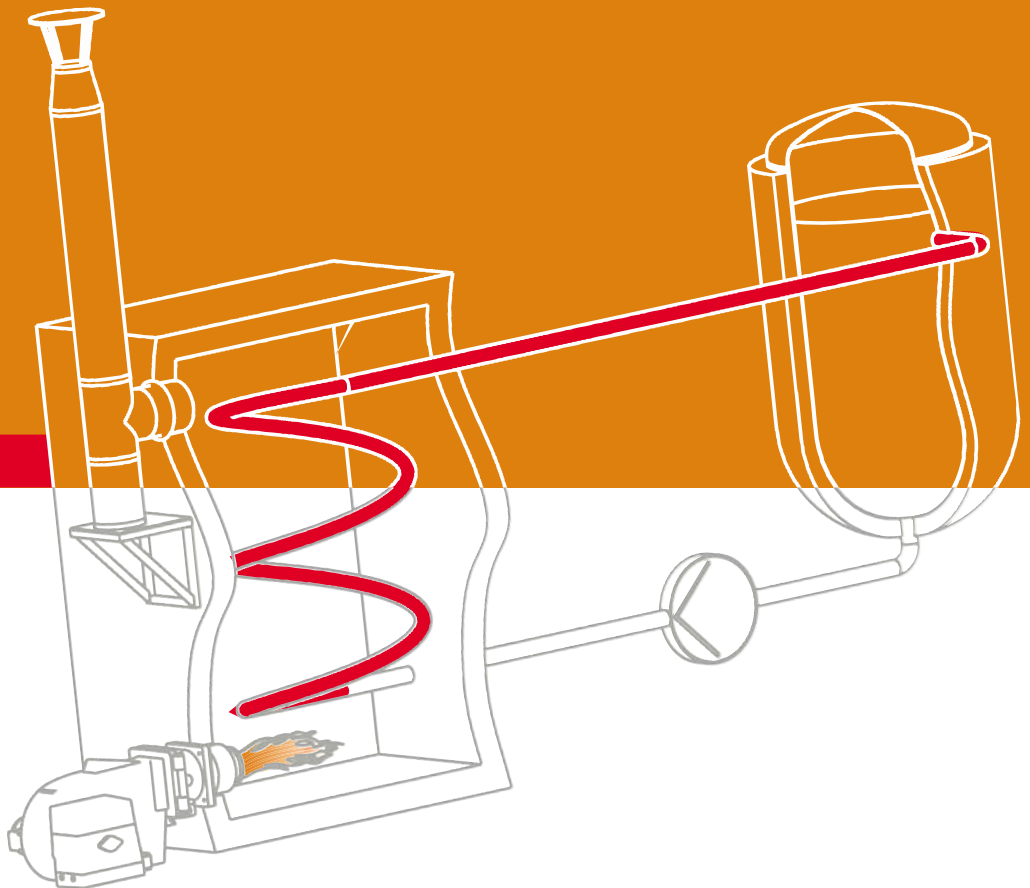


Walter Wagner

HEAT TRANSFER TECHNIQUE

WITH ORGANIC FLUIDS



A book by

PRO·CESS
Chemie · Pharma · Verfahrenstechnik

Dipl.-Ing. Walter Wagner

Heat Transfer Technique with Organic Fluids

3rd Edition



Dipl.-Ing. WALTER WAGNER

Born in 1941; following an apprenticeship as a technical draughtsman, he completed his mechanical engineering studies and was a plant design engineer in the atomic reactor industry during the period of 1964 to 1968; as of 1968 he was the technical director in plant construction, boiler construction and heat transfer technology. During the period of 1974 to 1997 Walter Wagner worked as a lecturer at the Technical College of Heilbronn, from 1982 to 1984 in addition at the Technical College of Mannheim, and from 1987 to 1989 at the Mosbach Vocational Academy. In the period of 1988 to 1995 he was the managing director of Hoch-Temperatur-Technik Vertriebsbüro Süd GmbH. Since 1992 he has been head of consulting and seminars for plant engineering: WTS Wagner-Technik-Service. In addition, Walter Wagner was also chairman of various DIN standards committees and an authorized specialist in heat transfer fluid technology, thermal plant construction and piping engineering.

WALTER WAGNER is the author of the following specialist books (currently only available in German language):


- Festigkeitsberechnungen im Apparate- und Rohrleitungsbau
- Kreiselpumpen und Kreiselpumpenanlagen
- Lufttechnische Anlagen
- Planung im Anlagenbau
- Regel- und Sicherheitsarmaturen
- Rohrleitungstechnik
- Strömung und Druckverlust
- Wärmeaustauscher
- Wärmeträgertechnik
- Wärmeübertragung
- Wasser und Wasserdampf im Anlagenbau
- DIETZEL/WAGNER: Technische Wärmelehre
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Preface

Heat transfer fluids in a broader sense as well as water and water steam include: refrigerants, organic fluids, salt melts, liquid metals and hot gases. This technical operation specialist book describes plants that primarily use organic heat transfer fluids. To date, it is the only standard text that comprehensively describes this technology.

The first heat transfer plants had already been used successfully in industry when the 1st edition of "Heat Transfer Technique" appeared in the summer of 1973. The technology has developed consistently and steadily since then, new regulations and rules have been created, and new areas of application have opened up. In particular, the ability to transfer and finely control temperatures of up to 400 °C – virtually without pressurization – is exactly one of the advantages of heat transfer technology using organic fluids.

In addition to the fundamental principles that continue to apply, it became necessary to adapt the edition to the new boundary conditions resulting from European legislation. When referring to regulations and standards, it is important to make sure that the newest version is used in each case. Biomass CHP plants have been added in Chapter 12. For Chapters 13 and 14, in addition to the project design equations, important diagrams, tables and figures have been incorporated for project planning. In Chapter 15 the material data tables for the fluids were revised and expanded with additional diagrams and equations.

The book can also provide valuable assistance as a reference guide in the planning of plant projects where precision must meet increasingly rigorous requirements, as an aid to design and calculations, and as an aid for questions and problems from everyday practice in process engineering.

By delivering "Heat Transfer Technique" to the Vogel Communications Group all of my books are now consolidated with one publisher. The specialist books that I have published, with special process engineering topics (see imprint), contain many examples, e.g. regarding combustion chamber calculations, pumps, the design of control valves, safety valves and heat exchangers etc., as well as pressure loss calculations that are important for heat transfer technology. I am always happy to receive feedback regarding the book and the solution methods provided, since a lively exchange of knowledge always motivates and inspires research and learning. The fastest contact will always be by email: wagner@wts-online.de.

I would like to thank Dr. Dietmar Hunold, heat 11, for numerous suggestions and contributions. I would also like to thank Vogel Communications Group for the excellent cooperation as always.

St. Leon-Rot

Walter Wagner

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Description of the Most Important Symbols

The following most important symbols are used, in principle, wherever possible, deviations from these symbols are always indicated at the relevant equations and illustrations.

| Symbol | Description | Unit | Remarks |
|--------|-----------------------------|------------------------------------|---|
| A | area, area of cross-section | m ² | |
| B | B | kg | |
| | \dot{B} | kg/s | |
| C | C | W/(m ² K ⁴) | |
| | C _s | W/(m ² K ⁴) | C _s = 5.67 · 10 ⁻⁸ W/(m ² K ⁴) |
| D | diameter | m | |
| E | E | Pa | preferred: N/mm ² |
| | E | J | |
| F | force | N | |
| G | force of weight | N | G = m · g |
| H | H | m | |
| | H _o | J/kg | |
| | H _u | J/kg | |
| I | I | m ⁴ | |
| | I | Ns | |
| I | current | A | |
| L | length | m | |
| M | moment | N m | |
| O | surface | m ² | |
| P | output | W | 1 W = 1 J/s |
| Q | Q | J | 1 J = 1 Ws |
| | \dot{Q} | W | |
| R | specific gas constant | J/(kg K) | |
| S | safety factor | – | |
| T | thermodynamic temperature | K | |
| U | U | m | |
| | U | V | |
| V | V | m ³ | |
| | \dot{V} | m ³ /s | |
| W | W | J | |
| | W | m ³ | |

| Symbol | | Description | Unit | Remarks |
|-------------|------------|--|---------------------------------|----------------------------------|
| a | a | acceleration | m/s^2 | |
| | a | temperature conductivity | m^2/s | |
| c | c | specific thermal capacity | $\text{J}/(\text{kg K})$ | |
| | c | spring coefficient | N/m | |
| d | | diameter | m | |
| f | f | factor | – | |
| | f | frequency | Hz | $1 \text{ Hz} = 1/\text{s}$ |
| g | | acceleration due to gravity | m/s^2 | $g_n = 9.80665 \text{ m/s}^2$ |
| h | h | enthalpy | J/kg | |
| | h | height | m | |
| i | | radius of moment of inertia | m | |
| k | k | overall heat transfer coefficient | $\text{W}/(\text{m}^2\text{K})$ | |
| | k | height of tube roughness | m | |
| m | m | mass | kg | |
| | \dot{m} | mass flow | kg/s | |
| | m | opening ratio | – | |
| n | n | number of revolutions | $1/\text{s}$ | |
| | n | air ratio | – | |
| P | | pressure | Pa | $1 \text{ Pa} = 1 \text{ N/m}^2$ |
| \dot{q} | | heat flux density | W/m^2 | |
| r | r | specific evaporation enthalpy | J/kg | |
| | r | radius | m | |
| s | | layer thickness, wall thickness | m | |
| t | | time | s | |
| v | | specific volume | m^3/kg | $v = 1/\rho$ |
| w | | velocity | m/s | |
| z | \dot{z} | decomposition rate | kg/s | |
| α | α | coefficient of longitudinal elongation | $1/\text{K}$ | |
| | α | flow rate | – | |
| | α | heat transfer coefficient | $\text{W}/(\text{m}^2\text{K})$ | |
| β | β | volumetric expansion coefficient | $1/\text{K}$ | |
| | β | angle | – | |
| δ | | thickness of boundary layer | m | |
| ϵ | ϵ | emission ratio | – | |
| | ϵ | elongation | – | |
| ζ | | resistance coefficient | – | |
| η | η | dynamic viscosity | Pa s | |
| | η | efficiency | – | |
| ϑ | | temperature °Celsius | $^{\circ}\text{C}$ | |

| Symbol | | Description | Unit | Remarks |
|-----------|-----------|----------------------------|-------------------|---------|
| λ | λ | thermal conductivity | W/(m K) | |
| | λ | ratio of slenderness | – | |
| | λ | tube friction coefficient | – | |
| μ | | friction coefficient | – | |
| ν | | kinematic viscosity | m ² /s | |
| ρ | | density | kg/m ³ | |
| σ | σ | tension, stress | N/m ² | |
| | σ | surface tension | N/m | |
| τ | | sheer stress, shear strain | N/m ² | |

Characteristic Quantities

| | | | |
|----|-----------------|---|--|
| Bi | Biot number | = | $\frac{\alpha \cdot s}{\lambda_w}$ |
| Gr | Grashof number | = | $\frac{d^3 \cdot g \cdot \beta \cdot \Delta\theta}{\nu^2}$ |
| Nu | Nusselt number | = | $\frac{\alpha \cdot d}{\lambda}$ |
| Pe | Péclet number | = | $\frac{w \cdot d}{a} = \frac{w \cdot d \cdot \rho \cdot c}{\lambda} = \text{Re} \cdot \text{Pr}$ |
| Pr | Prandtl number | = | $\frac{c \cdot \rho \cdot \nu}{\lambda} = \frac{\nu}{a}$ |
| Re | Reynolds number | = | $\frac{w \cdot d}{\nu}$ |

Data are required for the unambiguous description of a characteristic quantity, how the characteristic quantity used is defined and the temperature on which the properties are based.

Signs

| | |
|----------|--------------|
| Δ | difference |
| d | differential |
| Σ | sum |

Superior Indicator

| | |
|---|---------------------------|
| – | mean, average |
| · | quantity referred to time |
| ^ | maximum |
| ∨ | minimum |
| ~ | alternating |
| ' | liquid phase |
| " | vapor phase |

Inferior Indicator

| | |
|------------|----------------------------|
| F | flame |
| Fl | liquid, fluid |
| G | gas |
| Gr | boundary layer |
| K | convection |
| L | air, longitudinal, laminar |
| N | rated value |
| Q | cross-section |
| R | return |
| S | radiation |
| T | turbulent |
| V | flow, loss |
| W | wall |
| a | external |
| äq | equivalent |
| dyn | dynamic |
| ges | total |
| h | hydraulic |
| i | internal |
| log | logarithmic |
| n | normal condition |
| proj | projected |
| th | theoretical, thermal |
| ü | overpressure |
| x | direction |
| y | direction |
| zul / perm | permissible |
| ϑ | relating to temperature |
| ⊥ | vertical |
| | parallel |
| 0 | initial value |
| 1 | start, inlet |
| 2 | end, outlet |
| ∞ | infinite |

1 Introduction

Many technical processes require heating of the product to temperatures above the ambient temperature. Fundamentally, a distinction is made between two types of heating possibilities.

Direct Heating: The product is mainly and directly heated from outside by combustion gases or electric heating elements (Fig. 1.1).

Indirect Heating: A recirculating heat transfer fluid (the so-called “heat transfer fluid”) is used between heater and heat consumer (Fig. 1.2). This type of heating presents the basic principle of a heat transfer plant.

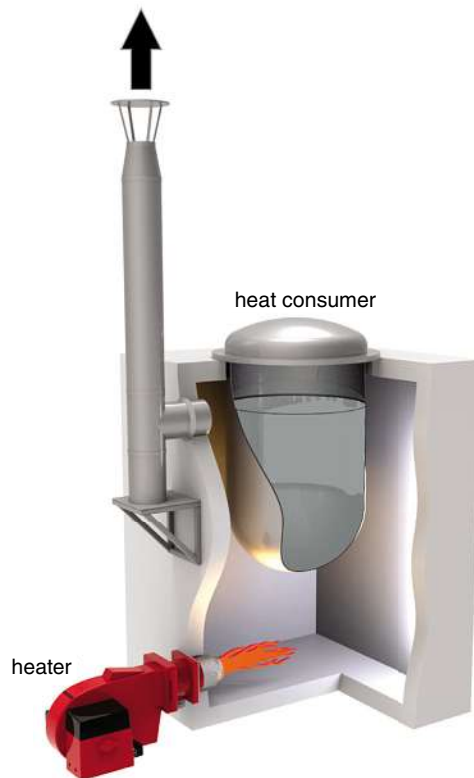


Fig. 1.1 Direct heating [heat 11]

A heat transfer plant is a plant in which the heat transfer fluid flows (from the heater to the heat consumer and back) between boundary walls at which heat is neither added nor removed (with the exception of natural losses). This intermediate boundary system for the transport of the heat transfer fluid defines the basis of the heat transfer plants described in this book.

Plants containing a heat transfer fluid but in which one of the boundary walls of the fluid is simultaneously a heat transfer surface to the heat consumer are thus not to be termed heat transfer plants.

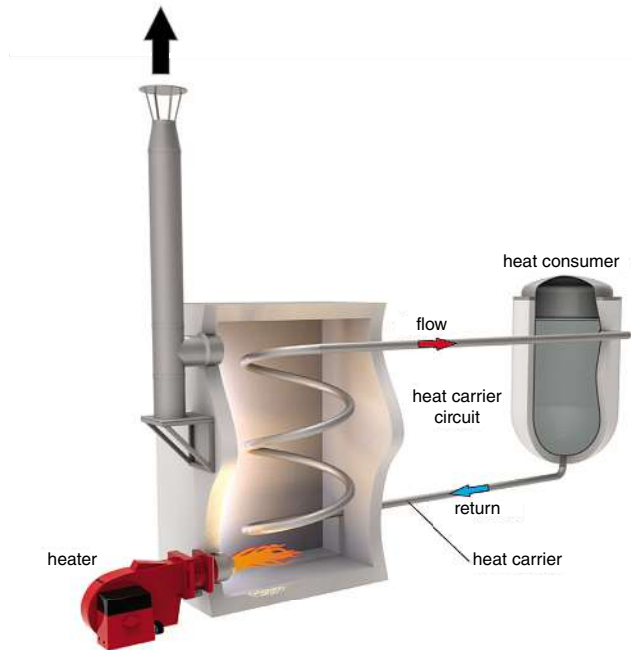


Fig. 1.2 Indirect heating [heat 11]

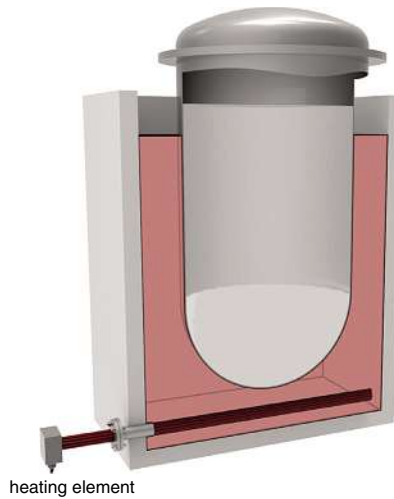


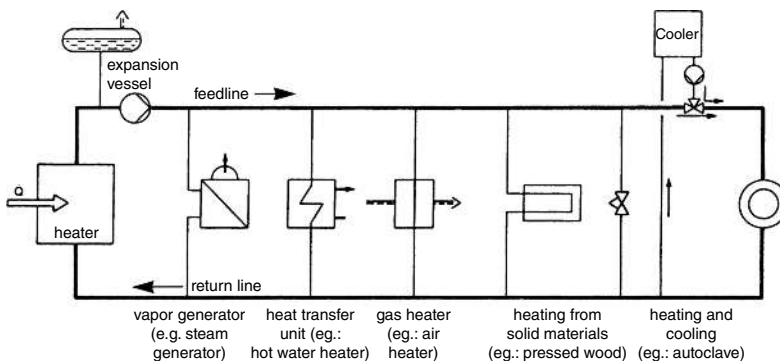
Fig. 1.3 Heating with contact fluid [heat 11]

An example of such a system is the heating shown in (Fig. 1.3). The contact liquid in the double jacket is electrically heated, flows upward by natural buoyancy and is cooled at the boundary wall. This does not meet the definition of a heat transfer plant and, in this case, it is not a heat transfer fluid but a contact fluid and not a heat transfer plant but a heat exchanger.

The Advantages of a Heat Transfer Plant in General Compared with the Directly Heated Plant

1. Local overheating of the product to be heated is avoided.
2. The temperature of the product can be controlled very accurately and uniformly. The flow temperature can also be adapted to the requirements.
3. A heater in the immediate neighbourhood of the consumer is not required (production safety rules with regard to ignition and explosion risk).
4. The heater can be arranged centrally and the heat transferred through the heat transfer fluid circuit to one or several consumer points.
5. The efficiency of the centrally heated plant is higher than that of individual directly heated consumers.
6. Feeding of fuel to each firing point and installation of flue gas chimneys are saved compared with individually fired plants.
7. Direct firing of individual consumers necessitates large maintenance expenditure.
8. The heat transfer conditions can be optimized at the consumer.
9. Heating and cooling processes can be carried out with the same heat transfer fluid.
10. Storage of heat is possible. This is advantageous where heat requirements fluctuate strongly with high short-term peak loads.
11. A central or local transformation to warm or hot water in heat exchangers, to steam in steam generators or hot air in air heaters is possible.
12. A conversion of the fuel system at the directly fired consumer to another fuel is very expensive.
13. The thickness of insulation at the consumer can be kept small and local excess temperatures at heat conducting bridges can be avoided.

The heating of heat consumers by a recirculating heat transfer fluid has found wide acceptance because of these advantages of the indirectly heated plant (Fig. 1.4).



| | | | |
|-----------------------------|------------------------------|-------------------------------|-------------------------------|
| Heat generation: | \dot{Q} | Electrical | Electrical resistance heating |
| Direct, by means of: | | Solid materials: | Wood Coal, etc. |
| heating oil: | EL M S | Indirect, by means of: | Liquids Gases Vapors |
| Gases: | Natural gas Synthetic gas | | |

Fig. 1.4 Basic design of a heat transfer plant

Historical Overview

- 1929 ■ Patent for the synthetic heat transfer fluid, the eutectic mixture – diphenyl/diphenyl oxide, with 26.5 % diphenyl by volume and 76.5 % diphenyl oxide by volume as an azeotropic liquid that can also be used in the vapor phase.
Registered by Dow Chemical.
IG-Farben (now Bayer) also registered a patent in 1931 retroactive to 1929.
- Application in the chemical industry.
- 1938 ■ Bertrams constructs the 1st installation for synthetic resin manufacture with $\vartheta_v = 320$ °C.
as of approx.
- 1960 ■ Mineral oil-based heat transfer fluids largely employed as an unpressurized heat transfer fluid up to 300 °C.
- 1st widespread application in laundries for ironing machines.
 - Application of synthetic heat transfer fluids as coolants in nuclear reactors.
- 1962 ■ Application in bakeries for baking ovens.
- 1964 ■ Application in the textile industry for clamping frames.
as of approx.
- 1965 ■ General application in a wide range of industrial sectors.
- Guideline no. 14 of BG-Chemie (chemical industry trade association).
 - Provisional guideline from the Berlin trade office regarding heating systems with hot oil.
- 1969 ■ 1st VDI conference on the subject of heat transfer fluids technology in Wiesbaden.
- 1970 ■ VDI guideline 3033.
- 1974 ■ DIN 4754.
- 1976 ■ VDI conferences in Ulm and Munich.
- 1980 ■ New edition of DIN 4754.
- 1990 ■ New edition of DIN 4754.
- 1993 ■ Accident prevention regulation (VBG 64) "Heat transfer fluid installations with organic heat transfer fluids."
- 1994 ■ New edition of VDI guideline 3033. New edition of DIN 4754.
- 1998 ■ Heat transfer fluids Q, DIN 51 522.
- Thermal stability, DIN 51 528.
- 2000 ■ Testing of used heat transfer fluids, DIN 51 529.
- 2002 ■ Pressure Equipment Directive 97/23/EG.
- Betriebssicherheitsverordnung (Industrial Safety Regulation).
 - Accident prevention regulation (BGV D3) "Heat transfer systems with organic heat transfer fluids."
 - Implementation instructions for BGV D3 (BGV D3 DA).
- 2015 ■ New edition of DIN 4754-1, DIN 4754-2 and DIN 4754-3.

2 Heat Transfer Fluids¹⁾

2.1 General

To utilize the advantages of indirect heating by heat transfer fluids for many applications, heat transfer fluids are used which are liquid or vaporous under operational conditions.

During operation with vaporous heat transfer fluids overpressure is generated and separating systems are required for evaporation and condensation so that this system is applied only there where a liquid recirculation plant is uneconomic.

The Main Requirements for Heat Transfer Fluids Are

1. high start of boiling at atmospheric pressure
2. low solidification temperature
3. good thermal stability
4. low viscosity over the whole temperature range (also in the startup condition)
5. good heat transfer properties
6. high specific evaporation enthalpy (when used as vapor)
7. low corrosion tendency for the materials of the apparatus
8. non-toxic and non-smelly
9. low sensitivity to impurities (e.g. oxygen)
10. low fire risk
11. low risk to the environment on leakage
12. possibility of easy destruction (passing used material into the natural circulation circuit)
13. economic purchase price

There is no single fluid meeting all requirements to a high degree. The heat transfer fluid must therefore be selected for the given operating conditions making possible the best compromise and the most economic solution. Data of the currently most frequently used heat transfer fluids in the various temperature ranges are given in Fig. 2.1.

- Water, warm water and hot water are primarily used in the range of +10 to 150 °C.
- Steam is used in the range of 150 to 200 °C, but at the upper temperature limit a system overpressure of merely 16 bar is necessary.
- Organic heat transfer fluids are used for heating processes in the range of 200 to 400 °C, where fluids can be run without overpressure up to 350 °C. System pressures of approx. 10 bar only are required for fluids at the highest application temperature.
- For even higher temperatures, aside from electrical heating, liquid salts (up to 550 °C), liquid metals (up to 750 °C) and, above this, gas heating, flame heating and radiant heating are used as heat transfer fluids.

1) In heat transfer technology different names are used for heat transfer fluids, e.g. Heat Transfer Media, Thermool, Thermaloil, Heat Transfer Oil, Heat Carrier Media.

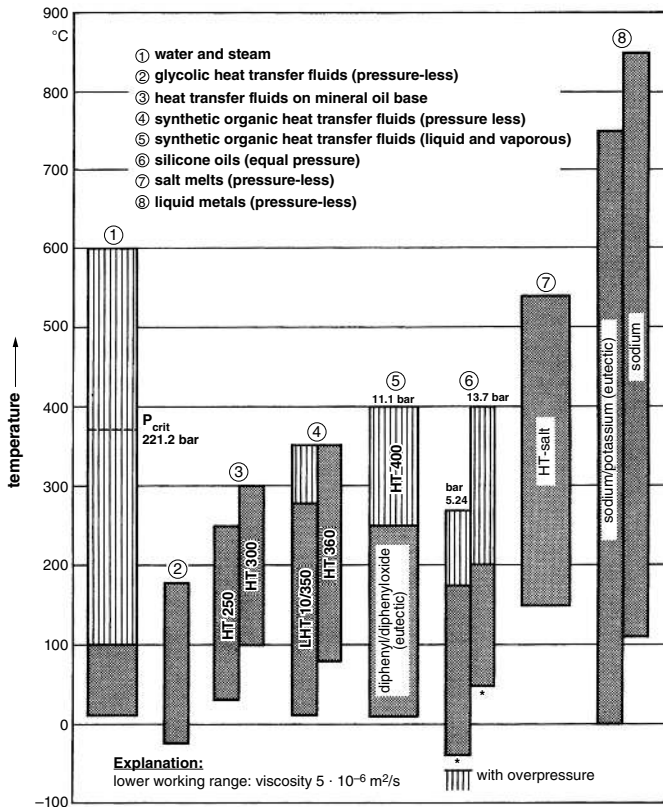


Fig. 2.1 Temperature ranges of liquid and vaporous heat transfer fluids

* On account of the limited increase in viscosity at low temperatures, in the case of silicone oils the working range can be lowered further.

2.1.1 Water and Steam

Water is the most widely used and best known heat transfer fluid. For the temperature range of 0 to 100 °C water is the ideal heat transfer fluid because it has optimum properties in this range (compared with other heat transfer fluids), i.e.

- evaporation temperature at atmospheric pressure = 100 °C
- freezing temperature = 0 °C
- high specific heat capacity $c \cdot \rho = 4200 \text{ kJ}/(\text{m}^3\text{K})$
- very high thermal conductivity $\lambda = 0.6 \text{ W}/(\text{mK})$
- low viscosity $\nu = 1 \cdot 10^{-6} \text{ m}^2/\text{s}$ at 20 °C
- economic application conditions, e.g. cheap, non-toxic, etc.

At temperatures above 100 °C the heat transfer fluid circuit must be pressurized (Fig. 2.2). This requires, already at 200 °C, a pressure-resistant plant designed for about 16 bar at least.

Expensive pressurized plants are therefore required necessitating extensive safety measures. Because of the high evaporation enthalpy of water of 2257.3 kJ/kg at 100 °C and the large heat transportability steam is generally preferred as heat transfer fluid instead of water from about 150 °C despite the condensate network required.

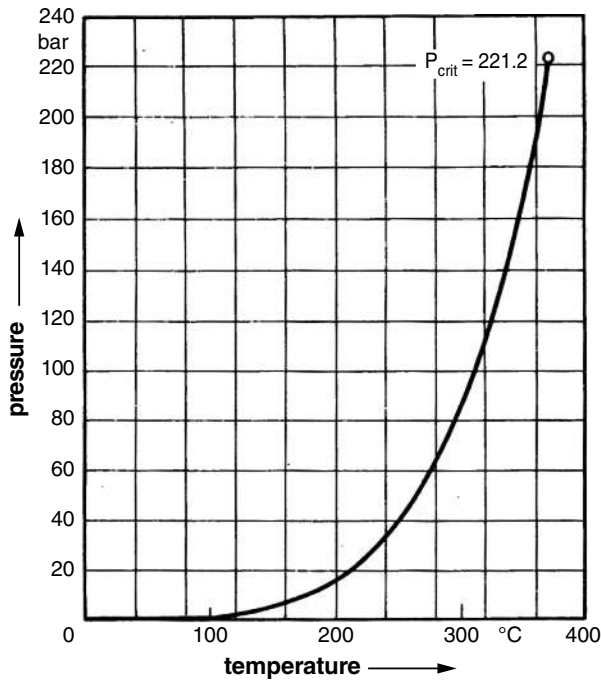


Fig. 2.2 Vapor pressure curve of water

Problems of corrosion, deposits and scaling occur in water and steam plants requiring water preparation plants because completely pure water does not exist in nature. Raw water and water supplied by waterworks always contains a more or less large number of admixtures and impurities. Water dissociates these impurities such as salts, acids and bases into ions.

The gases dissolved in the water, such as oxygen (O_2) and carbon dioxide (CO_2) are mainly responsible for the corrosion. The "pH", a measure of the hydrogen ion concentration in aqueous solutions, serves the assessment of the corrosion supporting behavior of water. The cause of corrosion is the same in all cases despite the many-sided corrosion phenomena: the corroding metal strives to change back into its chemical compound.

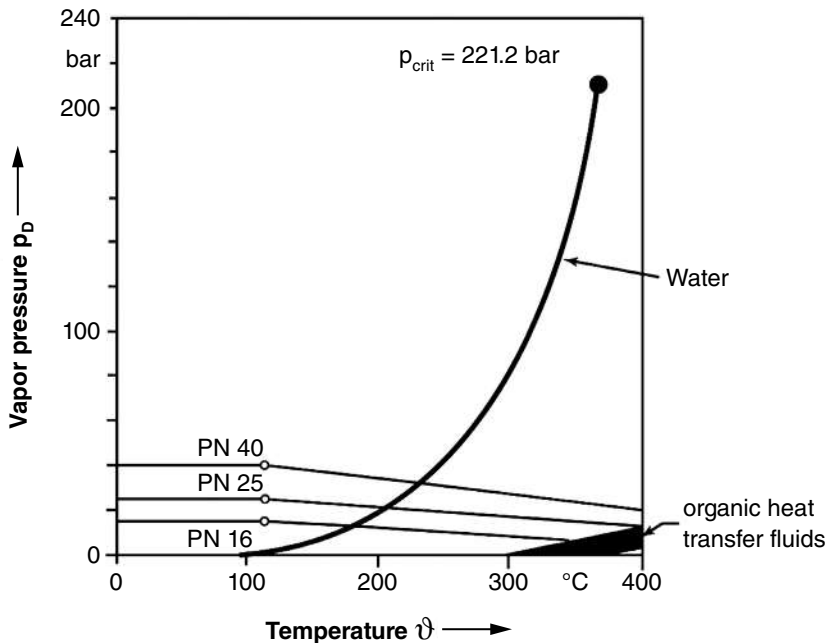
The scaling of the materials of the apparatus is due to the fact that boiler scale is formed on the walls on heating the water, particularly above 100 °C, which reduces the heat transfer and narrows the cross-section. Its cause is the salt content (dissolved salts of calcium and magnesium) of the water. A measure of the salt content is the "hardness". Further properties cf. [2.1].

Steam

It differs from gases only by being easily liquified. The equation of state for the ideal gas $p \cdot v = R \cdot T$ is valid only for steam at low pressure and high temperature. Further properties cf. e.g. [2.2].

Table 2.1 Different heat transfer fluids with the important material data for heat transfer

| Fluid | Spec. heat capacity c_p kJ/(kg · K) | Spec. evaporation enthalpy Δh_v kJ/kg | Thermal conductivity λ W/(m · K) |
|---|---|---|--|
| Water (H ₂ O) $\vartheta_{App.} \approx 5$ to 700 °C | 4.2 | 2257.3 at 100 °C | 0.65 |
| Organic liquids $\vartheta_{App.} \approx 50$ to 350 °C | 2.1 | - | 0.11 |
| Diphenyl/diphenyl oxide $\vartheta_{App.} \approx 20$ to 400 °C | 2.1 | approx. 300 at 256 °C | 0.10 |
| Salt melts $\vartheta_{App.} \approx 150$ to 500 °C | 1.55 | - | 0.3 |
| Liquid metals (Na) $\vartheta_{App.} \approx 100$ to 700 °C | 1.3 | - | 60 |
| Air and carbon dioxide $\vartheta_{App.} \approx -50$ to 1200 °C | 1.0 | - | ≈ 0.04 |
| Helium (He) $\vartheta_{App.} \approx -50$ to 1200 °C | 5.0 | - | ≈ 0.3 |

**Fig. 2.3** Vapor pressure curves for water and organic heat transfer fluids (with nominal pressure curves PN)

2.1.2 Salt Melts

Salt melts are increasingly used as heat transfer fluids in the temperature range of 400 to 550 °C. Most frequently used today is a ternary eutectic mixture of:

- 53 % potassium nitrate (KNO_3)
- 40 % sodium nitrite (NaNO_2)
- 7 % sodium nitrate (NaNO_3)

A binary mixture of substances such as, for example, sodium nitrite (45 %) and potassium nitrate (55 %) is sometimes used.

Important Properties and Data

- not combustible
- no risk of explosion
- escaping vapors are non-toxic

Melting Point:

$$\vartheta_{\text{Sch}} = 142 \text{ °C}$$

The freezing point of the indicated eutectic three-substance mixture is 142 °C and it changes depending on composition, as shown in Fig. 2.4.

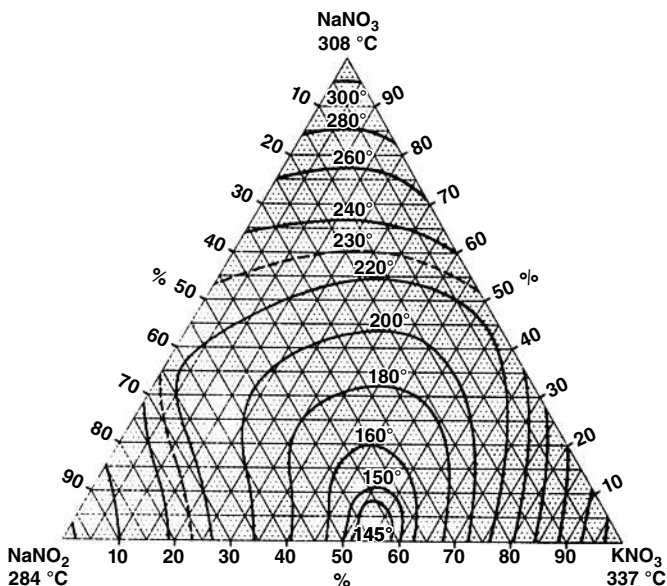


Fig. 2.4 Relationship between the melting temperature and the composition of the salt melt

Density:

$\rho = 2000 \text{ kg/m}^3$ at 150 °C and falls linearly to

$\rho = 1650 \text{ kg/m}^3$ at 600 °C.

Bulk density $\rho \approx 1200 \text{ kg/m}^3$

3 Design of Heat Transfer Plants

Design Principles

It is necessary to consider the specific properties of the heat transfer fluid to ensure safe operation and fault-free function of a heat transfer plant. All known heat transfer fluids whether water, steam, salt melts, liquid metals and gases, have advantages and disadvantages and specific application limits and organic heat transfer fluids are not excluded from these.

The disadvantages are the possibility of combustion, oxidation and thermal decomposition due to their chemical structure based on hydrocarbons (C–H).

The characteristics to be considered in the design of heat transfer plants are thus combustibility, oxidation and thermal stability. These three criteria form also the basis for safe design, construction and equipment of these plants as determined in the various specifications, standards and regulations.

- Combustibility determines the assessment of freedom from leakage of the plant components and the installation of heaters and vessels.
- Oxidation is the basis for the design of the system of separating the plant from the atmosphere.
- Thermal stability influences the permissible flow temperature and the design of the heat transfer fluid heater.

The optimum construction is thus a welded construction of the plant, wherever possible, with an expansion vessel covered by inert gas (purest nitrogen) and ensured flow rate. With such a planned plant the organic heat transfer fluid can be operated safely and without problems for many years, even at the thermal stability limit.

3.1 Symbols, Flow Sheets and Abbreviations

Symbols

Symbols are used to enable the presentation of plant schemes (flow sheets, flow schemes, heating circuits) by simple signs. They are presentations of individual parts in the plant schemes. The construction is taken into account only in so far as required for obtaining easily understood signs and illustrations. They can be inserted into the plans in any position corresponding to the course of the lines.

Design and kind of presentation of the individual plans must be freely adapted to the purpose. That kind of presentation is to be preferred which is most simple, clear and impressive. This target is approached most nearly, if the symbols are arranged within the plans in horizontal or vertical directions, if the curve of the main line contains few turns and if crossings are avoided as far as possible.

The symbols are taken from EN ISO 10 628.

Flow Sheets

The flow sheet is a drawing, simplified with the aid of symbols, of the design and function of the plant. Different kinds of flow sheets are used depending on information content and presentation; for heat transfer plants most widely used is: pipeline and instrument flow sheet (P + I flow sheet) to DIN 28 004 – flow sheets of processing plants, supplemented by DIN 19 227 (Fig. 3.1).

| | | | | | | | | |
|--|--|---|---|---|---|--|--|---|
| <p>heat transfer fluid flow system</p> | <p>TIC — FIZA-- — TI —</p> | <p>temperature control with gauge flow protection with gauge and borderline alarm for minimum permissible value temperature gauge</p> | <p>TI — LZA- — PI —</p> | <p>temperature gauge liquid level limiter with borderline value alarm at lower limit pressure gauge</p> | <p>TZA + — GOS - — LI — LI —</p> | <p>temperature limiting safety device with alarm at maximum allowable value position switch with indication of minimum permissible value level gauge level control-manual</p> | <p>TOS - — GSA - — GI — HOS —</p> | <p>temperature switch with indication of minimum permissible value position switch with borderline value alarm at lower limit position indicator manual switch, for example, for turning on pump</p> |
| <p>flue gas side</p> | <p>TZA + — GOS - — QS+ —</p> | <p>flue gas temperature limiter with borderline alarm position switch with indication of minimum permissible value position flue gas monitor for maximum permissible value</p> | <p>PSA + — TI — ... —</p> | <p>Pressure switch with alarm at maximum allowable value temperature gauge measuring point</p> | <p>PIC — TOS - —</p> | <p>pressure control with gauge temperature switch with indication of minimum permissible value</p> | <p>PZA + — TI —</p> | <p>safety pressure switch with alarm at maximum permissible value temperature gauge</p> |
| <p>fuel supply for solid fuel combustion</p> | <p>TIZA + — XZA - —</p> | <p>safety temperature limiter with alarm at maximum and minimum permissible values flame sensor with borderline alarm at lower limit</p> | <p>GSA - —</p> | <p>limit switch with alarm at minimum permissible value</p> | <p>LOS - —</p> | <p>level switch with indication of minimum permissible value</p> | <p>LOS + —</p> | <p>level switch with indication of maximum permissible value</p> |
| <p>combustion air</p> | <p>PZA + —</p> | <p>safety pressure switch with alarm at minimum permissible value</p> | | | | | | |
| <p>auxiliary systems</p> | <p>FSA + — PZA +- —</p> | <p>flow monitor with borderline value alarm safety pressure switch with alarm at maximum and minimum permissible values</p> | <p>LSA - —</p> | <p>level switch with alarm at minimum permissible value</p> | <p>LI —</p> | <p>level gauge</p> | <p>LC —</p> | <p>level control</p> |

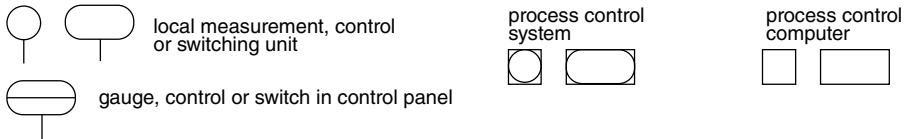


Fig. 3.1 Typical symbols for measurement, control and regulation in heat transfer technique.

The P + I flow sheet presents the technical equipment of a plant. The main flow direction should run from the left to the right hand. Flow lines or pipelines should not intersect if possible. If two lines intersect, one can be broken, for lines of different width always the narrower one and for lines of the same width it is recommended to draw the horizontal line as full line.

Abbreviations

Abbreviations in flow sheets serve the unambiguous characterization and identification of components; they are composed of:

- For apparatus and machines of:
 - a) the identification letter (cf. Table 3.1) and
 - b) a second identification letter (if applicable),
 - c) the internal plant counting figure and
 - d) if applicable, a figure is affixed or a letter, the affixed figure being separated from the counting figure by a stop.

Additional components belonging to an apparatus or a machine, such as drives, gears, etc., are given the same identification letter as the apparatus or machine, the identification letter for the additional component following as second letter.

- For fittings of:
 - a) the nominal size. For fittings the nominal size of which cannot be recognized from the following pipeline or which is not identical with the nominal size of its pipeline or cannot be associated with a pipeline, the nominal diameter must be put without key in front of the identification letter of the fitting,
 - b) the identification letter (cf. Table 3.1),
 - c) the identification figure (alpha-numerical) of the fitting which determines in detail the description of the fitting such as: type, nominal pressure, material and design details,
 - d) the identification letter of the fitting connection having a connection different from flanges with smooth sealing ledge (cf. Table 3.1),
 - e) the counting figure if required to be put below the abbreviation.

- For pipelines of:
 - a) the line figure. A counting figure determined by the user according to the requirements,
 - b) the nominal size,
 - c) the pipe class which is a concept for a fixed summary of all pipeline parts belonging to a pipeline. Within a pipe class the pipeline components associated with a nominal pressure and pipe material are always unambiguously determined in a design.
The description of a pipe class is composed of:
the nominal pressure to DIN 2401, sheet 1,
the identification letter (cf. Table 3.1) and
the pipe class figure identifying the mutually associated designs of pipeline components.

Table 3.1 Identification letters for apparatus, fittings and pipe materials

| Letter | Apparatus Machine Components | Fittings | Fitting Connections | Pipe Material Group |
|--------|----------------------------------|--------------------------|--|---------------------------------------|
| A | apparatus, machine general | drain | | cast iron |
| B | vessel | | | unalloyed steels |
| C | chemical reactor | | | low heat resistant steels |
| D | steam generator, furnace, heater | | | high heat resistant steels |
| E | | | | pressurized hydrogen resistant steels |
| F | filtering apparatus separators | filter dirt collector | | cold tough steels |
| G | gears | flow gauge | | surface protected steels |
| H | Lifting and transport equipment | tap | | stainless steels |
| I | | | | |
| J | | | flange with groove for ring joint to ASA | |
| K | column | flap | | non-ferrous metals |
| L | | | flange with groove for lense seal to DIN 2696 | plastics |
| M | electric motor | | flange with bevel for membrane weld seal to DIN 2695 | concrete |
| N | | | flange with groove to DIN 2512 | reinforced concrete |
| O | | | flange with groove for ring seal | |
| P | pump | | | prestressed concrete |
| Q | | | | asbestos cement |
| R | stirrer, mixer | return valve | set-off flange to DIN 2513 | earthenware |
| S | centrifuge, separator | slide valve | weld end | |
| T | drier, crystallizer | | | |
| U | | | | |
| V | compressor, blower | valve | | |
| w | heat exchanger | | | |
| X | other components | | | |
| Y | other drives | safety valve, burst disk | threaded socket | |
| Z | commutating machine | other fittings | threaded trunion | other materials |

Table 3.2 Identification letters for measuring, regulating and control technique (derived from ISA 5.1)

| Letter | Measured Quantity or Releasing Variable | Determining Variable | Indication of Measured Quantity | Initial Function | Determining Variable |
|--------|---|----------------------|---------------------------------|------------------|----------------------|
| A | analysis | | alarm | | |
| B | burner flame | | | | |
| C | elect. conductivity | | | control | |
| D | density | difference | | | |
| E | voltage | | primary element | | |
| F | flow rate | ratio | | | |
| G | measurement (thickness) | glass | | | |
| H | manual actuation | | | | high |
| I | elect. current | | indicating | | |
| J | power, energy | comparison | | | |
| K | time or time schedule | | | control station | |
| L | level | | light | | low |
| M | moisture | | | | centre |
| N | | | | | |
| O | | | orifice | | |
| P | pressure or vacuum | integrating | measuring point | | |
| Q | quantity or magnitude | | | | |
| R | radioactivity | | recording, printing | | |
| S | speed | safety | | switch | |
| T | temperature | | | transmitter | |
| U | multivariable | | multifunction | multifunction | multifunction |
| V | viscosity | | | valve | |
| w | weight, force | | well, sheath | | |
| X | | | | | |
| Y | | | | relay, computer | |
| Z | | | counter | drive | |

| Abbreviation | | | D1 | W1 | P1 | B1 | B2 |
|------------------------------------|-------------|-------------------|--------|---------------|------|------------------|-------------------|
| Name | | | Heater | Heat consumer | Pump | Expansion vessel | Collection vessel |
| Heat production | \dot{Q}_N | kW | 1000 | 800 | | | |
| Flow rate | \dot{v} | m ³ /h | | 100 | 100 | | |
| Pressure head | H | m | | | 45 | | |
| Output | P_M | kW | | | 18.5 | | |
| Revolutions | n | min ⁻¹ | | | 2900 | | |
| Permissible operating overpressure | | bar | 10 | 10 | | 1 | 1 |

4 Fundamentals of Flow and Heat

4.1 Flow Rate of Heat Transfer Fluid

The heat energy released in the heater is taken up by the heat transfer fluid and transferred to the consumer. In taking up heat the temperature level is increased. The material property for taking up heat is the specific heat capacity c , i.e. the amount of heat required to heat the mass 1 kg by 1 K.

c kJ/[kg K]

Due to the specific heat capacity usually not having a constant value a mean specific heat capacity must be used in the calculation. This depends primarily on the temperature. With a linear course of $c = f(\vartheta)$ or smaller temperature differences the arithmetic mean can be used. For the stated boundary conditions the true specific heat capacity can be used as mean specific heat capacity at the mean temperature.

$$c_{VR} = \frac{c_V + c_R}{2}; \text{ or } c_{VR} \text{ at } \frac{\vartheta_V + \vartheta_R}{2} \quad (4.1)$$

The amount of heat is then (Fig. 4.1):

$$\dot{Q}_H = \dot{m} \cdot c_{VR} \cdot \Delta\vartheta_{VR} \quad (4.2)$$

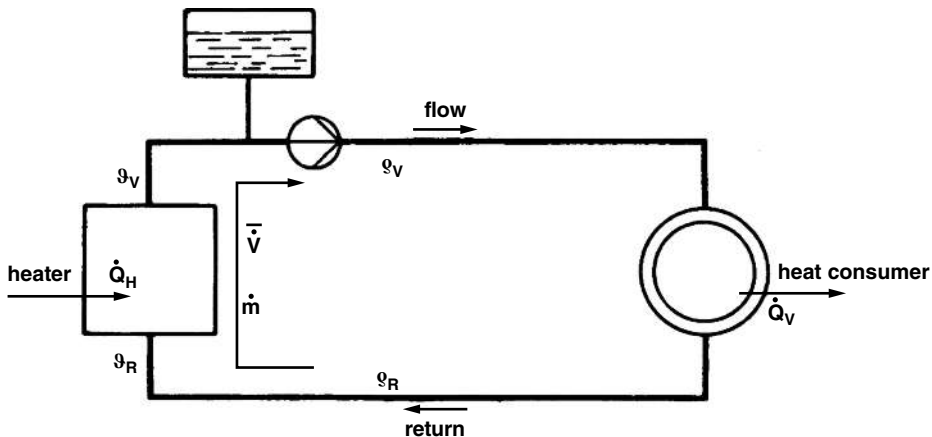


Fig. 4.1 Plant scheme for determining flow rate

At a given heat output and temperature difference the required mass flow is:

$$\dot{m} = \frac{\dot{Q}_H}{c_{VR} \cdot \Delta\theta_{VR}} \quad (4.3)$$

The mean volume flow to be circulated results:

$$\dot{m} = \bar{V} \cdot \rho_{VR} \quad (4.4)$$

$$\bar{V} = \frac{\dot{Q}_H}{\rho_{VR} \cdot c_{VR} \cdot \Delta\theta_{VR}} \quad (4.5)$$

The flow rate at the flow, return or at an intermediate temperature is

$$\dot{V}_\theta = \bar{V} \cdot \frac{\rho_{VR}}{\rho_\theta} \quad (4.6)$$

The mean flow velocity in a cross-section is:

$$\bar{w} = \frac{\dot{V}}{A_Q} \quad (4.7)$$

This equation is shown in Fig. 4.2 for pipes. The density is presented in Fig. 4.3 and the volumetric specific heat capacity $c \cdot \rho$ in Fig. 4.4 for determining the mean flow rate of the individual heat transfer fluids.

The exact numerical values of the individual fluids are given in Chapter 15. At the usual working temperatures between 200 and 320 °C a mean specific volumetric heat capacity of 1900 kJ/m³K can be used.

$$\bar{V} = \frac{\dot{Q}_H}{1900 \cdot \Delta\theta_{VR}} [\text{m}^3/\text{h}] \quad (4.8)$$

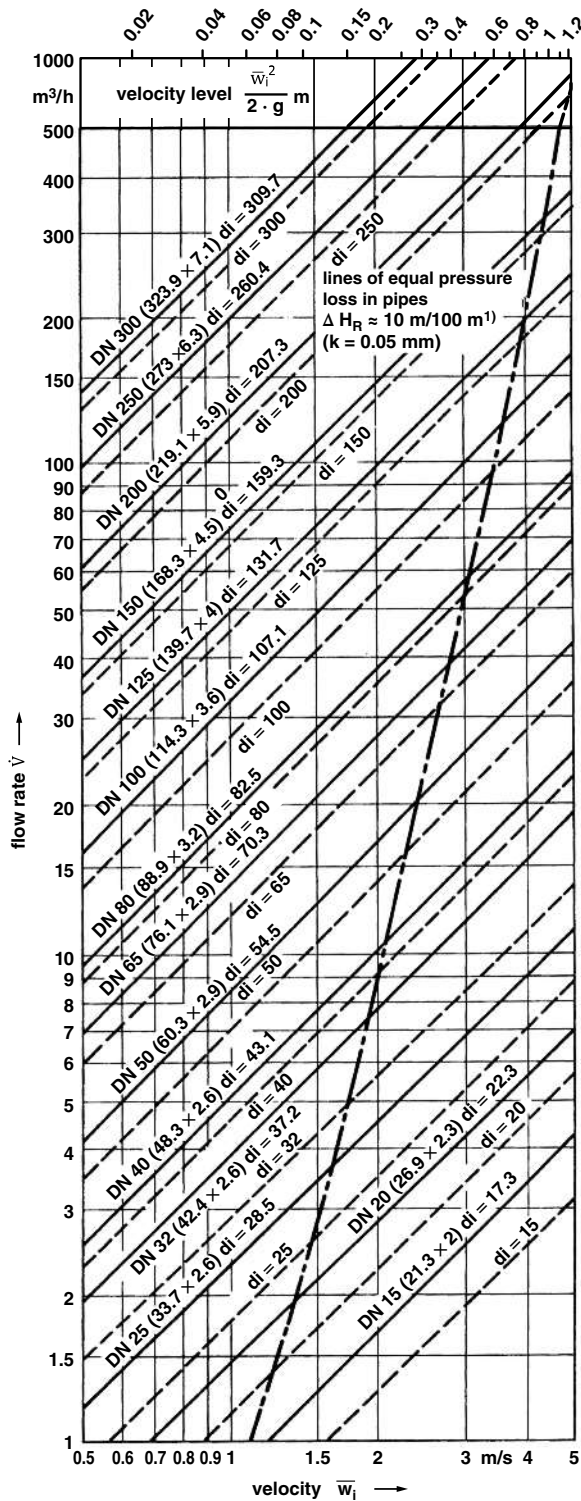


Fig. 4.2 Determination of average flow velocity \bar{w}_i in pipes
¹⁾ $H_R = 10 \text{ m}$ height loss per 100 m line length

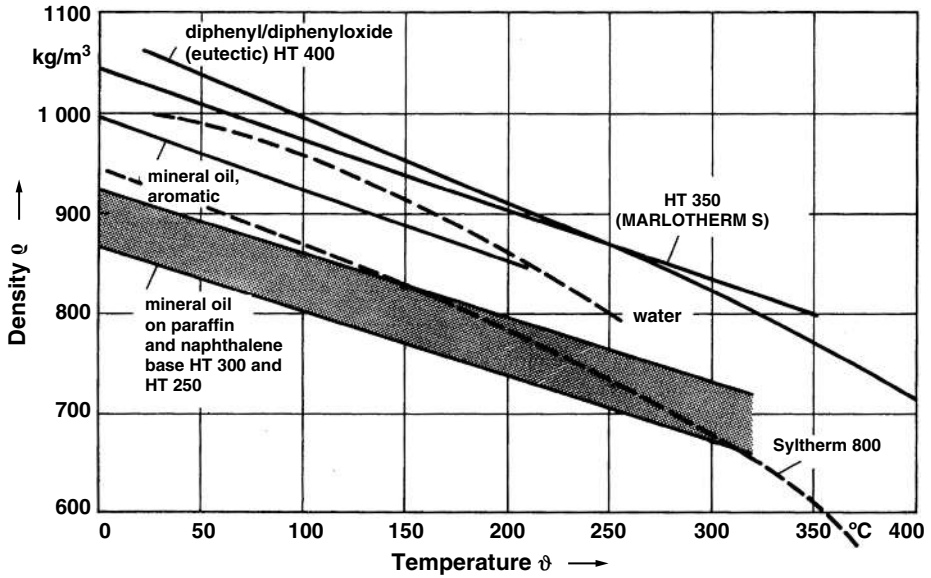


Fig. 4.3 Density ρ of heat transfer fluids

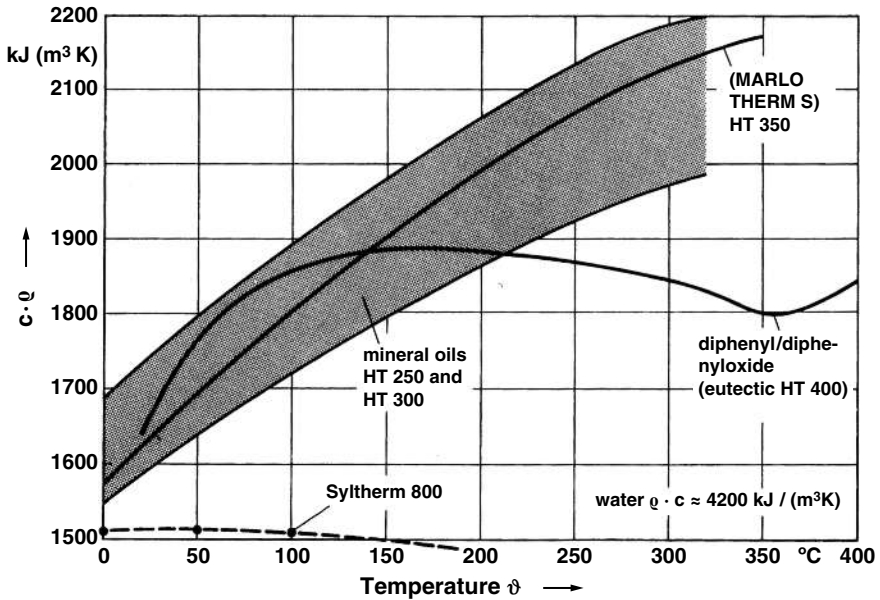


Fig. 4.4 Specific volumetric heat capacity $c \cdot \rho$ of heat transfer fluids

If the frequently used temperature difference of $\Delta\vartheta_{VR} = 20$ K is inserted, the following rule of thumb results:

$$\bar{V} \approx \frac{\dot{Q}_H}{38,000} [\text{m}^3/\text{h}] \quad \text{with } \dot{Q}_H \text{ in kJ/h} \quad (4.9)$$

or

$$\bar{V} \approx \frac{\dot{Q}_H}{10} [\text{m}^3/\text{h}] \quad \text{with } \dot{Q}_H \text{ in kW}$$

This relationship is shown graphically in Fig. 4.5.

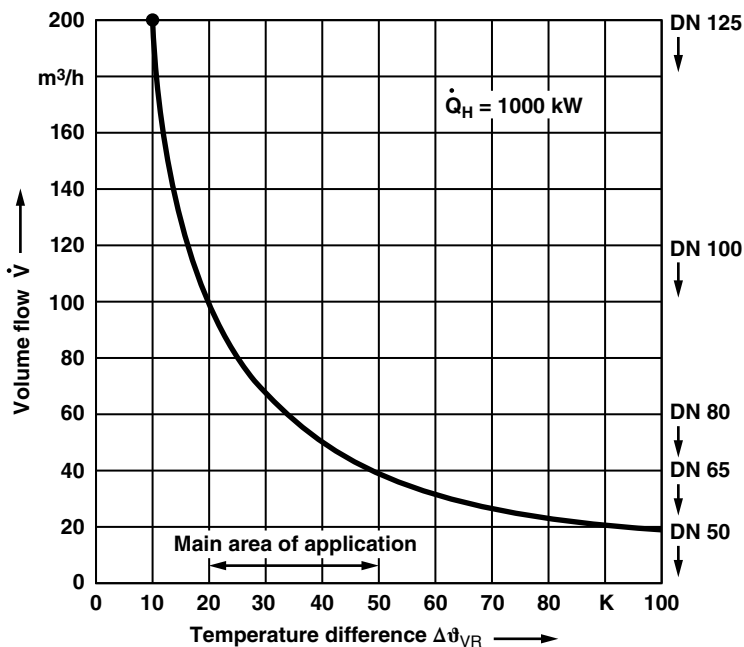


Fig. 4.5 Guide values for the volume flow \dot{V} depending on the temperature difference between the feed and the return $\Delta\vartheta_{VR}$ at a power of $\dot{Q}_H = 1000 \text{ kW}$. The required nominal diameter of the pipework is also given as a marginal value on the basis of Fig. 4.2.

4.2 Pressure Losses on the Heat Transfer Fluid Side

The heat uptake, heat transport and heat release is by the mass flow calculated in 4.1. The construction elements for carrying this mass flow are almost exclusively pipes or “hydraulically” equivalent components. As a substitute system a circuit consisting of pipes can be formed. The properties, such as flow form and specific resistance of this component, must be considered in more detail.

5 Heaters

Heaters in heat transfer plants have to raise the temperature level of the heat transfer fluid. The heaters have thus to meet requirements similar to those of e.g. a forced flow boiler for water. The organic heat transfer fluid having the property of decomposing from a certain temperature it is necessary not to exceed this temperature at any point in the heater.

This criterion runs as a "common theme" through all design specifications.

The term heater is defined as: the heat source is that point in the plant system at which the heat content (or its enthalpy) of the heat transfer fluid is increased. The heat flux to be transferred can either be generated in the heater itself, e.g. in fired or electrically heated heaters (heaters with heat generation) or is present already as heat enthalpy in the form of hot gases. In liquid circulation the temperature of the heat transfer fluid is increased by the amount:

$$\Delta\vartheta_{VR} = \frac{Q_H}{\dot{m} \cdot c_{VR}} \quad (5.1)$$

The energy can be added by physical or chemical heat generating processes.

During heating the heat is primarily transported in the thermal boundary layer. The temperature rise $\Delta\vartheta$ is a function of the heat flux density \dot{q} and the heat transfer coefficient α_i .

It is:

$$\Delta\vartheta = \frac{\dot{q}}{\alpha_i} \quad (5.2)$$

An example will illustrate the order of magnitude by which the maximum boundary layer temperature – equal to the internal tube wall temperature – increases over the average heat transfer fluid temperature $\bar{\vartheta}$, the heat flux density in a fired heater being assumed to be a maximum of 100,000 W/m². The heat transfer coefficient on the heat transfer fluid side is assumed to be 2000 W/(m² K). The temperature rise in the boundary layer is then 50 K.

5.1 Electric Heaters

An essential feature of electric heating is the defined heat flux density at the surface of the heating element. In view of the temperature increase $\Delta\vartheta$ on the surface of the heating element ($\Delta\vartheta = \dot{q}/\alpha$) the heat transfer fluid is the only remaining variable. Very different designs are possible depending on the choice of the two quantities. They are based on two design principles:

5.1.1 Electric Heaters in Container Design

Fig. 5.1 shows the principle. This design is mainly chosen for heat transfer fluids not subject to thermal decomposition.

This design principle is only conditionally applicable to heat transfer fluids having a high viscosity at the startup temperature and decomposing under thermal effect. Even using vanes and guide plates it is difficult to ensure a good flow around the whole surface of the heating element. If it is certain that the heat transfer fluid has a low viscosity on startup – or the heat release rate is reduced during this operational phase – organic heat transfer fluids, too, can be heated on this principle. The heat release rate should be limited to about 1 to 2 W/cm^2 in view of the not wholly unambiguous flow conditions and a safety margin should be present from the maximum permissible flow temperature.

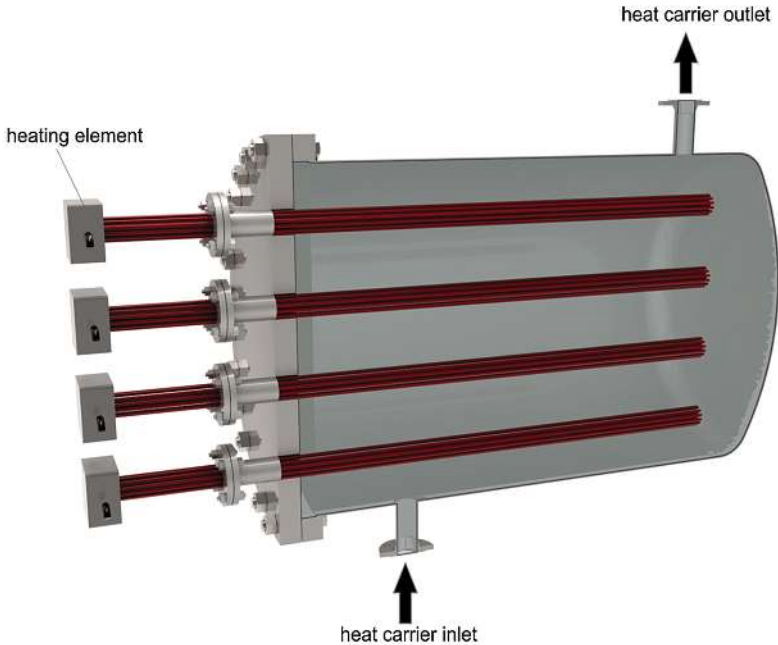


Fig. 5.1 Electrically heated heater in container design [heat 11]

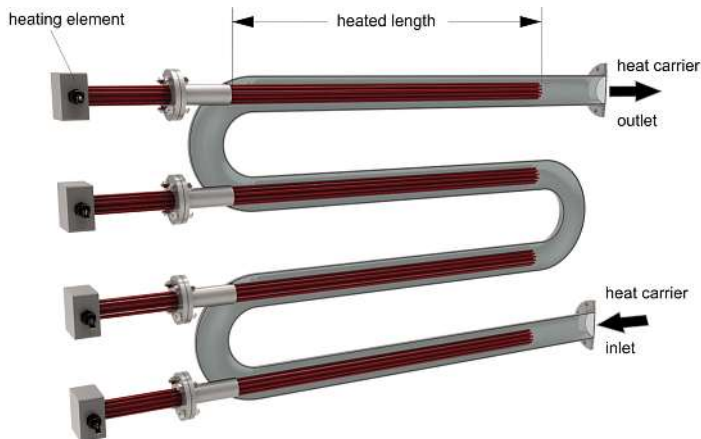


Fig. 5.2 Electrically heated heater in tubular design [heat 11]

5.1.2 Electric Heaters with Tube System

The flow conditions are in this case unobjectionable (Fig. 5.2). If the electrically heated surface is arranged in this flow zone, the two quantities determining the temperature rise at the heated wall are given and heat release rates up to e.g. 6 W/cm² are permissible. The cold start requires special attention. The viscosity may strongly increase with falling temperature and low heat transfer coefficients result. These may reach at room temperature sometimes only about 10 % of the design value. On commissioning the heating must be switched on only after venting the plant because gas bubbles may cause, because of their insulating effect, overheating damage of heating rods.

If a heat transfer coefficient of 2000 W/(m² K) at an operational temperature of 300 °C is assumed, analogous to the first example, an α value of only 200 W/(m²K) results on startup (at 20 °C). The temperature rise in the boundary layer is, at the chosen heat flux density of 6 W/cm² (corresponding to 60,000 W/m²), 300 K. The surface temperature of the heating element – identical to the maximum heat transfer fluid temperature – is 320 °C on startup. Since it is not desirable to approach the thermal limit of application of the heat transfer fluid on startup it is advantageous to start with a reduced heat release rate. If the upper limit of 2 W/cm² stated for the container design is chosen, the temperature rise in the boundary layer is only 100 K.

5.1.3 Heating Elements

If a voltage U is applied to an electrical conductor, a current I flows which generates at purely ohmic resistance a heat output of:

$$P = U \cdot I = I^2 \cdot R = \frac{U^2}{R} \quad (5.3)$$

For the generation of a limited heat output an electric resistance must be present resulting from the equation:

$$R = \varrho \cdot \frac{L}{A} \quad (5.4)$$

where ϱ = specific electric resistance (Ω mm²/m).

The electrical output in resistance heating with purely ohmic load is generated from equ. 5.3 and 5.4 according to the following basic equation:

$$P = \frac{A}{\varrho \cdot L} \cdot U^2 \quad (5.5)$$

where:

- L = length of heat conductor
- A = cross-section of heat conductor
- ϱ = specific ohmic resistance
- U = applied voltage

At a voltage of 230 or 400 V cross-section and length of conductor result for a certain material. The resistance wire is spirally wound and inserted into e.g. grooves of ceramic insulators. To safeguard the resistance material against vibrational fractures and short circuit it is usually embedded in a densified insulator (mostly magnesium oxide MgO) and surrounded by a metallic sheath. The resistance material should be applied so that a uniform heat flux density occurs on the tube surface. The average heat release rate is calculated from:

$$\dot{q} = \frac{P}{d \cdot \pi \cdot L} \quad (5.6)$$

Design Types

The following types of tubular heating elements exist:

- a) with inserted resistance spiral and an electrical connection at both sides (Fig. 5.4). This type is predominantly used for heating elements with an external diameter of 8.5 mm. To enable connection on one side they are turned around 180° and returned straight or spirally. The smallest bending roller diameter of steel sheaths is about 25 mm.

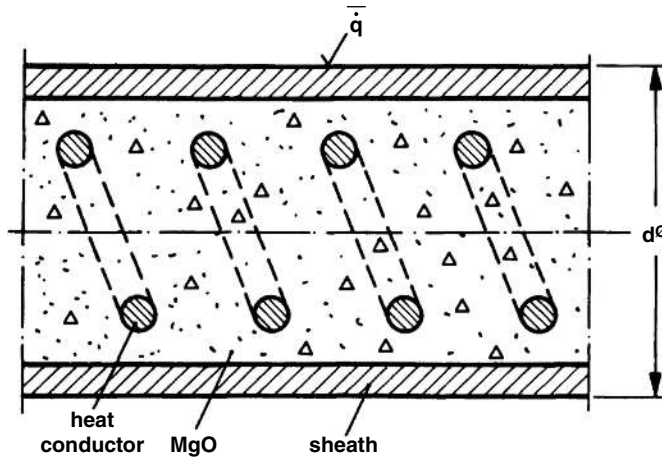


Fig. 5.3 Cross-section through an electric heating element

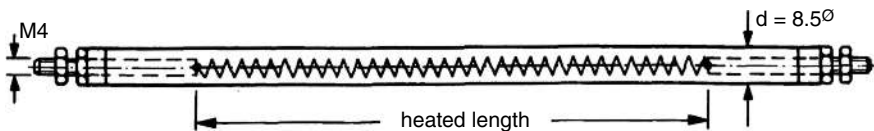


Fig. 5.4 Tubular electric heating element

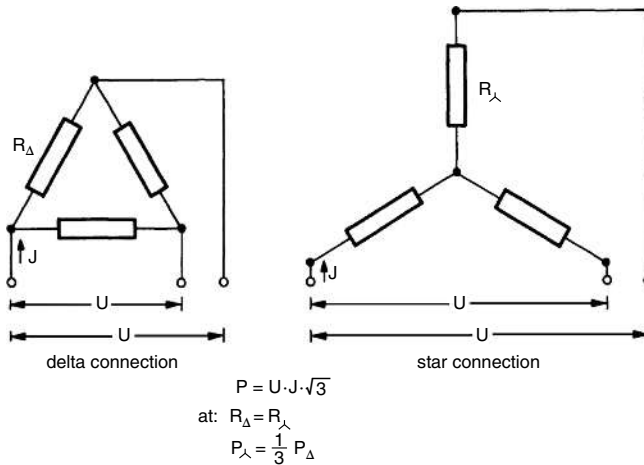


Fig. 5.5 Star and delta connection of electric heating elements

b) in the shape of heating rods in which the electric resistance conductor is returned in the tube. The external diameter is then larger than in a).

Delta and star connections are used. With type a) and combining three individual heating elements in a basic unit the switching over from e.g. star to delta can be carried out on these elements. With type b) the use of three heating spirals is directly possible and by taking out the star ends delta or star connection is possible (Fig. 5.5).

If a part voltage is required on the heating elements which cannot be obtained by switching over from delta to star, the heating elements must be arranged in series (Fig. 5.6).

5.1.4 Heating Element Installation

Heating elements should be installed in the heaters so that leakage of the heat transfer fluid is prevented and a replacement of the heating rods is possible without draining the heat transfer fluid. The two most usual installations are flange connection or welding (Fig. 5.7). The replacement of the heating elements without interruption of operation is obtained by the installation of sheaths around the heating rods.

In sheath construction the main resistance to the heat transport \dot{q} results from the air gap between heating rod and sheath, even with very small gap widths. If a gap width of 1 mm is assumed at a heat release rate of 6 W/cm² (Fig. 5.8) a heating spiral temperature of about 1000 °C occurs at a flow of the heat transfer fluid with $\alpha = 2000$ W/(m² K) around it. A heating rod directly immersed in the flowing heat transfer fluid attains a heating spiral temperature of only about 400 °C. The heat transfer in the air gap results from the fractions of thermal conduction, convection and radiation. The convection fraction can be neglected and it is, by iteration, the overall heat transfer in the gap about $\alpha_{sp} = 100$ W/(m² K) corresponding to a temperature increase of 600 K.

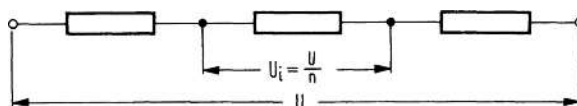


Fig. 5.6 Series connected electric heating elements

6 Plant Components

6.1 Heating

Furnaces which can be rapidly controlled and shut down are used for heating heaters, in addition to the electric heating elements and hot gases described in section 5. Fuels are mainly fuel oils and fuel gases.

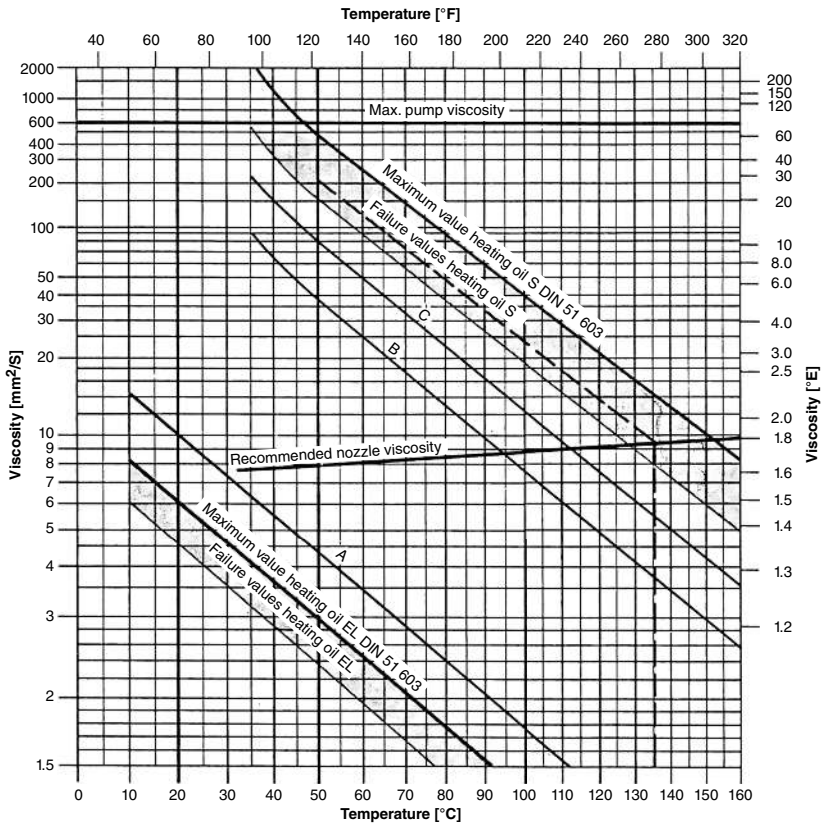
This already exists in molecular form for gaseous fuels. No fragmentation of the fuel is required, only mixing of combustion gas and combustion air.

Table 6.1 Minimum requirements for heating oils according to DIN 51603

| | | Heating oil EL DIN 51603 Part 1 | Failure value | Heating oil S DIN 51603 Part 2 | Failure value | Testing according to |
|---|--------------------|---------------------------------------|----------------|--------------------------------------|-----------------|---|
| Density at 15 °C | g/ml | max. 0.860 | 0.84-0.86 | no information | approx. 0.96 | DIN 51 757 |
| Flash point in closed crucible | °C | above 55 | 60-90 | above 65 | 90-180 | DIN 51 755 DIN 51 758 |
| Kinematic viscosity at 20 °C | mm ² /s | max. 6.0 | 4.5-6 | - | - | DIN 51 561 DIN 51 562 |
| at 50 °C | mm ² /s | - | - | max. 450 | approx. 150-450 | |
| at 100 °C | mm ² /s | - | - | max. 40 | approx. 18-40 | |
| Boiling range total evaporated volume proportion up to 250 °C | % | less than 65 | approx. 40 | - | - | DIN 51 751 ASTM D 86 |
| up to 350 °C | % | min. 85 | approx. 95 | - | - | |
| Pour point equal to or lower than | °C | - 6 | approx. - 10 | no information | 25-35 | EN 6 ISO 3016 |
| Coke residue according to Conradson | Mass % | max. 0.1 | approx. 0.05 | max. 15 | 9-15 | DIN 51 551 |
| Sulfur content | Mass % | max. 0.30 | approx. 0.3 | max. 2.8 | 1-2.5 | DIN 51 450 DIN 51 409 DIN 51 768 EN 41 DIN 51 400 |
| Water content not deductible | Mass % | max. 0.05 | Traces | max. 0.5 | approx. 0.1 | DIN 51 786 DIN 51 777 |
| Content of sediments | Mass % | max. 0.05 | Traces | max. 0.5 | approx. 0.1 | DIN 51 789 |
| Lower heating value | MJ/kg | min. 42.0 | approx. 42.7 | min. 39.8 | approx. 41 | DIN 51 900 |
| | kcal/kg | min. 10,000 | approx. 10,200 | min. 9,500 | approx. 9,800 | |
| | kWh/kg | min. 11.6 | approx. 11.8 | min. 11.0 | approx. 11.4 | |

Table 6.1 Minimum requirements for heating oils according to DIN 51603 – Continued

| | Heating oil EL DIN 51603 Part 1 | Failure value | Heating oil S DIN 51603 Part 2 | Failure value | Testing according to |
|--|---------------------------------------|---------------|--------------------------------------|---------------|----------------------|
| Ash (oxide ash) Mass % | max. 0.01 | Traces | max. 0.15 | approx. 0.04 | EN 7 |
| Part 1: Edition Dec. 81 Part 2: Edition Oct. 76 | | | | | |

**Fig. 6.1** Viscosity behavior of heating oils in relation to temperature [acc. Weishaupt]**Calculation of the preheating temperature**

The relevant preheating temperature can be found from the viscosity-temperature diagram. The viscosity of the heating oil in question must be known at a reference temperature. It must be taken into account that, due to heat loss between oil preheater and nozzle, the thermostat must be set higher to compensate.

Reading example

Known: MS-burner, oil throughput 150 kg/h, heating oil viscosity: 200 mm²/s at 50 °C. To be discovered: Preheating temperatures in °C. Procedure: Move vertically from the viscosity reference temperature of 50 °C up to the intersection point with the viscosity of 200 mm²/s, from there draw a line parallel to the nearest oil characteristic curve up to the "recommended nozzle viscosity" line and, from this intersection point, go vertically to the temperature axis. The sought preheating temperature of 135 °C is read here.

Recommendation

The following maximum viscosities should be considered in relation to nozzle size: In the area between the oil curves A and B nozzles > 0.85 USgph, B and C nozzles > 2.00 USgph and above C nozzles use > 5.00 USgph.

For liquid fuels, the increase in surface area is achieved via pressurized atomization or rotary atomization.

6.1.1 Liquid Fuels

The major proportion of liquid fuels are distillation products of mineral oil. The comments made in section 2 regarding the physical and chemical properties of organic heat transfer fluid on mineral oil base are thus valid. Attention is drawn in particular to the studies of these fluids described in section 2. Viscosity, boiling behavior and impurities form the essential differences between liquid fuels and organic heat transfer fluid. The minimum requirements for fuel oils are stated in DIN 51 613 (Table 6.1). The types EL (extra thinly liquid) and S (viscous) are mainly used. The application ranges result from the varying condition for preheating necessary for transport and combustion (Fig. 6.1).

Viscosity indicates the fluidity of the oil. Viscosity decreases on heating and increases on cooling. Viscosity is still measured in Stoke (St) or Centistoke (cSt). In foreign countries different units are used (Table 6.2), especially Sayboldt seconds in USA (SU = Sayboldt Universal) and Redwood seconds in Gt. Britain (R1 = Redwood seconds 1). For unproblematic atomization in pressure atomizing burners the fuel oil must have a viscosity of about 12 to 22 cSt, for rotary cup burners about 60 cSt. This is always the case for fuel oil EL. Fluids and heavy oils, on the other hand, must be heated before being burned. The maximum pump viscosity for gear pumps is about 800 cSt ($1 \text{ cSt} = 1 \cdot 10^{-6} \text{m}^2/\text{s}$).

Table 6.2 Comparison of different units of viscosity

| Kinematic viscosity cSt | Degree Engler E | Redwood 1 seconds (standard) | Sayboldt (SUS) (test) |
|-------------------------|-----------------|------------------------------|-----------------------|
| 1 | 1.00 | 28.5 | -- |
| 2 | 1.12 | 31.0 | 32.6 |
| 3 | 1.22 | 33.0 | 36.0 |
| 4 | 1.30 | 35.5 | 39.1 |
| 5 | 1.40 | 38.0 | 42.3 |
| 6 | 1.48 | 41.0 | 45.5 |
| 7 | 1.56 | 43.5 | 48.7 |
| 8 | 1.65 | 46.0 | 52.0 |
| 9 | 1.75 | 49.0 | 55.4 |
| 10 | 1.83 | 52.0 | 58.8 |
| 12 | 2.02 | 58.0 | 65.9 |
| 14 | 2.22 | 64.5 | 73.4 |
| 16 | 2.43 | 71.5 | 81.1 |
| 18 | 2.65 | 78.5 | 89.2 |
| 20 | 2.90 | 86 | 98 |
| 22 | 3.10 | 93 | 106 |
| 24 | 3.35 | 101 | 115 |
| 26 | 3.60 | 109 | 123 |
| 28 | 3.85 | 117 | 132 |

Table 6.2 Comparison of different units of viscosity – Continued

| Kinematic viscosity cSt | Degree Engler E | Redwood 1 seconds (standard) | Sayboldt (SUS) (test) |
|-------------------------|-----------------|------------------------------|-----------------------|
| 30 | 4.10 | 125 | 141 |
| 35 | 4.70 | 144 | 163 |
| 40 | 5.35 | 164 | 186 |
| 45 | 6.00 | 185 | 208 |
| 50 | 6.65 | 205 | 231 |
| 60 | 7.90 | 245 | 277 |
| 70 | 9.24 | 284 | 323 |
| 80 | 10.6 | 324 | 370 |
| 90 | 11.9 | 365 | 416 |
| 100 | 13.2 | 405 | 462 |
| 114 | 15 | 461 | 527 |
| 152 | 20 | 614 | 702 |
| 227 | 30 | 921 | 1053 |
| 303 | 40 | 1228 | 1404 |
| 379 | 50 | 1535 | 1756 |
| 400 | 53 | 1620 | 1848 |
| 520 | 69 | 2150 | 2500 |
| 620 | 82 | 2530 | 3000 |
| 720 | 96 | 2960 | 3500 |
| 900 | 120 | 3500 | 4000 |
| 1080 | 143 | 4435 | 5000 |

If fuel oils of different viscosity are mixed, the viscosity of the mixture can be determined from the "mixing diagram". Only mineral fuel oils can be mixed unobjectionably. When mixing mineral fuel oils with synthetic fuel oils, e.g. mineral fuel oil EL with coal tar oil, solid precipitates may form due to disturbances of the solubility equilibrium causing clogging in the burners and/or they partly burn only at 180 °C.

Burners for Liquid Fuels

Oil burners must atomize the fuel oil as finely as possible, mix it intensively with combustion air and ignite the fuel-air mixture. Stabilizers ensure stable burning of the flame. DIN 4787 contains the definitions, requirements and specifications for the design and testing. The majority of presently used burners are pressure and rotary cup atomizing burners.

Pressure Atomizing Burners

Pressure atomization only expanding the oil through a perforated nozzle is not used. The pressure is converted in tangential bores or channels of a swirl chamber partly into velocity (Fig. 6.2), the remaining amount generates the axial exit velocity. The high circumferential velocity results in fine atomization of the fuel oil, the resultant of circumferential and axial velocity determining the

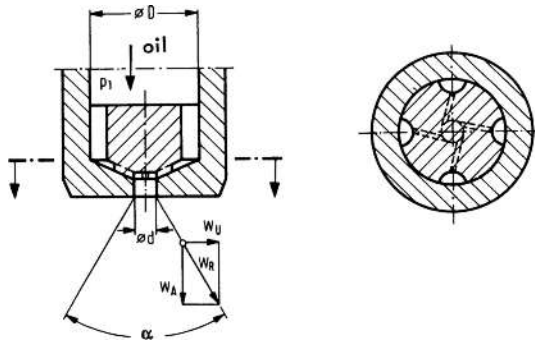


Fig. 6.2 Section through an oil burner nozzle

direction of the flying droplet and consequently the angle of the atomization cone. Swirl and fineness of atomization decrease with load (fuel oil volume) so that the control range of single nozzles is limited to 1 : 2. Due to the far advanced state of development of oil burner technique in USA the American technique was largely taken over on the introduction of liquid fuels in the German Federal Republic. The American definitions are still used for the characterization of oil burner nozzles. All nominal flow rates printed on the nozzles refer to an overpressure of 100 psi (7 bar). The flow rates obtained are measured in US gallons per hour (1 US gph \approx 3.785 liter/h). The flow rate is according to Bernoulli's equation:

$$V_{\text{Oil}} \sim \sqrt{\Delta p} \quad (6.1)$$

The nozzle size should be because of the danger of clogging:

- 0.85 US gph for fuel oils up to $38 \cdot 10^{-6} \text{ m}^2/\text{s}$ (at 50 °C)
- 2.0 US gph for fuel oils from 38 to $80 \cdot 10^{-6} \text{ m}^2/\text{s}$ (at 50 °C)
- 3.0 US gph for fuel oils above $80 \cdot 10^{-6} \text{ m}^2/\text{s}$ (at 50 °C)

It is necessary to fix the atomizing pressure when choosing the nozzle size. The following guiding values can be used (Fig. 6.3):

- fuel oil EL: 8 to 12 bar
- fuel oil S: 18 to 22 bar.

The average density of fuel oils can be based on a statistical determination for:

- fuel oil EL $\rho = 0.84 \text{ kg/l}$
- fuel oil M $\rho = 0.93 \text{ kg/l}$
- fuel oil S $\rho = 0.97 \text{ kg/l}$

The spray characteristic of nozzles is very variable in design, from full to hollow jet nozzles at a spray angle from 30 to 90°. It must be stated when ordering the nozzles. The full jet nozzle with a spray angle of 60° is most commonly used.

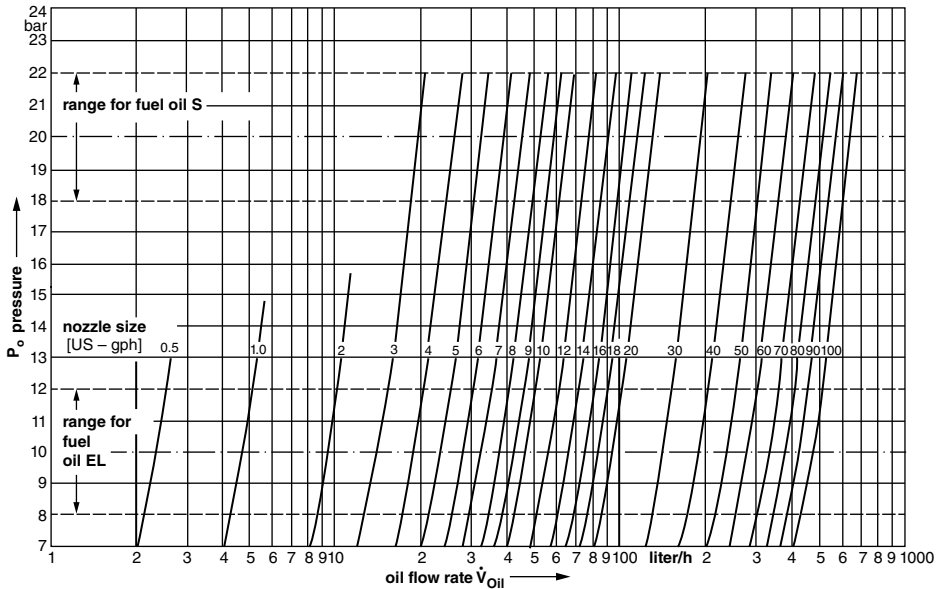


Fig. 6.3 Oil flow rate at burner nozzle as function of upstream oil pressure at nozzle (guiding value)

Example:

The total nozzle size is to be determined for a burner firing fuel oil EL at a rate of 117 kg/h (Fig. 5.80). The upstream pressure was measured to be $p_o = 15.5$ bar.

$$V_{\text{Oil}} = \frac{\dot{B}}{\rho}$$

The density ρ is taken from Table 6.1 at 0.86 kg/liter. The oil flow rate is thus:

$$V_{\text{Oil}} = \frac{117}{0.86} = 136 \text{ Liter/h} \cong \frac{136}{3.785} = 35.9 \text{ US-gph}$$

The nozzle size is then from equ. 6.1:

$$\text{total nozzle size} = V_{\text{Oil, US}} \cdot \sqrt{\frac{7}{p_o}} = 35.9 \sqrt{\frac{7}{15.5}} = 24 \text{ gal.}$$

The small control range of these nozzles can be extended by use of "recirculating nozzles". The control range is extended by maintaining the inlet flow rate constant and by making a bore (Fig. 6.4) in the swirl chamber a possibility in the nozzle to divert a partial flow of the incoming oil. This produces a control range of about 1 : 4. For this type of nozzle the flow rate must be determined by measuring the volume or through diagrams valid for a certain inlet pressure (Fig. 6.5).

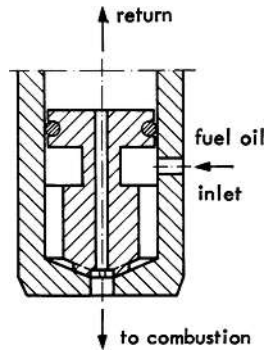


Fig. 6.4 Cross-section through an oil burner return nozzle

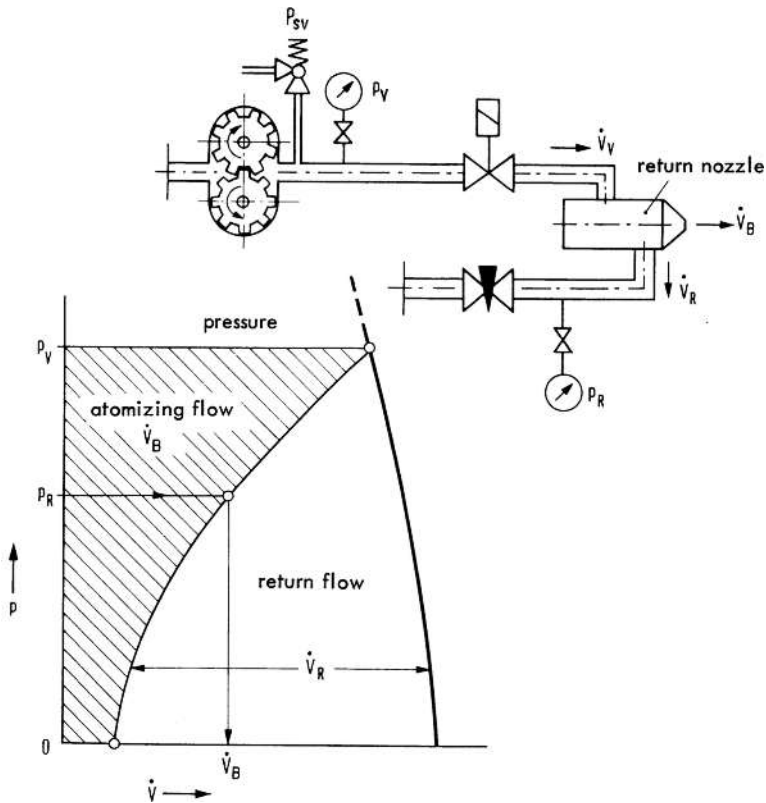


Fig. 6.5 Dependence, in principle, of atomizing flow of return nozzle on return pressure p_R

Function Schemes

The control of the fuel oil can be presented by three basic types:

1. single-stage burners
2. two-stage burners
3. continuously controllable burners

Light Oil Burners

Single-Stage Burner

Its characteristic is the on-off switching of the fuel oil flow (Fig. 6.6). The air volume is normally set once and for all at the oil nozzle damper. On switching on the control current the purging and preignition are started. The solenoid valve is still closed and the oil inlet to the nozzle remains blocked. After a certain time the solenoid valve opens and releases the oil flow to the nozzle, the flame appears.

Two-Stage Burner

Two nozzles are used for the oil atomization (Fig. 6.7). Two magnet valves are installed in the nozzle lines. Two-stage burners are used for oil flow rates from about 10 kg/h. The heat transfer fluid can thus be more slowly heated in the heaters than with single-stage burners and the startup impact is smaller. The oil flow rate of the first stage is usually between 20 and 60 % of the total amount and must be chosen so that on startup the heat transfer fluid is not overheated (section 5). In the startup position the magnet valve of the 1st stage opens first. Switching on and off of the magnet valve of the 2nd stage is done by control contacts independent of the 1st stage. The contact for the nozzle of the first stage is set to the upper limit value, the contact for the 2nd stage to the lower limit value. Each nozzle is correlated with a certain combustion air volume and, for this reason, these burners are equipped with special combustion air control devices. This can be done e.g. with an electric power cylinder operating the air damper in the air duct.

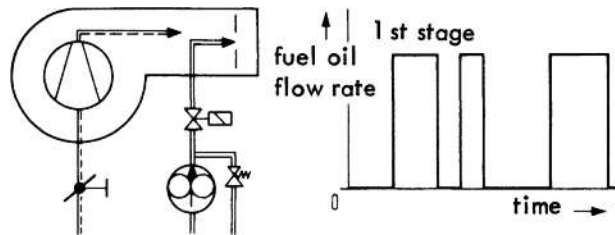


Fig. 6.6 Circuit diagram of a single-stage oil burner

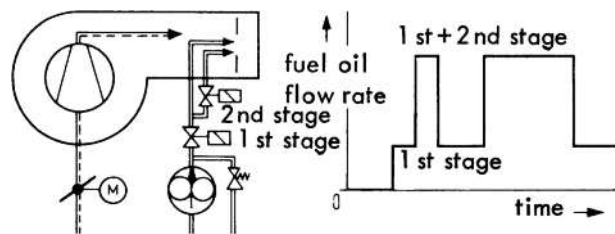


Fig. 6.7 Circuit diagram of a two-stage oil burner

Continuously Controllable Burners

A prestage is the sliding two-stage control. After switching on the 1st stage, the 2nd stage is steadily increased up to full load within a constant time (e.g. 1 minute). After reaching the rated temperature the 2nd stage is continuously reduced to the 1st stage. If the rated temperature is reached with the 1st stage, this basic stage is also switched off. Burners of this type have a recirculation nozzle and an automatic power cylinder for the air volume and oil flow rate. If the

sliding two-stage control is controlled by a stepless controller, the continuous, also called modulating control results (Fig. 6.8). The combined control of oil and air must ensure the combustion data at all load stages.

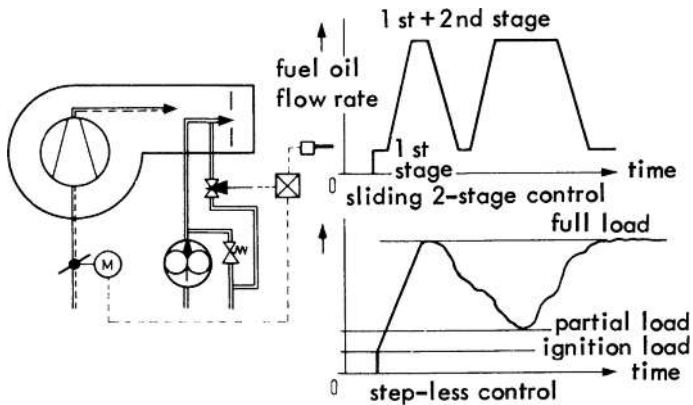


Fig. 6.8 Circuit diagram of a continuously controllable oil burner

Medium and Heavy Oil Burners

They function like light oil burners. Additional components are the oil preheater, nozzle heating and, if necessary, tracer heating. A release thermostat must be provided directly at the burner preventing the burner starting with cold oil. The burner control current is released only after the set minimum fuel oil flow temperature has been reached.

Rotating Cup Burners

Atomization is by centrifugal force at the rim of a fast ($\approx 6000 \text{ min}^{-1}$) rotating cup (Fig. 6.10) on the burner shaft. The fuel oil is drawn out to a uniform thin film detaching from the rim of the cup. Atomization is improved by the primary air introduced by a fan wheel. It forms from the disk-like oil film an atomizing cone so that it is also called "formation air".

The advantages over the pressure atomizer are:

- low oil overpressure (2 to 3 bar) because it only transports the oil
- low risk of fouling because of the absence of narrow bores
- viscosity of only $38 \text{ to } 55 \cdot 10^{-6} \text{ m}^2/\text{s}$ is required because irradiation into the cup results in additional preheating. This corresponds to $80 \text{ to } 90 \text{ }^\circ\text{C}$ for fuel oil S.

Burner Components

The main burner components are the fan, mixing device and oil pump.

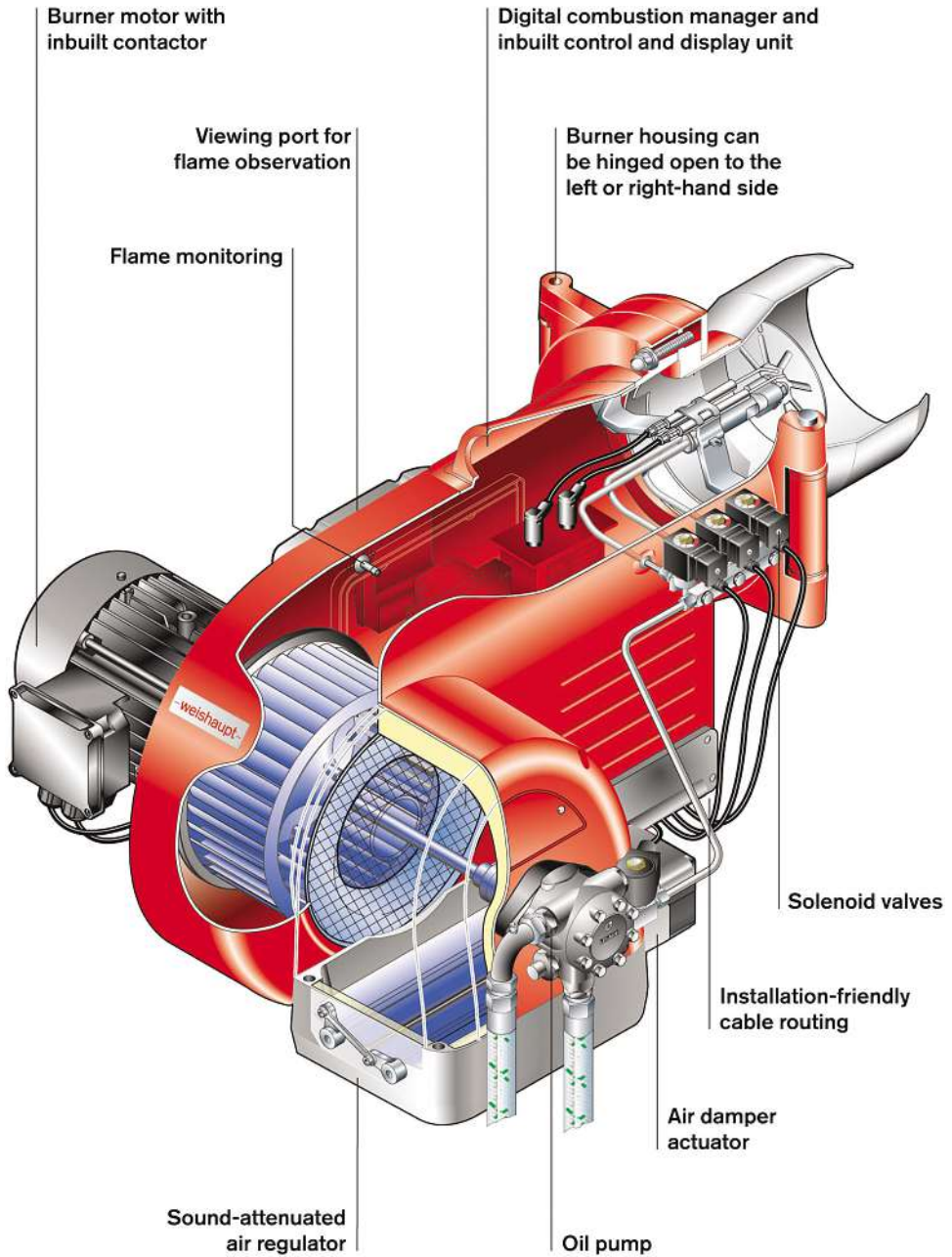


Fig. 6.9 Pressure-atomizing oil burners [Weishaupt, size: Monarch 5 to 11]

7 Heat Consumers

The preferred ranges of application of the heat transfer fluids technology are given by the different organic heat transfer fluids.

- Product heat generation at temperatures above 50 °C to 400 °C by a single primary heating system.
- Heating and cooling processes up to a coolant temperature of about –50 °C.
- Plants in which the risk of frost exists.
- Plants in which the preparation of the heat transfer fluid and/or the corrosion tendency of the material of the apparatus are to be avoided.

The heat consumers are generally heat exchangers in which the heat consuming fluid is separated by a wall from the heat transfer fluid.

During the thermal design it must be taken into account that the heater manufacturer rating plate cannot be used as the basis for the heat input temperature at the consumer. Differences arise here according to the circuit design of the system (see Fig. 7.1). In addition, a further temperature drop occurs due to heat losses, depending mainly on the thermal insulation and the pipework length.

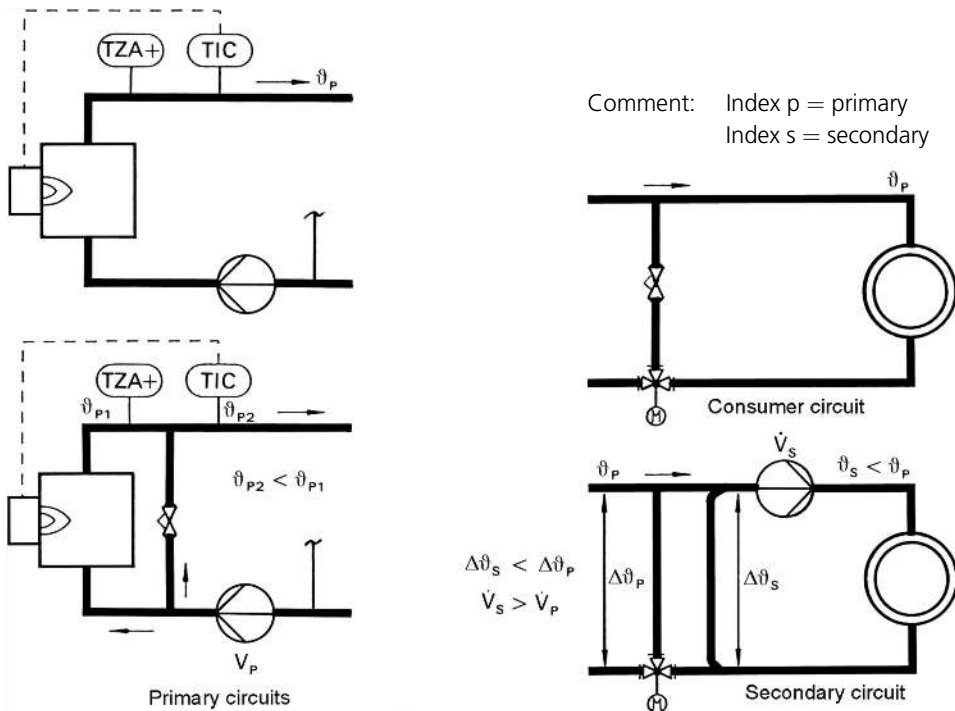


Fig. 7.1 Temperatures at the heater outlet and at the heat consumers

7.1 Determination of the Heat Requirement

When planning a heat transfer plant the heat requirement must be determined first. The temperature spread and the flow temperature at the heater can be determined for the product temperatures required by the process. The heat requirement results from the enthalpy difference of the mass between initial and final state per unit time plus the heat losses:

$$\dot{Q}_H = \frac{m \cdot \Delta h_{12}}{t} + \dot{Q}_v \quad (7.1)$$

This general equation can be subdivided for the practical use into continuous and discontinuous heating processes and the following equations are obtained for the usual determination of heat requirement:

7.1.1 Continuous Heating Processes of Fluids Not Changing Their Condition of State

This comprises e.g. heat exchangers for liquids and gases. The heat output is as shown in Fig. 7.2:

$$\dot{Q}_H = \dot{m} \cdot \bar{c} \cdot \Delta\theta_{12} + \dot{Q}_v \quad (7.2)$$

The heat losses \dot{Q}_v are assumed to be roughly 5 %. It is then:

$$\dot{Q}_H \approx 1.05 \cdot \dot{m} \cdot \bar{c} \cdot \Delta\theta_{12} \quad (7.3)$$

7.1.2 Continuous Heating Processes of Fluids Changing Their Condition of State

This applies e.g. for evaporators. The heat output is as shown in Fig. 7.3:

$$\dot{Q}_H = \dot{m} \cdot \bar{c} \cdot \Delta\theta_{12} + \dot{m} \cdot \Delta h_v + \dot{Q}_v \quad (7.4)$$

or:

$$\dot{Q}_H = \dot{m}(h' - h_1) + \dot{m} \cdot \Delta h_v + \dot{Q}_v \quad (7.5)$$

or:

$$\dot{Q}_H = \dot{m}(h'' - h_1) + \dot{Q}_v \quad (7.6)$$

The heat losses can be assumed to be about 10 %. It is then:

$$\dot{Q}_H \approx 1.1 \cdot \dot{m} \cdot (h'' - h_1)$$

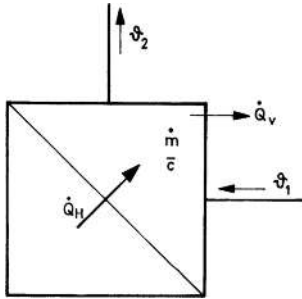


Fig. 7.2 Characteristic quantities for continuous heating

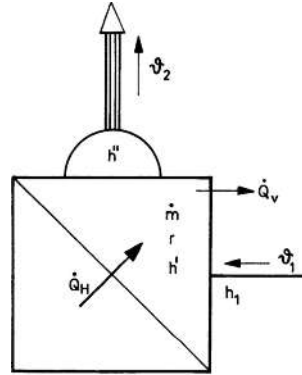


Fig. 7.3 Characteristic quantities for heating of steam generators

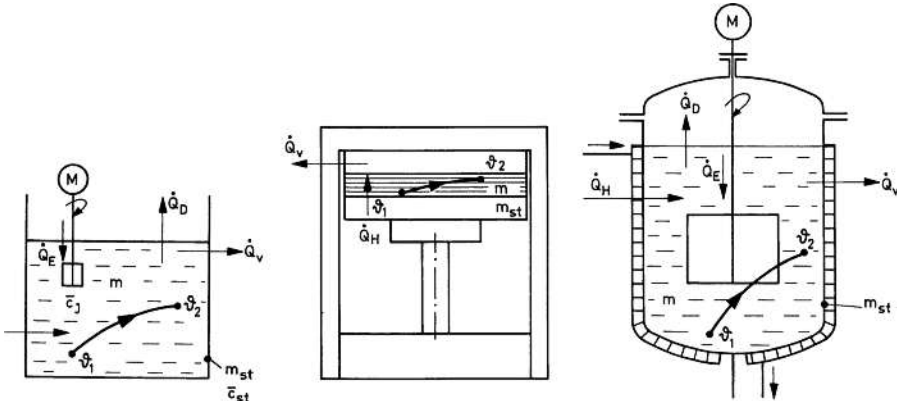


Fig. 7.4 Characteristic quantities for discontinuous heating

For continuous heating processes the heating surface required can be determined from the equation in section 4.3.5:

$$A = \frac{\dot{Q}_H}{k \cdot \Delta\vartheta_{\log}} \tag{7.7}$$

7.1.3 Discontinuous Heating Processes

This applies mainly for the heating of tanks, presses, containers and stirrer vessels. The heat output is as shown in Fig. 7.4:

$$\dot{Q}_H = \frac{m \cdot \bar{c} \cdot \Delta\vartheta_{12} + m_{st} \cdot \bar{c}_{st} \cdot \Delta\vartheta_{st} + m \cdot \Delta h_{sch} + m \cdot \Delta h_v}{t} + \dot{Q}_v + \dot{Q}_D - \dot{Q}_E \tag{7.8}$$

If the product passes through different conditions of state, the specific heats of the individual conditions must be inserted. The steel mass m_{st} includes also the proportion of the mass of insulation etc. The heat requirement equation must be matched for the individual case to the actual requirement according to equ. 7.8. The heat losses can be assumed to be about 20 %. The heat exchange surface is calculated from the data of Fig. 7.5:

$$A = \frac{\dot{Q}_H}{k \cdot \Delta\vartheta} \ln \frac{\bar{\vartheta}_{VR} - \vartheta_0}{\bar{\vartheta}_{VR} - \vartheta_E} \quad (7.9)$$

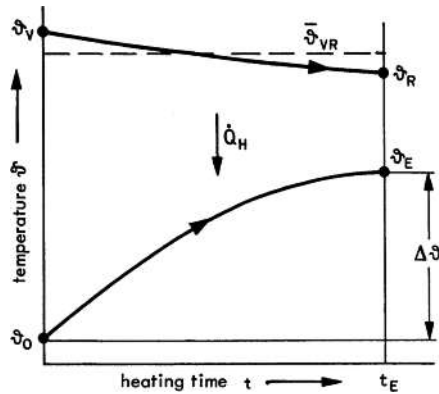


Fig. 7.5 Temperature data in discontinuous heating

For a given exchanger surface A the heating up time can be determined by inserting equ. 7.8 into equ. 7.9.

Determining the Heat Losses \dot{Q}_v

According to Chapter 4, the heat losses can be calculated as:

$$\dot{Q}_v = \dot{q}_{...} \cdot A$$

with: A = Surface

The heat flux density $\dot{q}_{...}$ can arise due to radiation, convection or thermal conduction.

Radiation:

$$\dot{q}_{\text{Rad}} = \varepsilon_1 \cdot \varepsilon_2 \cdot \varphi_{12} \cdot C_s \cdot (T_w^4 - T_u^4)$$

$$T = \vartheta + 273$$

$$C_s = 5.67 \cdot 10^{-8} \text{ W}/(\text{m}^2 \cdot \text{K}^4)$$

$$\varepsilon_1 \cdot \varepsilon_2 \cdot \varphi_{12} = 0.8 \quad \text{set in Fig. 7.6}$$

Convection:

$$\dot{q}_{\text{Conv}} = \alpha \cdot (\vartheta_w - \vartheta_u) (\text{W}/\text{m}^2)$$

With natural convection, this yields: $\alpha = K \cdot \sqrt[4]{\vartheta_w - \vartheta_u}$

K = geometry and material value factor

$K \approx 1.8$; α in (W/(m² · K))

Thermal Conductivity (Insulation):

$$\dot{q}_{\text{insulating}} = \frac{\lambda_D}{s_D} \cdot (\vartheta_{w,i} - \vartheta_0) \quad (7.10)$$

$\lambda_D = 0.1$ W/(m · K) and

$s_D = 100$ mm set in Fig. 7.6

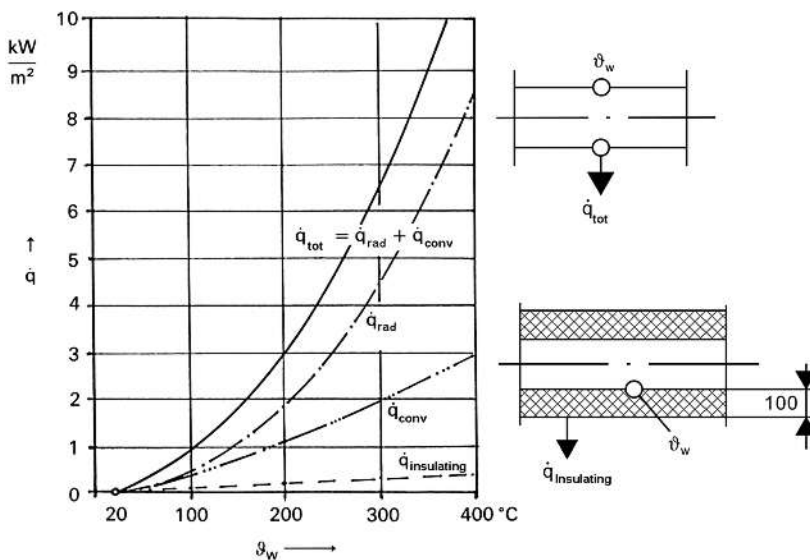


Fig. 7.6 Heat losses on uninsulated and insulated surfaces in relation to the temperature ϑ_w (for thermally insulated components ϑ_w is the internal temperature of the insulation.)

7.2 General Design Rules

The specific properties of organic heat transfer fluids must be taken into account in the design of the heating surface. Heat exchanger tubes, should, on principle, be welded to tube sheets. Because of the absence of corrosion on the heat transfer fluid side it must only be considered with regard to the design of the heating surface that it is well accessible and can be easily cleaned on the product side. The heat transfer fluid space must be provided with a vent so that the heating surface can be completely utilized and the heat transfer is not impaired by air or gas bolsters. In addition, a drainage possibility must be arranged.

A significant feature of the design are the high operating temperatures and the temperature differences. If the free expansion of the heating surface elements is not taken into account, very quickly stress values (section 6.4.2) arise which are above the yield point of the material and may cause fracture of the component at the appropriate number of load cycles. Care must be taken

8 Measuring, Control and Circuit Technology

Measurements in heat transfer plants are concerned with the determination of temperatures, pressures, levels and flows. From these measured quantities the control and limit quantities for influencing the heating and changes of the control organs are derived.

8.1 Measuring Technology

8.1.1 Temperature Measurement

The direct installation of temperature indicators at the measuring point is not usual because of the relatively high temperatures. The temperature value is transmitted to remote indicators through measuring sensors inserted into welded sheaths at the measuring points. Liquid thermometers with capillary tube are used only for subordinate temperature measurements because of their errors of indication if the conditions of the surroundings change. Electric contact thermometers are most frequently used.

Resistance Thermometers

The measured quantity is the change of resistance of metals (mainly platinum Pt) with increasing temperature. The measuring voltage at the required current source is usually 6 V. The basic resistance coefficient of platinum is 100 Ω at 0 $^{\circ}\text{C}$ (Pt-100); it changes by 38 Ω per 100 K.

The resistance at the measured temperature ϑ is calculated from:

$$R_{\vartheta} = R_0(1 + \alpha \cdot \Delta\vartheta) \quad [\Omega] \quad (8.1)$$

The temperature is normally not determined point-wise but as average value of the temperature over the whole winding length of 12 to 95 mm depending on type. The wires of 0.05 to 0.3 mm diameter are placed in the shape of a spiral on to cylindrical glass or ceramic insulators of 2 to 6 mm.

The resistance of the wires between measuring resistance and measuring instrument is included in the measurement and must be taken into account in equilibrating the measuring instrument. The supply wire resistance is standardized at 10 Ω (to DIN 43709). When commissioning a measuring installation the resistance coefficient of the supply wire must be equilibrated to 10 Ω .

The most usual is the two-wire circuit of Fig. 8.1; the equilibration resistance is installed in one of the supply wires. A low current cable with good insulation (if possible 1.5 mm² Cu per conductor) is sufficient for the supply wire. 100 m of supply and return wire with a cross-section of the wires of 1.5 mm² Cu each have a resistance of about 2.5 Ω at 20 $^{\circ}\text{C}$ room temperature. The temperature dependence of the electrical resistance becomes noticeably disturbing with the feeding. The three-wire circuit (Fig. 8.2) is generally used where the ambient temperature fluctuates.

Closed welded immersion sheaths are preferably used as protective tube to obtain a completely tight connection. The temperature sensor (Fig. 8.3) can then be replaced during operation.

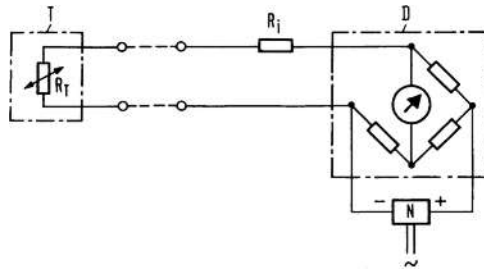


Fig. 8.1 Temperature measurement with resistance thermometer (two-wire circuit)

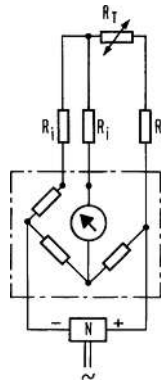


Fig. 8.2 Temperature measurement with resistance thermometer (3-wire circuit)

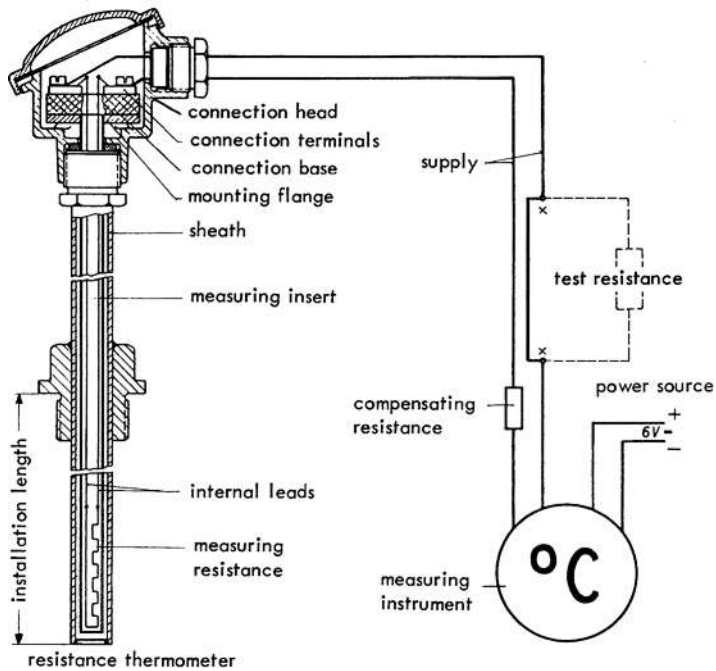


Fig. 8.3 Protective sheath for resistance thermometer

Thermocouples

Two wires of different metals or alloys welded or soldered at one end produce a thermal e.m.f. This depends on the metal wire pairing and the temperature difference between measuring point (soldering pearl) and indicating point. The material pairing for the temperature range of interest here is iron/constantan (Fe/const) or NiCr/Ni. A voltage source is not required.

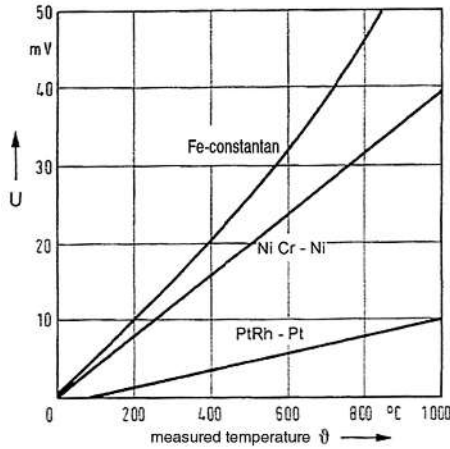


Fig. 8.4 Thermal e.m.f. of thermocouples as function of measuring temperature

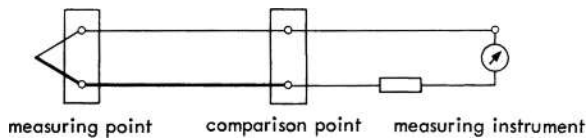


Fig. 8.5 Basic measuring arrangement of thermocouples

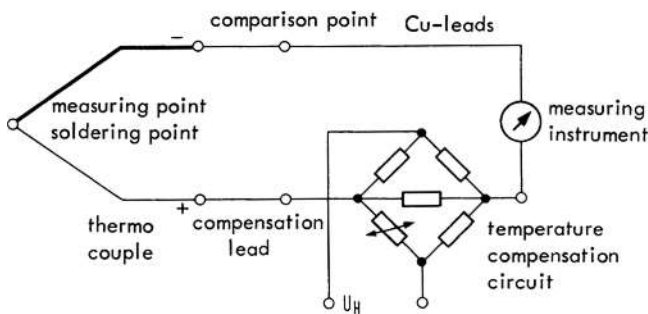


Fig. 8.6 Measuring arrangement of thermocouples with temperature compensation circuit of the comparison point

The thermal e.m.f. U is calculated from:

$$U = K \cdot (\vartheta - \vartheta_0) \quad [\text{mV}] \quad (8.2)$$

K is a factor depending on the thermocouple and to a slight degree on temperature. The thermo e.m.f. is shown in Fig. 8.4. The voltage level thus depends not only on the temperature ϑ at the measuring point but also on the temperature ϑ_u of the free ends of the thermowires. The temperature at the comparison point (Fig. 8.5) must be known and constant, if the measured result is to be correct. The ends of the thermowires (near the measuring point) are, for this reason, extended by wires of the same thermoelectric behavior (equalization wires) up to the point of constant temperature (temperature of comparison point). The comparison temperatures are usually 0 °C, 20 °C and 50 °C. If the reference comparison temperature of 20 °C (ambient temperature) cannot be assumed to be about constant, a temperature compensation of the comparison point is made in the form of an electric compensation bridge (Fig. 8.6) compensating the errors caused by temperature fluctuation at the comparison point. The protective sheath requirements are the same as for resistance thermometers and the arrangement with closed protective sheath is shown in Fig. 8.7.

Measuring Arrangement

Only a very small air gap should exist between the measuring sensor and the protective sheath welded into the system. The sensor and sheath should dip as far as possible into the fluid to be measured so that the measuring error due to thermal conduction through the protective sheath does not become excessive. The "soldering pearl" of thermocouples should always show against the direction of flow. Further hints will be found in the VDE/VDI specification 3512 "Measuring Arrangements for Temperature Measurements".

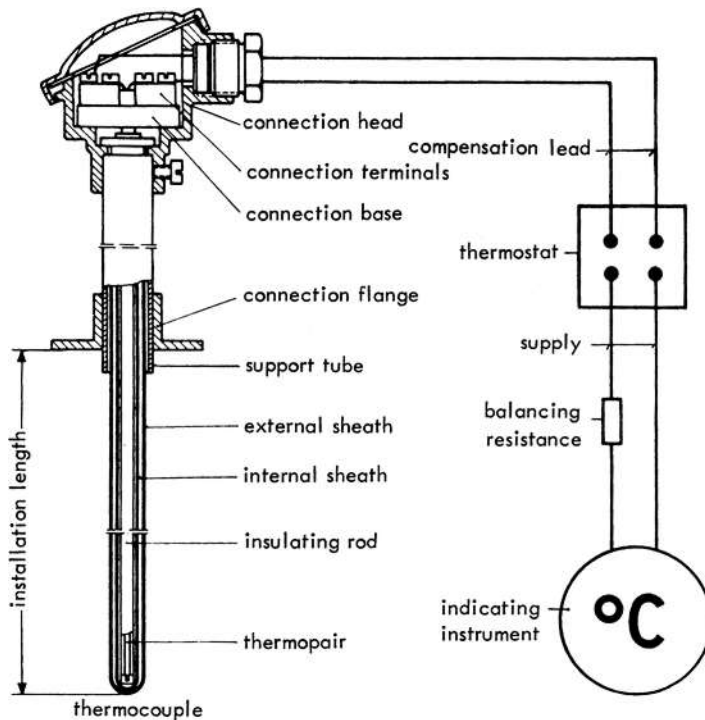


Fig. 8.7 Protective sheath for thermocouple

Installation Errors for Measurements in Gas Ducts

An installation error in flue gas and air ducts can occur when the measurement sensor participates in radiation exchange with walls that are normally colder. This error can be estimated using the following relationship.

Heat transfer due to radiation from the sensor to the walls (for small temperature differences):

$$\text{Emission ratio: } \epsilon_{\text{tot}} = 0.7$$

$$\text{Angular ratio: } \psi_{1.2} = 1.0$$

$$C_s = 5.67 \cdot 10^{-8} \cdot \text{W/m}^2\text{K}^4$$

$$\alpha_{\text{rad}} = 4 \cdot \epsilon_{\text{tot}} \cdot \psi_{1.2} \cdot C_s \cdot T_{\text{sens}}^3$$

As a 1st approximation, the sensor temperature can be set equal to the gas temperature. Heat transfer due to convection (estimated) cross-flow:

$$\text{Flue gas flow velocity: } w$$

$$\text{Outer diameter of sensor: } d_{\text{sens}}$$

$$\alpha_{\text{conv}} = 6 \cdot \frac{w^{0.67}}{\left(d_{\text{sens}} \cdot \frac{\pi}{2}\right)^{0.33}} \cdot \frac{W \cdot s^{0.67}}{K \cdot m^{2.34}}$$

The difference between gas- and sensor temperature is:

$$\Delta\vartheta_{\text{gas,sens}} = \frac{\alpha_{\text{rad}}}{\alpha_{\text{rad}} + \alpha_{\text{conv}}} \cdot (\vartheta_{\text{gas}} - \vartheta_w)$$

Example

A temperature sensor that exchanges radiation with the thermal oil tube basket heating surface is located at the outlet of the heater.

Question: How many °C lower is the temperature shown by the sensor as compared to the flue gas temperature?

$$\text{Tube basket heating surface: } \vartheta_w = 350 \text{ }^\circ\text{C}$$

$$\text{Temperature sensor diameter: } d_{\text{sens}} = 15.7 \text{ mm}$$

$$\text{Flue gas temperature: } \vartheta_{\text{gas}} = 450 \text{ }^\circ\text{C}$$

$$\text{Flow velocity: } w = 8 \text{ m/s}$$

$$\text{1st iteration: } \vartheta_{\text{sens}} = \vartheta_{\text{gas}}$$

Heat transfer through radiation:

$$\alpha_{\text{rad}} = 4 \cdot \epsilon_{\text{tot}} \cdot \psi_{1.2} \cdot C_s \cdot (\vartheta_{\text{sens}} + 273 \cdot \text{K})^3 \quad \alpha_{\text{rad}} = 60 \text{ W/m}^2\text{K}$$

Heat transfer through convection:

$$\alpha_{\text{conv}} = 6 \cdot \frac{w^{0.67}}{\left(d_{\text{sens}} \cdot \frac{\pi}{2}\right)^{0.33}} \cdot \frac{W \cdot s^{0.67}}{K \cdot m^{2.34}} \quad \alpha_{\text{conv}} = 82.1 \text{ W/m}^2\text{K}$$

9 Installation Specifications

9.1 Installation of the Heaters

Heaters in heat transfer plants must be installed so that the employees and the surroundings are not endangered by fire, explosions or hot heat transfer fluid.

These conditions are met, if the heaters are installed outdoors or in a special heater room and if, where the contents of the heater exceed 500 liters (to UVV-VGB 64), the installation outdoors has a safety distance of 10 m to buildings (the walls of which are not fire resistant) and to other plants. If insufficient space is available for installation outdoors, a fire-proof wall may be erected in place of the safety distance.

Heaters for the individual heating of machines and apparatus (to DIN 4754) may be installed in workshops, if continuous supervision is ensured during operation and the total contents of the heater including the containers do not exceed

- 2000 liters in fired heaters and
- 5000 liters in electrically heated heaters.

Installation above or below living rooms is not permitted.

For the installation of heaters the building supervisory specifications of the various States to be generally observed are valid. The expert commission building supervision of ARGEBAU produced a summary of the technical requirements of heater rooms (ZTA – heater rooms) enabling the States to make uniform requirements of heater rooms. The following statements are based on these ZTA heater rooms and on DIN 4754.

9.2 Heater Room Specifications

9.2.1 Heater Room Arrangement

The following boundary conditions decide the arrangement of the heater room within a building:

- As central a position as possible to all buildings connected to the central heater plant
- Suitable position to the fuel storage
- Sound insulation to the neighbourhood
- Arrangement of the expansion vessel at the highest point of the heat transfer plant
- Favourable position of the stack

9.2.2 Heater Room Design

It should be considered generally that:

- The heater room must be designed so that leaking heat transfer fluid or fuel cannot pass outdoors or into other rooms (e.g. by a threshold, a recess or a trough).
- No special foundations are generally necessary for the heaters.
- Each heater has its own stack.
- A craneway must be provided for larger plants.

a. Dimensions

a.1 The heater room must have a volume of at least 8 m³ and be dimensioned so that the furnace can be properly serviced and maintained, in particular a sufficient free space must be available in front, behind and above the furnaces for easy cleaning and maintenance.

a.2 The internal height of the heater room must be at least 2.00 m.

a.3 If the furnaces have to be entered from above, a distance of at least 1.50 m must be available for furnaces above 145 kW rated heating output and a distance of at least 1.80 m for furnaces above 349 kW between furnace and ceiling, support or pipelines.

a.4 Working platforms must be made of gridirons so that they do not impair the ventilation of the heater room.

b. Walls, Ceilings and Floors

b.1 The heater room must not be directly connected to social rooms.

b.2 The walls and supports of the heater room and the ceiling between the heater room and other rooms must be fire resistant. Walls and ceilings between the heater room and social rooms must be thermally insulated. Separating walls between the heater room and rooms belonging to the operation of the heating plant, except separating walls between heater room and oil storage room, can be made of nonflammable materials.

b.3 The floor must be safe for traffic and of nonflammable materials.

b.4 Openings for heating pipes or other pipes in walls, ceilings and floors must be closed.

c. Windows, Doors and Exits

c.1 The heater room must have at least one window opening to the open air. Cross-ventilation is to be aimed at. The inner dimensions of the window area should be at least $\frac{1}{12}$ of the floor area of the heater room. The devices for opening and closing the window must be at a suitable height and be easily operable.

c.2 Doors of heater rooms must open to the outside. Doors not leading to the outside must be at least fire retarding. Doors in separating walls, to section b.2 sentence 3 must be of nonflammable materials.

c.3 Heater rooms for furnaces with a total rated heating output of more than 349 kW must have, if possible, opposed exits. One of these exits must lead directly or through a hall not connected to other rooms into the open air; it may be designed as an exit from a window. If necessary, climbing irons must be provided.

d. Ventilation Plants**d.1** Ventilation

d.1.1 The heater room must have at least an equipment through which fresh air is drawn from outside and passed to the heater room (fresh air opening).

d.1.2 Suction openings lying on public or private traffic areas and less than 2 m above ground must be provided with an impact resistant grating.

Suction openings must have sufficient distance from unprotected openings of rooms with increased risk of fire or explosions.

d.1.3 Fresh air openings should be arranged near the floor.

d.1.4 For furnaces with a total rated output of 47 kW circular or rectangular openings leading directly into the open air must be at least 300 cm² and the length of the larger side must not be more than 1.5 times that of the shorter side. For each further 1.163 kW total rated heating output the openings of sentence 1 must be enlarged by 3 cm².

The free cross-section must be larger by 20 % for openings with crossbars.

Rectangular openings without crossbars where the length of the larger side is up to 5 times or 10 times that of the shorter side must be larger by at least 10 or 25 %, respectively.

d.1.5 If the fresh air is taken from a shaft arranged on the building wall, its cross-section must be at least 1.5 times the cross-section of the suction opening.

d.2 Vents

The flow rate of the venting plants must be at least 0.6 m³/h per each 1163 kW total rated heating output of the furnace, (if the combustion air is taken from the heater room) and at least 0.9 m³/h, if the combustion air is introduced through pipelines from outside.

The exact determination of the venting air cross-section can be determined from "ZTA heater rooms". A guiding value for natural venting is the size of the stack cross-section, at least, however, 180 cm².

e. Sound and Vibration Protection

Vibrations and noises originating from furnaces, ventilators or other installations must be damped so that dangers or excessive nuisances do not arise. The soundproofing of the heater room and the ventilation plants, in particular, must meet the requirements of DIN 4109 – soundproofing in buildings – in furnace plants requiring the licensing to paragraph 16 of the trade regulations and also the technical instructions for noise protection (TA Lärm).

f. Electrical and Fire Protection Installations

f.1 Electrical plants must meet the regulations to VDE 0100.

f.2 Outside the heater room at an easily accessible place not at risk a danger switch must be installed for those electrical operating means, the further operation of which may be a cause of danger in the event of trouble. The operating means which still have to operate effectively, such as flight path illumination, must be explosion proof.

f.3 A storage of combustible materials in the neighbourhood of the heat consumer as far as they are necessary for the operational process must be restricted to a minimum or avoided, if there is no automatic fire extinguishing installation.

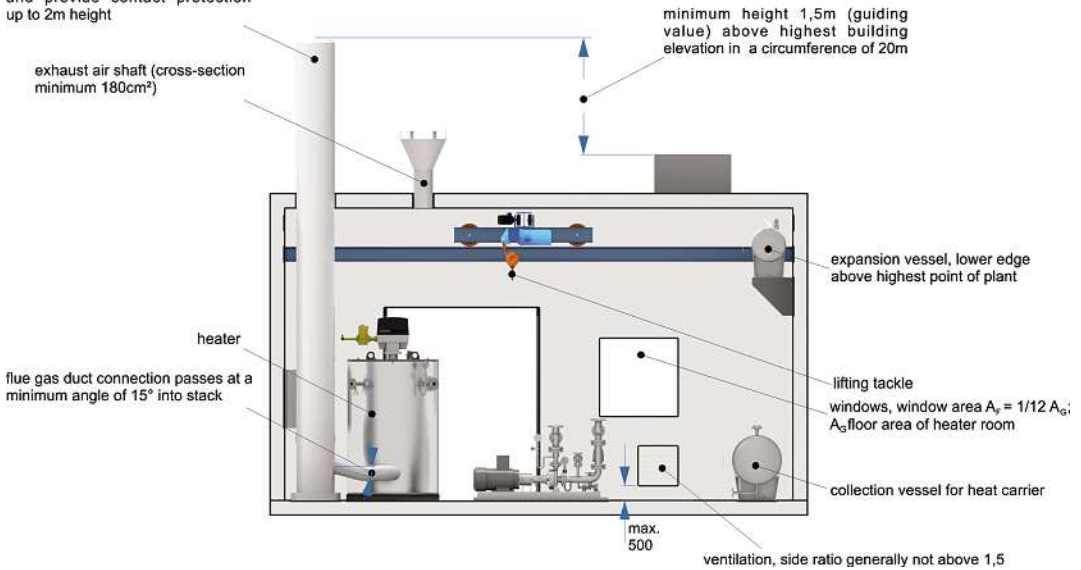
f.4 Fire extinguishers to DIN 14 406, fire class B, must be available in sufficient numbers near the heater.

9.2.3 Heater Room Design Example

If the design specifications of section 9.2.2 are taken into account, the basic design of central heating plants shown in Fig. 9.1 is obtained (cf. specification VDI 2050).

The installation of the expansion vessel decides the total height of the plant. The minimum heat transfer fluid level in the expansion vessel should be slightly above the highest point of the plant so that with the circulating pump at standstill vacuum cannot occur in the plant circuit system and consequently release of gas.

stack, consider temperature drop of flue gases, insulation in heater room and provide contact protection up to 2m height



Doors must open to the outside; the largest component must be considered when planning doors

danger switch switching off safety circuit on control circuit

electrical switch installation

stack cleaning opening

stack, consider temperature drop of flue gases, insulation in heater room and provide contact protection up to 2m height

flue gas duct, the plate thickness should be 2mm up to 200mm i.d., 3mm up to 300mm i.d and 4mm over 300mm i.d.

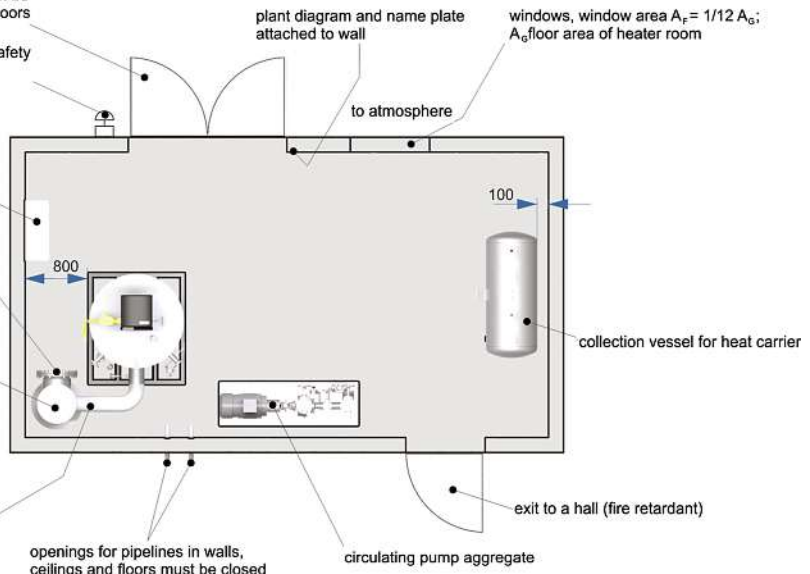


Fig. 9.1 Example of design of a central heating installation [heat 11]

A guiding value for the dimensioning of the ventilation cross-section is given in Fig. 9.2 as function of the total heater output.

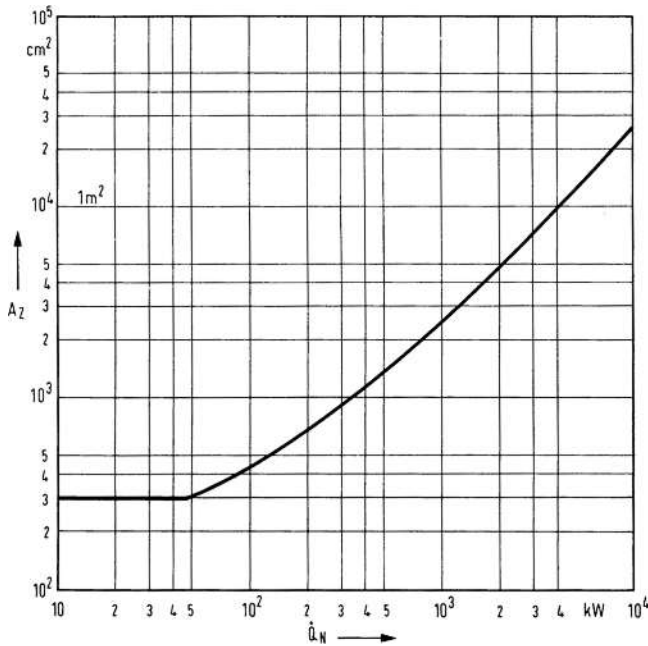


Fig. 9.2 Fresh air cross-section as function of total rated heating output

9.3 Stacks

9.3.1 Emission and Immission

Stacks must pass the flue gases to the atmosphere without risk.

Emissions are substances contaminating the air passing into the atmosphere after leaving the stack. The substances are stated in mg/m^3 (referred to standard conditions). The "TA Luft" contains indications on limiting the emissions and calculated examples for taking into account these limit values.

Immissions are substances contaminating the air which may occur near the effect stage and influence humans or vegetation. Maximum values are fixed for the immission relating to dusts, gases and vapors. The immission of sulfur dioxide has to be considered in particular. Sulfur dioxide base load values have been determined for various countries and regions. The additional immission produced by a new plant must not exceed the still permissible difference between immission limit value and base load.

The immission concentration as consequence of the emission of a stack primarily depends on climatic conditions at the stack location and stack height. These relations permit the determination of the minimum stack height based on data of existing and additional immission (specification VDI 2289).

10 Operation of Heat Transfer Plants

10.1 Acceptance Test

The system must be subjected to an acceptance test after completing the installation of the plant. It has to be observed that a heat transfer fluid is not yet filled in and the insulation not yet applied. The acceptance test is to ensure the agreement with the planning data (plant diagrams, installation plans, current flow diagrams) and thus to guarantee the working capability of the plant. The following points are important:

1. Check of the pipelines by means of the plant diagrams.
2. Verification of the laying of the pipelines with regard to fixed points, loose points and expansion possibilities.
3. Verification of the provision of vents at the highest points of the system and of drains at the lowest points.
4. Verification of the system that screwed connections with seals in the thread are not installed in the "hot" part of the plant (specially at the heat consumers).
5. Checking of the fuel supply system.
6. Check of the electrical lines that they are protected against effects of leaking heat transfer fluid.
7. Checking of vent lines from the expansion vessel or collection vessel that these lead into the open air.
8. Testing the installation position of fittings, dirt catchers and control valves (especially of the three-way control valves).
9. Checking that all measuring sockets are welded.
10. Check of pump cooling water and inert gas supply systems, if they exist.
11. Testing the design and execution of the heater room that this meets the heater room specification. Attention has to be paid in particular to the room ventilation and venting, the danger switch and the fire extinguishers.
12. Testing of the heat consumers.

10.2 Cleaning

The plant must, as far as possible, be cleaned of impurities such as welding beads, scaling and rust deposits.

Chemical cleaning with agents requiring subsequent neutralization and drying of the system is only done in large plants. It is usual, though, to purge the pipelines with compressed air and to inspect all vessels.

Dirt particles collecting in the dirt catchers and low-lying plant components and also impurities in the vessels must be carefully removed.

10.3 Leak Test

Heat transfer plants must be tested for tightness. The test fluid may be liquids or gases. When using water it must be taken into account that it can generally not be completely removed from the plant and longer "boiling out" times are required on commissioning. This is the reason that at present air is used almost exclusively as test fluid. The test pressure is then adjusted to about 1.5 to 2 bar.

All flanges and connection points and all weld seams must be brushed with soap solution and leakage points are indicated by the formation of soap bubbles. To increase the test pressure above 2 bar makes little sense because the danger exists that gas flows through the soap solution under the pressure and consequent high velocity without forming bubbles. A strength test cannot be obtained at an increased pressure because the main stresses occurring in operation (resulting from thermal stresses) cannot be represented. To prevent a permanent deformation of the pipes at room temperature a wall thickness of about 1 mm is sufficient for e.g. DN 200 and a test pressure of 11 bar, if the quality of the connection is assumed to be 0.5.

To restrict to a minimum the storage of compressible fluids for safety reasons the components already subjected to a pressure test to the pressure vessel regulations in the manufacturer's workshops, such as vessels, larger headers and distributors and also the heater and possibly the heat consumer should be excluded from the tightness test with air. This can be done with blind flanges on these components.

Circulating pumps with stuffing box should also be excluded because there is a risk that the test pressure cannot be maintained due large gas leakage at the stuffing box.

10.4 Filling of the Plant with Heat Transfer Fluid and Pressure Test

Care must be taken, in principle, that the plant may be filled only with the heat transfer fluid noted on the name plate. Filling can be done from barrels or tankers. It is most suitable to fill slowly from the lowest point of the plant so that the air can escape through the opened vent lines. The filling process must be continued until the minimum level is reached in the expansion vessel. If a cold fluid receiver is provided in the collection vessel, the corresponding heat transfer fluid volume must be filled into this vessel. A sample of 1 liter must be taken from the freshly filled heat transfer fluid and passed on for examination. The analysis data resulting from this examination present the data of the "fresh" heat transfer fluid shown in the operational control sheet of Fig. 2.55.

After filling the plant the components exposed to pressure must be subject to a pressure test with the heat transfer fluid.

The test overpressure at room temperature (20 °C) should be:

$$p_{\text{test}} = 1.43 \cdot p_{\text{PS}} \quad (10.1)$$

$$p_{\text{test}} = 1.25 \cdot p_{\text{PS}} \cdot \frac{K_{20}}{K_{\emptyset, \text{calc}}} \quad (10.2)$$

The higher test pressure should be used.

The value of 1.43 results from the ratio of the safety factors: $\frac{S}{S_{\text{test}}} = \frac{1.5}{1.05}$

with:

p_{PS} = maximum permissible working pressure PS
(see manufacturer's rating plate)

K_{20} = Material strength value at 20 °C

$K_{\theta, \text{calc}}$ = Strength value of the material at calculation temperature

This temperature derives from the maximum permissible temperature TS (see manufacturer's rating plate) as well as an uplift for the heating type

Comment

For components whereby the creep strength needs to be taken into account in the design, it must be ensured that 95 % of the cold yield point is not exceeded during the test pressure calculation.

Example

A fired heater has a pressure vessel designed for: PS = 10 bar and TS = 300 °C. The test pressure is to be determined for:

- a) Pressure vessel in P235GH and alternatively
- b) Pressure vessel in 1.4571

Solution

Because the pressure vessel has fired heating, if the temperature is not verified via calculation or measurement, the calculation temperature is:

$$\vartheta_{\text{calc}} = \text{TS} + 50 = 350 \text{ °C}$$

for a) the test pressure according to Equ. 10.1 is: $\rho_{\text{test}} = 1.43 \cdot 10 = 14.3 \text{ bar}$

for a) the test pressure according to Equ. 10.2 is with: $K_{20} = R_{p,0.2} = 235 \text{ N/mm}^2$
 $K_{\theta, \text{calc}} = R_{p,0.2} = 120 \text{ N/mm}^2$

$$\rho_{\text{test}} = 1.25 \cdot 10 \cdot \frac{235}{120} = 24.5 \text{ bar}$$

for a) the test pressure according to Equ. 10.1 is: $\rho_{\text{test}} = 1.43 \cdot 10 = 14.3 \text{ bar}$

for a) the test pressure according to Equ. 10.2 is with: $K_{20} = R_{p,1.0} = 225 \text{ N/mm}^2$

Material cold-worked

$$K_{\theta, \text{calc}} = R_{p,1.0} = 161 \text{ N/mm}^2$$

$$\rho_{\text{test}} = 1.25 \cdot 10 \cdot \frac{225}{161} = 17.5 \text{ bar}$$

10.5 Function Test

The function test comprises the controls to be carried out prior to starting the heating, e.g. the following work:

1. Setting the measuring, control, monitoring and limiting apparatus.
2. Setting the motor protective switches to the rated currents.
3. Lubricating the valve spindles with molybdenum disulfide, testing the stuffing boxes and smooth running.
4. Checking that all valves in the circuit system are opened.
5. Checking the lubrication of the circulating pumps bearings.
6. Checking the easy rotation of the pump shaft and coupling running.
7. Switching on the main switch in the electrical switch cubicle.
8. Starting the main circulating pumps and check of their direction of rotation.

9. Venting of the plant components and topping up the heat transfer fluid if necessary.
10. Shutting down the circulating pump after about half an hour and cleaning the dirt catchers. This operating cycle:
 - a) starting the pumps
 - b) venting (Fig. 10.1)
 - c) cleaning the dirt catchers
 must be repeated until the plant is completely vented and deposits are longer found in the dirt catchers. It is advantageous to install fine mesh filters for the startup operation to be replaced after the cleaning cycle by coarse mesh filters.
11. Check, if fuel supply is ready for operation.
12. The disturbance signalling system must be unlocked prior to starting the heating and a check must be made that all monitoring apparatus do not indicate an optical or acoustic signal.

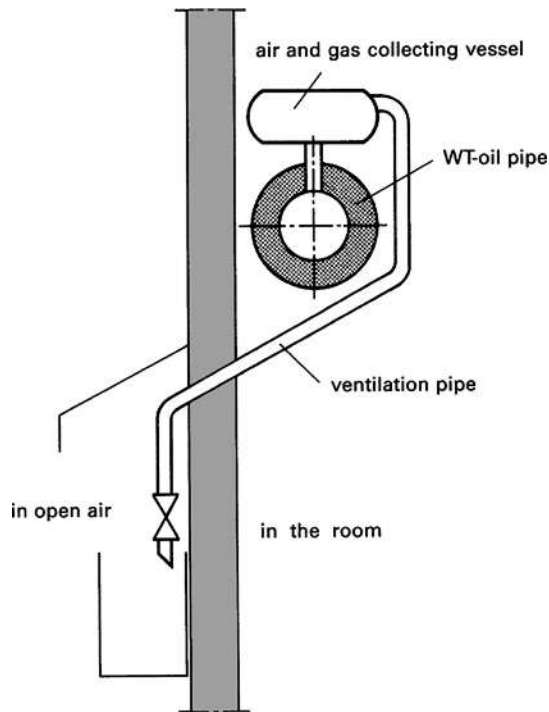


Fig. 10.1 Ventilation unit

10.6 Commissioning

The first "thermal" commissioning of the plant starts with switching on the heating. The flow temperature controller must be set to about 100 °C. As soon as this temperature is reached, the heating switches off and the plant must again be vented and checked for tightness. The expansion vessel must from now on be opened (also in systems with inert gas cover), if there is no direct connection to atmosphere. This is necessary because from about 100 °C water in the plant and heat transfer fluid starts evaporating. Due to the pressure superposition of the pump the evaporation temperature is shifted to usually a maximum of 200 °C. To relatively quickly remove gases

and vapors in the system a gas separator should be installed at the connection of the expansion vessel with the circuit system.

The danger exists during the ascension of the gases in the expansion line and into the expansion vessel that the vapors are condensed by cooling. It is therefore advantageous to operate the expansion vessel hot through a small bypass from the pressure side of the pump during the "boiling out" period.

The control contact for the flow temperature must be slowly increased in the range between 100 and 200 °C until the pressure fluctuations no longer occur in the system and it is ensured that gases and vapors are no longer present in the plant.

The expansion vessel in plants with inert gas cover must then be closed again. In plants with cold fluid receiver the expansion vessel must also be closed but after the plant has cooled a single pressure equalization has to be made because otherwise a vacuum would exist in the expansion vessel in the operating condition at $\vartheta_v \leq 200$ °C. The oxygen of the air of the volume between collection vessel with cold fluid receiver and expansion vessel must not, even once, react with the heat transfer fluid so that it is advantageous to replace this intermediary air by inert gas.

An intermediate check of the system for tightness, pipe expansion, pump cooling water temperature etc. must now be made. The plant can subsequently be slowly run up to the operating temperature.

When the operating temperature has been reached and the plant once more checked for tightness and proper conditions (tighten flange connections), the safety apparatus must be checked for proper functioning by changes of the switching point and be set to the limit design data of the plant. The essential checks and settings are:

1. Safety temperature limiter (TSB)
2. Flue gas temperature limiter
3. Flow limiter
4. Level limiter in expansion vessel
5. Pressure limiter (in overpressure plants)
6. Release thermostat in heavy fuel oil firing
7. Cooling water monitor at cooled pumps
8. Leakage control in heater if provided
9. Testing the safety valves if provided
10. Testing the level in the expansion vessel

A record of the measurements on the plant must be made.

10.7 Record of Measurements

The most important data at rated load operation must be contained in the record of measurements. These are generally the flow and return temperatures and the pressures at the pump and heater. The number of operating hours and the flow rate of the heat transfer fluid must also be recorded. For electrically heated heaters the power consumption data and for fuel gas heated heaters the gas temperature data must be recorded.

Flame heated heaters require a considerably more extensive measuring technique. It is essential to record the following:

1. Type of fuel
2. Calorific value

3. Fuel flow rate
4. Flame geometry
5. Flue gas data (cf. section 6.1.5) such as:
 - a) Flue gas temperature
 - b) CO₂ content
 - c) Co content for gas burners
 - d) Soot number for oil burners
 - e) NO_x content if required
 - f) Flue gas pressures
6. Determination of the combustion efficiency for determining the rated output

During the measurements the load conditions must be constant.

10.8 Shutting-Down

Switching off the heating shuts down the plant thermally. The pump must be switched off only, if an overheating of the heat transfer fluid on standstill of the flow in the plant (especially by heat storing masses in the heater) is excluded with certainty. Attention must be paid to the instruction of the manufacturer of water-cooled pumps. It may be necessary for the pump to continue running until it is cooled to a certain lower temperature.

It is recommended to close down the fuel supply and to switch off the main switch, if the plant is closed down for a longer standstill.

10.9 Maintenance

Relevant regular maintenance services of the system components are necessary and to be carried out by experienced personnel (cf. specification VDI 3033, Table 1) to ensure trouble-free operation of the plant over a longer period.

10.9.1 Heat Transfer Fluid

The heat transfer fluid must be checked for further use at least once yearly. After the first commissioning and after plant modifications this period should be shortened once to about 3 months to recognize early changes in the heat transfer fluid (Fig. 2.55). An important condition for the assessment is sampling uniformly to obtain a representative sample at about the same point and under the same conditions. The sampling point recommended is the flow side near the heater (Fig. 10.2). The design of the sample container should enable safe sampling and prevent influencing the analysis data (e.g. flash point and boiling behavior). This chance occurs easily, if sampling without cooling is done at a "hot" plant in an open vessel, and the low boiling components can escape. The samples should mainly be taken by means of closed "cooling traps" or from a cold plant (Fig. 10.3).

To avoid aging of the heat transfer fluid the exclusion of air in directly open plants the temperature in the expansion vessel should be checked. Heat transfer fluid leaked from the plant should not be filled in again. If the leakage volume is larger, this can be done, if an unobjectionable condition has been proved by an analysis of the heat transfer fluid.

11 Statutes, Ordinances, Regulations, Standards and Specifications of the EU and DIN

Among statutes, regulations and standards, it is statutes, or laws, which play the most important role. In every orderly community the body of laws is the only possible measure against which naturally opposing individual interests can be objectively measured and put into priority order. The authoritative law concerning heat transfer plants is the Pressure Equipment Directive (PED) (2014/68/EU). However, in addition, there is also environmental, water conservation, and building and construction law. Tables 11.1, 11.2 represent an overview and Table 11.3 an overview in particular for the pressure equipment.

Table 11.1 Overview of safety objectives in the legal areas which apply to heat transfer plants

| Legal area | Safety objective |
|------------------------------|---|
| Equipment safety statute | Protection of employees and third persons against fire, explosion, poisoning, or the effects of toxic agents |
| Water conservation law | Protection against contamination within the meaning of the "concern principle", or the principle of the greatest possible safety |
| Building construction law | Physical stability, sturdiness of construction, loading, preparation of building materials in view of physical and chemical operating conditions, fire resistance |
| Environmental protection law | Protection against contamination of water and air, against improper use or destruction of nature and landscape; protection against noise and shock |

Table 11.2 Overview of German statutes and ordinances

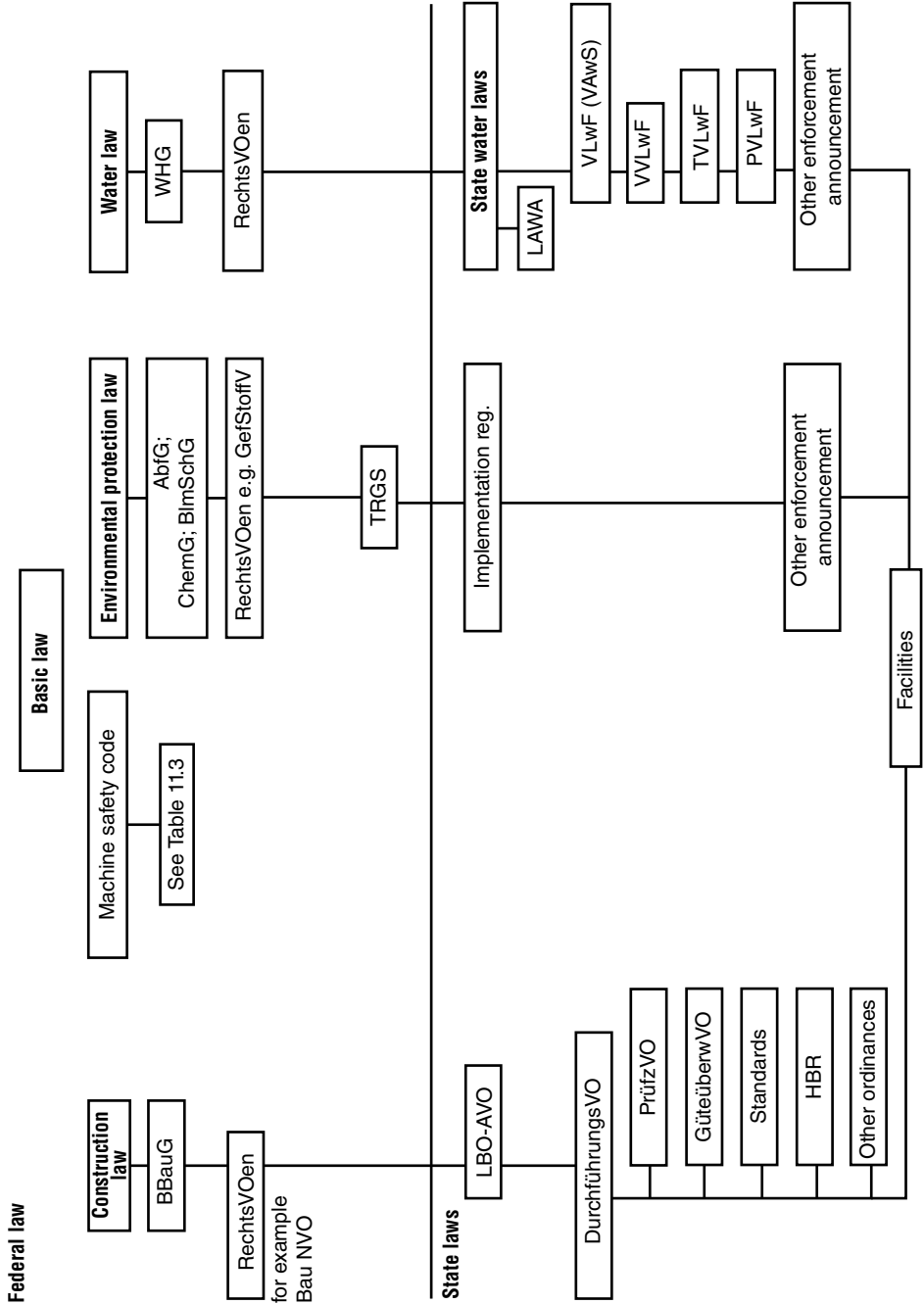


Table 11.3 Overview of statutes and ordinances

| GPSTG | Equipment and Product Safety Code |
|---|---|
| Section 1 Article 1 Scope of application | (1) Applies to the sales and distribution of products ... (2) Also applies to the installation and operation of plants requiring approval ... |
| Article 2 Terms | (7) Plants requiring approval... (16) Harmonized standard is a non-binding technical specification |
| Section 5 Article 14 Empowerment for the issuance of legal directives | (1),4 ... that such installations are subject to testing prior to commissioning, recurring testing ... |
| Article 16 ... approved inspection body | ... and provide information and submit documents as may be required for their activities. |
| Article 17 ... Testing and monitoring (7) | The accreditation of approved inspection bodies is the task of the authority identified as responsible according to federal law. |
| GPSTG | Pressure Equipment Directive |
| Article 1 Scope of application | (1) Applies to the sales and distribution of new pressure vessels ... |
| DGRL | Directive Regarding Pressure Vessels, Pressure Equipment Directive (PED) 2014/68/EU |
| Basis (4) | ... applies to equipment with a maximum permissible (over-) pressure (PS) of more than 0.5 bar. |
| Basis (5) | This directive also applies to assemblies... This directive, in contrast, does not apply to the assembly of pressure equipment carried out on the user's site.... |
| Basis (16) | European-level harmonized standards are helpful to make it easier to provide evidence of compliance with the fundamental requirements, because if they are complied with, it can be assumed that a product meets the basic requirements. The European standards are developed by private organizations and must remain voluntary (discretionary). |
| Article 1 Applicability | This directive applies to the design, manufacture and conformity assessment of pressure equipment and assemblies ... (2.1.1) "Vessel" a sealed component ... (2.1.2) "Pipework" specific piping sections for the routing of fluids ... Heat exchangers made of pipes for cooling or heating air are given the same status as pipework (2.1.5) "Assemblies" multiple items of pressure equipment which have been connected by a manufacturer into an integrated functional unit. |
| (3) Not included within the application area: | (3.21) Vessels for liquids with a gas pressure above the liquid of maximum 0.5 bar. |
| Article 3 Technical requirements | Pressure equipment must comply with the requirements set out in Appendix I. |
| Article 4 Free distribution of goods | (1) 1.1 ... Member states are not permitted to ban the distribution and the commissioning of pressure equipment or assemblies ... due to pressure-related risks, ... |
| Appendix I Fundamental safety requirements | (3.2.2) Pressure test The acceptance testing of the pressure equipment must incorporate a structural pressure test, which normally takes the form of a hydrostat pressure test, ... |

Table 11.3 Overview of statutes and ordinances – Continued

| | |
|--|---|
| Appendix II Conformity assessment diagrams | Categories |
| Appendix III Conformity assessment process | Modules |
| Appendix VII Declaration of conformity | |
| BetrSichV | Operational Safety Directive |
| Section 3 | Special regulations for plants requiring monitoring |
| Article 14 Tests prior to commissioning ... | ... carried out by an approved inspection body ... covering the assembly, installation, installation conditions and safe function. For exceptions see (3) |
| Article 15 Recurring tests (10) | For internal and external testing, inspections may be replaced by other processes of equivalent effectiveness, and for the structural testing, pressure tests may be replaced by an equivalent non-destructive process. |
| Article 17 Testing of special pressure equipment, Appendix 5 | 18. Pressure equipment in heat transfer plants (1) Heaters and heat consumers (2) Check plant for leaks prior to commissioning (3) Annual heat transfer fluid testing |
| TRBS | Technical Rules for Operational Safety (TRBS) |
| | Testing of working equipment and plants requiring monitoring |

Technical Rules

In contrast to statutes and regulations, which represent binding law, the “Technical Rules” are only recommendations. No one can be forced to apply these rules. Anyone may deviate from their stipulations, but he must then prove that his solution is at least of equal value by comparison with the case covered by the rules. Whoever follows the “Technical Rules”, however, fulfills, by general agreement, the engineer’s responsibility for diligence, which now also requires that the current “state of the art” of the technology be taken into account.

1. The DIN 4754 Standard

DIN 4754-1: Heat transfer installations working with organic heat transfer fluids –

Part 1: Safety requirements, test

DIN 4754-2: Part 2: Draught diverter

DIN 4754-3: Part 3: Level controller

This standard (2015 edition) applies to plants in which liquids are heated to temperatures below as well as above the boiling point at atmospheric pressure. The technical safety requirements for such plants are defined, for the most part, in this standard.

The materials used, and the design and manufacture of plant components are covered to a great extent by the pressure vessel data sheets, which were already mentioned under the ordinance on pressure vessels and are officially recognized as “Technical Rules”. The remaining sections of this standard have already been dealt with in previous chapters, and detailed treatment here would be superfluous. Several key points, however, should be mentioned:

1. The plants must be fitted with a heater that can be controlled and shut down quickly.
2. The liquid used as a heat transfer fluid must be serviceable for at least 1 year.

3. The heaters must be designed in such a way that at no point the walls and the heat transfer fluid will be heated above their permissible temperatures.
4. The manufacturer of the heater must be able to furnish evidence, at least by calculation, of the level of the film temperature.

In addition to the Technical Rules (pressure vessel data sheets) mentioned in the Ordinance Concerning Pressure Vessels, there is also VDI Instruction 3033 included in the DIN 4754 standard.

2. The DIN 51 522, DIN 51 528 and DIN 51 529 Standards

DIN 51 522: Heat transfer fluids Q

Specifications, test

DIN 51 528: Testing of mineral oils and related products –

Determination of thermostability of unused heat transfer fluids

DIN 51 529: Testing of mineral oils and related products –

Testing an evaluation of used heat transfer fluids

12 Biomass CHP Plants with Organic Fluids (ORC Installations)

These plants for producing electrical power and heat must be designed to the extent possible for fault-free operating periods of approx. 8000 h/a (up to the next maintenance/repair period of approx. two weeks). The installations consist of four main sections:

1. Energy from wood combustion
2. Intermediate circuit heat transfer fluid system with the fluid: Partially hydrated terphenyl (PHT)
3. ORC turbine circuit for generating power using the fluid: Silicone oil (octamethyltrisiloxane)
4. Hot water circuit (e.g. for district heating)

The principle of generating power using the ORC process corresponds to the conventional Rankine process. The key difference is that instead of water, an organic working fluid with specially assigned thermodynamic properties is used. Fig. 12.1 shows the different components of the ORC process as well as the integration in the overall plant.

The ORC installation is connected with the thermal oil heater via a thermal oil circuit, the ORC process itself is designed as a fully-sealed system and uses silicone oil as an organic working fluid. The pressurized silicone oil is evaporated by the thermal oil in the evaporator as well as slightly superheated and then expanded in a turbine that is directly coupled to an asynchronous generator. Before the expanded silicone oil reaches the condenser, it is routed to a regenerator (for internal heat recovery). Before the expanded silicone oil reaches the condenser, it is routed to a regenerator (for internal heat recovery).

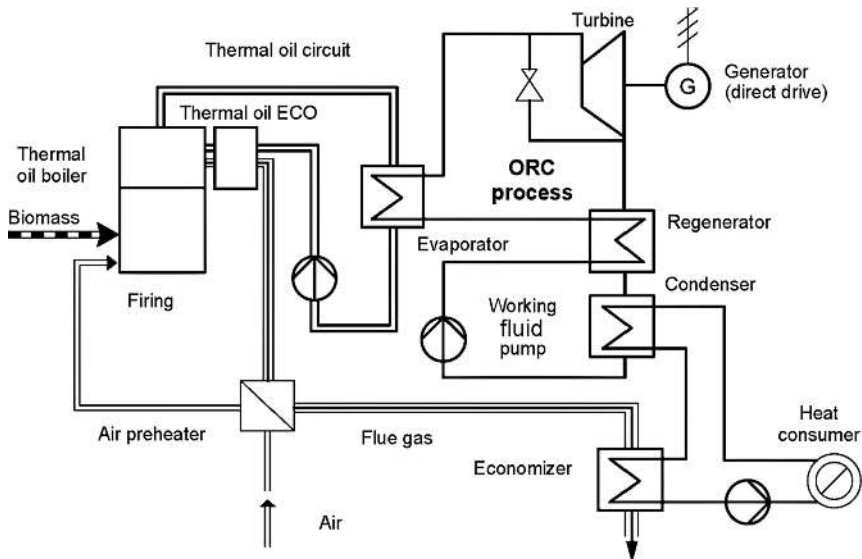


Fig. 12.1 Schematic diagram of the integration of the ORC process within the complete installation

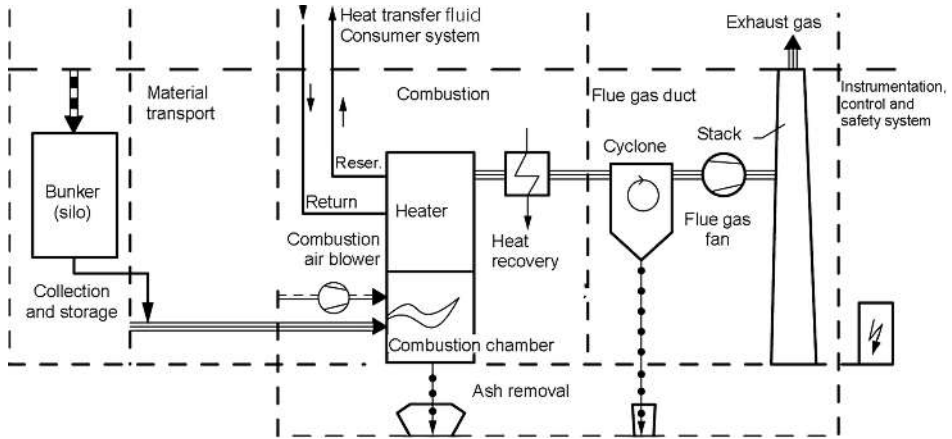


Fig. 12.2 The key assemblies and elements of a wood-fired installation

Working fluid condensation takes place at a temperature level that allows the extracted heat to be used as district heat or process heat (hot water feed temperature between 80...100 °C). Via a pump, the condensed silicone fluid is finally elevated back to the pressure level of the circuit working fluid to be evaporated, and after flowing through the regenerator reaches the evaporator once more via a preheater.

In order to achieve the highest possible electrical efficiency of the ORC installation (results from: electrical net power/added thermal power with the thermal oil), it is necessary to keep the backpressure of the turbine, and therefore the required district heating or process heating feed temperature, as low as possible. This allows optimized control of the plant hydraulics as well as of the district heating network. The ORC installation should always be connected directly to the return of the district heating or process heating network, and the feed exiting the ORC condenser should have a water temperature that is kept as low as possible. Reheating the feed is then possible using the flue gas thermal oil economizer described in Fig. 12.1 or other coupled heating systems (for example a second biomass hot water heater or a peak load boiler).

The net electrical efficiency of the ORC process is approx. 18 %, and the electrical efficiency of the overall installation (results from: electrical net power/added fuel thermal power $\{H_{U}\}$) is roughly 15 %.

General Notes Regarding the ORC Circuit

- Fluids:** More than 10 organic fluids have already been used depending on the application. Turbine inlet temperature up to 300 °C, latest developments up to 400 °C.
- Turbine:** 2 to 4 stages and up to 20 MW. The turbine efficiency can be more than 90 %.
- ORC circuit:** The ORC circuit is as per the RC circuit (water-steam circuit), but with an organic fluid (oil) being evaporated. Due to the higher molecular mass, this leads to lower vapor velocities and turbine speeds. This means that no erosion occurs on the metallic surfaces and blades.

12.1 Energy from Wood Combustion

12.1.1 Key Assemblies in a Wood-Fired Plant

As per Fig. 12.2, a wood-fired installation consists primarily of the following assemblies:

1. Material collection and storage
2. Transport of materials to the wood-fired installation
3. Wood-fired installation including the flue gas paths (heater and if required heat recovery from the flue gases)
4. Flue gas extraction and flue gas dust separation
5. Ash removal system
6. Instrumentation, control and safety system
7. Consumer system

12.1.2 Material Collection and Storage

Operational circumstances sometimes lead to very different forms of material collection and storage. The key difference is due to the types of wood (fresh (green) or dry wood).

Collection of Fresh Wood

The stored tree trunks (Fig. 12.3) are pre-treated for cleaning them, for example following moisture treatment (warm water bath or steam blasting), and debarked for mechanical processing. The bark content can be up to 10 % of the total weight of the tree trunks. The debarked tree trunk is then shredded, producing waste in the form of trimming (trunk waste) and chips and shavings. The moisture content of the waste is around approx. 40...60 % by weight (67...150 atmo %), and already has a heating value of 5000...10,000 kJ/kg. The waste is usually stored in bunkers (silos), to achieve independence between arriving materials and consumed materials, with the bunker being sized (storage capacity) according to the ratio of incoming and used materials.

Collection of Dry Wood

Dry wood for combustion purposes is waste that arises following the drying of sawed timber and/or chip drying. If the sawed timber drying takes place in air, the moisture content is 12...25 % by weight (14...33 atmo %), which leads to a heating value of 13,000...16,000 kJ/kg. With machine-dried wood, the remaining moisture content is only 5...10 % by weight (5.3...11 atmo %) and the heating value is increased to 16,300...17,300 kJ/kg.

Waste from refined products, such as grinding dust as well as production waste from chipboard and furniture manufacturing, presents particularly difficult challenges for combustion, because additives are contained within this waste (glue, binding agents, corundum, plastic particles etc.). This results in different combustion behavior (caking) and ash-formation behavior (reduction of the ash softening point to 800 °C). The organic constituents (that have a heating value of approx. 40,000 kJ/kg) increase the heating value of the "wood" to approx. 18,000...21,000 kJ/kg (depending on proportion).

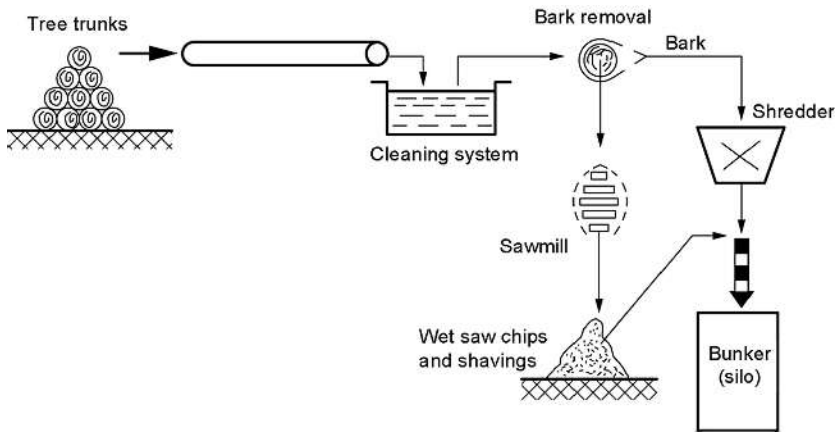


Fig. 12.3 Example of the treatment of “fresh” wood

Storage

The primary method of storing wood waste is flat-bottomed silos with a mechanical discharge system. An entire range of silo discharge systems have become well-proven, depending on the waste material and mix. Flat, moist wood waste in particular is one of the most difficult materials, providing considerable resistance to being removed from the bunker. The impairment to extraction results from overlaid particles that compress under their own weight and, depending on the structure, can “hook together”. This means that solid regions and sections can form within columns of the material, blocking free flow and impeding controlled discharge if no suitable countermeasures are taken.

As the fill height increases, the vertical and horizontal pressures within a material column increase, with the specific wall friction angle being the critical factor for the pressure ratios, and being related to the internal frictional angle. Once the horizontal pressure reaches a critical value, dynamic arches can form within a material column, which can become static arches once the flow has stopped. Material underneath can be removed, without the material above falling down. The risk of blockage is reduced if the upward-pointing structures that can cause blockages are converted into downward-pointing structures. This can be achieved, for example, by reducing the flow velocity at the bunker wall through an increased coefficient of friction.

The key difference between the silo designs lies in the type of discharge equipment that routes the material away from the extraction point. The material removal is usually done using an auger screw. The discharge systems are mainly divided into rotating or oscillating (moving back and forth) designs.

Pressure Relief Openings

When storing dust and for wood waste with dust particles (almost always present when storing dry material), protective measures must be put in place to prevent or control wood dust explosions. An explosion can occur only if fuel, oxygen and an ignition source are present. Fuels are only explosive within specific explosion limits. With wood dust, however, the distribution is usually uneven and can form an explosive mixture even due to local disturbances. Therefore, compliance with the

limits is not guaranteed and only of any importance if the lower explosive limit is exceeded by a substantial margin.

One of the safest explosion protection measures is inertizing, whereby the total oxygen concentration for wood dusts needs to remain limited at a maximum of 10 % by volume. This is very difficult to achieve when storing wood waste, however. The key protection objective must always be: Switching off sources of ignition (smoking, naked flames, fire, welding work, grinding work etc.)! To prevent static electricity, all conductive plant components should be grounded. Mechanical moving plant parts must not tend to run hot or lead to arcing and friction sparking.

Since it will not be possible to comply with preventative explosion protection measures with certainty, bunkers should have explosion pressure relief openings fitted. When defined openings are exposed, the excess pressure on the bunker resulting during the explosion is limited by the relief system. The bunker with relief system needs to be designed only strong enough to resist the reduced explosion pressure that occurs with the relief function.

Achieving a match between the static trigger pressure p_{stat} of the relief system and the reduced explosion pressure p_{red} can be achieved only with extremely large relief openings, which are almost impossible to implement in practice. For this reason, at least $p_{\text{stat}} = 1.1$ bar (absolute) and $p_{\text{red}} = 1.2$ bar are assumed.

For the dust characteristic value, the maximum pressure rise in the 1 m³ tank is selected (K_{St} value in bar · m/s), where for wood dust of dust explosion class St I max. $K_{\text{St}} = 200$ (bar · m)/s can be assumed and dust grain sizing of < 63 μm is present in general. The size of the pressure relief surface required is described in VDI guideline VDI-3673, which can be divided into multiple, individual surfaces and, where possible, should be located on the top. Combustible materials should not be located near the blow-out openings.

Material Transport

Material transport can be done mechanically, pneumatically or, for small plants, by manual feed.

Mechanical Transport

Constant conveyors that work continuously are mainly used. The key characteristic for performance is the material flow \dot{V} in m³/h or mass flow \dot{m} in kg/h or t/h.

$$\dot{m} = \dot{V} \cdot \rho \quad (12.1)$$

The bulk density ρ in kg/m³ can be taken from Fig. 12.4 in relation to the material as well as the water content. Limit values can be considered to be shavings with $\rho \approx 80$ kg/m³ as well as wet birch bark with $\rho \approx 800$ kg/m³. With mixed material with average moisture, a guideline value for planning is $\rho \approx 300$ kg/m³.

The material flow \dot{V} is calculated from the material cross-section A and the material velocity w .

$$\dot{V} = A \cdot w \cdot 3600 \text{ (m}^3\text{/h)} \quad (12.2)$$

with:

$$w = \text{m/s}$$

$$A = \text{m}^2$$

For screw conveyors, this yields the values according to Table 12.1.

13 Equations, Diagrams and Tables for Project Planning for Heat Transfer Fluid Installations in the Liquid Phase

For project planning and optimization of heat transfer fluid installations, it is useful to use simple equations that allow complex relationships to be represented in a simple way and allow the determining factors to be identified.

Where possible, rounded figures are used to make this clearer still. In special projects, however, the numbers can be adapted specifically to the heat transfer fluid and the design parameters.

In individual cases, it may be necessary to deviate from the usual, proven practical values, but it is then important to be aware of the "special design". The equations also serve as cross-checking calculations to determine whether the order of magnitude is correct.

The fundamental design of a heat transfer fluid installation is shown in Fig. 13.1.

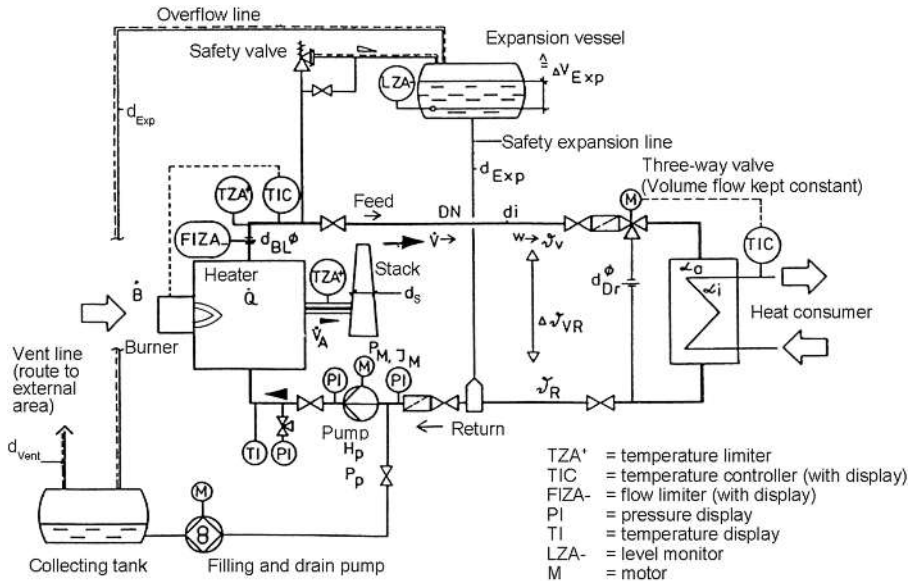


Fig. 13.1 Typical plant schematic

13.1 Volume Flow \dot{V}

To transmit a heat flow \dot{Q} the following volume flow \dot{V} is required:

$$\dot{V} = \frac{\dot{Q}}{\rho \cdot c \cdot \Delta\theta_{VR}}$$

In the upper temperature application range for organic heat transfer fluids, on average it can be assumed that:

Average density: $\bar{\rho}_{VR} \approx 700 \text{ kg/m}^3$

Average spec. heat capacity: $\bar{c}_{VR} \approx 2.7 \text{ kJ/(kg} \cdot \text{K)}$

The specific volumetric heat capacity therefore becomes:

$$(\bar{\rho} \cdot \bar{c})_{VR} = 1900 \text{ kJ/(m}^3 \cdot \text{K)}$$

with this:

$$\dot{V}_{(\text{m}^3/\text{h})} = \frac{\dot{Q}_{(\text{kW})} \cdot 3600}{1900 \cdot \Delta\theta_{VR}}$$

rearranged and rounded:

$$\dot{V}_{(\text{m}^3/\text{h})} \approx \frac{\dot{Q}_{(\text{kW})} \cdot 2}{\Delta\theta_{VR}}$$

Finally as a project planning equation:

Standard volume flow

$$\dot{V}_{(\text{m}^3/\text{h})} = \frac{\dot{Q}_{(\text{kW})}}{\Delta\theta_{VR}/2} \quad (13.1)$$

with: $\Delta\theta_{VR}$ = temperature difference between feed and return in °C or K

See Table 13.1

Table 13.1 Volume flow in relation to the thermal power and the temperature difference between feed and return

| $\dot{Q}_{(\text{kW})} \rightarrow$ | 100 | 500 | 1000 | 2000 | 4000 |
|---|---|-----|------|------|------|
| $\Delta\theta_{VR}(\text{°C}) \downarrow$ | $\dot{V}(\text{m}^3/\text{h}) \downarrow$ | | | | |
| 20 | 10 | 50 | 100 | 200 | 400 |
| 30 | 6.7 | 33 | 67 | 133 | 267 |
| 40 | 5 | 25 | 50 | 100 | 200 |
| 50 | 4 | 20 | 40 | 80 | 160 |

Limit Fluids*Silicone oil:*

$$\bar{\rho} = 600 \text{ kg/m}^3$$

$$\bar{c} = 2.0 \text{ kJ}/(\text{m}^3 \cdot \text{K})$$

$$\dot{V}_{\text{m}^3/\text{h}} = \frac{\dot{Q} \cdot 3600}{1200 \cdot \Delta\theta_{\text{VR}}}$$

$$\dot{V}_{\text{m}^3/\text{h}} = \frac{\dot{Q}_{(\text{kW})} \cdot 3}{\Delta\theta_{\text{VR}}}$$

$$\dot{V}_{(\text{m}^3/\text{h})} \approx \frac{\dot{Q}_{(\text{kW})}}{\Delta\theta_{\text{VR}}/3} \quad (1.5 \times \text{standard volume flow}) \quad (13.2)$$

Hot Water:

$$\bar{\rho} = 900 \text{ kg/m}^3$$

$$\bar{c} = 4.2 \text{ kJ}/(\text{kg} \cdot \text{K})$$

$$\dot{V}_{\text{m}^3/\text{h}} = \frac{\dot{Q} \cdot 3600}{3780 \cdot \Delta\theta_{\text{VR}}}$$

$$\dot{V}_{(\text{m}^3/\text{h})} \approx \frac{\dot{Q}_{(\text{kW})}}{\Delta\theta_{\text{VR}}} \quad (0.5 \times \text{standard volume flow}) \quad (13.3)$$

13.2 Pipework Inner Diameter d_i

From the volume flow and the flow velocity, this results in a pipe diameter of:

$$A = \frac{\dot{V}}{w}$$

$$\frac{d_i^2 \cdot \pi}{4} = \frac{\dot{V}}{w}$$

$$d_i = \sqrt{\frac{\dot{V} \cdot 4}{\pi \cdot w}} \quad (13.4)$$

$$d_{i(\text{mm})} = 18.8 \cdot \sqrt{\frac{\dot{V}_{(\text{m}^3/\text{h})}}{w_{(\text{m/s})}}}$$

The speed, which is the value still missing, can be assumed with the upper limit as per [13.1]:

$$w_{(\text{m/s})} = 0.3 \cdot \sqrt{d_{i(\text{mm})}} \leq 5 \text{ m/s} \quad (13.5)$$

This equation also applies for silicone oil and water. If this value is put in the equation above, this yields:

$$d_{i(\text{mm})} \approx 18.8 \cdot \sqrt{\frac{\dot{V}_{(\text{m}^3/\text{h})}}{0.3 \sqrt{d_{i(\text{mm})}}}}$$

$$d_{i(\text{mm})} \approx \frac{34 \cdot \dot{V}^{0.5}}{d_{i(\text{mm})}^{0.25}}$$

$$d_{i(\text{mm})}^{1.25} = 34 \cdot \dot{V}^{0.5}$$

$$d_{i(\text{mm})} \approx 17 \cdot \dot{V}_{(\text{m}^3/\text{h})}^{0.4}$$

This equation can also be represented as an approximation by the simplified equation:

$$d_{i(\text{mm})} \approx 10 \cdot \sqrt{\dot{V}_{(\text{m}^3/\text{h})}} \quad (13.6)$$

Or instead, the volume flow resulting from this:

$$\dot{V}_{(\text{m}^3/\text{h})} \approx \left(\frac{d_{i(\text{mm})}}{10}\right)^2 \quad (13.7)$$

valid for: $\dot{V} \leq 1000 \text{ m}^3/\text{h}$

Table 13.2 Upper limit and economic speeds in relation to the pipe inner diameter

| | | | | | | | |
|-------------------------|------|------|-----|-----|-----|-----|-----|
| $d_{i(\text{mm})}$ | 15 | 25 | 50 | 100 | 150 | 200 | 300 |
| $w_{(\text{m/s})}^*$ | 1.16 | 1.5 | 2.1 | 3.0 | 3.7 | 4.2 | 5 |
| $w_{(\text{m/s})}^{**}$ | 1.0 | 1.25 | 1.8 | 2.5 | 3.0 | 3.5 | 4.3 |

* Upper limit (Pre-exponential factor in Eq. 13.5: 0.3)

** Economic (Pre-exponential factor in Eq. 13.5: 0.25)

Table 13.3 Upper limit for the volume flow as a function of the nominal diameter

| | | | | | | | | |
|---|----|----|----|----|-----|-----|-----|-----|
| $\dot{V}_{(\text{m}^3/\text{h})} \rightarrow$ | 15 | 20 | 40 | 60 | 100 | 150 | 220 | 400 |
| $d_{i(\text{mm})} \rightarrow$ selected: | 35 | 45 | 63 | 77 | 100 | 122 | 149 | 200 |
| DN \rightarrow | 40 | 50 | 65 | 80 | 100 | 125 | 150 | 200 |

The diameter d_i must be rounded up to the next nominal diameter DN. The constant factor should be increased to 15 with volume flows of $\dot{V} \leq 10 \text{ m}^3/\text{h}$.

Caution: It is common to go up one nominal diameter level (due to NPSH) for the pipework from the connection point of the expansion line to the suction nozzle of the pump.

13.3 Contents /

13.3.1 Pipe Contents I_P

$$I_P = \frac{d^2 \cdot \pi}{4} \cdot L_P$$

with the economical pipe diameter $d_{(\text{mm})} \approx 10 \cdot \sqrt{\dot{V}_{(\text{m}^3/\text{h})}}$, become, for the range $\dot{V} = 10$ to $1000 \text{ m}^3/\text{h}$:

$$I_{P(\text{Liters})} = \frac{(10 \cdot \sqrt{\dot{V}_{(\text{m}^3/\text{h})}})^2 \cdot \pi}{10^6 \cdot 4} \cdot L_{(\text{m})} \cdot 1000 \frac{\text{Liters}}{\text{m}^3}$$

$$I_{P(\text{Liters})} = \frac{100 \cdot \dot{V}_{(\text{m}^3/\text{h})} \cdot \pi}{1000 \cdot 4} \cdot L_{(\text{m})}$$

$$I_{P(\text{Liters})} = 0.08 \cdot \dot{V}_{(\text{m}^3/\text{h})} \cdot L_{(\text{m})}$$

As a function of $L = 1 \text{ m}$ pipe length:

$$I_{P,1\text{m}(\text{Liters})} \approx \frac{\dot{V}_{(\text{m}^3/\text{h})}}{10} \quad (13.8)$$

See Table 13.4

Table 13.4 Pipe contents in relation to volume flow

| | | | | | | | | | | | |
|---|-----|-----|----|-----|-----|-----|-----|-----|-----|-----|-----|
| $\dot{V}_{(\text{m}^3/\text{h})} \rightarrow$ | 4 | 6 | 10 | 16 | 25 | 42 | 64 | 100 | 160 | 225 | 400 |
| $I_{P(\text{Liters/m})}$ | 0.4 | 0.6 | 1 | 1.6 | 2.5 | 4.2 | 6.4 | 10 | 16 | 23 | 40 |

Pipe fittings such as valves, compensators and pumps can be taken into account in a way similar to pipes, but using the equivalent pipe length or with added factors. Or with the thermal power:

$$I_{P,1\text{m}(\text{Liters})} \approx \frac{\dot{Q}_{(\text{kW})}}{10 \cdot \Delta\theta_{\text{VR}}/2}$$

13.3.2 Equipment - Contents I_A [13.2]

$$\dot{Q} = k \cdot A \cdot \Delta\theta_m$$

and therefore the heat transfer surface area:

$$A_a = \frac{\dot{Q}}{k_a \cdot \Delta\theta_m}$$

14 Overview Diagrams of Assemblies: Form Sheets with Sample Calculations

14.1 Form Sheets

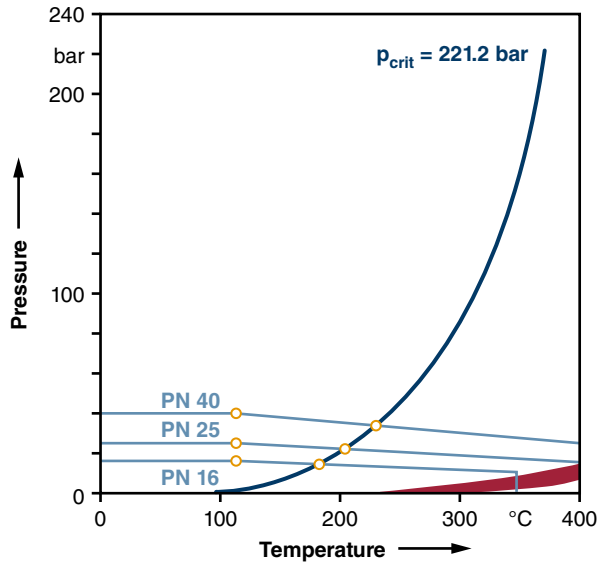


Fig. 14.1 Steam pressure diagram of water and heat transfer oils

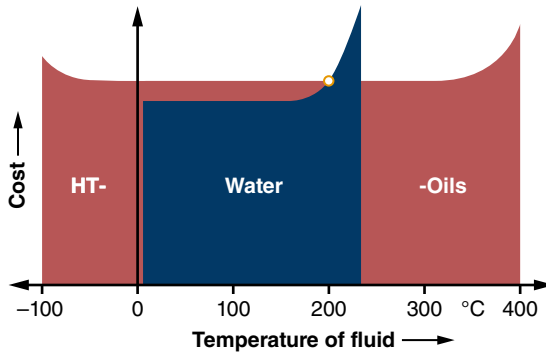
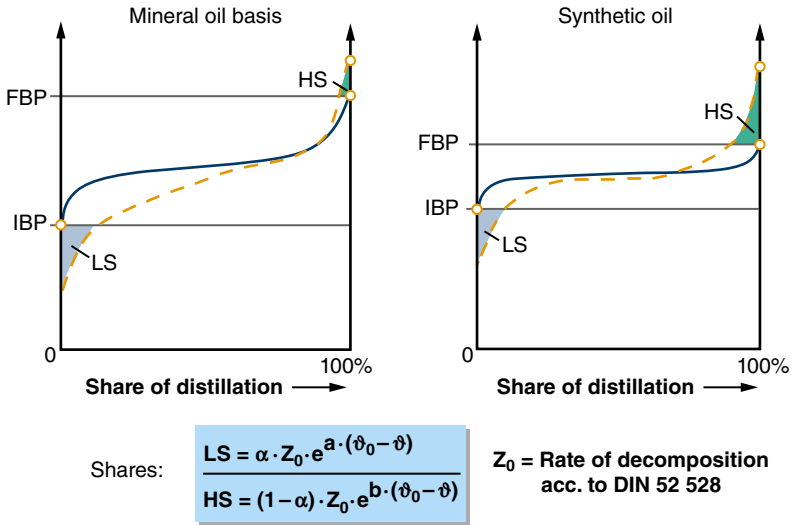


Fig. 14.2 Fields of application water and oil systems



With mineral oil-basic oils can approximate value $\alpha = 1$ be set, if the permissible film temperature is not exceeded.

Fig. 14.3 Development of low-(LS) and high-boiling point compounds (HS)

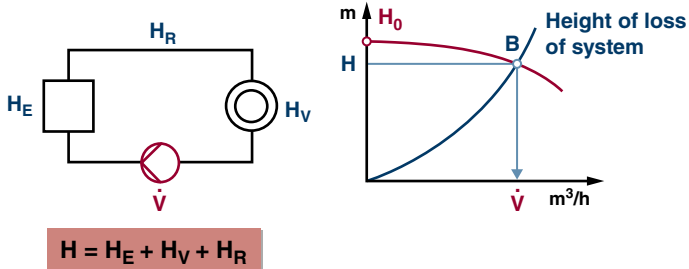


Fig. 14.4 Volume flow

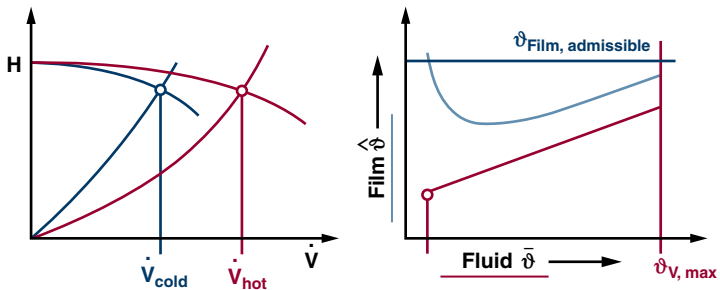


Fig. 14.5 Viscosity influence at start-up

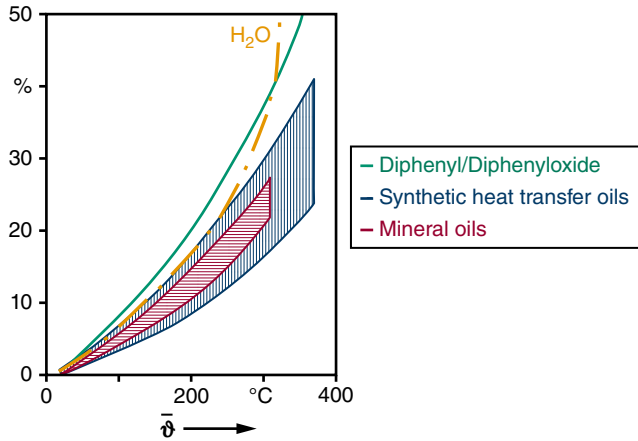
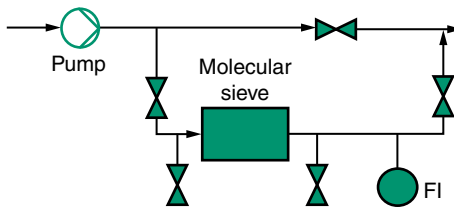


Fig. 14.6 Volume expansion

- Residual water removal for pump protection
- Protection against blocked filters, erosion protection
- Increase in long-term reliability



Conditions:

- 3 to 4 Angstrom
- Only during start up
- Min. 10 % flow
- At room temperature
- Karl Fischer titration (ASTM D 6304-00)

Fig. 14.7 Molecular sieve (low-temperature application: $T < 0\text{ }^\circ\text{C}$)

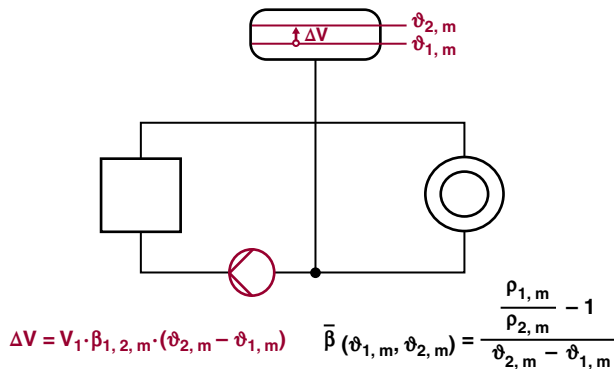


Fig. 14.8 Fluid systems/volume expansion

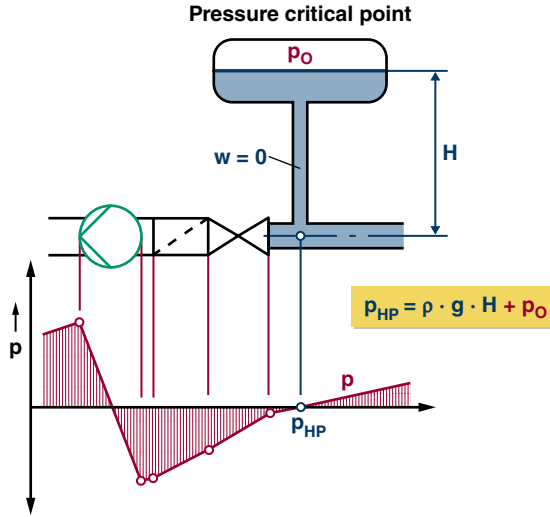


Fig. 14.9 Pressure critical point

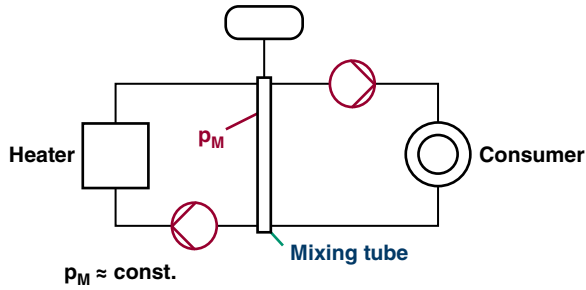


Fig. 14.10 Dual circuit plant

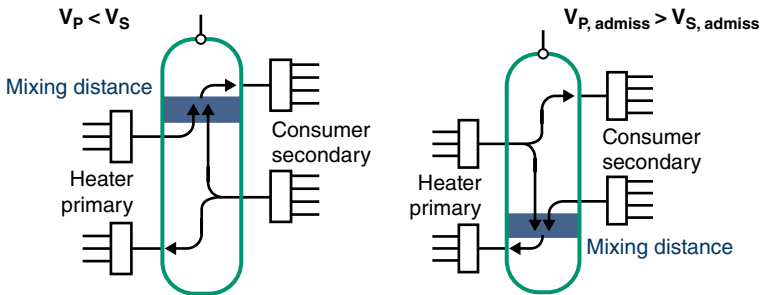


Fig. 14.11 Mixing tube

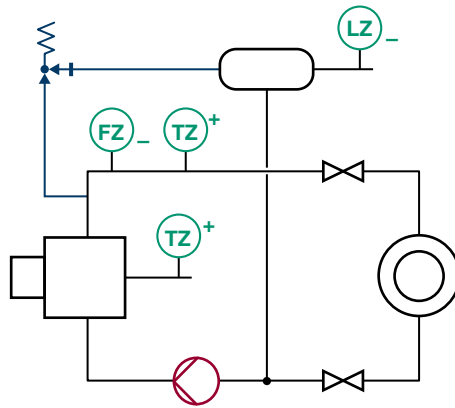


Fig. 14.12 Safety engineering

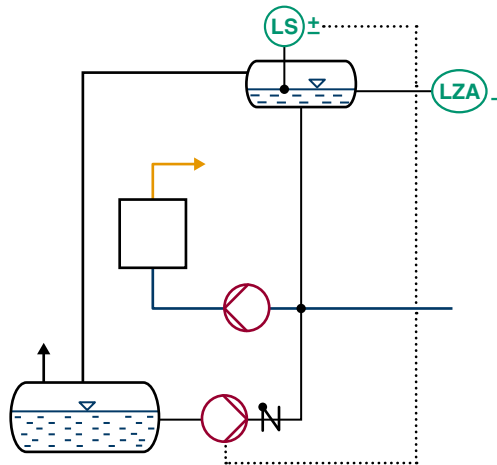


Fig. 14.13 Absorb overflow system

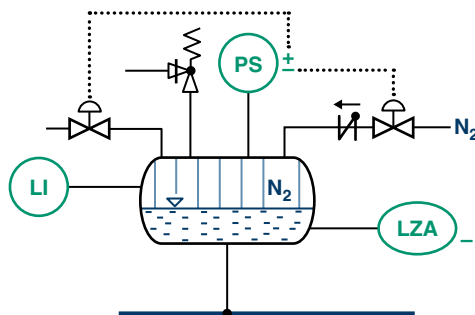


Fig. 14.14 Closed expansion tank

15 Properties of Organic Heat Transfer Fluids

Table Directory

1. Mineral Oil-Based Organic Fluids

Application Area: Approx. 50...250 °C

Product

| | |
|--|-----|
| ARAL Farolin T | 891 |
| AVIA Thermofluid A | 893 |
| BP Transcal LT | 895 |
| ESSO Thermal Oil S | 897 |
| EXXON MOBIL Mobiltherm 594 | 899 |
| FRAGOL FRAGOLTHERM Q-7 | 901 |
| FUCHS Renolin Therm 250 | 902 |
| GLOBAL HEAT TRANSFER Globaltherm LT Plus | 904 |

Application Area: Approx. 100...300 °C

| | |
|--|-----|
| ARAL Farolin U | 905 |
| AVIA Thermofluid B | 907 |
| BP Transcal N | 909 |
| EXXON MOBIL Mobiltherm 603 | 911 |
| EXXON MOBIL Mobiltherm 605 | 913 |
| FINKE Finkotherm M-300 | 915 |
| FRAGOL FRAGOLTHERM Q-32-N | 917 |
| GLOBAL HEAT TRANSFER Globaltherm M | 918 |
| GLOBAL HEAT TRANSFER Globaltherm MXT | 919 |
| MINERA Thermal Oil 36 | 920 |
| SHELL Heat Transfer Oil S2 | 922 |

2. Synthesis-Based Organic Fluids

Application Area: Approx. 50...max. 400 °C

Product

| | |
|-------------------------------|-----|
| DOW Dowtherm A | 924 |
| DOW Dowtherm G | 926 |
| DOW Dowtherm MX | 929 |
| DOW Dowtherm Q | 931 |
| DOW Dowtherm T | 933 |
| DOW Dowtherm RP | 935 |
| DOW Syltherm 800 | 937 |
| EASTMAN Therminol 54 | 939 |
| EASTMAN Therminol SP/55 | 940 |
| EASTMAN Therminol 58 | 942 |
| EASTMAN Therminol 59 | 943 |
| EASTMAN Therminol 62 | 944 |
| EASTMAN Therminol 66 | 945 |

| | |
|---|-----|
| EASTMAN Therminol 68 | 946 |
| EASTMAN Therminol 72 | 947 |
| EASTMAN Therminol 75 | 948 |
| EASTMAN Therminol VP-1 | 949 |
| EASTMAN Therminol VP-3 | 951 |
| EASTMAN Marlotherm LH | 953 |
| EASTMAN Marlotherm N | 955 |
| EASTMAN Marlotherm SH | 957 |
| FRAGOL FRAGOLTHERM DPO | 959 |
| FRAGOL FRAGOLTHERM 660 | 960 |
| FRAGOL FRAGOLTHERM 550 | 961 |
| FRAGOL FRAGOLTHERM X-75-A | 962 |
| GLOBAL HEAT TRANSFER Globaltherm HF | 963 |
| GLOBAL HEAT TRANSFER Globaltherm NF | 964 |
| GLOBAL HEAT TRANSFER Globaltherm Omnitech | 965 |
| GLOBAL HEAT TRANSFER Globaltherm RP | 966 |
| GLOBAL HEAT TRANSFER Globaltherm S | 967 |
| GLOBAL HEAT TRANSFER Globaltherm Syntec | 968 |
| LANXESS Diphyl | 969 |
| LANXESS Diphyl DT | 971 |
| LANXESS Diphyl KT | 973 |
| LANXESS Diphyl THT | 975 |
| TEXACO Texatherm HT 22 | 977 |

3. Organic Fluids for Heating-Cooling-Chilling Processes

Application Area: Approx. -50...300 °C

Product

In cooling processes, the viscosity value for the lower application limit can be increased to 7...10 mm²/s (then check the flow condition).

| | |
|--|------|
| DOW Dowtherm J | 979 |
| DOW Syltherm HF | 981 |
| DOW Syltherm XLT | 983 |
| EASTMAN Therminol ADX-10 | 985 |
| EASTMAN Therminol D-12 | 986 |
| EASTMAN Therminol LT | 988 |
| EASTMAN Therminol RD | 991 |
| EASTMAN Therminol VLT | 992 |
| EASTMAN Marlotherm XC | 994 |
| EASTMAN Marlotherm XD | 995 |
| FRAGOL FRAGOLTHERM X-T12 | 996 |
| FRAGOL FRAGOLTHERM F-12 | 997 |
| FRAGOL FRAGOLTHERM F-LT | 998 |
| GLOBAL HEAT TRANSFER Globaltherm D | 999 |
| GLOBAL HEAT TRANSFER Globaltherm J | 1000 |

4. Organic Fluids for the Food Industry**Application Area: Approx. 100...300 °C****Product**

| | |
|---|------|
| EASTMAN Therminol XP | 1001 |
| EASTMAN Marlotherm FG | 1002 |
| FRAGOL FRAGOLTHERM FG-8 | 1003 |
| FRAGOL FRAGOLTHERM FG-35 | 1004 |
| GLOBAL HEAT TRANSFER Globaltherm FG | 1005 |

5. Heat Transfer Fluids Based on Polyalkyleneglycol Derivatives**Application Area: Approx. 100...250 °C****Product**

| | |
|---------------------------------|------|
| FRAGOL FRAGOLTHERM S-250 | 1006 |
| FRAGOL FRAGOLTHERM S-20-A | 1007 |
| FRAGOL FRAGOLTHERM S-15-A | 1008 |

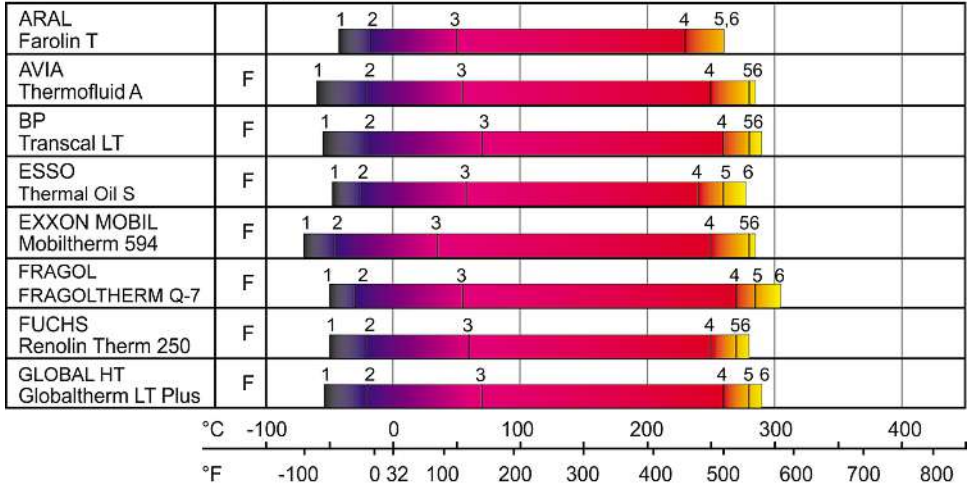
6. Organic Fluids for the Cleaning and Rinsing of Heat Transfer Installations 1009**Product**

| |
|---|
| EASTMAN Therminol FF |
| FRAGOL FRAGOLTHERM Cleaning Fluid 300 |
| FRAGOL FRAGOLTHERM Cleaning Fluid 330 |
| FRAGOL FRAGOLTHERM Cleaning Fluid 350 |
| FRAGOL FRAGOLTHERM Cleaning Fluid FG-33 |
| GLOBAL HEAT TRANSFER Globaltherm C1 |

Comments on the Following Properties of Organic Heat Transfer Fluids:

1. With the material values it must be noted that most of the physical data for temperatures above the initial boiling point are not measured, but extrapolated. Deviations of up to 10 % are possible.
2. The lower economical application temperature is assumed as a viscosity of 5 mm²/s. In individual cases this viscosity can be increased to up to approx. 10 mm²/s if acceptable heat and flow data are achieved.
3. The latest data sheets of the suppliers are valid.

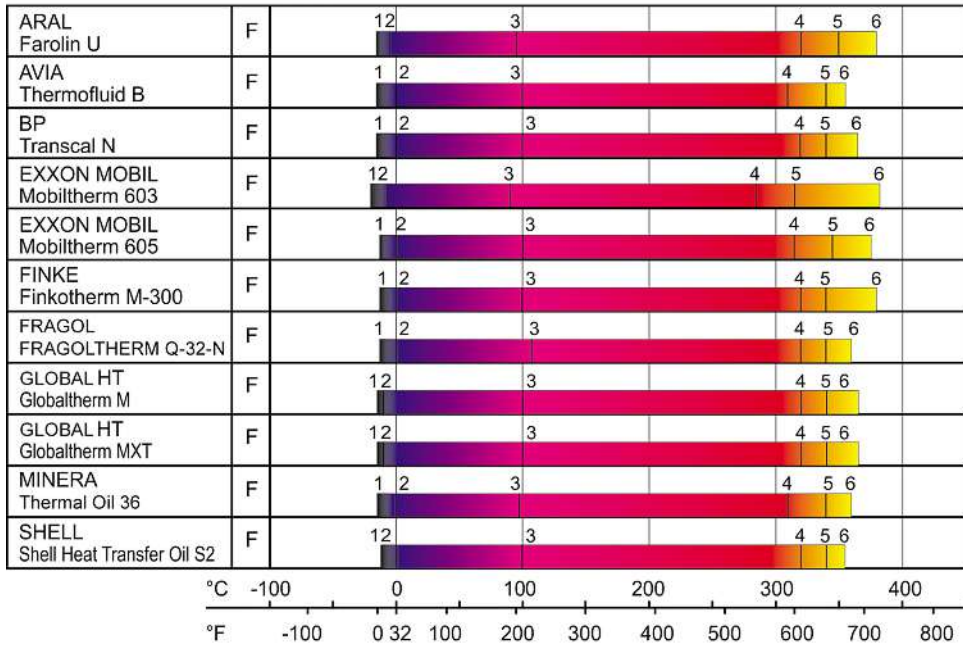
Ranges of application for mineral heat transfer fluids, scope of application approx. 50....250°C

 F = liquid
 F/D = liquid/vapor


- 1) flow limit (Pourpoint)
- 2) viscosity $\nu = 300 \text{ E-06 m}^2/\text{s}$ (filling)
- 3) viscosity $\nu = 5 \text{ E-06 m}^2/\text{s}$ (operation)
- 4) permissible flow temperature
- 5) permissible film temperature
- 6) boiling point at 1.013 bar

Ranges of application for mineral heat transfer fluids, scope of application approx. 50....300°C

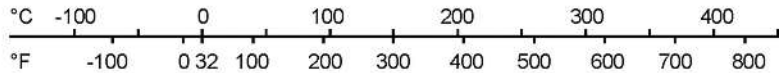
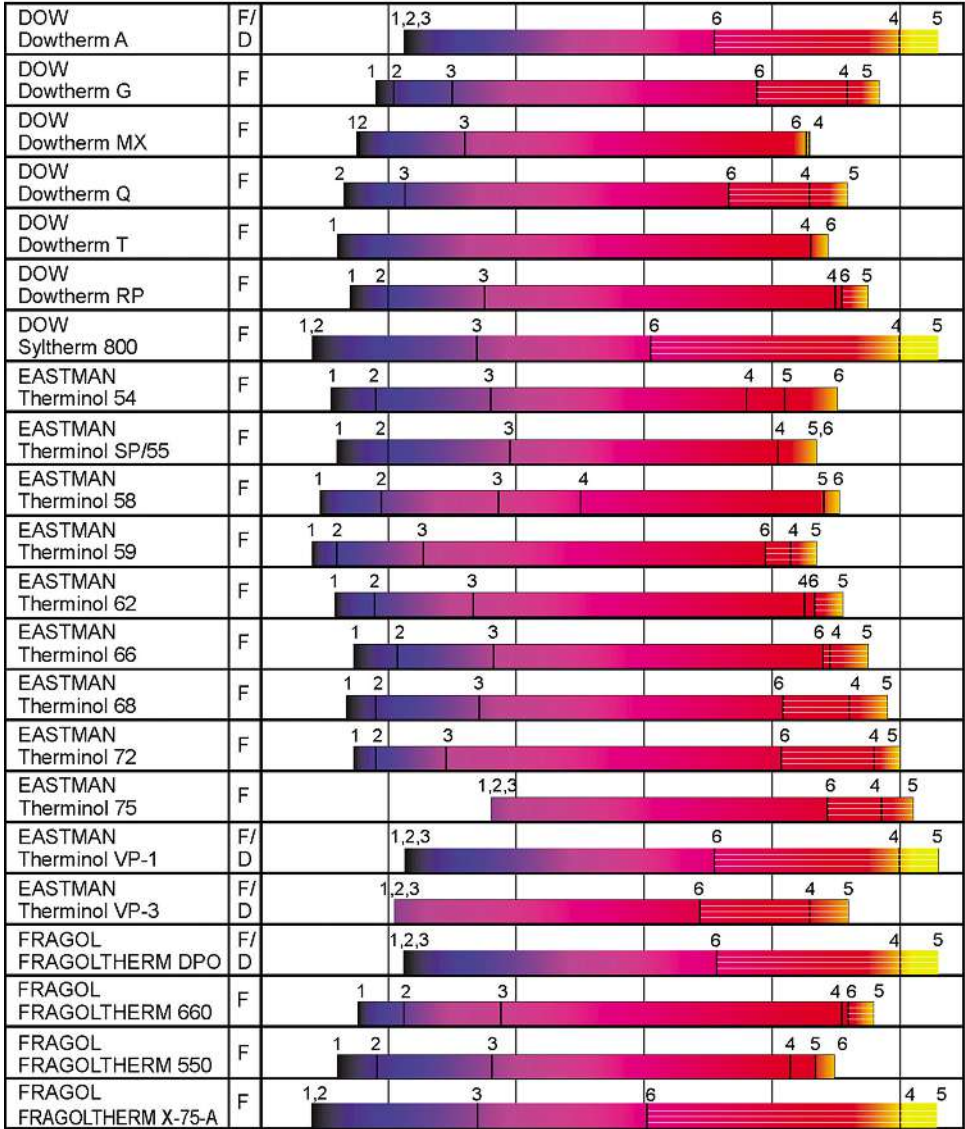
F = liquid
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- 1) flow limit (Pourpoint)
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- 3) viscosity $\nu = 5 \text{ E-06 m}^2/\text{s}$ (operation)
- 4) permissible flow temperature
- 5) permissible film temperature
- 6) boiling point at 1.013 bar

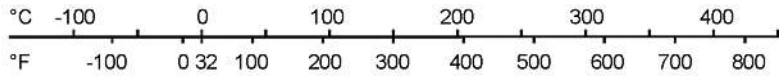
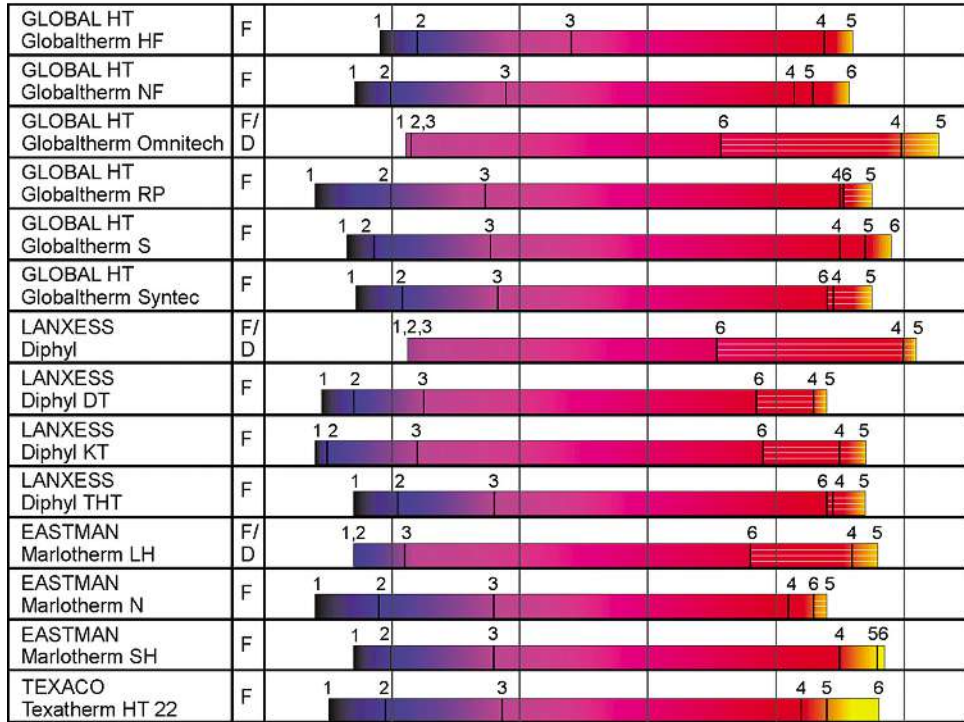
Heat transfer fluids usable at flow temperatures up to max. 400°C

F = liquid
F/D = liquid/vapor



- 1) flow limit (Pourpoint)
- 2) viscosity $\nu = 300 \text{ E-06 m}^2/\text{s}$ (filling)
- 3) viscosity $\nu = 5 \text{ E-06 m}^2/\text{s}$ (operation)
- 4) permissible flow temperature
- 5) permissible film temperature
- 6) boiling point at 1.013 bar

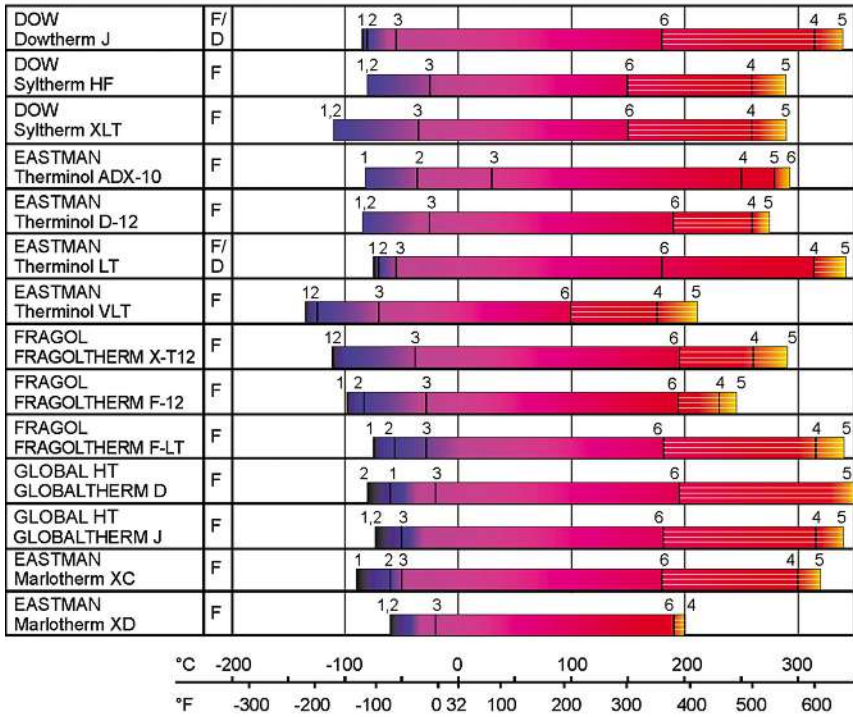
Heat transfer fluids usable at flow temperatures up to max. 400°C – Continued

 F = liquid
 F/D = liquid/vapor


- 1) flow limit (Pourpoint)
- 2) viscosity $\nu = 300 \text{ E-06 m}^2/\text{s}$ (filling)
- 3) viscosity $\nu = 5 \text{ E-06 m}^2/\text{s}$ (operation)
- 4) permissible flow temperature
- 5) permissible film temperature
- 6) boiling point at 1.013 bar

Heat transfer fluids usable for combined heating-cooling-chilling processes, scope of application approx. -50....300°C

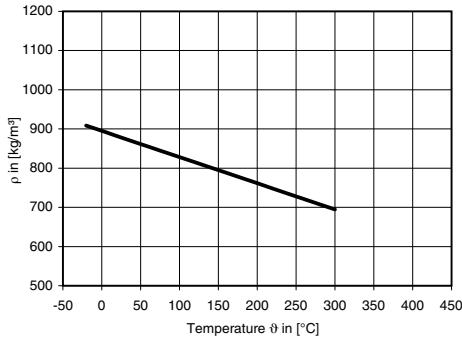
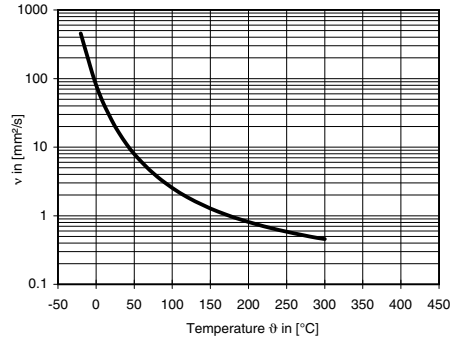
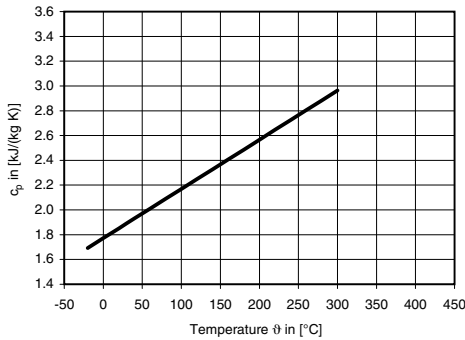
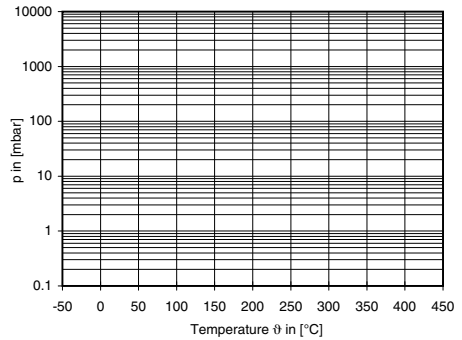
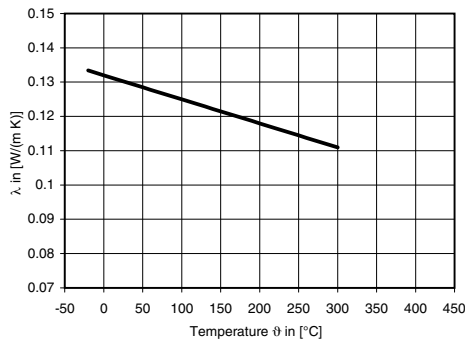
F = liquid
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- 1) flow limit (Pourpoint)
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- 3) viscosity $\nu = 5 \text{ E-06 m}^2/\text{s}$ (operation)
- 4) permissible flow temperature
- 5) permissible film temperature
- 6) boiling point at 1.013 bar

Mineral oil-based organic heat transfer fluids

Material data tables

Trade name: *Farolin T*Density $\rho = f(\vartheta)$ Kinematic viscosity $\nu = f(\vartheta)$ Spec. heat capacity $c_p = f(\vartheta)$ Vapor pressure $p = f(\vartheta)$ Thermal conductivity $\lambda = f(\vartheta)$ 

Equations and constants

| | Density ρ [kg/m³] | Spec. heat capacity c_p [kJ/(kg K)] | Kinematic viscosity ν [mm²/s] | Thermal conductivity λ [W/(m K)] | Vapor pressure p [mbar] |
|-------|------------------------------|---|---|--|---------------------------------|
| a_0 | 895.05 | 1.772 | -2.491 | 0.132 | |
| a_1 | -0.668 | 3.97E-03 | 681.961 | -7.02E-05 | |
| a_2 | | | 99.216 | | |
| a_3 | | | | | |
| a_4 | | | | | |

$$\nu, p(\vartheta) = \exp(a_0 + a_1 / (a_2 + \vartheta))$$

$$\rho, c_p, \lambda(\vartheta) = a_0 + a_1 \vartheta + a_2 \vartheta^2 + a_3 \vartheta^3 + \dots$$

16 Units and Conversion Tables

A new development was initiated by the law on units in the measuring technique of July 2, 1969¹⁷⁾ and the "execution decree" of June 26, 1970 which is to be welcomed in the interest of international cooperation. This law demands a rethinking of a series of previously usual ideas and also conversion of tabulated values, but the usefulness is clearly recognizable. The law is now in force and a transition time was provided for some units. The change-over to the system of units must have been completed by December 31, 1977.

No longer permitted with immediate effect are:

1. The unit kg as unit of force
2. The use of substance based units, such as the caloric or thermochemically defined calorie or the standard cubic meter
3. ata, atü, Ma (Mach), mile, micron with 1μ for 0.001 mm and some other more rarely used units

Legal "basic units" are the basic units of the "International Unit System" (SI) as follows:

| Basic unit | Unit symbol | Basic quantity |
|------------|-------------|---------------------------|
| meter | m | length |
| kilogram | kg | mass |
| second | s | time |
| ampere | A | electric current |
| Kelvin | K | thermodynamic temperature |
| mole | mol | quantity of a substance |
| candela | cd | light intensity |

The derived units of a system of units can be formed from these basic units, e.g. for pressure, mechanical stress, velocity, force, output, quantity of heat and others. For derived SI units the following names and unit symbols are valid:

| | | |
|---------|------|-----------------------|
| Coulomb | 1 C | = As |
| Farad | 1 F | = 1 As/V |
| Henry | 1 H | = 1 Wb/A = 1 Vs/A |
| Hertz | 1 Hz | = 1 s ⁻¹ |
| Joule | 1 J | = 1 Nm = 1 Ws |
| Lumen | 1 lm | = 1 cd sr |
| Lux | 1 lx | = 1 lm/m ² |

17) In the Federal Republic of Germany

| | | |
|-----------|-------|------------------------------------|
| Newton | 1 N | = 1 kg m/s ² |
| Ohm | 1 Ω | = 1 V/A |
| Pascal | 1 Pa | = 1 N/m ² |
| Radian | 1 rad | = 1 m/m |
| Siemens | 1 S | = 1/Ω |
| Steradian | 1 sr | = 1 m ² /m ² |
| Tesla | 1 T | = 1 Wb/m ² |
| Volt | 1 V | = 1 W/A |
| Watt | 1 W | = 1 J/s = 1 Nm/s |
| Weber | 1 Wb | = 1 Vs |

For unit samples the following relations are valid:

$$N = \text{kg m/s}^2$$

$$\text{Pa} = \text{kg}/(\text{m s}^2)$$

$$J = \text{kg m}^2/\text{s}^2$$

$$W = \text{kg m}^2/\text{s}^3$$

$$1 \text{ bar} = 10^5 \frac{\text{kg}}{\text{m s}^2}$$

The legally fixed nuclear physics units are: atomic mass unit (u) for the particle mass and electron volt (eV) for the energy. The decimal multiples and decimal fractions of units play a larger role than before:

Multiples

| | | |
|------------------|---------------------------|--------------|
| 10 ¹ | times has the prefix deca | with sign da |
| 10 ² | " " " " hecto | with sign h |
| 10 ³ | " " " " kilo | with sign k |
| 10 ⁶ | " " " " mega | with sign M |
| 10 ⁹ | " " " " giga | with sign G |
| 10 ¹² | " " " " tera | with sign T |

Fractions

| | | |
|-------------------|---------------------|---------------|
| 10 ⁻¹ | has the prefix deci | with symbol d |
| 10 ⁻² | " " " centi | with symbol c |
| 10 ⁻³ | " " " milli | with symbol m |
| 10 ⁻⁶ | " " " micro | with symbol μ |
| 10 ⁻⁹ | " " " nano | with symbol n |
| 10 ⁻¹² | " " " pico | with symbol p |
| 10 ⁻¹⁵ | " " " femto | with symbol f |
| 10 ⁻¹⁸ | " " " atto | with symbol a |

Very “unhandy” numerical values result from the conversion from the “old” system of units to the SI units by e.g. the factor 9.81 or 1.02 but they can be memorized by rounding off. The inaccuracies caused by this are bearable for the technical range discussed here. It must not be overlooked, though, in exact calculations that efforts are currently made to replace the permissible strength values of materials by conversion from kp/mm^2 to N/mm^2 partly by multiplication with 10 and all pressure data in the control works by 1 at = 1 bar.

Tables 16.1 to 16.4 contain the most important conversion factors of both systems of units. Table 16.5 contains as supplement a comparison with the Imperial system of units. Tables 16.6 and 16.7 contain conversions of various units and an example.

Table 16.1 Conversion factors and change-over advice to the SI system of units

| Type | Basic SI Unit to DIN 1301 | It is to be used | | Conversions |
|-----------------|---------------------------------------|---------------------|--|--|
| | | Instead of | In future | |
| Pressure | N/m ² ; Pa | kp/cm ² | N/m ² ; Pa | 1 kp/cm ² = 1 at = 0.9807 bar = 980.665 mbar ≈ 10 ⁵ Pa |
| | | or at | or bar | |
| | 1 Newton/m ² = 1 Pascal | mWC | bar | 1 mWC = 0.1 at = 0.0981 bar = 98.0665 mbar ≈ 10 ⁴ Pa |
| | | mmWC | mbar | 1 mmWC = 0.09807 mbar ≈ 10 Pa |
| | Torr | mbar | 1 Torr = 1 mm Hg = 0.00133 bar = 1.33332 mbar ≈ 133 Pa | |
| Energy | J; Nm; Ws | | | |
| | 1 Joule | cal | Ws | 1 cal = 4.1868 Ws |
| | = 1 Newton-meter | kcal | kWh | 1 kcal = 4186.8 Ws = 0.001163 kWh |
| | = 1 Wattsecond | | | |
| Output | W | | | |
| | 1 Watt | kcal/h | W | 1 kcal/h = 1.163 W = 0.001163 kW |
| | = 1 Joule/ second | Mcal/h | kW | 1 Mcal/h = 1163 W = 1.163 kW |
| | = 1 Newton-meter/second | Gcal/h | MW | 1 Gcal/h = 1163000 W = 1163 kW = 1.163 MW |
| Calorific value | J/kg; J/m ³ ; | kcal/kg | J/kg | 1 kcal/kg = 4.1868 kJ/kg = 4186.8 J/kg = 1.163 Wh/kg |
| | J/mol | | or Ws/kg | 1 kcal/kg = 4186.8 Ws/kg = 0.001163 kWh/kg |
| | | kcal/m ³ | J/m ³ | 1 kcal/m ³ = 4.1868 kJ/m ³ = 4186.8 J/m ³ = 1.163 Wh/m ³ |
| | | kcal/mol | J/mol | 1 kcal/mol = 4.1868 kJ/mol = 4186.8 J/mol = 1.163 Wh/mol |

Table 16.2 Conversion of various pressure units

| Pressure | N/m ² = Pa | bar | mbar | mmWC | kp/cm ² = at | atm | Torr |
|----------------------|-------------------------|-------------------------|-------------------------|------------------------|-------------------------|-------------------------|--------------------------|
| 1 N/m ² | 1 | 10 ⁻⁵ | 10 ⁻² | 0.102 | 1.02 · 10 ⁻⁵ | 9.87 · 10 ⁻⁶ | 7.5 · 10 ⁻³ |
| 1 Pa | 1 | 10 ⁻⁵ | 10 ⁻² | 0.102 | 1.02 · 10 ⁻⁵ | 9.87 · 10 ⁻⁶ | 7.5 · 10 ⁻³ |
| 1 bar | 10 ⁵ | 1 | 10 ³ | 1.02 · 10 ⁴ | 1.020 | 0.987 | 750 |
| 1 mbar | 100 | 10 ⁻³ | 1 | 10.20 | 1.02 · 10 ⁻³ | 9.87 · 10 ⁻⁴ | 0.750 |
| 1 mmWC | 9.81 | 9.81 · 10 ⁻⁵ | 9.81 · 10 ⁻² | 1 | 10 ⁻⁴ | 9.68 · 10 ⁻⁵ | 7.355 · 10 ⁻² |
| 1 kp/cm ² | 9.81 · 10 ⁴ | 0.981 | 981 | 10 ⁴ | 1 | 0.968 | 735.5 |
| 1 at | 9.81 · 10 ⁴ | 0.981 | 981 | 10 ⁴ | 1 | 0.968 | 735.5 |
| 1 atm | 1.013 · 10 ⁵ | 1.013 | 1013 | 10,330 | 1.033 | 1 | 760 |
| 1 Torr | 133.3 | 1333 · 10 ⁻³ | 1.333 | 13.6 | 1.36 · 10 ⁻³ | 1.32 · 10 ⁻⁵ | 1 |

Table 16.3 Conversion of various heat quantity units

| Quantity of heat | J = Nm = Ws | MJ | kWh | cal | kcal | Mcal |
|------------------|--------------------------|---------------------------|--------------------------|-------------------------|--------------------------|--------------------------|
| 1 J = 1 Ws | 1 | 10 ⁻⁶ | 2.778 · 10 ⁻⁷ | 0.2388 | 2.388 · 10 ⁻⁴ | 2.388 · 10 ⁻⁷ |
| 1 MJ | 10 ⁶ | 1 | 0.2778 | 2.388 · 10 ⁵ | 238.8 | 0.2388 |
| 1 kWh | 3.6 · 10 ⁶ | 3.6 | 1 | 8.6 · 10 ⁵ | 860 | 0.860 |
| 1 cal | 4.1868 | 4.1868 · 10 ⁻⁶ | 1.163 · 10 ⁻⁶ | 1 | 10 ⁻³ | 10 ⁻⁶ |
| 1 kcal | 4186.8 | 4.1868 · 10 ⁻³ | 1.163 · 10 ⁻³ | 1000 | 1 | 10 ⁻³ |
| 1 Mcal | 4.1868 · 10 ⁶ | 4.1868 | 1.163 | 10 ⁶ | 1000 | 1 |

Table 16.4 Conversion of various heat output units

| | W = 1 Nm/s = J/s | kW | MJ/h | kcal/h | Mcal/h | Gcal/h |
|-------------|-------------------------|--------------------------|---------------------------|-----------------|--------------------------|--------------------------|
| 1 W = 1 J/s | 1 | 10 ⁻³ | 3.6 · 10 ⁻³ | 0.860 | 0.860 · 10 ⁻³ | 0.860 · 10 ⁻⁶ |
| 1 kW | 1000 | 1 | 3.6 | 860 | 0.860 | 860 · 10 ⁻⁶ |
| 1 MJ/h | 277.8 | 0.2778 | 1 | 238.8 | 0.2388 | 238 · 10 ⁻⁶ |
| 1 kcal/h | 1.163 | 1.163 · 10 ⁻³ | 4.1868 · 10 ⁻³ | 1 | 10 ⁻³ | 10 ⁻³ |
| 1 Mcal/h | 1163 | 1.163 | 4.1868 | 1000 | 1 | 10 ⁻⁶ |
| 1 Gcal/h | 1.163 · 10 ⁶ | 1163 | 4186.8 | 10 ⁶ | 1000 | 1 |

Table 16.5 Conversion of the Imperial system of units to metric units

| Units of length (Brit., USA): | | | | |
|-------------------------------|--------------------------------------|---------------|------|---|
| 1 in. = 1" | = 1 Inch (divided into 16 divisions) | = 2.539954 cm | 1 cm | = 0.3937" \approx ²⁵ / ₆₄ " |
| 1 ft. = 1' | = 1 Foot (Foot= 12") | = 0.30479 m | 1 m | = 3.2809' |
| 1 yd. | = 1 Yard (Yard = 3') | = 0.91438 m | 1 m | = 1.0936 yds. |

Table 16.5 Conversion of the Imperial system of units to metric units – Continued

| | | | | |
|---------------------------------------|---|---|-------------------|-----------------------|
| Surface measures (Brit., USA): | | | | |
| 1 sq.in. | = 1 Square inch | = 6.4516 cm ² | 1 cm ² | = 0.155 sq.ins. |
| 1 sq.ft. | = 1 Square foot (144 sq.ins.) | = 0.0929 m ² | 1 m ² | = 10.764 sq.ft. |
| 1 sq.yd. | = 1 Square yard (9 sq.ft.) | = 0.8360 m ² | 1 m ² | = 1.196 sq.yds. |
| Cubic measures (Brit): | | | | |
| 1 cu.in. | = 1 Cubic inch (Cubic Inch) | = 16.3870 cm ³ | 1 cm ³ | = 0.061 cu.ins. |
| 1 cu.ft. | = 1 Cubic foot (1728 cu. ins.) | = 0.0283 m ³ | 1 m ³ | = 35.317 cu.ft. |
| 1 cu.yd. | = 1 Cubic yard (27 cu.ft.) | = 0.7646 m ³ | 1 m ³ | = 1.308 cu.yds. |
| 1 Imp.gal. | = 1 Gallon (277.27 cu.ins.) | = 4.5461 l | 1 l | = 61.024 cu.ins. |
| | | | | = 0.220 Imp.gals. |
| Cubic measures (USA): | | | | |
| q pt. | = 1 Pint (1/8 gals.) | = 0.4732 l | 1 l | = 2.112 pts. |
| 1 qt. | = 1 Quart (1/4 gals.) | = 0.9464 l | 1 l | = 1.056 qts. |
| 1 gal. (Winchester- or US-gal.) | = 1 Gallon (231 cu.ins.) | = 3.7871 l | 1 l | = 0.264 gals. |
| 1 bbl. | = 1 Barrel (31.5 gals. ¹⁸⁾) | = 1.1924 hl | 1 hl | = 0.839 bbls. |
| Weights, forces (Brit., USA): | | | | |
| 1 oz. | = 1 Ounce | = 28.2495 g | 1 g | = 0.0353 ozs. |
| 1 lb. | = 1 Pound (Pound = 16 ozs.) | = 0.4536 kg | 1 kg | = 2.2046 lbs. |
| 1 qr. | = 1 Quarter (28 lbs.) | = 12.7006 kg | 1 kg | = 0.0767 qrs. |
| 1 cwt. | = 1 Hundredweight (4 qrs.) | = 50.8023 kg | 1 kg | = 0.0197 cwts. |
| 1 Ton (long ton) | = 1 Tonne (20 cwts.) | = 1.0161 Mg(t) | 1 Mg (t) | = 0.9842 Tons |
| Pressures (Brit., USA): | | | | |
| 1 lb./sq.in. | = 1 Pound/Square inch (144 lbs./sq.ft.) | = 703.3254 mmWC (kp/m ²) | 1 mmWC | = 0.2048 lbs./sq.ft. |
| | | = 51.7151 mmQS | 1 mmQS | = 2.7851 lbs./sq.ft. |
| | | ≈ 2" Qs | 1 at | = 14.2233 lbs./sq.in. |
| 1 lb./sq.ft. | = 1 Pound/Square foot (0.006944 lbs./sq.in.) | = 4.8824 mmWC (kp/m ²) | 1 Atm | = 14.6960 lbs./sq.in. |
| Units of work (Brit, USA): | | | | |
| 1 ft.lb. | = 1 Foot-pound | = 0.1395 kp m | 1 kp m | = 7.1697 ft.lbs. |
| 1 HPh | = 1 horsepower-hour | = 1.014 PSh | 1 PSh | = 0.9862 HPh |
| | | = 0.7453 kWh | 1 kWh | = 1.3417 HPh |

18) For mineral oil it is 1 bbl = 42 US gals = 158.958 l = 1.5896 hl and in England 1 bbl = 163.7 l (dry measure).

17 Comparison of German and Foreign Rules and Standards²¹⁾

It will facilitate the use of this book, if the most important international standards are stated and compared.

17.1 Pipeline Components and Materials

Petrochemical engineering being based worldwide on American standards these have particular importance for all pipeline components, such as pipes, flanges and shaped components.

American Standards

The American National Standards Institute (ANSI, previously USASI, formerly ASA) is the American membership organization in ISO. It issues, similarly to the German Institute for Standardization (DIN), standards, and their designation consists of ANSI followed by a capital letter for the subject field – e.g. B for engineering –, followed by a number for the subject branch within the subject field – e.g. 16 for pipeline flanges and fittings –, a current standard sheet number separated by a stop and the year of publication or new edition separated by a dash.

The American Society for Testing and Materials (ASTM) is concerned with the preparation and publishing of material standards, technical supply conditions and test standards. It annually publishes standard books, the individual volumes containing all pertinent ASTM standards according to subject fields.

The American Petroleum Institute (API) issues standards for the field of the mineral oil and natural gas industry which have gained worldwide importance in connection with the technical development of deep boreholes and mineral oil technology in the USA. The API standards had great effects on the international standardization work of ISO (TC 67 equipment for the mineral oil and natural gas industry).

Sizes

The work with American size standards is made difficult by the inch unit which does not raise clear comparison images in Continental Europeans. In Germany, nominal size and external diameter always differ, in America the outside diameter = nominal pipe size (NPS) from 14".

The sizes familiar in EN are based on ISO unit series containing American sizes so that good agreement exists between DIN and ANSI nominal sizes.

It is e.g. NPS 6" = DN 150 with an outside diameter of 168.3 mm.

The conversion from inch to mm is simple, if it is roughly 4" = 100 mm. It is slightly more difficult for standard wall thicknesses for which two types of description exist. They are given either as standard wall (standard wall thickness), extra strong and double extra strong or as schedule 40, 60, 80 or 160. A statement in inches is not generally made. To make these matters still more difficult the standard wall does not always correspond to a definite schedule but is, depending on NPS, either schedule 40 or schedule 30 or is between schedule 10 and 20. The same is true for extra strong and double extra strong.

The steps of the wall thicknesses are coarser than in EN and also extend over a larger range. ANSI B 36.10 covers not only the wall thicknesses but extends far above the maximum value (25 mm).

21) Regarding the standards presented it must always be ensured, that the latest valid version of the standards is used.

A kind of nominal pressure stages exists for flanges but in contrast to German practice this is not related to 20 °C but to temperatures above 250 to 500 °C. The permissible operating pressure of a DIN flange being at maximum equal to, generally though lower than, the nominal pressure, the permissible operational pressure of an ANSI flange can be considerably higher than the nominal pressure stage expresses. ANSI B 16.5 contains seven "nominal pressure stages":

| | |
|-------------------------------------|---------|
| 150 lb/sq.in. corresponds to about | 10 bar |
| 300 lb/sq.in. corresponds to about | 20 bar |
| 400 lb/sq.in. corresponds to about | 28 bar |
| 600 lb/sq.in. corresponds to about | 40 bar |
| 900 lb/sq.in. corresponds to about | 65 bar |
| 1500 lb/sq.in. corresponds to about | 100 bar |
| 2500 lb/sq.in. corresponds to about | 176 bar |

ANSI Standards

Important ANSI standards for steel pipes are:

B 16 – Pipeline Flanges and Shaped Components

| | |
|---------|--|
| B 16.5 | steel pipe flanges and shaped components with flanges |
| B 16.9 | shaped components of steel with welding necks |
| B 16.11 | shaped components of wrought steel with welding neck sockets and threads |
| B 16.20 | ring joint seals and grooves for steel pipe flanges |
| B 16.25 | welding necks for pipes, flanges and shaped components |

B 31 – Pressurized Pipelines

| | |
|--------|--|
| B 31.1 | pipelines in power plants |
| B 31.2 | pipelines for combustible gases |
| B 31.3 | pipelines in mineral oil refinery plants |
| B 31.4 | pipelines for mineral oil transport |
| B 31.8 | pipelines for gas transport and gas distribution |

B 36 – Pipes of Iron and Steel

| | |
|---------|--|
| B 36.3 | technical data for seamless carbon steel pipes for high temperatures (ASTM A 106) |
| B 36.4 | technical data for electrically fusion welded steel pipes of plates (ASTM A 134) |
| B 36.5 | technical data for electrically resistance welded steel pipes (ASTM A 135) |
| B 36.9 | technical data for electrically fusion welded steel pipes (ASTM A 139) |
| B 36.11 | technical data for electrically fusion welded steel pipes for high temperatures (ASTM A 155) |
| B 36.16 | technical data for helically seam welded steel pipes (ASTM A 211) |
| B 36.17 | technical data for seamless boiler, superheater and heat exchanger tubes (ASTM A 213) |
| B 36.19 | tubes of stainless steel |

| | |
|---------|--|
| B 36.28 | technical data for seamless cold drawn heat exchanger and condenser tubes (ASTM A 179) |
| B 36.30 | technical data for seamless steel tubes for use in refineries (ASTM A 200) |
| B 36.33 | technical data for welded austenitic steel tubes for boilers, super-heaters and heat exchangers (ASTM A 249) |
| B 36.38 | technical data for seamless and welded austenitic steel tubes for the food industry (ASTM A 270) |
| B 36.39 | technical data for seamless austenitic steel tubes for mineral oil processing (ASTM A 271) |
| B 36.42 | technical data for seamless steel tubes for high temperature use (ASTM A 335) |
| B 36.43 | technical data for seamless austenitic steel tubes for high temperatures (ASTM A 376) |
| B 36.44 | technical data for seamless steel tubes with special heat treatment or high temperatures (ASTM A 405) |
| B 36.47 | technical data for electrically fusion welded austenitic steel tubes for high temperatures (ASTM A 358) |
| B 36.49 | technical data for arc welded steel tubes for high pressures (ASTM A 381) |
| B 36.50 | technical data for welded austenitic steel tubes of large diameter (ASTM A 409) |
| B 36.56 | technical data for seamless carbon steel pipes for product lines (ASTM A 524) |
| B 36.58 | technical data for seamless, cold drawn feedwater pipes (ASTM A 556) |
| B 36.60 | technical data for electrically welded steel pipes for the chemical industry (ASTM A 587) |

API Standards

The most important standards of API in the field of steel pipes are:

| | |
|----------|--|
| Std 5 A | pipes, pipe linings and borehole tubings |
| Std 5 AC | pipes and pipe linings of grades C 75 and C 95 |
| Std 5 AX | high strength pipes and pipe linings |
| Std 5 L | conduit pipes |
| Std 5 LS | helically seam welded conduit pipes |
| Std 5 LX | conduit pipes for high stresses |

ASTM Standards

The material grades contained in the various ASTM standards are compared with the comparable materials of EN standards (see Table 17.1).

In Table 17.1 the most important materials used in heat transfer fluids technology are compared.

Tables 17.2 to 17.11 contain the American standards for:

Table 17.2 welded and seamless pipes

Table 17.3 stainless steel pipes

Table 17.4 general survey of flanges, welding ends and facings

Table 17.5 pressure-temperature table for carbon steel

Table 17.6 welding neck flanges 150 lb/sq. in.

Table 17.7 welding neck flanges 300 lb/sq. in.

Table 17.8 welding neck flanges 400 lb/sq. in.

Table 17.9 welding neck flanges 600 lb/sq. in.

Table 17.10 welding neck flanges 900 lb/sq. in. including example for marking flanges

Table 17.11 facings with: large and small male/female facings, large and small groove/ tongue facings

Table 17.1 Material comparison

1. Unalloyed Steel

| Germany | | | Europe | | USA | |
|--------------|-------------------------------|--------------|--------------------|------------------|--------------------|---------------|
| Material no. | DIN- and material designation | Standard DIN | Designation grades | Standard EN | Designation grades | Standard ASTM |
| | P195T1 | -EN ... | P195T1 | 10216-1 | | |
| | P195T1 | -EN ... | P195T1 | 10217-1 | F | A 53 |
| | P195T1 | -EN ... | P195T1 | 10217-1 | | |
| | P195T2 | -EN ... | P195T2 | 10216-1 | | |
| | P195T2 | -EN ... | P195T2 | 10217-1 | | |
| 1.0211 | St 30 Si | 17120/21 | S215GSiT | - | | |
| 1.0212 | St 30 Al | 17120/21 | S215GAlT | - | | |
| 1.0254 | St 37.0 | 1629 | P235T1 | 10216-1 | A A | A 53 A 106 |
| 1.0254 | St 37.0 | 1626 | P235T1 L235 | 10217-1 10224 | A | A 53 |
| 1.0255 | St 37.4 | 1630 | P235T2 | 10216-T1 | A | A 135 |
| 1.0255 | St 37.4 | 1628 | P235T2 | 10217-T1 | Y 35 | A 381 |
| 1.0038 | RSt 37-2 | 17121 | S235JR | - | 1 | A 252 |
| 1.0038 | RSt 37-2 | 17120 | S235JR | - | A | A 500 |
| 1.0039 | S235JRH | -EN ... | S235JRH | 10210-1 | | |
| 1.0116 | St 37-3 | 17121 | S235J2 | - | | |
| 1.0116 | St 37-3 | 17120 | S235J2 | - | | |
| 1.0256 | St 44.0 | 1629 | P275T1 | 10216-1 | B B | A 53 A 106 |
| 1.0256 | St 44.0 | 1626 | P275T1 | 10217-1 | B | A 53 |
| 1.0257 | St 44.4 | 1630 | P275T2 | 10216-1 | B | A 135 |
| 1.0257 | St 44.4 | 1628 | P275T2 | 10217-1 | Y 42 | A 381 |
| 1.0044 | St 44-2 | 17121 | S275JR | - | 2 | A 252 |
| 1.0044 | St 44-2 | 17120 | S275JR | - | B | A 500 |
| 1.0149 | S275J0H | -EN ... | S275J0H | - | - | A 501 |
| 1.0144 | St 44-3 | 17121 | S275J2 | | | |
| 1.0144 | St 44-3 | 17120 | S275J2 | | | |
| 1.0138 | S275J2H | -EN ... | S275J2H | | | |
| 1.0421 | St 52.0 | 1629 | P355N | 10216-3 | Y 52 | A 381 |
| 1.0421 | St 52 0 | 1626 | P355N | 10217-3 | | |
| 1.0581 | St 52.4 | 1630 | P355N | 10216-3 | | |
| 1.0581 | St 52.4 | 1628 | P355N | 10217-3 | | |
| 1.0547 | S355J0H | -EN ... | S355J0H | 10210-1 | 3 | A 252 |
| 1.0570 | St 52-3 | 17121 | S355J2 | - | III | A 618 |
| 1.0570 | St 52-3 | 17120 | S355J2 | - | III | A 618 |
| 1 0576 | S355J2H | -EN ... | S355J2H | 10210-1 | III | A 618 |
| 1.0576 | S355J2H | -EN ... | S355J2H | 10210-1 | III | A 618 |

Parabolic Trough Solar Power Plants – The Largest Thermal Oil Plants in the World

Dr. Dietmar Hunold

1. Background

Already in the middle of the 80's of the last century parabolic trough solar power plants with a total electric capacity of more than 350 MW were erected in the Californian Mojave Desert. These plants have been steadily in operation until today.

Since the middle of 2007, the power generation using solar thermal power plants has been subsidized in Spain by a feed-in tariff of 0,12 €/kWh above the respective market price. And this is assured by law for a period of 20 years. After that still 80 % of the last subsidy will be guaranteed unlimited in time for the remaining running time of the power plants. Through this a real "solar boom" was triggered in Spain: As per March 2011 already more than 14 power plants with a total of 700 MW of electric peak capacity are in operation in Spain, additional 28 power plants (1.400 MW) are under construction as well as additional 19 power plants (950 MW) are in concrete planning. Up to year 2014 in Spain, the installed electric total capacity of solar thermal power plants was thereby increased to more than 3.000 MW. This corresponds approximately with the capacity of four conventional coal-fired power plants or three nuclear power plants. At an approximately power plant price of 250 million € the total investments accumulate to around 15 billion €. With this the solar thermal power generation has reached big industrial character also in Europe now. The used technology of the solar thermal power generation thereby is extensively based on the power plants in California, for the most part however supplemented by thermal energy storages (see below).

The first CSP plants are already in operation also in African countries (Egypt, Algeria, Morocco – NOOR complex – and South Africa). Here the values of irradiation (DNI = Direct Nominal Irradiation in kWh/year) are considerably higher than the corresponding values of the south of Spain which increases the efficiency of solar thermal power plants in direct proportion (refer Fig. 1).

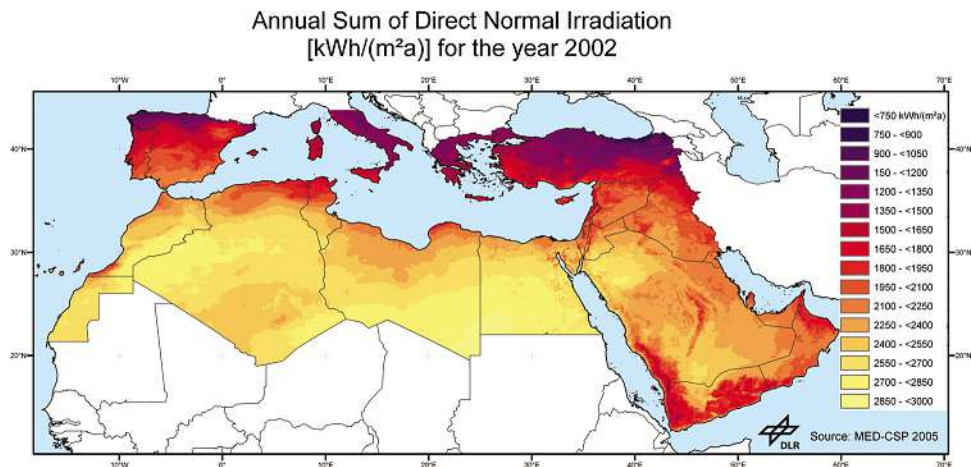


Fig. 1 Annual solar direct irradiation in Southern Europe and the MENA countries in kWh/year. [Source: DLR]

The Desertec Industrial Initiative (DII) develops in a study the technological and geopolitical conditions for a significant supply of Southern and Central Europe with renewable generated electricity from North Africa. In the coastal regions this should be done by wind farms, in the Sahara by solar thermal power plants. The power transmission to Europe can take place by high-voltage direct-current transmission (HVDC). This technology to the relatively low-loss transport of electricity over long distances is already available and used in Northern and Central Europe. For example, hydropower plants in Norway and off-shore wind farms being built in increasing numbers in the North Sea, are connected to the German power grid by HVDC.

2. The Technology of Parabolic Trough Solar Power Plants

In parabolic trough solar power plants, the sunlight is concentrated by parabolic shaped mirrors onto a vacuum-insulated absorber tube along the focal line, in which a special heat transfer fluid (HTF) flows and which is heated up to maximum 400 °C – the thermal stability limit of the HTF. Collectors of the current design have an opening (aperture) of just under 6 m and concentrate the solar radiation by about a factor of 80 (cp. Fig. 2).

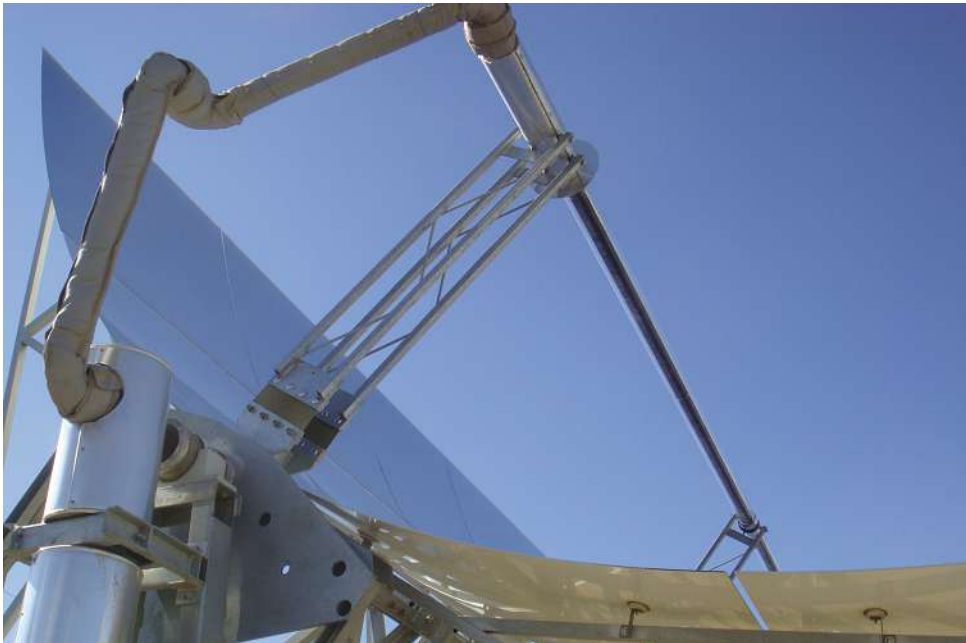


Fig. 2 Parabolic trough collector [Source: LUZ]

The HTF releases the thermal energy via heat exchangers (preheater, vaporizer, super heater, re-heater) to a conventional water / steam circuit with single reheating (steam parameters 371 °C / 100 bar / 30 bar) operating a turbine with generator. The overall process is shown schematically in Fig. 3.

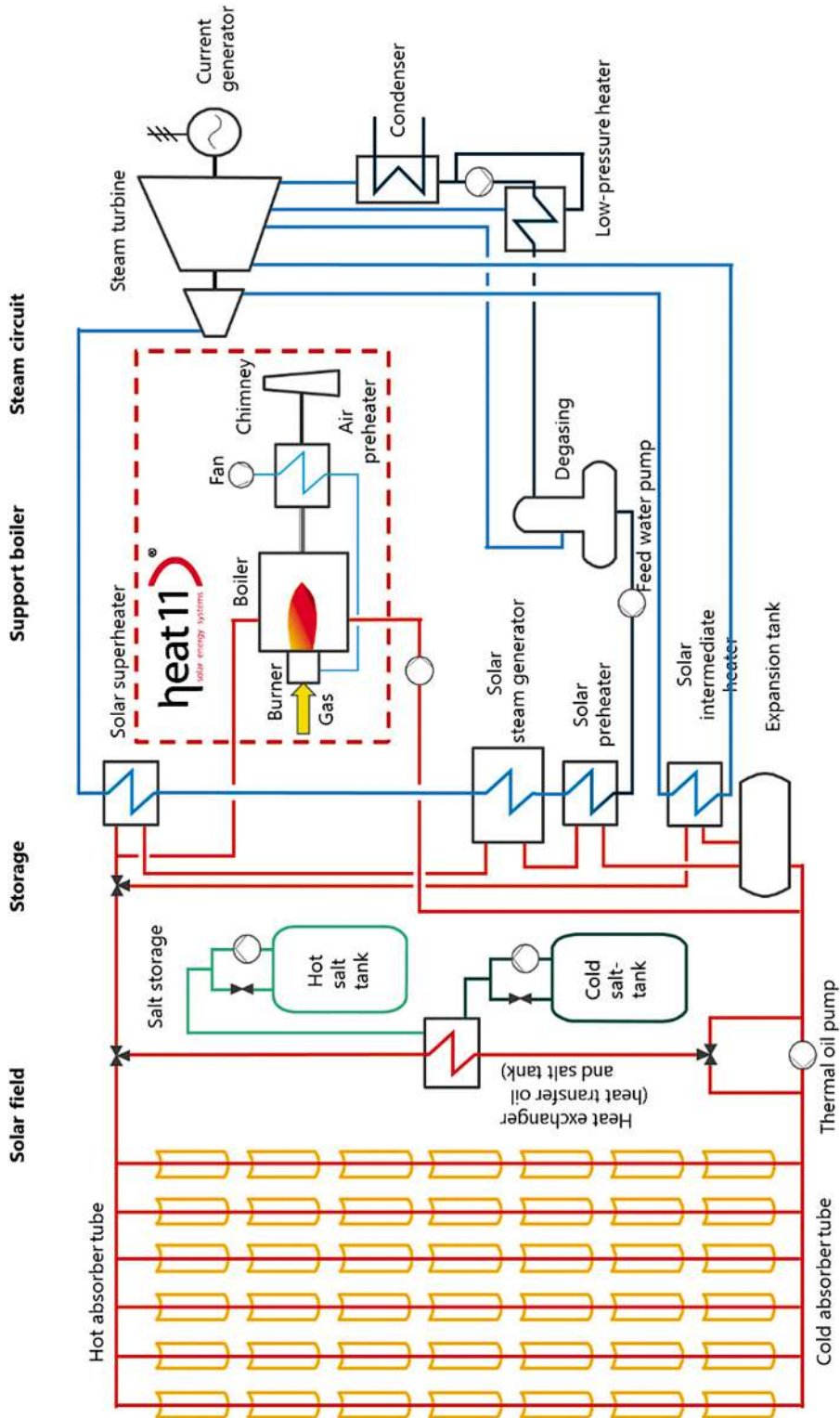


Fig. 3 Schematic diagram of a parabolic trough solar power plant [Source: heat 11]

Some Technical Boundary Conditions:

Due to the overall efficiency of the plant, around 160 MW of thermal capacity must be made available from the solar field to reach an electric capacity of 50 MW. For a location as in the south of Spain around 330.000 m² reflecting surface will be required. This results in a demand of space of approximately 1 km². Fig. 4 presents the three parabolic trough solar power plants Solnova 1, 3 and 4 near Seville with each 50 MW of electric capacity. Two solar tower power plants with 10 respectively 20 MW of electric capacity are operated also there.



Fig. 4 Parabolic solar power plants Solnova 1, 3 and 4 (3 x 50 MW_{el}) in San Lucar near Seville. In the upper part of the picture there are the two solar tower power plants PS 10 and PS 20 with a capacity of 10 respectively 20 MW. [Source: Wikipedia]

Thermal solar power plants have a number of advantages in comparison to photo-voltaic systems (PV) where solar radiation is directly converted into electricity:

1. The specific investment costs and electricity production costs are significantly below those of PV systems.
2. Due to the relatively large thermal inertia of the entire plant resulting from the widely use of large HTF volume (>1000 tons), the performance of a solar power plant does not break down immediately at short-term fluctuations of the radiation intensity, as it occurs for example in clouds passes.

To illustrate this, a small numerical example: If the temperature of the entire thermal oil filling (assumption: 1000 tons) of the power plant drops down at a temperature difference of 50 K within ½ hour due to lack of sunlight, a capacity of just under 70 MW will be released in this period which still corresponds to approximately half of the thermal capacity of the solar field.

1. The integration of thermal energy storage (TES) offers in particular the possibility to decouple in time the provision of the solar process heat from the conversion into electricity (“load

management”). That means the power generation can be moved in peak load times when higher feed-in tariffs are paid. The power generation even in times of little or lack of solar radiation is possible due to TES (securing base load and peak load covering). For this reason, numerous solar power plants are equipped with energy storages on the basis of molten salt-hot/cold storages. For a storage capacity of approximately 8 hours full-load operation of a 50 MW_{el} power plant around 30.000 tons of salt are required. In this case the size of the solar field grows from around 330.000 m² to around 520,000 m² up to 530.000 m² (based on radiation conditions as they are found in the south of Spain).

By using TES, thermal solar power plants – unlike PV systems – can be integrated very well in existing electric grid infrastructures. Especially the growing proportion of renewable technologies in electricity generation poses increasingly a challenge for energy providers (utilities) to ensure coverage of stable power supply with a highly variable offer of renewable generated electricity.

2. Solar thermal solar power plants furthermore can be fitted easily with fossil supplementary firing in form of gas- or oil-fired thermal oil boilers to allow an additional flexible operation of the power plant (to say the power generation) and a longer use of the capital-intensive power plant block that leads to an increase of profitability. More over the morning start-up procedure of the steam power process can be expedited by the thermal oil boilers with which an improved use of the solar field is attained. By means of the fired thermal oil boilers, the warm-up of the appliances (steam generator, super heater etc.) is already before the sunrise and just after the sunrise the thermal energy of the solar field can be used directly to generate electricity.

The total capacity of the thermal oil boilers with 45 to 50 MW comes to around 1/3 of the thermal capacity of the solar field (ref. Fig. 5).

Due to manufacturing issues, limitations resulting from the road transport, and to ensure sufficient redundancy the total capacity of the fossil supplementary firing is divided on two or three boilers. By the use of air preheaters and burners with O₂ control, these thermal oil boilers attain a thermal efficiency higher than 90 % despite of an inflow temperature of 395 °C.



Fig. 5 Three bottom-fired thermal oil heaters (fired boilers) with a capacity of 17 MW each, combustion air pre-heater, stack for CSP plant “Los Arenales” nearby Seville in Spain [Source: heat 11].

Apart from this field of application fired thermal oil heaters are also be used as so-called “night heaters” or “anti-freeze heaters” in order to ensure that the HTF (thermal oil) can be maintained above the freezing point of +12 °C. Even at nighttime or during longer periods in which the solar field is not in operation (due to maintenance or in rain periods). In this case the heating capacity is smaller.

As an example: The capacity of a “anti-freeze heater” the 100 MW parabolic trough solar power plant »Ilanga CSP 1 / Karashoek Solar One« located 25 km east of the South African city of Upington is just 2 x 7 MW (Fig. 6).



Fig. 6 Two bottom-fired thermal oil heaters (fired boilers) with a capacity of 7 MW each, combustion air pre-heater, stack for 100 MW CSP plant “Ilanga” nearby Upington in South Africa [Source: heat 11].

3. Qualification of Parabolic Trough Collector Panels – Mobile Test Unit

For the thermal and fluid dynamical qualification of parabolic trough collector loops a so-called Mobile Test Unit (MTU) can be used. Thereby different operating parameters (mass or volume flow through the collector panel, temperature level of the HTF) are varied under consideration of the current meteorological data (DNI, air humidity and temperature, wind speed) to determine the energy efficiency of collector loops.

The main components of such a plant are:

1. Speed controlled circulating pump with which the volume flow can be variably adjusted through the absorber tube in the range of six to 45 m³/h.
2. Combined expansion and collecting tanks for receiving the HTF.
3. Air-cooled re-cooler with maximum 2500 kW of capacity which releases the heat that is absorbed in the collector loops to be tested to the ambient. Here the capacity control is also via a variable speed of the cooling fans.
4. Control and measured value data-processing unit in which in particular the thermodynamical calculation of the collector loop is made.

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