



UNIVERSITÀ
DEGLI STUDI
FIRENZE

FLORE

Repository istituzionale dell'Università degli Studi di Firenze

A parametric library for the simulation of UIC pneumatic braking system

Questa è la Versione finale referata (Post print/Accepted manuscript) della seguente pubblicazione:

Original Citation:

A parametric library for the simulation of UIC pneumatic braking system / L. PUGI; M. MALVEZZI; B. ALLOTTA; L. BANCHI; P. PRESCIANI. - In: PROCEEDINGS OF THE INSTITUTION OF MECHANICAL ENGINEERS. PART F, JOURNAL OF RAIL AND RAPID TRANSIT. - ISSN 0954-4097. - STAMPA. - 218 part F:(2004), pp. 117-132. [10.1243/0954409041319632]

Availability:

The webpage <https://hdl.handle.net/2158/212608> of the repository was last updated on 2016-09-08T18:00:40Z

Published version:

DOI: 10.1243/0954409041319632

Terms of use:

Open Access

La pubblicazione è resa disponibile sotto le norme e i termini della licenza di deposito, secondo quanto stabilito dalla Policy per l'accesso aperto dell'Università degli Studi di Firenze (<https://www.sba.unifi.it/upload/policy-oa-2016-1.pdf>)

Publisher copyright claim:

La data sopra indicata si riferisce all'ultimo aggiornamento della scheda del Repository FloRe - The above-mentioned date refers to the last update of the record in the Institutional Repository FloRe

(Article begins on next page)

A parametric library for the simulation of a Union Internationale des Chemins de Fer (UIC) pneumatic braking system

L Pugi¹, M Malvezzi^{1*}, B Allotta¹, L Banchi¹ and P Presciani²

¹Dipartimento di Energetica 'Sergio Stecco', Sezione di Meccanica Applicata, University of Florence, Italy

²UTMR Trenitalia SPA, Florence, Italy

Abstract: European trains are equipped with a pneumatic braking system that has to respect severe specifications concerning both performances and safety. The pneumatic braking system is composed of hundreds of different pneumatic components that reproduce the prescribed response by a complex logic of pneumatic and mechanical elements. In this paper a tool for a complete simulation of the pneumatic braking system is described, it was developed using the Matlab–Simulink numerical environment.

The tool is composed of three different libraries of pneumatic components. The first includes the elementary components such as pipes, orifices, valves and the reservoir. By assembling elementary components, an advanced user can build a customized version of general pneumatic components or plants. Complex components of general use for railway pneumatic brake such as brake cylinders, distributors, pressure transformers and brake valves are available in a second library that can be used to assemble a customized braking plant for a vehicle. The last library is composed of macro-pneumatic subsystems that reproduce the braking system of a typical railway vehicle. Many common plant layouts are reproduced in this library (freight car, passenger coaches, locomotives, etc.).

The pneumatic brake system of a train can be simulated by assembling in a single Matlab–Simulink model the elements of the library.

In this paper the main features of this numerical tool and the test procedures developed to validate the software are described. Experimental data have been kindly supplied by Trenitalia SPA and they are referred to several test campaigns managed by Italian railway in order to verify and release existing components of the pneumatic brake.

Keywords: pneumatic brake, simulation, computational fluid dynamics (CFD)

NOTATION

a acceleration of the train
 A equivalent area of the pipe
 b_{line} equivalent resistance factor introduced by the line slope and curves
 $b_{\text{res}}(v_{\text{train}})$ equivalent dissipative factor due to vehicle's internal friction and aerodynamics; resistances, usually a polynomial function of the train travel speed

B longitudinal braking force exchanged between the vehicle and the rail
 B_i^{electr} equivalent longitudinal forces due to electric or electromagnetic braking on vehicle i
 B_{extras} external longitudinal forces applied on the complete train
 C_{friction} equivalent mechanical damping of the cylinder
 C_M Mach correction factor of mass flow
 C_q empirical correction factor of mass flow
 CaA1, CaA2 accelerator chambers
 CaC chamber of the distributor main device
 CaP pilot chamber of the pressure transformer
 CF brake cylinder
 CFD computational fluid dynamics

The MS was received on 15 April 2003 and was accepted after revision for publication on 28 November 2003.

* Corresponding author: Dipartimento di Energetica 'Sergio Stecco', Sezione di Meccanica Applicata, Università di Firenze, via S. Marta 3, 50139 Firenze, Italy.

CG	brake pipe
CP	auxiliary brake pipe
d	pipe diameter
F	clamping force exerted by the pneumatic cylinder
$F_{\text{pre-load}}$	pre-load of the cylinder
h_0	total enthalpy per unit mass
k	$1/(RT)$ (assuming that $T = 293 \text{ K}$) $= 1.189 \text{ e}^{-5} \text{ s}^2/\text{m}^2$
K_{spring}	stiffness of the cylinder pre-load spring
m_i^{vehicles}	equivalent inertial mass of vehicle i
m_{train}	equivalent inertial mass of the complete train
\dot{m}	mass flow
M	equivalent inertia of the piston and connected transmission elements
n_{cylinder}	number of brake cylinders placed on the vehicle
n_{disc}	number of brake discs applied to an axle
n_{pad}	number of pads applied on a single disc
n_{wheelset}	number of axles braked by a single cylinder
p	pressure
p_d	outlet pressure
$p_{t,z}$	discretized pressure at time t and node z
p_u	inlet pressure
P	vehicle weight
\dot{q}	specific thermal power per volume unit
R	constant for ideal gas
R_{brake}	equivalent braking radius of the disc
R_{wheel}	rolling radius of the wheel
Re	Reynolds number
SA	auxiliary reservoir
SC	command reservoir
SP	main reservoir (locomotive)
T	temperature
u	air velocity
$u_{t,z}$	discretized velocity at time t and node z
v_{train}	train speed
V	volume
X	piston stroke
z	position along the pipe length
γ	isentropic exponent
η	efficiency of brake rigging
λ	loss factor
μ_{pad}	friction factor between pad and disc
μ_{rail}	friction factor between rail and wheel
ρ	density
τ	pressure loss in the pipe per unit length
τ_b	mechanical transmission rate of rigging (ratio of the normal pressure on each brake pad to the force exerted by each brake cylinder)

1 INTRODUCTION: THE UNION INTERNATIONAL DES CHEMINS DE FER PNEUMATIC RAILWAY BRAKE

The braking system of a train is a complex plant whose dynamic behaviour involves the interaction of pneumatic, mechanical and electronic components. Performance, safety and reliability specifications of the braking system can really affect many different aspects of the management of an entire railway system, and in particular:

- safety,
- traffic intensity,
- maximum commercial velocity, management of the signalling system, and
- length, weight, number and kind of trailer vehicles, etc.

The braking system of every single vehicle interacts with the complete pneumatic plant of the train. Very demanding specifications prescribed by regulations in force [1, 2] have to be met in order to assure safety and interoperability between different kinds of vehicle.

The braking system of a single vehicle can be modelled in three main-subsystems (as can be seen in Fig. 1):

- Union International des Chemins de Fer (UIC) pneumatic brake.* A pneumatic plant is used to control the braking system of each vehicle of the train. The braking command of the driver is transmitted along the train as pressure reference. A complex system of mechanical and pneumatic components usually called the UIC pneumatic brake is used to regulate the clamping forces of the brakes from this pressure reference. The main features of this kind of brake are well known among the community of railway researchers:
 - Pneumatic.* Each actuator or control component works with pressurized air.
 - Continuous.* A single pressure signal propagates the brake-release command along the train.
 - Automatic (safe).* If the train integrity is interrupted, the vehicles automatically brake.
 - Inexhaustible.* The brake is released only if the air reservoirs of the vehicle are filled.
- Mechanical brake on braking surfaces.* A pneumatic cylinder is used to clamp the sliding surfaces of a mechanical brake. Disc or block brakes are the most common solutions.
- Anti-skid device.* The adhesion between the rolling surfaces of rails and wheels is quite low. When the braking forces are higher than the available adhesion, the rolling surfaces of wheel and rail begin to skid. The anti-skid device is an electro-pneumatic system able to regulate the clamping force exerted on every

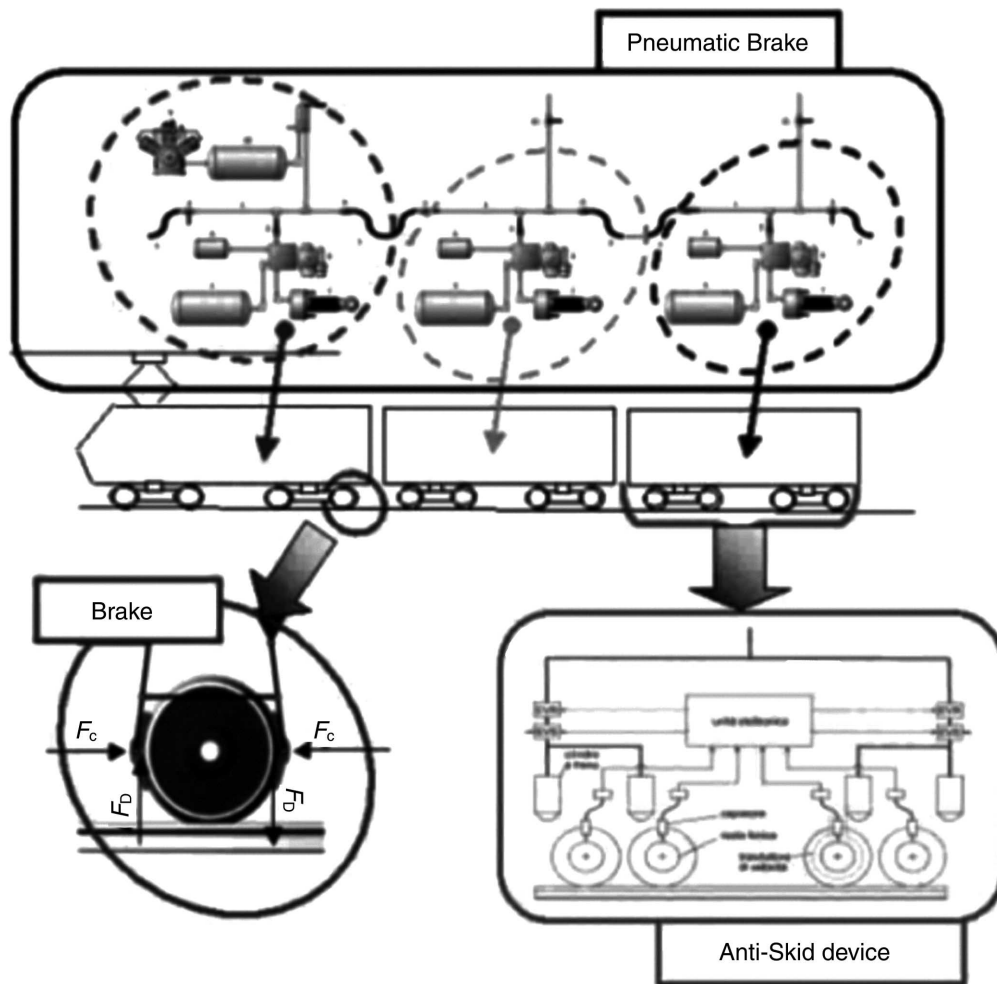


Fig. 1 Simplified scheme of the braking system of a modern train

axle in order to prevent skid and to minimize the space required to stop the vehicle. All vehicles with a maximum speed above certain levels are equipped with anti-skid devices, and in particular high-speed trains.

In this work, the present authors have focused their attention on the first two subsystems, neglecting the effects of anti-skid device. This approach may be useful in order to simulate braking performances of a train with optimal adhesion conditions when this kind of device does not intervene.

The dynamic behaviour of the pneumatic plant may delay the propagation of the braking–release command along the train. This delay could cause heavy longitudinal loads along the train, which may cause damage or even derailment. In order to avoid this dangerous condition the pneumatic components of the plant are often regulated with complex systems of calibrated orifices, springs and reservoirs.

The optimization of this system is quite difficult and so a numerical tool can really contribute to reduce testing and development costs. Also the complexity of

the resulting pneumatic plant may introduce undesired or unpredictable responses of the systems such as uncontrolled partial brake or release; the numerical simulation may help this to be understood.

The paper is organized as follows. Section 2 describes the main features of a pneumatic plant. In section 3 the developed software is described, and in particular the main blocks contained in the toolbox are shown. Section 4 presents a first validation of the model, obtained by comparing the simulation results with the experimental data. In section 5 the conclusions of the work and a brief description of possible improvements are presented.

2 PNEUMATIC PLANT DESCRIPTION

2.1 General layout

Figure 2 shows the simplified scheme of the UIC pneumatic brake [3]. The brake pipe CG (1) links all the pneumatic circuits of the train through a system of

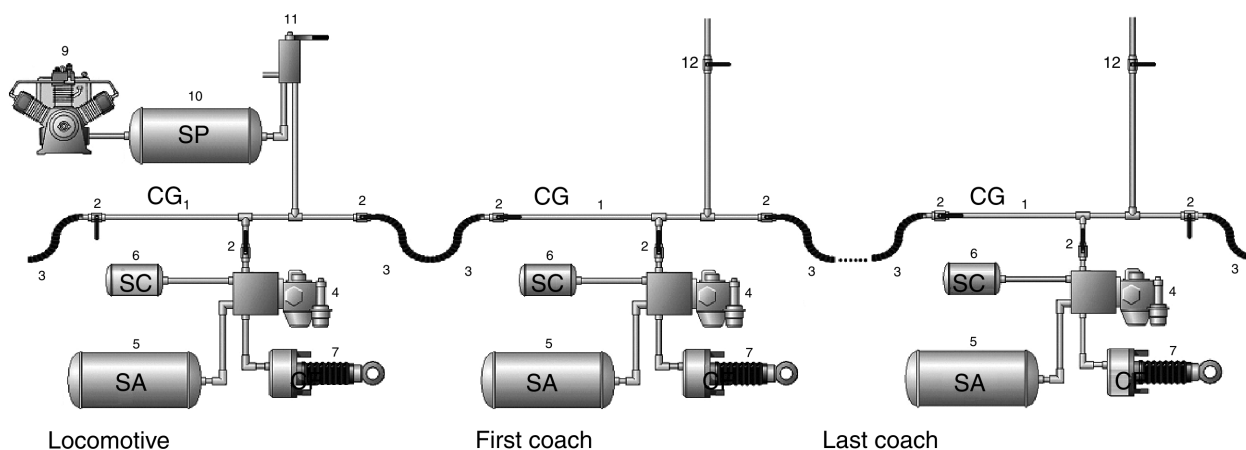


Fig. 2 Simplified pneumatic layout of the UIC brake

cocks (2) and coupled hoses (3) which are used to connect the vehicles. The distributor (4) is connected to the CG.

The distributor evaluates the difference between the constant pressure of the command reservoir SC (6) and the reference signal of CG. If the pressure of the brake pipe is lower than the SC pressure, the distributor regulates the pressure of the braking cylinder CF (7) with a pressure which is approximately proportional to the pressure drop of the brake pipe. CG feeds the distributor with pressurized air which is stored in the auxiliary reservoir SA (5). If the CF pressure rises above the desired level, the distributor connects the cylinder with the atmosphere.

SA and SC are fed by the pneumatic brake during the release phase of the brake; in this phase the pressure of the brake pipe returns to its nominal value. The complete plant is fed by a compressor (9) and a main reservoir SP (10) placed on the locomotive. The reference pressure signal of the brake pipe is controlled by the driver's brake valve (11) placed on the locomotive.

In order to obtain a shorter brake release phase, an auxiliary pressurized pipe may be directly connected to the reservoir SA; this second pipe CP (1) is directly fed by the main reservoir. Figure 3 shows a more detailed scheme of a pneumatic brake with an auxiliary pipe. The distributor is a totally pneumatic control element with a very complex internal structure.

In this study, attention was focused on a particular kind of distributor, the Westinghouse U distributor, which is quite common on Italian trains. Figure 4 shows a detailed scheme of this type of distributor.

2.2 Westinghouse U distributor

The Westinghouse U distributor it is composed of five main subsystems:

1. *Main device.* This is composed of three pistons with the aim of regulating the air pressure inside the brake cylinders, in order to satisfy the reference pressure signal of the brake pipe CG.

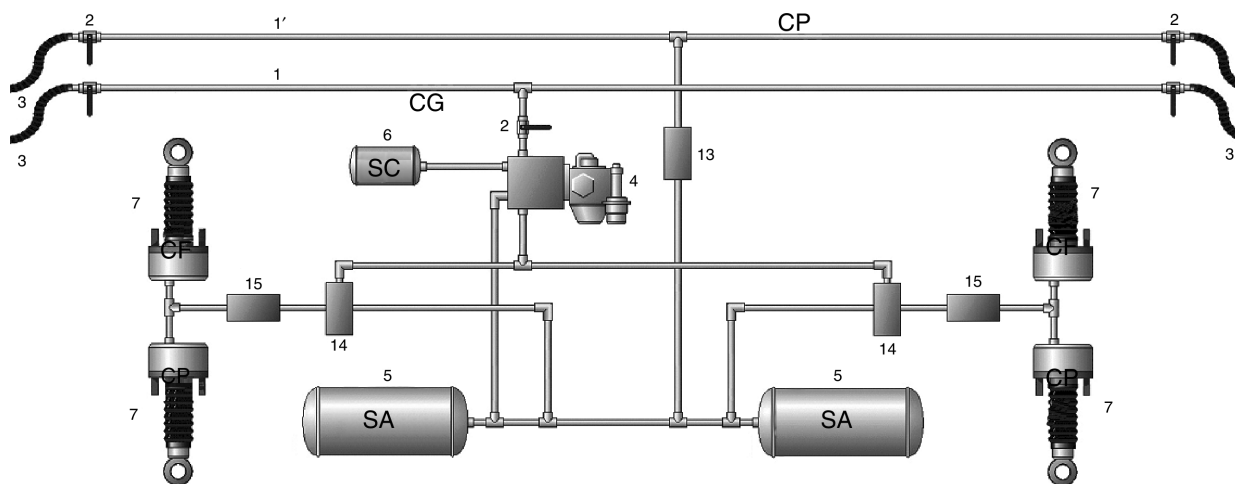


Fig. 3 Pneumatic brake scheme with the auxiliary pipe

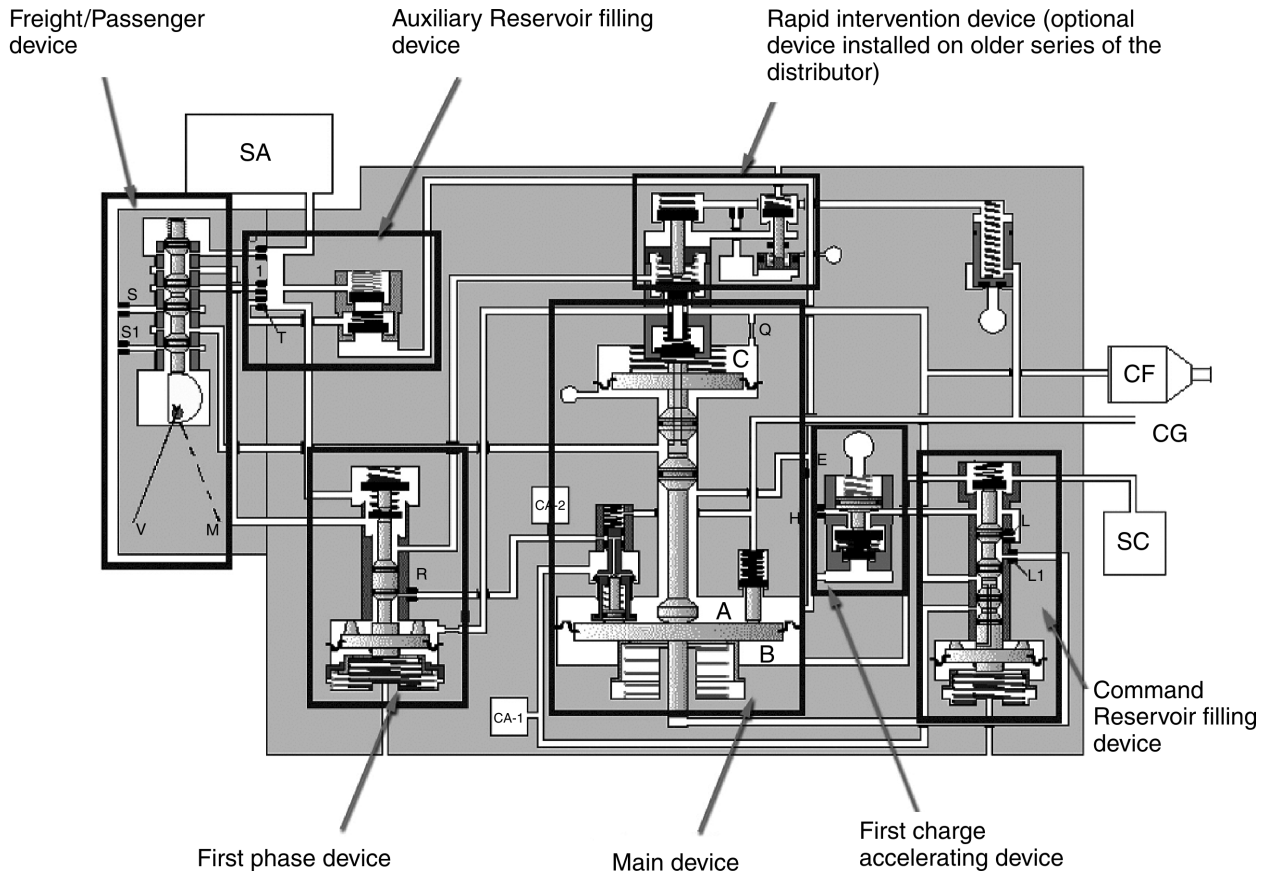


Fig. 4 Detailed pneumatic scheme of the Westinghouse U distributor

2. *Command reservoir filling device and first charge accelerating device.* These two devices are devoted to regulating the correct and faster feeding of the command reservoir, in order to assure correct intervention of the distributor.
3. *First phase device.* This pneumatic device introduces a non-linear correction of the distributor response. For low braking pressures (lower than 1.6 bar) the feeding of the cylinder is faster. Also a system of auxiliary capacities is used to accelerate the propagation of the pressure drop signal along the brake pipe. When high braking pressures occur, this device introduces an upper saturation of cylinder pressure corresponding to a value of 4.85 bar.
4. *Auxiliary reservoir filling device.* This device is designed in order to assure a faster and safer feeding of the auxiliary reservoir SA. When the pressure drop between SA and the brake pipe is higher than 0.4 bar, this device assures fast recharge of the auxiliary reservoir.
5. *Freight-passenger device.* Regulation in force establishes that passenger coaches and freight cars must have different braking performances. This is due to several technical reasons which are well known among the community of railway researchers:

- (a) different train lengths (freight trains may be much longer than passenger trains);
- (b) different commercial speeds (passenger trains are characterized by higher speeds);
- (c) the high weight variability of freight trains with inanimate loads that are often more sensitive to acceleration variations;
- (d) different braking performances.

For these reasons freight train brake systems have very different performances from passenger train brake systems. This pneumatic device is used to obtain a different dynamic response of the distributor according to the kind of vehicle (freight or passenger).

3 MATHEMATICAL MODELLING

A library of pneumatic models for the Matlab-Simulink environment has been developed in order to simulate the dynamic behaviour of the UIC railway brake. This library is composed of three main subsets of models:

1. *Elementary components.* A set of models is used that is able to simulate simple physical problems such as

pressure propagation on a constant-section pipe or the flux through an orifice. These components are defined as elementary because almost all the components of a pneumatic plant can be built by assembling these simple blocks.

2. *Complex components.* Brake cylinders, distributors, pressure transformer, brake valves and other complex devices are defined as an array of elementary components whose dynamic interaction is controlled by a finite state machine modelled with Stateflow, a Matlab–Simulink dedicated tool.
3. *Full vehicle plant.* The pneumatic plant of a single vehicle (passenger coach, freight car, locomotive, etc.) is modelled by assembling ‘elementary’ and ‘complex’ pneumatic components. By assembling different models of ‘vehicles’, the user can easily build up the complete pneumatic plant of a train. The subsystem also includes a simple model of the longitudinal dynamic of the train according to the applied braking forces.

3.1 Elementary components

This library is mainly composed of five components:

1. *Pipe calculus node.* Long pipes such as CG are discretized using the *finite difference* technique with a sequence of calculus nodes.
2. *Pipe capacity node.* This node is used as a fitting or a connector to link two or more components of the plant.
3. *Orifice.* This simple element is used to model calibrated holes or valves (treated as orifices with a variable area).
4. *Chambers.* This component is used for a simple and fast calculation of the capacity elements placed inside complex pneumo-logic elements such as distributors.
5. *Terminators and modified blocks.* Terminator blocks are used to introduce boundary conditions of the plant such as imposed pressures or mass flows. Other components can be modelled as modified versions of the previously described blocks. For example, in the cylinder subsystem, some customized elements have been inserted in order to model a continuously variable capacity.

3.1.1 Pipe calculus node

A pipe is modelled as a constant-area one-dimensional duct. Air is assumed to be an ideal gas; all the transformations are isothermal ($T = 293$ K) and pressure loss are modelled according to Darcy–Weisbach formulation. This approach is quite common in the literature [4–8]. The flow cannot be modelled as adiabatic, due to the high irreversibility of thermodynamic transformations and the large thermal

exchange surfaces involved. The polytropic transformation that represents a better approximation of the phenomenon is then isothermal. This model represents a good compromise between computational load and reliability of simulation results.

The following equations can be written in order to describe the dynamics of the flow in the pipe. The conservation of the mass in the tube can be expressed by means of the differential equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial z} = 0 \quad (1)$$

while the equilibrium balance can be written as

$$\rho \frac{du}{dt} = -\frac{\partial p}{\partial z} - \tau \quad (2)$$

The total enthalpy balance can be expressed as

$$\rho \frac{dh_0}{dt} = \frac{\partial p}{\partial t} + \dot{q} \quad (3)$$

The fluid involved in the flow is air, and then the following relation holds:

$$p = \rho RT \quad (4)$$

The pressure loss in the pipe is calculated by means of the expression

$$\tau = \lambda \frac{\rho}{d} \frac{u^2}{2} \quad (5)$$

where λ is calculated, according to references [4] to [6], as a function of the Reynolds number Re , using the expressions

$$\begin{aligned} Re < 2000, & \quad \lambda = \frac{64}{Re} \\ 2000 < Re < 4000, & \quad \lambda = \frac{0.0027}{Re^{0.222}} \\ Re > 4000, & \quad \lambda = \frac{0.316}{Re^{0.25}} \end{aligned}$$

In this analysis, the enthalpy balance (4) was not considered and isothermal conditions were supposed. In this case the flow can be described by the difference equations

$$\frac{\partial p}{\partial t} = -u \frac{\partial p}{\partial z} - p \frac{\partial u}{\partial z} \quad (6a)$$

$$\frac{\partial u}{\partial t} = -u \frac{\partial u}{\partial z} - \frac{1}{\rho} \frac{\partial p}{\partial z} - \frac{\lambda}{d} \frac{u^2}{2} \quad (6b)$$

These equations were solved using a numerical approach. The pipe was divided into a series of calculus nodes. The derivative terms in the differential equation

were approximated as

$$\begin{aligned}\frac{\partial p_{t,z}}{\partial t} &= \frac{p_{t+\Delta t,z} - p_{t,z}}{\Delta t} \\ \frac{\partial p_{t,z}}{\partial z} &= \frac{p_{t,z} - p_{t,z-\Delta z}}{\Delta z} \\ \frac{\partial u_{t,z}}{\partial z} &= \frac{u_{t,z} - u_{t,z-\Delta z}}{\Delta z} \\ \frac{\partial u_{t,z}}{\partial t} &= \frac{u_{t+\Delta t,z} - u_{t,z}}{\Delta t}\end{aligned}\quad (6c)$$

Then, from equations (6a) and (6b) the following difference equations are obtained:

$$P_{t+\Delta t,z} = p_{t,z} \left(1 - k \frac{\Delta t}{\Delta z} (u_{t,z} - u_{t,z-\Delta z}) \right) - u_{t,z} \frac{\Delta t}{\Delta z} (p_{t,z} - p_{t,z-\Delta z}) \quad (7a)$$

$$u_{t+\Delta t,z} = u_{t,z} \left(1 - \frac{\Delta t}{\Delta z} (u_{t,z+\Delta z} - u_{t,z}) \right) - \frac{\Delta t}{\Delta z} \frac{(p_{t,z+\Delta z} - p_{t,z})}{k p_{t,z}} - \frac{\lambda}{d} \Delta t \frac{u_{t,z}^2}{2} \quad (7b)$$

3.1.2 Pipe capacity node

The *pipe capacity node* is modelled as a constant isothermal volume with various inlet and outlet mass flows. Imposing mass conservation and neglecting kinetic energy the following equation is obtained:

$$\frac{\partial}{\partial t} \left(\int_{V_c} \rho_c dv \right) = \sum_{i=1}^n \frac{\pi(d_i)^2}{4} \rho_i u_{t,i} \quad u_{t,i} > 0 \Rightarrow (\text{inlet flow}) \quad (8)$$

In this calculus node the flows are characterized by the same temperature (due to the isothermal assumption) and locally have the same pressure; then the same constant density is assumed for every flow ($\rho_i = \rho_c$). According to these hypotheses the mass balance can be simplified as

$$\frac{\partial p_c}{\partial t} = \frac{\pi p_c}{4 V_c} \sum_{i=1}^n d_i^2 u_{t,i} \quad (9)$$

V_c represents the local equivalent capacity of the pipe. Usually, because of the dimension of the discretization grid its value is *small*. In other words, the pipe capacity node is a fictitious numerical solution to model fittings and connector by imposing a simple mass balance, while V_c models the damping in the pressure drop propagation due to the fluid compressibility.

Lumped pressure losses associated with the connectors are evaluated in the pipe calculus nodes, linked to the pipe capacity node. Equation (9) is then discretized following the finite differences approach, giving the equation

$$p_{t+\Delta t,c} = p_{t,c} \left(1 + \frac{\pi \Delta t}{4 V_c} \sum_{i=1}^n d_i^2 u_{t,i} \right) \quad (10)$$

3.1.3 Orifice

Calibrated holes or valves are modelled using the *orifice* component. The block orifice calculates the mass flow using well-known empirical correlations [9–11] as a function of inlet p_u and outlet pressure p_d :

$$\dot{m} = A C_q C_M \frac{p_u}{\sqrt{T_u}} \quad (11)$$

where C_M is a correction factor that is calculated from an isentropic approach to the problem (following the procedure described in the ISO 6358: 1989 regulation [12]):

$$C_M = \begin{cases} \sqrt{\frac{2\gamma}{R(\gamma-1)}} \sqrt{\left(\frac{p_d}{p_u}\right)^{2/\gamma} - \left(\frac{p_d}{p_u}\right)^{(\gamma+1)/\gamma}}, & \text{subsonic} \\ \sqrt{\frac{\gamma}{R} \left(\frac{2}{\gamma+1}\right)^{(\gamma+1)/(\gamma-1)}} \\ = \sqrt{\frac{2\gamma}{R(\gamma+1)} \left(\frac{2}{\gamma+1}\right)^{1/(\gamma-1)}}, & \text{supersonic} \end{cases} \quad (12)$$

However, the real flow is not necessarily isentropic (due to the high irreversibility of the thermodynamic processes), therefore an empirical correction factor C_q evaluated according to the Perry correlation is introduced:

$$C_q = 0.8414 - 0.1002 \left(\frac{p_d}{p_u}\right) + 0.8415 \left(\frac{p_d}{p_u}\right)^2 - 3.9 \left(\frac{p_d}{p_u}\right)^3 + 4.6001 \left(\frac{p_d}{p_u}\right)^4 - 1.6827 \left(\frac{p_d}{p_u}\right)^5 \quad (13)$$

The orifice area can be changed in order to simulate the behaviour of a valve.

There are no derivative terms in equation (11) and so the mass flow can be directly computed without any kind of discretization from the values of inlet and outlet pressures. From a physical point of view, an orifice block represents a lumped pressure drop associated with a part of the modelled pneumatic component.

3.1.4 Chamber

Inlet and outlet pressures of orifice elements are evaluated using *chambers*. A chamber is a fictitious volume capacity, which is introduced in order to calculate inlet and outlet pressures of the connected orifice blocks. Chambers components evaluate the pressure inside a known volume by means of a simple mass balance of inlet and outlet flows:

$$\frac{\partial}{\partial t} \left(\int_{V_c} \rho_c dv \right) = \sum_{i=1}^n \dot{m}_{t,i}$$

Air is modelled as an ideal gas and the process is assumed to be isothermal; then the following relation holds:

$$\frac{\partial}{\partial t} (kp_c V_c) = \sum_{i=1}^n \dot{m}_{t,i}$$

from which the pressure derivative term can be evaluated:

$$\frac{\partial p_c}{\partial t} = \frac{\sum_{i=1}^n \dot{m}_{t,i}}{kV_c} \quad (14)$$

The following difference equation can be obtained by discretizing equation (14):

$$p_{t+\Delta t,c} = p_{t,c} + \frac{\Delta t}{kV_c} \sum_{i=1}^n \dot{m}_{t,i} \quad (15)$$

Equations (14) and (15) are quite similar to the pipe capacity node equations (9) and (10). However, the ways in which the chamber and pipe capacity node equations are used are quite different; the capacity V_c parameter of the chamber elements is not necessarily small, since it represents the equivalent lumped capacity of the modelled part (such as a reservoir). For complex components such as distributors, inaccuracy in the simulation results may be introduced by unmodelled phenomena due to the geometrical layout of the components or to secondary elements that have been neglected. Then the value of the V_c parameter must be tuned in order improve the correspondence to experimental data.

3.2 Complex components

This library is composed of several different submodels that are able to reproduce the pneumatic behaviour of the brake plant components:

1. *Cylinder*. This model reproduces the dynamic response of the brake cylinder and calculates the resulting clamping forces according the mechanical brake layout (number of discs, leverage ratio and efficiency, etc.). The equivalent capacity of the

cylinder is modelled by means of a special chamber with a continuous volume V_c .

2. *Brake valves*. Models of the valves are used to regulate the pressure of the brake pipe CG.
3. *Distributor*. Complex pneumatic-logic components are used to regulate the clamping pressure of brake cylinders from the reference signal of pipe CG.
4. *Pipes and connectors*. Arrays of pipe calculus nodes and pipe capacity nodes are available in order to simulate pipes and common kind of connectors (three-way T-connectors, cross connectors, coupled hoses between different diameters, etc.).
5. *Other components*. Several minor components are available as lightly modified existing models.

3.2.1 Cylinder

The brake cylinder (whose section is shown in Fig. 5) is modelled as a special chamber with a variable volume whose value depends on the piston stroke X . Pneumatic behaviour is evaluated with a simple mass balance

$$\frac{\partial}{\partial t} \left(\int_{V_c} \rho_c dv \right) = \dot{m}_{t,i}$$

By assuming an ideal gas with isothermal conditions this equation can be rewritten as

$$\begin{aligned} \frac{\partial}{\partial t} (kp_c V_c) &= \dot{m}_{t,i} \\ \frac{\partial}{\partial t} [kp_c AX(t)] &= \dot{m}_{t,i} \\ \frac{\partial}{\partial t} [p_c X(t)] &= \frac{\dot{m}_{t,i}}{Ak} \end{aligned} \quad (16)$$

Equations (16) can be integrated according to the initial condition P_0 when the cylinder stroke is equal to X_0 . The pressure p of the cylinder can be then evaluated as a function of the mass flow and the piston stroke $X(t)$:

$$p_c(t) = \frac{1}{X(t)} \left(X_0 p_0 + \frac{1}{kA} \int_0^t \dot{m}(t) dt \right) \quad (17)$$

The piston dynamics and the force exerted by the actuator are calculated using different relations that depends on the value of piston stroke. At the beginning of the braking phase there is no contact between pad and disc, and so the piston motion is described by the equation

$$\begin{aligned} M\ddot{X}(t) &= A[p_c(t) - p_a] - C_{\text{friction}}\dot{X}(t) \\ &\quad - K_{\text{spring}}X(t) - F_{\text{pre-load}} \end{aligned} \quad (18)$$

When the piston reaches the position X_{contact} , which



Fig. 5 Brake cylinder

corresponds to the contact mate of pad and disc, the motion suddenly stops. As this second condition is achieved, the force F exerted by the actuator is evaluated according to the static equilibrium condition:

$$F(t) = A[p_c(t) - p_a] - K_{\text{spring}}X_{\text{contact}} - F_{\text{pre-load}} \quad (19)$$

This approach introduces an error during piston deceleration transients, because the contribution of the inertial term $M\ddot{X}(t)$ disappears in the transition between equation (18) and equation (19) and there is a discontinuity in the equation. However, its effect on the simulation results can be considered to be negligible.

3.2.2 Brake valve

A driver's brake valve is used to regulate the pressure of the brake pipe CG. Several kinds of valve are modelled in the block library developed. In this work, the authors have focused their attention on the self-regulating cock. This cock has six working positions that corresponds to different pressures of the brake pipe:

1. *Rapid charging position.* The brake pipe is over-fed with a pressure that is higher than the nominal value (6.5 bar; this value is given by the sum of the nominal value defined in the UIC 540-03 regulation [13] and the atmospheric pressure) in order to accelerate release of the brake.
2. *Filling position.* The brake pipe pressure is regulated at the nominal pressure (6 bars).
3. *Low service braking.* A light pressure drop (5.5 bars) is imposed on the brake pipe.
4. *Maximum service braking.* The pressure is regulated to 4.5 bars. Full service braking is the maximum braking power that is achieved with this pressure.

5. *Full service braking.* Some special vehicles and locomotives may have an extra braking level by lowering the pressure to 3.9 bars.
6. *Emergency (rapid) braking.* To obtain a faster response in emergency situation, the brake pipe is connected directly to the atmosphere.

This device is modelled with a lumped capacity of the brake valve CaR and its scheme is shown in Fig. 6.

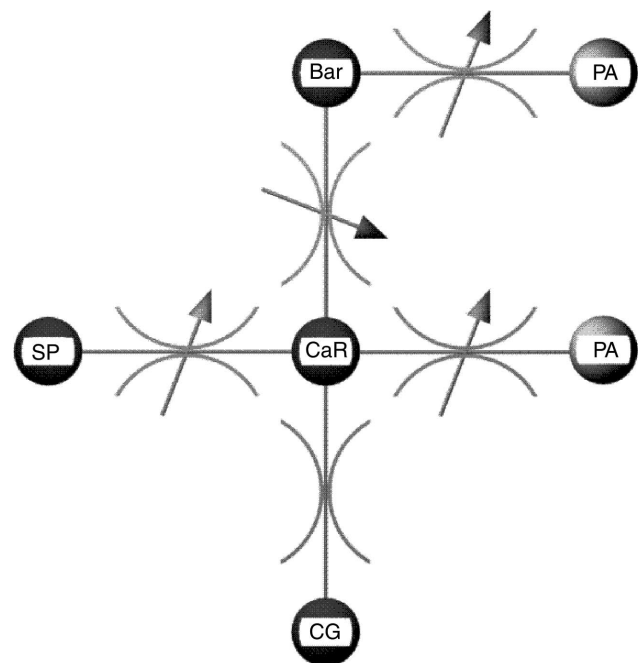


Fig. 6 Simplified pneumatic scheme of the driver's brake valve

1. The CaR element is connected by an orifice to a terminator block PA which sets an outlet pressure equal to the atmospheric pressure. When emergency braking is activated, this orifice is open and the air can leave the pipe. CaR is also connected to three chambers elements.
2. SP represents the lumped capacity of the main reservoir of the locomotive that is fed by a compressor.
3. BaR is the equivalent capacity of the equalizing reservoir, which is connected through a calibrated hole to atmosphere.
4. CG is a chamber element used to connect the valve to the brake pipe.

Orifice equivalent diameters are dynamically regulated by a Matlab–Simulink state-flow chart that simulates the behaviour of the valve according to the selected working position.

3.2.3 Distributor

The pneumatic behaviour of the distributor is modelled as a lumped system of chambers (lumped capacity) and orifices (lumped pressure drop). In Figs 7 and 8 the simplified schemes adopted to model the Westinghouse U distributor are shown. The flow through some orifice elements is regulated by a state-flow chart [14–16], which is able to reproduce the logic of this device.

The distributor model is connected to the pipe block through a *T connector* simulated by a fictitious pipe capacity node. Air flows through the main chamber CaA of the distributor. In addition, the accelerator capacities CaA1 and CaA2 are modelled in order to simulate how this device really speeds up pressure drop propagation along the brake pipe.

CaA is connected to the equivalent capacities of auxiliary (SA) and command (SC) reservoirs. SA and CaA2 are linked to chamber CaC. The latter simulates the equivalent capacity of the distributor chamber (labelled C in Fig. 4) which regulates the brake cylinder pressure (CF). Often distributors are equipped with a secondary pneumatic device called a *pressure transformer* that has two main purposes:

1. Friction between the braking surfaces (block and wheel or pads and disc) decreases as the sliding velocity between the braking surfaces increase. In order to prevent this undesirable phenomenon, pressure transformer behaves like a variable-gain amplifier that corrects the response of the distributor according train speed. If its value is lower than 60 km/h, the pressure of the cylinder is decreased in order to compensate the higher friction factor between the sliding surfaces of the brake.
2. When a pressure transformer is installed, the distributor controls the pressure of a pilot capacity CaP instead of those relative to the brake cylinder CF. The reference signal of CaP is used by the

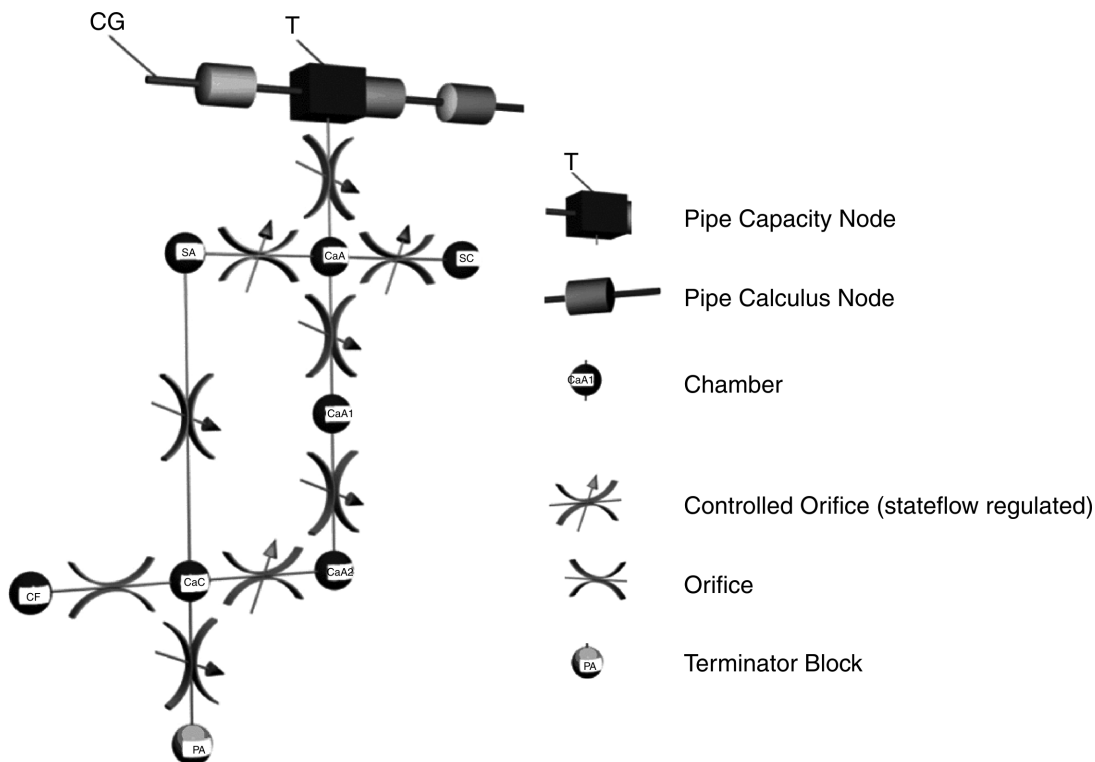


Fig. 7 Simplified pneumatic scheme of the Westinghouse U distributor

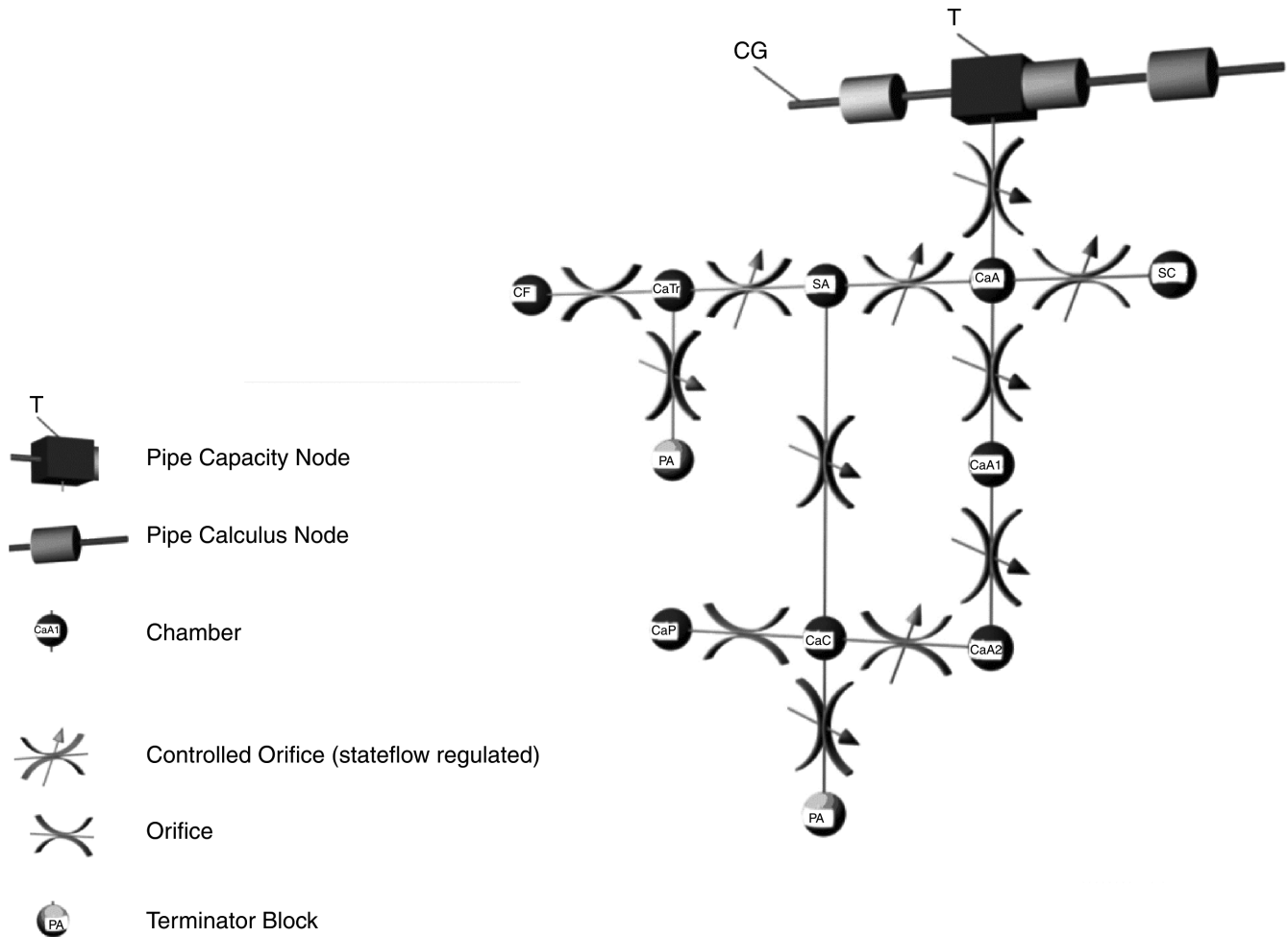


Fig. 8 Westinghouse U distributor and pressure transformer

pressure transformer to regulate the brake cylinder CF using the pressurized air of the auxiliary reservoir SA (the power storage element). So the pressure transformer works as a kind of pneumatic-impedance adapter that makes possible the use of the same distributor on vehicles that have cylinders with different sizes and capacities.

In this work, the pressure transformer is modelled as a set of lumped capacities connected with orifices regulated by a finite state machine implemented with a state flowchart.

3.3 Full vehicle plant

A small library of vehicles that can be used easily to assemble the complete pneumatic plant of a custom-defined train was developed. Figure 9 shows the library elements, while Fig. 10 shows a simple assembly of an

entire train. The library is composed of three kinds of block:

- head vehicles*, which are a set of blocks able to simulate vehicles such as locomotives with an autonomous source of pressurized air and a driver's brake valve to control CG pipe;
- trailer vehicles*, which are vehicles as freight car or passenger coaches that depend on the head vehicles for pneumatic power supply;
- signal vehicles*, which are fictitious vehicles blocks that the user may connect to the assembled train to impose boundary conditions and to perform some complementary computations that need data from every vehicle such as the longitudinal behaviour of the train.

Pneumatic plant of a vehicle is a model that is created by assembling a large number of elementary and complex elements. Each model is completed by a subsystem that calculates the equivalent longitudinal braking force according to the main mechanical features

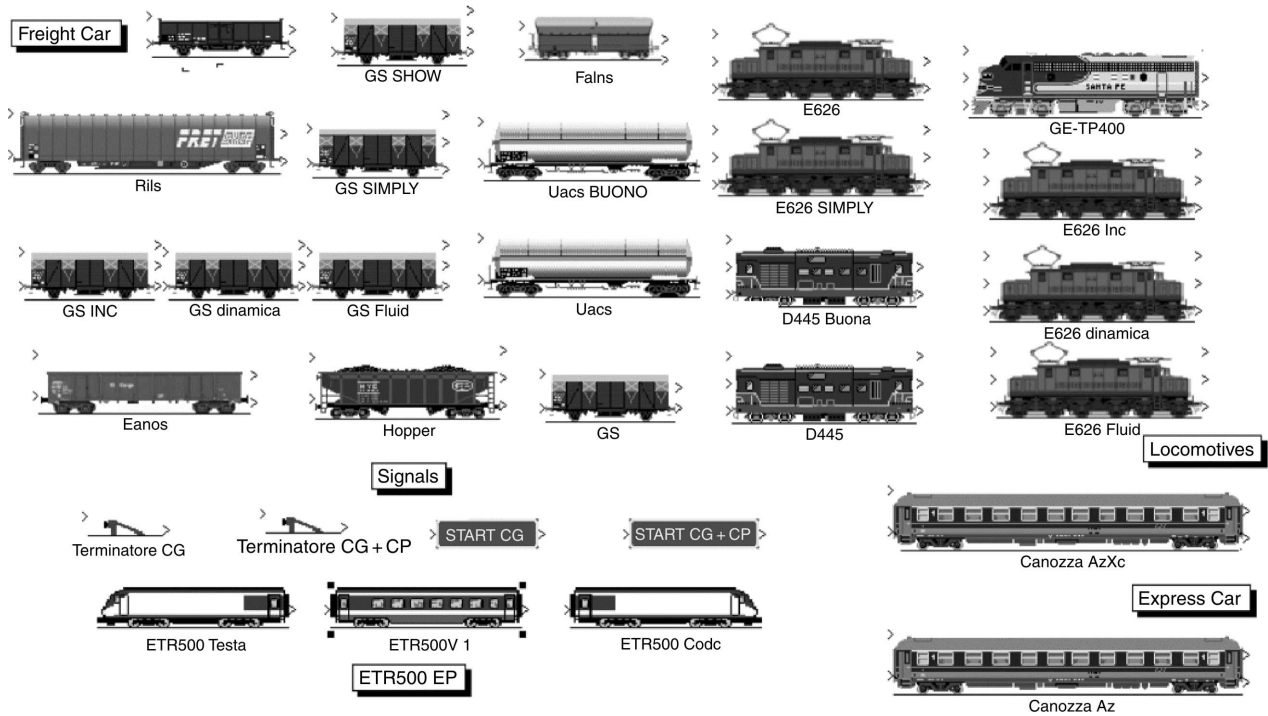


Fig. 9 Complete vehicle plant library

of the vehicle brake:

$$B_i = \begin{cases} n_{\text{wheelset}} n_{\text{pad}} n_{\text{disc}} \tau_b \eta \mu_{\text{pad}} \frac{R_{\text{brake}}}{R_{\text{wheel}}} \sum_{k=1}^{n_{\text{cylinder}}} F_k(t) & (B_i < \mu_{\text{rail}} P_i) \\ \mu_{\text{rail}} P_i & (B_i \geq \mu_{\text{rail}} P_i) \end{cases} \quad (20)$$

The dynamic behaviour of the vehicle suspension system is neglected in order to reduce the computational burden of the system. In order to have a rough estimation of the global braking performances, longitudinal behaviour is analysed by modelling the train as a single rigid body that translates along the line:

$$m_{\text{train}} a = a \sum_{i=1}^{n_{\text{vehicles}}} m_i^{\text{vehicles}} = B_{\text{extras}} + \sum_{i=1}^{n_{\text{vehicles}}} B_i(t) \quad (21)$$

An auxiliary force B_{extras} is introduced to take into account the contribution of various external force that may influence the longitudinal dynamic behaviour of the train:

$$B_{\text{extras}} = [b_{\text{line}} + b_{\text{res}}(v_{\text{train}})] \sum_{i=1}^{n_{\text{vehicles}}} m_i^{\text{vehicles}} + \sum_{i=1}^{n_{\text{vehicles}}} B_i^{\text{electr}}$$

4 FIRST MODEL VALIDATION AND RESULTS

Validation tests have been carried out on both single components and train plants. This approach is justified by a simple reason: errors on single components may be negligible, but performance of the complete plant model may be negatively affected by stability or convergence

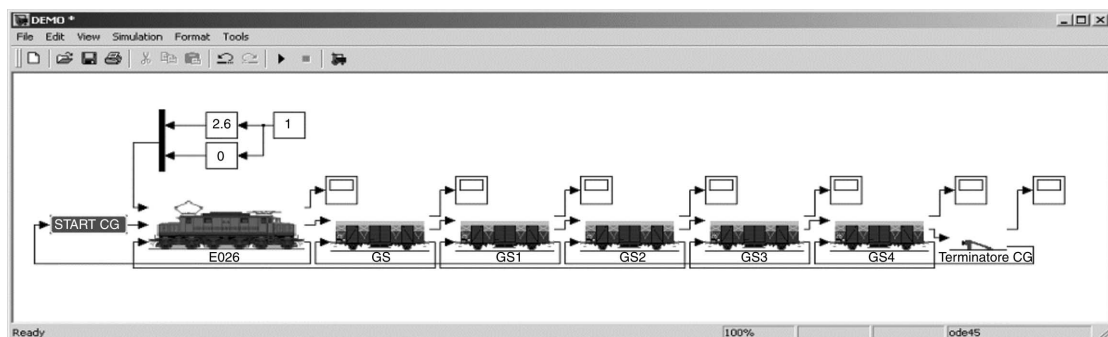


Fig. 10 A complete train model created by assembling a vehicle plant model

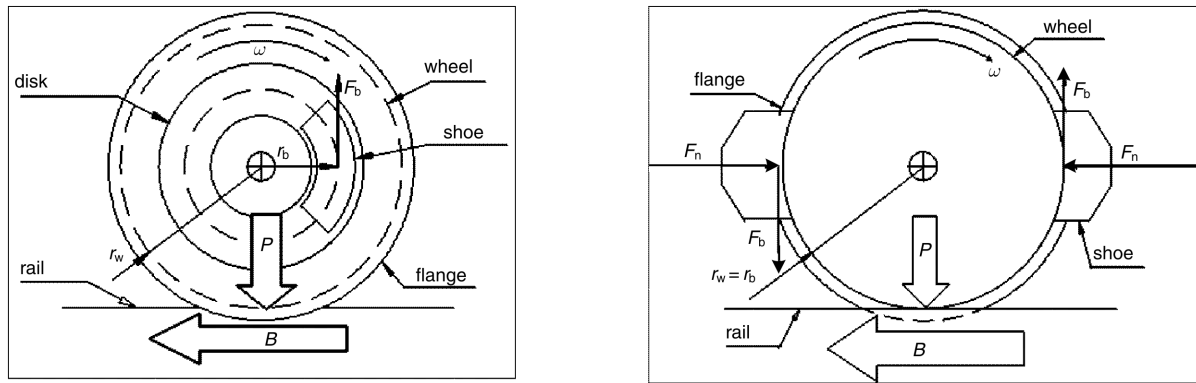


Fig. 11 Disc and block brake

problems due to the numerical stiffness of some decision elements. The distributor is one of the most critical components that have to be tested, and so a wide range of simulations has been performed in order to verify how the simulation results fit the experimental data.

Test and validation procedures have been managed according to the availability of experimental data from Trenitalia SPA, and so these results are referred to specific plant layouts. However, a large number of successful simulations have been performed up to now; therefore the simplified approach followed in this study seems to work correctly.

4.1 Westinghouse U distributor

Trenitalia SPA has a testing facility in Florence (Italy) for experimental activities on pneumatic components. This test rig is a modular system of pneumatic components that can be assembled in order to simulate a wide range of different plant layouts.

Accurate experimental data are available for the Westinghouse U distributor. Reference experimental data have been produced with a layout composed of three Westinghouse U distributors connected to the same brake pipe CG. Every distributor is connected with

a command reservoir (15 litres), an auxiliary reservoir (125 litres) and a brake cylinder. Every distributor is equipped with a pneumatic device called a pressure transformer. At the end of the brake pipe is connected a reservoir of 25 litres in order to increase the equivalent capacity of the plant.

The reference experimental test simulates a rapid emergency braking followed after about 80 s by complete release of the brake. The scheme of the device is shown in Fig. 12.

A Simulink system was assembled in order to validate the pneumatic behaviour of the distributor model. This validation model was composed of the following blocks:

- a *Westinghouse U distributor* with a 'pressure transformer' (the reference pressure signal of the brake pipe CG is an interpolated pressure profile produced from real experimental data produced using the experimental rig described above);
- a *command reservoir* connected to the distributor;
- an *auxiliary reservoir*, that feeds the pressure transformer;
- a *cylinder* filled by the pressure transformer.

Simulation results show good agreement with experimental data (they are summarized in Fig. 13). Higher errors are visible on rapid transitions such as the filling

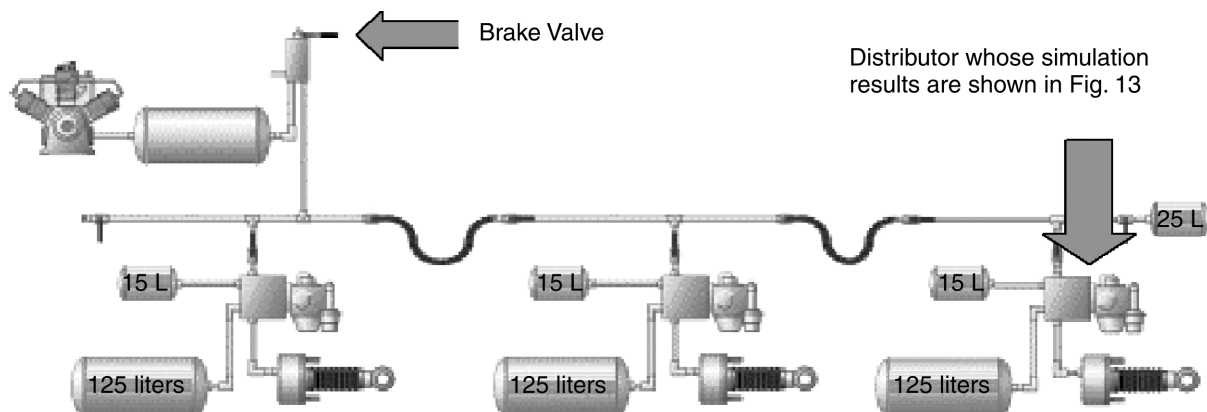


Fig. 12 Scheme of the tested and simulated device

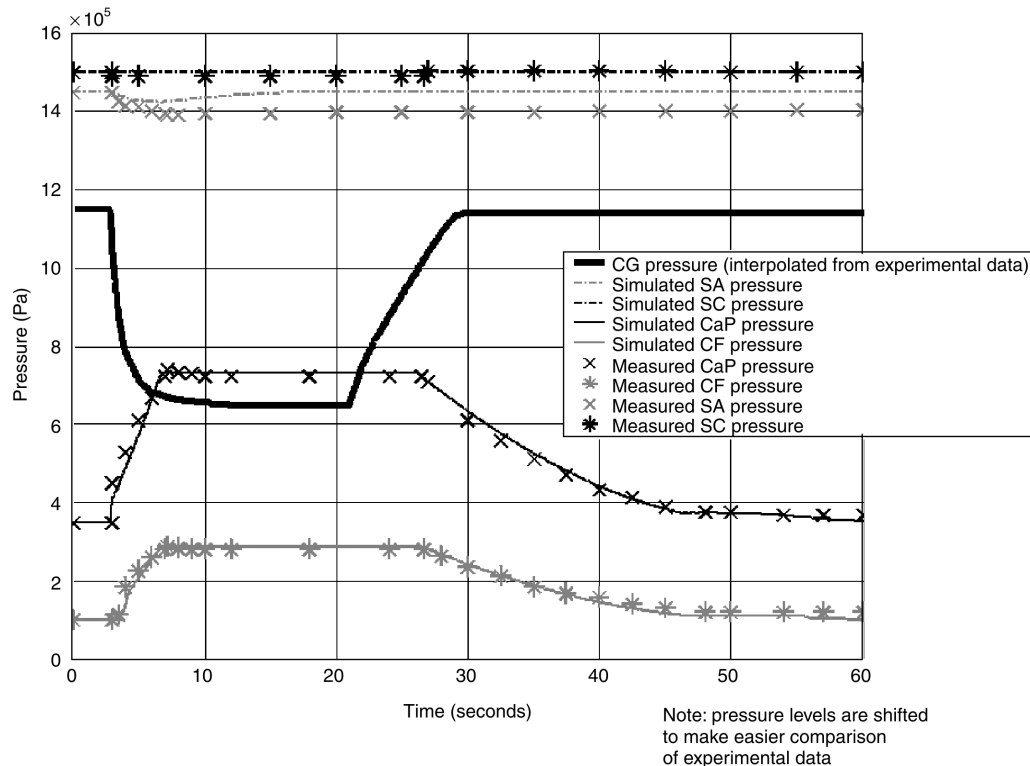


Fig. 13 Comparison between the numerical results and the experimental response of a Westinghouse U distributor

of the accelerator capacities CaA1 and CaA2. However, these errors are compatible with uncertainties in both model and experimental data.

4.2 Complete train simulation

In past experimental campaigns, Trenitalia SPA have made braking tests on special trains equipped with sensors able to measure the pressure of both the brake pipe CG and the cylinder CF of various vehicles. Reference experimental data have been produced on a train with a locomotive and 33 freight cars. The purpose of the experimental test was the release of some locomotive pneumatic components (driver's brake valve). The experimental test reproduces a sequence of emergency manoeuvres: a rapid braking followed by a complete release.

The tests have been performed with several different freight cars equipped with various distributors in order to simulate realistic train layouts. When the present authors performed validation tests, only a reduced set of library elements was validated with experimental data. For this reason the simulation model has been assembled with a reduced set of validated library elements.

The reference simulation model is composed of 33 GS freight cars equipped with Westinghouse U distributors. In Figs 14 and 15, simulation results and experimental

data are compared. Higher errors are visible on the calculated cylinder pressure during the release phase. However, simulation results are quite good according to the known approximations described. The model is able to reproduce typical behaviours of the pneumatic brake:

1. The delay introduced into signal propagation along the brake pipe is modelled with a good approximation.
2. The effect of the accelerator chambers CaA1 and CaA2 is clearly recognizable on both the cylinders and the brake pipe.
3. The gradual release of the brake due to the regulated behaviours of distributors is qualitatively reproduced.

5 CONCLUSIONS AND FUTURE IMPROVEMENTS

This work describes a numerical model that was designed to simulate the behaviour of a pneumatic brake system. The model is composed of a library of elements that can be assembled in order to schematize all the elements of the brake.

Experimental activities are indispensable in order to verify and release pneumatic components of railway brake. However, computational fluid dynamics (CFD) simulation can really contribute to optimizing

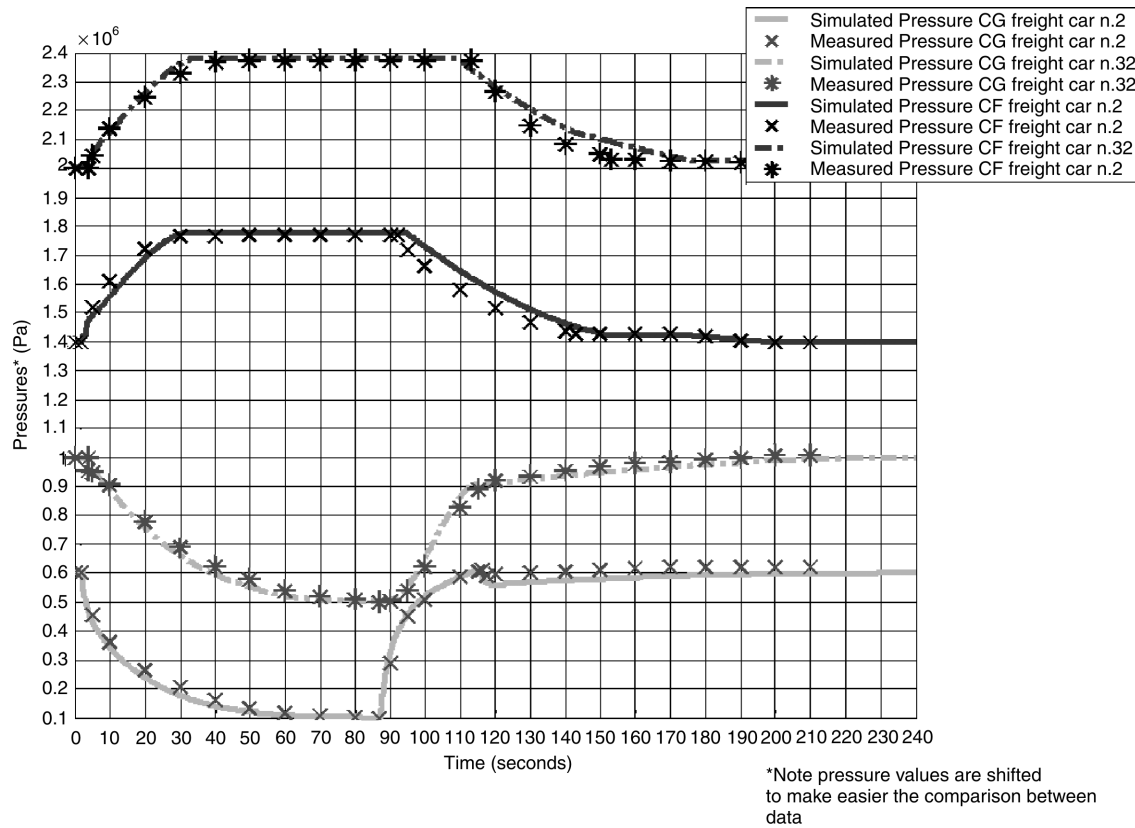


Fig. 14 Comparison between the experimental data and the numerical simulations for a 'full train layout'

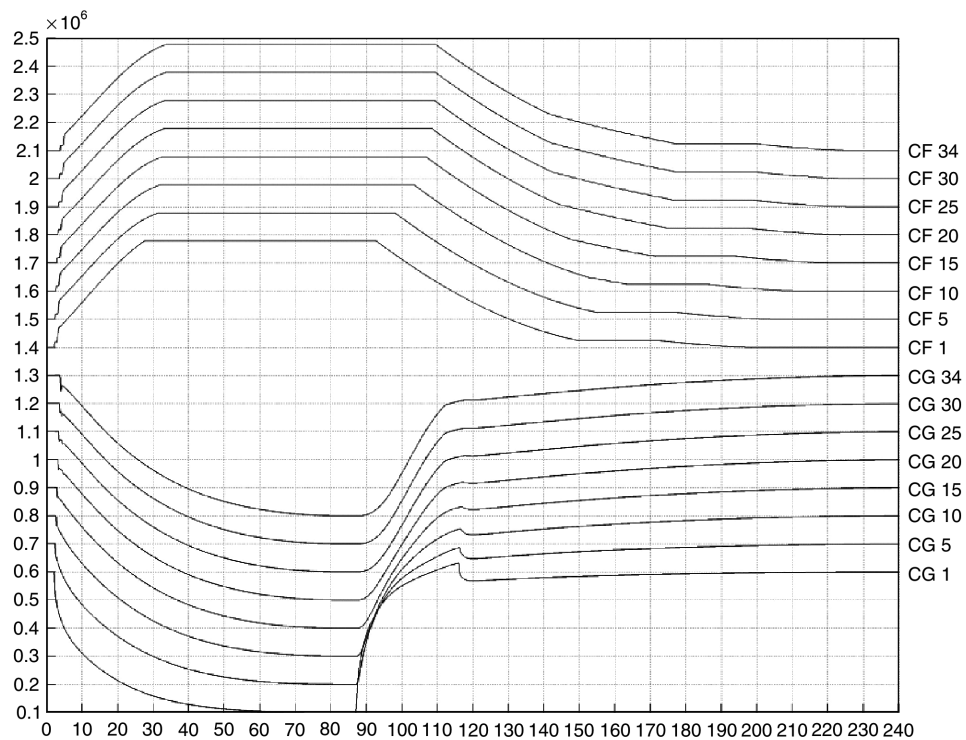


Fig. 15 Simulation results: comparison of the pressures of the brake pipe and cylinders along the train (the pressure curves are shifted in order to make the comparison between data easier)

experimental activities in order to obtain accurate results with lower investments. Once the simulation model has been properly validated by means of a set of experimental data, it may be used in the development and layout of braking systems. A numerical model allows the effect of parameter variations on the plant behaviour to be rapidly investigated and then could be used for optimization processes.

Availability of a fast code prototyping environment such as Matlab-Simulink can really speed up the development of model libraries for custom-defined pneumatic components. Computational speed, user-friendliness and reliability are indispensable requirements for profitable use of CFD tools. The mixed approach proposed in this paper seems to be a reasonable compromise between different requirements.

The finite difference approach applied to a one-dimensional problem assures good simulation of the plant pipes; complex decision elements that may need a heavy computational load are solved through an optimized mix of lumped pneumatic components and finite state machines. The resulting computational load is relatively small and complex plants composed of hundreds of different components can be simulated on a commercial personal computer. Simulation speed can also be increased using an automatic Matlab-C compiler like the Simulink Accelerator. Validation tests show substantial agreement between the numerical simulations and the experimental results obtained by Trenitalia SPA.

The present authors are currently working on feasible improvements to the library:

- (a) a higher number of pre-defined and validated pneumatic components and vehicles;
- (b) further numerical optimization of blocks in order to improve simulation efficiency (speed and accuracy);
- (c) a more sophisticated approach to the dynamic behaviour of the train in order to calculate longitudinal forces generated by different braking forces due to the behaviour of the pneumatic plant.

At the moment, only the pneumatic elements of the plant have been modelled. However, because of the versatility of Matlab-Simulink environment, the electro-pneumatic components could be introduced. The modelling of a electro-pneumatic plant presents a reduced numerical effort compared with a totally pneumatic plant.

REFERENCES

- 1 *Code UIC 540 Frein à Air Comprimé pour Trains de Marchandises et Trains de Voyageurs*, 3rd edition, 1 January 1982 (Union Internationale des Chemins de Fer).
- 2 Union Internationale des Chemins de Fer *Code UIC 547 Frein à air Comprimé Programme—Type d'Essais*, 4th edition, 1 July 1989 (Union Internationale des Chemins de Fer).
- 3 **Matricardi, A. and Tagliaferri, A.** *Il Freno: Corso Multimediale d'Istruzione sul Freno Installato a Bordo Dei Rotabili*, draft version of internal document, 1997.
- 4 **Murtaza, M. A. and Garg, S. B. L.** Brake modelling in train simulation studies. *Proc. Instn Mech. Engrs, Part F: J. Rail and Rapid Transit*, 1989, **203**(F2), 87–95.
- 5 **Murtaza, M. A. and Garg, S. B. L.** Transient performance during a railway air brake release demand. *Proc. Instn Mech. Engrs, Part F: J. Rail and Rapid Transit*, 1990, **204**(F1), 31–38.
- 6 **Murtaza, M. A. and Garg, S. B. L.** Parametric study of railway air brake system. *Proc. Instn Mech. Engrs, Part F: J. Rail and Rapid Transit*, 1992, **206**(F1), 21–36.
- 7 **Murtaza, M. A. and Garg, S. B. L.** Railway air brake simulation: an empirical approach. *Proc. Instn Mech. Engrs, Part F: J. Rail and Rapid Transit*, 1993, **207**(F1), 51–56.
- 8 **Bharath, S., Nakra, B. C. and Gupta, K. N.** Mathematical model of a railway pneumatic brake system with varying cylinder capacity effects. *J. Dynamic Systems, Measmt, Control*, 1990, **112**, 456–462.
- 9 **Hong, T. and Tessmann, R. K.** *The Dynamic Analysis of Pneumatic Systems using HyPneu*, 1996 (BarDyne, Inc.).
- 10 **Hong, T. and Tessmann, R. K.** *A Unified Approach for Design and Analysis of Engineered Systems Using HyPneu*, 1996 (BarDyne, Inc.).
- 11 **IMAGINE S.A.** *AMESim Pneumatic Library, Overview*, 1995 (IMAGINE S.A.).
- 12 **ISO 6538:1989** *Pneumatic Fluid Power—Components Using Compressible Fluids—Determination of Flow Rate Characteristics*, edition G, 21 September 1989 (International Standardization Organization, Geneva, Switzerland).
- 13 *Brakes—Regulations Concerning Manufacture of the Different Brake Parts—Driver's Brake Valve*, UIC Leaflet 541–03, 1st edition, 1 January 1984 (Union Internationale des Chemins de Fer).
- 14 **Math Works** *Matlab-Simulink and Stateflow Manuals and Technical Reference Release 6.1*, 2000 (The Math Works, Inc., Natick, Massachusetts).
- 15 **Harel D.** Statecharts: a visual formalism for complex systems. *Sci. Computer Programming*, 1987, **8**, 231–274.
- 16 **Hatley, D. J. and Imtiaz, A. P.** *Strategies for Real Time Specification*, 1988 (Dorset House Publishing, New York).