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ASSESSING STEAM LOCOMOTIVE DYNAMICS AND RUNNING SAFETY BY COMPUTER SIMULATION

Summary. Steam locomotives are preserved on heritage railways and also occasionally used on mainline heritage trips, but since they are only partially balanced reciprocating piston engines, damage is made to the railway track by dynamic impact, also known as hammer blow. While causing a faster deterioration to the track on heritage railways, the steam locomotive may also cause deterioration to busy mainline tracks or tracks used by high speed trains. This raises the question whether heritage operations on mainline can be done safely and without influencing the operation of the railways. If the details of the dynamic interaction of the steam locomotive's components are examined with computerised calculations they show differences with the previous theories as the smaller components cannot be disregarded in some vibration modes. A particular narrow gauge steam locomotive Gr-319 was analyzed and it was found, that the locomotive exhibits large dynamic forces on the track, much larger than those given by design data, and the safety of the ride is impaired. Large unbalanced vibrations were found, affecting not only the fatigue resistance of the locomotive, but also influencing the crew and passengers in the train consist. Developed model and simulations were used to check several possible parameter variations of the locomotive, but the problems were found to be in the original design such that no serious improvements can be done in the space available for the running gear and therefore the running speed of the locomotive should be limited to reduce its impact upon the track. The developed theory and calculation model allows an assessment of locomotive's current design running behaviour, and can be used to evaluate possible improvements to steam locomotive dynamics.

OCENY DYNAMIKI ORAZ BEZPIECZEŃSTWA PRACY LOKOMOTYWY PAROWEJ NA PODSTAWIE SYMULACJI KOMPUTEROWEJ

Streszczenie. Parowozy traktowane są na kolei jako jej dziedzictwo. Wykorzystuje się je okazjonalnie, głównie do wycieczek. Dzieje się tak ze względu na ich częściowe wyważenie silników tłokowych, które poprzez dynamiczne oddziaływanie na tory kolejowe przyczynia się do ich uszkodzeń. Tego typu działania mogą doprowadzić również do pogorszenia się stanu głównych linii kolejowych używanych przez pociągi dużych prędkości. Stąd właśnie rodzi się pytanie, czy możliwe jest wykorzystanie dziedzictwa jakim są parowozy w taki sposób, aby było to bezpieczne i nie wpływało na funkcjonowanie współczesnej kolei. Podczas badania oddziaływania dynamicznego elementów parowej lokomotywy, wyniki otrzymywane z obliczeń komputerowych pokazują różnice w teoriach. Analizie została poddana lokomotywa parowa Gr-319. Stwierdzono, że lokomotywa wykazuje duże wartości sił dynamicznych na torze, które są o wiele większe niż podane w danych projektowych. W związku z tym bezpieczeństwo jazdy jest zmniejszone. Stwierdzone duże niesymetryczne wibracje wpływają nie tylko na wytrzymałość zmęczeniową lokomotywy, ale również mają znaczenie dla komfortu załogi i pasażerów znajdujących się w pociągu. Opracowany model i symulacje zostały wykorzystane do sprawdzenia kilku możliwych wariantów parametrów lokomotywy. Jednak problemy w pierwotnym projekcie okazały się takie, że żadne poważne ulepszenia nie mogą być wykonane w przestrzeniach dostępnych dla podwozia. Opracowana teoria i model obliczeniowy pozwalają na ocenę zachowania lokomotywy i mogą być wykorzystane do oceny ewentualnych ulepszeń jej pracy.

1. INTRODUCTION

1.1. Motivation

If steam locomotives preserved in working order are facing concerns from railway infrastructure managers about the damage they may cause to the track, this raises fears that restrictions in the future might be adopted for steam locomotives using main and high speed lines.

Many steam locomotives around the globe are preserved in museums and museum railways and some are used regularly on heritage railways, and on the mainline, on special charter trains, "retro specials" and dedicated steam events. Both the operators of the heritage railways and the infrastructure owners of the mainline are facing the fact that steam locomotives are not as "track-friendly" as diesel and electric locomotives.

All piston engines have rotational and reciprocation imbalance. The former are caused by uneven weight distribution of rotational masses and object rotation around axes, not coincident with centre of gravity, like wheelsets and coupling rods connecting different wheels, causing unbalanced centrifugal forces to be present. All of these can be corrected by static and dynamic balancing with additional balance weights on the components. Reciprocating components cause additional planar inertial forces, which act, for example, along a piston axis or can be present together with centrifugal forces to cause complex orientation of inertial forces. This will happen with the connecting rod, connecting piston rod to the driving wheelset. These forces vary over time and cannot be corrected completely by adding additional counterweights, as the inertial forces mainly act unequally, in the horizontal direction, along the longitudinal axis of the locomotive and in the vertical direction as a much smaller force. By adding additional counterweights, centrifugal forces are introduced, which are directed opposite to the large horizontal inertial forces. The unequal action of the inertial forces means that the additional counterweights create an overcorrection causing too large an inertial force in the vertical direction.

Inertial forces of the steam piston engine are similar to those exerted in diesel engines, although all the parts involved are much heavier and forces are higher. Diesel engines are suspended on flexible mounts to absorb the internal vibrations, and if direct drive, they are connected with the gearbox and wheelsets by cardan shafts thus avoiding the transmission of vibrations to bogies or track. With the reciprocating steam engine no such elastic links exist between the wheels and rail track to lessen the inertial forces in vertical direction (so called "hammer blow"). Thus reciprocating balancing is a trade-off between low "hammer blow" (small additional counterweights, but large horizontal forces still unbalanced) and low longitudinal vibrations (generous counterweights leaving small horizontal forces, although with large unbalanced vertical forces). The means that even good design will not be able to achieve full balancing, but must aim to find the optimal solution with minimum hammer blow and reasonable longitudinal vibrations, although real designs differ from the optimum. In the worst scenario, large longitudinal vibrations can be present at the same time as large hammer blow forces - and this will be shown.

There is a danger that if unchecked these balancing problems will cause heritage railway operators unnecessary track maintenance and expense. Modern mainline operation involves very strict maintenance standards and high speed operation (160 km/h and more), and with the risk of increased track deterioration from steam locomotives, line operators might choose to ban steam locomotive operation, with the loss to everyone that this would entail.

1.2. Objectives & research aim

With the power of modern day computers it has proved possible to create a model to review the design philosophy, to research the steam locomotive running dynamics, and to find the actual scale of the forces involved - something that hitherto has never been achieved. The review of the previous theory has revealed that previous work has always concentrated on the heaviest parts of the steam locomotive running gear, ignoring the smaller ones. Yet, the smaller parts are not insignificant. Also no previous work has tried to evaluate the role of primary suspension on the lessening of the vibrations transmitted to the mainframe of the engine and only vibrations measured during trials have been presented, no attempts have been made to calculate locomotive mainframe vibrations before actual locomotives are built.

This work will address a 750 mm narrow gauge steam locomotive Gr-319, built in Lokomotivfabrik Karl Marx, Babelsberg, Germany in 1951 as per drawing albums [1, 2] for the Soviet Union. Several locomotives of this class are still in regular heritage operation, particularly in Germany, Ukraine and Russia, Fig. 1.

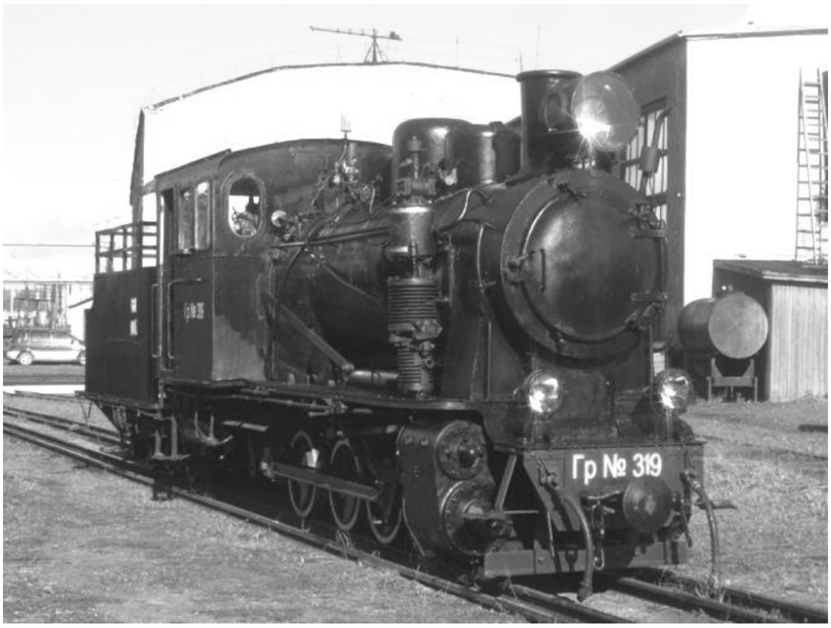

Fig. 1. Steam locomotive Gr-319
Rys. 1. Parowóz Gr-319

During the research the steam locomotive was under restoration in Gulbene - Aluksne narrow gauge railway, Latvia, led by the author, so direct measurements of the parts (size, weight, etc.) could be done to calculate the differences between drawings and the locomotive as built, and to measure the impact of these differences. Additionally the locomotive was built with the counterbalances (balance weights) of the driving wheels cast incorrectly (Fig. 2). Coupled wheels counterbalance was casted correctly – shifted clockwise to the side of the opposite wheel's crank, providing dynamical balancing. Driving wheels counterbalance was casted incorrectly – shifted counter clockwise, leaving wheels dynamically unbalanced and adding additional unbalanced inertial force couple. Calculations were made to evaluate the effect of this fault and the seriousness of the rotational unbalance caused.

Fig. 2. Comparison of coupled wheel and driving wheel counterbalances, showing that the counterbalance must be shifted clockwise

Rys. 2. Porównanie połączenia koła i koła napędowego pokazuje, że przeciwwagą musi zostać przesunięta w prawo

1.3. Hypothesis & results

The hypothesis of this work is that the actual dynamic forces exerted by steam locomotive are much more complex and at a higher level than calculated by previous theories, and that the actual forces developed by existing steam locomotives can differ from those calculated against the drawing parameters. It questions whether steam locomotives currently in operation also suffer from bad original designs that used faulty assumptions.

This work will show that modern computer simulation can provide a simple new tool, to explore ways of improving existing steam locomotives to minimise their impact on the track without extensive on-road testing.

Hammer blow forces for the narrow gauge Gr locomotive were found to be far too high, in its current state, endangering use of the locomotive at speeds higher than 25 km/h, and seriously overloading the track. This was attributed to flaws in the original design, although the confined space in the running gear leave no room for further improvements for this locomotive without serious redesign. Thus the work done offers a tool to allow improvement in steam locomotive technology and offers the means of setting safe limits on the speed of operation.

2. BACKGROUND RESEARCH

Mainstream development of steam locomotives stopped in the 1950s, so the previous design methods used had to be explored using books collected in railway museums, libraries, etc. No articles are published in this topic. The last books which mention it in some detail are dated from the 1960s. There are many books about the steam locomotive and its components, but dynamic behaviour and balancing of the steam locomotive is seldom mentioned in detail; only the general idea is described. Most of the books are written for the drivers or those repairing steam locomotives, and these people do not need to know this topic in detail, as it was the task of the designer to create a well balanced piston engine.

A simple and easy-to-understand description of the dynamic forces and balancing is given by [3], but without theory or calculations. A peculiar, but simple method of graphical determination of unbalanced forces is described in [4], which represents the technology of state-of-the-art in 1930s. A book of approximately the same age [5] gives the deepest available insight in the nature of dynamic forces and the theory behind them. It represents the different levels of scientific approaches also in different countries. The second most detailed research is described in [6].

All of these sources above state that complete balancing cannot be done and that it is a particular task to find the combination of design parameters with the lowest overall misbalance of vertical "hammer blow" and horizontal vibrations, as these parameters are generally contrary to each other.

Chapelon [7] stressed that attention to this topic was paid "in the United States where loss of track alignment and damage to the road bed has occurred with high speed running, due to excessive rotating overbalance".

Porta [8] stressed that modern steam locomotives, especially with crank axles should not be designed without adequate balancing, and Wardale [9] proposed improved balancing for existing QJ class steam locomotives, while admitting that "existing computer techniques for predicting railway vehicle stability would not be suitable for steam locomotives because of the radically different nature of a steam locomotive's chassis compared to the usual double bogie arrangement".

The only practical method of measurement of wheel counterweights is described in [10], this is the method also used during this work to measure the exact misbalance, and to compare it with the calculated.

Even if the theory of balancing of steam locomotives can be found in literature, almost no details are given of the measurement tools or methods used to check the steam locomotives after construction or general overhaul. In the author's opinion this explains why steam locomotives were not successfully balanced, and why the tools were not widely available.

3. THEORY

Previous knowledge of the fundamental theory describing steam locomotive dynamics is underdeveloped, so the theory and model (Fig. 3) was developed from scratch, using these principles:

- all components (parts in motion) are described, except for those parts that are stationary and do not exert dynamical forces such as components of the valve gear. This is in contrast to previous work, where only the largest and heaviest components (pistons, crossheads, connecting and coupled rods) were analyzed;
- all trajectories of moving parts are evaluated using the real geometrical sizes, mechanical link types and associated constraints from them. No simplification is made;
- linear and angular accelerations were calculated from the geometrical trajectories of parts. These were transformed into inertial forces using known weights and the inertial moments calculated;
- all dynamical interactions between parts in the links were evaluated, including the valve gear, without any simplifications.

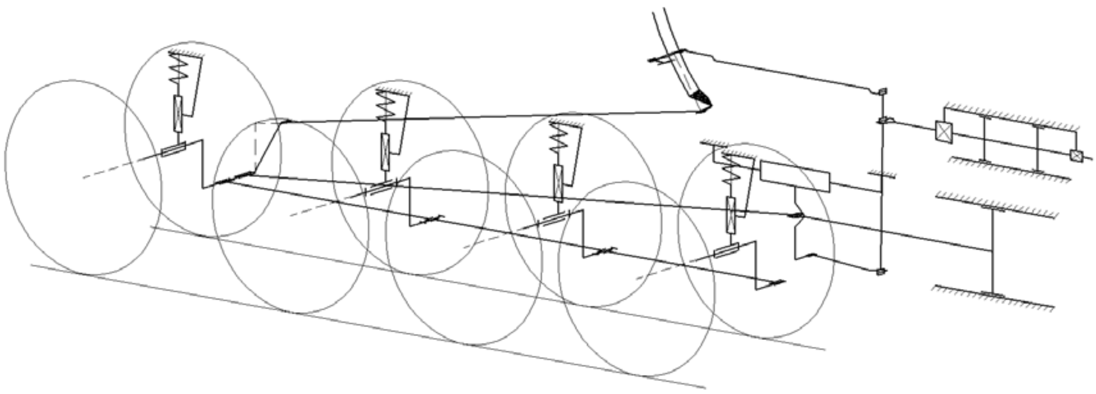

Fig. 3. Half model of the steam locomotive's Gr-319 running gear and piston engine, valve gear (for clarity, all wheels are shown)

Rys. 3. Połowiczny model podwozia i silnika tłokowego rozrządu parowozu Gr-319 (dla jasności, wszystkie koła zostały pokazane)

Motion of the steam locomotive is investigated at a constant maximum allowed speed (35 km/h for Gr-319, giving rotational speed of 232 rpm), at this speed the inertial forces and force couples are the largest. Small variations of the running speed are disregarded, as the theory states that a mechanical system, which is in balance at the constant speed, will remain in balance also during a change of speed.

Calculated inertial forces from the piston engine actually cause small variations of the locomotive running speed, as the wheels paired with rails represent a frictional reaction base, but this creates a recursive link in the calculations, as these small variations induce further inertial forces and geometrical movements. To simplify the already complex calculation model it was assumed that these variations are negligible.

The motion of the steam locomotive is evaluated under coasting conditions. Only inertial forces are analysed and balanced; no traction, braking or steam pressure forces are evaluated, as these forces are not constant, and can have different orientations. This is in opposition to internal combustion engines in diesel locomotives that run mainly in nominal power mode and only in one direction.

Balancing is carried out in mid-gear, as incorrect balancing against one of the end positions would cause double misbalance in the opposite valve gear position.

As the task was to explore the dynamic behaviour of the steam locomotive where inertial forces play the main role, track was assumed to be flat and straight. Vertical stiffness rate of railway track was assumed as being infinite (i.e. track is absolutely stiff), as the steam locomotive's plate spring's stiffness rate is more than 19 times lower. As was later found during the calculations, no wheelset flange-rail contact is established during coasting, so horizontal track stiffness was not evaluated.

No gravitational forces of the running gear were evaluated; only the gravitational forces of the wheelsets and the locomotive’s upper structure (frame, boiler, etc) were evaluated, as the inertial forces in the running gear far exceed the gravitational forces, as will be shown.

4. METHODS & MATERIALS

Kinematics and dynamics of the steam locomotive running gear were evaluated by using PTC MathCAD software, due to its user friendly interface and minimum learning curve. In total 51 bodies were included in the calculation, with 80 kinematic and dynamic interaction points, 134 forces and 15 force couples evaluated.

The amount of bodies and forces involved makes it complicated to solve the task in differential equation form, and to find an analytical solution to the problem. As PTC MathCAD is a general purpose mathematical software, and not a multi-body analysis tool, it was chosen to solve the task iteratively, by using conventional kinematic equations for determination of body coordinates, speeds and accelerations, and dynamic equations for body equilibrium, employing inertial forces and force couples known from the kinematic results. The resulting unbalanced forces applied to the locomotive upper structure are recalculated as unbalanced chassis accelerations, by using the D'Alembert's principle.

Exact algorithm consists of:

• Assumption of start time t_{sak} and time increment value ξ, so that time at any calculation step is determined as (i – current iteration step, starting from zero):

$$t = t_{sak} + i \cdot \xi \tag{1}$$

• Assumption of starting coordinates for all bodies, as per geometrical data of components and current chassis coordinates (assumed zero for first five iterations, to enable recognition of acceleration, as stated below):

$$x_{AK},\, y_{AK},\, \varphi_{AK},\, \psi_{AK},\, \theta_{AK} \tag{2}$$

• For first five iterations also chassis velocities (linear or angular) around axis are assumed as zero, as they are unknown;

• By using given starting coordinates (2), time as per (1) and given constant speed of rotation of wheelsets, coordinates for all bodies and kinematic pairs are determined;

• By using data from last five iteration steps (amount of data points established as minimum necessary to give approximation error less than 1%), an cubic spline approximation is carried out on the coordinates of body centres of gravity. The approximated equation allows to carry out second order derivation and to recognize the acceleration of the bodies for the last calculation step;

• Acceleration data derived is feed to dynamical equations, to find all interaction forces and also unbalanced forces transferred to track and chassis. By using the D'Alembert's principle, linear and angular acceleration of chassis centre of gravity $ax_{AK}, ay_{AK}, e\varphi_{AK}, e\psi_{AK}, e\theta_{AK}$ is determined;

• Chassis coordinate bx and velocity vx at the end of the iteration are determined as (shown for one mode of freedom):

$$bx_{AK} = x_{AK} + vxI_{AK} \cdot \xi + \frac{ax_{AK} \cdot \xi^2}{2} \tag{3}$$

$$vx_{AK} = vxI_{AK} + ax_{AK} \cdot \xi \tag{4}$$

• Results obtained are fed as input data for next iteration, with a new time value as per (1), and calculation is restarted, until a predetermined count if iterations are finished.

To optimize memory allocation, calculation was divided across two files – “master controller” and “model”, “master” being responsible for iteration loop control, cubic spline approximation and collection of results, while the “model” carried out the whole kinematic and dynamic calculations for a given iteration. In this way results for chassis and forces transferred to track where collected for all iterations, but only the necessary amount of data for five iterations was kept for the rest of the bodies, to carry out the cubic spline approximation. Calculations results were saved up to 5 significant digits. Internal calculations in the “model” were carried out up to 12 digits.

The measurement data from the drawing albums (Lova, 1951) was found to be accurate, and so it was used. The weight of the parts was measured with an accuracy of 0,1 kg, except for the wheelsets, which were weighted with an accuracy of 0,5 kg. All parts were weighted in actual assembly state, with all bolts and details attached.

Serious differences were found between the weight indicated in drawings and that measured in reality, for example the coupled wheelsets weighed 15% or 127 kg less than shown and the driving wheelset was 10% or 97 kg lighter than indicated in the drawings. Connecting rods were found to be 30% lighter than indicated in the drawings. This inaccuracy may be explained by faulty weighing in 1951 during construction or rough estimation, as the drawing dimensions are correct. The outcome of the mistake was to cause balance weights of the wrong value to be fitted. To explore these deviations several cases for calculations were defined:

• Case A (default): calculation of the locomotive dynamics according to the actual component weight measured, with the counterbalance of driving wheels cast incorrectly;

• Case B: calculation of the locomotive dynamics according to the actual component weight measured, except with the counterbalance cast correctly;

• Case C: calculation of the locomotive dynamics according to the weight indicated in the drawings, with the counterbalance cast correctly;

Misbalance at the crank pin of the wheelset's counterbalance weights was determined as described in [10]. This implies placing the wheelset on a special, yet simple rolling rig, which can be made on place, and levelling the wheelset by attaching balance weights to one side of the wheelset and measuring the balance weight necessary to keep the opposite crank in vertical plane, so that the wheelset's opposite side counterbalance weight cannot influence the side being measured (the lever distance is zero). Actual accuracy with the friction losses during the setting out of the vertical level was measured in the range 0,2...0,3 kg.

As no special measurement tools were available to measure the angle between the crank and the counterbalance, it was taken from the drawings, as no large differences of the overall sizes were found.

As there are limited methods available to measure the moment of inertia, and as the details to be measured range in sizes up to 3 m, and with weight up to 106 kg, the moment of the inertia was calculated theoretically from the sizes of the parts, as most of the details (rods, parts of the valve gear) are close to the form of solid rod. Moments of inertia of the upper structure (mainframe, boiler, etc.) were calculated using formula for cuboid of the overall size of locomotive. The centre of gravity of the parts was obtained with a balancing accuracy of 2 mm.

The stiffness rate of the steam locomotives springs was measured after the springs were repaired and lubricated, but still, serious differences were found. As all springs are interchangeable, the average value was used for the calculations.

5. INTERMEDIATE CORRECTIONS OF THE CALCULATIONS

As the calculations were carried out in the steady running state, the initial conditions of locomotive mainframe speed was set the same as the locomotive running speed and the acceleration was assumed to be zero, although calculations for the first revolutions of the wheel revealed that the initial conditions were assumed incorrectly. In some modes of vibrations (longitudinal and yaw) calculated results consisted of vibrations superimposed over a movement of constant speed (e.g. resulting in an ever increasing yaw angle). With these results the initial conditions - mainframe longitudinal movement speed deviation and yaw rotation speed - were calculated, and the calculations repeated.

Another effect to be accounted for was the resonance of the mainframe on the springs in the vertical direction, as initially no damping was accounted for and the inertial forces were exerted with a frequency close to the resonance frequency. Calculated frictional forces of the plate springs were found to be very large, and also resulted in a statically undetermined system of eight friction forces, so they were not implemented, instead friction forces between the axle box liners and mainframe pedestals were included, as being statically determined by the horizontal inertial forces pressing the axle boxes against the front or the back pedestals.

First results of the calculations were compared with results given in [5] but as no exact results of the locomotive under research were available, only the overall mode of vibrations (Fig. 4) and the magnitude of accelerations, forces in respect of the size and weight of the locomotive were evaluated, and found to be in good relation.

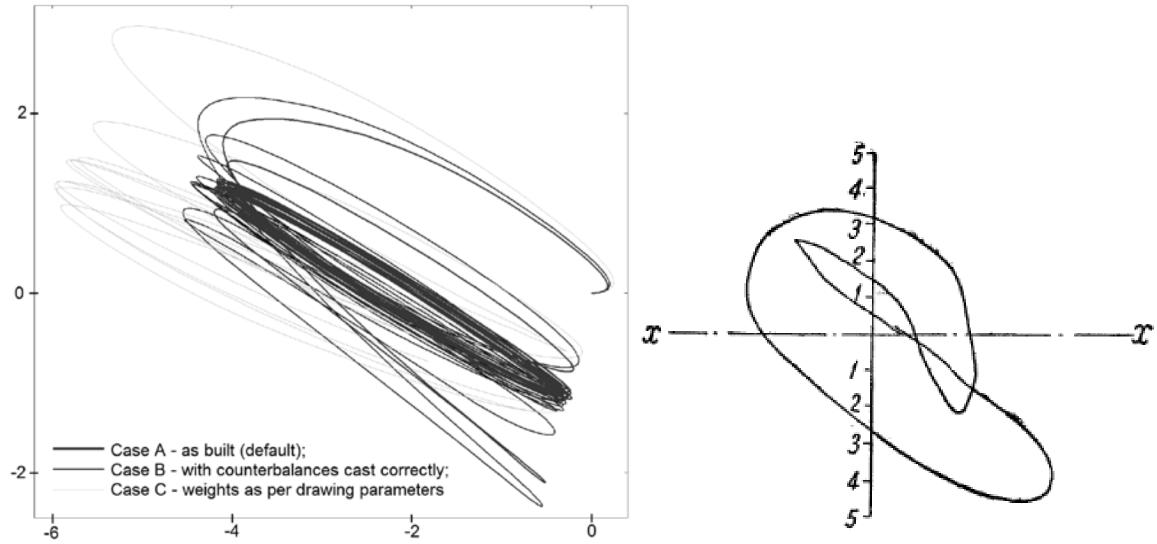

Fig. 4. Left – calculated trajectory of the driver's seat along x and y axis (mm), right – for comparison data from locomotive E Nr. 5570, as given in [5]

Rys. 4. Lewy – obliczona trajektoria fotela kierowcy wzdłuż osi x i y (mm), po prawej – dla porównania dane z lokomotywy E Nr. 5570, jak przedstawiono w [5]

Calculation time step was evaluated in the range 0,001 - 0,012 s (Fig. 5), it was found, that differences between results in the time step range 0,001 - 0,004 s were not negligible (~15%), although the overall results of vibrations were correlating around the same values and the differences were cyclically cancelling each other. To speed up the calculations, a time step 0,004 s was chosen, as each iteration step lasted ~30 seconds on a 3,07 GHz CPU.

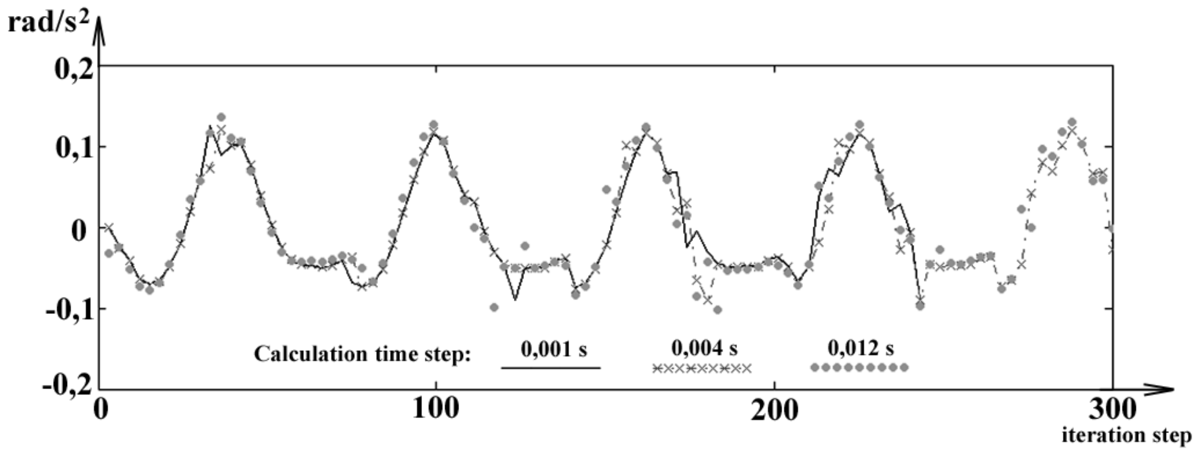

Fig. 5. Angular acceleration of the locomotive upper structure (roll vibrations), rad/s^2
Rys. 5. Przyspieszenie kątowe górnej struktury lokomotywy (wibracje rolki), rad/s^2

6. RESULTS

Final calculations were carried out with a calculation step of 0,004 seconds, and were made for 1519 steps, which represent more than 24 complete wheel revolutions, giving more precise results than a time step of 0,012 s, although at a more reasonable speed than time step of 0,001 s. Results are summarized in Table 1.

Pitch vibrations show unstable oscillations at the first revolutions of the wheel that gradually disappear. This can be attributed to the unknown initial conditions and can be explained by a decrease in the natural oscillations, leaving only forced oscillations in further steps of the calculations. Roll vibrations show a similar trend, with slowly fading natural oscillations, which do not fade completely during the period of calculations. It should be pointed out that the remaining oscillations are almost stabilized. In the table only the amplitude and acceleration of stabilized oscillations are given.

The main outcome is the large amplitude of chassis vibrations, and the most important outcome is the high level of hammer blow - wheel overload/under load, as determined by deviation from static reaction forces from the rail track. This is the most harmful force transmitted by a locomotive to the track. The largest acceleration of running gear parts was found for the piston, rod and crosshead - 134 m/s^2 (equal to 13,7 g), proving that weight of running gear components is insignificant in comparison to inertial forces. The largest angular acceleration was found for the expansion link - 390 rad/s^2.

To compare this work with previous theories, a comparison calculation was made taking into account only the largest components (pistons, crossheads, connecting and coupling rods) - four of the vibration modes explored showed small differences (from 3,2 to 6,1%), although the roll mode showed a difference of 14,5%, which cannot be assumed to be negligible.

Trajectories of different locomotive points give visual representations of the complex vibrations exerted to the upper structure. For example, the centre of gravity trajectory shows the resulting movements of longitudinal and vertical oscillations (Fig. 6), while the trajectory of the driver's seat shows the resulting movements of longitudinal, vertical and pitch oscillations (Fig. 4). These trajectories also show that natural oscillations fade away and the trajectory stabilizes leaving only the forced oscillations.

Table 1

Results of calculations for steam locomotive Gr-319

Parameter	Case A (default)	Case B	Case C
Longitudinal vibrations			
Amplitude of oscillations, maximum, mm	±2,40	±2,50	±3,17
Acceleration rate of mainframe, maximum, m/s^2	+1,45/-1,78	+1,56/-1,93	+1,96/-2,52
Unbalanced inertial force, maximum, kN	46,5	50,4	65,8
Vertical vibrations			
Amplitude of oscillations, maximum, mm	±0,612 [a]	±0,681 [a]	±0,559 [a]
Acceleration rate of mainframe, maximum, m/s^2	+0,498/-0,702	+0,507/-0,553	+0,719/-0,664
Unbalanced inertial force, maximum, kN	15,9	12,6	16,3
Pitch vibrations			
Amplitude of oscillations, maximum, rad	$\pm 6{,}16 \cdot 10^{-4}$	$\pm 6{,}60 \cdot 10^{-4}$	$\pm 6{,}78 \cdot 10^{-4}$
Angular acceleration rate of mainframe, rad/s^2	±0,365	+0,416/-0,439	+0,443/-0,428
Unbalanced inertial force couple, kN·m	47,1	56,6	57,1
Roll vibrations			
Amplitude of oscillations, maximum, rad	$\pm 2{,}22 \cdot 10^{-4}$	$\pm 2{,}13 \cdot 10^{-4}$	$\pm 2{,}57 \cdot 10^{-4}$
Angular acceleration rate of mainframe, rad/s^2	+0,131/-0,096	+0,131/-0,094	+0,162/-0,121
Unbalanced inertial force couple, kN·m	4,27	4,27	5,28
Yaw vibrations			
Amplitude of oscillations, maximum, rad	$\pm 4{,}06 \cdot 10^{-4}$	$\pm 3{,}45 \cdot 10^{-4}$	$\pm 4{,}32 \cdot 10^{-4}$
Angular acceleration rate of mainframe, rad/s^2	+0,229/-0,224	+0,22/-0,215	+0,274/-0,269
Unbalanced inertial force couple, kN·m	31,4	30,1	37,5
Reaction forces from the rail track (deviations from the static force)			
First coupled wheelset, left side, %	+17,5/-17,2	+19,8/-19,9	+14,8/-13,8
First coupled wheelset, right side, %	+7,9 / -8,4	+12,2/-12,3	+10,8/-8,1
Last coupled wheelset (driving), left side,%	+33,2/-26,7	+32,1/-30,4	+37,2/-40,0
Last coupled wheelset (driving), right side, %	+30,8/-27,1	+33,2/-28,8	+44,3/-38,9

[a] amplitude calculated is below the sensitivity level (1,49 mm) of plate springs due to friction.

7. DISCUSSION

Vertical vibrations were found to be less than the sensitivity level of plate springs (calculated as 1,49 mm). As a result, inertial forces would be transferred through springs without damping because of the "friction lock" of the plate springs, so actual vertical unbalanced forces would be even higher than the calculated (15,9 kN), as springs were assumed elastic in calculations.

The unbalanced longitudinal forces (calculated as 46,5 kN) reflect 18% of the locomotive's weight. The amplitude of longitudinal vibrations that result (±2,40 mm) are transmitted via the drawgear to the carriages and cause discomfort to the passengers, but it is the loco crew who are exposed to the highest level of overall vibrations.

The calculated maximum yaw angles at the maximum speed lead to lateral displacement of not more than 0,591mm for the far most wheelsets. This does not lead to flange to rail contact, but under traction or at other speeds the swaying couple of the traction forces may offer different outcomes.

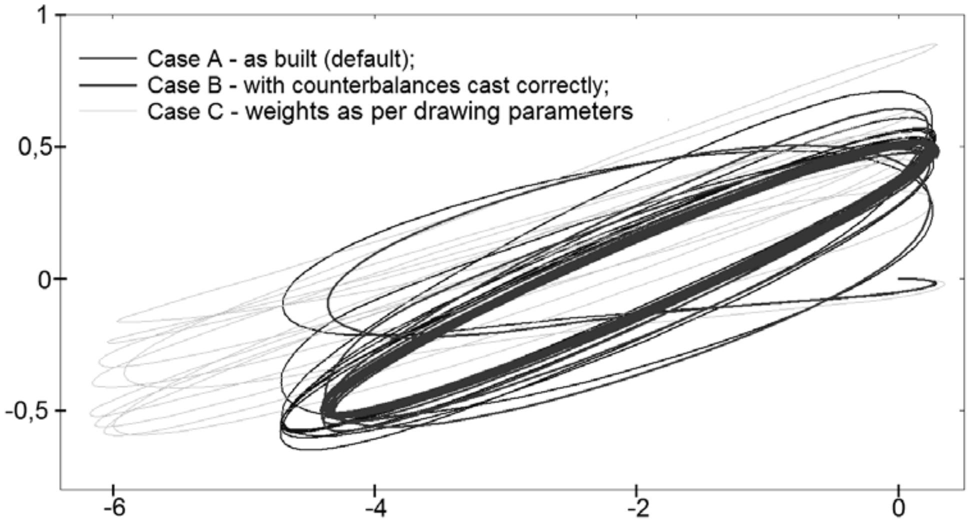

Fig. 6. Locomotive's centre of gravity trajectory along x and y axis, mm (start of calculations – point 0, 0)
Rys. 6. Trajektoria środka ciężkości lokomotywy wzdłuż osi x i y, mm (początek obliczeń – punkt 0, 0)

The maximum hammer-blow forces for the actual condition of locomotive reaches 33,2% overload and 27,1% unload; with an axle load of 6,70 t, the calculated wheel load varies between 8,92 and 4,88 tonnes. So the locomotive even exceeds the line axle load limit of 8 t/axle.

Apart from exceeding the axle load limit of the line, the calculated deviations also exceed design recommendations [5, 6, 9], that the deviation of the coupled wheels shouldn't exceed 30..35%, except for the leading and the trailing coupled wheels, for which a lower value of 20..25% is allowable. As can be seen, the trailing wheelset, which at the same time is the driving wheelset, is the most over/unloaded, exceeding the recommended rate and giving a warning that locomotive must run in reverse with caution, as the increased unloading of wheelset might lead to wheel climb and derailment of the locomotive.

To prove the hypothesis that actual forces are much higher than calculated by previous theories, calculated results are compared with the data given by the builder in the drawing album [1]. In introduction of the drawing album [1] it is stated, that the piston engine is very well balanced and actual vertical misbalance of wheelsets do not exceed 15%.

Additional calculations were carried out according with the cases B (with counterbalances cast correctly) and C (weights as per drawing parameters), but these show even worse results. In case B, as the correct alignment of counterweights would improve cross balancing of the locomotive, smaller yaw mode vibration amplitude is observed (-15,0%), as anticipated, although pitch vibrations increase by 7,1% and longitudinal oscillations increase by 4,2% and track reaction force deviations are found to be 33,2% overload (the same as in case A) and 30,4% under load (+12,2% increase in comparison to case A).

In case C with track reaction force deviations rising to 44,3% overload (+33,4% increase in comparison to case A) and -40,0% unload (+47,6% increase), it can be concluded, that if the locomotive would have been made fully according to the weight indicated in the drawings, even worse running behaviour would be experienced, showing that theory of locomotive balancing used by the designers had serious flaws.

To assess possible improvements of the running behaviour and to maintain a safe ride, the author found that the first coupled wheelset had excessive counterbalances which cause the excessive hammer blow, as the counterbalances were cast in the same size as for the second and for the third wheelset, and this could be easily corrected by lightening the counterbalances (creating hollow volumes in the counterbalances by boring), although the second and third wheelset was found to have a too little counterbalance weight, giving complete rotational balance and thus small hammer blow, but insufficient reciprocation balance, thus contributing to the large longitudinal vibrations of the locomotive.

The last coupled wheelset (driving wheelset) was seriously underbalanced. Large hammer blow excited by the last wheelset can be attributed to the unbalanced inertial forces of the piston engine, not because of excessive counterbalances, as the vertical reaction force is phase shifted in comparison to all other wheelsets. This in turn is also a reason for large horizontal forces being unbalanced and causing large longitudinal vibrations, together with second and third wheelsets. Additional counterbalance weight cannot be added to the last wheelset without redesigning the whole mainframe and rod assembly because of the constrained space (moving cylinder blocks further apart for additional space could be one of the solutions, although complicated), and other means of balancing (boring holes and filling them with lead, etc.) where found to correct only very small proportion of the missing counterbalance weight.

To approve safety of the ride, additional calculations were made according to the case A with the speed lowered to 25 km/h, obtaining that deviations of the track reaction forces at the driving wheelset decreased to +23,9%/-21,0%, which are within the recommended range of 20...25% [5, 6], concluding that safe ride in backward direction, especially in curves should be done with speed limited to 25 km/h. With speed limited hammer blow would be lowered also for the first wheelset, thus eliminating need for modifications, and no modifications were found to be practical for the second and third wheelset due to the confined space. Calculations for the speed of 25 km/h also illustrate the complex nature of vibrations - if overall unbalanced forces exerted in the running gear are calculated with speed as a factor in square, then lowering speed from 35 km/h to 25 km/h should result in lowering of driving wheelset overload from 33,2% to 16,9%, in fact, it lowers only to 23,9%, due to pitch vibrations adding additional load to the wheelsets, these vibrations are higher in 25 kph than in 35 kph.

All calculations were made using MathCAD, although during the calculations it was found, that MathCAD is not well suitable for recursive calculations and for the amount of data being used and for further work other calculation environments should be evaluated, this should also improve the speed of calculations.

Work was carried out on a small narrow gauge locomotive with a design speed of only 35 km/h, although with the driving wheels 0,8 m in diameter this corresponds to 232 revolutions per minute, which represent the same angular speed as, for example, standard gauge express passenger locomotive with 2,03 m driving wheels travelling at 89 km/h.

Although these results cannot be directly related to other steam locomotives of other sizes and dimensions, results prove that steam locomotives with serious design faults were built and are in use, so caution must be exercised, although "hammer blow" is hard to be noticed in service and no conclusions of a given locomotive type can be given without detailed examination and simulation of the locomotives design, indirect conclusions can be made from track deterioration and increased wear of parts only if locomotive is used regularly. It is known that the railway track slowly deteriorates under the load of any train, although as higher the forces, as faster it does, and the peak forces exerted only by few rare wheelsets (for example, with a flat wheel, developing similar under/overloading forces) are the most dangerous.

This research showed that modern computer simulation technique can be used to calculate exact dynamic forces of steam locomotives, although the calculations are recursive and carried out only in one way (from input data to results), complicating the task of running gear parameter optimisation (manual change of input parameters has to be carried out and calculations repeated), so further work should be necessary to develop analytical solutions, more suitable for automatic optimisation tasks, after which the final parameters can be profoundly verified using the theory and model developed in this work. Also further development of present model could imply modelling of plate springs with non static frictional coefficient, thus more exactly modelling vibrations of locomotive upper structure.

8. CONCLUSIONS

Steam locomotive running dynamics can be modelled using computer simulation. Patterns of results obtained coincide with data from real measurements for other locomotives, although no measurements are available for the locomotive type under question.

Although steam locomotives of this type have been in operation for more than 60 years, it was found that the dynamic behaviour of the original design is faulty, necessary level of balancing has not been achieved, as shown by the increased "hammer blow" or deviation from static reaction forces from rail track reach as much as 33,2% overload and 27,1% under load. As the driving wheels are also the last coupled wheels, leading the locomotive when running tender first, a speed restriction of 25 km/h in backwards running has to be implied to safeguard running both in tangent and especially in curved track.

Although steam locomotives are no longer used in mainline freight and passenger services, many steam locomotives are preserved in working condition and are used for heritage trains also on the mainline. Developed theory and calculation model allows assessment of locomotive's current design and can be used to evaluate possible improvements to the locomotive's dynamics - influence of modified component weight, corrected balance weight on the overall running behaviour.

As showed by the simulation, not only the driver, fireman and the overall locomotive components are exerted by large vibrations, but there are also large unbalanced longitudinal vibrations left to be transferred through the drawgear to the first carriages, affecting passengers ride comfort of those on the train.

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