

Analysis of coupling rod operating conditions in double slip turnout operation

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ABSTRACT

Despite operational limitations and high maintenance requirements, double slip turnouts still account for a significant percentage of track components in European railway networks. The need for their continued operation stems primarily from spatial constraints in station areas, which prevent the use of simpler track layouts for routing train traffic to platform tracks. Key operational challenges include speed restrictions on divergent routes and difficulties in maintaining track geometric stability. An area of particular structural sensitivity is the interaction place between the switch point machine, locking devices, and switch coupling rods, which are responsible for the switching kinematics and ensuring the proper closing and opening of switch rails against stock rails. Literature analysis indicates a significant research gap regarding the assessment of actual operating conditions of the switch coupling rod in the turnout and the consequences of insufficient track tamping in this area. This article attempts to address the indicated issue by combining an empirical approach (field tests) with numerical modeling using the finite element method (FEM). Specifically, the purpose of the work is to evaluate the mechanical behavior of the switch coupling rod and the specific consequences of inadequate track tamping. The novelty of the work is the comprehensive integration of empirical field test data with advanced finite element modeling to analyze the relationship between track maintenance conditions and turnout switching kinematics, offering a modern perspective on the optimization of safe railway traffic management.

Keywords: double slip, railway infrastructure, coupling rod, track maintenance, field research, turnouts.

INTRODUCTION

Double and single slip turnouts constitute a numerous family of turnouts on the railway network in Poland. According to data published by PKP Polskie Linie Kolejowe S.A., the total length of operated railway lines in 2024 amounted to over 18,806 km, which translates to over 36,326 km of tracks [1]. An essential element of linear infrastructure, affecting its structure and operational parameters, is railway turnouts [2]. Within the operated railway lines managed by PKP Polskie Linie Kolejowe, there are 37,352 turnouts installed, including approximately 3.287 double slip turnouts and single slip turnouts. They account for 8.8% of all turnouts on railway lines, as summarized in Table 1.

We distinguish between two types of slip turnouts: single slip turnouts and double slip turnouts, in which:

- a) the switch rails are located within the boundaries of the turnout quadrangle, i.e., they lie between lines perpendicular to the bisectors of the angles of the common crossings, drawn through their mathematical points,
- b) the switch rails lie outside the boundaries of the quadrangle, i.e., outside the lines perpendicular to the bisectors of the angles of the double or triple crossings, drawn through the mathematical points of these crossings [3].

A double slip turnout commonly used on the railway lines of PKP Polskie Linie Kolejowe is a structure with switch rails located within the

Table 1. Number of slip turnouts operated on railway lines managed by PKP Polskie Linie Kolejowe S.A.

Single slip and double slip	
Type	Quantity
60E1-190-1:9	905
49E1-190-1:9	2382

boundaries of the turnout quadrangle, allowing vehicles to travel in four directions:

- (A-B) along the diverging track with a maximum speed restriction dependent on the radius of the diverging curve, typically 190 meters,
- (A-D) along the main track,
- (C-B) along the main track,
- (C-D) along the diverging track with a maximum speed restriction dependent on the radius of the diverging curve, typically 190 meters.

A double slip turnout is obtained by combining two pairs of switch units that direct traffic in opposite directions and the crossing of tracks A-D and B-C into intersecting straight tracks, as illustrated in Figure 1.

Key design elements of the double slip turnout consist of 3 sections as:

- 2 common crossings with guard rails,
- 4 switches with locking devices included 2 point machines and
- 2 obtuse crossings with check rails [4].

The geometric layout of the double slip turnout is shown in Figure 2, while the basic construction and geometric parameters characterizing the tested double slip type: 60E1-190-1:9-S-b/(d)-1435mm are presented in Table 2 as, among others [5, 6]:

- 60E1 – type of rail profile [7],
- 190 – radius of diverging track,
- 1:9 – angle,
- S – welded version,

- b/(d) – assembled on prestressed concrete „b” or wooden „d” sleepers,
- 1435 – track gauge.

The purpose of this article is to analyze the operational behavior of railway turnouts, focusing specifically on the interaction between the switch machine and the moving elements of a double slip turnout, with particular emphasis on the performance and structural response of the switch coupling rod. Specifically, the paper examines the mechanical behavior of the switch coupling rod and the consequences of insufficient track tamping by means of combined empirical field tests and finite element method (FEM) simulations.

DOUBLE SLIP TURNOUT WITH SWITCH OPERATING DEVICES

The primary task of the switch operation in a double slip turnout is to change the direction of travel of a rail vehicle from the main track to the diverging track, which is achieved through moving switch elements such as the curved switch rail and the straight switch rail, controlled by a switch point machine via the locking devices [8–10].

To ensure proper adherence of the switch rail to the stock rail and to prevent it from spontaneously moving away from the stock rail under passing vehicles, locking devices are used. For these structures, functional requirements should be formulated, including in particular:

- resistance to temperature changes,
- tolerance of longitudinal switch rail movement,
- protection against external environmental factors,
- Maintenance and service capabilities.

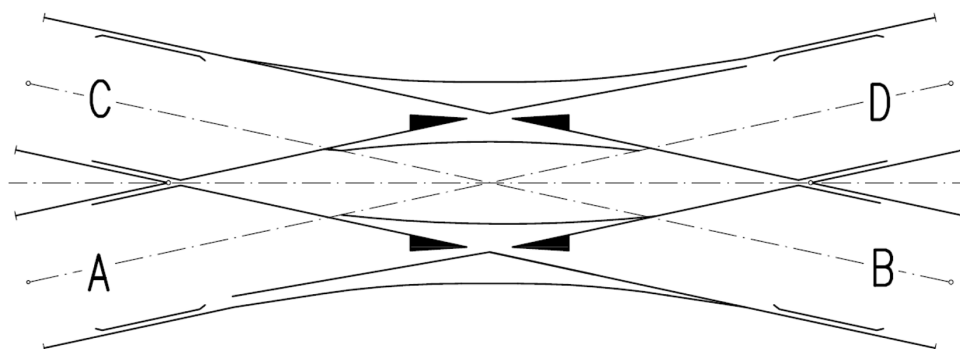


Figure 1. Track connection diagram in a double slip turnout

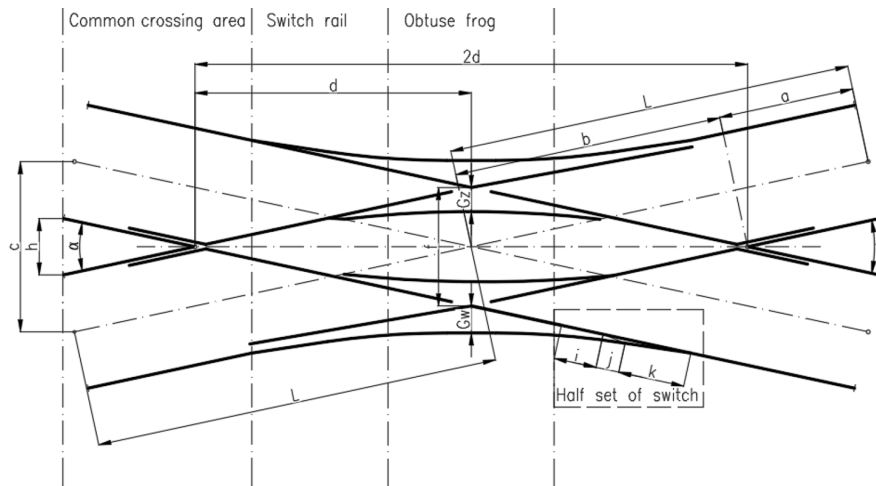


Figure 2. Geometric layout of the double slip turnout type 60E1-190-1:9

Table 2. Geometric dimensions of the double slip turnout type 60E1-190-1:9

Double slip 60E-190-1:9		
a	Distance from the start of the turnout to the nose of the crossing	3.660 mm
b	Distance from the crossing nose to the geometric center of the turnout measured along the track axle	12.954 mm
c	Distance of track axles at the end of the turnout	1837 mm
d	The distance from the crossing nose to the geometric center of the turnout measured along the turnout axle	12.974 mm
2d	Twice the dimensions "d"	25.949 mm
f	Distance between the noses of the obtuse crossings	1.437 mm
Gw	Width of the groove in the frog	300 mm
Gz	Check rails gauge in the frog area	292 mm
h	Distance between the rolling edges of the inner rails at the beginning of the turnout	2.825 mm
i	Distance between the beginning of the switch rail and the track gauge 1435mm	3.014 mm
j	Distance between the beginning of the inner edge and the beginning of the arc	100 mm
k	Track gauge extension length from 1435mm to 1445mm	1.560 mm
L	Distance from the start of the turnout to the geometric center of the turnout	16.615 mm
α	Turnout angle	6° 20' 24.69"

Depending on the design of the locking device, this system consists of the following basic components:

- two lock brackets attached to the stock rails by means of bolts with nuts, enabling proper guidance of the slide rod and the dovetail buckle,
- two dovetail buckle mounting clamps equipped with a set of bushes, safety pins, and eccentric adjusting pins,
- two dovetail buckles,
- a slide rod (insulated) complete with safety elements is shown in Figure 3 [11].

Locking devices are located at the beginning of the switch rail and are positioned between the

second and third turnout sleeper, counting from the switch joint, as shown in Figure 4 [12, 13].

The switch operates with a track switching device, such as a point machine, whose task is to move the switch rail into the end position and to determine the direction of travel of the rail vehicle [14]. Additionally, point machines are equipped with switch rail detection devices. The term point machine is understood as an assembly of elements including: the electric drive, the point machine bed with fastening elements to the turnout, the throw rod, and two detection rods, whose main task is to monitor the correct end position of the switch rail, and in particular to ensure its proper adherence to the stock rail [15, 16].



Figure 3. Slide rod for R190m radius double slip turnout

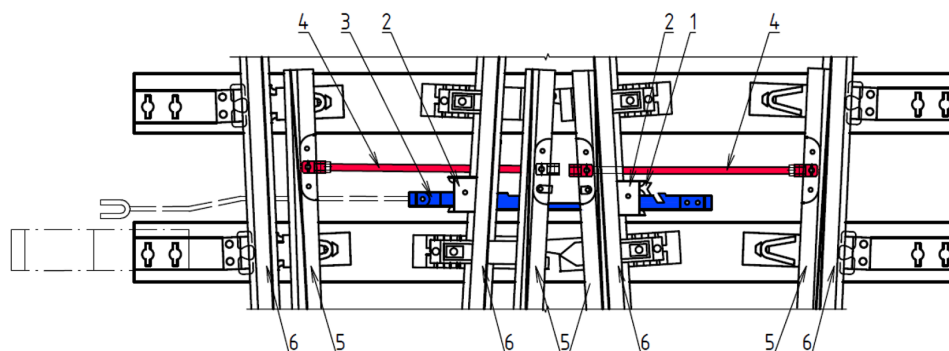


Figure 4. Locking device for double slip turnout type 60E1-190-1:9: 1) dovetail buckle, 2) locking bracket, 3) slide rod, 4) coupling rod, 5) switch rail, 6) stock rail

The control unit operating two pairs of switches, apart from the point machine, consists of: a slide-type locking device and two switch coupling rods in the form of a straight bar below the stock rail foot by dimension „z”, as illustrated in Figure 5.

The locking device in double slip turnouts most frequently operates in three closing phases, and the travel of the switch point machine operating rod is 220 mm [17, 18]. Two inner switch rails in turnouts are equipped with a locking device that is most commonly sensitive to climatic conditions and longitudinal displacements of the switch rails [19]. Locking devices that are insensitive to the mutual displacement of rail elements forced by changes in ambient temperature are increasingly applied.

When railway rolling stock travels through a double slip turnout, the internal switch rail is maintained in the ‘closed’ position, while the external switch rail is maintained in the ‘open’ position at a constant distance relative to the internal switch rail by means of the switch coupling rods.

Due to the static working nature, the switch coupling rod is an auxiliary device whose design should ensure high operational reliability throughout the entire life cycle of the turnout [20].

Design of the adjustable switch coupling rod

The switch coupling rod is an element coupling the internal switch rail with the external switch rail in a double slip turnout at the locking device position. The design of the switch coupling rod comprises four basic elements: the rod (most frequently straight, although bent versions are also increasingly common, as illustrated in Figures 6 and 7, respectively), the fixed switch rail foot bracket, the adjustable switch rail foot bracket (using a bolted connection) and the adjusting pin.

Tasks performed by the switch coupling rod

The switch assembly in a double slip turnout consists of eight half-switches: four internal and

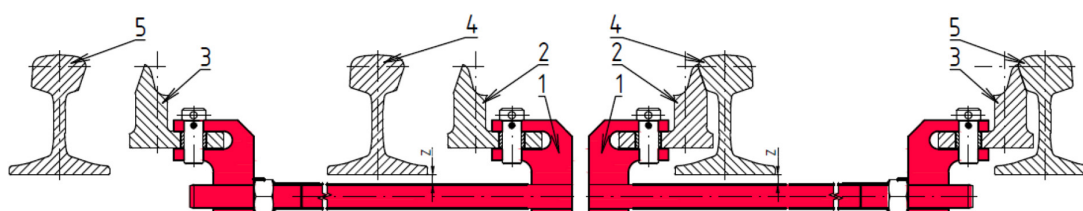


Figure 5. Locking device set: 1) coupling rod, 2) internal switch rail, 3) external switch rail, 4) internal stock rail, 5) external stock rail



Figure 6. Adjustable coupling rod, straight type: 1) straight rod, 2) fixed bracket, 3) adjustable bracket, 4) adjustment pin

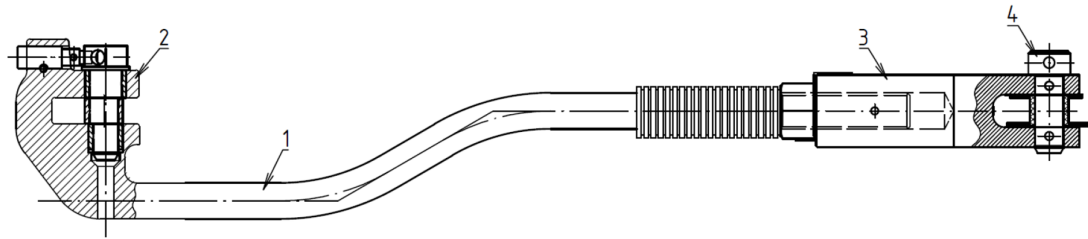


Figure 7. Adjustable coupling rod, bent type: 1) bent rod, 2) fixed bracket, 3) adjustable bracket, 4) adjustment pin

four external. A locking device is installed within the area of the internal half-switches, and four switch coupling rods are used, which allow for the connection of the external half-switches with the internal ones. The task of the switch coupling rod is to ensure a rigid connection between the external switch rail and the internal switch rail of the opposite track and to transmit the throw force from the internal switch rails to the external switch rails (these are horizontal forces acting parallel to the axis of the drive operating rod, as illustrated in Figure 8. Furthermore, operational conditions reveal impacts resulting from the uneven elongation of switch rails caused by thermal effects or creep (Figure 9), as well as from uneven tamping of the double slip turnout, mainly in

the area of the locking device, which results in increased vertical displacements of the stock rail and the switch rail [21, 22].

Unlike the track rail, which is fastened to every sleeper, the switch rail in its beginning part lacks fastenings to the switch sleepers due to the necessity of horizontal movement during switch throw. Additionally, it does not feature continuous longitudinal resistance. For this reason, the creep of the switch rail is not a process distributed along its length, but takes the form of a concentrated displacement of its end. The equation for the longitudinal (thermal) force for the switch rail cross-section A_{switch_rail} is given by Equation 1.

$$F_{term} = E \cdot A_{switch_rail} \cdot \alpha \cdot \Delta t \quad (1)$$

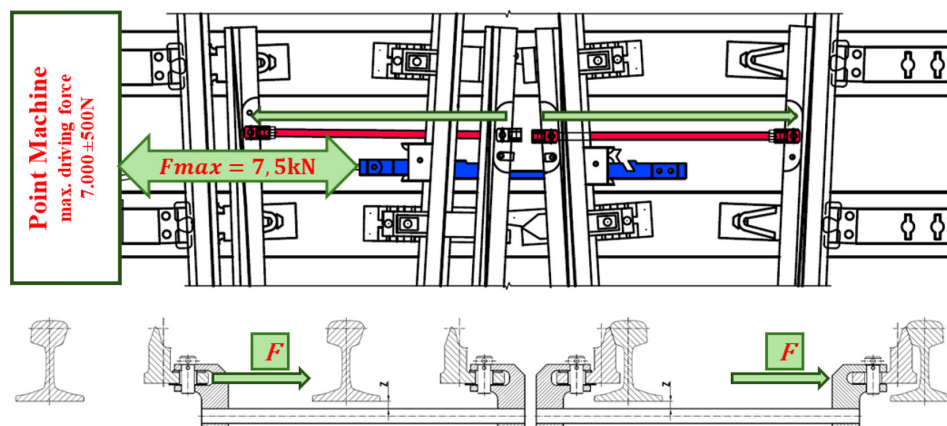


Figure 8. Point machine – double slip turnout cooperation

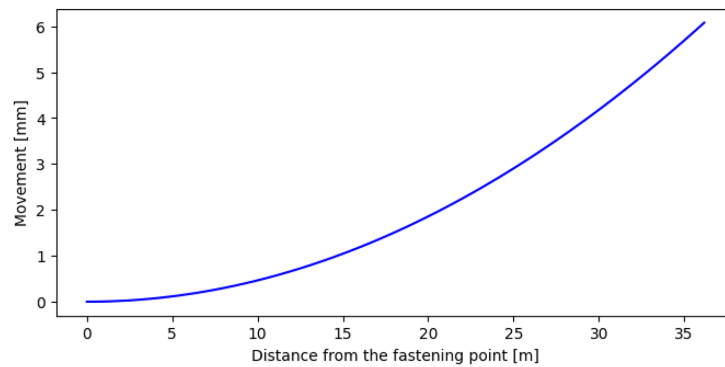


Figure 9. Graph of the switch rail longitudinal displacement versus switch rail length

where: F_{term} – the value of the thermal force [N], E – Young’s modulus for steel (approx. 2.1×10^5 MPa), A_{switch_rail} – cross-section of the switch rail [mm²], α – Coefficient of linear thermal expansion for rail steel (approx. 1.12×10^{-5} 1/°C), Δt – Current rail temperature minus neutral temperature (the temperature at which thermal forces inside the rail are zero) [°C].

Tendency of the switch rail to expand due to temperature changes, the switch rail encounters the resistance of slide plates (due to friction), the fastening system and its end section, as well as the locking device cooperating with the switch machine. Therefore, the equation for the switch rail displacement U_{switch_rail} takes the value according to Equation 2.

$$U_{switch_rail} = \Delta L_{term} - \frac{F_{point_machine}}{f} \quad (2)$$

where: $\Delta L_{term} = L_{switch_rail} \cdot \alpha \cdot \Delta t$ (theoretical elongation), f – coefficient of point machine and switch rail fastening stiffness in the attached section.

An increase in the value of the thermal force leads to accelerated wear of turnout components, particularly the switch coupling rod, which is exposed to longitudinal forces and torsional moment.

The switch coupling rod performs tasks that have a direct impact on railway traffic safety within railway stations, and its design should ensure the transmission of loads generated under normal turnout operating conditions without damage or permanent deformation. Furthermore, the influence of variable climatic conditions, including

ambient temperature fluctuations (summer–winter), on the operation of the switch coupling rod should be kept to a minimum. Due to the specifics of the tasks performed, various types of damage to the component coupling the internal switch rail with the external switch rail have been observed. The most frequent destructive failures that prevent further repair include:

- e) cracking of the rod at the thread location in the fixed bracket, as shown in Figure 10 and
- f) plastic deformations resulting from the pressure of the stock rail foot on the switch coupling rod during railway vehicle passage, as illustrated in Figure 11.

Example photos of the switch coupling rods in operated double slip turnouts, which were damaged during operation, are illustrated in Figure 12.

Point machine impact on switch rails operation

The maximum driving force „ F ” generated by the point machine is 7.5 kN [17]. The work is



Figure 10. An example of a broken coupling rod at the fixed bracket location

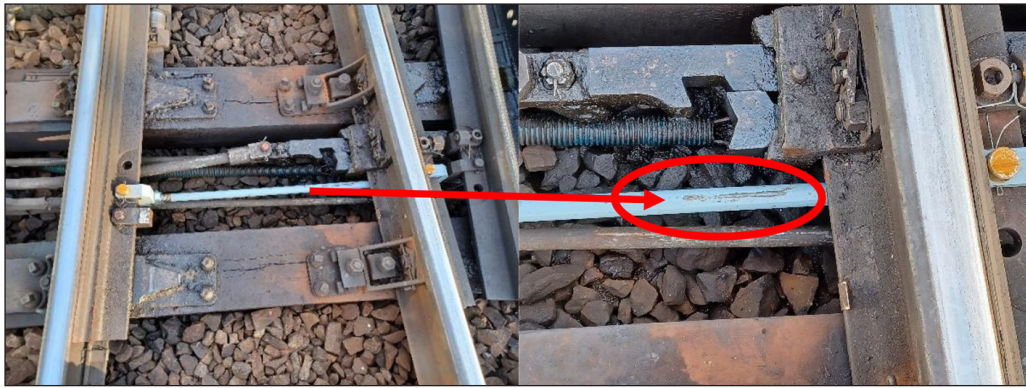


Figure 11. Location of stock rail foot pressure on the coupling rod

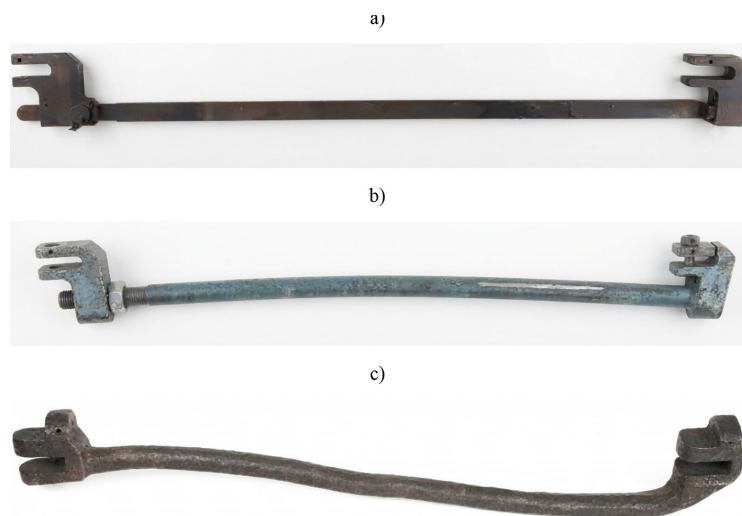


Figure 12. Typical coupling bar failures during operation: a) cracked rod in the fixed bracket at the thread, b) cracked fixed bracket, visible contact points with the stock rail foot, c) rod deformation along its entire length

performed by means of the drive slider and acts on the throw bar of the locking device. The operation of the switch locking device causes the switch rail to move from the open position to the closed position (or vice versa). The internal switch rails are coupled to the external switch rails by means of switch coupling rods, supporting the switch throw process. The active side shown in Figure 13 is the internal switch rail controlled directly by the operation of the point machine, whereas the passive side is moved as a result of the switch coupling rod operation. The dependency mechanism of the point machine and turnout interaction is shown in Figure 14.

Empirical testing of switch parameters for a double slip turnout

In order to proceed with testing the impact of railway rolling stock on the switch coupling rod, the geometry of the switches, turnout and the

point machine-turnout interaction were checked in accordance with the technical operation conditions of the turnout, containing measurement sheets and instructions describing the relevant parameters [11, 17] in two double slip turnouts within a single railway station. Selected parameters of this interaction are presented in Table 3.

To measure the slider stroke, the point machine housing was adopted as the reference criterion, from which the slider extension was measured. The result is the difference in the slider extension values between the two end positions, as illustrated in Figure 15. The switch rail adherence/opening parameter was measured at the point-locking location. The resistance paths measured in accordance with the Id-4 instruction [11] for slider locking devices insensitive to longitudinal movement of the movable switch elements.

The locking devices in the tested double slip turnout work properly with the point machines.

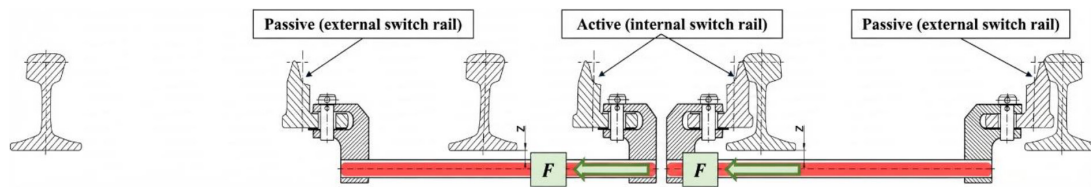


Figure 13. Locking device during operation. The force F comes from the point machine

Multiple check throws of the switch rails in the turnout showed no negative interactions in the point machine-turnout operation. The measurement of switch rail throwing resistance forces in the turnout was carried out using a PAMAR MS meter and a measuring pin, which is a strain gauge transducer measuring the mechanical deformation value of the pin's shape, as illustrated in Figure 17. An example chart of switch throwing resistance forces as a function of time is shown in Figure 16 [23]. During the measurements, a measuring pin connected to a data logger was inserted in place of the pin connecting the point machine slider with the locking device slider, and the turnout was thrown in the "+" and "-" directions. The obtained values of the switch throwing resistance forces did not exceed the permissible values applicable on PKP PLK SA lines and meet the requirements specified in the Id-4 instruction (Table 4) [11].

Identification of the research problem

The most frequent irregularities in the maintenance processes of double slip turnouts, which may cause additional negative impacts on the switch coupling rod are:

- Improper tamping of the turnout within the locking device area, causing mutual vertical displacement of elements and possible contact of the stock rail foot with the switch coupling rod,
- Non-adherence of the switch rail to the slide plates within the locking device area, causing vertical movements of the switch rail under the pressure of railway rolling stock wheels,
- Non-adherence of the switch rail to the distance blocks, causing horizontal movement of

the switch rail under the pressure of railway rolling stock wheels,

- Wear of wooden sleepers at the slide plate locations, causing changes in track cant and twist,
- Poor technical condition of the slide plate fastenings (loose screws, resilient fastening of the stock rail),
- High switch throwing resistance forces, inconsistent with the parameters contained in the infrastructure manager's instructions [11],
- Creeping of the switch rail and angular movement of the operating rod relative to the point machine slider,
- Improper plastic working of the switch rail (bending-straightening) after mechanical processing.

The identification of potential irregularities in maintenance processes necessitated undertaking innovative research that has no precedent in international scientific and technical literature. The analysis of the switch coupling rod operating parameters was divided into two research stages: a simulation approach using the finite element method (FEM) and field tests based on an original comparative method. The carried out measurements and detailed analysis of the switch coupling rod operating conditions in operated double slip turnouts will constitute the basis for optimizing the design of this element (taking into account the availability parameters of the railway line and the turnout in the planned life cycle [20]). This study takes into account the specific operating conditions of the railway trackage, including climatic factors and the quality of track tamping in the area of locking devices.



Figure 14. Switch control dependency schematic

Table 3. Selected parameters of point machine – double slip interaction

Double slip	Parameter	Measuring tool	Nominal dimension [mm]	Measured dimension [mm]
Locking device_1	Switch rail opening	Steel tape measure	160 ±5	162
	Switch rail opening	Steel tape measure	160 ±5	161
	Width groove	Caliper	min. 58	58
	Width groove	Caliper	min. 58	59
	Resistance path	Caliper	46 (+5/-10)	39
	Drive stroke	Steel tape measure	220 ±2	218
Locking device_2	Switch rail opening	Steel tape measure	160 ±5	161
	Switch rail opening	Steel tape measure	160 ±5	163
	Width groove	Caliper	min. 58	58
	Width groove	Caliper	min. 58	58
	Resistance path	Caliper	46 (+5/-10)	40
	Drive stroke	Steel tape measure	220 ±2	218
Locking device_3	Switch rail opening	Steel tape measure	160 ±5	164
	Switch rail opening	Steel tape measure	160 ±5	161
	Width groove	Caliper	min. 58	58
	Width groove	Caliper	min. 58	60
	Resistance path	Caliper	46 (+5/-10)	40
	Drive stroke	Steel tape measure	220 ±2	218
Locking device_4	Switch rail opening	Steel tape measure	160 ±5	162
	Switch rail opening	Steel tape measure	160 ±5	162
	Width groove	Caliper	min. 58	58
	Width groove	Caliper	min. 58	58
	Resistance path	Caliper	46 (+5/-10)	40
	Drive stroke	Steel tape measure	220 ±2	218



Figure 15. Drive rod stroke measurement

Stress analysis using the finite element method (FEM)

The analysis was performed in Ansys Mechanical 2025R2 with large deflection and nonlinear contacts assumptions. The applied boundary conditions are presented in Figure 18. The forces and displacements were applied to the rail surface.

The starting baseline for the simulation studies was an adjustable straight rod with a diameter of Ø27 mm, featuring a lowered position relative to the stock rail foot by the dimension „z” = 11 mm, as illustrated in Figure 18.

As a result of contact interactions and damage to the switch coupling rod, mainly at the threaded section, tests were carried out using the advanced finite

Table 4. Switch throw resistance results

Type of Turnout	Requirements	Date of test	No of sample	Direction of point machine movement	
				„+“ [kN]	„-“ [kN]
Double Slip side a-b	max. 5.1 kN	22.08.2025	1	4.63	2.53
			2	4.52	2.62
			3	4.58	2.54
		9.01.2026	1	2.92	4.15
			2	2.97	4.11
			3	2.88	4.22
Double Slip side c-d	max. 5.1 kN	22.08.2025	1	4.93	3.89
			2	4.88	3.89
			3	4.92	3.85
		9.01.2026	1	3.45	2.85
			2	3.44	2.82
			3	3.46	2.85

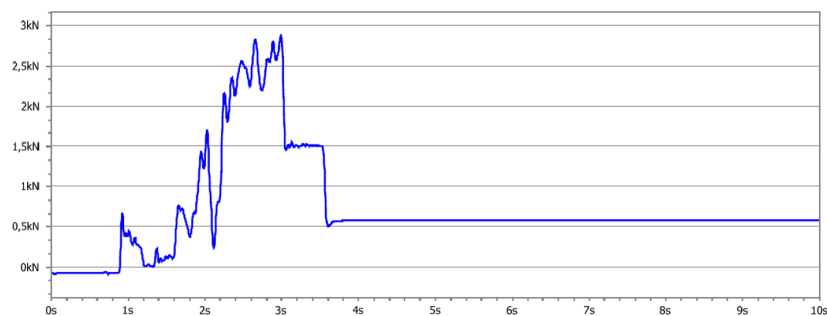


Figure 16. Example switch throw resistance graph over time



Figure 17. Switch throw resistance measurement device

element method (FEM), numerical problem solving and comparisons for 4 technological solutions:

- Straight coupling rod Ø27 mm, adjustable with dimension „z” from the stock rail foot,
- Straight coupling rod Ø27 mm, adjustable with dimension „z+10 mm” from the stock rail foot,
- Straight coupling rod Ø30 mm, adjustable with dimension „z” from the stock rail foot,

- Bent coupling rod Ø30 mm, adjustable with dimension „z” from the stock rail foot, as illustrated in Figure 18.

The numerical model was discretized using a combination of second-order tetrahedral elements and first-order hexahedral elements. The mesh appearance at the key locations is presented in Figures 19 and 20.

The contact modelling was implemented to achieve a compromise between computational efficiency and solution accuracy. Table 5 and 6 present the locations where nonlinear frictionless contact was defined, while the remaining contact regions were modelled as bonded contacts.

In order to perform a strength analysis – both static and fatigue – for the S355J2 steel rod, a bilinear stress-strain characteristic model for S355J2 steel describing the material behavior during static tension [24], shown in Figure 21,

and the SN (Stress-Number of Cycles) fatigue curve [25], shown in Figure 22 were utilized. It should be noted that the threaded region involves complex stress concentrations that cannot be strictly equated to standard structural or welded details. Therefore, these baseline parameters serve primarily for a comparative trend analysis.

In the case of the bent rod with a diameter of Ø30 mm, an M30×1.5 fine thread was used, while in the case of the straight rods with diameters of

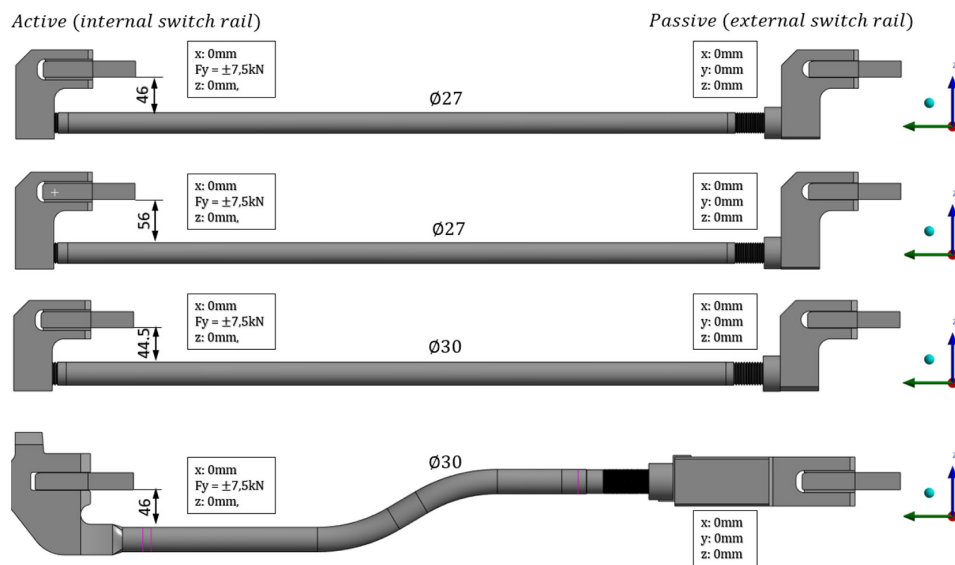


Figure 18. Variation analysis of coupling rod geometries using the finite element method

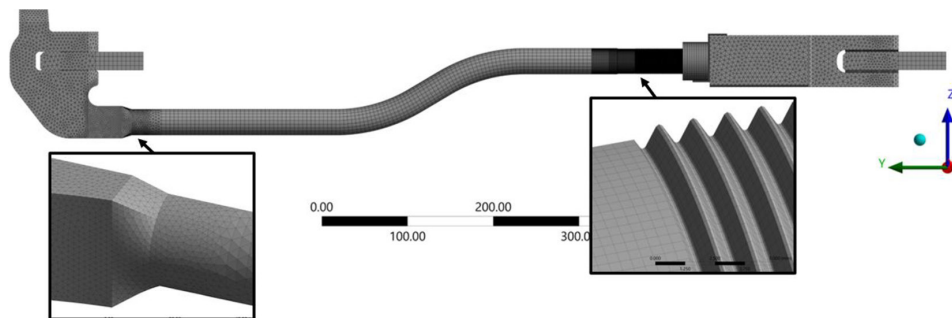


Figure 19. Representation of the mesh element for bent coupling rod

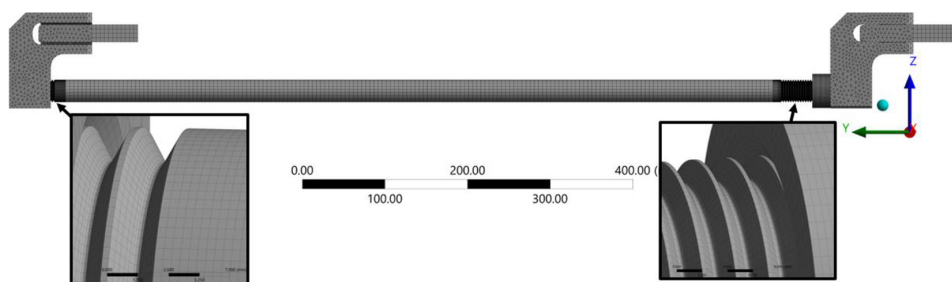


Figure 20. Representation of the mesh element for straight coupling rod

Table 5. The thread areas of frictionless contacts

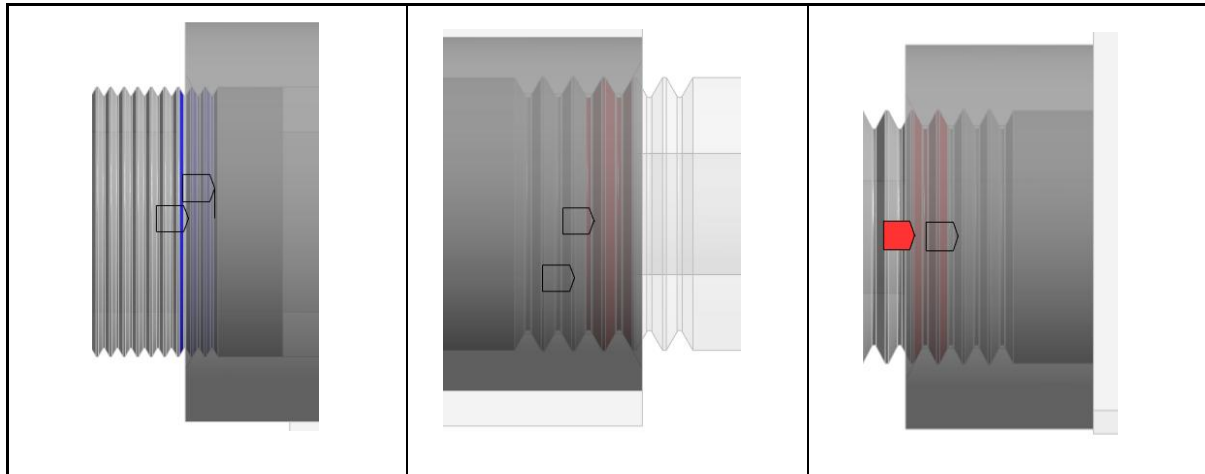


Table 6. The clamp areas of frictionless contacts

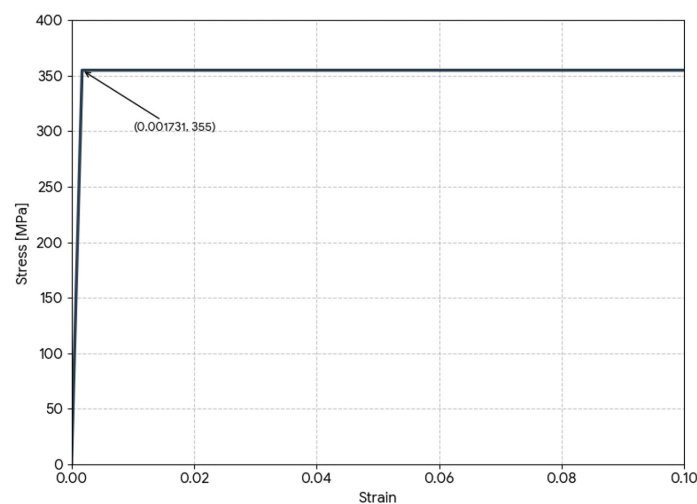
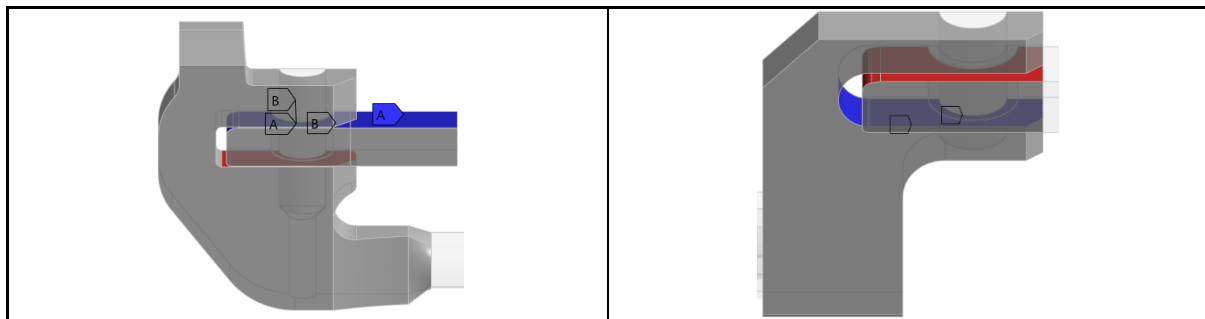


Figure 21. Bilinear stress-strain characteristic model for S355J2 steel [24]

Ø27 mm and Ø30 mm, standard metric threads M27 and M30 were used. The coupling rods were subjected to a tensile test by applying the maximum driving force from the point machine, i.e., 7.5 kN in two directions designated as „ $F_y = \pm 7.5$ kN” in Figure 18. The remaining parameters in the „x” axis and the „z” axis were assumed as

reference values. The results of the static analysis are presented in Table 7 and 8.

Based on the obtained maximum principal stress values and the S–N curve, the predicted fatigue life is estimated to be in the range of (10^5) to (10^7) cycles. An increase in the value of (z) leads to a deterioration of the results, whereas an

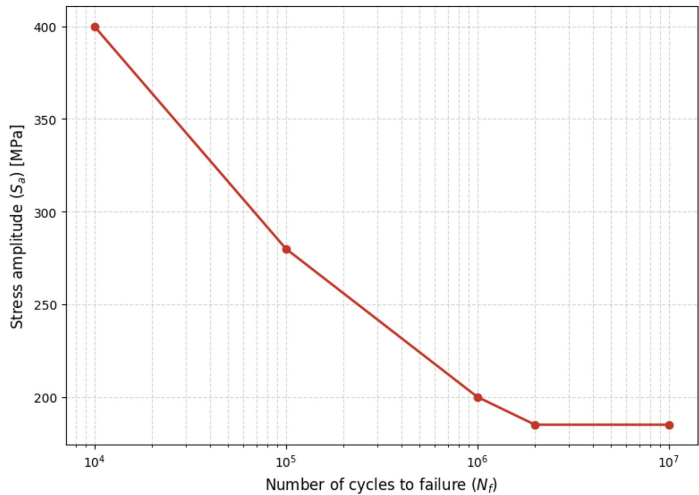
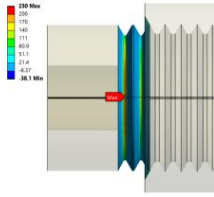
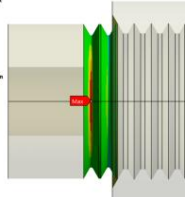
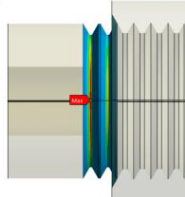
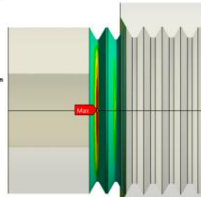
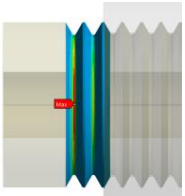
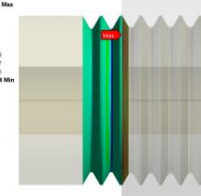
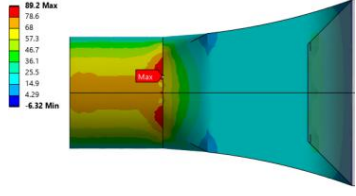
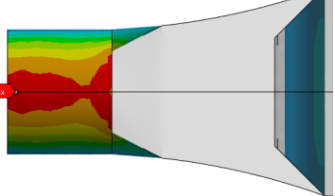


Figure 22. Fatigue curve (Wöhler / S-N curve) for S355J2 steel [25]

Table 7. Maximum stress results on the passive sides of the coupling rod

Type of coupling rod	Passive (top/bottom views)	
	+7.5kN (max throw force of point machine)	-7.5kN (max throw force of point machine)
	Mark „+“ or „-“ indicate the direction of movement of the point machine driving rod	
$\varnothing 27$ z=11 mm (46 mm)	87 MPa 	46 MPa
$\varnothing 27$ z=21 mm (56 mm)	72 MPa 	46 MPa
$\varnothing 30$ z=9.5 mm (44.5 mm)	76 MPa 	42 MPa
$\varnothing 30$ z=11 mm (46 mm)	189 MPa 	113 MPa

Table 8. Maximum stress results on the active sides of the coupling rod

Type of coupling rod	Active (top/bottom views)	
	+7.5 kN (max throw force of point machine)	-7.5 kN (max throw force of point machine)
	Mark „+“ or „-“ indicate the direction of movement of the point machine driving rod	
$\varnothing 27$ $z=11$ mm (46 mm)	230 MPa 	76 MPa 
$\varnothing 27$ $z=21$ mm (56 mm)	260 MPa 	109 MPa 
$\varnothing 30$ $z=9.5$ mm (44.5 mm)	216 MPa 	49 MPa 
$\varnothing 30$ $z=11$ mm (46 mm)	89 MPa 	54 MPa 

increase in the rod diameter has a beneficial effect on the predicted fatigue performance.

It should be noted that the boundary conditions require further refinement and validation in future studies, as the analysed system is complex and the current results should be treated primarily as a basis for a general comparison of the coupling rods rather than as an indication of specific, quantitative performance values. Therefore, the presented analysis should be regarded as a starting point for further investigations at a more detailed level.

Empirical field tests during operation

Field tests focused on assessing the impact of railway rolling stock on the operation of the coupling rod. For this purpose, a vision system was used to record the vertical movements of the stock

rail profile in the observed double slip turnout, as illustrated in Figure 23. The neutral position of the coupling rod is shown in Figure 24a, while the largest amplitude of vertical stock rail displacements was recorded during the passage of the first axles of passenger rolling stock, as illustrated in Figure 24b. The video footage (recorded via smartphone) is included for qualitative visualization of the operational movement. The vertical movement of the rail components and the reduction of the distance between the stock rail foot and the coupling rod from the base dimension of $z = 11$ mm to 5 mm resulted primarily from the maintenance condition (track tamping quality) in the locking device area. A simplified chart of the vertical displacements of the stock rail relative to the coupling rod is presented in Figure 25. The operating parameters of the point machine-turnout system complied with



Figure 23. Location of the camera setup for observing vertical movements of the stock rail in a double slip turnout during the passage of passenger rolling stock

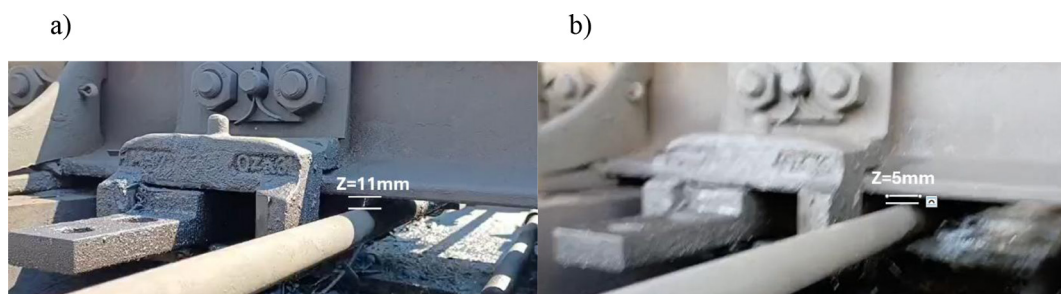


Figure 24. Coupling rod position: a) before train passage $z = 11$ mm; b) during train passage $z = 5$ mm

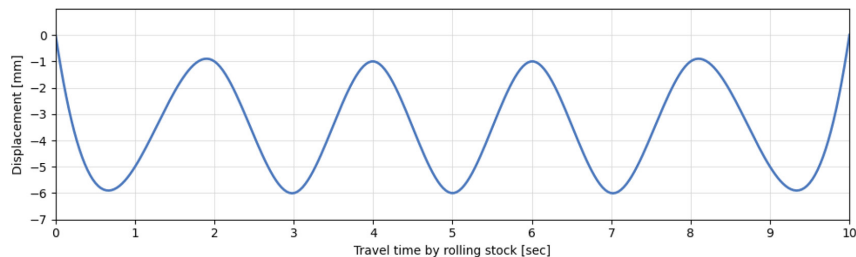


Figure 25. Simplified chart of stock rail vertical displacements relative to the coupling rod

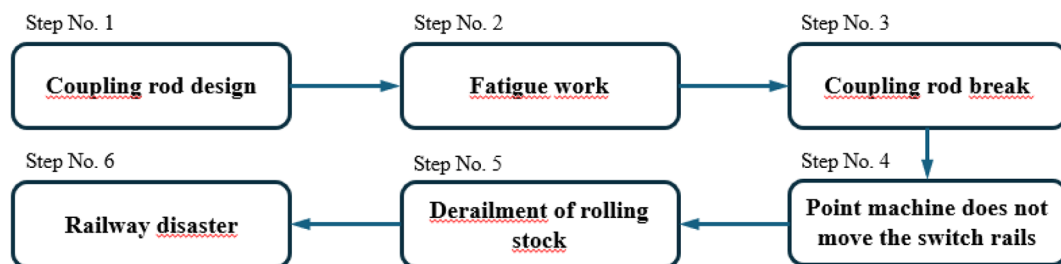


Figure 26. Development of damage from a minor defect to derailment of the rolling stock

the infrastructure manager's guidelines and did not have a significant impact on the operation of the coupling rod itself. The field tests were conducted during the passage of passenger trains with axle loads up to 180 kN and maximum speeds of

100 km/h. A total of 5 consecutive passages were recorded under clear weather conditions at an ambient temperature of 20 °C.

The vertical displacements of stock rails and switch rails [28, 29] in railway turnouts during

the passage of rolling stock are a commonly observed phenomenon. In the case of a double slip turnout, particularly in locations difficult to tamp (such as the area of the locking device), the phenomenon of uncontrolled deflections becomes critical for the correct operation of the coupling rods. Depending on the design solution, the coupling rod is subjected to the action of forces originating from the point machine, as demonstrated in the static analysis of the switch rail throwing process. This also applies to fatigue loads during operation, resulting, among other things, from the vertical displacement of rail components in the locking device area as a consequence of insufficient track tamping. Furthermore, as a result of thermal effects and the thermal elongation of the switch rail, the operating angle of the coupling rod deviates from parallelism with respect to the slide rod. This causes an increase in the forces acting on the point machine, which in extreme cases can lead to its failure to operate. Poor operating conditions of the coupling rod can result in its fracture, most frequently located within the fixed bracket or the threaded connection. This failure prevents the proper movement of the external switch rail, leading to the loss of control over its position in the turnout, incorrect setting of the route for train traffic and in a critical scenario – to the derailment of a railway vehicle and consequently, a railway transport disaster [30, 31]. The diagram of the sequence of negative event consequences is presented in Figure 26.

CONCLUSIONS

A significant contribution of the conducted research is the development of the conceptual foundations for the design and implementation of a predictive maintenance model for double slip turnouts equipped with coupling rods. Based on simulation studies, vision measurements, and wear trends, the model integrates key operating parameters, including rail elongation, operating force, movement resistance, geometry, tamping condition, environmental factors, and heating system efficiency.

The use of simulation and diagnostic data within the described model will allow for further research, which will focus on the selected coupling rod variant, including validation through comparative studies that also include turnouts with the 49E1 rail profile.

Observations of switch coupling rod operation in existing double slip turnouts, combined with the analysis of failed straight rod components over the past few years, allow for the identification of the nature and root causes of these fractures. Diagnostics of failure causes revealed the fatigue nature of the cracks, which indicates the dynamic loading of these components during operation. Particular attention is drawn to cracks located in the thread area, which are not subjected to pure tension or compression – this fact must be taken into account in strength calculations when designing new coupling rod structures.

In the course of the conducted work, the following key conclusions were formulated:

1. The primary condition for the reliable operation of a double slip turnout, including its coupling rod, is proper tamping of the turnout at the point of the locking device.
2. Creep of the double slip switch rail, especially under continuous welded rail conditions, generates higher torsional moments in the coupling rod structure.
3. Areas particularly susceptible to cracking are threaded connections, with special emphasis on the first threads.
4. The bent rod, due to the linear nature of its connection with the switch rail foot, exhibits lower stresses during the switch throwing process.
5. Lowering the straight rod by 10 mm relative to the stock rail foot does not negatively affect the physical operating properties of the coupling rod, while simultaneously eliminating the possibility of collision (contact) between the stock rail foot and the rod, which is of particular importance during the operation of the turnout on worn wooden sleepers.

Numerical simulations and field tests, including a comprehensive assessment of geometry, confirm that under normal operating conditions, the coupling rod can operate throughout the entire life cycle of the turnout with a substantially reduced risk of unexpected failure. Further research should focus on validating the operation of the coupling rod for double slip turnouts installed on pre-stressed concrete and wooden sleepers, as well as for 60E1 and 49E1 rail profiles, which will allow for increased operational durability under varying turnout geometries and maintenance processes. The developed predictive model will be used to identify operational hazards of the turnout and optimize the frequency of maintenance work.

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