Air-Cooled Heat Exchangers for General Refinery Service

API Standard 661, Fifth Edition
March 2002

ISO 13706: 2000, Petroleum and Natural Gas Industries—Air-cooled Heat Exchangers

American Petroleum Institute

Helping You Get The Job Done Right.SM

SPECIAL NOTES

API publications necessarily address problems of a general nature. With respect to particular circumstances, local, state, and federal laws and regulations should be reviewed.

API is not undertaking to meet the duties of employers, manufacturers, or suppliers to warn and properly train and equip their employees, and others exposed, concerning health and safety risks and precautions, nor undertaking their obligations under local, state, or federal laws.

Information concerning safety and health risks and proper precautions with respect to particular materials and conditions should be obtained from the employer, the manufacturer or supplier of that material, or the material safety data sheet.

Nothing contained in any API publication is to be construed as granting any right, by implication or otherwise, for the manufacture, sale, or use of any method, apparatus, or product covered by letters patent. Neither should anything contained in the publication be construed as insuring anyone against liability for infringement of letters patent.

Generally, API standards are reviewed and revised, reaffirmed, or withdrawn at least every five years. Sometimes a one-time extension of up to two years will be added to this review cycle. This publication will no longer be in effect five years after its publication date as an operative API standard or, where an extension has been granted, upon republication. Status of the publication can be ascertained from the API Downstream Segment [telephone (202) 682-8000]. A catalog of API publications and materials is published annually and updated quarterly by API, 1220 L Street, N.W., Washington, D.C. 20005.

This document was produced under API standardization procedures that ensure appropriate notification and participation in the developmental process and is designated as an API standard. Questions concerning the interpretation of the content of this standard or comments and questions concerning the procedures under which this standard was developed should be directed in writing to the director/general manager of the Downstream Segment, American Petroleum Institute, 1220 L Street, N.W., Washington, D.C. 20005. Requests for permission to reproduce or translate all or any part of the material published herein should also be addressed to the director.

API standards are published to facilitate the broad availability of proven, sound engineering and operating practices. These standards are not intended to obviate the need for applying sound engineering judgment regarding when and where these standards should be utilized. The formulation and publication of API standards is not intended in any way to inhibit anyone from using any other practices.

Any manufacturer marking equipment or materials in conformance with the marking requirements of an API standard is solely responsible for complying with all the applicable requirements of that standard. API does not represent, warrant, or guarantee that such products do in fact conform to the applicable API standard.

API FOREWORD

This standard is under the jurisdiction of the API Committee on Refinery Equipment. This API standard is identical with the English version of ISO 13706. ISO 13706 was prepared by Technical Committee ISO/TC 67, *Materials, equipment and offshore structures for petroleum and natural gas industries*, SC 6, *Processing equipment and systems*.

This standard shall become effective on the date printed on the cover but may be used voluntarily from the date of distribution.

API publications may be used by anyone desiring to do so. Every effort has been made by the Institute to assure the accuracy and reliability of the data contained in them; however, the Institute makes no representation, warranty, or guarantee in connection with this publication and hereby expressly disclaims any liability or responsibility for loss or damage resulting from its use or for the violation of any federal, state, or municipal regulation with which this publication may conflict.

Suggested revisions are invited and should be submitted to Downstream Segment, API, 1220 L Street, NW, Washington, DC 20005.

ISO FOREWORD

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 3.

Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

Attention is drawn to the possibility that some of the elements of this International Standard may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

International Standard ISO 13706 was prepared by Technical Committee ISO/TC 67, *Materials, equipment and offshore structures for petroleum and natural gas industries*, Subcommittee SC 6, *Processing equipment and systems*.

Annexes A, B and C of this International Standard are for information only.

Contents

Page

Introduction

This International Standard is based on API standard 661, fourth edition, November 1997.

Users of this International Standard should be aware that further or differing requirements may be needed for individual applications. This International Standard is not intended to inhibit a vendor from offering, or the purchaser from accepting, alternative equipment or engineering solutions for the individual application. This may be particularly applicable where there is innovative or developing technology. Where an alternative is offered, the vendor should identify any variations from this International Standard and provide details.

Petroleum and natural gas industries — Air-cooled heat exchangers

1 Scope

This International Standard gives requirements and recommendations for the design, materials, fabrication, inspection, testing and preparation for shipment of air-cooled heat exchangers for use in the petroleum and natural gas industries.

This International Standard is applicable to air-cooled heat exchangers with horizontal bundles, but the basic concepts may also be applied to other configurations.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this International Standard. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this International Standard are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

ISO 76, *Rolling bearings — Static load ratings.*

ISO 281, *Rolling bearings — Dynamic load ratings and rating life.*

ISO 286 (all parts), *ISO system of limits and fits.*

ISO 1081, *Belt drive — V-belts and V-ribbed belts, and corresponding grooved pulleys — Vocabulary.*

ISO 1459, *Metallic coatings — Protection against corrosion by hot-dip galvanizing — Guiding principles.*

ISO 1461, *Hot-dip galvanized coatings on fabricated iron and steel articles — Specifications and test methods.*

ISO 2491, *Thin parallel keys and their corresponding keyways (dimensions in millimetres).*

ISO 3744, *Acoustics — Determination of sound power levels of noise sources using sound pressure — Engineering method in an essentially free field over a reflecting plane.*

ISO 4183, *Belt drives — Classical and narrow V-belts — Grooved pulleys (system based on datum width).*

ISO 4184, *Belt drives — Classical and narrow V-belts — Lengths in datum system.*

ISO 5287, *Narrow V-belt drives for the automotive industry — Fatigue test.*

ISO 5290, *Belt drives — Grooved pulleys for joined narrow V-belts — Groove sections 9J, 15J, 20J and 25J (effective system).*

ISO 8501-1, *Preparation of steel substrates before application of paints and related products — Visual assessment of surface cleanliness — Part 1: Rust grades and preparation grades of uncoated steel substrates and of steel substrates after overall removal of previous coatings.*

ISO 9563, *Belt drives — Electrical conductivity of antistatic endless synchronous belts — Characteristics and test method.*

ISO 10436, *Petroleum and natural gas industries — General-purpose steam turbines for refinery service.*

AGMA 6001[1)], *Design and selection of components for enclosed gear drives.*

AGMA 6010-E, *Practice for enclosed speed reducers or increasers using spur, helical, herringbone and spiral bevel gears*.

ICBO[2)], *Uniform Building Code.*

3 Terms and definitions

For the purposes of this International Standard, the following terms and definitions apply.

3.1
bank
one or more items arranged in a continuous structure

3.2
bare tube surface
total area of the outside surfaces of the tubes, based on the length measured between the outside faces of the header tubesheets

3.3
bay
one or more tube bundles, serviced by two or more fans, including the structure, plenum and other attendant equipment

NOTE Figure 1 shows typical bay arrangements.

3.4
finned surface
<of a tube> total area of the outside surface exposed to air

3.5
forced-draught exchanger
exchanger designed with the tube bundles located on the discharge side of the fan

3.6
induced-draught exchanger
exchanger designed with the tube bundles located on the suction side of the fan

3.7
item
one or more tube bundles for an individual service

1) American Gear Manufacturers' Association, 1500 King Street, Suite 201, Alexandria, VA 22314, USA.

2) International Conference of Building Officials, 5360 South Workman Mill Road, Whittier, CA 90601, USA.

3.8
item number
purchaser's identification number for an item

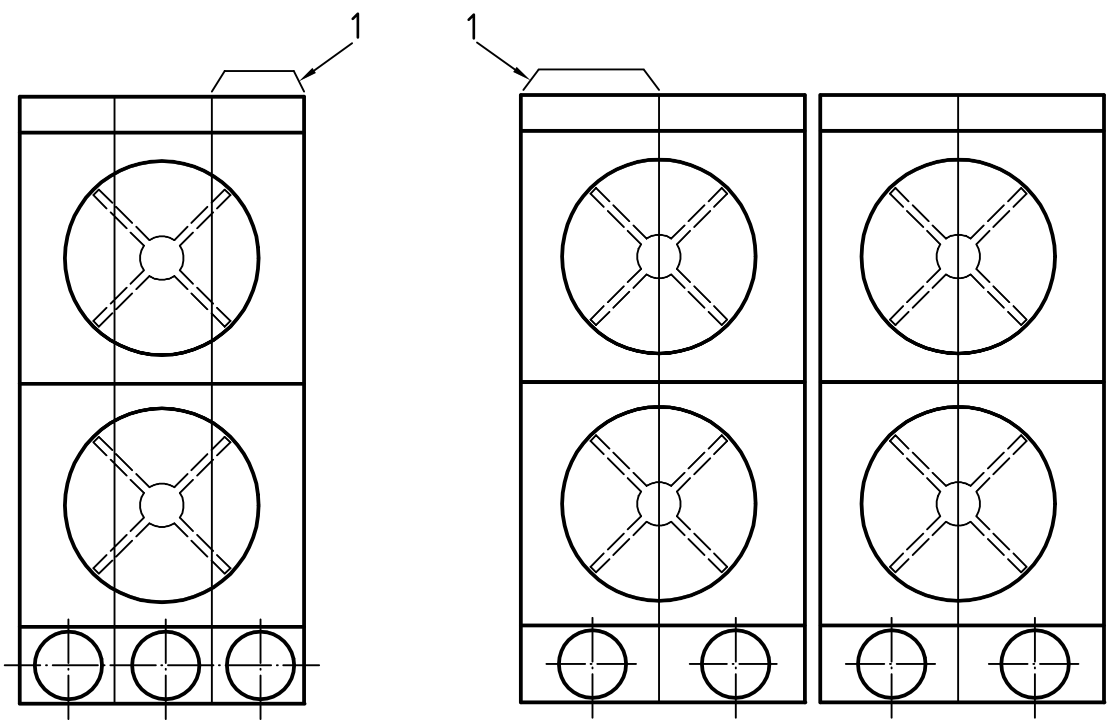
3.9
pressure design code
recognized pressure vessel standard specified or agreed by the purchaser

EXAMPLE ASME VIII.

3.10
structural code
recognized structural standard specified or agreed by the purchaser

EXAMPLES AISC M011 and AISC S302.

3.11
tube bundle
assembly of headers, tubes and frames



a) One-bay **b) Two-bay**

Key
1 Tube bundle

Figure 1 — Typical bay arrangements

4 General

● **4.1** The pressure design code shall be specified or agreed by the purchaser.

Pressure components shall comply with the pressure design code and the supplemental requirements given in this International Standard.

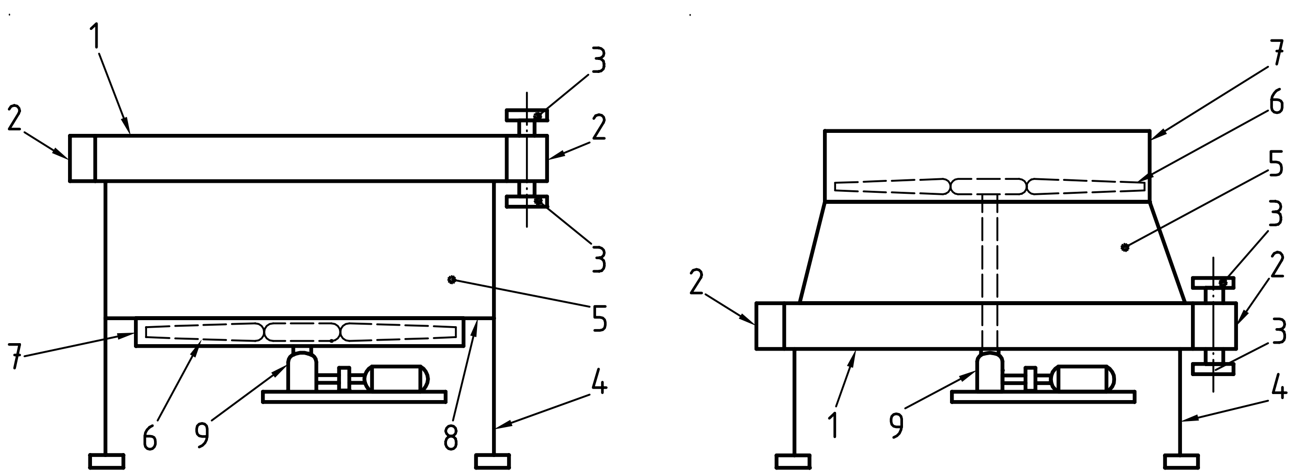
NOTE A round bullet (●) at the beginning of a subclause indicates a requirement for the purchaser to make a decision or provide information (see checklist in annex B). A triangular bullet (▲) at the beginning of a subclause indicates that this detail is included on the air-cooled heat exchanger data sheet (see annex B).

4.2 The air-cooled heat exchanger shall be either a forced-draught exchanger or an induced-draught exchanger and shall include the components shown in Figure 2 and any auxiliaries such as ladders, walkways and platforms.

4.3 Annex A, which may be consulted if required, includes for information some recommended mechanical and design details. Annex A also includes precautions for consideration when specifying certain design aspects, including temperature limitations, type of extended surface, tube support methods, type of air-cooled heat exchanger, materials of gasket construction and operational considerations such as walkway access.

● **4.4** The vendor shall comply with the applicable local regulations specified by the purchaser.

4.5 In this International Standard, where practical, U.S. Customary units are included in brackets for information.



a) **Forced draught** b) **Induced draught**

Key

1 Tube bundle
2 Header
3 Nozzle
4 Supporting column
5 Plenum
6 Fan
7 Fan ring
8 Fan deck
9 Drive assembly

Figure 2 — Typical components of an air-cooled heat exchanger

5 Proposals

5.1 The vendor's proposal shall include a completed data sheet for each item (see annex B).

5.2 A proposal drawing shall be furnished which shows the major dimensions in plan and elevation, and the nozzle sizes and their orientation.

5.3 The proposal shall state whether vertically mounted electric motors shall be shaft up or shaft down.

5.4 The fabrication procedure and welding procedure shall be furnished for welded tube-to-tubesheet joints.

5.5 The proposal shall fully define the extent of shop assembly and include a general description of the components to be assembled in the field.

5.6 Any proposal for a design that is not fully described in this International Standard shall include additional drawings sufficient to describe the details of construction.

5.7 The proposal shall include a detailed description of any exceptions to the specified requirements.

● **5.8** The proposal shall include noise data. The proposal shall include a noise data sheet (see annex B) if specified by the purchaser.

● **5.9** The proposal shall include fan performance characteristic curves if specified by the purchaser.

5.10 The proposal shall include details of the method used to secure the fin ends (7.1.11.7).

5.11 The vendor shall inform the purchaser if the vendor considers that the requirements specified by the purchaser are in conflict with, or are not suitable for, the intended purposes or operation of the unit.

6 Documentation

6.1 Approval information

● **6.1.1** For each item number, the vendor shall produce documents which include the following information. The purchaser shall specify which documents shall be submitted and which of them shall be subject to approval.

a) The purchaser's item number, the service, the project name and location, the purchaser's order number and the vendor's shop order number;

b) design pressure, maximum allowable working pressure, test pressure, maximum and minimum design temperature, and corrosion allowance;

c) any applicable codes and purchase specifications of the purchaser;

d) material specifications and grades for all pressure parts;

e) overall dimensions;

f) dimensions and locations of supports and sizes of holding-down bolts;

g) nozzle size, rating, facing, location, projection beyond header surface, allowable loadings (forces and moments) and direction of flow;

h) drive mount details;

i) masses of the tube bundle, the exchanger empty and full of water, and the mass of the heaviest component or combination of components intended by the vendor to be handled in a single lift;

j) column reactions for each load type listed in 7.3.3;

k) post-weld heat treatment requirements;

l) radiographic and other non-destructive examination requirements;

m) surface preparation and painting requirements;

n) design exposure temperatures for mechanical and instrumentation components;

o) nameplate and its position;

p) tube-to-tubesheet joint and details of joint preparation.

6.1.2 The vendor shall also furnish gasket detail drawings, field assembly drawings, and drawings for all auxiliary equipment and controls furnished. Drawings shall show electrical and control connections, including those of motive and signal air for any pneumatically actuated louvres or fans. The gasket details shall include type and material, and shall be shown on a separate drawing.

● **6.1.3** Calculations required by the pressure design code shall be made for the design of pressure components, including header boxes, tubes and tube joints. Additionally, sufficient detail shall be supplied for any non-standard pressure boundary components, such as swage-type transition nozzles. If specified by the purchaser, the calculations shall be submitted for approval.

● **6.1.4** If specified by the purchaser, weld maps, all proposed welding procedures and qualifications (including impact test results, if applicable) shall be submitted for approval prior to fabrication.

● **6.1.5** Further engineering information required from the vendor for installation, operation, maintenance, or inspection shall be a matter of agreement between the purchaser and the vendor.

6.2 Final records

6.2.1 The vendor shall maintain records of the materials used and fabrication details for at least 5 years.

● **6.2.2** The purchaser shall specify which of the following shall be furnished, and shall specify if any of them shall be in an electronic medium:

a) an “as-built” data sheet, including material specifications and grades for all pressure parts;

b) a manufacturer's data report in accordance with the pressure design code;

c) certified material test reports for all pressure parts;

d) fan and hub data, including shaft bore and keyway dimensions and coupling and sheave data;

e) a schematic diagram for automatically controlled fan pitch or louvre blade adjustment, if the controller is furnished by the vendor;

f) installation, operation and maintenance instructions, including the type of lubrication furnished for gears and bearings;

g) parts list;

h) a certified noise data sheet for the air-cooled heat exchanger with the fans operating at rated speed and at design conditions;

i) fan performance characteristic curves showing the operating point and shaft power consumption;

j) louvre characteristic performance curve;

k) temperature recorder charts made during postweld heat treatment of the headers.

7 Design

7.1 Tube bundle design

7.1.1 General

7.1.1.1 Tube bundles shall be rigid, self-contained, and designed for handling as a complete assembly.

7.1.1.2 The vendor shall make provision for lateral movement of exchanger tube bundles of at least 6 mm ($^1/_4$ inch) in both directions or at least 12 mm ($^1/_2$ inch) in only one direction, unless the purchaser and the vendor agree on a greater movement.

7.1.1.3 Provision shall be made to accommodate thermal expansion of tubes.

7.1.1.4 All tubes shall be supported to prevent sagging and meshing or deformation of fins. Tube supports shall be spaced not more than 1,8 m (6 ft) from centre to centre.

7.1.1.5 A hold-down member (tube keeper) shall be provided at each tube support. Hold-down members shall be attached to side frames by bolting.

7.1.1.6 Tubes of single-pass condensers shall be sloped downward at least 10 mm per metre ($^1/_8$ inch per foot) towards the outlet header.

7.1.1.7 Tubes of multipass condensers need not be sloped.

7.1.1.8 Air seals shall be provided throughout the tube bundle and the bay to minimize air leakage and bypassing. Any air gap that exceeds 10 mm ($^3/_8$ inch) in width shall be sealed.

7.1.1.9 The minimum thickness of metal used for air seal construction shall be 2,5 mm (12 gauge USS, 0,105 inch) within the bundle side frame and 2,0 mm (14 gauge USS, 0,075 inch) outside the bundle side frame.

7.1.1.10 Bolts for removable air seals shall be at least 10 mm ($^3/_8$ inch) nominal diameter.

● **7.1.1.11** Winterization shall be as specified or agreed by the purchaser. Annex C should be used.

● **7.1.1.12** The exchanger shall be designed for an internal steam-out operation at the temperature, pressure, and operating conditions specified by the purchaser.

7.1.2 Heating coils

7.1.2.1 Heating coils provided to protect the tube bundle against freeze-up shall be in a separate bundle, and not part of the tube bundle.

7.1.2.2 Heating coils shall cover the full width of the tube bundle.

7.1.2.3 The tube pitch of the heating coil shall not exceed twice the tube pitch of the tube bundle.

7.1.2.4 If steam is used as heating fluid, heating coils shall be single pass, and the tubes shall be sloped downward at least 10 mm per metre ($^1/_8$ inch per foot) towards the outlet.

7.1.2.5 Pipe-type headers with welded-in tubes may be used for steam service.

7.1.3 Tube bundle design temperature

● **7.1.3.1** The maximum and minimum design temperatures for pressure parts shall be as specified by the purchaser or, if not specified by the purchaser, the maximum design temperature shall be at least the specified process fluid inlet temperature plus 25°C (50°F).

● **7.1.3.2** The purchaser shall separately specify the maximum operating temperature to be applied for fin type selection (the fin design temperature). The design temperatures for pressure parts are not intended to govern fin type selection or to apply in determining exposure temperatures of mechanical and instrumentation components.

7.1.4 Tube bundle design pressure

● The design pressure shall be as specified by the purchaser or, if not specified, shall be the greater of the following:

a) the inlet pressure plus 10 %;

b) the inlet pressure plus 170 kPa (25 psi).

7.1.5 Corrosion allowance

▲ **7.1.5.1** The corrosion allowance shall be as specified by the purchaser for all surfaces exposed to the process fluid, except that no corrosion allowance shall be provided for tubes, gaskets or gasket contact surfaces. If not specified, a minimum corrosion allowance of 3 mm ($^1/_8$ inch) shall be provided for carbon and low-alloy steel components.

7.1.5.2 The corrosion allowance shall be provided on each side of pass partition plates or stiffeners.

7.1.5.3 A thickness equal to the depth of the pass partition groove may be considered as available corrosion allowance on grooved cover plate and tubesheet surfaces.

7.1.6 Headers

7.1.6.1 General

● **7.1.6.1.1** Headers shall be designed to prevent excessive warpage of tubesheets and/or leakage at tube joints. The analysis shall consider maximum operating temperature and maximum cooling conditions at minimum ambient air temperature. If specified by the purchaser, the analysis shall consider alternative operations such as low process flow at low ambient air temperature, freezing of fluids in tubes, steam-out, loss of fans due to power failure, and cycling conditions.

7.1.6.1.2 If the fluid temperature difference between the inlet and the outlet of a multi-pass bundle exceeds 110 °C (200 °F), U-tube construction, split headers or other methods of restraint relief shall be employed.

7.1.6.1.3 The need for restraint relief in single- or multi-pass bundles shall be investigated regardless of the fluid temperature difference between the inlet and outlet of the bundle. The designer shall provide calculations to prove the adequacy of the design. Calculations shall consider the following stress combinations:

a) For tube stress and/or tube joint stress:

   1) stress caused by pressure and temperature;

   2) stress caused by nozzle forces and moments;

   3) stress caused by differential tube expansion (including that caused by waxing or fouling) between rows/passes in the coil sections;

   4) stress caused by lateral header movement.

   Some of the above stresses are additive, and tube joint efficiency shall be considered.

b) For header and nozzle stress:

   1) stress caused by temperature and pressure;

2) stress caused by nozzle forces and moments;

3) stress caused by lateral header movement;

4) stress caused by differential tube expansion between rows/passes in the coil sections.

NOTE Set-in versus set-on nozzle attachments could greatly affect the above.

c) For header attachments and supports (including coil side frames and cooler structure):

1) stress caused by header mass and water;

2) stress caused by nozzle forces and moments;

3) stress caused by lateral header movement;

4) stress caused by tube expansion.

NOTE There may be additional loads and stresses imposed on the tube bundle which may not have been stated above (e.g. seismic).

7.1.6.1.4 Headers shall be designed so that the cross-sectional flow area of each pass is at least 100 % of the flow area in the corresponding tube pass.

7.1.6.1.5 The lateral velocity in the header shall not exceed the velocity in the nozzle. Multiple nozzles or an increased header cross-sectional area may be required.

7.1.6.1.6 The minimum nominal thickness of header components shall be as shown in Table 1.

Table 1 — Minimum nominal thickness of header components

Component	Minimum thickness	
	Carbon or low-alloy steel	High-alloy steel or other material
Tubesheet	20 mm ($^3/_4$ inch)	15 mm ($^5/_8$ inch)
Plug sheet	20 mm ($^3/_4$ inch)	15 mm ($^5/_8$ inch)
Top, bottom and end plates	12 mm ($^1/_2$ inch)	10 mm ($^3/_8$ inch)
Removable cover plates	25 mm (1 inch)	22 mm ($^7/_8$ inch)
Pass partition plates and stay plates	12 mm ($^1/_2$ inch)	6 mm ($^1/_4$ inch)

NOTE The thickness indicated for any carbon or low-alloy steel component includes a corrosion allowance of up to 3 mm ($^1/_8$ inch). The thickness indicated for any component of high-alloy steel or other material does not include a corrosion allowance. The thickness is based on an expanded tube-to-tubesheet joint with one groove.

7.1.6.1.7 Pass partitions used as stay plates for the tubesheet and plug sheet shall be made of one integral plate.

7.1.6.1.8 Header types other than those described in 7.1.6.2 or 7.1.6.3 may be proposed as an alternative design (see clause 12).

7.1.6.2 Removable cover plate and removable bonnet headers

7.1.6.2.1 The cover plate header design shall permit removal of the cover without disturbing header piping connections. Figure 3 shows typical construction of tube bundles with removable cover plate headers.

7.1.6.2.2 The bonnet header design shall permit removal of the bonnet with the minimum dismantling of header piping connections. Figure 3 shows typical construction of tube bundles with removable bonnet headers.

● **7.1.6.2.3** The use of either through-bolts or stud bolts for cover plates shall be agreed between the purchaser and the vendor. Bolted joints shall be designed with confined gaskets or unconfined full-faced gaskets.

Typical constructions are shown in Figure 4. The purchaser's enquiry shall specify the required design.

7.1.6.2.4 Gasket contact surfaces on cover plates, matching header box flanges and tubesheets shall be machined. The surface finish shall be appropriate for the type of gasket (annex A may be consulted for guidance on this).

7.1.6.2.5 Either jackscrews or a minimum clearance of 5 mm ($^{3}/_{16}$ inch) shall be provided at the cover periphery to facilitate dismantling.

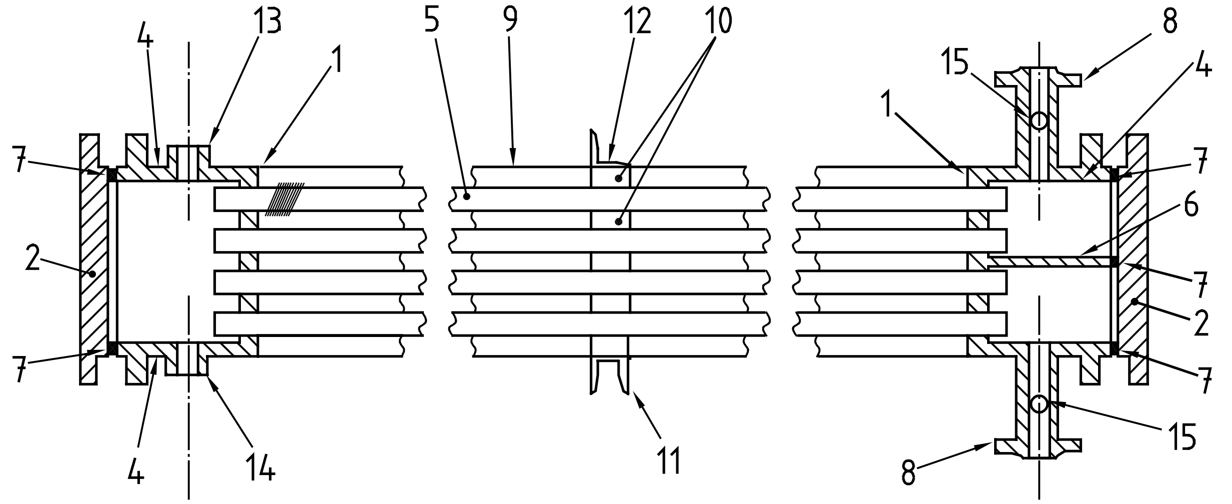
7.1.6.2.6 Stay-bolts shall not be used.

7.1.6.2.7 Provisions (e.g. sliding pins) should be made to prevent damage to the studs during handling of the cover plate.

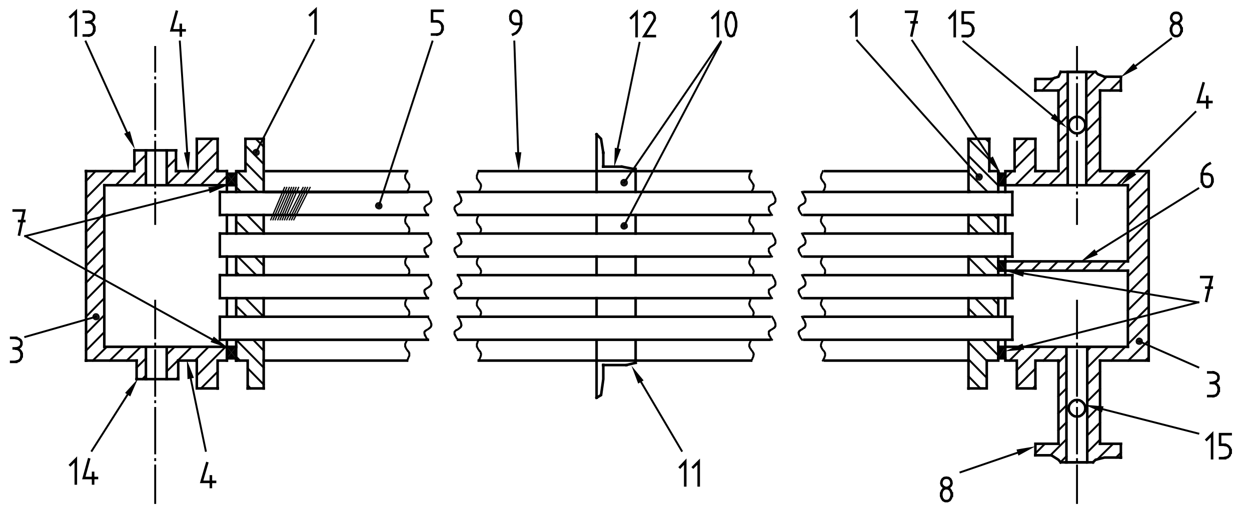
7.1.6.2.8 The minimum nominal diameter of stud bolts shall be 20 mm ($^{3}/_{4}$ inch). The minimum nominal diameter of through-bolts shall be 16 mm ($^{5}/_{8}$ inch).

7.1.6.2.9 The maximum spacing between bolt centres shall be in accordance with the pressure design code.

7.1.6.2.10 The minimum spacing between bolt centres shall be as shown in Table 2.



a) Removable cover-plate header

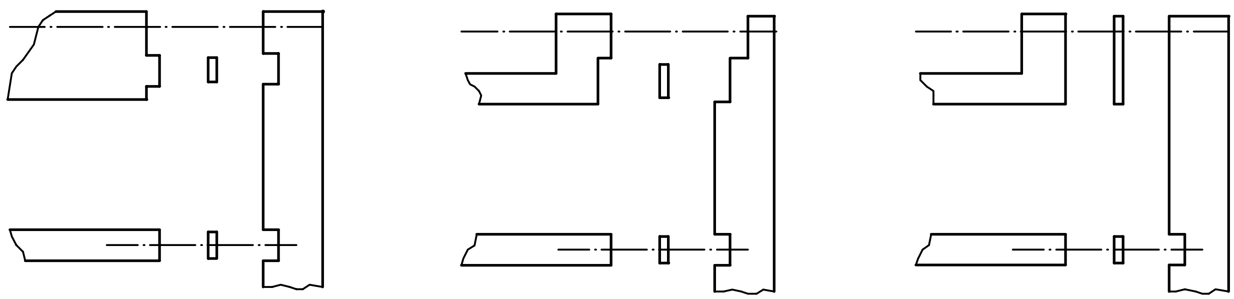


b) Removable bonnet header

Key

1 Tubesheet
2 Removable cover plate
3 Removable bonnet
4 Top and bottom plates
5 Tube
6 Pass partition
7 Gasket
8 Nozzle
9 Side frame
10 Tube spacer
11 Tube support cross-member
12 Tube keeper
13 Vent
14 Drain
15 Instrument connection

Figure 3 — Typical construction of tube bundles with removable cover plate and removable bonnet headers



a) Stud construction, confined gasket **b) Flanged construction, confined gasket** **c) Flanged construction, full-faced gasket**

Figure 4 — Typical confined and full-faced gasket joint details

Table 2 — Minimum flange bolt spacing

Nominal bolt diameter	Minimum bolt spacing
16 mm ($^{5}/_{8}$ inch)	38 mm (1 $^{1}/_{2}$ inch)
19 mm ($^{3}/_{4}$ inch)	44 mm (1 $^{3}/_{4}$ inch)
22 mm ($^{7}/_{8}$ inch)	52 mm (2 $^{1}/_{16}$ inch)
25 mm (1 inch)	57 mm (2 $^{1}/_{4}$ inch)
29 mm (1 $^{1}/_{8}$ inch)	64 mm (2 $^{1}/_{2}$ inch)
32 mm (1 $^{1}/_{4}$ inch)	71 mm (2 $^{13}/_{16}$ inch)
35 mm (1 $^{3}/_{8}$ inch)	76 mm (3 $^{1}/_{16}$ inch)
38 mm (1 $^{1}/_{2}$ inch)	83 mm (3 $^{1}/_{4}$ inch)
41 mm (1 $^{5}/_{8}$ inch)	89 mm (3 $^{1}/_{2}$ inch)
44 mm (1 $^{3}/_{4}$ inch)	95 mm (3 $^{3}/_{4}$ inch)
48 mm (1 $^{7}/_{8}$ inch)	102 mm (4 inch)
51 mm (2 inch)	108 mm (4 $^{1}/_{4}$ inch)

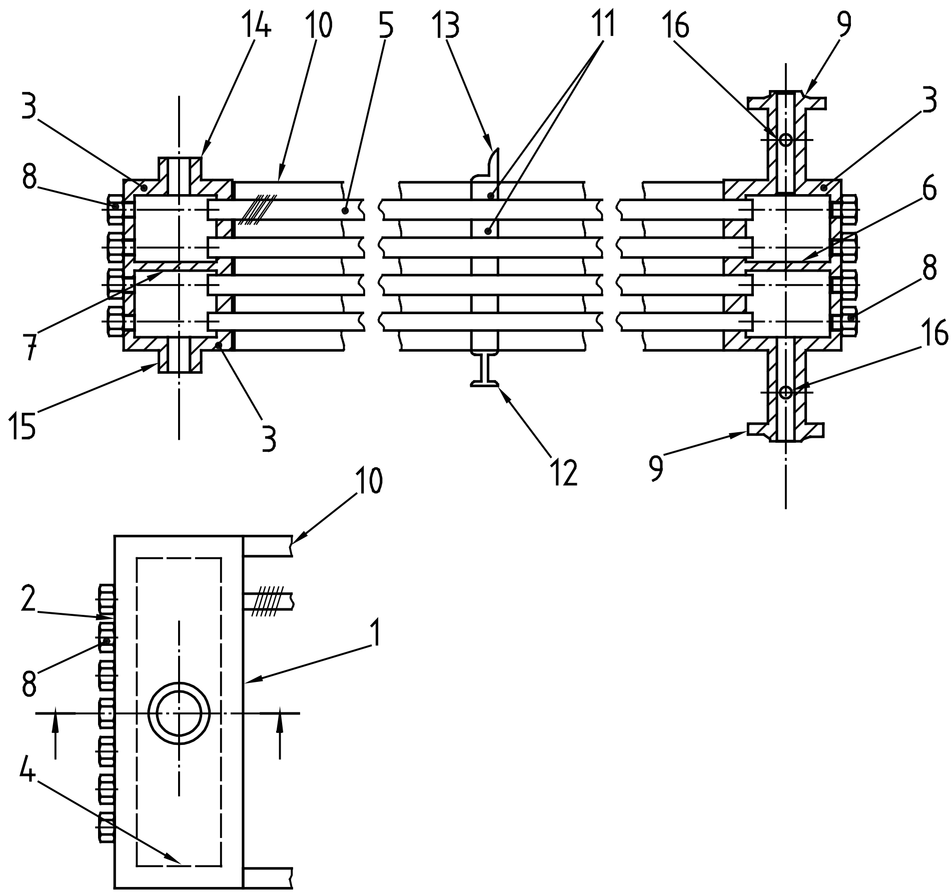
7.1.6.2.11 Spacing between bolts straddling corners shall be such that the diagonal distance between bolts adjacent to the corner does not exceed the lesser of the spacing on the sides or the ends.

7.1.6.3 Plug headers

7.1.6.3.1 Threaded plug holes shall be provided opposite the ends of each tube for access. Holes shall be threaded to the full depth of the plug sheet or 50 mm (2 inch), whichever is less. Figure 5 shows typical construction of a tube bundle with plug headers.

7.1.6.3.2 The diameter of the plug holes shall be equal to the nominal outside diameter of the tube plus at least 0,8 mm ($^{1}/_{32}$ inch).

7.1.6.3.3 Gasket contact surfaces of plug holes shall be spot-faced. The edges of the facing shall be free of burrs.



Key

1 Tubesheet
2 Plug sheet
3 Top and bottom plates
4 End plate
5 Tube
6 Pass partition
7 Stiffener
8 Plug
9 Nozzle
10 Side frame
11 Tube spacer
12 Tube support cross-member
13 Tube keeper
14 Vent
15 Drain
16 Instrument connection

Figure 5 — Typical construction of a tube bundle with plug headers

7.1.7 Plugs for tube access

7.1.7.1 Plugs shall be the shoulder type with straight-threaded shanks.

7.1.7.2 Hollowed plugs shall not be used.

7.1.7.3 Plugs shall have hexagonal heads. The minimum dimension across the flats shall be at least equal to the plug shoulder diameter.

7.1.7.4 The pressure seal shall be maintained by means of a gasket between the flange of the plug and the plug sheet.

7.1.7.5 Positive means (such as a self-centring taper) shall be provided to ensure seating of the gasket in the spot-faced recess.

7.1.7.6 Plugs shall be long enough to fill the plug sheet threads, with a tolerance of ± 1,5 mm ($^1/_{16}$ inch), except for galling materials or if the nominal plug sheet thickness is greater than 50 mm (2 inch), for which alternative designs may be used with the approval of the purchaser. Additional factors to consider in selecting the plug design are thread interference, erosion, crevice corrosion and retention of fluid in cavities.

7.1.7.7 The thickness of the plug head from its gasket surface to the top face shall be at least 50 % of the nominal tube outside diameter. Greater thickness may be required due to pressure rating and material considerations.

7.1.7.8 Threads of plugs having nominal diameters 30 mm ($1^1/_4$ inch) and smaller shall be fine series threads.

7.1.8 Gaskets

7.1.8.1 Plug gaskets shall be of the solid-metal or double-metal-jacketed, filled type, of the same general material classification as the plug.

7.1.8.2 Plug gaskets shall be flat and free of burrs.

7.1.8.3 The minimum thickness of solid metal plug gaskets shall be 1,5 mm (0,060 inch).

7.1.8.4 For joint type A in Figure 4, cover plate and bonnet gaskets shall be of the double-metal-jacketed, filled type. Filler material shall be non-asbestos and shall be suitable for sealing, exposure resistance and fire safety performance.

7.1.8.5 For joint type B in Figure 4, double-metal-jacketed, filled type gaskets or [at design pressures of 2 100 kPa gauge (300 psig) or less] compressed sheet composition gaskets suitable for the service shall be used. Gaskets shall be non-asbestos and shall be suitable for sealing, exposure resistance and fire safety performance.

7.1.8.6 For joint type C in Figure 4, compressed sheet composition gaskets suitable for the service may be used at design pressures of 2 100 kPa gauge (300 psig) or less. Gaskets shall be non-asbestos and shall be suitable for sealing, exposure resistance and fire safety performance.

7.1.8.7 The width of removable cover plate and removable bonnet gaskets shall be at least 9 mm ($^3/_8$ inch).

7.1.8.8 Gaskets shall be of one piece.

7.1.8.9 Annex A.8 may be consulted for further guidance on gaskets.

7.1.9 Nozzles and other connections

7.1.9.1 Flanges shall be in accordance with the pressure design code.

7.1.9.2 Connections of nominal pipe size DN 10 (NPS $^1/_2$), DN 32 (NPS $1^1/_4$), DN 65 (NPS $2^1/_2$) , DN 90 (NPS $3^1/_2$) or DN 125 (NPS 5) shall not be used.

7.1.9.3 Connections DN 40 (NPS $1^1/_2$) and larger shall be flanged.

7.1.9.4 In hydrogen service [i.e. if the partial pressure of hydrogen is greater than 700 kPa (100 psia)] all connections shall be flanged and slip-on flanges shall not be used.

7.1.9.5 If design conditions require the equivalent of PN 150 (ANSI 900) or higher flange ratings, all connections shall be flanged.

7.1.9.6 The minimum nozzle neck thickness, including corrosion allowance, of carbon steel and low-alloy steel flanged connections shall be as specified in Table 3.

Table 3 — Minimum nozzle neck thickness

Pipe size DN (NPS)	Minimum nozzle neck thickness mm (inch)
15 ($^1/_2$)	4,78 (0,188)
20 ($^3/_4$)	5,56 (0,219)
25 (1)	6,35 (0,250)
40 (1$^1/_2$)	7,14 (0,281)
50 (2)	8,74 (0,344)
80 (3)	11,13 (0,438)
100 (4)	13,49 (0,531)
150 (6)	10,97 (0,432)
200 (8)	12,70 (0,500)
250 (10)	15,09 (0,594)
300 (12)	17,48 (0,688)
NOTE The data in this table is taken from ASME B36.10M, using Schedule 160 for sizes up to DN 100 (NPS 4) and Schedule 80 for the larger sizes.	

● **7.1.9.7** The facing of process flanges shall be in a horizontal plane unless another arrangement is specified by the purchaser.

7.1.9.8 Flanged carbon steel connections shall be one of the following types:

a) a forged or centrifugally cast, integrally flanged welding neck;

b) a pipe welded to a forged or centrifugally cast welding neck flange;

c) a seamless transition piece attached to a forged or centrifugally cast welding neck flange;

● d) a cast or fabricated transition, if allowed by the purchaser;

e) a pipe or a transition welded to a forged slip-on flange.

7.1.9.9 If a transition is used, stay bars, greater header thickness or greater nozzle thickness may be required to provide adequate mechanical strength.

7.1.9.10 Except in hydrogen service (see 7.1.9.4), forged carbon steel slip-on flanges may be used on connections to headers that are limited to:

a) a maximum design pressure of 2 100 kPa gauge (300 psig);

b) a maximum design temperature of 450 °C (850 °F);

c) a maximum service corrosion allowance of 3 mm ($^1/_8$ inch).

7.1.9.11 Threaded connections shall be DN 25 (NPS 1), except that pressure gauge connections shall be DN 20 (NPS $^3/_4$).

7.1.9.12 Threaded connections shall be one of the following types and shall comply with the pressure design code:

a) forged steel full-coupling threaded one end only, with a suitable rating (e.g. ASME B16.11, class 6 000);

b) forged steel fitting with integral reinforcement;

c) tapped holes for vent and drain connections, where header plate thickness permits;

d) equivalent boss connection.

▲ **7.1.9.13** If a thermowell connection is specified, it shall be located in the nozzle unless the nozzle is smaller than DN 100 (NPS 4), in which case the connection shall be located on the header adjacent to the nozzle.

▲ **7.1.9.14** If a pressure gauge connection is specified it shall be located on the nozzle unless the nozzle is smaller than DN 80 (NPS 3), in which case the connection shall be located on the header adjacent to the nozzle.

7.1.9.15 Pipe threads shall be taper pipe threads (e.g. ASME B1.20.1) and shall comply with the pressure design code.

●▲ **7.1.9.16** The size, type and location of chemical cleaning connections shall be specified by the purchaser.

7.1.9.17 If specified, instrument connections shall be located in at least one inlet and outlet nozzle per bundle, except that none are required in intermediate nozzles of stacked bundles.

7.1.9.18 All threaded piping connections shall be closed with a round headed plug.

7.1.9.19 Flanged auxiliary connections, if any, shall be closed with blind flanges. The gasket and bolting materials shall be suitable for the specified operating conditions.

7.1.9.20 Vent and drain connections shall be provided at high and low points respectively on each header. Header nozzles installed at high and low points may serve as vents and drains. Connections serving as vents and drains shall not extend into the header beyond the inside surface.

7.1.9.21 If the header thickness will not permit minimum thread engagement of vent and drain plugs, couplings or built-up bosses shall be fitted.

7.1.9.22 Bolts between connecting nozzles of stacked tube bundles shall be removable without moving the bundles.

7.1.10 Maximum allowable moments and forces for nozzles and headers

7.1.10.1 Each nozzle, in its design corroded condition, shall be capable of withstanding the simultaneous application of the moments and forces defined in Figure 6 and Table 4.

7.1.10.2 The design of each fixed or floating header, the design of the connections of fixed headers to side frames, and the design of other support members shall ensure that the simultaneous application (sum) of all nozzle loadings on a single header will cause no damage. The components of the nozzle loadings on a single header shall not exceed the following values:

M_x 6 100 N · m (4 500 ft · lbf)

M_y 8 130 N · m (6 000 ft · lbf)

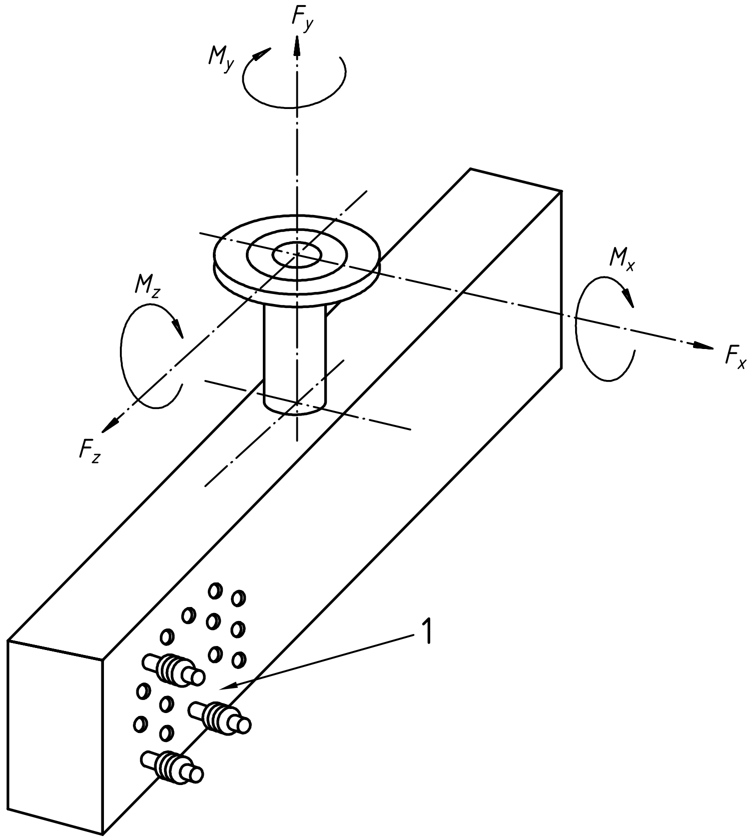
M_z 4 070 N · m (3 000 ft · lbf)

F_x 10 010 N [2 250 lbf]

F_y 20 020 N [4 500 lbf]

F_z 16 680 N [3 750 lbf]

NOTE The application of the moments and forces shown in Table 4 will cause movement that will tend to reduce the loads to the values given here.



Key

1 Fin tubes

Figure 6 — Nozzle loads

Table 4 — Maximum allowable nozzle loads

Nozzle size DN (NPS)	**Moments** N · m (ft · lbf)			**Forces** N (lbf)		
	M_x	M_y	M_z	F_x	F_y	F_z
40 ($1^1/_2$)	110 (80)	150 (110)	110 (80)	670 (150)	1 020 (230)	670 (150)
50 (2)	150 (110)	240 (180)	150 (110)	1 020 (230)	1 330 (300)	1 020 (230)
80 (3)	410 (300)	610 (450)	410 (300)	2 000 (450)	1 690 (380)	2 000 (450)
100 (4)	810 (600)	1 220 (900)	810 (600)	3 340 (750)	2 670 (600)	3 340 (750)
150 (6)	2 140 (1 580)	3 050 (2 250)	1 630 (1 200)	4 000 (900)	5 030 (1 130)	5 030 (1 130)
200 (8)	3 050 (2 250)	6 100 (4 500)	2 240 (1 650)	5 690 (1 280)	13 340 (3 000)	8 010 (1 800)
250 (10)	4 070 (3 000)	6 100 (4 500)	2 550 (1 880)	6 670 (1 500)	13 340 (3 000)	10 010 (2 250)
300 (12)	5 080 (3 750)	6 100 (4 500)	3 050 (2 250)	8 360 (1 880)	13 340 (3 000)	13 340 (3 000)
350 (14)	6 100 (4 500)	7 120 (5 250)	3 570 (2 630)	10 010 (2 250)	16 680 (3 750)	16 680 (3 750)

7.1.10.3 The total of all nozzle loads on one multi-bundle bay shall not exceed 3 times that allowed for a single header.

7.1.10.4 See 7.1.6.1.3 for further details.

7.1.11 Tubes

7.1.11.1 The outside diameter of cylindrical tubes should be at least 25 mm (1 inch).

● **7.1.11.2** The maximum tube length shall be as specified by the purchaser.

▲ **7.1.11.3** The wall thickness for tubes with an outside diameter of 25 mm (1 inch) to 38 mm ($1^1/_2$ inch) shall not be less than that specified in Table 5.

Table 5 — Minimum wall thickness of tubes

Tube material	Minimum wall thickness
Carbon steel or ferritic low-alloy steel (max. 9 % chromium)	2,0 mm (0,083 inch)
High-alloy [austenitic, ferritic and austenitic/ferritic (duplex)] steel	1,6 mm (0,065 inch)
Non-ferrous material	1,6 mm (0,065 inch)
Titanium	1,2 mm (0,049 inch)

For embedded fin tubes, this thickness shall be measured from the bottom of the groove to the inner wall.

NOTE Greater wall thickness may be required for severe services or certain tube configurations.

7.1.11.4 Tubes shall be furnished on either a minimum wall basis or an average wall basis, provided the wall thickness is nowhere less than that specified in 7.1.11.3.

▲ **7.1.11.5** Tubes may be finned or unfinned.

7.1.11.6 For a finned tube, the total unfinned length between tubesheets after assembly shall not exceed 1,5 times the thickness of one tubesheet.

7.1.11.7 Any finned tube construction shall be a matter of agreement between the purchaser and the vendor. The type of construction furnished shall be demonstrated by the vendor to be suitable for the intended service conditions (taking into account factors such as metal temperature, cycling, loss of cooling, effect of environment and any specified abnormal operating conditions). The following are descriptions of several types of commonly used finned tube construction.

a) *Embedded* — rectangular cross-section aluminium fin wrapped under tension and mechanically embedded in a groove 0,25 mm ± 0,05 mm (0,010 inch ± 0,002 inch) deep, spirally cut into the outside surface of a tube. Tube wall thickness is measured from the bottom of the groove to the inside diameter of the tube. The fin end at each end of the tube shall be secured to prevent loosening or unravelling of the fins; the vendor shall indicate the method used.

b) *Integral* — an aluminium outer tube from which fins have been formed by extrusion, mechanically bonded to an inner tube or liner.

c) *Overlapped footed* — L-shaped aluminium fin wrapped under tension over the outside surface of a tube, with the tube fully covered by the overlapped feet under and between the fins. The fin end at each end of the tube shall be secured to prevent loosening or unravelling of the fins; the vendor shall indicate the method used.

d) *Footed* — L-shaped aluminium fin wrapped under tension over the outside surface of a tube, with the tube fully covered by the feet between the fins. The fin end at each end of the tube shall be secured to prevent loosening or unravelling of the fins; the vendor shall indicate the method used.

e) *Externally bonded* — Tubes on which fins are bonded to the outside surface by hot-dip galvanizing, brazing or welding.

f) *Knurled footed* — L-shaped aluminium fin wrapped under tension over the outside surface of a tube, while the foot of the fin is simultaneously pressed into the ribbed outer surface of the tube. The fin end at each end of the tube shall be secured to prevent loosening or unravelling of the fins; the vendor shall indicate the method used.

7.1.11.8 For fins wrapped under tension or embedded, the minimum stock thickness shall be as follows:

— for fin heights not exceeding 12 mm ($^1/_2$ inch), the minimum stock thickness shall be 0,35 mm (0,014 inch);

— for fin heights exceeding 12 mm ($^1/_2$ inch), the minimum stock thickness shall be 0,40 mm (0,016 inch).

7.1.11.9 Flattening in the bend of U-tubes shall not exceed 10 % of the nominal outside diameter of the tube.

7.1.11.10 The minimum tube wall thickness at the U-bend shall be calculated as follows:

$$t_b = \frac{t}{\left[1 + \frac{OD}{4R_m}\right]}$$

where

t_b is the minimum tube wall thickness after bending;

t is the tube wall thickness before bending;

OD is the tube nominal outside diameter;

R_m is the mean radius of U-bend.

The calculated thickness, t_b, shall not be less than the specified minimum wall thickness.

7.1.11.11 If U-bends are formed from tube materials which are relatively non-work-hardening and are of suitable temper, tube-wall thinning in the bends should not exceed a nominal 17 % of the original tube wall thickness.

7.1.11.12 U-bends formed from tube materials having low ductility or materials which are susceptible to work hardening may require special consideration. If cold working induces embrittlement or susceptibility to stress corrosion in some materials or environments, then heat treatment should be considered.

● **7.1.11.13** Elliptical tubes may be used if agreed by the purchaser. A.1.4 may be consulted for further guidance on elliptical tubes.

7.2 Air-side design

7.2.1 General

● **7.2.1.1** Factors such as weather, terrain, mounting, environment and the presence of adjacent structures, buildings and equipment influence the air-side performance of an air-cooled heat exchanger. The purchaser shall supply the vendor with all such environmental data pertinent to the design of the exchanger. These factors shall be taken into account in the air-side design.

▲ **7.2.1.2** The need for air flow control shall be as defined by the purchaser on the basis of specific process operation requirements, including the effect of weather. Various methods of controlling air flow are available. The type ultimately selected depends on the degree of control required, the type of driver and transmission, the equipment arrangement, and economics. The various methods include simple on-off control, on-off step control (for multi-driver units), two-speed motor control, variable speed drivers, controllable fan pitch, manual or automatic louvres, and air recycling.

7.2.1.3 Fan selection at design conditions shall ensure that at **rated** speed the fan can provide, by an increase in blade angle, a 10 % increase in air flow with a corresponding pressure increase. Since this requirement is to prevent stall and inefficient operation of the fan, the resulting increased power requirement need not govern the driver rating.

7.2.1.4 The vendor may estimate the design exposure temperatures for mechanical components using conventional heat transfer analyses and shall submit the estimate for approval by the purchaser. Alternatively, these temperatures may be estimated by the following methods:

a) The design exposure temperature for mechanical and instrumentation components located above the tube bundle shall be equal to or greater than the following values:

   1) the maximum process inlet temperature less 60 °C (100 °F) (the maximum process inlet temperature is not the mechanical design temperature);

   2) the heating coil inlet temperature less 60 °C (100 °F) (the heating coil inlet temperature is not the heating coil mechanical design temperature);

   3) for units with exhaust air louvres, automatically controlled pitch fans, or two-speed fan motors, the inlet temperatures stated above less 30 °C (50 °F).

   In no case shall the minimum design exposure temperature be less than the design dry-bulb temperature.

b) The design exposure temperature for mechanical and instrumentation components located below the tube bundle shall be equal to or greater than the following values:

   1) the maximum process inlet temperature less 120 °C (200 °F) (the maximum process inlet temperature is not the mechanical design temperature);

   2) the heating coil inlet temperature less 120 °C (200 °F) (the heating coil inlet temperature is not the heating coil mechanical design temperature);

   3) for units with exhaust air louvres, automatically controlled pitch fans, or two-speed fan motors, the inlet temperatures stated above less 60 °C (100 °F).

   For items using air recirculation systems, design exposure temperatures for each operating mode (start-up, normal operation, shutdown, loss of power, stagnated air flow, one fan inoperable etc.) shall be examined.

7.2.2 Noise control

● **7.2.2.1** For a heat exchanger operating in the specified service with fans operating at design speed and pitch, the following noise limits shall be specified by the purchaser:

a) sound pressure level (SPL) values per fan at the location designated by the purchaser;

b) sound power level (PWL) values per fan.

NOTE A typical noise data sheet is shown in annex B.

The vendor shall submit sound power and sound pressure levels of the equipment, taking into account relevant information, e.g. inlet shape (type and dimension of bell or cone), obstructions, etc. Reference to fan includes driver, speed reducer, etc.

7.2.2.2 The order of preference for obtaining the required noise data shall be as follows:

a) actual testing of a representative bay installed either in an environment remote from other noise sources (shop or field tests) or installed in an operating plant;

b) derivation of noise data by testing similar equipment and adjusting the data for the actual equipment size and operating conditions. Both the measured data and the correction procedure shall be reported.

7.2.2.3 The procedure for determining noise levels shall be in accordance with ISO 3744, using the hemispherical method for determining sound power levels.

7.2.3 Fans and fan hubs

● **7.2.3.1** Two or more fans aligned in the direction of tube length shall be provided for each bay, except that single-fan arrangements may be used if agreed by the purchaser.

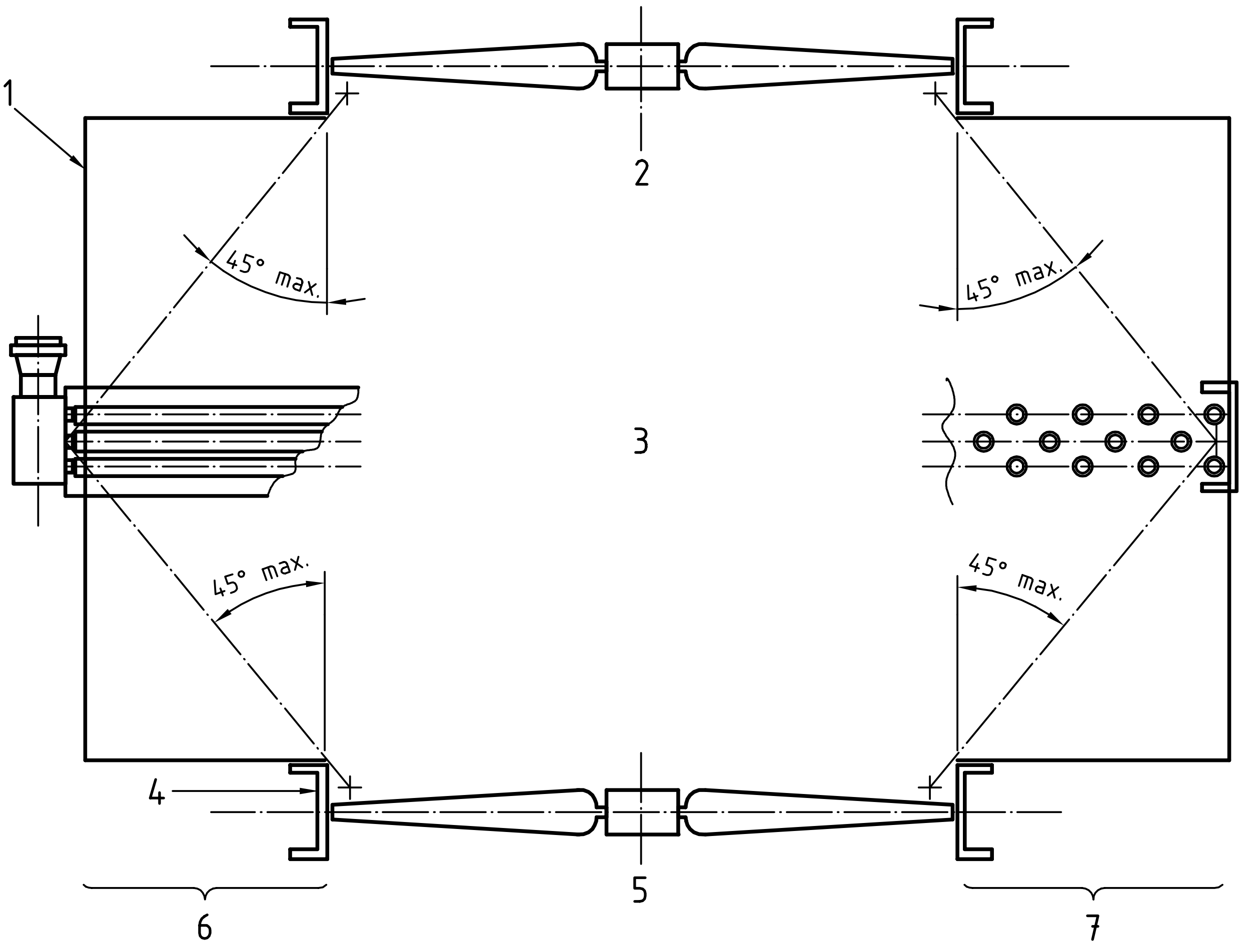
7.2.3.2 Fans shall be of the axial flow type.

7.2.3.3 Each fan shall be sized such that the area occupied by the fan is at least 40 % of the bundle face area served by that fan.

7.2.3.4 Each fan shall be located such that its dispersion angle shall not exceed 45° at the bundle centreline, as shown in Figure 7.

● **7.2.3.5** The fan tip speed shall not exceed the maximum value specified by the fan manufacturer for the selected fan type. Fan tip speed shall not exceed 60 m/s (12 000 ft/min) unless approved by the purchaser. In no case shall the fan tip speed exceed 80 m/s (16 000 ft/min). Noise limitations may require lower speeds.

7.2.3.6 The radial clearance between the fan tip and the fan orifice ring shall be as shown in Table 6.



Key

1 Plenum
2 Induced draught
3 Centreline of bundle
4 Fan ring
5 Forced draught
6 Side
7 Front

Figure 7 — Fan dispersion angle

Table 6 — Radial clearances

Fan diameter		**Radial clearance**	
m	(ft)	Minimum	Maximum
⩾ 1,0 and ⩽ 3,0	(⩾ 3 and ⩽ 9)	6 mm (1/4 inch)	13 mm (1/2 inch)
> 3,0 and ⩽ 3,5	(> 9 and ⩽ 11)	6 mm (1/4 inch)	16 mm (5/8 inch)
> 3,5	(> 11)	6 mm (1/4 inch)	19 mm (3/4 inch)

7.2.3.7 Detachable fan blades shall be moment-balanced against a master blade.

7.2.3.8 Each fan assembly shall be balanced by one of the following means:

a) dynamic balancing as an assembly;

b) dynamic balancing of the hub and static moment-balancing of the blades.

7.2.3.9 The fan assembly shall be designed to minimize reverse air flow at the hub.

▲ **7.2.3.10** For fans having a diameter larger than 1,5 m (5 ft), individual fan blades shall be manually adjustable for varying blade pitch. The use of automatic control for varying the blade pitch shall be as specified by the purchaser.

7.2.3.11 Fans equipped for pneumatically-actuated, automatically-controlled pitch adjustment of blades shall comply with the following.

a) If a single controller operates more than one actuator, the purchaser shall provide an isolating valve in the control signal line for each actuator, to allow maintenance.

b) The pneumatic actuator may be equipped with a positioner or a bias relay.

● c) If provided, the positioner or bias relay shall be designed to operate on a 20 kPa to 100 kPa gauge (3 psig to 15 psig) pneumatic control signal. Each change in the control signal shall result in a corresponding change in the fan blade pitch. The operating range of the positioner shall be adjusted so that the maximum pitch obtained is equal to the selected design blade angle setting. The fan manufacturer shall set maximum and minimum blade pitch limit stops. Unless otherwise specified by the purchaser, the minimum blade pitch limit shall result in an essentially zero air flow.

d) The vendor shall furnish a flexible tubing connection approximately 300 mm (12 inches) long for connecting to the purchaser's control-air line. The tubing shall connect to a rigid steel or alloy pipe or tube that terminates outside the fan enclosure. A terminal fitting for connection to the purchaser's control-air line shall be DN 8 (NPS $^1/_4$). Pipe threads shall be taper pipe threads.

▲ e) The purchaser shall specify the direction of change of the fan pitch with loss of control-air pressure.

7.2.3.12 Hub and fan assemblies with automatically controllable pitch adjustment employing lubricated joints shall be designed to minimize lubrication maintenance through the use of bearings not requiring periodic re-lubrication.

7.2.3.13 The fan characteristic performance curve shall relate static or total pressure, rate of flow, blade pitch and fan input shaft power, for dry-air standard conditions as stated in Table 7. The operating point and power for the specified design conditions shall be shown on the fan characteristic performance curve.

Table 7 — Dry-air standard conditions

Dry-bulb temperature	21,1 °C (70 °F)
Pressure	101,3 kPa (29,92 inches of mercury)
Density	1,2 kg/m^3 (0,075 lb/ft^3)

7.2.3.14 The natural frequency of the fan or fan components shall not be within 10 % of the blade-pass frequency. Blade-pass frequency (in passes per second) equals the number of blades multiplied by the fan speed (in revolutions per second). Slipping drive belts, low power supply voltage or variable fan-speed control operation will cause fan speeds lower than design values; if the blade pass frequency exceeds the natural frequency of the fan or component, the effect of such operation shall be evaluated.

7.2.3.15 Fan blades, hubs and blade retainers shall not be exposed to temperatures above the manufacturer's recommended operating limit, regardless of whether the fan is at rest or in operation. If this limit exceeds 105 °C (220 °F) for non-metallic components, or 145 °C (290 °F) for metallic components, the use of special materials and/or design features should be considered by the purchaser. Variable pitch fan components may further limit the exposure temperature, see 7.2.3.16.

7.2.3.16 Common elastomeric materials used in variable-pitch fan components are limited to maximum exposure temperatures as specified in Table 8.

Higher exposure temperatures require suitable materials and approval by the purchaser.

Table 8 — Maximum exposure temperature of elastomeric materials

For pneumatic diaphragm actuators	105 °C (225 °F)
For pneumatic positioners	80 °C (180 °F)
For rotary unions	120 °C (250 °F)

7.2.4 Fan shafts and bearings

7.2.4.1 Anti-friction shaft bearings shall have a calculated rating life, L_{10}, of 50 000 h at maximum load and speed in accordance with ISO 281 and/or ISO 76, where L_{10} is the number of hours at rated bearing load and speed that 90 % of a group of identical bearings will complete or exceed before the first evidence of failure.

7.2.4.2 The bearing design shall incorporate seals to prevent loss of lubricant and entry of foreign materials.

7.2.4.3 The fan shaft diameter shall suit the bearings. Bearings shall be sized in accordance with 7.2.4.1.

7.2.4.4 Fan shaft stresses shall not exceed the values given in AGMA 6001.

7.2.4.5 Fan shafts shall have key seats and fits in accordance with ISO 2491 and ISO 286 (tolerance N8) and ISO/R775.

7.2.4.6 Fan bearing exposure temperatures above 130 °C (260 °F) require one or more special features such as high-temperature seals, heat stabilization, retainers or modified internal clearances. Lubricants shall be suitable for the design exposure temperature plus any temperature due to friction and loading.

7.2.5 Lubrication facilities

Connections shall be provided outside the fan guards to permit grease lubrication of fan shaft bearings without shutdown of the equipment. Stainless steel tubing with an outside diameter of at least 6 mm ($^1/_4$ inch) shall be used for grease lines. The connections shall be accessible from grade or service platforms. The length of the grease lines should be minimized.

7.2.6 Fan guards

7.2.6.1 Removable steel fan guards shall be furnished on forced-draught exchangers.

7.2.6.2 The materials of fan blades and fan guards shall be a non-sparking combination.

7.2.6.3 Flattened expanded metal for fan guards shall not exceed 50 mm (2 inch) nominal mesh size.

7.2.6.4 The minimum thickness of expanded metal mesh shall be as shown in Table 9.

Table 9 — Minimum thickness of expanded metal fan guard mesh

Nominal size	Minimum thickness
40 mm ($1^1/_2$ inch)	2 mm (0,070 inch)
50 (2 inch)	3 mm (0,110 inch)

7.2.6.5 The openings in woven or welded mesh for fan guards shall not exceed an average area of 2 600 mm^2 (4 square inches) if the wire spacing in both directions exceeds 25 mm (1 inch).

7.2.6.6 The thickness of wire for welded or woven mesh shall be at least 2,8 mm (12 BWG, 0,109 inch).

7.2.6.7 Fan guards shall be designed with stiffening members so that a concentrated load of 1 000 N (200 lb) on any 0,1 m^2 (1 ft^2) shall not cause fastener failure or stiffener deflection greater than $L/90$, where L is the length of the span between points of support.

7.2.6.8 The distance from the fan guard to the fan blade at its maximum operating pitch shall be at least 150 mm (6 inch) or six times the smaller of the opening dimensions, whichever is less.

7.2.6.9 Gaps between the fan guard and equipment or between sections of the fan guard shall not exceed 13 mm ($^1/_2$ inch).

7.2.7 Drivers

7.2.7.1 General

● **7.2.7.1.1** The purchaser shall specify the type of drive system and the vendor's scope of supply.

7.2.7.1.2 For electric motor drivers, the rated shaft power available at the motor shaft shall be the greater of the terms on the right sides of the following equations. For steam turbine drivers, the rated shaft power at the turbine coupling shall be the greater of the following:

$$P_{dr} \geqslant 1{,}05\,(P_{f1}/E_m)$$

$$P_{dr} \geqslant 1{,}10\,(P_{f2})$$

where

P_{dr} = driver rated shaft power;

P_{f1} = fan shaft power operating at specified minimum design temperature with blade angle set for design dry-bulb temperature;

E_m = mechanical efficiency of separate power transmissions;

P_{f1} = fan shaft power operating at design dry-bulb temperature.

These requirements apply to fixed-pitch, variable-pitch and variable-speed fans unless otherwise specified.

7.2.7.2 Electric motor drivers

● **7.2.7.2.1** Electric motors shall be three-phase, totally enclosed, fan-cooled motors suitable for service in petrochemical installations and capable of full-voltage starting, full-phase inversion, continuous duty and designed for an 80 °C (140 °F) temperature rise over 40 °C (104 °F) ambient temperature at nameplate rating. The purchaser shall specify the voltage and frequency, the applicable motor specification, the hazardous area classification, the temperature classification and the insulation class.

7.2.7.2.2 The motor manufacturer shall be advised that the motor is intended for air-cooled heat exchanger service and operation outdoors, unprotected against the weather. If the motor is to operate vertically, the manufacturer shall verify in writing that the motor is suitable for vertical operation, either shaft up or shaft down.

7.2.7.2.3 Unless otherwise agreed by the purchaser, motor frames shall be of cast steel or corrosion-resistant cast iron, with integrally cast support feet.

7.2.7.2.4 The motor design loading shall exclude the service factor allowance.

7.2.7.2.5 Motors shall have grease-lubricated bearings designed for an L_{10} life of at least 40 000 h under continuous duty at rated load and speed (see 7.2.4.1 for the definition of L_{10}). If the motor is to be mounted vertically, the bearing lubrication system and seals shall be suitable for a vertically mounted motor.

7.2.7.2.6 If the motor is mounted in the shaft-up position, the belt sheave shall be designed as a shield to prevent water from accumulating and being directed down the motor shaft while the motor is either idling or running. Alternatively, an external conical slinger may be fitted to the shaft to prevent water from entering the housing along the shaft.

7.2.7.2.7 Motors shall have drains at the lowest point of the frame as mounted on the air-cooled heat exchanger.

7.2.7.2.8 Standard motors are designed for 40 °C (104 °F) ambient temperature and altitudes not exceeding 1 000 m (3 280 ft). Higher temperatures and/or altitudes (resulting in reduced air density) may require improved insulation or an increase in motor frame size. If the motor is required to be suitable for service exceeding the standard conditions, the motor manufacturer shall be notified.

7.2.7.2.9 If specified by the purchaser, a self-actuating braking device shall be installed to prevent reverse rotation of an idle fan and the connected driver caused by down-draughts.

7.2.7.3 Variable-speed drive systems

Requirements for variable-speed drive systems (VSDS) shall be agreed between the purchaser and the vendor.

7.2.7.4 Steam turbine drivers

Steam turbine drivers shall be in accordance with ISO 10436.

7.2.8 Couplings and power transmissions

7.2.8.1 General

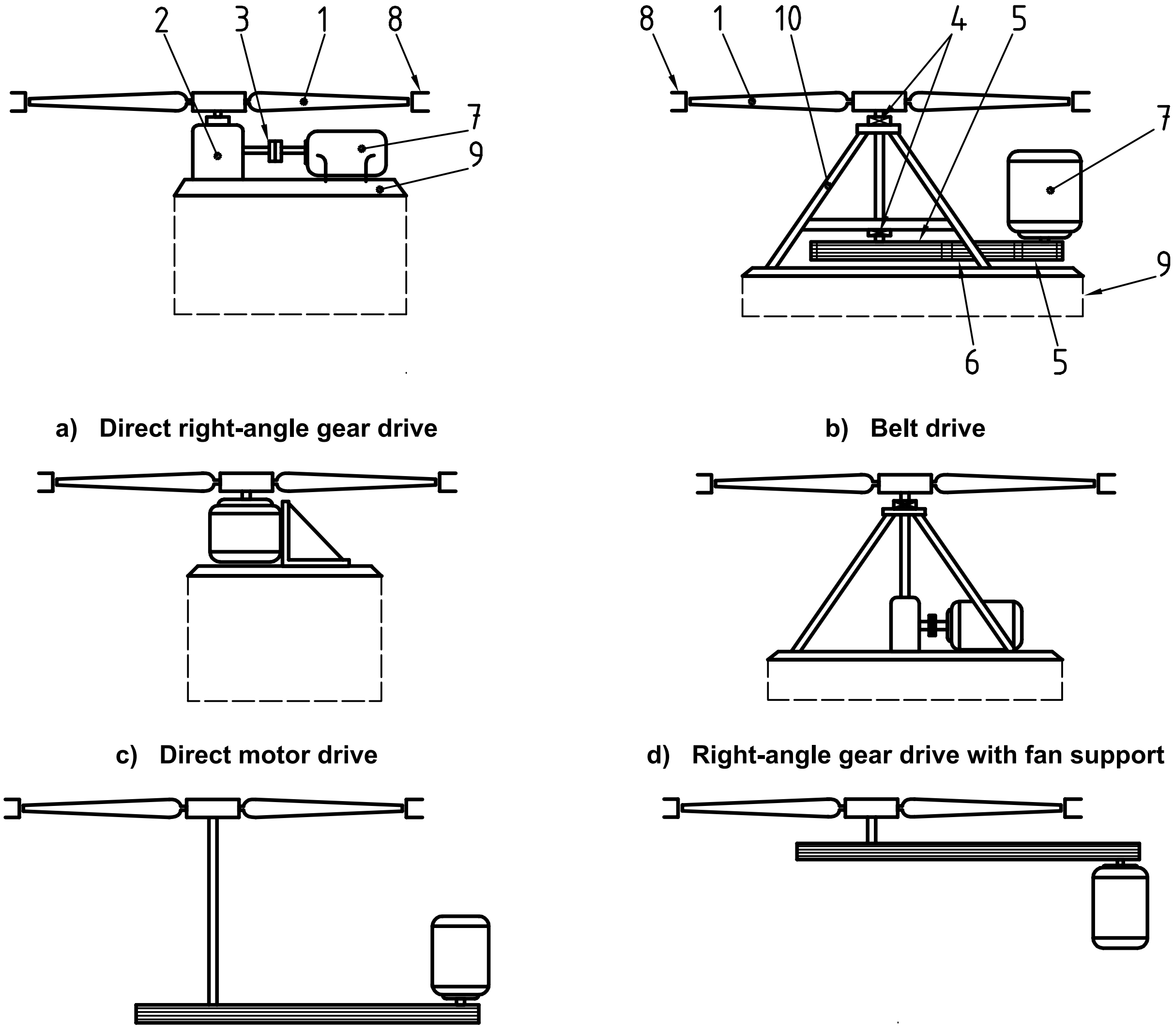
7.2.8.1.1 Bushings and couplings shall be either split taper or cylindrical fit and shall be keyed.

7.2.8.1.2 Power transmission components shall have a rated power for continuous service that is at least equal to the rated power of the actual driver multiplied by the component service factor.

7.2.8.1.3 Fan shaft and gear shaft couplings shall be the non-lubricated type and shall have a minimum service factor of 1,5.

7.2.8.1.4 Exposed moving parts shall have guards in accordance with 7.2.8.4.

7.2.8.1.5 Figure 8 shows typical drive arrangements.



a) **Direct right-angle gear drive**

b) **Belt drive**

c) **Direct motor drive**

d) **Right-angle gear drive with fan support**

e) **Suspended belt drive, motor shaft down**

f) **Suspended belt drive, motor shaft up**

Key

1 Fan
2 Gear box
3 Coupling
4 Bearing
5 Sheave
6 Belt drive
7 Motor
8 Fan ring
9 Base plate
10 Fan support

Figure 8 — Typical drive arrangements

7.2.8.2 Belt drives

7.2.8.2.1 Belt drives shall be either conventional V-belts or high-torque type positive-drive belts.

7.2.8.2.2 Belt drives in a heated air stream (such as top-mounted drives) shall not be used unless approved by the purchaser. If so approved, the belt design temperature shall take into account the maximum air temperature near the belt (or the maximum belt temperature possible due to radiation) under all conditions; decreased fan efficiency shall also be taken into account. The vendor shall indicate how the driver is suspended; the driver should not be located in the heated air stream (see also 7.2.7.2.8 and 7.2.8.2.13).

7.2.8.2.3 Belt drives shall be provided with guards in accordance with 7.2.8.4.

7.2.8.2.4 Belt drives shall be provided with jack screws or an equivalent means of initial belt-tensioning and/or re-tensioning.

7.2.8.2.5 V-belt drives shall be in accordance with ISO 1081, ISO 4183, ISO 4184, ISO 5287, ISO 5290 and/or ISO 9563 as applicable.

7.2.8.2.6 V-belts shall be either matched sets of individual belts or a multiple-belt section formed by joining a matched set of individual belts.

7.2.8.2.7 High-torque type positive-drive belts may be either one belt or a pair of matched belts.

7.2.8.2.8 V-belts shall have a minimum service factor of 1,4 based on driver rated power.

7.2.8.2.9 High-torque type positive-drive belts shall have a minimum service factor of 1,8 based on driver-rated power.

7.2.8.2.10 V-belt drive assemblies suspended from the structure may be used with motor drivers rated not higher than 30 kW (40 HP).

7.2.8.2.11 High-torque type positive-drive-belt drive assemblies suspended from the structure may be used with motor drivers rated not higher than 45 kW (60 HP)

7.2.8.2.12 The drive-belt jacket shall be oil resistant.

7.2.8.2.13 Standard drive-belt materials are limited to an exposure temperature of 60 °C (140 °F).

7.2.8.3 Gear drives

7.2.8.3.1 Electric motors rated higher than 45 kW (60 HP) shall use gear drives; motors rated not higher than 45 kW (60 HP) may use gear drives.

7.2.8.3.2 Gear drives for electric motors rated not higher than 45 kW (60 HP) may be suspended from the structure.

7.2.8.3.3 Steam turbine drivers shall use gear drives.

7.2.8.3.4 Gears shall be of the spiral bevel type. They shall have a minimum service factor of 2,0 in accordance with AGMA 6010.

7.2.8.3.5 Top-mounted gear drives shall not be used.

7.2.8.3.6 Gear boxes shall be provided with an external oil level indicator visible from the maintenance platform.

7.2.8.3.7 The vendor shall provide information concerning the expected gear lubrication oil temperature, the viscosity grade of oil provided initially, and other lubrication recommendations.

7.2.8.4 Mechanical power transmission guards

7.2.8.4.1 Guards shall be provided for moving components.

7.2.8.4.2 Guards shall be constructed to allow ready access for maintenance of the equipment.

7.2.8.4.3 Gaps between guards and equipment shall not exceed 13 mm ($^1/_2$ inch).

7.2.9 Vibration cut-out switches

▲ **7.2.9.1** One readily accessible, double-throw, two-contact vibration cut-out switch shall be provided for each fan driver unit if specified by the purchaser.

7.2.9.2 Vibration cut-out switches shall be of the manual, externally reset type not requiring dismantling of the switch for resetting and shall have sensitivity adjustment.

7.2.10 Louvres

7.2.10.1 The thickness of louvre blades manufactured from plain sheets shall be at least 1,5 mm (16 gauge USS, 0,060 inch) for carbon steel and 2,3 mm (0,090 inch) for aluminium. The thickness of extruded hollow-shaped aluminium blades shall be at least 1,5 mm (0,060 inch).

7.2.10.2 Frames of carbon steel shall be at least 3 mm (10 gauge USS, 0,135 inch) thick; frames of aluminium shall be at least 4 mm (0,160 inch) thick.

7.2.10.3 The unsupported louvre blade length shall not exceed 2,1 m (7 ft).

7.2.10.4 The deflections of louvre blades and side frames shall not exceed the values given in Table 10.

Table 10 — Maximum allowable louvre deflection

	Maximum deflection
Louvre blades in closed position with design load of 2 000 N/m^2 (40 lb/ft^2)	L/180
Louvre side frames with uniform design load of 1 000 N/m^2 (20 lb/ft^2)	L/360
L = length of the span between points of support.	

7.2.10.5 The deflection of louvre blades and side frames shall be evaluated at a metal temperature equal to the higher of the following:

a) the maximum process inlet temperature less 30 °C (50 °F);

b) the specified air inlet dry-bulb temperature.

7.2.10.6 The gap between the louvre blade and the frame at the header ends shall not exceed 6 mm ($^1/_4$ inch).

7.2.10.7 The gap between the louvre blades and the frame at the louvre sides shall not exceed 3 mm ($^1/_8$ inch)

7.2.10.8 Louvre blade pivot pins shall be designed for their load but, in any case, shall be at least 9 mm ($^3/_8$ inch) in diameter.

7.2.10.9 Bearings designed for exposure temperature in accordance with 7.2.1.4 shall be provided at all pivot points, including control arm, torque rod and blade pivot pins. Bearings shall not require lubrication. The exposure temperature shall not exceed 150 °C (300 °F) for polytetrafluoroethylene (PTFE) base composite bearing material in accordance with 8.3.2. Higher-temperature bearing materials are available but may be used only with the approval of the purchaser.

7.2.10.10 Louvre linkages shall be designed so that equal movement of all louvre blades results from a change of actuator position. The maximum allowable deviation shall be 3 mm ($^1/_8$ inch), measured as a gap between any two blades with louvre actuator in the fully closed position. The means of transmitting force between the louvre actuator and the blades shall be adequate to withstand, without damage, the maximum possible force that may be applied by the actuator in any blade position and in either direction.

7.2.10.11 Actuation of louvre sections shall require a torque of not more than 7 N·m for each square metre (6 inch-pounds of work per square foot) of face area to achieve full travel. The handling force to operate the louvres shall not exceed 250 N (56 lb)

7.2.10.12 The travel of louvre blades from fully closed to fully open shall be at least 70°.

7.2.10.13 All shaft connections shall be attached at adjustable linkage points by keys, splines or equivalent positive methods. Set-screw connections shall not be used.

▲ **7.2.10.14** If used for automatic control, louvre actuators shall be designed to operate with a 20 kPa to 100 kPa gauge (3 psig to 15 psig) pneumatic control signal. If supplied with design motive-air pressure, actuators shall be sized to supply at least 150 % of the necessary force for full-range louvre blade travel. Design motive-air pressure shall be 410 kPa gauge (60 psig) unless otherwise specified.

▲ **7.2.10.15** A positioner shall be provided at each actuator unless otherwise specified.

7.2.10.16 If a single controller operates more than one actuator, the purchaser shall provide an isolating valve in the signal line for each actuator, to allow maintenance.

7.2.10.17 The location of the actuator and positioner assembly shall not interfere with access to the header, and both shall be readily accessible for maintenance from a service platform (if available). The assembly shall not be in the hot-air stream if the exit air temperature at any condition exceeds 70 °C (160 °F). Alternative materials shall be selected for higher exposure temperatures.

▲ **7.2.10.18** The louvre position upon loss of control-air pressure shall be specified by the purchaser.

7.2.10.19 All louvres not automatically or otherwise remotely operated shall be provided with extensions or chains to permit manual operation from grade or platform, except that extensions or chains shall not be used if longer than 6 m (20 ft). Handles for manual operators shall not project into walkways or access ways in any operating position.

7.2.10.20 A locking device shall be provided for manual operators to maintain louvre position. Set-screw or thumb-screw locking devices shall not be used. A means shall be provided to indicate whether the louvres are open or closed.

7.2.10.21 The louvre characteristic performance curve shall relate the percentage of air flow to the angle of the louvre blade.

7.2.10.22 All requirements apply to both parallel- and opposed-action louvres, unless otherwise specified.

7.2.10.23 Due to the nature of their design, louvres are vulnerable to damage during handling. Spreader bars and anti-racking procedures should be used. Specific handling instructions shall be included on the louvre assembly drawing. A suitable handling procedure shall be marked on the louvre at one lift point.

7.2.10.24 Pin-type retainers shall be used to hold manual control levers of louvres in a set position; butterfly-type locking nuts shall not be used.

7.2.10.25 All linkage joints shall be through-bolted or pinned; friction-type joints shall not be used. The bolting or pinning shall be done after final linkage adjustment.

7.2.11 Screens

● The purchaser shall specify if screens are required and, if so, shall specify the type (hail screens, insect screens and/or lint screens).

7.3 Structural design

7.3.1 General

● **7.3.1.1** The structural code shall be specified or agreed by the purchaser. Structural steel design, fabrication and erection shall be in accordance with the structural code.

7.3.1.2 Bolts for load-bearing members shall be designed and installed in accordance with the structural code.

7.3.1.3 Weld-metal design stress shall conform to the structural code.

7.3.1.4 Structural members should be designed without the need for field welding.

7.3.1.5 For induced-draught exchangers, tube bundles shall be removable without removing the platforms, unless otherwise specified by the purchaser. For forced-draught exchangers, the bundles shall be removable without separately supporting or dismantling the fan, plenum or platforms and without disturbing the structure or adjacent bays.

7.3.1.6 Suspended drives shall be attached to the structure by through-bolts to permit dismantling.

7.3.2 Vibration testing

7.3.2.1 Structural members shall be designed to minimize vibration. The maximum amplitude of vibration over the design fan-speed range shall be 0,15 mm (0,006 inch) from peak to peak, as measured on primary structural members and machinery mountings.

● **7.3.2.2** The purchaser shall specify if a shop test is required to verify compliance with the vibration limits.

7.3.2.3 Wind velocity at test conditions shall not exceed 5 m/s (10 mph).

7.3.2.4 The effective vibration velocity (r.m.s.), measured on the bearings perpendicular to the fan shaft centreline, shall not exceed 6,3 mm/s ($^1/_4$ in/s) up to 10 r/s and 3,0 mm/s ($^1/_8$ in/s) above 10 r/s.

7.3.3 Structural design loads and forces

7.3.3.1 General

The design shall take into account the loads and forces defined in 7.3.3.2 through 7.3.3.13.

7.3.3.2 Dead loads

● Dead loads shall consist of the total mass of the material furnished by the vendor plus the mass of any fireproofing. If fireproofing is to be applied, the purchaser shall state the extent and mass.

7.3.3.3 Live loads

Live loads shall consist of movable loads (including personnel, portable machinery, tools and equipment) and operating loads in equipment and piping. Design live loads on platforms, columns and walkways (exclusive of loads from piping and equipment in place) shall be as specified in Table 11.

Table 11 — Live loads on platforms, columns and walkways

	Average load	Concentrated load
Floor plate or grating	4 900 N/m^2 (100 lb/ft^2)	
Floor framing	2 450 N/m^2 (50 lb/ft^2)	2 250 N (500 lb)
Columns and brackets	1 200 N/m^2 (25 lb/ft^2)	2 250 N (500 lb)
Ladders and stairways		2 500 N (500 lb)

7.3.3.4 Impact loads

The vertical design impact load for lifting devices furnished by the vendor shall be 2,0 times the mass of the heaviest piece of equipment to be lifted. The lateral impact load shall be 0,35 times the mass to be lifted.

7.3.3.5 Thermal forces

Thermal forces shall include forces caused by partial or complete anchorage of piping or equipment, sliding or rolling friction of equipment and expansion or contraction of the structure. The purchaser and the vendor shall agree on acceptable thermal forces.

7.3.3.6 Test load

The test load is the load due to the filling of equipment with water for testing.

7.3.3.7 Wind load

The wind design load shall be in accordance with the structural code.

7.3.3.8 Earthquake forces

Earthquake design shall be in accordance with ICBO *Uniform Building Code* unless otherwise specified.

7.3.3.9 Nozzle loads

Nozzle loads shall include all forces and moments applied to the nozzle face, such as deadweight of pipe, thermal forces, the mass of fluid in the piping, etc. The total magnitude and direction of these forces and moments shall be in accordance with 7.1.10 unless otherwise specified.

7.3.3.10 Fan thrust

Fan thrust shall be based on the maximum thrust. If velocity pressure is not included, then fan thrust shall be based on the static pressure shown on the data sheet multiplied by 1,25.

7.3.3.11 Snow load

● The purchaser shall specify the snow load, if any, to be applied to the total air-cooled heat exchanger plot area.

7.3.3.12 Other loads

● Loads, forces and moments other than those described in 7.3.3.3 through 7.3.3.11 which are to be supported by, or applied to, the air-cooled heat exchanger shall be specified by the purchaser in terms of exact type, location, magnitude and direction. Examples of such loads are special transportation loads, auxiliary pipe supports, ladders and walkways furnished by others, and temporary scaffolding supports. Structural and nozzle loads imposed by movement of the structure or installation (e.g. floating production system) on which the exchanger is mounted shall be specified by the purchaser in terms of the exact type, location, magnitude and direction (e.g. pitch, roll, yaw, heave, surge and sway).

7.3.3.13 Loading combinations

All structural components shall be designed to support combinations of the loads and forces to which they may be subjected during erection, testing or flushing of the equipment or during normal operations. The following combination of loads and forces shall be considered in the design of columns, bracing, anchor bolts and foundations and in checking stability against overturning. However, loading conditions of a special nature shall at all times receive proper consideration. (All loads and forces are additive).

a) Erection:

   1) dead load of the structure, less fireproofing;

   2) the greater of the following:

   i) dead load of equipment, less piping;

   ii) dead load of equipment, less platforms supported by the equipment.

   3) full wind load or earthquake load, whichever is greater.

b) Testing or flushing equipment:

   1) dead load of the structure, plus fireproofing;

   2) dead load of equipment, including platforms supported by the equipment;

   3) nozzle loads;

   4) test loads;

   5) wind load of 500 N/m^2 (10 lb/ft^2);

   6) applicable live loads from platforms and walkways specified in 7.3.3.3. However, these live loads shall not be included in either the design of anchor bolts or the check for stability against wind or earthquake.

c) Normal operations:

   1) dead load of structure;

   2) dead load of equipment, including platforms supported by the equipment;

   3) nozzle loads;

   4) operating weight of fluid in equipment;

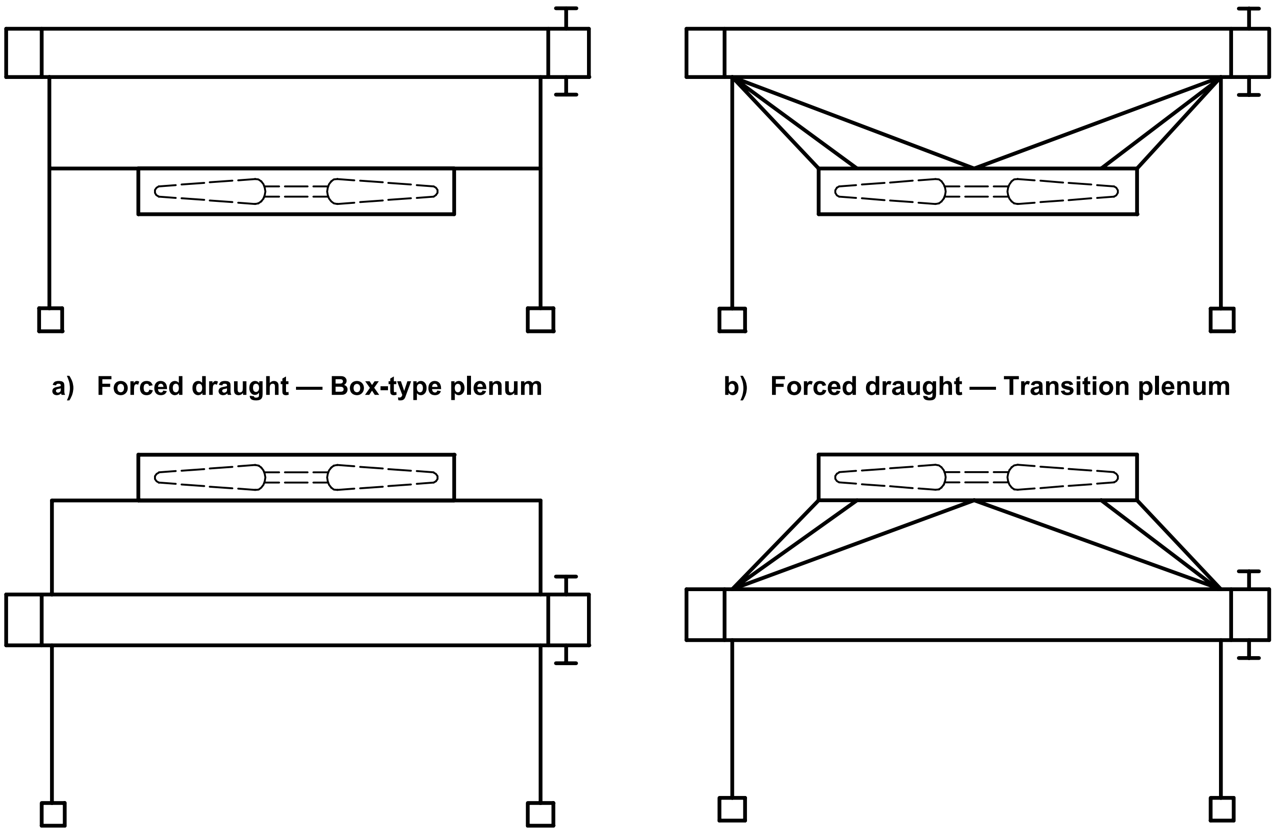
   5) unbalanced forces from impact.

   6) applicable live loads specified in 7.3.3.3. However, these live loads shall not be included in either the design of anchor bolts or the check for stability against wind or earthquake.

   7) full wind load or earthquake load, whichever is greater.

7.3.4 Plenums

7.3.4.1 Figure 9 shows typical plenum arrangements.



a) Forced draught — Box-type plenum

b) Forced draught — Transition plenum

c) Induced draught — Box-type plenum

d) Induced draught — Transition plenum

Figure 9 — Typical plenum arrangements

7.3.4.2 Box-type plenums employing panel construction shall be designed to form an integral part of the structure.

7.3.4.3 Bank arrangements for field-assembled units may be designed with common walls between adjacent plenums.

7.3.4.4 The plenums between the fan and the bundle shall be partitioned to prevent recirculation of air from operating fans through non-operating fans.

● **7.3.4.5** Plenum partition requirements for recirculation systems shall be as specified or agreed by the purchaser.

NOTE Annex C may be consulted for guidance.

7.3.4.6 The thickness of steel sheet material used in the construction of plenums shall be at least 2,0 mm (14 gauge USS, 0,075 inch) flat or 1,5 mm (16 gauge USS, 0,060 inch) ribbed.

7.3.4.7 The minimum plenum height shall be obtained from information provided in 7.2.3.4 and Figure 7.

7.3.4.8 Fabricated steel fan decks shall be designed for a live load of 2 500 N/m^2 (50 lb/ft^2) but in any case the metal thickness shall be at least 2,7 mm (12 gauge USS, 0,105 inch).

7.3.5 Mechanical access facilities

● **7.3.5.1** The number and location of header access platforms, interconnecting walkways and ladders shall be specified by the purchaser.

▲ **7.3.5.2** If specified, maintenance platforms shall be provided beneath each drive assembly to provide access for removal and replacement of all drive components. An unobstructed platform area shall be provided, extending at least 0,6 m (2 ft) in any plan dimension on all sides of the driver and drive components. However, such platforms shall not extend beyond the bay plan limits.

7.3.5.3 Platforms shall have a clear width of at least 0,75 m (2 ft 6 inches).

7.3.5.4 The floor of the walkways, platforms, etc. shall be a raised-pattern solid plate with drain holes, expanded metal or grating. If raised-pattern steel is used, the thickness shall be at least 6 mm ($^{1}/_{4}$ inch). Glass-reinforced plastic (GRP) may be used if specified or agreed by the purchaser.

7.3.5.5 Ladders, railings, toe plates, safety cages, etc. shall be of steel construction or, if specified or agreed by the purchaser, GRP in accordance with local regulations. The following shall apply:

a) Safety cages shall be provided for ladders over 3 m (10 ft) high.

b) Chains with safety hooks, or safety gates, shall be provided across ladder openings at platforms.

c) Ladders over 2 m (6 ft) high shall provide for side-step access to platforms unless otherwise specified.

7.3.5.6 Header platforms shall be provided with a toeboard on the side next to the exchanger. If the gap between the platform and the exchanger is greater than 150 mm (6 inches), a knee rail shall be fitted.

7.3.5.7 If steel pipe railings are not galvanized, they shall be sealed to prevent internal corrosion.

● **7.3.5.8** The purchaser shall specify requirements, if any, for personnel protection against high air-outlet temperatures and hot surfaces.

7.3.6 Lifting devices

7.3.6.1 At least two lifting lugs shall be provided on each side frame of tube bundles and each louvre section side frame. Lifting lugs on side frames of adjacent bundles shall be located so as not to interfere with bundle installation.

7.3.6.2 Two lifting lugs shall be provided on each removable cover plate and each removable bonnet.

7.3.6.3 Solid-forging or plate-type lifting lugs shall be used for tube bundle side frames, louvre side frames, cover plates and bonnets. The opening in the lug shall be at least 40 mm ($1^{1}/_{2}$ inch) diameter.

7.3.6.4 Sufficient lifting eyes shall be provided on each driver and gear to allow safe installation and dismantling. Provisions shall be made in the structure to accommodate lifting tools.

7.3.6.5 Lug or eye design shall be based on a total load equal to twice the weight of the lift.

7.3.6.6 A structural member shall be provided with load attachment points for removal and replacement of driver components.

8 Materials

8.1 General

8.1.1 Materials for pressure components shall be in accordance with the pressure design code.

8.1.2 Cast iron shall not be used for pressure components in flammable, lethal or toxic service.

8.1.3 Structural supports, such as side frames and beams, that are part of the tube bundle and not accessible for maintenance shall be galvanized unless otherwise specified.

8.1.4 Galvanizing of structural steel shall be in accordance with ISO 1459 and ISO 1461.

8.1.5 Galvanized materials or zinc-containing paints etc., should not be used on or directly above exposed austenitic stainless steel or high nickel alloy pressure components.

8.1.6 Combinations of construction materials shall be compatible such that electrolytic (galvanic) cells are minimized.

8.2 Headers

8.2.1 External load-bearing parts welded to headers shall be of a material allowed by the pressure design code.

8.2.2 Welded header pass partitions and stiffeners shall be of the same material specification as the header plate except that, with the purchaser's approval, welded pass partitions or stiffeners of suitable alloy steel may be used in carbon steel headers to avoid excessively thick sections.

8.2.3 Tube access header plug material shall be compatible with the header material. Cast iron shall not be used for plug material.

8.2.4 Tube access header plugs of carbon steel bar-stock material or individual forged plugs shall be of a material allowed by the pressure design code (e.g. ASME II, SA-105).

8.2.5 If the header material is solid stainless steel, precautions should be taken to avoid galling between the plugs and the plug sheet.

8.3 Louvres

8.3.1 Louvre blade pivot pins shall be austenitic stainless steel or UNS A96063 aluminium alloy in the T6 temper condition.

8.3.2 Louvre bearings shall be of either polytetrafluoroethylene (PTFE) base composite material containing at least 20 % fill for exposure temperatures of up to 150 °C (300 °F), or an approved alternative if required for a higher design temperature.

8.3.3 Steel louvre blades and frames shall be galvanized. If mill-galvanized material is used, all cut and punched edges shall be protected by a zinc-rich coating.

8.4 Other components

▲ **8.4.1** Fin material shall be aluminium unless otherwise specified or agreed by the purchaser.

▲ **8.4.2** Fan blades shall be of aluminium alloy or GRP unless otherwise specified.

8.4.3 Plugs for threaded connections shall be of a material with an alloy content at least equal to that of the connection.

8.4.4 Plenums, fan decks, partitions, platforms and fan rings shall be of carbon steel unless otherwise specified.

8.4.5 Metal gasket material shall be softer than the gasket contact surface.

8.4.6 Solid metal gaskets for shoulder plugs shall have a Brinell hardness no greater than 120 HB for carbon steel or 160 HB for austenitic stainless steel and austenitic/ferritic (duplex) stainless steel.

9 Fabrication of tube bundle

9.1 Welding

9.1.1 General

9.1.1.1 Welding procedures and welders shall be qualified in accordance with the pressure design code. Welding shall be performed in accordance with the pressure design code.

9.1.1.2 All header welds subject to pressure shall have full penetration and full fusion. All header welds other than connection-to-header welds shall be double-side welded joints, except that if one side of a weld on a pressure part is not accessible, single-side welded joints may be applied provided full penetration is obtained.

9.1.1.3 The root pass of single-side welded joints without backing strips shall be made using gas metal arc welding (GMAW), gas tungsten arc welding (GTAW) or low-hydrogen shielded metal arc welding (SMAW).

9.1.1.4 Enclosed spaces between any welded attachment and the headers shall be vented by a 3 mm ($^1/_8$ inch) diameter drilled hole.

9.1.2 Plug headers

9.1.2.1 Partition plates shall be seal-welded to abutting plates and shall be welded from both sides; full-penetration attachment welding may be used.

9.1.2.2 If pass partition plates are also used as stiffeners, a full-penetration attachment weld shall be used.

9.1.3 Removable cover plate and removable bonnet headers

9.1.3.1 Removable cover plate flanges and removable bonnet header flanges shall be installed with full penetration welding.

9.1.3.2 Partition plates and stiffeners shall be welded from both sides, along the full length of the three edges.

9.2 Postweld heat treatment

9.2.1 All carbon steel and low-alloy steel headers shall be subjected to postweld heat treatment. Welded tube-to-tubesheet joints shall be excluded from postweld heat treatment.

9.2.2 Gaskets made of ferritic materials and fabricated by welding shall be fully annealed after welding.

9.3 Tube-to-tubesheet joints

9.3.1 Tube hole diameters and tolerances

9.3.1.1 Tube holes in tubesheets shall be finished to the sizes and undertolerances shown under “Standard fit” in Table 12.

● **9.3.1.2** For work-hardening materials used for corrosion resistance, a closer fit between tube outside diameter and tube-hole inside diameter may tend to reduce work hardening (which can result in a loss of corrosion resistance). The closer fit shall be provided as shown under “Special close fit” in Table 12 if specified by the purchaser.

9.3.1.3 No more than 4 % of the total number of tube holes in a tubesheet may exceed the overtolerances shown under “Overtolerance” in Table 12. No tube holes shall exceed the nominal tube-hole diameter given in Table 12 by more than 0,25 mm (0,01 inch).

Table 12 — Nominal tube hole diameters and tolerances

Dimensions in millimetres (inches)

Nominal tube OD	Standard fit		Special close fit		Overtolerance
	Nominal tube hole diameter	Undertolerance	Nominal tube hole diameter	Undertolerance	
19,05 ($^3/_4$)	19,30 (0,760)	0,10 (0,004)	19,25 (0,758 in)	0,05 (0,002)	0,05 (0,002)
25,40 (1)	25,70 (1,012)	0,10 (0,004)	25,65 (1,010)	0,05 (0,002)	0,05 (0,002)
31,75 ($1^1/_4$)	32,11 (1,264)	0,15 (0,006)	32,03 (1,261)	0,08 (0,003)	0,08 (0,003)
38,10 ($1^1/_2$)	38,56 (1,518)	0,18 (0,007)	38,46 (1,514)	0,08 (0,003)	0,08 (0,003)
50,80 (2)	51,36 (2,022)	0,18 (0,007)	51,26 (2,018)	0,08 (0,003)	0,08 (0,003)

9.3.2 Tube-hole grooving

9.3.2.1 All tubesheet holes for expanded joints in tubesheets less than 25 mm (1 inch) thick shall be machined with one groove approximately 3 mm ($^1/_8$ inch) wide and 0,4 mm ($^1/_{64}$ inch) deep. A second groove shall be provided for tubesheets 25 mm (1 inch) or greater in thickness.

9.3.2.2 Tube-hole grooves shall be square-edged, concentric and free of burrs.

9.3.2.3 Grooves shall be located at least 3 mm ($^1/_8$ inch) plus the corrosion allowance from the process face of the tubesheet and at least 6 mm ($^1/_4$ inch) from the air-side face of the tubesheet.

9.3.3 Expanded tube-to-tubesheet joints

9.3.3.1 Tubes shall be expanded into the tubesheet for a length at least the smaller of the following:

a) 50 mm (2 inch);

b) the tubesheet thickness less 3 mm ($^1/_8$ inch).

In no case shall the expanded portion extend beyond the air-side face of the tubesheet.

9.3.3.2 The expanding procedure shall provide essentially uniform expansion throughout the expanded portion of the tube without a sharp transition to the unexpanded portion.

9.3.3.3 The ends of tubes shall extend at least 1,5 mm ($^1/_{16}$ inch) and not more than 9 mm ($^3/_8$ inch) beyond the tubesheet.

9.3.4 Welded tube-to-tubesheet joints

9.3.4.1 If approved by the purchaser, tube-to-tubesheet joints may be welded if tubes and tubesheets (or tubesheet facing) are of suitable materials.

9.3.4.2 If welding is used for sealing the tube-to-tubesheet joint and customary tube loads are carried by the expanded joint (seal-welded joint), the joints shall comply with 9.3.1, 9.3.2 and 9.3.3.

9.3.4.3 If welded tube joints are used as a complete substitute for expanded, strength-welded joints, the requirements of 9.3.1, 9.3.2 and 9.3.3 may be modified if agreed between the vendor and the purchaser.

9.4 Gasket contact surfaces

9.4.1 Final machining of gasket contact surfaces for removable cover plates shall be done after any postweld heat treatment.

9.4.2 Gasket contact surfaces of removable bonnet headers and removable cover plate headers shall be true planes within 1 mm ($^1/_{32}$ inch). The flatness of tubesheet gasket contact surfaces shall be measured after expanding or welding of the tubesheet joints.

9.4.3 Plug gasket contact surfaces shall be machined to a finish of average roughness between 3,2 µm and 6,3 µm (125 micro-inches and 250 micro-inches).

● **9.4.4** Special finish, if required, shall be specified by the purchaser.

9.5 Thread lubrication

9.5.1 Plug threads shall be coated with a suitable thread lubricant.

9.5.2 Header flange bolting shall be assembled using a thread lubricant suitable for the operating temperature.

9.6 Alignment and tolerances

9.6.1 Standard tolerances for the dimensions of air-cooled heat exchangers and for nozzle locations are shown in Figure 10. Tolerances apply to both forced-draught and induced-draught exchangers.

9.6.2 Header warpage shall be not more than 12 mm ($^1/_2$ inch) or 5 mm/m ($^1/_{16}$ inch/ft), whichever is less.

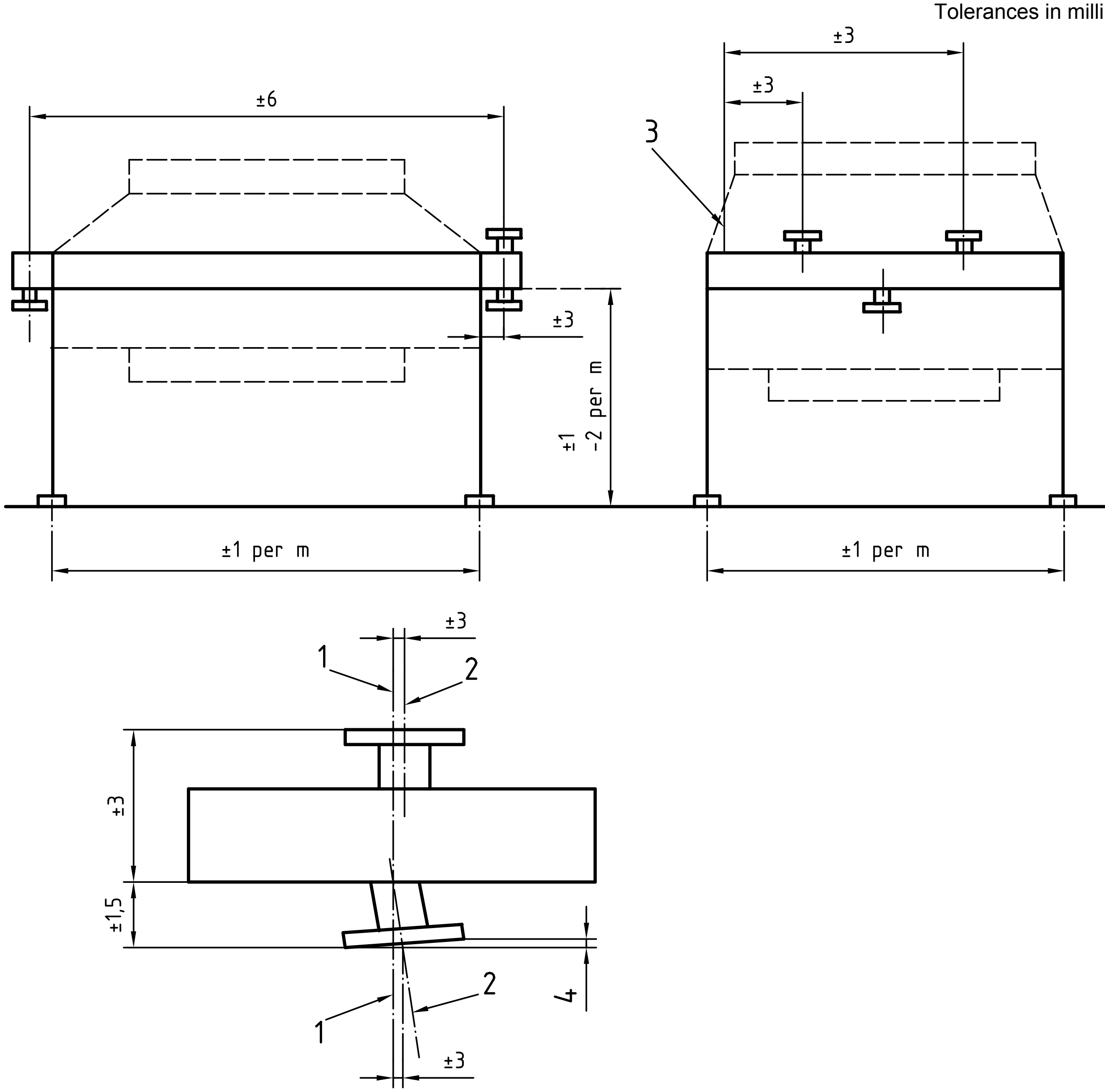
9.6.3 Bundles that are to be stacked in service shall be shop-checked for match tolerance.

9.6.4 Manufacturing tolerances shall be such that nominally identical parts shall be interchangeable.

9.7 Assembly

Air-cooled heat exchangers shall be completely assembled for shipment, except that if complete assembly is impractical they shall be partially shop-assembled into the largest practical sub-items to minimize field assembly work. The completeness of assembly for shipment shall be agreed between the purchaser and the vendor (see 5.5).

Tolerances in millimetres



NOTE 1 mm = 0,04 inch

Key

1 Centreline header
2 Centreline nozzle
3 Reference line
4 G = Out-of-plane tolerance

Nominal nozzle size DN (NPS)	**Maximum out-of-plane tolerance** G
50 to 100 (2 to 4)	1,6 mm ($^{1}/_{16}$ inch)
150 to 300 (6 to 12)	2,4 mm ($^{3}/_{32}$ inch)
> 300 (> 12)	4,8 mm ($^{3}/_{16}$ inch)
stacked, all	0,8 mm ($^{1}/_{32}$ inch)

Figure 10 — Standard tolerances

10 Inspection, examination and testing

10.1 General

● **10.1.1** If specified by the purchaser, the materials, fabrication, conformance with design, and testing of heat exchangers shall be subject to inspection for acceptance by the purchaser, his designated representative, or both.

10.1.2 The inspector designated by the purchaser shall be permitted entry to the vendor's shop where and while the work is being performed. The vendor shall afford the inspector reasonable facilities to satisfy him that the exchangers are being furnished in accordance with the requirements specified in the order.

10.1.3 All certification of materials, shop test data, and so forth needed to verify that the requirements of the specification are being met shall be available to the purchaser.

10.1.4 No unit shall be released for shipment without the approval of the purchaser or his representative.

10.2 Quality control

10.2.1 On components subject to full radiography, nozzle attachment welds that cannot be readily examined by radiography in accordance with the pressure design code shall have their root pass and final pass fully examined by the magnetic-particle or liquid-penetrant method after back-chipping or gouging (where applicable).

10.2.2 If full radiographic or ultrasonic examination is not specified, at least one spot radiographic or ultrasonic examination shall be made of a longitudinal outside pressure weld and an end-closure weld for each header. Process nozzle attachment welds shall be examined by the magnetic-particle or liquid-penetrant method. Examination shall apply to the root pass after back-chipping or flame-gouging (where applicable) and to the completed weld.

10.2.3 Spot radiographic or ultrasonic examinations shall include each start and stop of weld made by the automatic submerged arc-welding process and repaired areas of burn-through.

10.2.4 Spot radiographic or ultrasonic examinations shall cover either a length of at least 250 mm (10 inch) or the full length if the weld is less than 250 mm (10 inch) long.

10.2.5 For stainless steel and for ferritic alloy steel with a chromium content greater than 0,5 %, the root pass and final passes of welds not subject to full radiography shall be examined by the magnetic-particle or liquid-penetrant method.

10.2.6 If the plates are not fully examined for laminations by ultrasonic examination and if set-on connections are used, the edge of the hole in the plate to which the connections are attached shall be examined for laminations by magnetic particle or liquid penetrant. Indications found shall be cleared to sound metal and then back-welded.

10.2.7 Non-destructive examinations and acceptance criteria shall comply with the pressure design code.

10.2.8 Weld hardness testing shall be as follows.

a) Weld metal and heat-affected zones of pressure-retaining welds in components made of a carbon, Cr-Mo and 11/13/17 % chromium steels shall be hardness tested.

b) Hardness testing shall be performed using the Vickers or Rockwell test method.

c) Examination shall be made after any required postweld heat treatment.

d) Hardness shall not exceed 225 HB for carbon steel and Cr-Mo steels up to 1,25 % chromium; hardness shall not exceed 240 HB for other Cr-Mo steels and 11/13/17 % chromium steels.

e) Representative welds, including connection-to-header welds, shall be examined. Examination shall be made of one longitudinal weld, one weld at an end closure, and each connection-to-header weld if the connection is DN 50 (NPS 2) or larger. At least one header per item and every tenth header shall be examined.

10.2.9 For tubes with circumferential welds, the vendor shall demonstrate by means of a qualification procedure that weld-root reinforcement on the tube inside diameter will not exceed 1,5 mm ($^{1}/_{16}$ inch). Permanent backing rings shall not be used.

10.2.10 Inspection of tubes with circumferential welds shall be as follows.

a) At least 10 % of the welded joints selected at random shall be examined using radiography. One double-wall elliptical exposure for double-wall viewing shall be taken for each joint.

b) Sampling shall be done progressively throughout the period of fabrication.

10.2.11 Bar stock material for tube access plugs shall be radially examined by an ultrasonic or radiographic method. There shall be no linear indications exceeding 9 mm ($^{3}/_{8}$ inch).

10.2.12 Individually forged tube access plugs, either hot- or cold-forged, need not be examined in accordance with 10.2.11.

10.3 Pressure test

10.3.1 Hydrostatic tests shall be in accordance with the pressure design code.

10.3.2 Hydrostatic test pressure shall be maintained for at least 1 h.

10.3.3 Water used for hydrostatic testing of units in which austenitic stainless steel or Ni-Cu alloy materials will be exposed to the test fluid shall be potable water with a chloride content less than 50 mg/kg (50 parts per million by weight).

10.3.4 Unless otherwise specified by the purchaser, paint or other coatings may be applied over welds prior to the final pressure test.

10.3.5 Joints taken apart after the pressure test shall be reassembled with new gaskets.

● **10.3.6** Other types of test, such as halogen tests, shall be specified by the purchaser.

10.4 Shop run-in

● The extent of shop run-in tests of the driver, the drive assembly, and the fan of shop-assembled units shall be a matter of agreement between the purchaser and the vendor.

10.5 Nameplates

10.5.1 An austenitic stainless steel nameplate shall be affixed to the inlet header of each tube bundle indicating the item number, marks required by the pressure design code and any other information specified by the purchaser.

10.5.2 The nameplate shall be permanently mounted on a bracket welded to the top of the header.

10.5.3 The following parts shall be stamped with the vendor's serial number:

a) header;

b) cover plate flange of cover plate headers;

c) tubesheet flange of bonnet headers.

11 Preparation for shipment

11.1 General

11.1.1 All liquids used for cleaning or testing shall be drained from units before shipment.

11.1.2 Tube bundles shall be free of foreign matter prior to shipment.

11.1.3 Exposed flanged connections shall be protected by either of the following:

a) gasketed steel covers fastened by the greater of the following:

   1) 50 % of the required flange bolting;

   2) four bolts.

b) commercially available plastic covers specifically designed for flange protection.

● **11.1.4** The extent of skidding, boxing, crating, protection or coating for shipment shall be specified or agreed by the purchaser.

11.1.5 Each loose piece or assembly shall be properly protected to prevent damage during normal shipping and handling.

11.2 Surfaces and finishes

11.2.1 Surfaces to be painted shall be degreased and cleaned by wire brushing or a similar means to remove loose scale, dirt, and other foreign materials.

11.2.2 Machined surfaces that will be exposed to the atmosphere in transit and subsequent storage shall be protected with an easily removable rust preventative.

▲ **11.2.3** Unless otherwise specified, carbon steel headers shall be blast-cleaned in accordance with ISO 8501-1, grade Sa $2^1/_2$, and then coated with an inorganic zinc-rich primer to a dry-film thickness of at least 50 µm (0,002 inch).

11.2.4 Other than surfaces of tubes, all exposed ferrous surfaces not otherwise coated shall be given one coat of the manufacturer's standard shop primer before shipment.

11.3 Identification, conditioning and notification

11.3.1 All parts shall be marked for identification and conditioned for shipment.

11.3.2 The vendor shall advise the purchaser if bundles are temporarily fixed to bundle frames for shipping purposes. Transit and erection clips or fasteners shall be clearly identified on the equipment and the field assembly drawings to ensure removal before commissioning of the exchanger.

12 Supplemental requirements

12.1 General

● This clause provides additional design, fabrication and examination requirements that shall apply if specified by the purchaser. In general, these supplemental requirements should be considered if the design pressure exceeds 14 000 kPa gauge (2 000 psig), if the plate thickness of a box-type header of an air-cooled heat exchanger exceeds 50 mm (2 inch) or if an exchanger is to be placed in a critical service.

12.2 Design

12.2.1 Alternatives to plug header construction may be proposed for design pressures exceeding 20 700 kPa gauge (3 000 psig).

NOTE Threads are susceptible to deterioration and possible failure.

12.2.2 Header corner-joint design shall provide for clear interpretation of weld quality in accordance with the pressure design code. The vendor shall include in the proposal a drawing showing full details of the proposed welded joint design.

12.2.3 All tubes not strength-welded to the tubesheet shall be expanded and seal-welded.

12.2.4 If recessed-type tube-to-tubesheet welds (in the tube holes) are used, additional tubesheet thickness may be required to provide for the integrity of the expanded joint.

12.2.5 Nozzle connections to headers shall be made with full-penetration welds.

12.3 Examination

12.3.1 Ultrasonic examination shall be performed on plates and forgings welded to other components if the thickness exceeds 65 mm ($2^1/_2$ inch).

12.3.2 Ultrasonic examination shall be performed on all forgings exceeding 100 mm (4 inch) thickness, except for bolted flat covers and standard flanges.

12.3.3 Ultrasonic examination shall be performed on welds exceeding 65 mm ($2^1/_2$ inch) in thickness.

12.3.4 Ultrasonic examination shall be performed on all nozzle attachment welds.

12.3.5 After ultrasonic examination of plates, forgings and welds has been performed, the purchaser shall be supplied with a report which includes diagrams of the surfaces scanned, the indications obtained, the areas repaired, the nature of defects repaired and the repair procedures used. The following information shall be provided:

a) the pulse-echo unit manufacturer, model, and damping control setting;

b) the search unit manufacturer, model, dimensions, and the substance (such as oil or water) that is used to couple the transducer with the material being inspected;

c) the frequency used and the test angle on the component's surface;

d) the wedge medium for angle-beam examination.

12.3.6 All header welds and nozzle-attachment welds shall be 100 % radiographed. The root and final weld passes shall be examined by the magnetic-particle or liquid-penetrant method. For all welds not examined by radiography, ultrasonic examination shall be substituted.

12.3.7 Ultrasonic examination shall be performed on all weld repairs after postweld heat treatment.

12.3.8 Prior to welding, a magnetic-particle or liquid-penetrant examination shall be performed on all edges and plate openings prepared for welding. Defects found shall be cleared to sound metal.

12.3.9 A magnetic-particle or liquid-penetrant examination shall be performed on all attachment welds (e.g. supports).

12.3.10 A magnetic-particle or liquid-penetrant examination shall be made of areas where temporary lugs have been removed; these areas shall be prepared for examination by grinding.

12.3.11 After hydrostatic testing, all exterior pressure-retaining welds and all interior nozzle welds that are accessible without disassembly shall be examined by the liquid-penetrant method.

12.3.12 For pipe-manifold-type header construction, all boss-to-tube and tube-to-U-bend welds shall be 100 % radiographed. Boss-to-header welds shall be examined externally by the magnetic particle or liquid penetrant method.

12.3.13 Non-destructive examinations and acceptance criteria shall comply with the pressure design code.

12.4 Testing

12.4.1 A shop-air test at 170 kPa gauge (25 psig) shall be applied after tube-to-tubesheet welding and prior to tube expansion. Tube-to-tubesheet joints shall be examined for leaks by applying a soap solution.

12.4.2 After the final pressure test, plug joints and all gasketed joints shall be air-tested at 170 kPa gauge (25 psig), testing for leaks either by applying a soap solution or by total immersion in a water tank.

Annex A
(informative)

Recommended practices

A.1 Tubes and finning

A.1.1 The maximum design process temperature for various types of fin bonding should be as given in Table A.1.

Table A.1 — Maximum process temperature for fin bonding types

Fin bonding type	Maximum design process temperature
Mechanically embedded fins	400 °C (750 °F)
Hot-dip galvanized steel fins	360 °C (680 °F)
Extruded fins (aluminium fins)	300 °C (570 °F)
Footed fins (single L) and overlap footed fins (double L)	130 °C (270 °F)
Knurled/aluminium fin, either single L or double L	200 °C (390 °F)
Laser-welded fins	> 400 °C (750 °F) (maximum should be agreed by purchaser)
NOTE Except where stated otherwise, the above limits are based on a carbon steel core tube and aluminium fins; different materials for the core tube and/or the fins may result in a different temperature limit.	

A.1.2 Serrated, segmented and louvred fins, and fins with spacing tabs, have a slightly higher air-side film coefficient. The disadvantage however is that they are more susceptible to air-side fouling and are more difficult to clean due to the sharp edges at the discontinuities, so they should be considered only for low-fouling duties.

A.1.3 Tube supports should be designed such that mechanical loads are transferred to the core of the tube.

A.1.4 Elliptical tubes

The minimum tube wall thickness for elliptical tubes shall be as specified in 7.1.11.3.

The minimum dimensions of the elliptical tube to be used shall be: short axis 14 mm ($^{9}/_{16}$ inch), long axis 36 mm ($1^{7}/_{16}$ inch).

The maximum allowable temperatures for the type of fin bonding shall be in accordance with A.1.1.

A.2 Fans and drivers

Variable-speed (speed frequency controlled, SFC) fans or automatic variable-pitch (AVP) fans may be used for process control.

If there are stringent noise limitations during night-time and if, due to a lower air-inlet temperature at night-time, the air flowrate could be reduced, variable-speed fans should be used.

A.3 Walkways and platforms

Open grating should be used for the maintenance floor underneath the fan inlets to reduce air-side pressure drop. If solid plate is used, the effect on air-side pressure drop should be taken into account. To minimize this effect, a larger distance from maintenance floor to fan inlet may be required.

A.4 Selection of header type

The header types should be selected in accordance with Table A.2.

Table A.2 — Header selection

	Design pressure
Plug type headers or removable cover plate headers	< 3 000 kPa gauge (435 psig)
Plug type headers	≥ 3 000 kPa gauge (435 psig) and/or for hydrogen service

For fluid streams with a fouling resistance greater than 0,000 34 m^2·K/W (0,001 93 °F·ft^2·h/BTU), or if fouling layers are expected that cannot be removed by chemical means, the bundle construction shall be suitable for mechanical cleaning.

In heat exchangers having a condensing duty, the passes for the condensing phase should extend over the full width of the bundle. In case of total condensation, the size of the outlet nozzles should be such that flooding of the bottom rows of tubes cannot occur.

A.5 Air design temperature

To determine the air design temperature, the higher of the following temperatures may be used for non-critical processes:

— the highest air temperature that is exceeded for 400 h per year;

— the highest air temperature that is exceeded for 40 h per year, less 4 °C (7 °F).

For critical processes, the air design temperature shall be the highest air temperature which is exceeded for 40 h per year.

For an optimum design, the following temperatures should be specified, together with alternative process conditions specified in 7.1.6.1.1:

— minimum design metal temperature;

— design metal temperature;

— minimum ambient temperature;

— design ambient temperature; and

— fin selection temperature.

A.6 Bearing lubrication

To allow proper lubrication, a relief device should be fitted which ensures that the new grease displaces the maximum amount of old grease and automatically ejects any surplus to the outside.

A.7 Gaskets for bonnet or cover-plate type headers

Gasket types are given in Table A.3 and the required gasket contact face surface finish is given in Table A.4.

Gaskets shall not contain asbestos.

Service conditions are listed in Table A.5 and the gaskets should be selected using Table A.6.

Table A.3 — Gasket types

	Description	Minimum width	Minimum thickness
1	Aramid-fibre-filled NBR, oil and acid resistant	9,5 mm ($^3/_8$ inch)	2 mm ($^5/_{64}$ inch)
2	Compressed sheet composition, oil or acid resistant	9,5 mm ($^3/_8$ inch)	1,6 mm ($^1/_{16}$ inch)
3	Flat metal-jacketed, soft iron, filled	12,5 mm ($^1/_2$ inch)	3,2 mm ($^1/_8$ inch)
4	Flat metal-jacketed, stainless, filled	12,5 mm ($^1/_2$ inch)	3,2 mm ($^1/_8$ inch)
5	Metal-reinforced PTFE layers	9,5 mm ($^3/_8$ inch)	1,6 mm ($^1/_{16}$ inch)
6	Metal-reinforced expanded graphite	9,5 mm ($^3/_8$ inch)	1,6 mm ($^1/_{16}$ inch)
7	Grooved gaskets with graphite layers	12 mm ($^1/_2$ inch)	6 mm ($^1/_4$ inch)

Table A.4 — Gasket contact-surface finish

Gasket type	R_a value	
	µm	(micro-inch)
1, 2, 5, 6, 7	3,2 to 6,3	(125 to 250)
3, 4	0,8 to 1,6	(32 to 64)

Table A.5 — Service conditions

Service condition	Description
I	Non-corrosive and mildly corrosive
II	Hydrocarbon streams containing sulfur compounds and naphthenic acids with an acid value exceeding 300 mg/kg KOH (300×10^{-6} mass fraction KOH), and for maximum operating temperatures above 230 °C (446 °F)
III	Hydrocarbon streams containing sulfur compounds and naphthenic acids with an acid value not exceeding 300 mg/kg KOH (300×10^{-6} mass fraction KOH) and for maximum operating temperatures above 330 °C (626 °F)
IV	Hydrocarbons containing hydrogen
V	Non-corrosive cooling water below 50 °C (122 °F)
VI	Mildly corrosive cooling water below 50 °C (122 °F)
VII	Corrosive cooling water below 50 °C (122 °F)
VIII	Frequent changes in temperature and pressure, (e.g. hot washing, dewaxing, chilling) and frequent cleaning (i.e. more than twice a year under all conditions I to VII).

Table A.6 — Gasket selection

Service condition	Design temperature		Maximum design pressure		Recommended type	Alternative type
	°C	(°F)	kPa (ga)	(psig)		
I	–200 to 0	(–300 to 32)	3 000	(435)	6	4
	0 to 150	(32 to 300)	2 000	(290)	1	2, 5, 6
	0 to 240	(32 to 460)	3 000	(435)	6	3
	240 to 450	(460 to 840)	3 000	(435)	6	3
II	0 to 150	(32 to 300)	2 000	(290)	1	2, 5, 6
	0 to 240	(32 to 460)	3 000	(435)	6	3
	240 to 450	(460 to 840)	3 000	(435)	6	4
III	330 to 450	(630 to 840)	3 000	(435)	6	
IV	0 to 450	(32 to 840)	3 000	(435)	6	
V, VI, VII	0 to 50	(32 to 120)			1 [3,2 mm ($^{1}/_{8}$ inch) thick]	2, 5, 6
VIII	0 to 450	(32 to 840)	6 000	(870)	4	

A.8 Selection of induced draught or forced draught

Forced-draught fans should be used, except that induced-draught fans should be considered for the following situations:

a) if temperature control of the process is critical and sudden downpour of rain (i.e. excessive cooling) would cause operating problems;

b) to minimize the risk of hot-air recirculation, especially for large installations and for services requiring a close approach of outlet process temperature to inlet air temperature;

c) on sites where air-side fouling is a significant problem, requiring bundles to be washed;

d) to improve thermal performance in the event of a fan failure (due to the stack effect);

e) in hot climates, where the fan plenum chamber will shield the bundle from the sun.

Annex B
(informative)

Checklist, data sheets and electronic data exchange

B.1 Contents and usage

The checklist and data sheets in this annex provide the data necessary for the description and design of air-cooled heat exchangers for petroleum and natural gas services.

The checklist is used to note the specific requirements the purchaser shall make in response to the clauses and subclauses in this International Standard alongside which bullets (●) are used to indicate that more information is required or a decision must be made.

Completion of the checklist is the responsibility of the purchaser. Completion of the data sheets is the joint responsibility of the purchaser and the vendor. The purchaser is responsible for the process data on the data sheets.

The transport properties shall be based on the total composition of each of the phases (water, steam, air, and hydrogen or another permanent gas) if these components are parts of a homogeneous phase. If the liquid has immiscible phases, the liquid properties shall be separately and completely specified for each phase. If the transport properties do not include the mentioned components if they are present, their concentrations in the process stream shall be stated. In the simple case of a well-defined, no-change-of-phase service, the purchaser may use the data sheets as the only document for data transmittal.

The purchaser may submit the checklist and data sheets to the vendor in a form other than that indicated herein.

After exchanger fabrication, the vendor shall complete the data sheets to make a permanent “as-built” record that accurately describes the equipment.

In clause B.2 of this annex, a standardized electronic data exchange file specification is provided and described.

AIR-COOLED HEAT EXCHANGER CHECKLIST	Job No. ____ Item No. ____ Page: 1 of 2 By ____ Date ____ Revision ____ Proposal No. ____ Contract No. ____ Inquiry No. ____ Order No. ____

Subclause No.			
4.1	Pressure design code:		
4.4	Applicable local regulations:		
5.8	Is noise data sheet required?	Yes	No
5.9	Are fan performance characteristic curves required?	Yes	No
6.1.1	Which documents are to be submitted? Which documents are subject to purchaser's approval?		
6.1.3	Are calculations to be submitted for approval?	Yes	No
6.1.4	Are welding data to be submitted for approval?	Yes	No
6.1.5	Additional engineering information required:		
6.2.2	Records to be furnished and whether they shall be in electronic form:		
7.1.1.11	Winterization methods required:		
7.1.1.12	Temperature, pressure and operating conditions of internal steam-out design:		
7.1.3.1	Maximum design temperature: Minimum design temperature: Minimum design metal temperatures:		
7.1.3.2	Maximum operating temperature for fin selection:		
7.1.4.1	Design pressure of tube bundle:		
7.1.6.1.1	Is an analysis required of alternative operating conditions in design of headers?	Yes	No
7.1.6.2.3	Cover plate bolting type:	Through bolts	Stud bolts
7.1.9.7	Plane of process flanges if not horizontal:		
7.1.9.8 (d)	Is a cast or fabricated transition allowed?	Yes	No
7.1.9.16	Chemical cleaning connection size, type, and location:		
7.1.11.2	Maximum tube length:		
7.1.11.13	May elliptical tubes be used?	Yes	No
7.2.1.1	Special environmental factors affecting air-side design:		
7.2.2.1	Location of noise level values:		
7.2.3.1	Is a single-fan arrangement for each bay acceptable?	Yes	No

AIR-COOLED HEAT EXCHANGER CHECKLIST	Job No. ______	Item No. ______
	Page: 2 of 2	By ______
	Date ______	Revision ______
	Proposal No. ______	Contract No. ______
	Inquiry No. ______	Order No. ______

Clause No.			
7.2.3.5	Is a fan tip speed between 60 m/s and 80 m/s acceptable? Acceptable speed:	Yes	No
7.2.3.11 c	Any special blade pitch limit stop setting:		
7.2.7.1.1	Type of drive system: Drive equipment supplier:	Purchaser	Vendor
7.2.7.2.1	Electric motor construction; supply and classification:		
7.2.11	Are screens required? Type:	Yes	No
7.3.1.1	Structural code:		
7.3.2.2	Is shop test for vibration check required?	Yes	No
7.3.3.1	Extent and mass of fireproofing:		
7.3.3.10	Snow load:		
7.3.3.11	Exact type, location, magnitude, and direction of other design loads:		
7.3.4.5	Plenum partition requirements for recirculation systems:		
7.3.5.1	Number and location of header access platforms, interconnecting walkways and ladders:		
7.3.5.8	Are there any special requirements for personnel protection against high air-outlet temperature? If yes, state:	Yes	No
9.3.1.2	Are special close-fit tolerances required?	Yes	No
9.4.4	Is there any special finish for gasket contact surfaces? If yes, state:	Yes	No
10.1.1	Level of inspection by purchaser:		
10.3.6	Are special tests required? Details:	Yes	No
10.4	Are shop run-in tests required? Details:	Yes	No
11.1.4	Extent of skidding, boxing, crating, protection or coating for shipment:		
12.1	Supplemental requirements of clause 12 that apply:		

AIR-COOLED HEAT EXCHANGER DATA SHEET (SI UNITS)

Job No.		Item No.	
Page	1 of 2	By	
Date		Revision	
Proposal No.		Contract No.	
Inquiry No.		Order No.	

Manufacturer		Heat exchanged, kW	
Model No.		Surface/item-finned tube, m²	
Customer		Bare tube, m²	
Plant location		MTD, eff., °C	
Service		Transfer rate-finned, W/m²·K	
Type draught	O Induced O Forced	Bare tube, service, W/m²·K	
Bay size (W × L), m ______ No. of bay/items ______		Clean, W/m²	

Basic design data

Pressure design code		Structural code	
Tube bundle code stamped	O Yes O No	Flammable service	O Yes O No
Heating coil code stamped	O Yes O No	Lethal/toxic service	O Yes O No

Performance data — Tube side

			In	Out
Fluid name		Temperature, °C		
Total fluid entering, kg/s		Total flow rate (liq./vap.), kg/s	/	/
Dew/bubble point, °C	/	Water/steam, kg/s	/	/
O Pour point O Freeze point, °C		Noncondensable, kg/s	/	/
Latent heat, kJ/kg		Molecular. wt. (vap./non-cond.)	/	/
Inlet pressure O kPa (ga) O kPa (abs)		Density (liq./vap.), kg/m³	/	/
Pressure drop (allow./calc.), kPa	/	Specific heat (liq./vap.), kJ/kg·K	/	/
Velocity (allow./calc.), m/s	/	Thermal conductivity (liq/vap.), W/m·K	/	/
Inside foul res., m²·K/W		Viscosity (liq./vap.), mPa·s	/	/

Performance data — Air side

Air inlet temperature (design dry bulb), °C		Face velocity, m/s	
Air flow rate/item, (kg/s) (m³/s)		Min. design ambient temp., °C	
Mass velocity (net free area), kg/s·m²		Altitude, m	
Air outlet temperature, °C		Static pressure, kPa	
Air flowrate/fan, m³/s			

Design, materials and construction

Design pressure, kPa (ga)		**Heating coil**	
Test pressure, kPa (ga)		No. of tubes ______ O.D., mm	
Design temperature, °C		Tube material	
Min. design metal temperature, °C		Fin material and type	
		Thickness, mm	
Tube bundle		Pressure design code	
Size (W × L), m		Stamp?	O Yes O No
No./bay ______ No. of tube rows		Heating fluid ______ Flow, kg/s	
Bundles in parallel ______ In series		Temperature (in/out), °C	/
Structure mounting	O Grade O Pipe rack O Other	Inlet pressure, kPa (ga)	
Pipe-rack beams (distance C-C)		Pressure drop (allow./calc.), kPa	/
Ladders, walkways, platforms	O Yes O No	Design temp., °C, des. press., kPa (ga)	/
Structure surf. prep./coating		Inlet/outlet nozzle, DN	/
Header surf. prep./coating		**Header**	
Louvre		Type	
Material		Material	
Action control:	O Auto O Manual	Corr. allow., mm	
Action type:	O Opposed O Parallel	No. of passes*	

* Give tube count of each pass if irregular.

AIR-COOLED HEAT EXCHANGER DATA SHEET (SI UNITS)

Job No.	______	Item No.	______
Page	2 of 2	By	______
Date	______	Revision	______
Proposal No.	______	Contract No.	______
Inquiry No.	______	Order No.	______

Header (continued)

Slope, mm/m ______

Plug material

Gasket material

Nozzle	No.	Size, DN	Rating and facing
Inlet	____	____	______
Outlet	____	____	______
Vent	____	____	______
Drain	____	____	______

Misc. conn's: TI ______ PI ______

Chemical cleaning ______

Min. wall thickness, mm ______

Tube

Material ______

O.D., mm. ______ Min. wall thickness, mm. ______

No./bundle ______ Length, m ______

Pitch, mm ______

Layout ______

Fin

Type ______

Material ______

Stock thickness, mm ______

Selection temperature, °C ______

O.D., mm. ______ No./m ______

Customer specification ______

______

______

______

______

Mechanical equipment

Fan

Manufacturer & model ______

No./bay ______ Speed, r/min ______

Diameter, m ______ No. of blades ______

Angle ______

Pitch adjustment: O Manual O Auto

Blade material ______ Hub material ______

kW/fan.@des.temp. ______ @min.amb. ______

Max. allow./calc.tip speed, m/s ______/______

Driver

Type

Manufacturer & model ______

No./bay ______ Driver kW ______

Speed, r/min ______ Service factor ______

Enclosure ______

Volt ______ Phase ______ Cycle ______

Fan noise level (allow./calc.), dB(A), @m ______/______

Speed reducer

Type ______

Manufacturer & model ______

No./bay ______

Service factor ______ Speed ratio ______/1

Support: O Structure O Pedestal

Vib. switch: O Yes O No

Enclosure ______

Controls air-side

Air recirculation: O None O Internal O External

Over: O Side O End

Degree control of outlet process temp.

(max. cooling), +/– °C ______/______

Action on control signal failure ______

Fan pitch: O Minimum O Maximum O Lockup

Louvres: O Open O Close O Lockup

Actuator air supply ______

Fan: O None O Positioner O Bias relay

Louvres: O Inlet O Outlet O Bypass

Positioner: O Yes O No

Signal air pressure, kPa (ga)

From ______ To ______

From ______ To ______

Supply air pressure, kPa (ga)

Max. ______ Min. ______

Max. ______ Min. ______

Shipping

Plot area (W × L), m ______

Bundle mass, kg ______

Bay ______

Total ______

Shipping, kg ______

AIR-COOLED HEAT EXCHANGER DATA SHEET (US CUSTOMARY UNITS)				
	Job No.	____	Item No.	____
	Page	1 of 2	By	____
	Date	____	Revision	____
	Proposal No.	____	Contract No.	____
	Inquiry No.	____	Order No.	____

Manufacturer	____	Heat exchanged, BTU/h	____
Model No.	____	Surface/item-finned tube, ft^2	____
Customer	____	Bare tube, ft^2	____
Plant location	____	MTD, eff., °F	____
Service	____	Transfer rate-finned, BTU/(h·ft^2·°F)	____
Type draught O Induced O Forced		Bare tube, service, BTU/(h·ft^2·°F)	____
Bay size (W × L), ft ____	No. of bay/items ____	Clean, BTU/h·ft^2·°F	____

Basic design data

Pressure design code	____		Structural code	____	
Tube bundle code stamped	O Yes	O No	Flammable service	O Yes	O No
Heating coil code stamped	O Yes	O No	Lethal/toxic service	O Yes	O No

Performance data — tube side

			In	Out
Fluid name	____	Temperature, °F	____	____
Total fluid entering, lb/h	____	Total flow rate (liq./vap.), lb/h	___/___	___/___
Dew/bubble point, °F	____/____	Water/steam, lb/h	___/___	___/___
O Pour point O Freeze point, °F	____	Noncondensable, lb/h	___/___	___/___
Latent heat, BTU/lb	____	Molecular. wt. (vap./non-cond.)	___/___	___/___
Inlet pressure O psig O psia	____	Density (liq./vap.), lb/ft^3	___/___	___/___
Pressure drop (allow./calc.), psi	____/____	Specific heat (liq./vap.), BTU/lb·°F	___/___	___/___
Velocity (allow./calc.), ft/s	____/____	Thermal conductivity (liq/vap.), BTU/(h·ft·°F)	___/___	___/___
Inside foul res., h·ft^2·°F/BTU	____	Viscosity (liq./vap.), cP	___/___	___/___

Performance data — air side

Air inlet temperature (design dry-bulb), °F	____	Face velocity, sfpm	____
Air flowrate/item, (lb/h) (scfm)	____	Min. design ambient temp., °F	____
Mass velocity (net free area), lb/h·ft^2	____	Altitude, ft	____
Air outlet temperature, °F	____	Static pressure, inches water	____
Air flowrate/fan, acfm			

Design, materials and construction

Design pressure, psig	____	**Heating coil**	
Test pressure, psig	____	No. of tubes ____ O.D., in	____
Design temperature, °F	____	Tube material	____
Min. design metal temperature, °F	____	Fin material and type	____
		Thickness, in	____
Tube bundle		Pressure design code	____
Size (W × L), ft	____	Stamp? O Yes	O No
No./bay ____	No. of tube rows ____	Heating fluid ____	Flow, lb/h ____
Bundles in parallel ____	In series ____	Temperature (in/out), °F	____/____
Structure mounting O Grade	O Pipe rack O Other	Inlet pressure, psig	____
Pipe-rack beams (distance C-C)	____	Pressure drop (allow./calc.), psi	____/____
Ladders, walkways, platforms	O Yes O No	Design temp., °F, des. press., psig	____/____
Structure surf. prep./coating	____	Inlet/outlet nozzle, nps	____/____
Header surf. prep./coating	____	**Header**	
Louvre		Type	____
Material	____	Material	____
Action control: O Auto	O Manual	Corrosion allow., inches	____
Action type: O Opposed	O Parallel	No. of passes*	____

* Give tube count of each pass if irregular.

AIR-COOLED HEAT EXCHANGER DATA SHEET (US CUSTOMARY UNITS)	
	Job No. ____ Item No. ____
	Page 2 of 2 By ____
	Date ____ Revision ____
	Proposal No. ____ Contract No. ____
	Inquiry No. ____ Order No. ____

Header (continued)
Slope, in/ft ____
Plug material
Gasket material

Nozzle	No.	Size, NPS	Rating and facing
Inlet	____	____	____
Outlet	____	____	____
Vent	____	____	____
Drain	____	____	____

Misc. conn's: TI ____ PI ____
Chemical cleaning ____
Min. wall thickness, in ____

Tube
Material ____
O.D., in ____ Min. wall thickness, in ____
No./bundle ____ Length, ft ____
Pitch, in ____
Layout ____

Fin
Type ____
Material ____
Stock thickness, in ____
Selection temperature, °F ____
O.D., in ____ No./in ____
Customer specification ____

Mechanical equipment

Fan
Manufacturer and model ____
No./bay ____ Speed, rpm ____
Diameter, ft ____ No. of blades ____
Angle ____
Pitch adjustment: O Manual O Auto
Blade material ____ Hub material ____
kW/fan.@des.temp. ____ @min.amb. ____
Max. allow./calc.tip speed, fpm ____/____

Driver
Type
Manufacturer and model ____
No./bay ____ Driver HP ____
Speed, r/min ____ Service factor ____
Enclosure ____
Volt ____ Phase ____ Cycle ____
Fan noise level (allow./calc.), dB(A), @ft ____/____

Speed reducer
Type ____
Manufacturer & model ____
No./bay ____
Service factor ____ Speed ratio ____/1
Support: O Structure O Pedestal
Vib. switch: O Yes O No
Enclosure ____

Controls air-side

Air recirculation: O None O Internal O External
Over: O Side O End
Degree control of outlet process temp.
(max. cooling), +/– °F ____/____
Action on control signal failure ____
Fan pitch: O Minimum O Maximum O Lockup
Louvres: O Open O Close O Lockup
Actuator air supply ____
Fan: O None O Positioner O Bias relay

Louvres: O Inlet O Outlet O Bypass
Positioner: O Yes O No
Signal air pressure, psig
From ____ To ____
From ____ To ____
Supply air pressure, psig
Max. ____ Min. ____
Max. ____ Min. ____

Shipping

Plot area (W × L), ft ____
Bundle mass, lb ____
Bay ____
Total ____
Shipping, lb ____

AIR-COOLED HEAT EXCHANGER NOISE DATA SHEET	Job No. ______ Item No. ______ Page: 1 of 1 By ______ Date ______ Revision ______ Proposal No. ______ Contract No. ______ Inquiry No. ______ Order No. ______

1	**Noise data**	**Purchaser specification**	**Purchaser specification**	**Vendor guarantee**	**Vendor guarantee**
2	Octave bands centre	SPL at designated location	PWL per fan	SPL at designated location	PWL per fan
3	63				
4	125				
5	250				
6	500				
7	1 000				
8	2 000				
9	4 000				
10	8 000				
11	dB(A)				
12	Overall unit PWL				
13	dB(A)				

14	NOTE Unless otherwise specified: SPL is the sound pressure level measured in dB, reference 2×10^{-5} N/m^2. PWL is the sound power level measured in dB, reference 1×10^{-12} W. For forced-draught fans, the SPL is measured at the centreline of the fan 1 m below the inlet of the fan. For induced-draught fans, the SPL is measured 1 m below the bundles. Noise of equipment shall include noise from speed reducer and motor. The upper tolerance for noise levels is +0 dB(A). If tonal noise is present, then the specified overall noise levels shall be interpreted as 5 dB(A) more stringent.
15	Description of designated location:
16	Specification of special requirements (with/without acoustic measures, special low-noise fans):
17	
18	
19	
20	

B.2 Standardized electronic data exchange file specification

B.2.1 Scope

This clause provides a standardized file format for the electronic storage and transmittal of the data contained in the data sheets. This standard format is also known as the neutral data exchange file format.

The neutral data exchange file format allows groups with different operating systems, software, hardware, and data sheet forms to electronically exchange the data contained on a data sheet. Unlike printed data sheet forms, it is possible to import electronic data into design programs or other software systems.

Purchasers and manufacturers are encouraged to use this specification to transfer data. The method of data transfer, such as Internet FTP, e-mail, bulletin board etc., is not addressed by this specification. The parties exchanging data should agree on the transfer method.

The legal ramifications of exchanging data electronically are subject to the policies established between the data-exchanging parties. The parties may also require data sheets in paper format as legal documents.

B.2.2 File format

B.2.2.1 The neutral data exchange file is an ASCII text file. The exchange file only uses ASCII decimal codes 0 through 127, as these codes are common to multiple computer operating systems.

B.2.2.2 The data exchanged is defined in Table B.1. Each data field is separated by a carriage return (ASCII code 13) and line feed (ASCII code 10) combination. If a data field is unknown, not applicable or contains a null value, the carriage return and line-feed codes that would normally follow it shall be included in the file. The neutral data exchange file therefore has one line for each data field. The storage order shall also follow the data order defined in this specification. For example, data field number 20, defined as Item Number in this specification, will always be the 20th field, or line 20, in all neutral data exchange files.

B.2.2.3 Table B.1 defines the maximum length for each data field. Data exceeding the defined maximum length may not be processed correctly by the receiver's program.

B.2.2.4 Each data field in this specification is characterized as numeric, integer, date or character, in order to facilitate data-processing by the receiver's program. These field types are shown in the "Data Type" column of Table B.1 as follows:

a) "C" is a character field. The contents are any ASCII character from 0 to 127.

b) "D" is a date field. Dates are in the form YYYYMMDD, with YYYY representing the year, MM the month, and DD the day. For example, April 1, 1996 is represented as 19960401.

c) "I" is an integer field.

d) "N" is a numeric field. The contents are numeric values expressed in integer, floating point, or exponential format.

Data not conforming to the assigned data type may not be processed correctly by the receiver's program.

B.2.2.5 The units of measure for numeric data are defined for SI units (with US Customary units in brackets) in the "Units/Contents" column of Table B.1. The neutral file shall conform to one of these two measurement systems. Data field number 3, System of Units, defines the measurement system used in the neutral file.

B.2.2.6 Character type data which have well-defined options shall conform to a common nomenclature so they can be easily processed by the receiver's programs. The column "Units/Contents" in Table B.1 defines the nomenclature for these data fields. Each option is separated by a semicolon (";") in this column. For example, a raised face weld neck flange can be expressed many different ways on a data sheet. To facilitate the correct interpretation by the receiver's program, the sender must use the terminology "RFWN" to represent raised face weld neck in the neutral data exchange file (reference data field number 94). If the sender's data does not match one of the defined options, the field contents should be set to “OTHER” and the data output with a description to one of the remark fields.

B.2.2.7 Each item number, or data sheet, has its own neutral data exchange file.

B.2.2.8 The use of a standard file naming convention minimizes the possibility of duplicate file names and facilitates identification of file contents. The file naming convention used for the neutral data exchange file should be agreed to by the data-exchanging parties. However the naming convention shown below is recommended because it is supported by many operating systems. It is based on the DOS naming convention, which allows names up to eight characters in length with an additional three-character extension.

a) 1 to 6 characters for equipment item number. If the item number is longer, use the last 6 characters, excluding hyphens, commas, etc.;

b) +1 character for revision number;

c) +1 character for sequence number, such as alternative-design identifier. Use ‘0’ (zero) if no sequence number;

d) +‘.’

e) +3-character mnemonic representing sender’s name.

For example, if the ABC Manufacturing Company sent a neutral exchange file to Company X, and it contained the data for Item 20-E-43089, data sheet revision B, then the file name would be “E43089B0.ABC”. If they also submitted alternative design number 1 for this item number and revision, then the file name for it would be “E43089B1.ABC”.

B.2.3 Revisions to neutral data exchange file format

This specification defines version 1.0 of the neutral data exchange file format. Data field number 2, File Format Version Number, is currently equal to “V1.0”. Each revision to this specification will have its own unique File Format Version Number. The data-exchanging parties are responsible for maintaining compatibility with the latest format version.

Table B.1 — Neutral data exchange file specification

Data No.	Description	Data width	Data type	Units/contents	
1	File identifier	20	C	Air-cooled exchanger	
2	Neutral file format version number	10	C	V1.0	
3	System of units	2	C	SI	(US)
4	Vendor's name	30	C		
5	Model number	30	C		
6	Alternative design identifier	1	C	This field is blank for base designs, and is only used when transmitting an alternative design, e.g. A, B, or 1	
7	Purchaser's name	100	C		
8	Purchaser's job number	20	C		
9	Purchaser's reference number	40	C		
10	Purchaser's inquiry number	30	C		
11	Purchaser's purchase order	40	C		
12	Vendor's job number	20	C		
13	Vendor's reference number	40	C		
14	Vendor's proposal number	30	C		
15	Contact or sender's name	30	C		
16	Plant location	60	C		
17	Revision date	8	D	YYYYMMDD	
18	Revision	2	C		
19	Service of unit	60	C		
20	Item Number	50	C		
21	Tube-side fluid name	25	C		
22	Tube-side total flow	13	N	kg/s	(lb/h)
23	Tube-side vapour flow in	13	N	kg/s	(lb/h)
24	Tube-side vapour flow out	13	N	kg/s	(lb/h)
25	Tube-side liquid flow in	13	N	kg/s	(lb/h)
26	Tube-side liquid flow out	13	N	kg/s	(lb/h)
27	Tube-side steam flow in	13	N	kg/s	(lb/h)
28	Tube-side steam flow out	13	N	kg/s	(lb/h)
29	Tube-side water flow in	13	N	kg/s	(lb/h)
30	Tube-side water flow out	13	N	kg/s	(lb/h)
31	Tube-side non-condensable flow in	13	N	kg/s	(lb/h)
32	Tube-side non-condensable flow out	13	N	kg/s	(lb/h)
33	Tube-side temperature in	13	N	°C	(°F)
34	Tube-side temperature out	13	N	°C	(°F)
35	Tube-side liquid density in	13	N	kg/m^3	(lb/ft^3)
36	Tube-side liquid density out	13	N	kg/m^3	(lb/ft^3)
37	Tube-side vapour density in	13	N	kg/m^3	(lb/ft^3)
38	Tube-side vapour density out	13	N	kg/m^3	(lb/ft^3)
39	Tube-side liquid viscosity in	13	N	mPa·s	(cP)
40	Tube-side liquid viscosity out	13	N	mPa·s	(cP)

Data No.	Description	Data width	Data type	Units/contents	
41	Tube-side vapour viscosity in	13	N	mPa·s	(cP)
42	Tube-side vapour viscosity out	13	N	mPa·s	(cP)
43	Tube-side vapour molecular weight in	13	N		
44	Tube-side vapour molecular weight out	13	N		
45	Tube-side non-condensable molecular weight in	13	N		
46	Tube-side non-condensable molecular weight out	13	N		
47	Tube-side liquid specific heat in	13	N	kJ/kg·K	(BTU/lb·°F)
48	Tube-side liquid specific heat out	13	N	kJ/kg·K	(BTU/lb·°F)
49	Tube-side vapour specific heat in	13	N	kJ/kg·K	(BTU/lb·°F)
50	Tube-side vapour specific heat out	13	N	kJ/kg·K	(BTU/lb·°F)
51	Tube-side liquid thermal conductivity in	13	N	W/m·K	(BTU/h·ft·°F)
52	Tube-side liquid thermal conductivity out	13	N	W/m·K	(BTU/h·ft·°F)
53	Tube-side vapour thermal conductivity in	13	N	W/m·K	(BTU/h·ft·°F)
54	Tube-side vapour thermal conductivity out	13	N	W/m·K	(BTU/h·ft·°F)
55	Tube-side latent heat	13	N	kJ/kg	(BTU/lb)
56	Dew point	13	N	°C	(°F)
57	Bubble point	13	N	°C	(°F)
58	Pour point	13	N	°C	(°F)
59	Freeze point	13	N	°C	(°F)
60	Tube-side pressure in	13	N	kPa (ga)	(psig)
61	Tube-side allowable velocity	13	N	m/s	(ft/s)
62	Allowable velocity minimum or maximum	3	C	min; max	
63	Tube-side calculated velocity	13	N	m/s	(ft/s)
64	Tube-side pressure drop allow	13	N	kPa	(psi)
65	Tube-side pressure drop calculate	13	N	kPa	(psi)
66	Tube-side fouling resistance	13	N	$m^2·K/W$	($h·ft^2·°F/BTU$)
67	Air flowrate/item, mass	13	N	kg/s	(lb/h)
68	Air flowrate/item, volumetric	13	N	m^3/s	(SCFM)
69	Mass velocity (net free area)	13	N	$kg/s·m^2$	($lb/h·ft^2$)
70	Air flowrate per fan, actual conditions	13	N	m^3/s	(ACFM)
71	Face velocity	13	N	m/s	(SFPM)
72	Air inlet temperature	13	N	°C	(°F)
73	Air outlet temperature	13	N	°C	(°F)
74	Minimum design ambient	13	N	°C	(°F)
75	Altitude	13	N	m	(ft)
76	Static pressure	13	N	kPa	(inches water)
77	Air-side fouling resistance	13	N	$m^2·K/W$	($h·ft^2·°F/BTU$)
78	Heat exchanged	13	N	W	(BTU/h)
79	MTD	13	N	°C	(°F)
80	MTD type: corrected or weighted	4	C	Corr; WTD	
81	Transfer rate, finned	13	N	$W/m^2·K$	($BTU/h·ft^2·°F$)
82	Bare tube rate, service	13	N	$W/m^2·K$	($BTU/h·ft^2·°F$)

Data No.	Description	Data width	Data type	Units/contents	
83	Bare tube rate, clean	13	N	W/m^2·K	(BTU/h·ft^2·°F)
84	Tube-side design pressure	13	N	kPa (ga)	(psig)
85	Tube-side vacuum pressure (include negative sign)	13	N	kPa (ga)	(psig)
86	Tube-side test pressure	13	N	kPa (ga)	(psig)
87	Tube-side minimum design metal temperature	13	N	°C	(°F)
88	Tube-side maximum design temperature	13	N	°C	(°F)
89	Number of tube passes	35	C	include tube count of each pass if irregular	
90	Tube-side corrosion allowance	13	N	mm	(inch)
91	Number of tube-side inlet connections	5	I		
92	Tube-side inlet connection size	13	N	mm	(inch)
93	Tube-side inlet connection rating	5	I	150; 300; 600; 900; 1 500; 2 500	
94	Tube-side inlet connection facing	6	C	RFWN; RFLWN; RFSO; LJ; RTJWN; RTJLWN; BW; FFWN; FFLWN; OTHER	
95	Number of tube-side outlet connections	5	I		
96	Tube-side outlet connection size	13	N	mm	(inch)
97	Tube-side outlet connection rating	5	I	150; 300; 600; 900; 1 500; 2 500	
98	Tube-side outlet connection facing	6	C	RFWN; RFLWN; RFSO; LJ; RTJWN; RTJLWN; BW; FFWN; FFLWN; OTHER	
99	Number of tube-side vent connections	5	I		
100	Tube-side vent connection size	13	N	mm	(inch)
101	Tube-side vent connection rating	5	I	150; 300; 600; 900; 1 500; 2 500; 3 000; 6 000	
102	Tube-side vent connection facing	7	C	RFWN; RFLWN; RFSO; LJ; RTJWN; RTJLWN; BW; FFWN; FFLWN; CPLG; THDOLET; NPT; SCOLET; DRL&TAP; WLDBOSS; OTHER	
103	No. of Tube-side drain connections	5	I		
104	Tube-side drain connection size	13	N	mm	(inch)
105	Tube-side drain connection rating	5	I	150; 300; 600; 900; 1 500; 2 500; 3 000; 6 000	
106	Tube-side drain connection facing	7	C	RFWN; RFLWN; RFSO; LJ; RTJWN; RTJLWN; BW; FFWN; FFLWN; CPLG; THDOLET; NPT; SCOLET; DRL&TAP; WLDBOSS; OTHER	
107	No. of T.I. connections	5	I		
108	T.I. connection size	13	N	mm	(inch)
109	T.I. connection rating	5	I	150; 300; 600; 900; 1 500; 2 500; 3 000; 6 000	
110	T.I. connection facing	7	C	RFWN; RFLWN; RFSO; LJ; RTJWN; RTJLWN; BW; FFWN; FFLWN; CPLG; THDOLET; NPT; SCOLET; DRL&TAP; WLDBOSS; OTHER	
111	No. of P.I. connections	5	I		
112	P.I. connection size	13	N	mm	(inch)
113	P.I. connection rating	5	I	150; 300; 600; 900; 1 500; 2 500; 3 000; 6 000	

Data No.	Description	Data width	Data type	Units/contents	
114	P.I. connection facing	7	C	RFWN; RFLWN; RFSO; LJ; RTJWN; RTJLWN; BW; FFWN; FFLWN; CPLG; THDOLET; NPT; SCOLET; DRL&TAP; WLDBOSS; OTHER	
115	No. of chemical cleaning connections	5	I		
116	Chemical cleaning connection size	13	N	mm	(inch)
117	Chemical cleaning connection rating	5	I	150; 300; 600; 900; 1 500; 2 500; 3 000; 6 000	
118	Chemical cleaning connection facing	7	C	RFWN; RFLWN; RFSO; LJ; RTJWN; RTJLWN; BW; FFWN; FFLWN; CPLG; THDOLET; NPT; SCOLET; DRL&TAP; WLDBOSS; OTHER	
119	Minimum wall thickness, nozzle	13	N	mm	(inch)
120	Bay size (W x L)	20	C	m	(ft)
121	Draught type	7	C	Forced; induced	
122	No. of bays/item	5	I		
123	Bundles per bay	5	I		
124	Bundle size (W x L)	20	C	m	(ft)
125	Number of bundles connected in parallel	5	I		
126	Number of bundles connected in series	5	I		
127	Surface/item, finned	13	N	m^2	(ft^2)
128	Surface/item, bare tube	13	N	m^2	(ft^2)
129	Number of tubes/bundle	5	I		
130	Number of tube rows	5	I		
131	Tube outside diameter	13	N	mm	(inch)
132	Tube wall thickness	13	N	mm	(inch)
133	Type tube wall thickness	3	C	Avg; Min	
134	Tube length	13	N	m	(ft)
135	Tube pitch	13	N	mm	(inch)
136	Tube pattern	10	C	TRIANGULAR; SQUARE; OTHER	
137	Fin type	10	C	EMBEDDED; EXTRUDED; OVERLAPPED, FOOTED; NONE; OTHER	
138	Number of fins	13	N	m^{-1}	(per inch)
139	Tube material	20	C		
140	Tube to tubesheet joint	24	C	ROLLED; FULL ROLLED; SEAL WELDED & ROLLED; STRENGTH WELDED; STRENGTH WELDED & ROLLED; OTHER	
141	Fin material	30	C		
142	Fin stock thickness	13	N	mm	(inch)
143	Fin outside diameter	13	N	mm	(inch)
144	Fin selection temperature	13	N	°C	(°F)
145	Header type	40	C		
146	Header material	20	C		
147	Plug material	20	C		
148	Gasket material	20	C		

Data No.	Description	Data width	Data type	Units/contents	
149	Bundle slope	13	N	mm/m	(in/ft)
150	Pipe-rack beams, C-C	15	C	m	(ft)
151	Structure mounting	8	C	Grade; piperack; other	
152	Ladders, walkways, platforms	3	C	Yes; no	
153	Structure surf. prep.	40	C		
154	Header surf. prep.	40	C		
155	Fan manufacturer & model	50	C		
156	No. fans/bay	5	I		
157	Fan rotational speed	13	N	r/min	(rpm)
158	Fan diameter	13	N	m	(ft)
159	No. of blades	5	I		
160	Pitch adjustment	6	C	Manual; auto; fixed; other	
161	Percent automatic pitch	13	N	%	
162	Blade angle	13	N		
163	Blade material	20	C		
164	Hub material	20	C		
165	Power/fan at design temp.	13	N	kW	(bhp)
166	Power/fan at min. ambient	13	N	kW	(bhp)
167	Maximum allowable tip speed	13	N	m/s	(ft/min)
168	Calculated tip speed	13	N	m/s	(ft/min)
169	Fan noise level, allowable	100	C		
170	Fan noise level, calculated	100	C		
171	Driver type	15	C	Electric motor; hydraulic motor; steam turbine; other	
172	Driver manufacturer and model	40	C		
173	No. drivers/bay	5	I		
174	Driver power	13	N	kW	(bhp)
175	Driver rotational speed	13	N	r/min	(rpm)
176	Service factor	13	N		
177	Enclosure	30	C		
178	Voltage	3	I	V	(volt)
179	Phase	1	I		
180	Frequency	2	I	Hz	
181	Speed reducer type	16	C	V-BELT; COG BELT; RIGHT ANGLE GEAR; DIRECT; OTHER	
182	Manufacturer and model	30	C		
183	Speed reducer, no./bay	5	I		
184	Service factor	13	N		
185	Speed ratio	10	C		
186	Speed reducer support	9	C	Structure; pedestal; other	
187	Vibration switch	3	C	Yes; no	
188	Vib. switch enclosure	30	C		
189	Louvre material	20	C		
190	Louvre action control	6	C	Manual; auto	

Data No.	Description	Data width	Data type	Units/contents	
191	Louvre action type	8	C	Opposed; parallel	
192	Heating coil, no. of tubes	5	I		
193	Tube outside diameter	13	N	mm	(in)
194	Tube material	20	C		
195	Fin material and type	30	C		
196	Fin thickness	13	N	mm	(in)
197	Heating coil, code stamp	3	C	Yes; no	
198	Heating fluid	20	C		
199	Heating fluid flow	13	N	kg/s	(lb/h)
200	Heating fluid temp. in	13	N	°C	(°F)
201	Heating fluid temp. out	13	N	°C	(°F)
202	Inlet pressure	13	N	kPa (ga)	(psig)
203	Pressure drop allowed	13	N	kPa	(psi)
204	Pressure drop calculated	13	N	kPa	(psi)
205	Heating coil design temp.	13	N	°C	(°F)
206	Heating coil design pressure	13	N	kPa (ga)	(psig)
207	Inlet nozzle size	13	N	mm	(inch)
208	Outlet nozzle size	13	N	mm	(inch)
209	Air recirculation	18	C	None; internal; external over side; external over end; other	
210	Degree of control of outlet process temperature, (+)	13	N	°C	(°F)
211	Degree of control of outlet process temperature, (-)	13	N	°C	(°F)
212	Action of fan pitch on control signal failure	7	C	Minimum; maximum; lockup	
213	Action of louvres on control signal failure	6	C	Open; close; lockup	
214	Actuator air supply	20	C		
215	Fan actuator	10	C	None; positioner; bias relay	
216	Louvre location	6	C	Inlet; outlet; bypass	
217	Louvre positioner	3	C	Yes; no	
218	Louvres: Signal air pressure, from	13	N	kPa (ga)	(psig)
219	Louvres: Signal air pressure, to	13	N	kPa (ga)	(psig)
220	AV Fans: Signal air pressure, from	13	N	kPa (ga)	(psig)
221	AV Fans: Signal air pressure, to	13	N	kPa (ga)	(psig)
222	Louvres: Supply air pressure, max.	13	N	kPa (ga)	(psig)
223	Louvres: Supply air pressure, min.	13	N	kPa (ga)	(psig)
224	AV Fans: Supply air pressure, max.	13	N	kPa (ga)	(psig)
225	AV Fans: Supply air pressure, min.	13	N	kPa (ga)	(psig)
226	Tube bundle code stamp	3	C	Yes; no	
227	Plot space (W × L)	20	C	m	(ft)
228	Mass per bay	13	N	kg	(lb)
229	Mass per bay filled with water	13	N	kg	(lb)
230	Mass of bundle	13	N	kg	(lb)
231	Total Mass	13	N	kg	(lb)

Data No.	Description	Data width	Data type	Units/contents	
232	Shipping Mass	13	N	kg	(lb)
233	Remarks	100	C		
234	Remarks	100	C		
235	Remarks	100	C		
236	Remarks	100	C		
237	Remarks	100	C		
238	Remarks	100	C		
239	Remarks	100	C		
240	Customer specifications	100	C		
241	Pressure Design Code	50	C		
242	Structural Code	50	C		
243	Service flammable	3	C	Yes; no	
244	Service lethal/toxic	3	C	Yes; no	

Annex C
(informative)

Winterization of air-cooled heat exchangers

C.1 General

C.1.1 Scope

This annex presents design features and other considerations that contribute to satisfactory functioning of air-cooled heat exchangers operating with low-temperature inlet air. Data related to structural materials for low-temperature service and safety precautions related to the accumulation of snow and ice are not included.

C.1.2 Definitions of terms

Terms used in this annex are defined, for information only, in C.1.2.1 through C.1.2.10.

C.1.2.1
winterization
provision of design features, procedures, or systems for air-cooled heat exchangers to avoid problems with the process fluid as a result of low-temperature inlet air. Problems related to low-temperature inlet air include fluid freezing, cooling to the pour point, wax formation, hydrate formation, laminar flow, and condensation at the dew point (which may initiate corrosion)

C.1.2.2
inlet air
atmospheric or ambient air that enters the air-cooled heat exchanger

C.1.2.3
exhaust air
air that is discharged from the air-cooled heat exchanger to the atmosphere

C.1.2.4
recirculated air
air that has passed through the process bundle and is redirected to mix with and heat the inlet air

C.1.2.5
external recirculation
process that uses an external duct to carry recirculated air to mix with and heat the inlet air

C.1.2.6
internal recirculation
process that uses fans (possibly with louvres) to recirculate air from one part of the process bundle to the other part

C.1.2.7
minimum design air temperature
specified inlet air temperature to be used for winterization

C.1.2.8
critical process temperatures
temperatures related to important physical properties of a process stream, such as freezing point, pour point, cloud point, hydrate formation temperature and dew point

C.1.2.9
specified minimum tube-wall temperature
critical process temperature plus a safety margin

C.1.2.10
wind skirt
vertical barrier either above or below an air-cooled heat exchanger that minimizes the effect of wind

C.2 Winterization problem areas

C.2.1 General

The purposes of this clause are as follows:

a) to identify reasons for winterization of air-cooled heat exchangers;

b) to outline general design data requirements for winterization and guidelines for supplying such data;

c) to review heat losses and general problem areas;

d) to establish process categories that may require winterization and safety margins for each category.

C.2.2 Reasons for winterization

Winterization is generally applied to maintain the tube-wall temperature at or above a specified minimum tube-wall temperature to prevent operating problems. The specified minimum tube-wall temperature is the point at which the tube-wall temperature approaches the fluid's critical process temperature. Critical process temperatures include the freezing point, pour point, wax point, dew point (if condensation causes corrosion), hydrate formation point and any other temperature at which operating difficulties may occur.

C.2.3 General design data requirements

The following data should be determined and specified to the designer:

a) the specified minimum tube-wall temperature, which should include a safety margin as discussed in C.2.7;

b) the minimum design air temperature;

c) all alternative process conditions. including reduced flow (turndown) operations;

d) the design wind velocity and the prevailing wind direction;

e) the availability of steam or another source of heat for start-up in cold weather. (If steam is available, the steam pressure should be specified.)

C.2.4 Heat losses

The effect of heat losses (by conduction and convection, louvre leakage and natural draught) on the tube-wall temperature during start-up, shutdown and standby operating conditions should be considered when the requirements for a heating coil are determined.

C.2.5 General problem areas

The exit fluid temperature for any single row of any given pass may not be the same as the average exit fluid temperature for that pass. To avoid potential problems in the field, the exit fluid temperature for each row should be calculated separately to determine the lowest tube-wall temperature.

For critical services, it may be desirable to monitor the tube-wall temperature in the coldest zone. This may be done by installing thermocouples at critical points.

Maldistribution of the process fluid or airstream may also cause problems that should be considered in the design of the equipment.

C.2.6 Process categories

C.2.6.1 General

Most winterization problems fall into one of the following six categories:

a) problems with water and dilute aqueous solutions;

b) problems with total steam condensers;

c) problems with partial steam condensers;

d) problems with condensing process fluids containing steam with or without noncondensables;

e) problems with viscous fluids and fluids with high pour points;

f) problems resulting from freezing, hydrate formation, and corrosion caused by condensate.

These categories and the ways in which they apply to typical operating cases are described in C.2.6.2 through C.2.6.7.

C.2.6.2 Category 1: water and dilute aqueous solutions

Water and dilute aqueous solutions have high tube-side heat transfer coefficients, resulting in relatively high tube-metal temperatures. When these fluids are present, simple winterization systems, such as airflow control systems, are indicated. Start-up and shutdown at extremely low temperatures may require additional measures.

C.2.6.3 Category 2: total steam condensers

Total steam condensers that are single pass may be subject to a backflow of steam from the outlet end of the upper (hotter) tube rows into the outlet end of the lower (colder) tube rows. This usually leads to noncondensable contaminants collecting near the outlet end of the colder tubes. The presence of noncondensables results in diminished performance and in subcooling and possible freezing of condensate in the colder tubes. Corrosion may also occur.

In quite a few installations, a particular set of conditions has caused rapid perforation of tube walls. The perforations occur near the exit end of the lower (colder) tube rows. When this happens, a repetitive knocking or clicking noise called *water hammer* is always present. These failures, which have occurred in numerous locations, have the following common characteristics:

a) one pass with four or more rows of tubes whose outside diameter is 25,4 mm (1 inch) and whose length is 11 m to 16 m (36 ft to 52 ft);

b) inlet steam pressure between 0 kPa and 170 kPa gauge (0 psig and 25 psig).

The perforations have occurred as quickly as within 1 d of service on tubes with a wall thickness of 0,89 mm (0,035 inch) and as slowly as three months on tubes with a wall thickness of 2,11 mm (0,083 inch). The rapidity of failure appears to be related to the severity of the water hammer.

Measures to prevent this type of failure are all aimed at reducing or eliminating the quantity of steam back-flowing into the colder tubes. For instance, in a four-row, one-pass condenser, limiting the tube length to 360 times the tube outside diameter seems to suffice [for example, a length of 9 m (30 ft) for tubes with an outside diameter of 25,4 mm (1 inch)]. Alternatively, the rear header can be separated into four non-communicating compartments with drains provided for each compartment. Another method is to use restriction orifices in the tube inlets; however, this measure may not be completely effective at all flowrates.

C.2.6.4 Category 3: partial steam condensers

In Category 3 process streams, the quantity of outlet vapour is large enough that backflow cannot occur and steam exits continuously from the outlet ends of all tube rows. The quantity of outlet vapour is typically 10 % to 30 % by mass of the total inlet flow. Outlet quantities below 10 % by mass are characteristic of Category 2 condensers. The exact quantity of outlet vapour should be established by calculation, with consideration given to the mode of operation at the minimum ambient temperature. If calculations show that backflow will not occur, simple winterization systems, such as airflow control, are indicated. If calculations indicate that backflow will occur, moderate to extensive protection systems may be indicated.

C.2.6.5 Category 4: condensing process fluids containing steam with or without noncondensables

Category 4 is an extension of Category 3. Category 4 highlights the effects of other condensables on the tube-wall temperature. Prediction of the tube-side flow regime is essential for an accurate evaluation of tube-wall and fluid temperatures. Consider, for example, a stream containing steam, condensable hydrocarbons and non-condensables. Annular flow may exist at the condenser inlet, with a liquid hydrocarbon annulus being formed on the cold tube wall and surrounding a gas core. Stratified flow may exist at the condenser outlet, with water and liquid hydrocarbons draining from the bottom of the tube while steam condenses on the upper portion. Simple winterization systems are usually indicated when these conditions are present.

C.2.6.6 Category 5: viscous fluids and fluids with high pour points

When a viscous fluid is flowing through a number of parallel paths, local variations in cooling may cause a drastic reduction in velocity in some of the flow paths. This phenomenon is called *unstable flow.* Unstable flow is caused when, under certain conditions of bulk viscosity, wall viscosity and pressure drop, the increase in pressure drop resulting from a higher viscosity (caused by the additional cooling allowed by a lower velocity) offsets the decrease in pressure drop resulting from the lower velocity. This can occur only when the fluid is in laminar flow.

When unstable flow occurs, the velocities in parallel tubes within a pass can differ by as much as 5:1. As a result, the exchanger's overall tube-side pressure drop may increase by up to 100 % and the heat removal may decrease to less than 50 % of that possible if the fluid were equally distributed among the tube paths. This flow maldistribution is a major factor in many cases of diminished performance of viscous and high-pour-point fluid coolers.

At present, only general guidelines exist for avoiding such maldistribution. These guidelines are as follows.

a) The bulk viscosity of the process fluid at the outlet temperature should not exceed 50 mPa·s (50 cP).

b) The ratio of wall viscosity to bulk viscosity should not exceed 3:1.

The following additional factors should be given extra emphasis in both design and fabrication for this type of service:

a) Air-side flow distribution and temperature distribution should be as uniform as possible. External recirculation over only one side may cause nonuniform airflow and air temperature to the bundle.

b) Air bypassing the bundle between the side frames and tubes should be minimized by conforming to a maximum gap of 10 mm ($^3/_8$ inch) as specified in 7.1.1.8.

c) Allowable process fluid pressure drop should be high. Pressure drops of 275 kPa (40 psi) or higher are common.

d) Tube-side flow should be uniformly distributed within the headers. This may require additional nozzles and/or external insulation of the headers.

There may be cases in which successful operation can be achieved while violating these guidelines. However, when successful experience is lacking, it is risky to ignore these recommendations. Alternative designs that should be considered include indirect systems and air-cooled heat exchangers with serpentine coils.

C.2.6.7 Category 6: freezing point, hydrate formation point and dew point

Category 6 process streams are characterized by a discrete critical process temperature. For such streams, the calculation of wall and fluid temperatures tends to be straightforward. Depending on design conditions, recommended winterization systems include the full range outlined in C.3.

C.2.7 Safety margins

So many variables are involved in the process streams described in C.2.6.2 through C.2.6.7 that establishing a fixed safety margin (the tube-wall temperature minus the critical process temperature) is difficult. Each problem should be analyzed on an individual basis.

In the absence of more specific information, the safety margins given in Table C.1 should be added to the critical process temperature to determine the specified minimum tube-wall temperature.

Table C.1 — Safety margins for different process categories

Category	Safety margin °C (°F)
1	8,5 (15)
2	8,5 (15)
3	8,5 (15)
4	8,5 (15)
5	14 (25)
6	11 (20)

NOTE See C.2.6 for description of categories.

C.3 Methods of winterization

C.3.1 Airflow and air temperature control systems

C.3.1.1 System A: airflow control

System A generally uses automatically controlled variable-pitch fans, as shown in Figure C.1, and/or automatically or manually controlled louvres, as shown in Figure C.2, to control airflow.

Automatically controlled variable-pitch fans offer the following advantages over louvres:

a) better airflow control, providing more sensitive control of process temperatures at or near design conditions;

b) lower power requirements at reduced ambient temperatures .

Automatically controlled variable-pitch fans have the following disadvantages:

a) less precise airflow control when the required airflow is less than 30 % of the full airflow;

b) more sensitivity to wind effects at lower airflows.

Louvres offer the following advantages over automatically controlled variable-pitch fans:

a) more precise airflow control when the required airflow is less than 30 % of the full airflow;

b) less sensitivity to wind effects;

c) capability of full closure for warming the unit at start-up and shutdown.

Louvres have the following disadvantages:

a) less precise control when the required airflow is more than 30 % of the full airflow;

b) potentially inoperable linkages as a result of ice, snow, corrosion or wear.

In addition to the systems shown in Figures C.1 and C.2, airflow can be controlled by using variable-speed drives. Airflow control is used primarily to control process temperatures and offers the least winterization protection.

C.3.1.2 System B: airflow control plus air temperature control using a noncontained internal recirculation system

In System B, the automatically controlled variable-pitch fan near the process outlet reverses airflow when the inlet air temperature is low. The air heated from flow over the tubes enters a zone beneath the tube bundle that is protected to some degree from wind effects by downward-projecting wind skirts. Part of the heated air is then mixed with inlet air as shown in Figures C.3 and C.4. This system may be subject to uneven air mixing below the tube bundle and does not provide a positive method of controlling the mixed air inlet temperature. In addition, wind may adversely affect the circulation of the hot air. Care should be taken in selecting mechanical equipment installed below the downflow fan because of the higher air temperature. This system is not generally recommended but has been used for heat exchangers requiring moderate winterization protection.

C.3.1.3 System C: airflow control plus air temperature control using a contained internal recirculation system

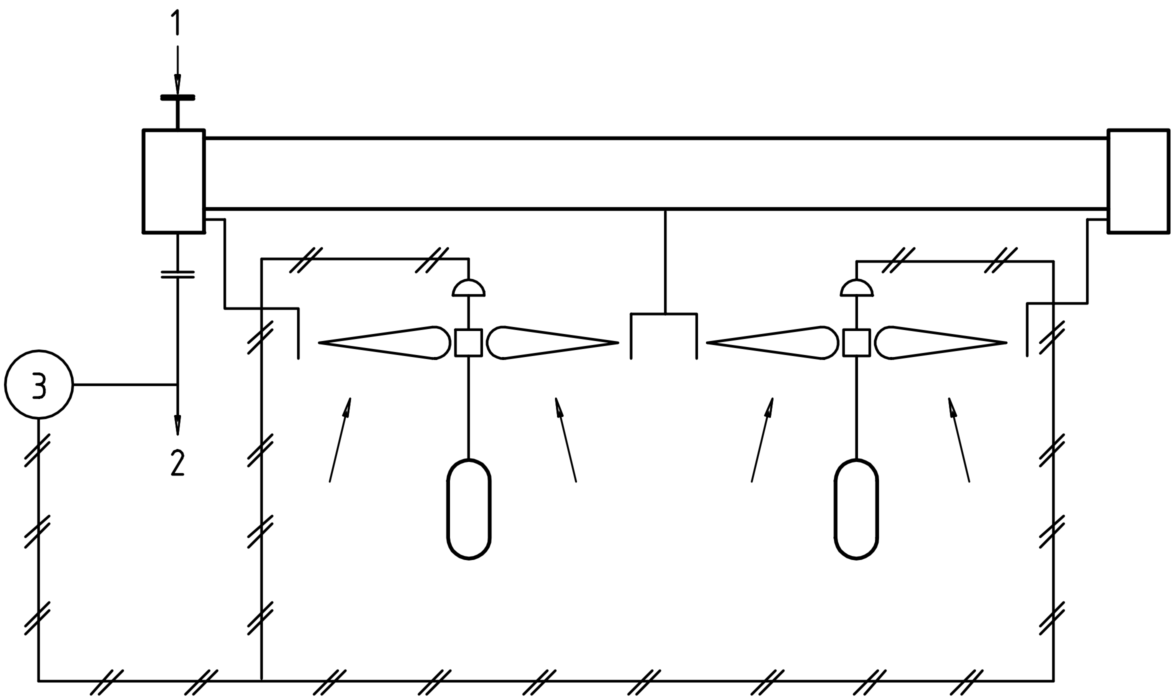
In System C, the automatically controlled variable-pitch fan near the process outlet reverses when the inlet air temperature is low to direct airflow downward while the exhaust louvres partially close, as shown in Figure C.5. Simultaneously, the vertical bypass louvres above the tube bundle open to redirect part of the exhaust air along the length of the tube bundle. This air is mixed above the downdraught side of the tube bundle with incoming ambient air. Only enough air is directed through the bypass louvres to ensure that the mixed air temperature above the downdraught fan is above a preset level. For certain design cases, wind skirts may be required below the tube bundle. The disadvantage of this system is that areas of the bundle may be exposed to low air temperatures as a result of uneven air mixing. Care should be taken in selecting mechanical equipment installed on and below the downflow fan because of the higher air temperature. This system offers an additional degree of winterization protection, compared with the systems described in C.3.1.1 and C.3.1.2.

C.3.1.4 System D: airflow control plus air temperature control using an external recirculation system

In System D, hot exhaust air is recirculated through an external recirculation duct to be mixed with inlet air when the inlet air temperature is low. The amount of air recirculated and the temperature of the mixed stream are controlled by partially closing the exhaust louvres while modulating the inlet and bypass louvres. This system normally includes a floor so that the unit is completely enclosed, thus providing positive control of the entering airflow. Although the temperature of the process fluid can be controlled by louvre action alone, automatically controlled variable-pitch fans may be used to control the process temperature more precisely. Automatically controlled variable-pitch fans may also enable the user to reduce the fan power required at lower ambient temperatures.

Figure C.6 illustrates recirculation over both sides of the unit. Some units may have a recirculation duct over one side only. Alternatively, a recirculation duct may be placed at one or both ends of the unit to minimize the width of the bay or to provide an enclosed heated area for headers and header walkways. In addition, various combinations and locations of inlet louvres can be used to maximize mixing of hot and cold airstreams.

This system provides maximum winterization protection, compared with the systems described in C.3.1.1 through C.3.1.3.



NOTE The air-cooled heat exchanger may be of the forced- or induced-draught type.

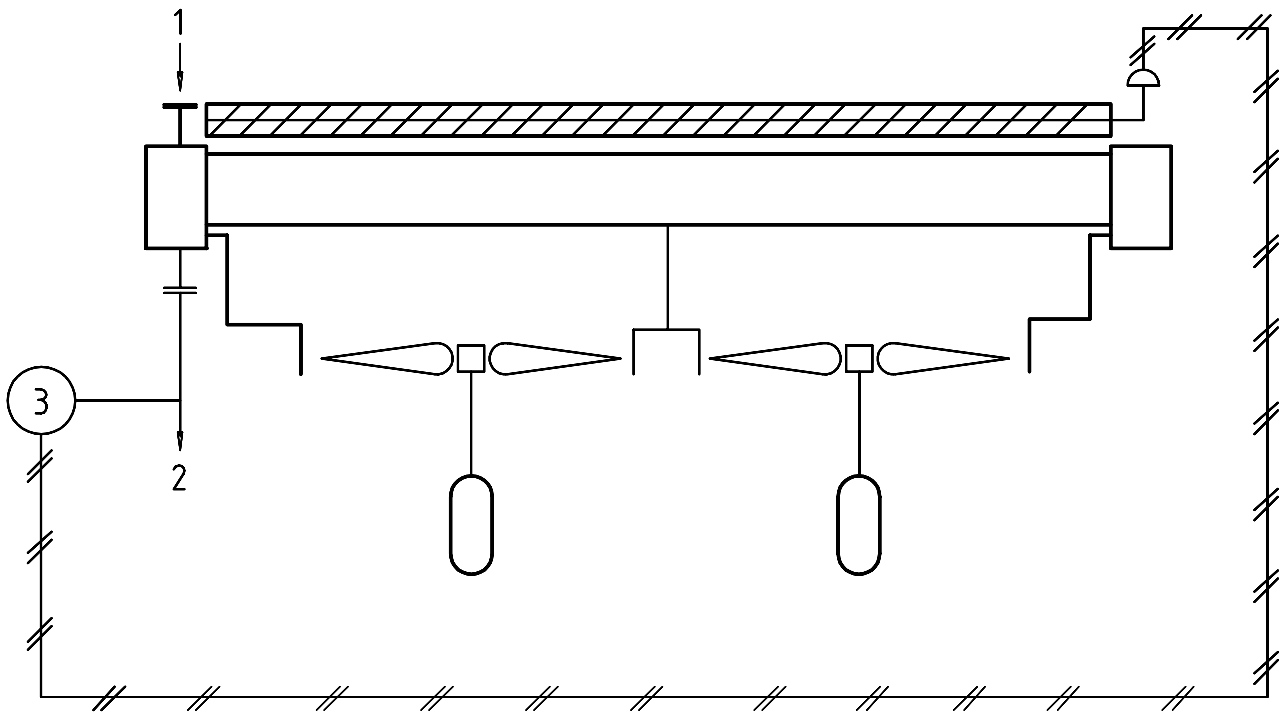
Key

1 Process in

2 Process out

3 Temperature-indicating controller

Figure C.1 — System A: Airflow control with automatically controlled variable-pitch fans



NOTE The air-cooled heat exchanger may be of the forced- or induced-draught type.

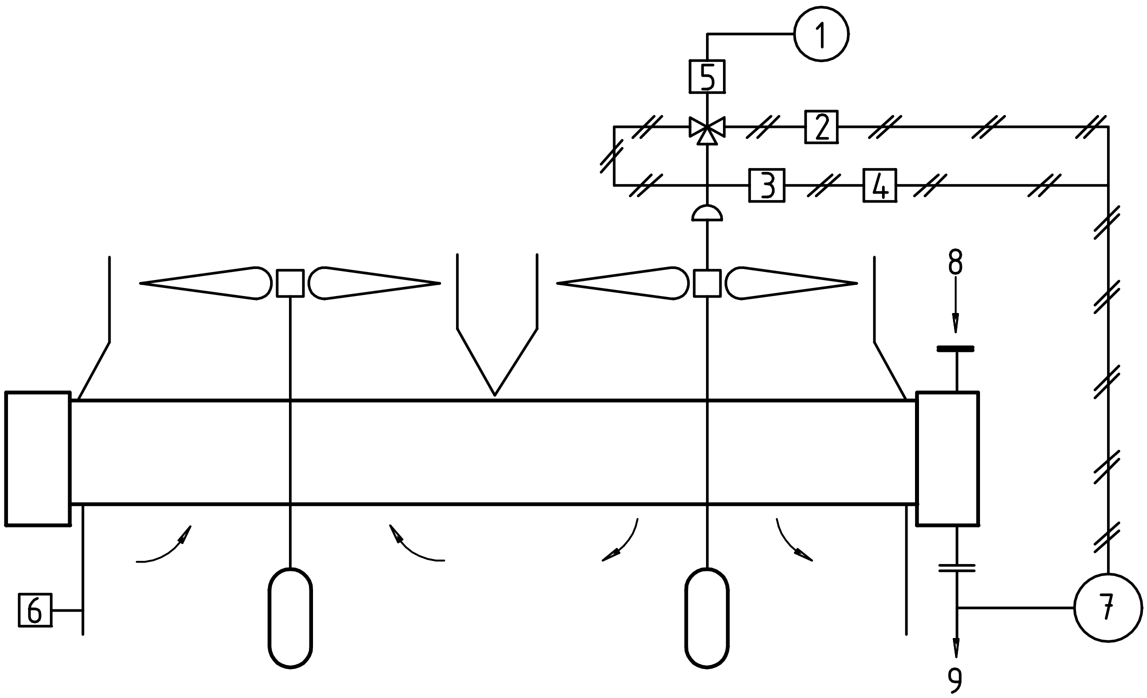
Key

1 Process in

2 Process out

3 Temperature-indicating controller

Figure C.2 — System A: Airflow control with automatically controlled outlet louvres and fixed-pitch fans

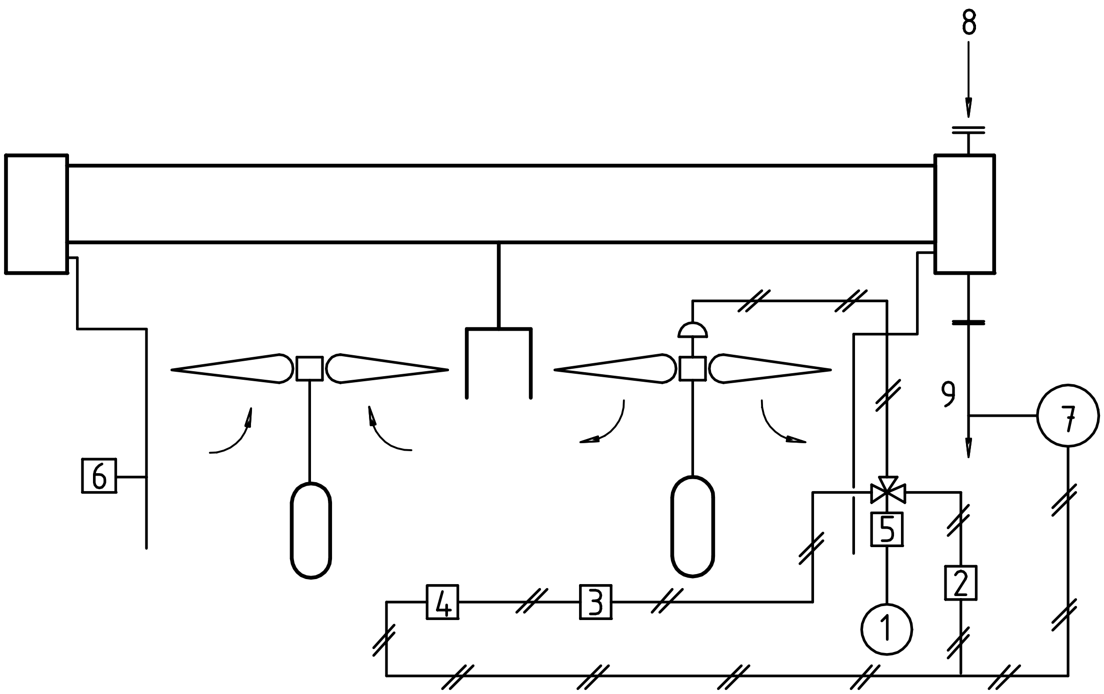


NOTE All fans can be of the automatically controlled variable-pitch type.

Key

1 Ambient air temperature switch
2 High-limit relay
3 Reversing relay
4 Low-limit relay
5 Three-way solenoid switch
6 Wind skirt
7 Temperature-indicating controller
8 Process in
9 Process out

Figure C.3 — System B: Induced draught with non-contained internal air recirculation

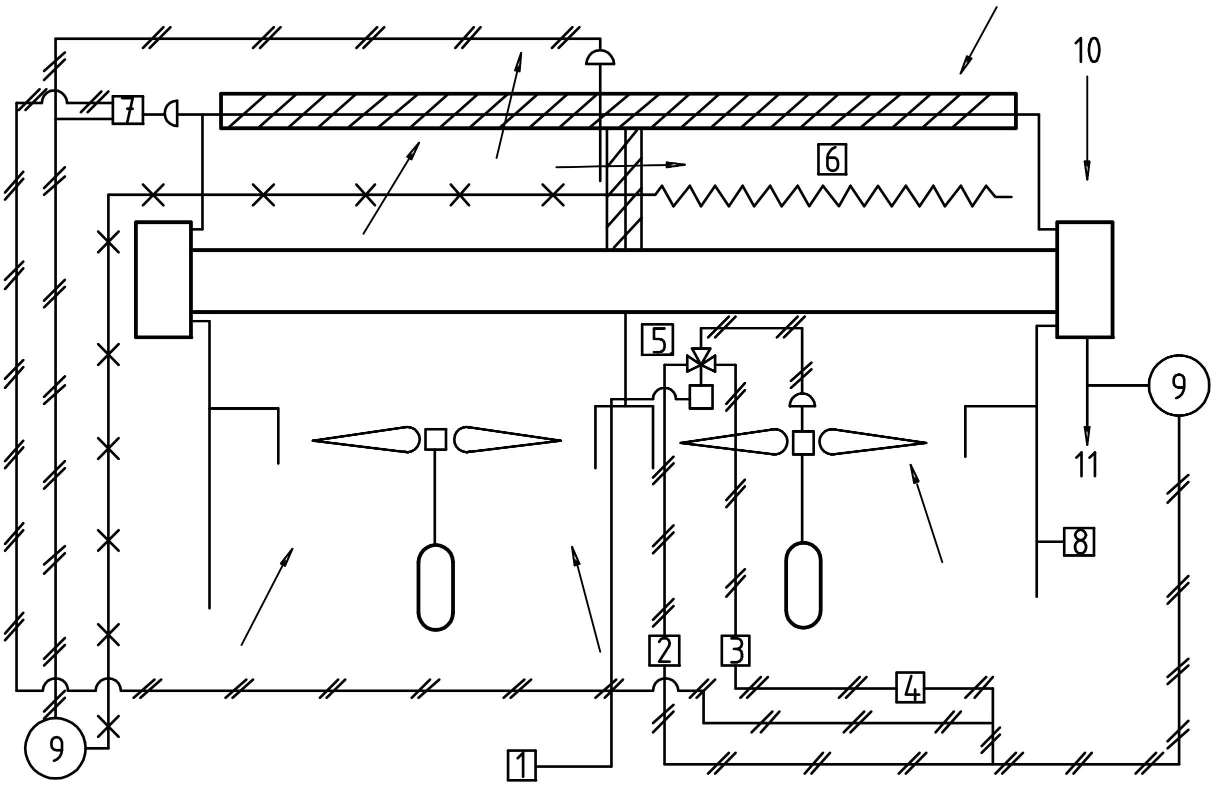


NOTE All fans can be of the automatically controlled variable-pitch type.

Key

1 Ambient air temperature switch
2 High-limit relay
3 Reversing relay
4 Low-limit relay
5 Three-way solenoid switch
6 Wind skirt
7 Temperature-indicating controller
8 Process in
9 Process out

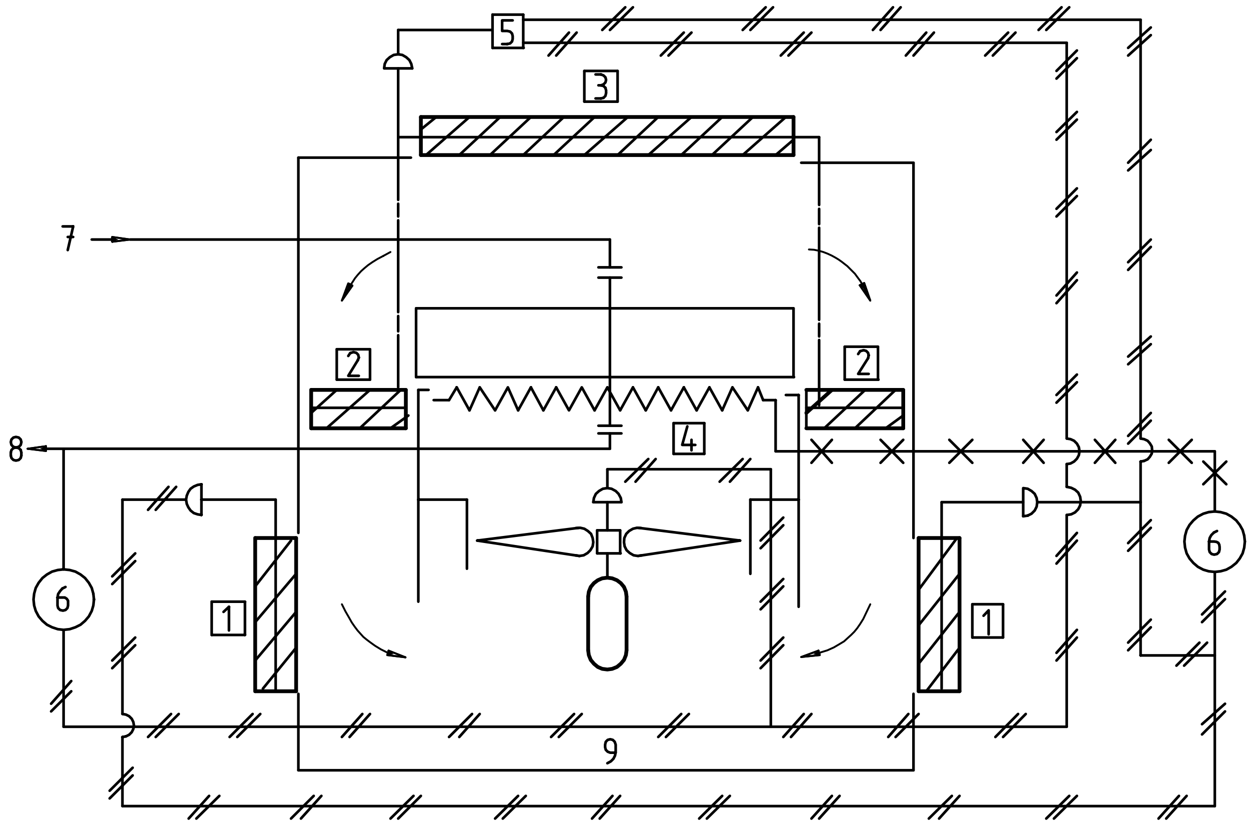
Figure C.4 — System B: Forced draught with non-contained internal air recirculation



Key

1 Ambient air temperature switch
2 High-limit relay
3 Reversing relay
4 Low-limit relay
5 Three-way solenoid switch
6 Temperature-sensing capillary tube
7 Low- or high-pressure selector
8 Wind skirt (optional)
9 Temperature-indicating controller
10 Process in
11 Process out

Figure C.5 — System C: Typical forced-draught air-cooled heat exchanger with contained internal air recirculation



NOTE 50 % to 100 % of the fans should be of the automatically controlled variable-pitch type.

Key

1 Intake louvres
2 Bypass louvre
3 Exhaust louvres
4 Temperature-sensing capillary tube
5 Low- or high-pressure selector switch
6 Temperature-indicating controller
7 Process in
8 Process out
9 Floor plate or grade

Figure C.6 — System D: Typical forced-draught air-cooled heat exchanger with external air recirculation

C.3.2 Concurrent flow

In a concurrent flow system, the process fluid begins its flow path at the bottom row of tubes, which is incident to the inlet air, and exists at the top row, which is swept by warm air. Thus, the coldest fluid exchanges heat with the warmest air, resulting in wall temperatures higher than those in a countercurrent arrangement. The design of the air-cooled heat exchanger can thus be kept simple, with no recirculation system required if the tube-wall temperature is kept above the specified minimum.

The main disadvantage of a concurrent flow system is that additional surface area is required as a result of the reduced mean temperature difference between the process fluid and the cooling air.

C.3.3 Bare tubes or reduced fin density

For viscous fluids and other fluids that exhibit low tube-side heat transfer coefficients, the tube-wall temperature can be increased by reducing the fin density or by using bare tubes. If the tube-wall temperature is thus maintained at or above the specified minimum value, no additional winterization protection is required.

C.3.4 Varying the active heat transfer surface

Heat rejection from the process fluid can be controlled by removing bundles from service, usually by the use of valves, as the ambient temperature drops. This system minimizes heat losses and maintains a higher tube velocity in the active bundles. The higher tube velocity results in a higher tube-wall temperature.

The effectiveness of this system depends on whether the reduction in active surface area results in a tube-wall temperature above the specified minimum value. Pumping capacity should be checked to ensure it is adequate for the additional pressure drop that results.

The contents of tubes in idle bundles may have to be purged or displaced. The stepwise nature of the operation may limit its applicability.

C.3.5 Serpentine coils

A serpentine coil design uses a single or limited number of continuous flow paths from the inlet to the outlet. Because of the limited number of flow paths, tubes with a large diameter [generally 50 mm to 150 mm (2 inches to 6 inches)] are used. The tubes can be finned or bare, depending on economics and the specified minimum tube-wall temperature that must be maintained.

This system usually requires high pressure drops, but the design is frequently favoured for viscous fluids because there is little or no possibility of maldistribution.

C.3.6 Indirect cooling

The normal design of an indirect (closed-loop tempered-water) cooling system uses recirculated water in a conventional shell-and-tube heat exchanger to cool the process fluid. The recirculated water is in turn cooled to a controlled temperature in an air-cooled heat exchanger that uses a fairly simple form of airflow control (automatically controlled variable-pitch fans or louvres) but is not winterized. For ambient temperatures above 0 °C (32 °F), ordinary condensate or treated water may be used. For ambient temperatures less than or equal to 0 °C (32 °F). the water should be mixed with an amount of antifreeze, such as ethylene glycol, sufficient to lower the solution's freezing point to the lowest expected air temperature.

This system is usually applied to fluids with a high viscosity or a high pour point. The system offers several advantages for these fluids:

a) better process temperature control;

b) less chance of process fluid maldistribution;

c) better general operability;

d) usually more economical operation.

However, an economic comparison should be made. C.10 provides an example of such a comparison.

C.3.7 Duty separation

The duty separation approach divides the process heat duty into two separate services. The intermediate temperature between the services is chosen to ensure that the tube-wall temperature in the upstream unit will be above the specified minimum tube-wall temperature for the full range of ambient air temperatures. The upstream unit does not require winterization; only the downstream unit is winterized.

C.3.8 Combinations

Depending on minimum air temperatures and specified minimum tube-wall temperatures, various combinations of the protection methods described in C.3.1 through C.3.7 may prove economically attractive. The following are examples of combination protective methods:

a) a combination of cocurrent and countercurrent bundles;

b) cocurrent bare-tube bundles;

c) duty separation, with varying fin densities, in subsequent bundles in series.

C.3.9 Instrumentation

C.3.9.1 General

Figures C.1 through C.6 illustrate typical instrumentation schemes for the systems described in C.3.1.1 through 3.1.4. The instrumentation methods shown are only suggestions.

C.3.9.2 System A

Typical instrumentation for System A (see Figures C.1 and C.2) consists of a temperature sensor in the exit fluid stream and a controller that receives a signal from the sensor and sends signals to one or more devices that control the airflow. These devices may be outlet louvres with a louvre actuator, automatically controlled variable-pitch fan hubs, or variable-speed fan drivers.

The most commonly used arrangements include one or more of the following components:

a) louvres with pneumatic operators, including valve positioners;

b) automatically controlled variable-pitch fans responding to a pneumatic signal;

c) pneumatic controllers having at least proportional-band and reset features. A manual/automatic setting is very desirable.

Electronic controllers and sensing elements may be used instead of pneumatic controllers. They usually require an electronic-to-pneumatic conversion at the fan hub or louvre actuator.

C.3.9.3 Systems

C.3.9.3.1 Induced draught

An induced-draught system typically employs several of the components discussed in C.3.9.2 [items b) and c)]. However, using the simple types of automatically controlled variable-pitch fans usually makes it necessary to use half the signal range from the controller for upflow and half for downflow. It is also necessary to cause a reversal of either the upflow or the downflow portion of the signal range. A simple way of reversing the signal is to use a reversing relay in conjunction with a low-limit relay, as shown in Figure C.3. Since the system must operate in two modes (upflow and downflow), an ambient temperature sensor and a selector valve are commonly used for mode selection. The high-limit relay shown in Figure C.3 is required to cause the split-range operation to occur. The exit end of the last pass is normally the most vulnerable to winterization problems and should be located under the downflow fan in the warmest air.

C.3.9.3.2 Forced draught

A forced-draught system, illustrated in Figure C.4, employs the same components discussed in C.3.9.3.1.

C.3.9.4 System C

Typical instrumentation for System C (see Figure C.5) consists of a temperature sensor in the exit fluid stream, a controller that receives a signal from the sensor and sends signals to the exhaust louvres, and one or more automatically controlled variable-pitch fans. Another temperature sensor (usually a long averaging bulb) is placed in the airstream above the bundle segment most vulnerable to freezing or other problems. A second controller receives a signal from this sensor and sends a signal to the partition louvres and the exhaust louvres. The exhaust louvres thus receive two control signals and will respond to the one requiring the more closed position. A high- or low-pressure selector relay is typically used to determine which signal reaches the exhaust louvres. It is not good practice to delete either the partition louvre or a separate actuator for the partition louvre. Deletion of the partition louvre leads to maximum cross-flow at all conditions and sacrifices heat transfer capability in certain ranges of operation while reducing cost very little.

The control elements between the controller and the automatically controlled variable-pitch fan are the same as those discussed in C.3.9.3.1 and function in the same manner.

C.3.9.5 System D

The typical instrumentation for System D (see Figure C.6) consists of a temperature sensor in the exit fluid stream and a controller that receives a signal from the sensor and sends signals to the automatically controlled variable-pitch fans and, optionally, to the exhaust louvres. A second temperature sensor (usually a long averaging bulb) is placed in the airstream below the bundle segment most vulnerable to freezing or other problems. A second controller receives a signal from this sensor and sends a signal to the exhaust louvres, the bypass louvres (if separately actuated), and the inlet louvres. Some of the inlet louvres may be manually operated.

C.4 Critical process temperatures

C.4.1 Pour points of hydrocarbon liquid mixtures

Air-cooled heat exchangers that handle gas oil and residuum cuts may require winterization. The pour points of these hydrocarbon liquid mixtures vary from –51 °C to 63 °C (–60 °F to 145 °F).

The pour point of a fraction of a hydrocarbon liquid cut with a known pour point cannot be predicted mathematically. The only realistic method of establishing the pour point of such a fraction is by measurement, using ASTM D 97.

The pour point of a blend of two hydrocarbon liquid cuts with known pour points can be approximated by calculation. Because of the imprecision of such calculations, however, when the actual pour point of the blend cannot be measured, a safety margin that respects the consequences of an air-cooled heat exchanger freezing up should be added to any predicted value.

NOTE Numbers in brackets in this annex refer to references in the bibliography.

C.4.2 Freezing points of hydrocarbons and other organic liquid pure compounds

Table C.2 lists the freezing points of frequently encountered refinery hydrocarbon and organic liquid pure compounds. Air-cooled heat exchangers that process these liquids may require winterization.

C.4.3 Water solutions of organic compounds

Water solutions of some of the organic compounds in Table C.2 are also subject to freezing in air-cooled heat exchangers. Freezing-point-concentration relationships for these materials are valid only for very dilute solutions. Figures C.7 through C.9 give measured values for freezing points over the entire concentration range.

Table C.2 — Freezing points of frequently encountered liquid pure components

Compound	Molecular weight	Freezing point	
		°C	(°F)
Water	18,0	0,0	(32,0)
Benzene	78,1	5,6	(42,0)
o-Xylene	106,2	– 25,2	(– 13,3)
p-Xylene	106,2	13,3	(55,9)
Cyclohexane	84,1	6,6	(43,8)
Styrene	104,1	– 30,6	(– 23,1)
Phenol	93,1	40,9	(105,6)
Monoethanolamine	61,1	10,3	(50,5)
Diethanolamine	105,1	25,1	(77,2)
Glycerol	92,1	18,3	(65,0)
Ethylene glycol	62,1	– 13,0	(8,6)
Naphthalene	128,2	80,3	(176,5)

C.4.4 Ammonium bisulfide

Solid ammonium bisulfide (NH_4HS) can be deposited by gas or vapour streams when the product of the partial pressures of ammonia (NH_3) and hydrogen sulfide (H_2S) exceeds the dissociation constant, K_d, at the temperature of interest and no liquid water is present.

Figure C.10 is a plot of K_d versus temperature. Deposition is not a problem in all-hydrocarbon streams, since the solubility of NH_4HS is negligible in hydrocarbons.

C.4.5 High pressure gases

Certain gases at high pressure, including C_1-C_4 paraffins and olefins, hydrogen sulfide and carbon dioxide, can form hydrates when saturated with water at temperatures above water's freezing point. These hydrates are solid crystals that can collect and plug the tubes of air-cooled heat exchangers. Figure C.11 shows the hydrate-formation conditions for these pure gases. Reference [12] gives semi-empirical methods for predicting hydrates in gas mixtures.

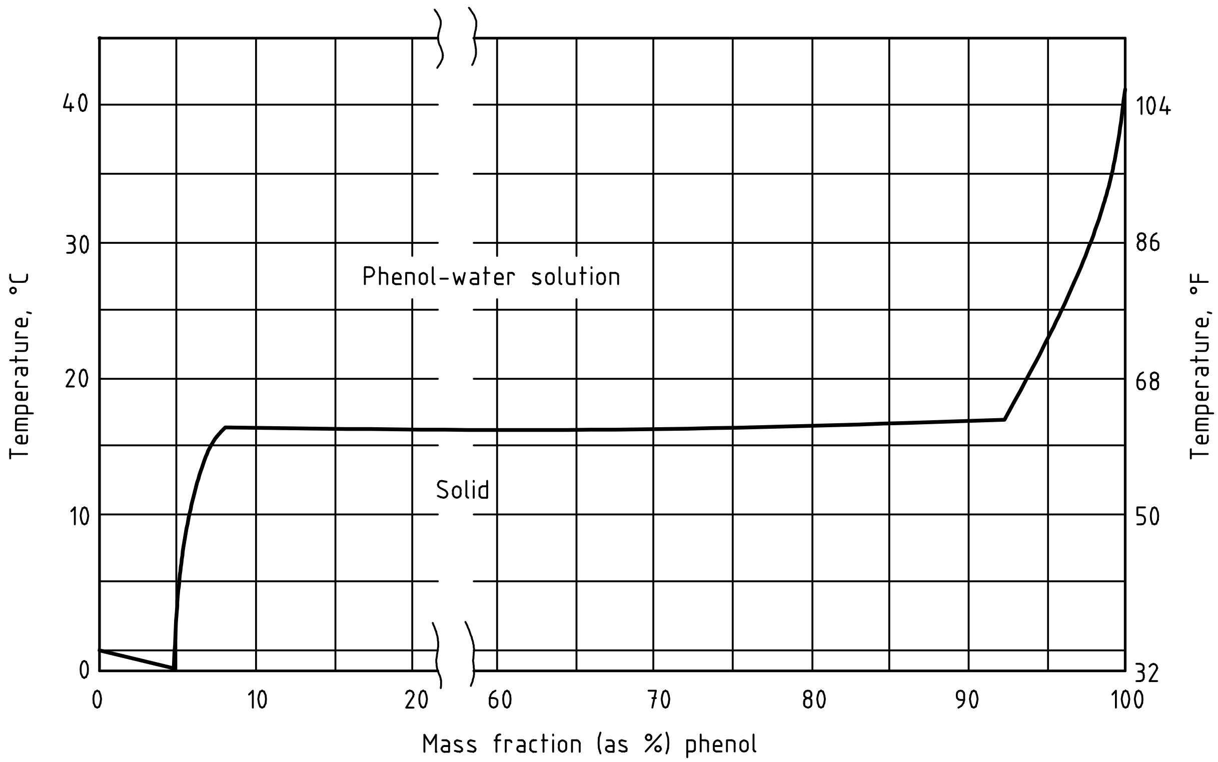


Figure C.7 — Freezing points of phenol-water solutions

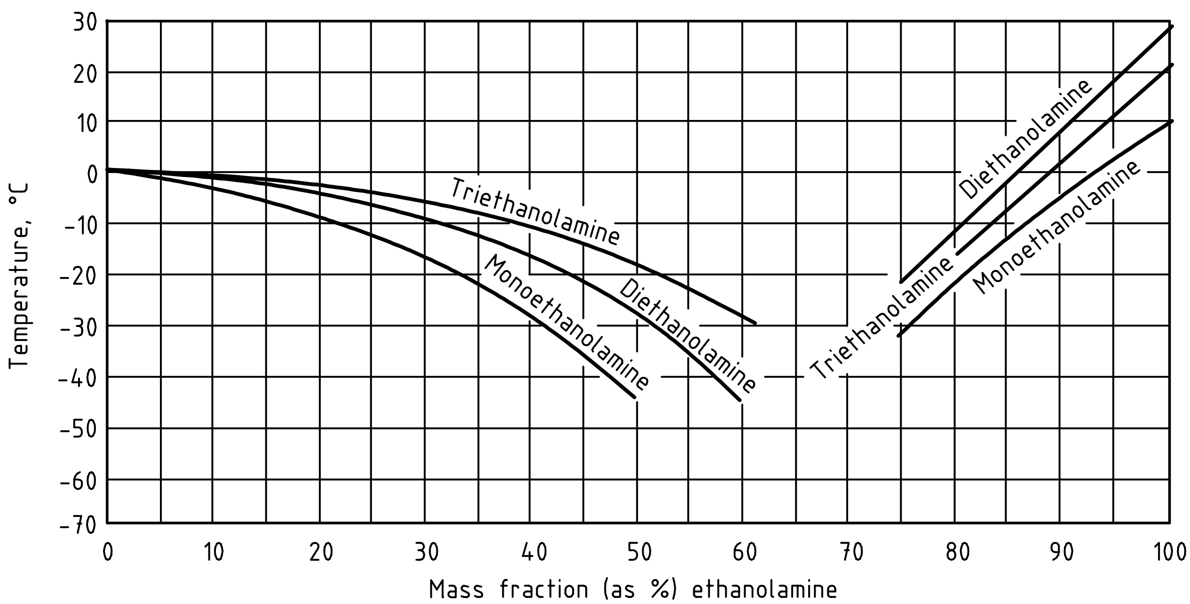


Figure C.8 — Freezing points of ethanolamine-water solutions

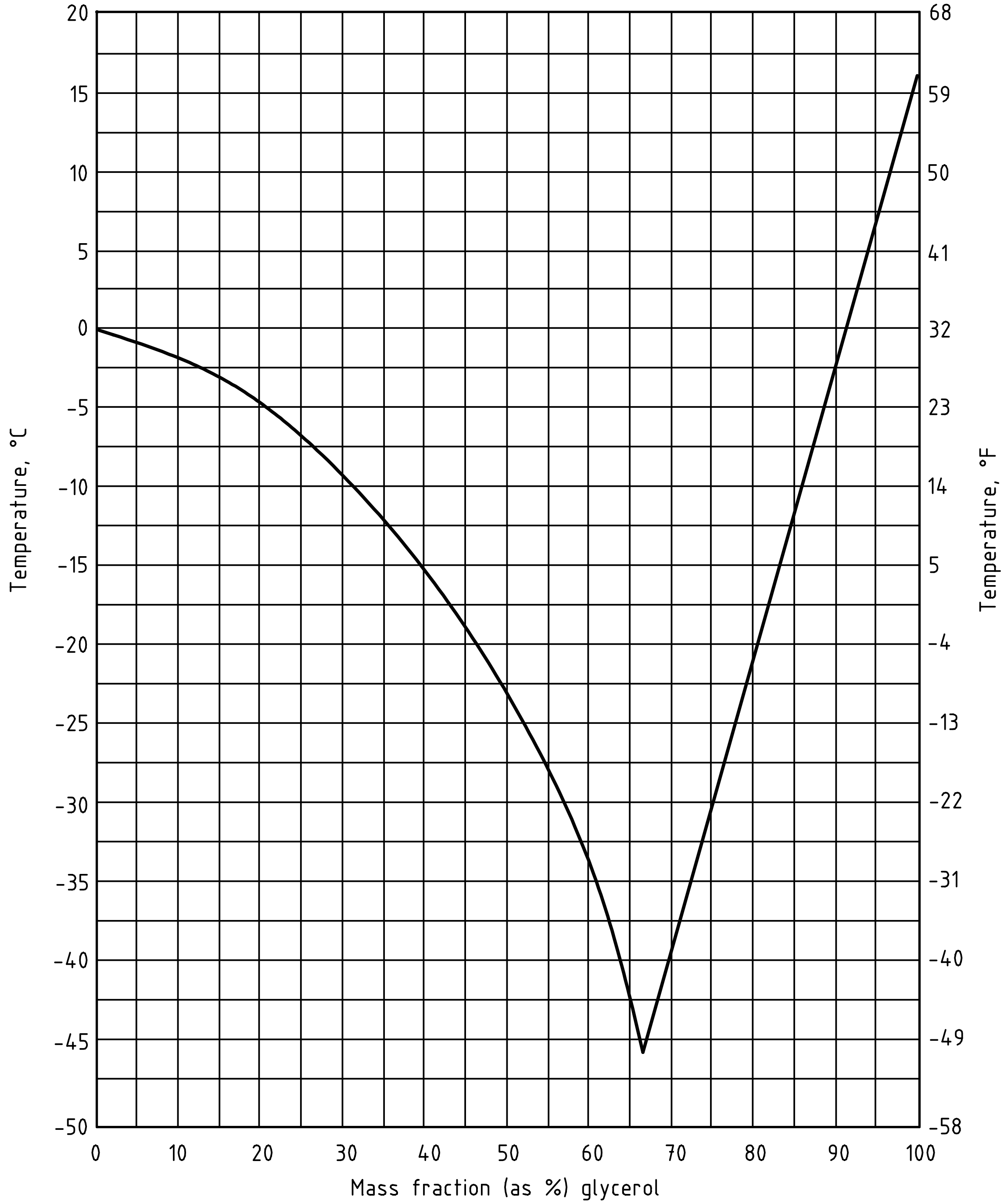


Figure C.9 — Freezing points of glycerol-water solutions

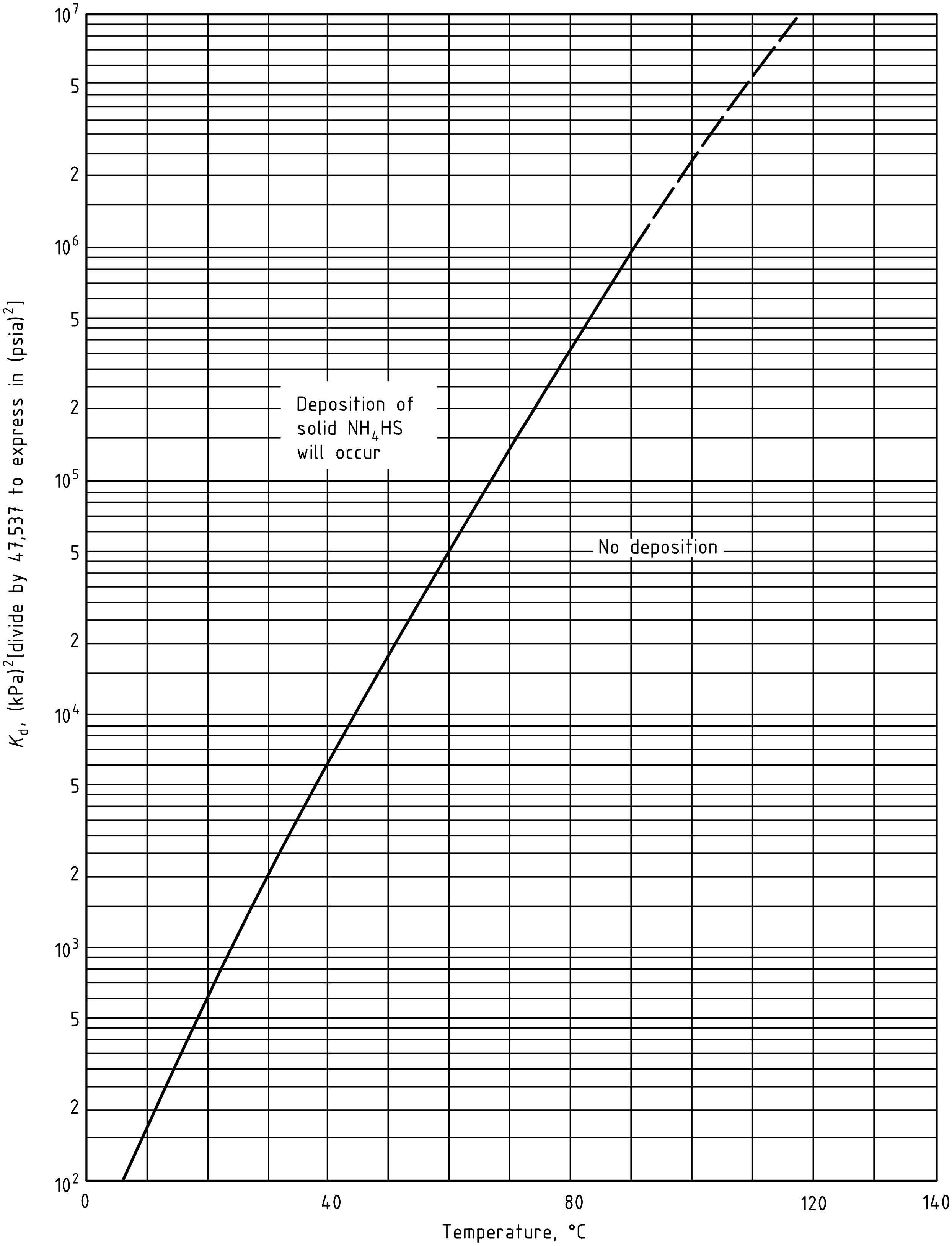
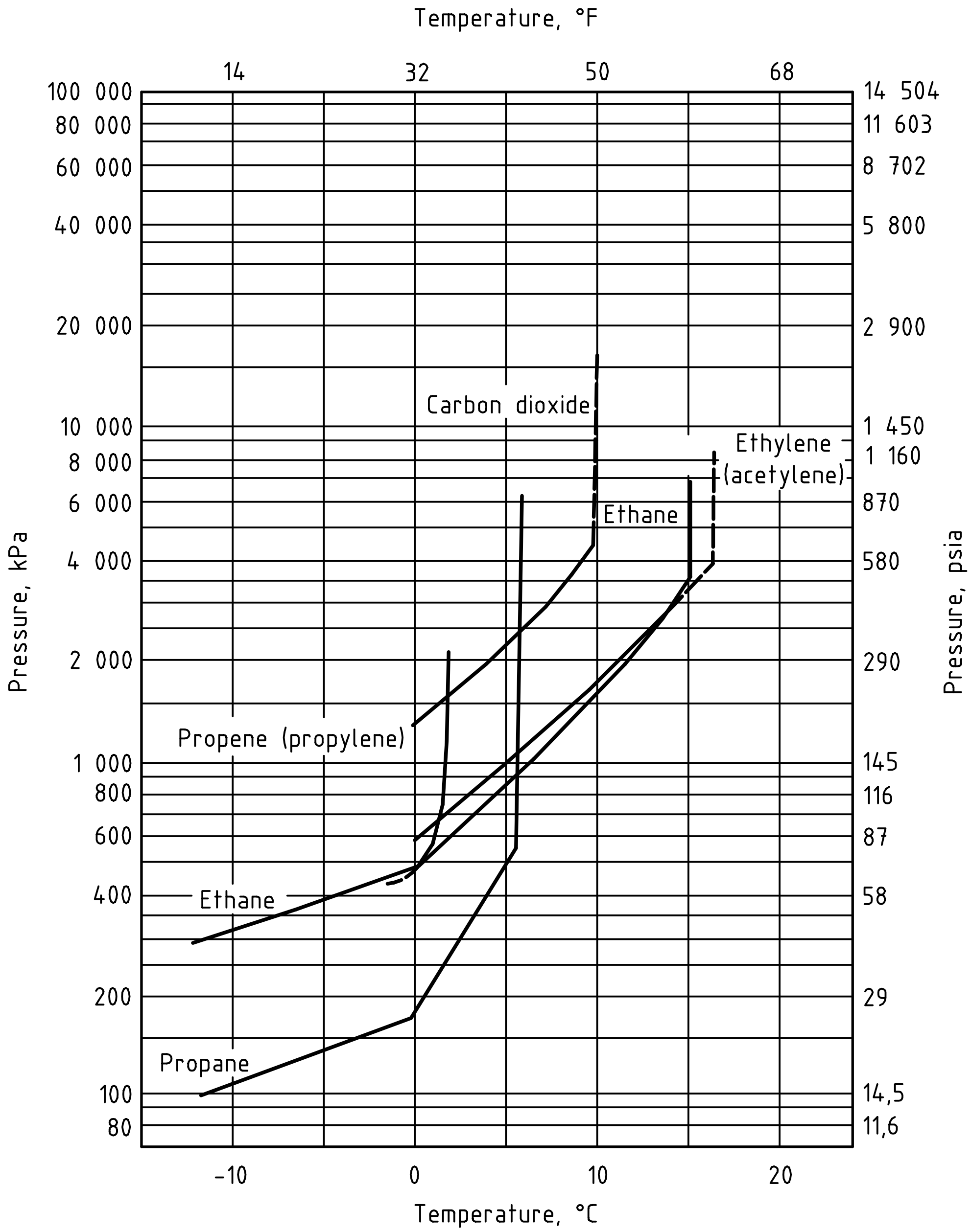


Figure C.10 — Dissociation constant of ammonium bisulfide (NH_4HS)



NOTE 1 The curves in the figure indicate the pressure-temperature regions in which gas hydrate formation is favoured (above and to the left of the appropriate curve).

NOTE 2 Equilibrium conditions are shown in the figure; however, since hydrate systems typically exhibit metastable tendencies, a metastable hydrate phase can exist far out of the hydrate region. In addition, hydrate formation will not always occur in the region in which formation is favored.

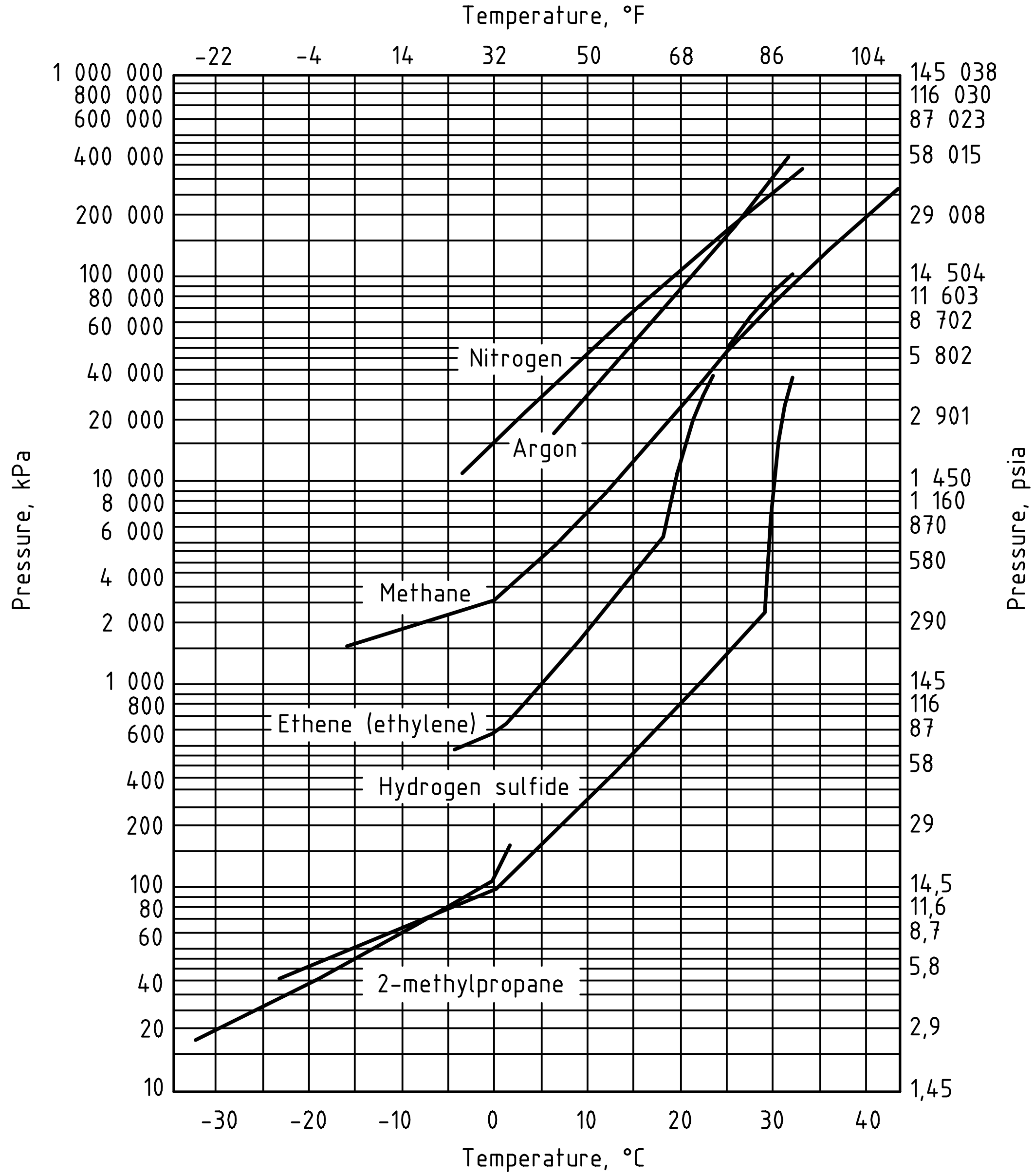
NOTE 3 The figure shows the equilibrium hydrate conditions to within 2 % of the pressure.

NOTE 4 The discontinuities in the lines correspond to changes in phase of the nonhydrate phases. For multi-component systems, hydrate formation conditions should be determined using the procedures outlined in [12]. It should be noted that small amounts of impurities can exert a very strong influence on hydrate formation conditions.

NOTE 5 Sources of hydrate equilibrium data for these figures are listed in [12].

NOTE 6 The figure and these notes are taken from Chapter 9 of [12].

Figure C.11 — Hydrate pressure-temperature equilibria



NOTE 1 The curves in the figure indicate the pressure-temperature regions in which gas hydrate formation is favoured (above and to the left of the appropriate curve).

NOTE 2 Equilibrium conditions are shown in the figure; however, since hydrate systems typically exhibit metastable tendencies, a metastable hydrate phase can exist far out of the hydrate region. In addition, hydrate formation will not always occur in the region in which formation is favored.

NOTE 3 The figure shows the equilibrium hydrate conditions to within 2 % of the pressure.

NOTE 4 The discontinuities in the lines correspond to changes in phase of the nonhydrate phases. For multicomponent systems, hydrate formation conditions should be determined using the procedures outlined in [12]. It should be noted that small amounts of impurities can exert a very strong influence on hydrate formation conditions.

NOTE 5 Sources of hydrate equilibrium data for these figures are listed in [12].

NOTE 6 The figure and these notes are taken from Chapter 9 of [12].

Figure C.11 *(continued)*

C.5 Tube-wall temperature calculations

C.5.1 General

C.5.1.1 The need for winterization of air-cooled heat exchangers is a function of the tube-wall temperature resulting from the inlet air temperature and the critical process temperature of the fluid. Consideration should be given to the type of design, the operating modes, and the fluid flow regime to predict tube-wall temperatures accurately.

C.5.1.2 In cross-flow countercurrent air-cooled heat exchanger bundles, the worst condition usually exists at the outlet of the bottom row of tubes. At this location, the air that comes in contact with the tube is at its lowest temperature, and the tube-side fluid is also at the lowest possible temperature. This is usually the critical location, but other locations may also need to be considered. Axial-flow fans do not provide completely even airflow distribution. The designer should add at least 20 % to the air-side heat transfer rate to account for areas of high airflow (see factor f_a in C.5.2). The designer should also ensure good tube-side flow distribution within the tube bundle.

To calculate the tube-wall temperature it is necessary to determine the air-side and tube-side resistances at each location under consideration. Such information may be obtained from the original manufacturer or another suitable source.

C.5.2 Symbols

A total outside surface area of the bottom layer of tubes, in m^2 (ft^2)

A_b outside surface area of the bare tube per unit length, in m^2/m (ft^2/ft)

A_f surface area of the fin per unit length of the tube, in m^2/m (ft^2/ft)

A_i inside surface area of the tube per unit length, in m^2/m (ft^2/ft)

A_o outside surface area of the finned tube per unit length, in m^2/m (ft^2/ft)

f_a air-side heat transfer coefficient multiplier to account for airflow maldistribution (the recommended minimum value is 1,2)

r local overall thermal resistance, in m^2·K/W (°F·ft^2·h/BTU)

r_c local clean overall thermal resistance, in m^2·K/W (°F·ft^2·h/BTU)

r_{ds} air-side fouling resistance, in m^2·K/W (°F·ft^2·h/BTU)

r_{dt} tube-side fouling resistance, in m^2·K/W (°F·ft^2·h/BTU)

r_{fs} local air-side resistance, in m^2·K/W (°F·ft^2·h/BTU)

r_{ft} local tube-side resistance, in m^2·K/W (°F·ft^2·h/BTU)

r_m total metal resistance of the tube, in m^2·K/W (°F·ft^2·h/BTU)

r_{mf} fin metal resistance, in m^2·K/W (°F·ft^2·h/BTU)

r_{mt} tube metal resistance based on the inside surface area of the tube, in m^2·K/W (°F·ft^2·h/BTU)

NOTE An exact calculation of r_{mt} would require the tube metal resistance to be based on the logarithmic mean surface area of the tube; however, the relatively insignificant magnitude of the error caused by basing the tube metal resistance on the inside surface area of the tube does not justify the complexity introduced by the use of the logarithmic mean surface area.

T_B bulk temperature of the tube-side fluid at the location where the wall temperature is to be calculated, in °C (°F)

t_B bulk temperature of the air at the location where the wall temperature is to be calculated, in °C (°F)

T_w tube-wall temperature, in °C (°F)

U local overall heat transfer coefficient referred to the outside surface, in W/m²·K (BTU/h·ft²·°F)

ϕ fin efficiency

ϕ_o efficiency of the finned tube surface

C.5.3 Calculations

C.5.3.1 Once the various resistances have been determined, the wall temperature can be predicted by prorating the resistances relative to the temperature at the outlet or other critical areas. This can be done using the following equations:

$$U = 1/r \tag{C.1}$$

$$r = (r_{ft} + r_{dt}) (A_o / A_i) + r_{mt} (A_o / Ai) + (1 / \phi_o)(r_{fs} / f_a) + r_{ds} \tag{C.2}$$

C.5.3.2 Fins do not cover all of the tube surface, and since the exposed bare tube surface may be considered to have an efficiency of 1,00, the efficiency of the finned tube surface is always higher than that of the fins alone. Thus,

$$\phi_o = 1 - (A_f / A_o)(1 - \phi) \tag{C.3}$$

C.5.3.2.1 Fin efficiency is a complex calculation. It can, however, be replaced with an equivalent fin metal resistance, designated as r_{mf}. This metal resistance reaches a constant maximum value for an outside resistance above some value determined by fin height, thickness, and thermal conductivity. For standard aluminium fins, 90 % of this value is reached at air-side resistances (r_{fs}) that are lower than those normally encountered. The fin metal resistance for these calculations may be considered constant with negligible error.

The overall resistance equation can therefore be rewritten in the following form:

$$r = (r_n + r_{dt}) (A_o / A_i) + r_m + (r_{fs} / f_a) + r_{ds} \tag{C.4}$$

where

$$r_m = r_{mf} + r_{mt} (A_o / A_i)$$

$$r_{mf} = \{(1 - \phi_o)/ \phi_o \}(r_{fs} / f_a) + r_{ds}$$

C.5.3.2.2 Tables C.6 and C.7 provide values of r_m for tubes of several common materials with an outside diameter of 25,4 mm (1 inch) and aluminium fins 0,4 mm (0,016 inch) thick and 15,9 mm ($^5/_8$ inch) high. Other sizes and fin materials require a fm efficiency calculation to define r_{mf} for the combination. For fin efficiency curves, refer to textbooks such as [13, 14, 15].

C.5.3.3 Initially, it is best to assume that the unit is clean. Tube-side fouling would increase the surface temperature, since the fluid contact would then be on the surface of the fouling material. The basic resistance equation can be rewritten in the following form:

$$r_c = r_{ft} (A_o / A_i) + r_m + (r_{fs} / f_a) \tag{C.5}$$

The equations are the same for bare-tube exchangers, except that $\phi_o = 1$ and A_o is the outside surface area of the bare tube, A_b. Since the performance of bare-tube exchangers is sensitive to pitch arrangement, the designer should refer to bare-tube correlations such as those described in [16] for air-side heat transfer coefficient calculations.

The overall resistance for bare tubes is therefore calculated as follows:

$$r = (r_{ft} + r_{dt})(A_b / A_i) + r_{mt} + (A_b / A_i) + (r_{fs} / f_a) + r_{ds} \tag{C.6}$$

For a clean bare-tube unit, this equation reduces to:

$$r_c = r_{ft}(A_b / A_i) + r_{mt}(A_b / A_i) + (r_{fs} / f_a) \tag{C.7}$$

C.5.3.4 The tube-wall temperature can be calculated on the basis of a prorated portion of the clean overall resistance:

$$T_w = T_B - (r_{ft} / r_c)(A_o / A_i)(T_B - t_B) \tag{C.8}$$

Sample calculations are given in C.11.

C.5.3.5 Single-pass, multiple-row air-cooled heat exchangers are more susceptible to freezing and pour-point problems because of variations in the layer-to-layer mean temperature difference, with the bottom row exchanging more heat than any of the upper rows. This means that the mixed outlet fluid temperature cannot be used safely; instead, the bulk tube-side fluid outlet temperature should be calculated for each row of concern.

Two-phase fluids in a single pass with multiple rows require a more complete analysis that recognizes the separation of phases in the header. The problem becomes more complex when the units are not designed with equal flow areas in each pass. With viscous fluids, the problem of extreme flow maldistribution arises. This is difficult to calculate, and these fluids should be handled in as few parallel passes as possible. A single continuous serpentine coil is the ideal approach.

C.5.3.6 When the tube-wall temperature is calculated, the following operating questions should be considered:

a) At lower temperatures, how much less airflow is needed to remove the required heat?

b) Is the unit to operate with fans off or on?

c) Does the unit have louvres?

d) Has an automatically controlled variable-pitch fan or another means been provided to reduce airflow?

e) Is the unit operating at partial load so that the tube-side flow conditions affect the wall temperature?

C.6 Heat losses

C.6.1 General

C.6.1.1 Air-cooled heat exchangers are usually large pieces of equipment that are not well suited to being enclosed. Where airflow must be contained or controlled, louvres or sheet metal panels are normally used. Provision should be made for shutting down, starting up, or holding such equipment at standby conditions during periods of minimum air temperature. Under these conditions, the process fluid may be cooled below its critical process temperature unless airflow through the bundle is nearly stopped and an auxiliary source of heat is provided.

C.6.1.2 Unless the amount of heat that will be lost by louvre leakage and by conduction through enclosing panels can be determined, there is no certainty that enough heat can be added by auxiliary means. Thus, the problem of evaluating the auxiliary heat source must begin with determining how much heat may be lost under a particular set of circumstances. Examples of heat loss calculations are given in C.12.

C.6.1.3 The most important case to be considered is that of no process flow with fans off, minimum air temperature and fairly high wind velocity. One should assume that it is necessary to maintain the bundle at least 11 °C to 17 °C (20 °F to 30 °F) above the critical process temperature.

C.6.1.4 A less important case is that of short-duration heat loss when there is no process flow with fans on, minimum air temperature and fairly high wind velocity. These conditions should occur only during the transition period from operation to shutdown or vice versa, so an example of this case is not given in C.12. The equations of the "fans-off" example can be used to find the louvre leakage by using the pressure drop that would exist with fans on instead of the pressure drop resulting from the effect of the hot-air column.

C.6.1.5 Several factors should be considered when the auxiliary heat source mentioned in C.6.1.1 and C.6.1.2 is installed. A choice should be made about what fluid will be used (usually steam but occasionally an antifreeze solution). The location of the heat source should also be decided. A separate coil that is one row deep is usually placed immediately below the process bundle; however, special considerations may dictate less effective placement, such as inside the recirculation duct.

C.6.2 Louvre leakage

Louvres of standard manufacture, maintained in good condition, will have a leakage area of not more than 2 % of the face area when closed. This can be reduced to not more than 1 % if special, more costly designs are used. The air leakage rate may be calculated for either case. (See C.12.1.3 or C.12.2.3 for a sample calculation.) Tests on standard louvres indicate that an average louvre will have only about half the leak area predicted by maximum tolerances.

C.6.3 Surface heat loss

The heat loss from the sheet metal panels that form the enclosure is a function of the air velocity both inside and outside, as well as the temperature differential between the enclosed air and the ambient air. (The overall heat transfer coefficient for this surface is calculated for a range of wind velocities in C.12.1.4 and C.12.2.4.) Calculations of this type can also be used to determine the heat loss from the hot air being recirculated through the recirculation duct during normal operation. The heat loss calculation for the duct can be used to assure that the required air temperature to the bundle will be maintained when the recirculated air is blended with the cold inlet air.

C.7 Guidelines

C.7.1 General

Air-cooled heat exchangers are normally designed to dissipate a given heat duty in summer conditions and also dissipate the same heat duty (or more) in winter conditions. Additional measures are taken to assure proper operation during periods of minimum air temperature. These measures include recirculating a fraction of the air so that it will mix with and heat the incoming cold air. Ducts and louvres are required to direct this recirculation.

Provisions to achieve thorough mixing of the recirculated air with the cold inlet air would be prohibitively expensive. The set point for the average temperature of this mixed airstream should therefore be above the critical process temperature. For instance, the set point for vacuum steam condensers is usually 1,5 °C to 4,5 °C (35 °F to 40 °F). It is important to measure the average air temperature in these systems with an averaging bulb 4 m to 6 m (12 ft to 20 ft) long that spans the airstream, and not with a sensor that measures temperature at only one point.

C.7.2 Design methods

C.7.2.1 System C: contained internal circulation

System C (see Figure C.5) operates in two modes, the summer mode and the winter mode. In the summer mode, both fans move air upward and no air is recirculated. In the winter mode, one fan (normally on the exit end of the unit) moves air downward. This also causes a part of the air that is moved upward through the bundle (on the end opposite the exit) to flow horizontally across the top of the bundle through a bypass louvre and then downward through the bundle. Only enough air makes this journey to cause the average temperature of the mixed air entering the bundle on a downward traverse to satisfy a preset value. The duct above the bundle should be adequately sized for the maximum quantity of air that must make the journey. A conservative design rule is to size the duct cross-section based on a linear air velocity of 305 m/min (1 000 ft/min), using the quantity of air that passes through the bypass louvres. In no case should the duct cross-section exceed that required to recirculate 100 % of the heated air.

An alternative method that has proved to be adequate is to make the height of the duct space above the top of the side frame one-tenth of the tube length, rounding to the nearest 0,15 m (0,5 ft). This would require a 1,2 m (4 ft) height for tube bundles 12 m (40 ft) in length, and a 1,1 m (3,5 ft) height for tube bundles 11 m (36 ft) in length.

C.7.2.2 System D: external recirculation

System D (see Figure C.6) operates in only one mode. This means that the air movement is always upward through the bundle. When inlet air temperatures are low enough, however, part of this air leaves the bundle and returns to the fan inlet by passing over the side or the end of the bundle through a duct with a bypass louvre.

This external recirculation duct may be conservatively sized using the same rules as for the internal recirculation duct described in C.7.2.1. The application of these rules will usually result in a duct with a cross-sectional area equal to 20 % to 30 %of the bundle face area. When more than 75 % of the heated air must be recirculated, however, the duct size may approach 40 % of the bundle face area.

C.8 Mechanical equipment

C.8.1 General

When mechanical equipment is to be operated in an extremely cold or hot environment, care should be taken that the equipment is specified and designed for the temperature extremes to which it will be exposed. It is possible that two heat exchangers located side by side will have different design temperature considerations if one has only airflow control and the other has an external recirculation system.

C.8.2 Design temperatures

C.8.2.1 Unless otherwise agreed upon, the minimum design temperature for the mechanical equipment should be the minimum design air temperature.

C.8.2.2 Unless otherwise agreed upon, the maximum design temperature for mechanical equipment in the airstream exiting the heat exchanger should be equal to the maximum process or auxiliary heating fluid temperature. The maximum air temperature may occur when the fans are not operating and the louvres are closed.

C.8.2.3 The maximum design temperature for the mechanical equipment in the inlet or recirculation airstream depends on the type of winterization system, as described in C.8.2.3.1 through C.8.2.3.3. Each operating mode (start-up, normal operation, and shutdown) should be examined to determine the design temperature to be used.

C.8.2.3.1 In an airflow control system (see Figures C.1 and C.2), the maximum design temperature for equipment in the inlet airstream is the design dry-bulb air temperature.

C.8.2.3.2 In a system with noncontained or contained internal recirculation (see Figures C.3, C.4 and C.5), the maximum design temperature for the equipment in the inlet airstream should be the temperature of the air exiting the reversed-airflow fan during recirculation, plus a safety factor of 14 °C (25 °F).

C.8.2.3.3 Since the air will not be completely mixed in an external recirculation system (see Figure C.6), the design temperature for the mechanical equipment in the inlet airstream should be chosen carefully. The danger lies in exposing the mechanical equipment to hot stratified air that has not been mixed with the cooler inlet air. This problem is most prevalent during start-up.

C.8.3 Design temperature range

Most mechanical equipment will operate satisfactorily between air temperatures of – 29 °C (– 20 °F) and 40 °C (104 °F) without any modifications. However, since material selection and design techniques are not standardized for most components of mechanical equipment, the standardized operating ranges vary among manufacturers.

C.8.4 Typical characteristics and operating ranges for standard mechanical equipment

C.8.4.1 General

The characteristics and air temperature ranges given in this section are typical and are not intended to limit the application of any equipment. The suitability of continuously operating a particular piece of equipment at a specified design temperature should be confirmed with the manufacturer.

C.8.4.2 Fans with manually adjustable pitch in continuous operation [– 54 °C to 121 °C (– 65 °F to 250 °F)]

For best results in cold weather, fans with manually adjustable pitch should have hubs made of ductile iron, aluminium, or another material with good ductility. The blade material should exhibit similar characteristics.

C.8.4.3 Fans with automatically controlled variable pitch in continuous operation
[– 32 °C to 121 °C (– 25 °F to 250 °F)]

The criteria in C.8.4.2 for hubs and blades for fans with manually adjustable pitch also apply to automatically controlled variable-pitch fans. Since the automatic pitch device for each manufacturer's fans is different, the actual operating temperature range and recommendations for extending the range should be obtained from the manufacturer.

C.8.4.4 Electric motors [– 30 °C to 40 °C (– 22 °F to 140 °F)]

The upper limit on operating temperature for electric motors may be raised by substituting an insulation system and a bearing lubricant with a higher temperature rating. In most cold weather applications, space heaters are provided in the motors to maintain the internal air temperature above the dew point.

C.8.4.5 V-belts [– 40 °C to 60 °C (– 40 °F to 140 °F)] **and high-torque-type positive-drive belts**
[– 34 °C to 85 °C (– 30 °F to 185 °F)]

Belt life is reduced when belts are operated outside the temperature ranges given above. Special belts are available for operation above and below these ranges. The life expectancy of special belts may be shorter than that of standard belts.

C.8.4.6 Gear drives [– 18 °C to 77 °C (0 °F to 170 °F)]

Operation of gear drives below the range given above would require changing to a lubricant suitable for the temperature and possibly adding an oil heater. Actual temperature ranges and recommendations to extend the temperature range should be obtained from the gear drive manufacturer.

C.8.4.7 Bearings [– 45 °C to 121 °C (– 50 °F to 250 °F)]

For bearings, the temperature range above can be extended by substituting a lubricant suitable for the required temperature range.

C.8.4.8 Steel or aluminium louvres [– 40 °C to 121 °C (– 40 °F to 250 °F)]

Louvres should be designed for the expected loads during operation at low temperatures. This may require selecting a more ductile material. Snow and ice loads, as well as the effect of ice on the design and operation of the linkage, should be considered in the design. The temperature range can be extended by selecting different bearing materials.

C.8.4.9 Pneumatic diaphragm actuators [– 40 °C to 82 °C (– 40 °F to 180 °F)]**, pneumatic piston actuators** [– 34 °C to 79 °C (– 30 °F to 175 °F)] **and pneumatlc positioners** [– 40 °C to 71 °C (– 40 °F to 160 °F)]

The temperature range given above for pneumatic actuators and positioners can be extended by changing materials of several of the components, including but not limited to diaphragms and O-rings.

C.8.5 Auxiliary heating equipment

C.8.5.1 When steam coils are used with any of the winterization systems, the maximum design temperature for the exposed mechanical equipment can be determined by the steam saturation temperature. The radiation effect of the steam coil is negligible and may be omitted when the design temperatures of mechanical equipment located below a steam coil are defined. Steam coils are normally used during start-up and shutdown, but not during general operation. A steam trap that fails in the open position should be used to avoid freezing of the steam coil.

C.8.5.2 Other types of auxiliary heating equipment, such as glycol/water coils, heat-transfer fluid coils, electric heaters and space heaters, are being used successfully.

C.9 Start-up and shutdown procedures

C.9.1 General

The procedures in this section are intended to supplement users' established procedures, not to replace them. The procedures apply only to air-cooled heat exchangers with some degree of winterization, from the simplest (airflow control only) to the most complex (full external air recirculation). The procedures apply only to start-up and shutdown during cold weather.

C.9.2 Start-up procedures

C.9.2.1 Before start-up, any snow or ice or protective coverings that may affect louvre or fan operation should be removed. Depending on the particular weather conditions, snow and ice can sometimes be removed by activating the start-up heating coil. Care should be taken not to damage the top louvres. Workers should not walk on the louvres.

C.9.2.2 The instrument air supply should be checked to ensure that it is functioning and free from water.

C.9.2.3 Instruments and control valves should be checked for satisfactory operation.

C.9.2.4 The operation of all louvres, linkages, and automatically controlled variable-pitch fans (if used) should be checked.

C.9.2.5 The louvres should be closed, and the start-up heating coil (if not already activated) should be activated. If the heating system is a steam coil, the steam trap should be checked to ensure that it is functioning satisfactorily. The tube bundle and the air surrounding it should be at a temperature higher than the critical process temperature before the bundle is placed in service.

C.9.2.6 For systems with internal circulation (see Figures C.3, C.4 and C.5), the control system should be verified as being in the winter mode, that is, with the fan nearest the process outlet pitched to blow air down through the bundle and the other fan pitched to force air up through the bundle. Both fans should be set at their maximum airflow position.

C.9.2.7 For systems with external recirculation (see Figure C.6), the following steps should be taken.

a) When a linkage between the top louvres and the bypass louvres is provided, it should be checked to ensure that the bypass louvres are working as intended.

b) The operation and means of actuation of the inlet louvres should be checked.

c) The exchanger's enclosure should be checked to ensure that no large openings are allowing ambient air into the enclosure.

C.9.2.8 Normal procedures should be followed when the unit is started up; however, certain process conditions may necessitate special start-up requirements. For instance, steam condensers or viscous liquid coolers at moderate temperatures should generally have the process stream introduced at or near the full flowrate. In contrast, process streams at high temperatures should be introduced to the exchanger gradually to minimize high thermal stresses that might cause mechanical failure.

C.9.2.9 The fans should be turned on, the louvres and automatically controlled variable-pitch fans should be placed on automatic control, and the heating coil should be shut off when normal operating conditions are reached.

C.9.3 Shutdown procedures

C.9.3.1 Before shutdown, the fans should be shut off, the louvres should be closed, and the heating coil, if provided, should be activated.

C.9.3.2 The normal plant procedure for shutdown should be followed.

C.9.3.3 If steam purging is required, caution should be exercised to assure that the condensate is thoroughly drained.

C.9.3.4 The heating coil should be turned off.

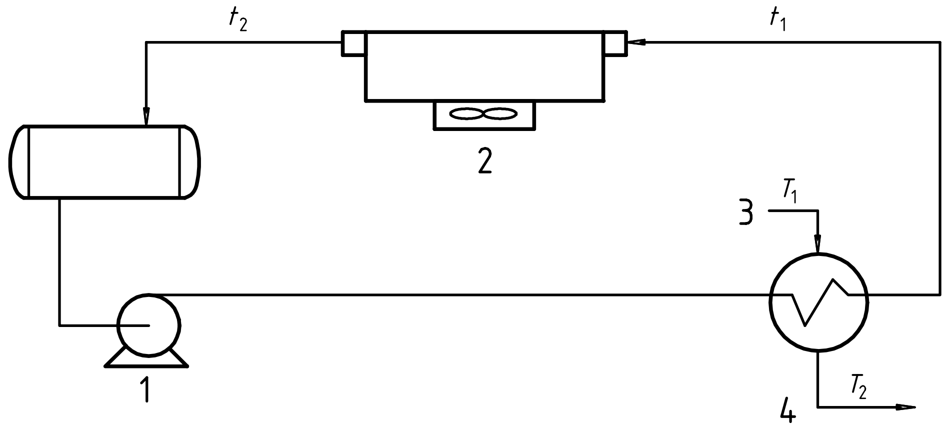
C.9.3.5 The normal plant procedures for protecting the equipment during shutdown periods should be followed.

C.10 Sample economic comparison of indirect (tempered-water) versus direct air cooling for systems requiring winterization

C.10.1 General

Before an economic comparison can be made between indirect and direct air-cooling for systems requiring winterization, it is necessary to select the operating temperatures for the tempered water. This is an important factor in the economics of the system. The outlet temperature from the air-cooled heat exchanger, t_2 (see Figure C.12), is a function of the design dry-bulb air temperature and of the critical process and tube-wall temperatures in the shell-and-tube heat exchanger. Temperature t_1 should be set above the critical process temperature. The temperature range, $t_1 - t_2$, affects the size of the tempered-water air-cooled heat exchanger, the shell-and-tube exchanger, and the circulating pump, and determines the number of shells in series. The temperature range should be selected to optimize the components for the particular system.

The conclusions of the example in C.10.2 and C.10.3 are specific for this application only. Each application should be investigated.



Key

1 Circulating pump

2 Water cooler

3 Process fluid

4 Process cooler

Figure C.12 — Tempered-water system

C.10.2 Example (SI units)

A residuum cooler is designed to cool 363 000 kg/h of atmospheric residuum from 171 °C to 93 °C. The design dry-bulb air temperature is 49 °C, and the minimum design air temperature is 0 °C. The residuum pour point is 35 °C, and the inlet and outlet viscosities are 6,0 and 48,0 centipoises, respectively. The inlet and outlet temperatures of the circulating water were selected as 57 °C and 123 °C, respectively, giving a flowrate of 227 000 kg/h. Note that the problem selected is based on a pressurized water system. Table C.3 describes the two systems, and Table C.5 compares their costs.

NOTE The direct air-cooled heat exchanger system described in this paragraph, although an actual application, may not represent the optimum design. In addition, the equipment costs given in this example are based on 1981 data and are for purposes of illustration only; when an actual economic comparison is made, current cost data should be used.

C.10.3 Example (US customary units)

A residuum cooler is designed to cool 800 000 lb/h of atmospheric residuum from 340 °F to 200 °F. The design dry-bulb air temperature is 120 °F, and the minimum design air temperature is 32 °F. The residuum pour point is 95 °F, and the inlet and outlet viscosities are 6,0 and 48 centipoises, respectively. The inlet and outlet temperatures of the circulating water were selected as 135 °F and 254 °F, respectively, giving a flowrate of 500 000 lb/h. Note that the problem selected is based on a pressurized water system. Table C.4 describes the two systems, and Table C.5 compares their costs.

NOTE The direct air-cooled heat exchanger system described in this paragraph, although an actual application, may not represent the optimum design. In addition, the equipment costs given in this example are based on 1981 data and are for purposes of illustration only; when an actual economic comparison is made, current cost data should be used.

Table C.3 — Description of indirect and direct air cooling systems (SI units)

Item	Indirect system	Winterized air-cooled heat exchanger
	Equipment	
Shell-and-tube exchanger	1 486 m^2	—
Air-cooled heat exchanger	1 208 m^2	[a]
Pump	227 m^3/h	—
Piping/surge tank	DN 150/1,89 m^3	—
	Installation cost factors [b]	
Shell-and-tube exchanger	2,7	—
Air-cooled heat exchanger	1,8	1,8
Pump	2,5	—
Piping/surge tank	2,5	—
	Driver requirements	
Air-cooled heat exchanger fans	111,9 kW	160,3 kW
Pump	18,6 kW	—

NOTE DN = Nominal Diameter.

[a] The process duty of the winterized air-cooled heat exchanger is divided into two services, with a different exchanger configuration to handle each. Both configurations have full external recirculation and steam coils. For the 171,11 °C – 126,67 °C cooling range, the configuration consists of two bays 5,79 m wide and 9,14 m long, with two bundles per bay. Each bundle has five rows and 12 passes . The tubes have an outside diameter of 50,8 mm and have 394 aluminium fins (15,88 mm high) per metre. The tubes are in an equilateral triangular pattern and have a 101,6 mm pitch. The extended surface area is 12,289 square metres (806 square metres of bare tube surface area). For the 226,67 °C – 93,33 °C cooling range, the configuration consists of three parallel bays 4,88 m wide and 9,14 m long, with six bundles per bay. There are two parallel strings of three bundles in series. Each bundle has six rows and three passes. The tubes are bare, with an OD of 25,4 mm and a wall thickness of 2,77 mm. The tubes are in an equilateral triangular pattern and have a 44,45 mm transverse pitch. The surface area is 4 133 m^2.

[b] The installation cost factors may vary depending on location, labour costs, and the like.

Table C.4 — Description of indirect and direct air cooling systems (US customary units)

Item	Indirect system	Winterized air-cooled heat exchanger
	Equipment	
Shell-and-tube exchanger	16 000 ft^2	—
Air-cooled heat exchanger	13 000 ft^2	[a]
Pump	1 000 gal/min	—
Piping/surge tank	NPS 6/500 gal	—
	Installation cost factors [b]	
Shell-and-tube exchanger	2,7	—
Air-cooled heat exchanger	1,8	1,8
Pump	2,5	—
Piping/surge tank	2,5	—
	Driver requirements	
Air-cooled heat exchanger fans	150 brake HP	215 brake HP
Pump	25 brake HP	—

NOTE NPS = nominal pipe size.

[a] The process duty of the winterized air-cooled heat exchanger is divided into two services, with a different exchanger configuration to handle each. Both configurations have full external recirculation and steam coils. For the 340 °F to 260 °F cooling range, the configuration consists of two bays 19 ft wide and 30 ft long, with two bundles per bay. Each bundle has five rows and 12 passes. The tubes have an outside diameter (OD) of 2 inches and have ten $^5/_8$-inch-high aluminium fins per inch. The tubes are in an equilateral triangular pattern and have a 1 inch transverse pitch. The extended surface area is 142 800 ft^2 (8 671 ft^2 of bare tube surface area). For the 260 °F to 200 °F cooling range, the configuration consists of three parallel bays 16 ft wide and 30 ft long, with six bundles per bay. There are two parallel strings of three bundles in series. Each bundle has six rows and three passes. The tubes are bare, with an OD of 1 inch and a wall thickness of 0,109 inch. The tubes are in an equilateral triangular pattern and have a 1 $^3/_4$-inch transverse pitch. The surface area is 44 485 ft^2.

[b] The installation cost factors may vary depending on location, labour costs, and the like.

Table C.5 — Comparison of costs of indirect and direct air cooling systems

Item	Indirect system (US Dollars)	Winterized air-cooled heat exchanger (US Dollars)
	Equipment costs [a]	
Shell-and-tube exchanger	150 000	—
Air-cooled heat exchanger	287 000	1 136 000
Pump	10 000	—
Piping/surge tank	20 000	—
	Installed costs [b]	
Shell-and-tube exchanger	405 000	—
Air-cooled heat exchanger	516 000	2 045 000
Pump	25 000	—
Piping/surge tank	50 000	—
	Power costs [c]	
Air-cooled heat exchanger fans	225 000	322 000
Pump	37 000	—
	Evaluated total cost [d]	
System	1 258 000	2 367 000

[a] The equipment costs are based on 1981 data.

[b] The installed cost is obtained by multiplying the installation cost factor by the purchase price of the equipment in question. The cost includes associated piping, excavation, concrete, structural steel, electrical work, instrumentation, painting, and insulation.

[c] The electric power costs were calculated based on 1 500 US Dollars per brake horsepower (2 011 US Dollars per kilowatt).

[d] The evaluated total cost is equal to the sum of the installed costs and the power costs.

C.11 Calculation of minimum tube-wall temperature

C.11.1 Sample calculation of minimum tube-wall temperature for finned tubes (US customary units)

C.11.1.1 General

The forced-draught unit specified in C.11.1.2 is designed to cool 116 000 lb/h of gas oil product (with a gravity of 21,4° API and a UOP K of 11,5) from 290 °F to 160 °F, with an air inlet temperature of 90 °F. The designer desires to calculate the minimum tube-wall temperature at the outlet of the bottom row of tubes for an air inlet temperature of 10 °F and a minimum airflow of 147 000 lb/h, which is required to maintain the design process outlet temperature. The pour point of the gas oil is 50 °F.

C.11.1.2 Unit description

The unit consists of one bay containing one tube bundle that is 9,5 ft wide and 30 ft long, with 7 rows and 7 passes. The bundle contains 319 carbon steel tubes with an outside diameter of 1 inch and a wall thickness of 0,109 inch, arranged in an equilateral triangular pattern on a $2^1/_2$ inch pitch. The tubes have 10 aluminium fins per inch; the fins are $^5/_8$ inch high and 0,016 inch thick.

C.11.1.3 Data

The variables for which values are given below are defined in C.5.2.

A_i 0,204 7 ft²/ft

A_o 5,5 ft²/ft

f_a 1,2

r_{fs} 0,336 7 (°F)(ft²)(h)/BTU

r_{ft} 0,038 9 (°F)(ft²)(h)/BTU

r_m 0,025 1 BTU^{-1} (from Table C.6)

r_c $r_{ft}(A_o/A_i) + r_m + (r_{fs}/f_a)$

0,038 9 (5,5/0,204 7) + 0,025 1 + (0,336 7/1,2)

1,351 (°F)(ft²)(h)/BTU

t_B 10 °F

T_B 160 °F

T_w $T_B - (r_{ft}/r_c)(A_o/A_i)(T_B - t_B)$

160 – (0,038 9/1,351) (5,5/0,204 7) (160 – 10)

44 °F

The calculated tube-wall temperature is well below the recommended temperature of 75 °F (50 °F plus 25 °F for the safety margin). Consequently, the designer must consider a winterization method that will adequately protect this design against pour-point problems. In this situation, System C or D offers possible solutions to be investigated; however, the designer decides to solve this problem in another way, as shown in C.11.2.

C.11.2 Sample calculation of minimum tube-wall temperature of bare tubes (US customary units)

C.11.2.1 General

The designer decides to design the unit using both finned and bare tubes. The new design features a tube bundle with the upper rows of finned tubes and the lower rows of bare tubes. The amount of air at 10 °F required to satisfy the design process outlet temperature of 160 °F is the same as for the design in C.11.1.1. The designer now calculates the minimum tube-wall temperature at the outlet of the bottom row of bare tubes.

C.11.2.2 Unit description

The unit consists of one bay containing one tube bundle that is 9,5 ft wide and 30 ft long, with 8 rows and 8 passes. The top six rows of the bundle contain 273 carbon steel tubes with an outside diameter of 1 inch and a wall thickness of 0,109 inch, arranged in an equilateral triangular pattern on a $2^1/_2$ inch pitch. These tubes have 10 aluminium fins per inch; the fins are $^5/_8$ inch high and 0,016 inch thick. In addition, the bundle contains 166 carbon steel bare tubes in the bottom two tube rows. These tubes also have an outside diameter of 1 inch and a wall thickness of 0,109 inch and are arranged in an equilateral triangular pattern on a $1^3/_8$ inch pitch.

C.11.2.3 Data

The data for the bottom row of bare tubes are as follows:

A_b 0,261 8 ft^2/ft

A_i 0,204 7 ft^2/ft

r_{fs} 0,153 (°F)(ft^2)(h)/BTU

r_{ft} 0,058 (°F)(ft^2)(h)/BTU

r_{mt} 0,000 4 (°F)(ft^2)(h)/BTU

r_c r_{ft} $(A_b/A_i) + r_{mt} + (r_{fs}/f_a)$

0,058 (0,261 8/0,204 7) + 0,000 4 + (0,153/1,2)

0,202 (°F)(ft^2)(h)/BTU

T_w $T_B - (r_{ft}/r_c)\ (A_b/A_i)\ (T_B - t_B)$

160 – (0,058/0,202) (0,261 8/0,204 7) (160 – 10)

105 °F

The calculated tube-wall temperature at the outlet of the bottom row of bare tubes is well above 75 °F, and no gas oil freeze-up is anticipated. To verify that the finned section of the bundle is also protected, the designer decides to calculate the tube-wall temperature of the outlet of the sixth pass (the row of finned tubes immediately above the bare tubes). From the thermal design calculations, the designer obtains the following data:

r_{fs} 0,335 6 (°F)(ft^2)(h)/BTU

r_{ft} 0,023 6 (°F)(ft^2)(h)/BTU

r_c r_{ft} $(A_o/A_i) + r_m + (r_{fs}/f_a)$

0,023 6 (5,5/0,204 7) + 0,025 1 + (0,335 6/1,2)

0,939 (°F)(ft^2)(h)/BTU

t_B 31,3 °F (air temperature leaving the bare tube section)

T_B 173,4 °F (gas oil temperature leaving the sixth pass)

T_w $T_B - (r_{ft}/r_c)\ (A_o/A_i)\ (T_B - t_B)$

173,4 – (0,023 6/0,939) (5,5/0,204 7) × (173,4 – 31,3)

77,4 °F (> 75 °F)

It appears that this design is safe against a potential freeze-up for continuous operation. As in all viscous fluid coolers, however, an auxiliary heating coil is recommended for cold start-up.

C.11.3 Sample calculation of minimum tube-wall temperature for finned tubes (SI units)

C.11.3.1 General

The forced-draught unit specified in C.11.3.2 is designed to cool 52 618 kg/h of gas oil product (with a gravity of 21,4° API and a UOP K of 11,5) from 143 °C to 71 °C, with an air inlet temperature of 32 °C. The designer desires to calculate the minimum tube-wall temperature at the outlet of the bottom row of tubes for an air inlet temperature of – 12 °C and a minimum airflow of 66 679 kg/h, which is required to maintain the design process outlet temperature. The pour point of the gas oil is 10 °C.

C.11.3.2 Unit description

The unit consists of one bay containing one tube bundle that is 2,90 m long, with 7 rows and 7 passes. The bundle contains 319 carbon steel tubes with an outside diameter of 25,4 mm and a wall thickness of 2,77 mm, arranged in an equilateral triangular pattern on a 63,5 mm pitch. The tubes have 394 aluminium fins per metre; the fins are 15,9 mm high and 0,4 mm thick.

C.11.3.3 Data

The variables for which values are given below are defined in C.5.2.

A_i 0,062 4 m^2/m

A_o 1,68 m^2/m

f_a 1,2

r_{fs} 0,059 3 m^2·K/W

r_{ft} 0,006 85 m^2·K/W

r_m 0,004 43 m^2·K/W (from Table C.7)

r_c $r_{ft}\,(A_o/A_i) + r_m + (r_{fs}/f_a)$

0,006 85 (1,68/0,062 4) + 0,004 43 + (0,059 3/1,2)

0,238 m^2·K/W

t_B – 12 °C

T_B 71 °C

T_w $T_B - (r_{ft}/r_c)\,(A_o/A_i)\,(T_B - t_B)$

71 – (0,006 85/0,238) (1,68/0,062 4) [71 – (– 12)]

6,7 °C

The calculated tube-wall temperature is well below the recommended temperature of 23,9 °C (10 °C plus 13,9 °C for the safety margin). Consequently, the designer must consider a winterization method that will adequately protect this design against pour-point problems. In this situation, System C or D offers possible solutions to be investigated; however, the designer decides to solve this problem in another way, as shown in C.11.4.

C.11.4 Sample calculation of minimum tube-wall temperature of bare tubes (SI units)

C.11.4.1 General

The designer decides to design the unit using both finned and bare tubes. The new design features a tube bundle with the upper rows of finned tubes and the lower rows of bare tubes. The amount of air at – 12 °C required to satisfy the design process outlet temperature of 71 °C is the same as for the design in C.11.3.1. The designer now calculates the minimum tube-wall temperature at the outlet of the bottom row of bare tubes.

C.11.4.2 Unit description

This unit consists of one bay containing one tube bundle that is 2,90 m wide and 9,14 m long, with 8 rows and 8 passes. The top six rows of the bundle contain 273 carbon steel tubes with an outside diameter of 25,4 mm and a wall thickness of 2,77 mm, arranged in an equilateral triangular pattern on a 63,5 mm pitch. These tubes have 394 aluminium fins per metre; the fins are 15,9 mm high and 0,4 mm thick. In addition, the bundle contains 166 carbon steel bare tubes in the bottom two tube rows. These tubes also have an outside diameter of 25,4 mm and a wall thickness of 2,77 mm and are arranged in an equilateral triangular pattern on a 34,9 mm pitch.

C.11.4.3 Data

The data for the bottom row of bare tubes are as follows:

A_b 0,079 8 m^2/m

A_i 0,062 4 m^2/m

r_{fs} 0,062 9 $m^2 \cdot K/W$

r_{ft} 0,010 2 $m^2 \cdot K/W$

r_{mt} 0,000 070 $m^2 \cdot K/W$

r_c $r_{ft}(A_b/A_i) + r_{mt} + (r_{fs}/f_a)$

$0{,}010\,2\,(0{,}079\,8/0{,}062\,4) + 0{,}000\,070 + (0{,}026\,9/1{,}2)$

0,035 5 $m^2 \cdot K/W$

T_w $T_B - (r_{ft}/r_c)(A_b/A_i)(T_B - t_B)$

$71 - (0{,}010\,2/0{,}035\,5)(0{,}079\,8/0{,}062\,4) \times [71 - (-12)]$

40,5 °C

The calculated tube-wall temperature at the outlet of the bottom row of bare tubes is well above 23,9 °C, and no gas oil freeze-up is anticipated. To verify that the finned section of the bundle is also protected, the designer decides to calculate the tube-wall temperature at the outlet of the sixth pass (the row of finned tubes immediately above the bare tubes). From the thermal design calculations, the designer obtains the following data:

r_{fs} 0,059 $m^2 \cdot K/W$

r_{ft} 0,00415 $m^2 \cdot K/W$

r_c $r_{ft}(A_o/A_i) + r_m + (r_{fs}/f_a)$

$0{,}004\,15\,(1{,}68/0{,}062\,4) + 0{,}004\,4 + (0{,}059/1{,}2)$

0,165 3 $m^2 \cdot K/W$

t_B – 0,4 °C (air temperature leaving the bare tube section)

T_B 78,5 °C (gas oil temperature leaving the sixth pass)

T_w $T_B - (r_{ft}/r_c)(A_o/A_i)(T_B - t_B)$

$78{,}5 - (0{,}004\,15/0{,}165\,3)(1{,}68/0{,}062\,4) \times [78{,}5 - (-0{,}4)]$

25,2 °C (> 23,9 °C)

It appears that this design is safe against a potential freeze-up for continuous operation. As in all viscous fluid coolers, however, an auxiliary heating coil is recommended for cold start-up.

Table C.6 — Values for r_m referred to the total outside area for a tube with an outside diameter of 1 inch and aluminium fins $^5/_8$ inch high and 0,016 inch thick, in (degrees Fahrenheit) (square foot) (hour) per British thermal unit

Tube material	k[a]	No. of fins per inch	Tube wall thickness (inches) 0,035	0,049	0,065	0,083	0,109	0,134
			r_m (°F·ft²·h/BTU)					
Admiralty	70	7	0,016 77	0,017 08	0,017 45	0,017 87	0,018 52	0,019 18
		8	0,017 00	0,017 35	0,017 77	0,018 25	0,018 98	0,019 43
		9	0,017 20	0,017 60	0,018 06	0,018 60	0,019 42	0,020 25
		10	0,017 39	0,017 82	0,018 33	0,018 93	0,019 84	0,020 76
		11	0,017 55	0,018 03	0,018 59	0,019 25	0,020 24	0,021 25
Aluminium	90	7	0,016 55	0,016 77	0,017 02	0,017 32	0,017 78	0,018 24
		8	0,016 75	0,017 00	0,017 29	0,017 63	0,018 14	0,018 66
		9	0,016 92	0,017 20	0,017 52	0,01,90	0,018 48	0,019 06
		10	0,017 08	0,017 38	0,017 74	0,018 16	0,018 79	0,019 44
		11	0,017 22	0,017 55	0,017 94	0,018 40	0,019 09	0,019 80
Carbon steel	26	7	0,017 89	0,018 67	0,019 59	0,020 67	0,022 30	0,023 96
		8	0,018 27	0,019 16	0,020 20	0,021 42	0,023 27	0,025 15
		9	0,018 62	0,019 61	0,020 78	0,022 14	0,024 21	0,026 31
		10	0,018 96	0,020 05	0,021 34	0,022 85	0,025 13	0,027 45
		11	0,019 27	0,020 47	0,021 88	0,023 53	0,026 03	0,028 58
Stainless steel (Types 302, 304, 316, 321 and 347)	9,3	7	0,020 68	0,022 63	0,024 93	0,027 62	0,031 69	0,035 84
		8	0,021 43	0,023 64	0,026 25	0,029 30	0,033 92	0,038 62
		9	0,022 15	0,024 63	0,027 55	0,030 95	0,036 11	0,041 37
		10	0,022 86	0,025 59	0,028 82	0,032 58	0,038 29	0,044 09
		11	0,023 55	0,026 54	0,030 08	0,034 20	0,040 45	0,046 81

NOTE The tabulated values are based on an assumed maximum air-side resistance (r_{fs}, or $r_{fs} + r_{ds}$) of 0,15. Assume fin efficiency ϕ = 1.

[a] k = thermal conductivity, in British thermal units (hour) (square foot) (degree Fahrenheit) per foot.

Table C.7 — Values for r_m referred to the total outside area for a tube with an outside diameter of 25,4 mm and aluminium fins 15,9 mm high and 0,4 mm thick, in (square metres) (kelvin) per watt

Tube material	k[a]	No. of fins per metre	Tube wall thickness (mm)					
			0,89	1,24	1,65	2,10	2,77	3,40
			r_m ($m^2 \cdot K/W$)					
Admiralty	121	276	0,002 95	0,003 01	0,003 07	0,003 15	0,003 26	0,003 38
		315	0,002 99	0,003 06	0,003 13	0,003 21	0,003 34	0,003 42
		354	0,003 03	0,003 10	0,003 18	0,003 28	0,003 42	0,003 57
		394	0,003 06	0,003 14	0,003 23	0,003 33	0,003 49	0,003 66
		433	0,003 09	0,003 18	0,003 27	0,003 39	0,003 56	0,003 74
Aluminium	155,6	276	0,002 91	0,002 95	0,003 00	0,003 05	0,003 13	0,003 21
		315	0,002 95	0,002 99	0,003 04	0,003 10	0,003 19	0,003 29
		354	0,002 98	0,003 03	0,003 09	0,003 1S	0,003 25	0,003 36
		394	0,003 01	0,003 06	0,003 12	0,003 20	0,003 31	0,003 42
		433	0,003 03	0,003 09	0,003 16	0,003 24	0,003 36	0,003 49
Carbon steel	45,0	276	0,003 15	0,003 29	0,003 45	0,003 64	0,003 93	0,004 22
		315	0,003 22	0,003 37	0,003 56	0,003 77	0,004 10	0,004 43
		354	0,003 28	0,003 45	0,003 66	0,003 90	0,004 26	0,004 63
		394	0,003 34	0,003 53	0,003 76	0,004 02	0,004 43	0,004 83
		433	0,003 39	0,003 60	0,003 85	0,004 14	0,004 58	0,005 03
Stainless steel	16,08	276	0,003 64	0,003 99	0,004 39	0,004 86	0,005 58	0,006 31
(Types 302,		315	0,003 77	0,004 16	0,004 62	0,005 16	0,005 97	0,006 80
304, 316		354	0,003 90	0,004 34	0,004 85	0,005 45	0,006 36	0,007 29
321 and 347)		394	0,004 03	0,004 51	0,005 08	0,005 74	0,006 74	0,007,6
		433	0,004 15	0,004 67	0,005 30	0,006 02	0,007 12	0,008 24

NOTE The tabulated values are based on an assumed maximum air-side resistance (r_{fs}, or $r_{fs} + r_{ds}$) of 0,026 4. Assume fin efficiency $\phi = 1$.

[a] k = thermal conductivity, in watts per (metre kelvin).

C.12 Sample calculation of heat losses

C.12.1 Sample calculation of heat losses (US customary units)

C.12.1.1 General

In the examples in this clause, the temperature differential between the ambient air and the enclosed air must be established. In the calculation determining the minimum heat-input requirement for an auxiliary heating coil, the temperature of the air enclosed in the plenum surrounding the process coil should be the temperature to which it is desired to warm the process bundle.

Within a heated enclosure, air near the top will be hotter than air near the bottom. An inside air temperature will be assumed for the top of the enclosure as well as the bottom. These assumed air temperatures are not recommended air temperatures but are simply assumed values used to illustrate the calculation procedure.

Note that the louvre area is assumed to be the same as the face area of the tube bundle — this is not always the case.

C.12.1.2 Nomenclature

C.12.1.2.1 Symbols

A_l louvre leakage area, in square feet. (In the calculations below, A_l is assumed to be 2 % of the tube bundle face area.)

c_p average specific heat capacity, in British thermal units per (pound) (degree Fahrenheit) (taken as 0,24 for air)

F_p pressure promoting leakage, in feet of fluid

g acceleration due to gravity

= 32,17 ft/s^2

h height of the hot air column, in feet

K local heat transfer coefficient, in British thermal units per (hour) (square foot) (degree Fahrenheit)

ϕ heat loss per unit time, in British thermal units per hour

T temperature, in degrees Fahrenheit

U thermal transmittance, in British thermal units per (hour) (square foot) (degree Fahrenheit)

v velocity, in feet per second

q_m flow per unit time, in pounds per hour

ρ density of air, in pounds per cubic foot

C.12.1.2.2 Subscripts

o relating to conditions outside the air-cooled heat exchanger

i relating to conditions inside the air-cooled heat exchanger

C.12.1.3 Louvre leakage

C.12.1.3.1 Air that is warmer than ambient air and is contained in an unsealed enclosure tries to rise within that enclosure and exerts a pressure on the upper surface. This causes leakage when the upper surface consists of non-sealing louvre blades. The pressure promoting leakage may be expressed as follows:

$$F_p = [h(\rho_o - \rho_i)] / \rho_i \tag{C.9}$$

The velocity through the leak areas (assuming a loss of 1,5 velocity heads) is:

$$v = (2gF_p / 1,5)^{1/2} \tag{C.10}$$

The rate at which warm air leaks through the louvres is:

$$q_m = 3\,600\ v\rho_i A_i \tag{C.11}$$

The rate of heat loss due to louvre leakage is:

$$\phi = q_m c_p (T_i = T_o) \qquad \text{(C.12)}$$

A sample calculation of heat loss due to louvre leakage is presented in C.12.1.3.2.

C.12.1.3.2 Assume the following conditions: a totally enclosed air-cooled heat exchanger is 14 ft wide, 36 ft long, and 8 ft high. The inside air temperature T_i is 100 °F, and the outside air temperature T_o is 0 °F. Assuming that the perfect-gas laws apply, the air density can be determined from equation (C.13):

$$\rho = (\mathrm{M}p) / (\mathrm{R}T) \qquad \text{(C.13)}$$

where

M = molecular weight of air = 28,96

p = pressure, in pounds per square inch absolute

R = gas constant
= 10,73 (cubic feet) (pounds per square inch absolute) per (pound-mole) (degrees Rankine)

T = temperature of air, in degrees Rankine

Therefore, the outside air density is

$$\rho_o = [(28,96)\ (14,70)] / [(10,73)\ (0 + 459,67)]$$
$$= 0,0863 \text{ lb/ft}^3.$$

The inside air density is

$$\rho_i = [(28,96)\ (14,70)] / [(10,73)\ (100 + 459,67)]$$
$$= 0,070\ 9 \text{ lb/ft}^3.$$

The pressure promoting leakage through the louvres is determined using equation (C.9) as follows:

$$F_p = [h(\rho_o - \rho_i)] / \rho_i$$
$$= [8\ (0,086\ 3 - 0,070\ 9)] / 0,070\ 9$$
$$= 1,738 \text{ ft of air.}$$

The velocity through the louvre leakage area resulting from this pressure is calculated using equation (C.10) as follows:

$$v = (2gF_p/1,5)^{1/2}$$
$$= [(2)\ (32,17)\ (1,738)/1,5]^{1/2}$$
$$= 8,63 \text{ ft/s.}$$

The air leakage rate through the louvres is calculated using equation (C.11) as follows:

$$q_m = 3\ 600\ v\rho_i A_i$$
$$= (3\ 600)\ (8,63)\ (0,070\ 9)\ [(0,02)\ (14)\ (36)]$$
$$= 22\ 203 \text{ lb/h.}$$

The rate of heat loss resulting from louvre leakage is then determined using equation (C.12) as follows:

$$\phi = q_m c_p (T_i - T_o)$$

$$= (22{,}203)\,(0{,}24)\,(100 - 0)$$

$$= 532\ 900 \text{ BTU/h.}$$

C.12.1.4 Surface heat loss

C.12.1.4.1 The heat lost by convection from the exterior surfaces of the plenum is a function of temperature difference, wind velocity, and surface area. For velocities less than 16 ft/s, the following equation from [13] is recommended for determining the heat transfer coefficient K for airflow parallel to flat surfaces:

$$K = 0{,}99 + 0{,}21v \qquad \text{(C.14)}$$

NOTE This equation is derived for vertical surfaces; for simplicity, however, it is used here for all surfaces, since most are vertical.

For velocities of 16 ft/s and higher, the following equation is recommended:

$$K = 0{,}5\,(v)^{0{,}78} \qquad \text{(C.15)}$$

A sample calculation of heat loss by convection from an air-cooled heat exchanger is presented in C.1.4.2.

C.12.1.4.2 Assume the following conditions: A totally enclosed air-cooled heat exchanger is 18 ft wide, 36 ft long, and 16 ft high (from grade to the top of the louvres). The inside air temperature varies linearly from 100 °F at the top to 50 °F at the bottom. The outside air temperature is 0 °F. The inside air velocity is 2 ft/s. The outside wind velocity is 30 ft/s.

The thermal transmittance, U, is calculated as follows:

$$K_i = 0{,}99 + 0{,}21v$$

$$= 0{,}99 + (0{,}21)\,(2)$$

$$= 1{,}41 \text{ BTU/(h·ft}^2\text{·°F).}$$

$$K_o = 0{,}5\,(v)^{0{,}78}$$

$$= 0{,}5\,(30)^{0{,}78}$$

$$= 7{,}09 \text{ BTU/(h·ft}^2\text{·°F).}$$

$$U = 1/[(1/K_i) + (1/K_o)]$$

$$= 1/[(1/1{,}41) + (1/7{,}09)]$$

$$= 1{,}176 \text{ BTU/(h·ft}^2\text{·°F).}$$

The total surface heat loss rate is determined as follows:

$$\phi = UA\,(T_i - T_o)$$

$$= (1{,}17)\,\{(18)\,(36)\,(100 - 0) + (16)\,(18 + 18 + 36 + 36)\,[(100 + 50)/2 - 0]\}$$

$$= 227\ 400 \text{ BTU/h.}$$

C.12.1.5 Total heat loss calculation

C.12.1.5.1 Forced draught with external recirculation

Assume that an enclosure is 18 ft wide, 36 ft long, and 18 ft high. The dimensions of the top louvre area are 14 ft by 36 ft. Inlet air louvres are located on the sides near the bottom. There is a hot air column that is 9 ft high between the heating coil and the top louvres. Assume an outside air temperature of 0 °F and an inside air temperature that varies linearly from 100 °F above the heating coil to 50 °F at the bottom of the enclosure.

The heat loss through the top louvres may be calculated as follows:

ρ_o = 0,086 3 lb/ft^3.

ρ_i = 0,070 9 lb/ft^3

The pressure promoting leakage resulting from the effect of the hot-air column above the heating coil is determined as follows:

$$F_{p1} = [h(\rho_\mathrm{o} - \rho_\mathrm{i})] / \rho_\mathrm{i}$$

$$= [(9)\ (0{,}086\ 3 - 0{,}070\ 9)] / 0{,}070\ 9$$

$$= 1{,}95 \text{ ft of air.}$$

The air density at the average temperature below the heating coil is

$$\rho_\mathrm{i} = (\mathrm{M}p)/(\mathrm{R}T_\mathrm{i})$$

$$= [(28{,}96)\ (14{,}7)] \div \{(10{,}73)\ [(100 + 50)/2 + 459{,}67]\}$$

$$= 0{,}074\ 2 \text{ lb/ft}^3.$$

The pressure promoting leakage below the heating coil is determined by

$$F_{p2} = [h(\rho_\mathrm{o} - \rho_\mathrm{i})] / \rho_\mathrm{i}$$

$$= [9\ (0{,}086\ 3 - 0{,}074\ 2)] / 0{,}074\ 2$$

$$= 1{,}47 \text{ ft of air.}$$

The total pressure promoting leakage is the sum of the pressures above and below the heating coil:

$$F_{p\mathrm{d}} = F_{p1} + F_{p2}$$

$$= 1{,}95 + 1{,}47$$

$$= 3{,}42 \text{ ft of air.}$$

This pressure drop, however, assumes no restriction of the inlet air to the exchanger. Since the entire exchanger is enclosed, the entering air must come through the inlet louvres. The quantity of inlet air must equal the quantity of exhaust air. It can be assumed that the effective pressure promoting leakage, F_p* will be equally divided between the inlet and exhaust louvres. Therefore:

$$F_p^* = 3{,}42/2$$

$$= 1{,}71 \text{ ft of air.}$$

The velocity through the louvre leak area is

$$v = (2gF_p^*/1{,}5)^{1/2}$$

$$= [(2)\ (32{,}17)\ (1{,}71)/1{,}5]^{1/2}$$

$$= 8{,}56 \text{ ft/s.}$$

The heat loss resulting from louvre leakage is then:

$$\phi = q_\mathrm{m} c_p\ (T_\mathrm{i} - T_\mathrm{o})$$

$$= [(3\ 600)\ (8{,}56)\ (0{,}070\ 9)]\ [(0{,}02)\ (14)\ (36)]\ [(0{,}24)\ (100 - 0)]$$

$$= 528\ 600 \text{ BTU/h.}$$

The thermal transmittance, U, is 1,176 BTU/(h·ft^2·°F). The surface heat loss is:

$$\begin{aligned}\phi &= UA\,(T_\text{i} - T_\text{o})\\ &= 1{,}176\,[(9)\,(18 + 18 + 36 + 36)\,(100 - 0)]\\ &\quad + 1{,}176\,\{(9)\,(18 + 18 + 36 + 36)\,[(100 + 50)/2 - 0]\}\\ &\quad + 1{,}176\,[(18)\,(36)]\,(100 - 0)\\ &= 276\,200 \text{ BTU/h.}\end{aligned}$$

C.12.1.5.2 Forced draught without louvres

This case is discussed to show that heat loss is from four to eight times greater without top louvres than when louvres are present. This loss is caused by an unimpeded natural draught of air through the tube bundle. Under such conditions, it is reasonable to assume an air velocity of 50 ft/min at the bundle face. Assume the following air-cooled heat exchanger geometry, as used in the example in C.12.1.3: A totally enclosed air-cooled heat exchanger is 14 ft wide, 36 ft long, and 8 ft high. Also assume that the air is heated from 0 °F to 100 °F.

The heat loss is calculated as follows: The warm air loss through the bundle is:

$$\begin{aligned}q_\text{m} &= 3\,600\, v\rho_\text{i}A\\ &= (3\,600)\,(50/60)\,(0{,}070\,9)\,[(14)\,(36)]\\ &= 107\,200 \text{ lb/h.}\end{aligned}$$

This leads to the following heat loss:

$$\begin{aligned}\phi &= q_\text{m}c_p\,(T_\text{i} - T_\text{o})\\ &= (107\,200)\,(0{,}24)\,(100 - 0)\\ &= 2\,572\,000 \text{ BTU/h.}\end{aligned}$$

For other designs that may require analysis, it is recommended that the principles shown in these examples be applied.

C.12.2 Sample calculation of heat losses (SI units)

C.12.2.1 General

In the examples in this clause, the temperature differential between the ambient air and the enclosed air must be established. In the calculation determining the minimum heat input requirement for an auxiliary heating coil, the temperature of the air enclosed in the plenum surrounding the process coil should be the temperature to which it is desired to warm the process bundle.

Within a heated enclosure, air near the top will be hotter than air near the bottom. An inside air temperature will be assumed for the top of the enclosure as well as the bottom. These assumed air temperatures are not recommended air temperatures but are simply assumed values used to illustrate the calculation procedure.

Note that the louvre area is assumed to be the same as the face area of the tube bundle — this is not always the case.

C.12.2.2 Nomenclature

C.12.2.2.1 Symbols

A_l louvre leakage area, in square metres. (In the calculations below, A_l is assumed to be 2 % of the tube bundle face area.)

c_p average specific heat capacity, in kJ/(kg·K) (taken as 1,005 for air)

F_p pressure promoting leakage, in metres of fluid

g acceleration due to gravity

= 9,807 m/s^2

h height of the hot air column, in metres

K local heat transfer coefficient, in W/(m^2·K)

ϕ heat loss per unit time, in W

T temperature, in °C

U thermal transmittance, in W/(m^2·K)

v velocity, in m/s

q_m flow per unit time, in kg/h

ρ density of air, in kg/m^3

C.12.2.2.2 Subscripts

o relating to conditions outside the air-cooled heat exchanger

i relating to conditions inside the air-cooled heat exchanger

C.12.2.3 Louvre leakage

C.12.2.3.1 Air that is warmer than ambient air and is contained in an unsealed enclosure tries to rise within that enclosure and exerts a pressure on the upper surface. This causes leakage when the upper surface consists of nonsealing louvre blades. The pressure promoting leakage may be expressed as follows:

$$F_p = [h(\rho_o - \rho_i)] / \rho_i \tag{C.16}$$

The velocity through the leak areas (assuming a loss of 1,5 velocity heads) is:

$$v = (2gF_p/1,5)^{1/2} \tag{C.17}$$

The rate at which warm air leaks through the louvres is:

$$q_m = 3\,600\ v\rho_i A_i \tag{C.18}$$

The rate of heat loss due to louvre leakage is:

$$\phi = q_m c_p\ (T_i - T_o) \tag{C.19}$$

A sample calculation of heat loss due to louvre leakage is presented in C.12.2.3.2.

C.12.2.3.2 Assume the following conditions: A totally enclosed air-cooled heat exchanger is 4,27 m wide, 10,97 m long, and 2,44 m high. The inside air temperature T_i is 37,78 °C, and the outside air temperature T_o is – 17,78 °C. Assuming that the perfect-gas laws apply, the air density can be determined from equation (C.20):

$$\rho = (\mathrm{M}p)/(\mathrm{R}T) \tag{C.20}$$

where

M = molecular weight of air
= 28,96

p = absolute pressure, in kilopascals

R = gas constant
= 8,31 J/(mol·K)

T = temperature, in kelvins

Therefore, the outside air density is:

$$\rho_o = [(28{,}96)\ (101{,}33)] / [(8{,}31)\ (-\ 17{,}78 + 273{,}15)]$$
$$= 1{,}383\ \mathrm{kg/m^3}$$

The inside air density is

$$\rho_i = [(28{,}96)\ (101{,}33)] / [(8{,}31)\ (37{,}78 + 273{,}15)]$$
$$= 1{,}136\ \mathrm{kg/m^3}$$

The pressure promoting leakage through the louvres is determined using equation (C.16) as follows:

$$F_p = [h(\rho_o - \rho_i)] / \rho_i$$
$$= [2{,}44\ (1{,}383 - 1{,}136)] / 1{,}136$$
$$= 0{,}531 \text{ metres of air.}$$

The velocity through the louvre leakage area resulting from this pressure is calculated using equation (C.17) as follows:

$$v = (2gF_p/1{,}5)^{1/2}$$
$$= [(2)\ (9{,}807)\ (0{,}531)/1{,}5]^{1/2}$$
$$= 2{,}63\ \mathrm{m/s.}$$

The air leakage rate through the louvres is calculated using equation (C.18) as follows:

$$q_m = 3\,600\ v\rho_i A_i$$
$$= (3\,600)\ (2{,}63)\ (1{,}136)\ [(0{,}02)\ (4{,}27)\ (10{,}97)]$$
$$= 10\,076\ \mathrm{kg/h.}$$

The rate of heat loss resulting from louvre leakage is then determined using equation (C.19) as follows:

$$\phi = q_m c_p\,(T_i - T_o)$$
$$= (10\,076)\ [(1{,}005)\ (100/3\,600)]\ [37{,}78 - (-\ 17{,}78)]$$
$$= 156\,300\ \mathrm{W.}$$

C.12.2.4 Surface heat loss

C.12.2.4.1 The heat lost by convection from the exterior surfaces of the plenum is a function of temperature difference, wind velocity, and surface area. For velocities less than 4,88 m/s, the following equation from [13] is recommended for determining the heat transfer coefficient for airflow parallel to flat surfaces:

$$K = 7{,}88 + 0{,}21v \qquad \text{(C.21)}$$

NOTE This equation is derived for vertical surfaces; for simplicity, however, it is used here for all surfaces, since the majority are vertical.

For velocities of 4,88 m/s and higher, the following equation is recommended:

$$K = 7{,}17(v)^{0{,}78} \qquad \text{(C.22)}$$

A sample calculation of heat loss by convection from an air-cooled heat exchanger is presented in C.12.2.4.2.

C.12.2.4.2 Assume the following conditions: A totally enclosed air-cooled heat exchanger is 5,49 m wide, 10,97 m long, and 4,88 m high (from grade to the top of the louvres). The inside air temperature varies linearly from 37,78 °C at the top to 10,0 °C at the bottom. The outside air temperature is – 17,78 °C. The inside air velocity is 0,61 m/s. The outside wind velocity is 9,14 m/s.

The thermal transmittance, U, is calculated as follows:

$$\begin{aligned} K_{\mathrm{i}} &= 7{,}88 + 0{,}21v \\ &= 7{,}88 + (0{,}21)\,(0{,}61) \\ &= 8{,}01\ \mathrm{W/m^2{\cdot}K} \end{aligned}$$

$$\begin{aligned} K_{\mathrm{o}} &= 7{,}17\ (v)^{0{,}78} \\ &= 7{,}17\ (9{,}l4)^{0{,}78} \\ &= 40{,}26\ \mathrm{W/m^2{\cdot}K} \end{aligned}$$

$$\begin{aligned} U &= 1/[(1/K_{\mathrm{i}}) + (1/K_{\mathrm{o}})] \\ &= 1/[(1/8{,}01) + (1/40{,}26)] \\ &= 6{,}68\ \mathrm{W/m^2{\cdot}K} \end{aligned}$$

The total surface heat loss is determined as follows:

$$\begin{aligned} \phi &= UA\ (T_{\mathrm{i}} - T_{\mathrm{o}}) \\ &= 6{,}68\ \{[(5{,}49)\ (10{,}97)]\ [37{,}78 - (-17{,}78)] + [(4{,}88)\ (5{,}49 + 5{,}49 + 10{,}97 + 10{,}97)]\ [(37{,}78 + 10)/2 - (-17{,}78)]\} \\ &= 67\ 000\ \mathrm{W}. \end{aligned}$$

C.12.2.5 Total heat loss calculation

C.12.2.5.1 Forced draught with external recirculation

Assume that an enclosure is 5,49 m wide, 10,97 m long, and 5,49 m high. The dimensions of the top louvre area are 4,27 m by 10,97 m. Inlet air louvres are located on the sides near the bottom. There is a hot air column that is 2,74 m high between the heating coil and the top louvres. Assume an outside air temperature of – 17,78 °C and an inside air temperature that varies linearly from 37,78 °C above the heating coil to 10,0 °C at the bottom of the enclosure.

The heat loss through the top louvres may be calculated as follows:

ρ_o = 1,383 kg/m^3

ρ_i = 1,136 kg/m^3

The pressure promoting leakage resulting from the effect of the hot air column above the heating coil is determined as follows:

$$F_{p1} = [h(\rho_o - \rho_i)] / \rho_i$$
$$= [2{,}74\ (1{,}383 - 1{,}136)] / 1{,}136$$
$$= 0{,}596 \text{ metres of air.}$$

The air density at the average temperature below the heating coil is

$$\rho_i = (\mathrm{M}p)/(\mathrm{R}T)$$
$$= [(28{,}96)\ (101{,}33)] \div \{(8{,}31)\ [(37{,}78 + 10{,}0)/2 + 273{,}15]\}$$
$$= 1{,}189 \text{ kg/m}^3$$

The pressure promoting leakage below the heating coil is determined by

$$F_{p2} = [h(\rho_o - \rho_i)] / \rho_i$$
$$= [2{,}74\ (1{,}383 - 1{,}189)] / 1{,}189$$
$$= 0{,}447 \text{ metres of air.}$$

The total pressure promoting leakage is the sum of the pressures above and below the heating coil:

$$F_{pd} = F_{p1} + F_{p2}$$
$$= 0{,}596 + 0{,}447$$
$$= 1{,}043 \text{ metres of air.}$$

This pressure drop, however, assumes no restriction of the inlet air to the exchanger. Since the entire exchanger is enclosed, the entering air must come through the inlet louvres. The quantity of inlet air must equal the quantity of exhaust air. It can be assumed that the effective pressure promoting leakage, $F_p{}^*$, will be equally divided between the inlet and the exhaust louvres. Therefore,

$$F_p{}^* = 1{,}043/2$$
$$= 0{,}522 \text{ metres of air.}$$

The velocity through the louvre leak area is

$$v = (2gF_p{}^*/1{,}5)^{1/2}$$
$$= [(2)\ (9{,}807)\ (0{,}522)/1{,}5]^{1/2}$$
$$= 2{,}61 \text{ m/s.}$$

The rate of heat loss resulting from louvre leakage is then

$$\phi = q_m c_p\ (T_i - T_o)$$
$$= [(3\ 600)\ (2{,}61)\ (1{,}136)]\ [(0{,}02)\ (4{,}27)\ (10{,}97)]\ [(1{,}005)\ (1\ 000/3\ 600)]\ [37{,}78 - (-\ 17{,}78)]$$
$$= 155\ 100 \text{ W.}$$

The thermal transmittance, U, is 6,68 W/m^2·K. The rate of surface heat loss is

$$\begin{aligned}\phi &= UA\,(T_\mathrm{i} - T_\mathrm{o})\\ &= 6{,}68\,(2{,}74)\,(5{,}49 + 5{,}49 + 10{,}97) \times [37{,}78 - (-17{,}78)] + 6{,}68\,(2{,}74)\,(5{,}49 + 5{,}49 + 10{,}97 + 10{,}97)\\ &\quad \times [(37{,}78 + 10{,}0)/2 - (-17{,}78)] + 6{,}68\,[(5{,}49)\,(10{,}97)]\,[(37{,}78 - (-17{,}78)]\\ &= 80\,900 \text{ W.}\end{aligned}$$

C.12.2.5.2 Forced draught without louvres

This case is discussed to show that heat loss is from four to eight times greater without top louvres than when louvres are present. This loss is caused by an unimpeded natural draught of air through the tube bundle. Under such conditions, it is reasonable to assume an air velocity of 15,24 m/min at the bundle face. Assume the following air-cooled heat exchanger geometry, as used in the example in C.2.3: A totally enclosed air-cooled heat exchanger is 4,27 m wide, 10,97 m long, and 2,44 m high. Also assume that the air is heated from – 17,78 °C to 37,78 °C.

The heat loss is calculated as follows.

The rate of warm air loss through the bundle is:

$$\begin{aligned}q_\mathrm{m} &= 3\,600\; v\rho_\mathrm{i}A\\ &= (3\,600)\,(15{,}24/60)\,(1{,}136)\,[(4{,}27)\,(10{,}97)]\\ &= 48\,700 \text{ kg/h.}\end{aligned}$$

This leads to the following rate of heat loss:

$$\begin{aligned}\phi &= q_\mathrm{m}c_p\,(T_\mathrm{i} - T_\mathrm{o})\\ &= (48\,700)\,[(1{,}005)\,(1\,000/3\,600)]\,[(37{,}78 - (-17{,}78)]\\ &= 754\,700 \text{ W.}\end{aligned}$$

For other designs that may require analysis, it is recommended that the principles shown in these examples be applied.

Bibliography

[1] AISC M011, *Manual of steel construction — Specification for the design, fabrication and erection of structural steel for buildings.*

[2] AISC S302, *Code of standard practice for steel buildings and bridges*, American Institute of Steel Construction, 400 N. Michigan Avenue, Chicago, IL 60611, USA.

[3] ASME II, *ASME Boiler and Pressure Vessel Code*, Section II, Materials.

[4] ASME VIII, *ASME Boiler and Pressure Vessel Code*, Section VIII, Rules for construction of pressure vessels, Division 1.

[5] ASME B1.20.1, *Pipe threads, general purpose (inch).*

[6] ASME B16.11, *Forged fittings, socket-welding and threaded.*

[7] ASME B36.10M, *Welded and seamless wrought steel pipe.*

[8] ASTM D 97, *Test methods for pour point of petroleum oils.*

[9] REID, E.B. and ALLEN, H.I., Estimating Pour Points of Petroleum Distillate Blends, *Petroleum Refiner*, 1951, **30**, No. 5, pp. 93-95.

[10] Index Predicts Cloud, Pour, and Flash Points of Distillate Fuel Blends, *Oil and Gas Journal*, November 9, 1970, **68**, No. 45, pp. 66-69.

[11] NELSON, W.L., Is There a Practical Pour-Point Correlation?, *Oil and Gas Journal*, May 14, 1973, **71**, No. 20, p. 108.

[12] Phase Equilibria in Systems Containing Water, Chapter 9, *Technical Data Book–Petroleum Refining*, 4th edn., American Petroleum Institute, Washington, D.C., 1983, pp. 9-23-9-43.

[13] MCADAMS, W.H., *Heat Transmission*, 3rd edn., McGraw-Hill, New York, 1954, pp. 268-271.

[14] KERN, D.Q., *Process Heat Transfer*, McGraw-Hill, New York, 1950, pp. 538-541.

[15] GARDNER, K.A., Efficiency of Extended Surface, *Trans. ASME, Series C: J. Heat Transfer*, 1945,. **67**, No. 8, pp. 621-628.

[16] GRIMISON, E.D., Correlation and Utilization of New Data on Flow Resistance and Heat Transfer for Cross Flow of Gases over Tube Banks, *Trans. ASME, Series C: J. Heat Transfer*, 1937, **59**, No. 7, pp. 538-594.

Additional copies are available through Global Engineering Documents at (800) 854-7179 or (303) 397-7956

Information about API Publications, Programs and Services is available on the World Wide Web at: http://www.api.org

American
Petroleum
Institute

1220 L Street, Northwest
Washington, D.C. 20005-4070
202-682-8000

Product No. C66105