SCR Piping Guide

For high-pressure SCR systems

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WIN GD Winterthur Gas & Diesel	Product **W-2S**	**SCR PIPING GUIDELINE** **for HP-SCR Systems**

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Table of contents

1 Disclaimer

The purpose of this document is to provide advice for the planning of the installation of a pre turbocharger SCR system into a vessel (arranged between the exhaust receiver of a WinGD 2-stroke engine and the turbocharger inlet of the same engine).
By the information and recommendations provided in this document, a ship designer or ship yard should be enabled to make design and cost investigations for the installation of a pre turbocharger SCR system onto a WinGD 2-stroke engine.

If not agreed otherwise, WinGD is not responsible for the final design of the SCR system arrangement in the ship hull, the support structure of the exhaust piping system, the dimensioning of the piping system and the correctness of any other information provided in this document.

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3 Introduction

This guideline provides basic information and recommendations for the layout, design and calculation of the exhaust gas piping elements of a high-pressure SCR system which should be arranged in or close to the engine room of a vessel.

This guide does not cover information and recommendations of how to arrange the SCR reactor or Mixing Pipe in the ship hull. This arrangement has to be done in cooperation with Ship yard or Ship Designer and the Suppliers for SCR Reactor and Mixing Pipe. This SCR system layout with the exact placement of SCR Reactor and Mixing Pipe has to be done in advance of the design of the piping for which this guide is valid. This guide only covers the design of the piping which connects Engine with Mixing Pipe, Mixing Pipe with SCR Reactor and SCR Reactor with the Engine.

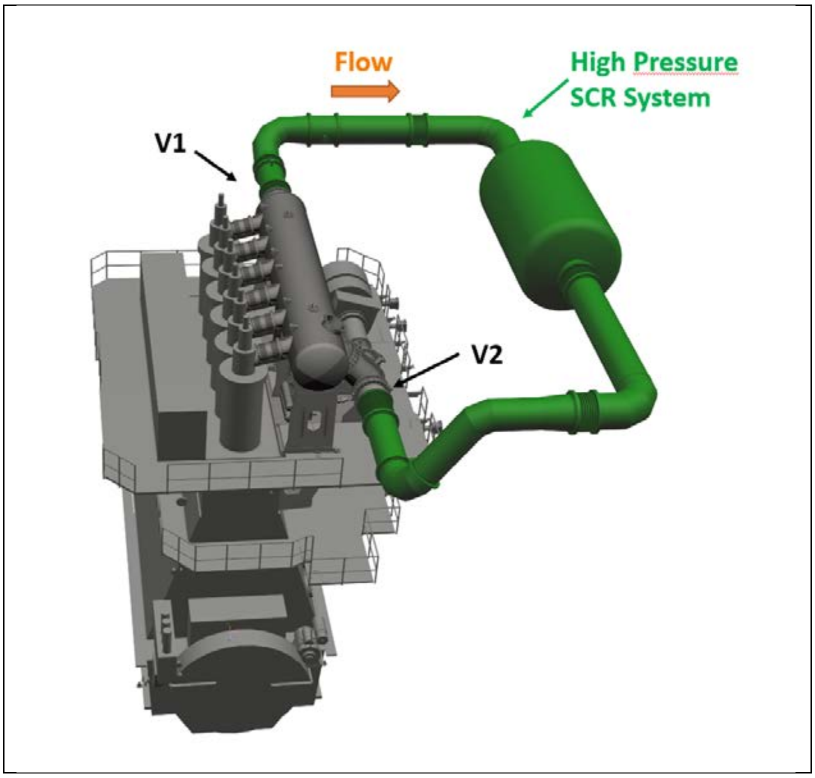
After describing the definition of parts for which this guideline is valid in more detail, information and boundary conditions for the design of the pipe segments is provided. With these information the reader is enabled to create a basic mechanical design of the SCR system piping segments. In the later chapters the basic knowledge for the layout and design of the whole SCR piping system is provided.

The order of the chapters is chosen to provide also the order of logical steps during the design phase of a SCR system piping. After considering the necessary sizing and material of the piping the thermal elongation of the piping has to be calculated to determine where expansion bellows are needed. After that the support for the piping segments can be designed for which at first knowledge about the forces acting on the piping is required. When the first design for the SCR system piping is done a calculation of the pressure drop can be executed which can lead to a next iteration of the design. When design and pressure drop is finished a vibration analysis can be performed to validate the design or show weak spots that have to be redesigned.

4 SCR Piping System

4.1 SCR System

The inlet of a high pressure SCR system is connected to the exhaust gas receiver of the engine and can be cut-off by a butterfly valve which is in the following being called V1. The outlet of the SCR system is connected to a pipe on the engine which leads to the turbocharger. The exhaust gas flow through the SCR system outlet to the engine can be cut-off by a butterfly valve which is in the following being called V2. In the following the terminology High Pressure SCR System or simply SCR System is being used for the whole system which is between V1 and V2 (marked in green in the image below).



Example Image of a High-Pressure SCR System

4.2 SCR System Layout and Design Features

A SCR system has 3 main design features. These are the SCR reactor (1), a pipe section for the urea injection and evaporation in the following called Mixing Pipe (2) and the interfaces to the engine (3) which are the connections to the valves V1 (3a) and V2 (3b).

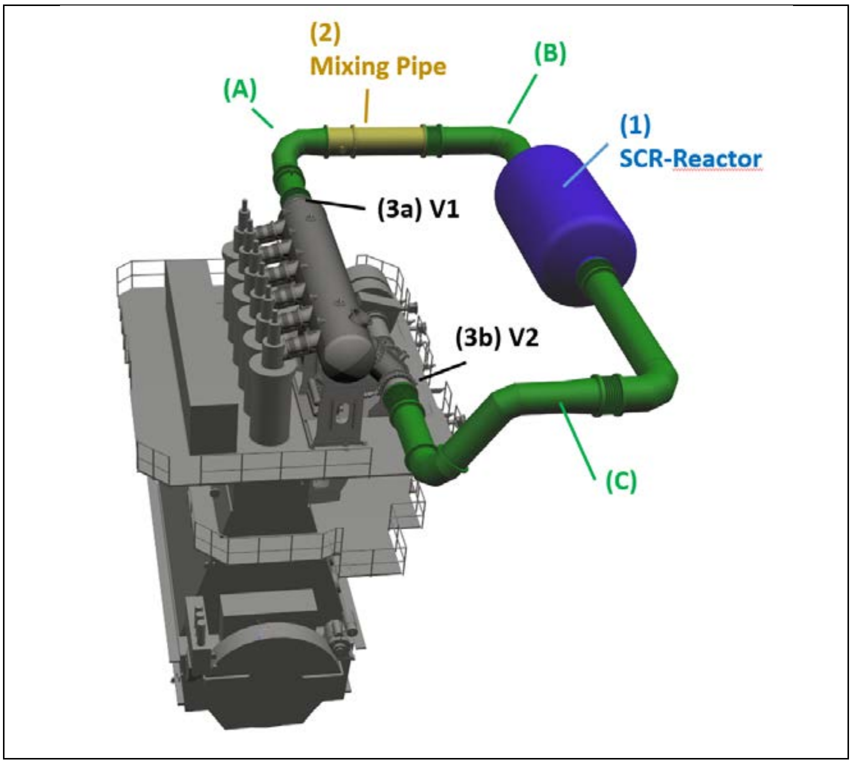


Image of a High-Pressure SCR System layout

This guide provides recommendations for the design of the piping segments A, B and C (marked in green) which connect the 3 main features of the SCR system.

Piping Segment A:
Connects Valve V1 (3a) with the mixing pipe (2).

Piping Segment B:
Connects the mixing pipe (2) with the SCR reactor (1).

Piping Segment C:
Connects the SCR reactor (1) with the Valve V2 (3b).

5 Piping Design

5.1 Boundary Conditions

5.1.1 Exhaust Gas Temperature in SCR System

The minimum and maximum operation temperatures depend on the SCR system and have to be provided by SCR system supplier.

In very rare cases the exhaust gas temperature of an aged 2-stroke RT-flex-engine can reach values up to 520°C before turbine, i.e. when an engine is operated close to the 110% load, i.e. for testing purposes. The catalyst elements and the whole SCR design must withstand such temperature for at least 20 minutes without any damage.

5.1.2 Exhaust Gas Pressure in the SCR System

The highest pressure for a specific engine can be found in WinGD general technical data. Under certain conditions the exhaust gas pressure at 100% engine load can exceed even 4.8 bar absolute.

The recommended minimum design pressure for the SCR system piping is 5 bar absolute pressure, which equals 4 bar overpressure in comparison to atmospheric pressure.

5.2 Piping Material

All materials used for components of the SCR system must be capable to withstand the environmental conditions. These conditions are determined by the impact from the process condition, sulphur content in fuel, corrosive fluids like urea solution and exhaust gas with high ammonia (NH3) concentration, and from the reactive chemical substances of the catalytic reduction process.

From the point of injection of the reducing agent a high concentration of ammonia (NH3) is present in the exhaust gas until the end of the SCR catalyst. A material with good corrosion resistance should be used in this area. After the SCR catalyst where the slip of NH3 is considered to be equal or below 10 ppm, this concentration can be considered as uncritical for corrosion.

Also if urea water solution (UWS) is used as reducing agent, it can happen that liquid UWS can hit the wall of the piping. At low wall temperatures this contact can lead to corrosive deposits on the wall. It needs to be clarified with the mixing pipe manufacturer if this must be considered for the material selection.

Depending on the content of sulphur in fuel, SO_2 and SO_3 is produced during combustion. As SO3 reacts easily with the water (H2O) in the exhaust gas, for lower exhaust temperatures it is mostly present in the form of gaseous sulphuric acid (H2SO4). After a vanadia based SCR catalyst the SO_3 concentration and therefore also the gaseous sulphuric acid (H2SO4) concentration can be higher than in front of the reactor, because for high vanadia contents and high exhaust gas temperatures, the SO2 in the exhaust gets oxidized to SO3 as a side reaction to the SCR reaction.

When the SCR system cools down after shut-down, the gaseous sulphuric acid can condense on the walls of the SCR system if the temperature drops below the dew point temperature. If no optional venting system or other countermeasures are being used, the walls of the piping system must be designed to withstand this high corrosive environment. Also with an optional venting system it is recommended to use a material with improved corrosion resistant properties.

Recommendations for the material selection

If specific class rules for the SCR piping material exist, these rules must be fulfilled.

If no specific class rules apply, it is recommended to use austenitic steel with proper corrosion and heat resistant properties, which can withstand the corrosive atmosphere of a high ammonia (NH3) concentration in the exhaust gas. In case that liquid urea-water-solution (UWS) could hit the inner wall of the piping a stainless steel material is highly recommended. But the mixing pipe supplier has to give information if liquid UWS could also hit the walls of the piping after the mixing pipe.

In DIN EN 1092-1 such a material is part of material class group 1 (e.g. 10E0, 13E0,…).
This industrial standard is designated especially for piping flanges, but the same material requirements can be also applied for the piping.

Recommended minimum material requirements

A material of the material group 4E0 (lower alloyed steels with about 0.3% molybdenum), according DIN EN 1092-1 Annex G2 is the recommended minimum quality. See image below.

Group	Material	Material No.	EN	R_p/R_e	Creep	Notes
3E0	P245GH	1.0352	10222-2	$R_{p0,2\,t}$	X	
3E1	P280GH	1.0426	10222-2	$R_{p0,2\,t}$	X	
4E0	16Mo3	1.5415	10222-2	$R_{p0,2\,t}$	X	
5E0	13CrMo4-5	1.7335	10222-2	$R_{p0,2\,t}$	X	
6E0	11CrMo9-10	1.7383	10222-2	$R_{p0,2\,t}$	X	
6E1	X16CrMo5-1+NT	1.7366	10222-2	$R_{p0,2\,t}$	X	
A1) 7E3 (A1	13MnNi6-3	1.6217	10222-3	$R_{p0,2\,t}$	—	f
7E1	P355 NL1, P355 NL2	1.0566 1.1106	10028-3	$R_{p0,2\,t}$	—	a, g
7E2	15NiMn6	1.6228	10222-3	$R_{p0,2\,t}$	—	f
7E3	12Ni14	1.5637	10222-3	$R_{p0,2\,t}$	—	f
A1) 7E3 (A1	X8Ni9	1.5662	10222-3	$R_{p0,2\,t}$	—	f
8E2	P285NH	1.0487	10222-4	$R_{p0,2\,t}$	—	c, d
8E3	P355NH	1.0565	10222-4	$R_{p0,2\,t}$	—	b, d, e
9E0	X20CrMoV11-1	1.4922	10222-2	$R_{p0,2\,t}$	X	
9E1	X10CrMoVNb9-1	1.4903	10222-2	$R_{p0,2\,t}$	X	

List of materials for non-austenitic materials
(Acc. Table G.1.1-1, DIN EN 1092-1:2007+A1:2013)

But it is recommended to use at least a material with also improved anti-corrosive properties. Such a material could be S355J2G1W (according DIN EN 10025-1:2005-02). This material is widely available and builds a protective layer on its surface which improves the anti-corrosive properties. In the ANNEX a more detailed specification for this material is provided, including corresponding material descriptions for other standards.

5.3 Nominal Pressure

Due to the recommended design pressure, design temperature and piping material limits, the recommended nominal design pressure can be derived from DIN EN 1092-1, Tables G.2.1. (see also image below). In the following a design temperature of 490°C, a material of type 4E0 and a maximum system over pressure of 3.7 bar are used as an example. The real design conditions must be provided by the SCR system supplier.

In the tables below, the first nominal pressure value that can withstand 3.7 or more bar overpressure for at least 490°C with a 4E0 material is PN 10.

If another design temperature, design pressure and/or is chosen the according nominal pressure can be selected accordingly.

If the nominal pressure for the SCR system is selected, all other necessary design information like the necessary wall thickness of the piping and the required flange sizes and types can be derived from DIN EN 1092-1.

Table G.2.1-2 — PN 6

PN	Group	v_R (mm)	max. allowable temperature TS °C																							
			RT	100	150	200	250	300	350	400	450	460	470	480	490	500	510	520	530	540	550	560	570	580	590	600
			max. allowable pressure PS bar																							
	3E0	≤ 50	6,0	5,5	5,2	5	4,5	4,1	3,8	3,5	1,9															
	3E0	50 < v_R ≤ 150	6,0	5,1	5	4,6	4,2	3,8	3,6	3,4	1,9															
	3E1	≤ 50	6,0	6,0	6,0	6,0	5,8	5,2	4,8	4,4	2,4															
	3E1	50< v_R ≤ 150	6,0	6,0	5,7	5,4	5,0	4,6	4,2	3,8	2,4															
	4E0	≤ 60	6,0	6,0	6,0	6,0	5,8	5,1	4,8	4,4	4,1	3,8	3,5	3,2	2,9	2,6	2,1	1,6	1,3							
	4E0	60 < v_R ≤ 90	6,0	6,0	6,0	6,0	5,5	4,8	4,5	4,1	3,8	3,6	3,3	3,1	2,8	2,6	2,1	1,6	1,3							
	4E0	90 < v_R ≤ 150	6,0	6,0	6,0	5,5	5,1	4,5	4,2	3,8	3,5	3,3	3,1	3,0	2,8	2,6	2,1	1,6	1,3							
	5E0	≤ 60	6,0	6,0	6,0	6,0	6,0	6,0	5,7	5,4	5,0	4,8	4,5	4,3	4,0	3,9	3,3	2,6	2,2	1,7	1,4	1,1	0,9			
	5E0	60 < v_R ≤ 90	6,0	6,0	6,0	6,0	6,0	5,8	5,4	5,0	4,7	4,5	4,3	4,1	3,9	3,9	3,3	2,6	2,2	1,7	1,4	1,1	0,9			
6	5E0	90 < v_R ≤ 150	6,0	6,0	6,0	6,0	6,0	5,5	5,0	4,7	4,4	4,2	4,1	4,0	3,9	3,9	3,3	2,6	2,2	1,7	1,4	1,1	0,9			
	6E0	≤ 150	6,0	6,0	6,0	6,0	6,0	6,0	5,8	5,5	5,2	5,0	4,7	4,4	4,1	3,8	3,3	2,9	2,5	2,2	1,9	1,6	1,4	1,2	1,0	0,9
	6E1	≤ 150	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	5,0	4,1	3,2	2,7	2,3	2,0	1,6	1,4	1,2	1,0			
	8E2	35 < v_R ≤ 70	6,0	6,0	6,0	5,6	5,1	4,4	3,9	3,3																
	8E2	70 < v_R ≤ 100	6,0	6,0	5,6	5,2	4,7	3,9	3,3	2,8																
	8E2	100 < v_R ≤ 150	6,0	5,8	5,3	4,7	4,2	3,3	2,8	2,2																
	8E3	50 < v_R ≤ 100	6,0	6,0	6,0	6,0	6,0	5,8	5,4	4,7																
	8E3	100 < v_R ≤ 150	6,0	6,0	6,0	6,0	6,0	5,6	5,0	4,2																
	9E0	≤ 150	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	5,3	4,7	4,2	3,6	3,1	2,7	2,3	1,9	1,6
	9E1	≤ 130	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	5,7	5,2	4,7	4,2	3,8	3,4	3,0	2,6

Table G.2.1-3 — PN 10

PN	Group	v_R (mm)	max. allowable temperature TS °C																					
			RT	100	150	200	250	300	350	400	450	460	470	480	490	500	510	520	530	540	550	560	570	580	590	600
			max. allowable pressure PS bar																							
10	3E0	≤ 50	10,0	9,2	8,8	8,3	7,6	6,9	6,4	5,9	3,2															
	3E0	50 < v_R ≤ 150	10,0	8,5	8,3	7,7	7,0	6,4	6,0	5,7	3,2															
	3E1	≤ 50	10,0	10,0	10,0	10,0	9,7	8,8	8,0	7,3	4,0															
	3E1	50< v_R ≤ 150	10,0	10,0	9,5	9,0	8,3	7,6	7,0	6,4	4,0															
	4E0	≤ 60	10,0	10,0	10,0	10,0	9,7	8,5	8,0	7,4	6,9	6,4	5,9	5,4	4,9	4,4	3,5	2,8	2,2							
	4E0	60 < v_R ≤ 90	10,0	10,0	10,0	10,0	9,2	8,0	7,6	6,9	6,4	6,0	5,6	5,2	4,8	4,4	3,5	2,8	2,2							
	4E0	90 < v_R ≤ 150	10,0	10,0	10,0	9,2	8,5	7,6	7,0	6,3	5,9	5,6	5,3	5,0	4,7	4,4	3,5	2,8	2,2							
	5E0	≤ 60	10,0	10,0	10,0	10,0	10,0	10,0	9,5	9,0	8,4	8,0	7,6	7,2	6,8	6,5	5,5	4,4	3,7	2,9	2,3	1,9	1,5			
	5E0	60 < v_R ≤ 90	10,0	10,0	10,0	10,0	10,0	9,7	9,0	8,3	7,8	7,5	7,2	6,9	6,6	6,5	5,5	4,4	3,7	2,9	2,3	1,9	1,5			
	5E0	90 < v_R ≤ 150	10,0	10,0	10,0	10,0	10,0	9,1	8,4	7,9	7,3	7,1	6,9	6,7	6,5	6,5	5,5	4,4	3,7	2,9	2,3	1,9	1,5			
	6E0	≤ 150	10,0	10,0	10,0	10,0	10,0	10,0	9,7	9,2	8,8	8,3	7,8	7,3	6,9	6,4	5,6	4,9	4,2	3,7	3,2	2,7	2,4	2,0	1,8	1,6
	6E1	≤ 150	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	8,4	6,9	5,3	4,5	3,8	3,3	2,8	2,3	2,0	1,7			
	8E2	35 < v_R ≤ 70	10,0	10,0	10,0	9,3	8,6	7,4	6,5	5,6																
	8E2	70 < v_R ≤ 100	10,0	10,0	9,4	8,6	7,9	6,5	5,6	4,6																
	8E2	100 < v_R ≤ 150	10,0	9,7	8,8	7,9	7,0	5,6	4,6	3,7																
	8E3	50 < v_R ≤ 100	10,0	10,0	10,0	10,0	10,0	9,8	9,0	7,9																
	8E3	100 < v_R ≤ 150	10,0	10,0	10,0	10,0	10,0	9,3	8,4	7,0																
	9E0	≤ 150	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	8,9	7,9	7,0	6,0	5,2	4,5	3,8	3,2	2,8
	9E1	≤ 130	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	9,5	8,7	7,9	7,1	6,3	5,7	5,0	4,4

P/T-rating table for PN 6 and PN 10
(Acc. Table G.2.1-2/3, *DIN EN 1092-1:2007+A1:2013)*

5.4 Sizing

5.4.1 Diameter of the Piping

The diameter of the piping is being defined by 3 parameters:

- Sizes of the Valves V1 and V2 (provided by the project specific WinGD drawing set)
- Necessary diameter for the mixing pipe (provided by the supplier of the urea injection system)
- Needed inlet and outlet diameter of the SCR reactor so that the desired ammonia uniformity can be reached (provided by the SCR reactor supplier)

Piping Segment A (from Valve V1 to the Mixing Pipe inlet)

The pipe size of the connection to Valve V1 has to be the same as the nominal diameter of V1 (see project specific WinGD drawing set).

If the necessary diameter of the mixing pipe is bigger or smaller than the nominal diameter of Valve V1, the diameter has to be adapted somewhere in the piping segment A from Valve V1 to the mixing pipe. It is recommended that this adaption is being executed by a cone.

Piping Segment B (from Mixing Pipe outlet to SCR reactor inlet)

The connection to the mixing pipe has to have the same diameter as the mixing pipe outlet.

If the necessary diameter of the SCR reactor inlet is bigger or smaller than the nominal diameter of the mixing pipe outlet, the diameter has to be adapted somewhere in the piping segment B from mixing pipe outlet to SCR reactor inlet. It is recommended that this adaption is being executed by a cone.

Piping Segment C (from SCR reactor outlet to Valve V2)

The connection to the SCR reactor outlet has to have the same diameter as the required SCR reactor outlet diameter.

If the necessary diameter of connection to Valve V2 is bigger or smaller than the nominal diameter of the SCR reactor outlet, the diameter has to be adapted somewhere in the piping segment C from the SCR reactor outlet to Valve V2. It is recommended that this adaption is being executed by a cone.

5.5 Piping Layout

5.5.1 Piping Segment A (from V1 to the Mixing Pipe)

From WinGD it is required to have a bellow/compensator between the engine side interface connection at Valve V1 and the piping segment A. Please see the project specific WinGD drawing set for further information.

It has to be ensured that the exhaust gas flow has the correct properties for a good evaporation and mixing of the urea solution and the risk of deposit creation is reduced as much as possible. Therefore the supplier of the mixing pipe has to provide the necessary flow properties (e.g. flow uniformity) for his specific mixing pipe design.

If the piping length is short, the piping segment A has a bend and or a cone shortly in front of the mixing pipe, the necessary flow requirements probably will not be achieved without any countermeasures to optimize the flow in the pipe segment. Such countermeasures can be the introduction of guide vanes in the pipe bend and introduction of perforated plates to unify the flow.

In a situation where high requirements are given for the flow and countermeasures are necessary to achieve these requirements it is recommended to perform at least a steady-state CFD analysis of the piping segment to ensure that the requirements are being achieved. It is also recommended to get an approval from the mixing pipe supplier for the final flow quality.

5.5.2 Piping Segment B (from Mixing Pipe Outlet to SCR Reactor Inlet)

It has to be ensured that the exhaust gas flow has the correct properties for a good SCR performance. Therefore the supplier of the SCR reactor (if different also the supplier of the catalytic elements) has to provide the necessary flow properties (e.g. ammonia uniformity, flow velocity uniformity, etc.) for his SCR reactor design.

If the piping length is short, the piping segment B has a bend and or a cone shortly in front of the SCR reactor, the necessary flow requirements probably will not be achieved without any countermeasures to optimize the flow in the pipe segment. Such countermeasures can be the introduction of guide vanes in the pipe bend and introduction of perforated plates to unify the flow or a special mixing device if the ammonia uniformity cannot be reached as intended.

In a situation where high requirements are given for the flow and countermeasures are necessary to achieve these requirements it is recommended to perform at least a steady-state CFD analysis of the piping segment to ensure that the requirements are being achieved. It is also recommended to get an approval from the SCR reactor supplier (if different also from the catalytic element supplier) for the final flow quality.

It can be reasonable to consider a bellow/compensator between the piping segment B and the SCR reactor inlet.

5.5.3 Piping Segment C (from SCR Reactor Outlet to Valve V2)

It can be reasonable to consider a bellow/compensator between the SCR reactor outlet and the piping segment C.

If there are special flow requirements for the exhaust gas flow from the SCR reactor at the inlet to Valve V2, these requirements are mentioned in the project specific WinGD drawing set.
If there are no special flow requirements for the inlet flow into Valve V2, but for the piping segments A and/or B a CFD analysis has been made it is recommended to also include piping segment C to the CFD analysis and provide the flow data at the inlet to Valve V2 to WinGD as an information.

From WinGD it is required to have a bellow/compensator between the engine side interface connection at Valve V2 and the piping segment C. Please see the project specific WinGD drawing set for further information.

5.5.4 General Considerations for the Layout

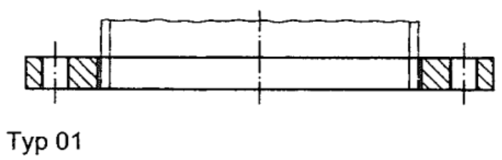
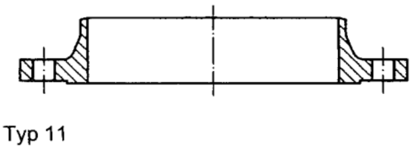
Necessary space requirements for crane operation, maintenance of the engine, turbocharger, SCR reactor, and other auxiliary systems of engine and SCR system have to be considered in the SCR System piping design.

For further information on SCR system related requirements please contact SCR system component suppliers, especially for SCR reactor and catalytic elements, soot blowing system, mixing pipe and urea solution pumping, dosing and injection system as well as for the control system of the SCR system.

For further engine related maintenance and other space requirements see WinGD project specific drawing set, WinGD Marine installation manual (MIM) or contact WinGD Licensee, Shipyard or WinGD for further advice.

5.6 Flanges

The flange type and dimensions for the selected nominal pressure and pipe diameter can be derived from DIN EN 1092-1.

	
“Standard” plain flange, Type 01	**Welding neck flange, Type 11**

Some examples for flange types according to DIN EN 1092-1

If the nominal pressure for the SCR system is PN16, the plain flange type 01 is only available until nominal diameter DN600, according to DIN EN 1092-1. Therefore it is recommended to choose a welding neck flange according type 11 or similar for all flange connections in the SCR system.

If it is desired to choose a plain flange or other flange type which is not valid in DIN EN 1092-1 for the chosen nominal pipe diameter, it is recommended to verify this design by a FEM calculation.

5.7 Gaskets

It has to be ensured that the system and especially all flange connections are always tight so that no exhaust gas can leak into the engine room.

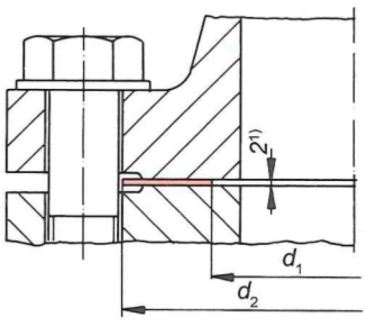
Due to manufacturing tolerances and small possible misalignments during assembly it is recommended to use gaskets for the flange connections.

Gasket material

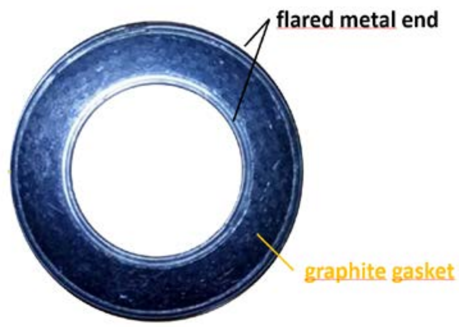
For the selection of the gasket, the chosen design temperature, design pressure and also the corrosive atmosphere in the SCR system has to be taken into consideration. Especially the possible formation of sulphuric acid is a limitation for the gasket material.

It is recommended to use a graphite based gasket material or similar.

Gasket Type



Standard graphite gasket pressed between two flanges

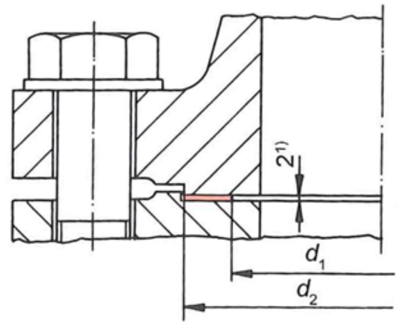


Spiral wound gasket with flared stainless steel ends

Examples for gaskets

The gasket is being pressed between the flanges. One option is to use a spiral wound gasket with a flared stainless steel end. This type allows an easier assembly and prevents the direct contact of the gasket with the corrosive atmosphere in the SCR system due to the inner stainless steel ring.

If a standard graphite gasket is selected it is recommended to have a flange with a recess for the gasket for better assembly properties. See also DIN 28090-2 for further information.



Standard graphite gasket pressed between a normal flange and a flange with recess

Example of flange with recess

Design considerations

For the design of the piping it has to be considered that the gasket is being compressed in the mounted condition when all bolts are properly tightened. This compression can be for example from 2 mm in normal condition to 1 mm in mounted condition. The information about the compression for a chosen gasket is being provided by the supplier of the gasket.

For bellows/compensators it is recommended to consider the mounted condition of the gaskets for the design, but to take the normal condition for the gasket for the definition of the pre-tensioning of the bellows. If the mounted condition of the gasket is considered for the pre-tensioning, it can lead to difficulties at assembly.

Further information

For further information about gaskets see DIN 28090-2 and/or DIN EN 1514-1 or contact the gasket supplier.

5.8 Bolting of Flanges

The bolting of the flanges is very important to ensure that the connection is tight. Only with the proper tightening force the gasket can be tight.

Also the material quality of the bolting is important because high stresses due to thermal differences can occur. During ramp up of the engine to a higher load the temperature rises and first heats the pipe and the flange which can elongate and put very high stresses on the bolting. Later the bolting is being heated as well by conduction and these thermal stresses are being relieved again.

Sizing of the bolting

The diameter and amount of the bolts for a certain flange is given in DIN EN 1092-1 in dependence from the nominal diameter of a flange and the nominal pressure.

The length of the bolts is depending case by case for each flange connection.

Material of Bolts and Nuts

Because of the high temperatures of the flanges it is necessary to use heat resistant bolts and nuts. It is recommended to use an austenitic material.

Due to possible different thermal properties it is important that the material is the same for the bolts, nuts and if they are being used also washers and distance pieces.

The standard EN 1515-1 helps with the selection of the bolting material. Depending from nominal pressure and the desired temperature range a list with different possible materials is provided there.

A material which is able to fulfil more than the necessary requirements for a SCR system is **X5CrNiMo17-12-2 (1.4401)** according to EN 10269. This material can be applied for nominal pressure up to PN 40 and temperatures from -200°C to +550°C.

Tightening

A proper tightening of the bolting is important. The following rules should be obeyed:

1. Tightening should **always** be executed from one bolt to a bolt across the flange and **never** from one to the one next to it.

2. The threads should always be lubricated well before tightening. For the SCR system bolting a heat-resistant lubricant has to be used.

3. Tightening has to be always executed with the correct tools which can ensure that the tightening torque has the correct value as designed.

If flanges are misaligned during assembly it is not allowed to try to fix the misalignment by additional tightening of the bolts. This can lead to bolting failure and leakage of the connection.

For the calculation of the necessary tightening force/torque please see reference list (Wagner, Festigkeitsberechnungen in Apparate- und Rohrleitungsbau, 2011) or other literature.

Forces on the bolting

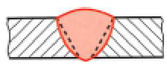
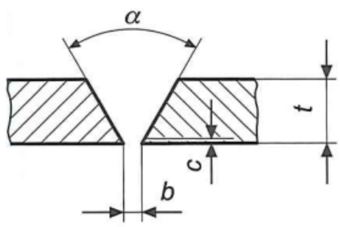
If forces (like gas forces or weight forces) are acting on a flange connection the bolting has to be calculated to be able to withstand these forces without allowing the flange connection to open a gap.
If the calculations show that the size or amount of the bolting is not sufficient, the size of the bolting and possibly also the design of the flange has to be adapted in a way which is different from given standards.
For such non-standard situations it is recommended to verify the design also by a FEM calculation.

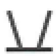
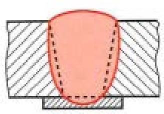
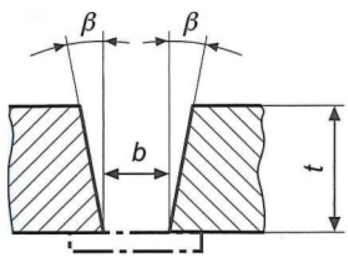
5.9 Welding of the Piping

It is recommended to use either V-seam welding or open single V-seam welding depending on the material thickness of the piping. For all case applicable a weld pool backup is also recommended so that it can be ensured that the weld seams can withstand the highest possible forces without cracking or starting to leak.

For guidance regarding welding see WinGD document 107.345.444B (Welding and quality instructions) and ISO 4063 for the correct welding process.

For the design of the welding seam preparation see DIN EN 29692. In the following table the welding seam preparation (acc. DIN EN 29692) for the recommended weld seams are displayed.

Welding type: V-seam		
Pipe wall thickness: $3 \leq t \leq 10$ **Opening angle:** $40° < \alpha < 60°$ **Gap:** $b \leq 4$ **Nose:** $c \leq 2$ **Symbol acc. ISO 2553**: 		

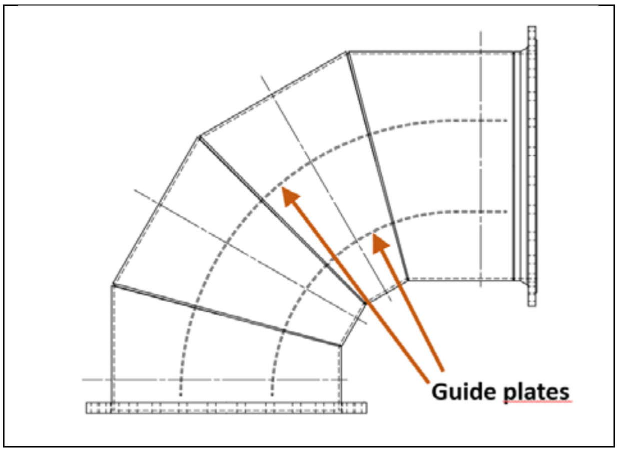
Welding type: open single V-seam		
Pipe wall thickness: $t > 16$ **Opening angle:** $5° \leq \beta \leq 20°$ **Gap:** $5 \leq b \leq 15$ **Symbol acc. ISO 2553:** 		

Properties of the recommended weld seams (acc. DIN EN 29692)

5.10 Piping elements for flow optimization

If a CFD analysis shows that the flow distribution over the SCR catalyst does not meet the requirements of the SCR supplier, measures need to be introduced to improve the flow distribution. In general, the service provider of the CFD analysis can help to define these measures.

Examples for such measures can be the introduction of static mixers, perforated plates, cone elements or guide plates in pipe bends (see example below).



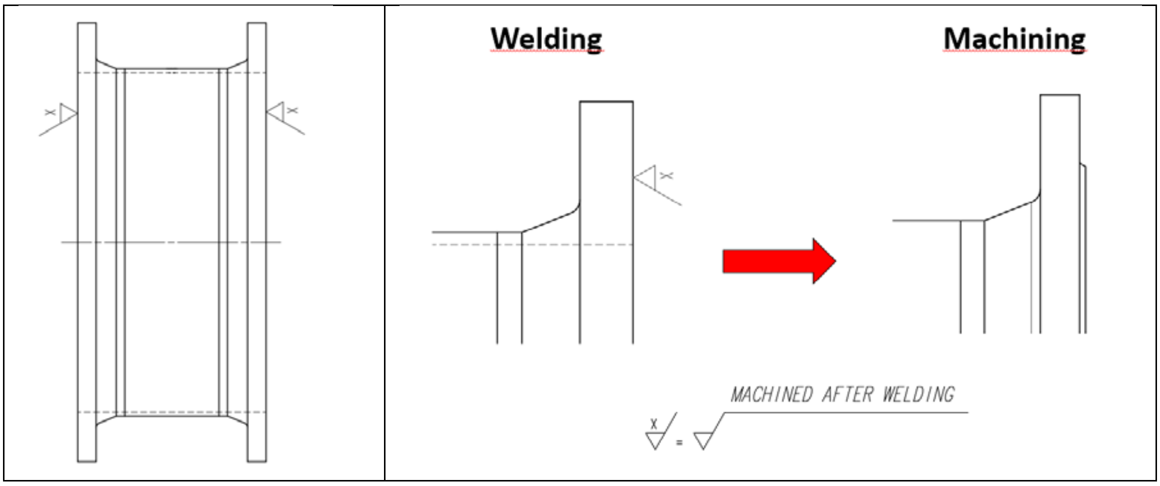
Pipe bend with guide plates

5.11 Further Design Recommendations

Machining and Welding

It is recommended to add enough allowances for proper machining to all flanges of pipe elements.

Due to thermal deformation during welding it is recommended to machine all single pipe elements after welding. This ensures the correct dimensions and tolerances for the piping and also a good welding quality.



Example for the use of "Machined after welding" on drawings

T_PC-Drawing_portrait | Made by: Y. Keel, S. Knecht | Released by: K. Moor | First released: 29.07.2010 | Release: 1.29 (2017-05-03)

Orientation of welded pipes

For large pipe diameters the pipe pieces are often manufactured by rolling and welding.

As it is possible that condensation of water or sulphuric acid, etc. can occur in the SCR system while cooling down after operation, this condense fluid can corrode the walls of the piping and weld seams.

To minimize the risk of corrosion of the weld seams it is recommended that for a not vertical pipe the manufacturing weld seam of the pipe is not at the bottom of the pipe after assembly (if applicable). It is best if this weld seam is at the top.

Example image for the orientation of the manufacturing weld seams of the piping after assembly

5.12 Assembly recommendations

It is recommended after the final assembly to check that at least the inside of the Piping Segment C from SCR Reactor to the Valve V2 on the engine is cleaned from any possible source of debris that can get loose and severely damage the turbocharger. Examples for such debris are remaining (not removed) slack from welding, forgotten assembly equipment in the piping or damaged (during assembly) inner sleeve of expansion bellow which could get loose.

The reason for this recommendation is that in the SCR Reactor a grid has to be mounted to protect the turbocharger from debris (see also DAAD075623, Turbocharger Protection Instruction), but in the piping after the SCR Reactor no such protection is possible and therefore this Piping Segment has to be checked with care.

6 Thermal Elongation

6.1 Calculation of the thermal elongation

The thermal elongation of a part (e.g. piping piece between to expansion bellows) is always expanding from the fix point of the support of the part. The thermal elongation at the fix point, per definition is 0 mm.

From the fix point, the thermal elongation Δl can be calculated as follows:

$$\Delta l = \alpha \cdot \Delta T \cdot l \tag{1}$$

With the variables:

Δl : Total amount of thermal elongation in $[mm]$

α : Thermal expansion coefficient in $\left[\frac{1}{K}\right]$

ΔT : Temperature difference in $[K]$

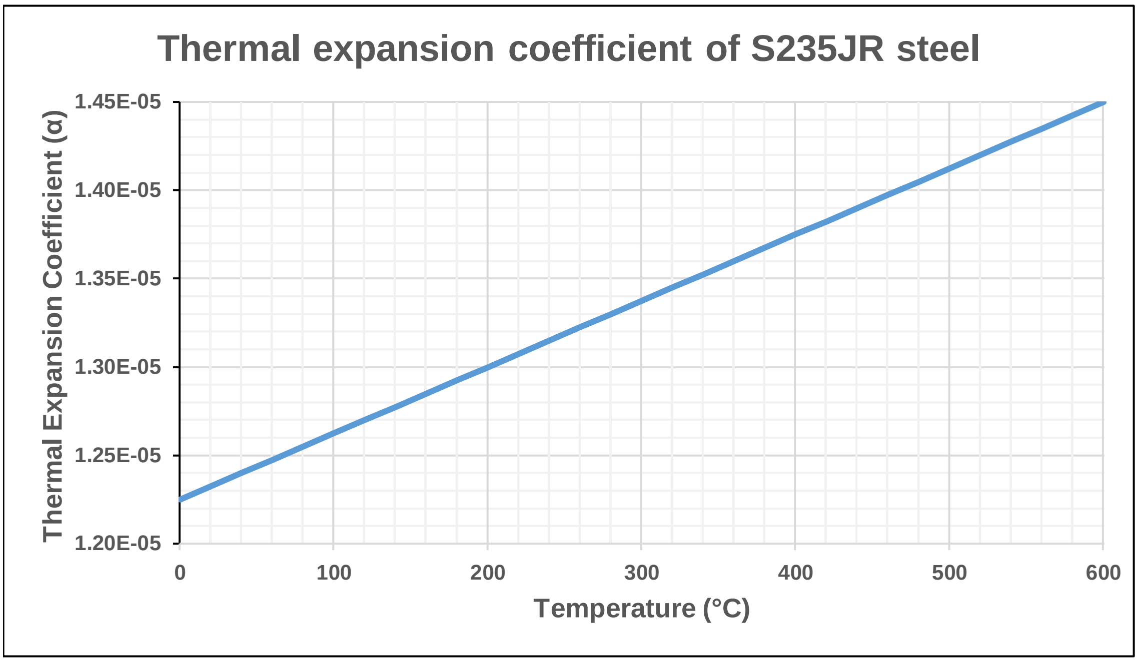
l : Length of the part from fix point of support to the maximum edge of the part in the direction in which the thermal elongation wants to be calculated. In $[mm]$.

6.2 Thermal expansion coefficient for thermal elongation calculation

The thermal expansion coefficient depends on the material of the piping system to be calculated. It is recommended to get this information directly from the supplier of the material.

The thermal expansion coefficient is not a constant value, but changes with the temperature. For steel the coefficient gets bigger with temperature. This means that the thermal elongation gets longer the higher the temperature is.

In the chart below the thermal behaviour of the thermal expansion coefficient is being displayed for S235JR steel as an example[1]:



Thermal expansion coefficient of S235JR Steel

If a simplified result is desired, for a material as shown in the chart above, the thermal expansion coefficient can be taken for the highest and lowest temperature and the arithmetic average can be used in the above mentioned equation.

$$\bar{\alpha} = \frac{\alpha_{T_{min}} + \alpha_{T_{max}}}{2} \qquad (1a)$$

If a more accurate result is needed, the thermal expansion has to be determined by integration of the following differential equation.

$$\alpha = \frac{1}{l} \cdot \frac{dl}{dT} \qquad (1b)$$

6.3 Temperature Difference for thermal Elongation Calculation

[1] This is just an example. True expansion coefficient data has always to be obtained from material supplier.

The temperature difference is the substraction of the minimum occurring temperature from the maximum occurring temperature in the system.

The temperature difference can be calculated as follows:

$$\Delta T = T_{max} - T_{min} \tag{2}$$

<u>With the variables:</u>

ΔT :	Temperature difference, in $[K]$
T_{max} :	Maximum occurring temperature in the system, in $[°C]$
T_{min} :	Minimum occurring temperature in the system, in $[°C]$

6.4 Minimum Temperature of the System for thermal Elongation Calculation

According to the standard DIN ISO 286-1 the reference temperature for all dimensions according to ISO standard is 20°C. For that reason the machining temperature of all pipes must be the same temperature.

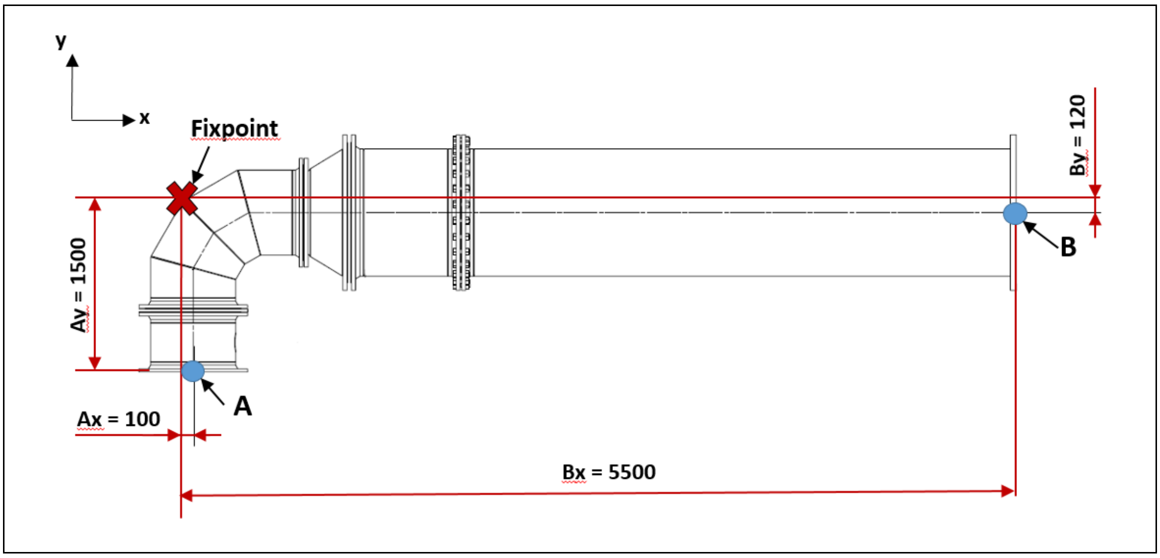
For that reason the minimum temperature for thermal elongation calculations for SCR piping should always be 20°C.

6.5 Maximum Temperature of the System for thermal Elongation Calculation

As already mentioned for the piping requirements, the recommended design temperature for the SCR piping has to be provided by the SCR system supplier.

To simplify the calculation of the maximum possible thermal elongation, and if no information about the maximum temperature is available, it is recommended to use 520°C as a maximum occurring temperature for all engines.

6.6 Calculation example of thermal elongation of a pipe segment



Example drawing of a piping element with fixpoint at pipe bend

As example assume a pipe segment with a 90° pipe bend and a longer straight pipe as shown above. With the same material S235JR as for which the thermal expansion is plotted as an example above. The minimum temperature is 20°C and the maximum temperature is 500°C. The points A and B should be considered to determine how big the thermal displacements of the connecting flanges are. The distance of point A from the fixpoint in x-direction is Ax and in y-direction it is Ay. The distances for point B are Bx and By. The simplified approach for the calculation of the thermal elongation should be used.

Step 1: Determine the average expansion coefficient $\bar{\alpha}$

From the graph above for the expansion coefficient, the expansion coefficients for 20°C and 500°C are:

$$\alpha_{T_{min}=20°C} = 1.23 \cdot 10^{-5}$$
$$\alpha_{T_{max}=500°C} = 1.41 \cdot 10^{-5}$$

From equation (1a) the average expansion coefficient can be calculated:

$$\bar{\alpha} = \frac{\alpha_{T_{min}} + \alpha_{T_{max}}}{2} = \frac{1.23 \cdot 10^{-5} + 1.41 \cdot 10^{-5}}{2} = \frac{2.64 \cdot 10^{-5}}{2} = 1.32 \cdot 10^{-5}$$

Step 2: Determine the temperature difference ΔT

From equation (2) the temperature difference can be calculated:

$$\Delta T = T_{max} - T_{min} = 500°C - 20°C = 480°C$$

Step 3: Calculation of the different thermal elongations

With the average expansion coefficient and temperature difference known, the thermal elongations can be calculated with equation (1), by replacing the length l with the distances of the points A and B from the fixpoint.

$$\Delta Ax = \bar{\alpha} \cdot \Delta T \cdot Ax = 1.32 \cdot 10^{-5} \cdot 480 \cdot 100\ mm = 0.63\ mm$$
$$\Delta Ay = \bar{\alpha} \cdot \Delta T \cdot Ax = 1.32 \cdot 10^{-5} \cdot 480 \cdot 1500\ mm = 9.5\ mm$$
$$\Delta Bx = \bar{\alpha} \cdot \Delta T \cdot Ax = 1.32 \cdot 10^{-5} \cdot 480 \cdot 5500\ mm = 34.8\ mm$$
$$\Delta By = \bar{\alpha} \cdot \Delta T \cdot Ax = 1.32 \cdot 10^{-5} \cdot 480 \cdot 120\ mm = 0.76\ mm$$

7 Compensators / Expansion Bellows

Expansion bellows are used to compensate thermal elongation and vibrations in the piping system.

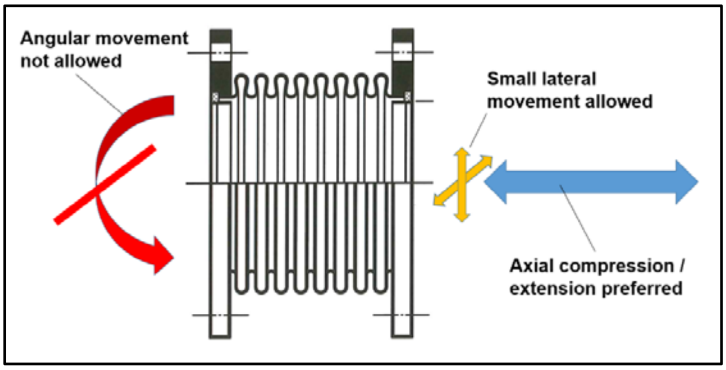
For detailed information and advice about expansion bellows it is recommended to contact the supplier.

It is recommended that all SCR system bellows should have an inner sleeve to protect the waves from turbulence induced vibrations due to the flow velocities occurring in a pre-turbocharger SCR system. These vibrations can lower the designed lifetime of a bellow significantly.

7.1 Types of Expansion Bellows

Many different types of expansion bellows exist for different load cases. A small selection of the most important types are described in the following. For other executions it is recommended to contact the supplier for information.

7.1.1 Axial Expansion Bellow



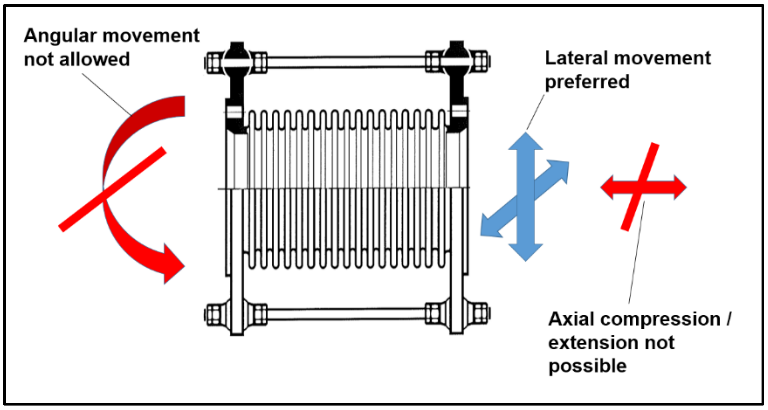
Axial Expansion Bellow

An axial expansion bellow is able to compress or extend in axial direction. A small lateral movement in addition to the axial compression is allowed.

An angular movement is not allowed. A maximum angular displacement of 1° due to manufacturing displacement is allowed for some manufacturer. It is recommended to request a detailed datasheet from the supplier.

An axial expansion bellow is transmitting axial and lateral forces to the piping where it is connected to.

7.1.2 Lateral Bellow

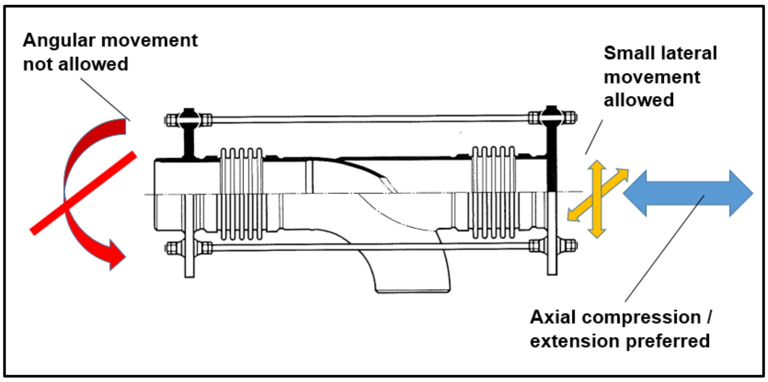


Lateral Bellow

The flanges of a lateral bellow are able to move relatively to each other in lateral direction of the bellow. An axial compression is not possible.

A lateral bellow is not transmitting any axial forces to the piping which it is connected to. Only lateral forces are being transmitted.

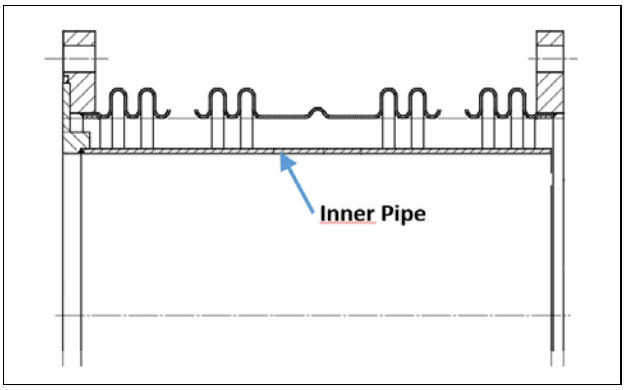
7.1.3 Pressure relieved Bellow



Pressure relieved Bellow

A pressure reliefed bellow is possible in different executions. The main feature is that it is not transmitting any forces on the connected piping. It can be very helpful for positions of the piping where a fixed support is not possible for a pipe bend. But the cost are significantly higher than a normal setup with fixpoint and axial bellows.

7.2 Expansion Bellow with inner pipe



Drawing of expansion bellow with inner pipe

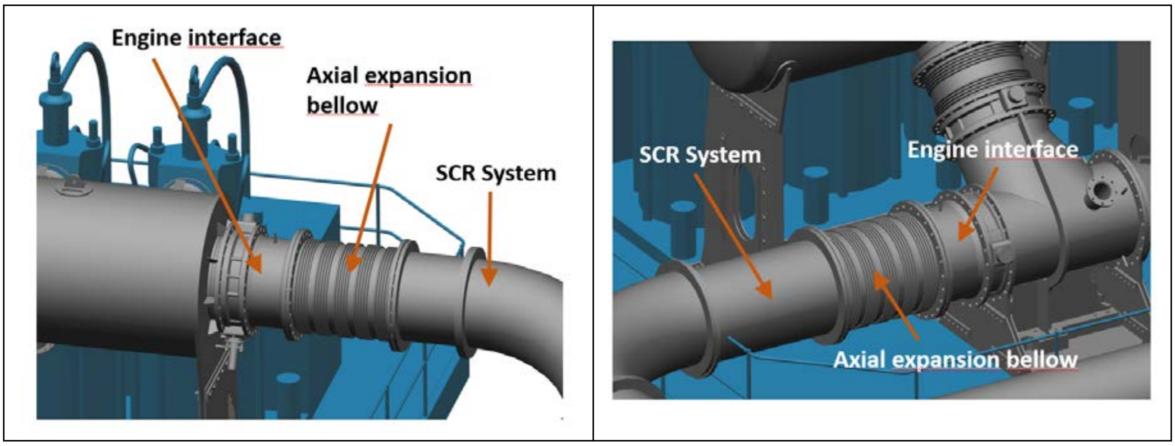
Some expansion bellows have inner pipes. The reason is either to have a smaller pressure drop over the expansion bellow or to protect the waves of the bellow from a high turbulent flow. The higher the exhaust gas velocity is, the heavier are the vibrations of the waves of the bellow caused by turbulences due to the form of the waves. These vibrations can lower the lifetime of the bellow, especially the longer the bellow is. It is recommended to ask the bellow supplier for advice if an inner pipe is needed or not, depending on the exhaust gas velocity.

7.3 Connection of engine and SCR system

To connect the interface of the engine with the SCR system, an axial expansion bellow is always needed between the engine and the SCR system. Also on the SCR system piping a fixpoint must be introduced as close as possible to this expansion bellow.

The reason are the vibrations of the engine. If the SCR system is not decoupled from the engine by an expansion bellow and fixpoint, the vibrations of the engine can translate into the SCR system and cause severe damage. If no fixpoint is used next to the expansion bellow, also the bellow itself can get heavily damaged because of the induced vibrations of the SCR system piping.

Examples for a correct arrangement of the bellow on the interface of engine an SCR system are shown in the pictures below.



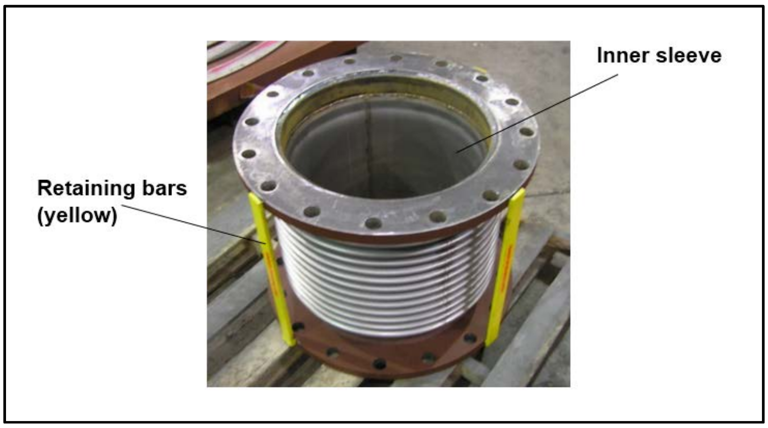
Interface between engine and SCR system with axial expansion bellow

7.4 Assembly of Expansion Bellows

For the assembly of the expansion bellows always the assembly instructions of the bellow supplier apply.

In addition WinGD recommends to keep to the following rules for the expansion bellow assembly:

- All expansion bellows are being delivered in a pre-tensioned state. The flanges of the bellow are connected with retaining bars to keep the bellow in its' pre-tensioned state. These retaining bars have to stay mounted on the bellow in their delivered state until the complete SCR piping system is assembled, correctly aligned and all bolts are tightened. As a last step in the whole SCR system assembly the retaining bars can be removed.
- The removal of the expansion bellows has to happen with a grinding device. A torch is not allowed at all close to a bellow. The hot flame can damage the thin sheet metal of the waves of the bellow.
- During assembly it has to be assured that the bellow has no torsional displacement. The torsion of a bellow will destroy it or limit its' lifetime dramatically.
- If there are any alignment issues during assembly the retaining bars are not allowed to be removed from the bellow to compensate big misalignment.



Axial Bellow with inner sleeve, before mounting

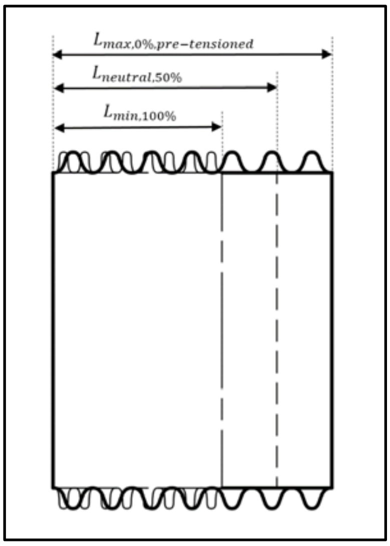
7.5 Pre-tensioning of Expansion Bellows

Pre-tensioning of an expansion bellow means that it is not assembled into the system in its' neutral state, but is compressed or extended for a certain percentage (e.g. 50%) of its' maximum stroke or for its' full stroke (100%). In general the expansion bellow is already being delivered in its' pre-tensioned condition which is also the mounting condition when the SCR system is being assembled.
The purpose of pre-tensioning of a bellow is that the deformation of the bellow is nearly zero for the designed operational working point of the engine (SMCR). For the load range in which the engine is running the longest time a bellow should be close to its' neutral state where the least forces and stresses are acting. This ensures the longest possible lifetime for the bellow.

A pre-tensioning of the bellow is always recommended. If the bellow is not pre-tensioned and assembled in its' neutral state it is always loaded in operation and the load on the bellow rises with the engine load. This means the bellow is experiencing the highest stresses for 100% where the worst operating conditions for the bellow take place. This operation method of the bellow should be always avoided.

If the engine running is running regularly with full load it is recommended to pretension it with its' maximum stroke.

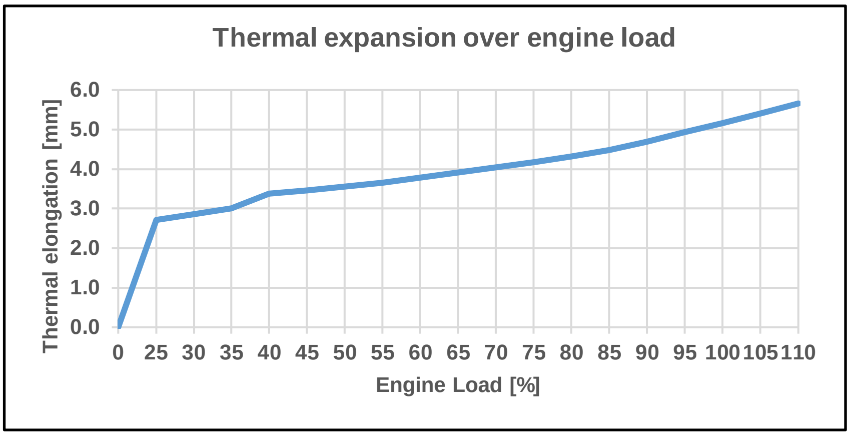
If the engine is regularly running with 50% load the bellow should be pre-tensioned for this load point.



Axial Bellow, pretensioned for 50% of stroke

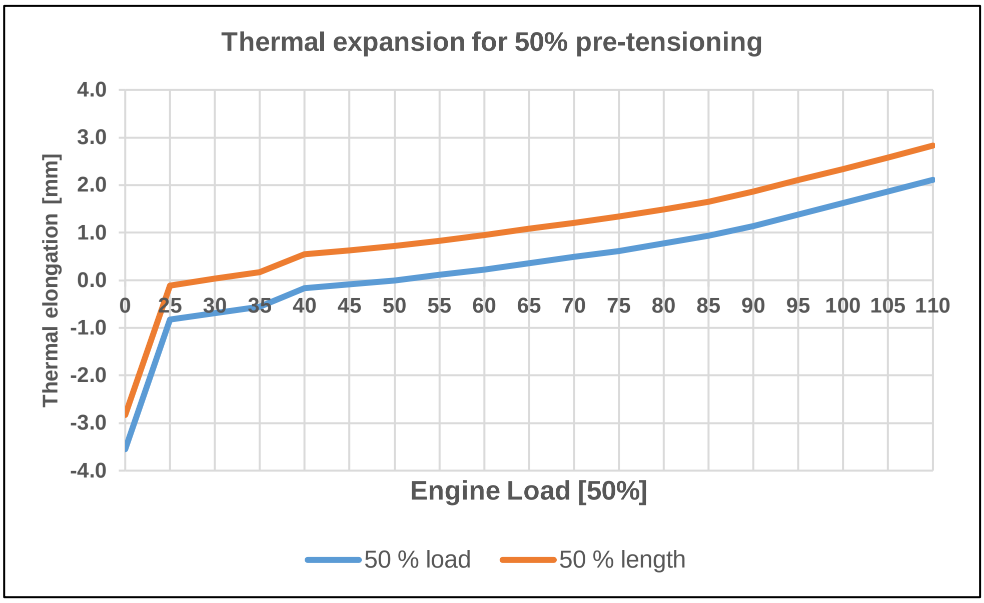
7.6 Choosing the pre-tensioning length of a bellow

If a bellow should be pre-tensioned for a certain load point of the engine (as an example in the following 50% engine load is being used) it has to be considered that the thermal elongation of a pipe is not exactly linear with the load range of the engine. For an example pipe piece the thermal elongation over the load range of an example engine is being displayed.



Thermal expansion over engine load

This behaviour implies that the pre-tensioning value for 50% of the engine load (so that the bellow is in its' neutral state for 50% engine load) is not the same as if the bellow would be pre-tensioned for 50% of the maximum thermal elongation of the pipe. In the chart below the comparison between pre-tensioning for 50% engine load and 50% maximum thermal elongation is being displayed. If the bellow is pre-tensioned for 50% of the maximum thermal elongation it reaches its' neutral state at around 30% engine load



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Modif	Number	Drawn Date	Number	Drawn Date	Number	Drawn Date	Number / Drawn Date
WIN GD Winterthur Gas & Diesel	Product W-2S		SCR Piping Guide for HP-SCR Systems				
Made	29.02.2016	M.Brutsche	Main Drw.	Page 38 / 90	Material ID PAAD219883		
Chkd	29.02.2016	D.Kadau	Design Group 8159	Drawing ID DAAD064155		Rev	
Appd	29.02.2016	M. Graf					

Thermal expansion for 50% Bellow pre-tensioning

7.7 Further Information

For further information please directly contact the supplier of the expansion bellows. Some suppliers have detailed design guidelines available, see for example list of references (BOA-Group, 2015).

Recommended supplier with experience for HP-SCR Systems:

Europe	Asia
SB Broneske GmbH (www.broneske.de) Ernst-Abbe-Strasse 9 D-25451 Quickborn/Hamburg Germany	**SB Broneske China Ltd.** (www.broneske.de) 7/F, China Overseas Building 76 Yanji Rd, Shibei District Qingdao China

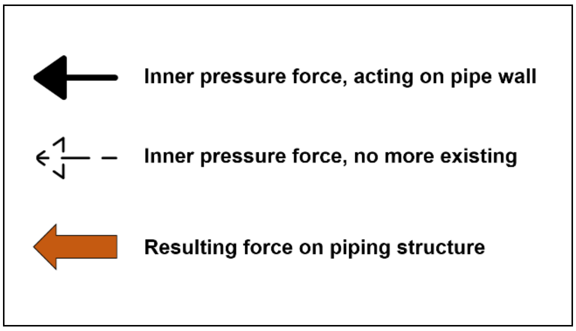
8 Pipe Forces

In comparison to a Post-Turbo SCR system in which the system contains atmospheric pressure the pressure in a Pre-Turbo SCR system is much higher. The overpressure can be several bar for a Pre-Turbo SCR system.
This overpressure can induce pressure forces on the SCR system piping that have to be considered for the support of the piping.

8.1 Forces due to inner Pressure

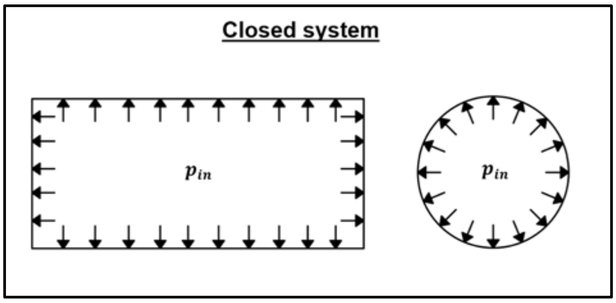
8.1.1 Definitions

For the explanations in this chapter the following different forces are considered:



8.1.2 Inner Pressure in a closed System

A system of a straight pipe is imagined which has a blind flange at both ends and is fully air tight. The system is considered rigid, so that it is not deformed by the force of the inner pressure. For this system, the forces due to the inner pressure (p_{in}) are cancelling each other as they are the same on each opposite side. No resulting force is acting on such a system.

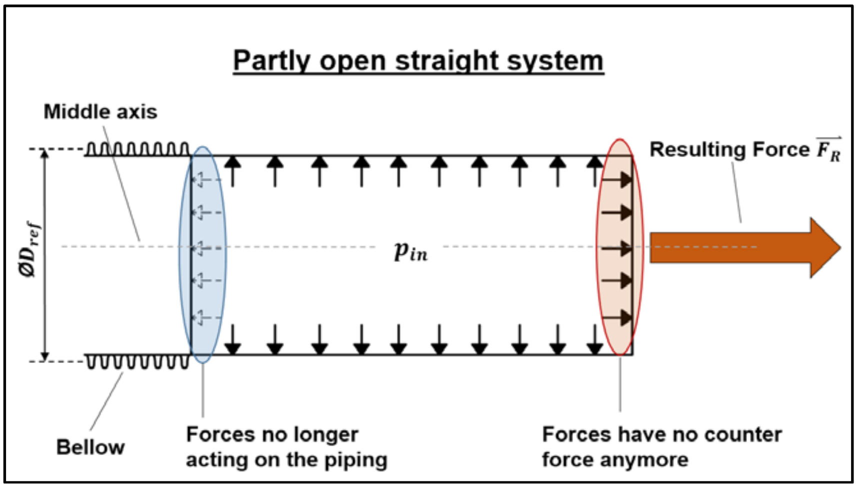


Example for a closed system

8.1.3 Inner Pressure in a partly open straight System

It is considered that one blind flange is removed from the closed system. If this face of the system stays open or is connected to another system with a bellow (that is not pressure reliefed), the inner pressure forces which acted on the blind flange disappear.
The inner pressure forces acting on the still existing blind flange have no counterpart anymore. This pressure force now cause a resulting force $\overrightarrow{F_R}$ which wants to push the system in the direction of the resulting force.

The force is acting along the middle axis of the system.



Partly open straight System

$$\overrightarrow{F_R} = p_{in} \cdot 100'000 \cdot \frac{\pi \cdot D_{ref}^2}{4} \qquad (3)$$

<u>With the variables:</u>

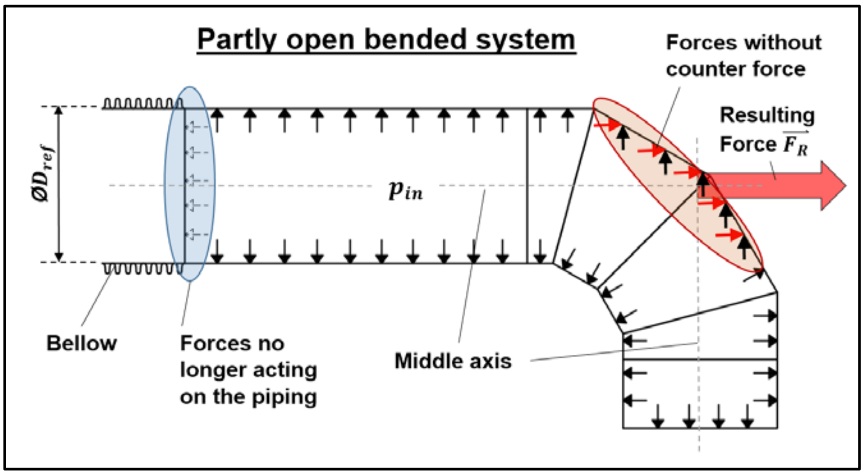
$\overrightarrow{F_R}$: Resulting force in $[N]$

p_{in} : Inner pressure in $[barg]$
Note: This is the relative overpressure in relation to the atmospheric pressure on the outside of the piping.

D_{ref} : Reference diameter of the bellow, in $[m]$
For a rough estimation, D_{ref} can be chosen as the diameter of the pipe, in $[m]$
For a precise calculation of the resulting force, the correct diameter D_{ref} has to be calculated as explained later.

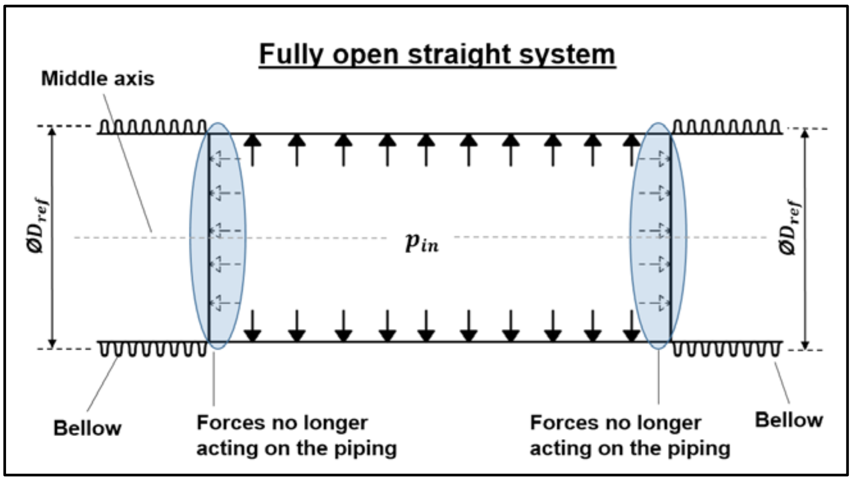
8.1.4 Inner Pressure in a partly open bended System

For a partly open bended system the same principles apply as for partly open straight systems. The resulting force on the piping can be calculated according to equation **(4)**.



Partly open bended System

8.1.5 Inner Pressure in a fully open straight System



Fully open straight System

Substitute for:				PC	Q-Code	X	X	X	X	X
Modif	Number	Drawn Date		Number	Drawn Date		Number	Drawn Date	Number	Drawn Date

WIN GD Winterthur Gas & Diesel	Product W-2S	SCR Piping Guide for HP-SCR Systems	
Made 29.02.2016 M.Brutsche	Main Drw.	Page 43 / 90	Material ID PAAD219883
Chkd 29.02.2016 D.Kadau	Design Group 8159	Drawing ID DAAD064155	Rev
Appd 29.02.2016 M. Graf			

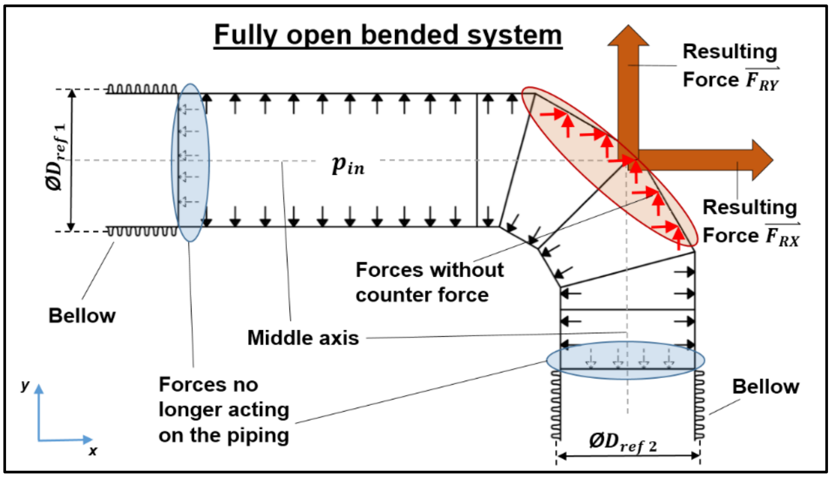
If the remaining blind flange is being replaced by a second bellow (with the same reference diameter than the first one) the system is now fully open. The pressure force which was pushing on the blind flange are now also removed and all inner pressure forces on the walls are cancelling each other out again. The result is no resulting force.

Note:

If the bellows don't have equal reference diameters a resulting force would act on the system.

8.1.6 Inner Pressure in a fully open bended System

In contrast to a fully open straight system the forces aren't cancelling each other out. Each opening creates a resulting force in the direction of its' middle axis. The resulting force depends on the value of the inner pressure and the area of the opening, which depends on the diameter of the opening.



Fully open bended System

In the displayed example the bend has a 90 degree angle and the coordinate axis x and y have been introduced for a better overview. Please consider that if the bend angle is not 90 degree, but the coordinate axis still are perpendicular to each other, the resulting forces have to be splitted in the direction of axis.

The inner pressure forces in a fully open bended system like it is displayed can be calculated as follows.

The resulting force in x-direction is:

$$\overrightarrow{F_{RX}} = p_{in} \cdot 100'000 \cdot \frac{\pi \cdot D_{ref\ 1}^2}{4} \quad (4)$$

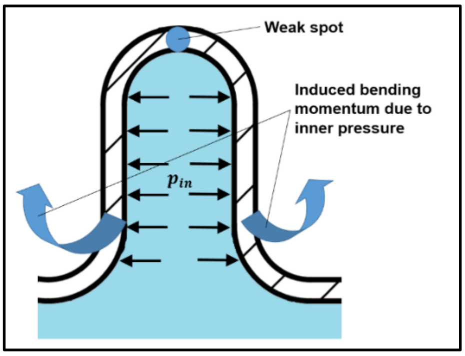
The resulting force in y-direction is:

$$\overrightarrow{F_{RY}} = p_{in} \cdot 100'000 \cdot \frac{\pi \cdot D_{ref\ 2}^2}{4} \quad (5)$$

Both equations (4) and (3) have the same variable definition as (3).

8.1.7 Determination of the correct Reference Diameter

The reference diameter for the calculation in the equations (3) to (5) does not depend on the inner pipe diameter (although it is a good rough estimation), but on the geometry of the expansion bellow (as long as no thrust pressure restraint bellow is being used).
The reason is, that the bellow is not rigid enough so that the inner pressure forces in the waves of the bellow are not cancelling each other out (Wagner, Rohrleitungstechnik, 2012). If a bellow would be closed, the tips of the waves of the bellow are weak spots where sheet metal would be bended around because of the momentum which is being induced by the inner pressure. If the bellow is open (which is always the case) this “deformation energy” is being also induced to the system it is connected to and adds up to the resulting force $\overrightarrow{F_R}$.



Weak Spot in Wave of Expansion Bellow

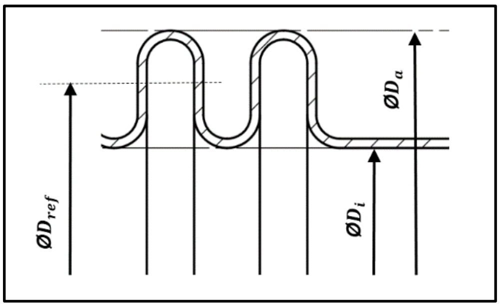
Because of that the theoretical area on with the inner pressure is acting is bigger than the area of the inner pipe diameter. This theoretical area is called “active Area” of the bellow. This value can be provided by the supplier of the expansion bellows.

If the active area of a bellow is known, the reference diameter can be calculated as follows.

$$D_{ref} = \sqrt{\frac{4 \cdot A_0}{\pi}} \quad (6)$$

With the variables:

D_{ref} : Reference diameter, in $[m]$
A_0 : Active area of the bellow, in $[m^2]$



Definition of Reference Diameter of Expansion Bellow

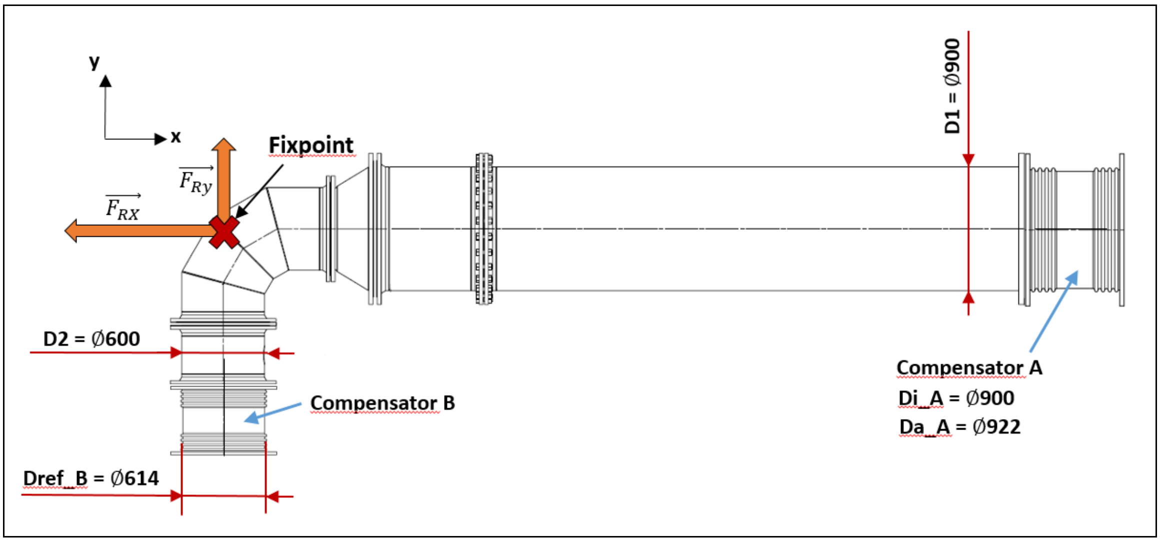
If the active area is not known but dimensions for the waves of the bellow are provided, the reference diameter can be also calculated as follows.

$$D_{ref} = \sqrt{\frac{D_a^2 + D_a \cdot D_i + D_i^2}{3}} \quad (7)$$

With the variables:

D_{ref} : Reference diameter, in $[m]$
D_a : Outer diameter of the waves of the bellow, in $[m]$
D_i : Inner diameter of the waves of the bellow, in $[m]$

8.1.8 Calculation example of resulting force on the fixpoint of fully open bended system



Example drawing of a fully open bended system including compensators at each end

As example assume a fully open bended system (piping segment) with a 90° pipe bend and a longer straight pipe as shown above. At each end a compensator is shown. For the compensator A only the inner and outer diameter of the bellow is known and for compensator B the reference diameter is known.

Assume the pressure in the piping system p_{in} to be 3.5 bar relative pressure to the outer pressure of 1 bar absolute (=1 atmosphere, atm).

The goal is to find the resulting forces on the fixpoint in x- and y-direction.

The fixpoint in the drawing is exactly in the longitudinal axis of compensator A and also exactly in the longitudinal axis of compensator B. This is why the force from compensator A is acting only in x-direction and the force from compensator B is only acting in y-direction. Hence, the resulting forces are equal to their corresponding forces from the compensators.

Please note, that if the fixpoint is not exactly in the longitudinal axis, the forces from the compensators are creating a momentum around the fixpoint.

Step 1: Calculate the reference diameter of compensator A

The reference diameter for compensator A can be calculated according equation (7), please note that it is recommended to use m as unit instead of mm:

$$D_{ref\ A} = \sqrt{\frac{D_a^2 + D_a \cdot D_i + D_i^2}{3}} = \sqrt{\frac{0.922^2 + 0.922 \cdot 0.900 + 0.900^2}{3}} = 0.911\ m$$

Step 2: Calculate the resulting force in x-direction

The resulting force in x-direction is equal to the force from the compensator A. It can be calculated according equation (4) by using the reference diameter of compensator A, as calculated in step 1:

$$\overrightarrow{F_{RX}} = p_{in} \cdot 100'000 \cdot \frac{\pi \cdot D_{ref\ A}^2}{4} = 3.5 \cdot 100'000 \cdot \frac{\pi \cdot 0.911^2}{4} = 228'136\ N = 228.1\ kN$$

Please note that the direction of the force is in negative x-direction, hence it is pushing onto the fixpoint.

Step 3: Calculate the resulting force in y-direction

The resulting force in y-direction is equal to the force from the compensator B. It can be calculated according equation (5) by using the known reference diameter of compensator B:

$$\overrightarrow{F_{RY}} = p_{in} \cdot 100'000 \cdot \frac{\pi \cdot D_{ref\ B}^2}{4} = 3.5 \cdot 100'000 \cdot \frac{\pi \cdot 0.614^2}{4} = 103'632\ N = 103.6\ kN$$

Please note that the direction of the force is in positive y-direction, hence it is pushing onto the fixpoint.

8.2 Dynamic Pressure Forces on a bended System

For the calculation of the force which is acting on the system due to pressure only the inner pressure was considered in the last subchapter. But for bended piping systems also the dynamic pressure which depends on the velocity of the flow is acting on the system wall. If a system is fully open and the exhaust gas can flow through the system this dynamic pressure force should also be considered in the force calculation.

However for 2-stroke pre turbine (high pressure) SCR systems this component of the force is very small. For the simplicity of the calculation the dynamic force can be neglected for piping diameters over DN 250 which would apply for the whole WinGD 2-stroke engine portfolio.

8.3 Forces acting on the System due to Bellow spring forces

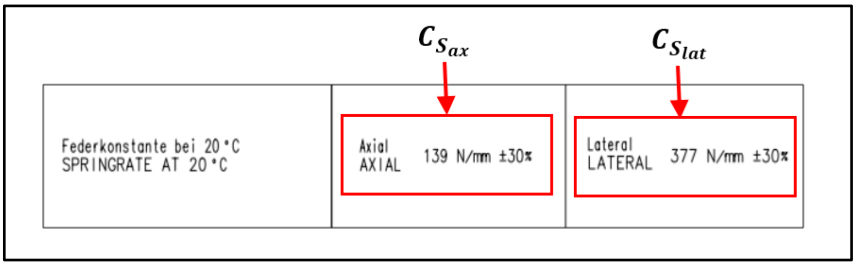
8.3.1 Calculation of the spring force of a Bellow

A bellow is similar to a spring. If a bellow gets compressed or extended from its' neutral position (either axial or lateral) it wants to get back in its' neutral position with a certain reset force F_S. The value of this force depends on the spring constant C_S or spring rate of the bellow and the length by which it gets compressed or expanded.

This reset force F_S also acts on both systems which the bellow is connecting.

The value of the spring constant C_S of a bellow is being provided by the supplier of the bellow.

A bellow has a spring constant $C_{S_{ax}}$ for axial compression or extension and a spring constant $C_{S_{lat}}$ for lateral movements of the bellow.



Example for spring constants of an Expansion Bellow

If the spring constants are known, the reset force F_S can be calculated as follows:

$$F_S = C_S \cdot \Delta l \quad (8)$$

With the variables:

F_S : Spring reset force of the bellow either in axial or lateral direction, in $[N]$

C_S : Spring constant of the bellow in $\left[\frac{N}{mm}\right]$, either $C_{S_{ax}}$ for axial direction or $C_{S_{lat}}$ for lateral direction.

Δl : Axial compression or extension length, or lateral movement, in $[mm]$

8.3.2 Direction and value of the spring force depending on pre-tensioning

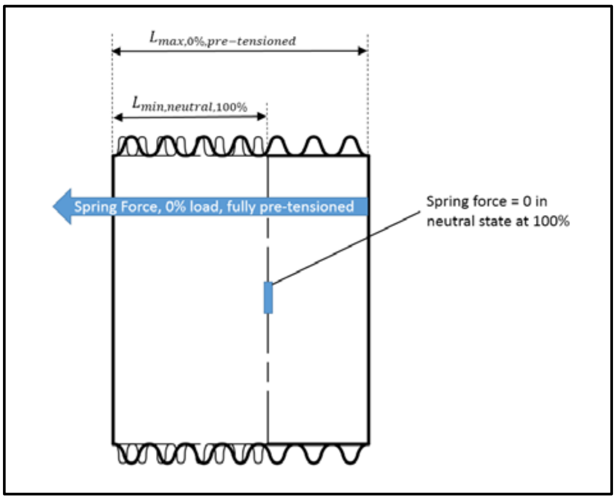
A bellow is made to compensate thermal elongations of piping systems. The spring force depends on the compression length of the bellow, which depends on the temperature of the system. This implies that the spring force of the bellow is not constant for the whole load range of the system, but is different for each load point.
As a bellow should always be assembled in a pre-tensioned state, the spring force which varies over the load range has a starting point that depends on the pre-tensioning of the bellow.

8.3.2.1 Direction and value of spring force for 100% pre-tensioning

If the bellow is pre-tensioned with 100% of the maximum thermal elongation length the spring force is maximal for the cold 0% load state. When the temperature is maximal for the 100% load state, where the pressure force on the system is highest, the spring force is being reduced to 0.

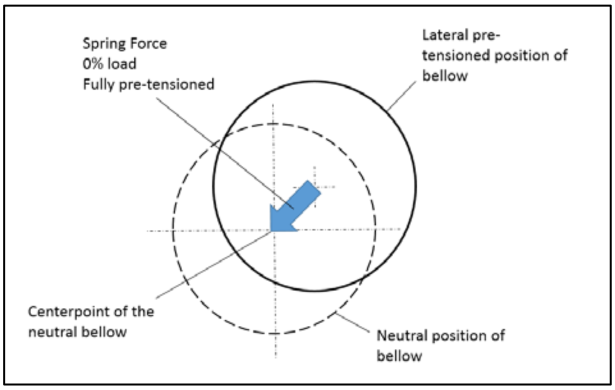
The spring force in pre-tensioned state is always pointing in the direction of the neutral (tensionless) state of the bellow.

For an axial pre-tensioned bellow, the force wants to contract the bellow. The force is pointing towards the bellow, whereas the pressure force is always pointing away from the bellow. This means the pre-tensioned spring force is reducing the force due to pressure that is acting from the bellow to the connected systems.



Spring force for 100% pre-tensioned axial bellow

For a lateral pre-tensioned bellow the spring force in pre-tensioned state is always pointing towards the center point of the neutral bellow.

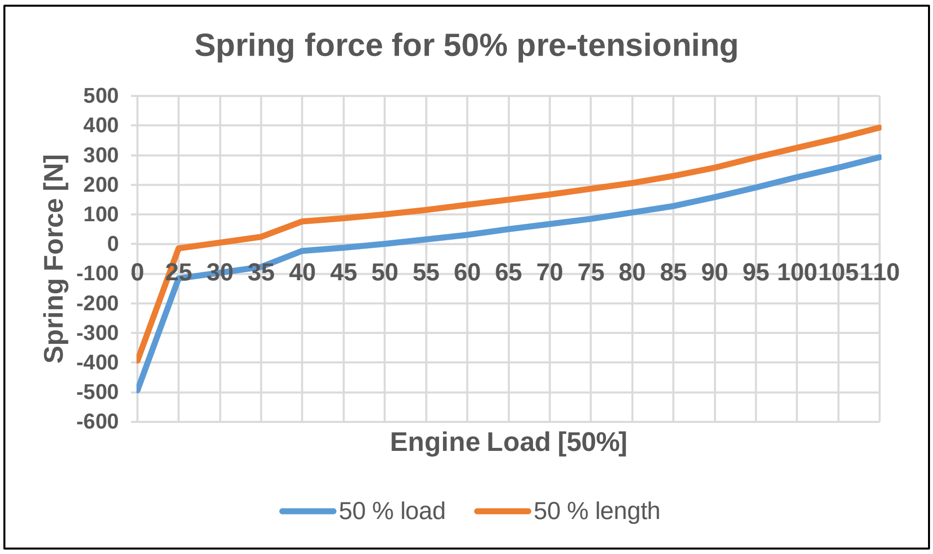


Spring force of a lateral pre-tensioned bellow

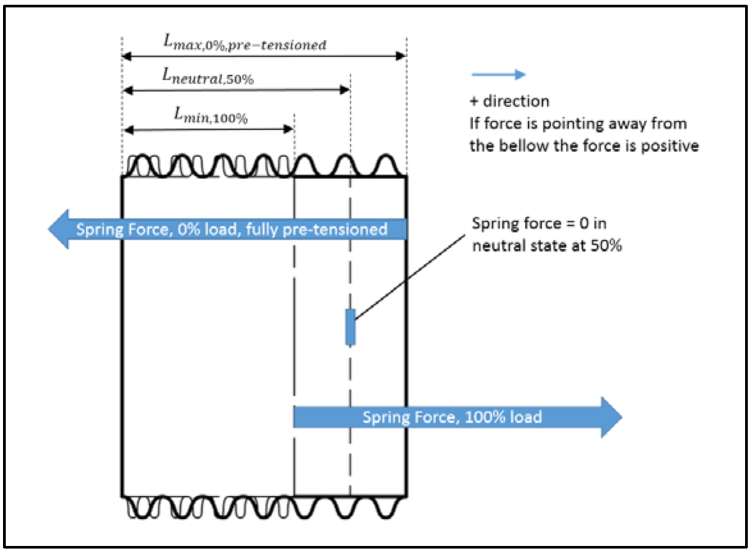
If the spring force of the bellow causes problems for the piping support it is recommended to pretension the bellow in this way.

8.3.2.2 Direction and value of axial spring force for 50% pre-tensioning

To pretension the bellow for 50% load is not the same as to pretension it for 50% of the maximum thermal expansion length. For pre-tensioning for 50% load the spring force is 0 N for 50% load. If the bellow is pre-tensioned for 50% of the maximum elongation the spring force is 0 N for a lower load. In the example chart it would be for approximately 30% load.



Spring force for 50% pre-tensioning of a bellow



Spring force of a 50% pre-tensioned bellow

Substitute for:					PC	Q-Code	X	X	X	X	X
Modif	Number	Drawn Date	Number	Drawn Date	Number	Drawn Date	Number	Drawn Date			
WIN GD Winterthur Gas & Diesel	Product W-2S	SCR Piping Guide for HP-SCR Systems									
Made	29.02.2016	M.Brutsche	Main Drw.	Page 52 / 90	Material ID PAAD219883						
Chkd	29.02.2016	D.Kadau	Design Group 8159	Drawing ID DAAD064155	Rev						
Appd	29.02.2016	M. Graf									

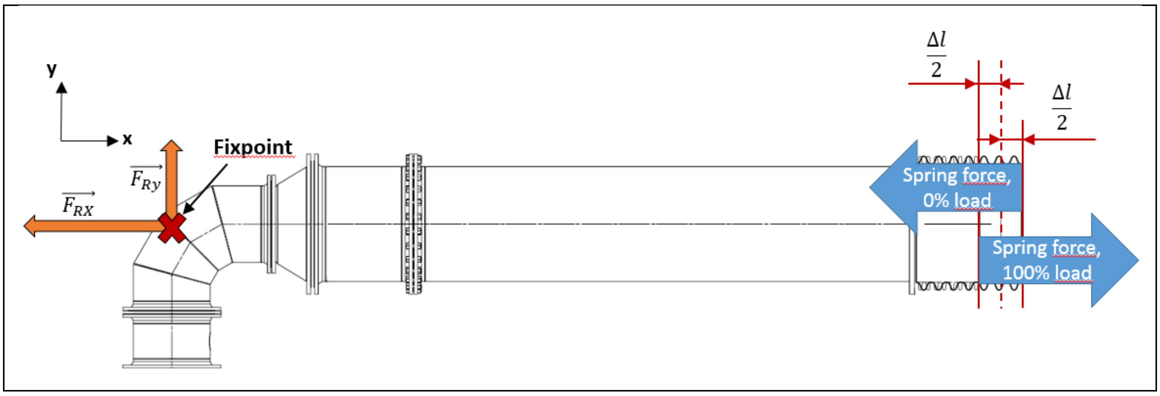
If the bellow is still in elongated in its pre-tensioning direction, the spring force is negative and pointing towards the bellow. This force pulls on the connected piping systems in the other direction of the inner pressure force. This spring force gets subtracted from the pressure force.

If the bellow is being compressed from its neutral state the force is positive and pointing away from the bellow. This force is pushing on the connected piping systems in the same direction as the inner pressure force. This force adds up to the inner pressure force.

8.3.2.3 Direction and value of lateral spring force for 50% pre-tensioning

For a lateral loaded bellow the same principles apply as for an axial loaded bellow, but the force has only one direction which is always pointing towards the center point of the bellow.

8.3.3 Calculation example for a 50% length pre-tensioned bellow



Example drawing of a piping segment with a 50% length pre-tensioned bellow

Assume a pipe segment as also shown earlier. an axial compensator with an axial spring constant $C_{S_{ax}} = 140\ N/mm$

As example assume a fully open bended system (piping segment) with a 90° pipe bend and a longer straight pipe as shown above. At the end of the straight pipe a compensator is mounted. This compensator has an axial spring constant $C_{S_{ax}} = 140\ N/mm$.

The total thermal elongation is assumed to be $\Delta l = 34.8\ mm$. The compensator is pre-tensioned by 50% ($\frac{\Delta l}{2}$) so that there is no spring force at all acting on the piping if the thermal elongation of $\frac{\Delta l}{2}$ is reached. The spring force of this compensator is pointing along the x-axis. The spring force has no component in y-direction, so the force on the fixpoint in y-direction due to the spring force of the compensator is 0. For 0% load the spring force is pointing in negative x-direction towards the fixpoint and for 100% the spring force is pointing in positive x-direction.

Calculation of the spring force acting on the fixpoint

For 0% as well as for 100% load the absolute value of the spring force is the same. Only the direction is different. The spring force can be calculated according equation (8):

$$F_S = C_{S_{ax}} \cdot \frac{\Delta l}{2} = 140 \cdot \frac{34.8}{2} = 4'872\ N$$

8.4 Forces acting on Fix Points of the Piping Support

All forces which are acting on the piping system which were described in the last subchapters have to be supported by the fix point support of a piping segment which is separated by two bellows.

For the calculation of the force that is acting on the fix point, all mentioned forces have to be added up.

8.4.1 Forces acting on a Fix Point due to friction in Sliding Supports

The weight force which is acting on the sliding supports (which are supporting the weight) of a piping segment that is separated by two bellows is causing a friction force. The counter force of this friction force is acting on the fix point of the supported piping segment.

The amount of the friction force depends on the amount of the weight force which is resting on a sliding point. If several sliding points are supporting a piping segment the friction force has to be calculated for each sliding point separately and then all the forces have to be added up to get the force acting on the fix point.

To calculate the weight which is acting on a sliding point it is recommended to use common literature about “Structural Analysis” or “Statics”.

If the weight which is supported by a sliding point is known, the friction force can be calculated as follows:

$$F_f = \mu \cdot m \cdot g \qquad (9)$$

<u>With the variables:</u>

F_f : Calculated friction force, in $[N]$

μ : Friciton coefficient, depending on the material combination of the materials which are in contact with each other. Unless it can be shown that the choice of sliding surfaces will provide a smaller coefficient consistently over the specified operating life of the piping, the friction coefficient shall be 0.3 (EN 13480-3, S. 13.5.5.5). Dimensionless.

m : Weight supported by a sliding point, in $[kg]$

g : Gravitational acceleration constant, recommended to use 9.81, in $\left[\frac{m}{sec^2}\right]$

If a sliding support does not support the weight of the piping, e.g. it is mounted vertically and tensioned to keep the piping from vibrating and tilting, the weight force ($m \cdot g$) has to be replaced by the tensioning force of the support. The friction force for this case can be calculated as follows:

$$F_f = \mu \cdot F_t \quad (10)$$

With the variables:

F_f : Calculated friction force, in $[N]$

μ : Friciton coefficient, depending on the material combination of the materials which are in contact with each other. Unless it can be shown that the choice of sliding surfaces will provide a smaller coefficient consistently over the specified operating life of the piping, the friction coefficient shall be 0.3 (EN 13480-3, S. 13.5.5.5). Dimensionless.

F_t : Tensioning force of the sliding support which is acting on the piping, in $[N]$

8.4.2 Weight force acting on a Fix Point

As a fix point always has to support also the weight of a pipe, this weight force has to be added to all the forces which are acting on the piping.

If a piping segment is supported by a fix point and one or several sliding points which are also supporting the weight of the segment, the partition of the weight which is acting on the fix point has to be calculated. For the method how this partition is calculated it is recommended to use common literature about “Structural Analysis” or “Statics”.

9 Piping Support

9.1 Layout of Piping Support

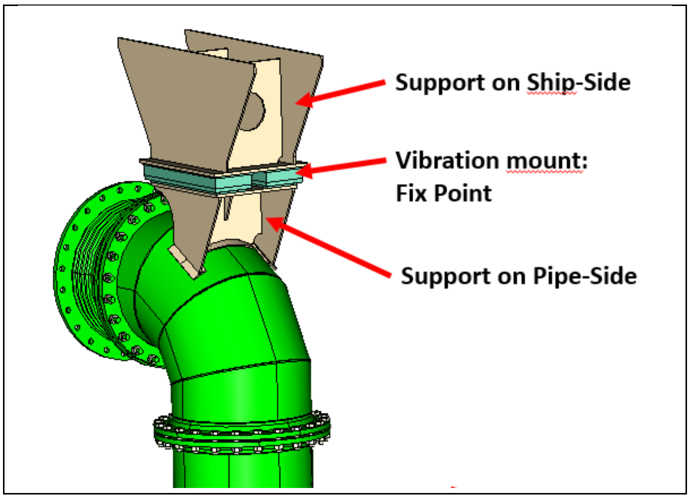
As the thermal expansion of the piping has to be compensated in the system by expansion bellows the SCR system has to be separated in smaller subsystems which are connected with bellows.
A subsystem has to be always supported by one fixed pipe support. If the piping of the subsystem could bend due to its' own weight and the possibility exists that an expansion bellow could carry weight forces of the piping, one or more sliding pipe supports have to be added to decrease the weight load on an expansion bellow to zero.

Composition of a piping support

A piping support consist of a support part which is mounted on the pipe side and another support part which is mounted on the ship side. Both support parts are bolted with each other.

Vibration mount

As the vibrations of the SCR system piping can be transmitted to the ship structure it is recommended to put a vibration mount at the interface between ship-side support and pipe-side support. This can be executed by welding the vibration mount to the ship-side support and mount the pipe-side support to the vibration mount by bolting.

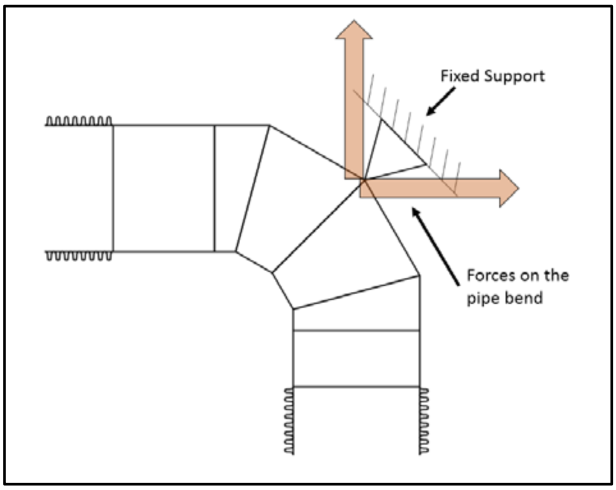


Layout of Piping Support

9.2 Support of Pipe Bend

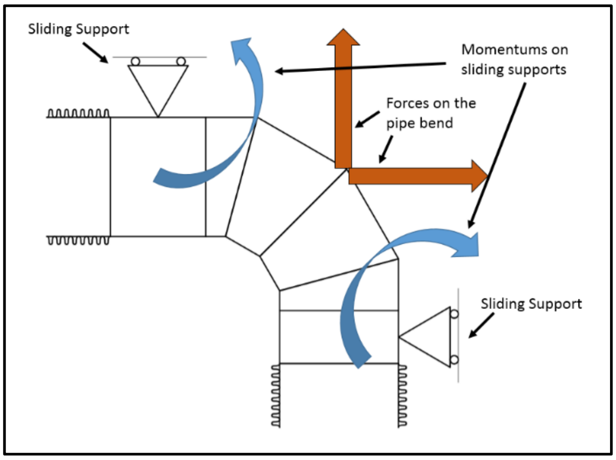
The most critical parts of the SCR piping system in relation to piping support are pipe bends. Due to the overpressure in the system very high forces can act on pipe bends (see also chapter about pipe forces). These forces have to be supported either by a fixed support or two sliding supports.

It is recommended to support a pipe bend always with a fixed support which is as close as possible to the spot where the forces are acting on the piping. This reduces momentum on the support itself.



Fixed Support of a Pipe Bend

If two sliding supports are used to support the forces on a pipe bend, momentums are induced on the sliding support which also counteract on the piping system. It is not recommended to use this method.



Double sliding support of Pipe Bend

Other supporting methods are not recommended as this would imply that the pipe bend is not support in both directions.
If a pipe bend is not supported in both directions the bend can move in one direction and the bellow which is perpendicular to this moving direction is therefore unnecessarily loaded in lateral direction.

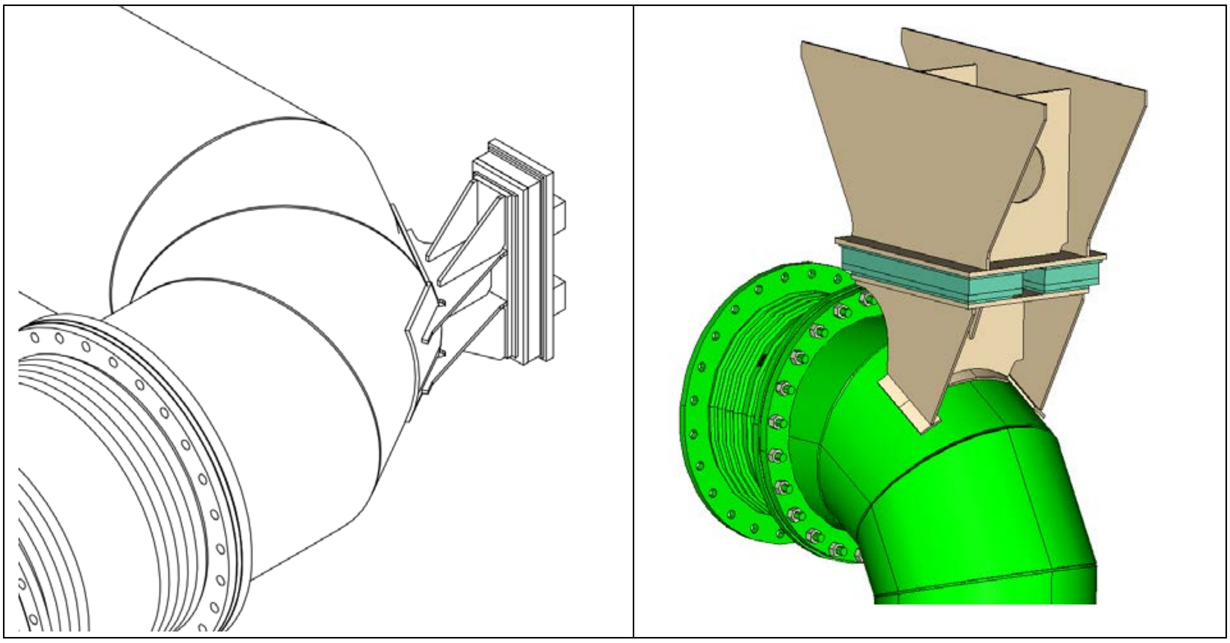
9.3 Fixed Pipe Support

A fixed support has to be designed strong enough to support the forces and the weight of the piping system without to allow the pipe to move in any direction. At the position of the fixed support all degrees of freedom of the pipe have to be limited. Also it should not be too rigid so that the pipe system is being deformed and huge stresses are introduced into the system.

For the ship-side support the structure has to be designed well to be capable to withstand the required load. A proper FEM calculation of the support structure is recommended.

The pipe-side support has to be designed well to be capable to withstand the required load. But also the connection to the pipe has to be designed and carefully calculated that no deformations or damages are caused to the pipe during operation. A proper FEM calculation is essential.

For a fixed pipe support the vibration mount is bolted to the pipe-side support with bolts mounted in normal bores (instead in long holes, etc. like for sliding support).



Examples for Fixed Support Executions

9.4 Sliding Pipe Support

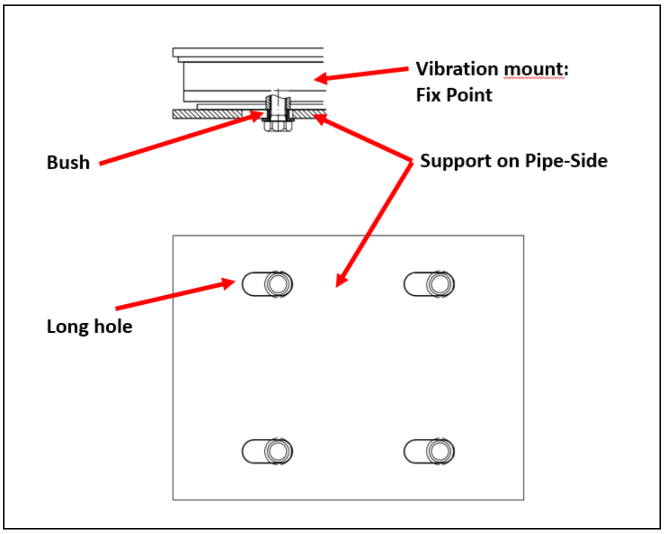
A sliding support only limits the movement of the piping in one or maximum two directions and allows the pipe to move freely in at least one direction.

In this free direction the thermal expansion of the pipe can take place.

For the ship-side support the structure has to be designed well to be capable to withstand the required load. A proper FEM calculation of the support structure is recommended.

The pipe-side support has to be designed well to be capable to withstand the required load. But also the connection to the pipe has to be designed and carefully calculated that no deformations or damages are caused to the pipe during operation. . A proper FEM calculation is essential.

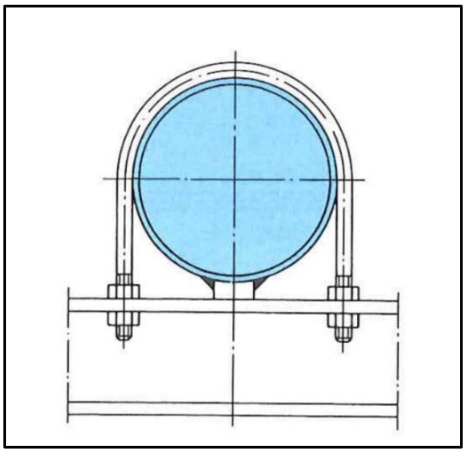
For a sliding pipe support the vibration mount is bolted to the pipe-side support with bolts mounted in bushes which can slide in long holes in the pipe-side support.



Example for a Sliding Pipe Support

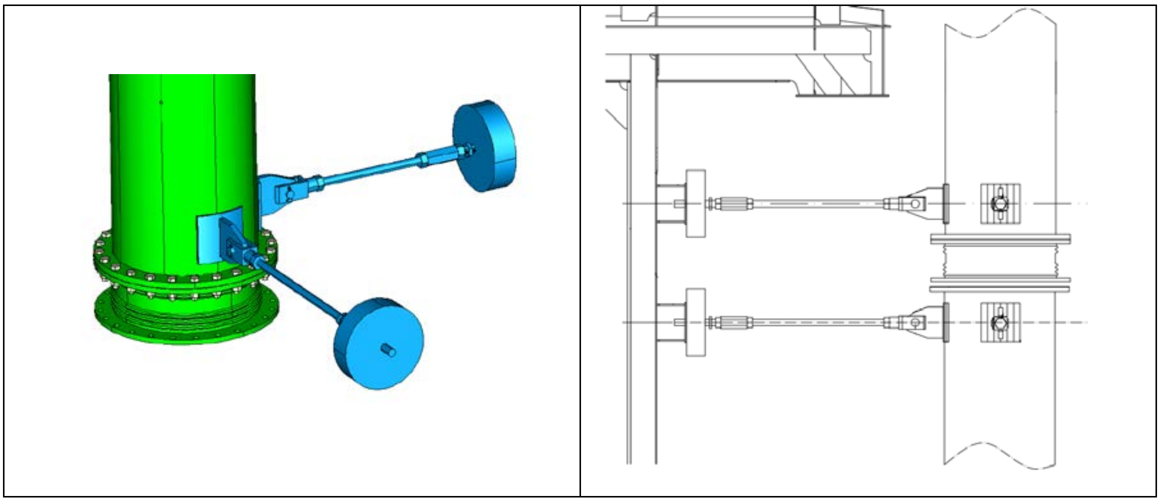
If it is not desired to use vibration mounts, for a horizontal pipe which is supported from below a sliding support it is possible to restrict the vertical elongation and movement of the pipe by a bended stay.

But the impact on the pipe and also the structure is also recommended to be calculated by FEM.



Example for a typical Sliding Support Execution

Another option as sliding point is an anchorage vibration mount. This consists of a stay which is connected to the pipe with a hinge. On the ship-side the stay is screwed into a vibration mount which is fixed on the ship-side stay.



Examples for Anchorage Vibration Mount which limits movement of the pipe in two directions

9.5 Further Information

For further information about piping support design and vibration mounts contact the desired supplier.

Recommended supplier for vibration mounting with experience in HP-SCR Systems:

Europe	Asia
SB Broneske GmbH (www.broneske.de) Ernst-Abbe-Strasse 9 D-25451 Quickborn/Hamburg Germany	**SB Broneske China Ltd.** (www.broneske.de) 7/F, China Overseas Building 76 Yanji Rd, Shibei District Qingdao China

10 Thermal Insulation

10.1 General Insulation Specification

For the SCR system insulation all SOLAS (Safety Of Life At Sea) and class rules have to be fulfilled.

For a detailed specification in regard of sound and heat insulation please see also WinGD drawing 107.079.071.500.

Generally, according to SOLAS, a temperature exceeding of 200°C is not accepted at all on any spot of the engine (SCR system included).

During operation, a maximum temperature of below 80°C is acceptable in areas where direct contact with the outer insulation sheet is possible.

10.2 SCR System Insulation Recommendations

It is recommended that the surface temperature of the of the insulation should never exceed 60°C. For calculations the 100% load point for tropical conditions should be considered. Detailed information about insulation has to be provided by SCR system supplier.

11 Pressure Drop over Piping Elements

Introduction

The installation of the SCR system between engine exhaust gas manifold and turbocharger inlet will cause a pressure drop, which can result in a loss of turbocharger efficiency, a drop in the mass flow of scavenge air, and an increased heat load of the components in the combustion chamber if the pressure drop is too high. Also the fuel consumption can be affected negatively. Consequently the maximum acceptable pressure drop of the entire SCR system (including valves V1 and V2) over its lifetime must be kept below a maximum acceptable limit.

Maximum allowable pressure drop

The value of the limit required from WinGD is **70 mbar** over the SCR system.

Definition of pressure drop over SCR system

The pressure drop over the SCR system is defined as the pressure drop from downstream of valve V1 to upstream of valve V2.

Further information and recommendations

In the following chapters methods are explained how the pressure drop over the SCR system can be calculated manually for a good estimation. But it is recommended that the calculated pressure drop is being validated by computational methods (CFD) after the design phase of the SCR system is finished. The final evaluated pressure drop has to be communicated to WinGD.

11.1 Calculation of the total Pressure Drop Δp_{total} over the SCR System

The total pressure drop Δp_{total} (in Pa) can be calculated as the sum of all pressure drops of the components of the SCR system. The important components for the pressure drop calculation are Mixing Pipe Δp_{MP}, SCR reactor $\Delta p_{Reactor}$ and the piping that connects all the parts Δp_{Pipe}.

$$\Delta p_{total} = \Delta p_{MP} + \Delta p_{Reactor} + \Delta p_{Pipe} \qquad (11)$$

11.2 Pressure Drop over Mixing Pipe Δp_{MP} and SCR Reactor $\Delta p_{Reactor}$

The pressure drop over the mixing pipe and SCR reactor depends on the inner design of these components, the urea injection nozzle design, inner components to direct the flow in order to get a good flow uniformity, specification and amount of catalyst elements, etc. A general calculation of the pressure drop over these components is not possible. The information of the pressure drop over these components have to be provided by the supplier of the mixing pipe and SCR reactor.

11.3 Pressure Drop Δp_{Pipe} over SCR Piping

The pressure drop Δp_{Pipe} (in Pa) is the total pressure drop of all pipe elements of the SCR system, including expansion bellows. To calculate this pressure drop the specific pressure drops for each pipe element is being calculated and all the separate values are then added up to the total pressure drop of the piping.

$$\Delta p_{Pipe} = \sum_{N=1}^{K} \Delta p_{element_N} \quad (12)$$

With the variables:

$\Delta p_{element_N}$:	Pressure drop over the N-th piping element, in $[mbar]$
N :	Index number of pipe elements of SCR piping system, dimensionless
K:	Total number of pipe elements of SCR piping system, dimensionless

In the following chapters, methods are explained how to calculate the pressure drop over different piping elements.

11.4 Pressure Drop over a straight Pipe

Due to friction at the inner wall of a straight pipe a pressure loss occurs.
This pressure loss (in Pa) can be calculated as follows:

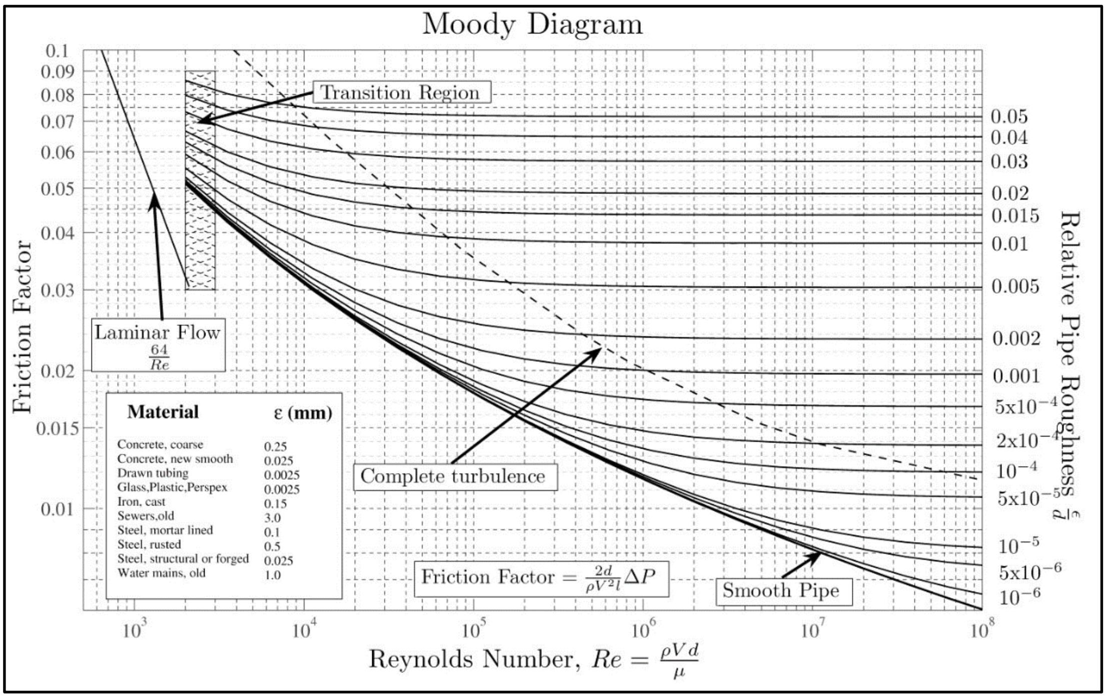
$$\Delta p_{straight} = \lambda \cdot \frac{L}{d_i} \cdot \rho \cdot \frac{\overline{w}^2}{2} \qquad (13)$$

With the variables:

λ : Pipe friction factor, dimensionless
L : Length of the straight pipe element, in $[mm]$
d_i : Inner diameter of the straight pipe element, in $[mm]$
$\overline{w}$: Medium flow velocity over the straight pipe element, in $\left[\frac{m}{sek}\right]$
For the calculation of $\overline{w}$ please see Annex.

11.4.1 Pipe Friction Factor λ

The pipe friction factor can be extrapolated from the so called “Moody Diagram” (see image below).



Moody Diagram

Substitute for:							PC	Q-Code	X	X	X	X	X
Modif	Number	Drawn Date	Number	Drawn Date	Number	Drawn Date	Number	Drawn Date					

WIN GD Winterthur Gas & Diesel	Product W-2S	SCR Piping Guide for HP-SCR Systems	
Made 29.02.2016 M.Brutsche	Main Drw.	Page 66 / 90	Material ID PAAD219883
Chkd 29.02.2016 D.Kadau	Design Group 8159	Drawing ID DAAD064155	Rev
Appd 29.02.2016 M. Graf			

To be able to derive the pipe friction factor from the Moody diagram the Reynolds number (Re) and the Relative Pipe Roughness (R_{rel}) have to be known. For the calculation of the Reynolds number please see the Annex of this guide.

The relative pipe roughness R_{rel} can be calculated as follows:

$$R_{rel} = \frac{\varepsilon}{d_i} \tag{14}$$

With the variables:

ε : Absolute inner surface roughness of the pipe, in $[mm]$
For an aged steel pipe (slightly corroded) for SCR application a factor of 0.2 can be used.

d_i : Inner diameter of the pipe, in $[mm]$

11.4.2 Calculation example for pressure drop over a straight pipe

Assume a new straight pipe segment with a surface roughness of $\varepsilon = 0.12\,mm$, a nominal diameter of DN900 and nominal pressure of PN10 with an inner diameter of $d_i = 894\,mm$ and the length of $L = 2.4\,m$. The flow velocity for which the pressure drop should be calculated is $\bar{w} = 35\ {}^{m}/_{sec}$ and the density of the exhaust gas is $\rho = 1.5\ {}^{kg}/_{m^3}$. The Reynolds number is $Re = 1.4 \cdot 10^6$.

To obtain the pipe friction factor, the relative pipe roughness has to be calculated according eq. (14):

$$R_{rel} = \frac{\varepsilon}{d_i} = \frac{0.12}{894} = 1.3 \cdot 10^{-4}$$

With this value and the Reynolds number the estimated pipe friction factor according the Moody diagram is:

$$\lambda \approx 0.014$$

With the pipe friction factor, the pressure drop over the straight pipe segment can be calculated:

$$\Delta p_{straight} = \lambda \cdot \frac{L}{d_i} \cdot \rho \cdot \frac{\bar{w}^2}{2} = 0.014 \cdot \frac{2.4}{0.894} \cdot 1.5 \cdot \frac{35^2}{2} = 34.5\ Pa = 0.345\ mbar$$

11.5 Pressure Drop over a Pipe Element

Similar to the friction factor λ of a straight pipe element all other pipe element that are not straight (like pipe bends, diffusor, etc) have a characteristic coefficient (ζ) which describes the pressure loss over this element.

The pressure drop (in Pa) over any not straight pipe element can be calculated as follows:

$$\Delta p_{elem} = \zeta \cdot \rho \cdot \frac{\overline{w}^2}{2} \qquad (15)$$

<u>With the variables:</u>

ζ : Resistance coefficient, specific for a pipe element, dimensionless

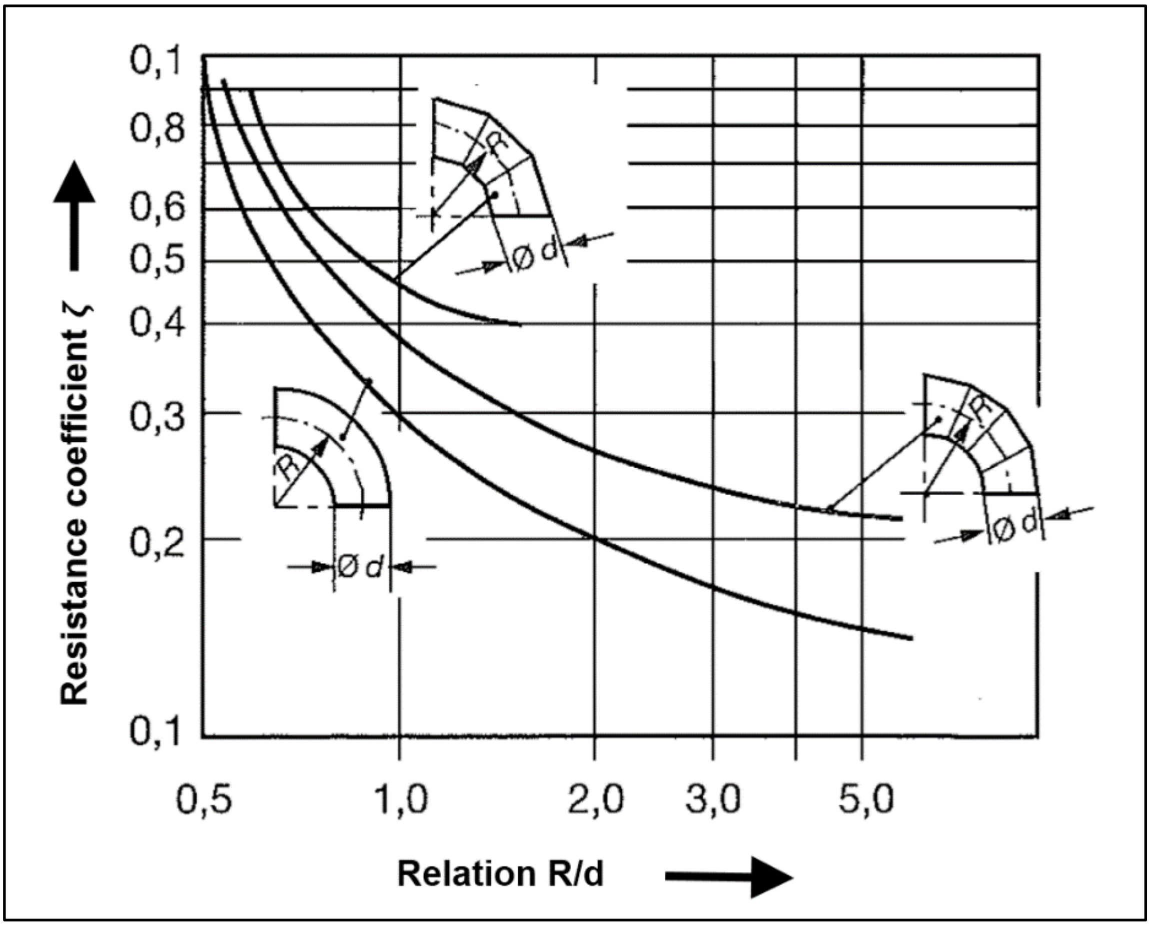
ρ : Density of the exhaust gas, in $\left[\frac{kg}{m^3}\right]$

$\overline{w}$: Medium exhaust gas flow velocity, in $\left[\frac{m}{sec}\right]$

11.5.1 Resistance Coefficient for Pipe Bends

The resistance coefficient for pipe bends depends on the Reynolds Number Re and the Relative pipe roughness R_{rel}, same as the pipe friction factor λ for straight pipes.

To make the calculation more simple the Resistance coefficient for pipe bends can be derived from the chart below with sufficient accuracy for a quick manual calculation. This chart is only valid for high Reynolds numbers ($Re > 2 \cdot 10^5$). For lower Reynolds numbers please see specific literature (Idelchik, 2007).



Resistance Coefficient for Pipe Bends

11.5.2 Calculation example for the pressure drop over a pipe bend

Assume a 90° pipe bend with 4 segments, a pipe diameter of $d = 914\ mm$ and a bend radius of $R = 1828\ mm$. The flow velocity is $\overline{w} = 35\ {}^{m}/_{sec}$ and the exhaust gas density is $\rho = 1.5\ {}^{kg}/_{m^3}$. The Reynolds number is $Re = 1.4 \cdot 10^6$.

The relation of bending radius to the diameter of the pipe bend is:

$$\frac{R}{d} = \frac{1828}{914} = 2.0$$

The resistance factor can be estimated with this relation from the above diagram:

$$\zeta \approx 0.25$$

With eq. (15) the pressure drop over this pipe bend can be calculated:

$$\Delta p_{elem} = \zeta \cdot \rho \cdot \frac{\overline{w}^2}{2} = 0.25 \cdot 1.5 \cdot \frac{35^2}{2} = 229.7\ Pa \approx 2.30\ mbar$$

11.5.3 Resistance Coefficient for conical Diffusor

The resistance coefficient depends on the opening angle φ of the diffusor and relation of the inlet and outlet diameter, as well as on the Reynolds Number.

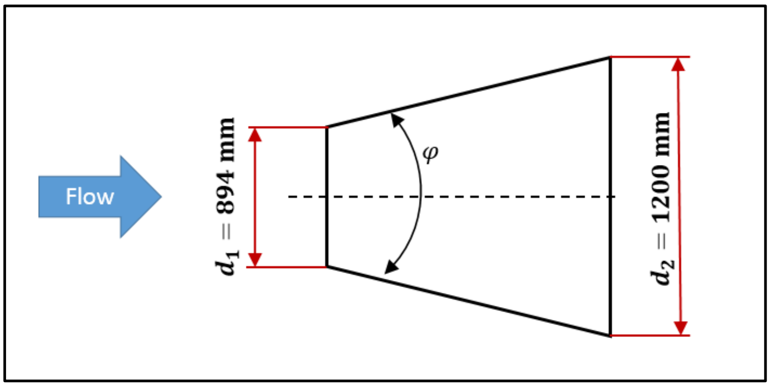
For a diffusor with an opening angle of $0 < \varphi < 40°$ the resistance coefficient can be calculated as follows (Bohl & Elmendorf, 2005, p.192):

$$\zeta_{Diffusor} = 3.2 \cdot \tan\frac{\varphi}{2} \cdot \sqrt[4]{\tan\frac{\varphi}{2}} \cdot \left(1 - \frac{d_1^2}{d_2^2}\right)^2 + \frac{\lambda}{8 \cdot \sin\frac{\varphi}{2}} \cdot \left[1 - \left(\frac{d_1^2}{d_2^2}\right)^2\right] \tag{16}$$

<u>With the variables</u>:

φ :	Opening angle of the diffusor, in [°]
d_1 :	Inner pipe diameter at the inlet of the diffusor, in $[mm]$
d_2 :	Inner pipe diameter at the outlet of the diffusor, in $[mm]$
λ :	Pipe friction factor for a straight pipe piece with the same inner pipe diameter as d_1 and the same Reynolds Number as for a straight pipe at the inlet of the diffusor. Dimensionless.

11.5.4 Calculation example for the pressure drop over a conical diffusor



Assume a conical diffusor with an opening angle of $\varphi = 30°$, an inner inlet diameter of $d_1 = 894\,mm$, an outlet diameter of $d_2 = 1200\,mm$ and an inner surface roughness of $\varepsilon = 0.12\,mm$. The flow velocity for which the pressure drop should be calculated is $\overline{w} = 35\ {}^{m}/_{sec}$ and the density of the exhaust gas is $\rho = 1.5\ {}^{kg}/_{m^3}$. The Reynolds number is $Re = 1.4 \cdot 10^6$.

To obtain the pipe friction factor, the relative pipe roughness has to be calculated according eq. (14):

$$R_{rel} = \frac{\varepsilon}{d_i} = \frac{0.12}{894} = 1.3 \cdot 10^{-4}$$

With this value and the Reynolds number the estimated pipe friction factor according the Moody diagram is:

$$\lambda \approx 0.014$$

With the pipe friction factor, the resistance factor for the conical diffusor can be calculated according eq.(16):

$$\zeta_{Diffusor} = 3.2 \cdot \tan\frac{30}{2} \cdot \sqrt[4]{\tan\frac{30}{2}} \cdot \left(1 - \frac{894^2}{1200^2}\right)^2 + \frac{0.014}{8 \cdot \sin\frac{30}{2}} \cdot \left[1 - \left(\frac{894^2}{1200^2}\right)^2\right]$$

$$\zeta_{Diffusor} = 0.127$$

With eq. (15) the pressure drop over this conical diffusor can now be calculated:

$$\Delta p_{elem} = \zeta \cdot \rho \cdot \frac{\overline{w}^2}{2} = 0.127 \cdot 1.5 \cdot \frac{35^2}{2} = 116.7\, Pa \approx 1.17\, mbar$$

11.5.5 Resistance Coefficient for Pipe Nozzle

For a pipe nozzle the resistance coefficient depends on the cone angle φ of the nozzle, the relation of the inlet and outlet diameter, as well as on the Reynolds Number.

As the flow is getting accelerated, the resistance coefficient splits in two parts, an acceleration part and a wall friction part. The total resistance can be calculated as follows:

$$\zeta = \zeta_a + \zeta_f \quad (17)$$

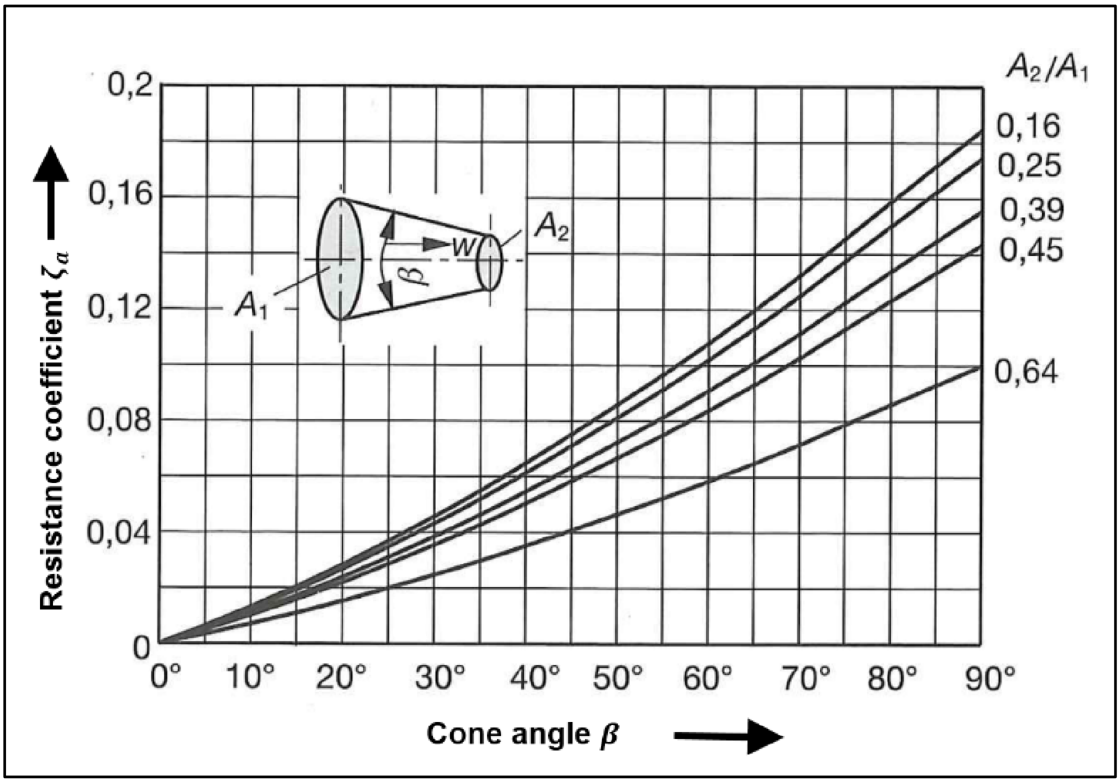
With the variables:

ζ_a : Resistance coefficient part due to acceleration of the exhaust gas, dimensionless

ζ_f : Resistance coefficient part due to friction on the inner wall of the pipe

The resistance coefficient due to acceleration ζ_a can be derived from the chart below. Please note, that the ratio of the areas is the same as the ratio for the squared diameters:

$$\frac{A_2}{A_1} = \frac{d_2^2}{d_1^2}$$



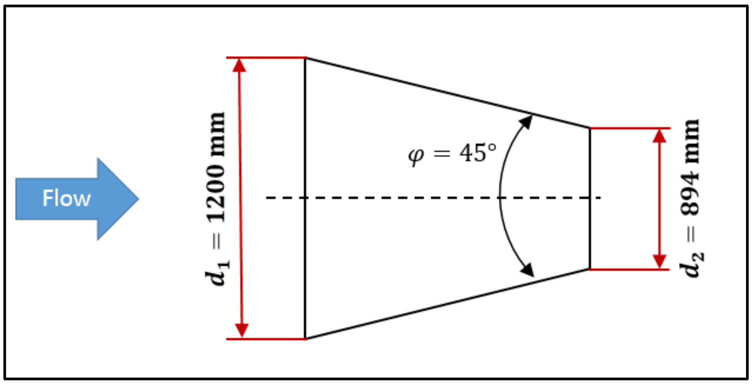
The resistance coefficient due to friction ζ_f can be calculated as follows:

$$\zeta_f = \frac{\lambda}{4 \cdot tan\frac{\beta}{2}} \cdot \frac{1 - \sqrt{\frac{d_2^2}{d_1^2}}}{2 + \sqrt{\frac{d_2^2}{d_1^2}}} \cdot \left(1 + \frac{d_2^2}{d_1^2}\right) \qquad (18)$$

With the variables:

λ :	Pipe friction factor for a straight pipe piece with the same inner pipe diameter as d_2 and the same Reynolds Number as for a straight pipe at the outlet of the nozzle. Dimensionless.
β :	Cone angle of the nozzle, in [°]
d_1 :	Inner pipe diameter at the inlet of the nozzle, in $[mm]$
d_2 :	Inner pipe diameter at the outlet of the nozzle, in $[mm]$

11.5.6 Calculation example for the pressure drop over a pipe nozzle



Assume a pipe nozzle with an angle of $\varphi = 45°$, an inner inlet diameter of $d_1 = 1200\,mm$, an outlet diameter of $d_2 = 894\,mm$ and an inner surface roughness of $\varepsilon = 0.12\,mm$. The flow velocity for which the pressure drop should be calculated is $\overline{w} = 35\ {}^{m}/_{sec}$ and the density of the exhaust gas is $\rho = 1.5\ {}^{kg}/_{m^3}$. The Reynolds number is $Re = 1.4 \cdot 10^6$.

To obtain the pipe friction factor, the relative pipe roughness has to be calculated according eq. (14):

$$R_{rel} = \frac{\varepsilon}{d_i} = \frac{0.12}{894} = 1.3 \cdot 10^{-4}$$

With this value and the Reynolds number the estimated pipe friction factor according the Moody diagram is:

$$\lambda \approx 0.014$$

To obtain the resistance coefficient due to acceleration ζ_a the ratio of the squared diameters has to be calculated:

$$\frac{d_2^2}{d_1^2} = \frac{894^2}{1200^2} \approx 0.56$$

From the diagram above, the resistance factor can be estimated:

$$\zeta_a \approx 0.05$$

The resistance coefficient due to friction ζ_f can be calculated according eq. (18):

$$\zeta_f = \frac{0.014}{4 \cdot tan\frac{45°}{2}} \cdot \frac{1 - \sqrt{\frac{894^2}{1200^2}}}{2 + \sqrt{\frac{894^2}{1200^2}}} \cdot \left(1 + \frac{894^2}{1200^2}\right) = 0.0012$$

The total resistance factor can be calculated according eq.(17):

$$\zeta = \zeta_a + \zeta_f = 0.05 + 0.0012 = 0.0512$$

Now, the pressure drop over the pipe nozzle can be calculated according eq. (15):

$$\Delta p_{pipe\ nozzle} = \zeta \cdot \rho \cdot \frac{\overline{w}^2}{2} = 0.0512 \cdot 1.5 \cdot \frac{35^2}{2} = 47.04\ Pa \approx 0.47\ mbar$$

11.6 Pressure Loss over Expansion Bellow

If pressure loss is an issue, it is recommended to have an inner sleeve for all bellows of the SCR system. The pressure loss can be calculated in the same way as for a straight pipe. As diameter for the calculation the inner diameter of the inner sleeve has to be taken.

If a bellow without inner sleeve is being used the supplier of the bellow has to be contacted for advice regarding the calculation of the pressure loss.

11.7 Further Information about Pressure Drop

For further detailed information regarding pressure drop calculation as well as for geometries not covered in this document or cases for which the provided equations do not apply, please see special literature. Some references can be found in the reference list (Idelchik, 2007), (Wagner, Strömung und Druckverlust, 2001).

12 Vibrations

12.1 Modalanalysis

For assuring that none natural frequency of the SCR's piping system is excited by engine-born stimulations it is recommended to perform a modal analysis of the piping structure. Its first eigenfrequency should be larger than the 18th order stimulation frequency at full engine load:

$$f_{Eigen,1st} > 18 \cdot \frac{rpm_{Engine}}{60} \quad (19)$$

With the variables:

$f_{Eigen,1st}$: Calculated first Eigenfrequency of the piping segment, in $[Hz]$

rpm_{Engine} : maximum engine speed in rpm at 100% engine load, in $\left[\frac{1}{min}\right]$

12.2 Forced Response Analysis

In addition to the modal analysis a forced response analysis is recommended. This type of calculation reveals how the structure reacts to stimulating forces and momentums and allows to predict resonance amplitudes. The simulation allows to estimate if the SCR system and its components withstand vibrations that are excited by e.g. diesel engines, propellers, and other stimulating sources. If not requested differently (by yard, classes, or ship designer) the following limits are recommended for SCR system piping structure which is not mounted on the engine (i.e. mounted on ship side):

Frequency range (in any direction)	Sinusoidal vibration (zero to peak)	Root Mean Square (RMS)
2.0 Hz ... 6.4 Hz	amplitude ± 1.0 mm	0.71 mm
6.4 Hz ... 27.3 Hz	velocity ± 40 mm/sec	28 mm/sec
27.3 Hz ... 100 Hz	acceleration ± 0.7 g	0.5 g

For effective vibration (broadband) measurement the RMS vibration levels shall be applied as limits.
These vibration levels correspond to ISO 10055 and IACS UR E10, Category 1 "other equipment and machinery components" with the only difference that the limit values for the mid-frequency range are reduced to 40 mm/sec.

13 Design Standards

Bellow:	EJMA standard (www.ejma.org), ASTM F1120-87, DIN EN 13480-3 Annex C
Bolting:	EN 1515-1
Bolting material:	EN 10269
Flanges:	DIN EN 1092-1
Flanges/Gaskets:	DIN 28090-2
Gaskets:	DIN EN 1514-1
Piping:	EN 13480-3
Tolerances	DIN ISO 286-1
Valves:	VDI 2173 DIN EN 60534-1
Vibrations:	ISO 10055 IACS UR E10
Welding preparation:	DIN EN 29692
Welding Process:	ISO 4063
Welding symbols:	ISO 2553

14 Abbreviations

BSEF	Break specific exhaust gas flow
BSFC	Break specific fuel consumption
CFD	Computational Fluid Dynamics
FEM	Finite Element Method
HFO	Heavy Fuel Oil
HNCO	Isocyanic acid
MDO	Marine Diesel Oil
MGO	Marine Gas Oil
NECA	NOx Emission Control Area
NH_3	Ammonia
NOx	Nitrogen Oxides
pExh	Exhaust gas pressure
RMS	Root mean square
SAC	Scavenge Air Cooler
SAR	Scavenge Air Receiver
SCR	Selective Catalytic Reduction
SCR-VCS	SCR Valve Control System
SI	French: Système International d'unités, English : International Unit System
SO_2	Sulfur dioxide
SO_3	Sulfur trioxide
SOLAS	Safety Of Life At Sea, UN convention
tEbT	Temperature of exhaust before turbocharger
UWS	Urea water solution
WinGD	Winterthur Gas & Diesel Ltd.

15 List of References

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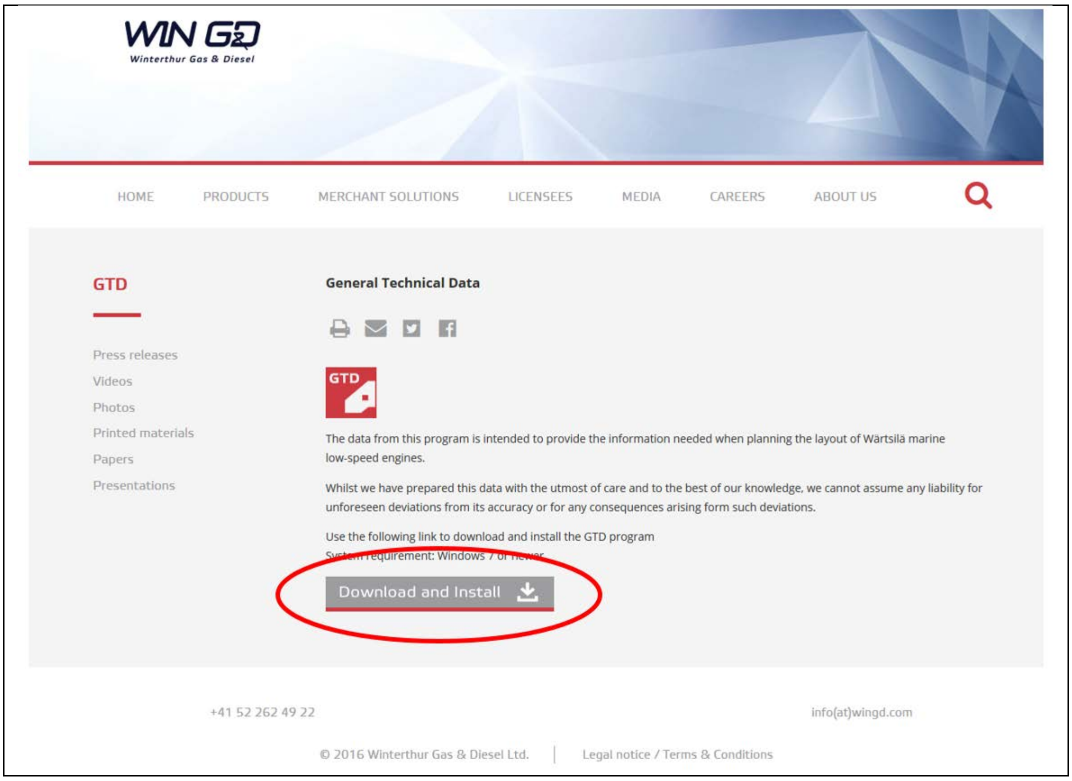
16 ANNEX

16.1 How to get WinGD General Technical Data Tool

The WinGD General Technical Data Tool can be downloaded and installed from the following site:

https://www.wingd.com/en/media/general-technical-data/



Substitute for:							PC	Q-Code	X	X X X X
Modif	Number	Drawn Date		Number	Drawn Date		Number	Drawn Date	Number	Drawn Date

WIN GD Winterthur Gas & Diesel	Product **W-2S**		**SCR Piping Guide** **for HP-SCR Systems**
Made 29.02.2016 M.Brutsche	Main Drw.	Page 82 / 90	Material ID **PAAD219883**
Chkd 29.02.2016 D.Kadau	Design Group **8159**	Drawing ID **DAAD064155**	Rev
Appd 29.02.2016 M. Graf			

16.2 Deriving Exhaust Gas Pressure p and Temperature T from General Technical Data

The exhaust gas pressure p and temperature T can be derived for ISO and other conditions from the Exhaust Gas Data Sheet of the General Technical Data Tool for the specific project.

ISO | Design | Winter

Exhaust Gas

coolant temperature before SAC: 29 °C
air temperature before compressor: 25 °C

Power %	Power kW	Speed rpm	Bypass %	tEbT °C	tEaT °C	tEbE °C	pExh bar	Steam kg/h	Urea l/h
110	11000	108.4	4.0	475	287	293	4.28	3901	173
100	10000	105.0	4.0	447	269	274	4.01	3116	171
95	9500	103.2	4.0	432	261	266	3.85	2740	170
90	9000	101.4	4.0	416	253	257	3.66	2382	169
85	8500	99.5	4.0	400	245	249	3.46	2046	167
80	8000	97.5	0.0	381	228	226	3.59	1357	164
75	7500	95.4	0.0	368	222	220	3.40	1150	162
70	7000	93.2	0.0	355	219	217	3.18	983	160
60	6000	88.6	0.0	335	216	214	2.75	794	152
50	5000	83.3	0.0	322	221	219	2.36	777	140
40	4000	77.4	0.0	315	237	235	1.93	885	125
30	3000	70.3	5.6	310	269	269	1.54	1091	104
25	2500	66.1	4.6	310	273	272	1.37	947	91

The exhaust gas pressure p can be derived from the column "pExh" for the the load that wants to be calculated. The displayed value is the absolute pressure $[bara]$. If the relative pressure to the atmosphere $[barg]$ is needed, 1 bar has to be substracted from this value.

The exhaust gas temperature T can be derived from the column "tEbT" for the load that wants to be calculated.

16.3 Calculation of the Mass Flow Rate $\dot{m}$ of the Exhaust Gas

The unit of the mass flow rate is $\left[\frac{kg}{h}\right]$.

The mass flow rate is provided by WinGD in the Engine Performance Data sheet of the General Technical Data Tool for the specific project.

Engine Performance

Power	Performance Power	Speed	MEP	BSFC	ISO BSEF	tEaTm	BSFC	Design BSEF	tEaTm	BSFC	Winter BSEF	tEaTm
%	kW	rpm	bar	g/kWh	kg/kWh	°C	g/kWh	kg/kWh	°C	g/kWh	kg/kWh	°C
110	11000	108.4	19.08	168.6	6.81	295	171.6	6.39	335	166.4	7.08	274
100	10000	105.0	17.90	167.6	7.16	276	170.6	6.69	317	165.4	7.47	255
95	9500	103.2	17.30	165.4	7.30	268	168.4	6.81	309	163.2	7.63	246
90	9000	101.4	16.69	163.2	7.42	259	166.2	6.90	301	161.0	7.78	237
85	8500	99.5	16.07	161.8	7.51	251	164.8	6.97	294	159.6	7.90	228
80	8000	97.5	15.43	160.6	8.03	228	163.6	7.45	267	158.4	8.43	206
75	7500	95.4	14.78	160.0	8.22	222	163.0	7.60	263	157.8	8.66	200
70	7000	93.2	14.12	159.3	8.31	219	162.3	7.66	260	157.1	8.79	196
60	6000	88.6	12.74	157.5	8.49	216	160.5	7.76	260	155.4	9.02	192
50	5000	83.3	11.28	159.0	8.73	221	162.0	7.91	268	158.3	8.95	215
40	4000	77.4	9.72	161.7	8.77	237	164.7	7.86	287	161.3	8.78	239
30	3000	70.3	8.02	166.6	8.83	271	166.5	9.01	267	166.5	8.71	276
25	2500	66.1	7.11	167.1	8.90	274	167.6	8.95	276	167.0	8.76	279

With these information the mass flow rate for a given state can be calculated as follows:

$$\dot{m} = P_{engine} \cdot BSEF \qquad (20)$$

<u>With the variables:</u>

P_{engine} : Power of the engine in $[kW]$ for the state that wants to be calculated.

$BSEF$: Brake Specific Exhaust gas Flow in $\left[\frac{kg}{kW \cdot h}\right]$, that value which corresponds to the chosen Power of the engine has to be used. Unless it is wanted to calculate a specific case, it is recommended to use the BSEF value for Reference condition (ISO condition).

16.4 Calculation of the Density ρ of the Exhaust Gas Flow

The unit of the exhaust gas density is $\left[\frac{kg}{m^3}\right]$. The exhaust gas density for a given state depends on the pressure and the temperature of the exhaust gas. The density is being calculated as follows:

$$\rho = \frac{p \cdot 100'000}{R \cdot (273.15 + T)} \qquad (21)$$

With the variables:

p : Absolute pressure of the exhaust gas in $[bar]$
R : Specific gas constant of the exhaust gas, recommended to use 288 $\frac{J}{kg \cdot K}$
T : Temperature of the exhaust gas in $[°C]$

16.5 Calculation of the Volume Flow Q of the Exhaust Gas

Q is the volume flow rate in $\left[\frac{m^3}{h}\right]$ for the state of the valve that wants to be calculated.

The volume flow rate can by calculated from the mass flow rate as follows:

$$Q = \frac{\dot{m}}{\rho} \qquad (22)$$

16.6 Calculation of the medium Exhaust Gas Velocity $\overline{w}$ in a Pipe

$\overline{w}$ is the medium exhaust gas velocity (in $\left[\frac{m}{sec}\right]$) in a pipe.
It can be calculated as follows:

$$\overline{w} = \frac{4}{\pi} \cdot \frac{Q}{d_i^2} \qquad (23)$$

With the variables:

Q: Volume flow rate of the exhaust gas flow, in $\left[\frac{m^3}{sec}\right]$
d_i : Inner diameter of the pipe, in $[m]$

16.7 Calculation of Kinematic Viscosity υ of the Exhaust Gas

There is not an exact formula for calculation of the kinematic viscosity for the exhaust gas of a 2-stroke Diesel engine. But a calculation formula for air (Bohl & Elmendorf, 2005) is providing a sufficient precision for a manual calculation.

$$\nu = \frac{418.45}{p} \cdot \frac{(T + 273.15)^{\frac{5}{2}}}{T + 383.55} \cdot 10^{-6} \ \left[\frac{m^2}{sec}\right] \quad (24)$$

With the variables:

p : Absolute exhaust gas pressure, in $[Pa]$
T : Exhaust gas temperature, in $[°C]$

16.8 Calculation of Reynolds Number Re in a Pipe

The Reynolds number in a pipe can be calculated as follows:

$$Re = \frac{\overline{w} \cdot d_i}{\nu} \quad (25)$$

With the variables:

$\overline{w}$: Medium flow velocity in the pipe, in $\left[\frac{m}{sec}\right]$
d_i : Inner diameter of the pipe, in $[m]$
υ : Kinematic viscosity of the exhaust gas, in $\left[\frac{m^2}{sec}\right]$

16.9 Determination of the thermal expansion of a material

The solution of the differential equation (1b) can be used to determine the thermal elongation of a material. Separation of variables and integration of equation (1b) leads to:

$$\int_{T_{min}}^{T_{max}} \alpha(T)\, dT = [\ln l]_{T_{min}}^{T_{max}} \tag{30}$$

Please note the absence of an integration constant. By choosing the boundary condition $\Delta l = 0$ if $\Delta T = 0$ it can be shown, that the coefficient vanishes.

If it is assumed, that the temperature dependent function of α is of the linear form:

$$\alpha(T) = k_1 \cdot T + k_2 \tag{31}$$

The solution of equation (30) is:

$$k_1 \cdot (\Delta T)^2 + k_2 \cdot \Delta T = \ln l_{T_{max}} - \ln l_{T_{min}} \tag{32}$$

By using logarithm rules and the fact that $l_{T_{min}}$ is the original length l, as well as $l_{T_{max}} = l + \Delta l$ (with Δl as the thermal elongation of the material):

$$k_1 \cdot (\Delta T)^2 + k_2 \cdot \Delta T = \ln \frac{l + \Delta l}{l} = \ln\left(1 + \frac{\Delta l}{l}\right) \tag{33}$$

The solution of the thermal expansion for a material with a linear characteristic of the thermal expansion coefficient like in equation (31) therefore is:

$$\Delta l = \left[e^{\left(k_1 \cdot (\Delta T)^2 + k_2 \cdot \Delta T\right)} - 1\right] \cdot l \tag{34}$$

The determination of the thermal expansion for a material with a non-linear behaviour of the thermal elongation coefficient can be conducted analogous.

16.10 Material specification of S355J2G1W

Ferrous Materials Unalloyed (FU) — W-FU-355-J2W

Keywords
- Structural steel with improved corrosion resistance
- Normalised steel

Info

If the responsible classification society rules or specific requirements on drawing or component related documents deviate from values specified in this data sheet, then these rules or requirements must be followed.
For further general information regarding this document, please refer to DAAD799998.

Chemical Composition

The following data cover all materials included in "Similar Standards". Within the grey area the values are given as defined in the highlighted standard in "Similar Standards".
Acc. min./max.: accepted limits for chemical composition according to "Similar Standards", as long as mechanical properties are fulfilled

[% by mass]	C	Si	Mn	P	S	Cr	Mo	Ni	V	Ti	Cu	Al	Nb					
Acc. Min.		0,15	0,50			0,30				0,020	0,20	0,020	0,015					
Min.		**0,30**	**0,80**			**0,40**			**0,020**		**0,25**							
Max.	**0,19**	**0,65**	**1,25**	**0,035**	**0,035**	**0,65**		**0,40**	**0,10**		**0,65**							
Acc. Max.			1,40	0,040	0,050	0,80	0,30	0,65	0,12	0,10		0,060	0,060					

Comments:
- Fe: remainder

Mechanical Properties

The raw material can be purchased with or without heat treatment. Heat treatment which determines the final mechanical properties can also be performed in pre-machined state.
Unless different values are given on the drawing or in the material specification, all mechanical properties are mandatory for final products.
Only in case tensile tests or impact tests are requested by Wärtsilä, the values in 1) are mandatory. In any other case the values in 2) apply and the values in 1) are only for information.

1)										
Material Condition										
d / t	[mm]									
R_m	[MPa]		≥ 485							
Rp0.2	[MPa]		≥ 345							
A	[%]	long.	≥ 19							
		tang.								
		trans.								
Z	[%]	long.								
		tang.								
		trans.								
ISO-V [a]	[J]	long.								
		tang.								
		trans.								
Hardness										

[a] *Temperature for measurement of impact energy:*

2)	*R_m values converted into HBW according to ISO 18265:2004*									
R_m	[MPa]									
Hardness		calc.								

Comments:

Modif									
	Number	Drawn Date		Number	Drawn Date		Number	Drawn Date	

WÄRTSILÄ	Product: Basic Mat	Material Specification W-FU-355-J2W	
Made	23.04.2014 sth017 S. Thalmann	Main Drw	Page 1/2	Material ID PAAD700672
Chkd	23.04.2014 wlu003 W. Luft	Design Group 0200	Drawing ID DAAD700672	Rev -
Appd	24.04.2014 rbi002 R.Bianchin			

Forming and Heat Treatment

Only heat treatment processes have to be performed that are mentioned on the drawing and/or in the material and testing specification of the part.

The given temperatures are for guidance only. The temperature must be chosen part-related.

Process	Description

Similar Standards

The similar standards below are acceptable as long as the mandatory requirements in chemical composition and mechanical properties section are fulfilled.

Grade	Standard	Comments
CORTEN B	not standardized	
S355J2G1W	DIN EN 10025-1:2005-02	
SMA490CW	JIS G 3114:2008-11	C-equi. min: 0,113 C-equi. max:0,617
15MnCuCr	GB 4172:2000	C-equi. min: 0,323 C-equi. max:0,563
A588GradeA	ASTM A 588/A 588M:2010	C-equi. min: 0,234 C-equi. max:0,602

Obsolete Standards

Grade	Standard	Comments

Additional Information

Attribute	Description
Forming	well suitable for bending, beading and similar forming
Weldability	weldable (see Comments Similar Standards)

Modif		Number	Drawn Date		Number	Drawn Date		Number	Drawn Date

WÄRTSILÄ

Product: Basic Mat	Material Specification W-FU-355-J2W		
Made	23.04.2014 sth017 S. Thalmann	Main Drw	Page 2/2	Material ID PAAD700672
Chkd	23.04.2014 wlu003 W. Luft	Design Group 0200	Drawing ID DAAD700672	Rev -
Appd	24.04.2014 rbi002 R.Bianchin			

Substitute for:									PC	Q-Code	X	X	X	X	X
Modif	Number	Drawn Date		Number	Drawn Date		Number	Drawn Date		Number	Drawn Date				

WIN GD Winterthur Gas & Diesel	Product W-2S		SCR Piping Guide for HP-SCR Systems	
Made	29.02.2016	M.Brutsche	Main Drw.	Page 90 / 90	Material ID PAAD219883
Chkd	29.02.2016	D.Kadau	Design Group 8159	Drawing ID DAAD064155	Rev
Appd	29.02.2016	M. Graf			

CONCEPT-GUIDANCE_WinGD-2S_SCR-Installation

TRACK CHANGES

DATE	SUBJECT	DESCRIPTION
2017-08-07	GUIDANCE	First web upload
2017-09-05	DAAD064155	Guidance – new document revision