Colantuono, Thomas, Grossoni, Ilaria, Allen, Paul, Molyneux-Berry, Paul and Borczyk, Philip

The optimal strength of shear pins: Requirements in point run-throughs

Optimal Shear Pin Strength: Requirements in Point Run-Throughs

Journal:	*Part F: Journal of Rail and Rapid Transit*
Manuscript ID	JRRT-15-0185
Manuscript Type:	Article
Date Submitted by the Author:	21-Oct-2015
Complete List of Authors:	Colantuono, Thomas; University of Huddersfield-Institute of Railway Research, School of Computing & Engineering Grossoni, Ilaria; University of Huddersfield, School of Computing and Engineering Allen, Paul; University of Huddersfield, IRR Molyneux-Berry, Paul; University of Huddersfield, Computing and Technology Borczyk, Philip; London Underground, Track Engineering
Keywords:	switch and crossing, shear pin, run-through, derailment, climb-out, switch blade
Abstract:	Mathematical modelling work has been carried out to investigate the potential for a revision to the strength of shear pins used in London Underground Switch and Crossing drives to protect against run-through derailments. The revision has been requested based on a need to increase the life of the pins which are prone to fail due to service fatigue loadings. The study has been accomplished in two stages: analytical modelling to establish the stiffness of the switch blade, followed by vehicle dynamics modelling to analyse wheel climb-out events and the resultant loadings on the shear pin. A range of shear pin strengths and vehicle-track parameters have been considered. A post-processing routine has been developed to enable a thorough study of the relationship between vehicle-track parameters, shear pin strength, failure rates and vehicle derailment risk. A good agreement between the modelling results and the current shear pin's ability to protect against derailments has been found. Ultimately the potential to increase the shear pin strength has been analysed in order to protect against fatigue failure in service.

Journal of Rail and Rapid Transit 0(0)

Optimal Shear Pin Strength: Requirements in Point Run-Throughs

Thomas Colantuono[1], Ilaria Grossoni[1], Paul Allen[1*], Paul Molyneux–Berry[1*] and Philip Borczyk[2]

[1]Institute of Railway Research, University of Huddersfield, Huddersfield , UK
[2]London Underground Ltd, Track Engineering, London, UK

[*]**Corresponding authors:**
Institute of Railway Research, University of Huddersfield, Queensgate, Huddersfield HD1 3DH, UK. email: p.d.allen@hud.ac.uk
Institute of Railway Research, University of Huddersfield, Queensgate, Huddersfield HD1 3DH, UK. email: p.b.molyneux-berry@hud.ac.uk

Abstract

Mathematical modelling work has been carried out to investigate the potential for a revision to the strength of shear pins used in London Underground Switch and Crossing drives to protect against run-through derailments. The revision has been requested based on a need to increase the life of the pins which are prone to fail due to service fatigue loadings. The study has been accomplished in two stages: analytical modelling to establish the stiffness of the switch blade, followed by vehicle dynamics modelling to analyse wheel climb-out events and the resultant loadings on the shear pin. A range of shear pin strengths and vehicle-track parameters have been considered. A post-processing routine has been developed to enable a thorough study of the relationship between vehicle-track parameters, shear pin strength, failure rates and vehicle derailment risk. A good agreement between the modelling results and the current shear pin's ability to protect against derailments has been found. Ultimately the potential to increase the shear pin strength has been analysed in order to protect against fatigue failure in service.

Keywords

Switch and crossing, shear pin, run-through, derailment, climb-out, switch blade

1 Introduction

A shear pin is a mechanical safety device incorporated into the design of Point Operating Equipment (POE) which provides the actuation, locking and detection throughout the length of the switch rail [1]. Run-through is the event during a 'trailing' move (the switch is approached from one of the two converging routes) in which the vehicle drives through a trailing switch against the direction set by the POE [1]. Theoretically, in such a case, the shear pin fails and the switch blades will move to allow the passage of the vehicle without causing a derailment.

London Underground (LU) is one of the few administrations that require shear pins in this situation, owing to the severe consequences of a derailment in the confined tunnels. The nature of LU operation is such that run-through events are extremely unlikely, especially for passenger trains.

There is presently an incompatibility between the specified 35kN shear strength of the pin for this requirement, and the desired fatigue life of the pin in normal operation. Fatigue failure leads to operational and reliability problems which can cause knock-on safety risks with overcrowding.

For this reason it is necessary to investigate the shear strength requirements of the pin to determine whether the current designs are appropriate for LU operation.

The following paper presents the results of modelling work which has been carried out in two stages. The first stage involved analytical modelling to establish the lateral structural flexibility of the switch blade. The second stage of work has utilised the resultant flexibility information, expressed as a function of distance along the switch within the VAMPIRE vehicle dynamics simulation software. This study has considered only the case whereby the wheelset fit within the track gauge dictates that the wheel flange is trapped between the converging closed switch rail and the adjacent stock rail. 1200 wheel-rail interaction simulations have been carried out to analyse wheel climb-out events and the resultant loadings on the shear pin. The analysis work has considered a range of shear pin strengths and vehicle-track parameters. Using post-processing routines written within Matlab, the relationship between these parameters, shear pin strength, failure rates and vehicle derailment risk have been studied in detail.

2 Literature Review

In an attempt to establish the origin of the 35kN shear pin breakage limit, a literature review exercise was carried out, including direct liaison with the LU and Network Rail (NR) Switch & Crossing teams. The LU requirements for points operating equipment [1] include a maximum strength of 35kN in the point operating equipment linkages; this is derived from a 35kN limit used on NR and its predecessors [2]. It is believed that the maximum force limit was intended to satisfy a requirement included in Railway Group Standards, first appearing in 2000 in [3] and later in [4] that specified:

> '*Points and their associated operating equipment shall be designed so that a trailing movement passing through a point end which is in other than the correct position does not result in a derailment.*'

The 35kN limit is provided within [1] and [2] without the provision of reference information or background calculations. While a literature survey has identified several papers related to wheel/switch forces in normal operation [5], [6] and the strength of various POE designs when run-through [7],[8], no evidence has been found to support the 35kN figure.

A new design of POE was introduced by NR in the early 2000s which did not comply with the 35kN requirement. It was granted a temporary non-compliance [9] and ultimately the requirement quoted above was removed from Railway Group Standards [10].

The current relevant EN standard [11] makes a distinction between 3 categories of POE:

- 'Trailable' devices which permit trailing as a non-standard operation. In this case, parts of the switch may be slightly damaged. The switches and crossings will only be released for further operation after full inspection of switch and actuator.
- 'Trailable' devices which permit trailing as standard operation.
- 'Non-trailable' devices which do not permit the trailing of the switches and crossings by a vehicle.

It is believed that the third case is the most common form installed in mainland Europe. The relevant case here is the first one listed. A method for acceptance testing of trailable devices is included in the EN standard, and it is recognised that the performance is influenced by speed, axle load and wheel profile. It requires that:

> *In case of a trailable switch for non-standard operation no deformations of the parts of the switch or crossing may appear. Deformations, breakage or wear on the actuator or locking system should be reported in the test report.*

No strength figure is quoted, but it is implied that the actuator and/or locking system of the trailable device should yield to permit the vehicle to pass without damage to the switch blade and without derailment.

No clear justification for the existing 35kN limit has been found. Further, there is no clear requirement in EN standards or current Railway Group Standards for any maximum switch restraining force, or for a run-through switch to prevent derailment.

3 Modelling of Switch Blade Structural Flexibility

3.1 Modelling of the Switch Blade and Linkages

A mechanical model of the switch blade lateral behaviour has been created in order to calculate the effective lateral stiffness of the closed switch blade which the wheel is trying to force open. The switch considered is a 113A BVS switch with drive to a single stretcher bar near the toe (i.e. no supplementary drive). CAD drawings provided by LU are shown in Appendix 1 and were used as the basis for the switch geometry; note that the second stretcher bar has been neglected.

A sketch of the simplified model is presented in Figure 1.

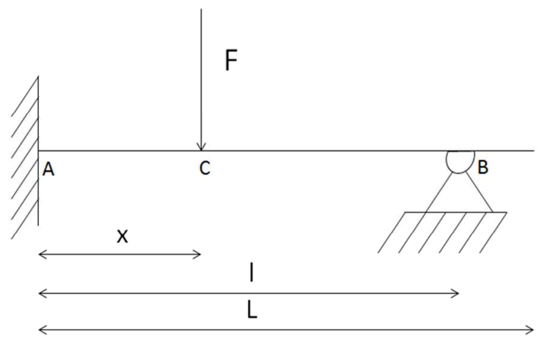


Figure 1. Mechanical model of the switch blade lateral behaviour.

The model assumes the heel of the switch is encastré (Point A), with the blade section area reducing through the planing radius. The pin joint at Point B represents the stretcher bar and location of the shear pin, in proximity to the switch toe.

The sliding friction of the switch blade on its slide baseplate is neglected. In the case examined, no wheels are running on the switch blade being forced to move laterally, so the loads on these friction surfaces are likely to be small in comparison with the other forces during the run-through.

The beam has been modelled as a Timoshenko beam. In order to describe the beam constraints, the penalty method has been considered as being valid, so that constraint A has been represented with a very stiff vertical and rotational spring and constraint B with a vertical spring. A summary of the main parameters are reported in Table 1 below.

Table 1. Summary of main parameters used in the model.

Variable	Value	Units
Beam total length L	8.90	m
Distance of the constraint l	8.41	m
Number of elements	600	-
Young modulus	2.10e11	N/m^2
Beam shear modulus	8.10e10	N/m^2
Poisson's coefficient	0.34	-
Steel density	7850	kg/m^3
Vertical stiffness in the constraints A and B	10e10	N/m
Rotational stiffness in the constraint A	10e10	N/rad
Section area	variable	m^2
Section principal moment of inertia	variable	m^4

The section properties, including the area and the principal moments of inertia, are variable along the beam length. As a result, these have been interpolated from the 3D switch CAD drawing at 100 mm intervals. A static analysis has been performed and the point of application of the force has been varied along the beam length. The equivalent stiffness $K_{eq}(x)$ has been calculated as:

$$K_{eq}(x) = \frac{F(x)}{d(x)} \tag{1}$$

F(x) is the applied lateral force at location x (Figure 1) and d(x) the lateral displacement due to the applied force in the same location x.

3.2 Results of switch blade and linkage modelling

The results in terms of equivalent stiffness and displacement have been evaluated based upon the switch blade and linkage model; these are shown in Figure 2.

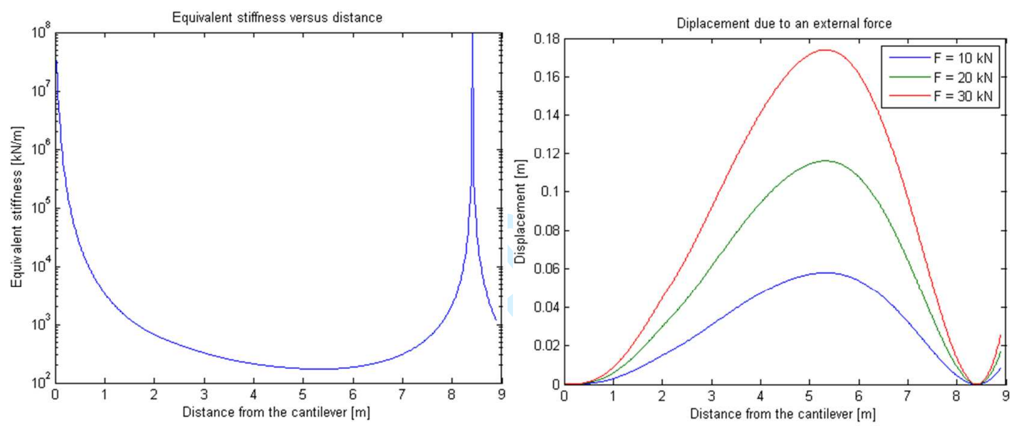


Figure 2. (a) Equivalent stiffness along the beam; (b) displacement along the beam in correspondence of three force levels (10kN, 20kN, 30kN).

Comparing the two plots of Figure 2, it can be seen that there is no displacement where the stiffness is maximum, i.e. in correspondence of constraints A (heel of the switch) and B (stretcher bar). The displacement is also maximum where the stiffness is minimum; that location is not exactly in the middle-span, as in case of a uniform simply supported beam, but nearer to the constraint B. Moreover, the displacement curve in correspondence of constraint A is characterised by a zero value tangent because it is clamped - as expected for the beam theory.

The varying equivalent stiffness presented in Figure 2(a) was then processed for inclusion in a VAMPIRE vehicle dynamics simulation to establish the forces at the point of wheel-rail climb-out. These will be used as an indicator of the shear pin strength requirements.

4 Wheel-rail interaction: Simulation and post-processing

4.1 Overview

Wheel-rail interaction simulation work has been carried out, by means of VAMPIRE modelling techniques, to determine the forces acting on the switch blade in a run-through situation. In this condition, the wheel of interest approaches the incorrectly set switch blade in a trailing move. The flange enters a converging geometry whereby ultimately it becomes trapped between the stock rail and the back of the switch blade. A simple cross-sectional diagram of the run-through model is shown in Figure 3. Forces and system intrinsic characteristics have been taken into account in the model. These are: the horizontal force of the switch rail (F_1) acting on the flange back, the lateral force (F_2) and friction (f_2) between the stock rail and the wheel, the lateral stiffness (k) and damping (c) between the switch rail and the stock rail, the lateral stiffness (k_1) and damping (c_1) between the stock rail and the sleeper. Other features have not been modelled. These are: stock rail and switch rail masses (M_1 and M_2), friction (f_1) between the switch rail and the flange back, friction (f_3) between the switch rail and the sleeper.

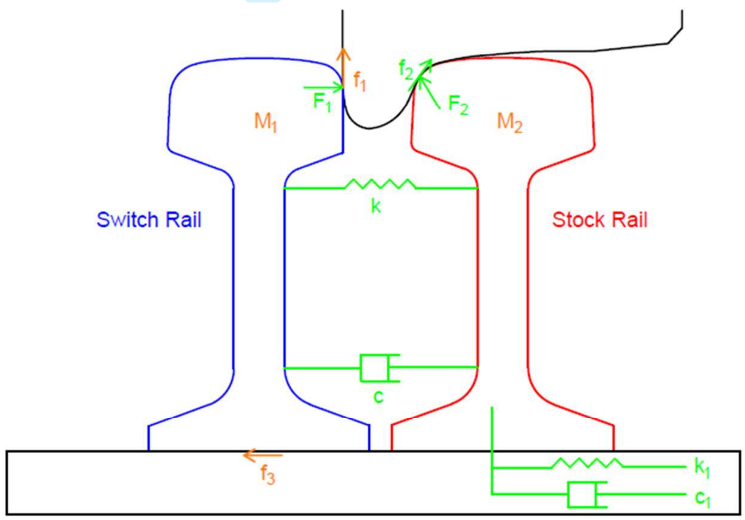


Figure 3. Cross-sectional diagram of run through model.

In order to increase the accuracy of the simulations in these conditions, the work has considered a switch blade with varying lateral stiffness along its length, as described in the previous section.

The VAMPIRE analysis has allowed the location and magnitude of the peak lateral load on the switch blade prior to wheel climb to be determined. From these values, the force in the shear pin has been estimated, and compared to the current shear pin strength requirement of 35kN. This process has made it possible to determine:

- wheel climb events which generate sufficient force to break the shear pin and avoid derailment
- events which generate insufficient lateral force to break the shear pin, and hence derail.

By considering a range of higher strength shear pins, the optimal strength level, offset against derailment risk has been predicted. Details of the VAMPIRE simulations and the resultant post-processing are presented in the following sections.

4.2 Vampire Wheel-Rail Interaction Simulations

The VAMPIRE vehicle dynamics simulation software has been used to determine the levels of lateral force acting on the switch blade as the wheelset passes in a trailing run-through move. The switch blade has been modelled in the simulation using the VAMPIRE checkrail element within the track model. This is a single degree of freedom (lateral direction) element with a linear stiffness characteristic which can be varied with respect to distance along the track. The element allows the structural flexibility described above to be integrated within the model and hence provide a realistic checkrail (switch blade) response to the passage of the wheelset.

The converging geometry seen by the wheelset as it approaches the incorrectly set switch has been modelled based upon the BVS switch drawing within Appendix 1. The geometry has been implemented within VAMPIRE by varying the checkrail clearance with respect to distance, up until the point where it becomes zero. As in the real case, this results in the wheel becoming increasingly constrained by the checkrail until it is effectively ejected out of the gap between the checkrail and the running rail. By setting up the dynamic simulations in this manner, a climb-out of the wheel on to the top of the rail is assured in all simulation cases. The key output of the simulations being the lateral force in the checkrail element and whether this leads to sufficient load at the shear pin location to break the pin in reality. The simulation environment is shown in Figure 4.

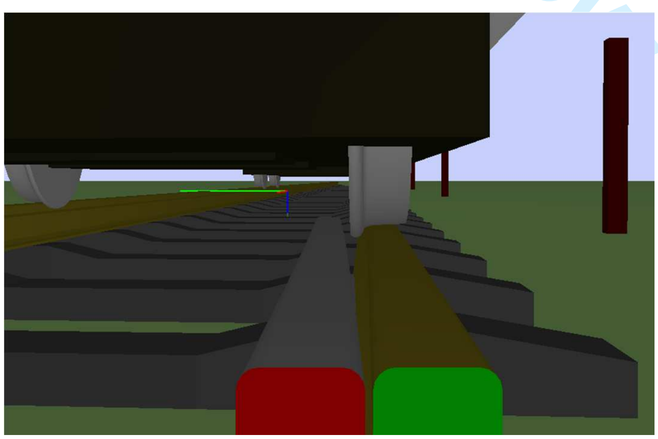


Figure 4. Visualisation of Simulation Environment.

A 1972 Tubestock vehicle, configured in both tare and crush laden conditions has been used for the vehicle model. This type of vehicle operates on the Bakerloo Line which features some of the more sharply curved route sections on London Underground.

The following parameters have been varied to ensure a wide range of operating conditions:

- vehicle load: laden and tare;
- state of wheel profile: 1 new profile and 5 worn profiles;
- track type: 1 straight and 4 curved (turnout radius: 500m, 200m, 100m, 50m);
- friction coefficient: 0.1, 0.2, 0.3, 0.4, 0.5;
- train speed: 1m/s, 2m/s, 5m/s, 10m/s.

Variation of the parameters listed above has resulted in 1200 VAMPIRE simulations. Post-processing of this data has required the development of Matlab code to automate the analysis.

4.3 Simulation Data Post-Processing

In order to analyse the VAMPIRE simulation data, the required outputs have been defined and the features related to the peak climb-out force prior to the wheel being ejected from the converging rails have been identified. The checkrail lateral force [kN], wheel lift [mm], left flangeway clearance [mm] and dynamic left flangeway clearance [mm] have been selected, analysed and plotted using Matlab. The location and magnitude of the maximum value of the switch blade lateral force has also been established. An example plot showing all the above outputs in the case of the left wheel of the leading wheelset is represented in Figure 5. Note that the distance reference frame differs from that used in the modelling of switch blade structural flexibility section; the stretcher bar location has been marked on all plots to allow cross-reference.

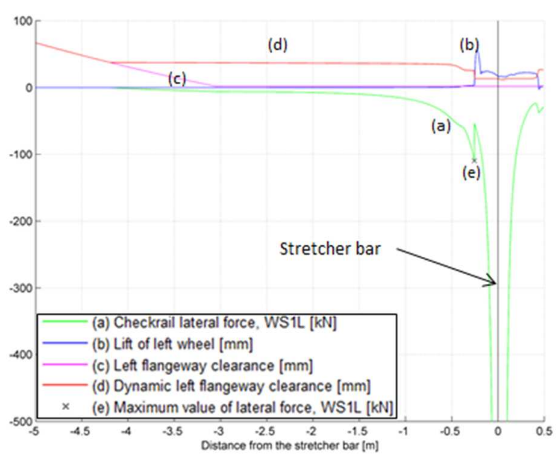


Figure 5. Trends of interest versus distance from the stretcher bar.

The left-hand side of Figure 5 shows the converging closed switch blade. At approximately -4.2m, the flangeway between switch and stock rail becomes smaller than the flange width. The trapped flange forces the switch blade across to maintain the flangeway. Until -3m, this requires increasing levels of force, largely from the increased switch blade displacement required. From then on, the displacement required remains the same but the lateral stiffness of the switch blade increases - Figure 2(a) - thereby causing a further increase in the lateral force.

At approximately -0.25m, the lateral forces are high enough that the wheel climbs out and the switch blade closes underneath it. Beyond this point, the simulation results become invalid.

The value marked with “x” is the maximum lateral force generated prior to the wheel climbing out and onto the rail head. The vertical line represents the position of the stretcher bar and also the location of the shear pin. It should be noted that depending on the parameter set being studied and its influence on the wheel-rail forces, the point of climb-out relative to the stretcher bar changes.

Figure 6 shows the maximum lateral force reacted at the switch blade contact just point prior to climb-out (derailment) for the entire simulation data set. The figure considers only the small distance over which the derailments occurred, which is within 1m of the stretcher bar.

The simulation cases which climb out late and close to the stretcher bar result in very high lateral forces. This is due to the geometry of the contact and the rapid increase in rail stiffness at the stretcher bar location itself. As the shear pin is currently configured at only 35kN, it has been deemed reasonable to only consider values of the maximum lateral force less than 500kN. By excluding the very high force values, the number of simulation results for analysis has been reduced from 1200 to 1055. It has also been noted that the majority of the rejected cases are within the outlying regions of the parameter studies, i.e. very low friction levels and high speed.

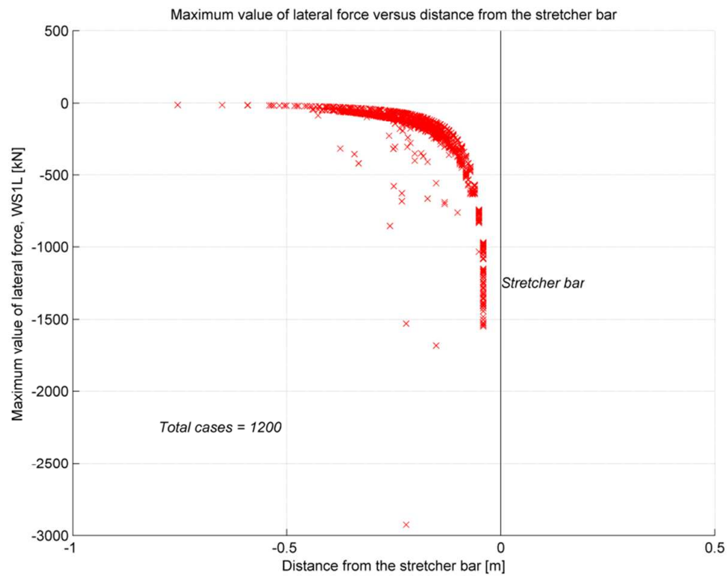


Figure 6. Maximum values of lateral force versus distance from the stretcher bar

Figure 7 shows the reduced data set for the maximum lateral force but with the addition of a number of limit lines (blue). These lines represent the force required to break a shear pin of a given strength with respect to distance from the stretcher bar. The limiting force varies due to the influence of the geometry between the encastre support at the switch heel and the stretcher bar.

As an example, considering the 35kN at the top of the plots (noting the negative axis scale) a red maximum lateral force data point above the line, would be insufficient to break a 35kN shear pin, and would thus cause a derailment. However, a red data point below the line would exceed the breaking load and hence the pin would shear, preventing a derailment.

The same approach is applied for all of the blue shear pin lines. Using this method, it is possible to identify the cases whereby the lateral forces at wheel climb-out are not sufficient to break a given shear pin and hence a derailment would occur. Similarly, cases whereby the lateral forces are sufficient, resulting in the failure of the shear pin and the safe passage of the wheel behind the switch blade are also identified.

The values of shear pin strength which have been considered are 35kN, 40kN, 45kN, 50kN, 55kN, 60kN, 65kN, 70kN; the region of interest is more clearly shown in Figure 7(b).

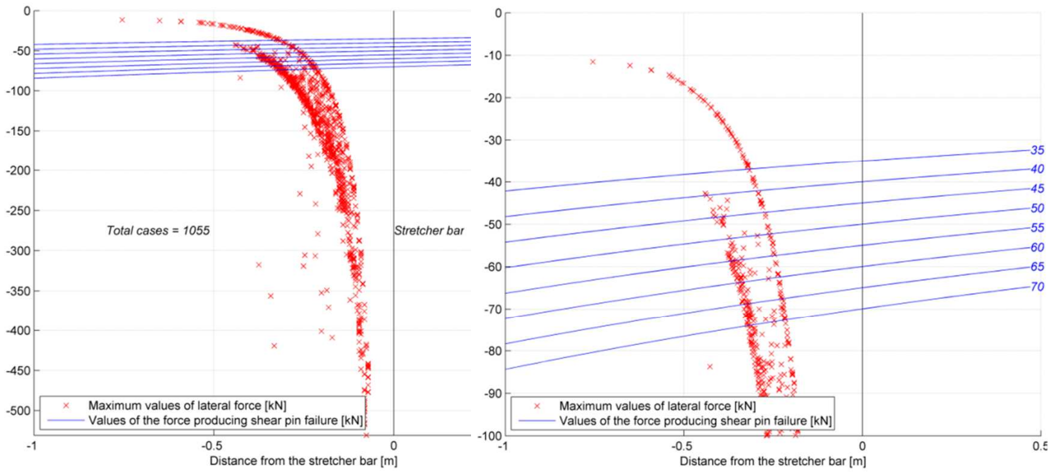


Figure 7. (a) Lateral force and shear pin failure forces versus distance from the stretcher bar; (b) expansion of relevant part of Figure 7(a)

5 Analysis of Results

5.1 Influence of Shear Pin Strength

The graphs presented in Figure 7 indicate the conditions where both the existing and increased strength shear pins would, or would not, prevent a derailment.

In order to further analyse the data and better understand the relationships between the simulation parameters studied, the results presented above have been expressed in terms of the number of simulated vehicle run-through events which generate insufficient lateral force to break a shear pin of a given strength.

From now on, if not explicitly stated, the word 'derailment' infers the case whereby a shear pin of given strength does not fail and hence the wheel would climb out of the flangeway onto the rail head. The complete results data set of 1055 simulations is reported in Figure 8(a). It shows the percentage of simulations indicating derailments before shear pin failure for 8 shear pin strengths, from 35kN (current shear pin) through to 70kN. In practice, the strength of other parts of the POE linkage limits the peak force to 70kN [12].

With a 35kN shear pin, just over 5% of the 1055 cases would generate insufficient lateral force to break the shear pin. Applying a stronger shear pin, e.g. 50kN, increases the proportion of cases which would derail to 8.6%. Moving to the extreme case, with a strength of 70kN, the value rises to 22.2%.

It was observed early in the analysis that a large proportion of the derailed cases occurred in the 50m radius turnout. This is due to the high wheelset angle of attack and hence earlier onset of flange climb as the wheelset is subjected to the forces imposed by the converging switch and stock rail. As 50m radius turnouts are not particularly representative of LU infrastructure, it was appropriate to consider the results excluding the 50m radius: the number of cases analysed decreases from 1055 to 828. The results in terms of percentages are reported in Figure 8(b): the risk of derailment before shear pin failure is significantly reduced. A summary of the derailment percentage and number of derailments is reported in Table 2. When excluding the 50m radius, the current 35kN shear pin was successful in preventing derailment in all cases simulated.

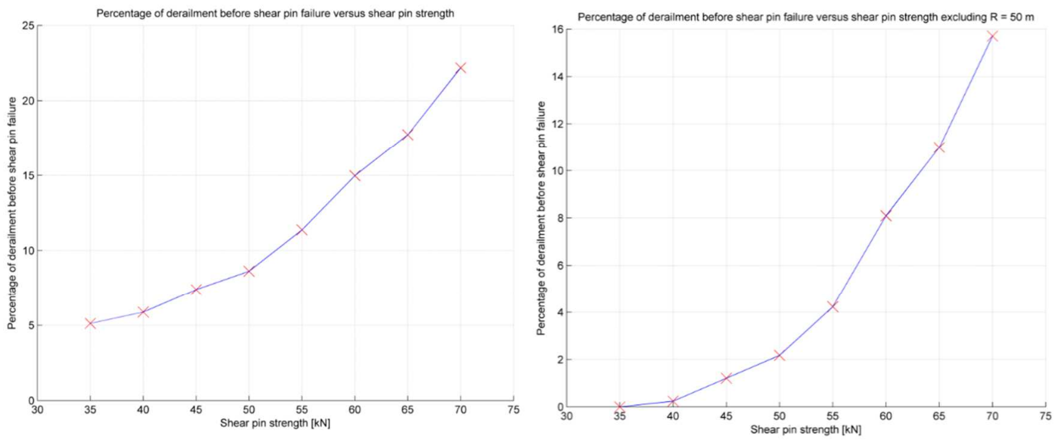


Figure 8. (a) Percentage of derailment before shear pin failure versus shear pin strength; (b) percentage of derailment before shear pin failure versus shear pin strength excluding the turnout radius equal to 50m.

When excluding the 50m curve radius, it is clear that there is a significant increase in the gradient of the derailment trend above 50kN. This suggests that any increase in shear pin strength above 50kN might begin to lead to an unacceptable increase in derailment risk.

The improved fatigue life and service reliability of a 50kN shear pin could well justify the small increase in derailment risk observed. This suggests that 50kN may be a sensible target value for a future shear pin design.

Shear Pin Strength (kN)	All Curve Radii		Excluding 50m Radius	
	Number of Derailments	Percentage of Total Cases	Number of Derailments	Percentage of Total Cases
35	54	5.1%	0	0.0%
40	62	5.9%	2	0.2%
45	78	7.4%	10	1.2%
50	91	8.6%	18	2.2%
55	120	11.4%	35	4.2%
60	158	15.0%	67	8.1%
65	187	17.7%	91	11.0%
70	234	22.2%	130	15.7%
	Total Cases		Total Cases	
	1055	100%	828	100%

Table 2. Derailment Trends with Shear Pin Strength

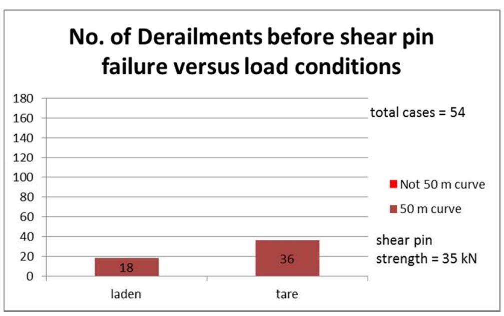
In order to achieve a more detailed analysis of the number of derailments before shear pin failure, the dependence between each of the studied parameters (vehicle load and profile conditions, turnout radius, speed and friction coefficient) and shear pin strength (from 35kN to

70kN) has been considered. In this final analysis all the considered turnout curve radii have been included, together with the case whereby 50m radius curves are excluded (highlighted in red). Based upon the above analysis of the gradient of the failure trend curves, the results are only presented here for the 35kN and 50kN shear pins.

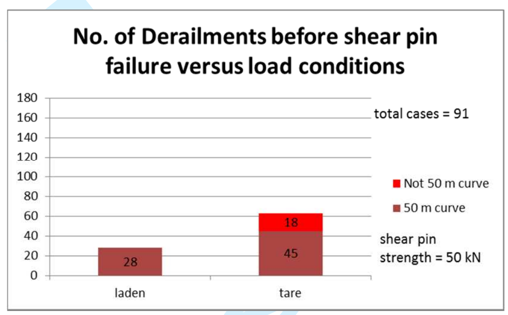
5.2 Influence of Vehicle Load and Profile Conditions

The number of derailments versus load condition and profile conditions for shear pins of 35kN and 50kN are reported in Figure 9. It is clear from Figure 9 (a) and (b) that increasing the shear pin strength from 35 to 50kN increases the number of derailment cases (from 54 to 91). The majority of derailments occur in the tare condition as the wheel is carrying less vertical load and hence the lateral component required to break the shear pin is lower. For the 35kN shear pin, derailments only occur on the 50m radius case, whilst for the 50kN shear pin, there are 18 derailment cases on larger radii which are all in tare condition.

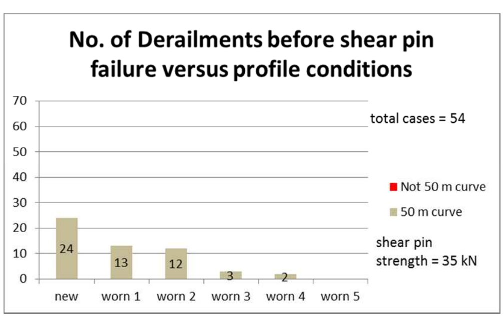
In Figure 9(c) and (d) 6 wheel profiles have been considered, in different wear states: profiles are ordered from least worn (new) to the most worn (worn 5). It is noticeable that the more heavily worn wheels are associated with a reduced risk of derailment. A more severely worn wheel often has a steeper flange angle, requiring a higher lateral force to initiate wheel climb. Therefore worn wheel profiles are more likely to break the shear pin before climbing out.



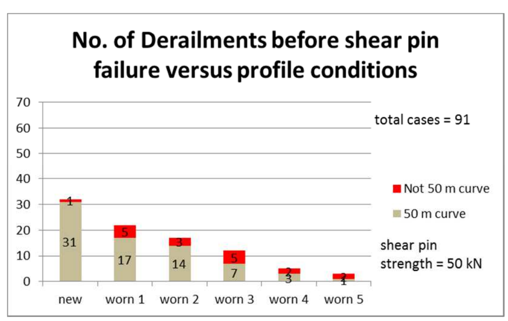
(a)



(b)



(c)



(d)

Figure 9. (a) Number of derailment versus load conditions (shear pin strength 35kN); (b) number of derailment versus load conditions (shear pin strength 50kN); (c) number of derailment versus profile conditions (shear pin strength 35kN); (d) number of derailment versus profile conditions (shear pin strength 50kN);

5.3 Influence of turnout radius, speed and friction coefficient

The number of derailments versus turnout radius, speed and friction coefficient for shear pins of 35kN (current shear pin) and 50kN are reported in Figure 10.

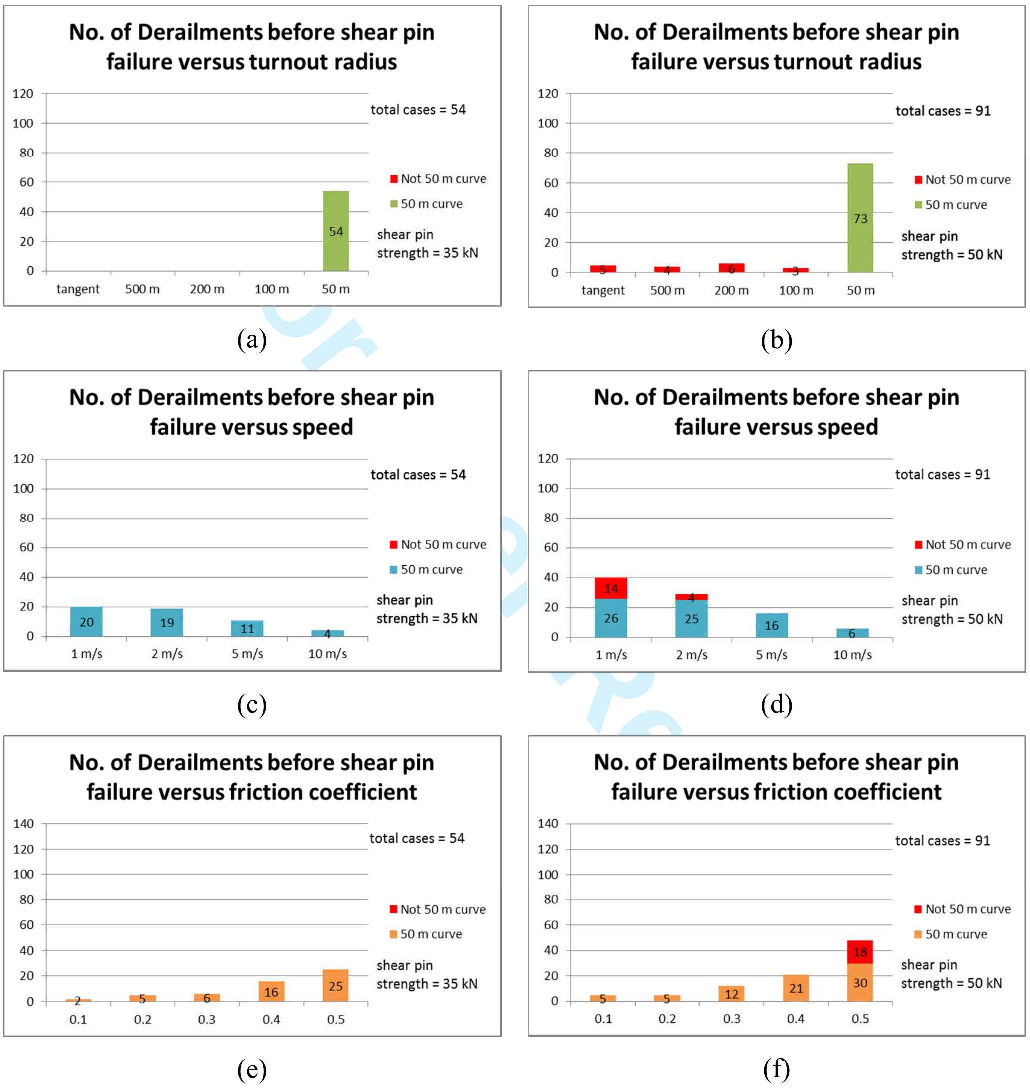


Figure 10. (a) Number of derailment versus turnout radius (shear pin strength 35kN); (b) number of derailment versus turnout radius (shear pin strength 50kN); (c) number of derailment versus speed (shear pin strength 35kN); (d) number of derailment versus speed (shear pin strength 50kN); (e) number of derailment versus friction coefficient (shear pin strength 35kN); (f) number of derailment versus friction coefficient (shear pin strength 50kN);

Figure 10(a) demonstrates that with the 35kN shear pin, all derailment cases are on the 50m radius turnout. Strengthening the shear pin to 50kN (Figure 10(b)) increases the number of derailments on the 50m curve but also introduces a few derailment cases on all other radii considered.

From Figure 10(c) and (d) it can be observed that increasing the speed decreases the number of derailments. When the wheel climbs out at a higher train speed, the vertical force required to accelerate the wheel upwards is greater. This tends to favour shear pin breakage over derailment. For a shear pin strength of 50kN, there are 18 derailment cases for radii over 50m, and these are concentrated at the low speeds of 1m/s and 2m/s. This point is useful to note, as in effect, the increased likelihood of derailment occurs only at very low speed and hence the potential consequences of derailment are likely to be relatively low.

From Figure 10(e) and (f) one can observe that increasing the friction coefficient increases the number of derailments. Higher wheel/rail friction enables the wheel to climb out at a lower lateral force, hence promoting derailment over shear pin breakage. With a 50kN shear pin and ignoring the 50m curve radius, all derailments occur with the highest friction coefficient of 0.5.

6 Discussion

6.1 Implications of the Results

The most sharply-curved standard turnout found on LU and NR routes is the AV6, which has a turnout radius of 97.6m out of straight. In restricted locations where similar flexure turnouts are installed, the inner curve radius could be sharper than this. However, such cases with turnout radii significantly less than 100m are rare on LU running lines.

LU provided data on the distribution of turnout radius for their running-line turnouts. This is shown in Table 3. Turnouts within depots and sidings do not have shear pins so they are not included in these statistics.

Table 3. Distribution of Radius on LU Running Line Turnouts

Turnout Radius (m)	Percentage of Turnouts
1 to 50m	0.0%
50 to 100m	7.4%
100 to 200m	37.8%
200 to 500m	45.0%
500m+	9.8%

Based on these figures, the 50m case may be excluded, and the 100m case considered representative of the sharpest turnouts on LU. The existing 35kN shear pin will then prevent derailment in 100% of the remaining cases analysed.

Strengthening the shear pin to 50kN prevents derailment in 97.8% of the cases analysed. The 2.2% 'at-risk' cases are all in the tare condition, at speeds ≤2m/s, and at the highest wheel/rail friction coefficient of 0.5.

Such high flange friction on a sharp curve would only be achieved where flange lubrication systems had become ineffective, reducing the likelihood of these circumstances. The consequences of a derailment at very low speed are much less severe than a high-speed derailment, and in tare condition the safety risk to passengers would also be minimised.

6.2 Limitations of the Analysis

Certain assumptions and simplifications were made during the analysis. The effect of these is considered here.

1. The situation analysed a wheel flange becoming trapped between the stock rail and the back of the closed switch blade. Depending on the turnout geometry, track gauge and wheel wear state, it is possible that the wheelset flanges could become trapped in the converging gauge between the stock rail and the gauge face of the opposite (open) switch blade. The latter scenario has not been considered to date.
2. The model did not consider the friction between the switch rail and the flange back, or the friction between the switch rail and the sleeper. In practice these interfaces are usually effectively lubricated, and the closed switch rail is not carrying a wheel. The frictional forces in these contacts are therefore likely to be small compared to the other forces considered in the analysis. However, their influence would be to slightly increase the likelihood of derailment.
3. The model did not consider the mass of the rails. At the low speeds considered, the inertial effect of moving the rails would be negligible in comparison to the other forces.
4. A single track/switch blade geometry (BVS Switch) in new condition has been used to calculate the stiffness along the length of the switch and to simulate all the derailment cases, despite different switch radii being evaluated. In practice this is equivalent to a BVS switch installed in turnouts with varying degrees of contrary or similar flexure. Longer switches are likely to represent a lower derailment risk as the blade is more flexible and a greater proportion of the wheel/rail forces will be carried by the shear pin; they also tend to be installed on a larger curve radius. Conversely, the shortest switch (AVS) may represent a higher risk of derailment, but these switches are rare on running lines.

6.3 Potential Future Work

Some turnouts are designed to be run-through in normal operation. Future work could include modelling a turnout of this type and carrying out tests to validate the model of the switch blade and wheel trapping behaviour. This would also indicate the significance of the simplifications made in the model, such as neglecting frictional forces on the switchblade.

Strengthening the existing 35kN shear pin to 50kN would slightly increase the risk of derailment in the event of run-through, while improving system reliability. These effects could be compared using a quantitative risk assessment to justify a change in standards. Run-through events on LU are very rare, and it may be that a quantitative risk assessment could justify dispensing with the shear pin entirely.

7 Conclusions

Mathematical modelling work has been carried out to investigate the potential for a revision to the strength of shear pins used in LU S&C drives to protect against run-through derailments. The revision has been requested based on a need to increase the service life of the pins which are prone to fail due to service fatigue loadings.

Based upon the work presented in this report, it has been possible to demonstrate that the current shear pin (35kN) is effective in preventing all simulated derailments for turnout radii representative of those used on LU running lines. This conclusion provides confidence both in the modelling approach and also in the current specification of the shear pin in terms of its ability to protect against run-throughs.

In terms of the potential to increase the strength of the shear pin to protect against fatigue failure, the simulation work has identified a shear pin with a strength of 50kN as being good compromise in terms of minimal increase in derailment risk, whilst offering what would be a useful increase in fatigue life. The performance of the 50kN shear pin in terms of the studies performed in this work can be summarised below:

1. 2.2% risk of derailment due to non–failure of the shear pin (18 in 828 cases). Compares to 0% (0/828) cases for the current 35kN shear pin.
2. No derailments cases occur in the fully laden vehicle condition.
3. Derailment cases are concentrated at low speeds (1m/s and 2m/s). In terms of safety risk analysis, this reduces the potential severity of outcome.
4. All the predicted derailments occur when assuming a high value of wheel-rail friction coefficient (0.5); thus an effective lubrication regime could help further mitigate the risk of increasing the strength of the shear pin.

When it is also considered that run-through events on LU are very rare, the overall derailment risk from strengthening the shear pin strength to 50kN is likely to be more than offset by the improvement in reliability of the network and reduction in other forms of safety risk.

It is possible that a quantitative risk assessment could justify dispensing with the shear pin entirely.

References

1. 'Surelock Ground Connections Development wide to gauge points: Product Requirements (PR)', London Underground Limited, HOE-TRK-RS-2013-0133.
2. 'Requirements for Powered Point Operating Equipment', Railtrack Company Product Specification RT/SRS/2001 Issue 2
3. 'Requirements for the Design, Operation and Maintenance of Points', Railway Group Standard GI/RT7004 issue 1, 2000 (withdrawn)
4. 'Track System Requirements', Railway Group Standard GC/RT5021 Issue 3, 2007 (superseded)
5. Clark, R. A., 'Measurement of Wheel/Rail Contact Forces at a Selection of Switches and Crossings using HSFV1 Equipped with Load Measuring Wheelsets', British Rail Research TM-DA-39, 1981
6. Clark, R. A., 'Measured Wheel/Rail Contact Forces Produced by Locomotive 56042 at a Selection of Switches and Crossings', British Rail Research TM-VDY-024, 1988
7. Smallwood, R., 'The Effects of Traffic, Track Conditions and S&T Adjustments on the Strain Environment of Point Clamp Lock Components', British Rail RR-MF-047, 1993
8. 'Rail Clamp Point Locks', Railway Group Standard GK/RC 0774 Issue 1, 1995 (withdrawn)
9. 'Certificate of Temporary Non-Compliance: HPSS Point Systems', RSSB, 03/261/TNC, 2003.
10. 'Impact Assessment for Changes to Railway Group Standard GC/RT5021', RSSB, 09IA12, 2009.
11. 'Railway applications - Track - Switches and crossings - Part 4: Actuation, locking and detection', BS EN 13232-4:2005+A1:2011
12. Scoble, M., 'Surelock Points System Run Through Event Performance Assessment', London Underground Limited, HOE-TRK-REP-2015-0351 Issue 1.3, 22/01/15

Appendix 1

Switch blade: BVS Switch – Assembly and Machining.

The reference of the following CAD drawing is RE/PW/1601.