

Navigation

- [next](#)
- [previous](#) |
- [Pneumatic Library](#) »
- [8. Modeling Pneumatic pipes](#)

8. Modeling Pneumatic pipes

8.1. Equations

The physical properties of a gas, and in particular:

- its density ρ ,
- its compressibility B ,
- its viscosity μ

influence the transient behavior in pipes in ways which depend on whether they are all important or some of them dominate. Thus, the gas in a pipe can be assumed to be purely compressive, dissipative or inertial or various combinations of the three.

The phenomena introduced by long pipes may have crucial consequences in some industrial applications. In the truck or railway industry, for example, brake systems use pneumatic energy and the dynamics of this system are vital for the safety and the robustness of the vehicle.

The equations used to model pneumatic pipes are known as the conservation equations of a compressible fluid. In the one-dimensional case:

Mass conservation

$$(8.1) \quad \frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} = q_s$$

where ρ is the gas density, u the gas velocity and q_s the flow source term.

Momentum conservation

$$(8.2) \quad \frac{\partial \rho u}{\partial t} + \frac{\partial \rho u^2}{\partial x} + \frac{\partial P}{\partial x} = \frac{\partial \tau}{\partial x}$$

where:

$$(8.3) \quad \frac{\partial \tau}{\partial x} = -\rho \cdot f \cdot \frac{u |u|}{2d}$$

Energy conservation

$$(8.4) \quad \frac{d\rho \cdot e}{dt} = -P \cdot \frac{\partial u}{\partial x} + R + \Phi + \lambda \cdot \frac{\partial^2 T}{\partial x^2} - h_T \cdot (T - T_{\text{out}})$$

where:

- e is the internal energy,
- R is the chemical, radiation, phase change power,
- F is the power of the local viscosity forces,
- λ is the thermal conduction,
- h_T is the thermal exchange coefficient.

The lumped parameter pneumatic pipe models use a cascade of pressure drops, compressibility and inertia effects.

8.1.1. Dissipative effect R

Submodels belonging to this category are used to model resistance to flow in straight tubes and conduits. The pressure losses along a straight tube of constant cross-sectional area are calculated from the Darcy-Weisbach equation:

$$(8.5) \quad \Delta p = \lambda \frac{l}{D_h} \cdot \frac{\rho Q^2}{2A_{\text{min}}^2}$$

where:

- λ is the friction coefficient of the segment of relative unit length $\frac{l}{D_h} = 1$;
- D_h , the hydraulic or equivalent diameter;
- and l the length of flow segment.

The total friction factor ζ is defined by:

$$(8.6) \quad \zeta = \lambda \frac{l}{D_h}$$

In straight tubes, the resistance to the motion of a liquid or a gas under conditions of **laminar flow** is due to the force of internal friction. This happens when one layer of the liquid (or gas) has a relative motion compared to the others. These viscosity forces are proportional to the first power of the flow velocity. We then have:

$$(8.7) \quad \lambda \equiv \lambda(Re)$$

As the Reynolds number increases, the inertia forces, which are proportional to the velocity squared, begin to dominate. As **flow becomes turbulent**, there is a significant increase in the resistance to the motion. Part of this increase is due to the roughness of the wall surface.

Therefore we have:

$$(8.8) \quad \lambda \equiv \lambda(Re, rr)$$

where rr is the relative roughness calculated as the ratio of the average height of asperities to the tube diameter. See details in [Fig. 8.1](#):

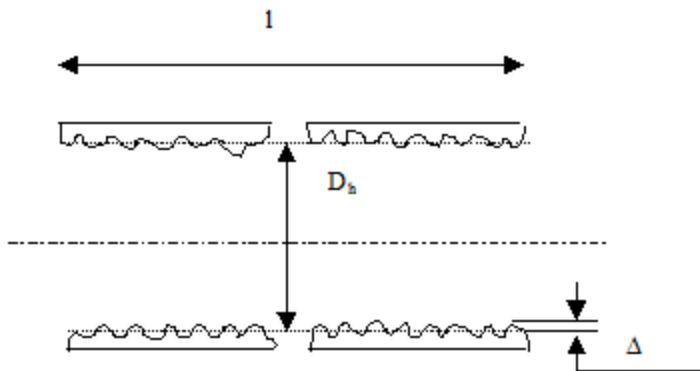


Fig. 8.1 relative roughness definition [1](#)

The relative roughness of a pipe is given by:

$$(8.9) \quad rr = \frac{\Delta}{D_h}$$

where:

- Δ is the equivalent uniform roughness of wall;
- and D_h the hydraulic diameter of the pipe.

A sampling of absolute pipe roughness Δ for new clean pipes is proposed by Binder [\[1\]](#):

Table 8.1 absolute roughness for various
pipe materials [1](#)

Pipe material Absolute roughness

Drawn brass 1.5 μm

Drawn copper 1.5 μm

Commercial steel 45 μm

Wrought iron 45 μm

Asphalted cast iron 120 μm

Galvanized iron 150 μm

Cast iron 260 μm

Wood stave 0.2 to 0.9 mm

Concrete 0.3 to 3.0 mm

Riveted steel 0.9 to 9.0 mm

The dependence of the friction coefficient λ on the Reynolds number and the relative roughness as shown in [Fig. 8.2](#) is often known as the harp of Nikuradse.

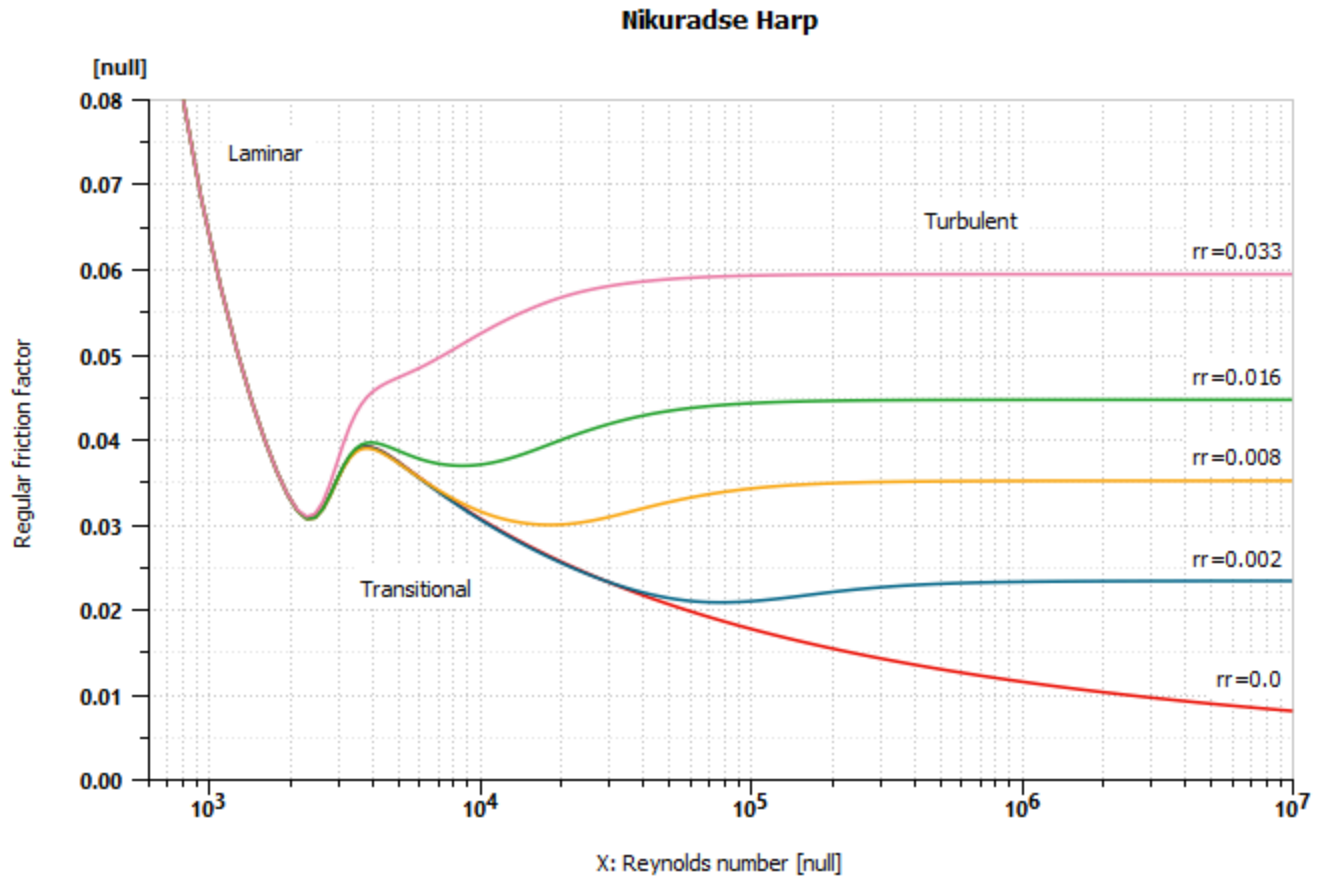


Fig. 8.2 evolution of the frictional drag factor with the Reynolds number and the relative roughness [1](#)

8.1.2. Compressive effect [C1](#)

The compressibility of the gas in the cell is taken into account using either a polytropic model or a heat exchange model (preferred).

The polytropic approach is much easier for analysis of the dynamic behavior of the pneumatic pipes. Indeed, in this particular case, the speed of sound in the gas only depends on its temperature T . However, for general studies, it is less representative of reality than the heat exchange approach.

By analogy with hydraulic systems, the bulk modulus of the gas can be estimated by differencing the polytropic law $P \cdot V^k = \text{constant}$.

$$\frac{dP}{P} + K \cdot \frac{dV}{V} = 0 \text{ and therefore, since } \frac{dP}{B} = \frac{dV}{V}, \text{ we find: } B = -K \cdot P.$$

The speed of sound in the gas is $c = \sqrt{\frac{B}{\rho}}$.

For a gas, $\rho = \frac{P}{r \cdot T}$, then, $c = \sqrt{K \cdot r \cdot T}$.

With the heat exchange model, the estimation of the bulk modulus is more complex, but also more realistic. The speed of sound varies from $c = \sqrt{\gamma \cdot r \cdot T}$ to $c = \sqrt{r \cdot T}$ depending on whether the system is respectively adiabatic or isothermal.

8.1.3. Inertial effect [1](#)

This is calculated for a single cell considering Newton's equation that leads to:

$$(8.10) \quad \Delta P = \frac{L}{A} \cdot \frac{d\dot{m}}{dt}$$

where L is the length of cell, A the flow area and $\dot{m}$ the mass flow through the cell.

This equation is linked to the assumption that the density is constant in the cell. This leads to a limitation on the validity of current **Pneumatic** pipe submodels that should be used when the velocity is well below the speed of sound, typically when the Mach number is below 0.3.

8.2. Response to a pressure step [1](#)

In order to illustrate the influence of the pipe sampling, we compare the evolution of pressure in a *10 m* long pipe (diameter *10 mm*) supplied with a *10 barA* pressure source. The pipe is closed at one end and a pressure step is imposed at the other end. Three different pipe samplings are considered.

Bearing type	Series	Bearing factor f_c		Bearing factor f_c
		Grease, oil mist	Oil bath, recirculating oil	
Spherical roller bearings	213	2,3	3,5	$0,0005 \cdot (P_d/C_2)^{0,22}$
	222	2,7	4	
	223	3	4,5	$0,0008 \cdot (P_d/C_2)^{0,22}$
	230			$0,00075 \cdot (P_d/C_2)^{0,22}$
	239			
	231	3,7	5,5	$0,0012 \cdot (P_d/C_2)^{0,22}$
	232	4	6	$0,0016 \cdot (P_d/C_2)^{0,22}$
	240	4,3	6,5	$0,0012 \cdot (P_d/C_2)^{0,22}$
	241	4,7	7	$0,0022 \cdot (P_d/C_2)^{0,22}$

Fig. 8.3 Model for pneumatic pipe simulation [1](#)

This model comprises components from the **Pneumatic** library. Fill in the parameters of this model as shown in the following table, leaving the other parameters at their default values and run a simulation for *0.5 seconds* with a **Print interval** of *0.005 seconds*.

Table 8.2 parameters for [Fig. 8.3](#) [1](#)

Icon

Submodel

Main parameters

0

Bearing type	Series	Bearing factor f_0		Bearing factor f_1
		Grease, oil mist	Oil bath, recirculating oil	
Cylindrical roller bearings, full complement	SL18 18	3	5	0,0005
	SL18 29	4	6	
	SL18 30	5	7	
	SL01 48, SL02 48	6	9	
	SL01 49, SL02 49	7	11	
	SL19 23	8	12	
	SL18 50	9	13	

PN_AIR

Default gas properties parameters

1

Bearing type	Series	Bearing factor f ₀		Bearing factor f ₁	
		Grease, oil mist	Oil bath, recirculating oil		
Radial ball bearings double row	42	2,3	3,5		
	43	4	6		
Angular contact ball bearings single row	70..BE 2Z	1,3	2	0,001·(P ₀ /C ₀) ^{2/3}	
	718..				
	72..B				
	72..BE				
	73..B	2	3		
Angular contact ball bearings, double row	30..2Z	2,3	3,5		
	32..2Z				
	32..B				
	38..2Z				
	39..2Z				
	33	4	6		
	33..B 2Z				
	33..DA				

PNCS001

pressure and temperature source
pressure = 10 barA

2

PNPL01

No parameter

zero
pneumatic
flow source

Icon

Submodel

Main parameters

Bearing type	Series	Bearing factor f ₂		Bearing factor f ₁
		Grease, oil mist	Oil bath, recirculating oil	
Self aligning ball bearings	12	1	1,5	0,0003·(P _d /C ₂) ^{2,4}
	13	1,3	2	
	22	1,7	2,5	
	23	2	3	
Four point contact bearings		2,7	4	0,001·(P _d /C ₂) ^{2,22}

3

Bearing type	Series	Bearing factor f ₂		Bearing factor f ₁
		Grease, oil mist	Oil bath, recirculating oil	
Axial ball bearings	511	1	1,5	0,0012·(F _d /C ₂) ^{2,22}
	512			
	513			
	514			
	522	1,3	2	
	523			
	524			

PNND005

2-ports node

No parameter

4

(1) For the wider series, use the larger value

B : Bearing width (mm) - d : Shaft diameter (mm)

P : Equivalent dynamic bearing load (N) - C_d Equivalent dynamic bearing load (N)

F_a Axial component of bearing load (N) - F_r Radial component of bearing load (N)

line

diameter = 0 mm

length = 10 m

thermal exchange coefficient = 0 J/m**2/K/s

5

(1) For the wider series, use the larger value

B : Bearing width (mm) - d : Shaft diameter (mm)

P : Equivalent dynamic bearing load (N) - C_d Equivalent dynamic bearing load (N)

F_a Axial component of bearing load (N) - F_r Radial component of bearing load (N)

PNL0005

line

diameter = 10 mm

length = 5 m

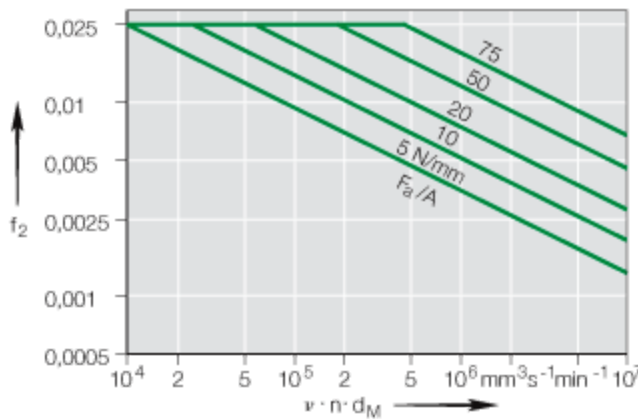
thermal exchange

	Icon	Submodel	Main parameters
			coefficient = 0 J/m**2/K/s
			diameter = 10 mm
6		PNL0006	length = 5 m
	(1) For the wider series, use the larger value B : Bearing width (mm) - d : Shaft diameter (mm) P : Equivalent dynamic bearing load (N) - C _d Equivalent dynamic bearing load (N) F _a Axial component of bearing load (N) - F _r Radial component of bearing load (N)	line	thermal exchange coefficient = 0 J/m**2/K/s
			diameter = 0 mm
7		PNL0020	length = 10 m
	(1) For the wider series, use the larger value B : Bearing width (mm) - d : Shaft diameter (mm) P : Equivalent dynamic bearing load (N) - C _d Equivalent dynamic bearing load (N) F _a Axial component of bearing load (N) - F _r Radial component of bearing load (N)	line	thermal exchange coefficient = 0 J/m**2/K/s

When this is done, compare the pressures at the closed end of the pipe as presented in [Fig. 8.3](#) for the different submodels, and consequently for the different sampling schemes:

- one *PNL0004*, 1 cell
- one *PNL0005* + one *PNL0006*, 3 cells
- one *PNL0020*, 6 cells

These submodels include resistance, inertia and compressibility and therefore generate wave dynamics. The submodels *PNL0004*, *PNL0005* and *PNL0006* are intended mainly to determine if the wave dynamics are significant. *PNL0020* is a much more accurate submodel for wave dynamics. By combining simpler submodels separated by nodes, *PNL0005* and *PNL0006* in this case, an intermediate level of complexity is obtained. Results are shown in the next figures.



$$M = M_{rr} + M_{sl} + M_{seal} + M_{drag}$$

Using only one *PNL0004* submodel, the significance of wave dynamics over the time range of 0.5 seconds is clearly demonstrated. However, results are very different from those obtained using *PNL0005* + *PNL0006* or *PNL0020* submodels. It is obvious that the more often the pipe is discretized, the more accurate the results will be. However, the price in terms of calculation time could be very high, since the number of state variables will increase rapidly.

The ideal solution is the simplest submodel providing a reasonable picture of the behavior of the pipe in the interaction with the components attached to its extremities. Perhaps in this case *PNL0005* + *PNL0006* is the best compromise.

This problem is more difficult in the case of a pneumatic network. Some considerations may be used to help select submodel simplifications:

- If all the pipes of the network have the same diameter, the sampling criteria will be in relation to the length of each branch.
- If a branch in a network presents a large diameter, in relation to the length, the inertial and dissipative effects are low, and may be ignored. In this case, it is possible to reduce this branch to a simple compressive effect submodel.
- It is important to understand that in some situations wave dynamics can be ignored. The simple rule used in hydraulic systems also works well in pneumatics:
- If the time taken for wave to travel the length of the pipe is significant when compared with the **Print interval**, it is important to take wave dynamics into account.

In this case with a length of 10 m and a speed of sound of about 300 m/s, the wave takes about 0.03 seconds to travel along the pipe. The integration range is 0.5 seconds with a **Print interval** of 0.005 seconds which means wave dynamics are significant. If the simulation range had been 500 seconds with a **Print interval** of 5 seconds, we could have used *PNL0001* to *PNL0003* or *PNL0010* to *PNL0012*.

8.3. Frequency domain [1](#)

A powerful solution for analyzing the system is to work in the frequency domain since it can yield important information with very small calculation times. Below, we give a brief introduction.

8.3.1. Theoretical aspect ¶

A popular model of wave dynamics is very useful because the resulting equations can be solved exactly. This gives frequencies for pipes with pure boundary conditions such as, with K being between I and γ depending on the thermal heat exchange. A simple polytropic assumption leads to $\sqrt{\frac{B}{\rho}} = \sqrt{K \cdot r \cdot T}$.

- if the line is open at both ends

$$(8.11) \quad f_i = i \cdot \frac{1}{2L} \sqrt{K \cdot r \cdot T} \quad \text{for } i = 0, 1, 2, \dots$$

- if the line is closed at both ends

$$(8.12) \quad f_i = i \cdot \frac{1}{2L} \sqrt{K \cdot r \cdot T} \quad \text{for } i = 1, 2, 3, \dots$$

- if the line is open at one end and closed at the other one

$$(8.13) \quad f_i = \frac{2i-1}{2} \cdot \frac{1}{2L} \sqrt{K \cdot r \cdot T} \quad \text{for } i = 1, 2, 3, \dots$$

An open boundary condition is similar to a temperature / pressure source (constant pressures and temperatures are imposed, equivalent to connecting to an infinite volume).

A closed boundary condition is similar to an enthalpy flow / mass flow source (zero flux is imposed, equivalent to a blocked end).

Note

These theoretical results come from an **approximate** model. However, the first few frequencies are known to be reasonably accurate. It is interesting to compare the simulated mode with these theoretical ones. We choose a *10 m* long pipe with a *10 mm* diameter.

In **Simcenter Amesim**, the values of the modes of the pneumatic pipes are extracted from the eigenvalues of the system. See the **Linear Analysis** documentation for more explanations about the linear approach.

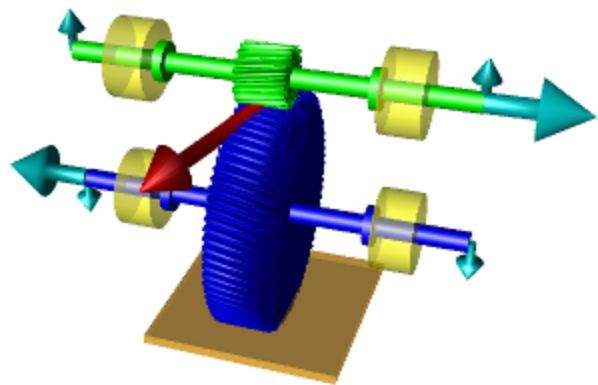


Fig. 8.4 Pure boundary conditions

To obtain the eigenvalues of your system, define an **LA Time** at *0 seconds* in the **Linear Analysis Menu**, and run a simulation with the **Stabilizing + Dynamic Run** option. Check that your system is perfectly equilibrated. Clicking on the **Eigenvalues** button will produce the eigenvalues of your system. Conjugate eigenvalue pairs correspond to oscillatory modes.

With $K = 1.35$ and $T = 293\text{ K}$, we have for the first three modes:

Pipe submodel	Extreme conditions	Simulated modes [Hz]	Theoretical modes [Hz]	Difference [%]
PNL0004	opened / closed	5.3	8.4	57
PNL0005	opened / opened	0	0	0
		10.7	16.8	57
PNL0006	closed / closed	10.7	16.8	57
		8.4	8.4	0
PNL0020	opened / closed	24.5	25.3	3
		38.6	42.1	9
PNL0021	opened / opened	0	0	0
		16.7	16.8	1
		32.1	33.7	5

Pipe submodel	Extreme conditions	Simulated modes [Hz]	Theoretical modes [Hz]	Difference [%]
PNL0022	closed / closed	45.5	50.5	11
		16.7	16.8	1
		32.1	33.7	5
		45.5	50.5	11

Note

The 57% difference in the case of a one-cell sampled pipe corresponds to the usual $\frac{\pi}{2}$ factor.

The first mode is predicted well with *PNL002x* series.

For a more complex network, use the same linear approach to ensure you make the right sampling.

8.3.2. Modal shapes [1](#)

Another powerful tool to observe your network is to use the **Modal Shapes** function. This consists in plotting the eigenvectors of your system for specific observed variables. See the **Linear Analysis** documentation for more explanations about the modal shapes.

To obtain the eigenvectors of your system, define an **LA Time** at *0 seconds* in the **Linear Analysis Menu**, and observed variables (mass flow rates for example) by clicking on the corresponding variables. Run a simulation with the **Stabilizing + Dynamic Run** option. Check that your system is in a state of equilibrium. Selecting the **Eigenvalues > ModalShapes** button will produce the eigenvalues of your system. Selecting a specific eigenvalue will produce a **Modal Shape Analysis** popup. You can then chart the associated modal shape considering the *plot magnitude* option.

The modal shape of the first mode is presented in [Fig. 8.5](#).

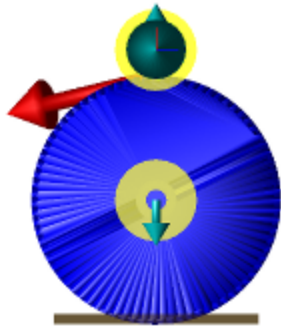


Fig. 8.5 *PNL0022*, closed / closed, first mode [9](#)

These techniques are well known in the mechanical field. You can apply the same approach to more general problems to better understand the dynamic behavior of your network.

[Table of Contents](#)

- [1. Introduction](#)
- [2. Getting started with the **Pneumatic** library](#)
- [3. The gas properties](#)
- [4. Running multi-gas simulations](#)
- [5. Running multi-domain simulations: gas-liquid heat exchangers](#)
- [6. Important rules for modeling a system with pneumatic components](#)
- [7. Brief review of the theory: pneumatic calculations](#)
- [8. Modeling **Pneumatic** pipes](#)
 - [8.1. Equations](#)
 - [8.1.1. Dissipative effect R](#)
 - [8.1.2. Compressive effect C](#)
 - [8.1.3. Inertial effect I](#)
 - [8.2. Response to a pressure step](#)
 - [8.3. Frequency domain](#)
 - [8.3.1. Theoretical aspect](#)
 - [8.3.2. Modal shapes](#)
- [9. Brief review of the theory: thermal calculations](#)
- [10. Heat exchanger models based on effectiveness-NTU method](#)
- [Appendix A: some gas properties](#)
- [Appendix B: oxygen properties](#)
- [Appendix C: Nusselt correlations](#)
- [Appendix D: heat exchanger effectiveness](#)
- [Appendix E: energy storage phenomena](#)
- [Bibliography](#)

Navigation

- [next](#)
- [previous](#) |
- [Pneumatic Library](#) »
- [8. Modeling **Pneumatic** pipes](#)

© Copyright Unpublished work. Copyright 2024 Siemens.