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Space Engineering — Thermal design handbook

Part 13: Fluid Loops

National foreword

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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

2 References

EN reference	Reference in text	Title
EN 16601-00-01	ECSS-S-ST-00-01	ECSS System - Glossary of terms
TR 17603-31-08	ECSS-E-HB-31-01 Part 8	Thermal design handbook – Part 8: Heat Pipes
TR 17603-31-09	ECSS-E-HB-31-01 Part 9	Thermal design handbook – Part 9: Radiators
TR 17603-31-10	ECSS-E-HB-31-01 Part 10	Thermal design handbook – Part 10: Phase-Change Capacitors
TR 17603-31-14	ECSS-E-HB-31-01 Part 14	Thermal design handbook – Part 14: Cryogenic Cooling
TR 17603-31-15	ECSS-E-HB-31-01 Part 15	Thermal design handbook – Part 15: Existing Satellites

All other references made to publications in this Part are listed, alphabetically, in the **Bibliography**.

Terms, definitions and symbols

3.1 Terms and definitions

For the purpose of this Standard, the terms and definitions given in ECSS-S-ST-00-01 apply.

3.2 Abbreviated terms

The following abbreviated terms are defined and used within this Standard.

A/D	analog-to-digital converter
ARS	atmosphere revitalization subsystem
ATCS	active thermal control system
BOL	beginning of life
CACSD	computer-aided control system design
CAD	computer-aided design
CAPL	capillary pumped heat transport loop
CPL	capillary pumped loop
CPU	central processing unit
D/A	digital-to-analog converter
DCR	direct condensation radiator
ECLA	experiment cooling system with liquid to air heat exchanger
EDHX	experiment dedicated heat exchanger
EOL	end of life
FCL	freon coolant loop
FL	fully developed flow
GSE	ground support equipment

HP	heat pipe
HPSTM	high power spacecraft thermal management
HTEL	combined hydrodynamical and thermal entry length
HX	heat exchanger
I	integral controllers
I/O	input/output
Mac	maximum allowable concentration
MBA	multiple berthing adaptor
MPU	microprocessor unit
NPSH	net positive suction head
OA	orbiting astronomical observatory
P	proportional controllers
PC	phase change
PCCP	prototype capillary cold plates
PCM	phase change material
PD	proportional-derivative controllers
PI	proportional-integral controllers
PID	proportional-integral-derivative controllers
S/H	sample-and-hold
SLMS	seal leakage management subsystem
STS	space transportation system
TED	thermoelectric device
TEL	thermal entry length
TMS	thermal management system
TS	thermal storage
VCHP	variable conductance heat pipe

3.3 Symbols

A	heat transfer surface area, [m ²]
A_{FL}	internal cross-sectional area of a duct. [m ²]. Also called free flow area
A_{FR}	frontal area of a heat exchanger core, [m ²]
A_G	gas filled part of the cross-sectional area of a duct. [m ²]
A_L	liquid filled part of the cross-sectional area of a duct, [m ²]
A_f	fin area, [m ²]
A_k	wall cross-sectional area for longitudinal heat conduction, [m ²]
A_w	Clause 9: inside wall heat transfer surface area, [m ²] Clause 11: average wall area, [m ²]
C	Chaps 5 and 11: capacity rate of a flow stream, [W.K ⁻¹] $C = mc_p$ Clause 14: Chisholm parameter. Eq. [14-9] Clause 15: electrical capacitance, [F]
C^*	capacity rate of the in parallel stream, [W.K ⁻¹] refers to assemblies of heat exchangers in parallel in one of the streams
$Cost_{Ntu}$	degradation in the heat exchanger thermal performance, $Cost_{Ntu} = 1 - (N_{tueff}/N_{tu})$
D	diameter, [m]
D_E	equivalent or hydraulic diameter of a duct of Non-Circular Cross Section. [m]. $D_E = 4A_{FL}/\Pi$.
D_{Ea}	equivalent or hydraulic diameter of a finned duct, [m]
D_M	inner diameter of a rough tube defined on a volumetric basis, [m] $D_M = (4\dot{v}/\pi L)^{1/2}$
D_b	diameter of the largest cross section in an axial displacement heat transfer enhancement device. [m]
D_m	no-shear diameter in the flow through an annular duct, [m]
E	emissive power, [W.m ⁻²]
E_L	specific energy lost by a pump, [J.kg ⁻¹] it is defined per unit liquid mass

E_n	specific energy developed by a pump, [J.kg ⁻¹] it is defined per unit liquid mass
E_s	specific energy supplied to a pump, [J.kg ⁻¹] it is defined per unit liquid mass
E_v	specific suction energy of a pump, [J.kg ⁻¹] it is defined per unit liquid mass
F	Clause 9: fin factor. ratio of the total convective heat transfer to that transferred through the tube walls alone, Clause 10: cooling effectiveness of a fluid
F_{Low}	fraction of channels carrying lower-than-average flow on the nonuniform side of a split counterflow heat exchanger (fleming's model)
F_r	froude number, $F_r = V^2/gD$
F_1	factor in tube banks, it accounts for variable fluid properties
F_2	factor in tube banks, it accounts for abnormal number of rows, ($N \neq 10$)
F_3	factor in tube banks, it accounts for the effect of tube yaw
F_4	factor in tube banks, it gives the row to row variation of heat transfer coefficient
Gr	Grashof number. $Gr = \frac{\beta g D_E^3 \rho^2 T_w - T_b }{\mu^2}$
Gz	Graetz number, $Gz = \pi D_E / 4 L P r$ $Re = m c_p / k L$.
H	Clause 12: head, [m] Clause 14: liquid level in stratified flow, [m]
J_c	thermal capacitor stored energy, [J]
K	Clause 6: dean number, $K = Re (r/R)^{1/2}$ Clause 7: correction factor, it accounts for the fact that the cross section of the tube is not circular
L	length, [m]
L_E	equivalent length, [m]
L_{ent}	entrance length, [m]
M	Clause 7: mach number, $M = V/a$ Clauses 13 and 15: system mass, [kg]

M_{CP}	cold plate mass, [kg]
M_P	fluid driver mass, [kg]
M_{SP}	solar panel mass, [kg]
M_{wL}	(wet) plumbing mass, [kg]
M^+	mass penalty if a fluid loop, [kg] it includes the equivalent mass due to pumping power
N	Clause 6: number of rows in tube banks, the rows are counted in the flow direction Clause 11: passage count of a heat exchanger core, [m ⁻²]
N_{TD}	parameter in the Taitel-Duckler limit for stratified flow, Eq. (10.42)
N_{tu}	number of heat transfer units
N_{tu}^o	overall number of heat transfer units, refers to a liquid-coupled system of heat exchangers $N_{tu}^o = \left(\frac{1}{N_{tu1}} + \frac{1}{N_{tu2}} \right)^{-1}$
Nu	Nusselt number, $Nu = hD_E/k$.
Nu_T	Nusselt number for constant wall temperature
Nu_q	Nusselt number for constant heat flux through the wall
Nu_{11}	Nusselt number for the inner wall of a cylindrical annulus when the outer wall is insulated
Nu_{22}	Nusselt number for outer wall of a cylindrical annulus when the inner wall is insulated
P	power supplied to a pump, [W]
Pe	Peclet number, $Pe = PrRe$.
Pr	Prandtl number, $Pr = \mu c_p/k$.
Q	heat transfer rate, [W]
$Q(x)$	convective heat transfer rate in a tube, from onset of heat transfer to station x, [W]
Q_c	thermal capacitor energy storage rate, [W]
R	radius of the centerline in curved pipes, [m] Clauses 5, 11 and 13: capacity ratio, $R = C_1/C_2 \leq 1$

R_E	Wicks-Dukler entrainment parameter, [m ² .s] Eq. (10.29)
R	Reynolds number, $Re = \rho V D_E / \mu$
S	suction specific speed of a pump at given operating conditions, $S = \frac{\omega(m/\rho)^{1/2}}{E_v^{3/4}}$
Sf	Stefan number, $Sf = \frac{h_{fg}}{c_{pL}(T_{sat} - T_w)}$
St	Stanton number, $St = Nu/PrRe = h/\rho V c_p$
T	temperature, [K]
T_R	reference temperature, [K] Clause 15: radiator temperature, [K]
T_S	source temperature, [K]
T_{av}	fluid average temperature, $T_{av} = \frac{\int T dA_{FL}}{A_{FL}}$
T_b	fluid bulk temperature, $T_b = \frac{\int u T dA_{FL}}{V A_{FL}}$
T_{bam}	arithmetic mean bulk temperature [K], $T_{bam} = (T_{bi} + T_{bo})/2$.
T_c	Clauses 5, 11, 13 and 15: cold temperature, [K] Clause 9: cooling water temperature, [K]
T_{sat}	liquid-vapor temperature, [K]
T_w	wall temperature, [K]
T_s	fluid film temperature, [K], $T_s = (T_w + T_b)/2$
T^+	dimensionless temperature, $T^+ = \rho u^* c_p / h$
U	overall thermal conductance, [W.m ⁻² .K ⁻¹], $U = q/(T_h - T_c)$.
V	mean fluid velocity, [m.s ⁻¹]
We	Weber number, $We = \rho_c V^2 D / \sigma$.
X	Lockhart-Martinelli parameter. Eq. [14-3]

Z	influence coefficient, it relates the one-surface-heated Nusselt number for the cylindrical annulus to Nusselt numbers on both surfaces when they are heated with any ration of heat fluxes.
a	Clause 7: velocity of sound, [m.s ⁻¹]. Clauses 11 and 13: wall or plate thickness, [m]
a'	parting sheet thickness, [m]
b	protrusion spacing, [m] it can be measured in either axial or circumferential direction
c	mass fraction of a given species, when several species are present
c_K	pressure loss coefficient, $c_K = (p_1 - p_2)/(1/2\rho V^2)$
c_p	constant-pressure fluid specific heat, [J.kg ⁻¹ .K ⁻¹] for gases c_p is used in distinction to c_v , the constant-volume specific heat
d	Clause 7: diameter of an orifice, wire or tube placed inside a duct, [m] Clause 11: cell height, [m] Clause 15: interplate distance of a capacitor [m]
e	roughness height, [m]
e^+	dimensionless roughness height, $e^+ = eu^*/\nu = (e/D)Re(f/2)^{1/2}$
f	fanning friction factor, $f = \lambda/4$
f_c^*	fanning friction factor for the homogeneous core mixture flowing alone in the duct
f_F	fanning liquid film friction factor at the wall, annular flow with entrainment
f_{Gi}	fanning core friction factor at the interface, ideal annular flow
f_{Gi}^*	fanning core friction factor at the interface, annular flow with entrainment
f_L	fanning liquid film friction factor at the wall, ideal annular flow
f_{TP}	fanning friction factor at the wall, two-phase mixture model
g	acceleration due to gravity, [m.s ⁻²] Clause 5: dimensionless heat transfer similarity function

h	convective heat transfer coefficient, [W.m ⁻² .K ⁻¹]
h_c	thermal joint conductance, [W.m ⁻² .K ⁻¹]
h_{fg}	heat of vaporization or condensation, [J.kg ⁻¹]
h'_{fg}	heat of condensation corrected by partial subcooling effect, [J.kg ⁻¹] Eq. (10.34)
j	colburn factor for heat transfer, $j = StPr^{2/3}$
k	thermal conductivity, [W.m ⁻¹ .K ⁻¹] Clause 7: incremental pressure loss factor, it accounts for entrance effects
m	fluid mass flow rate, [kg.s ⁻¹] Clauses 6 and 9: exponent, it appears in the loading factor accounting for the influence of fluid temperature on heat transfer
m_E	mass flow rate of the fluid entrained in the core, [kg.s ⁻¹] annular flow with entrainment
m_F	mass flow rate of the liquid remaining in the film, [kg.s ⁻¹] annular flow with entrainment
n	Clause 6: number of tubes in tube banks, also number assigned to the nth row tubes in tube banks Clause 7: number of wires per unit length of wire gauze, [m ⁻¹] exponent Clauses 11 and 13: number of assembled heat exchangers Clause 12 and 15: rotating speed, [revolutions per minute]
n_s	(optimum) specific speed of a pump, [m ^{3/4} .s ^{-3/2}] $n_s = \frac{n(m/\rho)^{1/2}}{H^{3/4}} \Big _{opt}$
p	pressure, [Pa]
p_{sat}	liquid-vapor pressure, [Pa]
p_t	total pressure, [Pa]
q	heat flux, [W.m ⁻²]
r	radius or radial coordinate, [m]
r_f	fouling resistance, [K.m ⁻² .W ⁻¹]
s	protrusion thickness, [m] it is measured in axial direction

s_1	longitudinal pitch in tube banks, [m]
s'_1	diagonal pitch in staggered tube banks, [m]
s_t	transverse pitch in either in-line or staggered tube banks, [m]
t	Clauses 7 and 9: thickness, [m] Clause 11: temperature, [K] Clause 15: time, [s], [min], [h] or [yr]
u	local fluid velocity [m.s ⁻¹]
u^*	friction velocity, [m.s ⁻¹], $u^* = (\tau_w/\rho)^{1/2}$.
u^+_c	dimensionless velocity, $u^+ = u/u^*$
u^+	dimensionless friction factor similarity function
v	volume, [m ³]
w	vapor quality
x	axial distance, [m] Clause 11: mass flow fraction
x^+	dimensionless axial distance to duct entry, $x^+ = (x/r)/PrRe = \pi/2Gz$
y	Clause 9: tube diameter per 180° fin or tape twist Clause 14: distance to the wall of the duct, [m]
z	vertical distance above arbitrary datum plane, [m]
ΔT_m	mean wall-to-fluid bulk temperature differential, [K]
Δp	pressure loss, [Pa]
Γ	radiator area density, [kg.m ⁻²]
Λ	longitudinal wall conduction parameter, $\Lambda = (k/L)A_k/C_1$
Π	wetted perimeter of a duct, [m]
Φ	Chaps 5, 7 and 11: ratio of free-flow area to total frontal area, also called porosity Clause 14: Lockhart-Martinelli pressure loss multiplier Eq. [14-2]
Ψ	upstream to downstream area ratio, refers to sudden area changes in ducts, factor in tube banks, it accounts for the effect of tube yaw on pressure loss
Ω	(optimum) specific speed of a pump, ECUACION

α	<p>Clause 11: area density, [m².m⁻³], ratio of total heat transfer area on one side of a closed-type heat exchanger or recuperator to total exchanger volume, in the case of periodic type exchangers or regenerators α is based on the area of both sides</p> <p>Clauses 14 and 15: void fraction</p>
α_1	factor in bends, it accounts for the aspect-ratio of the bend cross section
α_2	factor in bends, it accounts for the bend angle
α_3	factor in orifices, it accounts for the relative thickness of an orifice plate when $t/d < 0,8$
α_4	factor in orifices, it accounts for the relative thickness of an orifice plate when $t/d \geq 0,8$
α_5	factor in screens, it accounts for low Reynolds number in round-wire gauzes
α_6	factor in valves, it accounts for partial opening
α_s	solar absorptance
β	<p>Clauses 5 and 11: area density, [m².m⁻³] ratio of total heat transfer area on one side of a plate-fin closed-type heat exchanger or recuperator to the volume between the plates on that side, in the case of periodic type exchangers or regenerators β is based on the area of both sides</p> <p>Clauses 6 and 9: coefficient of volumetric thermal expansion, [K⁻¹]</p> <p>Clause 7: factor in bends, it accounts for the interaction between two circular arc bends having a bend angle of 90° each</p> <p>Clause 15: 90° minus the angle between the line to the Sun and the normal to orbit plane</p>
γ	Ratio of specific heats in a gas
δ	<p>Clause 7: ratio of valve control travel from closed position to total valve control travel</p> <p>Clause 8: laminar sublayer thickness, [m]</p> <p>Chaps 14 and 15: film thickness, [m]</p>
δm	correction to allow for differences between dimensionless temperature and dimensionless velocity in the fully turbulent core in pipe flow, see clause 13.2.
$\delta\epsilon$	decrease in effectiveness of a heat exchanger because of longitudinal wall conduction

δ^*	dimensionless film thickness, $\delta^* = \delta u^*/\nu$
ε	Clauses 5, 11 and 13: exchanger heat transfer effectiveness Clause 9: radial gap between the twisted-tape insert used for enhancing the heat transfer and the tube wall, [m] Clause 15: hemispherical total emittance Clause 15: dielectric permittivity, [F.m ⁻¹]
ε_{cp}	cold plate effectiveness
ε_0	dielectric permittivity of vacuum, $\varepsilon_0 = 8,8542 \times 10^{-12}$ F.m ⁻¹
ε_t	overall heat transfer effectiveness of an association of heat exchangers
η	heat transfer effectiveness
η_f	fin effectiveness
η_p	overall efficiency of a pump
θ	Clause 6: yaw angle, [angular degrees] Clause 7: total angle turned through by a bend, [angular degrees]
λ	friction factor
λ_B	baker parameter
μ	fluid dynamic viscosity, [Pa.s] it is also known as viscosity coefficient
ν	fluid kinematic viscosity, [m ² .s ⁻¹]
ρ	fluid density, [kg.m ⁻³]
σ	surface tension, [N.m ⁻¹] Clause 15: Stefan-Boltzmann constant, $\sigma = 5,6697 \times 10^{-8}$ W.m ⁻² .K ⁻⁴
τ	shearing stress, [Pa]
ϕ	Clause 11: flow maldistribution parameter, it is defined, although not in a unique fashion, in terms of the mass flow rate Clause 14: polar angle of stratified flow interface, [angular degrees]
ψ_B	baker parameter
ω	rotating speed, [radians per second]

Subscripts

F	liquid film, annular flow with entrainment
G	Clause 7: gross value Clause 14: gas
Gi	gas at the interface
H	affected by hall's transformation
High	refers to channels with higher-than average flow on the nonuniform side of a split counterflow heat exchanger (fleming's model)
L	Clause 7: refers to length L Clause 11: refers to liquid stream in liquid-coupled heat exchanger Clause 14: liquid
Low	refers to channels with lower-than average flow on the nonuniform side of a split counterflow heat exchanger (Fleming's model)
M	phase-change
N	net value
P	evaluated at constant pumping power
R	radiator
S	refers to heat source
SG	gas flowing alone in the duct
SL	liquid flowing alone in the duct
T	refers to total (axial plus spiral) values in spiralling flow
TP	two phase in the mixture theory
a	heat transfer enhanced date Clause 15: Earth Albedo.
am	arithmetic mean value
av	average value
b	fluid bulk properties
c	Clauses 5, 11, 13 and 15: cold stream Clause 7: cylindrical tubes

	Clause 14: core, annular flow with entrainment
d	downstream tangent of a bend
e	spatial and temporal mean conditions at the tip level of the roughness elements Clause 15: space environment condition
equiv	equivalent
f	refers to fin or twisted-tape
f	hot stream
i	Clauses 5, 6 and 15: inlet conditions Clause 14: interface
im	inlet mean value
ir	infrared
lm	logarithmic mean value
m	mean value between commencement of heat transfer and station x
max	maximum value
n	refers to the nth component of an ensemble
o	outlet conditions Clause 9: heat transfer nonenhanced data
opt	optimum conditions
r	reference value for given thermal and/or geometrical characteristics of a system, it is omitted when no correction is made
s	refers to static pressure Clause 15: solar
t	refers to total pressure
tt	turbulent liquid, turbulent gas
tv	turbulent liquid, laminar gas
u	upstream tangent of a bend
vt	laminar liquid, turbulent gas
vv	laminar liquid, laminar gas
w	wall conditions

x	local conditions at section x of a duct
δ	conditions at the edge of the laminar sublayer
0	refers to an orifice plate of zero relative thickness
0,8	refers to an orifice plate if relative thickness $t/d = 0,8$
1	Clauses 5, 11 and 13: conditions for the stream with smaller capacity rate Clauses 6 and 9: inner wall conditions in a cylindrical annulus Clause 7: upstream reference conditions
2	Clauses 6 and 9: outer wall conditions in a cylindrical annulus Clause 7: downstream reference conditions Clause 11 conditions for the stream with larger capacity rate

Superscripts

-	mean value
*	entrainment effects included

General introduction

Fluids can be used to protect from thermal loads sensitive components of spacecraft systems.

This protection can be achieved on the basis of two completely different procedures.

1) The fluid is at rest although changing its phase, storing heat through melting (or evaporation) when the temperature increases, and restoring it through freezing (or condensation), for dissipation at a later time, when the temperature decreases. This is typical of phase change capacitors whose main features are discussed in [ECSS-E-HB-31-01 Part 10](#).

Single phase fluids at rest are not used for thermal control purposes because of the far better qualities of solid materials, either as thermal insulators or as thermal conductors.

2) The fluid is in motion, absorbing the heat at a relatively steady rate from the component whose temperature is controlled, transferring it to a heat sink which can be placed fairly apart from the source.

Heat transfer can be achieved either through sensible heat change or through phase change (latent heat):

2.1) Sensible heat change. A liquid or gas phase is used to transfer heat according to the equation $Q = mc_p \Delta T$. In order to increase the heat transfer rate, Q , either the mass flow rate, m , or the temperature difference, ΔT , is increased for a particular fluid. The specific heat, c_p , is more or less the same for all liquids and gases at normal conditions.

2.2) Phase change. The heat transferred is used to evaporate the working liquid according to the equation $Q = mh_{fg}$, where h_{fg} is the latent heat of vaporization. Now the heat transfer is much higher than in 2.1 for the same mass flow rate and the same type of fluid, whereas the temperature gradient along the duct is much smaller (system isothermality which allows the maintenance of isothermal heat load interfaces).

Forcing of the fluid through the duct can be performed by use either of:

2.3) A pumping device, centrifugal or positive displacement pump, normally driven by an electric motor, as in the case of both single and two phase fluid loops which are the subject of this Part.

2.4) Capillary forces, as in Heat Pipes. Such devices are discussed extensively in [ECSS-E-HB-31-01 Part 8](#).

A capillary pumped loop (CPL) is a two-phase heat transfer system which consists of multiple parallel evaporators and where the pumping function is achieved through a fine pore size capillary structure. Unlike conventional evaporators where heat is transferred through a liquid layer to the evaporating surface, in the CPL evaporator the heat is transferred directly to the vaporization surface. These devices present the same boiling, priming and capillary pressure limits associated with heat pipes (see [ECSS-E-HB-31-01 Part 8, clause 5.2](#)).

2.5) Short-term discharge of a stored fluid. This may be accomplished by use of either of (see [ECSS-E-HB-31-01 Part 14, clause 5.3](#)):

2.5.1) Sudden expansion of a gas stored in a high-pressure bottle, with the associated decrease in total temperature (Joule-Thomson process).)

2.5.2) Boiling of liquids such as water or cryogenic fluids. These liquids are selected on the basis of their high latent heats and (or) their low boiling points.

2.5.3) Sublimation of solids.

Because the large volume ratio associated with these open cycle processes, the fluid is vented out of the vehicle once it has absorbed the heat.

4.1 Fluid loops

The aim of the fluid loop, as of any other thermal control system, is to keep the temperature of a given component within the range which secures its correct functioning. To this goal, the heat flow rate evacuated should be equal to the heat rate dissipated by the component plus that coming from external sources.

According to their working mode, fluid loops may be either considered as:

1. Thermal insulators, the aim of which is to protect the component against a thermally hostile environment, or
2. Thermal acceptors transferring the exceeding thermal energy from the component to the thermal sink.

The coolant may circulate through a single-walled chamber enclosing the component, or through a double-walled component heat exchanger package. The single-walled jacket affords more efficient heat transfer than the heat exchanger, but the fluid can contaminate, corrode or chemically react with the materials of the component which is being thermally controlled.

Dual-loop configurations, consisting of two circulating coolant loops coupled by an interloop heat exchanger, are used very often. A dual-loop configuration offers a more precise temperature control than could a single-loop; in addition, it provides more flexibility to accommodate changes in the heat dissipating elements, spacecraft thermal loads, and coolant inlet temperature requirements.

The heat rejection depends on whether the coolant is expendable or non-expendable. An expendable coolant is rejected from the vehicle once it has accomplished its mission, while a non-expendable coolant is recirculated again after losing its thermal energy excess; in the last case the excess heat is radiated to the space via a radiator.

Single-phase fluid loops incorporating space radiators are fairly simple but massive. Two-phase systems are less massive and more flexible in the accommodation of system growth or changing heat loads, but require careful design to work properly under reduced gravity conditions (phase separation, good heat transfer, appropriate condensation rates, ...). Mass saving of two-phase systems are confined to pumps, fluid inventory and plumbing. Pipe diameters are also smaller than those corresponding to a single-phase liquid line, providing the possibility of retrofitting an existing single phase system to two-phase operation (Sadunas, Lehtinen & Parish (1985) [153]).

Major disadvantages of the two-phase pumped loop system are the lack of long-life qualified pumps and potentially complex valving, metering and control algorithms (Wise (1986) [192]).

When cryogenic cooling below 70 K is required, the direct radiation to the space is hardly possible, then the fluid loop incorporates a refrigerating system (Brayton, Stirling, or Vuilleumier cycles) which compresses the gas at ambient temperature and then expands it at a lower temperature. During the

expansion, heat is added to the gas providing the required cooling. This heat is radiated to the space. Several refrigerators used onboard satellites are described in [ECSS-E-HB-31-01 Part 14, clause 5.2](#).

4.2 Comparison between fluid loops and alternative systems

In order to define the field of application of fluid loops it should be convenient to compare them briefly with alternative insulating or refrigerating systems.

4.2.1 Passive thermal insulations

Fairly sophisticated, highly reliable, and extremely efficient passive thermal insulations are available for use in spacecraft. Data in [ECSS-E-HB-31-01 Part 7](#) are referred to for details on the subject. Although these systems can be used in conjunction with component precooling, they cannot insulate for long times components having low thermal capacity, unless requiring prohibitively thick insulations. Among these low-thermal- capacity components, mention should be made to heavy electronic devices, such as batteries and tape recorders, which have a specific heat of the order of $c = 850 \text{ J.kg}^{-1}.\text{K}^{-1}$, common to many electronic systems (Werth et al. (1966) [189]. In addition, thermal insulations are not appropriate for controlling components which operate in repeating on-off cycles, since in these cases it is required both to keep off the heat flowing outside and to evacuate the heat dissipated from inside.

In addition to being well suited to thermal protection of electronic and similar components, fluid loops afford a very precise control of the thermal energy being evacuated, and the flexibility to be adapted to a previously defined geometrical configuration.

4.2.2 Thermoelectric devices

Thermoelectric devices (TEDs), also called Peltier-effect devices, are low-voltage, high-current active thermal control modules that provide heating or cooling with the reliability associated with a purely electrical device. The relatively low performance efficiency of TEDs in conjunction with its high mass have prevented previous spacecraft application as a prime active thermal control system.

The comparison of TEDs with fluid loops within the normal range of application of these last devices makes little sense, the fluid loops being far superior. Nevertheless TEDs present several features which fluid loops cannot match. A few typical values could substantiate these assertions.

1. A typical thermoelectric module having an area of $0,03 \text{ m} \times 0,03 \text{ m}$ pumps 5 W of heat through a 50 K temperature differential (Scott (1974), p. 216). Assuming that the thermally controlled area equals the thermoelectric pellet area, the heat flux would be $5,5 \times 10^3 \text{ W.m}^{-2}$. This figure looks, however, very biased in favor of TEDs since several modules are placed a bit apart from each other on the surface to be controlled; a value of 110 W.m^{-2} is quoted as typical by Chapter & Johnsen (1973) [24]. On the other side, the laminar single-phase flow of water through a tube of cylindrical cross section, with an internal diameter of $5 \times 10^{-3} \text{ m}$ could remove $30 \times 10^3 \text{ W.m}^{-2}$ from the tube wall whose temperature is assumed to be 50 K above that of water, and this figure can be improved with turbulent single- phase flow and with two-phase flow.
2. A useful measure of the efficiency of a TED module is the "coefficient of performance", ratio of the heat pumped to the power that is supplied to operate the module. This