) [64].	189
Figure 10-20: Specific heat, c_p , of Coolanol 15, 25, 35 and 45 vs. temperature, T . From Filippi & Guerra (1977) [64].	190
Figure 10-21: Thermal conductivity, k , of Coolanol 15, 25, 35 and 45 vs. temperature, T . From Filippi & Guerra (1977) [64].	190
Figure 10-22: Dynamic viscosity, μ , of Coolanol 15, 25, 35 and 45 vs. temperature, T . From Filippi & Guerra (1977) [64].	190
Figure 10-23: Kinematic viscosity, ν , of DC 200 vs. temperature T . Numbers on curves indicate the standard viscosity in cs. From DOW CORNING (1972) [38].	191
Figure 10-24: Freezing point, T , of Water/Glycol Solutions vs. Glycol mass fraction, s . From Filippi & Guerra (1977) [64].	191
Figure 10-25: Vapor pressure, p_{sat} , of Water/Glycol Solutions vs. temperature, T . Numbers on curves indicate Glycol mass fraction, c . From Filippi & Guerra (1977) [64].	192

Figure 10-26: Density, ρ , of Water/Glycol Solutions vs. temperature, T . Numbers on curves indicate Glycol mass fraction, c . From Filippi & Guerra (1977) [64].	192
Figure 10-27: Specific heat, c_p , of Water/Glycol Solutions vs. temperature, T . Numbers on curves indicate Glycol mass fraction, c . From Filippi & Guerra (1977) [64].	193
Figure 10-28: Thermal conductivity, k , of Water/Glycol Solutions vs. temperature, T . Numbers on curves indicate Glycol mass fraction, c . From Filippi & Guerra (1977) [64].	193
Figure 10-29: Dynamic viscosity, μ , of Water/Glycol Solutions vs. temperature, T . Numbers on curves indicate Glycol mass fraction, c . From Filippi & Guerra (1977) [64].	194
Figure 10-30: Vapor pressure, p_{sat} , of Flutec PP-2, PP-9 and PP-50 vs. temperature, T . Data are from Dunn & Reay (1976) [40] except those corresponding to Flutec PP-50 which are from Wyn-Roberts (1974) [193].	194
Figure 10-31: Density, ρ , of Flutec PP-2, PP-9 and PP-50 vs. temperature, T . Data are from Dunn & Reay (1976) [40] except those corresponding to Flutec PP-50 which are from Wyn-Roberts (1974) [193].	195
Figure 10-32: Specific heat, c_p , of Flutec PP-50 vs. temperature, T . From Wyn-Roberts (1974) [193].	195
Figure 10-33: Thermal conductivity, k , of Flutec PP-2, PP-9 and PP-50 vs. temperature, T . Data are from Dunn & Reay (1976) [40] except those corresponding to Flutec PP-50 which are from Wyn-Roberts (1974) [193].	196
Figure 10-34: Dynamic viscosity, μ , of Flutec PP-2, PP-9 and PP-50 vs. temperature, T . Data are from Dunn & Reay (1976) [40] except those corresponding to Flutec PP-50 which are from Wyn-Roberts (1974) [193].	196
Figure 10-35: Vapor pressure, p_{sat} , of Freon 11, 12, 13, 21, 22, 113, 114 and 142 vs. temperature, T . From Vargaftik (1975) [183].	197
Figure 10-36: Density, ρ , of Freon 11, 12, 13, 21, 22, 113, 114 and 142 vs. temperature, T . From Vargaftik (1975) [183].	197
Figure 10-37: Specific heat, c_p , of Freon 11, 12, 13, 21, 22, 113, 114 and 142 vs. temperature, T . From Filippi & Guerra (1977) [64].	197
Figure 10-38: Thermal conductivity, k , of Freon 11, 12, 13, 21, 22, 113, 114 and 142 vs. temperature, T . From Vargaftik (1975) [183].	198
Figure 10-39: Dynamic viscosity, μ , of Freon 11, 12, 13, 21, 22, 113, 114 and 142 vs. temperature, T . Data are from Vargaftik (1975) [183] except those corresponding to Freon 13 which are from Filippi & Guerra (1977) [64].	198
Figure 10-40: Vapor pressure, p_{sat} , of Freon E1, E2, E3, E4 and E5 vs. temperature, T . From Filippi & Guerra (1977) [64].	198
Figure 10-41: Density, ρ , of Freon E1, E2, E3, E4 and E5 vs. temperature, T . From Filippi & Guerra (1977) [64].	199
Figure 10-42: Specific heat, c_p , of Freon E1, E2, E3, E4 and E5 vs. temperature, T . From Filippi & Guerra (1977) [64].	199
Figure 10-43: Thermal conductivity, k , of Freon E1, E2, E3, E4 and E5 vs. temperature, T . From Filippi & Guerra (1977) [64].	199
Figure 10-44: Dynamic viscosity, μ , of Freon E1, E2, E3, E4 and E5 vs. temperature, T . From Filippi & Guerra (1977) [64].	200

Figure 10-45: Vapor pressure, p_{sat} , of FC 75 vs. temperature, T . From Filippi & Guerra (1977) [64].	200
Figure 10-46: Density, ρ , of FC 75 vs. temperature, T . From Filippi & Guerra (1977) [64].	200
Figure 10-47: Specific heat, c_p , of FC 75 vs. temperature, T . From Filippi & Guerra (1977) [64].	201
Figure 10-48: Thermal conductivity, k , of FC 75 vs. temperature, T . From Filippi & Guerra (1977) [64].	201
Figure 10-49: Dynamic viscosity, μ , of FC 75 vs. temperature, T . From Filippi & Guerra (1977) [64].	201
Figure 10-50: Thermal conductivity, k , of Methanol/Water Solutions vs. temperature, T . Numbers on curves indicate Methanol mass fraction, c . From Vargaftik (1975) [183].	202
Figure 10-51: Dynamic viscosity, μ , of Methanol/Water Solutions vs. temperature, T . Numbers on curves indicate Methanol mass fraction, c . From Vargaftik (1975) [183].	202
Figure 10-52: Vapor pressure, p_{sat} , of Monsanto OS 59 vs. temperature, T . From Filippi & Guerra (1977) [64].	203
Figure 10-53: Density, ρ , of Monsanto OS 59 vs. temperature, T . From Filippi & Guerra (1977) [64].	203
Figure 10-54: Specific heat, c_p , of Monsanto OS 59 vs. temperature, T . From Filippi & Guerra (1977) [64].	203
Figure 10-55: Thermal conductivity, k , of Monsanto OS 59 vs. temperature, T . From Filippi & Guerra (1977) [64].	204
Figure 10-56: Dynamic viscosity, μ , of Monsanto OS 59 vs. temperature, T . From Filippi & Guerra (1977) [64].	204
Figure 10-57: Density, ρ , of Air at a pressure of 10^5 Pa vs. temperature, T . From Vargaftik (1975) [183].	212
Figure 10-58: Specific heat, c_p , of Air vs. temperature, T . From Vargaftik (1975) [183].	212
Figure 10-59: Thermal conductivity, k , of Air vs. temperature, T . From Vargaftik (1975) [183].	212
Figure 10-60: Dynamic viscosity, μ , of Air vs. temperature, T . From Vargaftik (1975) [183].	213
Figure 11-1: Typical regenerators. a) Rotary type. b) Valved type. From Kays & London (1964) [102].	214
Figure 11-2: Typical recuperators. a) Counterflow heat exchanger. b) Crossflow heat exchanger. From Welty, Wicks & Wilson (1969) [188].	215
Figure 11-3: Some typical examples of compact heat exchanger surfaces. From Kays & London (1964) [102]. a) Circular tube bundle. b) Finned-circular-tube surface. c) Finned-tube surface, flat tubes, continuous fins. d) Plate-fin arrangement. e) Strip-fin surface. f) Regenerator compact matrix.	216
Figure 11-4: a) Shell-and-tube exchanger with two shell passes and four tube passes. b) Schematic representation of the exchanger which will be used in clause 11.2.3.	217

Figure 11-5: Liquid-coupled indirect-transfer type of heat exchanger. From Kays & London (1964) [102].	217
Figure 11-6: Thermal conductivity, k , of several metals vs. temperature, T . From Kays & London (1964) [102].	219
Figure 11-7: Heat transfer effectiveness, η_f , of trapezoidal fins, vs. dimensionless fin length, $L[2h/k(\delta_r + \delta_b)]^{1/2}$. Calculated by the compiler after Jakob (1958) [99].	220
Figure 11-8: Heat transfer effectiveness, η_f , of circular fins, vs. dimensionless fin length, $(r_o - r_i)(h/k\delta)^{1/2}$. Calculated by the compiler after Jakob (1958) [99].	220
Figure 11-9: Heat transfer effectiveness, ε , vs. number of heat transfer units, N_{tu} , in a counterflow exchanger. From Kays & London (1964) [102].	222
Figure 11-10: Heat transfer effectiveness, ε , vs. number of heat transfer units, N_{tu} , in a parallel flow exchanger. From Kays & London (1964) [102].	222
Figure 11-11: Heat transfer effectiveness, ε , vs. number of heat transfer units, N_{tu} , in a crossflow exchanger with fluids unmixed. Calculated by the compiler after Mason (1954) [124].	223
Figure 11-12: Heat transfer effectiveness, ε , vs. number of heat transfer units, N_{tu} , in a crossflow exchanger with one fluid mixed. From Kays & London (1964) [102].	223
Figure 11-13: Heat transfer effectiveness, ε , vs. number of heat transfer units, N_{tu} , for the case of n-pass counter-crossflow exchangers, when fluid A is unmixed throughout and fluid B mixed throughout, and with passes connected in reverse order. Calculated by the compiler after Stevens, Fernandez & Woolf (1957) [170].	224
Figure 11-14: Heat transfer effectiveness, ε , vs. number of heat transfer units, N_{tu} , for the case of n-pass counter-crossflow exchangers, when fluid A is unmixed throughout and fluid B mixed throughout, and with passes connected in identical order. Calculated by the compiler after Stevens, Fernandez & Woolf (1957) [170].	225
Figure 11-15: Heat transfer effectiveness, ε , vs. number of heat transfer units, N_{tu} , in a multipass exchanger with 1 shell pass and 2 or more tube passes. From Kays & London (1964) [102].	226
Figure 11-16: Heat transfer effectiveness, ε , vs. number of heat transfer units, N_{tu} , in a multipass exchanger with 2 shell passes and 4, 8, 12, ... tube passes. From Kays & London (1964) [102].	226
Figure 11-17: Heat transfer effectiveness, ε , vs. number of heat transfer units, N_{tu} , in a multipass exchanger with 3 shell passes and 6, 12, 18, ... tube passes. From Kays & London (1964) [102].	227
Figure 11-18: Heat transfer effectiveness, ε , vs. number of heat transfer units, N_{tu} , in a multipass exchanger with 4 shell passes and 8, 16, 24, ... tube passes. From Kays & London (1964) [102].	227
Figure 11-19: Heat transfer effectiveness, ε , vs. number of heat transfer units, N_{tu} , in a multipass exchanger with 5 shell passes and 10, 15, 20, ... tube passes. From Kays & London (1964) [102].	228
Figure 11-20: Heat transfer effectiveness, ε , vs. number of heat transfer units, N_{tu} , for different number of shell passes, in a multipass exchanger with $R = 1$. The case $R = 0$ is also shown for comparison. Calculated by the compiler after Kays & London (1964) [102].	228

Figure 11-21: Optimum liquid flow capacity rate, C_1/C_{Lopt} , to maximize the heat transfer effectiveness vs. ratio of the number of heat transfer units N_{tu1}/N_{tu2} , of exchanger 1 to exchanger 2. Calculated by the compiler after Holmberg (1975) [90].	229
Figure 11-22: Heat transfer effectiveness, ε , vs. liquid flow capacity rate, C_L/C_{Lopt} , for different values of the overall number of heat transfer units, N_{tu}^o . $N_{tu1}/N_{tu2} = 1$. Calculated by the compiler after Holmberg (1975) [90].	230
Figure 11-23: Heat transfer effectiveness, ε , vs. liquid flow rate, C_L/C_{Lopt} , for different values of the ratio between the number of heat transfer units of exchanger 1 to exchanger 2, N_{tu1}/N_{tu2} . $N_{tu}^o = 2$. Calculated by the compiler after Holmberg (1975) [90].	230
Figure 11-24: Overall heat transfer effectiveness, ε , vs. area ratio between exchanger 1 and 2, $(A_1/A_2)/(A_1/A_2)_{opt}$, for the case of optimum liquid flow capacity rate and $R = 1$. Calculated by the compiler after Holmberg (1975) [90].	231
Figure 11-25: Overall heat transfer effectiveness, ε_t , of an assembly of n identical exchangers in parallel, vs. effectiveness, ε , of a single exchanger. Calculated by the compiler after Domingos (1969) [36].	232
Figure 11-26: Overall heat transfer effectiveness, ε_t , of an assembly of n identical exchangers in counterflow, vs. effectiveness, ε , of a single exchanger. Calculated by the compiler after Domingos (1969) [36].	233
Figure 11-27: Overall heat transfer effectiveness, ε_t , of an assembly of n identical exchangers in parallel in the stream of lower capacity rate, vs. effectiveness, ε , of a single exchanger. Calculated by the compiler after Domingos (1969) [36].	233
Figure 11-28: Overall heat transfer effectiveness, ε_t , of an assembly of n identical exchangers in parallel in the stream of higher capacity rate, vs. effectiveness, ε , of a single exchanger. Calculated by the compiler after Domingos (1969) [36].	234
Figure 11-29: Heat exchanger ineffectiveness, $1-\varepsilon$, vs. number of heat transfer units, N_{tu} , for a counterflow heat exchanger. Results are shown for $R = 1$ and different values of the wall conduction parameter, Δ . From Kroeger (1967) [110].	251
Figure 11-30: Heat exchanger ineffectiveness, $1-\varepsilon$, vs. number of heat transfer units, N_{tu} , for a counterflow heat exchanger. Results are shown for $R = 0,95$ and different values of the wall conduction parameter, Δ . From Kroeger (1967) [110].	251
Figure 11-31: Heat exchanger ineffectiveness, $1-\varepsilon$, vs. number of heat transfer units, N_{tu} , for a counterflow heat exchanger. Results are shown for $R = 0,90$ and different values of the wall conduction parameter, Δ . From Kroeger (1967) [110].	252
Figure 11-32: Heat exchanger ineffectiveness, $1-\varepsilon$, vs. dimensionless wall conduction, Δ , for a counterflow heat exchanger. Results are shown for $N_{tu} = 50$ and different values of the capacity-rate ratio, R . Calculated by the compiler after Kroeger (1967) [110].	252
Figure 11-33: Relative capacity, $\Sigma Q_n/Q$, of a simple two fluid heat exchanger vs. the maldistribution parameter, ϕ , for several values of the nominal number of heat transfer units, N_{tu} . From Weimer & Hartzon (1973) [187].	255

Figure 11-34: Relative surface requirements, $\Sigma A_n/A$, of a simple two fluid exchanger for fixed total duty vs. the maldistribution parameter, ϕ . Results are shown for several values of the nominal number of heat transfer units, N_{tu} . From Weimer & Hartzog (1973) [187].	255
Figure 11-35: Flow distribution model for the non-uniform side. From Fleming (1967) [65].	256
Figure 11-36: Overall effectiveness, ε , and effective number of heat transfer units, N_{tueff} , vs. the fraction, F_{Low} , of channels which carry lower-than-average flow on the nonuniform side of a "paired channels" heat exchanger. Results are shown for different values of the ratio of the capacity rate, C_{Low} , of a single channel with lower-than-average flow to the capacity rate, C_{High} , of a single channel with higher-than-average flow. (a) is for a nominal number of heat transfer units $N_{tu} = 10$ and (b) for $N_{tu} = 100$. From Fleming (1967) [65].	257
Figure 11-37: Overall effectiveness, ε , vs. the fraction, F_{Low} , of channels which carry lower-than-average flow on the nonuniform side of a heat exchanger with uniform side mixed. Results are shown for different values of the nominal number of heat transfer units N_{tu} , and of the ratio C_{Low}/C_{High} . From Fleming (1967) [65].	257
Figure 11-38: Ideal flow distribution in the shell side of shell-and-tube heat exchangers. ...	258
Figure 11-39: Schematic of the temperature distribution along the heat exchanger. (a) Balanced case. (b) Imbalanced case. From Cowans (1974) [28].	262
Figure 11-40: Flow imbalance compensating technique for gas to gas heat exchangers. From Cowans (1974) [28].	263
Figure 11-41: Plate spacing, (a), and fin spacing (b), type flow passage non-uniformities. From London (1970) [118].	264
Figure 11-42: Degradation in the heat exchanger thermal performance, measured by $Cost_{Ntu}$, and relative gain in pressure loss, $1-(\Delta p_1/\Delta p_n)$, as functions of the deviation in channel size, $1-(D_{E1}/D_{En})$. Results in (a) apply to any cylindrical passage provided that the non-uniformities are geometrically similar. Results in (b) are for non-uniformities of the fin-spacing type. From London (1970) [118].	266
Figure 11-43: Fin center of offset rectangular plat-fin surface. From Shah & London (1970) [164].	268
Figure 11-44: Assumed core geometries. From Shah & London (1970) [164].	269
Figure 11-45: Heat transfer, j , and friction, f , characteristics as functions of Reynolds number, Re , for surfaces 501 and 501 MOD. From Shah & London (1970) [164].	270
Figure 11-46: Flow area goodness factor, j/f , as a function of Reynolds number, Re , for surfaces 501 and 501 MOD. From Shah & London (1970) [164].	271
Figure 12-1: Typical characteristic curves of a centrifugal pump for a given rotating speed.	287
Figure 12-2: Rotodynamic pump impellers. From Nekrasov (1969) [132].	289
Figure 12-3: Characteristic curves of SEALED MOTOR CONSTRUCTION Centrifugal Pumps Cadet "Mini" and Cadet "S" pumping water. From Wyn-Roberts (1973) [194].	294
Figure 12-4: Characteristic curves of EURAMO Centrifugal Pumps Cadet MX 32-E and XA 15-R pumping water. From EURAMO-POMPES SALMSON (1977) [60].	294

Figure 12-5: Characteristic curve of the Centrifugal Pump devised by Engel & Walter Cadet "Mini" and Cadet "S" pumping water. From Wyn-Roberts (1973) [194].	295
Figure 12-6: Characteristic curves of Eastern-Iwaki Centrifugal Pumps MD-15T and MDR-30T pumping water. From GELBER (1976) [71].	295
Figure 12-7: Characteristic curve of Centrifugal Pump AC-3C-MD pumping water. From GELBER (1976) [71].	296
Figure 12-8: Characteristic curve of Positive Displacement Rotary Pump 413-7-1285 pumping water. From GELBER (1976) [71].	296
Figure 13-1: Schematic representation of the fluid loop considered by Barker, Stephens & Taylor (1967) [3].	299
Figure 13-2: Results of the liquid coolant system optimization analysis. a) gives the mass penalty, M^* , of the system components vs. Freon 21 mass flow rate, m_c . The mass penalty includes the equivalent mass due to pumping power. b) gives the system mass, M , vs. Freon 21 mass flow rate, m_c . From Baker, Stephens & Taylor (1967) [3].	301
Figure 13-3: Schematic representation of the fluid loop with EDHX. From Berner & Schleicher (1976) [13].	302
Figure 13-4: Mass penalty, M^* , of the supply and return plumbing system vs. source temperature, T_s . M^* includes mass of tubes and fluid plus the equivalent mass due to pumping. Heat load, $Q = 10^3$ W. From Berner & Schleicher (1976) [13].	304
Figure 13-5: Source temperature, T_s , vs. coolant mass flow rate, m_h , for different values of the heat transfer rate, Q . The interrupted lines are drawn through the points for which the power requirements, for a given D_i , equals 1 W. From Berner & Schleicher (1976) [13].	305
Figure 13-6: Schematic representation of basic ECLA system. The aim of the accumulator, which is not mentioned in the text, is to accommodate changes in coolant density with temperature. The coolant throttle is used for adjusting the coolant mass flow rate, m_h . From Berner & Schleicher (1976) [13].	306
Figure 14-1: Flow-pattern map for vertical upward air-water flow. Prepared by the compiler after Hewitt (1982) [86]. m_G and m_L are the gas and liquid mass flow rates, respectively. A_{FL} is the internal cross-sectional area of the duct.	309
Figure 14-2: Baker flow-pattern map for horizontal air-water flow. From Hewitt (1982) [86]. m_G and m_L are the gas and liquid mass flow rates, respectively. A_{FL} is the internal cross-sectional area of the duct.	310
Figure 14-3: Lockhart - Martinelli correlation for pressure loss multipliers. The figure has been drawn by use of Chisholm analytical representation. See text.....	313
Figure 14-4: Gas phase pressure loss multiplier, Φ_G , vs. Lockhart - Martinelli parameter, X , as deduced from Eqs. [14-9] and [14-10] and from experimental data. a) tt case; b) vt case; c) vv case. Experimental points are from Wallis (1969) [184].	315
Figure 14-5: Liquid fraction, $1-\alpha$, vs. Lockhart - Martinelli parameters, X . From Wallis (1969) [184].	316
Figure 14-6: Comparison of Lockhart - Martinelli with available experimental data. Points with C as per Eq. [14-13], with $\rho_G/\rho_L = 0$ are also shown. From Yang & Palen (1977) [195].	317

Figure 14-7: The annular flow configuration.....	318
Figure 14-8: Liquid film friction factor, f_L , as a function of liquid film Reynolds number, Re_L , for annular two-phase flow in cylindrical ducts of circular cross-section. From Hewitt (1982) [86]. The full lines correspond, respectively, to Hagen-Poiseuille formula and to Blasius formula. f_L is equal to the friction factor f_{sL} which corresponds to the single phase flow along the duct at the same Reynolds number except for an intermediate Re_L	321
Figure 14-9: Pressure gradient multipliers, Φ_L and Φ_G , dimensionless film thickness, δ/D , and liquid fraction, $1-\alpha$, according to Eqs. [14-34], [14-35] and [14-36], with $\rho_L/\rho_G = 1000$. Calculated by the compiler.	323
Figure 14-10: Comparison of the liquid fraction, $1-\alpha$, vs. Lockhart - Martinelli parameter, X , as deduced from Eqs. [14-34] and [14-35], with $\rho_L/\rho_G = 1000$, with experimental data from Wallis (1969) [184].	324
Figure 14-11: Comparison of the gas pressure gradient multiplier, $\Phi_G = x\Phi_L$, vs. Lockhart - Martinelli parameter, X , as deduced from Eqs. [14-34] and [14-36], with $\rho_L/\rho_G = 1000$, with experimental data from Wallis (1969) [184]. tt case.	324
Figure 14-12: The function $F(\delta/D)$ which appears in Eq. [14-62]. The effect of the liquid-gas density ratio is negligible for the range of values given in the figure. Compare the values given in this curve with those given by $2\delta/D$ vs. X in Figure 14-9.....	331
Figure 14-13: Martinelli parameter, X , vs. entrainment parameter, R_E , as deduced from air-water flow experiments. From Wicks & Duckler (1960) [190]. Note that R_E is not dimensionless.	332
Figure 14-14: Concentration of entrained droplets in the gas core, $\phi_G m_E/m_G$, vs. dimensionless number $\tau_i^* \delta/\sigma$. Replotted by the compiler after Hutchinson & Whalley (1973) [93]. Different symbols are used to indicate experiments by different authors.	333
Figure 14-15: Flow geometry when gravity is dominant (a) or negligible (b). The void fraction (vapor fractional area in the figure) is the same in both cases.	341
Figure 14-16: Geometry used in the model of stratified flow condensation.	342
Figure 14-17: Liquid fraction, $1-\alpha$, vs. vapor quality, w . For stratified condensing flow of several liquid along horizontal ducts. Calculated by the compiler.	348
Figure 14-18: Vapor quality, w , vs. dimensionless distance along the duct, x/D , for stratified condensing flow for several liquids along horizontal ducts. Calculated by the compiler.	349
Figure 14-19: The Taitel and Dukler limit for stratified flow in the w vs. $(1-\alpha)$ plane. Numbers on the curves represent constant values of the parameter $w^2(1-w)/\alpha^2(1-\alpha)$ which appears in the left hand side of Eq. [14-114].	350
Figure 14-20: Liquid-alone Nusselt number, Nu_{SL} , vs. liquid Reynolds number, Re_L , as given by different correlations in typical cases. Calculated by the compiler.	354
Figure 14-21: Liquid fraction, $1-\alpha$, as a function of vapor quality, w , for annular flow of several liquid along ducts. Calculated by the compiler.....	355
Figure 14-22: Vapor quality, w , as a function of dimensionless distance along the duct, x/D , for annular flow of several liquids along ducts. Calculated by the compiler.	356

Figure 15-1: Schematic of ATCS fluid loop in the Module-Pallet mode. From Owen, Sessions & Walker (1976) [139].	358
Figure 15-2: Schematic of three different types of two-phase flow loops. a) Parallel circuit. b) Series circuit. c) Series/parallel circuit.	359
Figure 15-3: Physical layout of 25 kW technology demonstrator (thermal bus). From Carlisle / Nolan (1987) [18].	360
Figure 15-4: Space platform configuration. From Sadunas, Lehtinen & Parish (1985) [153].	361
Figure 15-5: Study baseline centralized liquid loop external bus. From Sadunas, Lehtinen & Parish (1985) [153].	364
Figure 15-6: Concept III. Decentralized liquid loop. From Sadunas, Lehtinen & Parish (1985) [153].	366
Figure 15-7: Concept IV. Decentralized two-phase transport. From Sadunas, Lehtinen & Parish (1985) [153].	366
Figure 15-8: Trade study mass summary. From Sadunas, Lehtinen & Parish (1985) [153].	370
Figure 15-9: Decentralized systems with TS. a) Pumped liquid loop. b) Two-phase transport. From Lehtinen & Sadunas (1985) [114].	371
Figure 15-10: Radiator absorbed environmental heat flux, q_e , vs. dimensionless time, t/t_p . Orbital period, $t_p = 90$ min. Radiator limit: Black body emissive power. From Lehtinen & Sadunas (1985) [114].	371
Figure 15-11: Thermal performance of PC capacitors in different subsystems. a) Metabolic. b) Equipment. c) Fuel cell. Thermal performance is given in terms of: 1) energy storage rate, Q_c ; and 2) net stored energy, J_c . From Lehtinen & Sadunas (1985) [114].	374
Figure 15-12: Required specific storage capacity. J_c/Q_c , of typical TMSs vs. temperature, T . From Lehtinen & Sadunas (1985) [114].	376
Figure 15-13: Schematic of the fuel cell loop TMS. a) Pumped liquid loop. b) Two-phase transport. From Sadunas, Lehtinen & Parish (1986) [153].	377
Figure 15-14: Respond-to-demand temperature control scheme From Sadunas, Lehtinen, Nguyen & Parish (1986) [154].	379
Figure 15-15: Orbital-average temperature control scheme. From Sadunas, Lehtinen, Nguyen & Parish (1986) [154].	379
Figure 15-16: Instrumentation of the different control schemes. a) "Respond-to-demand" pumped-liquid-loop scheme. b) "Orbital-average" pumped-liquid-loop scheme. c) Two-phase transport loop scheme. From Sadunas, Lehtinen, Nguyen & Parish (1986) [154].	381
Figure 15-17: Evolution of the heat rejection rate and configuration of spacecraft thermal control systems. From Dexter & Haskin (1984) [34].	383
Figure 15-18: Schematic of the heat pump working principle. Adapted by the compiler after Rye & Steen (1986) [152].	383
Figure 15-19: Radiator mass, M_R , of four typical systems (see text above) for different values of the heat rejection rate, Q . Calculated by the compiler.	385
Figure 15-20: Mass breakdown of a heat pump augmented system for different values of the heat rejection rate, Q . $T_R = 300$ K. Calculated by the compiler.	386

Figure 15-21: Mass breakdown of a heat pump augmented system for different values of the heat rejection rate, Q . a) $T_R = 340$ K. b) $T_R = 380$ K. Calculated by the compiler.	387
Figure 15-22: Ratio of total mass, M , of the heat pump augmented system to the total mass, M_r , of the reference classical pumped-liquid loop vs. the inverse Carnot coefficient of performance $[T_R - T_c]/T_c$ for different values of the heat rejection rate, Q . Calculated by the compiler.	388
Figure 15-23: Effect of α_s/ε on radiator specific area, A_R/Q . From Lehtinen & Sadunas (1985).	389
Figure 15-24: Radiator specific area, A_R/Q , vs. refurbishment frequency, t . From Lehtinen & Sadunas (1985) [114].	390
Figure 15-25: Steerable to fixed radiator specific heat-rejection ration, $(Q/A_R)_{steer}/(Q/A_R)$, vs. radiator temperature, T_R . From Sadunas, Lehtinen & Parish (1985) [154].	392
Figure 15-26: Rotatable radial flow heat pipe. From Delil (1986a) [33].	396
Figure 15-27: Thermal joint based on a radial flow heat pipe. From Hinderer & Savage (1978) [88].	396
Figure 15-28: Heat transfer enhancement by flow oscillation demonstrator. From Kurzweg & Zhao (1984) [113].	397
Figure 15-29: Hemispherical heat pipe junction. From Delil (1986a) [33].	397
Figure 15-30: Cylindrical heat pipe junction. From Delil (1986a) [33].	397
Figure 15-31: Heat pipe in heat pipe joint. This concept is an off-spring of the cylindrical heat pipe joint. From Shaubach (1985) [165].	398
Figure 15-32: Flexible heat pipe consisting of a 0,15 m long rigid evaporator and a 0,20 m long rigid condenser. From Delil (1986a) [33].	398
Figure 15-33: Flexible cooper/acetone heat pipe. It features a 0,381 m long adiabatic section between an evaporator and a condenser both 0,178 m long. Inside diameter is 0,016 m. From Delil (1986a) [33].	398
Figure 15-34: Finned heat exchangers. Interstice either vacuum-filled or filled with a gas, a low melting point or a grease. From French (1985) [68].	398
Figure 15-35: Grease-filled heat pipe-heat pipe joint. Hinge joint-coaxial with the internal heat pipe and parallel to the plane of the radiator heat pipe. From Delil (1986a) [33].	399
Figure 15-36: Braided conductor flexible thermal joint. From Delil (1987a) [31].	399
Figure 15-37: Clamped joint contact conductor. Heat pipe penetrates heat exchanger. Fluid pressure-enhanced thermal contact. From Ellis & Rankin (1983) [42].	400
Figure 15-38: Self deployed membrane heat pipe radiator. Both deployed and undeployed configurations shown in the figure. From Delil (1986a) [33].	400
Figure 15-39: Internal details of the rotatable fluid transfer coupling. Dimensions are in mm. From Heizer, Goo, Rhodes, Thoreson & Parish (1986) [83].	401
Figure 15-40: Rotatable fluid transfer coupling. a) General view showing the liquid and vapour radial ports. b) Channel geometry. From Heizer, Goo, Rhodes, Thoreson & Parish (1986) [83].	401

Figure 15-41: Pressure drop, Δp , along the duct at one position of the rotary coupling for different values of the heat rejection rate, Q . From Heizer, Goo, Rhodes, Thoreson & Parish (1986) [83].	402
Figure 15-42: a) Schematic of a three-axis stabilized spacecraft in geosynchronous orbit. From Chalmers & Pustay (1986). b) Typical thermal load sharing of east-west faces. From Wise (1986) [192].	403
Figure 15-43: East-west radiator coupling. a) Based on HP technology. From Chalmers & Pustay (1986). b) Based on CPL technology. From Chalmers, Pustay, Moy & Kroliczek (1986) [23].	404
Figure 15-44: Schematic of CPL engineering model. In a) the liquid fills most of the loop, whereas in b) part of this liquid has been evaporated. From Chalmers, Pustay, Moy & Kroliczek (1986) [23].	405
Figure 15-45: Schematic of a basic CPL system. From Chalmers, Pustay, Moy & Kroliczek (1986) [23].	409
Figure 15-46: a) Standard CPL evaporator pump. From Chalmers et al. (1986) [23]. b) Heat flow in a cross section of a typical CPL evaporator. From Wise (1986) [192].	409
Figure 15-47: Prototype capillary cold plate (PCCP) design. Dimension in mm. From Chalmers, Pustay, Moy & Kroliczek (1986) [23].	410
Figure 15-48: CPL technology radiators. a) Direct condensation radiator. b) Heat exchanger-heat pipe radiator. From Chalmers, Pustay, Moy & Kroliczek (1985) [23].	410
Figure 15-49: Monogroove heat pipe. From Alario, Haslett & Kosson (1981) [1].	412
Figure 15-50: Osmotically pumped heat transfer system. From Tanzer, Fleischman & Stalmach (1982) [175].	413
Figure 15-51: Biomorph (Biomorph) pump. From Peterson (1987) [142].	413
Figure 15-52: Instrument-TMS interfaces. From Almgren et al. (1981) [2].	415
Figure 15-53: Schematic of a grooved cold plate/hot plate in the cold plate mode. From Hwangbo & McEver (1985) [94].	415
Figure 15-54: Two-Phase Mounting Plate (TPMP) development unit. The Lexan window has been incorporated to observe the flow during testing. All dimensions are in mm. From Grote & Swanson (1985) [77].	416
Figure 15-55: Void fraction sensors of the capacitance type. a) Single coaxial capacitor. b) Concave plate capacitor. c) Double helix capacitor. d) Film thickness gage. From Delil (1986b) [30].	418
Figure 15-56: Dimensionless capacitance as a function of void fraction, α . a) Annular flow, single coaxial capacitor. b) Annular flow, concave plate capacitor. From Delil (1986b) [30].	419
Figure 15-57: Dimensionless capacitance as a function of dimensionless film thickness, δ/d . Flat wall, film thickness gage. From Delil (1986b) [30].	420
Figure 15-58: Engagement sequence of the RSO disconnect. Disengagement is achieved through the reverse sequence. From MOOG [127].	421
Figure 16-1: Closed-loop control system block diagram.	424
Figure 16-2: Open-loop control system block diagram.	424
Figure 16-3: Typical block diagram of adaptative control systems.	425

Figure 16-4: Feedback system	426
Figure 16-5: Sketch of the gain of a system as a function of frequency.	427
Figure 16-6: Feedback system with two feedback loops.....	428
Figure 16-7: Feedback system with a noise signal.	429
Figure 16-8: Feedback structure of instruments and regulators. From Ogata (1990) [137].....	431
Figure 16-9: Block diagram of an industrial control system, which consists of an automatic controller, an actuator, a plant, and a sensor (measuring element). From Ogata (1990) [137].	432
Figure 16-10: a) Block diagram of an on-off controller; b) block diagram of an on-off controller with differential gap; c) output versus time curve. From Ogata (1990) [137].	433
Figure 16-11: Block diagram of a proportional controller. From Ogata (1990) [137].	434
Figure 16-12: Block diagram of an integral controller. From Ogata (1990) [137].	435
Figure 16-13: a) Block diagram of a proportional-integral controller; b) and c) diagrams depict a unit-step input and the controller output. From Ogata (1990) [137].	436
Figure 16-14: a) Block diagram of a proportional-derivative controller; b) and c) diagrams depict a unit-ramp input and the controller output. From Ogata (1990) [137].	437
Figure 16-15: a) Block diagram of a proportional-integral-derivative controller; b) and c) diagrams depict a unit-ramp and the controller output. From Ogata (1990) [137].	438
Figure 16-16: Block diagram of a digital control system. From Ogata (1987) [135].	440
Figure 16-17: Block diagram of a digital control system showing signals in binary or graphic form. From Ogata (1987) [135].	440
Figure 16-18: Schematic diagram of a pressure system. From Ogata (1990) [137].	441
Figure 16-19: Schematic diagram of a pneumatic actuating valve. From Ogata (1990) [137].	443
Figure 16-20: a) Dashpot; b) step change in x and the corresponding change in y plotted versus t ; c) block diagram of the dashpot. From Ogata (1990) [137].	444
Figure 16-21: a) Schematic diagram of a force-distance type pneumatic proportional controller; b) block diagram; c) simplified block diagram. From Ogata (1990) [137].	446
Figure 16-22: Schematic diagram of a force-balance pneumatic proportional controller. From Ogata (1990) [137].	447
Figure 16-23: a) Servomotor that acts as a proportional controller; b) block diagram of the servomotor. From Ogata (1990) [137].	448
Figure 16-24: a) Pneumatic proportional-derivative controller; b) step change in e and the corresponding changes in x and p_c plotted versus t ; c) block diagram. From Ogata (1990) [137].	449
Figure 16-25: a) Sketch of a hydraulic proportional-derivative controller; b) block diagram. From Ogata (1990) [137].	450
Figure 16-26: Hydraulic integral controller. From Ogata (1990) [137].	452

Figure 16-27: a) Pneumatic proportional-integral controller; b) step change in e and the corresponding changes in x and p_c plotted versus t ; c) block diagram on the controller; simplified block diagram. From Ogata (1990) [137].	453
Figure 16-28: a) Schematic diagram of hydraulic proportional-integral controller; b) block diagram. From Ogata (1990) [137].	454
Figure 16-29: a) Pneumatic proportional-integral-derivative controller; b) block diagram of the controller. From Ogata (1990) [137].	455
Figure 16-30: Fluidloop modelled as a control system.	458
Figure 16-31: Control block diagram of the fluid loop for cooling Spacelab experiments. From Microtecnica (1977) [126].	459
Figure 16-32: Instrumentation and control system schematics. From Sadunas et al. (1986) [154].	460
Figure 16-33: Block diagrams of automatic controllers with a) first-order sensor; b) overdamped second-order sensor; c) underdamped second-order sensor. From Ogata (1990) [137].	465
Figure 16-34: Block diagram of a control system.	465
Figure 16-35: Space radiator system. From Baker et al. (1967) [3].	472
Figure 16-36: Space radiator block diagram. From Baker et al. (1967) [3].	474

Tables

Table 7-1: Effective roughness height of a number of common surfaces.	95
Table 7-2: Constants for Power Law Approximation. Hydraulically Smooth Regime.	97
Table 7-3: Loading Factors Accounting for Temperature-Dependence of Diabatic Friction	98
Table 9-1: Heat Transfer Enhancement Techniques	130
Table 9-2: Ratio of heat transfer coefficients for constant wall temperature, Nu_T , and constant heat flux, Nu_q , for turbulent pipe flow. From Reynolds (1974) [149].	134
Table 10-1: Physical Properties of Typical Liquid Coolants	180
Table 10-2: Environmental Properties of Typical Liquid Coolants	183
Table 10-3: Properties of Dow Corning 200 Fluids (Dimethyl Siloxane Polymers)	205
Table 10-4: Corrosion and Oxidation Test Data for Coolanol Liquids.	207
Table 10-5: Toxicity of Several Freon Liquids	208
Table 10-6: Swelling of Elastomers in Several Freon Liquids ^a	208
Table 10-7: Compatibility of Freon E2 and FC-75 with Elastomers ^a	209
Table 10-8: Compatibility of Freon E3 with Elastomers, Plastics and Wire Coatings ^a	209
Table 10-9: Compatibility of Oronite Flo-Cool 100 with Elastomers	211
Table 11-1: Flow Inside Circular and Flattened Circular Tubes.	237
Table 11-2: Flow Normal to Banks of Bare Tubes	239
Table 11-3: Plate-Fin surfaces, plain fins.	240
Table 11-4: Plate-Fin surfaces, louvered fins.	242

Table 11-5: Plate-Fin surfaces, strip fins	243
Table 11-6: Plate-Fin surfaces, wavy fins	244
Table 11-7: Plate-Fin surfaces, perforated fins	244
Table 11-8: Plate-Fin surfaces, pin fins	245
Table 11-9: Finned tubes, circular tubes, circular fins	246
Table 11-10: Finned tubes, circular tubes, continuous fins	247
Table 11-11: Finned tubes, flat tubes, continuous fins	247
Table 11-12: Crossed-Rod, woven-screen and sphere matrices	248
Table 11-13: Fluid Paths in the Shell Side of Shell-and-Tube Exchangers ^a	259
Table 11-14: Engineering Practices for Reducing Maldistribution in the Shell Side of Shell-and-Tube Exchangers ^a	260
Table 11-15: Nominal Dimensions for Numerical Results	264
Table 11-16: Basic assumptions for the theoretical analysis	264
Table 11-17: Characteristics of Offset Rectangular Plate-Fin Surfaces	268
Table 11-18: Core 501 MOD Geometries Derived from Different Models	269
Table 11-19: Fouling Resistance of Several Common Materials ^a	272
Table 11-20: Fouling Mechanisms ^a	273
Table 11-21: Effects of Different Physical Parameters on Fouling ^a	274
Table 12-1: Rotodynamic Pumps	284
Table 12-2: Displacement Pumps. Reciprocating	284
Table 12-3: Displacement Pumps. Rotary	285
Table 12-4: Main Features of Typical Pumps	285
Table 12-5: Conversion Factors in the Deduction of Ω from n_s	288
Table 12-6: Characteristics of Several Commercially Available Pumps	292
Table 13-1: Optimization of the Liquid to Air Heat Exchanger Case 1) $m_c = 0,0303$ kg.s^{-1} , $T_S - T_{ci} = 20 \text{ K}$	307
Table 13-2: Optimization of the Liquid to Air Heat Exchanger Case 2) $m_c = 0,0817$ kg.s^{-1} , $T_S - T_{ci} = 10 \text{ K}$	308
Table 14-1: Typical Pressure Losses in Air-Water Annular Flow with Entrainment ($D =$ $25,4 \times 10^{-3} \text{ m}$, $p = 10^5 \text{ Pa}$, $T = 293 \text{ K}$).	338
Table 14-2: Geometry and Flow-Dependent Terms in Eqs. [14-107] and [14-109]	345
Table 14-3: Condensation in Ducts. Typical Fluid Properties. Assumed values: $T_{sat} =$ 300 K , $T_{sat} - T_w = 10 \text{ K}$	347
Table 14-4: Condensation in Ducts. Parameters Depending on m and D . Assumed values: $m h_{fg} = 1,5 \times 10^3 \text{ W}$, $D = 16,1 \times 10^{-3} \text{ m}$. Stratified case	348
Table 15-1: Evolution of Requirements.....	357
Table 15-2: Two-Phase Loop Line Patterns.....	359
Table 15-3: Power Dissipation by Module.....	362
Table 15-4: TMS Design Requirements.....	363

Table 15-5: TMS Design Concepts.....	365
Table 15-6: TMS Design Goals	365
Table 15-7: Evaluation of Concepts.....	368
Table 15-8: Characteristics of Single-Phase and Two-Phase TMSs with PC Capacitor.....	372
Table 15-9: PC Capacitor Performance.....	373
Table 15-10: Single-Phase and Two-Phase TMS Capacitor Specifications	375
Table 15-11: Fuel Cell Loop Design Requirements	378
Table 15-12: Orbital-Average Schema. Values of x for Off-Design Operation.....	380
Table 15-13: Two-Phase System. Radiator Temperatures for Off-Design Operation.....	380
Table 15-14: Component Mass of a Typical Pumped Loop TMS	382
Table 15-15: Heat Pump Augmented vs. Classical Fluid Loop Trade-Off	384
Table 15-16: EOL Radiator Area for 10-year Life.....	390
Table 15-17: Radiator Area vs. Design Life	391
Table 15-18: Comparison of Fixed and Steerable Radiator Areas	392
Table 15-19: Requirements for a Rotary Coupling Onboard. Space Station.	393
Table 15-20: Joints for Steerable Radiators.....	394
Table 15-21: NASA development efforts in CPL technology.....	406
Table 15-22: Comparative Summary of Pumping Systems.....	414
Table 16-1: Summary of basic control actions	438
Table 16-2: Summary of Implementation Techniques.....	456
Table 16-3: Control unit philosophy trade-off. From Microtecnica (1977) [126].....	461
Table 16-4: Space computers.....	463
Table 16-5: Available microprocessor options	464
Table 16-6: Characteristics of several temperature sensors.	466
Table 16-7: Characteristics of pressure sensors.....	467
Table 16-8: Characteristics of flow sensors.	468
Table 16-9: Characteristics of control valves. From Liptak (1969) [115].....	468
Table 16-10: Some MATLAB-Driven CACSD Software .From Cellier and Rimvall (1988) [21].....	470
Table 16-11: Some non-MATLAB CACSD Software Packages. From Cellier and Rimvall (1988) [21]	470
Table 16-12: A Brief Survey of 22 CACSD Packages. From Jamshidi et al. (1992) [100] ...	471
Table 16-13: System characteristic. From Backer et al. (1967) [3].....	473
Table 16-14: Control law for the block diagram in Figure 16-36.....	475

European Foreword

This document (CEN/CLC/TR 17603-31-13:2021) has been prepared by Technical Committee CEN/CLC/JTC 5 “Space”, the secretariat of which is held by DIN.

It is highlighted that this technical report does not contain any requirement but only collection of data or descriptions and guidelines about how to organize and perform the work in support of EN 16603-31.

This Technical report (TR 17603-31-13:2021) originates from ECSS-E-HB-31-01 Part 13A .

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. CEN [and/or CENELEC] shall not be held responsible for identifying any or all such patent rights.

This document has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association.

This document has been developed to cover specifically space systems and has therefore precedence over any TR covering the same scope but with a wider domain of applicability (e.g.: aerospace).

1

Scope

Fluid loops are used to control the temperature of sensitive components in spacecraft systems in order to ensure that they can function correctly.

While there are several methods for thermal control (such as passive thermal insulations, thermoelectric devices, phase change materials, heat pipes and short-term discharge systems), fluid loops have a specific application area.

This Part 13 provides a detailed description of fluid loop systems for use in spacecraft.

The Thermal design handbook is published in 16 Parts

TR 17603-31-01	Thermal design handbook – Part 1: View factors
TR 17603-31-02	Thermal design handbook – Part 2: Holes, Grooves and Cavities
TR 17603-31-03	Thermal design handbook – Part 3: Spacecraft Surface Temperature
TR 17603-31-04	Thermal design handbook – Part 4: Conductive Heat Transfer
TR 17603-31-05	Thermal design handbook – Part 5: Structural Materials: Metallic and Composite
TR 17603-31-06	Thermal design handbook – Part 6: Thermal Control Surfaces
TR 17603-31-07	Thermal design handbook – Part 7: Insulations
TR 17603-31-08	Thermal design handbook – Part 8: Heat Pipes
TR 17603-31-09	Thermal design handbook – Part 9: Radiators
TR 17603-31-10	Thermal design handbook – Part 10: Phase – Change Capacitors
TR 17603-31-11	Thermal design handbook – Part 11: Electrical Heating
TR 17603-31-12	Thermal design handbook – Part 12: Louvers
TR 17603-31-13	Thermal design handbook – Part 13: Fluid Loops
TR 17603-31-14	Thermal design handbook – Part 14: Cryogenic Cooling
TR 17603-31-15	Thermal design handbook – Part 15: Existing Satellites
TR 17603-31-16	Thermal design handbook – Part 16: Thermal Protection System