7 Basic calculations

7.1 Factors influencing pipe system design

Type of influence	Short symbol	Unit
Outside diameter	d_e	mm
Inside diameter	d_i	mm
Internal pressure	p_i	bar
External pressure	p_{ex}	bar
Underpressure	p_u	bar
Reference stress	σ_{ref}	N/mm²
Creep modulus	E_{cR}	N/mm²
Operating temperatures		
Lowest operating temperature	$T_{i\ min}$	°C
Highest operating temperature	$T_{i\ max}$	°C
Highest ambient temperature	$T_{a\ max}$	°C
Average installation temperature	T_{ins}	°C
Safety factor		
for load class I = 1.3	S	
for load class II = 1.8	S	
for load class III = 2.0	S	
for stability N = 2.0	S	
Calculated pipe wall temperature		
$T_W = 0{,}9 \cdot T_{Flow\ medium}$	$T_{W\ min}$ $T_{W\ max}$	°C
Maximum occurring temperature difference		
max $\Delta\vartheta = T_u - T_i$	$\Delta\vartheta$	K
T_0 = upper maximum value		
T_u = lower maximum value		
Load period:		
Load h/a (= hours per annum)		
for min T_W	-	h/a
for max T_W	-	h/a
for ρ_i	-	h/a
for ρ_a	-	h/a
Calculated service life for	t_{SL}	h
calculation of creep strength	t_{SL}	a
Reduction factors		
Time dependence (if applying appendix A1, this factor has already been taken into account)	A_1	-
Reduction factor		
(influence of ambient medium) according to DVS 2205 - Part 1	A_2	-
Reduction factor		
Temperature dependence (if applying appendix A1, this factor has already been taken into account!)	A_3	-
Reduction factor		
(influence of specific resilience) DVS 2205 - Part 1 (*See table 7.12*)	A_4	-
Flow medium		
Pressure loss per m pipe	Δp	bar/m
Flow velocity	w	m/s
Flow volume	$\dot{V}$	m³/h

Type of influence	Short symbol	Unit
Pressure Surge	p_{surge}	
Welding factors		
Short-term welding factor	f_s	-
Long-term welding factor	f_l	-
(The abbreviations f_s and f_l indicated here have been freely chosen by Akatherm. In the DVS 2205 Part 1 guideline they are shown as f_z (short-term welding factor) and f_l (long-term welding factor)!		
Connection method		
- *detachable connection*		
- unions		
- couplings		
- plug-in connections		
- flange connections		
- compression connections		
- *permanently attached connections*		
- welding		
o electrofusion	EF	
o butt-welding	BW	
o hot gas extrusion welding	HE	
o hot gas hand welding	HW	
o hot gas speed welding	HR	
Types of installation		
- above ground with linear compensation	☐	
- above ground without linear compensation	☐	
- concrete encased	☐	
- buried	☐	
- installation in the open	☐	
- installation in buildings	☐	
Additional influences		

Table 7.1

The listed data is meant to guide compilation of important data and variables important for the construction of a pipe installation.

Note:
The listed mathematical operations and relations have been simplified as much as possible. Plastic-specific parameters and generally valid factors are partly already integrated into calculation formulae. We have abstained from detailing the derivation or reproduction of single values to abbreviate this section.

7.2 Calculating pipe parameters

7.2.1 Explanation

The calculation of thermoplastic pipe systems is especially important for the engineer undertaking a project. This chapter presents the basic principles required for designing plastic pipe systems. However, the practitioner (user) should also be able to acquire the necessary data and standard variables for internally pressure-loaded pipe in a relatively simple manner without spending a lot of time. The mathematical operations are supported by diagrams in the appendices from which most values and data can be read off. Thermoplastic pipe calculations basically occur on the basis of long-term values. The reference stress (σ_{ref}) and mechanical (creep) strength (i.e. creep modulus (E_{cR})) of a pipe system in relation to temperature can be derived from the creep diagram in appendix A1 and the creep modulus curve diagram in appendix A2. The creep rupture curves are based on internal pressure tests on pipe samples filled with water and represent minimum values. If pipe systems are not intended for water but for other flow media, their effects on creep strength properties must be given special consideration.

Basic calculations

7.2.2 Hydraulic principles

- Calculating flow velocity

$$w = 354 \frac{\dot{V}'}{d_i^2} \quad \text{resp.} \quad w = 1275 \cdot \frac{\dot{V}''}{d_i^2}$$

Equation 7.1

d_i = inside diameter -> $d_s = d_e - 2 \cdot e$ (mm)
$\dot{V}'$ = flow volume (flow rate) (m^3/h)
$\dot{V}''$ = flow volume (flow rate) (l/s)
w = flow velocity (m/s)
Standard values for fluids:
w ≈ 0.5 - 1.0 m/s -> intake side
w ≈ 1.0 - 3.0 m/s -> pressure side

- Calculating the flow volume

The required pipe size is derived from the flow velocity of the medium and the clear cross-section of the pipe. The flow rate of fluids is calculated as follows:

$$\dot{V} = A \cdot w = const.$$

$$A = \frac{d_i^2 \cdot \pi}{4} \qquad d_i = d_e - 2 \cdot e$$

Equation 7.2

d_i = inside diameter of the pipe (mm)
d_e = outside diameter of the pipe (mm)
e = pipe wall thickness (mm)
A = cross-sectional area of flow channel (mm^2)
w = flow velocity (m/s)
$\dot{V}$ = flow volume (m^3/h, l/s)

- Calculating the inside diameter of the pipe

If the flow rate (flow volume) is known and the constants in the above-stated equations have been compiled, this yields a conventionally-sued formula for calculating the inside diameter of the pipe:

$$d_i = 18{,}8 \cdot \sqrt{\frac{\dot{V}'}{w}} \quad \text{resp.} \quad d_i = 35{,}7 \cdot \sqrt{\frac{\dot{V}''}{w}}$$

Equation 7.3

d_i = inside diameter of the pipe (mm)
w = flow velocity (m/s)
V' = flow volume (m^3/h)
V'' = flow volume (l/s)

The tomogram presented in the appendix A16 (along with the associated tables A16a, A16b and A16c) can be used to directly calculate the appropriate dimensions.

- Calculating the Reynolds number (Re)

The Reynolds number (Re) indicates the relationship of the inert forces affecting the flow particles to the forces of viscosity. The Reynolds number is dimensionless and is calculated as follows:

$$Re = \frac{w \cdot d_i}{10^3 \cdot \nu}$$

Equation 7.4

Re = Reynolds number (-)
ν = kinematic viscosity (m^2/s)
w = flow velocity (m/s)
d_i = inside diameter of the pipe (mm)

- Determining the pipe friction number (λ_P)

The dimensionless pipe friction number (λ_P) is required to calculate the hydraulic loss in the pipe. For laminar flows (Re < 2320) in circular cross-sections, the following holds:

$$\lambda_P = \frac{64}{Re}$$

Equation 7.5

Re = Reynolds number (-)
λ_P = pipe friction number (-)

In most cases, turbulent flows are created in pipes (Re > 2320).
The pipe friction number (λ_P) for turbulent flows can be determined with the aid of appendix A3. The pipe roughness value (k), which is crucial for the calculation, can be considered to be 0.01. In practice, $\lambda_P = 0.02$ can be used to make rough estimates of hydraulic loss.

- Equivalent pipe length

$$L_{tot} = L_{Pipe} + L_{Fitting} + L_{Valve}$$

Equation 7.6

$$L_{Pipe} = \text{total pipe length (straight pipe)}$$

$$L_{Fitting} = \frac{d_i}{10^3} \cdot \Sigma \xi_F$$

$$L_{Valve} = \frac{d_i}{10^3 \cdot \lambda_P} \cdot \Sigma \xi_A$$

Equation 7.7

L_{tot} = total pipe length (m)
ΣL_{Pipe} = sum of all pipe lengths in the system (m)
$\Sigma L_{Fitting}$ = sum of all fitting lengths in the system (m)
ΣL_{Valve} = sum of all valve lengths in the system (m)
d_i = inside diameter of the pipe (mm)
λ_P = pipe friction number (-)
$\Sigma \zeta_F$ = sum of the individual resistances of fittings (-)
$\Sigma \zeta_A$ = sum of the individual resistances of valves (-)

An exact indication of pressure loss at connections (welding joints, flange connections, unions) is not possible because the nature and manner of the connections vary. An overall addition of 10% to the calculated total loss would appear to be appropriate.

- Calculating total pressure loss

In calculating the total pressure loss, the sum of the respective individual pressure losses must be computed. The sum of the individual pressure losses is calculated by:

$$\Delta p_{tot} = \Sigma \Delta p_{RP} + \Sigma \Delta p_{RF} + \Sigma \Delta p_{RV} + \Sigma \Delta p_{RC}$$

Equation 7.8

$\Sigma \Delta p_{tot}$ = sum of the individual losses (bar)
$\Sigma \Delta p_{RP}$ = sum of the pressure losses in straight pipe lengths (bar)
$\Sigma \Delta p_{RF}$ = sum of the pressure losses in fittings (bar)
$\Sigma \Delta p_{RV}$ = sum of the pressure losses in valves (bar)
$\Sigma \Delta p_{RC}$ = sum of the pressure losses in connections (bar)

- Pressure loss in straight pipe sections

The pressure loss in straight pipe sections is calculated using the following equation:

$$\Delta p_{RP} = \lambda_R \cdot \frac{L_P}{d_i} \cdot \frac{\rho_F}{2 \cdot 10^2} \cdot w^2$$

Equation 7.9

$\Delta\rho_{RP}$ = pressure loss in individual pipe sections (bar)
λ_R = pipe friction number (-)
L_P = length of the pipe sections (m)
d_i = inside diameter of the pipe (mm)
ρ_F = density of the flow (kg/m^3)
w = flow velocity (m/s)

- Pressure loss in pipe fittings

Given that substantial losses result from friction, changes of direction and transfer loss, these individual losses represent a significant portion of the total pressure loss in a pipe system. The pressure loss is calculated with the following formula:

$$\Delta\rho_{RF} = \zeta_{RF} \cdot \frac{\rho_F}{2 \cdot 10^5} \cdot w^2$$

Equation 7.10

$\Delta\rho_{RF}$ = pressure loss in individual pipe fittings (bar)
ρ_F = density of the flow (kg/m^3)
w = flow velocity (m/s)
ζ_{RF} = resistance factors for pipe fittings (-)

The resistance factors for fittings will be identified in tables 7.2 to 7.7.

- Pressure loss in pipe connections

Pressure loss also occurs in pipe connections, although it cannot be precisely characterised, as the geometric dimensions vary a great deal (e.g. welding beads). DVS 2210 Part 1 recommends basing pressure loss calculations for connections in plastic pipe systems (such as butt, sleeve and flange joints) on a resistance factor of ζ_{RF} = 0.1 (-) . For pressures losses in connections, this means:

$$\Delta\rho_{RC} = \zeta_{RC} \cdot \frac{\rho_F}{2 \cdot 10^5} \cdot w^2$$

Equation 7.11

$\Delta\rho_{RC}$ = pressure loss in connections (bar)
ρ_F = density of the flow (kg/m^3)
w = flow velocity (m/s)
ζ_{RC} = resistance factor for pipe connections = 0.1 (-)

To obtain a rough estimate of the pressure loss in connections, it is sufficient to consider an addition of 15% to the pressure loss in straight sections ζ_P and fittings ζ_{PF}.

- Pressure loss in valves

The pressure loss in valves can be determined using the following equation:

$$\Delta\rho_{RV} = \zeta_{RA} \cdot \frac{\rho_F}{2 \cdot 10^5} \cdot w^2$$

Equation 7.12

$\Delta\rho_{RV}$ = pressure loss in valves (bar)
ρ_F = density of the flow (kg/m^3)
w = flow velocity (m/s)
ζ_{RA} = resistance factors for pipe valves (-)

The resistance factors for valves are shown in table 7.10.

- Resistance factors

The pressure loss in a pipe system is caused by friction, changes of direction and transfer loss. To determine the variables involving total pressure loss in a pipe system requires knowledge of the resistance factors (ζ_F) for fittings, pipe connections and valves. The most important resistance factors are indicated below.

- ζ_F - values for seamless pipe bends

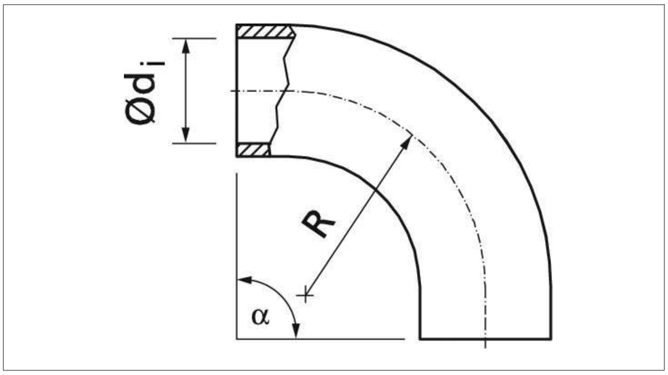

Figure 7.1 Seamless pipe bends

α \ R/d_i	1	1.5	2	4
30°	0.23	0.19	0.14	0.11
45°	0.34	0.27	0.20	0.15
60°	0.41	0.33	0.24	0.19
90°	0.51	0.41	0.34	0.23

Table 7.2 ζ_F value for seamless bends

- ζ_F values for segmented pipe bends

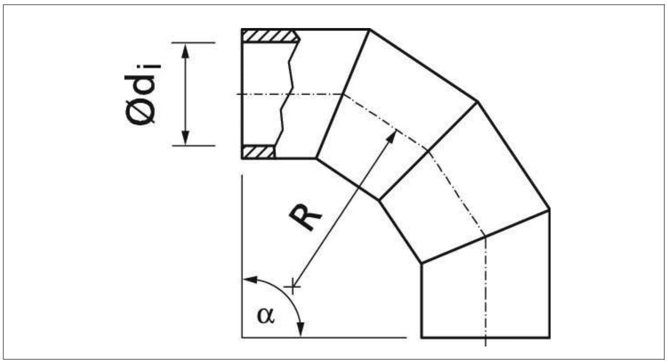

Figure 7.2 Segmented pipe bends

α \ R/d_i	1.5	3	4	6
30°	0.10	0.10	0.11	0.11
45°	0.14	0.15	0.16	0.17
60°	0.19	0.20	0.22	0.23
90°	0.24	0.26	0.28	0.29

Table 7.3 ζ_F values for segmented pipe bends

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Basic calculations

- ζ_F values for segmented pipe bends

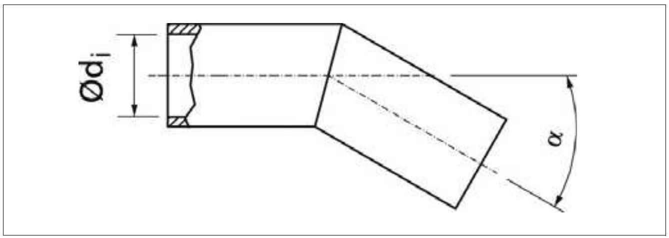

Figure 7.3 Elbows

α	10°	15°	20°	30°	45°
ζ_F	0.04	0.05	0.05	0.14	0.30

Table 7.4 ζ_F values for segmented pipe bends

- ζ_F values for tee sections

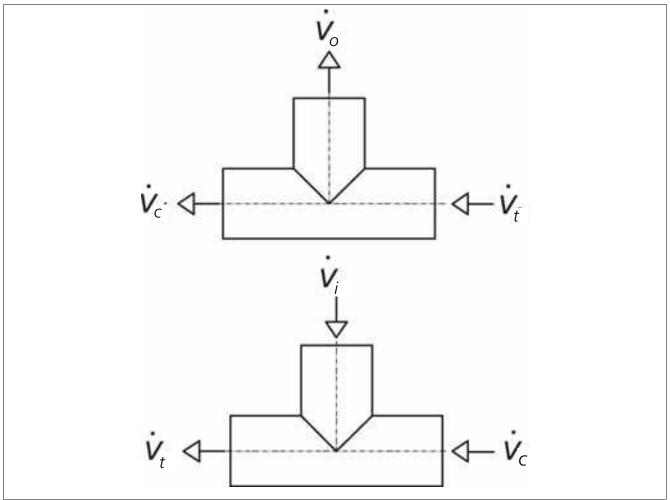

Figure 7.4 Tee for dividing and uniting flows

$\dot{V}_i/\dot{V}_t$	Uniting flows $\dot{V}_t=\dot{V}_o+\dot{V}_i$		$\dot{V}_o/\dot{V}_t$	Dividing flows $\dot{V}_t=\dot{V}_o+\dot{V}_c$	
	ζ_z	ζ_d		ζ_a	ζ_s
0.0	-1.20	0.06	0.0	0.97	0.10
0.2	-0.40	0.20	0.2	0.90	-0.10
0.4	0.10	0.30	0.4	0.90	-0.05
0.6	0.50	0.40	0.6	0.97	0.10
0.8	0.70	0.50	0.8	1.10	0.20
1.0	0.90	0.60	1.0	1.30	0.35

Table 7.5 ζ_F values for tees

$\dot{V}_t$ -> total flow volume (l/s - m^3/h)
$\dot{V}_c$ -> total continuous volume (l/s - m^3/h)
$\dot{V}_o$ -> outgoing volume (l/s - m^3/h)
$\dot{V}_i$ -> incoming volume (l/s - m^3/h)

Positive values indicate pressure loss, negative values pressure increases.

- ζ_F values for tanks and tank outlets

overlying -> ζ_F = 2.0
flush -> ζ_F = 0.3
rounded -> ζ_F = 0.05

- ζ_F values for cross-section changes

a) Pipe enlargement, perpendicular

d_{e1} / d_{e2}	1.2	1.4	1.6	1.8	2.0
ζ_F	0.07	0.15	0.24	0.46	0.56

Table 7.6 Pipe enlargement, perpendicular

b) Pipe constriction, perpendicular

d_{e1} / d_{e2}	1.2	1.4	1.6	1.8	2.0
ζ_F	0.10	0.22	0.29	0.33	0.35

Table 7.7 Pipe enlargement, perpendicular

c) Pipe enlargement, conical ζ value for λ_P = 0.025

	ζ_F values for angle of enlargement α_E		
d_{e2}/d_{e1}	4 - 8°	16°	24°
1.2	0.10	0.15	0.20
1.4	0.20	0.30	0.50
1.6	0.50	0.80	1.50
1.8	1.20	1.80	3.00
2.0	1.90	3.10	5.30

Table 7.8 Pipe enlargement, conical

α_E is the angle of enlargement. The ζ values are related to velocity (w_2). The conical length (l) is derived from the angle of enlargement (α_E) and the diameter ratio d_{e2}/d_{e1}. The best angle of enlargement in which the stream of the inflowing medium still does not lose contact with the pipe is about 7 to 8°.

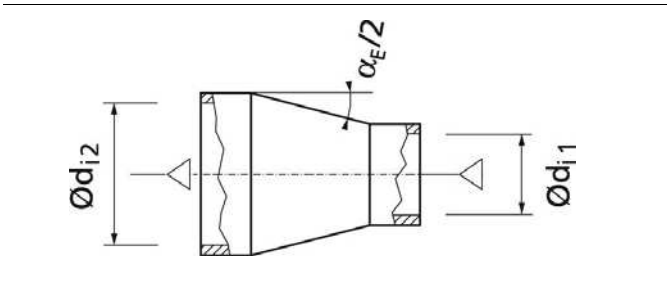

Figure 7.5 Conical enlargement of the flow cross section

d) Pipe enlargement, conical ζ value for λ_P = 0.025

	ζ_F values for angle of enlargement α_E		
d_{e2}/d_{e1}	4°	8°	24°
1.2	0.046	0.023	0.210
1.4	0.067	0.033	0.013
1.6	0.076	0.038	0.015
1.8	0.031	0.041	0.016
2.0	0.034	0.042	0.017

Table 7.9 Pipe enlargement, conical

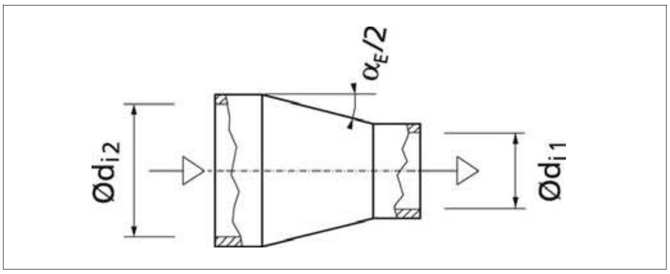

Figure 7.6 Conical enlargement of the flow cross section

- ζ_V values for valves

nominal diameter DN	diaphragm valve	flat-seat valve	ball valve	inclined seat valve valve without constriction	ball valve	butterfly valve	free-flow check valve	check valve
	Resistance factor (ζ)							
25	4.0	2.1	3.0	0.1...0.3	0.1...0.15	0.3...0.6	2,5	1,9
32	4.2	2.2	3.0	0.1...0.3	0.1...0.15	0.3...0.6	2.4	1.6
40	4.4	2.3	3.0	0.1...0.3	0.1...0.15	0.3...0.6	2.3	1.5
50	4.5	2.3	2.9	0.1...0.3	0.1...0.15	0.3...0.6	2.0	1.4
65	4.7	2.4	2.9	0.1...0.3	0.1...0.15	0.3...0.6	2.0	1.4
80	4.8	2.5	2.8	0.1...0.3	0.1...0.15	0.3...0.6	2.0	1.3
100	4.8	2.4	2.7	0.1...0.3	0.1...0.15	0.3...0.6	1.6	1.2
125	4.5	2.3	2.3	0.1...0.3	0.1...0.15	0.3...0.6	1.6	1.0
150	4.1	2.1	2.0	0.1...0.3	0.1...0.15	0.3...0.6	2.0	0.9
200	3.6	2.0	2.4	0.1...0.3	0.1...0.15	0.3...0.6	2.5	0.8

Table 7.10 ζ-values for valves

- ζ values for compensators

Surge suppressors : ζ = 2.0 per surge
Expansion joints : ζ = 3.0

- ζ values for welding seams

Electrofusion : ζ = 0.05
Butt-welding : ζ = 0.1

- ζ values for pipe bends in succession

180° bends : $\zeta = 2.0 \cdot \zeta$ 90°
Stage of 2 x 90° bends : $\zeta = 2.5 \cdot \zeta$ 90°
Expansion stage of 2 x 90° bends : $\zeta = 3.0 \cdot \zeta$ 90°

7.2.3 Investigating pressure surges

The change in the stationary conditions of a pipe system (e.g. due to activation of a butterfly valve or the failure of a pump) creates a pressure wave producing superpositions due to reflections in bends, reducers, etc. that can amount to many times the operating pressure. The largest pressure increase occurs when the closure procedure or stop command following pump failure falls under the so-called reflection time. If a pipe system contains externally controlled valves or trip mechanisms, it must be ensured that their closure time is larger than the reflection time.
If a pressure surge occurs, its effects must be investigated by means of a short-term stress analysis and a stability test.

Note:
The pressure surge is not always sufficiently expressed by the following equations. In particular, more precise investigations must be employed in longer or complexly branched pipe systems in order to consider and evaluate all the significant factors.

Significant factors are:
Operating pressure, pump pressure = p_O (bar)
Flow velocity = w_o (m/s)
E modulus of the pipe material = E_{cR} (N/mm^2)
(short-term E modulus = creep modulus between 1 and 10 min)
E modulus of the flow medium = E_F (N/mm^2)
(for water at 20°C -> $E_F \approx 2100$ N/mm^2)
Density of the flow medium = ρ_F (g/cm^3)

- Calculating pressure wave velocity (a)

$$a = \frac{a_0}{\sqrt{1 + 2 \cdot \frac{E_F}{E_{C_R}} \cdot \frac{\eta^2 + 1}{\eta^2 - 1}}}$$

Equation 7.13

The indicated resistance factors are reference values and are used in rough pressure loss calculations. Object related calculations are based on the data of the respected valve manufacturers.

$$a_0 = 31{,}3 \cdot \sqrt{\frac{E_F}{\rho_F}}$$

Equation 7.14

$$\eta = \frac{d_e}{d_i}$$

Equation 7.15

a = pressure wave velocity (m/s)
a_0 = speed of sound (m/s)
d_e = outside diameter of the pipe (mm)
d_i = inside diameter of the pipe (mm)
η = correction factor (-)
E_{cR} = creep modulus of the pipe material (N/mm^2)
E_F = E modulus of the flow medium (N/mm^2)
ρ_F = density of the flow medium (kg/m^3)

- Maximum pressure surge for $\Delta w = w_0$

$$p_{Surge} = p_0 \pm \frac{a}{100} \cdot \Delta w$$

Equation 7.16

a = pressure wave velocity (m/s)
p_{Surge} = pressure surge (bar)
p_O = pump pressure, operating pressure (bar)
Δw = $w_0 - w_1$ (with w_1 = flow velocity at the reflection position) (m/s)

For short pipes of a single length can be assumed that a surge degradation occurs.

$$L_{Pipe} \leq \frac{a \cdot t_s}{2}$$

Equation 7.17

t_s = closing time of the butterfly valve or delay of the pump stop command after failure (c)
L_{Pipe} = pipe length (m)
a = pressure wave velocity (m/s)

The surge degradation factor is:

$$Z_s = \frac{2 \cdot L_{Pipe}}{a \cdot t_s} \geq 1$$

Equation 7.18

Z_s = surge degradation factor (-)
L_{Pipe} = pipe length between valve mechanism or pump and reflection position (m)
t_s = closing time of the butterfly valve or delay for the pump stop command after failure (c)
a = pressure wave velocity (m/s)

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The complete pressure change ($\Delta\rho$) corresponding to the velocity change (Δ_w) velocity when the closing time (t_c) is smaller than the reflection time (t_R). The reflection time (t_R) is calculated by:

$$t_R = \frac{2 \cdot L_{Pipe}}{a}$$

Equation 7.19

t_R = reflection time (c)
L_{Pipe} = pipe length between valve mechanism or pump and reflection position (m)
a = pressure wave velocity (m/s)

The rough estimate of a pressure surge hazard can draw on the following rule of thumb:

$$K_s = \frac{L_{Pipe} \cdot w_0}{\sqrt{H_P}}$$

Equation 7.20

H_P = delivery height of the pump (m)
K_s = pressure wave size (-)
L_{Pipe} = pipe length between valve mechanism or pump and reflection position (m)
w_o = flow velocity (m/s)

For $K_s > 70$ and with simultaneous use of rapidly closing valves, a special pressure surge calculation is recommended.

- Calculating the pressure surge for the section involved $L_{Pipe} \leq 500 \cdot de$

Equation 7.21 is an empirical calculation formula. It should therefore only be used when rough estimates are to be made for simple pipe systems without fittings.

$$p_{Surge} = p_0 \cdot \left[1 + \frac{100 \cdot w \cdot p_F}{\varphi \cdot p_s} \cdot \left(1 + 0{,}2 \cdot \sqrt{p_0} \right) \right]$$

Equation 7.21

p_{Surge} = pressure surge (bar)
p = pump pressure, operating pressure (bar)
w = flow velocity (m/s)
ρ_F = density of the flow medium (g/cm^3)
φ = surge factor (-)

In general, the following values can be selected for φ:
φ = 25 for rapidly closing valves
φ = 50 for normally closing valves
φ = 75 for slowly closing valves

The loads often arising along with pressure surges due to vacuums can, under certain conditions, cause buckling of the pipe in specific situations. The stability (i.e. buckling resistance) of the pipe series in question must be verified in every case.

$$p_{Crit} = 2{,}9 \cdot E_{C_R} \cdot \left(\frac{2 \cdot e}{d_i} \right)^3$$

Equation 7.22

p_{Crit} = critical underpressure (bar)
E_{cR} = elasticity modulus (between 1…10 min) (N/mm^2), see appendix A2.
e = pipe wall thickness (mm)
d_i = inside diameter of the pipe (mm)

From this, it follows:

$$SF = \frac{p_{Crit}}{p_u} \geq 2$$

Equation 7.23

p_u = under pressure from pressure surge calculation (bar)
p_{Crit} = critical underpressure (bar)
SF = safety factor (-)

Note:
Appendices B9 to B10 indicate the acceptable external pressure load (p_{ex}) or the underpressure load (p_u) in short-term observations (t = 100 h) and long-term observations (t = 25a) for PE at various SDR levels.

7.2.4 Calculating acceptable stress

Calculation of acceptable stress requires the inclusion of various reduction factors. The significance of individual reduction factors will be discussed in section 7.2.6. The acceptable stress is determined by the following formula.

$$\sigma_{acc} = \frac{\sigma_{ref} \cdot f_{s,l}}{R_2 \cdot R_4 \cdot SF}$$

Equation 7.24

σ_{acc} = acceptable stress (N/mm^2)
σ_{ref} = reference stress (N/mm^2), see appendix A2.
f_l = long-term welding factor (-)
f_s = short-term welding factor (max. 1h) (-)
R_2 = reduction factor for the influence of the flow medium (see table 2.3) (-)
R_4 = reduction factor for the influence of the specific resilience of the material (-)
SF = safety factor for the respective loads (see table 7.13) (-)

7.2.5 Welding factors

The size of the welding factors depends on the specific welding procedures. There is also a distinction made between short-term (f_s) and long-term welding factors (f_l). Only long-term factors may be used in component calculations. The calculation of welding factors is performed in DVS 2203 part 2.
Table 7.11 contains the relevant factors for PE butt-weld (DVS 2203 part 1) and electrofusion (DVS 2212 part 1).
The original abbreviations f_s and f_l have been freely chosen by Akatherm.

Material	Welding factor	PE
Butt-welding	f_s	0.9
(BW)	f_l	0.8
Electro-fusion	f_s	0.9
(EF)	f_l	0.8

Table 7.11 Welding factors f_s and f_l

7.2.6 Reduction factors

- Reduction factor R_1

The reduction factor R_1 (time dependent) accounts for the dependence of material strength on stress duration. This value is displayed in the creep rupture curves (appendix A1). Use of this diagram supplants direct observation.

- Reduction factor R_2

This reduction factor designates the influence of the flow medium (chemical resistance factor) on the creep strength of thermoplastic. Relevant values can be taken from table 2.3 in chapter 2.

- Reduction factor R_3
The dependence of strength on the operating temperature is accounted for in reduction factor (R_3). This value is also displayed in the creep rupture curves (appendix A1). Use of this diagram supplants direct observation.

- Reduction factor R_4
This factor accounts for the specific resilience of the material as a function of the operating temperature and is therefore derived from impact strength values. The corresponding values for reduction factor (R_4) are found in table 7.12.

Material	Pipe wall temperature			
	-10°C	20°C	40°C	60°C
PE100	1.2	1	1	1

Table 7.12 Reduction factor R_4

Factors can be interpolated for intermediary temperatures.

7.2.7 Safety factor (SF)

Load Class		
I	Even load on the pipe in installations in building. If damage occurs, no possible danger to people, property and the environment.	1.3
II	Load on the pipe under changing operating conditions in installations outside buildings. If damage occurs, no damage to people is expected. Effects on property and the environment limited. (*)	1.8
III	Load on the pipe under changing operating conditions in installations inside and outside buildings. If damage occurs, there are possible hazards for people, property and the environment. Applied for monitoring and inspection facilities.	≥2.0

Table 7.13 Safety factors (SF) for various loads

(*) In contrast to general references in the technical literature (e.g. DVS 2205 part 1) in which only two loads are distinguished, Akatherm has introduced this additional load class for larger loads that could affect a pipe system in an installation outside buildings.

7.2.8 Minimum wall thickness

- Straight pipes

$$e_0 = \frac{d_e \cdot p_i}{20 \cdot \sigma_{acc} + p_i}$$

Equation 7.25

e_0 = minimum wall thickness (mm)
d_e = outside diameter of the pipe (mm)
p_i = Internal pressure (bar)
σ_{acc} = acceptable stress (N/mm^2)

- Seamless and welded segmented pipe bends
For the inside of bends, the following holds:

$$e_i = \frac{d_e \cdot p_i}{20 \cdot \sigma_{acc} + p_i} \cdot f_{Bi}$$

Equation 7.26

e_i = wall thickness on the inside of the bend (mm)
f_{Bi} = factor for the inside of the bend from table 7.14 (-)
p_i = Internal pressure (bar)
d_e = outside diameter of the pipe (mm)
σ_{acc} = acceptable stress (N/mm^2)

For the outside of bends, the following holds:

$$e_o = \frac{d_e \cdot p_i}{20 \cdot \sigma_{acc} + p_i} \cdot f_{Bo}$$

Equation 7.27

e_o = wall thickness on the outside of the bend (mm)
f_{Bo} = factor for the outside of the bend from table 7.14 (-)
p_i = internal pressure (bar)
d_e = outside diameter of the pipe (mm)
σ_{acc} = acceptable stress (N/mm^2)

- Factors (f_{Bi}) and (f_{Bo}) for determining wall thicknesses of bends

		Radius R			
		$1{,}0 \cdot d_e$	$1{,}5 \cdot d_e$	$2{,}0 \cdot d_e$	$2{,}5 \cdot d_e$
Seamless pipe-bends	f_{Bi}	1,27	1,22	1,15	1,12
	f_{Bo}	0,92	0,93	0,95	0,96
Segmented bends	f_{Bi}	1,59	1,50	1,44	1,40
	f_{Bo}	1,15	1,16	1,19	1,20

Table 7.14 Factors for determining wall thicknesses for bends

- Welded branches (tees)
In principle, branches in pipes represent weaknesses.
This condition has the consequence that unreinforced welded tee sections do not correspond to the nominal pressure of the pipe. Welded tees can only be exposed to a reduced pressure load. An improvement of the internal load capacity is achieved by increasing the wall thickness of the connection or by a corresponding strengthening of welded joints.

- Wall thickness of branch pipes:

$$e_0 = \frac{d_e \cdot p_i}{(20 \cdot \sigma_{acc} + p_i) \cdot W_{RF}}$$

Equation 7.28

e_o = wall thickness (mm)
d_e = outside diameter of the pipe (mm)
p_i = internal pressure (bar)
σ_{acc} = acceptable stress (N/mm^2)
W_{RF} = weakening factor of the fitting (-)

Values for the weakening factor (WRF) can be interpreted from appendix B2.

The reduced internal loads for fittings welded on pipe segments is shown in appendix B2.

7.2.9 Internal pressure capacity of welded and injection-moulded fittings.

- Moulded fittings
Moulded fittings do not generally require any restrictions regarding a minimum internal pressure load capacity. They therefore withstand the same internal pressure load as a pipe of the same SDR level.

- Segment-welded tee sections
The inclusion of unreinforced welded tees in any pipe system reduces the tolerable operating pressure. The corresponding reduction factor is strongly temperature-dependent, which means that the internal pressure capacity is lowered by rising operating temperatures. Information about the internal pressure load capacity of segment-welded tees is provided by appendix B2.

Basic calculations

- Segment-welded bends

Like the mentioned segment-welded tees, segment-welded bends (angle >30º) have a reduced internal pressure load capacity in comparison with a straight pipe. The weakening effect of butt-welded segmented bends is also dependent on the same temperature conditions affecting segment-welded tees. The actual internal pressure load capacities can be seen in appendix B2.

7.2.10 Length changes due to heating and internal pressure

- Thermal-conditioned length change

Exposing a pipe system to varying temperatures (e.g. ambient and operating temperatures) changes its status with regard to the expansion capacities of individual pipe sections. Pipe sections designate the distances between the system point in question to the relevant fixed point. The thermally conditioned length change (Δl_ϑ) of individual pipe sections is calculated as follows:

$$\Delta l_\vartheta = L_{Pipe} \cdot \alpha_\vartheta \cdot \Delta\vartheta_{max} \cdot 10^3$$

Equation 7.29

Δl_ϑ = thermal length change (mm)
L_{Pipe} = pipe length (m)
α_ϑ = thermal expansion coefficient (K-1)
$\Delta\vartheta_{max}$ = max. temperature difference (K)

Examples for the calculation of $\Delta\vartheta_{max}$:
$\Delta\vartheta_{max}$ = $T_{W\,max}$ - $T_{W\,min}$
or
$\Delta\vartheta_{max}$ = $T_{W\,max}$ - T_M
or
$\Delta\vartheta_{max}$ = $T_{W\,max}$ - T_a
$\Delta\vartheta_{max}$ = temperature difference (K)
$T_{W\,min}$ = minimum pipe wall temperature (°C)
T_M = average installation temperature (°C)
$T_{W\,max}$ = maximum pipe wall temperature (°C)
T_a = lowest ambient temperature (°C)

- Length change due to internal pressure loads

In any closed pipe string, a longitudinal expansion is produced by the occurrence of internal pressure loads as follows:

$$\Delta l_\rho = \frac{0{,}1 \cdot p_i}{E_{c_R}} \cdot \frac{(1-2\cdot\mu)}{\left(\frac{d_e}{d_i}\right) - 1} \cdot L_{Pipe}$$

Equation 7.30

Δl_ρ = length change due to internal pressure (mm)
p_i = internal pressure (bar)
E_{cR} = creep modulus of the pipe material (N/mm^2)
d_e = outside diameter of the pipe (mm)
d_i = inside diameter of the pipe (mm)
L_{Pipe} = pipe length (L_1, L_2,···, L_x) (mm)
μ = transverse contraction number (-)

- Determining expansion bend dimensions

Length change (Δl_ϑ):

$$\Delta l_\vartheta = L_{Pipe} \cdot \alpha_\vartheta \cdot \Delta\vartheta \cdot 10^3$$

Equation 7.31

L_{Pipe} = pipe length (m)
α_ϑ = thermal expansion coefficient (K-1)
Δ_ϑ = temperature difference (K)
Δl_ϑ = thermal length change (mm)

In calculating temperature difference ($\Delta\vartheta$), care should be taken that the lowest and highest temperatures (installation, operating and stationary) are used in the calculation. Approximations usually take just flow medium and ambient temperatures into account. They simplest type of compensation for length changes in thermoplastic pipe installations involves using L-form expansion bends (90° changes of direction). L-bends are also referred to as elbows. The minimum dimensions of an elbow as illustrated in figure 7.7 is indicated by the following equation:

$$L_{B1,2} = \sqrt{\frac{3 \cdot d_e \cdot L_{1,2} \cdot \varepsilon \cdot E_{c_R}}{\sigma_{sf\,acc}}}$$

Equation 7.32

L_B = length of the expansion bend (mm)
$L_{1,2}$ = system lengths of expansion bends (mm)
d_e = outside diameter of the pipe (mm)
ε = expansion (-)
E_{cR} = (average) bending creep modulus of the pipe material for t = 25a (N/mm2)
$\sigma_{sf\,acc}$ = acceptable flexural stress share for t = 25a (N/mm^2)

Expansion bend lengths for L, Z and U bends are illustrated in appendices B5 to B8, and the corresponding values can be read from these diagrams.

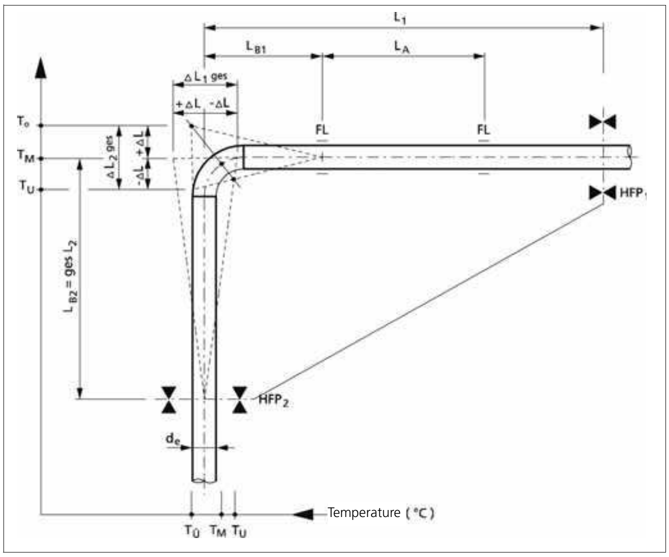

Figure 7.7 Elbow (L-bend)

- Pipe sections for absorbing length changes

As shown in figure 7.8, length changes subject pipe sections to bending. The dimensions of expansion bends (L_B) are established by the relationship:

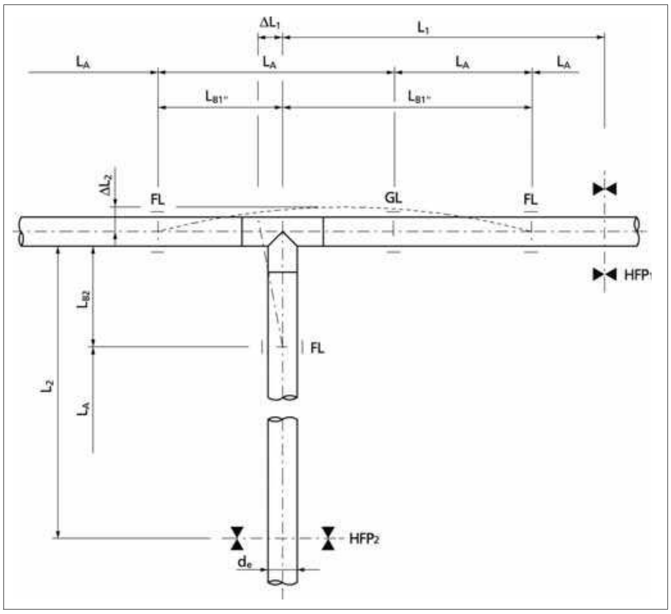

Figure 7.8 Pipe section for absorbing length changes

$$L_B = \sqrt{\frac{3 \cdot d_e \cdot L_{1,2} \cdot \varepsilon \cdot E_{C_R}}{\sigma_{f\,acc}}}$$

Equation 7.33

L_B = length of the expansion bend (mm)
d_e = outside diameter of the pipe (mm)
ε = expansion (-)
$L_{1,2}$ = system lengths of expansion bends (mm)
E_{cR} = (average) bending creep modulus of the pipe material for t = 25a (N/mm^2)
$\sigma_{f\,acc}$ = acceptable flexural stress (N/mm^2)

- Branch connections

In pipe sections subject to axial movement, it is also important to ensure that branching pipe strings are not exposed to high flexural tensions. The minimum distance between a tee section and the nearest guide bracket corresponds to the expansion bend length (L_B). With the aid of appendices B5 to B8, the required expansion bend lengths are visually illustrated.

7.2.11 Calculating forces at fixed points

Fixed points in pipe installations are brackets that fix the pipe in x, y and z directions. The forces at these points depend on the nature of individual pipe system. The "slacker" the pipe running between two fixed points is, the smaller the reaction forces generated by deformation effects. The exact calculation of the force components in any given pipe system is laborious. The calculation is made easier by employing the correspondingly suitable PC software for thermoplastic pipe systems.

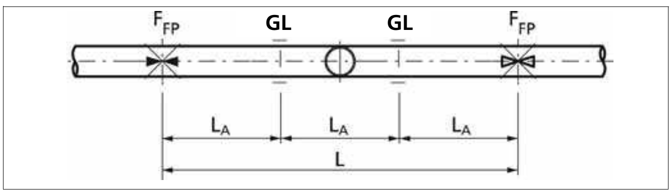

Figure 7.9 Firmly fastened pipe string

- Maximum force at fixed-points without compensation for length change

The largest fixed-point load occurs on a firmly fastened pipe string. It is calculated as follows:

$$F_{FP} = A_p \cdot \varepsilon \cdot E_{C_R}$$

Equation 7.34

Load from termal expansion: $\varepsilon = \alpha_\vartheta \cdot \Delta\vartheta$ (35)

Load from internal pressure: $\varepsilon = \dfrac{0,1 \cdot p_i \cdot (1-2 \cdot \mu)}{E_{C_R} \cdot \left(\dfrac{d_e^2}{d_i^2} - 1\right)}$ (36)

Load from swelling: $\varepsilon = 0,025 \ldots 0,040$ (37)

Equation 7.35-7.37

Explanation of symbols in equations 7.34 to 7.37
F_{FP} = force at fixed points in a firmly fastened pipe string (N)
A_P = area of pipe annular surface (mm^2)
ε = constricted length expansion (-)
α_ϑ = thermal expansion coefficient (K-1)
$\Delta\vartheta$ = maximum temperature difference between TL and TU (K)
p_i = internal pressure (bar)
E_{cR} = (average) creep modulus of the pipe material for t = 100 min (N/mm^2)
d_e = outside diameter of the pipe (mm)
d_i = inside diameter of the pipe (mm)
μ = transverse contraction number (-)

Since the creep modulus is dependent on time, temperature and stress, an average load period of 100 min is adopted for the calculations. The corresponding values can be derived from appendix A2. The forces at fixed points can also be interpolated from the appendix B9.

- System-dependent fixed-point force

Thermoplastic pipes are usually installed so that the compensation capacity of direction changes can be used to absorb length changes.
The resulting fixed-point forces are therefore system-specific. In most cases, they are smaller than the fixed-point load calculated in section 7.2.14. In practice however, the procedure is simplified by adopting the largest possible load to design the fixed-point structure. If, due to the load compensation parameters, the forces are prevented from being large enough (e.g. for pipe bridges, brickwork, containers, pump supports), a system-specific calculation is an essential part of a structural calculation. The PC calculation program can again serve this purpose.

Basic calculations

- Forms of compensation in level pipe systems

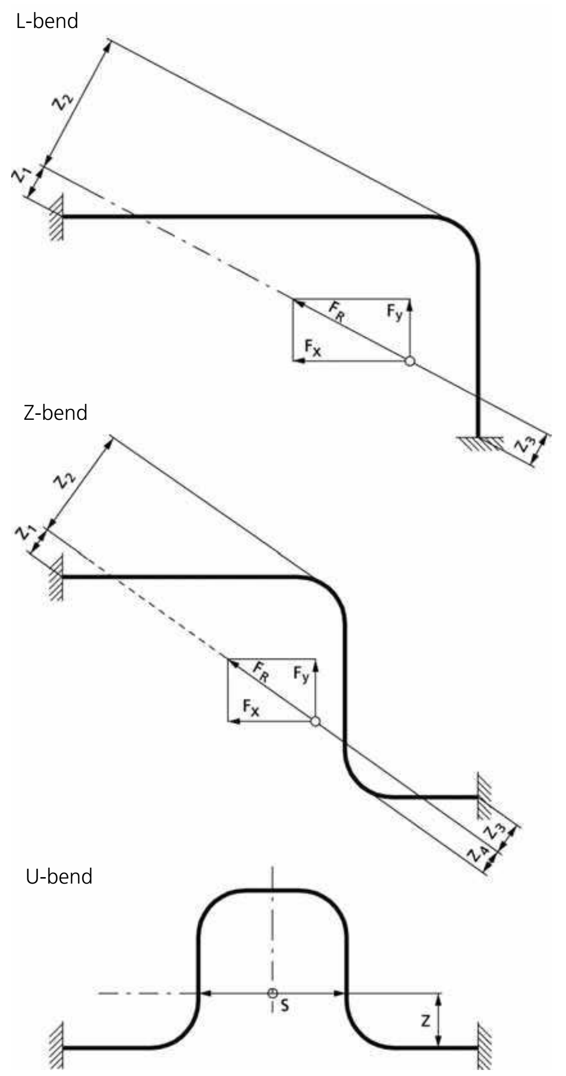

Figure 7.10 Forms of compensation in level pipe systems

Explanation of figure 7.10:

F_x, F_y, F_z = force components
F_y, F_x = F_T at fixed points "A" und "B"
F_T = transverse force
F_R = resulting total force
Z_1, Z_2, Z_3 = distances of the system points from the line of action
S = system centre of mass

The distance (Z) (lever) and the resulting total force (F_R) produces the moment (M)

$$M = F_R \cdot Z$$

Equation 7.38

at a given position of the individual pipe system

M = moment of torque (Nm)
F_R = resulting force (N)
Z = lever (m)

- Rough calculation of fixed-point forces in L-bends:

$$F_{LB} = \frac{12 \cdot \Delta l \cdot E_{c_R} \cdot J_P}{L_B^3}$$

Equation 7.39

F_{LB} = fixed-point load at (L, Z, U) bends -> F_y or F_x (N)
Δl = length change (mm)
E_{cR} = (average) creep modulus of the pipe material for t = 100 min (N/mm²)
J_P = pipe moment of inertia (mm⁴)
L_B = length of the expansion bend (mm)

$$J_P \approx \frac{d_e^4 - d_i^4}{20}$$

Equation 7.40

J_P = pipe moment of inertia (mm⁴)
d_e = outside diameter of the pipe (mm)
d_i = inside diameter of the pipe (mm)

- Fixed-point force in cases of constricted length change due to internal pressure load

$$F_{FP} = A_P \cdot \sigma_a \quad \text{with} \quad A_P = \frac{\left(d_e^2 - d_i^2\right) \cdot \pi}{4}$$

Equation 7.41

σ_a = axial stress (N/mm²)
A_P = area of pipe annular surface (mm²)
F_{FP} = force at fixed point (N)
d_e = outside diameter of the pipe (mm)
d_i = inside diameter of the pipe (mm)

- Fixed-point force in cases involving compensators or expansion couplers

$$F_{FP} = 0{,}1 \cdot A_K \cdot p_i \quad \text{with} \quad A_K = \frac{d_i^2 \cdot \pi}{4}$$

Equation 7.42

A_K = surface subject to pressure (mm²)
p_i = internal pressure (bar)
d_i = inside diameter of the pipe (mm)
F_{FP} = force at fixed point (N)

- Resulting force at fixed point

Brackets are subject to both horizontal and vertical loads. Simultaneous loads in both directions produce a resulting total force (F_R).

$$F_R = \sqrt{F_B^2 + F_V^2}$$

Equation 7.43

F_B = axial fixed-point force (F_{FP}) or friction force (F_μ) as a result of a length change in the pipe string
F_V = transverse force (F_{TV}) for fixed-point load and/or pipe weight (F_W) between two brackets with or without additional load.

7.2.12 Elasticity test

Elasticity tests on pipe systems serve predominantly to calculate force actions and stresses.
Stresses are created by:
- internal pressure loads
- constricted longitudinal movement under temperature change
- bending of bend sections when absorbing length changes

The forces affecting fixed points do not only cause bending moments in the bend sections but also in the firmly fastened pieces. In 3-D systems, the pipe string is also subject to torsion (twisting).
For facilities corresponding to load classes II and III (table 7.13) or requiring testing or monitoring in general, the elasticity calculation is not only a criterion for careful project planning but also partly prescribed.

7.2.13 Calculating the stresses arising in a pipe system

Pipe systems are usually exposed to a multi-axial state of stress. The mechanical stress measured in the internal pressure creep test (appendix A1) provides the basis for designing thermoplastic pipe systems. The layout of pipes is generally based on the internal pressure load, which corresponds to the load in the internal pressure creep test.

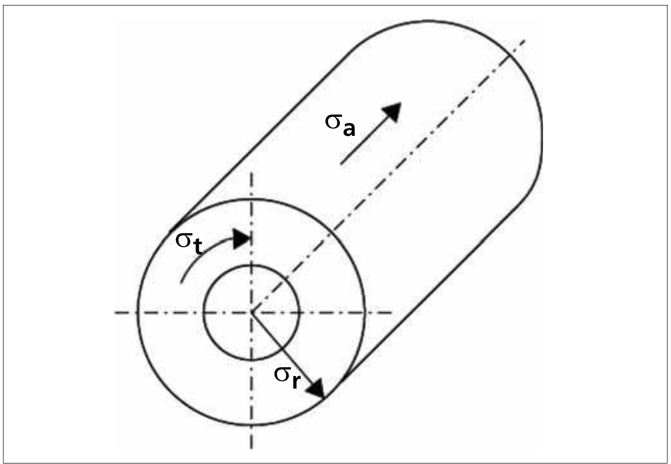

Figure 7.11 Radial (σ_r), Axial (σ_a) or Tangential (σ_t) stress

However, in addition to the mentioned stresses in axial, radial and tangential directions to the pipe axis, which are usually caused by heat or internal pressure loads, there are other stresses (e.g. flexural stresses) created in a pipe system, making it necessary to investigate all the stress components as part of a structural analysis. To establish the multi-axial state of stress requires calculation of the resulting stress (σ_{res}) and comparison with the value for (σ_{acc}). In particular, failure to account for flexural stresses around expansion bends and the tensile stresses resulting from impeded thermal expansion can result in failure of a pipe that is, otherwise, subject to a small internal pressure load.

- Axial stress (X axis) due to internal pressure

$$\sigma_a = \frac{p_i}{10} \cdot \frac{1}{\left(\frac{d_e}{d_i}\right)^2 - 1} \mathrel{\hat{=}} \sigma_1$$

Equation 7.44

- Tangential stress (Y axis) due to internal overpressure

$$\sigma_t = \frac{p_i}{10} \cdot \frac{\left(\frac{d_e}{d_i}\right)^2 + 1}{\left(\frac{d_e}{d_i}\right)^2 - 1} \mathrel{\hat{=}} \sigma_y$$

Equation 7.45

- Radial stress (Z axis due to inner over pressure)

$$\sigma_r = -\frac{p_i}{10} \mathrel{\hat{=}} \sigma_z$$

Equation 7.46

- Flexural axial stress (x axis) in a straight pipe between brackets

$$\sigma_f = \frac{q \cdot L_B^2}{8 \cdot W_R} \mathrel{\hat{=}} \sigma_x$$

Equation 7.47

- Flexural stress in bends

$$\sigma_f = \sigma_0 \cdot \frac{2}{1+12 \cdot \lambda_B^2} \cdot \left[9 \cdot \lambda_B + (1+6 \cdot \lambda_B^2) \cdot \frac{r_a}{R} \right]$$

Equation 7.48

M_b from the elasticy test

$$\sigma_0 = \frac{M_b \cdot d_e}{20 \cdot J_P}$$

Equation 7.49

$$\lambda_B = \frac{R \cdot e}{r_a^2}$$

Equation 7.50

$$r_m = \frac{d_e - e}{2}$$

Equation 7.51

- Stresses due to constricted thermal expansion

maximum stress value in a short-term perspective:

$$\sigma_{\vartheta max} = \pm \alpha_\vartheta \cdot \Delta\vartheta \cdot E_{C_R} \mathrel{\hat{=}} \sigma_x$$

Equation 7.52

in a long-term perspective:
Stress for temperature = constant

$$\sigma_\vartheta = 0{,}5 \cdot \sigma_{\vartheta max}$$

Equation 7.53

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Basic calculations

stress for temperature = variable

$$\sigma_{\vartheta} = 0{,}67 \cdot \sigma_{\vartheta max}$$

Equation 7.54

Tensile stress arises when the period of low operating temperature(t_{cold}) is shorter than the period of high operating temperature (t_{warm}) , i.e. $t_{cold} < t_{warm}$

$$\sigma_z = -0{,}33 \cdot \sigma_{\vartheta max}$$

Equation 7.55

- Residual stress due to temperature differences between inner pipe wall and outer pipe wall

$$\sigma_{\vartheta} = 0{,}81 \cdot E_{C_R} \cdot \alpha_{\vartheta} \cdot (T_{S_P} - T_{e_P})$$

Equation 7.56

- Calculating the resulting stress

$$\sigma_{res} = 0{,}71 \cdot \sqrt{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2}$$

Equation 7.57

$$\sigma_x \leq \sigma_{acc} = \sigma_a + \sigma_f \pm \sigma_{\vartheta}$$

Equation 7.58

$$\sigma_y \leq \sigma_{acc} = \sigma_u$$

Equation 7.59

$$\sigma_y \leq \sigma_{acc} = \sigma_r$$

Equation 7.60

$$\sigma_{res} \leq \sigma_{acc}$$

Equation 7.61

Explanation of symbols in equations 7.44 - 7.61

R_p = moment of resistance of the pipe (cm^3)
M_b = bending moment due to the tensile or compressive forces affecting branch sections
L_B = bracket distance (m)
J_P = pipe moment of inertia (cm^4)
R = bending radius of bends (mm)
r_a = average pipe radius (mm)
q = weight of the full pipes (N/m)
e = pipe wall thickness (mm)
d_e = outside diameter of the pipe (mm)
d_i = inside diameter of the pipe
E_{cR} = average creep modulus of the pipe material (N/mm^2)
ρ_i = internal pressure (bar)
σ_0 = stress in straight pipe (N/mm^2)
σ_a = axial stress (N/mm^2)
σ_t = tangential stress (N/mm^2)
σ_r = radial stress (N/mm^2)
σ_f = flexural stress (N/mm^2)
σ_{ϑ} = axial stress due to constricted thermal expansion (N/mm^2)
$\sigma_{\vartheta max}$ = maximum axial stress due to constricted thermal expansion (N/mm^2)
σ_{res} = resulting stress (N/mm^2)
$\sigma_{x, y, z}$ = direction-dependent stress (N/mm^2)
σ_{acc} = acceptable stress (N/mm^2)
T_{sP} = inner pipe wall temperature (°C)
T_{eP} = outer pipe wall temperature (°C)
λ_B = bend factor (-)
a_{ϑ} = thermal expansion coefficient (K-1)

7.2.14 Expansions

If pipe deformation exceeds certain creep limits, flow zones and microscopic cracks are created across the direction of expansion. For usual applications, pipe deformation is calculated in the following manner.

$$\varepsilon_{\sigma x} = \frac{100}{E_{C_R}} \cdot \left[\sigma_x - \mu \cdot (\sigma_y - \sigma_z)\right]$$

Equation 7.62

$$\varepsilon_{\sigma y} = \frac{100}{E_{C_R}} \cdot \left[\sigma_y - \mu \cdot (\sigma_x + \sigma_z)\right]$$

Equation 7.63

$\varepsilon_{\sigma x}$, $\varepsilon_{\sigma y}$ = expansion along the x and y axes of stress in a multi-axial load (-)
σ_x, σ_y, σ_z = stress along the x, y and z axes (N/mm^2)
E_{cR} = average creep modulus of the pipe material (N/mm^2)
μ = transverse contraction number = 0.38 (-)

The largest expansion is calculated on each occasion and compared with the threshold value (ε_F) in table 7.15. The requirement reads:

$$\frac{\varepsilon_F}{\varepsilon_{max}} \geq 1{,}3$$

Equation 7.64

ε_F = expansion threshold value
ε_{max} = $\varepsilon_{\sigma x}$ or $\varepsilon_{\sigma y}$ (-)

Material	PE
expansion threshold value ε_F	3%

Table 7.15 Expansion threshold value for PE

Should the correlation between stress and expansion be expressed, it can also be included in the following relationship:

$$\sigma_0 = \varepsilon \cdot E_{C_R} \text{ resp. } \varepsilon = \frac{\sigma_0}{E_{C_R}} \cdot 100\%$$

Equation 7.65

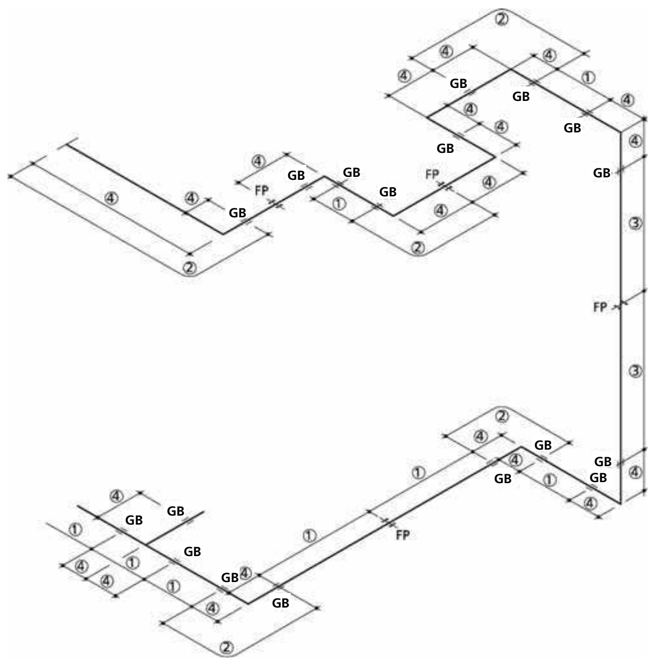
E_{cR} = average creep modulus of the pipe material (N/mm^2)
ε = expansion (-)
σ_0 = stress (N/mm^2)

7.2.15 Estimating service life

If the calculation variables, temperatures and their reaction time in relation to the assumed total load period are all available, the expected service life (life-span) can then be determined. The calculation of the service life however requires a large amount of mathematical complexity. For this reason, closer examination of the mathematical determination of a pipe system's service life will be set aside. In calculating service life, it should be noted that the life-span of the pipe system is also affected by chemical and/or physical conditions. A detailed treatment of the topic of service-life, complete with examples, is contained in the DVS 2205-1 guideline. The calculation of service lives involving varying loads requires firm knowledge about the handling of plastics. Valuable aids are PC calculating programmes, which can also provide support for less frequently used calculations.

7.2.16 Sample calculation

A number of values will be calculated for the isometric pipe system illustrated below.



GB = guide bracket (guide saddle), which is a fixing with an axial pipe guide

FP = fixed point, which is a fixing without any possibility for movement

① L_M = support distance for the middle of a continuous pipe

② L_E = support distance for a separate or end piece, as well as the self-supporting total pipe length of an L bend. If both bending lengths are, in total, longer than the acceptable support distance (L_F), an additional SB is to be placed between the pipe bend and the GB

③ L_V = guide bracket distance for an inclined pipe string. To absorb weight, a fixed point or bearing fixed point or support ring is to be provided for each string

④ L_B = bending lengths for absorbing length changes and operating conditions

Figure 7.12 Pipe isometrics

Explanation of terms ① ②③ ④ for equations 7.66 to 7.73. The numbering and short symbols provide information about the variables being calculated and where their location in the isometric diagram. The illustration is therefore a guide and helps the user to obtain an overview of a pipe system and its calculations.

1. Load on brackets

Brackets are subject to both horizontal and vertical loads. Simultaneous loads produce a resulting total force (F_R), which can be calculated using the following equation 7.66. Figure 7.13 shows a bracket with the effective forces and the calculated bracket distance (B_d).

$$F_R = \sqrt{F_B^{\,2} + F_V^{\,2}}$$

Equation 7.66

FB = axial fixed-point force (F_{FP}) or friction force (F_μ) as a result of a length change in the pipe string

FV = transverse force (F_{TV}) for fixed-point load and/or pipe weight (F_W) between two brackets with or without additional load

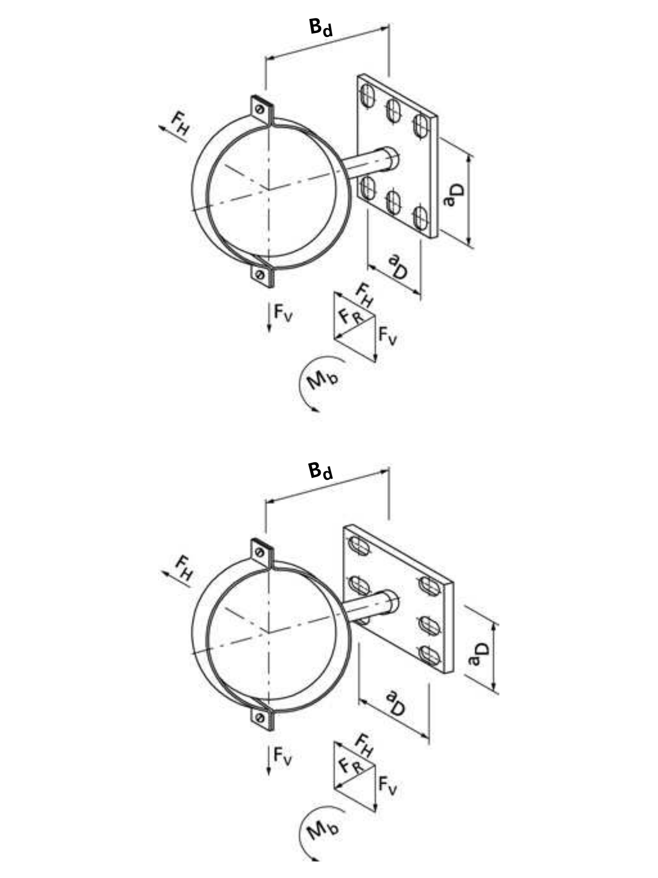


Figure 7.13 Pipe saddle load

Determination of the force components

F_{FP} = see equation 7.34

F_μ = $F_W \cdot \mu$

F_{TV} = based on system calculation

$F_W = F_{W\,(Pipe)} + F_{W\,(Fill\,med)} + F_{W\,(Add\,load)}$

Equation 7.67

μ_F	=	friction coefficient usually between 0.3 and 0.5
F_{FP}	=	force at fixed point (N)
F_W	=	weight (N)
F_{TV}	=	transverse force (N)
$F_{w\,(Pipe)}$	=	weight (pipe weight) (N)
$F_{W\,(Fill\,med)}$	=	weight (weight of fill medium) (N)
$F_{W\,(Add\,load)}$	=	weight (additional weight) (N)

2. Bracket distances (B_d)

The fixing distances of a plastic pipe are to be determined in such a way that no excessive stresses arise both under operating conditions and during testing. Similarly, consideration is also to be given to the deflection limits of the pipe. The arrangement of various brackets and their distances from each other can be seen in the illustrated example (figure 7.12) of pipe isometrics.

Basic calculations

3. Calculating acceptable bracket distances

$$L_M = 1{,}86 \cdot \sqrt[4]{\frac{E_{C_R} \cdot f_{acc} \cdot \left(d_e^4 - d_i^4\right) \cdot 10^5}{\left(d_e - d_i\right) \cdot e \cdot \rho_P + \frac{d_i^2}{2} \cdot \rho_F}}$$

Equation 7.68

L_M = support distance for the middle of a continuous pipe (mm)
E_{cR} = creep modulus of the pipe material for t = 25a (N/mm^2)
f_{acc} = acceptable pipe deflection according to the recommendations in table 7.16 (mm)
d_e = outside diameter of the pipe (mm)
d_i = inside diameter of the pipe (mm)
e = pipe wall thickness (mm)
ρ_P = density of the pipe medium (g/cm^3)
ρ_F = density of the flow medium (g/cm^3)

$$L_E = 0{,}67 \cdot L_M$$

Equation 7.69

$$L_V = Y \cdot L_M$$

Equation 7.70

Values for Y: $\leq \varnothing d_e$ 110 mm -> Y=1,1
$\varnothing d_e$125 - 200 mm -> Y=1,2
$\leq \varnothing d_e$ 225 mm -> Y=1,3

L_B = mathematical definition in equation 7.28 respectively with appendices B3 to B7

Equation 7.71

Explanation of symbols in equations 7.69 to 7.71:
L_M = support distance for the middle of a continuous pipe (mm)
L_E = support distance for a separate and end piece, as well as the self-supporting total pipe length of an L bend.
L_V = guide distance for an inclined pipe string (mm)
L_B = length of the expansion bend (mm)
Y = factor (dependent on pipe dimensions) (-)

To calculate the support distances, the following deflections are recommended as acceptable values:

$\varnothing d_e$	20 - 110	125 - 200	225 - 355	400 - 600
f_{acc}	2 -3 mm	3 - 5 mm	5 - 7 mm	7 - 10 mm

Table 7.16 Deflection f_{acc} for pipes

If larger deflections are permitted, the pipe should not be axially constricted.

4. Verifying the acceptable buckling length

If pipes are installed so that axial expansion of all or individual strings is no longer possible (axial constriction), the calculated fixing distance must be tested for its buckling resistance. To avoid the risk of buckling due to constricted thermal expansion, the length of pipe between two brackets must be no more than L_K .

$$L_{K_{acc}} \leq 1{,}12 \cdot \sqrt{\frac{d_e^2 - d_i^2}{\alpha_\vartheta \cdot \Delta\vartheta}}$$

Equation 7.72

See appendix B10.

$L_{K\,acc}$ = acceptable buckling length between two brackets (mm)
α_ϑ = thermal expansion coefficient (K^{-1})
Δ_ϑ = temperature difference (K)
d_e = outside diameter of the pipe (mm)
d_i = inside diameter of the pipe (mm)

The following applies to all pipe systems without linear compensation:

$$L_{B_{acc}} \leq L_{S_{acc}}$$

Equation 7.73

Values for LB are found in appendix B10:
$L_{B\,acc}$ = acceptable pipe length between two brackets (mm)
$L_{S\,acc}$ = available or calculated support distance according to equations ① ② ③ (equations 7.68 to 7.70)
d_e = outside diameter of the pipe (mm)
d_i = inside diameter of the pipe (mm)
α_ϑ = thermal expansion coefficient (K^{-1})
Δ_ϑ = temperature difference (K)

Note to equations 7.72 and 7.73:
Axially constricted pipe systems operated at raised temperatures or an expected reduction of the creep modulus (E_{cR}) as a result of chemical effects will both give rise to risks of buckling.
Raised operating temperatures are:
PE -> $T_{crit} \geq 45°C$
PP -> $T_{crit} \geq 60°C$
The buckling risk is strengthened due to bending along the pipe axis or insufficient pipe storage practices. Bending can result from too long support distances, improper storage of pipe and lasting pipe impacts during welding.
In such case, it is recommended that the buckling distances either calculated or interpolated from appendix B10 are reduced by a factor of 0.8.
Pipe systems with $d_e \leq 50$ mm should be equipped with continuous support for economic reasons.

5. Determining pipe deflection for calculated support distances

Pipe deflection for the calculated support distance (L_M) (from equation 7.68) is determined by means of the following equation.

Pipe deflection for the calculated support distance (L_E) (from equation 7.69) is also determined by means of equation 7.74.

$$① \quad f_D = \frac{2{,}6 \cdot q \cdot L_M^4}{E_{C_R} \cdot J_P} \qquad ② \quad f_D = \frac{5{,}4 \cdot q \cdot L_E^4}{E_{C_R} \cdot J_P}$$

Equation 7.74

q = weight of the filled pipe along with any insulation (N/mm^2)
E_{cR} = creep modulus of the pipe material (N/mm^2)
L_M, L_E = bracket distance (m)
L_B = acceptable bracket distance (m)
J_P = pipe moment of inertia (mm^4)
f_D = deflection (mm)

At position 3, the pipe is not subject to deflection but to buckling.