INFLUENCE OF HEAT TRANSFER ON RAILWAY PNEUMATIC SUSPENSION DYNAMICS

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Abstract.

This paper deals with the modelling of railway pneumatic suspension which is commonly used as the secondary suspension on passenger trains and involves a set of various pneumatic components such as tanks, pipes and valves. It focuses on the analysis of the heat transfer between the suspension system and the atmosphere and investigates its influence on the vehicle behaviour. First, pneumatic component models are presented and it is explained how the heat exchanges are taken into account. Then, the system composed of an airspring connected to an auxiliary tank is analysed and it is shown how the dynamic stiffness is affected by the heat transfer. Finally, a metro car is analysed for a stationary twist test called $\Delta Q/Q$ and a curve entry which show discrepancies between adiabatic and isotherm cases in term of safety, dynamics and air consumption.

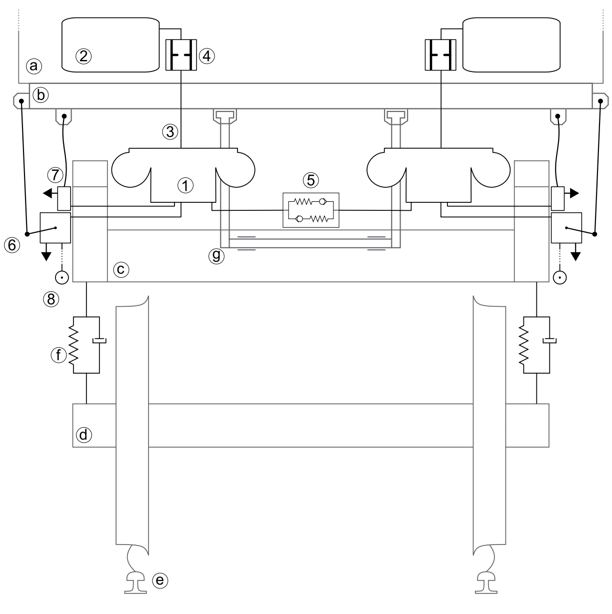
1 INTRODUCTION

Nowadays, most of passenger trains are equipped with a pneumatic secondary suspension. This system provides an increasing stiffness with increasing payload and affords to keep a constant height between the carbody and the bogie frame whatever the payload. Those capabilities are achieved by inflating or deflating the airsprings, the main components of the suspension, thanks to various pneumatic valves. Furthermore, the suspension stiffness can be adjusted by connecting auxiliary tanks to the bellows via a pipe on which an orifice can be added to increase the suspension damping. Thus, the pneumatic suspension system involves a complete suspension circuit as illustrated on figure 1.

From a modelling point of view, railway vehicles are often suitably modelled using multibody formalism whereas the pneumatic suspension requires a specific modelling task. Several airspring models exist in the literature, especially for the subsystem composed of a bellow connected to a tank via a pipe. For instance, Ref. [1] proposes an equivalent mechanical approach which takes into account rubber friction behaviour and non-linear damping in the pipe. Ref. [4] considers the air in the pipe as a constant oscillating mass which induces volume variations in the bellow and in the tank. Ref. [2] presents a model leading to a dimensionless analysis and design procedure while Ref. [3] analyses the effect of the pipe by using finite difference discretization. The SIMPACK multibody software also affords to connect several bellows together and to add control valves but the physical meaning of the model parameters is not straightforward (see Ref. [5]). In previous work, we use thermodynamical models for each component that can be easily combined in many ways in order to analyse the behaviour of the full pneumatic circuit. These models have been compared with the above-mentioned models of the literature in Ref. [8]. The refinement of pipe modelling was analysed in Ref. [9] and in Ref. [10], we use the flexibility of this approach to compare the so-called four-point suspension, for which there is one levelling valve per bellows, with the two-point suspension for which there is only one levelling valve for the two bellows of the same bogie.

In our previous work, it was assumed that the system is adiabatic, i.e. there is no heat exchange between the pneumatic circuit and the atmosphere and thus all energy dissipated by the suspension stays in the system and tends to increase the temperature. This assumption seems to be reasonable for short excitation problem where heat exchanges are small and do not have enough time to occur but it can be questionned for longer benchmark. Furthermore, for a cylindrical bellow with constant effective area, the stiffness can be approximated by $\gamma A^2 p_0/V$ in the adiabatic case and by $A^2 p_0/V$ in the isotherm case (with γ the Poisson coefficient, A the effective area, p_0 the initial pressure and V the volume). The goal of this paper is therefore to analyse whether this γ factor difference due to heat transfer could be significant for railway vehicle dynamics. The focus will be put on the heat exchange between the bellows or the tanks and the atmosphere, flow in the pipe being still assumed to be adiabatic.

In section 2, we first present the thermodynamical models and explain how the equations are completed to take the heat transfer into account. In section 3, we then focus on the subsystem composed of one bellow and one auxiliary tank connected via a pipe and we analyse the influence of heat exchange on the suspension dynamic stiffness.



Vehicle main components

a	Carbody
b	Bolster
c	Bogie frame
d	Wheelset
e	Track
f	Primary suspension
g	Anti-roll bar

Pneumatic components

1	Bellow
2	Auxiliary tank
3	Pipe
4	Pipe orifice
5	Differential pressure valve
6	Levelling valve
7	Safety valve
8	Pressure source

Figure 1. Illustration of a simplified bogie and the various components of its pneumatic suspension.

Finally, in section 4, the influence of heat transfer on the beaviour of a metro train is discussed for the $\Delta Q/Q$ test and a curve entry analysis.

2 PNEUMATIC SUSPENSION MODELLING

A railway pneumatic suspension can be modelled in various approaches, such as equivalent spring-damper, oscillating mass model, etc. Those are often limited to the analysis of a single airspring or an airspring connected to an additional tank, which can be plenty sufficient to study passenger comfort for instance. However, modelling the complete pneumatic suspension circuit, escpecially taking valves into account, requires the use of other methods like thermodynamical models presented in Ref. [8] . This section briefly presents the thermodynamical modelling approach for the various suspension components and finally shows how the heat transfer is taken into account in the bellow and tank equations.

2.1 Bellow and Tank

The bellow and the tank are modelled as pneumatic chambers in which the mass, the temperature and the pressure vary because of the flow coming from the connected pipes or from the valves.
The continuity equation imposes that the air mass variation with time is given directly by the total mass flow rate entering the chamber:

$$\frac{dM}{dt} = \dot{M} = \sum_i q_i \ , \tag{1}$$

where: M is the mass in the chamber;
q_i is an entering mass flow through port i.

The temperature variation is deduced from the internal energy variation using the first law of thermodynamics applied to an open system:

$$\frac{dT}{dt} = \dot{T} = \frac{\gamma-1}{RM}\left(\sum_i q_i h_i - \frac{RT}{\gamma-1}\sum_i q_i + \frac{dQ}{dt} - p\frac{dV}{dt}\right) \ , \tag{2}$$

where: T is the temperature in the chamber;
$q_i h_i$ is the total enthalpy flow rate at each pipe or valve connection;
dQ/dt is the heat flow rate;
p is the pressure in the chamber;
V is the volume of the chamber;
γ is the Poisson coefficient;
R is the perfect gas constant.

In the case of the tank, the volume variation with time, dV/dt, vanishes.
Given the temperature and the mass in the chamber, the pressure is deduced assuming a perfect gas situation:

$$pV = MRT \ . \tag{3}$$

For the bellow, the vertical component of the reaction force can be calculated as follow:

$$F = A_e(p_b - p_a) \ , \tag{4}$$

where p_b is the (absolute) pressure in the bellow $[Pa]$;
p_a is the atmospheric pressure $[Pa]$;
A_e is the bellow effective area $[m^2]$.

This approach allows us to connect several components to the bellow or the tank simply by considering several mass flows entering the chamber.

2.2 Pneumatic pipe

In order to be associated with the pneumatic chamber equations, the pipe model must provide a value of the flow at each of its ends given the pressure in the chambers to which its connected. Various approaches have been presented and compared in previous work (see Ref. [9]). One can first distingish between models using an algebraic relation between the flow and the pressures and models using a differential equation. In addition, models can be based on an incompressible or compressible flow. In all cases, it is assumed that there is no mass accumulation in the pipe, and thus, inflow in one end equals outflow at the other end. Furthermore, only adiabatic flow is considered. For the present work, the incompressible differential model is used, based on the following equation:

$$\dot{q} = \frac{A_p}{L_p}\left((p_2 - p_1) - \frac{1}{2\rho_p A_p^2}\left(\frac{\lambda L_p}{d_p} + \zeta\right) q^2 sign(q)\right) . \tag{5}$$

with: q, the mass flow rate in the pipe $[kg/s]$;
ρ_p, the air density $[kg/m^3]$;
L_p, the pipe length $[m]$;
d_p, the pipe diameter $[m]$;
A_p, the pipe section $[m^2]$;
λ, the distributed pressure drop coefficient $[-]$;
ζ, the lumped pressure drop coefficient $[-]$;
p_1, the pressure at port 1 $[Pa]$;
p_2, the pressure at port 2 $[Pa]$.

2.3 Valves

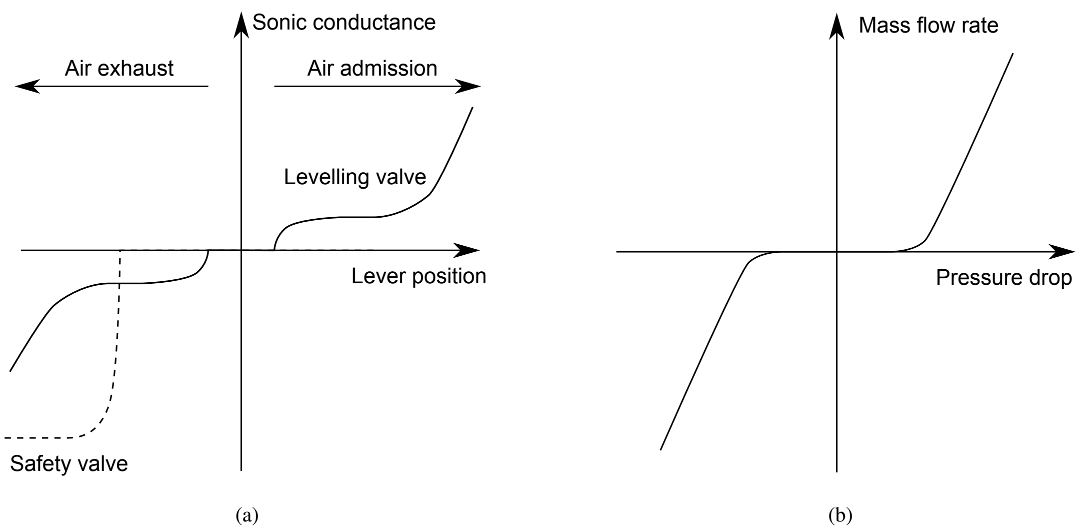


Figure 2. Qualitative valve characteristics. (a) Sonic conductance as a function of the lever displacement for levelling and safety valves. (b) Mass flow rate as a a function of pressure drop accross a differential pressure valve.

Levelling and safety valves Those two valves control the air quantity in the suspension by inflating or deflating the bellows when the carbody height is varying. Consequently, for those valves, the mass flow rate must be calculated given the upstream and downstream pressures and the valve lever displacement. The ISO 6358 standard is used to calculate the dependence with pressure:

$$q = Cp_1\rho_{ref}\sqrt{\frac{T_{ref}}{T_1}} \text{ if } \frac{p_2}{p_1} \leq b \tag{6}$$

$$q = Cp_1\rho_{ref}\sqrt{\frac{T_{ref}}{T_1}}\sqrt{1 - \left(\frac{\frac{p_2}{p_1} - b}{1-b}\right)^2} \text{ if } \frac{p_2}{p_1} \geq b \tag{7}$$

with: subscript 1: upstream conditions;

subscript 2: downstream conditions;
subscript ref: reference conditions ($T_{ref} = 293.15K$ et $p_{ref} = 1bar$);
b: critical pressure ratio $[-]$;
C: sonic conductance $[sm^4/kg]$.

The sonic conductance varies with the position of the valve lever and is given by a characteristic curve that can be calculated from experimental measurements (see figure 2(a)).

Differential pressure valve This valve is used in the four-point configuration and transfer air between the two bellows of a same bogie when the pressure difference is too large. The mass flow rate is thus a function of the two port pressures. An approximation consists in taking it as a function of the pressure difference as illustrated in figure 2(b).

2.4 Heat transfer modelling

In previous work, the whole system was considered to be adiabatic, i.e. there were no heat exchange with the atmosphere and thus all energy dissipated in the suspension, especially in the pipes, was staying in the system. The present work aims at questioning the validity of this hypothesis by considering the heat exchange between the pneumatic chambers (bellows and tanks) and the atmosphere. Therefore, the heat flow rate term $dQ = dt$ in Equation 2 which was considerd to be zero is now assumed to be proportional to the temperature difference between the pneumatic chamber and the atmosphere:

$$\frac{dQ}{dt} = k\,(T - T_{atm}) \ , \tag{8}$$

where: T is the temperature in the chamber $[K]$;
T_{atm} is the atmospheric temperature $[K]$;
k is a heat transfer coefficient $[J/s/K]$.

The key point is to estimate the heat transfer coefficient k. A null value corresponds to the adiabatic case whereas an infinite value corresponds to the isotherm case. The present work aims at showing the influence of this parameter.
As an illustration, figure 3 shows the reaction force of a bellow submitted to a 20 mm ramp displacement excitation for various value of the transfer coefficient. Between the adiabatic and isotherm cases, one can clearly distingish the intermediate behaviour of the bellow.

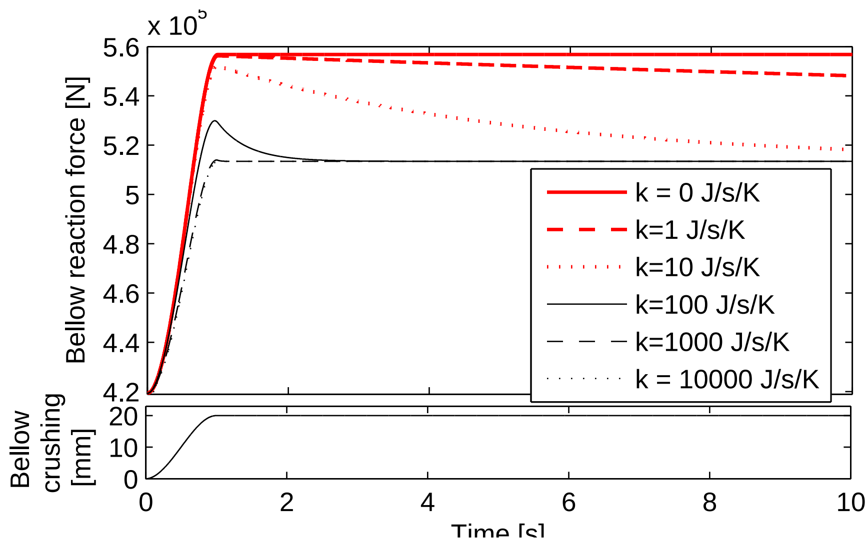
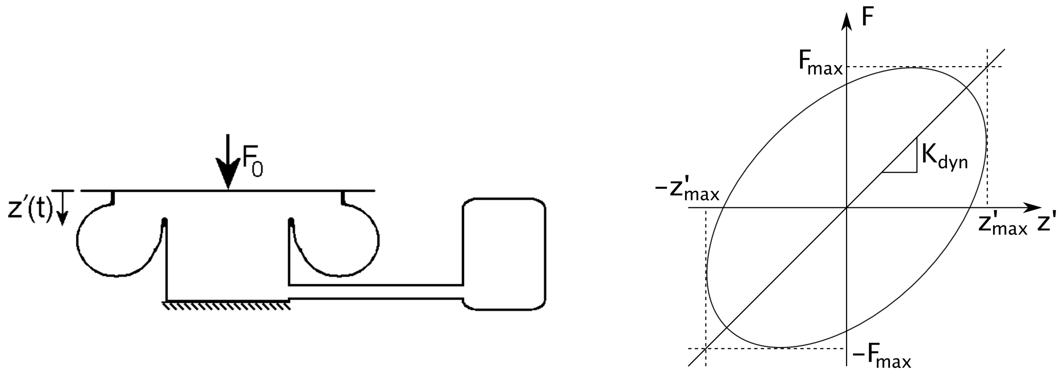


Figure 3. Reaction force of a bellow submitted to a 20 mm compression excitation.



(a) (b)

Figure 4. (a) Illustration of the bellow tank subsystem. (b) Force-displacement curves and definition of the dynamic stiffness (z' denotes the bellow crushing).

3 BELLOW-TANK SUBSYSTEM

This section focuses on the analysis of the subsystem composed of a bellow connected to an auxiliary tank via a pipe as illustrated in figure 4(a). This system forms the core structure of a pneumatic suspension and, as explained previously, it has been modelled in many ways. It is used in Ref. [8] to compare the thermodynamic models with those existing in the litterature.

The test consists in applying a sinusoidal displacement excitation to the bellow upper part and to calculate its reaction force time history. After stabilization, the force-displacement curves look like ellipses, as in figure 4(b) which illustrates the dynamic stiffness that can be defined as:

$$K_{dyn} = \frac{F_{max} - F_{min}}{2z'_{max}} \,, \tag{9}$$

where z' denotes the bellow crushing.
Figure 5 illustrates the temperature time history obtained for a 20 mm amplitude excitation at 1 Hz with parameters listed in table 1.
For a null value of the heat transfer coefficient k, i.e. the adiabatic case, the temperature increases and does not stabilize since all dissipated mechanical energy stays in the system. For $k = 1\ J/s/K$, the mean temperature first increases and then stabilizes after several excitation cycles. For $k = 100\ J/s/K$, the mean temperature is constant and the oscillation amplitude is similar to the cases $k = 0\ J/s/K$ and $k = 1\ J/s/K$. For $k = 1000\ J/s/K$, the oscillation amplitude decreases and the case $k = 10000\ J/s/K$ can be considered as isotherm due to the small temperature variations.
For higher frequencies, the behaviour tends toward the adiabatic case for coresponding k values. Furthermore, the mean temperature after stabilization is larger and is reached after a longer time since there is more energy injected in the system per second. For instance, for a 15 Hz excitation, the temperature reaches more than 6000 K for $k = 1\ J/s/K$ which is not realistic.

A similar behaviour is observed for the pressure time history. For $k = 0\ J/s/K$, the mean pressure increases indefinitely, inducing an increasing suspension static load. Moreover, the oscillation amplitude does not stabilize which make it difficult to calculate the dynamic stiffness.

Figure 5 shows the dynamic stiffness for various frequencies and various heat transfer coefficients. The cases $k = 0\ J/s/K$ and $k = 1\ J/s/K$ are not drawn because, as explained before, they induces a non realistic temperature increase.
Two constant levels at low and high frequencies can be observed. For low frequencies, the stiffness corresponds to the quasi-static stiffness of the bellow and the tank while, for high frequency, it tends to the quasi-stiffness of the isolated bellow since the pipe airflow saturates (see table 2).
Around 15 Hz, a resonance effect, due to the inertia of the air present in the pipe, can be observed. This effect is more important for small values of the heat transfer coefficient because the temperature elevation is more important

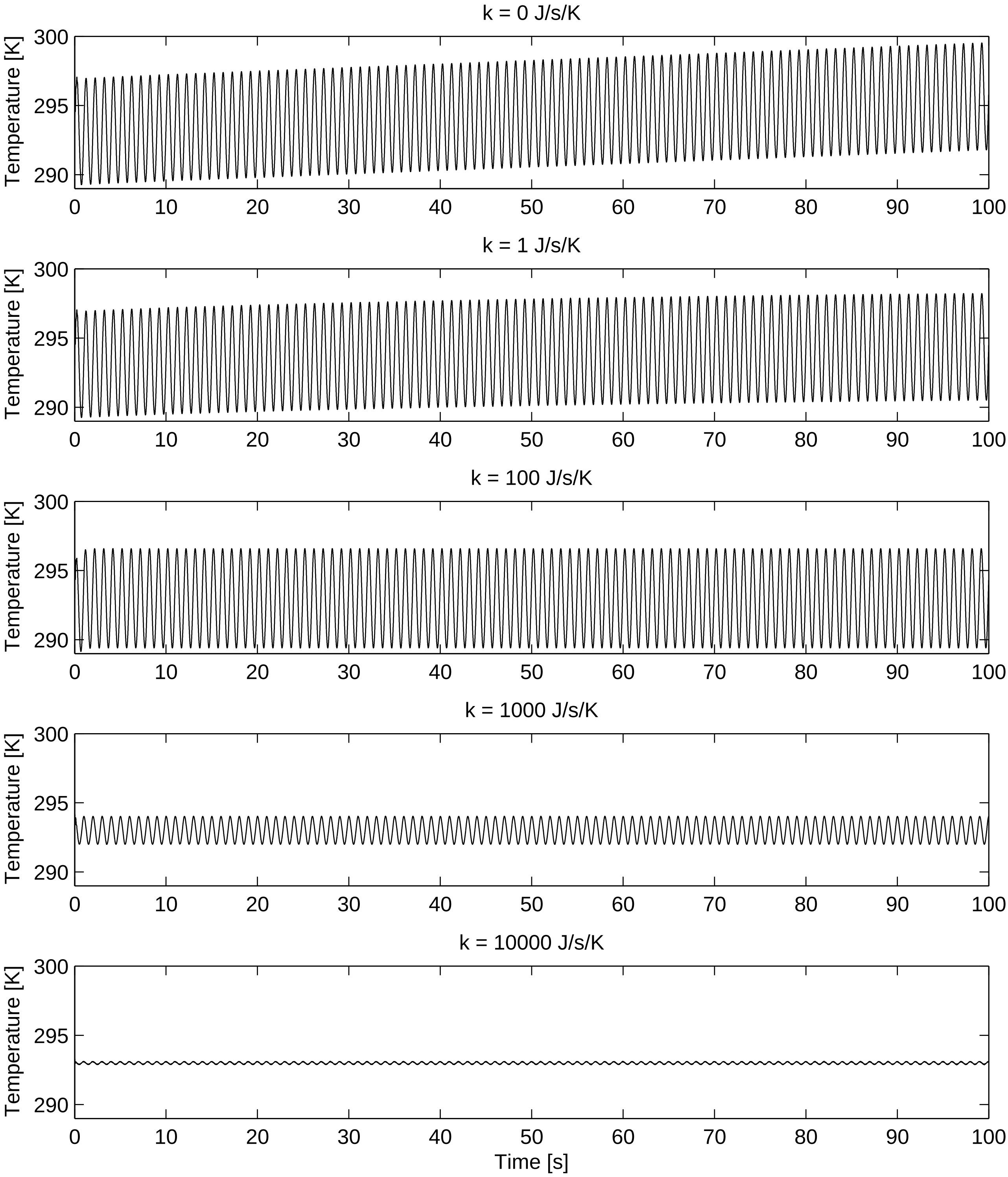


Figure 5. Bellow temperature time history when the bellow upper part is submitted to a sinusoidal displacement excitation for various heat transfer coefficients.

Nominal bellow volume	12	dm^3
Bellow volume gradient	0.13	m^2
Nominal effective area	0.13	m^2
Effective area gradient	0	m
Tank volume	27	dm^3
Initial pressure	$4\ 10^5$	Pa
Bellow-tank pipe:		
length	1 000	mm
diameter	18	mm
total pressure drop coefficient	1.11	$-$

Table 1. Pneumatic suspension parameters.

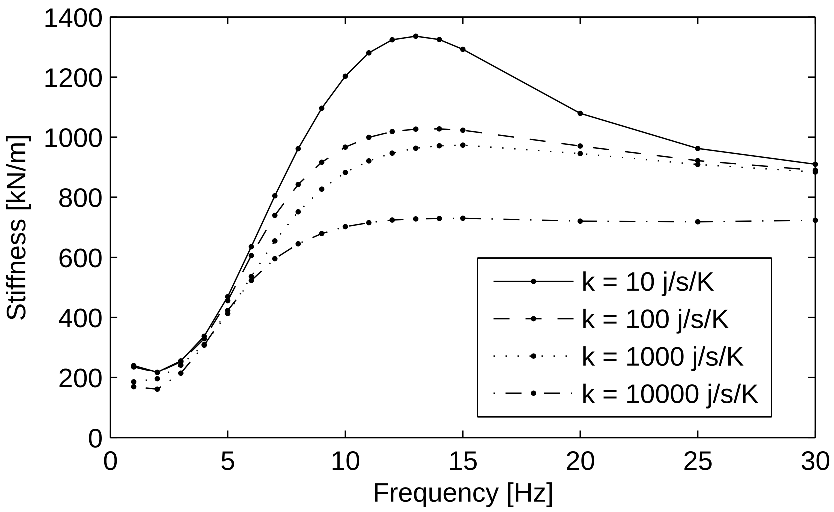


Figure 6. Dynamic stiffness of the bellow-tank subsystem for various values of the heat transfer coefficient k.

		Bellow and tank $V = V_b + V_t$	Bellow only $V = V_b$
Adiabatic	$\gamma \frac{A^2 p_0}{V}$	$243\ kN/m$	$816\ kN/m$
Isotherm	$\frac{A^2 p_0}{V}$	$173\ kN/m$	$582\ kN/m$

Table 2. Quasi-static stiffness of the suspension for the adiabatic and isotherm cases.

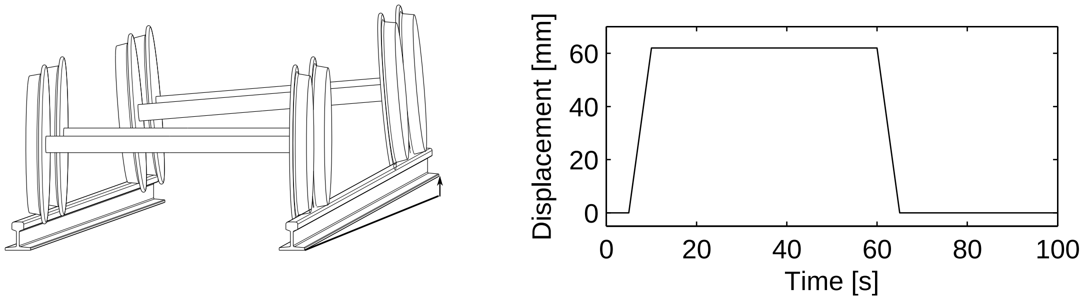
and tends to increase the mean pressure and the stiffness.
This section shows the importance of the heat transfer in the dynamic stiffness calculation, especially when using this test to adjust unknown model parameters to fit with ewperimental results. At the inverse, the measured temperature elevation could be used to estimate the heat transfer coefficient.

4 APPLICATION TO A METRO CAR

After analysing an isolated airspring and its auxiliary tank, we now present the behaviour of a metro car equipped with airsprings. The suspension is purely pneumatic: there is no anti-roll bar and no hydraulic dampers. The four-point configuration is considered, i.e. there are four levelling valves, one per bellow.
The vehicle dynamics are modelled in the multibody software SIMPACK while the penumatic equations are implemented in MATLAB/SIMULINK, the two integration process interacting via the SIMAT co-Simulation interface. The main parameters of the vehicle multibody model are listed in table 3 while the pneumatic suspension parameters corresponds to those used in section 3.

Carbody mass	17 000	kg
Bogie frame mass	1 900	kg
Wheelset mass	800	kg
Bogie center distance	10	m
Wheelset base distance	2	m

Table 3. Main parameters of the analysed vehicle.



(a) (b)

Figure 7. (a) Illustration of the twist excitation. (b) Vertical displacement imposed to the right wheels of the front bogie to simulate the twist excitation.

4.1 $\Delta Q/Q$ test

This test consists in measuring the variation of the wheel/rail force vertical component when the vehicle passes through a rail twist, i.e. the situation in which the two rails are not parallel. This analysis is conducted on a stationary train and the excitation is imposed by forcing the front wheelsets to follow a given trajectory illustrated in figure 7. As a result, the front right and rear left bellows are crushed and the front left and rear right ones are extended.

Figure 8 presents the wheel/rail force vertical component time history of the first and the fourth wheelsets wheels for several heat transfer coefficient values. As expected, the variations are more important for the adiabatic case ($k = 0\ J/s/K$) than the isotherm one ($k = 10000\ J/s/K$) since it corresponds to a higher suspension stiffness. For the intermediate values, the force first varies similarly to the adiabatic case when the twist is applied and then tends toward the isotherm case, more rapidly for larger heat transfer coefficients.

This figure higlights that, even it is not the more realistic case for the dynamic stiffness, the adiabatic case can be used for the $\Delta Q/Q$ test since it induces the more important wheel unloading, which is the most restrictive from a safety point of view. Furthermore, for small heat transfer coefficients, the maximum variations are similar to the adiabatic case.

4.2 Curve passing

We now investigate the influence of the heat transfer when the system is submitted to a roll loading due to a curve entry. The vehicle first follow a straight track at constant $10\ m/s$ before entering a $100\ m$ radius curve. The constant radius part has no cant elevation and is separated from the straight portion by a $15\ m$ length transition track.

Figure 9 presents the carbody roll angle time history. When valves are disconnected, discrepancies between the adiabatic and the isotherm cases clearly appear, the roll angle being larger in the second case due to the lower suspension stiffness. As for the $\Delta Q/Q$ test, the intermediate values of the heat transfer coefficient k are first close to the adiabatic case before tending to the isotherm case.

When valves are connected, the differences are less important because the levelling valves inflate the outer bellows and deflate the inner ones to keep a constant airspring height. In the cases $k = 1\ J/s/K$ and $k = 10\ J/s/K$, there is small oscillation when the train is in the curve because, as the temperature gradually decreases, the airspring stiffness also decreases and the valves are therefore engaged sporadically which prevents the system to reach an equilibirum. It can also be noticed in table 4 that the isotherm case induces a larger air consumption which is meausured by the total air quantity injected from the pressure source towards the bellows via the levelling valves.

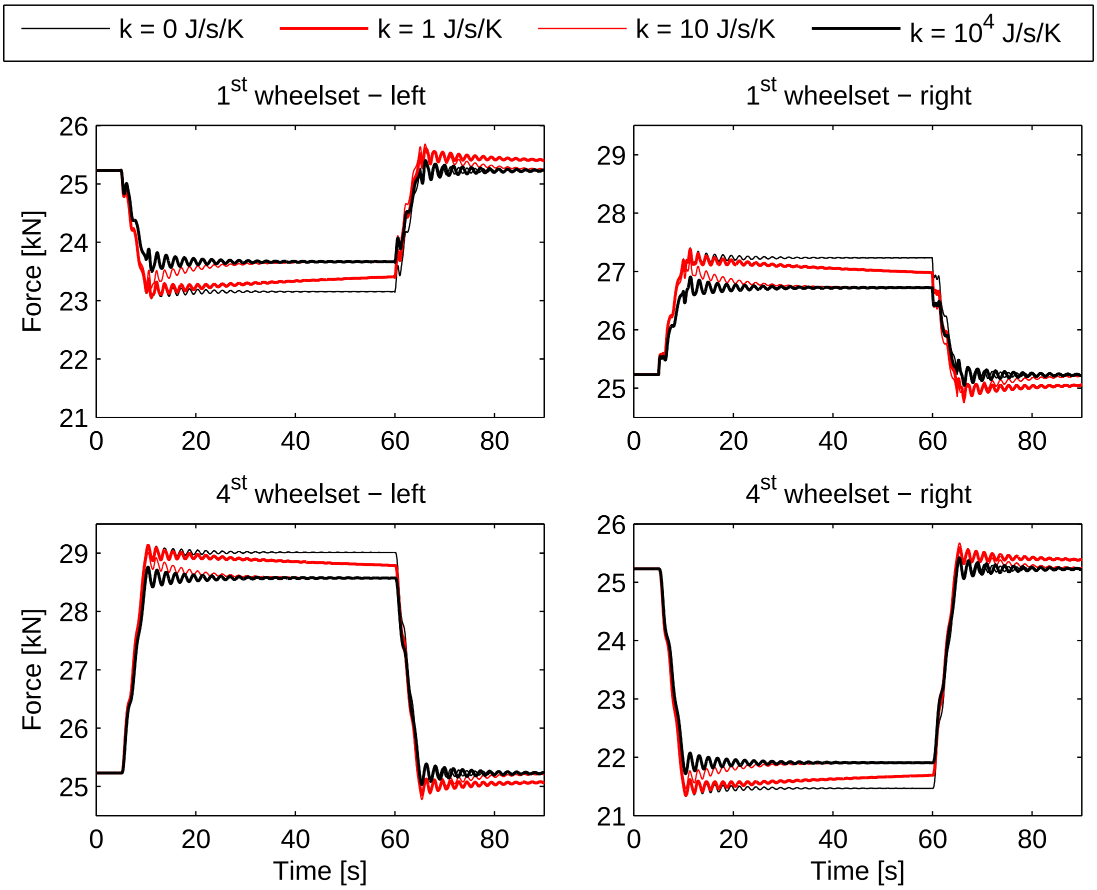


Figure 8. Vertical component of the wheel/rail forces during a $\Delta Q/Q$ test. Only the first and the fourth wheelset curves are shown, the curves of the second and third wheelsets being similar to those of first and fourth wheelsets, respectively.

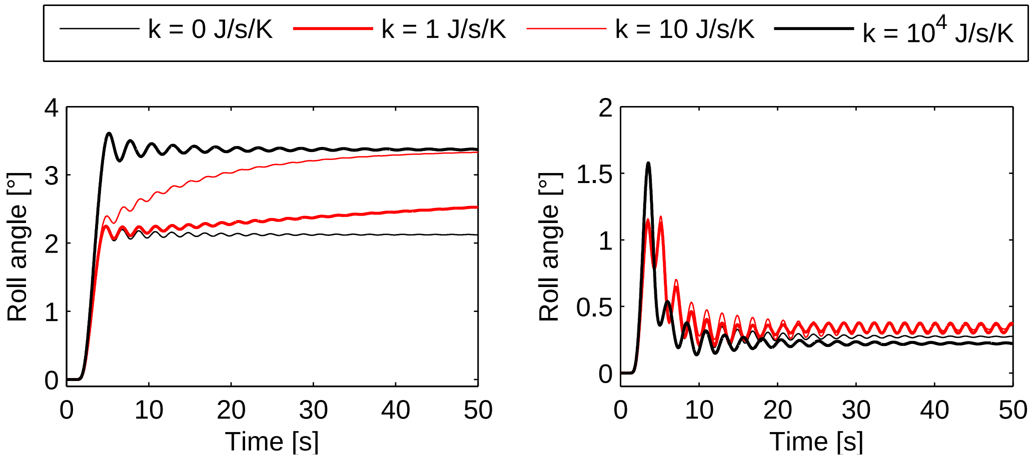


Figure 9. Carbody roll angle of a metro car entering a $100\ m$ *radius curve at* $10\ m/s$*. Left: valves disconnected. Right: valves connected.*

5 CONCLUSION

The present work is related to the modelling of railway pneumatic suspension. It first explained the thermodynamical models for each component of the pneumatic circuit which afford to analyse various suspension topologies. It was then presented how the heat transfer between pneumatic chambers (bellows or tanks) and the atmosphere

$k = 0\ J/s/K$	$k = 1\ J/s/K$	$k = 10\ J/s/K$	$k = 10^4\ J/s/K$
$0.025\ kg$	$0.029\ kg$	$0.035\ kg$	$0.035\ kg$

Table 4. Total airmass injected in the pneumatic suspension via the levelling valves during the curve entry for various heat transfer coefficient k values.

was taken into account, assuming a heat flow rate proportional to the temperature difference. Afterwards, those models were used to analyse the bellow-tank subsystem and discrepancies between adiabatic and isotherm cases were highlighted. It was also shown that the dynamic stiffness is larger for the adiabatic case than the isotherm one and that it presents a resonance effect for frequencies about $15\ Hz$ which is more important for small heat transfert coefficients. Finally, a metro car equipped with a fully pneumatic suspension for a $\Delta Q/Q$ test and a curve entry was analysed. For the $DeltaQ/Q$ test, the adiabatic case was the more critical since it corresponds to a larger suspension stiffness. Moreover, it was noticed for the the curve entry analysis that the air consumption evaluation depends on the heat transfer mode.
This paper has presented how the heat transfer can influence the suspension behaviour. However, it would be interesting to estimate values of the heat transfer coefficients on basis of experimental results. For this purpose, temperature measurement during the dynamic stiffness calculation would be interesting. Furthermore, pneumatic models can still be improved, for instance by analysing heat transfer between the pipes and the atmosphere or by refining the valve modelling.

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