IMPLEMENTATION OF ELECTRONICALLY CONTROLLED PNEUMATIC BRAKE FORMULATION IN LONGITUDINAL TRAIN DYNAMICS ALGORITHMS

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ABSTRACT

The main goal of this investigation is to integrate an ECP (electronically controlled pneumatic) brake model with efficient longitudinal train force algorithms based on the trajectory coordinate formulations. The ECP brake model, developed in this investigation consists of the train line (cable), locomotive automatic brake valve, air brake pipe, and ECP manifold. The train line, which covers the entire length of the train, allows the brake commands to be received by the car simultaneously. While pneumatic pressure is used to generate the braking forces, the brake pipe is no longer used to provide the brake level commands. Instead, the brake pipes are used to provide a continuous supply of compressed air stored in a reservoir mounted on each railcar. Using the ECP system to apply the brakes uniformly and instantaneously gives better train control, shortens the stopping distances, and leads to a lower risk of derailment. In this investigation, the fluid continuity and momentum equations are used to develop the governing air pressure flow equations. These partial differential equations are converted to a set of ordinary differential equations using the finite element (FE) method leading to an air brake force model that accounts for the effect of the air flow in long train pipes as well as the effect of leakage and branch pipe flows. The car brake forces are applied to the wheels using the ECP manifold located in each car. The ECP manifold used in this investigation has four valves; cut-off valve, vent valve, auxiliary valve, and emergency valve. The ECP manifold is connected to three main pneumatic components; the auxiliary reservoir, the emergency reservoir, and the brake cylinder. The reservoirs serve as the main storage of the pressurized air, while the brake cylinder and other mechanical components such as the rigging and the brake shoes transmit the brake force to the wheels. In this investigation, a mathematical model of the ECP manifold and its components is developed. The relationship between the main components of the ECP brake system and the train dynamics is discussed, and the final set of differential equations that integrates the ECP brake and train dynamics is presented. Different simulation scenarios are considered in this study in order to investigate the effect of the brake forces on the train longitudinal dynamics in the case of different braking scenarios. The performance of the developed ECP brake system are compared with the Association of American Railroads (AAR) safety and operation standards, and with experimental results published in the literature.

Keywords: Freight train braking, ECP brake system, nonlinear longitudinal train dynamics, in-train forces, multibody system dynamics.
1. INTRODUCTION

The influence of the braking forces on longitudinal dynamics and stability of railroad vehicles is significant. In conventional air braking systems, originally developed in the nineteenth century, the braking command is conveyed from the engine to the vehicles through a pressure drop signal that propagates along the train. As a consequence, braking forces are applied sequentially on each railcar causing the rear railcars to run into the railcars in front of them producing large coupler forces. This effect is particularly significant in freight trains that can be more than a mile in length. The coupler longitudinal compressive and tensile forces are the cause of the significant rolling stock and track repair cost, long stopping distances, and compromising train operation safety. The use of a braking system that allows applying all brake forces simultaneously can improve both train safety and operations by reducing in-train forces, making train handling simpler, providing shorter stopping distances regardless of the train length, and reducing coupler impacts during the brake applications.

During the past 15 years, there has been a slow progress in adopting a relatively new technology for controlling the brake system on freight trains. The new technology is called electronically controlled pneumatic braking commonly referred to as ECP braking. ECP braking uses electronic controls which make it possible to activate air-powered brakes on all the train cars at the same time. Applying the brakes uniformly and instantaneously in this way gives better train control, shortens the stopping distances, leads to a lower risk of derailment and/or coupler failure, improves train service cycle times, increases network capacity, enhances rail safety, reduces fuel consumption, and results in less wear and tear (Smith and Carlson, 2009).

The functional and performance requirements of the ECP brake system are given by AAR standard S-4200 (1999) which describes an ECP brake system as a train power brake.
system actuated by compressed air and controlled by electronic signals originated at the
locomotive for service and emergency applications. The system provides almost instantaneous
response to braking commands, including graduated brake releases and reapplications. The
system responds appropriately to undesired separation or malfunction of hoses, cabling, or brake
pipe (Sich, 2002). The brake pipe is still used to provide a continuous supply of compressed air
to be stored in a reservoir on each railcar. A rapid reduction of brake pipe pressure to zero will
still provide a pneumatic emergency brake application through the train. Both electrical power
and communications are transmitted via an electric cable which is known as the train line and
spans the entire length of the train (Formenton, 2009; AAR standards S-4200, 1999).

The main objective of this investigation is to develop an accurate ECP brake model and
integrate this model with an efficient nonlinear multibody system (MBS) train dynamics
algorithm based on the trajectory coordinate formulation. The first part of the paper describes
the main components and the operation of the ECP system. The ECP system model is then
presented focusing on the equations describing the dynamics of the pressurized air inside the
pipes, reservoirs, brake cylinders, and across the electronically controlled valves. Validation and
verification of the proposed ECP model will be discussed by comparing the obtained numerical
results with the requirements and experimental data taken from standards and technical literature.
In order to demonstrate the benefits of the ECP brake system, the performances of the
conventional air brake model developed by Specchia et al (2011) and Afshari et al (2011) and the
proposed ECP brake model are compared.

2. ECP TRAIN LAYOUT DESCRIPTION
AAR standard S-4200 (1999) provides a detailed description of all the devices used in an ECP brake system. In this section, a brief description of each major component and its purpose and functions is provided. A schematic diagram of such an EPC system is shown in Fig. 1. The *Head End Unit* (HEU), mounted within the locomotive cab, is the ECP brake system control device. It is used by the train operator for the application and release of train brakes, and allows the operator to manage and control various ECP components in the train. The *Power Supply Controller* (PSC), which is connected to the train line communication network and controls the train line power supply as commanded by the HEU, provides electrical power to all connected Car Control Device (CCD) and End-of-Train (EOT) units.

The *Locomotive ID Module* (LID) contains locomotive specific data which are provided to the HEU and PSC during the ECP operation. Similar to the LID, the *Car ID Module* (CID) stores railcar specific data used by the *Car Control Device* (CCD) for controlling the brakes on a vehicle correctly. The CCD is a device used to control all the pneumatic valves needed for the application and release of brakes. The *End-of-Train* (EOT) Device, designed to communicate with the HEU, is connected to the train line at the end of the train. The EOT is the last network node in the train and transmits, at a rate of once per second, a status message indicating that the train is intact.

### 3. ECP CAR LAYOUT DESCRIPTION

As shown in Fig. 2, several components are used in a car equipped with the ECP brake system including the auxiliary reservoir, emergency reservoir, brake cylinder, Car Control Device (CCD), an ECP manifold which contains the pneumatic valves, Battery, Car ID Module (CID), and pressure transducers. The main characteristic of an ECP system is that the information of a
desired control commanded from the HEU is conveyed through a cable called ECP train line. The same line is used also for charging the battery on board. This signal is designed to be easily understood by the microprocessor inside the CCD.

4. ECP OPERATION

Before presenting the mathematical model of the ECP brake system, it is important to understand the operation of the ECP brake system. A freight train typically includes one or more locomotives, a certain number of railcars, and several train lines. For a freight train led by a locomotive equipped with an ECP brake control system, the train lines include both pneumatic and electrical lines some of which run from the lead locomotive to the last rail vehicle in the train. In the following subsections the ECP brake modes, communications between different ECP units, as well as ECP brake application scenarios will be discussed. The description of the ECP operation presented in this section is a summary obtained from a published AAR report (American Association of Railroads, 1999).

4.1 ECP Brake Modes

The ECP brake control system in the locomotive includes a cab station unit and a master controller from which the brakes on the train are controlled (HEU) (American Association of Railroads, 1999). The train operator uses automatic brake handle in the cab station to directly control the brake application and release. The automatic brake handle, can be moved to and between several positions: release, minimum service, full service, suppression, and emergency. The brake can be incrementally increased or decreased between the minimum and full service position.

4.2 Communication between ECP Brake Units
As shown in Fig. 2, the ECP brake equipment on each rail vehicle typically includes a car control unit (CCU), several pressure transducers, various pneumatic and electro pneumatic valves, an auxiliary reservoir, an emergency reservoir, and at least one brake cylinder. The pressures in the brake pipe, the brake cylinders, and the two reservoirs are monitored by pressure transducers, which send to the CCU electrical signals with the pressure levels in each of them. On the other hand, the CCU includes a transceiver and a microprocessor. The transceiver converts the electrical brake commands into a form usable by the microprocessor. According to the brake commands and other electrical signals received by the microprocessor, it controls the electro pneumatic valves in a way similar to what is used in the existing airbrake control systems. Through controlling these electro pneumatic valves, a certain brake command can be applied. As shown in Fig. 3, four solenoids are mounted on the top of the cover plate. These solenoids are designed to control the flow of the pressurized air between different components of the ECP manifold based on the brake mode (Sich, 2002).

4.3 ECP Brake Applications

By moving the automatic brake handle into the service zone, for example, ECP brake control system issues a service-brake command along the ECP train line. The microprocessor on each railcar then routes the appropriate amount of air from the auxiliary reservoir and/or the emergency reservoir, through the appropriate electro pneumatic valve to the brake cylinder. Firstly, by selecting which brake control system will directly control the brakes, the train operator in the locomotive ultimately determines the state of the cut-out valve insert in ECP manifold. When the conventional braking mode is selected, the CCU on each railcar is not commanded via the ECP train line to energize the solenoid of the cut-out valve. This leaves cut-out valve in its cut-out state, with its lower valve closed and its upper valve open (Sich, 2002).
Consequently, whenever the conventional brake control system sends a brake application command along the brake pipe, the cut-out valve will allow pressurized air to flow through its upper valve and into the brake cylinders (conventional airbrake system). On the other hand, when the ECP braking mode is selected, the CCU on each railcar receives a signal via the ECP train line to energize the solenoid. In this case, the cut-out valve disconnects the brake cylinder from the passage in the service portion of the brake control valve (Sich, 2002). Furthermore, in the ECP braking mode, the brakes are released or applied only by exhausting or pressurizing the brake cylinder through the vent, auxiliary, and emergency valve. The CCU on each railcar receives the brake release and application commands via the ECP train line and energizes the solenoid above the appropriate valve insert.

As described by Sich (2002), whenever the ECP brake control system sends a brake application command along the ECP train line, for example, the CCU will energize the solenoid above either auxiliary valve or emergency valve or both. Pilot air from the auxiliary reservoir then acts against the top of the piston. As the pilot pressure builds, the piston moves downward against the spring force. This forces the emergency valve into its energized state, with its upper valve closed and its lower valve open. In this state, the emergency insert allows pressurized air to flow from the emergency reservoir through its lower valve and into the brake cylinder. On the other hand, in case of brake release command, the ECP brake control system sends a brake release command along the ECP train line; the CCU will energize the solenoid above vent valve. Consequently, pilot air from the auxiliary reservoir acts against the top of insert piston, as the pilot pressure builds, the piston moves downward against the force of the spring. This forces the vent valve into its energized state, where its upper valve will be closed and its lower valve will
be opened. In this state, the vent allows the pressurized air previously developed within the brake cylinder to flow through its lower valve to the atmosphere.

It is important to mention that, for safety purpose, emergency brake commands are conveyed to the railcars not only along the ECP train line but also pneumatically along the conventional airbrake system using the bake pipe. Moving the handle into the emergency position causes the pressure in the brake pipe to drop at an emergency rate. This drop in pressure then quickly propagates along the brake pipe to each railcar in the train. If the ECP equipment loses power or otherwise fails electrically, it will still respond pneumatically to the reduction in pressure that occurs in the brake pipe during an emergency. The ECP brake equipment is designed to respond to the emergency pressure drop by supplying pressurized air from both the auxiliary and emergency reservoirs to the brake cylinder and thereby cause an emergency application of the brakes.

As described by the American Association of Railroads (AAR) standard (1999), the brake application level applied by the ECP system is determined by the train brake commands (TBC). The TBC is expressed as a percentage from 0% to 100% of full-service braking force in 1% increments where a 100% TBC means a full-service application, and 0% means a full release application. The minimum TBC service is a 10% of the full-service brake force; which does not necessarily mean the brake cylinder pressure is a 10% of the full-service brake cylinder pressure, but it means the pressure in the brake cylinder is sufficient to get the brake blocks to be on contact with the wheel treads. The emergency brake application (whether intentional or as a result of a fault or penalty) results in a TBC of 120% of the full-service brake cylinder pressure setting. The CCD will continue to monitor the brake cylinder pressure and maintain the required brake cylinder pressure against leakage, during the brake application. As result of a brake
application, immediately the reservoirs on a railcar will be recharged. Also, the CCD will continue monitoring the brake pipe, reservoirs, and brake cylinder pressures. HEU sends a message every second to all network devices in the train, and in the same time receives an updated status report from each CCD. For safety purpose operations of the ECP system, the EOT sends a beacon message every second; in case a CCD confirmed a loss of 3 consecutive EOT beacon messages and other two CCDs confirmed a similar loss, the devices will issue a critical loss message to all other devices and activate an emergency brake application, while the HEU will command an ECP emergency brake application.

5. AIR BRAKE SYSTEM MATHEMATICAL MODEL

In this section the mathematical model for the air brake system developed by Specchia et al. (2011) and Afshari et al. (2011) is briefly reviewed. The air brake model, presented in this paper, consists of the locomotive automatic brake valve, air brake pipe, and car control unit (CCU) as shown in Fig. 3. The proposed air brake force model accounts for the effect of the air flow in long train pipes as well as the effect of leakage and branch pipe flows. For the air brake pipe model, the governing equations of the air pressure flow are developed using the general fluid continuity and momentum equations, simplified using the assumptions of one dimensional isothermal flow. Using these assumptions, one obtains two coupled air velocity/pressure partial differential equations that depend on time and the longitudinal coordinate of the brake pipes. The partial differential equations are converted to a set of first order ordinary differential equations using the finite element (FE) method. The resulting air brake ordinary differential equations are solved simultaneously with the train second order nonlinear dynamic differential equations of motion that are based on the trajectory coordinates.
The air pressure in the brake pipe system is controlled by the brake locomotive automatic brake valve. The mathematical model of the 26C locomotive valve developed by Abdol-Hamid (1986) is used in this investigation. In a conventional air brake model, the primary function of the automatic brake valve is to control air pressure in the brake pipe allowing for the application or release of the train and locomotive brakes. For the ECP brake model, the primary function of the automatic brake valve is to provide a continuous supply of compressed air to be stored in auxiliary and emergency reservoirs on each railcar. As shown in Fig. 4, the main components of the 26C automatic brake valve are the regulating valve, relay valve, brake pipe cut-off valve, vent and emergency valves, and suppression valve. Detailed description of the function and the mathematical model for each of these components can be found in the work of Abdol-Hamid (1986), Specchia et al. (2011), and Afshari et al. (2011).

5.1 Air Flow Equations

In this section, the basic continuum mechanics equations used in this investigation to study the air flow dynamics in train brake pipes are presented. These equations include the momentum, continuity, and constitutive equations. Using the divergence theorem, the continuity equation can be written as \[ \frac{\partial \rho}{\partial t} + \nabla (\rho \mathbf{v}) = 0. \] In this equation, \( \rho \) is the mass density, \( \mathbf{v} \) is the velocity vector, \( \nabla \) is the divergence operator. Since air flow is considered in this study, the density \( \rho \) cannot be treated as a constant, and therefore, the assumption of incompressibility is not used in this study.

Using the assumption of Newtonian fluid (shear stress is proportional to the rate of shear strain) and the continuum-mechanics partial differential equations of equilibrium, one obtains the following system of equations (Specchia et al., 2011):
\[
\frac{\partial (\rho \mathbf{v})}{\partial t} + (\rho (\nabla \mathbf{v}) + \nabla (\rho \mathbf{v})) \mathbf{v} = \{-\nabla (p I) + \lambda \nabla (\text{tr} (\mathbf{D})) + 2\mu \nabla (\mathbf{D})\}^T + \mathbf{f}_b
\]  

(1)

where \( p \) is the hydrostatic pressure, \( \mathbf{f}_b \) is the vector of body forces per unit volume, and \( \lambda \) and \( \mu \) are viscosity coefficients that depend on the fluid density and temperature. This equation governs the motion of general isotropic fluid in the special case of Newtonian fluid.

### 5.2 One-Dimensional Model

The assumption of one-dimensional air flow used in this study implies that the air flows in only one direction along the longitudinal axis of the train pipe, that is, the effect of the velocity components in the other directions are neglected. Such an assumption is justified because of the small pipe cross section dimensions compared to the pipe length. The magnitude of the flow velocity is also assumed to be uniform at any cross section, and consequently, shear stresses are neglected, leading to zero off-diagonal elements of the Cauchy stress tensor. Using the assumption of isothermal flow, one has the relationship \((p/\rho) = R_g \Theta\) (White, 2008), where \( R_g \) is the gas constant which has units J/(Kg\(^{\circ}\)K), and \( \Theta \) is the local temperature (\(^{\circ}\)K). This isothermal flow relationship can be used to eliminate the air density \( \rho \) from the continuity and momentum equations leading to the following coupled system of pressure/velocity equations (Specchia et al., 2011):

\[
\begin{align*}
\frac{\partial p}{\partial t} + \frac{\partial (pu)}{\partial x} + \gamma_r L &= 0 \\
\frac{\partial (pu)}{\partial t} + \frac{\partial (pu^2)}{\partial x} + \gamma_r \frac{\partial p}{\partial x} &= \gamma_r f_b
\end{align*}
\]

(2)

In this equation, \( \gamma_r = R_g \Theta \). Given the boundary and initial conditions, the preceding system of coupled partial differential equations can be solved for the pressure and velocity distributions using numerical methods as discussed in the following section.
5.3 FE Equations

Using the change of variables \( q = pu \) in the partial differential equations of Eq. 2, one obtains

\[
\begin{align*}
\frac{\partial p}{\partial t} + \frac{\partial q}{\partial x} &= -\gamma, L \\
\frac{\partial q}{\partial t} + \frac{\partial (qu)}{\partial x} + \gamma, \frac{\partial p}{\partial x} &= \gamma, f_b
\end{align*}
\]

(3)

These two partial differential equations are converted in this investigation to a system of ordinary differential equations using the FE method. The brake pipe is assumed to consist of \( m \) one-dimensional finite elements, and the domain of each element is defined by the spatial coordinate \( x = x^e, 0 < x^e < l^e \), where \( l^e \) is the length of the finite element. Over the domain of the finite element, the variables \( p \) and \( q \) are interpolated using the following field equations

\[
p^e(x,t) = S_p^e p^e, \quad q^e(x,t) = S_q^e q^e
\]

where \( e = 1, 2, \cdots, m \), \( S_p^e \) and \( S_q^e \) are appropriate element shape functions, and \( p^e \) and \( q^e \) are the vectors of nodal coordinates. While linear interpolation is assumed in the numerical study presented in this paper, higher orders of interpolations can also be used. Multiplying the first equation in Eq. 3 by the virtual change \( \delta p^e \) and the second equation by the virtual change \( \delta q^e \), integrating over the volume, using the relationship

\[
dV^e = A^e dx^e, \quad A^e \text{ is the cross section area; and using the FE assumed displacement field; one write the element first order ordinary differential equations as}
\]

\[
M^e \ddot{e}^e = Q^e
\]

(4)

The components of the coefficient matrix \( M^e \) and the vector \( Q^e \) are calculated as described in the literature (Specchia et al 2011). Assembling the FE equations, one obtains the first order
ordinary differential equations of the brake pipe system which can be written as $\mathbf{M}\dot{\mathbf{e}} = \mathbf{Q}$, where $\mathbf{e}$ is the vector of the system nodal coordinates, $\mathbf{M}$ is the brake pipe global coefficient matrix that results from assembling the $\mathbf{M}^e$ element matrices, and $\mathbf{Q}$ is the right hand side vector that results from the assembly of the $\mathbf{Q}^e$ element vectors.

6. CAR CONTROL UNIT MATHEMATICAL MODEL

The number of the car control units depends on the number of cars in the train, as depicted in Fig. 5. For the ECP brake system, the CCU includes various pneumatic and electro-pneumatic valves (ECP manifold), an auxiliary reservoir, an emergency reservoir, and at least one brake cylinder as shown in Fig. 6. The brake modes depend on the signal received by the CCD from the HEU as previously mentioned. The brake cylinder is the component where the pressurized air can push the cylinder piston to produce the axial force required for the brake application. The brake rigging is a leverage mechanism that transmits the piston axial force and converts it to magnified normal forces that press the brake shoes against the wheel to produce the required friction brake forces (Afshari et al., 2011). In order to study of different brake operation modes, models of the air flow between different CCU sections must be developed. These models can be defined using a set of first order ordinary differential equations that are combined with the automatic brake valve and the brake pipe differential equations in order to define the air brake mathematical model which can be integrated with the nonlinear train dynamic model that employs the trajectory coordinates.

6.1 Flow and Pressure Rates

In this investigation, the average density approach will be used to calculate the mass flow rate (Abdol-Hamid, 1986). The total mass flow rate to a component $e$ is denoted by $\frac{d\mathbf{m}_e}{dt} = \dot{m}_e$,.
while the mass flow rate from component $f$ to component $e$ is denoted by $\dot{m}_{e-f}$. In this paper, the letters $bp, x, c, u, em$, and $a$ are used to denote, respectively, the brake pipe, the auxiliary reservoir, the brake cylinder, the control unit, emergency reservoir, and the atmosphere. A schematic of the airflow between the above-mentioned components is shown in Fig. 7. The airflow to the auxiliary and emergency reservoirs of car $i$ from the brake pipe ($e = x, em$ and $f = bp$) is obtained using the following equation:

\[
\dot{m}_{(x-bp)_i} = 0.6 A_{(x-bp)_i} P_{ix} \sqrt{\frac{r_i^2 - 1}{R_g \Theta} \frac{|r_i - 1|}{r_i - 1}}
\]

\[
\dot{m}_{(em-bp)_i} = 0.6 A_{(em-bp)_i} P_{em} \sqrt{\frac{r_i^2 - 1}{R_g \Theta} \frac{|r_i - 1|}{r_i - 1}}
\]

where $r_i = P_{bp}(x_u,t)/P_{ix}$ ($e = x, em$), $P_{bp}(x_u,t)$ is the local brake pipe pressure that is function of the location of the CCU connection point with the pipe $x_u$ and time $t$, and $A_{(x-bp)_i}$ and $A_{(em-bp)_i}$ are the equivalent areas connecting the pertinent components (Afshari et al., 2011). For brake application, the airflow from the auxiliary and emergency reservoirs to the brake cylinder ($e = c$ and $f = x, em$) can be obtained from the following equation:

\[
\dot{m}_{(c-x)_i} = 0.6 A_{(c-x)_i} P_{ix} \sqrt{\frac{r_i^2 - 1}{R_g \Theta} \frac{|r_i - 1|}{r_i - 1}}
\]

\[
\dot{m}_{(c-em)_i} = 0.6 A_{(c-em)_i} P_{em} \sqrt{\frac{r_i^2 - 1}{R_g \Theta} \frac{|r_i - 1|}{r_i - 1}}
\]

where $r_i = P_{fi}/P_{vi}$ ($f = x, em$), and $A_{(c-x)_i}$ and $A_{(c-em)_i}$ are the equivalent connecting areas. For brake release, the air vented to the atmosphere ($e = a$) from the brake cylinder ($f = c$) is governed by the equation
\[
\dot{m}_{(a-c)_{i}} = 0.6A_{(a-c)_{i}}P_a \left( \frac{r_i^2 - 1}{r_i - 1} \right) 
\]

where, \( r_i = P_{c_i}/P_a \), and \( A_{(a-c)_{i}} \) is the equivalent areas of the components of car \( i \).

Assuming that all the processes taking place in the CCU are isothermal (\( \Theta \) is constant), one can obtain the time rate of the auxiliary reservoir pressure of car \( i \). Using the fact that the volumes of the auxiliary and emergency reservoirs are constant, one can write as

\[
\frac{dP_{x_i}}{dt} = \frac{1}{V_{x_i}} \left( R_g \Theta \frac{dm_{x_i}}{dt} - P_{x_i} \frac{dV_{x_i}}{dt} \right) = \frac{R_g \Theta}{V_{x_i}} \dot{m}_{x_i} 
\]

\[
\frac{dP_{em}}{dt} = \frac{1}{V_{em}} \left( R_g \Theta \frac{dm_{em}}{dt} - P_{em} \frac{dV_{em}}{dt} \right) = \frac{R_g \Theta}{V_{em}} \dot{m}_{em} 
\]

Similarly, the time rate of the brake cylinder pressure is given by

\[
\frac{dP_{c_i}}{dt} = \frac{1}{V_{c_i}} \left( R_g \Theta \frac{dm_{c_i}}{dt} - P_{c_i} \frac{dV_{c_i}}{dt} \right) = \frac{R_g \Theta}{V_{c_i}} \dot{m}_{c_i} 
\]

Unlike the auxiliary reservoir, the brake cylinder, in general, does not have a constant volume because of the piston movement. However, during the brake shoe force application, one can assume that the cylinder volume remains constant. The CCU parameters previously discussed, depending on the brake mode, can be function of time, the car location in the train, and other air brake parameters.

In this investigation, \( \dot{m}_{bp_{i}} \) is the rate of the pipe mass flow through the CCU of car \( i \) to its auxiliary reservoir, or to emergency reservoir. In the model developed in this study, \( \dot{m}_{bp_{i}} \) is the parameter that relates the brake pipe FE model to the CCU of car \( i \). In order to account for the effect of \( \dot{m}_{bp_{i}} \) in the FE model, the element where the car control valve is connected to the brake
pipe is determined. Then, $m_{bp}$, is distributed between the two nodes of the element and is considered as the element leakage in the equations presented by Specchia et al. (2011).

6.2 ECP Manifold Mathematical Model

The most relevant part of the model will be the formulation of the electro-pneumatic valves that are included in the ECP manifold as shown in Fig. 3. For modeling the valves, it is helpful to recall the equations of the mass flow rates, the pressure rates, and the connection areas presented previously in this paper. Variables associated with the cut-off valve are marked with number $1$, for the vent valve, $2$ for the auxiliary valve, and $4$ for the emergency valve.

**Cut-Out Valve Mathematical Model**

The main function of the cut-out valve, shown in Fig. 8a, is to enable or disable the ECP brake system on a single car. This valve is controlled by a solenoid valve which has only two possible positions, on or off. The solenoid valve controls the flow of air from the auxiliary reservoir to the upper chamber of the valve cylinder, this air flow passes through a small passage ($A_{i,2}$). The fluid network for the valve is shown in Fig. 8b. If the solenoid position is off the area $A_{i,1}$ is zero, which means that the passage is closed and the mass flow rate is also zero. When the position of the solenoid valve is off, the task of the valve is to connect the auxiliary reservoir, after a passage in the pneumatic control valve, with the brake cylinder. If the position of the solenoid valve is on the cut-off valve has to close the passage, with area $A_{i,2}$, which connects the pneumatic control valve with the brake cylinder, thereby enabling the ECP brake system. If the solenoid position is on, the area $A_{i,1}$ is not zero and in this case it is possible to determinate the air mass flow rate from the equation

$$\dot{m}_{aux \rightarrow 1} = 0.6 A_{i,1} P_1 \sqrt{ \frac{r^2 - 1}{r - 1} \frac{r - 1}{R_g \theta}}$$

(10)
where \( r = \frac{P_{aux}}{P_1} \). As the pressure in the chamber \( P_1 \) increases, the piston starts to move. In this case, the pressure rate is governed by the equation

\[
\frac{dP_1}{dt} = \frac{1}{V_1} \left( R_e \theta \frac{dm_{aux \rightarrow 1}}{dt} - P_1 \frac{dV_1}{dt} \right)
\]  

(11)

While \( V_1 \) is not a constant volume, the cylinder cross section is constant. It is clear that 

\[
V_1 = V_{1,0} + S_1 x_1(t),
\]

and therefore, the pressure rate equation becomes

\[
\frac{dP_1}{dt} = \frac{1}{V_{1,0} + S_1 x_1(t)} \left( R_e \theta \frac{dm_{aux \rightarrow 1}}{dt} - P_1 S_1 \frac{dx_1(t)}{dt} \right)
\]  

(12)

In order to determine the displacement \( x_1(t) \) and its derivative, the free body diagram of the piston shown in Fig. 8c is used to write the following equilibrium equation:

\[
m_1 \ddot{x}_1 = P_1 S_{1,1} - P_a S_{1,2} - P_{aux} S_{1,3} - kx_1 - L_1
\]  

(13)

where \( k \) is the stiffness of the spring of the valve, \( L_1 \) is the preload force of the spring, \( S_{1,1} \) is the area of the piston that faces the upper chamber, \( S_{1,2} \) is the area of the middle part of the piston, \( S_{1,3} \) is the smaller area of the piston in the lower chamber, and \( m_1 \) is the mass of the piston. Because the mass of the piston and its displacement are small, one can assume that the inertial force \( m_1 \frac{d^2x_1(t)}{dt^2} \) is negligible. Using this assumption, an approximation of \( x_1 \): can be obtained as 

\[
x_1 = \left( P_1 S_{1,1} - P_a S_{1,2} - P_{aux} S_{1,3} - L_1 \right) / k,
\]

and its derivative as 

\[
\dot{x}_1 = \left( \dot{P}_1 S_{1,1} - \dot{P}_{aux} S_{1,3} \right) / k.
\]

One can then write the rate of the mass flow from the auxiliary reservoir to the brake cylinder as

\[
\dot{m}_{aux \rightarrow bc} = 0.6 A_{1,2} P_{bc} \sqrt{\frac{\left| r^2 - 1 \right|}{R_e \theta r - 1}}
\]  

(14)
where \( r = P_{aux} / P_{be} \). Note that \( A_{1,2} = \pi D_1 (x_{1,\text{max}} - x_i) \) is not constant, and \( 0 < x_i < x_{1,\text{max}} \), with \( x_i \) is equal to zero when the pressure \( P_1 \) is equal to the atmospheric pressure and \( x_i \) is equal to \( x_{1,\text{max}} \) if \( P_1 \) is equal to \( P_{aux} \) (Aboubakr et al., 2015; Volpi, 2013).

**Vent, Auxiliary, and Emergency Valve Mathematical Model** The main function of the vent valve, shown in Fig. 9a, is to connect the brake cylinder to the atmosphere in the case of the release mode. This valve is controlled by a solenoid valve which has only two possible positions *on* - *off*. The solenoid valve controls the flow of air from the auxiliary reservoir to the upper chamber of the valve, this air flow passes through a small area \( A_{2,1} \). The fluid network for the valve is shown in Fig. 9b. If the solenoid position is *off*, the area \( A_{2,1} \) is zero, which means that the passage is closed and the mass flow rate is zero. When the position of the solenoid valve is *off*, the vent valve has no effect on the air flow and on the air brake system. If the position of the solenoid valve is *on*, the vent valve opens the passage \( A_{2,2} \), which connects the brake cylinder with the atmosphere causing a drop in the cylinder pressure which in turn causes the release of the brake. If the solenoid position is *on*, the area \( A_{2,1} \) is not zero and the air mass flow rate can be evaluated using the equation

\[
\dot{m}_{aux\rightarrow 2} = 0.6 A_{2,1} P_2 \sqrt{\frac{r^2 - 1}{r - 1}} \sqrt{r - 1} \tag{15}
\]

where \( r = P_{aux} / P_2 \). As the pressure in the chamber \( P_2 \) increases, the piston starts to move. In this case, one has

\[
\frac{dP_2}{dt} = \frac{1}{V_2} \left( R_g \frac{d\dot{m}_{aux\rightarrow 2}}{dt} - P_2 \frac{dV_2}{dt} \right) \tag{16}
\]
The volume $V_2$ is not constant and can be evaluated using the equation $V_2 = V_{2,0} + S_2 x_2(t)$. It follows that

$$\frac{dP_2}{dt} = \frac{1}{V_{2,0} + S_2 x_2(t)} \left( R \theta \frac{dm_{aux,2}}{dt} - P_2 S_2 \frac{dx_2(t)}{dt} \right)$$

(17)

The displacement $x_2$ is governed by the piston equilibrium equation $m_2 \ddot{x}_2 = P_2 S_{2,1} - P_a S_{2,2} - P_{aux} S_{2,3} - k x_2 - L_2$ as shown in Fig. 9c, where $k$ is the stiffness of the valve spring, $L_2$ is the preload force of the spring, $S_{2,1}$ is the area of the piston that faces the upper chamber, $S_{2,2}$ is the area of the middle part of the piston, $S_{2,3}$ is the smaller area of the piston in the lower chamber, and $m_2$ is the mass of the piston. Because the mass and displacement of the piston are small, one can assume that the inertial force $m_2 \left( \frac{d^2 x_2(t)}{dt^2} \right)$ is negligible. Using this assumption, one can have an approximation of $x_2$ as $x_2 = \left( P_2 S_{2,1} - P_a S_{2,2} - P_{aux} S_{2,3} - L_2 \right) / k$ and its derivative as $\dot{x}_2 = \left( \dot{P}_2 S_{2,1} - P_{aux} S_{2,3} \right) / k$. The rate of the mass flow from the auxiliary reservoir to the brake cylinder can then be written as

$$\dot{m}_{bc \rightarrow a} = 0.6 A_{2,2} P_a \sqrt{\frac{r^2 - 1}{r - 1}}$$

(18)

where $r = P_{bc} / P_a$, $A_{2,2} = \pi D_2(x_2)$, $0 < x_2 < x_{2,max}$, and $x_2$ is equal to zero when the pressure $P_2$ is equal to the atmospheric pressure and $x_2$ is equal to $x_{2,max}$ if $P_2$ is equal to $P_{aux}$ (Aboubakr et al., 2015; Volpi, 2013).

In case of brake applications, the auxiliary valve connects the auxiliary reservoir to the brake cylinder. While, in emergency applications, the emergency valve connects the emergency reservoir to the brake cylinder to increase its pressure in order to faster apply higher
forces. The mathematical models of the auxiliary and emergency valves can be developed using a procedure similar to the one used for the vent valve.

7. BRAKE FORCE

In order to determine the brake force, the brake cylinder piston axial force has to be defined using the piston displacement and brake cylinder pressure. To determine the piston displacement \( u_{p_i} \) and the brake cylinder pressure \( P_{c_i} \), the following set of two coupled differential equations; the piston equation of motion and the time rate of the cylinder pressure equation, must be formulated:

\[
\begin{align*}
\frac{d^2 u_{p_i}(t)}{dt^2} &= \left( P_{c_i}(t) - P_d \right) S_{p_i} - K_{c_i} u_{p_i}(t) - f_f - F_{p_i} \left( u_{p_i}(t) - u_{con_{c_i}} \right) \\
\frac{dP_{c_i}(t)}{dt} &= \frac{1}{V_{0_i} + S_{c_i} u_{p_i}(t)} \left[ R_g \Theta \frac{dm_{c_i}(t)}{dt} - P_{c_i}(t) S_{c_i} \frac{du_{p_i}(t)}{dt} \right] \\
\end{align*}
\]

where \( m_{p_i} \) and \( S_{p_i} \) are, respectively the piston mass and cross-section area, \( K_{c_i} \) is the stiffness constant of the cylinder spring, \( f_f \) is the friction force, \( F_{p_i} \) and \( u_{con_{c_i}} \) are the piston axial force and displacement when the brake shoes contact the wheels, \( h(u_{p_i}) \) is the Heaviside step function, \( V_{0_i} \) and \( S_{c_i} \) are, respectively, the initial volume and the cross-section area of the brake cylinder ( \( S_{c_i} \approx S_{p_i} \)), and \( m_{c_i}(t) \) is the cylinder mass flow rate introduced in the preceding section. When the brake shoes come into contact with the wheels, one can assume \( d^2 u_{p_i}(t)/dt^2 \approx du_{p_i}(t)/dt \approx 0 \), and \( u_{p_i}(t) \approx u_{con_{c_i}}, u_{p_i}(t) > u_{con_{c_i}} \) which lead to \( h\left(u_{p_i}(t) - u_{con_{c_i}}\right) = 1 \). Using these assumptions, the piston axial force can be calculated as \( F_{p_i}(t) = \left( P_{c_i}(t) - P_d \right) S_{p_i} - K_{c_i} u_{con_{c_i}} - f_f \). Furthermore, assuming small piston acceleration and velocity, when \( u_{p_i}(t) \leq u_{con_{c_i}} \), the spring displacement can
be defined as \( u_{p_i}(t) = ((P_{c_i}(t) - P_a) s_{p_i} - f_{f_i}) / K_{c_i} \). In order to take into account rigging mechanical energy loss, the rigging efficiency can be accounted for using a simplified approach (Afshari et al., 2011). Figure 10 shows a schematic of the mechanism used to apply the brake force on one car wheel. In longitudinal train dynamics algorithms, one coordinate is used for the forward motion of each railcar. In this case, frictional tangential brake force for each shoe can be calculated as follows (Sanborn et al., 2007):

\[
f_{\text{brake}_i}(t) = \eta_i \frac{\mu_i}{N_{s_i}} R_i F_{p_i}(t) = \eta_i \frac{\mu_i}{N_{s_i}} R_i \left( (P_{c_i}(t) - P_a) s_{p_i} - K_{c_i} n_{\text{con}_i} - f_{f_i} \right)
\]

(20)

In this equation, \( \eta_i \) is the rigging efficiency, \( \mu_i \) is the friction coefficient between the wheel tread and the brake shoe, \( R_i \) is the brake rigging leverage ratio, and \( N_{s_i} \) is the total number of the brake shoes of the car.

8. NONLINEAR TRAIN DYNAMIC EQUATIONS

In this section, the nonlinear dynamic equations of the train cars are developed using the trajectory coordinates. The formulation used in this investigation allows each railcar to have up to six degrees of freedom that describe arbitrary three-dimensional translational and rotational displacements. In developing the nonlinear train equations, it is assumed that the rail vehicle dynamics has no effect on the air flow in the brake pipe, while the braking forces can have a significant effect on the train longitudinal forces as the result of the brake application and release.

8.1 Position, Velocity, and Acceleration

Starting with the absolute coordinates, the position vector \( \mathbf{r}_i \) of an arbitrary point on the car body \( i \) with respect to the global coordinate system can be defined as shown in Fig. 11 as
\[ r_i' = R_i' + A_i' \bar{u}_i', \] where \( R_i' \) is the global position vector of the origin of the body coordinate system with respect to the global system, \( \bar{u}_i' \) is the position vector of the arbitrary point on the car body with respect to the body local coordinate system, and \( A_i' \) is the transformation matrix that defines the orientation of the car local coordinate system with respect to the global system (Shabana, 2010). In the case of rigid body dynamics, \( \bar{u}_i' \) is constant. Differentiating the position vector \( r_i' \) with respect to time, one obtains the absolute velocity vector \( \dot{r}_i' = \dot{R}_i' + \omega_i' \times u_i' \), where \( \omega_i' \) is the absolute angular velocity vector defined in the global coordinate system, and \( u_i' = A_i' \bar{u}_i' \).

Differentiating \( \dot{r}_i' \) with respect to time, one obtains the absolute acceleration vector \( \ddot{r}_i' = \ddot{R}_i' + \alpha_i' \times u_i' + \omega_i' \times (\omega_i' \times u_i') \), where \( \alpha_i' \) is the angular acceleration vector of the car body \( i \).

Alternatively, the acceleration vector can be written as \( \ddot{r}_i' = \ddot{R}_i' + A_i' (\ddot{\alpha}_i' \times \bar{u}_i' + \ddot{\omega}_i' \times (\ddot{\omega}_i' \times \bar{u}_i')) \), where \( \alpha_i' = A_i' \ddot{\alpha}_i' \) and \( \omega_i' = A_i' \ddot{\omega}_i' \). The angular velocity vectors can be written in terms of the time derivatives of the orientation coordinates \( \theta_i \) as \( \omega_i' = G_i' \dot{\theta}_i', \) \( \bar{\omega}_i' = \bar{G}_i' \bar{\dot{\theta}}_i', \) where \( G_i' \) and \( \bar{G}_i' \) are functions of the orientation parameters \( \theta_i \) (Shabana et al., 2008).

### 8.2 Trajectory Coordinates

In the trajectory coordinate formulation, a centroidal body coordinate system is used for each rail component. Additionally, a body/track coordinate system that follows the motion of the components is used. The location of the origin and the orientation of the body/track coordinate system are defined using one geometric arc length parameter \( s_i \) that defines the distance travelled by the component along the track. The centroidal body coordinate system is selected such that it has no displacement in the longitudinal direction of motion with respect to the body/track coordinate system, that is, two translational coordinates, \( y_i' \) and \( z_i' \); and three angles, \( \psi_i', \phi_i', \) and \( \theta_i' \), are used to define the position and orientation of the body coordinate
system with respect to the body/track coordinate system $X^iY^iZ^i$, as shown in Fig. 12. Therefore, for each component $i$, the following six trajectory coordinates are used $\mathbf{p}_i = [s^i \ y^i \ z^i \ \psi^i \ \phi^i \ \Theta^i]^T$. In terms of these coordinates, the global position vector of the body $i$ mass center can be written as $\mathbf{R}_i = \mathbf{R}_i^i + \mathbf{A}_i^{i^i} \mathbf{u}^i$, where $\mathbf{u}^i = [0 \ y^i \ z^i]^T$ is the position vector of the center of mass with respect to the body/track coordinate system, $\mathbf{R}_i$ is the global position vector of the origin of the trajectory coordinate system, and $\mathbf{A}_i^{i^i}$ is the transformation matrix that defines the orientation of the body/track coordinate system. The vector $\mathbf{R}_i$ and the matrix $\mathbf{A}_i^{i^i}$ are functions of only one time dependent arc length parameter $s^i$.

The transformation matrix $\mathbf{A}_i^{i^i}$ is a function of three predefined Euler angles $\psi^i$, $\phi^i$, and $\Theta^i$ which are used in the description of the track geometry. One can then write

$$\mathbf{A}_i^{i^i} (s^i) = \left[ \begin{array}{ccc} \psi^i (s^i) & \Theta^i (s^i) & \phi^i (s^i) \end{array} \right]^T$$ (Shabana et al., 2008). On the other hand, the matrix $\mathbf{A}_i^{i^i}$ that defines the orientation of the body coordinate system with respect to the body/track coordinate system can be written in terms of the three time dependent Euler angles

$$\mathbf{A}_i^{i^i} = \left[ \begin{array}{ccc} \psi^i & \Theta^i & \phi^i \end{array} \right]^T$$ previously defined.

### 8.3 Equations of Motion

In order to obtain the equations of motion in terms of the trajectory coordinates, a velocity transformation matrix that relates the absolute Cartesian accelerations to the trajectory coordinate accelerations is used. Using the velocity transformation and the Newton-Euler equations that govern the spatial motion of rigid bodies, the equations of motion of the car bodies expressed in terms of the trajectory coordinates can be developed (Shabana, 2010). To this end, the vector of absolute accelerations $\mathbf{a}_i = \left[ \mathbf{\ddot{R}}_i^i \ \mathbf{\ddot{u}}_i^i \right]^T$ is written in terms of the trajectory accelerations as
\( \mathbf{a}' = \mathbf{B}' \mathbf{\dot{p}}' + \mathbf{\gamma}' \), where \( \mathbf{B}' \) is a velocity transformation matrix, and \( \mathbf{\gamma}' \) is a vector that absorbs terms which are quadratic in the velocities (Shabana et al., 2008). Substituting \( \mathbf{a}' \) into the Newton-Euler equations and pre-multiplying by the transpose of the velocity transformation matrix \( \mathbf{B}' \), one obtains the dynamic equations expressed in terms of the trajectory coordinates, as described in detail in (Shabana et al., 2008).

9. NUMERICAL RESULTS

In this section, several simulation scenarios are considered in order to demonstrate the use of the formulation presented in this paper. First, the proposed ECP brake model will be evaluated according to the AAR standard for cable based ECP system (S-4200) (“Performance Requirements for Testing Electronically Controlled Pneumatic (ECP) Cable-Based Freight Brake System”). The goal of the simulations is to demonstrate that the proposed ECP system meets the performance requirements specified in the AAR standard. Furthermore, the results will be compared with experimental data relevant to the same test reported in “Safety Evaluation of TSM Prototype Electronically Controlled Pneumatic train brake system on brake rack” (1998) in order to provide a validation of the model developed in this study. In the second part of this section, additional numerical results will be presented to evaluate the effect of the ECP system on the longitudinal dynamics of long trains. The stopping distances of the same train equipped with the conventional and ECP braking systems will be compared to demonstrate the benefits of the ECP system. The in-train force results and the general performance predicted using the two systems (conventional and ECP) will also be compared in this section.
**Pneumatic Performance Test** As previously mentioned, this section will provide numerical and experimental results of some of the tests described in “Performance Requirements for Testing Electronically Controlled Pneumatic (ECP) Cable-Based Freight Brake System”, AAR S-4200 standard.

**Description of TSM System Test Conditions** The TSM systems consist of a Car Control Device (CCD) and a manifold. The manifold contains the solenoids valves and the pressure transducers which are used to fill and vent the brake cylinder and monitor the brake pipe and reservoir pressure. The manifold is mounted between the pipe bracket and the service portion of the pneumatic control valve. When the ECP system is energized, the manifold cuts off the communication between the service portion and the brake cylinder. The service portion, however, continues its function of charging the reservoir. The system is controlled by the head end unit (HEU) which is mounted on the top of the engineer control stand. The HEU consists of a control box which has push buttons and soft keys. Brake applications are made as a percentage of full service pressure, with full service being a 100-percent application, a minimum service being a 15-percent application, and an emergency being a 120-percent application. The brake can be applied and released in 1-percent increments from 0 to 100 percent. The car brake systems are connected to each other by a shielded two-conductor gage cable. The cable carries both power and signal. The system allows the train operator to directly control the brake cylinder pressure on every car in the train. The brake pipe is used only to charge the reservoirs. Fifty TSM manufactured ECP brake CCDs were installed on the 150-car brake rack. The TSM CCDs were installed on every third air-brake control valve. Unequipped air-brake control valves were cut out. This setup resulted in the equivalent of a 7500-foot train consisting of fifty 150-foot cars.
**Simulation Model** In order to obtain the numerical results, a 50-car train model will be considered. Each car will be equipped with the ECP brake system. In this series of simulations, the dynamics of the train is ignored because the aim of this part of the study is to evaluate the pneumatic performance of the system. Every ECP unit in the train model consists of a CCD and four valves (cut-out, vent, auxiliary, emergency) described previously in this paper. In the simulation model, the CCD is a logical controller, as described in Section 3 of this paper, and therefore, such a controller is not affected by any errors or inaccuracy that can affect a real CCD. The brake pipe and the brake cylinder models used are the same as the ones proposed by Specchia et al. (2011) and Afshari et al. (2011). Table 1 shows the parameters of the brake pipe while the detailed parameters of the CCU are presented in Table 2.

The tests considered are described in the standard “Performance Requirements for Testing Electronically Controlled Pneumatic (ECP) Cable-Based Freight Brake System”, AAR Specification S-4200. The aim of these tests is to verify the performance of the electro-pneumatic system, especially in order to estimate the time required to apply/release brakes on the vehicles. The tests refer to Section 4 and 5 of the S-4200 specification: Section 4 describes the single car test rack requirements while Section 5 describes the performance requirements during testing on an AAR approved 150-car test rack. The performance requirements cover the normal operation of the system.

**Full-Service Brake Application and Release Test** The first test is a “Full-Service Brake Application and Release Test” performed on the 50-car model. The test is performed by applying a full service brake force; hold the pressure in the brake cylinder for 20 seconds, and then a full release is commanded. The requirements for the successful completion of the test are as follows:
1. A pressure equal to 64 psig (542589.47 Pa) in the brake cylinder must be achieved in 10 seconds with a maximum inaccuracy of ±2 psig for at least 10 seconds in all units.

2. The full release, assuming a threshold equal to 5 psig (135798.7 Pa), must be reached within 15 seconds of the time at which the command is given.

Figure 13 shows the pressure in the brake cylinder and the auxiliary reservoir in cars 1 and 50, in order to show the worst possible case in the simulation test. The results presented in this figure show that the system fulfills the requirements: the build-up time is about 5 seconds for every car, and it is also clear that the pressure increases simultaneously in all the brake cylinders. The requirement for the release time is also achieved, with the threshold pressure of 5 psig reached after 12 seconds. In Fig. 14, a comparison between the experimental results of the previously cited paper and the numerical results is presented. This comparison shows good agreement between the results of the experiment and simulation. It is important to keep in mind, however, that the simulation model is not exactly the same as the system tested because some TSM parameters are not available. The main difference is in the phase when the pressure of the brake cylinder has to be maintained to the desired value: in this phase the TSM brake cylinder appears to have some leakage, prompting the system control center to fill the cylinder in order to re-establish the desired pressure. In the simulated system, it is assumed that there is no leakage in the brake cylinder, and therefore, the pressure remains stable once the desired level of application is reached.

**Minimum Service Application and Release Test** Minimum service requirements were tested in accordance with Section 4.1 of S-4200. With the reservoirs fully charged, a minimum service brake application is commanded. The pressure is maintained for 20 seconds and then the complete release is commanded. The only requirement for this test is to reach the desired
pressure (9.6 psig, 167514.6 Pa) and maintain it in the tolerance of ±2 psig for 20 seconds. As shown in Fig. 15, the simulation model meets the requirements and, as mentioned before, all the units reach the desired pressure at the same time. The experimental data presented in Fig. 16 show in general a good agreement with the simulation results. However, these experimental data suffer a problem of increased pressure which is quickly corrected by the controller. The increase in pressure has to be eliminated as the report states that “this over-pressure condition is undesirable and is being addressed by the equipment manufacturer”.

**Emergency Application**  Emergency application is tested in accordance with Section 4.3 of S-4200. An emergency application is commanded immediately after a release from full service; as specified in the S-4200, no release command is given. To perform this test successfully, the brake cylinder should reach the 120-percent of the full service application pressure (76.8 psig, 630842.36 Pa) within 10 seconds and hold this pressure indefinitely. The results of this test, shown in Fig. 17, demonstrate that the requirements were fulfilled with a build-up time of approximately 2 seconds. The experimental results for this test are not available because the TSM system fails to meet the requirements.

**Gradual Application and Release**  The gradual application and release test is performed in accordance with Section 5.2 of S-4200. Starting with the reservoirs fully charged, a minimum service application is commanded (15 percent of the full service brake pressure), then the pressure is reduced to 10 percent of full service. Then every 5 seconds, the commanded brake application is increased by 10-percent of full service until a full service brake application is reached. Afterwards the commanded brake application is reduced by 10 percent of full service at 5 seconds intervals until the brakes are fully released. The S-4200 standard requires that the pressure within the brake cylinder follows the commands quickly, and the required pressure with
a tolerance of ±2 psig must be ensured. As pointed out in Formenton (2009) and Smith and Carlson (1999), the gradual release of the brake is a new option that the ECP system offers. Gradual or incremental release of a brake application is a major benefit for North America freight rail system where existing equalization type systems are limited to direct release only. In fact, with the conventional air brake system, it is not possible to carry out a partial release of the brake, that is, once the braking is activated it is not possible to reduce its intensity but only release the brake completely or continue with the same intensity. With the new ECP system, it is possible to improve braking efficiency by ensuring greater control of the level of braking application. Such improvement in efficiency also allows reducing the fuel consumption and the amount of air required, thereby increasing the availability of reservoirs in the case of brake application. The simulated system meets the standard requirements as shown in Figs. 18 and 19. The TSM system also follows the standard requirements with more inaccuracy as compared to the proposed model.

**Repeated Full Service Brake Application Test** The repeated full service brake application test is performed in accordance with section 5.3 of S-4200. With the reservoir completely charged, a full-service application is commanded, when the desired pressure (64 psig, 542589.47 Pa) is reached, a complete release is commanded. After 15 seconds another full-service application is made. The standard requires both the first and the second application to reach the full-service pressure. The results of Fig. 20 show that the proposed model is able to meet the requirements. The data presented in Fig. 21 reveal that also the TSM test system meets the standard requirements and that the simulated system and the tested system display a similar behavior.
Summary    Considering the results presented, one concludes that the ECP brake model developed in this investigation meets the requirements of the standard S-4200. Furthermore, comparisons with the test results available in the literature, demonstrate that the behavior of the proposed model is consistent with an ECP system currently installed on freight trains in North America. It is important, however, to point out that the system parameters used in this numerical study were not obtained from an actual prototype but based on information gathered and data from the literature. This means that a system with different parameters may result in more or less differences as compared with the results presented in this study.

ECP Braking Performance

In this section, the performance of the ECP braking system will be tested and compared to that of a conventional air brake system. As it is known, the application of the conventional pneumatic brakes is dependent on the pressure difference between the auxiliary reservoir and the brake pipe, and the rate at which the brake pipe pressure drop is sensed by the triple valve. Due to the dependency on the triple valve to detect these changes in the brake pipe pressure, there is a delay from the engine braking command to the brake application on the cars. The delay period varies with the train length; the longer the train the longer the delay. The effect of this delay can be significant mainly for two reasons: the stopping distance, and the longitudinal forces exchanged between the adjacent cars, absorbed almost entirely by the couplers. When using the ECP brake system, the delay is eliminated since all CCDs in the train receive the broadcast Train Brake Command (TBC) from the HEU almost at the same time. In this section, the performance analysis will focus on the stopping distance, the in-train force distribution along the train, and the effect of continuous recharging.
**Stopping Distance**  The stopping distance results obtained for the same train with two different braking systems (ECP and conventional) are compared. A 75-car model for a train with 3 locomotives travelling on a tangent track is considered. It is assumed that all the cars and the locomotives have initial forward velocity of 29.1 m/s (65.0 mph), and the brake pipe length is 1059.5 m (3476.05 ft). The brake pipe is modeled using 150 finite elements. The data of the train such as mass, number of wheelset, and dimensions of the cars and locomotives are shown in Table 3. At the beginning of the simulation the train reservoirs are assumed to have operating pressure of 90 psig (721853.15 Pa), after 5 seconds, a full service braking is commanded followed by a full release after 80 seconds. The simulation continues until all the auxiliary reservoirs are fully recharged at the operating pressure. Figure 22 shows the time history of the positions of the center of mass of the train cars equipped with the ECP and conventional air brakes systems. The results presented in this figure show that the stopping distance can be reduced by approximately 40% when the ECP brake system is used. The train travels 1395 m before stopping when using the conventional brake system, while it travels only 895 m when the ECP brake system is used. These results confirm what reported by Formenton (2009) and Smith and Carlson (1999) with regard to the decrease in the stopping distance when the ECP brake is used.

**Longitudinal Couplers Force**  The same train model previously used to study the stopping distance is used to demonstrate the effectiveness of ECP system in reducing the coupling forces between vehicles. In the case of the conventional air brake, it is known that the longitudinal forces in a braking situation are for the most part compressive forces caused by the delay in the brake application. Another numerical study of this phenomenon was performed by Nasr et al. (2010). The coupler force clearly depends on the type of couplers used in the simulation (Cheli
In this case, an End of Car Cushioning (EOC) Device force model (Shabana et al., 2011; Mass et al., 2012) is adopted. As previously mentioned, one advantage of the ECP model is its ability to eliminate the delay in the brake application because of the use of the electronic communication system. Figures 23 to 26 show the results when the conventional air brake system is used. These results clearly show the effect of the time delay on the performance of the system. On the other hand, the results presented in Figs. 27 and 28 show the delay time is approximately zero when the ECP brake system is used. An analysis of the coupler force distribution along the train was performed. Nasr et al. (2010) analyzed the distribution of the couplers along the train in order to study the longitudinal dynamics of a freight train equipped with a conventional air brake system. In their model, the delay of application of the braking force was an input to the system. Nasr et al. attempted to determine the delay time by comparing the couplers force distribution along the train. In this study, on the other hand, the delay time is determined using the air flow finite element model of the brake pipe previously described in this paper. Figures 29 and 30 show the coupler forces along the train for the conventional and ECP systems, respectively. The horizontal axis represents the position of the $i$-th coupler divided by the train length, while the vertical axis represents the maximum value reached by the compression and traction forces. In a train equipped with the conventional air brake system, the central couplers of the train experience the highest increase in the forces because they are subjected to the most compression caused by the longer time delay of the last cars (Cheli et al., 2008). The results obtained in this numerical investigation confirmed this phenomenon. In the ECP model, on the other hand, (Figure 30) a completely different scenario is observed as shown in Fig. 30. Because of the simultaneous application of the brake force for all the cars, the coupler forces are significantly lower than the forces in the case of the conventional brake model.
Moreover, the distribution of the coupler forces along the train is approximately constant or, at least, with less variations.

**Charging** The numerical simulations can be used to predict the behavior of the braking system during the charging phase. As shown in Fig 23, the conventional system starts charging the reservoirs when the release command is given. The pressure in the auxiliary reservoirs mounted on the $i$-th car increases only when the pressure in the brake pipe at the corresponding connecting point is higher than the actual auxiliary reservoir pressure. As a result, the time needed to fill all the reservoirs of a 75-car train for the conventional air brake system is about 170 seconds. On the other hand, because the ECP brake system (Fig. 27) does not use the pressure drop along the brake pipe to transmit the signal, such a system can start to charge the reservoirs as soon as the pressure in the auxiliary reservoirs becomes lower than a certain value, and therefore, the charging starts automatically without any specific command from the HEU. This results in a faster recharging which is completed in about 90 seconds.

10. **SUMMARY AND CONCLUSIONS**

The development of an accurate nonlinear longitudinal train dynamic algorithm is necessary in order to better understand railroad vehicle dynamics. Such accurate models can be used efficiently to study the effect of different braking systems and scenarios on the stability and longitudinal dynamics of long trains. To this end, this investigation is focused on integrating an ECP (Electronically Controlled Pneumatic) brake model with efficient train longitudinal force algorithms based on the trajectory coordinate formulation. The ECP brake model, developed in this study consists of the train line (cable), locomotive automatic brake valve, air brake pipe, and ECP manifold. The train line covers the length of the train and allows the brake commands to be
received by each car simultaneously. Unlike the conventional air brake model, the brake pipe in the ECP system is no longer used to signal brake level commands. In the ECP system, the brake pipe provides a continuous supply of compressed air to be stored in a reservoir on each railcar. As previously mentioned and demonstrated in this paper, uniform and instantaneous application of the brakes on all railcars gives better train control, shortens the stopping distances, and leads to a lower risk of derailment, and/or coupler failure. The equations that govern the air pressure flow were developed in this paper using the general fluid continuity and momentum equations. These equations were simplified using the assumptions of one-dimensional isothermal flow. The FE method is used to convert the resulting partial differential equations to a set of first order ordinary differential equations which are integrated with the train differential equations of motion. The car brake forces depend on the signal from the Head End Unit (HEU) mounted within the locomotive cab which is the ECP brake system control device. These forces are applied to the wheels using the ECP manifold located at each car. The ECP manifold used in this investigation has four valves; cut-off valve, vent valve, auxiliary valve, and emergency valve. The ECP manifold is connected to three main pneumatic components; the auxiliary reservoir, the emergency reservoir, and the brake cylinder. A detailed mathematical model for the ECP manifold and its components is developed and its use is demonstrated in this paper using several dynamic simulation scenarios of short and long trains. The performance of the developed ECP brake system is compared with Association of American Railroads safety and operation standards, and such a comparison showed a good agreement. Furthermore, a detailed comparison between the ECP brake system and the conventional air brake system is presented in order to demonstrate the benefits of using the ECP brake system in the case of freight transportation. To this end, numerical simulations were carried out to compare the performance of conventional and
ECP braking on a 75-car freight train. The simultaneous application of ECP braking forces leads to reduction of the stopping distance by 40% in a full-service braking with an initial speed of 65 mile/hour. For the same reason, coupling forces among wagons can be drastically reduced when using the ECP system. Numerical results also showed the time required to charge all the reservoirs can be almost halved with the new braking system. In summary, the analysis presented in this paper shows the benefits the ECP braking system that allows controlling the longitudinal dynamics and contributes to achieving operation safety.
REFERENCES


**Abbreviations**

AAR  Association of American Railroads  
CCD  Car control device  
CCU  Car control unit  
CID  Car ID Module  
ECP  Electronically controlled pneumatic  
EOC  End of car cushioning  
EOT  End-of-Train  
FE  Finite element  
HEU  Head end unit  
LID  Locomotive ID Module  
PSC  Power Supply Controller  
TBC  Train brake commands

**Nomenclature**

$A^e$  Element cross section area  
$A^i$  Transformation matrix  
$A^{ir}$  Transformation matrix of the car coordinate system with respect to the track coordinate system  
$A^{ti}$  Transformation matrix of the track coordinate system  
$A_{e,f}$  Equivalent connecting areas between component $f$ and component $e$  
$A_{i,1}$  Passage area between valve $i$ and one of CCU components based on function of valve $i$  
$A_{i,2}$  Passage area between the auxiliary reservoir and valve $i$
\( \mathbf{a}^i \) Vector of absolute acceleration in terms of trajectory coordinates

\( \mathbf{B}^i \) Velocity transformation matrix

\( \mathbf{e} \) Vector of the system nodal coordinates

\( F_{p_i} \) Piston axial force of brake cylinder \( i \)

\( \mathbf{f}_b \) Vector of body forces per unit volume

\( f_{f_i} \) Friction force system

\( \mathbf{G}^i, \mathbf{\dot{G}}^i \) Velocity transformation matrices that relate the angular velocities to the time derivative of the orientation coordinates

\( h(u_{p_i}) \) Heaviside step function of brake cylinder \( i \)

\( K_{c_i} \) Spring stiffness of brake cylinder \( i \)

\( k \) Stiffness of the spring for ECP manifold valves

\( L_i \) Preload force of valve spring \( i \)

\( l^e \) Length of the finite element

\( \mathbf{M} \) Brake pipe global coefficient matrix

\( \mathbf{M}^i \) Element mass matrix

\( m_i \) Mass of the piston of the valve \( i \)

\( m_{p_i} \) Piston mass of brake cylinder \( i \)

\( N_{s_i} \) Total number of the brake shoes of the car

\( p \) Hydrostatic pressure

\( \mathbf{p}^i \) Trajectory coordinates of car body \( i \)
\( P_i \) Pressure in the chamber of valve \( i \)

\( P_{ci} \) Pressure of brake cylinder \( i \)

\( \mathbf{p}^e, \mathbf{q}^e \) Vectors of element nodal coordinates

\( \mathbf{Q} \) Brake pipe global right hand side vector

\( \mathbf{Q}^e \) Element right hand side vector

\( R_g \) Gas constant (\( \text{J}/(\text{Kg}^\circ\text{K}) \))

\( \mathbf{R}^i \) Global position vector of the origin of the car \( i \)

\( \mathbf{R}^d \) Global position vector of the origin of the trajectory coordinate system

\( R_i \) Brake rigging leverage ratio of brake cylinder \( i \)

\( \mathbf{r}^i \) Position vector of an arbitrary point on the car \( i \) with respect to global coordinate system

\( S_{ci} \) Cross-section area of the brake cylinder \( i \)

\( S_p^e, S_q^e \) Element shape functions

\( S_{i,1} \) Area of the piston that faces the upper chamber in valve \( i \)

\( S_{i,2} \) Area of the middle part of the piston in valve \( i \)

\( S_{i,3} \) Area of the piston in the lower chamber of valve \( i \)

\( S_{pi} \) Piston cross-section area of brake cylinder \( i \)

\( s^i \) Arc length

\( \mathbf{u}^i \) Position vector of the arbitrary point on the car \( i \) with respect to the car coordinate system

\( \mathbf{u}^{ir} \) Position vector of the center of mass of car \( i \) with respect to the track coordinate system

\( u_{pi} \) Piston displacement of brake cylinder \( i \)
\( u_{\text{con}, i} \)  \( \text{Piston axial displacement of brake cylinder} \ i \)

\( V^e \)  \( \text{Element volume} \)

\( V_i \)  \( \text{Volume of the chamber for valve} \ i \)

\( V_0 \)  \( \text{Initial volume of the brake cylinder} \ i \)

\( \mathbf{v} \)  \( \text{Velocity vector} \)

\( x_i \)  \( \text{Displacement of the piston of the valve} \ i \)

\( \dot{m}_e \)  \( \text{Mass flow rate to a component} \ e \)

\( \dot{m}_{e-f} \)  \( \text{Mass flow rate from component} \ f \text{ to component} \ e \)

\( \dot{m}_i \)  \( \text{Mass flow rate of brake cylinder} \ i \)

\( \mathbf{r}^i \)  \( \text{Absolute velocity vector of an arbitrary point on the car} \ i \)

\( \ddot{r}^i \)  \( \text{Absolute acceleration vector of an arbitrary point on the car} \ i \)

\( \alpha^i \)  \( \text{Absolute angular acceleration vector of car} \ i \)

\( \mathbf{\theta}^i \)  \( \text{Orientation coordinates vector of car} \ i \)

\( \Theta \)  \( \text{Temperature (°K)} \)

\( \rho \)  \( \text{Mass density} \)

\( \lambda, \mu \)  \( \text{Viscosity coefficients} \)

\( \eta_i \)  \( \text{Rigging efficiency of brake cylinder} \ i \)

\( \mu_i \)  \( \text{Friction coefficient between the wheel tread and the brake shoe of brake cylinder} \ i \)

\( \nabla \)  \( \text{Divergence operator} \)

\( \omega^i \)  \( \text{Absolute angular velocity of car} \ i \)
**Table 1**: Brake pipe properties (conventional air brake system)

<table>
<thead>
<tr>
<th>Parameter description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air temperature</td>
<td>300 °K</td>
</tr>
<tr>
<td>Air viscosity</td>
<td>$1.95 \times 10^{-5}$ Pa.s</td>
</tr>
<tr>
<td>Atmospheric pressure</td>
<td>101.325 kPa (14.7 psia)</td>
</tr>
<tr>
<td>Brake pipe diameter (1¼” Schedule 80 extra heavy pipe)</td>
<td>3.246 cm (1.278 inches)</td>
</tr>
<tr>
<td>Operating pressure</td>
<td>620.5 kPa gage (90 psig)</td>
</tr>
<tr>
<td>Main reservoir pressure</td>
<td>951.48 kPa gage (138 psig)</td>
</tr>
<tr>
<td>Parameter description</td>
<td>Value</td>
</tr>
<tr>
<td>--------------------------------------------------------------------------------------</td>
<td>-----------------------------</td>
</tr>
<tr>
<td>Number of brake cylinders per car</td>
<td>1</td>
</tr>
<tr>
<td>Number of brake shoes per car</td>
<td>8</td>
</tr>
<tr>
<td>Brake rigging leverage ratio</td>
<td>10:1</td>
</tr>
<tr>
<td>Brake rigging efficiency at full service brake</td>
<td>0.65</td>
</tr>
<tr>
<td>Brake shoe/wheel friction coefficient</td>
<td>0.35</td>
</tr>
<tr>
<td>Brake cylinder spring stiffness</td>
<td>14593 N/m</td>
</tr>
<tr>
<td>Brake cylinder maximum swept volume</td>
<td>10296 cm³</td>
</tr>
<tr>
<td>Brake piston effective frontal area</td>
<td>506.7 cm²</td>
</tr>
<tr>
<td>Brake cylinder and auxiliary reservoir equivalent connecting area</td>
<td>0.0236 cm²</td>
</tr>
<tr>
<td>Brake cylinder and emergency reservoir equivalent connecting area</td>
<td>0.0400 cm²</td>
</tr>
<tr>
<td>Brake cylinder and atmosphere equivalent connecting area</td>
<td>0.0446 cm²</td>
</tr>
<tr>
<td>Brake cylinder piping volume</td>
<td>4839.3 cm³</td>
</tr>
<tr>
<td>Auxiliary reservoir to brake valve piping volume plus quick service volume in pipe bracket</td>
<td>4439.1 cm³</td>
</tr>
<tr>
<td>Emergency reservoir to brake valve piping volume</td>
<td>3186.6 cm³</td>
</tr>
<tr>
<td>Auxiliary reservoir volume</td>
<td>40967 cm³</td>
</tr>
<tr>
<td>Auxiliary reservoir and brake pipe equivalent connecting area</td>
<td>0.0201 cm²</td>
</tr>
<tr>
<td>Emergency reservoir volume</td>
<td>57354 cm³</td>
</tr>
<tr>
<td>Auxiliary Reservoir and Cut-Out Valve upper chamber connecting area</td>
<td>0.00314 cm²</td>
</tr>
<tr>
<td>Diameter of the passage section Cut-Out Valve</td>
<td>1 cm</td>
</tr>
<tr>
<td>Maximum Displacement of the Cut-out Valve</td>
<td>0.1 cm</td>
</tr>
<tr>
<td>Upper chamber frontal area Cut-Out Valve</td>
<td>0.5 cm²</td>
</tr>
<tr>
<td>Lower chamber frontal area Cut-Out Valve</td>
<td>0.085 cm²</td>
</tr>
<tr>
<td>Initial Volume of the upper chamber of the Cut-Out Valve</td>
<td>7.6 cm³</td>
</tr>
<tr>
<td>Auxiliary Reservoir and Vent Valve upper chamber connecting area</td>
<td>0.0314 cm²</td>
</tr>
<tr>
<td>Diameter of the passage section Vent Valve</td>
<td>0.5 cm</td>
</tr>
<tr>
<td>Maximum Displacement of the Vent Valve</td>
<td>0.15 cm</td>
</tr>
<tr>
<td>Upper chamber frontal area Vent Valve</td>
<td>0.5 cm²</td>
</tr>
<tr>
<td>Lower chamber frontal area Vent Valve</td>
<td>0.085 cm²</td>
</tr>
<tr>
<td>Initial Volume of the upper chamber of the Vent Valve</td>
<td>7.6 cm³</td>
</tr>
<tr>
<td>Auxiliary Reservoir and Auxiliary and Emergency Valve upper chamber connecting area</td>
<td>0.0314 cm²</td>
</tr>
<tr>
<td>Diameter of the passage section Auxiliary and Emergency Valve</td>
<td>0.5 cm</td>
</tr>
<tr>
<td>Maximum Displacement of the Auxiliary and Emergency Valve</td>
<td>0.15 cm</td>
</tr>
<tr>
<td>Upper chamber frontal area Auxiliary and Emergency Valve</td>
<td>0.5 cm²</td>
</tr>
<tr>
<td>Lower chamber frontal area Auxiliary and Emergency Valve</td>
<td>0.085 cm²</td>
</tr>
<tr>
<td>Initial Volume of the upper chamber of the Auxiliary and Emergency Valve</td>
<td>7.6 cm³</td>
</tr>
<tr>
<td>Stiffness of the piston spring in the ECP valves</td>
<td>3500 N/m</td>
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Table 2: Car control unit properties
### Table 3: Train model data

<table>
<thead>
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<th>Parameter description</th>
<th>Value</th>
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<tbody>
<tr>
<td>Number of cars</td>
<td>75</td>
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<tr>
<td>Number of locomotives</td>
<td>3</td>
</tr>
<tr>
<td>Cars mass (full loaded)</td>
<td>129727.42 Kg</td>
</tr>
<tr>
<td>Locomotives mass</td>
<td>166921.99 Kg</td>
</tr>
<tr>
<td>Number of wheelsets for car</td>
<td>4</td>
</tr>
<tr>
<td>Number of wheelsets for locomotive</td>
<td>6</td>
</tr>
</tbody>
</table>
FIGURES

Figure 1: ECP train layout.

Figure 2: ECP car layout.
**Figure 3:** ECP manifold components.

**Figure 4:** 26C valve.
**Figure 5:** Main air brake components

**Figure 6:** Car control unit components

**Figure 7:** Mass flow between ECP manifold and the car control unit components
(a) Cut out valve scheme

(b) Fluid network

(c) Cut out valve free body diagram

Figure 8: Cut out valve
Figure 9: Vent Valve
Figure 10: Brake force applied on a car wheel

Figure 11: Coordinate systems
Figure 12: Trajectory coordinates

Figure 13: Full service application scenario

(■—Brake cylinder car 1 •—Brake cylinder car 50 ●—Auxiliary reservoir car 1 ▽—Auxiliary reservoir car 50)
**Figure 14:** Full service application comparison

(- - Experimental results  –– Simulation results)

**Figure 15:** Minimum service application scenario

( ■ Brake cylinder car 1  ■ Brake cylinder car 50  ■ Auxiliary reservoir car 1  ■ Auxiliary reservoir car 50)
Figure 16: Minimum service application comparison
(■ Experimental results ○ Simulation results)

Figure 17: Emergency application scenario
(■ Brake cylinder car 1 ▲ Brake cylinder car 50 ■ Emergency reservoir car 1 ▼ Emergency reservoir car 50)
**Figure 18:** Gradual application and release scenario
( ■ Brake cylinder car 1 ○ Brake cylinder car 50)

**Figure 19:** Gradual application and release scenario
( ■ Auxiliary reservoir car 1 ○ Auxiliary reservoir car 50)
Figure 20: Repeated full service application scenario
(● Brake cylinder car 1 ▲ Brake cylinder car 50 ● Auxiliary reservoir car 1
△ Auxiliary reservoir car 50)

Figure 21: Repeated full service application comparison
(● Experimental results ○ Simulation results)
Figure 22: Position of center of mass of the train

(  ECP model   Conventional air brake model)

Figure 23: Pressure in auxiliary reservoirs; conventional air brake model

( car 1   car 25   car 50   car 75)
Figure 24: Pressure in the brake cylinders; conventional air brake model

\(\text{Pressure [Pa]}\)

\(\text{Time (s)}\)

- O--- car 1
- ▽--- car 25
- ◀--- car 35
- ●--- car 50
- □--- car 75

Figure 25: Pressure in the brake pipe; conventional air brake model

\(\text{Pressure [Pa]}\)

\(\text{Time (s)}\)

- O--- node 1
- ▽--- node 20
- ◀--- node 60
- ●--- node 80
- □--- node 90
- ▶--- node 150
**Figure 26:** Couplers forces; conventional air brake model
(■ coupler 1 ■ coupler 25 ■ coupler 35 ■ coupler 40 ■ coupler 60)

**Figure 27:** Auxiliary reservoirs pressure; ECP system
(○ car 1 ▼ car 25 ▲ car 35 ▪ car 50 ■ car 75)
Figure 28: Brake cylinder pressure; ECP system

Figure 4 Maximum couplers forces; conventional air brake model
**Figure 30:** Maximum couplers forces; ECP system

(Compression force — Traction force)