DETERMINATION OF DYNAMIC CHARACTERISTICS OF DRAFT GEARS OF HEAVY HAUL TRAIN USING COLLISION SIMULATIONS

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SUMMARY

A heavy haul train model has been generated using the multi-body dynamics approaches in order to examine the dynamic performance of coupler draft gears through the simulations of a train model hitting a fixed block at a very low speed. The train model consists of a locomotive and several wagons, which are connected with auto-couplers with friction wedge type draft gear packages. Three types of coil spring stiffness characteristics in draft gears – linear, soften and harden nonlinear, are selected to investigate the dynamic performance of draft gears. The simulation results show that the crash speed must be less than 5km/h so that the draft gears work properly. Coil springs with soften nonlinear characteristic showed a significant reduction in the second impact forces on locomotive head.

INTRODUCTION

It is well known in heavy haul train transportation that the draft gear in a coupler plays an important role. The draft gears are designed not only to have the buffer capacity for smooth operation, but also to reduce the transient impact in-train forces through the couplers. Impact in-train forces are generally associated with run-in and run-out occurrences due to changes in locomotive power and braking settings, changes in grade and undulations. It is the draft gear that is used to absorb and dissipate the energy generated during coupler impacts between rail wagons without damaging the vehicle structure and loading.

As the heavy haul transportation industries around the world continuously improve their productivity and efficiency, the train axle load as well as train length are constantly increased. Many problems concerning coupler and draft gear systems arise during the development of heavy haul transportation such as decoupling, breakage and fatigue damage of either the couplers or draft gears and the fracture of knuckle, etc. Derailments may occur due to large coupler forces with the increase in axle load and running speed.

BHP-Billiton Iron Ore holds the world’s highest axle loads and longest train records. Their maximum wagon count leads by 640 wagons, and in terms of axle loads, the average increased by 16% to 37 tonnes per axle in the past 15 years. Data from coupler failure have indicated that there may exist a correlation between higher failure rate and increased train length at high axle loads [1].

It is obvious that the operation of longer and heavier heavy haul trains will generate larger in-train forces and more complicated force patterns. Practical experience indicates that the development of fatigue failure of coupling systems within modern heavy haul trains may differ from the conventional understanding. The current techniques in dynamics modeling of friction draft gears have been reviewed in order to improve existing or develop new models to achieve more accurate force amplitude and pattern predictions [2].

During the last few decades, the longitudinal dynamic performances of heavy haul trains, including couplers and draft gears, have been investigated and many methods have been put forward to improve their performance. A series of studies have been conducted each focusing on longitudinal train dynamics, wagon connection modeling, train management and comfort, and driving practices [3] ~ [4]. A train dynamic model was developed to study the dynamic performance of heavy haul locomotives. This model takes into account the use of different coupler and buffer systems under conditions of severe longitudinal coupler compressive forces [5] ~ [6]. From this study, it was found that large capacity elastic clay (or cement) buffers cannot satisfy the requirement of heavy haul locomotives.

The potential derailments occurred in recent years due to coupler jack-knifing, and the coupler jack-knifing behaviour and its implications for locomotive safety have been studied. From the
derailment experience, coupler angle self-lock behaviour was introduced and simulated. An approach to determine coupler angles in the jack-knifed position was derived [7]. Finite element methods were used to investigate dynamic behaviour of draft gear [8]. The longitudinal force that the draft gear absorbs usually leads the failure of its components, especially, the load bearing draft pads.

Auto-couplers with friction wedge type draft gear packages, as shown in Figure 1, are the most common wagon connection in the Australian and North American heavy haul train systems. Their modeling and simulation may present to be the most challenges due to the high nonlinearities of coupler slack, draft gear spring characteristic (polymer or coil springs), and stick – slip friction provided by a wedge system. Due to these complexities, the common auto-coupler-friction type draft gear wagon connection will be examined in this paper.

![Auto-coupler](image1.png)

**Figure 1 Auto-coupler**

The friction-type draft gears are the most widely used draft gears. The ever developing heavy haul transport environment requires further or new understanding of friction draft gear behaviour and its implications for train dynamics, as well as fatigue damage of rolling stock. However, modeling of friction draft gears is a highly nonlinear question. Especially the poor predictability, repeatability and the discontinuity of friction make this task more challenging. In this paper, a heavy haul train dynamic model is developed to examine the dynamic performance of draft gears. This takes into account the use of three types of draft gear systems under conditions of severe longitudinal coupler compressive forces – simulating the heavy haul train model hitting a fixed block. The train model consists of a locomotive and several wagons. The wagon connection modeling for auto-couplers with friction wedge type draft gear packages described in [3] is applied.

**NOTATION**

\[ F_{ci} \] – the \( i \)-th coupler force;
\[ N_{1} \] – the normal force on slope of wedge;
\[ \mu \] – the friction coefficient on slope surface;
\[ \phi \] – the wedge angle;
\[ F_{si} \] – the spring resilient force on the \( i \)-th coupler force;
\[ k(x) \] and \[ c(x) \] are the stiffness and damping coefficients of spring

\[ x_{i} \] and \[ x_{\dot{i}} \] are the displacement and velocity of \( j \)-th vehicle.

**DRAFT GEAR MODELING**

Figure 1 illustrates three states of a conventional auto-coupler and draft gear package [3]. The draft gear package can be considered as a single wedge spring system as shown in Figure 1. The free body diagram for increasing load (i.e., compressing) is shown in Figure 2.

![Free Body Diagram of Draft Gear](image2.png)

**Figure 2 Free Body Diagram of Draft Gear [3]**

The rollers on one top side of the compression rod represent that the multiple wedges are arranged symmetrically around the outside of the rod in the actual unit. Depending on the direction of motion, wedge angles, and surface conditions, the different equilibrium states are possible, as shown in Figure 3.
Figure 3 Force Equilibrium States

From Figure 3, there are two cases. Case 1 represents the sliding action in the direction for compression. Case 2 applies if a pre-jammed state exists. In this case, the rod is held in by the jamming action of the wedge. This case is beyond consideration of this paper, so that, only case 1 is only taken into account.

The $i^{th}$ coupler force for loading situation in Case 1:

$$F_{ci} = N_1 (\sin \theta + \mu_1 \cos \theta) \quad (1)$$

Further, the equations relating the wedge forces to the coupler force and the coil spring force are developed, again assuming saturated friction states and direction shown in Case 1, as following:

$$F_{ci} = F_{si}(\sin \theta + \mu_1 \cos \theta) / (\mu_1 - \mu_2) \cos \theta + (1 + \mu_1 \mu_2) \sin \theta \quad (2)$$

In Equation (2), $F_{si}$ is the coil spring force in the $i^{th}$ coupler.

If it is assumed that $\mu_1 = \mu_2 = \mu$, and that both surfaces are saturated, then the equation can be reduced to:

$$F_{ci} = F_{si}(\cot \theta + 1) / [1 + \mu^2] \quad (3)$$

The other assumption is made that there is no impending motion on the sloping surface due to the seating of the rod and wedge, the value assumed for $\mu_1$ is zero, thereby reducing Equation (3) to (including unloading situation):

$$F_{ci} = F_{si}(\tan \theta + \mu_2) \quad (4)$$

$$F_{si} = k(x) \cdot (x_i - x_{i-1}) + c(x) \cdot (\dot{x}_i - \dot{x}_{i-1}) \quad (i = 1, 2, 3 ...) \quad (5)$$

In Equation (4), $\mp$ represents that “+” is the loading process while “−” is the unloading process.

The three stiffness characteristics of coil spring $k(x)$ – linearity, soften and harden non-linearity are considered. The corresponding coupler forces calculated through Equation (4) are shown in Figure 4.

Figure 4 Three types of coupler forces

In Figure 4 (a), (b) and (c), the working region of all couplers is set up from -125 mm to 150 mm, and at the limit of working region the coupler forces are set up as -2000 kN and 2000 kN respectively. Over the limit of working region, a locked stiffness value in the order of 80 MN/m is selected. In Figure 4 (a), (b) and (c), the solid line is the loading process while the dash line is the unloading process.

HEAVY HAUL TRAIN MODELING

The heavy haul train model including one locomotive and several wagons are generated...
using Gensys software package, as shown in Figure 5.

![Figure 5 Heavy haul train model (4 wagons) using Gensys](image)

### 1. Loco Model

In the loco model, the loco body and the bogie frame are modelled as a single mass with 6 degrees of freedom respectively. The connections (the secondary suspensions) between the loco body and one bogie frame include:

- Four vertical coil spring elements,
- One spring element for the anti-roll bar, and one spring element and one damper with series flexibility for the traction rod in the direction specified by the coupling's attachment points,
- One lateral and two vertical bumpstops, two vertical viscous dampers, and two lateral viscous dampers and two yaw dampers with series flexibility in the direction specified by the coupling's attachment points respectively.

The wheelset is also modelled as a single mass with 5 degrees of freedom, not including the pitch rotation. The connections (the primary suspensions) between one bogie frame and three wheelsets include:

- 18 spring and damping elements in three X, Y and Z directions,
- Three lateral and six vertical bumpstops, and six vertical viscous dampers in the direction specified by the coupling's attachment points respectively.

### 2. Wagon Model

A typical wagon with two three-piece bogies is considered for modeling. In this model, it includes 11 masses – one wagon car body, 2 bolsters, 4 sideframes and 4 wheelsets, which are modelled as rigid bodies. The connections modelled are:

- The centre bowl is modelled as four-point contact between wagon car body and bolster – four vertical (Z) stiffness elements associated with two-dimensional (X-Y) friction block at each stiffness element, two longitudinal (X) stiffness elements associated with one-dimensional (Y) friction block at each stiffness element, and one lateral (Y) stiffness element and one three-dimensional damper including rotations due to consideration of material damping.
- The side bearer between wagon car body and bolster is modelled as one vertical stiffness element associated with two-dimensional (X-Y) horizontal friction block.
- The secondary suspension includes one coil-spring and one damper between bolster and sideframe, plus one longitudinal and one lateral bumpstops.
- The primary suspension includes two vertical contact stiffness elements laterally separated 120 mm on each axle box in order to give roll stiffness to side-frame and yaw friction moment, associated with two-dimensional (X-Y) horizontal friction block at each element, plus one longitudinal and one lateral bumpstops.

### SIMULATION RESULTS

The train model with five wagons is chosen for the simulations. The simulations have been conducted to let the locomotive front head collide a fixed rigid block on the track at a constant train speed. The collisions happen at the moment of 0.1 seconds. Such simulations only test the compressive characteristic of draft gears. After several trial simulations, it is found that when the crash speed is less than 5 km/h, the draft gear can work properly in its working region. Therefore, the crash speed of 3 km/h is chosen. Figure 6 (a) and (b) show the collision process (demonstrated using the train model with four wagons).
The locomotive head impact forces are shown in Figure 7. In this figure, there are several impact peaks. The first peak force is almost from the collision of locomotive. The second peak force is mainly from the crash of the first wagon behind the locomotive through the first coupler. It can be seen that the stiffness characteristic of draft gear plays a significant role. The linear stiffness of coil spring causes the largest impact force while the soften nonlinear stiffness generates the smallest impact force. The difference of these two impact forces is about 500 kN.
Figure 8 First coupler forces

Figure 8 (a) and (b) show the histories of first coupler forces with the time and with the relative longitudinal displacement (locomotive and the first wagon) respectively for these three types of coil springs. From Figure 8 (b), the coupler forces follow the patterns described in Figure 4 (a), (b) and (c). It is expected that the draft gear with soften nonlinear coil spring is the first to close to the work region limit while the ones with linear and harden nonlinear coil spring are the second and third. It can be seen that the maximum coupler force is generated by the draft gear with harden nonlinear coil spring. However, from Figure 8 (b), harden nonlinear coil spring only performs at approximately half way to the limit of the working region. This means that the draft gear with harden nonlinear coil spring has the larger potential to withstand the collision at a higher speed.

Figure 9 Coupler forces

The five coupler forces on the couplers with linear coil spring are shown in Figure 9. Due to the collision, the maximum impact forces on the first and the last (the fifth) couplers changes significantly, with the values of 1.42 MN and 0.73 MN respectively. The ones on the second, third and fourth couplers do not change much, with about 1.0 MN.

CONCLUSION

The modern technology of calculation and simulation using professional multi-body dynamics software packages can be used to investigate the dynamic performance of connection devices (coupler and draft gear) in heavy haul train as a substitute of traditional method – drop-hammer test or actual train collision experiment. The heavy haul train model developed in this paper using Gensys shows to be capable in its abilities. The collision simulations should be conducted at a low crash speed. However, the model needs to be improved to meet the requirements of high accuracy and efficiency.

Three types of draft dear coil springs are selected for the simulations. In the impact force on the locomotive head, the second peak is identified from the wagon impact through the first coupler. The linear stiffness of coil spring causes the largest impact force while the soften nonlinear stiffness generates the smallest impact force.

When the impact forces on the first coupler are examined, the maximum coupler force is generated by the draft gear with harden nonlinear coil spring. However, its deflection is only the half of the working region limit. The draft gear with harden nonlinear coil spring has the potential to withstand the collision at a higher speed.
REFERENCES


