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Optimal Vibration-Absorber Design: A Case Study on Railway Trailing Arm Bush

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Abstract

The wheel-rail surface damage in curves can be effectively reduced with a lower Primary Yaw Stiffness (PYS), while a lower PYS would compromise the passenger comfort with conventional trailing arm bushes (a type of railway vehicle primary suspension component). Previous studies showed that a hydraulic-damping-integrated trailing arm bush has the potential to combat the trade-off between surface damage and ride comfort. However, it only considered one specific parameter set (hence a specific PYS), while there are numerous design possibilities unexplored (e.g., considering multiple fluid passageways). Therefore, the maximum achievable PYS reduction by optimal trailing arm bush design is still unknown. To fully unlock the potential, this work applies an optimal vibration-absorber design methodology to design the trailing arm bush which can provide the minimum PYS without compromising ride comfort. This methodology enables a wide range of design possibilities to be systematically explored and the optimal design to be identified. The potential lifetime cost savings due to surface damage reduction caused by the optimal design have also been estimated. Considering a case study for a Mark 4 Coach, the results show that the optimal network-based bush solution can reduce PYS by 97% over the default without compromising the ride comfort. Two novel physical designs have been proposed accordingly to realise the optimal network properties, achieving up to £41 million saving in track and wheelset maintenance costs. Whilst this study has used the trailing arm bush and Mark 4 coach as an example, the vibration-absorber design methodology is directly applicable to other types of railway suspensions and vehicle types.

Keywords: Absorber design, network synthesis, wheel-rail surface damage, trailing arm bush

1. Introduction

The surface damage caused by wheel-rail contact forces during curving makes a significant contribution to the overall cost of maintaining railway system (e.g., track, wheels). Both active and passive control methods have been carried out to reduce the maintenance cost [1-3]. Among these solutions, passive methods are still regarded as an attractive approach and have been widely used in real-life applications due to their design simplicity, high reliability and low cost. In the previous research, designs of passive vehicle primary suspensions to minimise the wheel-rail surface damage have drawn lots of attention. The studies in [4,5] have shown that a reduced longitudinal static stiffness of the vehicle primary suspension, which takes a large portion of the vehicle primary yaw stiffness (PYS), will reduce the surface damage. This is because with a lower PYS, the wheelsets will have a better radial alignment in curves and the energy loss due to the friction would be reduced. However, a reduced PYS could cause the vehicle to react more strongly to track geometry irregularities, leading to higher lateral carbody acceleration, thus worsening passengers' ride comfort and cause instability. Therefore, a small PYS is needed in order to provide sufficient reduction in surface damage, and a large PYS is required to maintain the ride comfort and stability performance.

Trailing arm bushes are widely used on passenger vehicle bogies as a type of primary suspension component, and conventionally incorporating rubber bushes. These rubber bushes can provide stiffness and a limited level of damping, which cannot provide the desired frequency-dependent vehicle PYS. Hence, rubber bushes cannot combat the trade-off between surface damage and ride comfort. Alternatively, in the last 10 years, a hydraulic-damping-integrated bush, the HALL® bush [6], has been used on some railway vehicles, which can provide

desired small low-frequency stiffness and large high-frequency stiffness, therefore efficiently reduce the surface damage while maintaining ride comfort. However, the HALL® bush introduced in [6] only considered one specific parameter setting, hence a specific PYS. The achievable maximum PYS reduction by optimally designing trailing arm bush is still unknown. This is because there are numerous design possibilities existing (e.g., considering multiple fluid passageways), while the traditional absorber (i.e., trailing arm bush) design approach could only consider a limited number of possibilities, by making specific changes to the existing designs based on ‘trial and error’. Therefore, to fully unlock the potential of trailing arm bush, this work applies an optimal vibration-absorber design methodology [7, 8], which could enable the optimal bush designs to be constructed. A case study for optimally designing trailing arm bush for a Mark 4 Coach, with the aim to reduce surface damage without compromising ride comfort and stability performance, is carried out in this work. A high-level business case is also presented, where the potential lifetime cost savings due to surface damage reduction caused by the enhanced bush designs have also been estimated. This case study was carried out under a project funded by Rail Safety and Standards Board (RSSB) [9].

2. Optimal Absorber Design Methodology

To achieve the optimal vibration absorber design, there are two questions which need to be answered: 1) what the optimal properties are and 2) how to realise such properties. A design methodology [7, 8], which constructs the optimal absorber designs with tailored stiffness, damping and, where needed, inertance [10] properties, provides the answers to these two questions. This methodology can be summarised as a four-step framework:

Step 1: Optimal network identification. The totality of candidate network-represented absorber possibilities with a predetermined complexity (number and types of stiffness, damping and inertance [10]) will be enumerated via the network synthesis theory [7]. Via optimisation against the objective performance and constraints, the optimal network will be identified.

Step 2: Network topology realisation. The physical design which can realise the optimal network topology from Step 1 will be developed. Such realisation will employ the physical components from multiple domains including mechanical, hydraulic, pneumatic and electrical (such as hydraulic orifice and gas chamber).

Step 3: Network property realisation. A Multiphysics numerical model which can adequately characterise the behaviour of the physical design established at Step 2 will be developed. Using this model, the parameters of the physical design (such as the diameter of the orifice) will be identified to realise the desirable optimal network properties obtained from Step 1.

Step 4: Performance evaluation. The physical design developed in Step 3 will be assessed to examine whether all performance measures are still satisfactory. This will be achieved by implementing the numerical model of the physical design established in Step 3 into the hosting structure model.

3. A Case Study on Railway Trailing Arm Bush Design

3.1 Objective functions and constraints

This case study uses a four-axle two-bogie vehicle, the Mark 4 Coach, which is a typical vehicle used in intercity operations on the GB mainline and has the operational speed of up to 125 mph (55.5 m/s). A representative vehicle model ‘BogiePassenger_45t_40yaw.veh’ has been selected from the Vehicle Track Interaction Strategic Model (VTISM) library and be simulated using a commercial multi-body dynamics software, VAMPIRE®. A detailed plan view for the bogie has been shown in Figure 2, where only the primary suspension components act in the horizontal plane are shown (longitudinal X, lateral Y and yaw W). The primary suspension for Mark 4 vehicle consists of a trailing arm bush (highlighted in the dashed box in Figure 2, where its longitudinal contribution is in the blue box marked ① and its lateral contribution is in the red box marked ②), a primary vertical spring and a primary vertical damper (the vertical damper is not included in the figure). A list of the vehicle suspension components with their corresponding parameter symbols are provided in Table 1.

The total PYS of the Mark 4 Coach can be written as

$$PYS = 2a(K_x + K_{sx}) \quad (1)$$

where a ($=1$ m) represents the semi-wheelbase for each bogie. For the selected Mark 4 Coach, the total PYS is

40 MNm/rad, where K_x is 19.43 MN/m and K_{sx} is 0.57 MN/m. It can be seen that the main contribution to the PYS comes from the longitudinal static stiffness of the trailing arm bush K_x . Therefore, for the following optimisation, the longitudinal component of the trailing arm bush will be optimised and K_x will be considered as the objective function to be minimised. The lateral and vertical components will be kept as the default. The damping in the hydraulic bush, HALL® bush, is achieved by the fluid being compressed through the fluid passageway. Based on an initial calculation considering the potential fluid properties and component dimensions [11], the effective damping could reach to 7000 kNs/m. Therefore, for the optimal design, the parameter range for C_{dx} is up to 7000 kNs/m, and the dynamic stiffness K_{dx} is capped at 50 MN/m, matching the default bush.

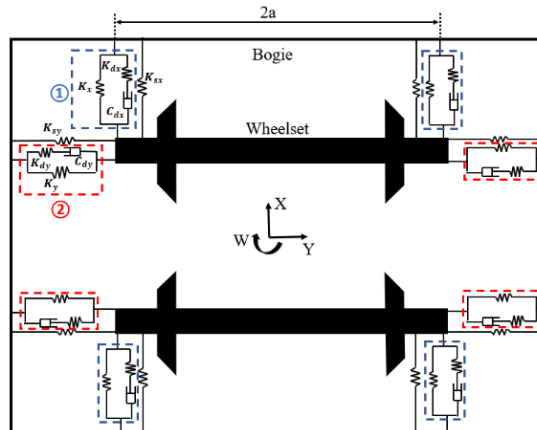


Figure 2: A schematic of one bogie for the ‘BogiePassenger_45t_40yaw.veh’ model in the horizontal plane (longitudinal X, lateral Y and yaw W).

Symbol	Suspension component
K_x	Longitudinal static stiffness from trailing arm bush
K_{dx}	Longitudinal dynamic stiffness from trailing arm bush
C_{dx}	Longitudinal damping from trailing arm bush
K_{sx}	Longitudinal shear stiffness from vertical spring
K_y	Lateral static stiffness from trailing arm bush
K_{dy}	Lateral dynamic stiffness from trailing arm bush
C_{dy}	Lateral damping from trailing arm bush
K_{sy}	Lateral shear stiffness from vertical spring

Table 1: Vehicle primary suspension components and symbols

When the K_x is minimised during the optimisation, it is necessary to make sure that the passenger comfort and stability performance are not compromised. To this end, the Root Mean Square (RMS) of the carbody lateral acceleration representing the level of ride comfort is assessed and selected as the optimisation constraint. A straight track, ‘track200.dat’ from VAMPIRE® library is selected for the simulation, which represents a typical GB track with the line speed of 200 kph (55.6 m/s). Combined operation conditions of three different speeds (33.3, 44.4, 55.6 m/s) and three different wheel-rail contact conicities (0.1, 0.3, 0.5), reflecting the wheel worn conditions [12], forming a total of nine cases, are considered to form a thorough analysis on vehicle performance assessment. Compared with the default rubber bush, the Mark 4 Coach with HALL® bush retrofitted provides slightly higher RMS acceleration in some cases, e.g., up to 5% at 55.6 m/s (see Section 3.4 for detailed comparison). Therefore, the constraints for the RMS acceleration are set with 5% relaxation over the default at the highest speed (55.6 m/s), and 10% relaxation for the lower speeds to allow wider parameter space. Note that the RMS carbody lateral acceleration at high speed and high conicity will inherently consider the vehicle stability performance, therefore, it is not directly used in the optimisation. However, a separate stability performance assessment will be carried out during the performance verification in Section 3.4. For the

optimisation, co-simulation between MATLAB® Optimisation Toolbox (Particle Swarm Optimisation is used) and VAMPIRE® is adopted to find the optimal solutions with the smallest K_x whilst maintaining comfort performance.

3.2 Step 1: Optimal network-represented absorber properties

Applying the network synthesis technique [7], numerous bush design possibilities in network representation are explored. The optimal network design is then identified, with the element parameter values summarised in Table 2 (together with the corresponding PYS) and the network layouts in longitudinal direction in Figure 3(a). It can be found that the optimal solution is with the same layout as the default but introducing a large damping could reduce K_x to zero. This solution (denoted as S1X) achieves a 97% PYS reduction compared with the default rubber bush, and 87% PYS reduction compared with the HALL® bush. Note that since K_x has been reduced to zero, the total PYS only comes from the longitudinal stiffness of the primary vertical spring acting in shear, K_{sx} .

Configuration	K_x (MN/m)	K_{dx} (MN/m)	C_{dx} (kNs/m)	Total PYS (MNm/rad)	PYS reduction (%)
Default Bush	19.43	38.86	19.43	40.00	-
HALL® Bush	3.90	25.00	6563.00	8.94	77.65
S1X	0.00	16.42	6686.00	1.14	97.15

Table 2: Parameter values for the default rubber bush, HALL® Bush and the optimised hydraulic bush, S1X, in the longitudinal direction

3.3. Steps 2&3: Network topology and property realisation

Following the Steps 2&3 of the design methodology, novel physical designs to realise the desired network topology and property of S1X have then been constructed. Specifically, a physical design with 1.6 MN/m longitudinal static stiffness (denoted as S1X_1.6) has been developed. For this design, the static stiffness K_x is realised by pure rubber, the dynamic stiffness K_{dx} and damping C_{dx} (highlighted in the dashed box in Figure 3 (a)) are realised by the volumetric stiffness of the fluid chambers, and the fluid being compressed through a passageway connecting the chambers respectively (see Figure 3 (b) for network topology in hydraulic domain and Figure 3 (c) for the schematic plot). The fluid passageway has been modelled in Simscape, where the length and diameter are optimised to realise the desirable damping of S1X. Meanwhile, the non-linear damping characteristics and the parasitic inertance effect due to the passageway, illustrated in Figure 3 (d), are also considered and the effect will be investigated during performance verification. The rubber hardness and dimensions are tuned from FEM analysis.

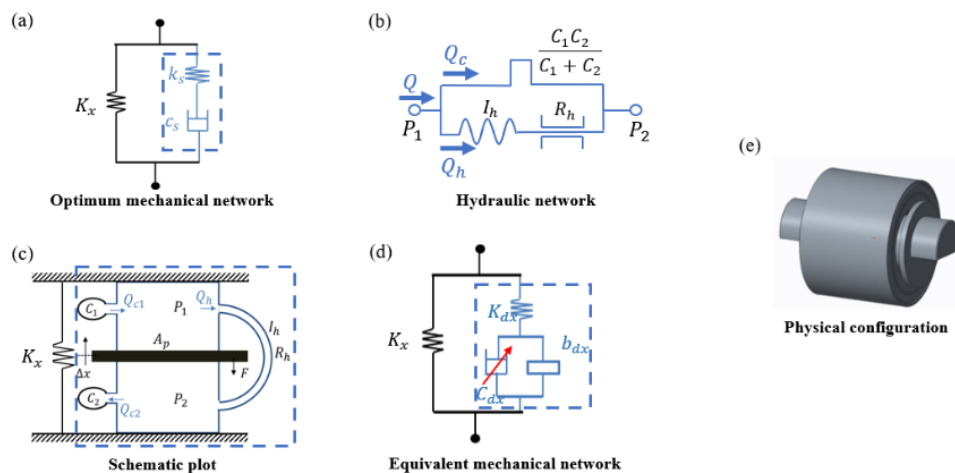


Figure 3: (a) Optimal bush network layouts in the longitudinal direction, (b) The network topology realisation in hydraulic domain, (c) The schematic plot for the S1X_1.6, (d) The equivalent network considering non-linearities and parasitic effects, (e) The outside look of the physical design S1X_1.6.

In addition, a near-zero K_x (0.22 MN/m) can also be achieved by another physical design using hydraulic and mechanic components, denoted as S1X_near0. The longitudinal static stiffness is also realised by pure rubber, while the series stiffness-damping is achieved by a block with an orifice and two connected coil spring at each end. The S1X_1.6 design has the identical size as the HALL® Bush and can be directly fitted into Mark 4 Coach, while the S1X_near0 has the same diameter but requires a 20% length increase (180 mm for S1X_near0, 150 mm for HALL® Bush). Based on the Mark 4 bogie design, this 30 mm length increase has the potential to be accommodated with modification to the bush mounting points. Due to confidentiality, detailed designs have been removed here (please contact University of Bristol for more information if interested) and the outside look of the S1X_1.6 is illustrated in Figure 3(e).

3.4. Step 4: Performance evaluation

The performance of the proposed physical designs needs to be assessed in VAMPIRE® for verification, with the non-linear characteristics and parasitic inertance effect implemented (shown in Figure 3 (d)). Note that this work only considered the non-linearity of the hydraulic damping (C_{sx}), while the stiffness of the rubber and the fluid chamber are considered as linear for simplicity at this stage. The performance of S1X_1.6 with the non-linear damping has been assessed as an example here. It can be seen from Figure 4 (a) that the ride comfort performance of S1X_1.6 have not been compromised even in the highest speed (55.6 m/s). Besides, a stability check using the ‘stability.dat’ file from VAMPIRE® library has been carried out, to assess the vehicle critical speed (when a limit cycle motion will be observed). It can be seen from Figure 4 (b) that across all conicity cases, the S1X_1.6 design can provide similar critical speeds with the default rubber bush and the HALL® Bush.

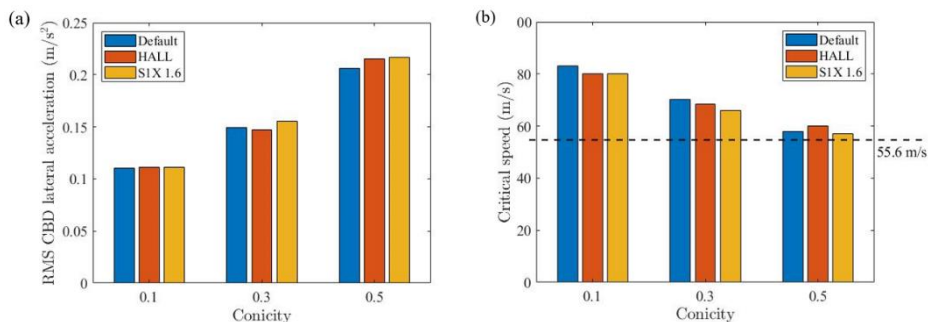


Figure 4: The comparison between the default, HALL® Bush and S1X_1.6, in terms of (a) the RMS carbody lateral acceleration at 55.6 m/s, (b) the critical speed.

4. A High-Level Business Case

To provide a more direct understanding of the benefits the enhanced bushing designs can bring, a high-level bushiness case is given here. The potential cost saving of the new designs over the default bush and the HALL® Bush is estimated, as summarised in Table 3. The Network Rail Variable Usages Charge (VUC) calculator and the VTISM – Wheelset Maintenance Model (WMM) calculator are applied, to obtain the track and wheelset maintenance cost respectively [13,14]. Fleet size of 302 vehicles is used (number of Mark 4 Coaches on the East Coast Main Line in 2019) and 20 years lifetime with a discounting factor of 3.5% applied as per UK Transport Appraisal Guidance [15]. As shown in Table 3, compared with the default bush, the S1X_1.6 can provide 39% VUC reduction and 27% WMM reduction, while the S1X_near0 can provide 41% and 35% reduction respectively. Further investigation on the lifetime cost saving of the Mark 4 fleet has also been carried out, based on the data given in [15] and the assumption that other costs (production, fitment, maintenance and lifespan) are the same. As shown in Table 3, compared with the default, S1X_1.6 can provide the total lifetime saving (due to wheel-rail maintenance) for Mark 4 fleet of approximately £37m. Further cost saving can be achieved by S1X_near0, which is up to £41m, at the expense of a slightly increase in the bush length.

Bush type	PYS (%)	VUC (p/vm)	WMM (p/vm)	Annual saving for single vehicle (£)	Lifetime saving for the fleet (£)
Default Bush	40.00 (-)	12.09 (-)	2.60 (-)	-	-
HALL® Bush	8.94 (77.65%)	8.82 (27.05%)	2.30 (11.54%)	6,024.55	24,669,001.82
S1X_1.6	4.34 (89.15%)	7.43 (38.54%)	1.90 (26.92%)	9,045.26	37,038,041.91
S1X_near0	1.58 (96.05%)	7.10 (41.27%)	1.70 (34.62%)	9,939.66	40,700,382.71

Table 3: Estimated potential savings in VUC and WMM from default bush compared to other configurations (The percentage shown in the bracket represents the reduction compared with the default bush.).

5. Conclusion

This work is the first time of applying the optimal vibration-absorber design methodology to systematically design railway vehicle suspension. With this methodology, a systematic investigation on the Mark 4 trailing arm bush has been carried out to fully explore the potential benefits it can bring in surface damage reduction. A 97% PYS reduction can be achieved with the optimal network solution and two novel designs have been proposed to realise the desired dynamic properties. A high-level business case is also introduced to estimate the lifetime saving in wheel and rail maintenance cost. The results show that the two novel designs can provide approximately £37m and £41m savings respectively. Whilst this study used the Mark 4 Coach as an example, the design methodology is directly applicable to other types of railway vehicles, where the potential savings/performance improvement could be more significant.

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