ADDIS ABABA UNIVERSITY

INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

SIMULATION OF ELECTRO-PNEUMATIC BRAKING SYSTEM RESPONSE IN LONG TRAINS

A thesis

Submitted to the school of graduate studies Addis Ababa University in partial fulfillment for the degree of masters of Science in rolling stock engineering

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SCHOOL OF GRDUATE STUDIES

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ABSTRACT

Electro-pneumatic braking system (epbs) components can be frequently used in any commercial automobile vehicles regardless of their size of outputs. Their difference is in the amount of longitudinal force developed inside these components according to the size of the automobile. The simulation of electro-pneumatic brake systems has of great importance in order to understand the fluid dynamics inside these brake components.

The electro pneumatic braking system is applicable in any automobiles. But this particular system is applicable in a train vehicle (coach). This paper presents a computational model based simulation of electro-pneumatic braking system used for prediction of fluid dynamic behavior inside the systems and to compare the simulated results with the pneumatic one.

The system is modeled and simulated in modeling and simulation environment called MATLAB SIMULINK. Considering the complexity of the electro pneumatic braking system components and the related phenomena of fluid motion inside these components, special models have been elaborated using flow chart algorithm.

The simulation result shows good agreement with measurement data found from different literature reviews but have a big difference in time response with the pneumatic braking system. The comparative measurement data (experimental data) found from different literature reviews and simulation result found from this simulation study confirms that the system is modeled correctly and this simulating tool can be effectively used in design, research and development of electro pneumatic braking systems.

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NOMENCLATURE

A_{bcp}	[m^2]	Area of brake cylinder piston
A_{bcpr}	[m^2]	Area of brake cylinder piston rod
A_{bv}	[m^2]	Maximum area of balance valve
A_{rv}	[m^2]	Maximum area of release valve
A_{mvlp}	[m^2]	Area of main valve piston rod
A_{mvlp}	[m^2]	Area of main valve large piston
A_{mvsp}	[m^2]	Area of main valve small piston
C_d		Discharge coefficient
C_A		Constant coefficient of orifice cross-section
d_{bc}	[m]	Brake cylinder inside diameter
d_{bcp}	[m]	Brake cylinder pipe diameter
d_{bcpr}	[m]	Brake cylinder piston rod diameter
d_{srp}	[m]	Service reservoir pipe diameter
d_{arp}	[m]	Auxiliary reservoir pipe diameter
d_{tp}	[m]	Train pipe line diameter
K_{bv}	[N/m]	Stiffness of balance valve spring
K_{rv}	[N/m]	Stiffness of release valve spring
K_{bc}	[N/m]	Stiffness of brake cylinder spring
M_{bcp}	[kg]	Mass of brake cylinder piston with its rod
M_{mvp}	[kg]	Mass of main valve piston
P_1	[pa]	Pressure in service reservoir
P_2	[pa]	Pressure in train pipe
P_3	[pa]	Pressure in auxiliary reservoir
P_4	[pa]	Pressure in front of brake cylinder piston
P_5	[pa]	Pressure at the back of brake cylinder piston
P_{atm}	[pa]	Atmospheric pressure out of the system
R	[m^2/s^2k^0]	General gas constant
T_{atm}	[k^0]	Atmospheric temperature
X_{bvp}	[m]	Maximum stroke of balance valve piston
X_{rvp}	[m]	Maximum stroke of release valve piston
Y	[m]	Brake cylinder piston maximum stroke

γ		Specific heat ratio for air
Ff	[N]	Frictional force
$\dot{m}$	[kg/s]	Mass flow rate
$\ddot{Y}$	[m/s^2]	Piston cylinder acceleration
$\dot{Y}$	[m/s]	Piston cylinder velocity
$\ddot{X}$	[m/s2]	Piston valve acceleration
$\dot{X}$	[m/s]	Piston valve velocity
β	[Ns/m]	Damping coefficient
V_{bc}	[m3]	Active volume of brake cylinder
V_{ar}	[lit]	Volume of auxiliary reservoir
V_{sr}	[lit]	Volume of service reservoir
M_p	[kg]	Mass of plunger
F_{pld}	[N]	Spring pre-load force
P_{sup}	[pa]	Supply pressure
P_u	[pa]	Upstream pressure
P_d	[pa]	Downstream pressure
P_r	[pa]	Critical pressure
X_p	[m]	Plunger position
A_v	[m^2]	Area of valve orifice
$\dot{m}_{bvo}$	[kg/s]	mass flow rate of balance valve orifice
$\dot{m}_{rvo}$	[kg/s]	mass flow rate of release valve orifice
$m_{bc(air)}$	[kg]	mass of brake cylinder air
p_{bc}	[pa]	brake cylinder pressure
A_{eep}	[m^2]	area of electromagnetic valve exhausting port
A_{evep}	[m^2]	area of emergency valve exhausting port
m_{eep}	[kg]	mass of air through electromagnetic valve exhausting port
m_{evep}	[kg]	mass of air through emergency valve exhausting port

CHAPTER-I

INTRODUCTION

1.1. Backgrounds

Train braking is a very complex process, specific to rail vehicles and has great importance to the safety of the traffic.

The purpose of braking action is to perform controlled reduction in velocity of the train, either to reach a certain lower speed or to stop to a fixed point. In general terms, this happens by converting the kinetic energy of the train and the potential energy in case of circulation on slopes in to mechanical work of braking forces which usually turns into heat, which dissipates into the environment.

As a consequence, along the time, for railway vehicles have been developed various brake systems, whose construction, design and operation depend on many factors such as running speed, axle load, type, construction and technical characteristics of vehicles, traffic conditions, etc.

Depending on how the command actions are performed and according to the mean that braking forces are basically developed, braking system is divided in to:

A. Handbrakes

Which are applied by hand action to a wheel or lever on the vehicle. Nowadays are generally used for securing unattended or unpowered vehicles against unplanned movement, not for braking actions while operating. That is why usually it is known as parking brakes.

B. Pneumatic brakes

This system uses air pressure variations both to command and to apply the brake blocks or pads, generating braking/releasing actions and forces. The vast majority of trains use compressed air, changing the level of air pressure in the brake pipe determining a change in the state of the brake on each vehicle.

In the case of straight air brake system, the increase of air pressure in the pipe determines the increase of air pressure in the brake cylinders of each vehicle. In the case of indirect air brake system, braking actions are commanded by decreasing the pressure in the train's pipe, generating by special features the increase of air pressure in the brake cylinders of each vehicle.

C. Electro-pneumatic brakes

That has an electrical command while compressed air is used to increase the pressure in the brake cylinders of each vehicle to apply the brake blocks or pads for generating braking forces;

D. Rail (track) brakes

Which are usually electric commanded and the braking force is achieved due to strong magnetic forces induced by large electromagnets hung under the vehicle's bogie, over the top surface of the rails. If the braking forces are generated by the frictional forces between the electromagnets and rails it is an electromagnetic rail brake. If the electromagnetic fields generate eddy currents in the rails, creating forces acting in the opposite direction of the movement of the train, it is a linear eddy current brake system.

On the same principle is the rotary eddy currents brake system, the metallic mass being either the wheels, or discs attached to the wheel sets.

E. Electric braking

Which is based on the reversibility of the electric engine, in particular the electric traction motors are reconnected in such a way that they act as generators which provide braking effort. Practically, the kinetic and/or the potential energy (while running on slopes) are converted into electric energy. If the power generated during braking is dissipated as heat through on-board resistors it is about rheostatic braking.
On electric railways, it is also possible to convert the energy of the train back into usable power by diverting the braking current into traction supply line, this being the case of regenerative braking.

F. Hydraulic brakes

Which act using hydraulic oil and, depending on the achievement of braking forces, there may be hydrostatic, when is due to oil pressure increasing or hydrodynamic when the kinetic energy of the vehicle is converted in the rotor of a hydraulic pump in heat which is dissipated through the oil cooling system.

From the above train braking systems let me discuss on the two braking systems i.e pneumatic and electro pneumatic braking systems. Because now a day they are widely used across the world they have almost the same components and working principles.

1.1.1. Pneumatic braking systems

As discussed above there are two types of pneumatic braking systems. These are the direct and indirect braking systems. Whatever the two systems are, imagine a vehicle that is a mile in length. It is so long that the front of the vehicle might be climbing a grade while the back is descending, or perhaps the front and back are turning left while the middle is turning right. This vehicle is more than 300 times as long as its width.

Now, visualize the vehicle traveling at 70 MPH and the operator wants to stop. This is a complex and challenging problem in air braking system **[27]**

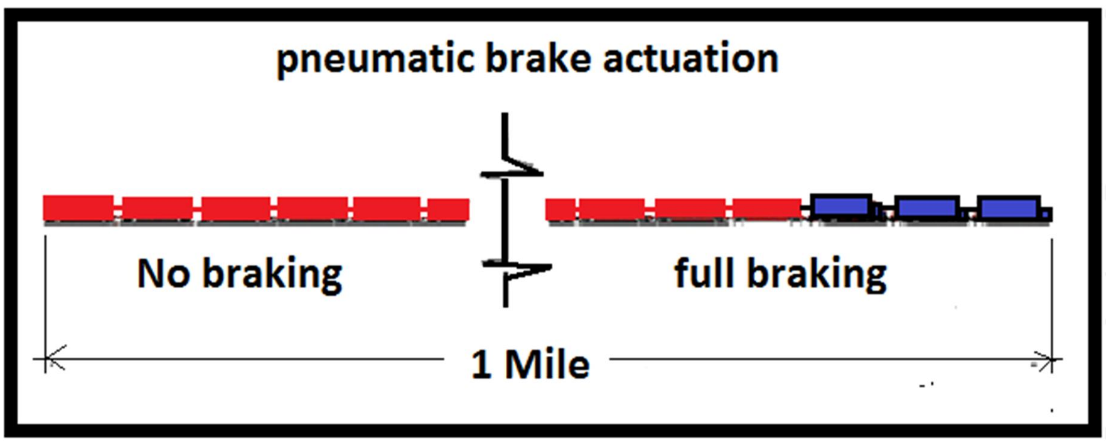


Figure 1.1 effect of time lag on pneumatic brake system **[27]**

Even though **fig 1.2** in the next page is indirect pneumatic braking system but regardless of their types purely air brake system of railway trains really has a problem on the transmission of an air signal along the brake pipe. This air signal is initiated from the front of the train and has to be sent to all vehicles along the train to the rear. There will always be a time lapse (called the propagation rate) between the reaction of the leading vehicle and the reaction of the one at the rear. This time lapse is a considerable restraint on operation. It causes the braking of vehicles to happen at different times along the train so that while some cars are slowing down; others are still trying to push, unbraked, from the rear as shown in fig 1.1 above. When releasing, the front of the train is pulling the rear, still braking, and causes stress to the draft gear and coupler assemblly mechanism. Another drawback is the lack of a graduated release. **[2]**

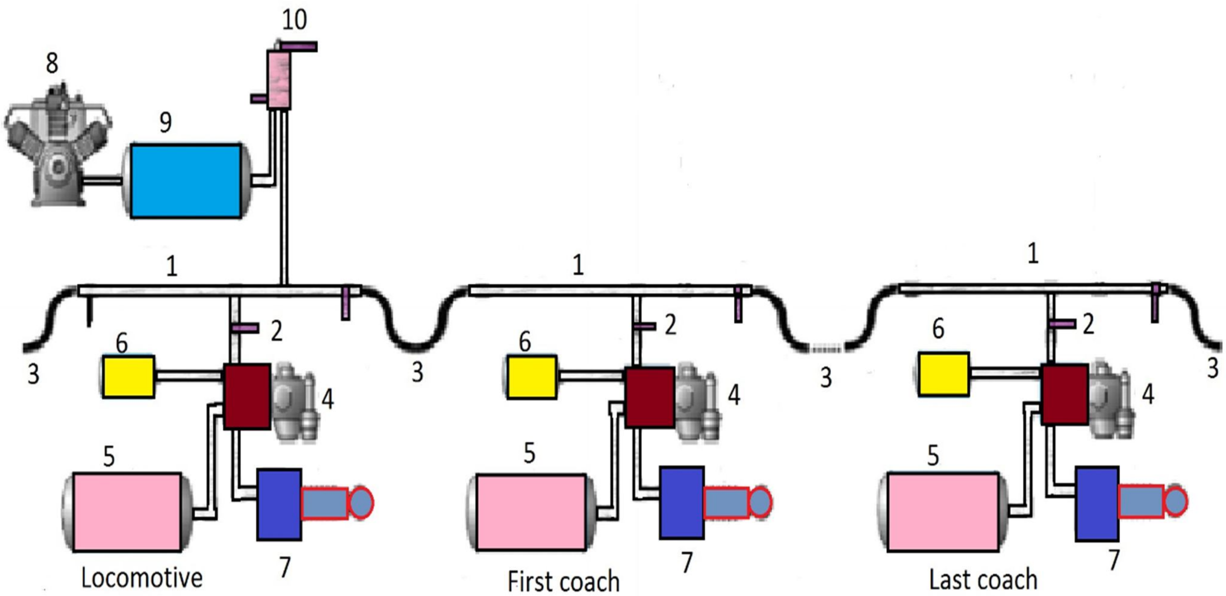


Figure 1.2 Simplified schematic of the pneumatic brake **[2]**

Component number with their respective names

1) Train pipe, 2) cut off cock 3) coupling hose 4) distributor valve 5) auxiliary reservoir

6) Service reservoir 7) brake cylinder 8) compressor 9) main reservoir 10) drivers brake valve

As observed in fig 1.2 above all the components of each coach are found on the locomotive. The only difference between them is that in the locomotive there are additional components which are not found on the coaches. These components are; compressor, main reservoir and drivers brake valve.

1.1.1.1. Working principle of pneumatic braking system

The working principle of the pneumatic braking system shown on fig 1.2 above is the same in all coaches and on the locomotive. at braking time when the driver select valve 10 to braking position, the train pipe pressure exhausts to the atmosphere as a result pressure difference is occurred between service reservoir and train pipe on the distributor valve so the main valve piston is displaced to the braking position causing compressed air to pass from auxiliary reservoir to the brake cylinder. At this time all the coaches including the locomotive decelerates or even stops at different time interval causing passenger discomfort and damaging of some components.

At release time when the driver put valve 10 on release position the train pipe is charged with compressed air from the main reservoir to a maximum pressure that causes a pressure difference between the service reservoir and train pipe on the distributor valve, as a result the main valve piston displaces to the release position causing to cut the passage between auxiliary reservoir and brake cylinder and to open a passage from the brake cylinder to the atmosphere. The same as in braking time here also there is a time delay of compressed air from the leading locomotive to the last coach.

Draw backs of pneumatic braking system

Uneven serial brake applications

In longer trains (more than 2 miles in length) this relatively slow release of air can cause the cars near the locomotive to begin brake application as much as 90 seconds sooner than cars near the end of the train **[29].**

Braking force that is not proportional to car loading

Regardless of their weight, all cars in a train will reach the same brake cylinder pressure for any given brake application. This can result in locked (and eventually flat) wheels in lighter cars and insufficient braking force in heavier cars **[29].**

Slack action due to in-train forces

As cars begin braking at different times and heavier cars maintain their speed longer than lighter cars, they will begin to run into each other, causing compression and stretching of the knuckles and couplers **[29].**

Lack of graduated release

Because all releases are full releases, an engineer who applies too much braking power in an attempt to control speed must fully release the brakes before he/she can re-apply them to the correct level. On graded terrain, this can cause the train to slow down too much, or even come to a complete stop, in the middle of the slope. The engineer must then apply locomotive power to pull the entire train off the slope while the cars are still braked. To avoid this situation, an engineer will often partially apply the air brakes and use the dynamic brake to “top off” the braking force. In either case, extra energy must be expended, in the form of locomotive engine fuel, to control the speed of the train. **[29]**

Lack of performance monitoring capabilities

The air communications in conventional pneumatic systems is one-way from the locomotive to the cars, making real-time brake system monitoring and diagnostics impossible. Due to this an improved type of braking system that over comes the above problem is created **[29].**

1.1.2. Electro pneumatic braking systems

Due to the above disadvantages or limitations listed under pneumatic braking system and the rise of rapid transit operations in cities, with their high volume and frequent stops and starts, that requires quick responses to brake commands and accurate stopping at stations it was an essential ingredient in getting more efficiency braking system. Due to this electro pneumatic braking system was first appeared in the US. They were also tried on the New York Subway in 1909 and then on London Underground in 1916 **[9].**

Electro-pneumatic braking system is originally designed for subways or metros, and it has more recently been used on main line passenger railways and some specialized freight operations.

Here the main difference and similarities between pneumatic and electro pneumatic braking systems are summarized in the table below.

Table 1.1 similarities and difference b/n pneumatic and Electro pneumatic with respect to their components.

Brake system / components	pneumatic	Electro pneumatic	remark
compressor	✓	✓	To compress air
Main reservoir	✓	✓	To hold air for the whole train
Drivers brake valve	✓	✓	For actuating the system
Electrical unit		✓	Developing power
Train pipe	✓	✓	Way for the air
Branch pipe	✓	✓	Branching air to each coach
Service reservoir	✓	✓	Holding air for individual coach
Auxiliary reservoir	✓	✓	Air for braking
Distributor valve	✓	✓	Pressure distributor

Electromagnetic valves		✓	For actuating the distributor valve
Cut off cocks	✓	✓	To cut the passage
Brake cylinder	✓	✓	Force developing area
Fluid (air) in the system	✓	✓	Input for the system
Command transmitted	pneumatically	electrically	

The components of electro pneumatic braking system with their arrangements are clearly shown in the diagram below.

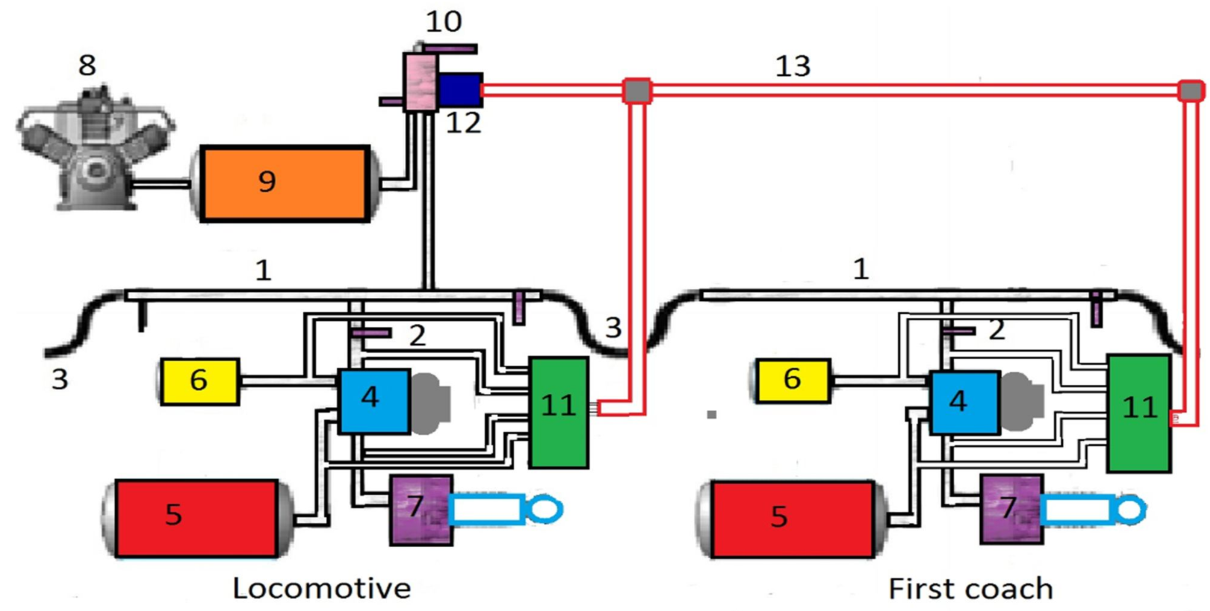


Fig1.3. Schematic diagram of train electro pneumatic braking system

1) Train pipe, 2) cut off cock 3) coupling hose 4) distributor valve 5) auxiliary reservoir 6) Service reservoir 7) brake cylinder 8) compressor 9) main reservoir 10) drivers brake valve 11) Electromagnetic valve box 12) electrical unit 13) cable holding pipe

The cable holding pipe 13 is a pipe which is used to hold the electrical wires throughout the length of the train to protect from any damaging equipments. And its detail schematic diagram is shown in fig 1.4 below

The cable connection and positions of electro pneumatic braking are shown in fig 1.4 below**.**

The numbers indicated on fig 1.4 shows

The 1st cable indicates service brake application position,

The 2nd cable indicates release application position, The 3rd cable indicates pressure holding position,

The 4th cable indicates emergency brake application position and

The 5th cable indicates the negative or ground

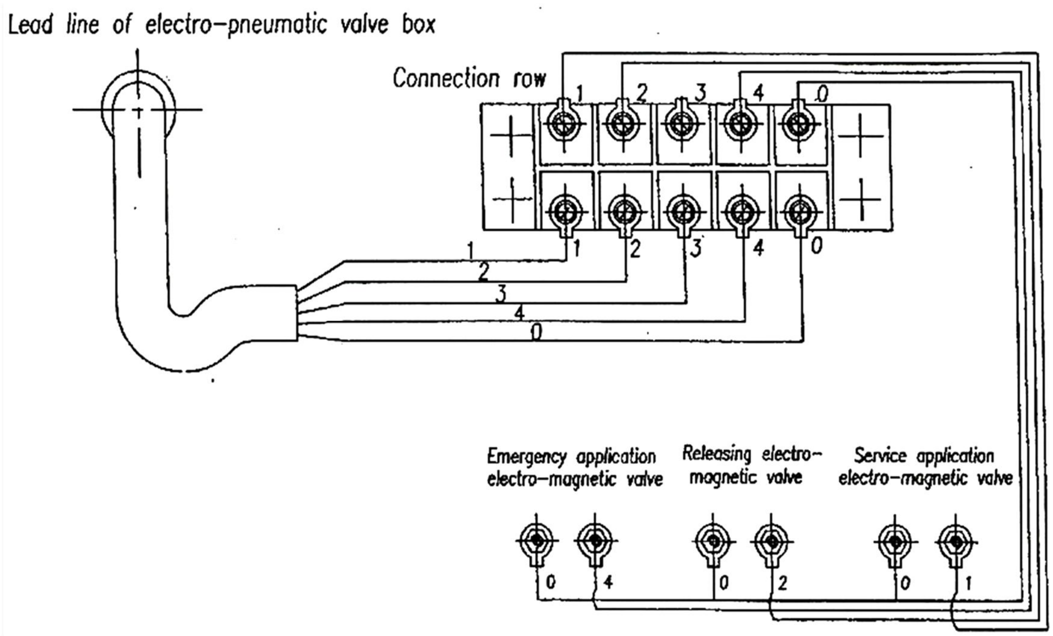


Fig1.4. the electrical systems of electro pneumatic brake **[9]**

The following table shows the relationship between function of electromagnetic valve and position of drivers brake valve.

Table1.2. relation between function and position of electromagnetic valve and drivers brake valve **[9]**

function / position	Service application	release	Emergency application
Service application Electromagnetic valve	✓		✓
Release electromagnetic valve		✓	
Emergency application electromagnetic valve			✓

In the above table (✓) indicates the indicated electromagnetic valve is excited

1.1.2.1. Working principle of train electro pneumatic braking system

The working principle of electro pneumatic braking system shown on fig 1.2 above is almost the same as that of pneumatic braking system except the command in electro pneumatic braking system is transmitted electrically and there are additional electromagnetic components in electro pneumatic braking system which is not found in pneumatic braking system as shown on the above table.

Service brake application position for the locomotive and all coaches

In service brake application when the driver select the drivers brake valve on service application position the service electromagnetic valve located on the locomotive and on all the coaches at different distance from the driver are excited electrically at the same time causing compressed air in the train pipe to exhausts to atmosphere through the service electromagnetic valve at the same time.

Since the pressure in service reservoir and in train pipe makes the distributors main valve piston in equilibrium, up on exhausting the train pipe pressure the main piston of distributor valve moves upward causing the passage between auxiliary reservoir and brake cylinder to open. Compressed air from auxiliary reservoir flows to brake cylinder as a result service braking action is performed.

Release application position for the locomotive and all coaches

In release application position when the driver select the drivers brake valve on release application position the release electromagnetic valve located on the locomotive and on all the coaches at different distance from the driver are excited electrically at the same time causing compressed air from train pipe and service reservoir flow to upper side of distributor main valve piston.

At this time the distributor main valve piston is in equilibrium by the dawn ward pressures of brake cylinder and train pipe and upward pressure of service reservoir. At this time the cumulative downward pressure is greater than up word pressure. As a result the main valve piston displaces downward causing the release valve located at the bottom of distributor valve to open. The compressed air in the brake cylinder exhausts to the atmosphere.

Emergency brake application position for the locomotive and all coaches

In emergency brake application when the driver select the drivers brake valve on emergency brake application position the service electromagnetic valve and emergency electromagnetic valve located on the locomotive and on all the coaches at different distance from the driver are excited electrically at the same time causing compressed air in the train pipe to exhausts to atmosphere through the service and emergency electromagnetic valves at the same time.

Since the pressure in service reservoir and in train pipe makes the distributors main valve piston in equilibrium, up on exhausting the train pipe pressure the main piston of distributor valve moves upward quickly causing the passage between auxiliary reservoir and brake cylinder to open. Compressed air from auxiliary reservoir flows to brake cylinder as a result service braking action is performed.

By observing from the above discussions and diagrams since the command is transferred from the leading locomotive to the last coach by electrical signals the same amount of pressure is developed in the brake cylinder of each coaches at the same time regardless of their position. So the same amount of longitudinal braking force is developed on each oaches causing the whole train set to brake at the same time. This too smooth braking minimizes the coupler and draft gear damage.

Advantage over the air brake

its speed of control**,** quick on-vehicle reaction times and giving instantaneous control of the whole train to the driver**,** reduce longitudinal train shock in braking, reduce the braking distance, improve the release performance of the train and if once the electronic control device is out of order; the brake system can still be controlled pneumatically. The charging and recharging of train pipe in electro pneumatic braking system is the same as the pneumatic braking system. Since the compressed air is feed from the main reservoir to all the coaches there is a time delay to propagate the air from the leading locomotive to the last coaches.

Therefore in this paper the electro pneumatic braking system of any coach is simulated in chapter four. And the detail system components used in the simulation with their working principles are discussed with clear diagrams in chapter three.

Table below, summarizes the features and benefits of the EP braking system over pneumatic braking system **[30]**

Table 1.3 Advantages of train E-P braking over pneumatic

S/N	Description of Feature and Benefit	pneumatic	Electro pneumatic
01	**Safety** - if all else fails, the train will stop	✓	✓
02	**Simultaneous braking** - because of the digital communications medium, all cars brake simultaneously, resulting in reduced brake shoe and wheel wear.		✓
03	**Shorter stopping distance** – simultaneous brake application means all cars will set quickly and the train will reach maximum target braking within seconds. Improves cargo throughput by allowing for greater running speeds within existing signal block territories.		✓
04	**Uniform braking** - each car brakes according to its weight, reducing in-train forces by avoiding wheel slip on lighter cars and unbalanced braking on heavier cars.		✓
05	**Graduated apply and release** - improved train handling, saves fuel costs by allowing air brake system to share more of the braking load		✓
06	**Reduced shock-** the train set is stopped at the same time causing coupler and draft gear assemblly shock to be reduced		✓
07	**Speed of control-** due to electrical command any accident is controlled in a short period of time		✓

1.2. Statement of the problem

The dynamic fluid movement inside the electro pneumatic braking system components is a complicated process that needs a special simulating tool to understand its motion.

The experimental values of the electro pneumatic braking system can be found from different literatures. But it is different for different drivers brake valve position. The variables of the system are not also clearly shown by how much they vary. So it is essential to simulate the system at these different derivers brake valve position to know whether the experimental values are the same with the simulated value or not and how the variables vary with drivers brake valve position. The time delay of compressed air from the locomotive to the last coach in pneumatic braking system is shown from previous researches and it has a negative impact on train system components like coupler and draft gear assembly. And this research will clarify the difference in time delay between pneumatic and electro pneumatic braking systems.

1.3. Objective of research

1.3.1. General objective:

The general objective of this research is to simulate the braking response of fluid dynamics on train coach electro-pneumatic braking system at different conditions of brake valve position located at a distance from the locomotive and comparing the simulated results with pneumatic braking system to see the effect of electric and air pressure command on time delay of compressed air to the braking chambers.

1.3.2. Specific objective

The specific objectives of this train vehicle electro-pneumatic braking system simulation include:

- ❖ To know the maximum longitudinal braking force developed in service and emergency brake application positions.
- ❖ To know the maximum pressure developed inside brake cylinder at service and emergency brake application positions and their respective time taken to develop those maximum pressures.
- ❖ To estimate the time required to start braking at service and emergency application position for the locomotive and coaches.

- To know the velocity and position of the piston cylinder on the braking chamber at different braking conditions.
- To know the relations between pressure in the system and main valve piston velocity and position.
- This also will show the volumetric orifice flow rate of compressed air at different drivers brake valve position indicated by:
    - Release position
    - Service brake application and
    - Emergency brake application

1.4. Research methodology

Research methodology as its name indicates is the method used to solve the problem identified in the statement of problem. So the methodology used to accomplish the specified problem here in is:
By browsing different literatures i.e pneumatic and electro pneumatic braking system related books, researches done before as indicated in the reference.

Procedures [understanding the system, modeling the system simply in block diagram to show fluid motion with its directions, modeling with matlab simulink, simulating the model, displaying the result and finally comparing the simulated result with the experimentally result found from the literature review].

Finally the general simulation output of the electro pneumatic braking system is compared with the pneumatic braking system to see the effect of commanding signals on time response of braking i.e to observe the time delay of braking system on long trains.

1.5. Scope of research

The electro pneumatic braking system is a complicated process due to the number and structure of brake system components. The whole system components could not cover in one thesis paper due to the above reason. So this paper will not cover the whole braking system components in the simulation part rather a selected brake system components that could achieve the specific objective are simulated.

In any electro pneumatic brake system all the coaches including the locomotive have the same components except some components that found only on the locomotive. And the commanding signal from the locomotive to each coach is electric current as a result no time delay to actuating the electromagnetic valve found on all coaches.

This paper is limited to simulation of fluid dynamics across the components of auxiliary reservoir, service reservoir, distributor valve and brake cylinder as if the electromagnetic valves are actuated by the driver on the selective brake valve position (service brake, release and emergency brake valve positions).

1.6. Limitation of the study

The limitation of this study is that there is no skill full person that can elaborate the working principle of all system components. And there are no enough reference books of railway technology. This paper has done by browsing different literatures from internet. Since the author haven’t seen the train electro pneumatic braking system practically it seems like theoretical thesis. All the electro pneumatic braking system components with all their dimensions could not found on one compiled document rather from different literature reviews.

1.7. Thesis organization

In chapter one (1) the background of train general braking systems and particularly pneumatic and electro pneumatic braking systems with their respective diagrams are discussed. In this chapter the problem of the statement and the main and specific objectives of this electro pneumatic brake simulation are also discussed clearly.

In Chapter two (2) a related literature review is given on simulation of train pneumatic braking system performance and longitudinal force evaluation on last and first coaches on different modeling and simulating environments.

In chapter three (3) important data and equations are collected for the simulation of electro pneumatic braking system and simple schematic diagram of the main electro pneumatic braking system components with their fluid directions at different brake valve positions is shown.

In chapter four (4) the braking system is modeled and simulated in matlab simulink modeling and simulating working environment.

In chapter five (5) the simulated result in chapter four is discussed clearly with the help of diagrams. In this chapter the simulated results are compared with the actual measured data found from experiment to state weather the simulated model is modeled correctly or not.

In chapter six (6) the whole research is concluded shortly and briefly. And there is a future work recommended to be done by anyone who has an interest on this field of specialization.

CHAPTER-II

LITERATURE REVIEW

Luca Pugi, Duccio Fioravanti, Andrea Rindi **[3]** have modeled the longitudinal dynamics of long freight trains during the braking phase.

The main aim of these guys was to know how train composition, buffers mechanical Features, braking plant design and other key parameters may influence the braking action in long trains.

In long trains compressed air is the only fluid for commanding the system components and applying the brake in brake cylinder from different reservoirs. The fluid in pneumatic braking system is distributed from the main reservoir located on the locomotive to all vehicles found at different position from it. So this fluid is propagating with a certain speed inside train pipe and takes a certain time to propagate from the front locomotive to the rear vehicle. At braking and releasing time due to the propagating time of compressed air inside train pipe the vehicles which are found at front and rear of the train brakes at different braking time.
As a result of this, the vehicles found at the front gets large braking force than at the rear that may cause damaging of wagons or even derailment of train.

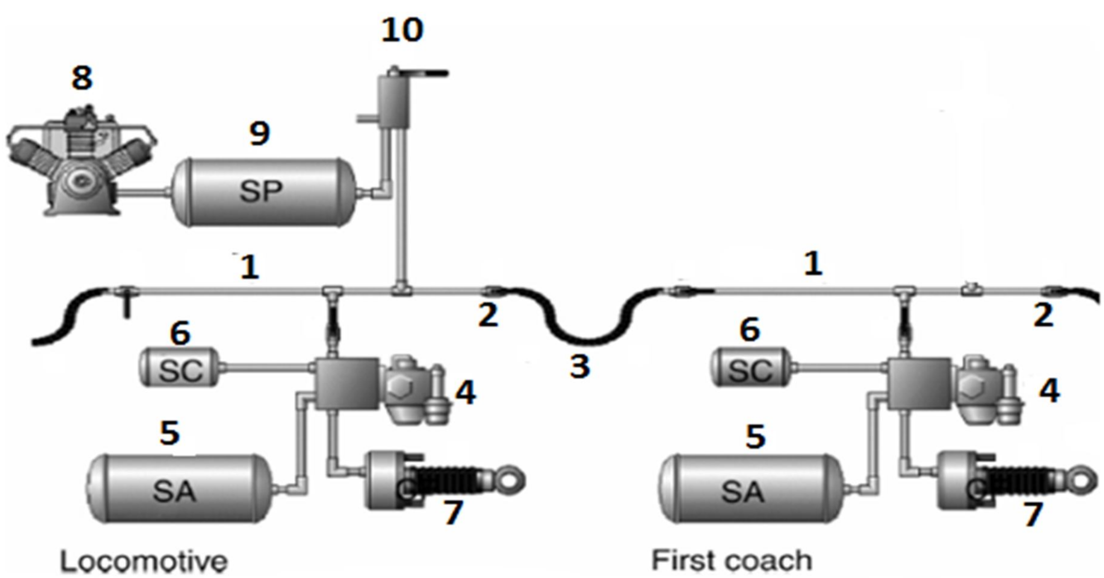


Figure 2.1 Simplified schematic of the pneumatic brake **[3]**

In this particular study two different aspects were emphasized

1) Modeling the braking signal along brake pipe as a depression wave.

2) Distributor response

The modeled train pipe here has a dimension of 300m length, 25mm inside diameter, 0.01 relative roughness and pipe pressure of 600kpa **[2].**

The pipe described in figure below is in effect a continuous system with infinite modal shapes and natural frequencies that can be evaluated by the following relation (adiabatic small perturbations)

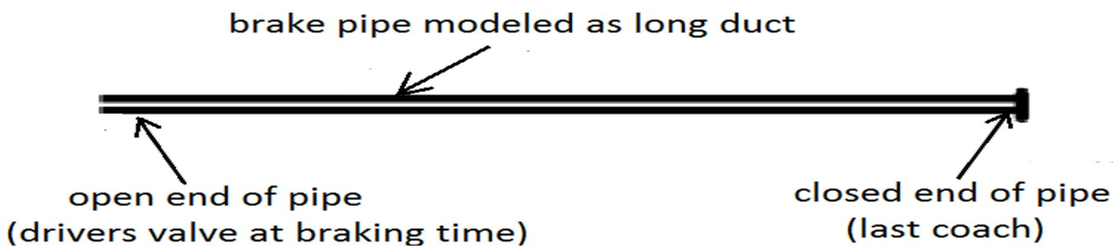


Figure 2.2 Simplified schematic of train pipe to be simulated **[2]**

Up on simulating the system the following results were found diagrammatically

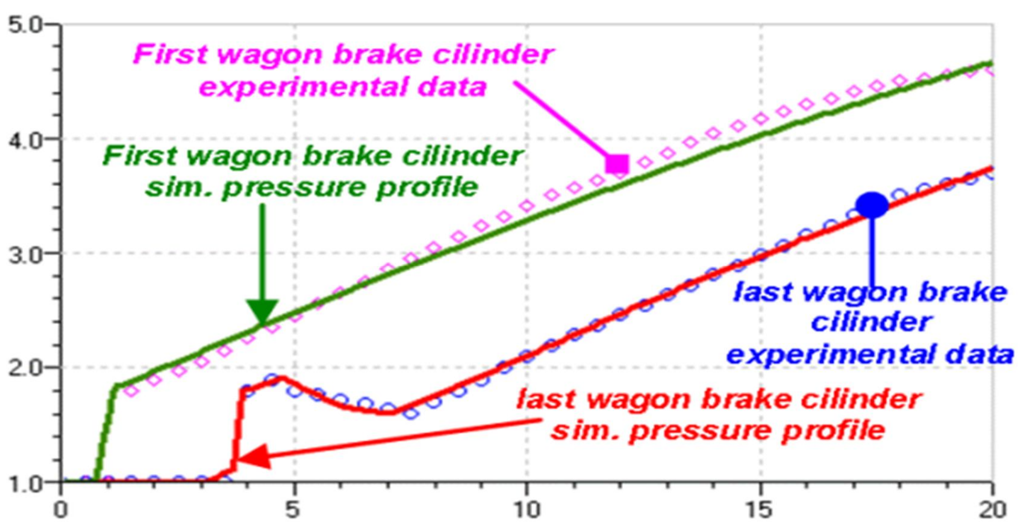


Fig.2.3 Comparison of brake cylinder pressures at braking time (experimental/simulated) with respect to simulation time **[2].**

The simulating environments here are advanced modeling environment for simulation (Amesim) and E-train^{M} soft ware.

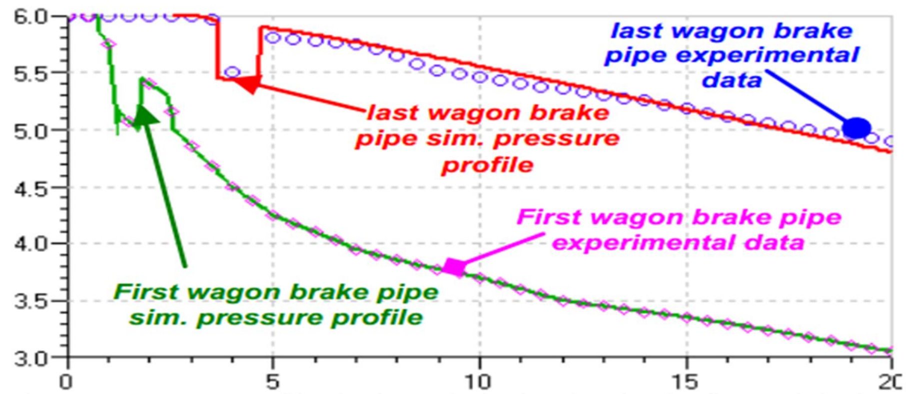


Fig2.4 Comparison of brake pipe pressures at releasing time (experimental/simulated) with respect to simulation time **[2].**

As shown on the above diagrams the braking and releasing action of air brake system is fast on the 1st coach and slower on the rear one.

From fig 2.3 the brake cylinder of the first wagon develops 4.4 bar of compressed air at a simulation time of 20 sec. but for the same simulation time the last wagon develops brake cylinder pressure of 3.75 bars. These simulated wagons are located around 300m from each other.

From fig 2.4 the maximum pressure of train pipe (6 bars) on the first wagon takes to release 20sec whereas on the last wagon on the same simulating time only 2.4 bar of compressed air is released.

On the same manner Luca Pugi , Alessandro Palazzolo, Paolo Presciani, Duccio Fioravanti **[2]** also has done on the Simulation and optimization of railway pneumatic braking system.

The aim of this paper was in the performance and reliability of railway braking systems that affects in many different way both vehicles and infrastructure design features. And the same results were found by modeling the system on different modeling and simulating environments like (E-train™ and AMESim) and by using the same boundary conditions.

Cantone L., Durand T. **[30]** has done a paper on longitudinal forces evaluation of SNCF trains. The aim of this paper was the investigation of Longitudinal Forces exchanged between two consecutive vehicles of a train in long train set.

This paper has tried assessing of train compositions, since it affects train suitable length, applicable traction power, load capacity and permissible speed, especially for heavy hauled freight trains.

The modeling was carried out on matlab simulink. At the end this author has found large longitudinal braking forces caused by time delay of compressed air from the leading locomotive to the last wagon. And he concludes especially for heavy hauled freight trains the coupler and draft gear mechanism that affects by the shock of large unbalance longitudinal force has to be manufactured from hard materials.

Viktor SZENTE, János VAD **[4]** has also modeled Computational and Experimental Investigation on dynamics of Electric Braking Systems. This Electric Braking System (EBS) presented herein comprises an air supply unit, an EBS modulator (distributor), piping, a diaphragm brake chamber, and the connected brake mechanism. This simulation is applied for commercial motor vehicle. The aim of this paper was to understand the fluid dynamics inside the above described components. In addition to this the appropriateness of the simulating environment is also evaluated.
The system is modeled in the simulating environment called AMESim. The paper presents a computational simulation tool used for prediction of dynamic behavior of EP brake systems.
Finally this paper concludes since the experimental results are the same with the results found from the simulation this simulating environment is appropriate for this particular brake system. And a clear view of fluid dynamics inside these brake system components is observed.

BehrouzNajjari, S. Masoud Barakati, Ali Mohammadi, Mohammad Javad Fotuhi, SaeidFarahat, and Mohammad Bostanian **[5]** have done a paper on modeling and Controller Design of Electro-Pneumatic Actuator Based on PWM. This paper has divided the whole system in to subsystems as electrical, magnetic, mechanical and fluid. The aim of this paper was to design a controller up on simulating the system. The control cylinder position is simulated Using programmable logic controller (PLC) and simulating environment of matlab simulink.

Finally the system is simulated including the proposed controller by using the given boundary conditions and compared with the experimental results.
Therefore since the simulated result is similar with the experimental found results the proposed controller design is appropriate for this particular system.
Izabella ICHIM, Catrina CHIVU **[6]** has also done a paper on the simulation of pneumatic directional valve.

In this paper the author wants to see the effect of accumulator volume on the response time of the directional valve.

Mainly the author was developed the circuit to compare the dynamic behavior of different pneumatic cylinders and their applicability on material handling.
From the simulation result the author observed as the diameter and stroke of the cylinder becomes larger and larger the response time of the directional control valve becomes higher and higher. So for the same supply pressure decreasing the volume of the actuator cylinder results in short response time.

The design and simulation is done using oriented software that includes predefined elements associated with pneumatic components. The simulation is done using a tool box of matlab software simscape.

Conclusion

From the above literature reviews, papers [2], [3] and [30] have discussed the effect of train length on the propagation of compressed air throughout the length of the train in pneumatic braking system.

Almost all the other papers discussed above try to understand the fluid dynamics around the critical components of pneumatic braking system.

Therefore in the present paper the simulation of train electro pneumatic braking system will be discussed to clearly show the advantage of electric command in electro pneumatic brake system rather than compressed air signal in pneumatic braking system in the three drivers brake valve position (service brake application, release application and emergency brake application). The same as air brake in braking time, the initial charging of electro pneumatic brake system components response time located on different coaches of the train set are different. That means there is a time lag in the initial charging of electro pneumatic braking system reservoirs located on each coach. Here the coach nears the locomotive charges before the one at the rear.

The main target here is to observe the response time braking at different brake valve positions and comparing with the pneumatic one.

CHAPTER-III

SYSTEM COMPONENTS AND THEIR WORKING PRINCIPLES

Before directly proceed to the simulation of the electro pneumatic braking system of train vehicle, it is better to discuss the structure and detailed sectional view of the components to be simulated.

3.1. System components

3.1.1. Distributing valve

Distributor valve is the most important functional component of the electro pneumatic brake system and is also sometimes referred to as the heart of the E.P brake system. The function of the distributor valve is to distribute compressed air received from brake pipe to all system components.

In addition to this it also senses drop and rise in brake pipe pressure for brake application and release respectively. It is connected to break pipe through branch pipe. Various other components connected to the distributor valve are auxiliary reservoir, brake cylinders and service reservoir.

3.1.1.1. Structure of Distributing Valve

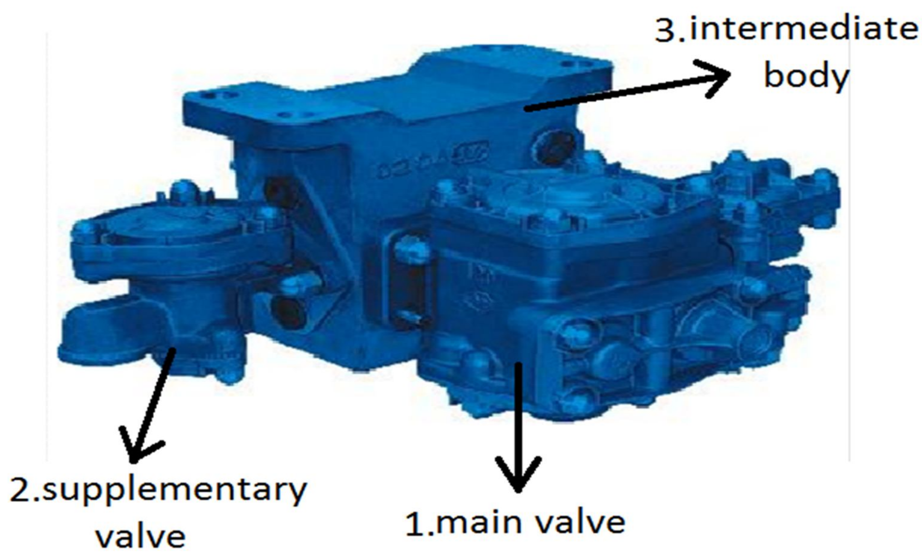


Fig 3.1 Structure of distributor valve with its main parts

Three important elements distributor valve

1. Main valve
2. Supplementary valve
3. Intermediate body

3.1.1.2. Sectional view of distributor valve

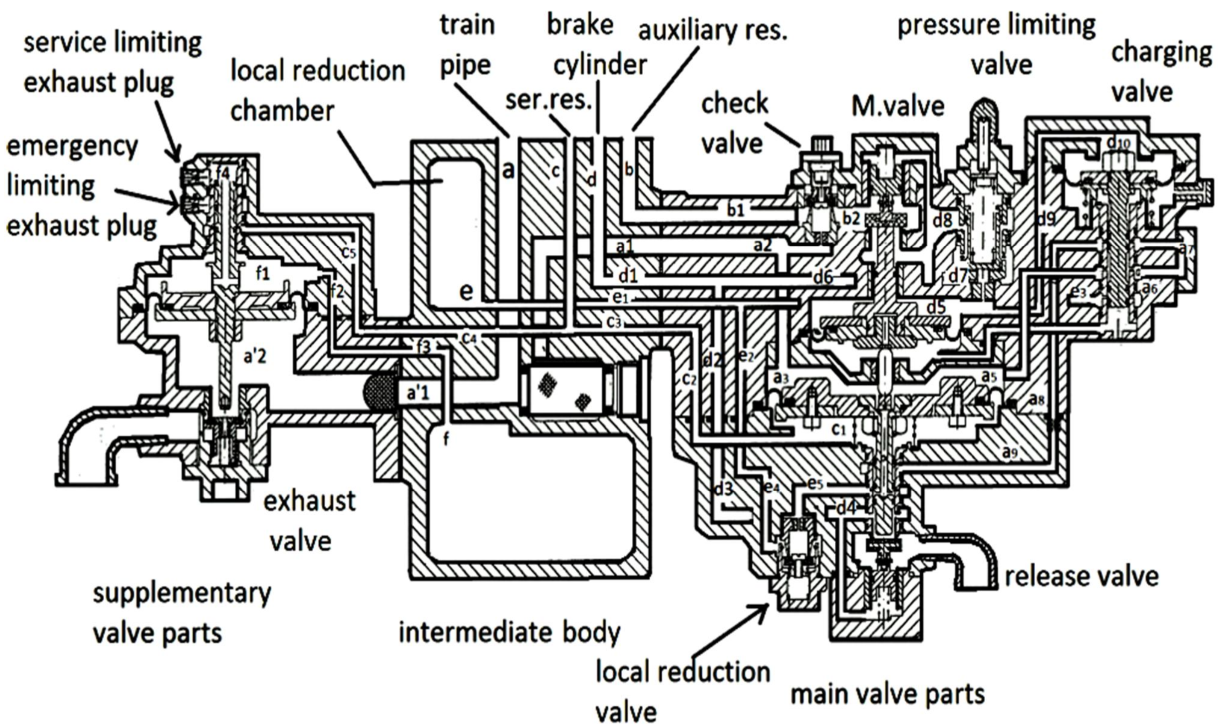


Fig 3.2 Sectional view of distributor valve **[9]**

The alphabets labeled on the above distributor valve diagram are used to show the directions of fluid flow inside the distributor valve.

On the system working principle these letters are omitted and these letters are replaced by coloring of fluids. So if any reader has a doubt on the direction of fluid motion up on braking releasing, it is better to refer to the above diagram in which the direction is clearly labeled by these letters and discussed on the working principles.

Main valve piston

The main operation of braking and releasing is performed on the main valve due to the triple pressure difference created on both sides of the distributor main valve piston. So the critical dimension of the main valve piston with its piston rod and diaphragms is labeled clearly on the following diagram. The labeled dimensions are given on chapter four of table 4.1.

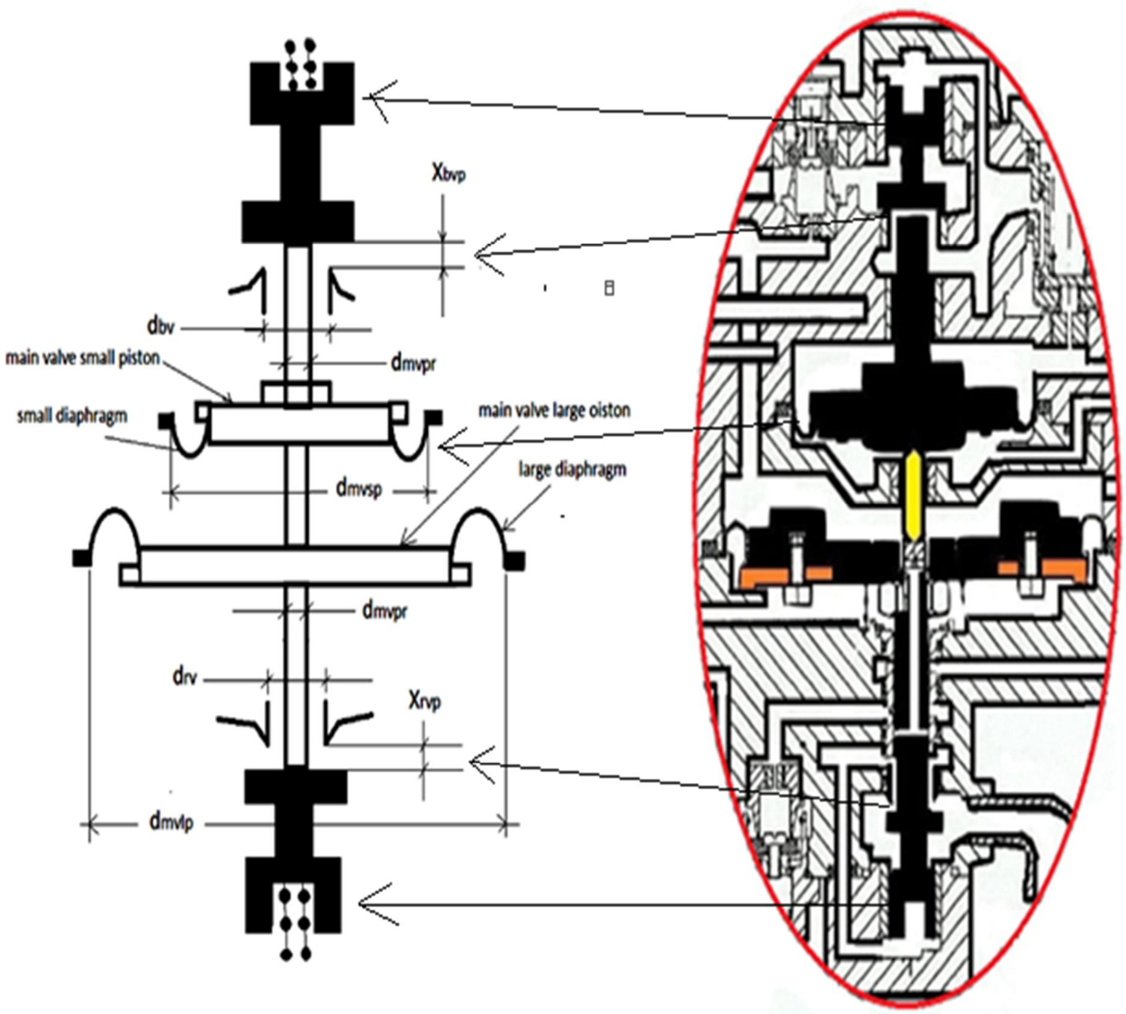


Fig 3.3 Schematic diagram of distributor valve main piston

3.1.1.3. Main valve

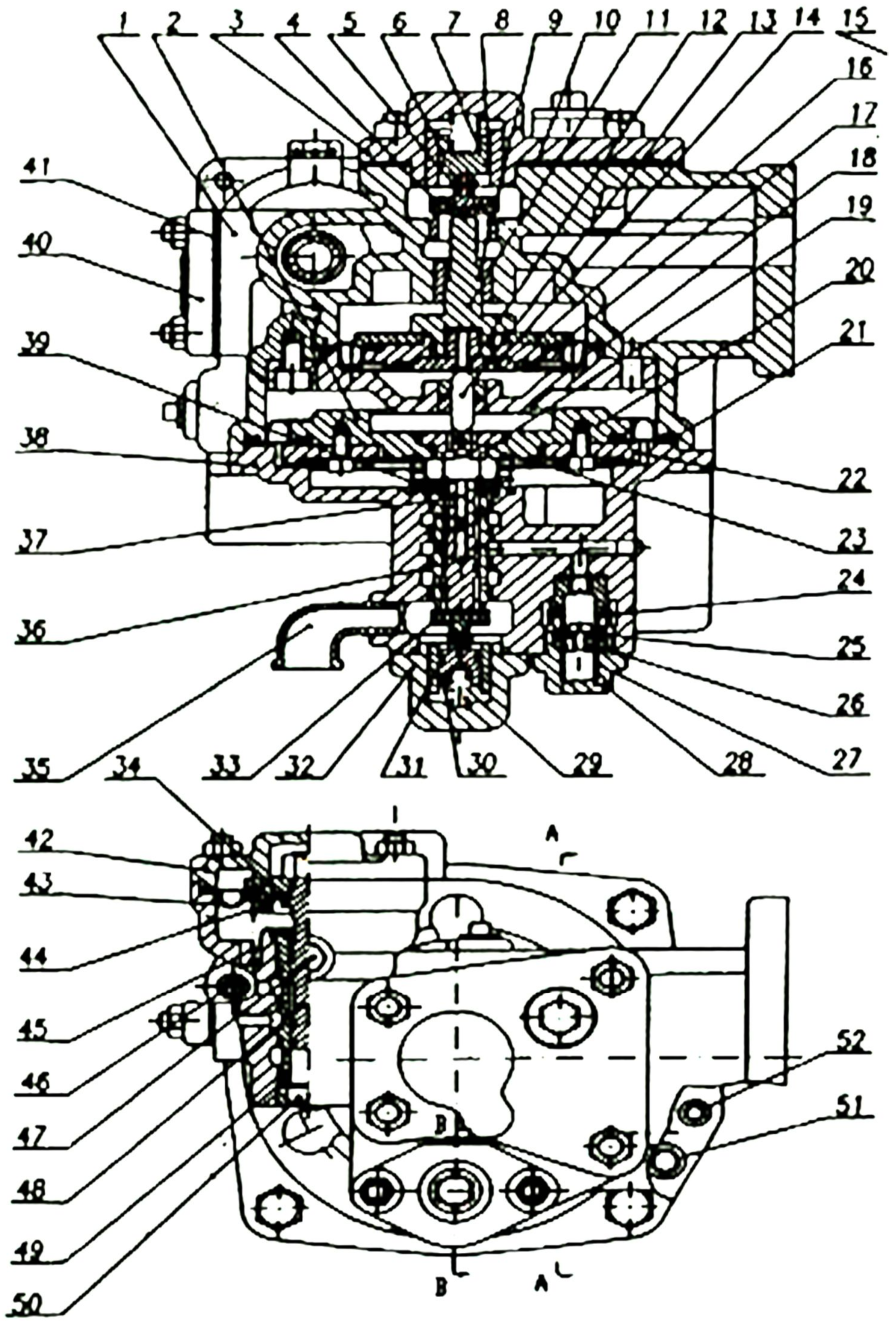


Fig3.4. Structure of main valve with its components **[9]**

The main valve part is composed of main valve, charging valve, pressure limiting valve, auxiliary reservoir charging check valve, local reduction valve, change-over cover plate etc.

Main valve

The main valve is composed of balance valve assembly 4, main valve rod 11, small piston 17, small diaphragm 18, main piston 20, large diaphragm 21, local reduction valve sleeve 36, and release plunger 37, brake spring 39, release valve 33 etc.

The main functional part of the distributor valve is the main valve and it is a triple pressure equilibrium mechanism, the both sides of the main piston are respectively compressed air of service reservoir and that of train pipe, the upper side of the small piston is the pressure of brake cylinder, the lower side communicates with atmosphere. Through equilibrium action of the triple pressures, the basic actions of brake application, holding and release occur.

Basic actions of the main valve:

Brake application action

After pressure reduction is applied on train pipe, certain pressure difference value is formed between service reservoir and train pipe respectively on both sides of main piston under the action of compressed air from service reservoir, the main piston is moved upward, carrying main valve rod 11 to move upward, and to open balance valve 4 resulting in compressed air from auxiliary reservoir entering brake cylinder.

Pressure holding action

After the train pipe stops to reduce pressure, (in case of graduated release, after train pipe stops to increase pressure), the pressure in brake cylinder rises (or drops) until equilibrium holds among pressures from service reservoir, train pipe and brake cylinder, then the balance valve 4 is closed, now the brake cylinder pressure is hold on certain value, and the main valve is thus in a pressure holding state.

Release action

When the pressure in train pipe is increased, the resultant force of train pipe and brake cylinder is greater than force of service reservoir air pressure acting on main piston 20, resulting in downward movement of main piston to open release valve, causing the compressed air from brake cylinder exhausting to atmosphere through release valve assembly 33, then the main valve is situated in release position.

Train pipe pressure reduction action

When pressure in train pipe is reduced and the main piston just begin to move upward, the release plunger 37 then moves upward to open communicating passage between train pipe and local reduction valve sleeve, the compressed air from train pipe goes through intermediate hole on release plunger, local reduction valve sleeve 36, opens check valve 26 and enters local reduction chamber in intermediate body resulting in occurrence of local pressure reduction action in train pipe so as to promote the main valve to take action quickly to enter brake application position , this can also promote transfer of train braking valve propagation

Change-over between direct release and graduated release

The hang-over cover plate 40 can be used on both positions for direct release or graduated release to change-over installation. When the cover plate 40 is installed in “graduated release position”» the distributing valve is provided with graduated release performance. When distributing valves without graduated release action is mixed with those with graduated release action to be operated in a train composite, the change-over cover plate 40 should be placed on “direct position”,

Now when release is applied, the compressed air from service reservoir can flow through local reduction valve sleeve 36, holes and slots on change-over cover plate and directly enters train pipe to attain the purpose of direct release. Due to the fact that in release, the compressed air from service reservoir directly and quickly enters train pipe, it not only raises the-recharging rate of train pipe, but also speeds up the release action of main valve, this is meaningful to speed up release on rear cars on longer composite train.

3.1.1.4. Supplementary valve

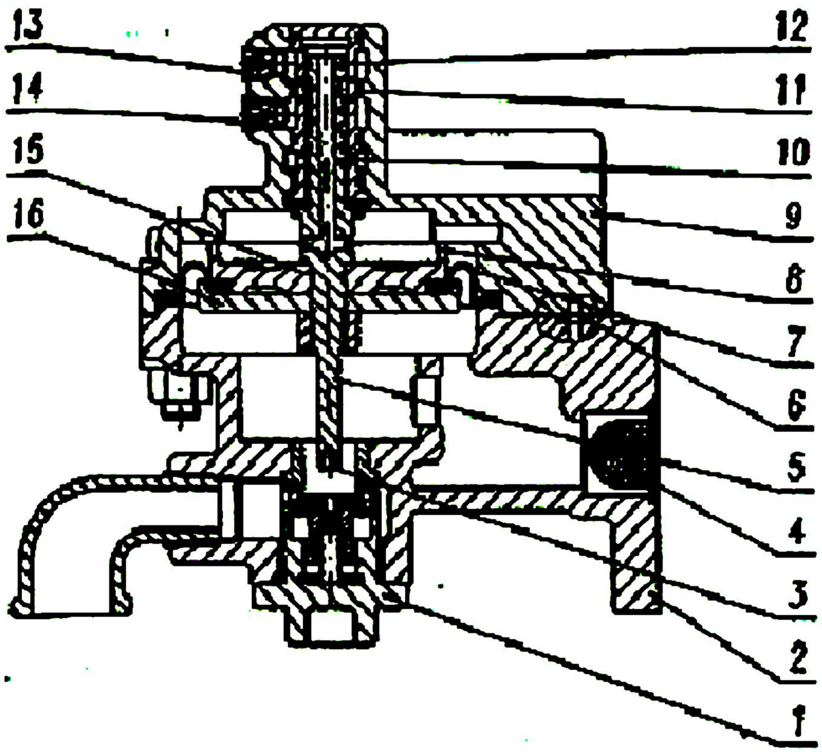


Fig 3.5 Supplementary Valve Part of Passenger Car distributing Valve F8 **[9]**

Where1.Emergency exhausts valve assembly, 2.Supplementary body assembly, 3.Contact, 4. Filter nut, 5.Supplementary valve rod, 6.Rubber shim, 7.Diaphragm, 8.Supplementary valve piston, 9.supplementary valve upper cover, 10.Supplementary valve sleeve, 11. Exhaust valve o-ring, 12.Valve rod o-ring, 13.Service exhaust plug, 14.Emergency exhaust plug, 15.Plunger o-ring, 16.Piston pressing plate.

The supplementary valve is mainly composed of supplementary valve piston 8, supplementary valve sleeve 10, O-ring 11 and 12, supplementary valve rod 5, service exhaust plug 13, emergency exhaust plug 14, emergency discharging valve assembly 1 etc.

The supplementary valve structure is a dual pressure balance valve structure, The upper side of supplementary valve piston is supplementary chamber compressed air, the lower side is train pipe compressed air, The compressed air coming from main valve can be charged into supplementary chamber and to upper side of diaphragm through passage on upper side of supplementary valve, the compressed air in supplementary chamber and that in train pipe becomes in equilibrium, In brake application due to pressure reduction in train pipe, the supplementary valve reaches brake application position, the compressed air in supplementary chamber exhausts to atmosphere through service exhaust plug% 13 and emergency exhaust plug.

Functions of supplementary valve

To speed up release action after brake application

During train pipe being charged and released, the compressed air of service reservoir a-gain charges into supplementary chamber through supplementary valve sleeve 10 and communicating slots on supplementary valve rod 5, the pressure in service reservoir drops rapidly, so that the release action of main valve is speeded up.

Emergency exhaust action

When train pipe exhaust with emergency exhaust rate, the supplementary chamber pressure pushes piston 8 and supplementary valve rod 5 to move downward to open emergency exhaust valve on lower side to produce emergency exhaust action.

3.1.1.5. Intermediate body part

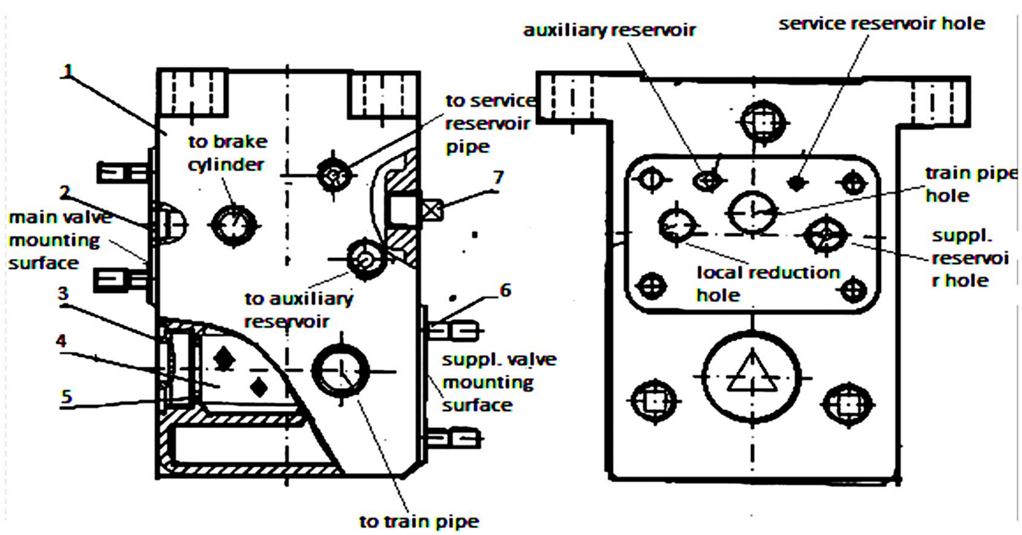


Fig.3.6. Intermediate Body Assembly **[9]**

The intermediate body part is mainly composed of intermediate body 1, threaded plug 2, threaded cover 3, filter valve 4, dust guard shim 4, stud etc.

In the intermediate body, there are two air chambers, one is local reduction chamber with capacity 0.8 liter another one is supplementary chamber with capacity3 liters. Besides there are several internal air passages. On internal air passage for train pipe, a round barrel from filter net is installed, for filtering compressed air which enters main valve- On outside of dust guard net, threaded cover is installed to be used for the convenience of cleaning the filter net.

The main valve and supplementary valve are separately installed on both sides of intermediate body, the upper plane of the intermediate body is provided with four holes which are the hanging holes for hanging the intermediate body under car body.

On one side of the intermediate body, there are pipe taper threaded holes for connecting pipe lines under vehicle; these individually connect train ‘pipe 25mm auxiliary reservoir 18mm, brake cylinder 18mm, and service reservoir 12mm.

3.1.2. Brake cylinder

The train brake cylinder is one of the most important components of the electro pneumatic braking system in which the compressed air is changed in to longitudinal braking force.

The type of brake cylinder is single acting actuator.

The vehicle brake actuator used for both locomotive and the coaches in this particular study is the same in all aspects. The brake cylinder has a volume (V_{bc}) and inside diameter (d_{bc}) of 15lit and 365mm respectively. And it has also piston diameter (d_{bcpr}) and maximum stroke (Y) of 28mm and 200mm dimensions respectively. The mass of brake cylinder piston with its piston rod is 8kg.

The diameter of brake cylinder pipe (d_{bcp}) is 18mm as shown in the diagram below [9]

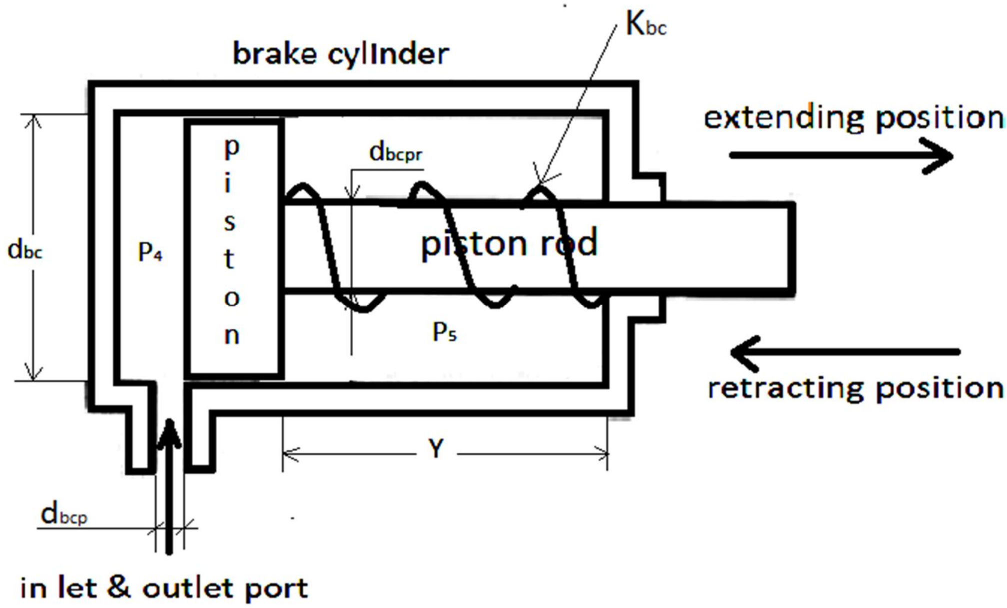


Fig.3.7. Brake cylinder

3.1.3. Auxiliary Reservoir / Brake Reservoir

This auxiliary reservoir is an air tank provided on each vehicle of a train equipped with air brakes to supply air for brake applications. The air in this reservoir is supplied from main reservoir located on the locomotive for all the coaches of the train.

The volume of this auxiliary reservoir (Var) and diameter of auxiliary reservoir pipe (d_{arp}) are 80lit and 18mm respectively [9]. The working principle how the auxiliary reservoir is charging compressed air from the main reservoir and feeding compressed air to the brake cylinder is clearly shown on system working principle at different brake valve positions. This reservoir is also called brake reservoir.

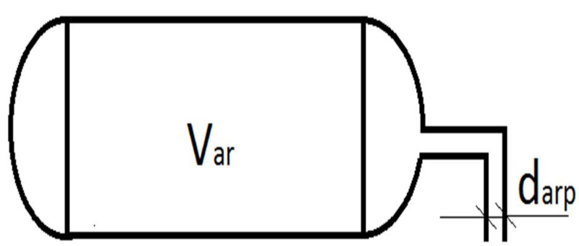


Fig.3.8. Schematic of auxiliary reservoir

3.1.4. Service reservoir

This service reservoir is an air tank provided on each vehicle of a train equipped with compressed air to make the balance valve in different working conditions (service brake, release and emergency brake).

The volume of this service reservoir (Vsr) and diameter of this service reservoir pipe (d_{srp}) are 11lit and 12mm respectively [9]. The working principle how this service reservoir is charging compressed air from the main reservoir and how it makes balance with the train pipe pressure is clearly shown on system working principle at different brake valve positions.

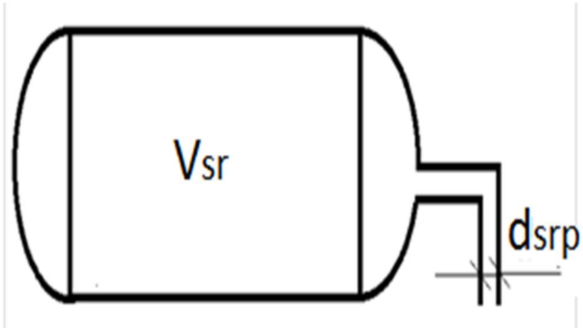


Fig.3.9. Schematic of service reservoir

3.1.5. Electromagnetic valve box

As shown in fig 3.9 below it is a box in which the following functional components are included in it

Service electromagnetic valve

Release electromagnetic valve and

Emergency electromagnetic valve

Service application electro-magnetic valve

It is actuated when the driver selects service brake application position to increase the exhausting speed of train pipe, accelerate the electro pneumatic braking.

Release electro-magnetic valve

In initial charging, to link up the passage between service reservoir and train pipe, to increase the initial charging speed of the service reservoir .After braking, in releasing, to increase the adverse air moving speed from the service reservoir to the train pipe, to acquire rapid releasing.

Emergency application electromagnetic valve

To increase the brake cylinder air intake speed in electro-pneumatic emergency application, to improve electro – pneumatic emergency application performance; In the mean time, to make the train pipe open to the air and produce additional exhausting in electro-pneumatic emergency application.

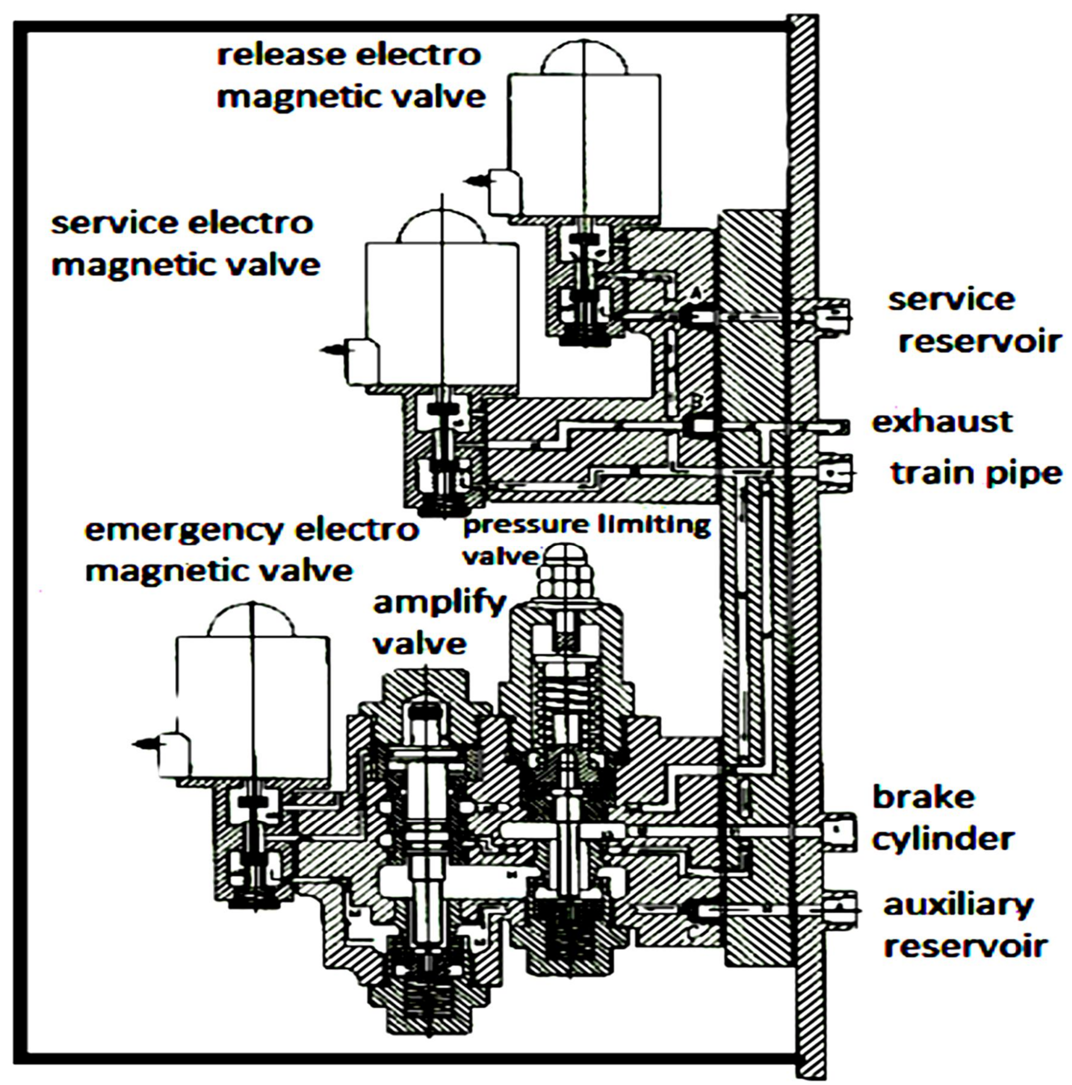


Fig.3.10. Sectional view of electromagnetic valve box **[9]**

3.2. Working principle of the electro pneumatic brake system

The distributing valve has the following functional positions, namely: charge and release position, service brake application position, brake application holding position, graduated release holding position and emergency brake application position.

Even though electro pneumatic brake system has the above listed functional positions but according to the scope of this research paper it is simulated in the following three selected drivers brake valve positions (service brake application, release application and emergency brake application).

3.2.1. Release application position

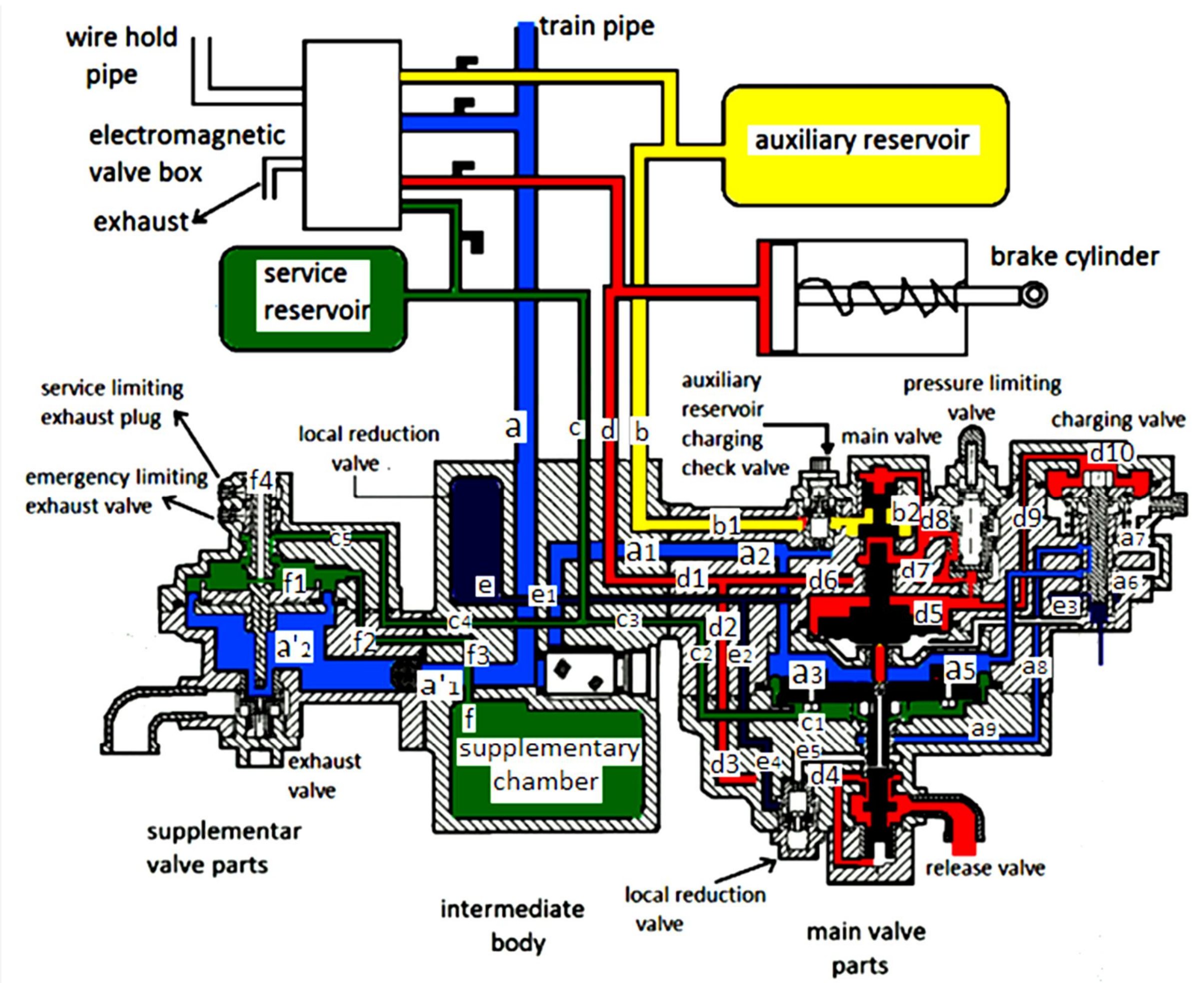


Fig 3.11 Electro pneumatic brake system at release position

Recharge

Recharge means charge after brake application. After pressure in train pipe has been increased, the triple pressure equilibrium state of original main valve is broken, the resultant force of train pipe pressure and brake cylinder pressure (downward) is larger than the force of service reservoir (upward), and the main valve enters charge and release position. Its charge passage is the same as that in initial charge, the release and exhaust passage is as follows.

Brake cylinder releases

After main valve reaches release position, the release valve on lower side of main valve is opened, the compressed air of brake cylinder flows through d-d_1-d_2-d_3-d_4-release valve to exhaust to atmosphere. Meanwhile, the compressed air on upper side d_5 of small piston of main valve also flows through large contracting plug on pressure limiting valve bottom-pressure limiting valve sleeve-pressure limiting valve slot-d_7-d_6-d_2-d_3-d_4-release valve to exhaust to atmosphere.

In applying graduated release operation (change-over cover plate in graduated release position), owing to that the train pipe ceases to increase pressure, the brake cylinder pressure on upper side of small piston of main valve drops until the upward force on main valve is slightly larger than downward force on it, then main piston moves upward to close release valve port and brake cylinder stops to exhaust. The triple pressure of main valve reaches a new equilibrium point, now brake cylinder still maintain certain braking force, its magnitude can be controlled by locomotive driver via controlling amount of pressure increase in train pipe, the locomotive driver can operate flexibly according to necessity.

In applying direct release (change-over cover plate in direct release position), when pressure in train pipe is increased, and after main valve reaches- release position, the compressed air of brake cylinder exhausts to atmosphere, at the same time the compressed air of service reservoir on lower side of main piston of main valve flows through C1-local reduction valve rod, and sleeve-a_9-a_8-a_7-a_6 to upper side a_3 of main piston, the pressure in service reservoir drops rapidly and charges train pipe, now, the main valve is impossible to again return to release and holding position, brake cylinder pressure is directly released entirely, but cannot realize graduated release action. Such action is quite necessary and also very important to operation of train, mixedly composed of distributing valves with and without graduated release. When disregarding operation of train mixedly composed of distributing valve without graduated release, the change-over cover plate can permanently be placed in graduated release position (indicating arrowhead indicates downward).

When the change-over cover plate is in graduated release position, locomotive driver can also use operation method for direct release, the train pipe pressure is increased directly to constant pressure, brake cylinder can be released directly entirely, the only difference lies in that in direct release position, the release time is faster.

Local reduction chamber exhausts

When brake cylinder pressure exhausts to below 20 kPa, charging valve piston moves downward under action of spring force and opens passage at tail part of charging valve rod, the compressed air of local reduction chamber exhausts to the atmosphere.

3.2.2. Service brake application position

The brakes used now a day's belong to automatic air brake, the braking action can automatically take place as soon as the pressure in train pipe is reduced, hence when train is separated into two sections, or train pipe is broken (including breaking of hose coupling), or conductor's valve is pulled to move or locomotive operates to make train pipe reduce pressure, automatic braking action can take place in either case to ensure running safety.

When train pipe pressure is reduced at certain rate, the train pipe pressure on upper side of main valve drops, while compressed air of service reservoir on lower side of main piston begins to back flow to train pipe, however due to restriction of contracting plug I , the pressure drops rather slowly in consequence certain pressure difference is formed on main piston , at the same time., under the action of brake spring force, the main piston is pushed to move, and carrying release valve rod, main valve rod etc to move upward and perform the following actions :

- To cut-off back-flow passage for the compressed air from service reservoir to back flow to train pipe» the main piston moves upward rapidly.
- Release valve closes communication between brake cylinder and atmosphere.
- Local pressure reducing occurs.

The compressed air of train pipe coming from upper side or main piston flows through a_3-a_4-release valve rod slot and sleeve-e_5-contracting plug IV in local reduction valve-e_4.e_3-e_2-e_1-e-

local reduction chamber. Meanwhile, it flows through passage e_3 and charging valve sleeve to exhaust to atmosphere, forming first stage local reduction action.
The local reduction action promotes sudden drop of train pipe pressure and further makes main piston to move upward.

Open balance valve on main valve, the compressed air from auxiliary reservoir flows through b-b_1-b_2-d_8-pressure limiting valve sleeve and pressure limiting valve-d_7-d_6-d1-d-brake cylinder, then the pressure in brake cylinder rises. At the same time, the compressed air from brake cylinder also flows through d_7-large contracting plug I –d_5-upper side of small piston of main valve. It also flows through d_9-di_0-one side of charging valve piston, compresses charging valve spring causing charging valve rod to move to cut off communication between local reduction chamber and atmosphere, and finishes first stage local reduction action.
The pressure in local reduction chamber continues to increase until it is in equilibrium with train pipe pressure, i. e- second stage local reduction action.

In service application, due to the fact that the train pipe pressure is lower than auxiliary reservoir pressure, the auxiliary reservoir check valve is closed.
In the initial period of service application, the air pressure in supplementary chamber is almost unchanged, after pressure reduction in train pipe, the pressures acting on supplementary valve piston 8 becomes out-of –equilibrium, when its pressure difference is greater than pressure difference valve for action of supplementary valve, the supplementary valve piston 8 together with supplementary valve rod 5 to move downward.

The communicating passage between service reservoir and supplementary chamber is cut-off first, (communication between c_5 and f_1), and then the passage between supplementary chamber and service exhaust plug is opened, the compressed air from supplementary chamber flows through f_3-f_2-fi-f_4-contracting plug V-atmosphere. Now the supplementary valve appears in service application position. The pressure in supplementary chamber drops closely following train pipe pressure reduction. Too large pressure difference cannot be formed between both, exhaust valve will no$ be opened, because the contracting plug V controls adequate exhaust rate, this controls pressure difference value between both sides of supplementary valve piston.

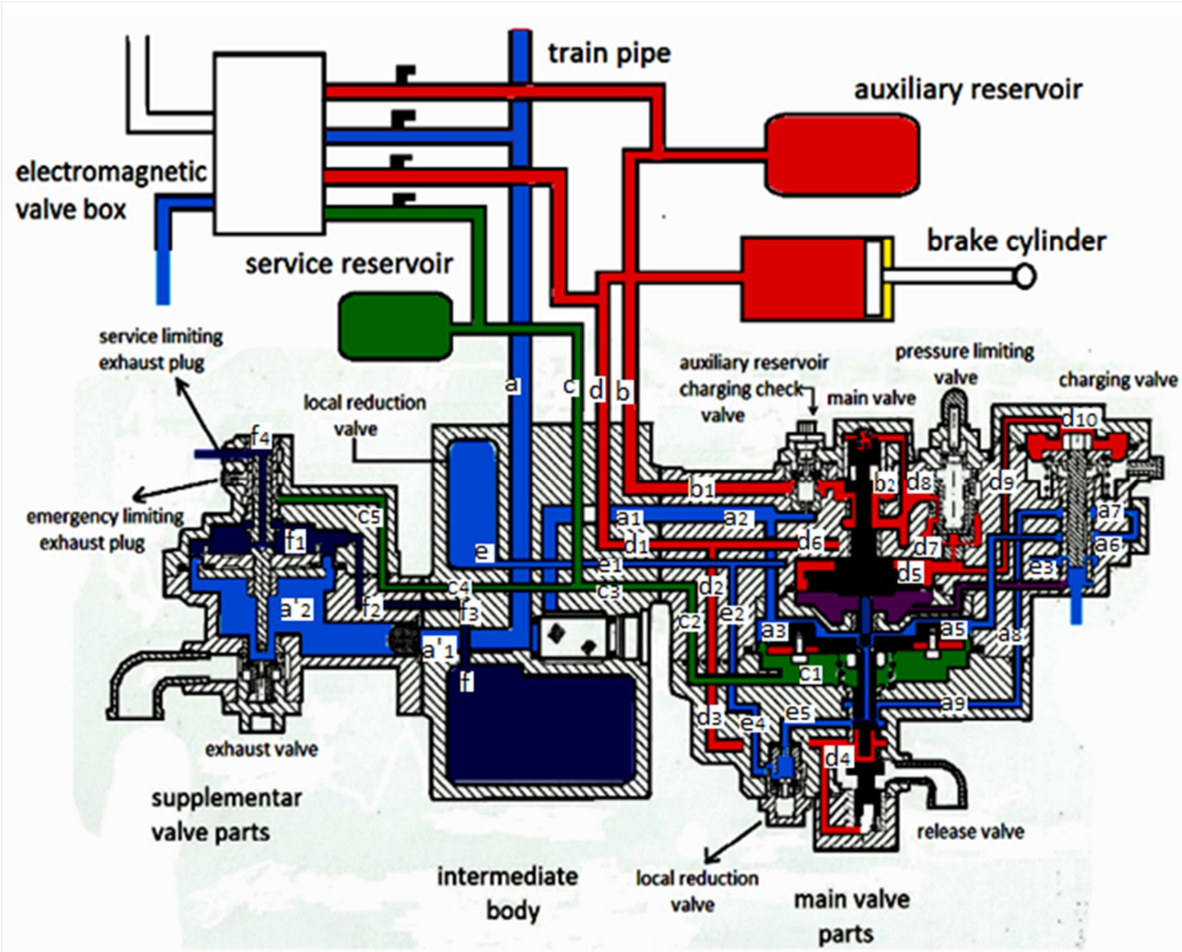


Fig 3.12. Electro pneumatic brake system at service brake application position

3.2.3. Emergency brake application position

In emergency application, the action process of main valve is the same as that, in service .application, only the action is faster because the rate of pressure reduction in train pipe is faster than that in service application. In addition, the pressure-difference on both sides of main piston is much faster than that-in service application, so the balance valve 4 is in fully opened state, compressed air from auxiliary reservoir enters brake cylinder rapidly, and brake cylinder pressure rises rapidly.

At the same time, larger pressure difference is also formed on both side of supplementary valve piston 8, the supplementary valve piston move downward rapidly, to open emergency exhaust valve, compressed air in train pipe exhausts to atmosphere rapidly to attain the purpose of emergency exhaust. Now the compressed air of supplementary chamber besides communicates to atmosphere through service exhaust plug, also through both passages of emergency exhaust plug exhausts to atmosphere, because these passages are also opened.

When brake cylinder pressure exceeds pre compressed force of pressure limiting valve spring 54, the pressure limiting valve 55 moves upward to close the passage from auxiliary reservoir to brake cylinder. Now, although the main valve is in emergency application state, brake cylinder pressure cannot continue to rise, the maximum pressure value of auxiliary reservoir is controlled by pressure limiting valve. The pressure limiting valve can be regulated according to different requirement.

After the train pipe pressure is diminished to zero, the compressed air in supplementary chamber continues to exhaust to atmosphere through two passages, it needs about 10-15s to exhaust to 0, in this process, the exhaust valve is opened continuously, train pipe is impossible to be recharged.

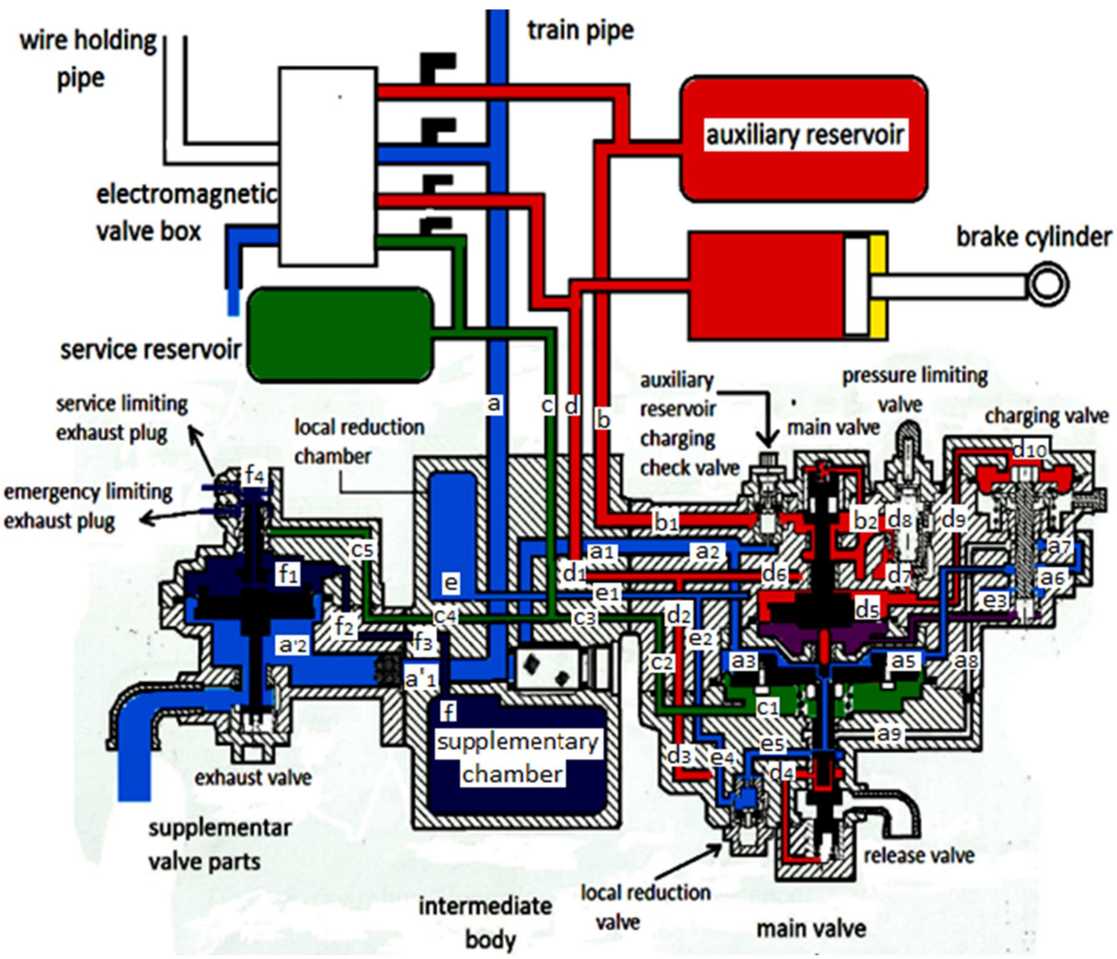


Fig 3.13. Electro pneumatic brake system at emergency brake application position

CHAPTER-IV

TRAIN E-P BRAKE SYSTEM SIMULATION

4.1. System to be simulated

Since all the braking system components of the locomotive and the coaches are the same in number and size and the commanding signal is electric current from the driver to the last coach as shown in fig.1.2, so by simulating the braking system of the locomotive and the last coach whatever the number of the coaches are, their output is the same.

This is for the three drivers brake valve positions i.e for service brake application position, release application, emergency brake application position and their hold positions (brake holding position and gradual release holding position).

4.2. Steps of simulation

Based on the steps of simulation written on the literature review the simulation of the system should have to pass the following steps.

4.2.1. Defining the system

The system to be studied or to be simulated is defined as the system of electro pneumatic braking system of train's vehicle. This system is composed of pneumatic and electrical subsystems in which the pneumatic braking is actuated electrically to reduce time delay of compressed air in long trains from the leading locomotive to the vehicle located at the rear part. The aim of this system is to stop all the vehicles of the train at the same time regardless of their position.

4.2.2. Identifying the system components

The system components involved in this simulation study of electro pneumatic brake system are: brake pipe, air distributor valve, auxiliary reservoir, electromagnetic valve box, service reservoir and brake cylinder.

4.2.3. Flow chart (Algorithm) of the system to be simulated

The simulation of the electro pneumatic brake system will be performed in three cases of drivers brake valve position. Namely

1. Service brake application position,
2. Release application position and
3. Emergency brake application position

For all the three cases illustrated in the diagrams below let the pressures in each of the system components are assumed to be:

P_1 = pressure in service reservoir

P_2 = pressure in train pipe

P_3 = pressure in auxiliary reservoir

P_4 = pressure in front of brake cylinder piston

P_5 = pressure at the back of brake cylinder piston and

P_{atm} = atmospheric pressure out of the system

4.2.3.1. Service brake application position

The actuation mechanism and sequential flow of compressed air in the braking system of service brake application position is shown in the following block diagram

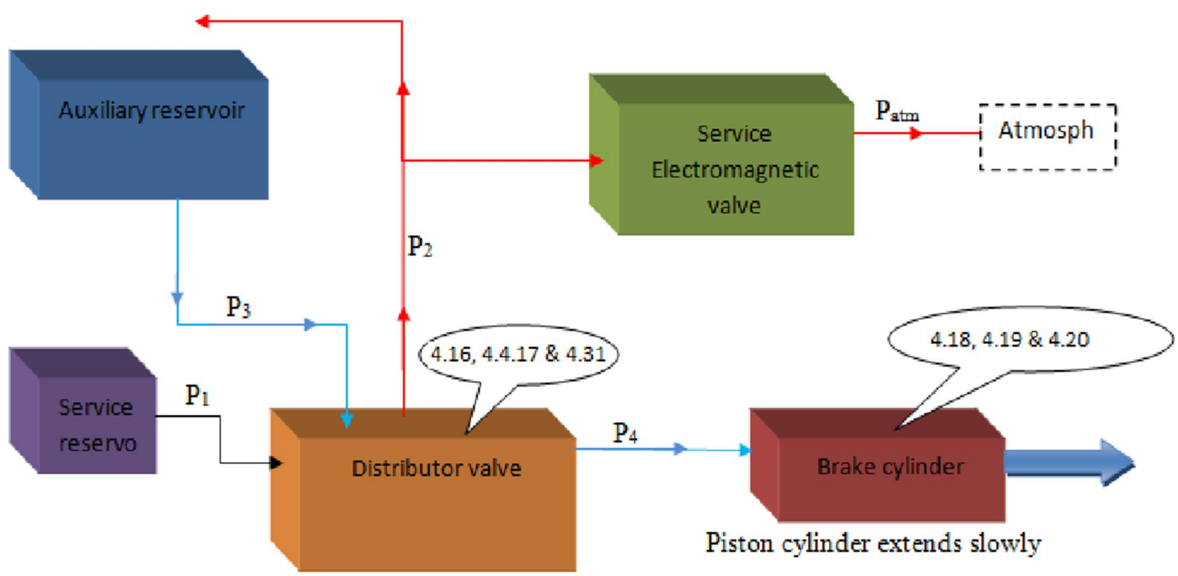
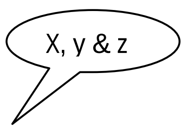


Fig4.1 Service brake flow chart

The system components are initially charged with compressed air except the brake cylinder which is exhausted to the atmosphere → the driver put the drivers brake valve on service brake application position → the service electromagnetic valve is excited → brake pipe pressure is being exhausted through service brake exhaust valve → pressure difference occur on the distributor main valve → the auxiliary reservoir and brake cylinder connect each other due to the position of main valve piston → compressed air enters to the brake cylinder from auxiliary reservoir causing service brake to apply.

Out puts of the simulation to be validated.

- ✓ longitudinal force due to the pressure in it,
- ✓ maximum pressure developed in brake cylinder
- ✓ time taken to start braking
- ✓ time taken to finish piston stroke,
- ✓ speed and acceleration of the piston
- ✓ Position of main valve piston at different time interval



where x, y & z are equation numbers on figures 4.1, 4.2 & 4.3

4.2.3.2. Release application position

The actuation mechanism and sequential flow of compressed air in the braking system of release application position is shown in the following block diagram

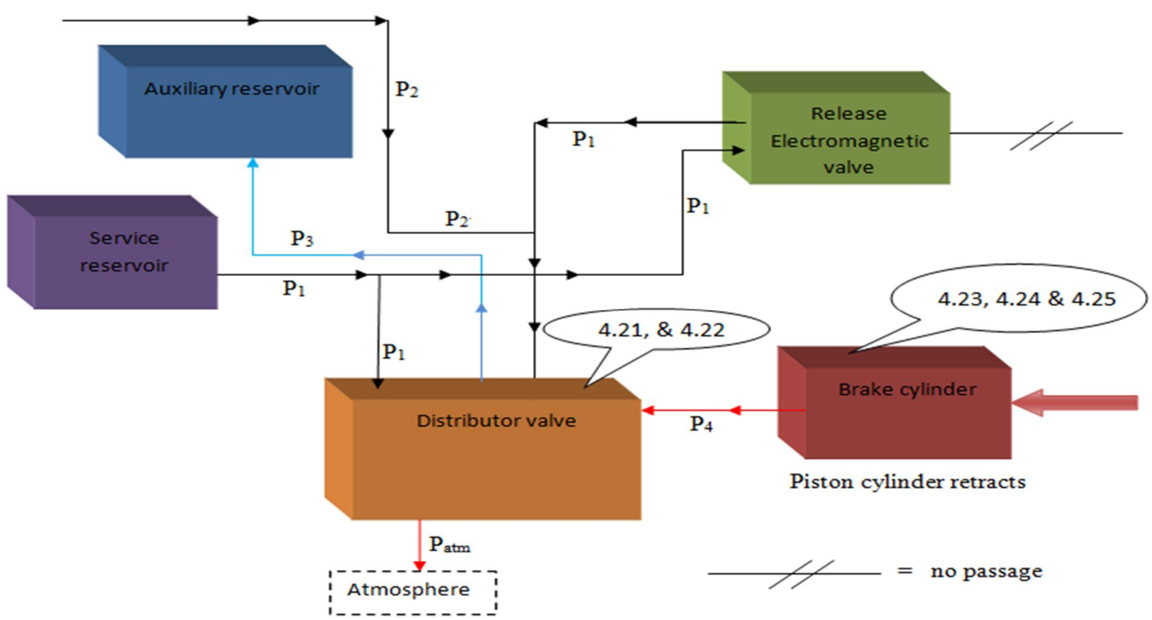


Fig4.2 Release brake flow chart

initially the brake pipe is exhausted to the atmosphere through service electromagnetic valve and brake cylinder is charged with compressed air from auxiliary reservoir→ the driver put the drivers brake valve on release application position → the release electromagnetic valve is excited → brake pipe pressure is being increased from the main reservoir located on the locomotive and compressed air from services reservoir is supplied to the brake pipe through release electromagnetic valve → pressure difference occur on the distributor main valve causing the piston to move down ward → the brake cylinder is connected to the atmosphere due to the position of main valve piston → compressed air from brake cylinder exhausts to the atmosphere through release valve inside distributor valve.

Out puts from this simulation

- ✓ Exhausting time of the brake cylinder.
- ✓ Flow rate of the compressed air from brake cylinder to the atmosphere.
- ✓ Position of main valve piston.
- ✓ Retracting speed and acceleration of the piston cylinder.

4.2.3.3. Emergency brake application position

The actuation mechanism and sequential flow of compressed air in the braking system of emergency brake application position is shown in the following block diagram.

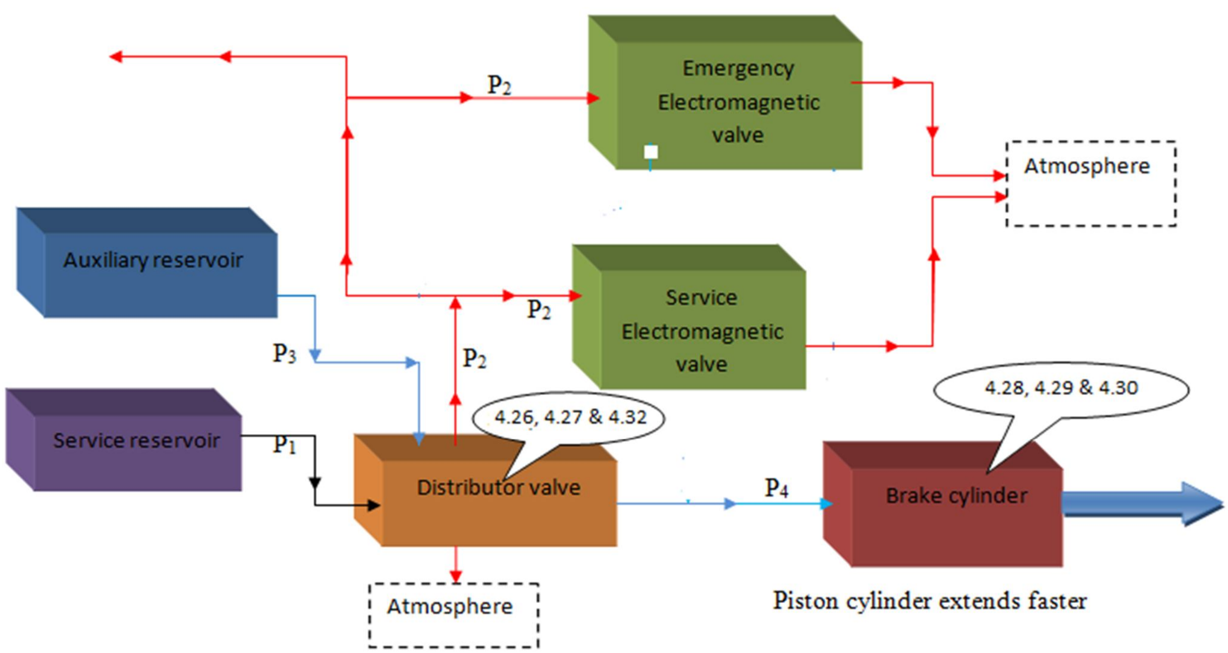


Fig4.3 Emergency brake application position flow chart

The system components are initially charged with compressed air except the brake cylinder which is exhausted to the atmosphere → the driver put the drivers brake valve on emergency brake application position → the service and emergency electromagnetic valves are excited →

brake pipe pressure is being exhausted through service and emergency brake exhaust valve → pressure difference occur on the main valve → the auxiliary reservoir and brake cylinder connect each other due to the position of main valve piston → compressed air enters to the brake cylinder from auxiliary reservoir quickly causing emergency brake to apply.

The out puts are the same as in service brake but what is needed here is that to compare the values between service and emergency.

Finally the general output results of the train electro pneumatic braking system will be compared with the pneumatic one to see the effects of time delay on due to commanding media.

4.2.4. Modeling the system with equations:

For all cases the system model consists of four sub systems and it is illustrated in the following block diagram. On this block diagram the electromagnetic subsystem is not included in the simulation part. It is assumed as if the driver actuates the electromagnetic valves on these selected drivers brake valve positions to perform the desired activities.

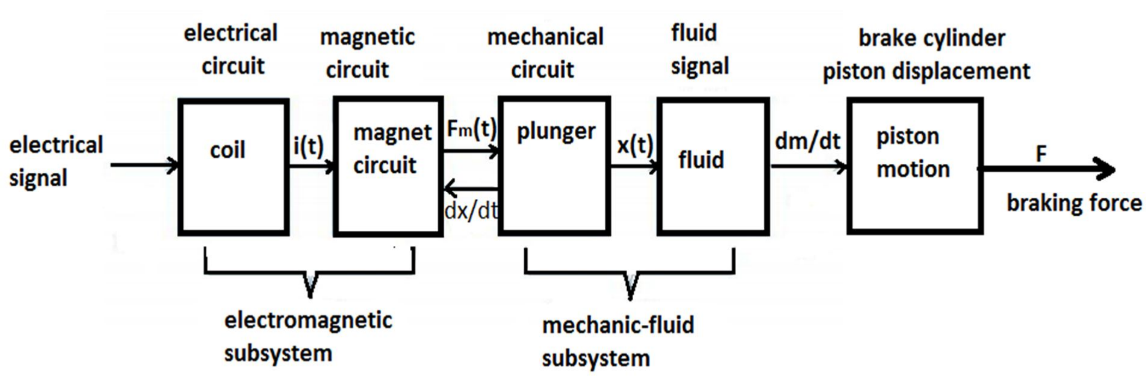


Fig4.4. General System model in block diagram **[5]**

Electrical model

This model represents the current passing through the coil within the electromagnetic valve in terms of applied voltage (U_{pwm}), plunger position & speed. This electrical current is an input for the electro magnet to open or close to perform the required functions.

Magnetic model

This model considers magnetic force applied into the plunger to open the air path out of the system in terms of current and plunger position.

Magnetic circuit comprises of the fixed core with a coil and plunger, which is moving by magnetic force as in figure below.

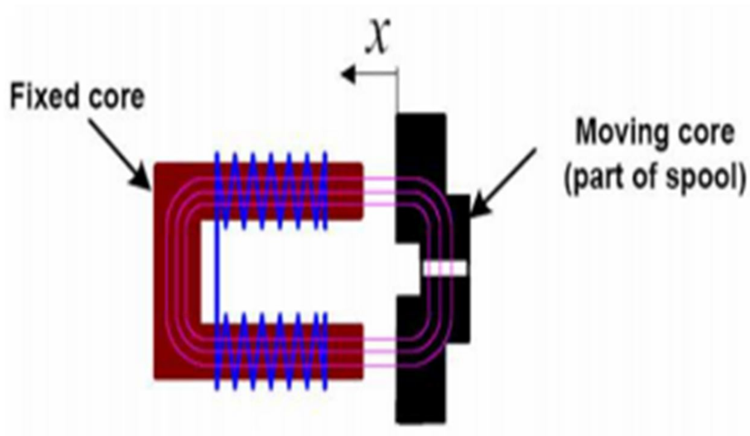


Fig4.5. Schematic of magnetic valve electrical connection **[5]**

The two subsystems of the electro pneumatic brake system described above do nothing on the fluid dynamics of the system. But they actuate the system or initiate the fluid to circulate inside. So more emphasize is taken to the following subsystems. And the equation below is considered to simulate the following sub systems as if the applied voltage is low or high.

$$U_{PWM} = \begin{cases} \text{high} \\ \text{low} \end{cases} \qquad 4.1$$

The above equation is high if the coil in the electromagnetic valve gets an electric current and becomes energized, and it becomes low if the coil in the electromagnetic valve does not get an electric current through it or if it is de-energized **[5].**

Mechanical model

On the schematic diagram of distributor valve it is difficult to label the dimensions of the distributor main valve piston. Therefore here the dimensions used in the simulation equation are labeled and their values are listed on table 4.6.1

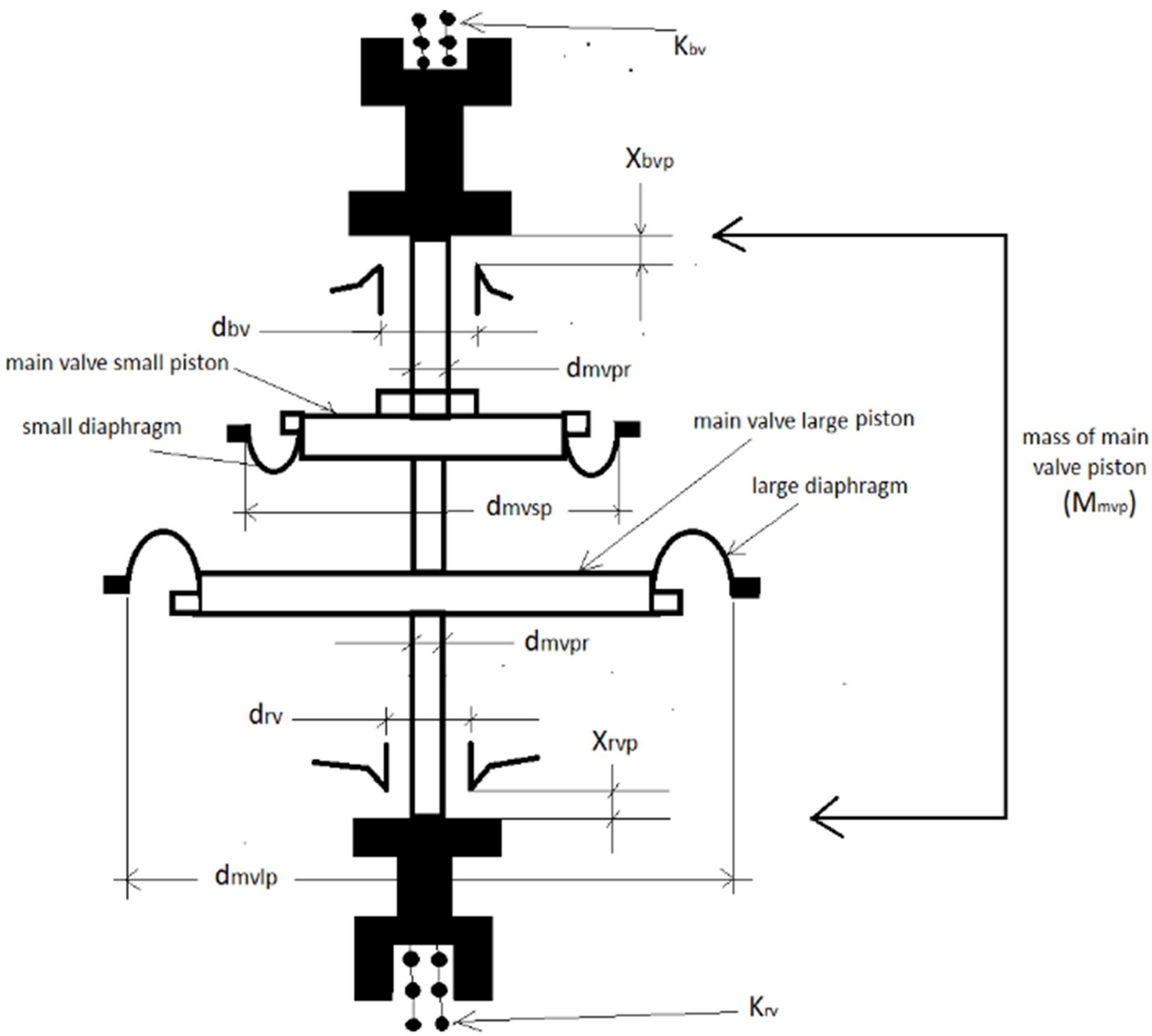


Fig 4.6. Schematic diagram of distributor valve main piston

This representation establishes the distributor main valve piston position (Xp), speed and acceleration in terms of the parameters listed in the equation below. The dynamic equation for the valve piston that makes it in equilibrium is formulated by the Newton’s second law as follows:

$$M_{mvp} \times \ddot{X}_{bvp} = F_{prs} - P_{tp} \times A_{mvlp} - K_{mv} \times X_{bvp} - b \times \dot{X}_{bvp} \quad 4.2$$

Where $$F_{prs} = P_{sup} \times A_{mvlp} \quad 4.3$$

P_{tp} = pressure of train pipe, M_{mvp} = mass of main valve piston, K = valve spring constant, b = valve damping coefficient, A_{mvlp} = area of main valve large piston, P_{sup} = pressure in to the valve and Ptp = train pipe pressure **[5]**

Fluid flow model

This model presents mass flow rate ($\dot{m}$) passing through the valve in terms of the plunger position (X_P) and the air pressure force (F_{prs}) .Plunger movement provides the required mass flow rate to control the piston cylinder motion.

Electromagnetic and mechanical components of the valve are used to control the fluid flow through the valve orifice using the plunger position. Turbulent air flow passing through the orifice is modeled by the Orifice equation (3.4). This relation is composed of two different equation flow regimes. As the fluid inside the pneumatic system is compressible, equation (3.6) below is used whereas if it is incompressible equation, (3.7) is used.

So since the fluid inside the pneumatic system is compressible the equation used to calculate f (P_r) is (3.6) below

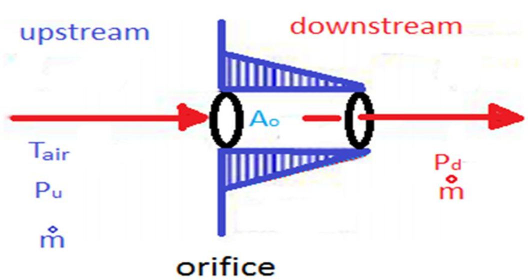


Fig 4.7. Schematic diagram of valve orifice

Standard equations to the mass flow rate passing through the valve orifice is described as

$$\frac{dm}{dt} = \dot{m} = \frac{C_d C_A A_v P_u}{\sqrt{RT_{air}}} f(P_r) \quad 4.4$$

$$P_r = \frac{P_{down}}{P_{up}} \quad 4.5$$

$$f(P_r) = \sqrt{\gamma\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}} \quad \text{If } P_r \le P_{cr} \qquad 4.6$$

$$f(P_r) = \sqrt{\frac{2\gamma}{\gamma-1}(P_r)^{\frac{2}{\gamma}} - (P_r)^{\frac{\gamma+1}{\gamma}}} \quad \text{If } P_r > P_{cr} \qquad 4.7$$

$$P_{cr} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} = 0.528 \qquad 4.8$$

Where C_d discharge coefficient, P_{up}, P_{down} upstream and downstream pressures of the valve (input and output) respectively, γ specific heat ratio for air and its value is equal to 1.4, R general gas constant equal to $287\frac{m^2}{S^2K^0}$, T_{air} system temperature, C_A constant coefficient discharge to orifice cross sectional area, A_v orifice cross sectional area, P_{cr} critical pressure inside the orifice.**[5]**

Cylinder model

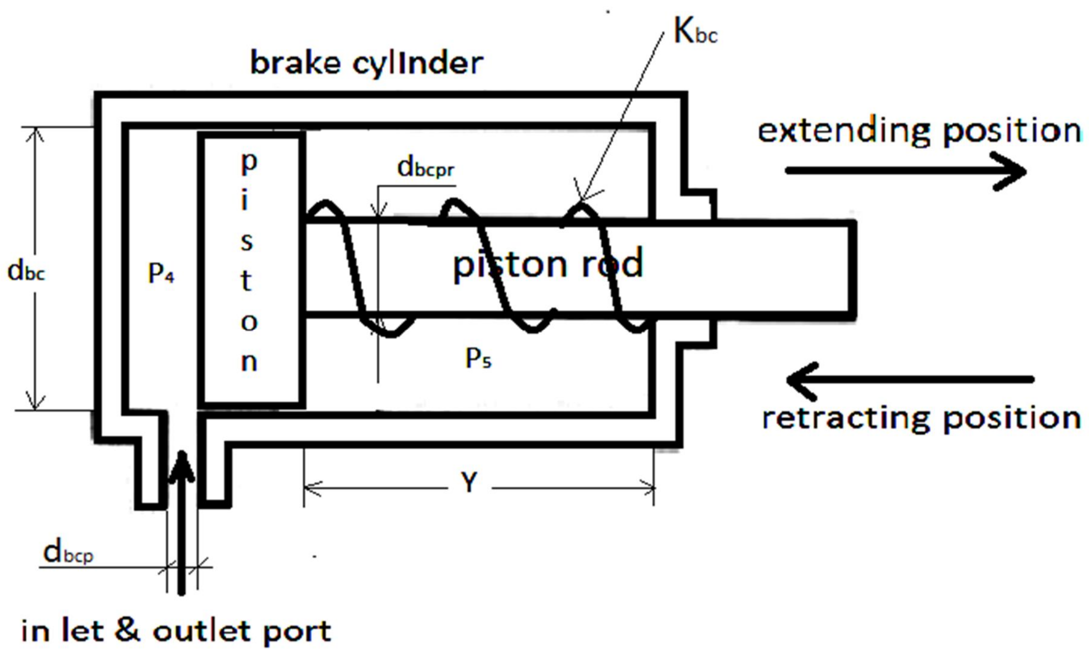


Fig 4.8. Schematic of brake cylinder

Equations governed by the pressure dynamics in the cylinder chambers, can be obtained through the thermodynamic analysis based upon energy conservation and continuity law. In cylinder flow modeling the flow model is obtained based on the following assumptions

- Air is thermally a perfect gas

- Pressure is homogeneously distributed throughout the chamber
- No heat loss occurs from the walls of the actuator to surrounding i.e. the system is adiabatic.
- There is no leakage throughout the system components.

Using the ideal gas law: **[5]**

$$P_i V_i = m_i R T_i \tag{4.9}$$

where, i=1,2 denotes indices to two chests of the cylinder piston, Pi, Vi, Ti and mi are pressure, volume, temperature and the mass on both piston sides. Deriving and incorporating the above into the energy equation, the differential equation below yields.

$$\frac{dp_i}{dt} = \frac{\gamma RT}{V_i} \times \frac{dm_i}{dt} - \frac{\gamma p_i}{V_i} \times \frac{dp_i}{dt} \tag{4.10}$$

Choosing the displacement coordinates origin at the stroke beginning, volume of each chamber can be written as:

$$V_i = V_{0i} + A_i \times y \tag{4.11}$$

$$V_j = V_{0j} + A_j \times (L - y) \tag{4.12}$$

Where, L cylinder course, y displacement of the cylinder piston, V_{0i}, V_{0j} dead volume in the first and second chest and A_i and A_j are cross sectional areas of the first and second chest and Pi=pressure in cylinder. Inserting the above equations and assuming the dead volumes on both sides of piston are negligible the pressure dynamics is described as

$$\frac{dp_i}{dt} = \frac{\gamma RT}{A_i * y} \times \frac{dm_i}{dt} - \frac{\gamma p_i A_i}{A_i * y} \times \frac{dy}{dt} \tag{4.13}$$

According to second Newton's law for the cylinder piston, movement equation turns out to be

$$M_{bcp} \times \ddot{Y} = P_{bc} \times A_{bc} - P_{atm}(A_{bc} - A_{rod}) - K_{bc} \times Y - \beta \times \dot{Y} - Ff \tag{4.14}$$

Where, M_{bcp} mass of piston, β brake cylinder damping coefficient, F_f brake cylinder frictional force on braking position and it is given by

$$F_f = P \times A - P_{atm}(A_{bc} - A_{rod}) - K \times Y - \beta \times \dot{Y} - m\ddot{Y} \tag{4.15}$$

Specific equations for each brake application position

Service brake application position

$$M_{mvp} \times \ddot{X}_{bvp} = P_1 \times A_{mvlp} - P_2 \times A_{mvlp} - K \times X_{bvp} - b \times \dot{X}_{bvp} \qquad 4.16$$

Where M_{bvp} = mass of balance valve piston and X_{bvp} = balance valve max. Piston stroke A_{mvlp} = area of main valve large piston.

$$\frac{dm_{bvo}}{dt} = \dot{m}_{bvo} = \frac{C_d C_A A_{bv} P_3}{\sqrt{RT_{air}}} * f(P_r) \qquad 4.17$$

Where $\dot{m}_{bvo}$ = mass flow rate of distributor balance valve orifice from auxiliary reservoir to brake cylinder

A_{bv} = area of balance valve given by $A_{bv} = \pi \times d_{bv*X_{bv}}$

$$\frac{dp_4}{dt} = \frac{\gamma RT}{A_{bc}*y} \times \frac{dm_{bvo}}{dt} - \frac{\gamma p_4 A_{bc}}{A_{bc}*y} \times \frac{dy}{dt} \qquad 4.18$$

$$M_{bcp} \times \ddot{Y} = P_4 \times A_{bcp} - P_{atm} \times (A_{bc} - A_{bcpr}) - K \times Y - \beta \times \dot{Y} - Ff \qquad 4.19$$

$$Ff = P_4 \times A_{bcp} - K \times Y - \beta \times \dot{Y} - M_{bcp} \times \ddot{Y} \qquad 4.20$$

Release application position

$$M_{mvp} \times \ddot{X}_{rvp} = P_2 \times A_{mvlp} + P_4 \times A_{mvsp} - P_1 \times A_{mvlp} - K \times X_{rvp} - b \times \dot{X}_{rvp} \qquad 4.21$$

Where M_{rvp} = mass of release valve piston and X_{rvp} = release valve piston stroke

$$\frac{dm_{rvo}}{dt} = \dot{m}_{rvo} = \frac{C_d C_A A_{rv} P_4}{\sqrt{RT_{air}}} * f(P_r) \qquad 4.22$$

Where $\dot{m}_{rvo}$ = mass flow rate of distributor release valve orifice from brake cylinder to atmosphere, A_{rv} = area of release valve given by $A_{rv} = \pi \times d_{rv} \times X_{rv}$

$$\frac{dp_4}{dt} = -\frac{\gamma RT}{A_{bc}} \times \frac{dm_{bc(air)}}{dt} + \frac{\gamma p_4 A_{bcp}}{A_{bc}} \times \frac{dy}{dt} \qquad 4.23$$

$$M_{bcp} \times \ddot{Y} = P_{atm} \times (A_{bc} - A_{bcpr}) + K \times Y + \beta \times \dot{Y} - P_4 \times A_{bcp} - Ff \qquad 4.24$$

$$Ff = K \times Y + \beta \times \dot{Y} - P_4 \times A_{bcp} - M_{bcp} \times \ddot{Y} \qquad 4.25$$

Emergency brake application position

$$M_{mvp} \times \ddot{X}_{bvp} = P_1 \times A_{mvlp} - P_2 \times A_{mvlp} - K \times X_{bvp} - b \times \dot{X}_{bvp} \qquad 4.26$$

Where M_{mvp} = mass of main valve piston and X = main valve piston stroke

$$\frac{dm_{bvo}}{dt} = \dot{m}_{bvo} = \frac{C_d C_A A_{bv} P_3}{\sqrt{RT_{air}}} * f(P_r) \qquad 4.27$$

Where $\dot{m}_{bvo}$ = mass flow rate of distributor balance valve orifice from auxiliary reservoir to brake cylinder

A_{bv} = area of balance valve given by $A_{bv} = \pi \times d_{bv} \times X_{bv}$

$$\frac{dp_4}{dt} = \frac{\gamma RT}{A_{bc} * y} \times \frac{dm_{bc(air)}}{dt} - \frac{\gamma p_4 A_{bcp}}{A_{bc} * y} \times \frac{dy}{dt} \qquad 4.28$$

$$M_{bcp} \times \ddot{Y} = P_4 \times A_{bcp} - P_{atm} \times (A_{bcpr} - A_{bcpr}) - K \times Y - \beta \times \dot{Y} - Ff \qquad 4.29$$

$$Ff = P_4 \times A_{bcp} - K \times Y - \beta \times \dot{Y} - M_{bcp} \times \ddot{Y} \qquad 4.30$$

The equations of service and emergency brake application looks like the same. The only difference between them is that the exhausting speed of compressed air from train pipe to the atmosphere. As a result of this the response time of braking is different for the two brake valve positions.

On service brake application position there is one exhausting port, which is through the service electromagnetic valve. So the compressed air is exhausted to the atmosphere slowly. The train pipe pressure on service application position is

$$\frac{dp_2}{dt} = \frac{\gamma RT}{A_{eep} * X} \times \frac{dm_{eep}}{dt} - \frac{\gamma p_2 A_{bv}}{A_{bv} * X} \times \frac{dX}{dt} \qquad 4.31$$

In case of emergency brake application there are two exhausting ports which are electromagnetic valve exhausting port and emergency valve exhausting port. So the equation of train pipe pressure on emergency brake application position is

$$\frac{dp_2}{dt} = 2\frac{\gamma RT}{A_{eep} * X} \times \frac{dm_{eep}}{dt} + \frac{\gamma RT}{A_{evep} * X} \times \frac{dm_{evep}}{dt} - \frac{\gamma p_2 A_{bv}}{A_{bv} * X} \times \frac{dX}{dt} \qquad 4.32$$

Generally the exhausting ports are clearly shown on figures 4.1 and 4.3.

4.2.5. Building system model with Simulink

The simulink model of electro pneumatic braking system of train vehicle at different driver's brake valve position (service, release and emergency brake application positions) is modeled. The actuation mechanisms and working principles of the electro pneumatic braking system is shown on chapter three of figures (3.11, 3.12 and 3.13).

Since the electro pneumatic braking system components are so complicated it is difficult to model the system in one simulink model. To simplify this difficulty MATLAB SIMULINK has sub system modeling technique.

In this paper the train electro pneumatic braking system is modeled in MATLAB SIMULINK sub system modeling. Here the motion of fluid flow with its direction is clearly shown. The system is divided in to four sub systems as shown on figures 4.9, 4.11 and 4.13.

Each sub system is linked internally by their inlet and outlet ports. And even though it is not as clear as sub system the detail modeling of these sub systems (figures 4.6, 4.11 & 4.13) is shown on figures 4.10, 4.12 & 4.14 respectively.

4.2.5.1. Simulink model of system on service application position

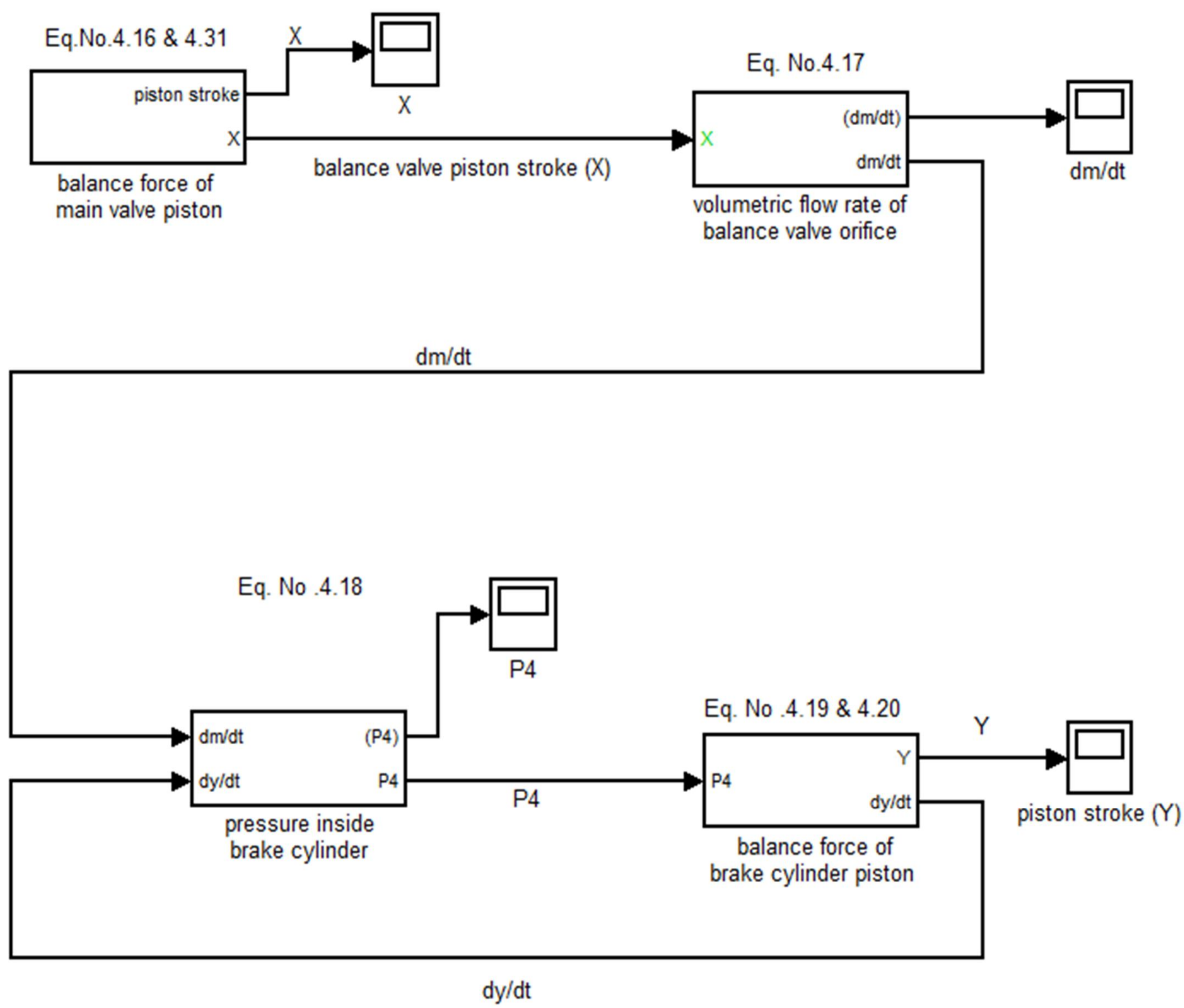


Fig.4.9. Sub-System simulink model on service brake application position

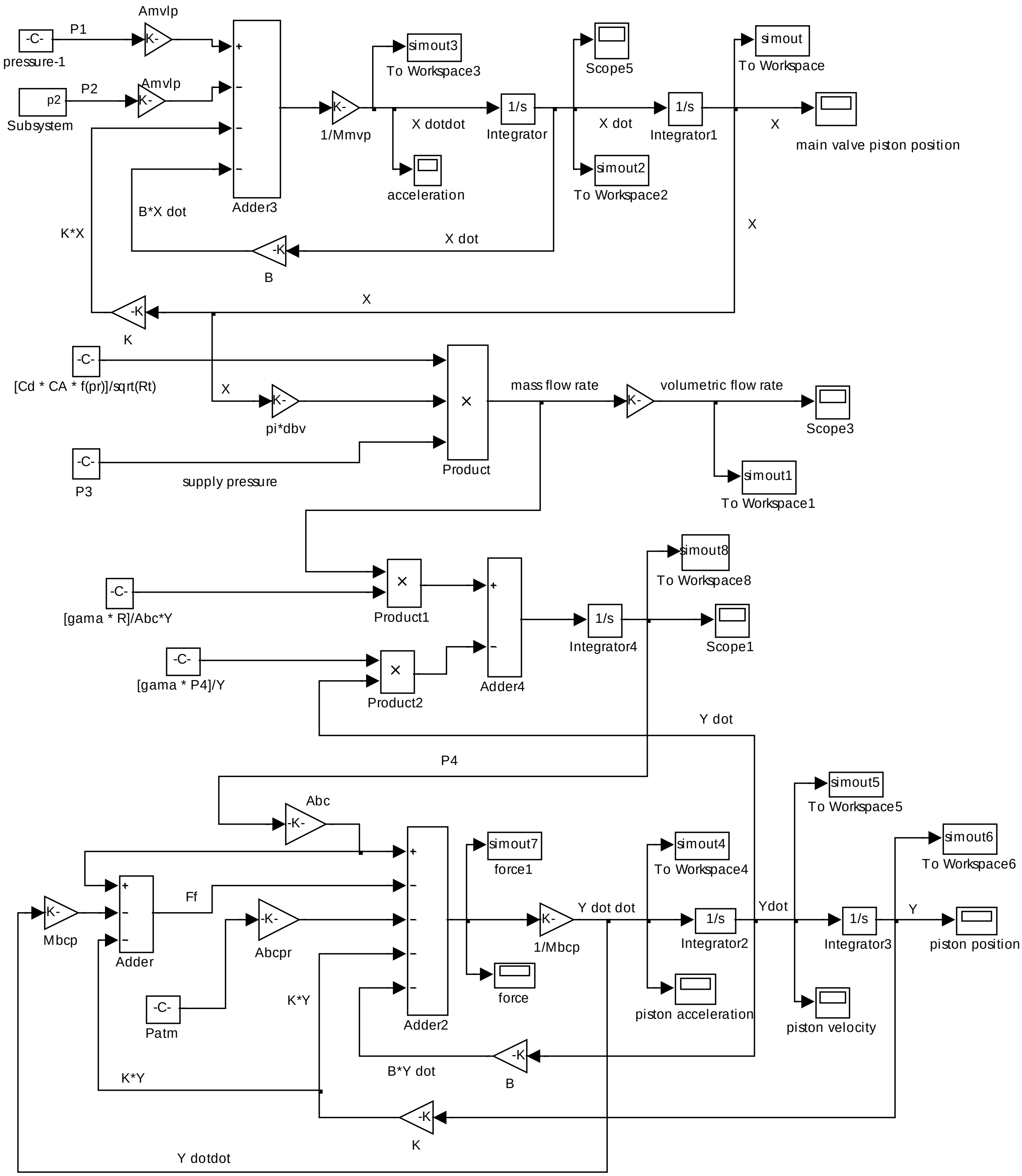


Fig.4.10. Detail System simulink model on service brake application position

4.2.5.2. Simulink model of the system on release application position

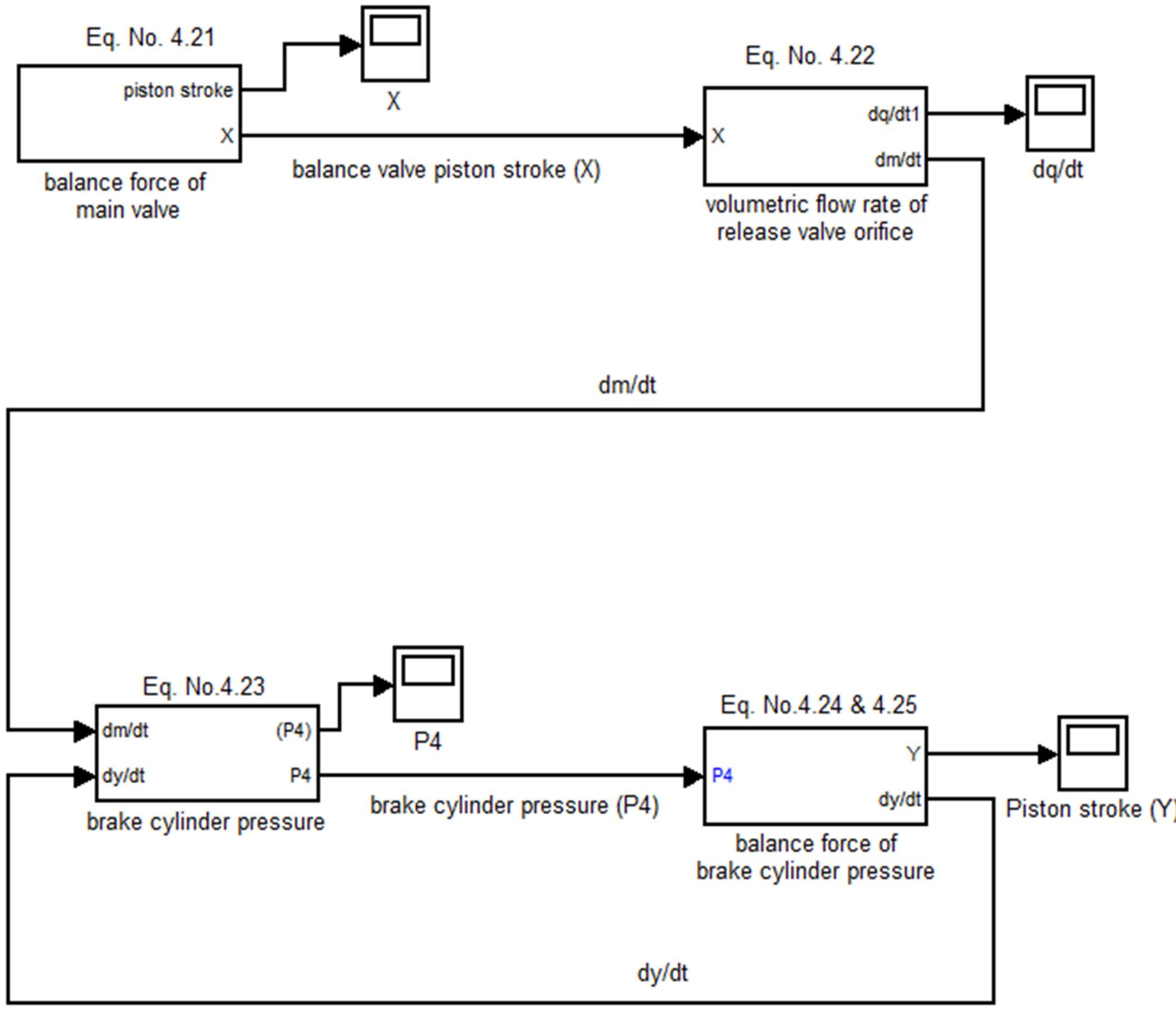


Fig.4.11. Sub-System simulink model on release application position

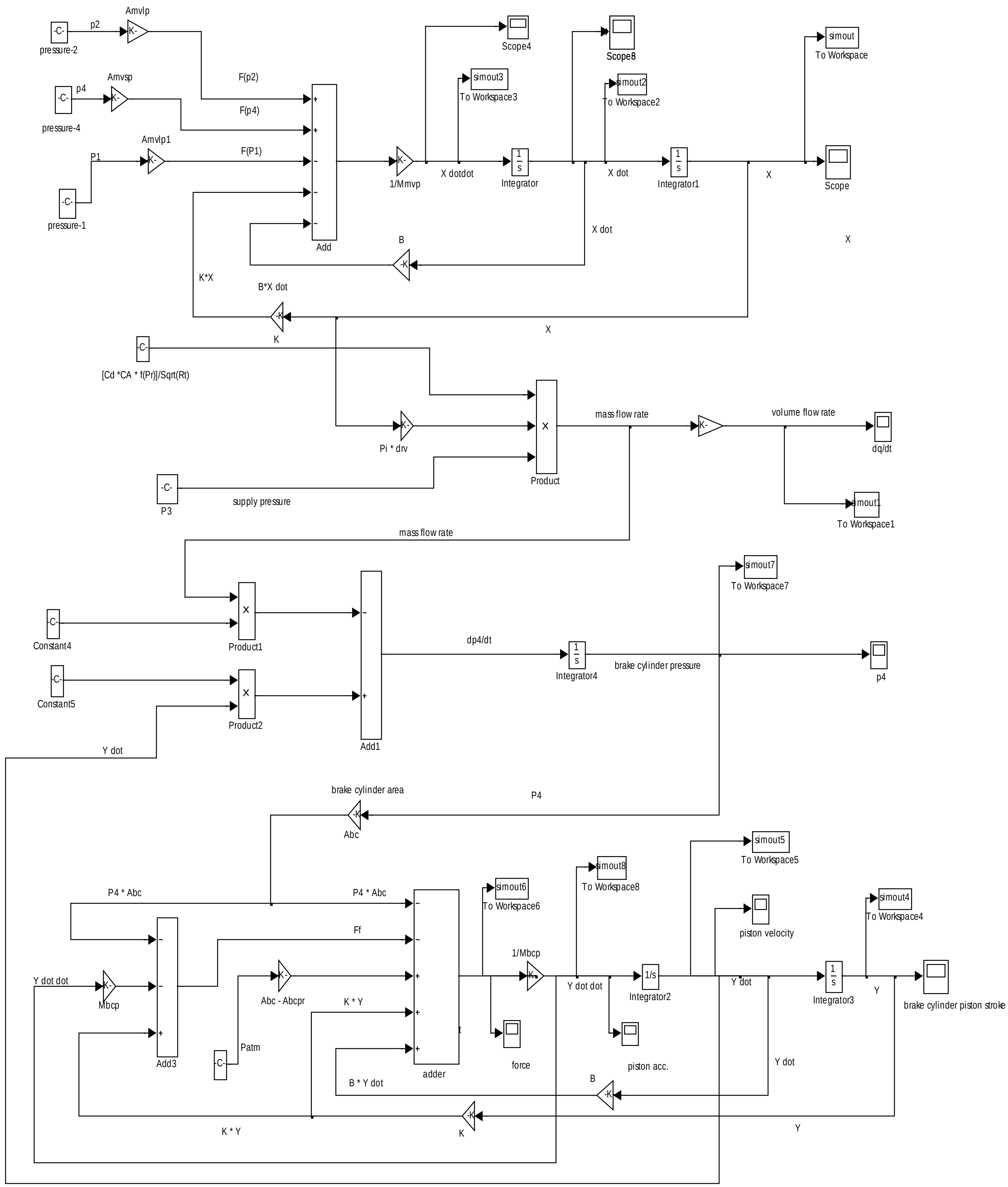


Fig.4.12. Detail System simulink model on release application position

4.2.5.3. Simulink model of system on emergency application position

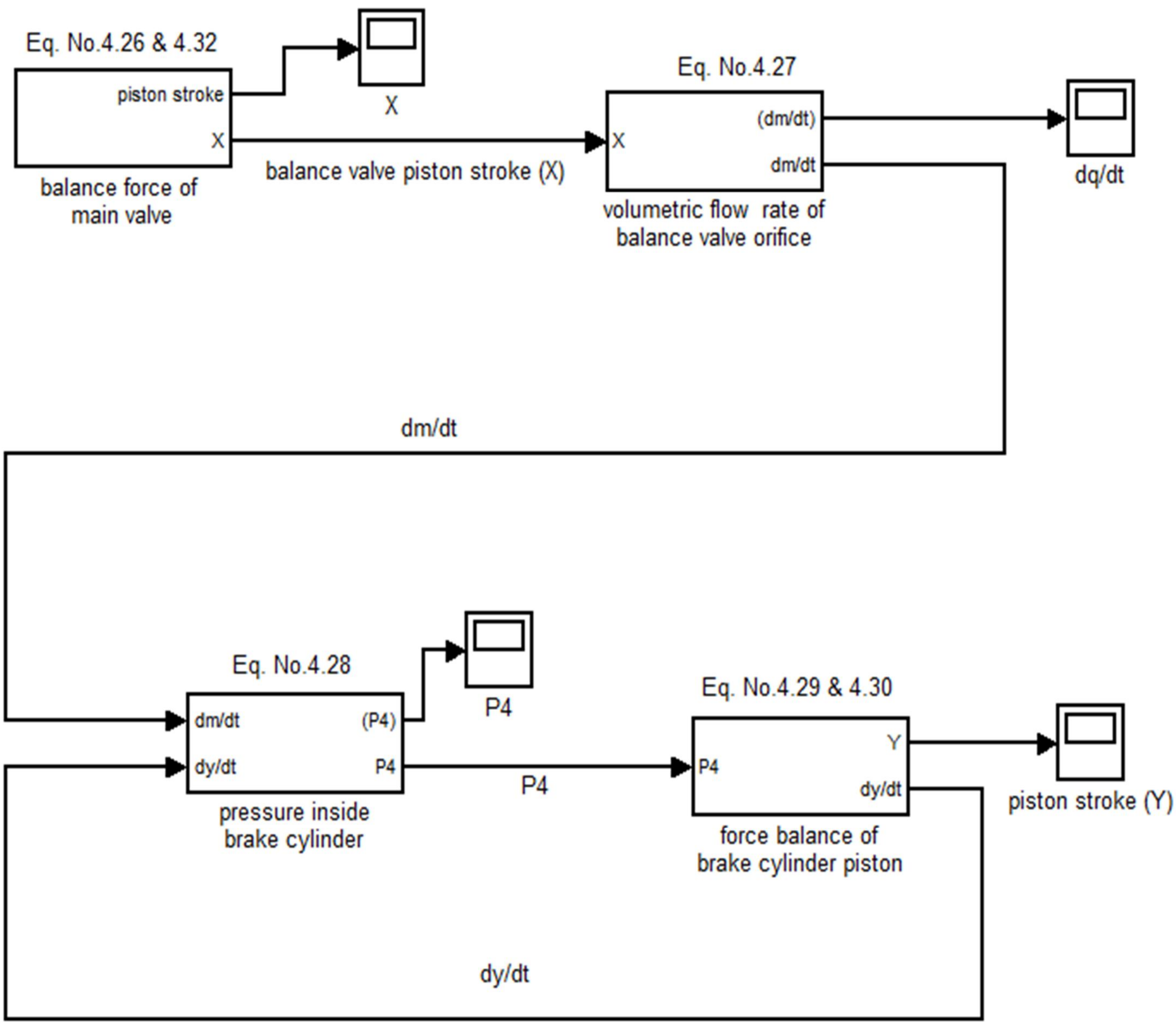


Fig.4.13 Sub-System simulink model on emergency brake application position

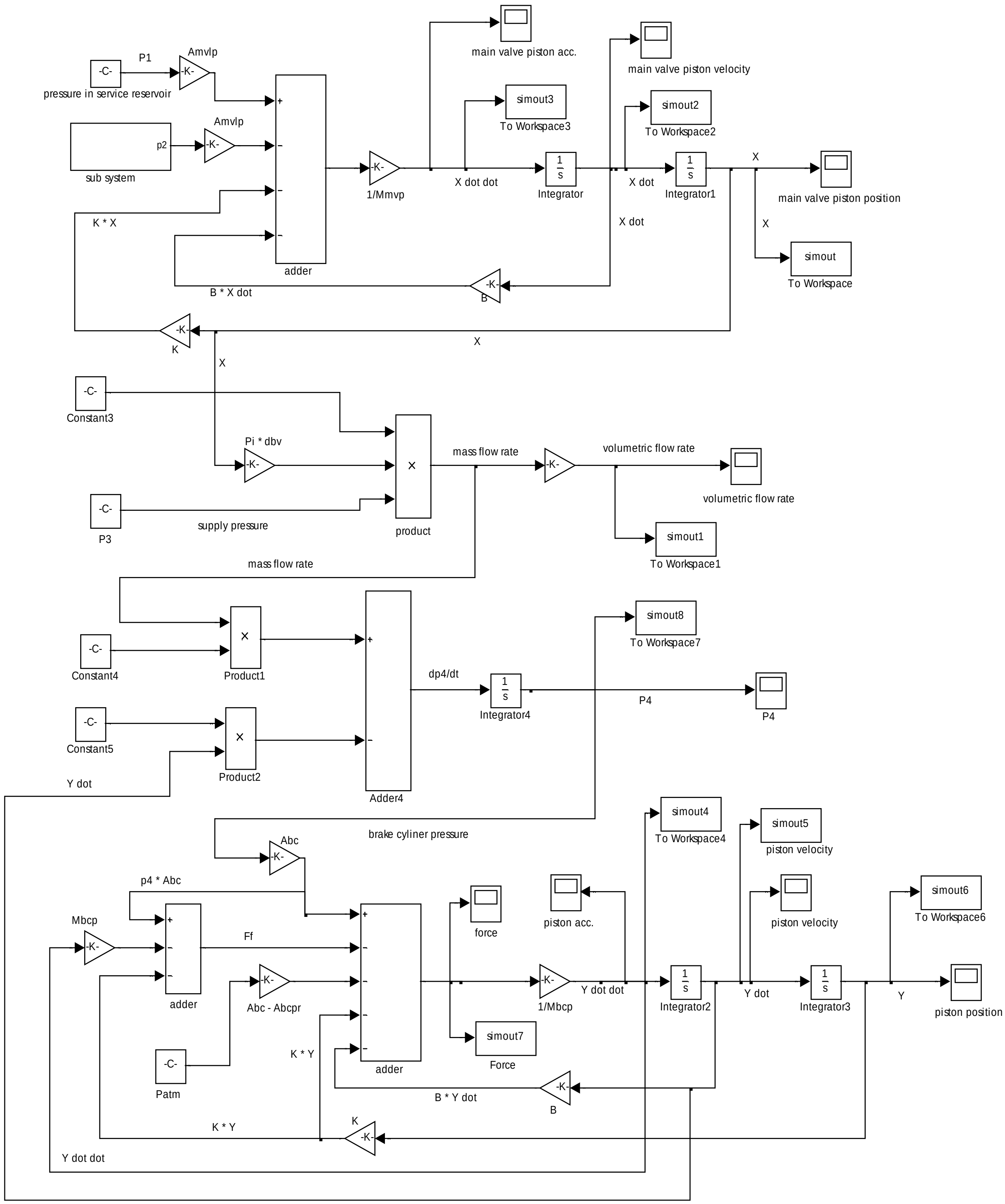


Fig.4.14 Detail System simulink model on emergency brake application position

4.2.6. Parameters and variables used in the simulation model

Parameters: the parameters of the electro pneumatic braking system of train vehicle simulation are listed in the table below from [9], [8], [19], [27], [25] and [28].

Table 3.1 parameters of the simulation model

parameters	symbol	unit	value
Train pipe pressure	P_2	kpa	600
Brake cylinder inside diameter	d_{bc}	m	0.365
Brake cylinder piston max. stroke	Y	m	0.200
Brake cylinder max. pressure (at service brake)	P_4	KPa	380
Mass of brake cylinder piston with its rod	M_{bcp}	kg	16
Active volume of brake cylinder	V_{bc}	lit	15
Brake cylinder pipe diameter	d_{bcp}	m	0.019
Brake cylinder piston rod diameter	d_{bcpr}	m	0.028
Max. pressure in service reservoir	P_1	KPa	600
Volume of service reservoir	V_1	lit	11
Service reservoir pipe diameter	d_{srp}	m	0.012
Max. pressure in auxiliary reservoir	P_3	kpa	600
Volume of auxiliary reservoir	V_3	lit	80
Auxiliary reservoir pipe diameter	d_{arp}	m	0.019
Train pipe line diameter	d_{tp}	m	0.025
Atmospheric pressure	P_{atm}	KPa	101.325
Atmospheric temperature	T_{atm}	K^0	293.15
Stiffness of brake cylinder spring	K_{bc}	N/m	16010
Stiffness of balance valve spring	K_{bv}	N/m	1500
Stiffness of release valve spring	K_{rv}	N/m	1500
Damping coefficient of brake cylinder	β_{bc}	NS/m	12
Damping coefficient of balance valve	β_{bv}	NS/m	5.0
Damping coefficient of release valve	β_{rv}	NS/m	5.0
Max. stroke of balance valve piston	X_{bvp}	m	0.008
Max. stroke of release valve piston	X_{rvp}	m	0.008
Mass of main valve piston	M_{mvp}	kg	3.0
Diameter of balance valve	d_{bv}	m	0.014
Diameter of release valve	d_{rv}	m	0.014
Main valve large piston diameter	d_{mvlp}	m	0.040
Main valve small piston diameter	d_{mvsp}	m	0.030
Diameter of main valve piston rod	d_{mvpr}	m	0.008
Area of brake cylinder piston	A_{bcp}	m^2	0.10463
Area of brake cylinder piston rod	A_{bcpr}	m^2	0.00062
Max. area of balance valve	A_{bv}	m^2	0.00035

Max. area of release valve	A_{rv}	m^2	0.00035
Area of main valve piston rod	A_{mvpr}	m^2	0.0000503
Area of main valve large piston	A_{mvlp}	m^2	0.00121
Area of main valve small piston	A_{mvsp}	m^2	0.00066
diameter of electromagnetic valve exhausting port	d_{eep}	m	0.002
diameter of emergency valve exhausting port	d_{evep}	m	0.004
Specific heat ratio for air	γ	……………	1.4
General gas constant	R	M^2/s^2k^0	287
Discharge coefficient	C_d	……………	0.4
Constant coefficient of orifice cross-section	C_A	……………	0.26

Variables of the simulation model

The variables of the simulation model are summarized in the table below

Table 3. 2 variables of simulation model

position / variables	Service brake Application	Release application	Emergency brake application
Balance valve piston position	✓		✓
Balance valve piston velocity	✓		✓
Balance valve orifice volumetric flow rate	✓		✓
release valve piston position		✓	
release valve piston velocity		✓	
release valve orifice volumetric flow rate		✓	
Brake cylinder piston position	✓	✓	✓
Brake cylinder piston velocity	✓	✓	✓
Longitudinal force developed	✓		✓
Brake cylinder pressure	✓	✓	✓

The values of these variables are determined from the simulation diagrams listed below.

CHAPTER-V

RESULT AND DISCUSSION

5.1. Results and discussion on service brake application position

5.1.1. Results on service brake application position

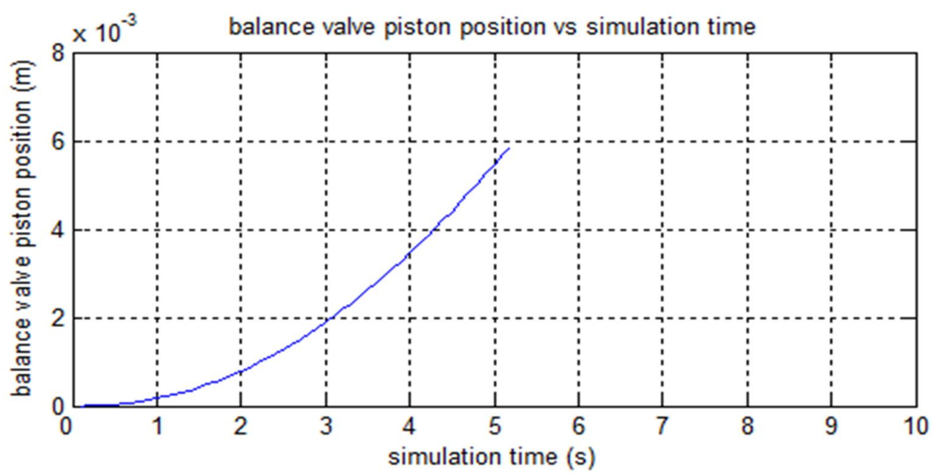


Fig.5.1 Balance valve piston position in service application position

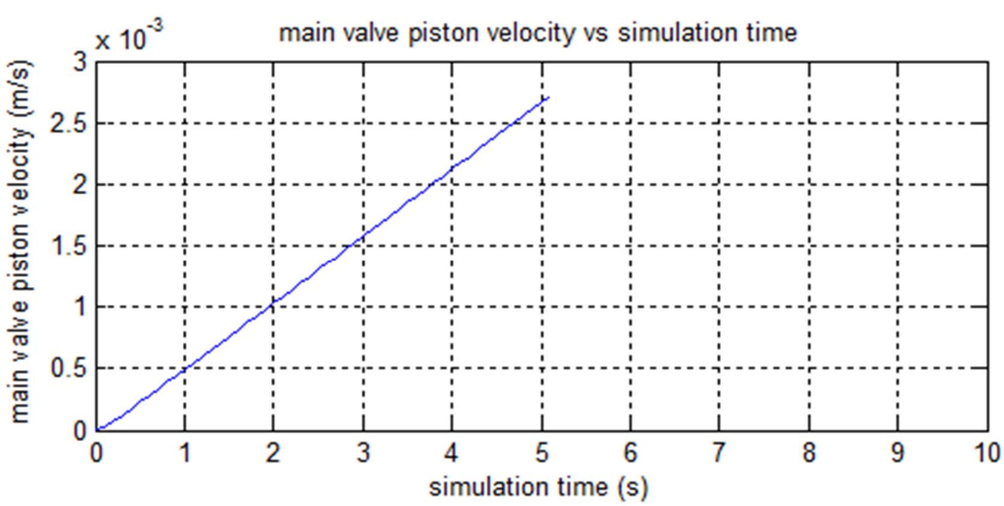


Fig.5.2 Balance valve piston velocity in service application position

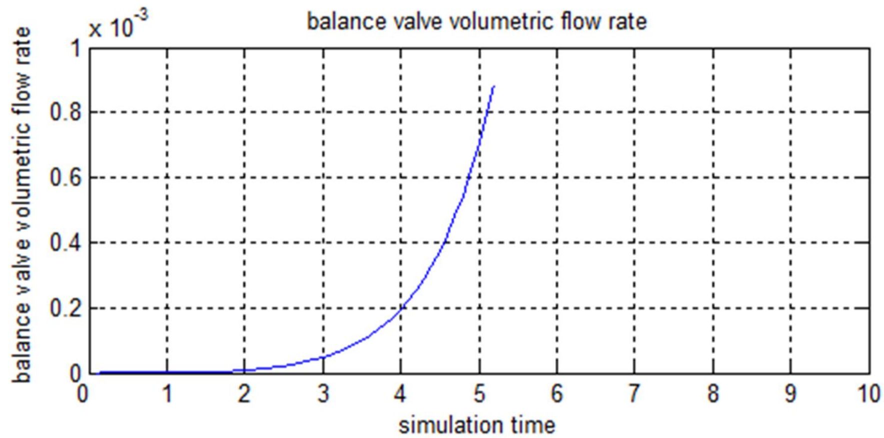


Fig.5.3 Balance valve orifice fluid volumetric flow rate in service application position.

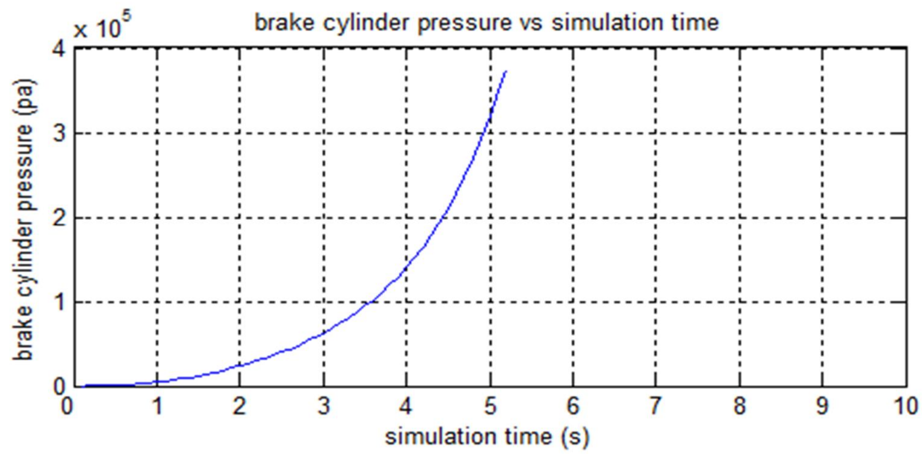


Fig 5.4 Service brake cylinder pressure in service application position

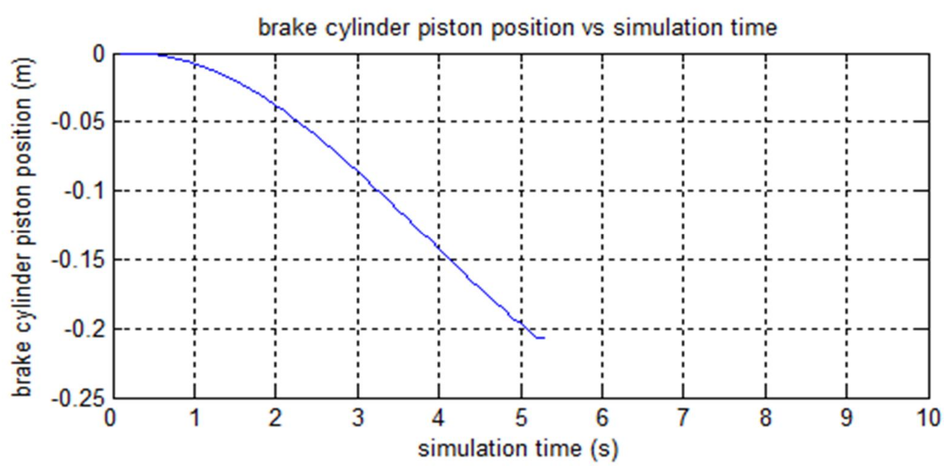


Fig.5.5 Brake cylinder piston position in service application position

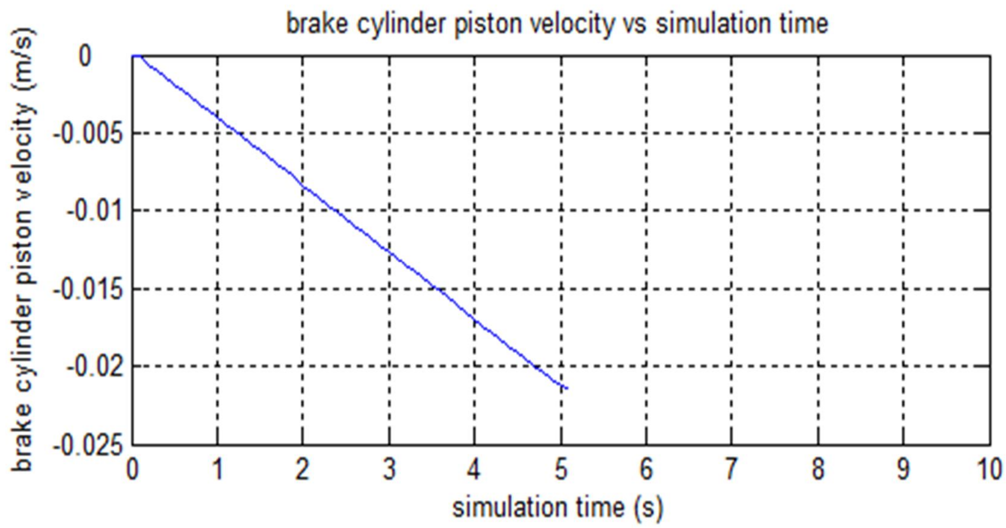


Fig.5.6 Brake cylinder piston velocity in service application position

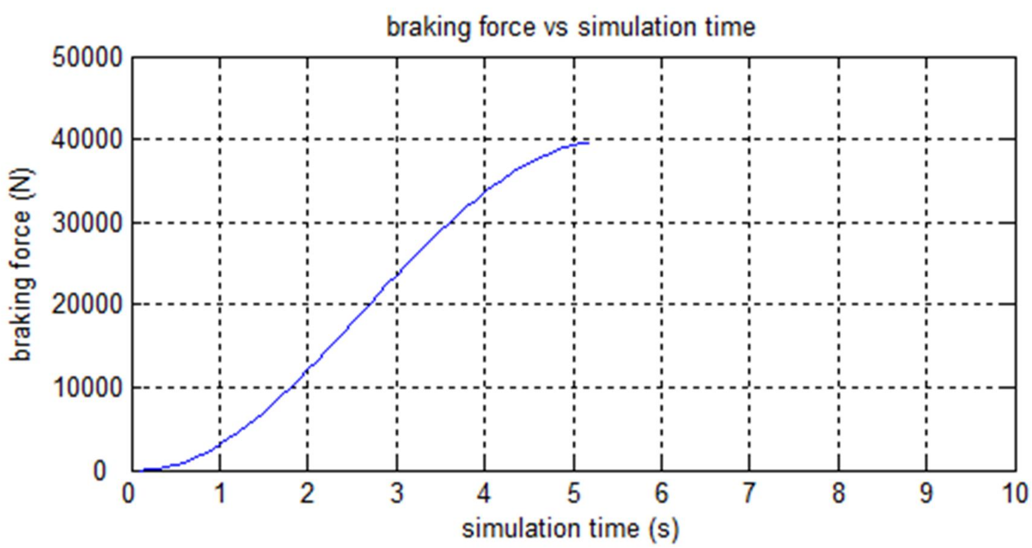


Fig5.7. Braking force developed in service application position

5.1.2. Discussion on Service brake application position

There is a standard piston cylinder travel (stroke) of 200mm and standard auxiliary reservoir pressure of 600kpa.so this research study fits to the large volume (80lit) of charged pressure (600kpa) flows to small volume (15lit) of empty pressure (atmospheric pressure). In this case since the flow is from large volume and high pressure to small volume and low pressure and even though equalization may not happen but braking should takes place at a pressure higher than half the original auxiliary reservoir pressure.

The maximum stroke of brake cylinder piston from previous literature is 200mm [8]. The brake is applied after the brake cylinder piston reaches its maximum stroke of 200mm.From the simulation result on fig 5. 5 the simulation time to reach this maximum stroke is around 5.2 sec. From the simulation result of fig 5.4 the maximum pressure in the brake cylinder chamber is 380KPa**.** This maximum brake cylinder pressure is obtained at the maximum simulation time of 5.2 sec. because the maximum brake cylinder stroke is obtained at the time when the brake cylinder gets its maximum pressure. As observed from fig 5.1 the balance valve orifice is not fully opened at the simulation time of 5.2 sec. the position of distributor main valve piston at that indicated simulation time is 0.006m which is not fully opened. Even though the system doesn't reach equalization point it is protected by adjustable pressure limiting valve that limits pressure more than that maximum value from entering to the brake cylinder. This is a safety valve protecting the wheel thread and rail flange from skidding.

The maximum balance valve orifice volumetric flow rate in service application position is found when the balance valve piston stroke reaches its maximum value. From fig 5.3 the valve is opened 0.006m from its closed position at simulation time of 5.2 sec and the flow rate at this instant is $0.9*10^{-3}m^3/s$. Finally from fig 5.7 the longitudinal braking force developed at service brake application position with max. Brake cylinder pressure of 380kpa is F = 39kN. This braking force is developed after a delay time of 5.2ses.
From previous literatures **[9]** full braking at service brake application positions is found after a time delay of 3.5sec – 5.5sec.

Therefore, since the simulated result is in the range with the experimentally found values the system is modeled correctly and this simulating software is appropriate for this particular paper.

5.2. Results and discussion on release application position

5.2.1. Results on release application position

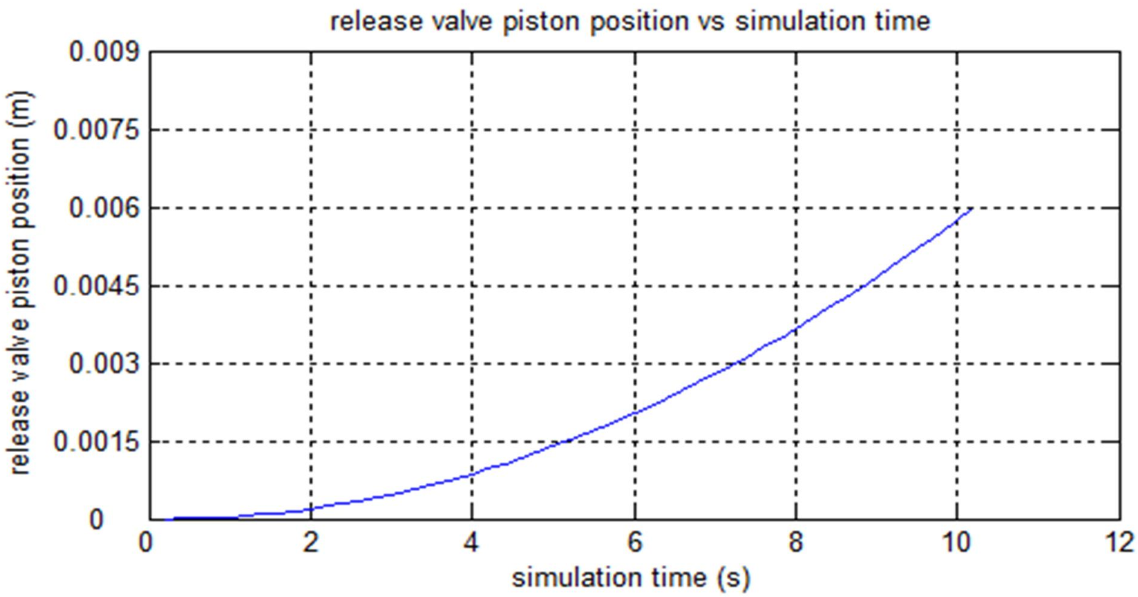


Fig.5.8 Release valve piston position in release application position

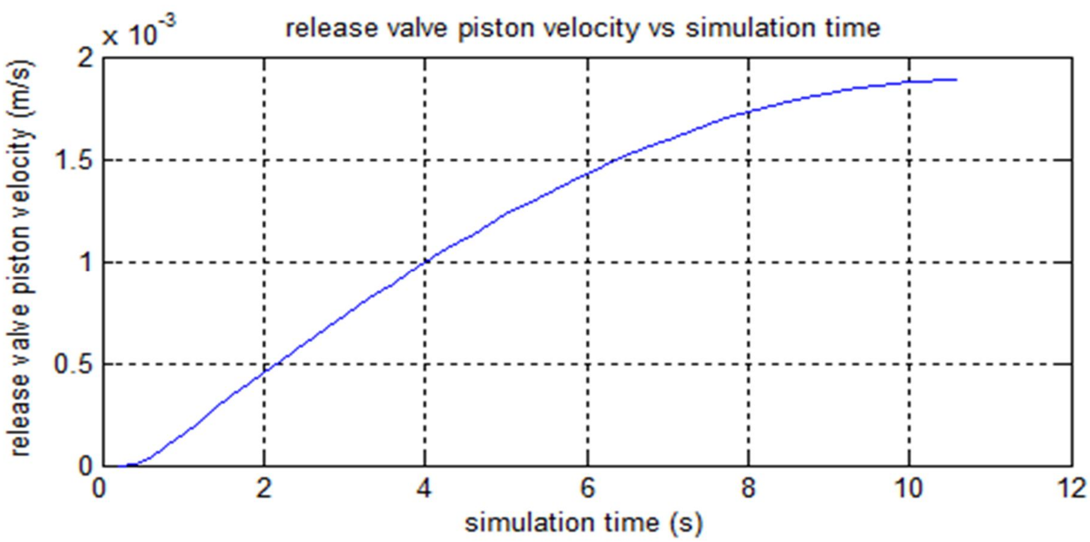


Fig.5.9 Release valve piston velocity in release application position

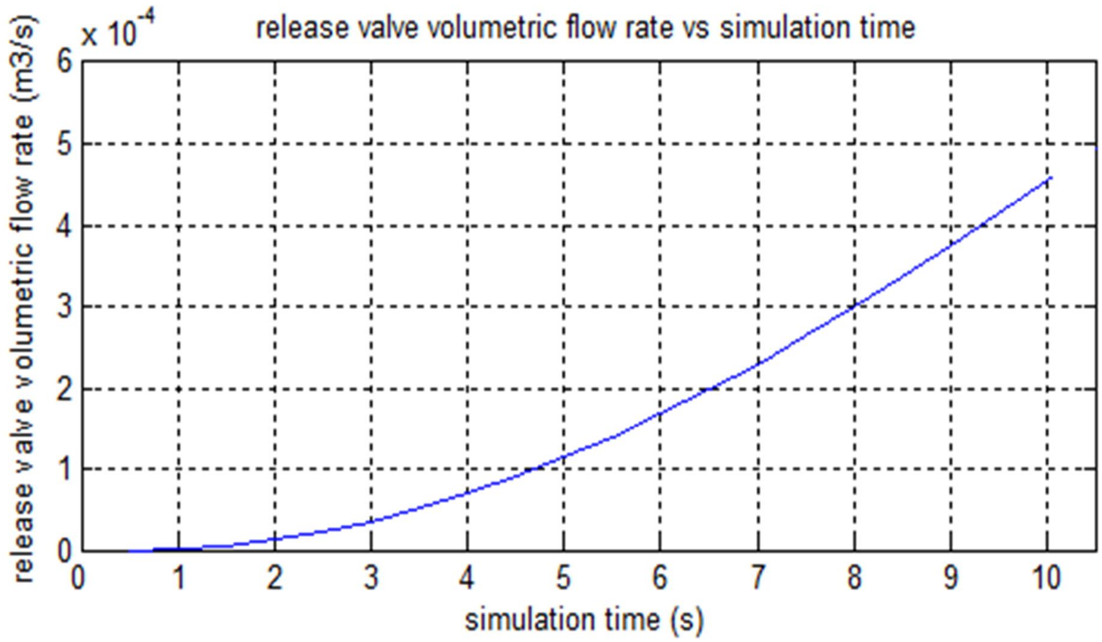


Fig.5.10 Release valve orifice fluid flow rate in release application position

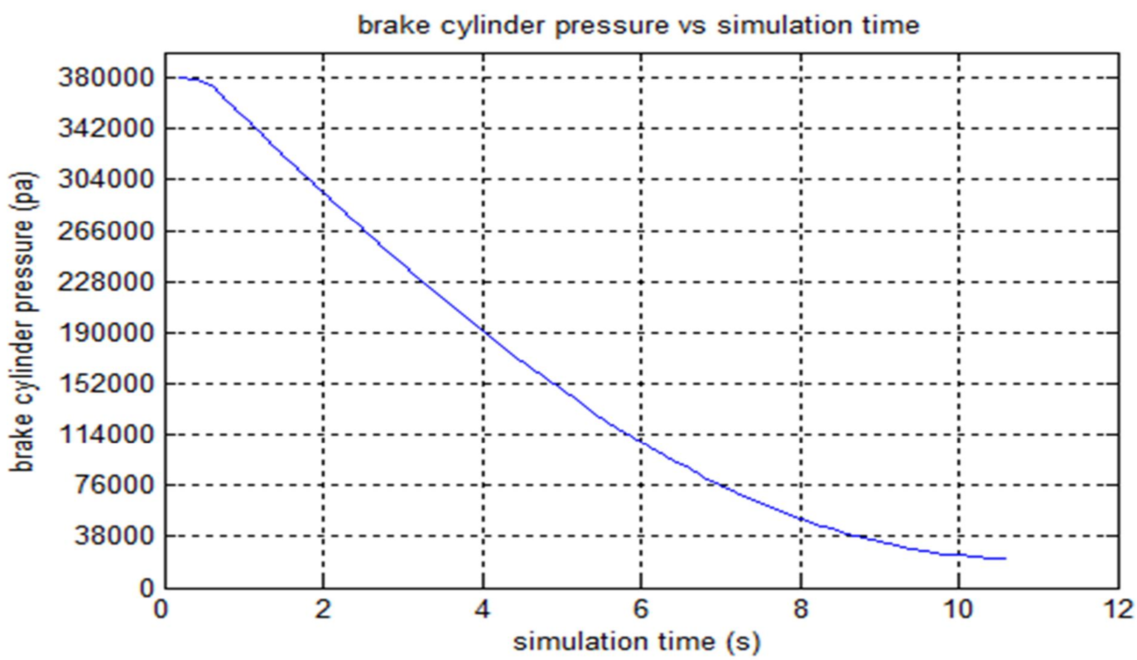


Fig 5.11 Release brake cylinder pressure in release application position

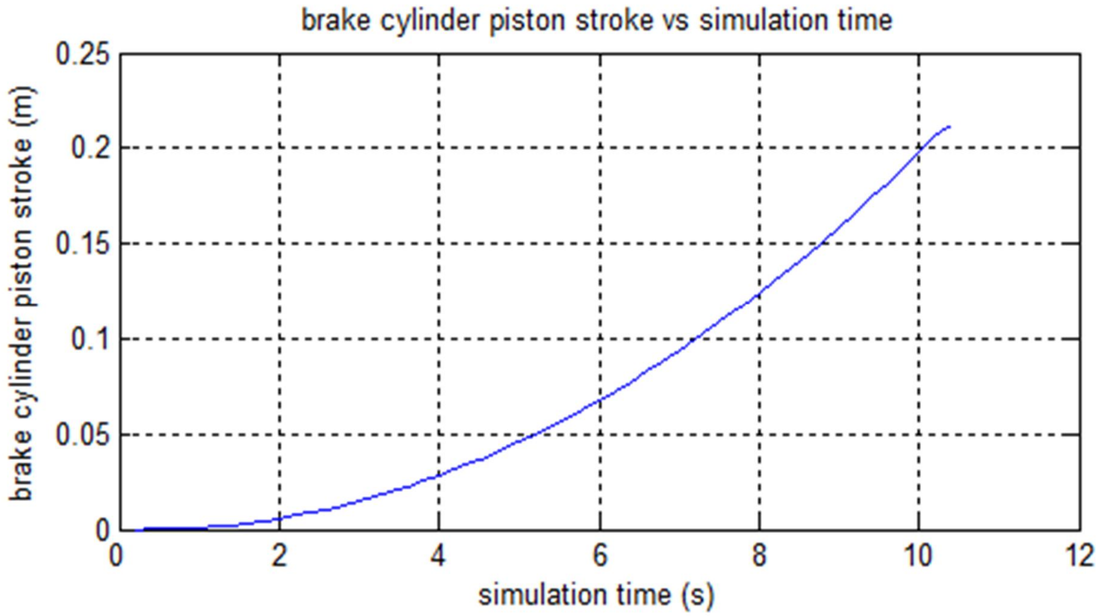


Fig.5.12 Brake cylinder piston position in release application position

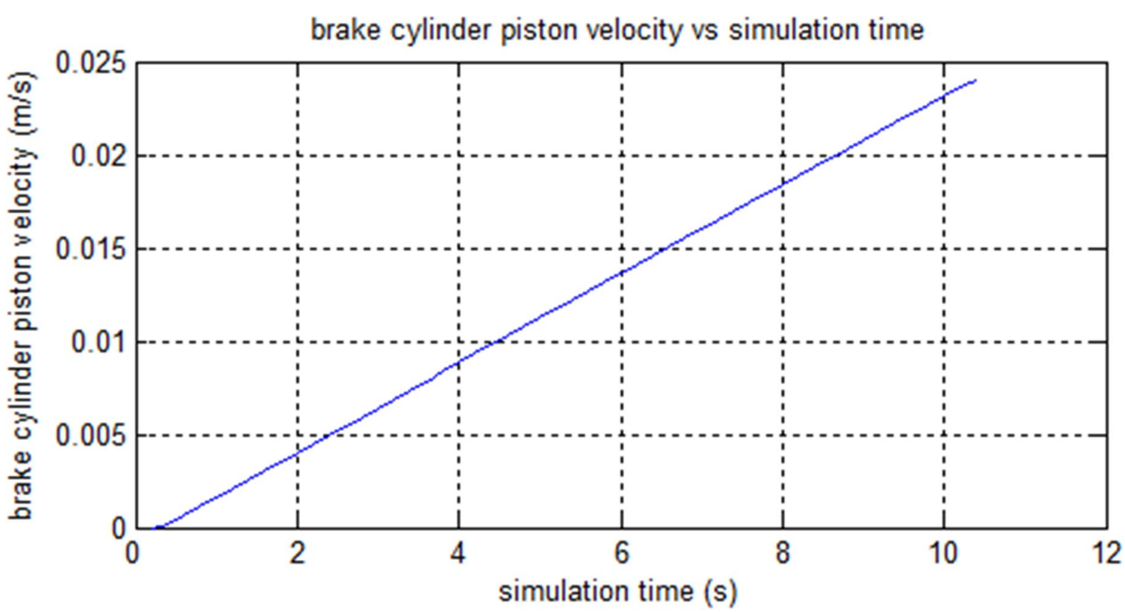


Fig.5.13 Brake cylinder piston velocity in release application position

5.2.2. Discussion on release application position

Here before release application initially the train pipe is exhausted to the atmosphere and the brake cylinder is filled with compressed air at service braking application time. So up on the application of release electromagnetic valve the reverse becomes true. That is the train pipe charges with compressed air and the brake cylinder releases to the atmosphere through release valve. The pressures that make the release valve in equilibrium are:

I. Upward pressure (P_1) i.e pressure in service reservoir and
II. Downward pressures (P_2 and P_4) i.e pressures in train pipe and brake cylinder respectively.

Since the main valve is a triple pressure balance valve, though it has good graduated release action, however, during release, it is needed to wait train pipe pressure to be charge to approach constant pressure, then brake cylinder pressure can be released entirely so that the release time is quite long compared to service brake and emergency brake application positions **[9].**

In release position as shown in fig.5.12 the full brake cylinder piston stroke is retracted with in a simulation time of 10.5 seconds unlike in service application position of 5.2 seconds. So the releasing action of compressed air from brake cylinder to atmosphere is slower than charging action of compressed air from auxiliary reservoir to brake cylinder. The reason is described on the above paragraph.

At the same time as shown on figures 5.3 and 5.10 the maximum volumetric flow rate of releasing valve in release application position is $0.45*10^{-3}m^3/s$ which is lower than the volumetric flow rate of balance valve at service application position. I.e the same volume of brake cylinder is released in 10.5seconds whereas in service application position the brake cylinder takes to charge is5.2 seconds. And from fig 5.4 the maximum pressure charged to brake cylinder in service application position is 380kpa. So the time needed to exhaust this amount of compressed air to the atmosphere 10.5sec.

So generally speaking the releasing action of compressed air from brake cylinder to the atmosphere is slower than charging action of compressed air from auxiliary reservoir to brake cylinder at service and emergency application positions. From previous literature of [9] the time needed to release the compressed air from brake cylinder to the atmosphere is on the range of 6sec-12sec. so the simulation result is between the recommended values.

5.3. Result and discussion on emergency brake application position

5.3.1. Results on emergency brake application position

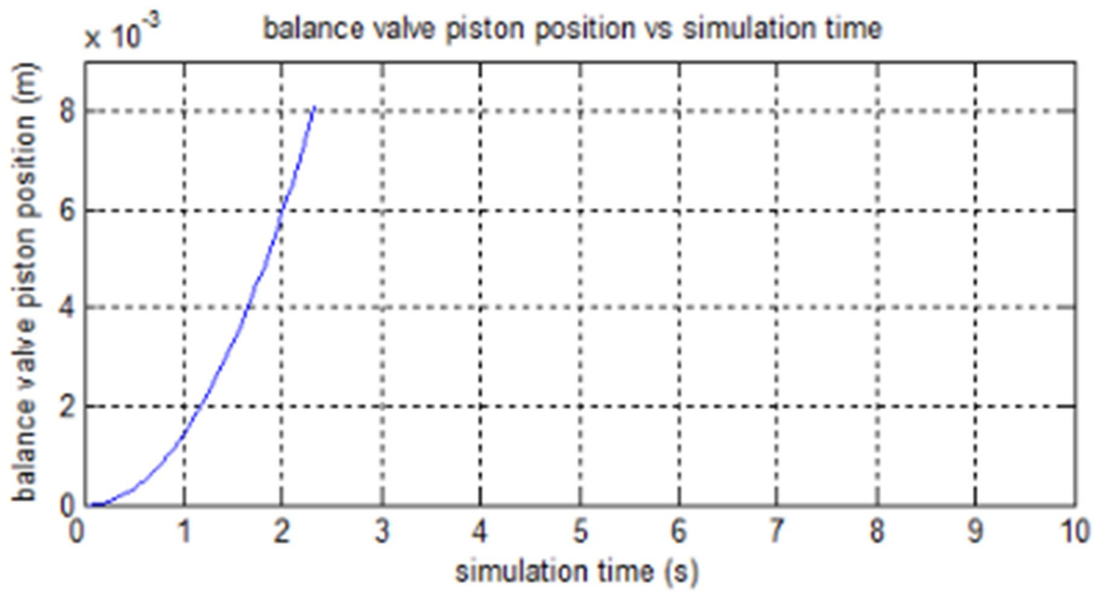


Fig.5.14 Balance valve piston position in emergency application position

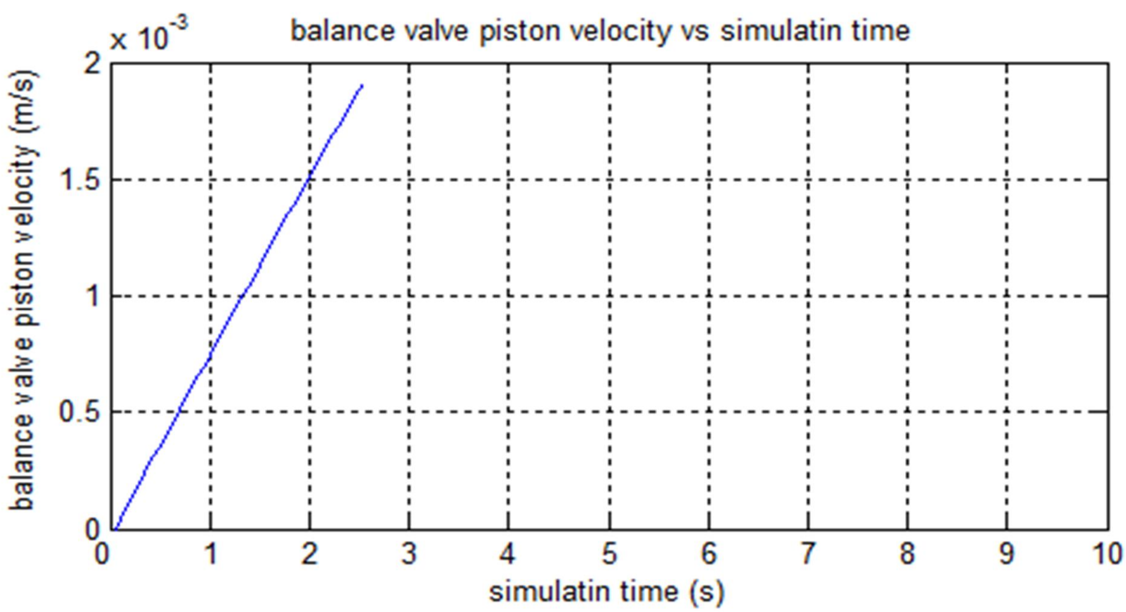


Fig.5.15 Balance valve piston velocity in emergency application position

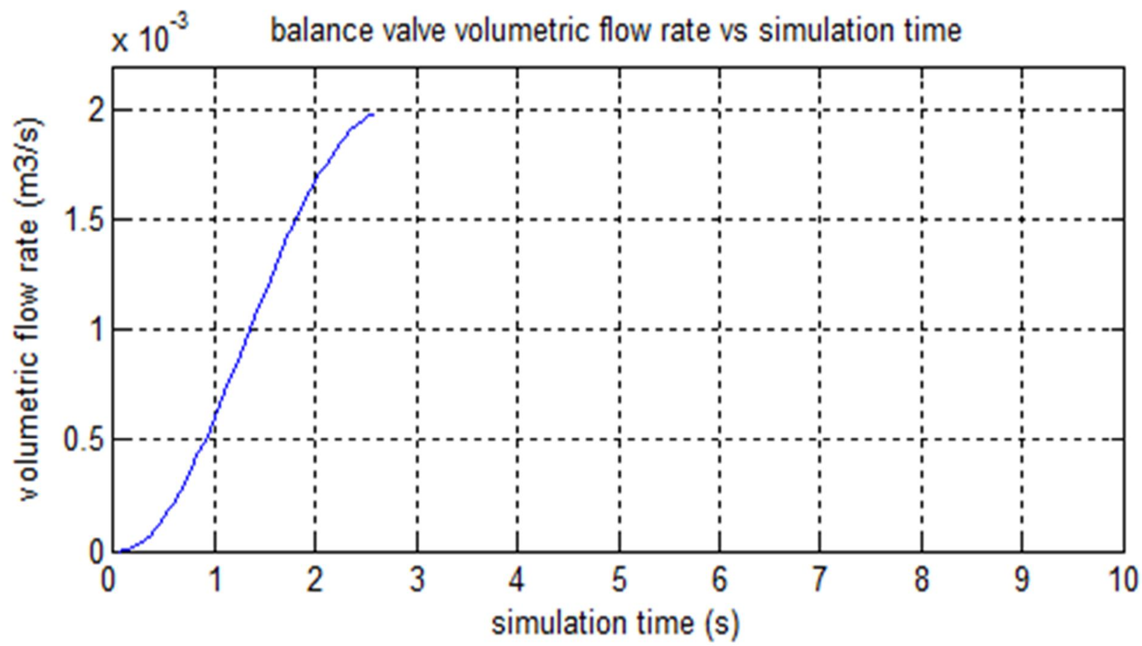


Fig.5.16 Balance valve volumetric flow rate in emergency application position

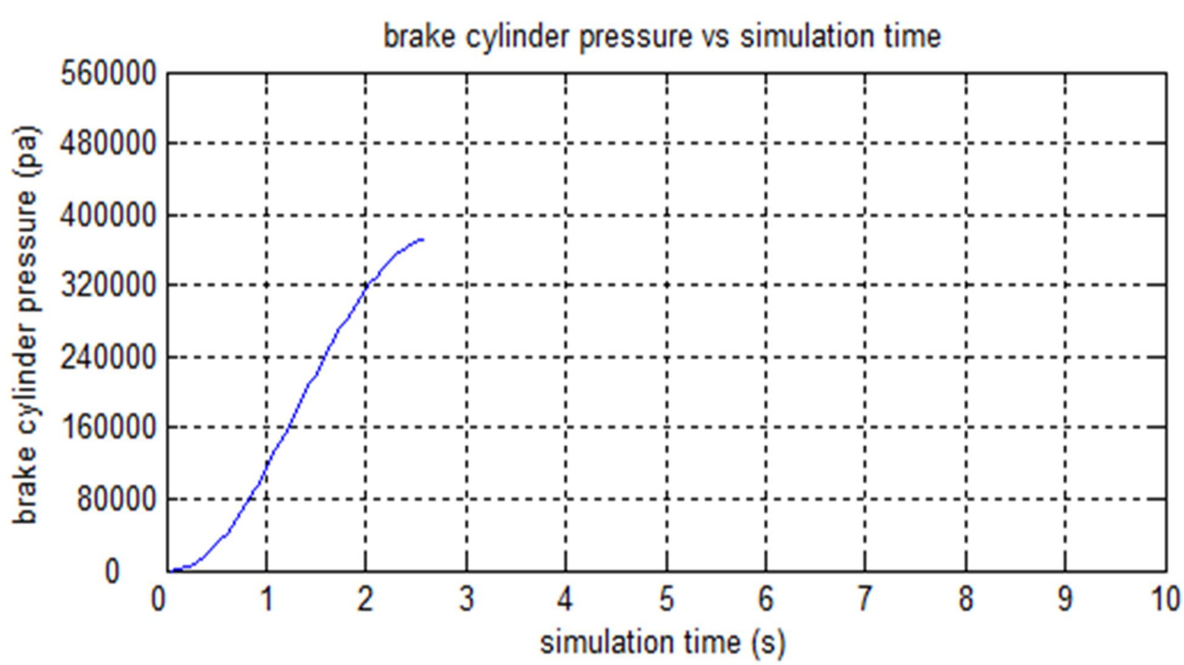


Fig 5.17 Brake cylinder pressure in emergency application position

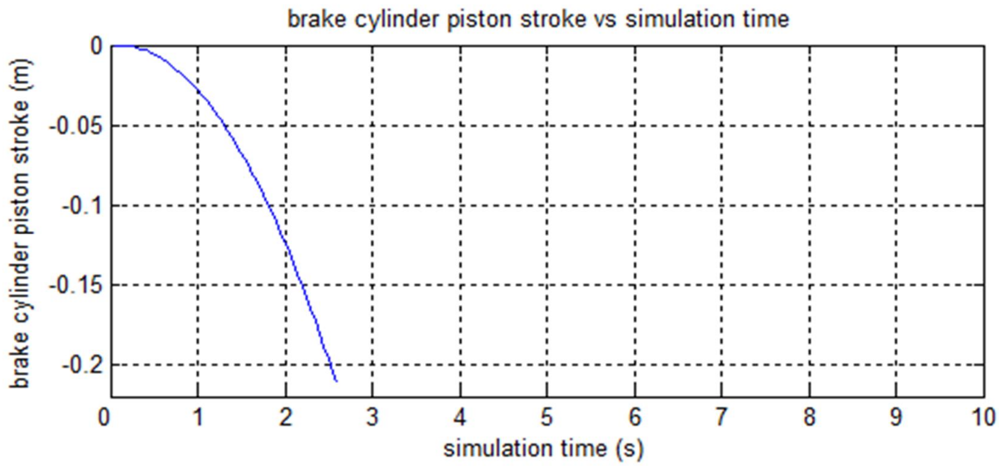


Fig.5.18 Brake cylinder piston position in emergency application position

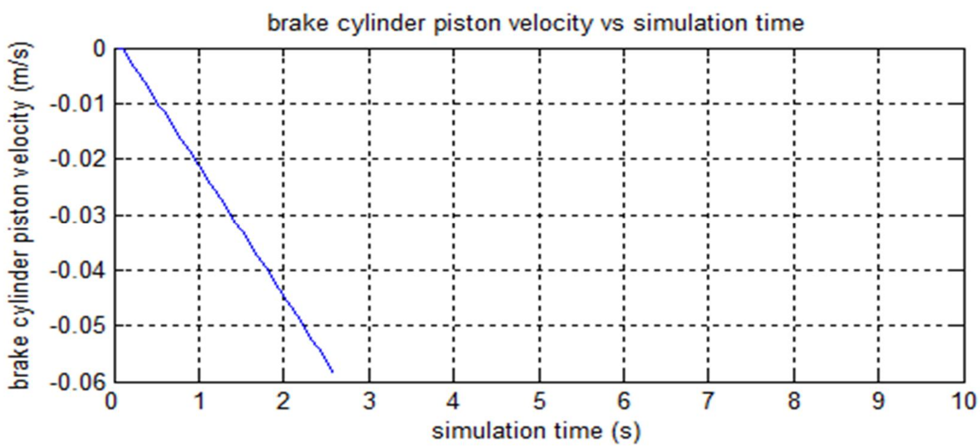


Fig 5.19 Brake cylinder piston velocity in emergency application position

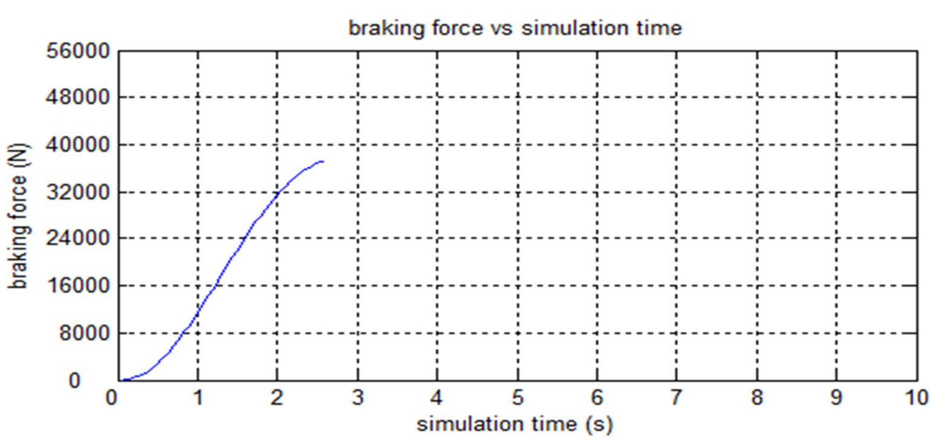


Fig 5.20 Brake cylinder force in emergency application position

5.3.2. Discussion on emergency brake application position

By the same reason as in service brake application position the direction of fluid is from large volume (80lit) of charged pressure (600kpa) i.e auxiliary reservoir to small volume (15lit) of empty pressure (atmospheric pressure) i.e brake cylinder. The only difference between service and emergency application positions is:

In service brake application position only service electromagnetic valve is exited and vented to the atmosphere at one position whereas in emergency application position both the service and emergency electromagnetic valves are excited and vented to the atmosphere in addition to emergency valve causing high pressure difference between train pipe and service reservoir.

If a brake pipe is vented from several places along its length the pressure drops much more quickly and the brake applies more rapidly compared to single vented pipe [8]. So without observing the simulated results in the service and emergency brake application positions the maximum stroke of the brake cylinder piston in emergency brake application position must attain within a short period of time due to high flow rate of balance valve orifice caused by maximum pressure difference.

The same as in service brake application position the maximum brake cylinder piston stroke is 200mm [8]. This maximum brake cylinder piston stroke is achieved in simulation time of 2.5sec (see fig 5.18). As observed from fig 5.17 the maximum brake cylinder pressure at simulation time of 2.5sec is 380kpa. Here the maximum pressure developed inside brake cylinder in service and emergency brake application positions is the same i. e p=380kpa. The only difference is the amount of time needed to develop that pressure. As observed from fig 5.14 the balance valve orifice is opened fully at simulation time of about 2.5sec.

The maximum balance valve orifice volumetric flow rate is found when the balance valve piston stroke reaches its maximum value or simply when the valve opens fully.
From fig 5.16 the maximum flow rate when the valve opens fully at simulation time of 2.5 sec is $2*10^{-3}m^3/s$ which is a high flow rate compared to service and release application positions.

Finally in fig 5.20 the maximum longitudinal braking force developed in this emergency brake application position is around F = 38.5kN which is approximately equal to 39kN in service brake application position. This is the maximum braking force. If the braking force is beyond this result wheel sliding will occur. But to prevent this problem pressure limiting valve is provided inside the distributor valve.

From [9] the time taken to charge the brake cylinder is on the range of 2sec-3sec. so the above result is on the range of that value.

Therefore, from the simulated results stated above the braking action in emergency application position is faster than in service application position.

CHAPTER-VI

CONCLUSION, RECOMMENDATION AND FUTURE WORK

6.1.Conclusion

As shown from previous literatures the air brake system on long train takes several times to propagate from the leading locomotive to the last coach **[2].**

From [2] simulation of compressed air propagation on train pipe is performed to see the effect of time delay on long trains. The simulated pipe length was 300m long. Up on simulating this system the time taken to propagate the compressed air from the main reservoir of the leading locomotive to the brake cylinder of the first coach and the coach 300m back is 23sec and 40sec respectively. This time taken is almost the same in all releasing, charging and braking processes because the commanding signal is compressed air unlike electrical current in electro pneumatic braking system.

On service and emergency brake application positions the response time of electro pneumatic braking is faster than pneumatic braking system. This is due to fast electrical command in electro pneumatic braking system rather than compressed air command in pneumatic braking system. From the simulation results of **fig 5.4** and **5.18** the response time of both the service brake application and emergency brake application is 5.2sec and 2.5sec respectively. This value is the same for the leading locomotive as well as the last coach due to electrical signal command. But on both brake application positions the maximum brake cylinder piston stroke is the same which is 200mm. the only difference between them is that the time taken to complete that stroke. On release time, even though compressed air is released from the service reservoir to the train pipe to accelerate releasing but train pipe is also charged from the main reservoir located on the locomotive as a result there is a little bit delay to charge it. As shown from the simulated result on **fig 5.12** the exhausting time of compressed air from brake cylinder to the atmosphere is 10.5sec which is greater than 2.5esc and 5.2sec on emergency and service application positions respectively.

Generally speaking from the above discussions charging of brake cylinder with compressed air is faster than exhausting in electro pneumatic braking system. And compared with the pneumatic one electro pneumatic braking is faster in time response.

6.2. Recommendation

From the simulation result the electro pneumatic braking system realizes all coaches of the train set brakes spontaneous compared to any other train braking system.

Since the entire Ethiopian railway corporation (ERC) lines are electrified lines it is recommended to buy trains that use electro pneumatic braking system to get immediate braking response throughout the train set due to electric command.

As a result there will be better passenger comfort and minimized component damage.

6.3. Future work

This paper presents the simulation of electro pneumatic braking system of the main components namely, Train pipe, Distributor valve, Auxiliary reservoir, Service reservoir and Brake cylinder. The whole system is actuated by the three electromagnetic valves electrically which are mounted on electromagnetic valve box.

These system components are simulated on three different driver's brake valve position. i.e on, service brake application position, release application position and emergency brake application position to see the response time of braking and also to compare the result with pneumatic braking system.

Therefore any one can continue this paper to simulate the whole system including

- The electrical circuit
- Electromagnetic valves on all drivers' brake valve positions and
- Other safety valves like
  - Pressure limiting valve to control the over flow of compressed air from entering to the brake cylinder to prevent wheel sliding and
  - Auxiliary reservoir charging check valve preventing back flow of compressed air from auxiliary reservoir.

CHAPTER-VII

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