



Pneumatic Library User's guide

Simcenter Amesim 2404

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1. Introduction

The **Simcenter Amesim Pneumatic** library is a set of component submodels ready for use with existing **Simcenter Amesim** components. It can be used in most engineering applications where gas flows take place.

This library was originally developed at the Laboratoire Ampère (INSA Lyon, France) and is based on the accumulated experience and knowledge of the Fluidetronic research team in pneumatics systems. Industrialization has been carried out by the **Simcenter Amesim** development staff.

Heat exchanges occur in most industrial processes either because they are desired (ovens, heat exchangers, ...) or because they are unavoidable (thermal shocks, thermal losses, friction, ...). These exchanges are likely to occur through mobile gases or between mobile gases and their environment as soon as temperature differences are encountered.

Pressured gas is used today in many mechatronic applications for power transmission. Most of the systems use compressed air, but applications with hot gas or inert gas are becoming more common. There are many advantages with systems based on compressed air:

- most production plants have a compressed air supply,
- pneumatic components are relatively inexpensive,
- pneumatic technology is very adaptable,
- pneumatic equipment is reliable and inexpensive to maintain,
- pneumatic tools such as drills and grinders, do not burn out when stalled,
- there is no fire hazard with pneumatic systems,
- pneumatic systems are clean.

A typical processing pneumatic plant is shown in Fig.1.1.

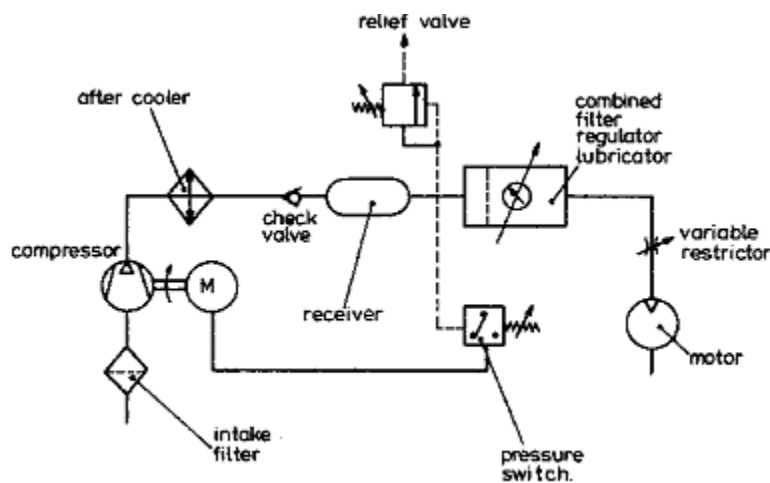


Fig. 1.1: Simple pneumatic system

This library can be used alone or can be coupled with the **Simcenter Amesim Pneumatic Component Design**, **Simcenter Amesim Thermal** and **Simcenter Amesim Thermal-Hydraulic** libraries. These libraries contain some components with thermal ports and others allowing the basic element approach. Temperature and heat flow information can be exchanged between components of these three libraries through their thermal ports. Building systems with components belonging to these three libraries makes it possible to study a wide range of gas systems applications.

We will work through a small tutorial example before describing the properties of the gas. Then a second example will be given followed by rules and advice. Finally, we will detail the theory and assumptions used in this library, the modeling of the lines and heat exchanges.

It is assumed that the reader is familiar with the use of **Simcenter Amesim** software. If this is not the case, we suggest that you do the exercises of the **Simcenter Amesim** tutorial manual before attempting the examples below.

2. Getting started with the Pneumatic library

The **Pneumatic** library comprises a set of basic components from which it is easy to model and observe the evolution of temperatures, pressures and mass flow rates in pneumatic systems. By modeling the system shown in Fig.2.1, you will find that several questions are raised. They will be answered in this manual. We recommend you do this example as a first introduction to the **Pneumatic** library.

2.1 Train pneumatic example

The system shown in Fig.2.1 describes a part of a pneumatic circuit for trains. Two pressurized accumulators discharge into a third one, and the duration of the process must be estimated. A good understanding of how it works is useful for predicting braking time response, while the knowledge of the exact process with the saturation effects due to sonic velocity makes it possible to optimize the system.

The system consists in three 800 L, 400 L and 100 L pneumatic accumulators connected through 500 mm² and 20 mm² orifices. The initial pressures are respectively 11 barA, 6 barA and 1 barA while the initial temperature is 293.15 K.

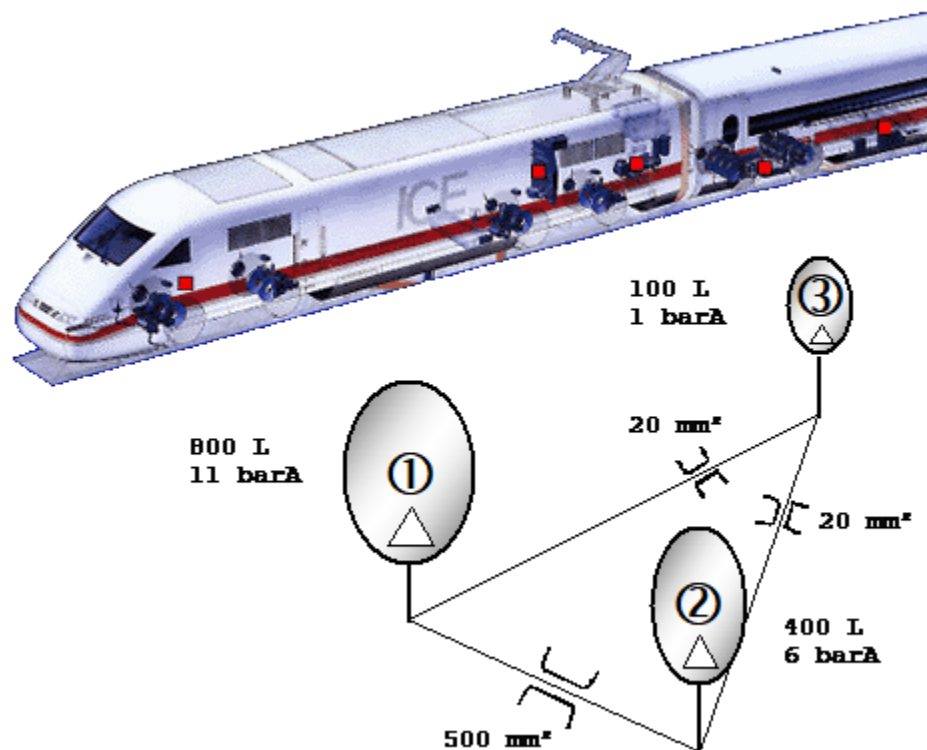


Fig. 2.1: Simplified part of a train pneumatic circuit

To model this system, select the **Pneumatic** library category icon shown in Fig.2.2.



Fig. 2.2: Pneumatic library category icons

This will produce the popup shown in [Fig.2.3](#).

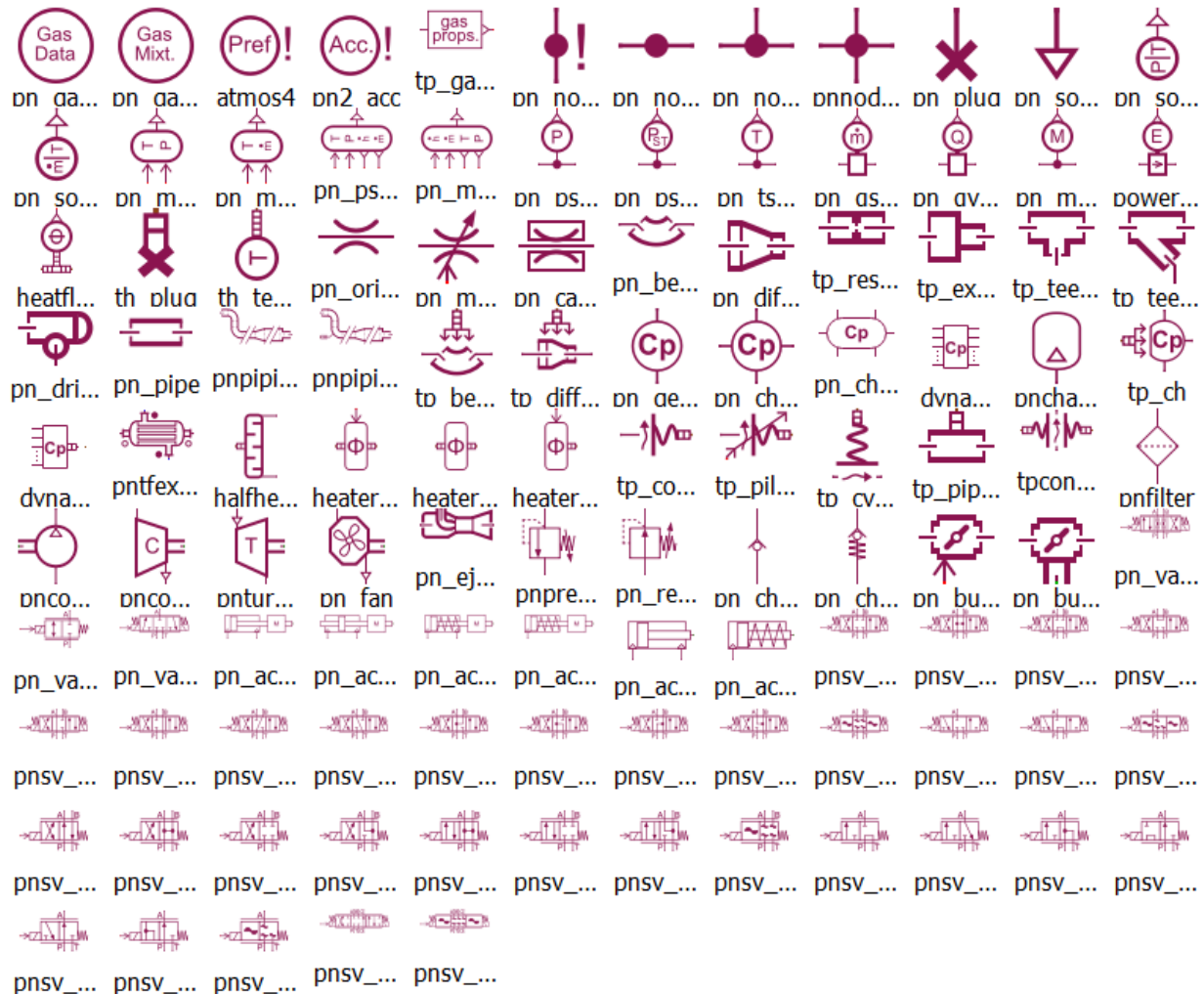
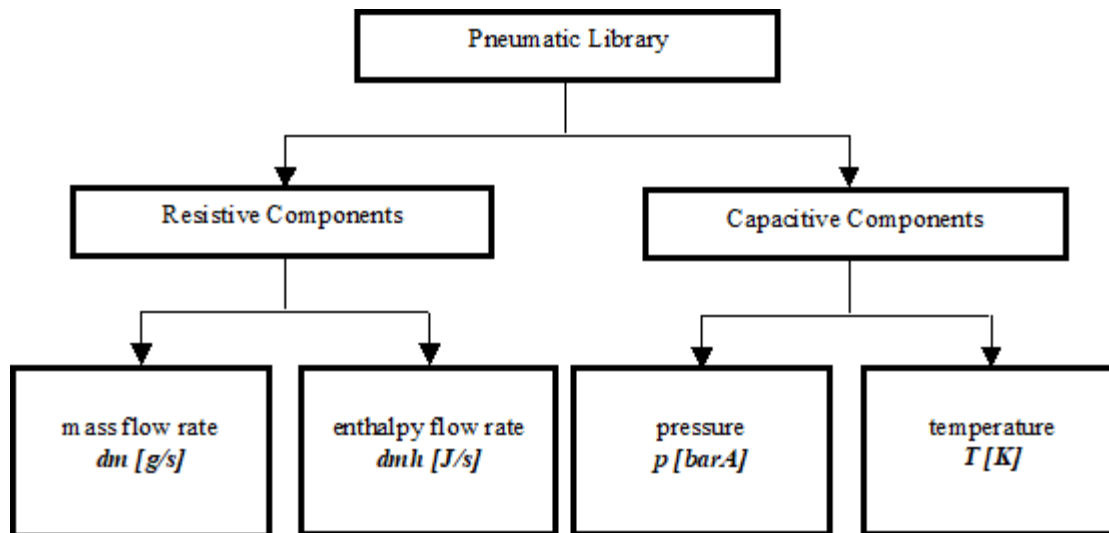


Fig. 2.3: Components of the Pneumatic library

First look at the components available in this library. Display the titles of each component by hovering the pointer over the icons.

The **Pneumatic** library can be basically represented as shown below:



The resistive submodels can be described as **steady-state submodels**. A more accurate description is that they are instantaneous submodels. By this, we mean that they are assumed to react instantaneously to the temperatures and pressures applied to them so that they are always in an equilibrium state.

The capacitive submodels have two state variables: the temperature and the pressure. This means that each of these variables is defined by a differential equation. These are very commonly used in **Pneumatic** library systems and we will describe them as **transient submodels**.

In addition, some components are both resistive and capacitive.

There are also some inertial submodels whose external variables are similar to those of the resistive submodels. They compute the mass flow rate and enthalpy flow rate depending on the temperatures and pressures applied to them.

The combination of resistive, capacitive and inertial components is also possible, as provided for example in the pneumatic pipe submodels.

Pneumatic library components have **Pneumatic**, **Thermal** and **Signal** ports.

Four variables are exchanged at **Pneumatic** ports:

- Temperature T [K]
- Enthalpy flow rate dmh [J/s]
- Pressure p [Pa]
- Mass flow rate dm [g/s]

The pressure at ports is a gauge pressure in Pa. A more commonly-used unit for the pressure in pneumatic systems is the absolute pressure in barA (absolute pressure = gauge pressure + atmospheric pressure). Verify that the conversion from Pa to barA is selected in the menu **Configure > Unit management... > Default system > hyd/pneu pressures & bulk modulus**. All pressures will then be displayed in barA.

Two variables are exchanged at thermal ports:

- Temperature T [degC]
- Heat flow rate dq [W]

Thermal ports are easily recognizable because of their special shape given in Fig.2.4 below:





The accumulators are assumed to be directly connected through orifices without considering any pneumatic pipes with propagation effects between them.

Table 2.1: components used in Fig. 2.5

Icon	Name	Main parameters
0 	PNGD_AIR gas properties	default parameters
1 	PNCH022 pneumatic chamber	pressure = 11 barA volume = 800 L
2 	PNCH022 pneumatic chamber	pressure = 6 barA volume = 400 L
3 	PNCH022 pneumatic chamber	pressure = 1 barA volume = 100 L
4 	PNOR001 pneumatic orifice	orifice area = 500 mm**2
5, 6 	PNOR001 pneumatic orifice	orifice area = 20 mm**2
7 	PN3NODE3 pneumatic node	no parameter
8 	PN3NODE2 pneumatic node	no parameter

Finally, run a simulation with a final time of 20 seconds.

The main point of interest in this example is to observe the evolution of the gas pressures inside the three accumulators as shown in Fig.2.6. The two higher pressure chambers first equilibrate before stabilizing with the third chamber. The final steady-state pressure is approximately 8.5 barA at 20 s.

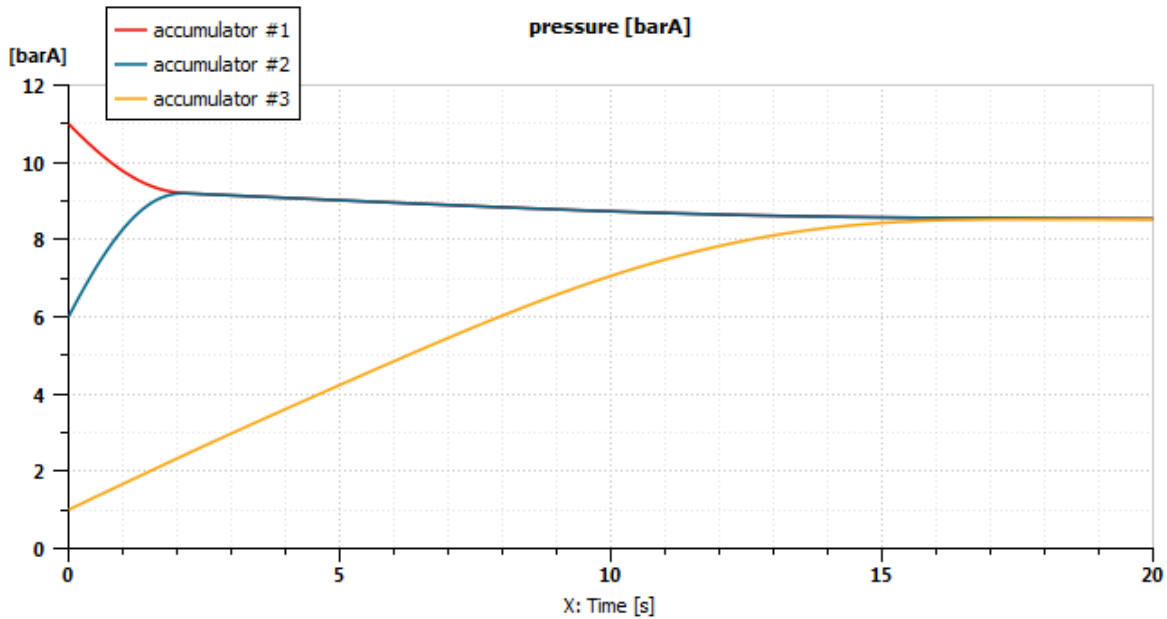


Fig. 2.6: Evolution of the gas pressures in the accumulators

The gas flows through accumulators 2 and 3 because of the large flow area of 500 mm^2 . The stabilizing process then takes a long time as a consequence of the small flow area of 20 mm^2 and of the high-pressure ratios between the accumulators that lead to a sonic saturated flow. The mass flow parameter C_m in Fig.2.7 highlights the sonic flow with a maximal value of 0.404 for air.

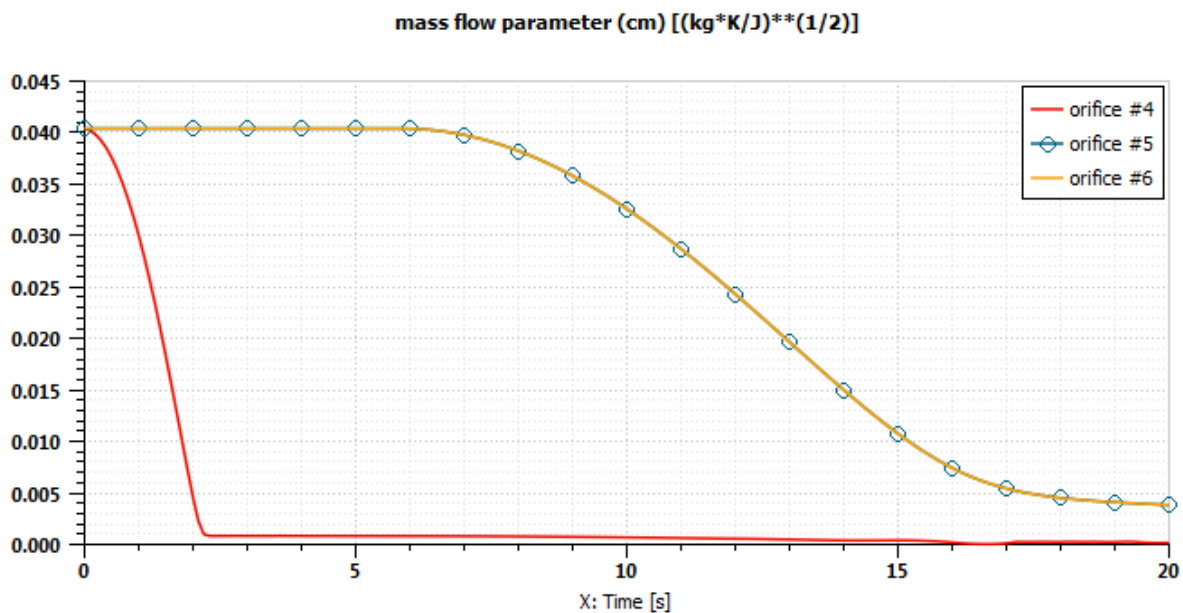


Fig. 2.7: Evolution of the mass flow parameter C_m in the orifices

Next, we can compare the evolution of the temperatures in the three accumulators as shown in Fig.2.8. It shows that the temperature of the depressurizing accumulator 1 decreases while

the temperatures of the pressurizing accumulators 2 and 3 increase.

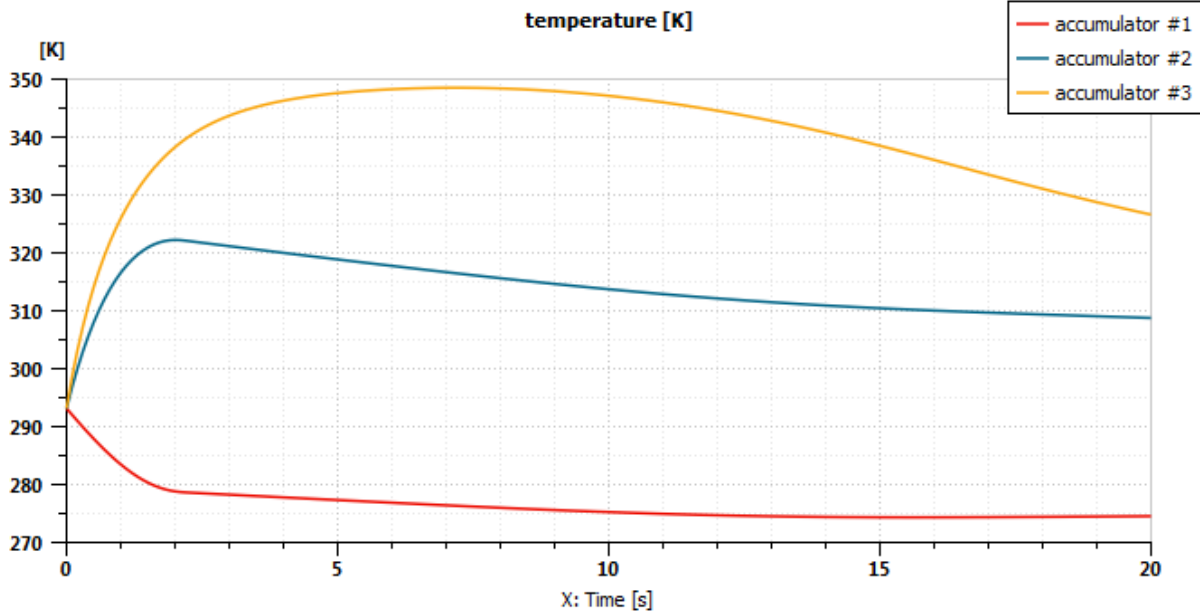


Fig. 2.8: Evolution of the gas temperatures in the accumulators

In this model, the use of submodel *PNGD_AIR* associated with the icon below is very important.



This submodel contains the pneumatic properties of the gas used for the simulation. More details on possible submodels of gas data will be given in [Section 3](#).

2.2 Some questions

Starting from this very basic example, the following questions are raised:

- Pneumatic gas property considerations:

How does the user get access to the properties of the gases and how are these properties dealt with in the **Pneumatic** library?

The **Pneumatic** library offers several levels of gas models defined as perfect, semi-perfect or ideal, real or through mixture of various gases. Some thermal characteristics for common gases (O_2 , N_2 , H_2 , ...) are also proposed as customized submodels.

Is it possible for the user to define his own gas thermal properties?

Yes. *PNGD_AIR* is a customized submodel of *PNGD00* which is a generic submodel. Default parameters of *PNGD00* offer the possibility to set a complete user-defined gas through data files in a specific format.

- Multi-gas considerations:

If the user has a system to model which contains several gases, is it possible to run multi-gas simulations?

Yes. In some cases, systems comprise different gases interacting with each other. The use of more than one instance of *PNGD00* submodels of gas data enables the handling of multi-gas systems. This is fully described in [Section 4](#).

- **Pneumatic considerations:**

How are the mass and enthalpy flow rates computed in the **Pneumatic** resistive submodels (R)?

Some submodels in the **Pneumatic** library compute mass and enthalpy flow rates due to frictional drag (distributed resistance) or to local changes in geometry (local resistance). In [Section 7.2](#), a brief review of the theory and the equations used is given.

How are the pressure and temperature transients computed in the **Pneumatic** capacitive submodels (C)?

Some submodels in the **Pneumatic** library compute the pressure and the temperature from their derivatives with respect to time. In [Section 7.3](#) a brief review of the theory and the equations used is given.

- **Pneumatic pipe considerations:**

Which assumptions are used concerning wave propagation equations in pneumatic pipes?

In [Section 8](#), a brief review of the theory and the equations governing these phenomena is given. How to model a pneumatic pipe network is introduced.

- **Thermal considerations:**

Which assumptions are used concerning heat transport in the gases, internal heat transfers between the gas circulating and the wall of the pipe, as well as heat exchanges with the outside?

In [Section 7](#), a brief review of the theory and the equations governing these phenomena is given.

3. The gas properties

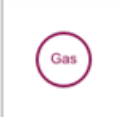
3.1 Various models of gas

The kinetic theory of gases makes it possible to define the pressure and temperature in relation to the statistical effect of the collision of molecules against solid walls and to kinetic energy of thermal agitation. Different models of gas are deduced from this theory. They are implemented in the *PNGD00* submodel that is linked with the following icon:



Several customized submodels of *PNGD00* are available.

Submodel summary



Component: pn_gas_data

Compatible submodel list

Name	Description	Submodel type
▼ PNGD00	generic gas definition	generic submodel
PNGD_AIR	gas definition for air	customized submodel
PNGD_AMMONIA	gas definition for ammonia (NH3)	customized submodel
PNGD_ARGON	gas definition for argon (Ar)	customized submodel
PNGD_BUTANE	gas definition for butane (C4H10)	customized submodel
PNGD_CARBDIOXIDE	gas definition for carbon dioxide (CO2)	customized submodel
PNGD CARBONMONOXIDE	gas definition for carbon monoxide (CO)	customized submodel
PNGD_DECANE	gas definition for decane (C10H22)	customized submodel
PNGD_ETHANE	gas definition for ethane (C2H6)	customized submodel
PNGD_ETHYLENE	gas definition for ethylene (C2H4)	customized submodel
PNGD_EXHAUST_STOICHIO	gas definition for exhaust (stoichio)	customized submodel
PNGD_HELIUM	gas definition for helium (He)	customized submodel
PNGD_HEPTANE	gas definition for heptane (C7H16)	customized submodel
PNGD_HEXANE	gas definition for hexane (C6H14)	customized submodel
PNGD_HYDROGEN	gas definition for hydrogen (H2)	customized submodel
PNGD_HYDROGENSULFIDE	gas definition for hydrogen sulfide (H2S)	customized submodel
PNGD_ISOBUTANE	gas definition for isobutane (C4H10)	customized submodel
PNGD_ISOPENTANE	gas definition for isopentane (C5H12)	customized submodel
PNGD_METHANE	gas definition for methane (CH4)	customized submodel
PNGD_NEON	gas definition for neon (Ne)	customized submodel
PNGD_NITRICACID	gas definition for nitric acid (HNO3)	customized submodel
PNGD_NITRICOXIDE	gas definition for nitric oxide (NO)	customized submodel
PNGD_NITROGEN	gas definition for nitrogen (N2)	customized submodel
PNGD_NITROGENDIOXIDE	gas definition for nitrogen dioxide (NO2)	customized submodel
PNGD_NITROUSOXIDE	gas definition for nitrous oxide (N2O)	customized submodel
PNGD_NONANE	gas definition for nonane (C9H20)	customized submodel
PNGD_OCTANE	gas definition for octane (C8H18)	customized submodel
PNGD_OXYGEN	gas definition for oxygen (O2)	customized submodel
PNGD_PENTANE	gas definition for pentane (C5H12)	customized submodel
PNGD_PROPANE	gas definition for propane (C3H8)	customized submodel
PNGD_SULFURHEXAFLUORIDE	gas definition for sulfur hexafluoride (SF6)	customized submodel
PNGD_SULFURICACID	gas definition for sulfuric acid (H2SO4)	customized submodel
PNGD_SULFURTRIOXIDE	gas definition for sulfur trioxide (SO3)	customized submodel
PNGD_WATERVAPOR	gas definition for water vapor (H2O)	customized submodel
PNGD_XENON	gas definition for xenon (Xe)	customized submodel

External variables
Help
View

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Fig. 3.1: PNGD00 and its customized submodels

3.2 The gas type index parameter

All submodels from the **Pneumatic** library require a parameter called the **gas type index**, whose function is to reference a particular gas.

In the first tutorial example, the gas type index was equal to 1. This index is used to make the link between the properties of the gas and the submodels of the system which requires these properties to compute the enthalpy flow rates, the mass flow rates, the pressures or the temperatures. Display the parameter pop-up of submodel *PNCH022-1* (representing the first chamber) of the first tutorial example as shown in Fig.3.2.

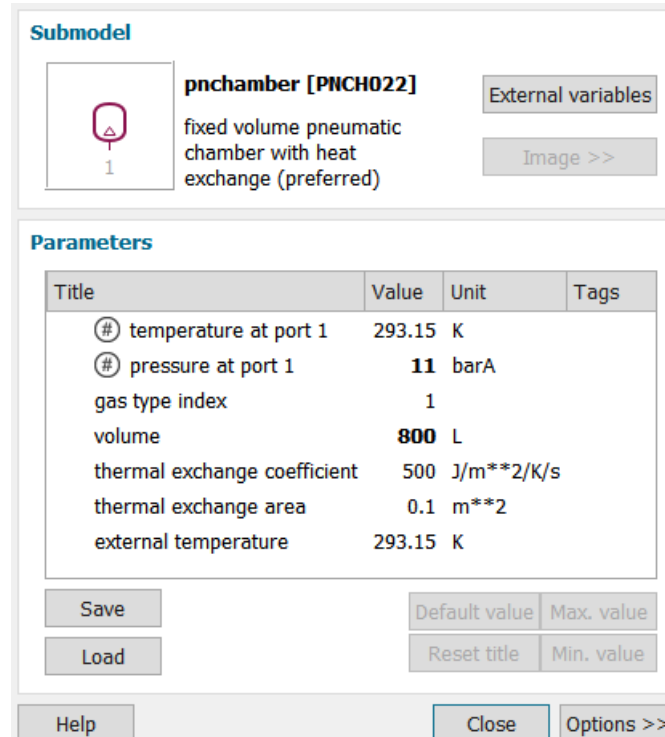


Fig. 3.2: Pneumatic chamber parameter pop-up

This submodel requires the gas type index, whose default value is 1. If you only have one gas in your system, you can leave the gas type index in *PNGD00* and in any other **Pneumatic** submodel at its default value.

The important rule is: **an icon associated with submodel *PNGD00* inserted on the sketch of a system refers to one and only one gas.** If your system has more than one gas, you must add the corresponding number of icons.

3.3 Defining the properties of the gas

Pneumatic submodels may be supplied with the properties of the gas to compute the different variables exchanged at ports. Submodel *PNGD00* and its customized submodels are used to provide access to various properties fully defining a gas: the **equation of state**, the **constant-pressure specific heat**, the **absolute viscosity** and the **thermal conductivity**.

In generic submodel *PNGD00*, users can set their own properties through various data files.

Title	Value	Unit	Tags	Name
gas type index	1			gi
fluid definition	user defined			fluidType
properties definition	semi perfect			eosType
molar mass	28.9651	g/mol		M
enthalpy of formation	-125.53	J/mol		enthForm
filename for Cp, h and s coefficients	\$AME/libpn/data/nasa_properties/air_cp_h_s.data			cphsFileName
filename for mu coefficients	\$AME/libpn/data/nasa_properties/air_mu.data			muFileName
filename for lambda coefficients	\$AME/libpn/data/nasa_properties/air_lambda.data			lambdaFileName
name of the gas	air (semi perfect gas)			gasName

Fig. 3.3: Parameters of generic *PNGD00*

In customized submodels of *PNGD00*, all properties are predefined. The user can only change the level of modeling.

Title	Value	Unit	Tags	Name
gas type index	1			gi
properties definition	semi perfect			eosType
	perfect			
	semi perfect			
	real: Van der Waals			
	real: Redlich - Kwong			
	real: Redlich - Kwong - Soave			
	real: Peng - Robinson			

Fig. 3.4: Parameters of customized submodels of *PNGD00*

There are three levels of modeling:

- **perfect gas:** the perfect equation of state is used. All other properties are constant.
- **semi-perfect gas:** the perfect equation of state is used. All other properties are temperature-dependent.
- **real gas:** a specific equation of state (among 4 predefined) is used. The absolute viscosity and the thermal conductivity are temperature-dependent. All other properties are temperature and pressure-dependent.

3.3.1 Perfect gas constant of a gas

In the **Pneumatic** library, the perfect gas constant r of a gas referenced by an index is assumed to be constant. Therefore, there is no variation of the perfect gas constant when the temperature varies.

It is used in the perfect gas equation:

$$p \cdot V = m \cdot r \cdot T \quad (3.1)$$

The perfect gas constant is linked with the molar mass M of the gas through the perfect gas constant $R = 8.3145 \text{ J/mol/K}$:

$$r \cdot M = R \quad (3.2)$$

The perfect gas constant of the gas is expressed in J/kg/K .

3.3.2 Equation of state of a gas

The equation of state is used to compute the density (ρ in kg/m^3) as a function of the pressure P and the temperature T .

3.3.2.1 Perfect gas or semi-perfect gas

Perfect or semi-perfect gases use the perfect gas equation of state:

$$\rho = \frac{p}{r \cdot T} \quad (3.3)$$

3.3.2.2 Real gas

Real gases use a specific equation of state from among four predefined:

- Van der Waals

$$\left(P + \frac{a}{V^2} \right) \cdot (V - b) - r \cdot T = 0 \quad (3.4)$$

where the attractive term a and the covolume b are functions of the critical pressure, the critical temperature, and the perfect constant as written hereafter:

$$a = \frac{27 \cdot r^2 \cdot T_c^2}{64 \cdot P_c} \text{ and } b = \frac{r \cdot T_c}{8 \cdot P_c} \quad (3.5)$$

- Redlich-Kwong

$$\left(P + \frac{a \cdot \alpha(T)}{V \cdot (V + b)} \right) \cdot (V - b) - r \cdot T = 0 \quad (3.6)$$

where a and b coefficients are evaluated with the following relationships:

$$a = 0.42748024 \cdot \frac{r^2 \cdot T_c^{2.5}}{P_c} \text{ and } b = 0.08664035 \cdot \frac{r \cdot T_c}{P_c} \quad (3.7)$$

the attractive term is temperature-dependent through the alpha-function:

$$\alpha(T) = \frac{1}{\sqrt{T}} \quad (3.8)$$

- Redlich-Kwong-Soave:

the equation of state is identical to the previous one proposed by Redlich-Kwong (Eq.3.6) but parameters and the alpha-function are different:

$$a = 0.42748024 \cdot \frac{r^2 \cdot T_c^2}{P_c} \text{ and } b = 0.08664035 \cdot \frac{r \cdot T_c}{P_c} \quad (3.9)$$

$$\alpha(T) = \left[1 + m \cdot \left(1 - \sqrt{\frac{T}{T_c}} \right) \right]^2 \quad (3.10)$$

with m calculated using a 2nd order polynomial function of the acentric factor:

$$m = 0.48 + 1.574 \cdot \omega - 0.176 \cdot \omega^2 \quad (3.11)$$

ω is the Pitzer's acentric factor, which is an intrinsic property of the gas considered, defined as follows:

$$\omega = -1 - \log_{10} \left(\frac{P_{\text{sat}}}{P_c} \right)_{T=0.7 \cdot T_c} \quad (3.12)$$

- Peng-Robinson

$$\left(P + \frac{a \cdot \alpha(T)}{V^2 + 2 \cdot b \cdot V - b^2} \right) \cdot (V - b) - r \cdot T = 0 \quad (3.13)$$

where a and b coefficients are evaluated with the following relationships:

$$a = 0.457235583 \cdot \frac{r^2 \cdot T_c^2}{P_c} \text{ and } b = 0.07779607 \cdot \frac{r \cdot T_c}{P_c} \quad (3.14)$$

The attractive term is temperature-dependent through the following alpha-function:

$$\alpha(T) = \left[1 + m \cdot \left(1 - \sqrt{\frac{T}{T_c}} \right) \right]^2 \quad (3.15)$$

$$m = 0.37464 + 1.54226 \cdot \omega - 0.26992 \cdot \omega^2 \quad (3.16)$$

This last model is the most accurate for processes occurring near the critical point.

Note:

It is pointless to perform a simulation using the real gas submodel and the mixture submodel and/or selecting the polytropic model in pipe or volume submodels. Indeed, these computations will not take the real gas behavior into account.

3.3.3 Constant-pressure specific heat of a gas

The base formulation of the constant-pressure specific heat of a gas is its term that depends on the temperature. In the **Pneumatic** library this term comes from NASA literature (see [2] and [7]).

$$C_{p,NASA} = r \cdot \left(\frac{a_1}{T^2} + \frac{a_2}{T} + a_3 + a_4 \cdot T + a_5 \cdot T^2 + a_6 \cdot T^3 + a_7 \cdot T^4 \right) \quad (3.17)$$

The coefficients $a_1' * to : math : 'a_7$ are constant. They can be defined differently in various temperature ranges. For example, for air:

```
# Table format: MULTILINES
# Gas: air (0.78109 N2, 0.20954 O2, 0.00937 Ar)
# Coefficients for specific heat (Cp in J/kg/K), specific enthalpy (h in J/kg) and specific entropy (s in J/kg/K)
# as a function of temperature (T in K) in the NASA 2002 format
# Ref: NASA Glenn Coefficients for Calculating Thermodynamic Properties of Individual Species, NASA/TP-2002-211556,
2002
200.0 1000.0 6000.0 ..... # boundary temperatures
# Temperature range 200.0 - 1000.0 K
1.009950e+04 -1.968276e+02 5.009155e+00 -5.761014e-03 1.066860e-05 -7.940298e-09 2.185232e-12 ..... # a1, a2... a7
in Cp0/r=a1/T^2+a2/T+a3+a4.T+a5.T^2+a6.T^3+a7.T^4
-1.767967e+02 ..... # a8 in H_0/rT=-a1/T^2-a2/T.ln(T)+a3+a4/2.T+a5/3.T^2+a6/4.T^3+a7/5.T^4+a8/T
-3.921504e+00 ..... # a9 in S_0/r=-a1/2/T^2-a2/T+a3.ln(T)+a4.T+a5/2.T^2+a6/3.T^3+a7/4.T^4+a9
# Temperature range 1000.0 - 6000.0 K
2.415214e+05 -1.257875e+03 5.144559e+00 -2.138542e-04 7.065228e-08 -1.071483e-11 6.577800e-16 ..... # a1, a2... a7
in Cp0/r=a1/T^2+a2/T+a3+a4.T+a5.T^2+a6.T^3+a7.T^4
6.462263e+03 ..... # a8 in H_0/rT=-a1/T^2-a2/T.ln(T)+a3+a4/2.T+a5/3.T^2+a6/4.T^3+a7/5.T^4+a8/T
-8.147412e+00 ..... # a9 in S_0/r=-a1/2/T^2-a2/T+a3.ln(T)+a4.T+a5/2.T^2+a6/3.T^3+a7/4.T^4+a9
```

Fig. 3.5: Example of definition for Cp for air

The specific heat of the gas is expressed in J/kg/K.

Note:

A quadratic extrapolation is done when the current temperature falls outside the definition range.

3.3.3.1 Perfect gas

The constant-pressure specific heat is a constant.

$$C_p(P, T) = C_{p,0} \quad (3.18)$$

3.3.3.2 Semi perfect gas

The constant-pressure specific heat is equal to the NASA formulation.

$$C_p(P, T) = C_{p,NASA}(T) \quad (3.19)$$

3.3.3.3 Real gas

The constant-pressure specific heat is equal to the NASA formulation plus a residual term that depends on the equation of state.

$$C_p(P, T) = C_{p,NASA}(T) + C_{p,res}(P, T) \quad (3.20)$$

3.3.4 Thermal conductivity of a gas

The formulation of the thermal conductivity of a gas is a term that depends on the temperature. This term comes from NASA literature (see [2] and [7]).

$$\ln(\lambda(T)) = \ln(\lambda_{\text{NASA}}) = a_\lambda \cdot \ln(T) + \frac{b_\lambda}{T} + \frac{c_\lambda}{T^2} + d_\lambda \quad (3.21)$$

a_λ to d_λ coefficients are constant. They can be defined differently in various temperature ranges. For example, for air:

```
# Table format: MULTILINES
# Gas: air (0.78109 N2, 0.20954 O2, 0.00937 Ar)
# Coefficients for thermal conductivity (lambda in microW/cm/K) as a function of temperature (T in K) in the NASA 96 format
# Ref: Computer Program for Calculation of Complex Chemical Equilibrium Compositions and Applications, NASA Reference Publication 1311, 1996
200.0 1000.0 5000.0 15000.0 ..... # boundary temperatures
# Temperature range 200.0 - 1000.0 K
8.351316e-01 8.379024e+01 -1.091253e+04 6.451952e-01 ..... # aLam, bLam, cLam, dLam in ln(Lam) = aLam.ln(T)+bLam/T+cLam/T^2+dLam
# Temperature range 1000.0 - 5000.0 K
8.943614e-01 1.931717e+02 -3.792499e+04 1.534105e-01 ..... # aLam, bLam, cLam, dLam in ln(Lam) = aLam.ln(T)+bLam/T+cLam/T^2+dLam
# Temperature range 5000.0 - 15000.0 K
1.406851e+00 -1.354762e+03 1.353235e+07 -4.438968e+00 ..... # aLam, bLam, cLam, dLam in ln(Lam) = aLam.ln(T)+bLam/T+cLam/T^2+dLam
```

Fig. 3.6: Example of definition for lambda for air

The thermal conductivity of the gas is expressed in W/m/K.

Note:

- Natural extrapolation (the same coefficients are used) is done when the current temperature falls outside the definition range.
- Thermal conductivity is constant ($a_\lambda = b_\lambda = c_\lambda$, no extrapolation) when the level of modeling is set to “perfect gas”.

3.3.5 Absolute viscosity of a gas

The formulation of the absolute viscosity of a gas is a term that depends on the temperature. This term comes from NASA literature (see [Ref. 20] and [Ref. 21]).

$$\ln(\mu(T)) = \ln(\mu_{\text{NASA}}) = a_\mu \cdot \ln(T) + \frac{b_\mu}{T} + \frac{c_\mu}{T^2} + d_\mu \quad (3.22)$$

a_μ to d_μ coefficients are constant. They can be defined differently in various temperature ranges. For example, for air:

```
# Table format: MULTILINES
# Gas: air (0.78109 N2, 0.20954 O2, 0.00937 Ar)
# Coefficients for absolute viscosity (mu in microP) as a function of temperature (T in K) in the NASA 96 format
# Ref: Computer Program for Calculation of Complex Chemical Equilibrium Compositions and Applications, NASA Reference Publication 1311, 1996
200.0 1000.0 5000.0 15000.0 ..... # boundary temperatures
# Temperature range 200.0 - 1000.0 K
6.216316e-01 -3.660056e+01 -1.396818e+03 1.815022e+00 ..... # aMu, bMu, cMu, dMu in ln(mu) = aMu.ln(T)+bMu/T+cMu/T^2+dMu
# Temperature range 1000.0 - 5000.0 K
8.388077e-01 4.714665e+02 -1.467807e+05 -4.785247e-02 ..... # aMu, bMu, cMu, dMu in ln(mu) = aMu.ln(T)+bMu/T+cMu/T^2+dMu
# Temperature range 5000.0 - 15000.0 K
8.509112e-01 7.817794e+02 -6.393888e+05 -1.933531e-01 ..... # aMu, bMu, cMu, dMu in ln(mu) = aMu.ln(T)+bMu/T+cMu/T^2+dMu
```

Fig. 3.7: Example of definition for mu for air

The absolute viscosity of the gas is expressed in $N.s/m^2$ or $kg/m.s$.

Note:

- Natural extrapolation (the same coefficients are used) is done when the current temperature falls outside the definition range.
- Absolute viscosity is constant ($a_\mu = b_\mu = c_\mu = 0$, no extrapolation) when the level of modeling is set to "perfect gas".

3.3.6 Other properties of a gas

Other properties of a gas are calculated out of the given equation of state, the constant-pressure specific heat, the absolute viscosity and the thermal conductivity.

- the constant-volume specific heat is computed with the generalized Mayer law:

$$C_V = C_P - T \cdot \left(\frac{\partial P}{\partial T} \right)_V \cdot \left(\frac{\partial V}{\partial T} \right)_P \quad (3.23)$$

- the kinematic viscosity of the gas:

$$\nu = \frac{\mu}{\rho} \quad (3.24)$$

- the specific heat ratio of the gas:

$$\gamma = \frac{C_P}{C_V} \quad (3.25)$$

- the isentropic calorific factor of the gas:

$$\gamma_S = \frac{C_V}{C_P} \cdot \gamma_T \quad (3.26)$$

where γ_T is the isothermal calorific factor of a gas which is defined hereafter:

$$\gamma_T = -\frac{P}{v} \cdot \left(\frac{\partial v}{\partial P} \right)_T \quad (3.27)$$

where v is the specific volume [m^3/kg].

These calorific factors are dependent on the gas model chosen or on the equation of state chosen to describe the gas behavior. For example, considering the perfect or semi-perfect (ideal) gas assumption, the isothermal calorific factor is equal to one, and so, the isentropic calorific factor is equal to the inverse of the specific heat ratio:

$$\gamma_S = \frac{1}{\gamma} \quad (3.28)$$

- the speed of sound:

$$c_s = \sqrt{\left(\frac{\partial P}{\partial \rho} \right)_S} = \sqrt{\gamma \cdot \left(\frac{\partial P}{\partial \rho} \right)_T} \quad (3.29)$$

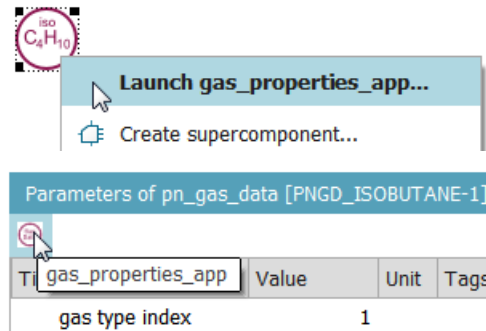
where S stands for entropy.

Considering a perfect or semi-perfect (ideal) gas behavior, the previous definition of the speed of sound becomes:

$$c_s = \sqrt{\gamma \cdot r \cdot T} \quad (3.30)$$

3.4 Setting/viewing gas properties in Simcenter Amesim

A specific app is available at *PNGD00* level to change and view the gas properties. This app can be launched directly when double-clicking or right-clicking on the *PNGD00* icon or in its parameters view.



The app shows properties as a function of temperature for a given pressure. The level of modeling can be changed to quickly assess the variations of properties it generates.

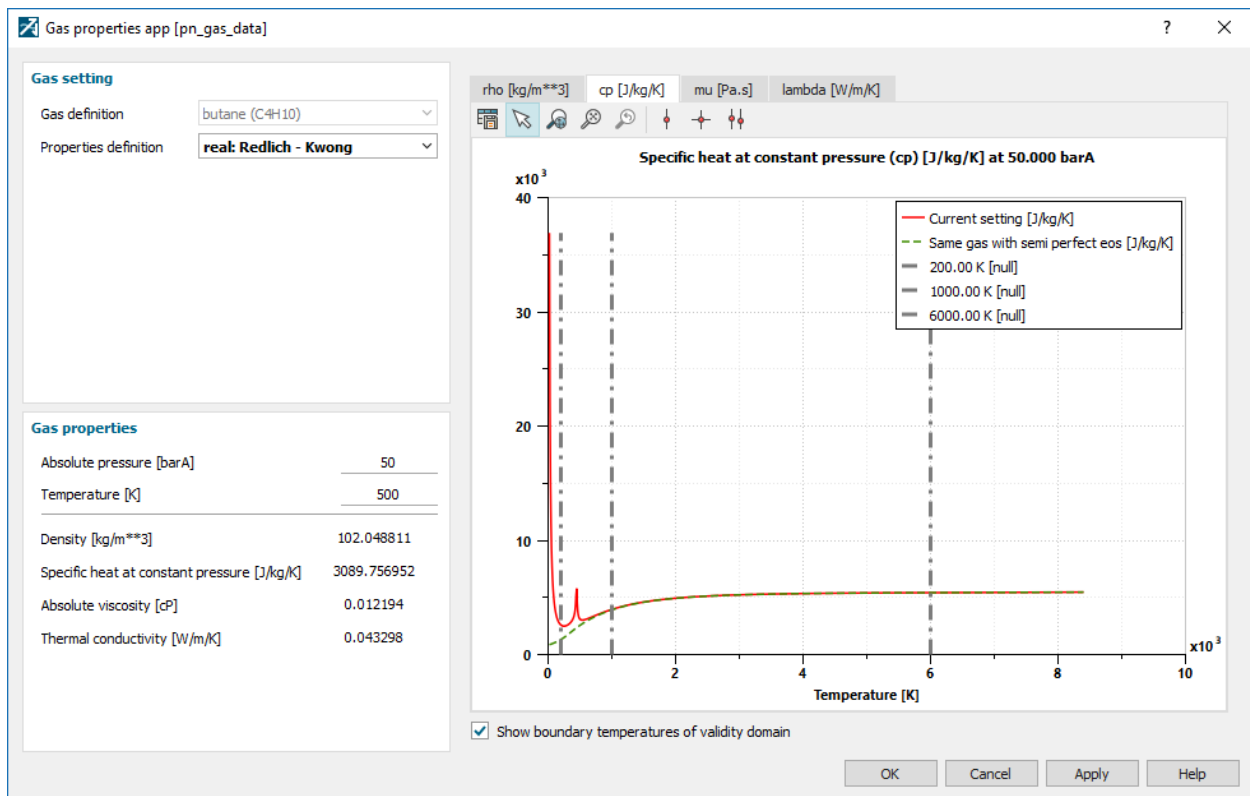


Fig. 3.8: “gas_properties_app”

As said above, the properties of a perfect gas are constant. Perfect gas constant and constant-pressure specific heat can be set through a pair of thermodynamics values.

Title	Value	Unit	Tags	Name
gas type index	2			gi
fluid definition	user defined			fluidType
properties definition	perfect			eosType
setting properties through	gamma + r			gasSetting
specific heat ratio (gamma)	gamma + r	null		gamma
perfect gas constant (r)	Cp + Cv	.../K		rGas
absolute viscosity (mu)	gamma + Cp	cP		mu
thermal conductivity	gamma + Cv	.../K		lam
enthalpy of formation	r + Cp	J/mol		enthForm
name of the gas	predefined hydrogen (H2)			gasName

Fig. 3.9: Setting a user-defined “perfect” gas

For predefined gas (customized submodels of *PNGD00*) it is possible to set the constant properties out of a given operating temperature and existing defined properties supposing a semi-perfect level of modeling.

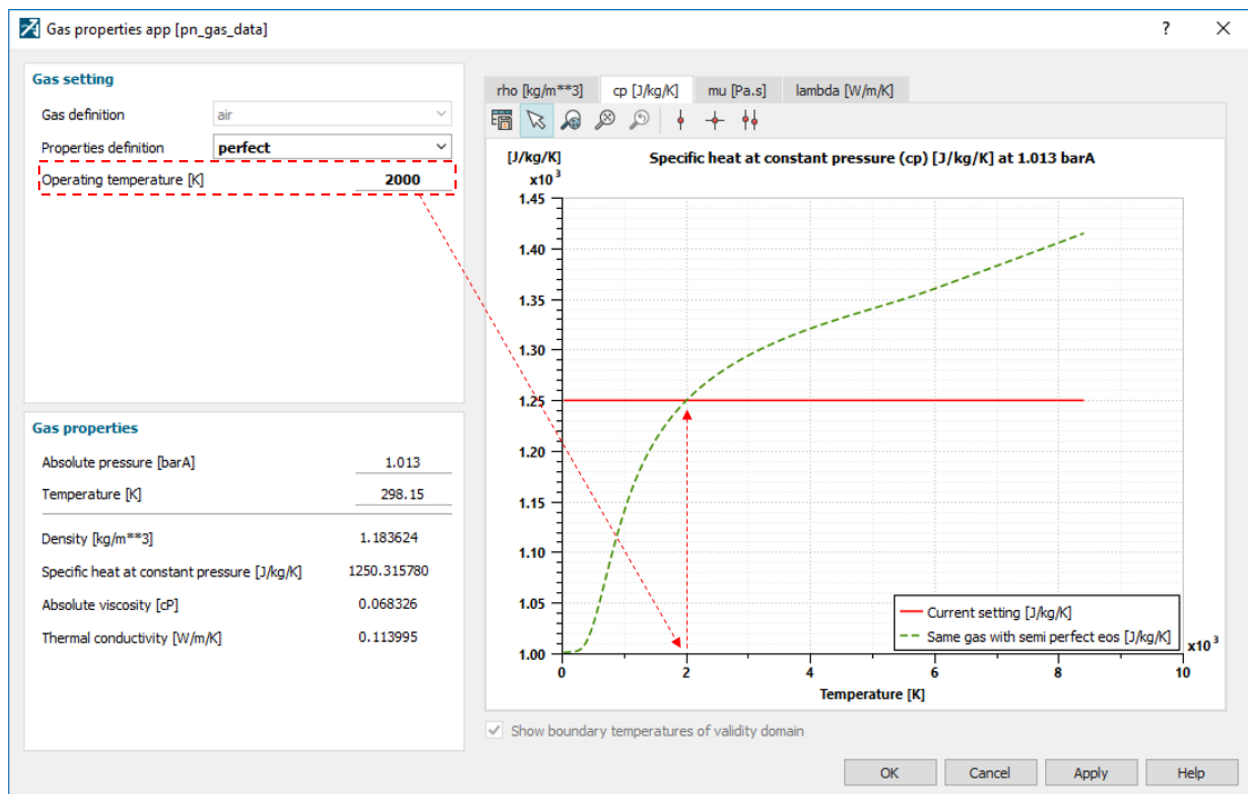


Fig. 3.10: Setting a predefined “perfect” gas

4. Running multi-gas simulations

In this section we describe how the **Pneumatic** library can handle **multi-gas simulations**. In some cases, several gases are involved in pneumatic systems. The rule that must be followed to run multi-gas simulations is:



Each icon represents a single gas.

The user must put on the sketch one gas properties icon for each gas involved in the system and set a different index for each icon. Note that perfect gas, semi-perfect or mixture gases can be used together. Finally, the gas type index in every other submodel of the system must be updated. This rule is applied in the following tutorial example.

4.1 Tutorial example

In order to illustrate a multi-gas simulation, we will compare the evolution of the pressures and temperatures of a 1 L accumulator submitted to a 7 barA charge and discharge, for two types of gas: hydrogen H_2 and oxygen O_2 . Why did we choose these two specific gases?

The reason is simple. They have thermal properties that are very different from one another: the constant-pressure specific heat of hydrogen is nearly fifteen times that of oxygen.

Table 4.1: Gas properties for hydrogen/oxygen at 293.15 K and at 1 barA

Gas	ρ [kg/m**3]	C_p [J/kg/K]	C_v [J/kg/K]	γ	r [J/kg/K]
Hydrogen	0.083	14287	10162	1.406	4124.5
Oxygen	1.313	917	656	1.395	259.8

4.1.1 Simulating gas properties

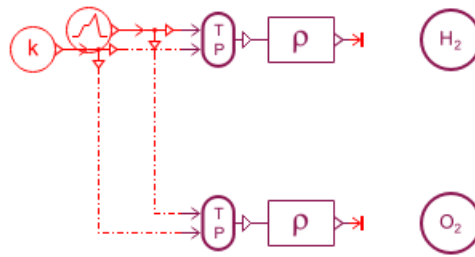


Fig. 4.1: Multi-gas example

You can use the pre-defined customized *PNGD_HYDROGEN* and *PNGD_OXYGEN* submodels which are described in the previous section.

Note:

You can either use existing customized submodels of PNGD00 (in **Submodel** or **Parameter** mode) or choose the adequate gas in the “fluid definition” enumeration of PNGD00 (in **Parameter** or **Submodel** mode).

Plotting the constant-pressure specific heat and the absolute viscosity of the two fluids with respect to the temperature leads to the following figures.

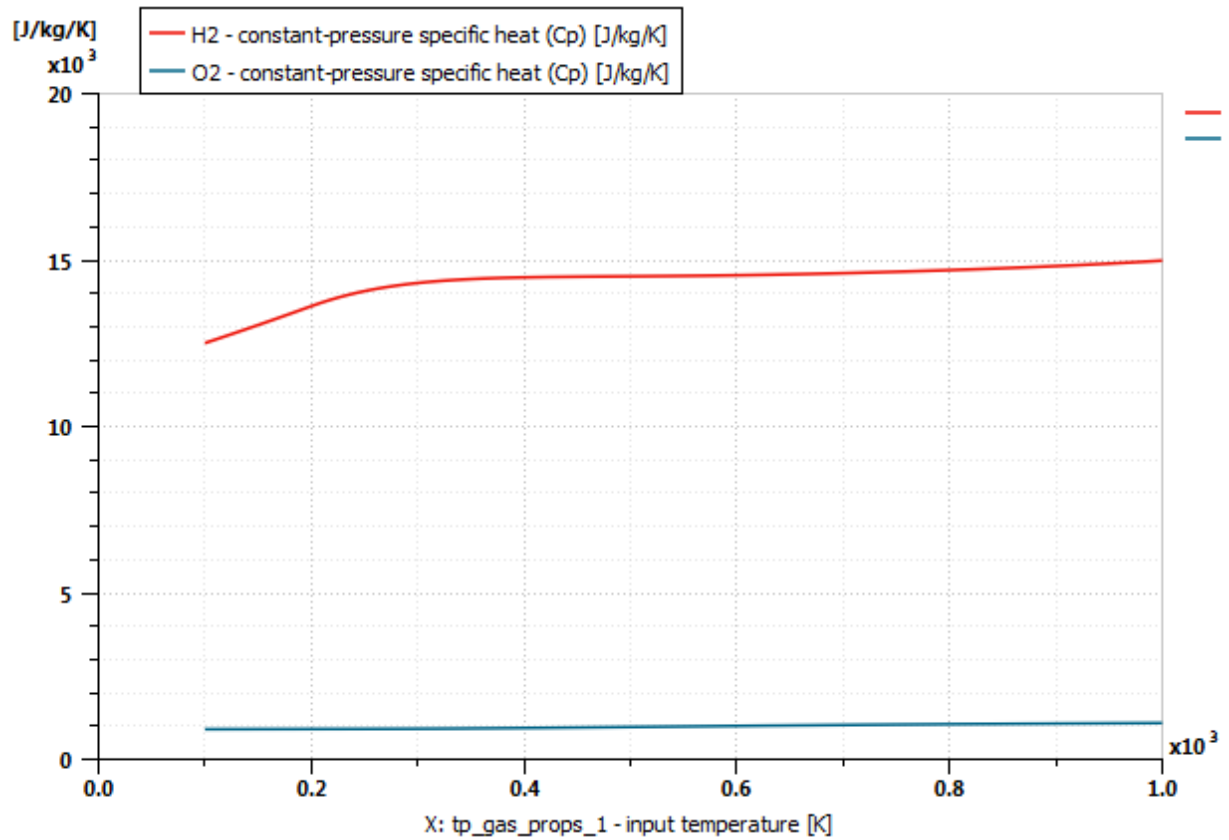


Fig. 4.2: Constant-pressure specific heat for hydrogen and oxygen

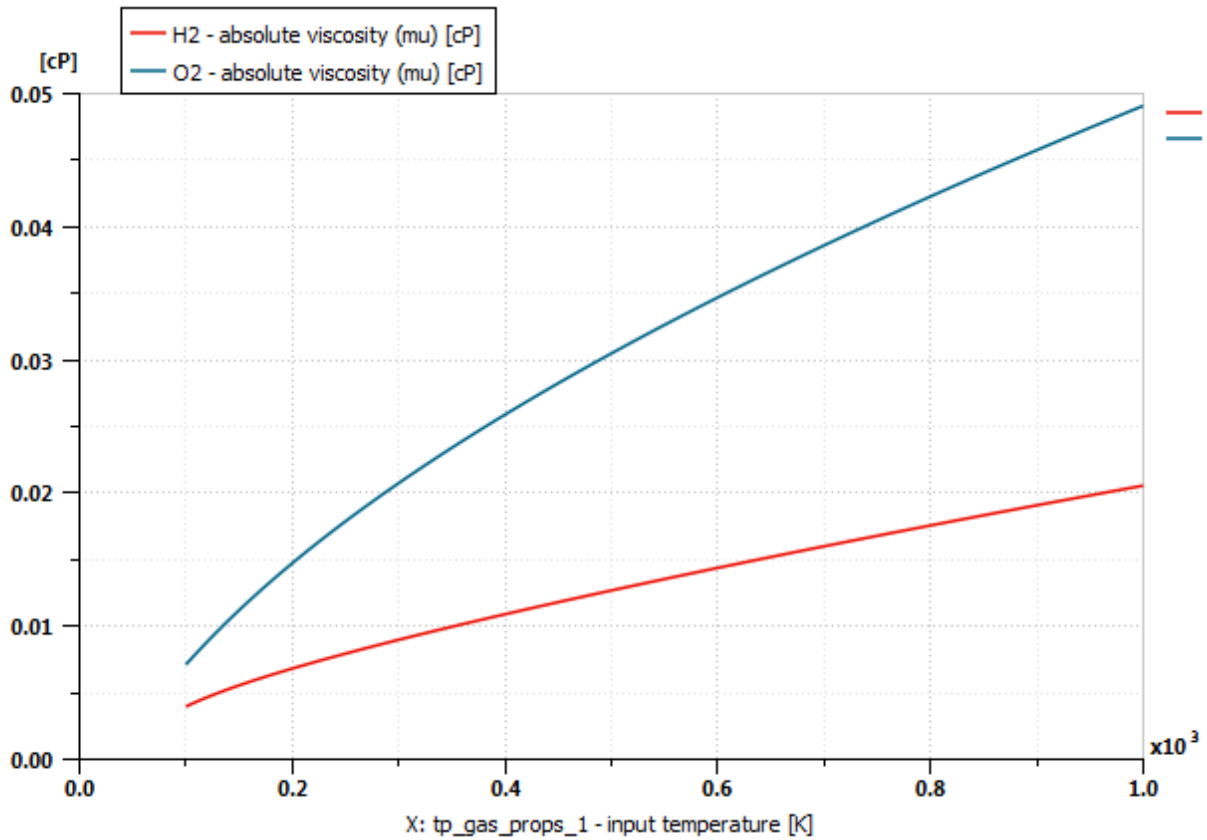


Fig. 4.3: Absolute viscosity for hydrogen and oxygen

4.1.2 Simulating the tank dynamics

Build the circuit shown in Fig.4.4:

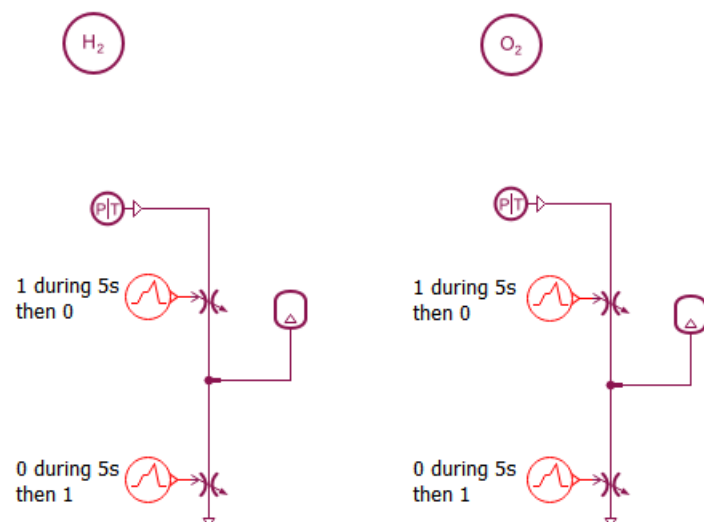


Fig. 4.4: Comparison of hydrogen and oxygen properties

This model comprises 2*8 components belonging to the **Pneumatic** and to the **Signal, Control** libraries. 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 *10 seconds* with a **Print interval** of *0.01 [s]*.

Note:

Do not forget to enter two different indexes for the different gases *PNGD_HYDROGEN* and *PNGD_OXYGEN* submodels. The index of the submodels in the hydrogen part of the model (a) should keep index 1, which is the default index, and the submodels in the oxygen part (b) should have (for example) index 2.

Table 4.2: parameters used in Fig. 4.4

Icon	Submodel	Category	Main parameters
	PNGD_HYDROGEN gas properties	Pneumatic	gas type index = 1
	PNGD_OXYGEN gas properties	Pneumatic	gas type index = 2
	PNCH022 pneumatic chamber	Pneumatic	default parameters
	PNCS001 pressure and temperature source	Pneumatic	pressure at port 1 = 7 barA
	PNVO001 modulated orifice	Pneumatic	default parameters
	PNAS001 atmospheric source	Pneumatic	default parameters

continues on next page

Table 4.2 – continued from previous page

Icon	Submodel	Category	Main parameters
	PN3NODE2 pneumatic node	Pneumatic	no parameter
	UD00 constant signal	Signal, Control	duration of stage 1 = 1 output at start of stage 1 = 1 output at end of stage 1 = 5
	UD00 constant signal	Signal, Control	duration of stage 1 = 5 output at start of stage 2 = 1 output at end of stage 2 = 1

When this is done, compare the pressures and temperatures of the pneumatic accumulators referenced by *1a* and *1b* as in [Fig.4.5](#) and [Fig.4.6](#).

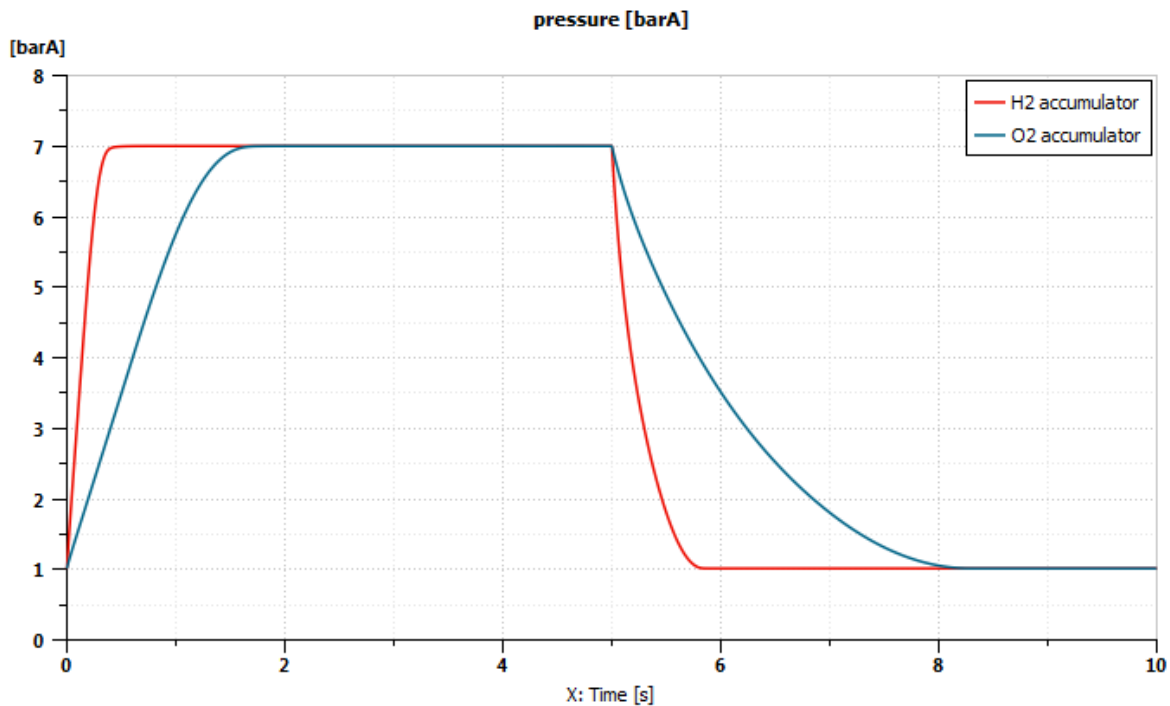


Fig. 4.5: Evolution of the pressures in hydrogen/oxygen accumulators

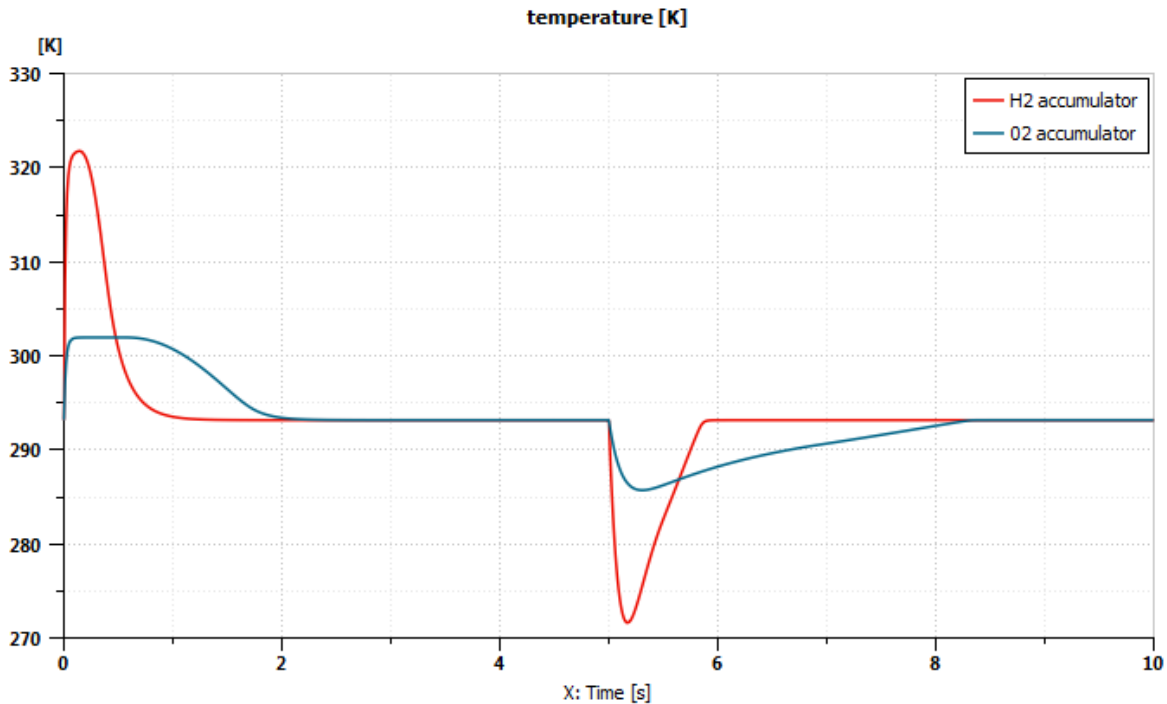


Fig. 4.6: Evolution of the temperatures in hydrogen/oxygen accumulators

You may have noticed that the hydrogen pressure and temperature reach their steady-state values faster than the oxygen. The higher specific heat and perfect gas constant explain the faster transient behavior of the hydrogen accumulator.

In the same way, the mass of gas inside the chamber is different due to the perfect gas constant differences as shown in [Fig.4.7](#).

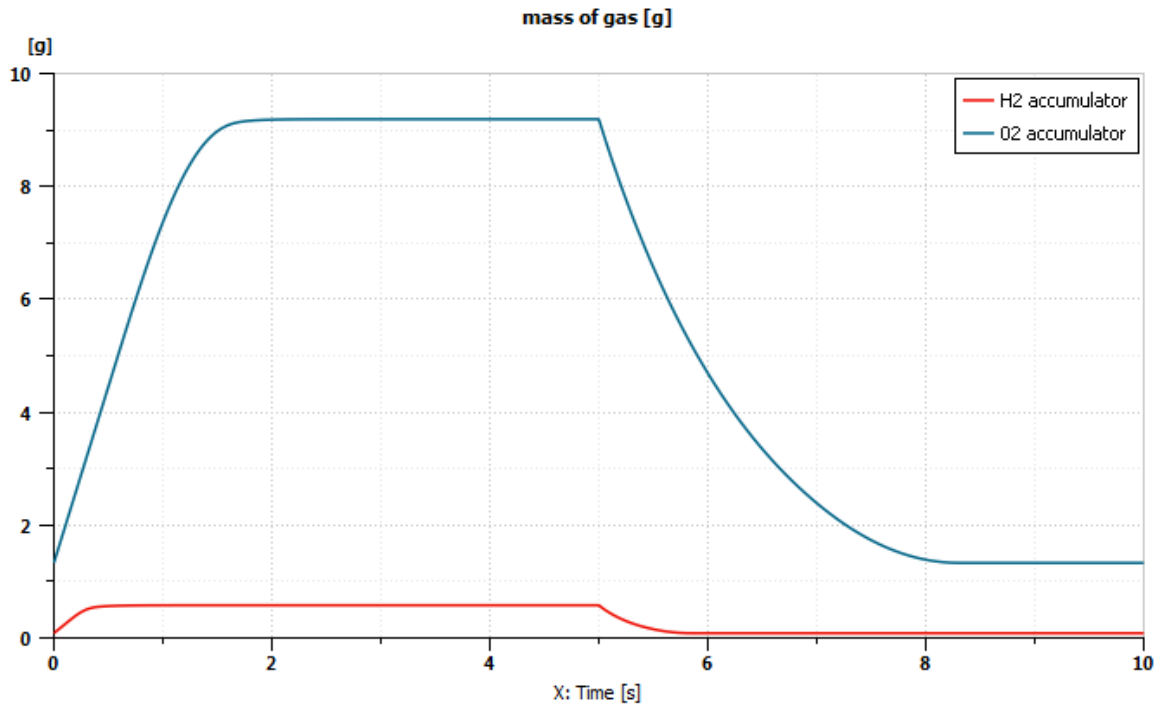


Fig. 4.7: Evolution of the mass of gas in hydrogen/oxygen accumulators

From this second basic example, we can understand how multi-gas simulations can be set up.

4.2 Using the gas mixture submodel

PNGM001 allows the user to define a predefined, chemically-inert, mixture of up to ten gases. All gases must share the same equation of state. The composition of the mixture is set through mass content for each gas.

Parameters of pn_gas_mix [PNGM001-1]

Title	Value	Unit	Tags	Name
index for gas mixture	10			gim
number of species	3			nspec
index of gas 1	11			gi1
index of gas 2	12			gi2
index of gas 3	13			gi3
mass content of gas 1	y0	null		pg1
mass content of gas 2	y1	null		pg2
mass content of gas 3	y2	null		pg3

Fig. 4.8: parameters of *PNGM001*

The properties of the PN mixture are defined through different equations and mixing rules as

in the **Gas Mixture** library. They are explained below. The only difference between a mixture in the **Pneumatic** library and a mixture in the **Gas Mixture** library is that the composition is fixed throughout the simulation in the former whereas it can vary during the simulation in the latter.

The normalized mass content gives the mass fractions which in turn gives the normalized molar fractions:

$$Y_i = \frac{m_i}{m_{\text{tot}}} \quad (4.1)$$

$$X_i = \frac{Y_i \cdot r_i}{r} \quad (4.2)$$

Where r , the perfect gas constant of the mixture is defined as:

$$r = \sum_i Y_i \cdot r_i \quad (4.3)$$

4.2.1 Density

The density is calculated with the pressure P and the temperature T through for a mixture of perfect or semi-perfect gases or through the resolution of the cubic equation of state for a mixture of real gases. In this latter case, special mixing rules are used to define the attractive term a , the covolume b and the acentric factor w .

$$a = \sum_i \sum_j Y_i \cdot Y_j \cdot \sqrt{a_i \cdot a_j} \quad (4.4)$$

$$b = \sum_i \sum_j Y_i \cdot Y_j \cdot \frac{b_i + b_j}{2} \quad (4.5)$$

$$w = \sum_i \sum_j X_i \cdot X_j \cdot \frac{w_i + w_j}{2} = \sum_i X_i \cdot w_i \quad (4.6)$$

The critical temperature T_c can be found out of the attractive term a and the covolume b as the following equalities must be verified for any equation of state:

$$\left. \frac{\partial P}{\partial v} \right|_T (P_c, T_c) = 0 \quad \text{and} \quad \left. \frac{\partial^2 P}{\partial v^2} \right|_T (P_c, T_c) = 0 \quad (4.7)$$

Which leads to the following definition for the critical temperature:

- Van der Waals:

$$T_c = \frac{8}{27} \cdot \frac{a}{r \cdot b} \quad (4.8)$$

- Redlich-Kwong:

$$T_c = \left(\frac{0.08664035}{0.42748024} \cdot \frac{a}{r \cdot b} \right)^{2/3} \quad (4.9)$$

- Redlich-Kwong-Soave:

$$T_c = \frac{0.08664035}{0.42748024} \cdot \frac{a}{r \cdot b} \quad (4.10)$$

- Peng-Robinson:

$$T_c = \frac{0.07779607}{0.457235583} \cdot \frac{a}{r \cdot b} \quad (4.11)$$

4.2.2 Specific heat and specific enthalpy

The specific heat and specific enthalpy are calculated through the simple mixing rules below for a mixture of perfect or semi-perfect gases.

$$C_p = \sum_i Y_i \cdot C_{p,i} \quad (4.12)$$

$$h = \sum_i Y_i \cdot h_i \quad (4.13)$$

The specific heat and specific enthalpy are defined with the same mixing rule plus a residue term function of the pressure and the temperature for a mixture of real gases.

$$C_p = \sum_i Y_i \cdot C_{p,i,NASA}(T) + C_{p,res}(P, T) \quad (4.14)$$

$$h(P, T) = \sum_i Y_i \cdot h_{i,NASA}(T) + h_{res}(P, T) \quad (4.15)$$

h_{res} results from the integration over temperature and pressure of $C_{p,res}$. This residue term depends on the equation of state.

- Van der Waals

$$C_{p,res} = \frac{-r}{1 - \frac{0.5 \cdot v^3 \cdot r \cdot T}{a \cdot (v-b)^2}} \quad (4.16)$$

- Redlich-Kwong

$$C_{p,res} = -r + \frac{3a}{4b \cdot T \cdot \sqrt{T}} \log\left(1 + \frac{b}{v}\right) - T \cdot \left(\frac{\partial P}{\partial T}\right)_v^2 \cdot \frac{\partial v}{\partial P}\bigg|_T \quad (4.17)$$

- Redlich-Kwong-Soave

$$C_{p,res} = -r + \frac{a \cdot T}{b} \left(\frac{m^2}{2T \cdot T_c} + \frac{m \cdot \left[1 + m \left(1 - \sqrt{\frac{T}{T_c}}\right)\right]}{\sqrt{\frac{T}{T_c}} \cdot \frac{T}{T_c} \cdot \frac{T_c}{2}} \right) \log\left(1 + \frac{b}{v}\right) - T \cdot \left(\frac{\partial P}{\partial T}\right)_v^2 \cdot \frac{\partial v}{\partial P}\bigg|_T \quad (4.18)$$

- Peng-Robinson

$$C_{p,res} = -r - \frac{a \cdot T}{2\sqrt{2} \cdot b} \left(\frac{m^2}{2T \cdot T_c} + \frac{m \cdot \left[1 + m \left(1 - \sqrt{\frac{T}{T_c}}\right)\right]}{\sqrt{\frac{T}{T_c}} \cdot \frac{T}{T_c} \cdot \frac{T_c}{2}} \right) \log\left(\frac{v + b(1 - \sqrt{2})}{v + b(1 + \sqrt{2})}\right) - T \cdot \left(\frac{\partial P}{\partial T}\right)_v^2 \cdot \frac{\partial v}{\partial P}\bigg|_T \quad (4.19)$$

4.2.3 Absolute viscosity and thermal conductivity

Whatever the level of modeling, the absolute viscosity of a mixture follows Eq.4.20; the thermal conductivity of a mixture follows Eq.4.22 (see [9], [8] and [6]).

The coefficients for these equations are recalculated so as to be equivalent to the following mixing rules:

$$\mu = \sum_i \frac{X_i \cdot \mu_i}{\sum_j X_j \cdot \varphi_{i,j}} \quad (4.20)$$

where:

$$\varphi_{i,j} = \frac{1}{\sqrt{8} \cdot \sqrt{1 + \frac{M_i}{M_j}}} \left[1 + \sqrt{\frac{\mu_i}{\mu_j}} \cdot \left(\frac{M_j}{M_i} \right)^{\frac{1}{4}} \right]^2 \quad (4.21)$$

and:

$$\lambda = \sum_i \frac{X_i \cdot \lambda_i}{\sum_j X_j \cdot \varphi_{i,j}} \quad (4.22)$$

where:

$$\varphi_{i,j} = \frac{1}{\sqrt{8} \cdot \sqrt{1 + \frac{M_i}{M_j}}} \left[1 + \sqrt{\frac{\lambda_i}{\lambda_j}} \cdot \left(\frac{M_j}{M_i} \right)^{\frac{1}{4}} \right]^2 \quad (4.23)$$

4.2.4 Example

As an example of the use of the gas mixture model, we will try to reproduce the thermal properties of air from its constituents.

The sea-level composition of air (in percent by volume at the temperature of 15 degC and the pressure of 1.01325 barA) is given below:

Name	Symbol	Percent by Volume
Nitrogen	N ₂	78.084 %
Oxygen	O ₂	20.9476 %
Argon	Ar	0.934 %
Carbon Dioxide	CO ₂	0.0314 %
Neon	Ne	0.001818 %
Methane	CH ₄	0.0002 %
Helium	He	0.000524 %
Krypton	Kr	0.000114 %
Hydrogen	H ₂	0.00005 %
Xenon	Xe	0.0000087 %

To simplify the example, we consider that air is constituted with only 78% N₂, 21% O₂ and 1% Ar.

In the system shown in Fig.4.9, three *PNGD00* semi-perfect submodels are included to access the customized N₂, O₂ and Ar properties (customized submodels *PNGD_NITROGEN*, *PNGD_OXYGEN* and *PNGD_ARGON*). The validity domain for all gases is [200 K; 20000 K] for constant-pressure specific heat and [200K ; 15000 K] for absolute viscosity and thermal conductivity. The mixture shares these values.

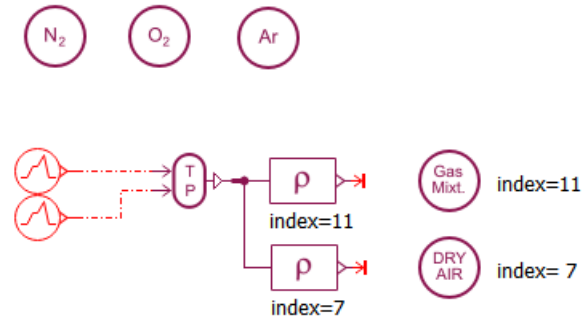


Fig. 4.9: Use of gas mixture submodel

The *PNGM001* gas mixture submodel allows us to compute the properties of air from the ratios of its constituents (up to 10). It creates a new gas which will be referenced by a new gas type index (11 by default).

Note:

The ratios are mass ratios whereas the compositions given in Table 4 are volume ratios. Therefore, the mass ratios must be computed from the volume ratios considering the densities of every constituent. That leads to 75 % *N2*, 23 % *O2* and 2 % *Ar* composition in mass.

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 *10 seconds*. Vary the input temperature and pressure and run a simulation for *10 seconds*.

Table 4.3: components used in Fig. 4.9

Icon	Submodel	Category	Main parameters
	PNGD_NITROGEN gas properties	Pneumatic	gas type index = 1
	PNGD_OXYGEN gas properties	Pneumatic	gas type index = 2
	PNGD_ARGON gas properties	Pneumatic	gas type index = 3

continues on next page

Table 4.3 – continued from previous page

Icon	Submodel	Category	Main parameters
	PNGM001 gas mixture	Pneumatic	mass content of gas 1 = 75 mass content of gas 2 = 23 mass content of gas 3 = 2
	PNGD_AIR gas properties	Pneumatic	gas type index = 7
	UD00 piecewise signal	Signal, Control	Stage 1 From 100 to 22000 in 10 s
	UD00 piecewise signal	Signal, Control	Stage 1 constant value = 1
	PNVS001 conversion of signals into temperature and pressure	Pneumatic	No parameter
	TPGPROP002 calculation of pneumatic properties	Pneumatic	gas type index = 11
	TPGPROP002 calculation of pneumatic properties	Pneumatic	gas type index = 7
	SSINK signal plug	Signal, Control	No parameter

When the simulation is over, click on the icon associated with submodel *TPGPROP02*, and plot the constant-pressure specific heat and the absolute viscosity of the air mixture with respect to temperature.

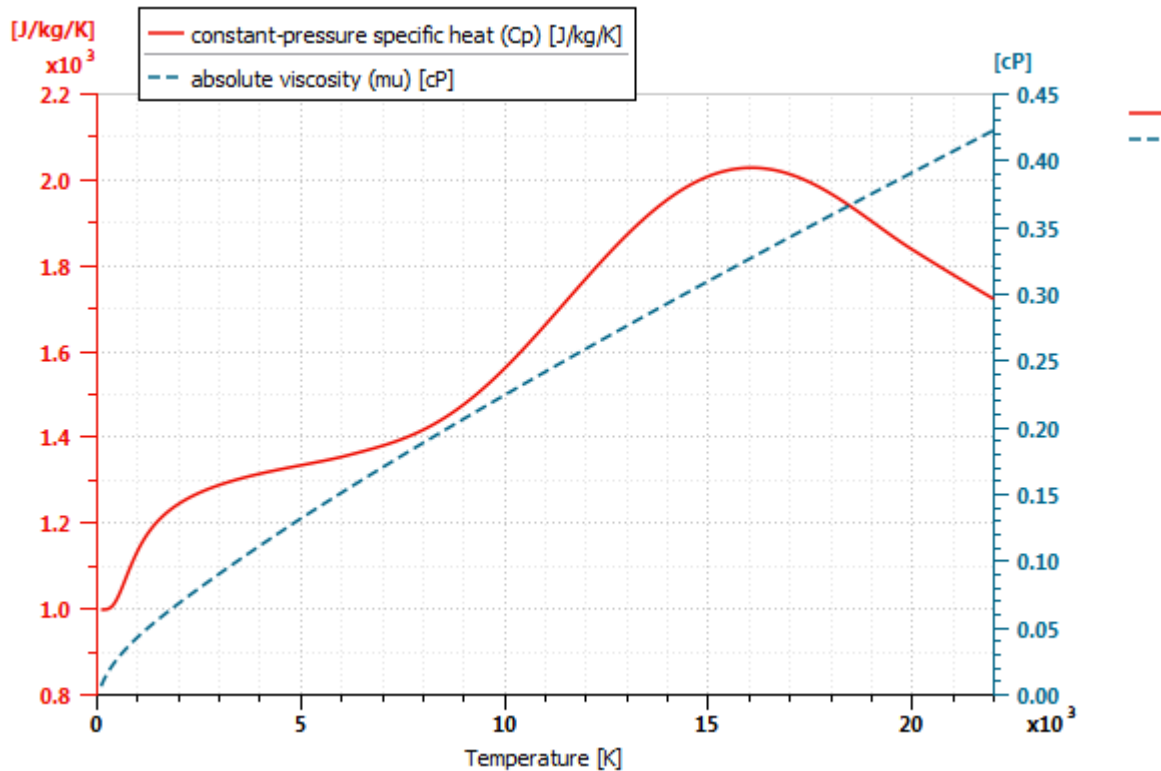


Fig. 4.10: Air mixture properties with respect to temperature

You can check that these properties are similar to the ones usually used for air. The results are different above 6000 K for constant-pressure specific heat because the validity domain of the predefined air is [200 K; 6000 K] and a quadratic extrapolation is made above the maximum temperature. Note that this limitation comes from water vapor. There is no difference for absolute viscosity because the validity domain of predefined air is the same as for the mixture: [200 K; 15000 K].

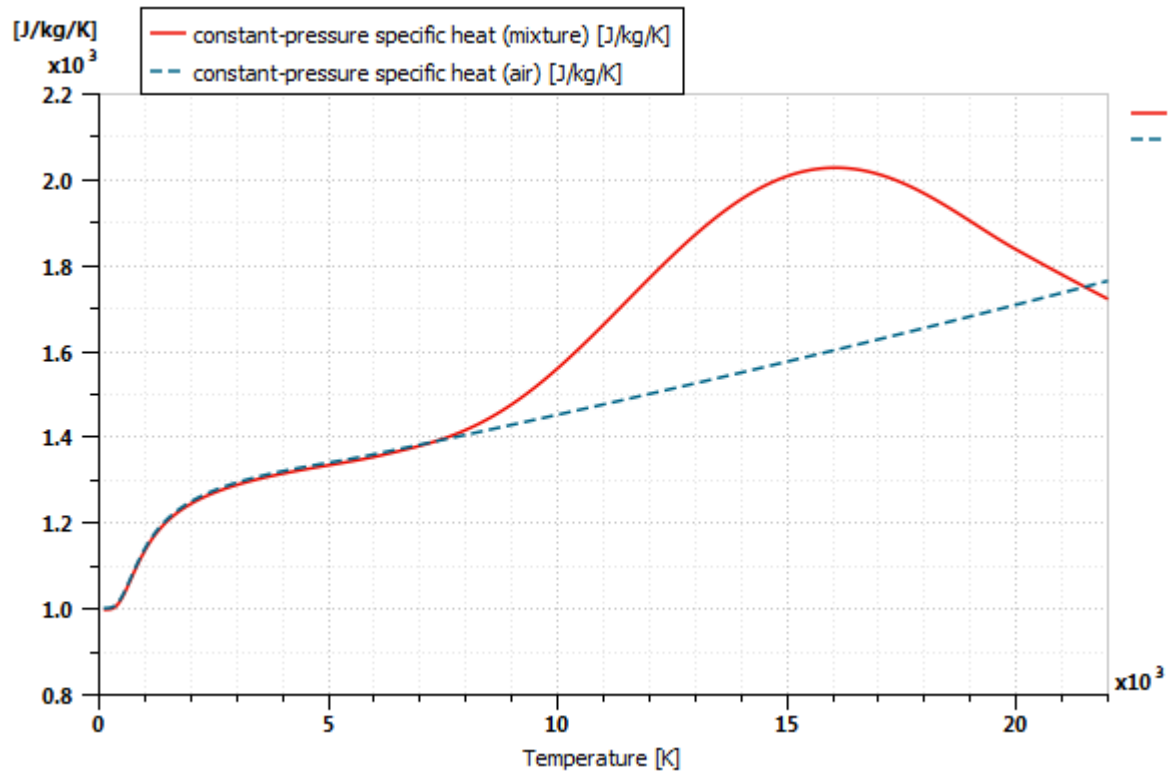


Fig. 4.11: Comparison of mixture and predefined air for C_p

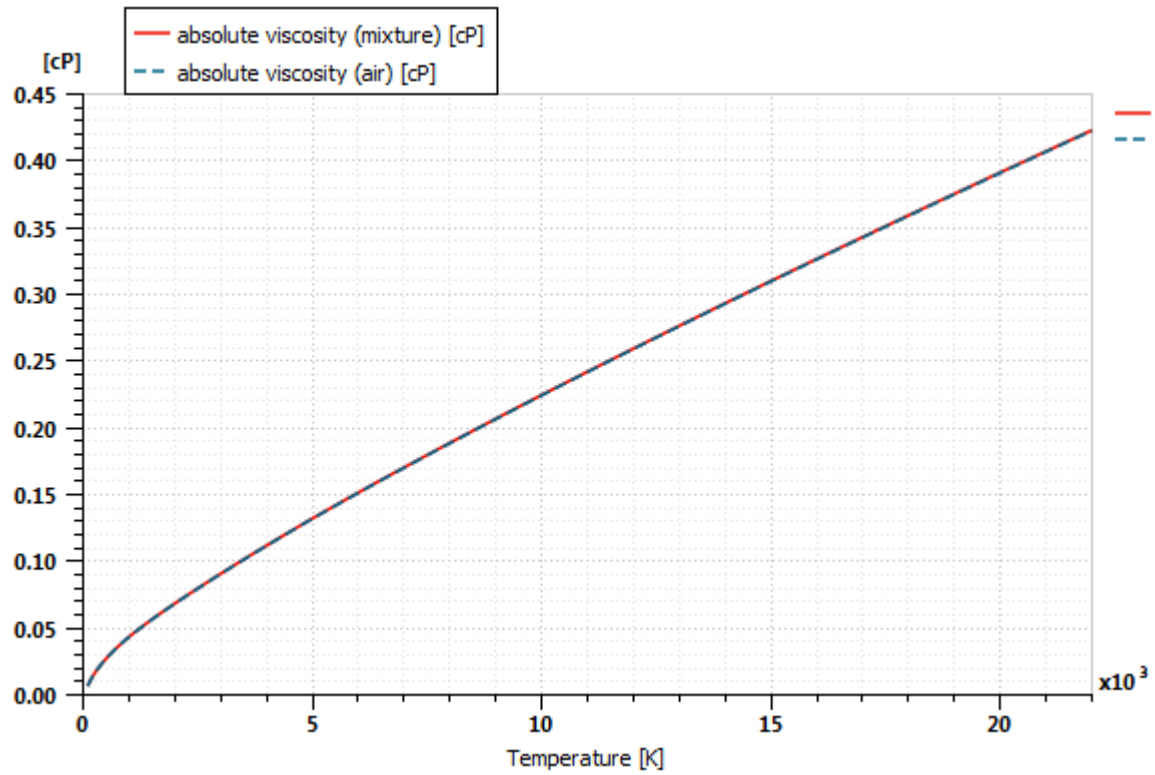


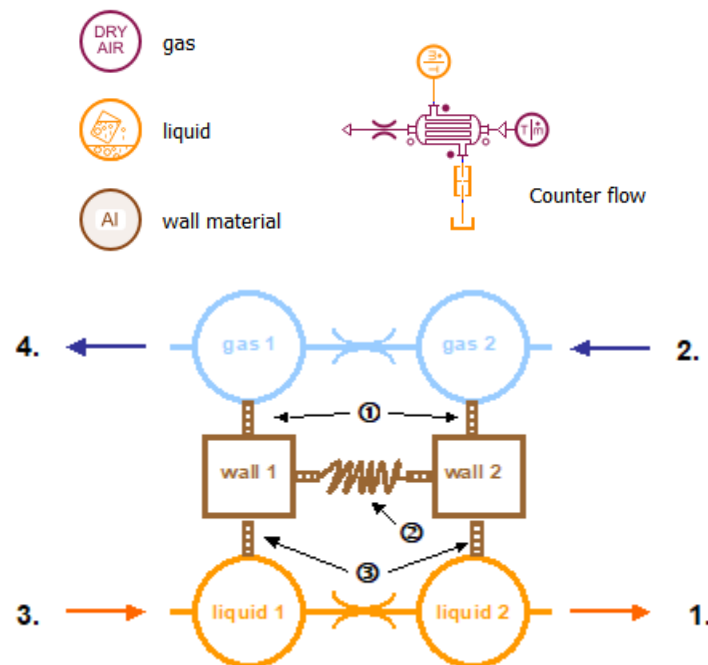
Fig. 4.12: Comparison of mixture and predefined air for viscosity

5. Running multi-domain simulations: gas-liquid heat exchangers

The process of heat exchange between two fluids (one gas and one liquid in our case) at different temperatures and separated by a solid wall occurs in many engineering applications. Heat exchangers are typically classified according to flow arrangement and type of construction. The simplest heat exchanger is one for which the hot and cold fluids move in the same or opposite directions in concentric tubes.

We propose to examine a gas-liquid heat exchanger since the corresponding model is built from the three **Pneumatic**, **Thermal-Hydraulic** and **Thermal** libraries.

The standard **Pneumatic** library provides a model of a heat exchanger for which the solid wall is divided into two sections to represent the evolution of the temperature along the tubes. The inner structure is presented in the next figures.



The heat exchanges considered are:

1. convection and radiation between the gas and the pipe walls,
2. conduction between the two solid parts of the pipes,
3. convection between the liquid and the pipe walls.

The interesting point here is to compare the difference in the heat exchanger performances due to mixing conditions.

This is illustrated with the proposed model in Fig.5.1. It comprises 16 referenced elements. It consists of a 0.001 kg/s , 200 degC hot liquid flow which is refreshed with a 10 g/s , 293.15 K fresh airflow. We consider a counter flow heat exchanger and a parallel flow heat exchanger.

Fill in the parameters of these components as described in the table below leaving the others at their default values. Finally, run a simulation with a final time of *200 seconds* with a **Print interval** of *1 second*.

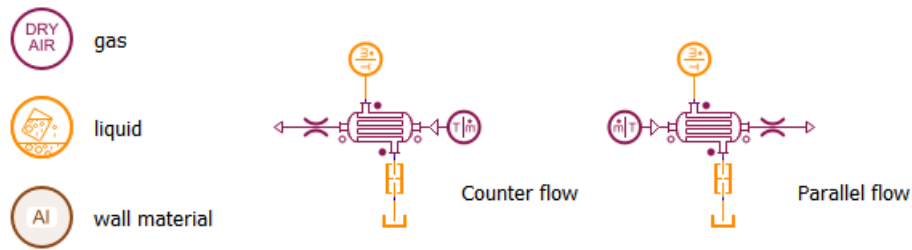


Fig. 5.1: Counterflow and parallel flow heat exchangers

Table 5.1: Parameters for counter flow and parallel flow heat exchangers

Icon	Name	Category	Main parameters
	PN_AIR pneumatic properties	Pneumatic	default parameters
	TFFD3_OIL_15W40 thermal-hydraulic properties	Thermal-Hydraulic	default parameters
	THSD00 thermal solid properties	Thermal	default parameters
	PNAS001 atmospheric source	Pneumatic	default parameters
	PNOR001 orifice	Pneumatic	orifice area = 100 mm ²
	TFORF0 orifice	Thermal-Hydraulic	default parameters

continues on next page

Table 5.1 – continued from previous page

Icon	Name	Category	Main parameters
	TFTK0 tank	Thermal-Hydraulic	default parameters
	PNCS002 mass and enthalpy flow rates source	Pneumatic	output mass flow = 10 g/s
	HEGL002 heat exchanger	Pneumatic	temperature at port 1 = 200 degC temperature at port 3 = 200 degC
	TFMTS mass and enthalpy flow rates source	Thermal-Hydraulic	mass flow rate at port 1 = 0.001 kg/s source temperature = 200 degC

The temperatures for the exchangers are then plotted on [Fig.5.2](#).

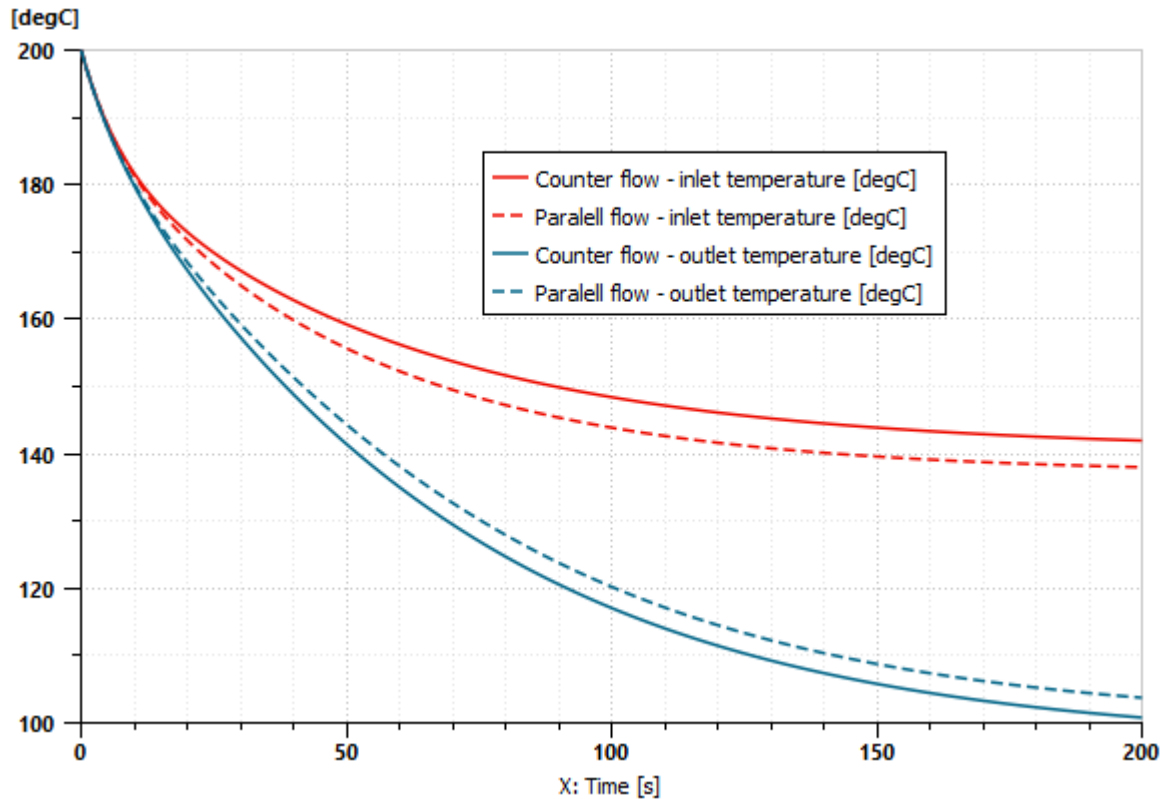


Fig. 5.2: Comparison of the liquid temperatures

Note:

For these two systems, the outlet liquid temperature is lower in the case of a counterflow heat exchanger than with a parallel flow heat exchanger.

6. Important rules for modeling a system with pneumatic components

This section will detail the important rules that should be respected to model a system with **Pneumatic** library components correctly.

6.1 Causality in the Pneumatic library

As previously stated, it is necessary to distinguish two types of component in the **Pneumatic** library.

- Capacitive components are volumes in which the temperature and the pressure are computed from the enthalpy and mass flow rates inputs at their ports.
- Resistive components are those in which the enthalpy and mass flow rates are evaluated from the temperature and pressure inputs at their ports.

This implies that a pneumatic model is always built in such a way that an extreme resistive element R is always connected to an extreme capacitive element C as shown in [Fig.6.1](#).

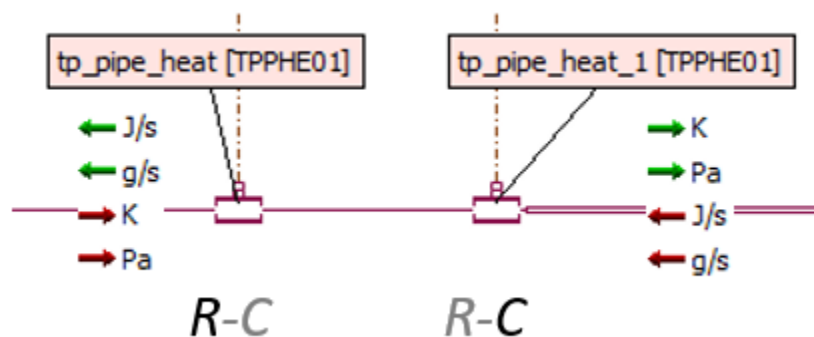


Fig. 6.1: Connection rule

For this reason, it is important for you to understand the inner structure of the submodels used (R , C - R , C - R - C , ...) to ensure correct connection.

6.2 Important rules

- The **Pneumatic** library deals with gases only.
- It is better if you build **the system slowly** and run a simulation after each portion of circuit is added. This normally leads to a better understanding of the system and the results. It is also less likely that the setting of parameters will be forgotten. The user will have to be very careful when running multi-gas simulations: the gas type indexes must be adjusted in the corresponding submodels.
- This version of the **Pneumatic** library includes more than 50 submodels. Full documentation of these submodels is available in HTML format or, more briefly, directly in **Simcenter Amesim**, in the description of each submodel.
- The **Pneumatic** library can be coupled with the **Thermal** and **Thermal-Hydraulic** libraries to study interactions between gases, liquids and solids as shown in the previous examples. In the **Pneumatic** library, thermal ports have been added to pneumatic components:

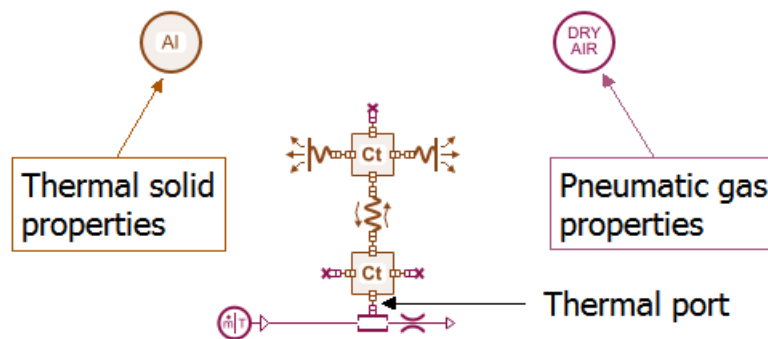


Fig. 6.2: Coupling thermal and pneumatic components

- In some extreme cases, you will have to change the value of the tolerance used by the integration algorithm to control the accuracy of the solution. You can access this parameter in the **Run Parameters** dialog box in **Simulation** mode. Its default value (10^{-7}) is satisfactory for most situations.

Sometimes, calculations may be unusually slow with a large number of discontinuities being handled during the simulation. This may be due to integration “noise” where small variations of some state variables are being greatly amplified and the resulting variations create discontinuities.

A reduced tolerance will reduce the noise and may cure the problem. Notice that it is normally unsatisfactory to set values for the tolerance higher than 10^{-3} or lower than 10^{-12} .

7. Brief review of the theory: pneumatic calculations

7.1 Pneumatic assumptions

The submodels in the **Pneumatic** library use the first and second law of thermodynamics.

The following assumptions are used in the submodels of the **Pneumatic** library:

- the velocity of gas is not supersonic,
- the flow is one-dimensional,
- the gas is considered to be a perfect, semi-perfect, or a real gas,
- the gravitational effect on the gas is not taken into account,
- the intensive variables, i.e. pressure and temperature, are assumed to be average values in the whole volume considered (lumped modeling),
- the kinetic energy is ignored in chamber submodels,
- reversible thermodynamic transformations are used in the orifice submodels.

7.2 Modeling pneumatic orifices

Almost all pneumatic systems you are likely to model will contain orifices. This section describes how they are modeled. The kinetic energy of the upstream flow is ignored. The mass flow rate dm through a pneumatic orifice is a function of input pressures (absolutes) and temperatures.

Assuming the gas is a compressible fluid, we have:

$$dm = A \cdot C_q \cdot C_m \cdot \frac{P_{up}}{\sqrt{T_{up}}} \quad (7.1)$$

where T_{up} is the upstream temperature and C_m is the flow parameter.

The flow coefficient C_q is introduced to adjust the theoretical relationship to experiments. This coefficient is used to include extra losses due to local friction and loss of kinetic energy. It may represent frictional drag in pipes for instance, or local resistance due to changes in geometry or in flow direction.

The flow parameter is determined using the following general expression:

$$C_m = \sqrt{\frac{2}{1 - \gamma_s} \cdot \frac{T_{up}}{v_{up} \cdot P_{up}}} \cdot \sqrt{\left(\frac{P_{dn}}{P_{up}}\right)^{2 \cdot \gamma_s} - \left(\frac{P_{dn}}{P_{up}}\right)^{1 + \gamma_s}} \quad (7.2)$$

where T_{up} , P_{up} and v_{up} are respectively the upstream temperature, pressure and specific volume, T_{dn} is the downstream temperature and γ_s the isentropic calorific factor.

Considering a perfect or semi-perfect (ideal) gas behavior, the flow parameters written as following:

$$C_m = \begin{cases} \sqrt{\frac{2 \cdot \gamma}{\gamma - 1}} \cdot \sqrt{\left(\frac{P_{dn}}{P_{up}}\right)^{\frac{2}{\gamma}} - \left(\frac{P_{dn}}{P_{up}}\right)^{\frac{\gamma+1}{\gamma}}} & \text{if } \frac{P_{dn}}{P_{up}} > P_{cr} \text{ (subsonic)} \\ \sqrt{\frac{\gamma}{r} \cdot \left(\frac{2}{\gamma-1}\right)^{\frac{\gamma+1}{\gamma-1}}} = \sqrt{\frac{2 \cdot \gamma}{\gamma-1}} \cdot \left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}} & \text{if } \frac{P_{dn}}{P_{up}} \leq P_{cr} \text{ (sonic)} \end{cases} \quad (7.3)$$

Therefore, when the flow is sonic, the flow parameter is constant and only a function of the gas properties, whereas in subsonic flow, the flow parameter is a function of the pressure ratio $\frac{P_{dn}}{P_{up}}$.

The critical pressure ratio at which the flow switches from sonic to subsonic mode can be calculated using the equation:

$$P_{cr} = \left(\frac{2 \cdot \gamma_s}{\gamma_s + 1} \right)^{\frac{1}{1-\gamma_s}} \quad (7.4)$$

Here again, considering a perfect or semi-perfect gas behavior, the previous equation becomes the well-known critical pressure ratio expressed as a function of the specific heat ratio:

$$P_{cr} = \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma-1}} \quad (7.5)$$

The evolution of the flow parameter for air is shown in Fig.7.1.

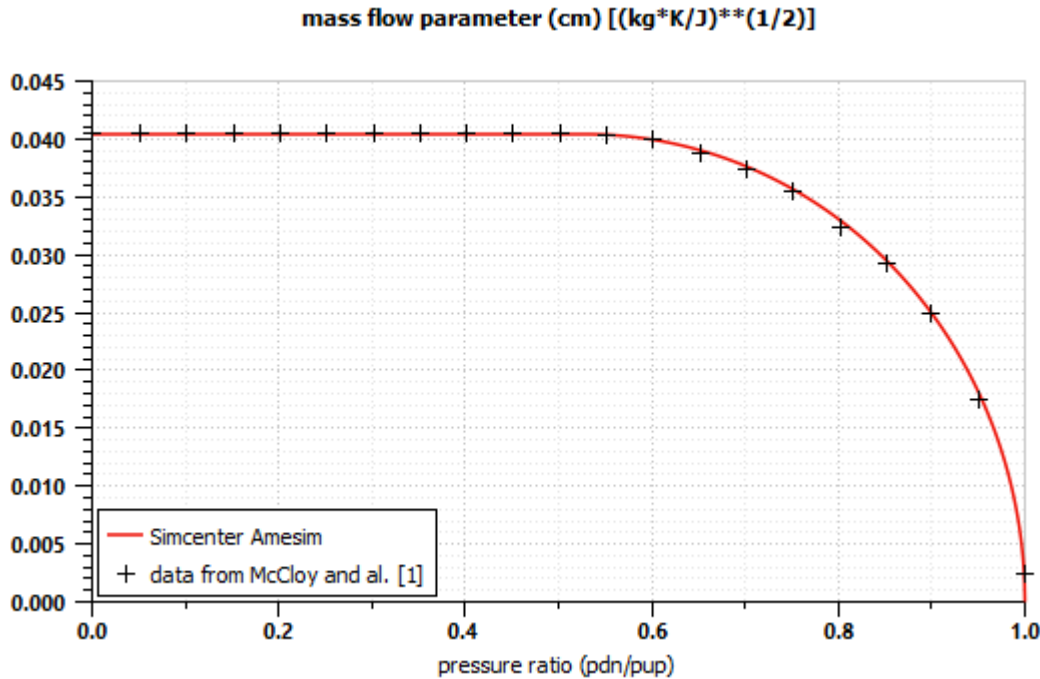


Fig. 7.1: Flow parameter as a function of pressure ratio

The enthalpy flow rate dmh is calculated as follows:

$$dmh = dm \cdot h(P_{up}, T_{up}) \quad (7.6)$$

where $h(P_{up}, T_{up})$ is the gas specific enthalpy at upstream conditions.

The specific enthalpy derives from the polynomial form of the specific heat of the gas:

$$dh = C_p \cdot dT \quad (7.7)$$

with:

$$C_p = C_{p0} \cdot [C_{pc} + C_{pt} \cdot \Delta T + C_{pt2} \cdot \Delta T^2] \quad (7.8)$$

After integration:

$$h = C_{pc} \cdot C_{p0} \cdot T_0 + C_{p0} \cdot \Delta T \cdot \left(C_{pc} + C_{pt} \cdot \frac{\Delta T}{2} + C_{pt2} \cdot \frac{\Delta T^2}{3} \right) \quad (7.9)$$

The *vena contracta* mean gas velocity is calculated from the mass flow rate, the cross-sectional area of the fitting and the flow coefficient C_q using the assumption that the evolution is isentropic (adiabatic and reversible) between the upstream port and the *vena contracta*.

Therefore, the *vena contracta* specific volume (or density) is a function of the upstream specific volume (or density) independently of the gas behavior:

$$v_{vc} = \left(\frac{2 \cdot \gamma_s}{\gamma_s + 1} \right)^{\frac{\gamma_s}{1-\gamma_s}} \cdot v_{up} \quad (7.10)$$

where v_{vc} is the *vena contracta* specific volume and v_{up} is the upstream specific volume.

The critical gas velocity corresponds to the speed of sound of the gas calculated at the *vena contracta* temperature:

$$v_{max} = \sqrt{\frac{2}{1 + \gamma_s} \cdot v_{up} \cdot P_{up}} \quad (7.11)$$

where v_{max} is the critical gas velocity, v_{up} is the upstream specific volume, P_{up} is the upstream pressure and γ_s the isentropic calorific factor.

Considering a perfect or semi-perfect gas behavior, the previous expression of the critical velocity becomes:

$$v_{max} = \sqrt{\frac{2}{\gamma + 1} \cdot \gamma \cdot r \cdot T_{up}} \quad (7.12)$$

where T_{up} is the upstream temperature, r is the perfect gas constant and γ the specific heat ratio.

This is equivalent to the critical velocity computed at the *vena contracta* temperature:

$$v_{max} = \sqrt{\gamma \cdot r \cdot T_{vc}} \quad (7.13)$$

where v_{max} is the critical gas velocity, T_{vc} is the *vena contracta* temperature, r is the perfect gas constant and γ the specific heat ratio.

7.2.1 Submodel with fixed flow coefficient

A straightforward approximation consists in taking the flow coefficient C_q as a constant.

7.2.2 Submodel with polynomial Perry's flow coefficient

Since the flow aspect changes with the pressure ratio, a slightly more complex hypothesis is to assume that the flow coefficient C_q is a polynomial function of the pressure ratio $\frac{P_{dn}}{P_{up}}$.

The Perry's flow coefficient is used for sharp-edged orifices and it can be approximated by a polynomial function of the pressure ratio $\frac{P_{dn}}{P_{up}}$:

$$C_q = \left[\left\{ \left[\left(-1.6827 \cdot \frac{P_{dn}}{P_{up}} + 4.6 \right) \cdot \frac{P_{dn}}{P_{up}} - 3.9 \right] \cdot \frac{P_{dn}}{P_{up}} + 0.8415 \right\} \cdot \frac{P_{dn}}{P_{up}} - 0.1 \right] \cdot \frac{P_{dn}}{P_{up}} + 0.8414 \quad (7.14)$$

The variation of flow coefficient C_q is shown in Fig.7.2.

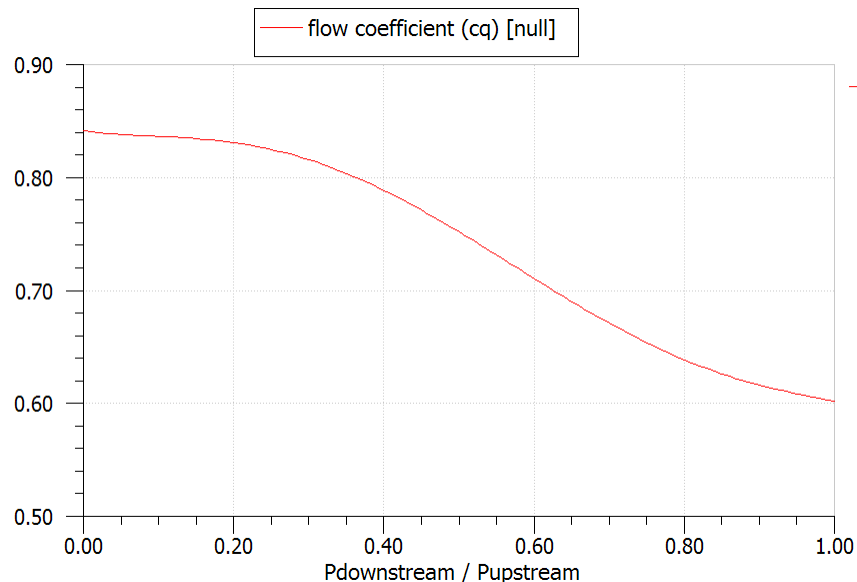


Fig. 7.2: Perry's flow coefficient

7.2.3 Submodel with ISO-6358 approximation

When some experimental measurements for an orifice are provided, it is more common to use the ISO 6358 norm (see [4]) which defines the static characteristics of a pneumatic orifice using two parameters only:

- The **critical pressure ratio b** is the pressure ratio which localizes the transition between subsonic and sonic flow regions.
- The **sonic conductance C** is the flow characteristics of the orifice (in $\text{s.m}^4 / \text{kg}$).

In the ISO 6358 norm, the mass flow rate is evaluated using the following equations:

$$dm = \begin{cases} C \cdot P_{up} \cdot \rho_0 \cdot \sqrt{\frac{T_0}{T_{up}}} \sqrt{1 - \left(\frac{P_{dn}}{P_{up}} - b \right)^2} & \text{if } \frac{P_{dn}}{P_{up}} > b \text{ (subsonic)} \\ C \cdot P_{up} \cdot \rho_0 \cdot \sqrt{\frac{T_0}{T_{up}}} & \text{if } \frac{P_{dn}}{P_{up}} \leq b \text{ (sonic)} \end{cases} \quad (7.15)$$

Where ρ_0 is the density of the gas at reference conditions: $T_0 = 293.15$ K, $P_0 = 10^5$ Pa and 65 % relative humidity

In this case, the *vena contracta* mean gas velocity cannot be estimated since the *vena contracta* density is unknown (no information is available on the evolution between upstream and *vena contracta* conditions, the equivalent flow area and the flow coefficient being unknown).

7.3 Modeling pneumatic chambers

The pneumatic chamber is the second main submodel used in the simulation of pneumatic systems. In this lumped parameter element, we calculate the temperature and the pressure which are average values in the control volume.

The temperature and pressure are state variables since they are computed from their time derivatives.

7.3.1 Mass balance

The mass variation can also be written as below:

$$\frac{dm}{dt} = \frac{d}{dt}(\rho \cdot V) = \rho \cdot \frac{dV}{dt} + V \cdot \frac{d\rho}{dt} = \rho \cdot \frac{dV}{dt} + V \cdot \left[\left(\frac{\partial \rho}{\partial P} \right)_T \cdot \frac{dP}{dt} + \left(\frac{\partial \rho}{\partial T} \right)_P \cdot \frac{dT}{dt} \right] \quad (7.16)$$

where:

- $\frac{dm}{dt}$ is the mass variation in the volume [kg/s],
- V is the volume [m³],
- ρ is the gas density [kg / m³].

7.3.2 Energy balance

The energy in the control volume is given by:

$$E = m \cdot u + \frac{mV^2}{2} + mgz \quad (7.17)$$

These three terms represent the internal energy, the kinetic energy and the potential energy respectively.

We simplify the previous equation introducing the assumption that **the kinetic and potential energies in the control volume are ignored** (see [appendix E](#)). This leads to:

$$E = mu = U \quad (7.18)$$

where u is the internal energy by mass unit, and U is the total internal energy.

We can express the variation of internal energy U using the first law of thermodynamics applied to an open system:

$$dU = \sum dm_i \cdot h_i + \delta Q + dW_{\text{ext}} \quad (7.19)$$

where $dm_i \cdot h_i$ is the incoming total enthalpy flow at port i during time dt , obtained by the product of the incoming mass flow during time dt and the specific enthalpy at port i , δQ is the heat exchanged through the control volume during dt and dW_{ext} is the work variation of the system.

The external work supplied by the system is due to the mechanical displacement of a piston connected to the volume, and therefore:

$$\frac{dW_{\text{ext}}}{dt} = -p \frac{dV}{dt} \quad (7.20)$$

where $\frac{dV}{dt}$ is the volume variation due to a piston displacement.

From the relation of internal energy U :

$$U = m \cdot u \quad (7.21)$$

The energy variation can also be written as follows:

$$dU = d(m \cdot u) = dm \cdot u + m \cdot du = \sum_i dm_i \cdot u + m \cdot du \quad (7.22)$$

Introducing the enthalpy $h = u + P \cdot v$ and using Eq.7.19 and Eq.7.20 leads to:

$$m \cdot dh - V \cdot dP = \sum_i dm_i \cdot h_i - \sum_i dm_i \cdot h + \delta Q \quad (7.23)$$

Considering variation over time, this leads to:

$$\left(m \cdot \left(\frac{\partial h}{\partial P} \right)_T - V \right) \cdot \frac{dP}{dt} + m \cdot \left(\frac{\partial h}{\partial T} \right)_P \cdot \frac{dT}{dt} = \sum_i \frac{dm_i}{dt} \cdot h_i - \sum_i \frac{dm_i}{dt} \cdot h + \delta Q \quad (7.24)$$

Temperature and pressure time derivatives are computed solving the following system composed by Eq.7.16 and Eq.7.24:

$$\begin{bmatrix} V \cdot \left(\frac{\partial \rho}{\partial P} \right)_T & V \cdot \left(\frac{\partial \rho}{\partial T} \right)_P \\ m \cdot \left(\frac{\partial h}{\partial P} \right)_T - V & m \cdot \left(\frac{\partial h}{\partial T} \right)_P \end{bmatrix} \cdot \begin{bmatrix} \frac{dP}{dt} \\ \frac{dT}{dt} \end{bmatrix} = \begin{bmatrix} \sum_i \frac{dm_i}{dt} - \rho \cdot \frac{dV}{dt} \\ \sum_i \frac{dm_i}{dt} \cdot h_i - \sum_i \frac{dm_i}{dt} \cdot h + \delta Q \end{bmatrix} \quad (7.25)$$

Note:

All the equations above are available for any equation of state.

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

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

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

Momentum conservation

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

where:

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

Energy conservation

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

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:

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

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:

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

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:

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

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:

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

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:

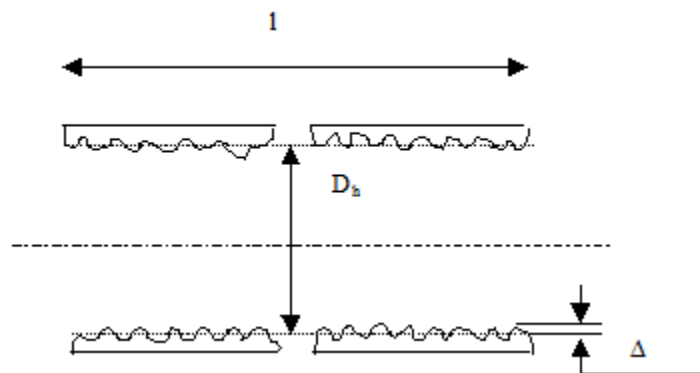


Fig. 8.1: relative roughness definition

The relative roughness of a pipe is given by:

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

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

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.

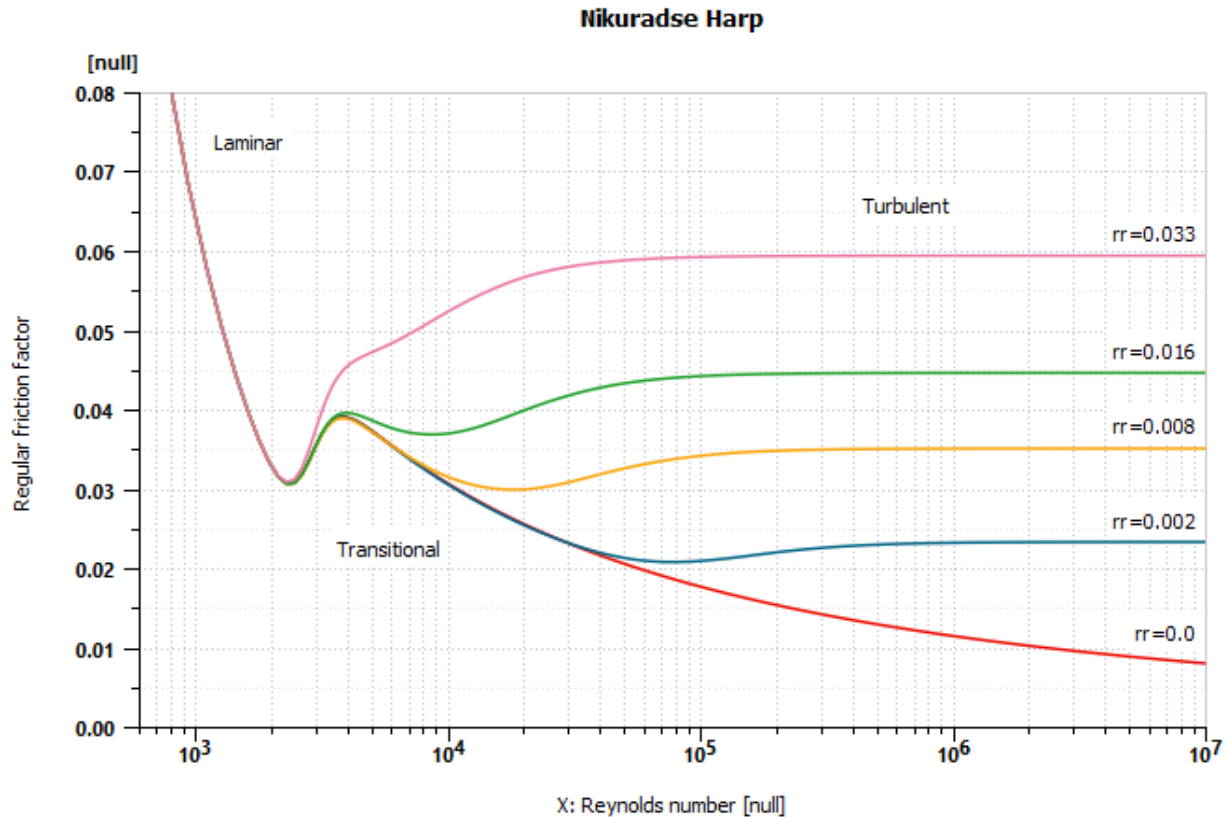


Fig. 8.2: evolution of the frictional drag factor with the Reynolds number and the relative roughness

8.1.2 Compressive effect C

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 I

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

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

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

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.

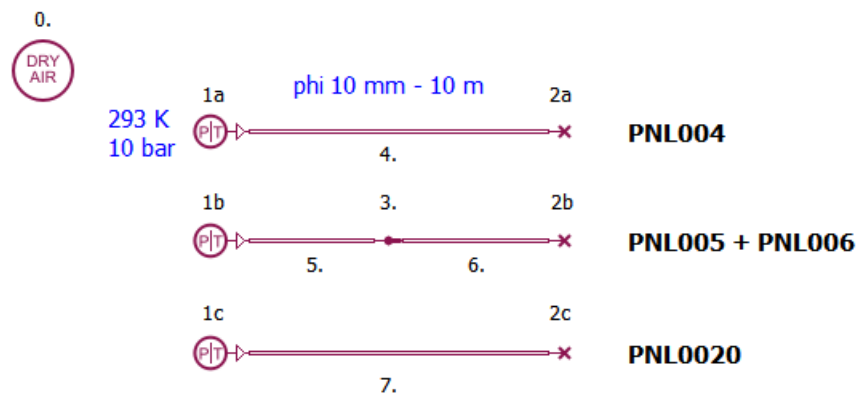


Fig. 8.3: Model for pneumatic pipe simulation

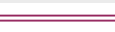
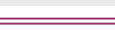
This model comprises components from the **Pneumatic** library. Fill in the parameters of this model as shown in the following table, leaving the other parameters at their default values and run a simulation for 0.5 seconds with a **Print interval** of 0.005 seconds .

Table 8.2: parameters for Fig. 8.3

Icon	Submodel	Main parameters
0 	PN_AIR gas properties	Default parameters
1 	PNCS001 pressure and temperature source	pressure = 10 barA

continues on next page

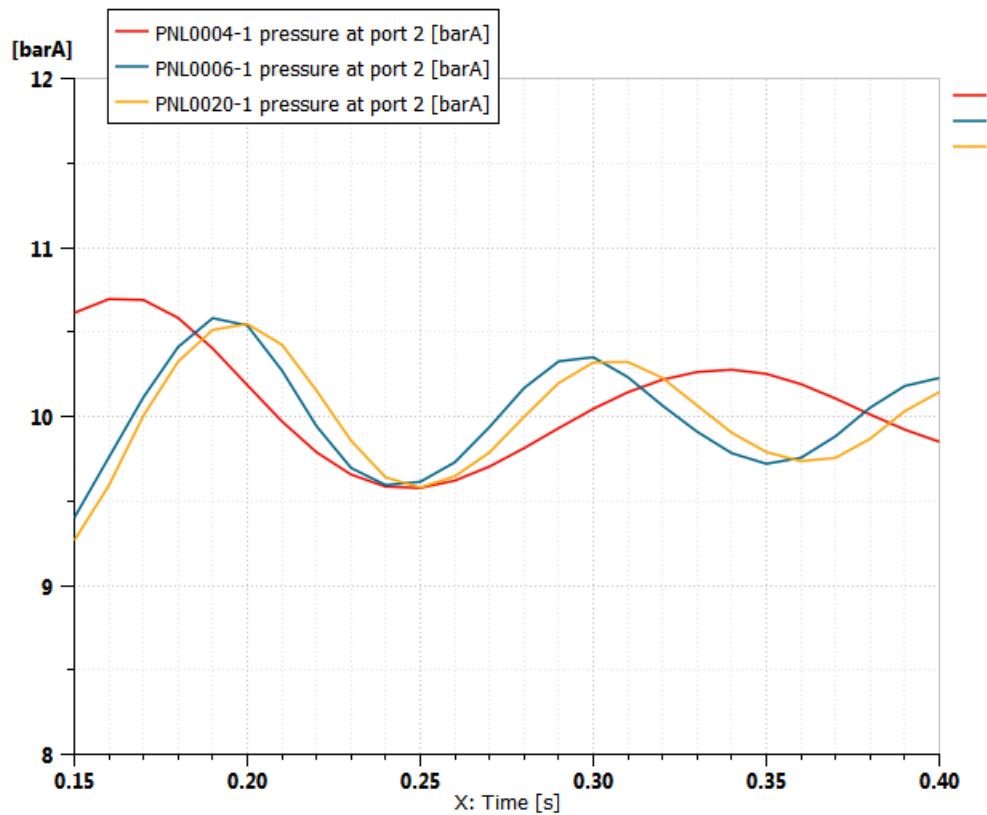
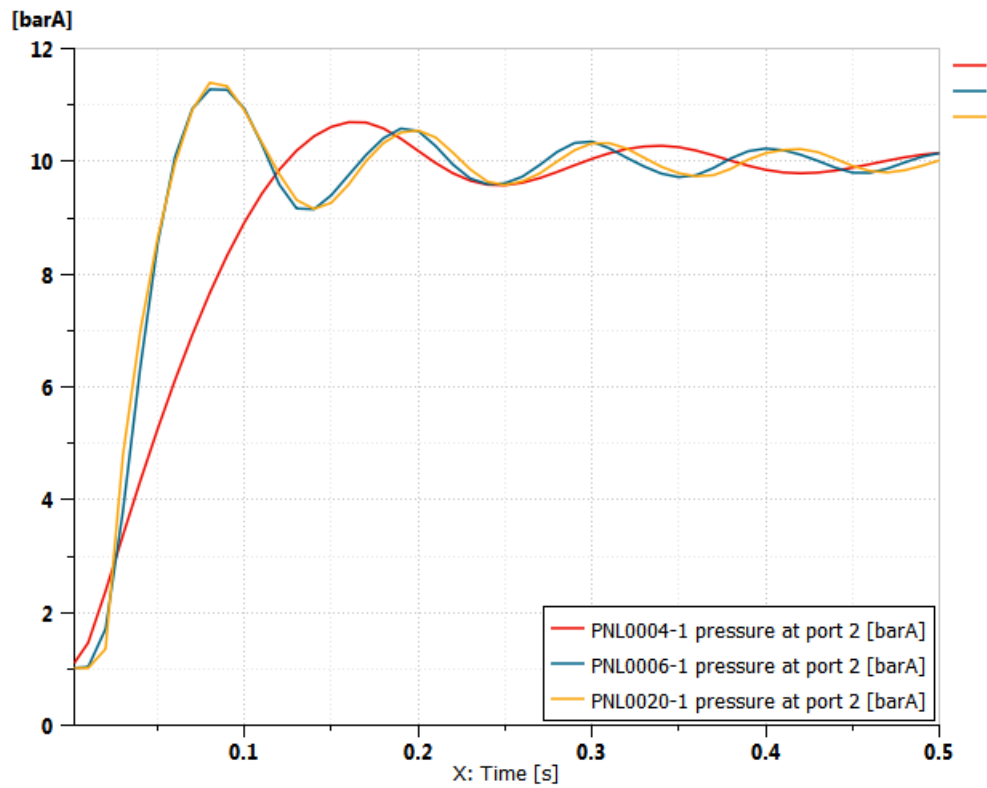
Table 8.2 – continued from previous page

Icon	Submodel	Main parameters
2 	PNPL01 zero pneumatic flow source	No parameter
3 	PNND005 2-ports node	No parameter
4 	PNL0004 line	diameter = 0 mm length = 10 m thermal exchange coefficient = 0 J/m**2/K/s
5 	PNL0005 line	diameter = 10 mm length = 5 m thermal exchange coefficient = 0 J/m**2/K/s
6 	PNL0006 line	diameter = 10 mm length = 5 m thermal exchange coefficient = 0 J/m**2/K/s
7 	PNL0020 line	diameter = 0 mm length = 10 m 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.



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

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 1 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

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

- if the line is closed at both ends

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

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

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

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.

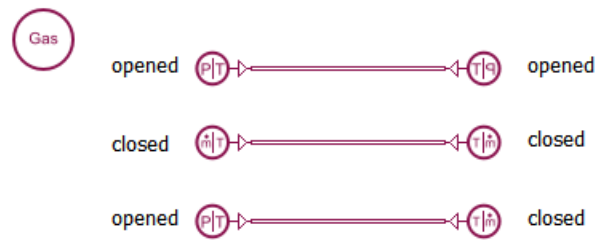


Fig. 8.4: Pure boundary conditions

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 10.7	0 16.8	0 57
PNL0006	closed / closed	10.7	16.8	57
PNL0020	opened / closed	8.4 24.5 38.6	8.4 25.3 42.1	0 3 9
PNL0021	opened / opened	0 16.7 32.1 45.5	0 16.8 33.7 50.5	0 1 5 11
PNL0022	closed / closed	16.7 32.1 45.5	16.8 33.7 50.5	1 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

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](#).

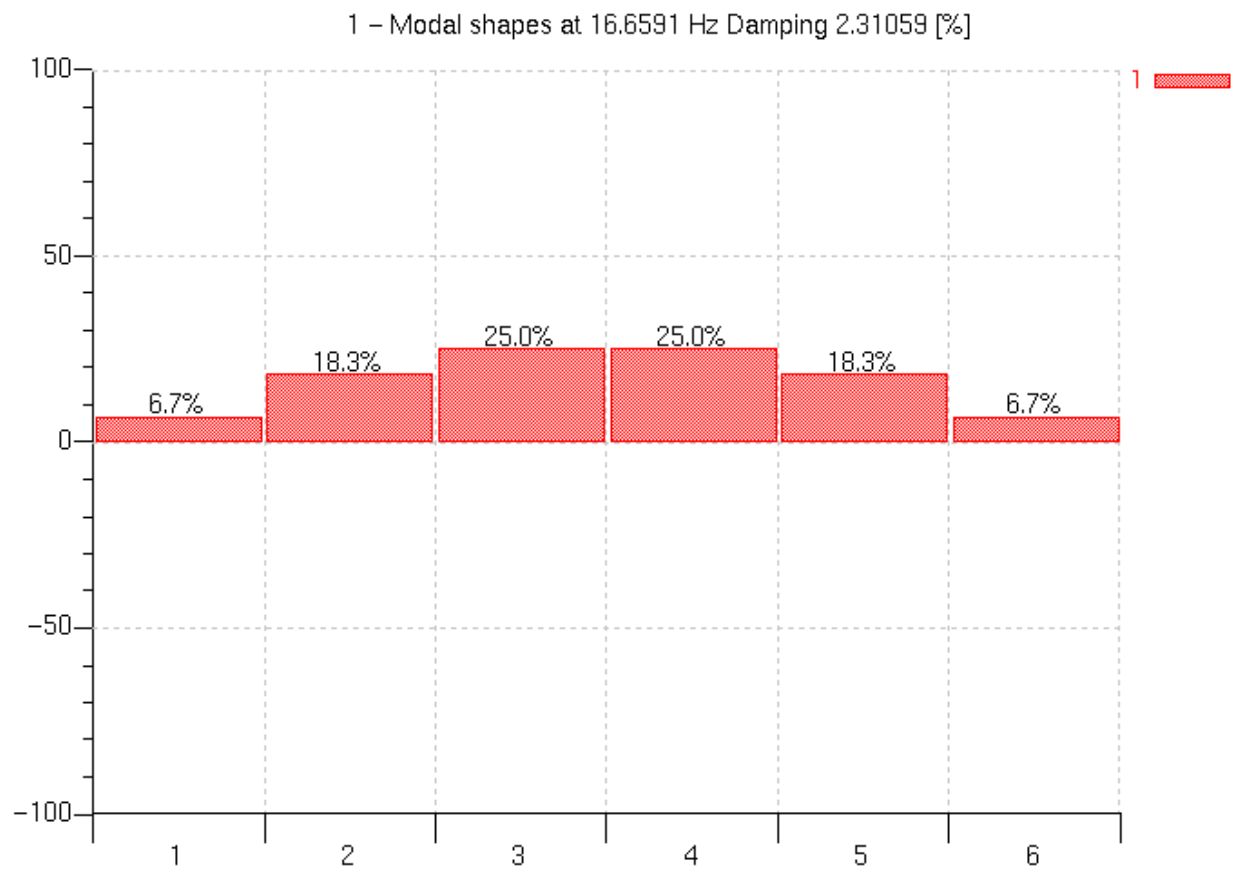


Fig. 8.5: PNL0022, closed / closed, first mode

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.

9. Brief review of the theory: thermal calculations

This section gives an overview of the thermal calculation theory.

9.1 Thermal assumptions

Here we present the basic equations used in the **Pneumatic** library (in global pipe submodels for example). A full description of the thermal phenomena is given in the **Thermal** library user's guide. A summary of this guide is given in what follows.

Remember that there are two major types of thermal component:

- thermal capacity submodel with state variables,
- heat transfer submodels which are instantaneous.

The thermal capacity submodel is based on the following statement: **the energy stored in a thermal capacity induces a variation in the temperature of this capacity**. A thermal capacity can be considered as a mass of material in which the temperature is computed.

Heat transfer submodels are based on the following statement: **heat transfer is energy transfer due to a temperature difference**. In these submodels, the heat flow rate is computed.

Three types of heat transfer processes can be found:

1. **Heat transfer by conduction:** this is a heat transmission induced by molecular vibrations. It is the result of a temperature gradient. This phenomenon is shown in Fig.9.1:

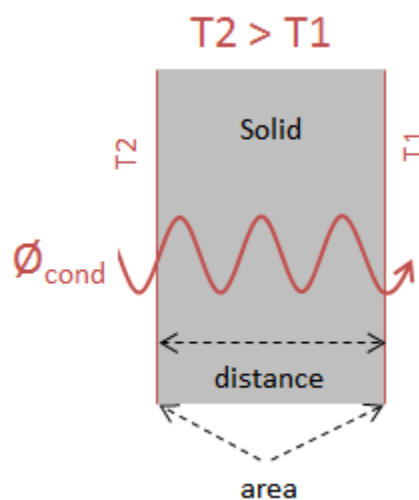


Fig. 9.1: Conduction phenomenon in a solid

Naturally if $T1$ is greater than $T2$ the heat flow rate will be in the opposite direction.

2. **Heat transfer by convection:** the key feature of this kind of heat exchange is the movement of a fluid. This phenomenon can be seen in Fig.9.2:

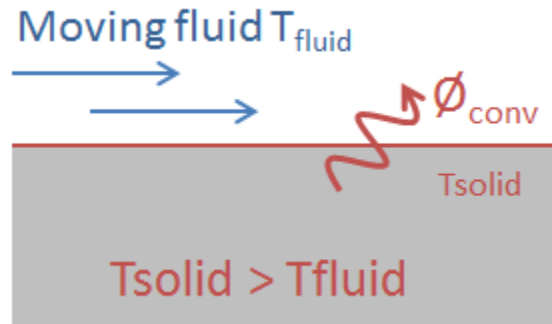


Fig. 9.2: Convection phenomenon between a solid and a fluid

3. **Heat transfer by radiation:** this is the most complex mode of heat transfer. Every liquid, solid or gaseous body emits electro-magnetic radiation. This energy emission results in a loss of the internal energy of the body. Furthermore, when studying the thermal equilibrium of a system, it is necessary to consider bodies as emitters (because a body is always emitting) as well as receivers (because a body receives radiation from surrounding bodies). See Fig.9.3:

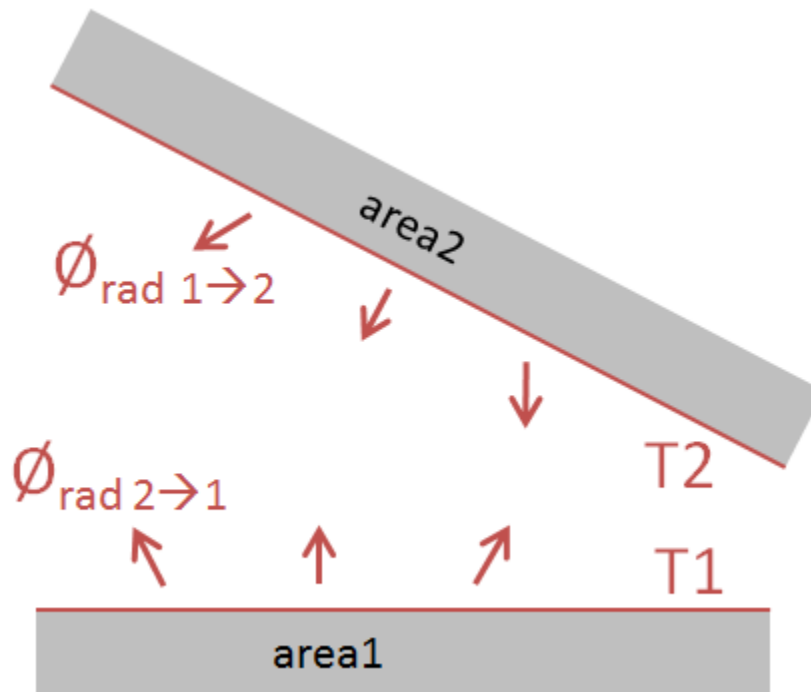


Fig. 9.3: Radiation phenomenon between two bodies

In each thermal component, the following assumptions are made:

- The heat flow is one-dimensional,

- The solids used are considered as isotropic; this means that their thermal properties are the same in all directions,
- The temperatures in the thermal capacities are homogeneous.

9.2 Thermal capacity equations

The thermal capacity can be described as a temperature node. The temperature is computed from its derivative with respect to time and is considered to be homogeneous in the mass of the solid involved. The temperature derivative is calculated as follows:

$$\frac{dT}{dt} = \frac{\Phi_{in} - \Phi_{out}}{m \cdot C_p} \quad (9.1)$$

where Φ_{in} is the incoming heat flow rate [W], Φ_{out} is the outgoing heat flow rate [W], m is the mass of material [kg] and C_p is the specific heat of the material [J/kg/K].

9.3 Convection heat flow rate

Convection occurs as soon as a fluid is moving along the wall of a solid. The convection phenomenon is the combination of two heat transfer modes:

- Conduction in the thickness of the laminar film of fluid in contact with the wall. In this film, it is considered that there is no mixture of matter.
- Outside this laminar layer, heat is transferred due to a mixture of the fluid particles in a flow assumed to be turbulent. We then talk about a mixing temperature T_m .

The general form of the equation expressing the convection heat flow rate is given by:

$$\Phi_{convection} = h \cdot A \cdot (T_m - T_w) \quad (9.2)$$

where h is the convective exchange coefficient, A is the exchange area, T_m is the mixing temperature of the fluid and T_w is the wall temperature. There are two kinds of convective exchange and a brief review of the assumptions and equations governing these phenomena is given below.

9.3.1 Forced convection

We talk about forced convection as soon as a fluid moves under the action of a mechanical external source and flows along a solid wall. The forced convection heat flow rate is computed using the following equation:

$$\Phi_{forced} = h_{forced} \cdot A \cdot (T_m - T_w) \quad (9.3)$$

where h_{forced} is the forced convective exchange coefficient, A is the exchange area, T_m and T_w are respectively the mixing temperature of the fluid and the temperature of the solid wall. The major problem is to determine h_{forced} , it is given below:

$$h_{forced} = \frac{Nu(Re, Pr) \cdot \lambda}{C_{dim}} \quad (9.4)$$

where Nu is a dimensionless number called the Nusselt number, λ is the thermal conductivity of the fluid and C_{dim} is a characteristic geometrical dimension of the exchange. The Nusselt

number can be interpreted as a dimensionless temperature gradient at the surface. It characterizes the thermal exchange between the fluid and the solid wall. It is usually taken to be a function of two dimensionless numbers which are the Prandtl number and the Reynolds number.

The Prandtl number is defined by the following relationship:

$$Pr = \frac{\mu \cdot C_p}{\lambda} \quad (9.5)$$

where μ is the absolute viscosity of the fluid, C_p is the specific heat of the fluid and λ is the thermal conductivity of the fluid. These thermal properties depend on pressure and temperature. The Prandtl number can be interpreted as the ratio of the momentum and thermal diffusivities. It characterizes the thermal properties of the fluid.

The Reynolds number is defined as follows:

$$Re = \frac{V \cdot D}{\nu} \quad (9.6)$$

where V is the velocity of the fluid, D is the hydraulic diameter of the flow and ν is the kinematic viscosity of the fluid. The Reynolds number can be interpreted as the ratio of the inertia and viscous forces. It characterizes the flow regime of the fluid.

There are many relationships given for the Nusselt number as a function of the Prandtl and the Reynolds number. These are based on experiment and are available either for external flow or internal flow conditions. Examples of functions commonly used are summarized in the [appendix C](#) of this manual.

9.3.2 Free convection

Free or natural convection occurs when the movement of the fluid is due to variations in its density induced by heat exchanges between the fluid and the solid wall. Basically, the free convection heat flow rate is computed using the following equation:

$$\Phi_{free} = h_{free} \cdot A \cdot (T_m - T_w) \quad (9.7)$$

where h_{free} is the free convective exchange coefficient, A is the exchange area, T_m and T_w are respectively the mixing temperature of the fluid and the temperature of the solid wall. The major problem is to determine h_{free} , it is given below:

$$h_{free} = \frac{Nu(Gr, Pr) \cdot \lambda}{C_{dim}} \quad (9.8)$$

where Nu is a dimensionless number called the Nusselt number, λ is the thermal conductivity of the fluid and C_{dim} is a characteristic geometrical dimension of the exchange. The Nusselt number can be interpreted as a dimensionless temperature gradient at the surface. It characterizes the thermal exchange between the fluid and the solid wall. It is usually taken to be a function of two dimensionless numbers which are the Grashof number and the Prandtl number.

The Grashof number is defined by the following relationship:

$$Gr = \frac{C_{dim}^3 \cdot \rho^2 \cdot g \cdot \alpha \cdot (T_f - T_w)}{\mu^2} \quad (9.9)$$

where μ is the absolute viscosity of the fluid, C_{dim} is a characteristic geometrical dimension of the exchange, ρ is the density of the fluid, g is the gravity acceleration, α is the volumetric

expansion coefficient of the fluid, T_f is the fluid temperature and T_w is the solid wall temperature.

In the case of a perfect or semi-perfect gas, the volumetric expansion coefficient α is computed as follows:

$$\alpha = \frac{1}{T_{\text{mean}}} = \frac{1}{\frac{T_f + T_w}{2}} \quad (9.10)$$

The Prandtl number is defined in Eq.9.5.

There are many relationships given for the Nusselt number as a function of the Prandtl and the Grashof number. These are based on experiment and are available either for external flow or internal flow conditions.

Examples of functions commonly used are summarized in [appendix C](#) of this manual.

9.3.3 Generic approach to model convective heat exchange in the thermal libraries

In **Simcenter Amesim** convective heat exchange by convection encompasses both forced and free convection as explained in the paragraphs above. There are several different submodels which implement such convective heat exchanges, they are either internal or external. They are very similar in the parameterizing as well as the underlying equations and physical assumptions. These submodels come with the following features:

- **Choice** between two geometries (plate or cylinder) with **predefined heat exchange correlations** (see [Fig.9.4](#)) or generic geometry with heat exchange characteristics that can be entirely defined as the user wishes.
- **Dynamic icon** to enhance the understanding of the **Simcenter Amesim** model.
- Unified way of parameterizing, fully similar to submodels dealing with convective heat exchange in the **Thermal-Hydraulic** library and the **Pneumatic** library (see [Fig.9.5](#)).
- **Extended means**, possibly driven using an external signal, to define the convective heat exchange coefficient.

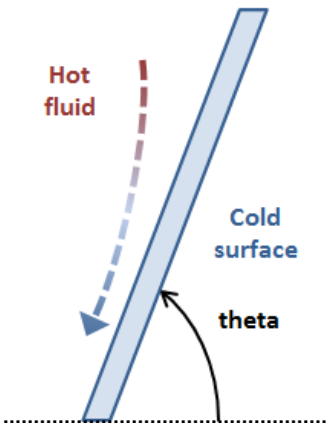
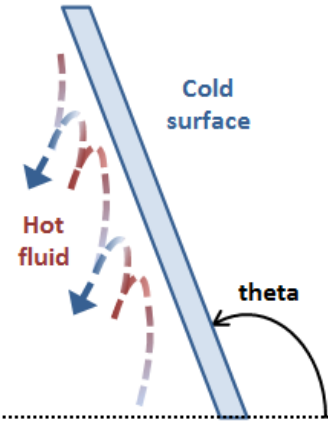
Inclination angle θ	Cold plate with hot fluid	
$\theta = 0$ degree		$Nu = 0.27 \cdot Ra^{1/4}$
$0 < \theta < 30$ degree		Sinusoidal trend using boundary $\theta = 0$ and $\theta = 30$ degree
$30 \leq \theta < 90$ degree		Same as $\theta = 90$ degree with $Ra = Ra \cdot \sin(\theta)$
$\theta = 90$ degree		$Nu = \left(0.825 + \frac{0.387 \cdot Ra^{1/6}}{\left(1 + \left(\frac{0.492}{Pr} \right)^{9/16} \right)^{8/27}} \right)^2$
$90 < \theta < 180$ degree		Sinusoidal trend between $\theta = 90$ and $\theta = 180$ degree
$\theta = 180$ degree		$\begin{cases} Nu = 0.54 \cdot Ra^{1/4} & \text{if } Ra < 4.74 \cdot 10^6 \\ Nu = 0.15 \cdot Ra^{1/3} & \text{otherwise} \end{cases}$

Fig. 9.4: example of implemented predefined correlations in convective heat exchanges within the Pneumatic library

 Thermal Sources, Sensors Material, Nodes, Masses, Ambient conditions Heat Exchanges Environmental Moist Air	 Thermal Hydraulic Fluids, Sources, Sensors Pressure Losses, Volumes, Nodes Heat Exchanges CFD-1D Lines, Discrete Partitioning Pumps, Motors Directional Control Valves Pressure Control Valves Flow Control Valves Linear Actuators	 Pneumatic Sources, Sensors Fluids, Nodes, Volumes Frictions Frictions with Thermal Exchanges Control Valves Control Valves Extended Linear Actuators Compressors and Motors Heat Exchanges
 THGCV0 th_convection  THGCV1 th_piloted_conv...  THGCV2 th_convection2p  THGCV3 th_pil_conv2p  THGCV4 th_internal_conv...	 TFGCV2 thf_convection...  TFGCV3 thf_piloted_co...  TFCVINT10 (C), 11 (CR), 12 (RC), 13 (CRC) thf_convection  TFPHE00 (C), 01 (CR), 02 (RC), 03 (CRC), 04 (RCR) thf_pipe_heat  TFCVINT01 (CR), 02 (RC), 03 (CRC) thfconvection2...	 TPGCV2 tp_convection...  TPGCV3 tp_piloted_conv...  TPCVINT10 (C), 11 (CR), 12 (RC), 13 (CRC) tp_cvnnode  TPPHE00 (C), 01 (CR), 02 (RC), 03 (CRC), 04 (RCR) tp_pipe_heat  TPCVINT01 (CR), 02 (RC), 03 (CRC) tpconvection2p...

Fig. 9.5: submodels dealing with generic convection in the Thermal library, the Thermal-Hydraulic library and the Pneumatic library

9.4 Gas/wall radiation exchange

Radiation is a complex method of exchanging heat. It depends on a large set of parameters. In the **Pneumatic** library, radiation is considered each time the gas is in contact with a wall material.

The radiation exchange equation used to model radiation between a wall and a gas is the following:

$$\Phi_{radiation} = \sigma \cdot \varepsilon_{gw} \cdot A \cdot (T_g^4 - T_w^4) \quad (9.11)$$

where σ is the Stefan-Boltzmann constant = $5,67 \cdot 10^{-8} \text{ W} / \text{m}^2 / \text{K}^4$, ε_{gw} is the equivalent emission factor of the combination gas/wall, A is the emitting area considered, T_g is the temperature of the gas and T_w is the temperature of the wall.

9.5 Radiation exchange between two pipe surfaces forming an enclosure

This characterization is used in global insulated pipe submodels to compute the radiation heat flow rate inside the air gap.

It is used to model two concentrically-located pipe surfaces that exchange radiation only with each other. In this case, the net rate of radiation heat transfer emitted by surface i must equal the net rate of radiation heat transfer received by surface j . The heat flow rate propagating by radiation in this case is computed as follows:

$$\Phi_{\text{radiation}} = \frac{1}{\frac{1-\varepsilon_i}{\varepsilon_i \cdot A_i} + \frac{1}{A_i} + \frac{1-\varepsilon_j}{\varepsilon_j \cdot A_j}} \cdot \sigma \cdot (T_i^4 - T_j^4) \quad (9.12)$$

where ε_i is the emission factor for body i , A_i is the emitting area of body i , and σ is the Stefan-Boltzmann constant.

9.6 Linear conduction

As previously-stated, heat flow rates propagate whenever a temperature difference occurs. Consequently, the conduction heat flow rate is computed from the temperature difference between two thermal nodes.

In this case of linear conduction in the pipe wall, the heat flow rate is evaluated as follows:

$$\Phi = \frac{\lambda \cdot A}{d} \cdot (T_1 - T_2) \quad (9.13)$$

where λ is the thermal conductivity of the material in which the heat flow rate is propagating, A is the exchange area, and d is the distance between the two temperature points under consideration.

The heat flow depends on one of the thermal properties of the material which is the thermal conductivity. This data varies with the mean temperature T_{mean} as follows:

$$\lambda \equiv \lambda(T_{\text{mean}}) = \lambda \left(\frac{T_1 + T_2}{2} \right) \quad (9.14)$$

10. Heat exchanger models based on effectiveness-NTU method

10.1 Effectiveness-NTU method

The effectiveness-NTU (Number of Transfer Units) method is widely considered for heat exchanger analysis. It is used when only the inlet temperatures are known.

The heat exchanger is modeled considering three elements:

- the separate cold and hot fluids that exchange with one another, represented with two similar parts,
- the link between both these parts with another intermediate element.

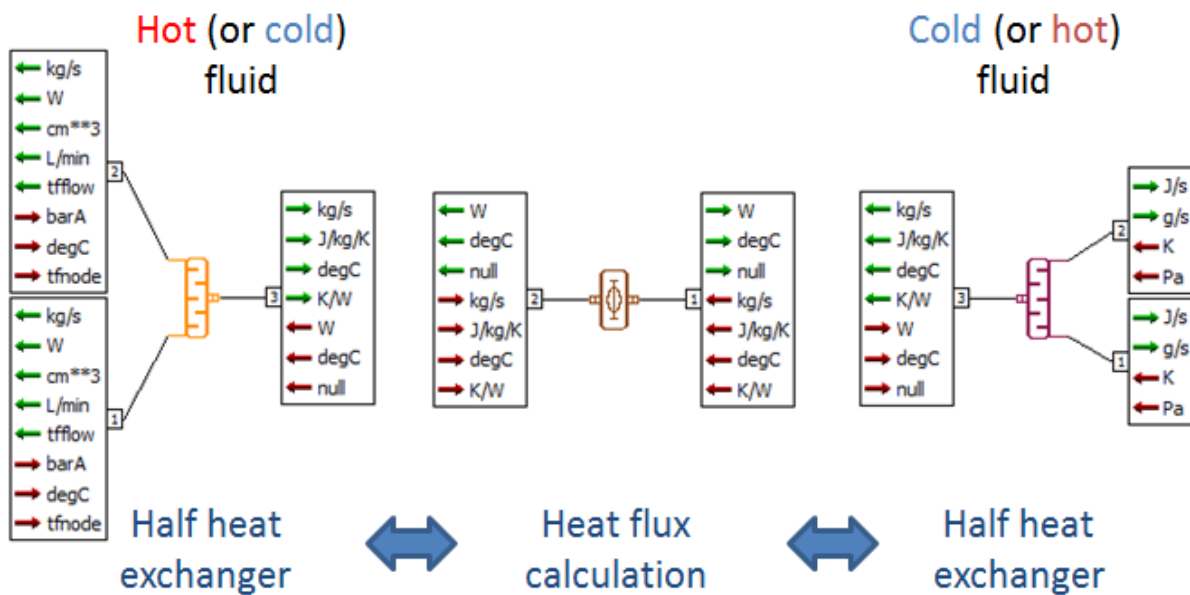


Fig. 10.1: Modeling of heat exchangers using the effectiveness-NTU method

In the effectiveness-NTU method, the actual heat flow rate is expressed as:

$$q = \varepsilon \cdot q_{max} \quad (10.1)$$

where q is the actual heat transfer rate, ε is the effectiveness and q_{max} is the maximum heat transfer rate.

The maximum heat flow rate considers temperature at inlet of both sides of the heat exchanger.

$$q_{max} = \min(dm_{cold} \cdot (h_{cold}(T_{hot,inlet}) - h_{cold}(T_{cold,inlet})), dm_{hot} \cdot (h_{hot}(T_{cold,inlet}) - h_{hot}(T_{hot,inlet}))) \quad (10.2)$$

The equation above is often written for perfect gas as:

$$q_{\max} = \min(dm_{\text{cold}} \cdot C_{p,\text{cold}}, dm_{\text{hot}} \cdot C_{p,\text{hot}}) \quad (10.3)$$

and are the heat capacities at both sides of the heat exchanger. In **Simcenter Amesim** this equation is also used considering the following relationships that are true whatever the level of modeling of the gas.

$$C_{p,\text{cold}} = \frac{\Delta h_{\text{cold}}}{\Delta T}, C_{p,\text{hot}} = \frac{\Delta h_{\text{hot}}}{\Delta T} \quad (10.4)$$

There are many relationships given for the heat exchanger effectiveness ε as a function of the NTU and $Cr=C_{\min}/C_{\max}$ numbers. These are based on analytical calculations available for various types of heat exchangers.

Examples of functions commonly-used are summarized in the [appendix D](#) of this manual.

The number of transfer units NTU is a dimensionless parameter and is defined as:

$$NTU = \frac{U \cdot A}{C_{\min}} \quad (10.5)$$

where U is the overall heat transfer coefficient, A is the total area and $C_{\min}$ is the minimum heat capacity rate.

The overall heat transfer $U \cdot A$ is computed as:

$$U \cdot A = \frac{1}{R_{th1} + R_{wall} + R_{th2}} \quad (10.6)$$

where R_{th1} and R_{th2} both correspond to the thermal resistance of each fluid, and R_{wall} to the wall thermal resistance.

The thermal resistance R_{th} is finally computed as:

$$R_{th} = \frac{1}{h_{\text{conv}} \cdot S_{hx}} \quad (10.7)$$

where h_{conv} and S_{hx} correspond respectively to the convective heat exchange coefficient and the exchange surface.

If the heat exchanger contains fins, the thermal resistance R_{th} is computed as:

$$R_{th} = \frac{1}{h_{\text{conv}} \cdot S_{dhx} \cdot (1 + \eta_{\text{fins}})} \quad (10.8)$$

where h_{conv} , S_{dhx} , η_{fins} correspond respectively to the convective heat exchange coefficient, the exchange surface and the finned surface efficiency.

In **Simcenter Amesim** there are three levels of modeling:

- **simple:** simple mathematical relationships are used to define the overall heat transfer $U \cdot A$. This is useful early in the project where the geometry of the heat exchanger and expressions of heat exchange coefficients are not known.
- **normal:** a fairly complete definition of the geometry as well as the form of the heat exchange (through Nusselt correlations) are used to fill in the equations above.
- **regression:** experimental results are required. Dedicated regression tools can then be used to automatically set adequate relationships characterizing heat exchanges that match the given experimental results.

Submodel summary		
	Alias: halfheaterthpn2	
	Component: halfheaterthpn2	
	Submodel name: TPEXSIMP01	
Compatible submodel list		
Name	Description	Submodel type
TPEXSIMP01	pneumatic half heat exchanger (simple thermal approach) (R-C-R)	generic submodel
TPEX0001	pneumatic half heat exchanger (heat exchange correlations and geometry) (R-C-R)	generic submodel
TPEXREG01	pneumatic half heat exchanger for the NTU regression tool (R-C-R)	generic submodel

Fig. 10.2: Three levels of modeling for effectiveness-NTU method

10.2 Gas-gas heat exchangers

The purpose of this section is to describe the method for modeling gas-gas heat exchangers considering various flow arrangements with:

1. the effectiveness fixed as a user-defined expression,
2. some cross-flow considerations,
3. some parallel-flow considerations,
4. some counter-flow considerations.

For the last three cases, the flow arrangement expressions are set in the submodels *TPPHI03* and are used to compute the heat flow rate through the heat exchangers as well as their effectiveness.

The interesting point here is to compare the difference in the heat exchanger performances due to the flow arrangements.

To model these four heat exchangers, build the circuit shown in [Fig.10.3](#). It consists of a *10 g/s and 573.15 K* hot exhaust gas flow rate which is refreshed with a *10 g/s and 308.15 K* fresh air flow rate:

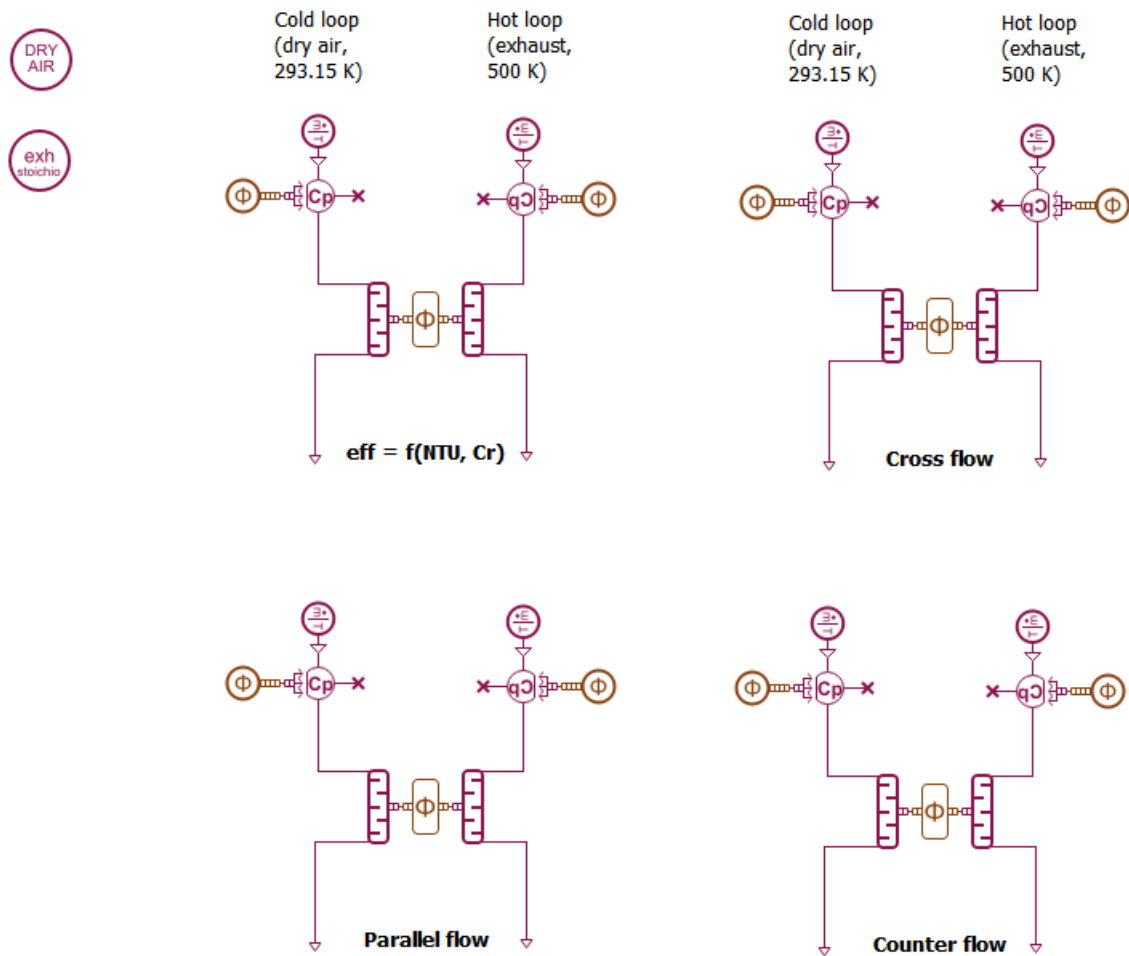


Fig. 10.3: Heat exchangers based on the NTU-effectiveness method

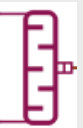
These models are comprised of components from the **Pneumatic** library. Fill in the parameters of these components as shown in the following table leaving the others at their default values. Finally, run a simulation with a final time of *10 seconds* with a **Print interval** of *0.1 seconds*.

Table 10.1: parameters for heat exchangers models (Fig. 10.3)

Icon	Submodel	Main parameters
	PNGD_AIR pneumatic properties	default parameters
	PNGD_EXHAUST_STOICHIO pneumatic properties	gas type index = 2

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Table 10.1 – continued from previous page

Icon	Submodel	Main parameters
	PNCS002 mass and enthalpy flow rates source	output mass flow = 10 g/s
	PNCS002 mass and enthalpy flow rates source	output mass flow = 10 g/s gas type index = 2 source temperature = 500 K
	TPCH013 pneumatic chamber	default parameter
	TPCH013 pneumatic chamber	gas type index = 2 source temperature = 500 K
	TPEX0001 pneumatic half heat exchanger	convective exchange area = 50000 mm**2 expression for turbulent Nusselt number = $0.027 \cdot re^{0.8} \cdot pr^{1/3}$
	TPEX0001 pneumatic half heat exchanger	heat exchanger outlet temperature = 500 K gas type index = 2 convective exchange area = 50000 mm**2 expression for turbulent Nusselt number = $0.027 \cdot re^{0.8} \cdot pr^{1/3}$
	THPHINTU01 pneumatic heat flow rate calculation	relation = <i>effectiveness expression or datafile</i> $eff=f(f(Ntu, cr)$ and effectiveness expression or datafile $eff=f(Ntu, cr) = 0.5$ or relation = <i>cross flow thermal effectiveness</i> or relation = <i>parallel flow thermal effectiveness</i> or relation = <i>counter flow thermal effectiveness</i>

When this is done, plot a graph of the outlet temperatures in component *TPEX0001* against time as shown in [Fig.10.4](#), [Fig.10.5](#) and [Fig.10.6](#).

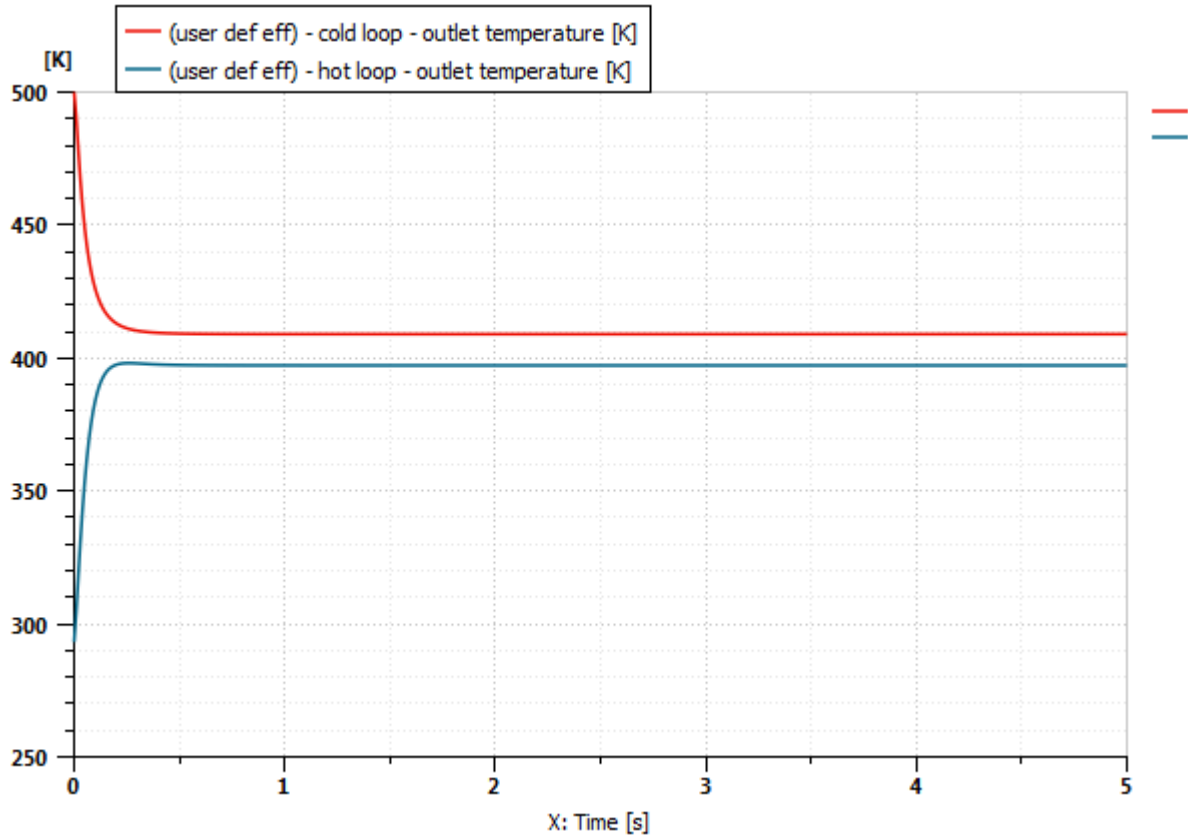


Fig. 10.4: outlet gas temperatures for the hot and cold loops, constant effectiveness

For the model with effectiveness directly fixed on the signal port, the hot exhaust gas is cooled from 500 K to ~397 K while the fresh air is warmed from 293 K to ~406 K.

Here it is interesting to observe the outlet temperatures of the hot and cold gases with the three proposed flow arrangements of cross, parallel and counter flows.

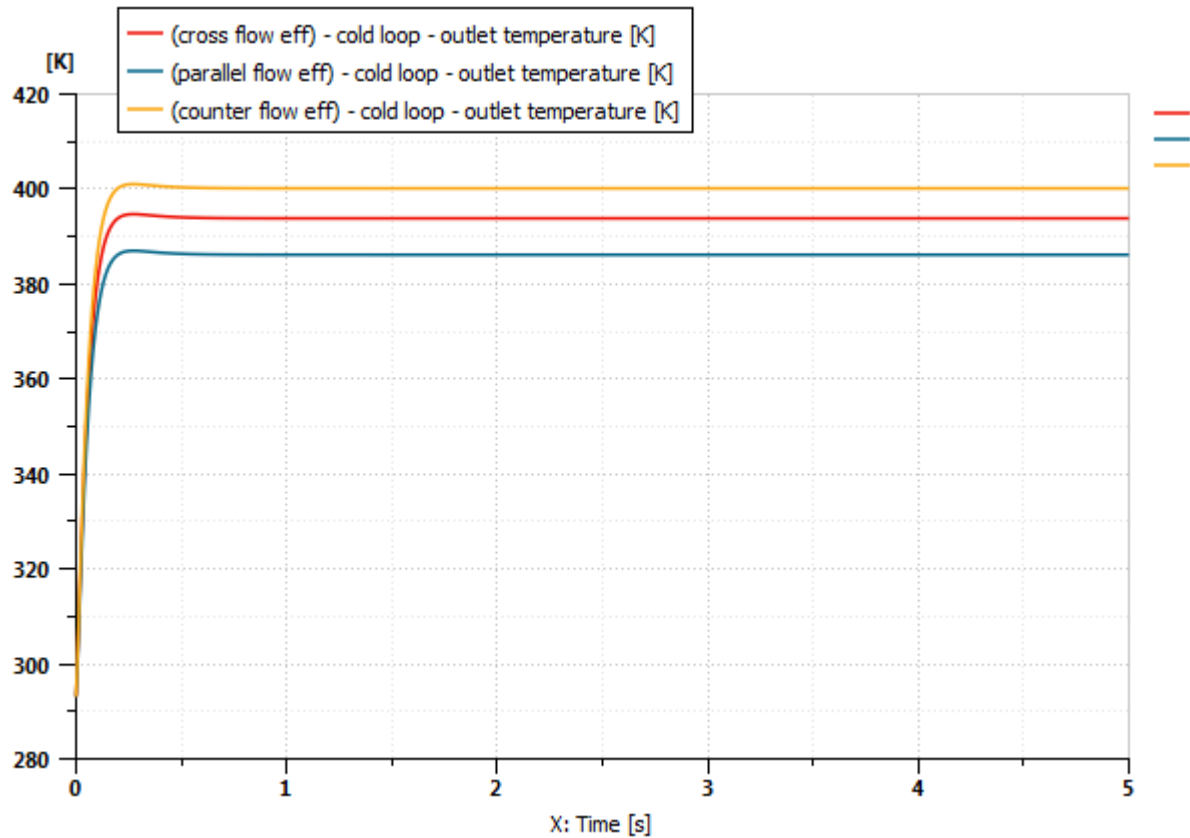


Fig. 10.5: Outlet gas temperatures (cold loop), cross, parallel and counter flows

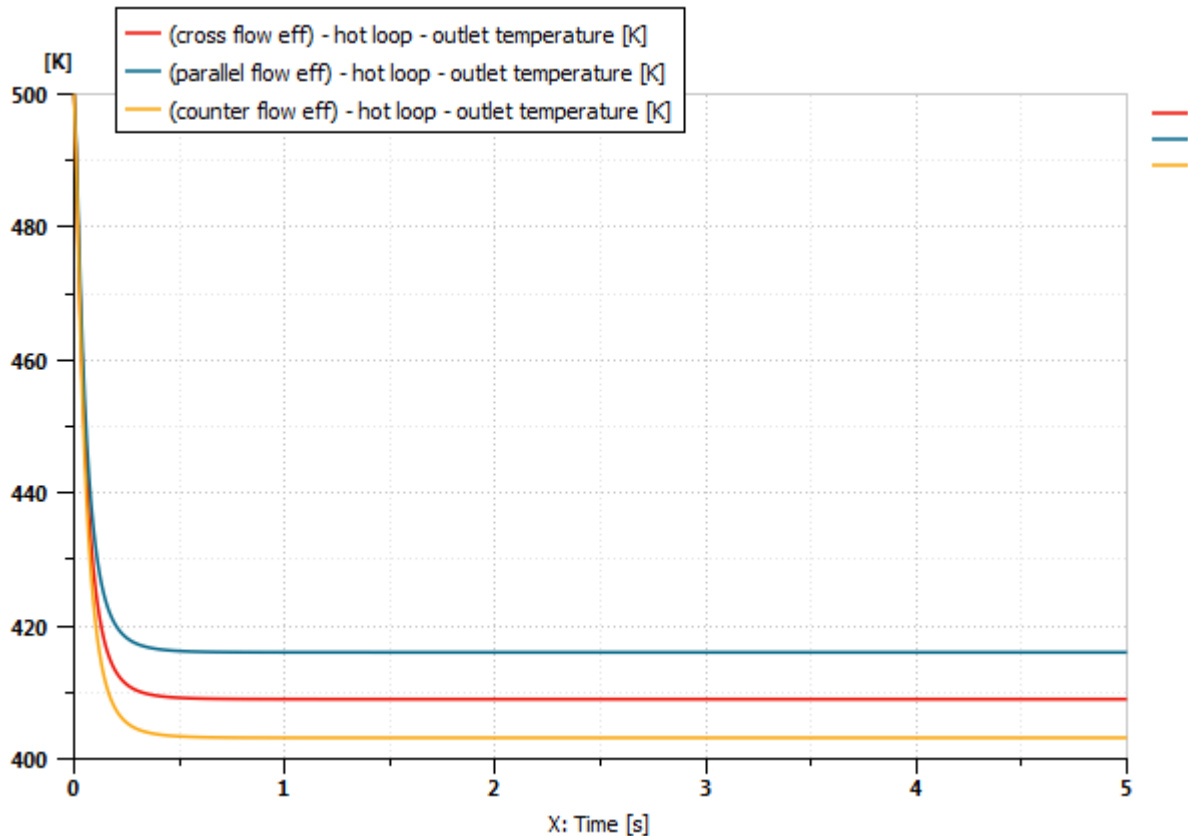


Fig. 10.6: Outlet gas temperatures (hot loop), cross, parallel and counter flows

We can easily see that every arrangement leads to different outlet temperatures for hot and cold gases. The most efficient arrangement is the counter-flow heat exchanger since the hot gas temperature is the lowest and the cold gas temperature the highest.

10.3 Gas - liquid heat exchangers

In the same way, it is possible to build gas-liquid heat exchangers using components from the **Pneumatic** and the **Thermal-Hydraulic** libraries. In this case, the two heat exchanger halves are submodels *TPEX0001* for the gas part and *TFOX0001* for the liquid part.

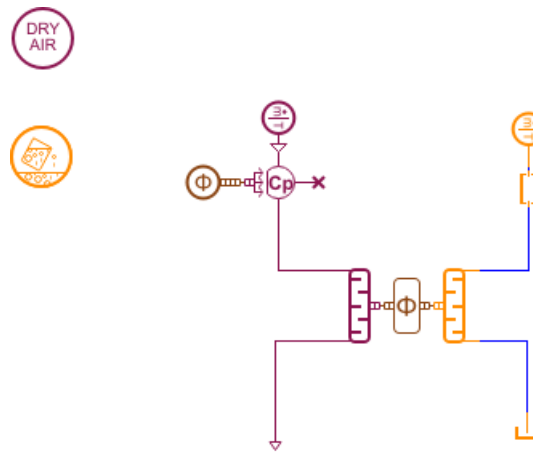


Fig. 10.7: Gas - liquid heat exchanger model

Refer to the **Thermal-hydraulic** manual for information on how to proceed for gas-liquid heat exchangers.

11. Appendix A: some gas properties

All the following values are extracted from data provided by NIST12, Thermodynamic and Transport Properties of Pure Fluids [5]

Table 11.1: Gas properties given at $T=293.15\text{ K}$ and $p=1.0\text{ barA}$:

Gas	ρ [kg/m ³]	C_p [J/kg/K]	C_v [J/kg/K]	γ	r [J/kg/K]
Ammonia NH ₃	0.707	2163.7	1641.7	1.318	522.0
Argon Ar	1.640	521.6	312.4	1.670	209.2
Butane C ₄ H ₁₀	2.463	1714.9	1548.4	1.108	166.5
Carbon dioxide CO ₂	1.815	846.0	652.4	1.297	193.5
Carbon monoxide CO	1.150	1040.3	741.9	1.402	298.4
Ethane C ₂ H ₆	1.244	1738.0	1452.2	1.197	285.8
Ethylene C ₂ H ₄	1.158	1520.9	1217	1.250	303.9
Helium He	0.164	5193.0	3116	1.667	2077.0
Hydrogen H ₂	0.083	14286.1	10160	1.406	4126.1
Isobutane C ₄ H ₁₀	2.452	1678.0	1514.6	1.108	163.3
Methane CH ₄	0.659	2220.5	1697.3	1.308	523.2
Neon Ne	0.828	1030.4	618.1	1.667	412.2
Nitrogen N ₂	1.150	1041.3	743.1	1.401	298.2
Oxygen O ₂	1.314	918.9	657.7	1.397	261.2
Propane C ₃ H ₈	1.840	1675.8	1472.6	1.138	203.2
Xenon Xe	5.417	160.1	95.4	1.678	64.7

and

Table 11.2: Gas properties given at $T=293.15\text{ K}$ and $p=1.0\text{ barA}$:

Gas	μ [Pa.s]	ν [m ² /s]	λ [W/m/K]	c [m/s]	Z
Ammonia NH ₃	9.91E-06	1.40E-05	1.40E-05	429.4	0.9889
Argon Ar	2.24E-05	1.37E-05	1.75E-02	319.0	0.9993
Butane C ₄ H ₁₀	7.28E-06	2.96E-06	1.57E-02	208.6	0.9683
Carbon dioxide CO ₂	1.47E-05	8.09E-06	1.62E-02	266.6	0.9947
Carbon monoxide CO	1.74E-05	1.52E-05	2.61E-02	349.2	0.9996

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Table 11.2 – continued from previous page

Gas	μ [Pa.s]	ν [m ² /s]	λ [W/m/K]	c [m/s]	Z
Ethane C ₂ H ₆	9.21E-06	7.40E-06	2.04E-02	309.0	0.9921
Ethylene C ₂ H ₄	1.01E-05	8.77E-06	1.97E-02	327.5	0.9940
Helium He	1.96E-05	1.20E-04	1.54E-01	1007.9	1.0005
Hydrogen H ₂	8.81E-06	1.07E-04	1.82E-01	1304.7	1.0006
Isobutane C ₄ H ₁₀	7.37E-06	3.01E-06	1.55E-02	209.5	0.9726
Methane CH ₄	1.10E-05	1.67E-05	3.36E-02	445.0	0.9982
Neon Ne	3.08E-05	3.72E-05	4.76E-02	448.9	1.0005
Nitrogen N ₂	1.76E-05	1.53E-05	2.54E-02	349.1	0.9998
Oxygen O ₂	2.04E-05	1.55E-05	2.57E-02	326.0	0.9993
Propane C ₃ H ₈	8.01E-06	4.35E-06	1.78E-02	246.5	0.9831
Xenon Xe	2.26E-05	4.18E-06	5.43E-03	175.5	0.9945

With:

- ρ the density [kg/m³]
- C_p constant-pressure specific heat [J/kg/K]
- C_v constant-volume specific heat [J/kg/K]
- γ specific heats ratio [null]
- r perfect gas constant [J/kg/K]
- μ absolute viscosity [Pa*s]
- ν kinematic viscosity [m²/s]
- λ thermal conductivity [W/m/K]
- c speed of sound [m/s]
- Z compressibility factor [null]

12. Appendix B: oxygen properties

An example of the oxygen compressibility factor Z and specific heat ratio as a function of pressure and temperature, from data provided by NIST12, Thermodynamic and Transport Properties of Pure Fluids [5].

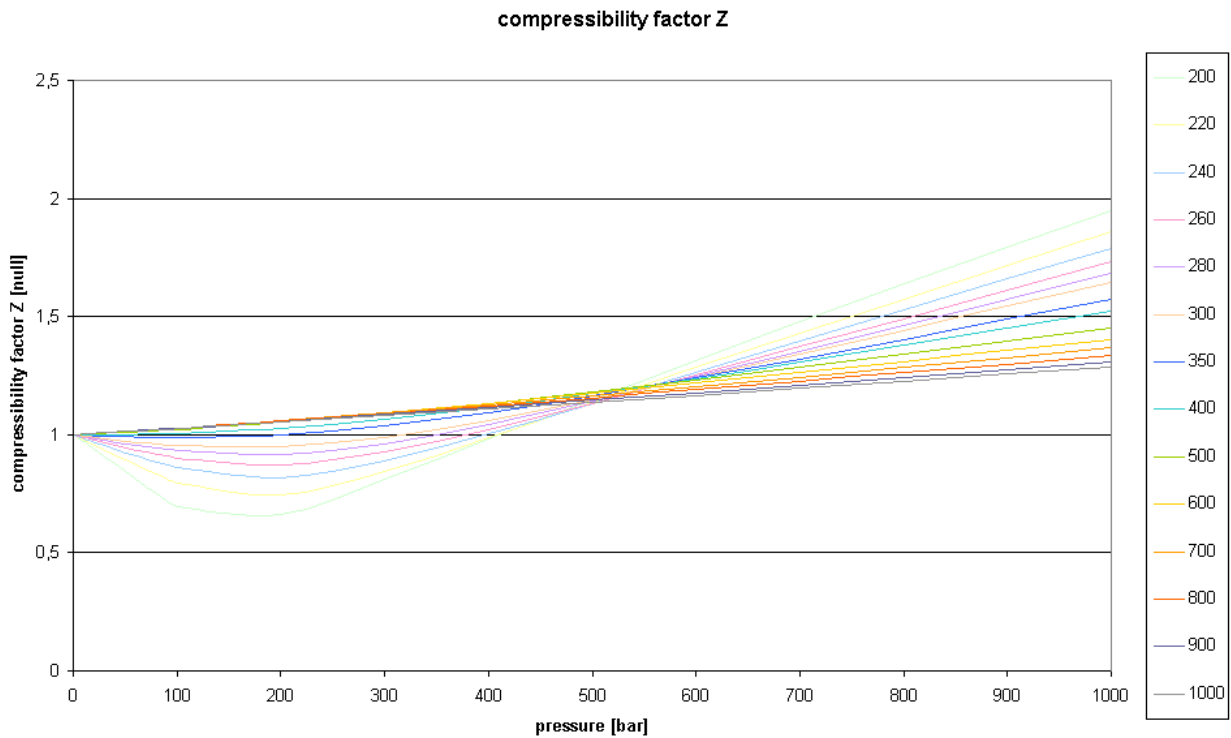
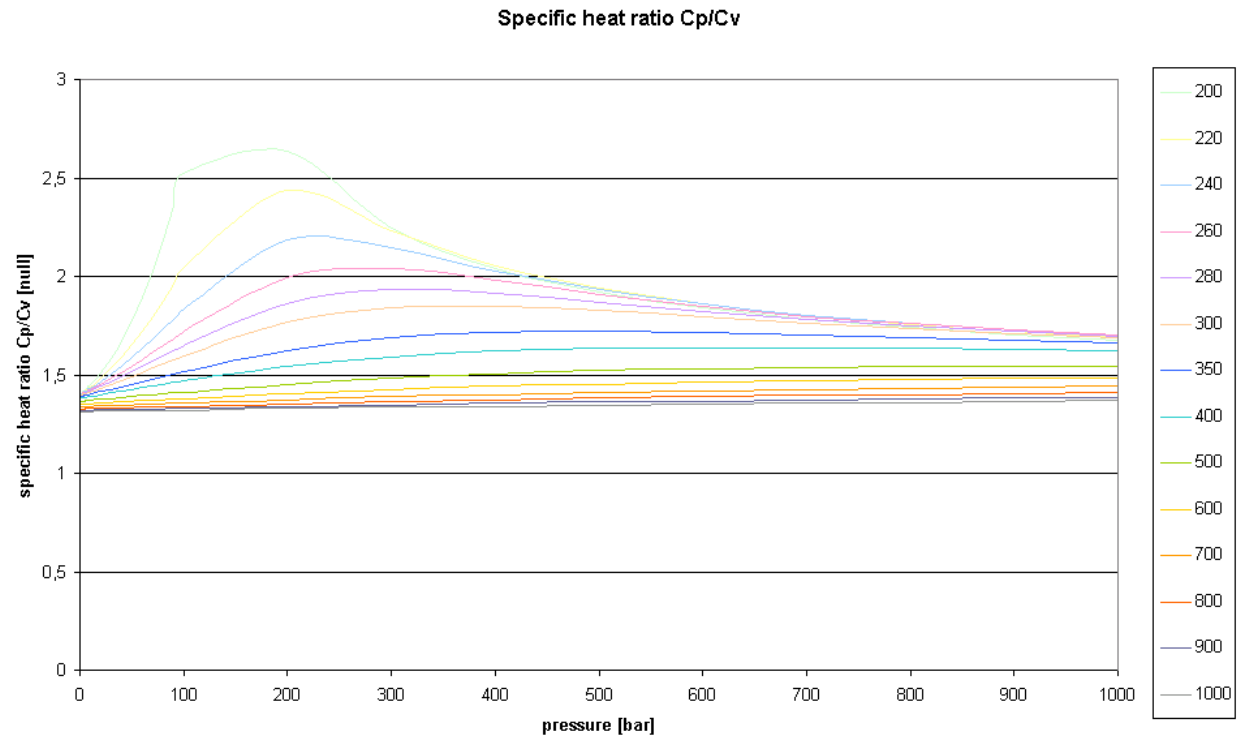


Fig. 12.1: Compressibility factor Z for oxygen

**Fig. 12.2: Specific heat ratio for oxygen**

13. Appendix C: Nusselt correlations

13.1 Summary of most common Nusselt correlation for external flow forced convection heat transfer

Correlation			Geometry	Conditions
$Nu = 0.332 \cdot Re^{0.5} \cdot Pr^{0.33}$ (13.1)			Flat plate	Laminar, local, $0.6 < Pr < 50$
$Nu = 0.664 \cdot Re^{0.5} \cdot Pr^{0.33}$ (13.2)			Flat plate	Laminar, average, $0.6 < Pr < 50$
$Nu = 0.0296 \cdot Re^{0.8} \cdot Pr^{0.33}$ (13.3)			Flat plate	Turbulent, local, $Re < 10^8$, $0.6 < Pr < 50$
$Nu = C \cdot Re^m \cdot Pr^{0.33}$ (13.4)			Cylinder	Average, $0.4 < Re < 4 \cdot 10^5$
Re	C	m		
0.4-4	0.989	0.33		
4-40	0.911	0.385		
40-4000	0.683	0.466		
4000-40000	0.193	0.618		
40000-400000	0.027	0.805		
$Nu = 2 + (0.4 \cdot Re^{0.5} + 0.06 \cdot Re^{0.66}) \cdot Pr^{0.4}$ (13.5)			Sphere	Average, $3.5 < Re < 7.6 \cdot 10^4$, $0.7 < Pr < 380$

13.2 Summary of most common Nusselt correlation for forced convection heat transfer flow in circular tubes

Correlation Conditions	
$Nu = 3.66$ (13.6)	Laminar, fully developed, uniform T, $Pr \geq 0.6$
$Nu = 1.86 \cdot$ (13.7)	Laminar, combined entry length, L = length of the cylinder, D = diameter of the cylinder
$Nu = 0.027$ (13.8)	Turbulent, fully developed, $0.7 < Pr < 16700$ $Re > 10000$, $L/D > 10$

13.3 Summary of most common Nusselt correlation for free convection heat transfer

A general form of the Nusselt number for free convection is given as follows:

$$Nu = C.(Gr.Pr)^n \quad (13.9)$$

where Gr is the Grashof number and Pr is the Prandtl number. It is explained in the theoretical section of this manual how these 2 numbers are computed.

Here are the values of C and n constant for different geometrical configuration of the free convection heat transfer:

- For laminar flow, **$n = 0.25$**
- For turbulent flow, **$n = 0.33$**

Geometry and orientation of the wall	Characteristic dimension of the exchange	C coefficient		
		Laminar convection $n = 0.25$	Turbulent convection $n = 0.33$	
Vertical plate or cylinder	Height	0.59 ($10^4 < Gr.Pr < 10^9$)	0.13 10^{13}	($10^9 < Gr.Pr < 10^{13}$)
Horizontal cylinder	External diameter	0.53 ($10^4 < Gr.Pr < 10^9$)	0.10 10^{13}	($10^9 < Gr.Pr < 10^{13}$)
Horizontal plate heating upwards	Width	0.54 ($10^4 < Gr.Pr < 10^9$)	0.14 10^{13}	($10^9 < Gr.Pr < 10^{13}$)
Horizontal plate heating downwards	Width	0.27 ($10^4 < Gr.Pr < 10^9$)	0.07 10^{13}	($10^9 < Gr.Pr < 10^{13}$)

14. Appendix D: heat exchanger effectiveness

Summary of the most common Heat Exchanger Effectiveness Relationships

All the following values are extracted from data provided by Incropera and Dewitt [3].

Flow arrangement:

1. Concentric tube

- Parallel flow

$$\varepsilon = \frac{1 - \exp[-NTU \cdot (1 + C_r)]}{1 + C_r} \quad (14.1)$$

- Counter flow

$$\varepsilon = \frac{1 - \exp[-NTU \cdot (1 - C_r)]}{1 - C_r \cdot \exp[-NTU \cdot (1 - C_r)]} \quad (C_r < 1) \quad (14.2)$$

$$\varepsilon = \frac{NTU}{1 + NTU} \quad (C_r = 1) \quad (14.3)$$

2. Shell and tube

- One shell pass (2, 4, ..., tube passes)

$$\varepsilon_1 = 2 \cdot \left\{ 1 + C_r + (1 + C_r^2)^{\frac{1}{2}} \cdot \frac{1 + \exp[-NTU \cdot (1 + C_r^2)^{\frac{1}{2}}]}{1 - \exp[-NTU \cdot (1 + C_r^2)^{\frac{1}{2}}]} \right\}^{-1} \quad (14.4)$$

- n shell pass (2n, 4n, ..., tube passes)

$$\varepsilon = \left[\left(\frac{1 - \varepsilon_1 C_r}{1 - \varepsilon_1} \right)^n - 1 \right] \cdot \left[\left(\frac{1 - \varepsilon_1 C_r}{1 - \varepsilon_1} \right)^n - C_r \right]^{-1} \quad (14.5)$$

3. Cross flow (single pass)

- Both fluids unmixed

$$\varepsilon = 1 - \exp \left[\left(\frac{1}{C_r} \right) \cdot (NTU)^{0.22} \cdot \{ \exp[-C_r (NTU)^{0.78}] - 1 \} \right] \quad (14.6)$$

- $C_{\max}$ (mixed), $C_{\min}$ (unmixed)

$$\varepsilon = \left(\frac{1}{C_r} \right) \cdot (1 - \exp \{ -C_r [1 - \exp(-NTU)] \}) \quad (14.7)$$

- $C_{\min}$ (mixed), $C_{\max}$ (unmixed)

$$\varepsilon = 1 - \exp(-C_r^{-1} \{ 1 - \exp[-C_r (-NTU)] \}) \quad (14.8)$$

4. All exchangers ($C_r=0$)

$$\varepsilon = 1 - \exp(-NTU) \quad (14.9)$$

where: NTU is the number of transfer units [null] and $C_r = \frac{C_{\min}}{C_{\max}}$ heat capacity ratio [null]

15. Appendix E: energy storage phenomena

In industrial pneumatic systems operating at low pressures, the typical order of magnitude of the specific energies for air are:

- Potential energy due to gravitation for a difference of level of 1 meter:

$$9.81.1 = 9.81 \text{ J/kg} \quad (15.1)$$

- Kinetic energy for a velocity of 100 m/s:

$$\frac{1}{2}v^2 = 5 \cdot 10^3 \text{ J/kg} \quad (15.2)$$

- Internal energy of air at 25 degC:

$$u = c_v T = \frac{1004}{1.4} \cdot 298 = 213708 \text{ J/kg} \quad (15.3)$$

- Pressure energy of air at 25 degC:

$$\frac{p}{\rho} = rT \approx 287 \cdot 298 = 85226 \text{ J/Kg} \quad (15.4)$$

- Specific enthalpy of air at 25 degC:

$$h = c_p T \approx 1004 \cdot 298 = 299192 \text{ J/Kg} \quad (15.5)$$

These values justify that in a control volume, the kinetic and potential energies can be ignored in comparison with the internal energy of the gas.

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