MACHINERY RESISTANCE IN STEAM LOCOMOTIVE TESTING AND THE ALTERNATING (OR FORE AND AFT) FORCE ALONG TRAINS GENERATED BY MOST STEAM LOCOMOTIVES

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This note is about the two subjects given in the title.

(1) MACHINERY RESISTANCE IN STEAM LOCOMOTIVE TESTING

For over a century, Machinery Resistance (MR) has been wrongly treated in reporting the results of, most locomotive testing, or was ignored in such results, no doubt because it is not directly measured. This started in the book by Prof W M Goss of 1907, *Locomotive Performance,* Chapter XIX, entitled *Machine Friction*, which chapter opens with the statement that on a Test Plant (TP), the only loss is machinery, meaning that the only loss in energy on such a plant is that between the ITE (indicated tractive effort, the effort in the cylinders), and that at the rear of the locomotive, the pull on the tender, the drawbar tractive effort (DTE)(on a TP, no tender is attached, the locomotive obtains its fuel and water from facilities in the plant, from which consumption is readily measured). There is no doubt that MR is part of that difference, but there is at least one other resistance in that difference. (On the road, the DTE is measured at the tender to train coupling).

Such machinery resistance (MR) losses arise from piston thrusts, friction among the parts, and in the work in movement and rotation of many parts, some even in opposing the work being done elsewhere. More detail follows below. Goss was wrong. In fact, as explained below, the MR as quoted by Goss includes the resistance of the coupled wheel bearings (CWBs). That is because on a TP the coupled wheels revolve because they are driven as part of the locomotive, and their bearings bear vertically much of the mass of the locomotive and horizontally the thrusts from the pistons, just as they do on the road, the two resolving to a single force on each stroke as vertical and horizontal forces acting simultaneously on a pin or bearing do.

Compared to road testing, where the DTE is measured at the tender drawbar, and made equivalent to running on the level, there is the resistance of moving the tender on the equivalent of level track. This note concerns TP results, however.

In addition, if the TP has been designed with the intention or hope of smoothing out or dampening the fluctuations in the DTE, those which can result from alternating horizontal forces (or fore and aft forces (F&A), or rake), felt in trains hauled by many steam locomotives (discussed below), if successful the dampening devices will absorb some of the energy produced at the CW rims, which will then be considered to be additional resistance between ITE and DTE, or apparent MR.

There is no need to include a resistance for the revolution of the journal in the bearing. That is already included in the resistance of rotating the journal.

Things are no better in the article of 1943 by Lawford H Fry, *Locomotive Machine Friction*, p 201 of the May 1943 *Railway Mechanical Engineer*. In addition, Fry adopted questionable statistical techniques to analyse the TP data, and considers the MR to be in part a multiple of adhesive weight and number of coupled wheels, which are interconnected variables so far as explaining MR is concerned, therefore potentially intercorrelated, so his estimating equation is invalid.

I confess to having done the same in the past as the above authors, influenced by Goss, and by the results of the UK Rugby Testing Station having been declared in terms of WR (coupled wheel rim) HP and indicated (cylinder) HP, which, reduced to TE (tractive effort) in each case, invite the conclusion that MR is ITE less WRTE. More recently, Doug Landau has placed data from results from the UK Rugby TP calculated in these terms (i.e. ITE less drawbar pull as MR) on the website of the Railway Performance Society in the UK, making the same error.

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Goss concluded that MR is approximately 5 lbs mean effective pressure (MEP), or 600 lbs at the drawbar of his test locomotive, approximately constant at all speeds, but falling with Cut Off. That falling off might indicate that it partly results from intensity of thrusts and speed of operation of the machinery. This was for a 4-4-0 of 1891, weighing without tender 38 long tons.

Goss remarks that the readings of DTE and ITE need to be accurate to obtain accurate readings of MR, a remark which confirms that he considers MR to be the difference between them. Goss was a pioneer of TPs, his TP at Purdue being a pioneer plant.

A summary of the forces concerned on a TP is:

ITE – MR – CWBR = CWRTE = DP

which assumes that the CWRTE is transmitted through the frames to the drawbar pull dynamometer on the rear of the locomotive without further loss than that incurred in MR and CWBR.

Although CWBR would usually be considered an element of the vehicle resistance (VR) of the locomotive, those bearings are on the routes of transmission from ITE to CWRTE, and from CWRTE to DP, and on a TP the CWBs carry some of the mass of the locomotive as well as incurring much of the MR. It is therefore impossible to ignore CWBR in obtaining the value of DP (and CWRTE). The Goss idea of obtaining MR inevitably results in obtaining CWBR as well, but not separately. In turn, obtaining CWBR in its own right is a matter of estimation, which requires knowledge of the mass of the coupled wheel and axle sets including rods, the CW bearing dimensions, the mass carried by each bearing, the resulting pressure [(the mass carried by the bearing)/(area of the journal)] (the area of the bearing taken as the area of the journal) and the appropriate coefficient of friction (Cf) for that pressure, plus a modest addition for the speed of rotation of the bearing obtained from the Davis formula (see below).

On a TP, the CWBs therefore resist two different types of force. One is that of carrying vertically some of the mass of the locomotive (on a locomotive with no carrying wheels, all of it, on most locomotives that not borne by the carrying wheels, i.e. that borne by the coupled wheels). The other is those forces from the piston thrusts which are conveyed to those bearings by the machinery and reach those bearings close to horizontally. The MR is the net effect of the resolution of the near horizontal forces, and the vertical forces.

What Goss calls MR is actually therefore MR + net CWBR. The mass of each coupled wheel and axle set of 63 ins diameter on his test locomotive has been taken as 4 tons. The weight of the locomotive carried by the CW bearings is 25 long tons. Each bearing therefore supports 6.25 tons; with 7 x 8 ins journal area the bearing pressure is 240 lbs/sq in, at which the coefficient of friction is .012, which is the minimum value (obtained from evidence of declared resistance of vehicles on many railways), for the vertical element of CWBR. There is also a part of the resistance varying with speed, which from the Davis formula, reduces to .0336VW, V in mph and W in long tons. The speed component of CWBR is then 90 lbs at 20 mph, 110 at 40 and 125 at 60 mph. As Goss's apparent MR of about 600 lbs at the drawbar, clearly includes CWBRV, V for vertical, the MR per se is say 500 lbs at 40 mph and 475 lbs at 60 mph and the remaining 100 to 125 lbs is from the horizontal forces. For Goss's test engine, the 500 lbs for MR per se is some 20 lbs of MR per short ton of adhesive weight.

US books on the steam locomotive in the twentieth century (see especially Johnson[1] and Hay[2]) give the Davis formula (determined originally for the resistance of electric locomotives - see reference below) beyond the motor gearing, as applying to steam locomotives as well, excluding MR, plus

[1] R P Johnson, *The Steam Locomotive*, 1945 and other editions.
[2] W W Hay, *Railroad Engineering*, Vol 1, 1953 and other editions.

20lbs per short ton of the adhesive weight of the locomotive for the MR thereof, apparently recommended by Cole, engineer of the American Locomotive Company, the basis of which is unknown, but probably one of the desirable features of "good" locomotives, which he promoted. That is presumed to apply whatever the CW diameter and the number of coupled axles, which cannot be uniformly so. The Queensland Railways used 25 lbs per long ton of adhesive weight as MR in locomotive resistance in calculating locomotive loads. I have used the same MR when calculating IHPs of Queensland, British and other locomotives, which I consider too high from first principles, and not really applying at all rates of working and rates of revolution, wishing I could use something more definitely established for MR. Hence this note, and earlier attempts. One intended improvement suggested by Prof W E Dalby, best known for his pioneering work on balancing reciprocating steam engines (not only locomotives), in his book *Steam Power* of 1920, is to introduce a constant, and driving wheel diameter as a divisor, in the 120AW/D he suggested, AW being adhesive weight in tons and D coupled wheel diameter in feet, source of the 120 not explained, but in my view that makes things worse in terms of acceptability or sense, i.e. it increases the MR.

I continue to work on what data there are on MR per se to obtain some general formula for MR. It is obvious that for a particular locomotive, the basis of MR is the result of the piston thrusts and the rpm of the mechanism squared. Data on the measurements and masses of the locomotive per se are also required. There are precious little data available, and finding generalisation among those locomotives for which there are some data so far elusive. One of these days I might have something to report on this matter.

The Davis Formula

This formula comes from the article by W J Davis Jr, *The Tractive Resistance of Electric Locomotives and Cars*, in the *General Electric Review*, Vol XXIX, 1926, p 685, the results of resistance tests in the USA and elsewhere. For electric locomotives he gave

$R = 1.3 + 29/W + .03V + .0024\ AV^2/Wn$, R the resistance at rail in lbs per short (2000lbs) ton, excluding motors, axles and gears, W the average short tons per axle, V mph, n number of axles, and A cross section area in sq ft.

For freight stock, the V term coefficient is .045 and the .0024 in the fourth term becomes .0005, while for vestibuled passenger stock, the coefficient of the V is .03, and that in the fourth term is .00034. These formulae are all for normal rolling stock with an axle load of 5 short tons or more. The article gives formulae for lighter axle loads and interurban vehicles, and for the area of exposed skin of cars as an alternative to cross section area. The formulae are convertible to use with Imperial tons by multiplying the coefficients of the first three terms in each equation by 1.12, the ratio of the mass of a long ton to that of a short ton.

It will be observed that the resistance varies with axle or journal load, falling in specific terms (per ton) as axle load increases, as occurs above with the friction coefficient for loads on bearings. Many railways in the world used this formula, and based train loads on actual loaded weights of vehicles. In the UK, resistance was always expressed in lbs/ton terms until some of the post-war work of Andrews[3] and Ell[4] (who retained that form of presentation, but whose results where conversion is possible are close to those of Davis from more than twenty years before and show universal lbs/ton form of resistance to be wrong).

[3] H I Andrews: *The measurement of Train Resistance, Jnl Instn Locom. Engineers*, 1954 p 91
4 S O Ell: *Developments in Locomotive Testing, Jnl Instn Locom Engineers*, 1953 p 561

First Principles of Machinery Resistance

The sources of this MR on a two-cylinder engine are:

(1) friction of items moving against one another: piston rings and valve rings on cylinder and valve chest walls; of the piston rod and valve rod at the gland leaving the cylinder and valve chest; given by forces resulting from steam pressure in the cylinder, as seen in a mean effective pressure diagram, including that in the opposite direction to the power stroke from compression into the clearance volume per stroke from the previous power stroke[5]; sliding of the crosshead block on the guide bars; the action at the crosshead of the piston rod on the connecting rod; the action of the connecting rod on the big end pin on the driving wheel; the action of the coupling rods on the pins on the driving and coupled wheels;

(2) the work in revolving various parts of the drive, some balanced, some partly balanced, and some not balanced at all; rotating part of the connecting rod and all the coupling rods and the balancing masses in the coupled wheels; the work in rotating the balancing masses for balancing part of the reciprocating masses; effect of changed cut off on valve travel (or similar for other valve gears and valves); the work done by the forces resulting from the pistons operating at 90/270 degrees to one another (in swaying the locomotive and in creating Fore and Aft forces along the train (see below)); the resistances on the CWBs from the positive and negative piston thrusts and bearing some of the mass of the locomotive, as resolved; and

(3) the work in reciprocating various parts.
In my view, the only completely satisfactory way to obtain LR and MR is from the results of road tests, with the locomotive indicated precisely and the pull at the tender drawbar accurately measured when and at a point where no F&A effect is present, with a large range of tests, at all useful cut offs, speeds at 10 mph intervals at least, at each of which ITE and EDBTE are obtained, hence LR under various V, Q (steam rates) and EDBTE (train loads) are obtained. Regressions would then be performed on the range of LR data at all speeds and loads and sources of MR, in multi variable regressions from which MR should emerge. It will be necessary to have good estimates of all other sources of resistance and results of the tests of reliability. As V^2 enters into both the atmospheric effects of VR (vehicle resistance) and the revolving masses effects of MR, the testing procedure should be designed with both variation in wind speed at constant train speed and variation in train speed with constant wind speed, with sufficient observations of each for the effect to register in the (multiple) regressions.

(2) THE ALTERNATING (OR FORE AND AFT) FORCE ALONG TRAINS GENERATED BY STEAM LOCOMOTIVES

I am not attempting to treat the balancing of the reciprocating parts of steam locomotives, the why and how, ably treated in many references (e.g. *Balancing Engines* by Professor W E Dalby), rather what forces remain after the part balancing, why F&A occurs, and whether that was a source of MR. It is commonly believed that F&A was more or less eliminated in the UK, but what follows will show that was not the case.

F&A was common in the BR standard locomotives as issued to traffic. In his *British Railways Standard Steam Locomotives*, E S Cox says (p 187) that when a two-cylinder engine is balanced to reduce vertical hammer blow on the rails to acceptable proportions, a longitudinal component remains, which can produce a shaking or F&A motion. The effect can be effectively smothered, he

[5] All mep diagrams show the expansion line of a power stroke intersecting the compression line of the return of the previous power stroke, in effect cancelling some of the mep of a power stroke and creating a negative thrust from what is by direction a positive thrust.

says, if the right relationship between unbalanced reciprocating weight and locomotive weight is established, provided there is no artificial magnifier along the train which picks up the smothered impulses, and by resonance magnifies them into perceptible shaking or F&A. The culprit in the BR case, according to Cox, was considered found to be the drawbar spring at the rear buffer beam of both tender and tank engines. Reduction in the initial compression with which these springs were assembled reduced the problem. Cox mentions that there were GW engines on which the percentage of reciprocating parts balanced was increased to 70, and in one case to 85, instead of the usual 50 as the remedy.

F&A is distinct from surging and jerking along trains. F&A is a large scale continuous vibration or shaking, while surging and jerking are effects on the train from variations in track alignment or from locomotive handling, especially the engine slipping. Surging is not often felt on short trains on well aligned lines. It happens as the result of a sudden, even deliberate, slowing. The buffers along the train compress, and are then released, and power is applied in large measure to resume line speed, even to surmount an incline. Acceleration of the train, especially the rear, is briefly sudden and high, a true surge. I understand surging, but wonder about F&A, and am interested in the experience of others and others' views on its origin and extent.

It was virtually unknown on the railway I grew up with, that of Queensland. Only the smaller suburban tank engine, the D17, accelerating hard, could have the passengers in at least the leading car nodding slightly at each other, involuntarily. The worst F&A I ever encountered, however, occurred just south, in New South Wales. The Brisbane Express via Wallan-garra on the inland connection between Brisbane and Sydney of the Queensland and NSW systems ran both ways between the border station at Wallan-garra and Tenterfield 12 miles south, where the New South Wales Railways had its local locomotive depot, behind the engine used for shunting at the border for the day, a D50 or D53 2-8-0 "standard goods" engine with 51 inches coupled wheels. The speed on that section was mostly about 35 mph, but the F&A in the train was extreme, and many passengers removed their luggage from the racks for safety. South of Tenterfield, the train was hauled by a C32 4-6-0 over the highest railway in Australia, about 4500 ft altitude, with no F&A.

But F&A remains. In February 2010, BR Standard Britannia *70013* on a circular tour of Kent hauling nine cars displayed it near Peckham Rye, leaving Bromley South, near Rye, and at 31 mph between West St Leonards and Crowhurst.

On the Mid Hants Railway in October 2014 GW Hall *6960* funnel first hauling 180 tons displayed it at 27 mph on the 1 in 60 climb from Alton, and downhill shut off on the same gradient to Ropley. On the return tender first there was no F&A uphill or downhill on 1 in 60s between Ropley and Alton.

In August 2006, GW 2-8-0T *5224* hauling seven cars on the North Yorkshire Moors Railway, over 250 tons, displayed bad F&A in both directions at 13 mph and double that speed both powering and coasting. The S&D 2-8-0 *53809* which was also working there then, displayed none, although its climb from Grosmont to Goathland was extremely slow – down to 8 mph.

In June 2002, four-cylinder (but not in line) GW King *6024* at a Mid Hants Railway gala, displayed F&A at 13 – 14 mph in the cutting leaving Alton funnel first, and tender first drawing up at Ropley but none at normal speeds, hauling nine cars. The railway's own 9F *92212* showed F&A chimney first with nine cars at 21-24 mph, but none steaming from Ropley uphill tender first.

In August 2005, I travelled on the Tal-y-llyn Railway (27 ins gauge) behind 0-4-0WT *Duncan* (aka *Douglas*) hauling 6 vehicles. Even at the 15 mph limit for the line, F&A was all but continuous, and considerable.

On 20 March 2022, Standard 4 2-6-0 *76017* on the Mid Hants Railway showed no F&A engine first uphill or downhill, or tender first hauling uphill, but tender first downhill conveyed a soft F&A to the leading car, no doubt through compressed buffing on both front of the locomotive and the leading end of the leading car.

I have travelled in long, heavy trains in the USA behind large two-cylinder steam locomotives. As these trains started, there was often F&A, but less than I had expected.

I have not observed deceleration as F&A sets in, even on a heavy train, nor acceleration on its ceasing.

Prof Dalby gives the technical basis for the value of this force on two-cylinder locomotives and its maximum value. The maximum arises from:

$1.7Mn^2r\cos(\theta+45^\circ)$ where

M is the lbs mass of the unbalanced reciprocating parts per side, r is the crank radius in feet (i.e. half the stroke), n the revolutions per second, and θ the crank angle.

This reaches a maximum in each direction when $\cos(\theta+45^\circ)$ reaches the maximum value of a cosine, 1, which applies at zero degrees for $\theta + 45^\circ$, on the side on which the crank leads.

He also gives an example, pp 80 to 81. With the unbalanced mass of the reciprocating parts per side of 600 lbs, 54 ins coupled wheels diameter, stroke 26 ins, $\cos\theta + 45^\circ$ has maxima on the side with the leading crank at crank angles of 315 (or -45) and 135 degrees, and minima at 45 and 225 degrees. The average value is the average value of a cosine over the whole range between 0 and 90 degrees in place of cos (θ+45) in the formula, which is 0.63. Its value is never zero, except at cos (45+45), i.e. $\cos 90^\circ$. The Dalby example gives the maximum unbalanced force at 38.5 mph as +/-17,613 lbs. At 54 mph (diameter speed, CW diameter in inches as mph) it is +/-34,712 lbs, and at 10 mph +/-1,191 lbs, the range coming from the n^2 term in the formula. (Dalby's formula already allows for the conversion of weight to mass, and he remarks that it is common for the values of the alternating force (i.e. the F&A) to exceed the average tractive effort exerted by the engine. Of course, as the alternating force is instantly followed by a force of the same value but in the opposite direction, the tractive effort is not impeded.) The same formula applies to unbalanced reciprocating masses even when say 50% of such are balanced, as is usual. At 38.5 mph there are 240 rpm, or 4 rps or 8 F&A return movements per second. As the vast majority of the alternations do not by observation result in F&A, and are absorbed by the inertia mass of the engine (perhaps engine and tender) and those I have felt have occurred at the rate of no more than about two per second, it is also clear that even if not absorbed in inertia mass, the remainder occur more as a vibration, each changing almost as soon as it is generated to an equal one in the opposite direction. Certainly on the locomotive itself, the F&A is felt mostly as a vibration, one of several felt thereon.

It is nevertheless not explained in mechanics, nor in Dalby's book, why the F&A forces, sufficient to be felt by passengers, suddenly emerge at a speed and rate of working the engine at which they have not occurred elsewhere on the same day, on the same train, or at the same place on another day, or a near day, in the same circumstances; and when the speed is very low as in the case of *5224* at 13 mph above. Is Cox's explanation, that they result from previously suppressed forces being recreated through resonance and re-emerging to shake up the passengers, the only or right one? For the effect to be felt at a lower rate in the train than it occurs in itself, is the occurrence in the train a matter of several repeated blows resulting in a breakthrough of some kind, then springs in the drawgear causing a return of the movement which has occurred? He does not consider the springs in the drawgear of the train, which have to act both ways to allow for the drawing out with acceleration, and the compression as the train bunches up on the locomotive with braking, and can be observed where the drawbar is attached to the underframe of the carriage. The amplitude of the T&F movement which is present in the Dalby formula is the length of the diameter of the crank, which on most medium and wider gauge locomotives is the stroke, from say 22 inches upwards. The F&A felt in carriages is never as great as that, nor as rapid as suggested above. These aspects of the F&A must be the result of the frequency being so high that before one movement can finish, another commences in the opposite direction.

The resistance from creating the alternating force must apply on average to the extent given in the Dalby formula for the average cosine, whether the force is absorbed in inertia mass, or emerges as F&A anywhere on the train.

Readers' experience of F&A and explanations of its (usually occasional) occurrence are welcome.

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