

Locomotive Counterbalancing

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The paper offers a method for computing and analyzing the counterbalance system of a locomotive. As a result of a straightforward simple series of computations, a clear and exact picture is obtained, showing the resultant forces set up by the rotating and reciprocating parts and the counterbalance in any given locomotive. A method is also provided for determining the proper counterbalance to meet any given conditions. The problems involved are of importance to the designer of modern locomotives of large size. The day has passed when it is satisfactory to provide static balance only for the rotating engine parts of a locomotive. Many locomotives now in service, which are balanced for static conditions only, have the main pair of wheels out of balance dynamically by as much as

400 lb. At "diameter speed," that is, at as many miles per hour as the driving wheels have inches of diameter, this lack of balance is sufficient to increase and decrease the axle load by over 20,000 lb during each revolution of the wheels. Up to the present there has been no adequate study of the effect of unbalanced horizontal forces on the locomotive. The proposed method of analysis offers a simple but accurate basis for such study. It seems probable that with proper cross-balancing a very large proportion of the mass of the reciprocating parts can be left unbalanced. In the case of the locomotive examined, over 80 per cent of the mass of the reciprocating parts is unbalanced and the locomotive is reported to ride very satisfactorily.



IN MODERN locomotive designing, the proper counterbalancing of the rotating and reciprocating parts deserves close attention. Methods used up to a few years ago are inexact and allow high unbalanced inertia forces, with destructive effect on the track. In Europe the civil engineers of the railroads require all possible protection for the permanent way, and as a result correct balancing of locomotives has been common practise. In recent years some locomotive engineers in this country have adopted methods of

cross-balancing which have reduced the unbalanced inertia forces and have thus cut down the dynamic augment of wheel loads produced by these forces. However, the great majority of locomotives designed ten years or more ago are balanced for static conditions only. In these locomotives the main axle load on the track may be increased and decreased by 20,000 lb or more during each revolution. Little argument should be needed to show the desirability of avoiding the large and unnecessary stresses thus imposed on the track.

Acceptance of this unnecessary dynamic augment in the axle load cannot be excused on the grounds of simplification of design. The great reduction in track stresses secured by elimination of unnecessary dynamic augment requires only that the counterbalance be set a few degrees off the center line of the crank. The necessary increase in weight of the counterbalance is small.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Some refinements are necessary in weighing the wheels to see that they are correctly balanced. These, however, should present no real difficulty with adequate mechanical engineering talent and proper shop management.

It is occasionally argued that it is unnecessary to consider cross-balancing for certain types of locomotives, because the wheel centers are too small to take all of the balance desired. This excuse is not valid. It is shown later that even with less than complete balance a shift of the center of gravity of the balance by about 7 degrees may reduce the dynamic augment on the main axle by over 10,000 lb. The possibility of this reduction should not be neglected.

It is not easy to see why American civil and mechanical railway engineers have neglected for so long the proper balancing of locomotives. Probably one reason for this neglect has been the lack of a simple method for analyzing the inertia forces and presenting the facts. The present paper attempts to fill this gap. It offers a complete and accurate method for analyzing the inertia effects of the rotating parts of a locomotive engine. The method serves a double purpose. In the first place, it shows exactly what unbalanced inertia forces are developed by a given locomotive. In the second place, it enables a designer to arrange the counterbalance of a locomotive so as to secure the best possible results for any given conditions.

The paper describes in detail all of the computations to be carried out and keeps the mathematics down to the inescapable minimum. The final results are presented in simple form so that they may be readily understood by executives too busy to unravel the usual intricacies of cross-balancing.

It is believed that general use of this method will accelerate the progress in improved balancing that has begun in the last few years. The present interest in the reduction of track stresses by proper balancing is directly traceable to the track stress measurements begun in 1913 by Prof. A. N. Talbot for the American Railway Engineering Association. Mr. C. T. Ripley, of the Atchison, Topeka & Santa Fe Railway, cooperated with Professor Talbot in this work and was much impressed by advantages to be gained by designing locomotives to reduce track stresses. Improved distribution of weight and elimination of flangeless tires were tried and found to be efficacious. Professor Talbot's tests also showed that large unnecessary stresses were due to imperfect balancing of locomotives. In 1924, Mr. Ripley arranged for the cross-balancing of a large Santa Fe type locomotive, and the author worked with Mr. Ripley on the preliminary computations of the balance for this engine. In 1926,

Mr. Ripley reported to the American Railway Association, Mechanical Division, that the experimental locomotive gave a satisfactory reduction in track stresses. On this foundation a considerable amount of improvement in locomotive balancing has been built. In 1930, the Committee on Locomotive Construction reported to the American Railway Association, Mechanical Division, a proposed method for dynamic or cross-balancing, and in 1932 the committee's method was adopted by the Association as recommended practise.

The present paper starts with the American Railway Association's method of computation and modifies and extends this. In addition, a method is provided for a complete analysis of the inertia forces of an existing locomotive. Special attention is called to the method of presenting the results of this analysis. Although entirely accurate, the results are presented in very simple form.

In each pair of wheels all of the inertia forces of the rotating parts and of the counterbalances are combined so that they are completely represented by two equivalent weights in each wheel. These weights act in the plane of rotation of the center of gravity of the counterbalance. In this plane one equivalent weight acts along the wheel diameter through the crankpin and the other perpendicular to this diameter. When this pair of representative equivalent weights is set down for each wheel, as shown in Fig. 2, the exact conditions of balance of the locomotive can be seen very readily. The equivalent weight acting along the crank diameter represents, in each wheel, the overbalance or underbalance acting to balance or to reinforce the inertia forces of the reciprocating parts. The equivalent weight acting at right angles to the crank is a parasitic effect due to incomplete cross-balancing. The resultant of the two equivalent weights at right angles to each other, shown by broken lines in Fig. 2, determines the maximum value of the dynamic augment. If the parasitic effect is large enough to produce an undesirably large increase in the dynamic augment, it can be reduced by cross-balancing.

With this as introduction, the method proposed by the author will be considered in more detail. The method is illustrated by using it to analyze the inertia forces set up by the rotating parts of an existing 4-8-4 type of locomotive. This engine, which has been in satisfactory service for about six years, has the main pair of driving wheels partly cross-balanced. The locomotive is of the 4-8-4 type, with dimensions as follows:

Cylinders, diameter, in.....	30
Cylinders, stroke, in.....	30
Driving wheel, diameter, in.....	73
Weight on first pair of drivers, lb.....	66,500
Weight on second pair of drivers (main axle), lb.....	70,500
Weight on third pair of drivers, lb.....	66,500
Weight on fourth pair of drivers, lb.....	66,500
Total weight on drivers, lb.....	270,000
Total weight of locomotive, lb.....	420,000

When the original design of the locomotive was under consideration, it was stipulated that at diameter speed, that is, at 73 mph, the combined static and dynamic rail load of any axle should not exceed 75,000 lb.

This is a highly intelligent method of setting the limits to be worked to in counterbalancing. After each static axle load has been established, the difference between this static load and the permissible maximum of 75,000 lb is the maximum allowable dynamic augment permitted for the axle in question. From this can be computed the amount of overbalance which may be put into each wheel to balance the reciprocating parts.

The locomotive under consideration had the main wheels partially cross-balanced. The analysis which follows shows that the final results in this pair of wheels differed somewhat from that aimed at. The deviation is due to two causes. In the first place, the eccentric cranks were assumed to be concentrated at the

crankpin, while the present more accurate analysis takes into account the fact that the center of gravity of the eccentric cranks does not fall on the main crank radius. In the second place, the position of the counterbalance to give correct cross-balance for the rotating parts was determined, and the overbalance for the reciprocating parts was then added without changing the position of the counterbalance.

The difference between the result aimed at and that obtained is not large, but is sufficient to show that any counterbalance scheme should be accurately analyzed before the locomotive design is accepted as satisfactory.

All computations necessary for an accurate analysis of the system of rotating parts are described in detail in the Appendix. A step-by-step method is used, so that those who carry out such computations infrequently may be able to follow the reasoning involved.

Before considering the application of the method to the example, a word of explanation of the term "equivalent weight" is in order. This term has been introduced to simplify the mathematics and to avoid the necessity for introducing the speed of the locomotive. Instead of calculating with "masses" or with "centrifugal forces," the "equivalent weights" are used. The inertia effect of each rotating mass is represented by its equivalent weight. Equivalent weight is defined as the weight of that mass which, rotating at crank radius about the axis of the axle, produces the same centrifugal force as the mass represented. The equivalent weight is assumed to act radially through the center of gravity of the mass it replaces.

If their positions, directions, and magnitudes are taken into account, the various equivalent weights can be resolved and combined just as though they were forces. They of course represent forces which are proportional to their magnitudes and to the square of the speed of rotation of the wheel. To speak mathematically, they are vector quantities.

We now consider the processes by which, in each pair of wheels, the inertia effects of all rotating parts, including the counterbalances, are resolved into and represented by two pairs of equivalent weights. One pair acts in each wheel in the plane of rotation of the center of gravity of the counterbalance. In each of the counterbalance planes, one equivalent weight of the pair acts along the diameter through the crankpin, with the other weight of the pair acting at right angles to this diameter.

The main pair of wheels is taken as an example. Weights and positions of the rotating masses are assumed to be as given in the Appendix. Steps in the analysis are illustrated in Fig. 1. First, by the method of section 1 of the Appendix, all rotating parts in one wheel, except eccentric cranks and counterbalance, are replaced by a single equivalent weight of 2490 lb acting in a plane 71.3 in. from the central plane of the counterbalance in the opposite wheel. (See Fig. 1a.) The distance between the central counterbalance planes is 62 in. This resultant equivalent weight of 2490 lb must be resolved into two components, one in each counterbalance plane, as described in section 2 of the Appendix. When this is done for both wheels of the pair, each wheel has an equivalent weight of 2870 lb acting along the crank radius and an equivalent weight of 380 lb at right angles to the crank, as shown in Fig. 1b. The inertia effect of the eccentric cranks must now be similarly resolved into two equivalent weights in each counterbalance plane. The method of computation is described in section 3 of the Appendix. The result is shown in Fig. 1c, together with the component equivalent weights already found for the other rotating parts. In the left main wheel the inertia effect of the eccentric cranks is represented by two equivalent weights. One weight, of 156 lb, acts along the main crank radius. The other, of 17 lb, acts along the radius 90 deg ahead of the crank. In the right wheel the components for the eccentric

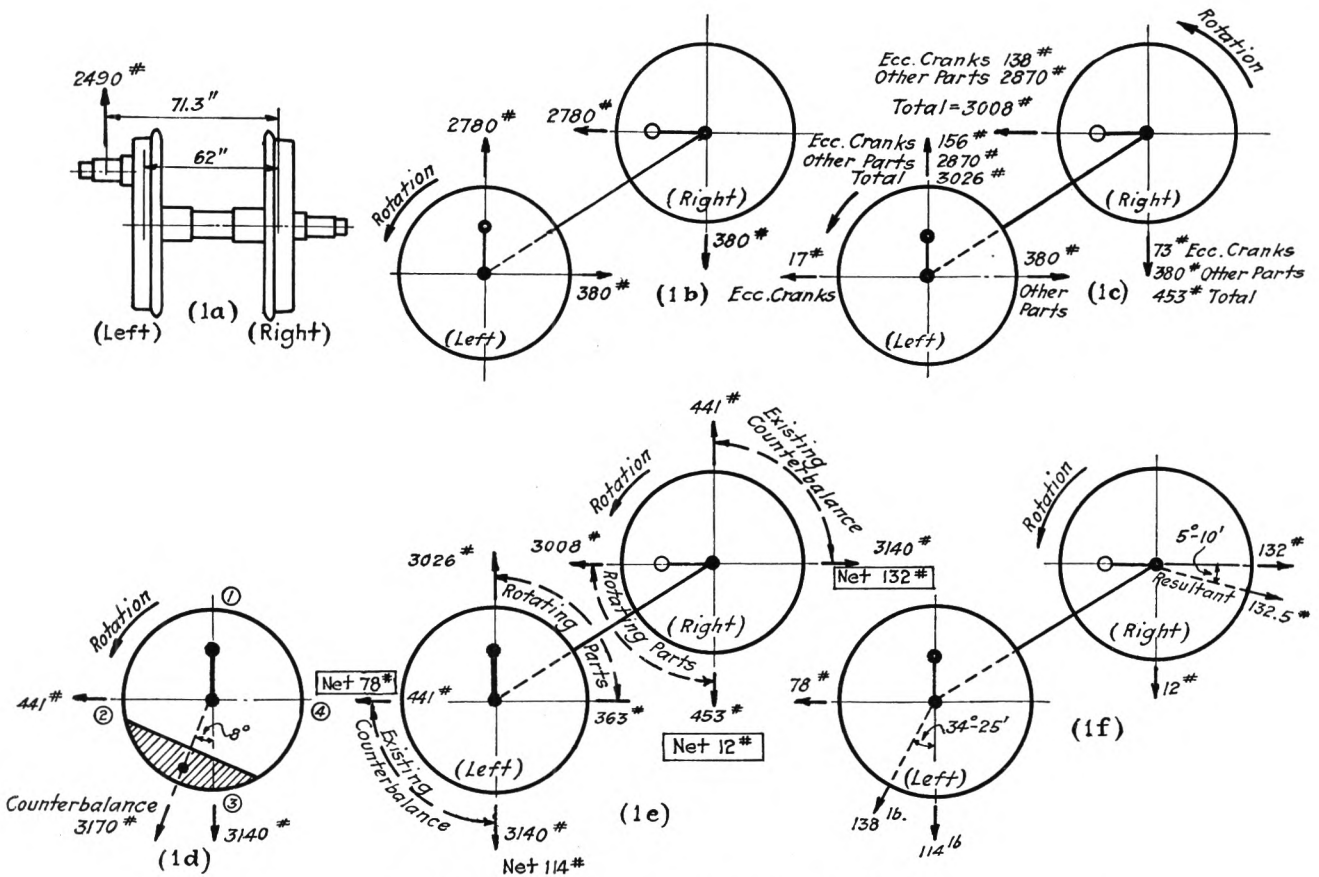


FIG. 1 ANALYSIS OF INERTIA EFFECTS IN MAIN WHEELS

(1a, equivalent weight representing all rotating parts in left main wheel except eccentric cranks and counterbalance. 1b, equivalent weight in counterbalance planes representing all rotating parts except eccentric cranks and counterbalances. 1c, equivalent weight representing eccentric cranks and all other rotating parts except counterbalances. 1d, equivalent weight and position of existing counterbalance and component equivalent weight representing counterbalance. 1e, component equivalent weights representing all rotating parts including counterbalances. 1f, component and resultant equivalent weights in counterbalance planes of main wheels representing combined effect of all rotating parts including counterbalances.)

cranks are 138 lb along the crank radius and 73 lb at 90 deg ahead of the crank. Owing to the position of the eccentric cranks, their effect is not symmetrical in the two wheels of the pair.²

Combining the components of eccentric cranks and other rotating parts, the result is set down as in Fig. 1c, and to this the inertia effect of the counterbalance as determined later is added. The locomotive under analysis had in each main wheel a counterbalance with an equivalent weight of 3170 lb, with its center of gravity set in each wheel 8 deg behind the crank diameter. (See Fig. 1d.) This can be resolved into two components at right angles to each other, as shown in Fig. 1e. The detailed method for this resolution of the counterbalance effect into two component equivalent weights is given in section 7 of the Appendix. These components are an equivalent weight of 3140 lb along the crank diameter opposite the main pin and another equivalent weight of 441 lb 90 deg behind the first.

It is now only a matter of simple subtraction to arrive at the final result of Fig. 1f. The unbalanced inertia effect of all rotating parts including eccentric cranks and counterbalance in the left counterbalance plane is represented by an equivalent weight of 114 lb along the crank diameter opposite the main pin and 78 lb at 90 deg back of this. In the right-hand counterbalance plane the unbalanced inertia effects are represented by an equivalent weight of 132 lb opposite the right-hand crankpin and

by another equivalent weight of 12 lb acting 90 deg back of this.

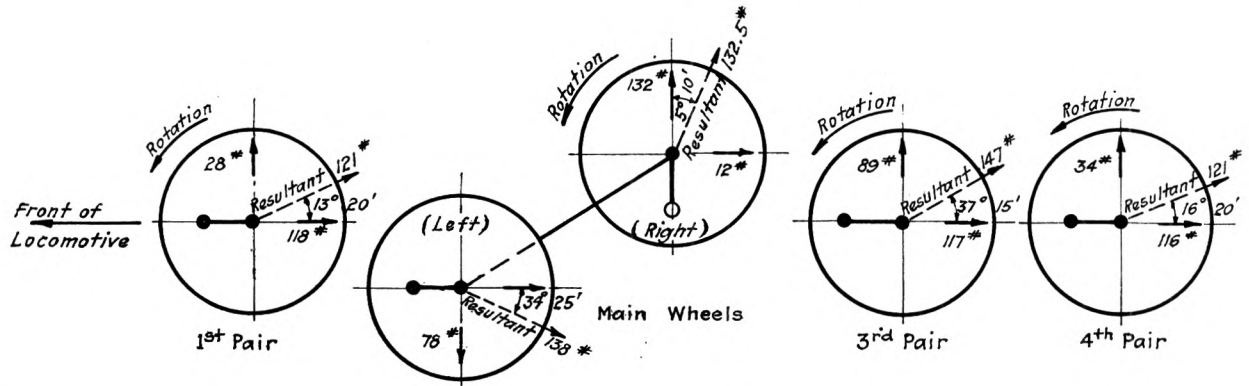
Fig. 1f gives a complete representation of the unbalanced inertia forces in the main wheels. All rotating parts, including eccentric cranks and counterbalances, are covered. In each counterbalance plane there are two component equivalent weights at right angles to each other. These can be combined into a single resultant as shown.

The main pair of wheels being disposed of, the other coupled wheels must be dealt with in the same way. It is convenient to use a blank form similar to Fig. 1 for setting down the various steps in the calculation, omitting details which pertain only to the main wheels. Details of the computations are not given here, but the final results for the four pairs of main and coupled wheels are shown in Fig. 2. Except in the case of the main pair of wheels, the right- and left-hand wheels on each axle have the same equivalent weights, and therefore only one wheel of each pair is shown. The lack of symmetry in the main wheels is due, as explained above, to the position of the eccentric cranks.

It is important to use a form similar to that of Fig. 2 for showing the results obtained, as this gives complete information regarding the unbalanced inertia forces in the simplest possible form. The information thus given puts the locomotive designer in a position to give critical consideration to the balance system which has been applied or which is proposed for application.

The balancing of the locomotive under consideration will now be reviewed. Two points are of major importance: the dynamic augment and the amount of balance provided for the reciprocating parts.

² The author is indebted to Mr. C. H. Bilty, mechanical engineer, Chicago, St. Paul & Pacific Railroad Company, for calling attention to the desirability of considering the eccentric cranks separately from the other revolving parts on the crankpin.



Line		1st Pair	Main Wheels		3rd Pair	4th Pair		
			Left	Right				
1	Resultant Equivalent Weight Producing Dynamic Augment	Lb	Lb	Lb	Lb	Lb		
		121	138	132.5	147	116		
2	Dynamic Augment at Diameter Speed	Per Wheel	5800	6630	6360	7050	5800	
3		Per Axle	8200	6570		9950	8200	
4	Static Axle Load	66,500	70,500		66,500	66,500		
5	Maximum Combined Static and Dynamic Axle Load at Diameter Speed	74,700	77,070		76,450	74,700	Total Overbalance	
6	Equivalent Weight Opposing Reciprocating Parts	118	114	132	117	116	465* Left 483* Right	

FIG. 2 COMPONENT AND RESULTANT EQUIVALENT WEIGHTS REPRESENTING UNBALANCED INERTIA FORCES

The force producing dynamic augment in each wheel is proportional to the resultant equivalent weight shown for that wheel in Fig. 2. This force is computed from the resultant equivalent weight by the usual formula for centrifugal force:

$$F = 0.000284 WRn^2$$

where

- F = force in pounds
- W = equivalent weight in pounds
- R = crank radius in inches
- n = revolutions per minute.

At diameter speed—that is, at a speed of as many miles per hour as the driving wheels have inches of diameter—the wheels make 336 revolutions per minute. At this speed the centrifugal force is:

$$F = 3.2 RW$$

For the locomotive under consideration, which has a crank radius of 15 in., this equation takes the form

$$F = 48 W$$

That is, the maximum dynamic augment in each wheel is 48 times the resultant equivalent weight in that wheel. Values corresponding to this are shown in line 2 of the tabulation in Fig. 2 for all of the wheels. The values here are the maximum values of the force which alternately increases and decreases the wheel load on the rail during each revolution. The maximum increase, or maximum dynamic augment, occurs in that position of the wheel in which the resultant equivalent weight is directed vertically downward. It is evident that the maximum values for right- and left-hand wheels on one axle do not occur at the same time. Consequently, the maximum increase in axle load is not the sum

of the maximum increases in the two wheels. Section 5 of the Appendix gives a method for determining the maximum axle load and the corresponding position of the wheels when the resultant equivalent weights in the two wheels are different and are set at any angle. If the resultant equivalent weights are the same in both wheels of a pair and act at 90 deg apart, the maximum axle load occurs when the resultants in both wheels are directed downward at an angle of 45 deg from the vertical. The maximum dynamic augment for the axle is then 1.414 times the maximum for each wheel. In the locomotive under consideration, this is the case for all of the coupled wheels except the main pair.

In the main pair of wheels, the planes through the resultant weights and the axis of the axle stand at an angle of 119 deg 15 min, and the resultants have the values of 138 lb in the left and 132.5 lb in the right wheel. By applying the method of section 5 of the Appendix, it is found that the equivalent weight producing the maximum axle load is 137 lb. This does not differ greatly from the individual resultants in the wheels.

It should be noted that the relation between maximum axle load and maximum wheel load depends on the angle between the wheel resultants. The extreme cases are, (1) with zero angle between the resultants the increase in axle load is twice the wheel load, and (2) with 180 deg between the resultants the increase in axle load is zero.

If the angle between the resultants is large, so that the increase in axle load is small, it will probably be desirable to consider the increase in wheel load as the limiting factor rather than the increase in axle load. The axle load measures the influence of the locomotive on a track unit such as a bridge, while the wheel load determines the influence on an individual rail.

Returning to a consideration of Fig. 2: Line 3 shows the maximum dynamic augment per axle at diameter speed and line 4 shows the static axle load. Line 5, the sum of the two preceding lines, gives the maximum combined static and dynamic axle load at diameter speed. As mentioned above, it was intended when designing the locomotive that this combined axle load should not exceed 75,000 lb. This limit is observed in the front- and back-wheel pairs, but is slightly exceeded in the main and third pairs of wheels. If an analysis had been made in this form before the locomotive had been built, it would have been a simple matter to determine the weight and position of counterbalances which would keep to the desired axle load.

The general method is described in section 6 of the Appendix. In the main axle, the maximum dynamic augment permissible is 75,000 - 70,500 = 4500 lb. This corresponds to an equivalent weight of 94 lb on the axle. If the resultants in the wheels are equal and act at 90 deg apart, the value of each will be 66.3 lb.³ Assume then that each of the main wheels is to be balanced so that the resultant producing dynamic augment in the wheel is 66 lb. It is desirable to eliminate any parasitic effect and to have the full weight of this resultant acting opposite the crankpin so as to be available for balance of the reciprocating parts.

In the left main wheel, the rotating parts are represented as shown in Fig. 3a by equivalent weights of 3026 lb along the crank diameter and 363 lb acting 90 deg back of this. To produce the desired balance, the components of the counterbalance must be 363 lb opposing the same weight at right angles to the crank, and

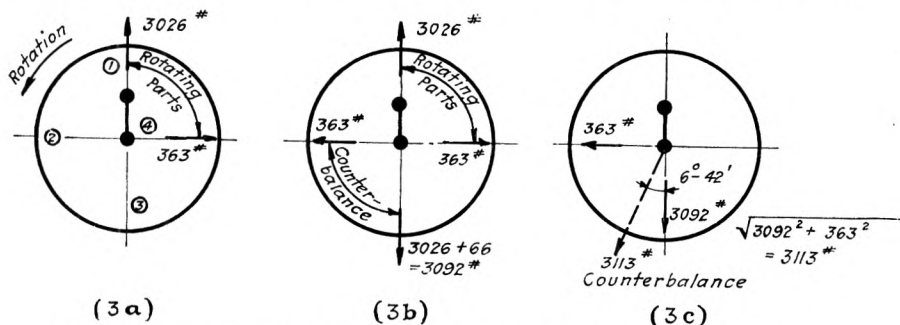


FIG. 3 DETERMINATION OF COUNTERBALANCE TO GIVE COMPLETE CROSS-BALANCE AND DESIRED OVERBALANCE OF 66 LB

(3a, equivalent weights representing rotating parts to be balanced in left main wheel. 3b, component equivalent weights representing rotating parts and desired counterbalance. 3c, components and resultant equivalent weight giving counterbalance desired.)

3026 + 66 = 3092 lb opposite the pin, as shown in Fig. 3b. These components, when combined as in Fig. 3c, show that the counterbalance itself must have an equivalent weight of 3113 lb, with its center of gravity set 6 deg 42 min back of the crank diameter. Comparing this with the counterbalance as applied, it is seen that by reducing the equivalent weight from 3140 lb to

³ 66.3 × 1.414 = 94.

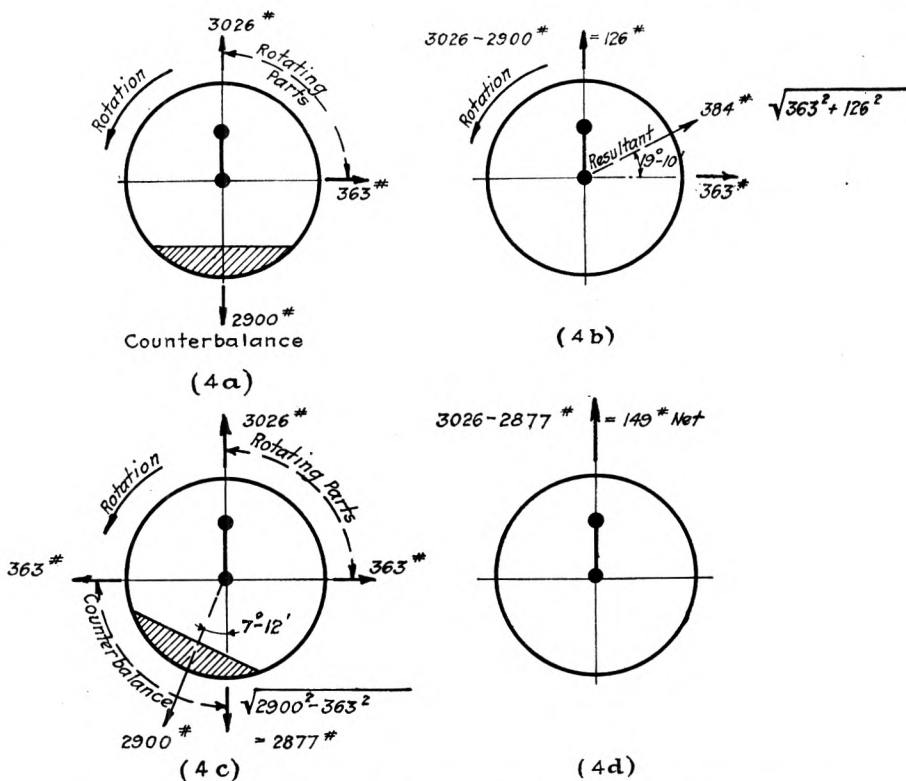


FIG. 4 INCORRECT AND CORRECT POSITIONS FOR INSUFFICIENT COUNTERBALANCE (4a, component equivalent weights with insufficient counterbalance placed opposite crankpin. 4b, component and resultant equivalent weights representing unbalanced inertia forces of arrangement 4c. 4c, position of insufficient counterbalance to give least possible unbalanced inertia force. 4d, net equivalent weight representing unbalanced inertia force resulting from arrangement 4c.)

3113 lb and changing the angle from 8 deg to 6 deg 42 min, the unbalanced resultant equivalent weight is reduced from 138 to 66 lb, with a reduction of 3350 lb in the dynamic augment of the wheel at diameter speed.

This illustration shows that when the present method is used to resolve the inertia effects into two equivalent weights in each wheel, it is a simple matter to determine the balance required to meet any desired conditions.

By applying the same method to the right-hand main wheel, it is found that for complete balance, except for an overbalance of 66 lb opposite the pin, the counterbalance must have an equivalent weight of 3107 lb set 8 deg 23 min off the diameter. The difference between the counterbalance found for the right- and left-hand main wheels is not large. It is desirable for practical reasons of manufacture to make both wheels the same. A satisfactory balance can be obtained by giving the actual counterbalance in both wheels an equivalent weight of 3110 lb set at an angle of 7 deg 30 min off the diameter. This weight and angle are obtained by averaging the values found as above for the right- and left-hand wheels. With this counterbalance the left wheel will have an overbalance of 56 lb and a resultant of 71 lb, while the right wheel will have an overbalance of 74 lb and a resultant of 87 lb. The left wheel is slightly underbalanced and the right wheel slightly overbalanced, but the difference is not serious.

In addition to the main pair of wheels, it will be seen from Fig. 2 that the third pair also exceeds the proposed limit of 75,000 lb for combined static and dynamic load. The dynamic augment exceeds the estimated figure by 1450 lb, because the wheels were not cross-balanced. Cross-balancing would have involved changing the equivalent weight of the counterbalance from 1130 to 1133 lb and moving its center of gravity 4 deg 30 min away from the crank diameter. This would eliminate the parasitic effect of 89 lb, leaving the overbalance the same, 117 lb, and reducing the resultant producing dynamic augment from 147 to 117 lb. This reduction of 20 per cent in the dynamic augment is well worth considering if the locomotive is being built to a closely restricted weight.

In the fourth pair of wheels, the desired overbalance opposite the crankpin is 116 lb. The parasitic effect which could be eliminated by cross-balancing is 34 lb. These produce a resultant of 121 lb. Cross-balancing by eliminating the parasitic 34 lb would reduce the equivalent weight, producing dynamic augment from 121 to 116 lb—that is, by only 4.1 per cent. The reduction is hardly enough to be worth while.

As a general rule, it may be noted that unless the parasitic effect to be eliminated by cross-balancing amounts to more than 30 per cent of the overbalance the reduction in the resultant produced by cross-balancing will not be more than 4.5 per cent, and therefore hardly justifies the additional complication.

Before leaving the balance of rotating parts, consideration must be given to the desirability of cross-balancing when the wheel centers are too small to allow the full amount of balance to be applied. Assume that in a main wheel the component equivalent weights representing the rotating parts are as found for the left main wheel of the locomotive already examined—that is, 3026 lb along the crank radius and 363 lb at right angles to this. Assume also that the wheel design limits the equivalent weight of the counterbalance to 2900 lb. If this is put exactly opposite the pin, as in Fig. 4a, it is obvious that the unbalanced components will be as in Fig. 4b, 126 lb along the crank and 363 lb at 90 deg, giving an unbalanced component of 384 lb, producing dynamic augment. This can be reduced materially by shifting the equivalent weight of 2900 lb 7 deg 12 min off the crank diameter. As shown in Fig. 4c, this gives components of 2877 lb opposite the pin and 363 lb at right angles to this. The position is chosen so that the component at right angles to the crank diameter is just equal and opposite to the component of the rotating parts acting at right angles to the crank. The simple calculation required to determine the other component is obvious from Fig. 4c. The components being 363 and 2877 lb, the tangent of the angle at which the counterbalance must be set is $363/2877 = 0.1265$, which is the tangent of 7 deg 12 min. Combination of the four components in Fig. 4c shows that the net unbalanced equivalent weight is 149 lb, as in Fig. 4d, instead of 384 lb when the counterbalance was directly opposite the pin, as in Fig. 4a. This large reduction in the unbalanced force producing dynamic augment makes cross-balancing well worth while, even though full balancing is not possible.

Consideration must now be given to the overbalance provided for the reciprocating parts. Fig. 2 shows that the overbalance amounts to 465 lb on the left-hand and 483 lb on the right-hand side. The locomotive under consideration had reciprocating parts weighing 2241 lb on each side of the engine. Approximately 80 per cent of the weight of the reciprocating parts is unbalanced. This is very much more than the 50 per cent set up by the American Railway Association as recommended practise. In spite of this, the locomotive rides satisfactorily.

The fact is that the A.R.A. practise needs further consideration. In the first place, there is no logical reason for specifying the overbalance as a percentage of the weight of the reciprocating parts.

The proper course is to consider the unbalanced weight in comparison with the total weight of the locomotive. The unbalanced portion of the reciprocating parts tends to shake the locomotive. This shaking is resisted by the inertia of the locomotive as a whole. Consequently, the stability of the locomotive is determined by the relation of the mass of the whole locomotive to the mass of the unbalanced parts. This principle was stated by George R. Henderson over twenty-five years ago, but has not received the recognition it deserved.

Henderson suggested that one-four hundredth of the weight of the locomotive might remain unbalanced on each side. That would be 2.50 lb unbalanced per 1000 lb of locomotive weight. The 4-8-4 type locomotive analyzed has 4.2 lb unbalanced per 1000 lb. Other locomotives analyzed in the same way show unbalanced weights of 5.4 lb per 1000 for a 2-8-4 type and 3.8 lb per 1000 for a 4-6-4 type. All are reported to ride satisfactorily.

The conclusion to be drawn is that if the rotating parts are properly balanced, it is only necessary to balance a comparatively small portion of the reciprocating parts.

Further study of results obtained in practise is desirable, but if the reports of such results are to have any value they must be based on an accurate analysis of the balance similar to that which has been described.

To complete this examination of the inertia forces it is necessary to take into account the fact that the inertia forces of the reciprocating parts act in the vertical plane through the center of the main rod, while the inertia force of the overbalance acts in the central plane of the counterbalance. This difference in plane can usually be neglected. Its effect is to tend to turn the locomotive about a vertical axis. This tendency will have no perceptible effect on a long modern locomotive. In adjusting the overbalance to offset the inertia forces of the reciprocating parts, it is only necessary to consider the longitudinal forces. The couple about the vertical axis due to the difference between the planes of action can be neglected. Cross-balance is therefore not necessary for the overbalance for the reciprocating parts.

In conclusion, the author repeats what he has said before in addressing the Society. It is important for railroad engineers to adopt an accurate method for analyzing the balance of a locomotive. It is also highly important that the results which are obtained by the analysis be stated in a simple manner free from involved mathematics. If this is done, the advantages of proper balancing methods will be self-evident and improvements will follow.

Appendix

Locomotive Counterbalancing

SCHEDULE OF DATA AND COMPUTATIONS FOR DETERMINING THE PROPER COUNTERBALANCE FOR LOCOMOTIVE DRIVING WHEELS

THE principles of the method and the results to be obtained are discussed in the body of the paper. This schedule is drawn up to present the detailed methods of computation to be followed in order to secure the results desired.

Values corresponding to an actual locomotive are inserted for each item, and the schedule is arranged so that if it is rewritten with blank spaces for the values, it can be used as a work sheet for computing any other example.

The notes at the end of each section define any special terms used and provide any necessary discussion of the methods.

The data as to weights and dimensions called for by the schedule must of course be obtained from the design of the locomotive.

SECTION 1—TO FIND THE EQUIVALENT WEIGHT AND PLANE OF ACTION OF THE ROTATING PARTS IN EACH WHEEL

The schedule given here is drawn to cover the main pair of driving wheels. It should be applied successively to each of the other pairs of drivers, dropping out the items which do not apply. This section of the schedule is illustrated by Figs. 5 and 6.

1. E = distance in inches between central planes of right- and left-hand counterbalances. 62 in.
2. W_1 = equivalent weight in pounds of crankpin hub, together with part of pin encircled by hub⁴. 500 lb
3. A = distance in inches between centers of gravity of right- and left-hand crankpin hubs. 71 in.
4. M_1 = moment of crankpin hub parts about center plane of opposite counterbalance
 $M_1 = W_1 \times (A + E)/2 = 33,200$ in-lb
5. W_2 = equivalent weight in pounds of side rod carried on crankpin,⁴ together with weight of part of crankpin encircled by side-rod bearing. 550 lb
6. B = distance in inches between center planes of side rods 77 in.
7. M_2 = moment of side-rod parts about center plane of opposite balance
 $M_2 = W_2 \times (B + E)/2 = 38,200$ in-lb
8. W_3 = equivalent weight in pounds of back end of main rod,⁴ together with weight of part of crankpin encircled by main-rod bearing. 1440 lb
9. C = distance in inches between center planes of main rods. 85 in.
10. M_3 = moment of main-rod parts about center plane of opposite counterbalance
 $M_3 = W_3 \times (C + E)/2 = 106,000$ in-lb
11. W_t = sum of all rotating equivalent weights, except those for eccentric cranks
 $W_t = \text{items } (2 + 5 + 8)$
 $= W_1 + W_2 + W_3 = 2490$ lb
12. M_t = sum of all moments of rotating parts, except eccentric cranks, about center plane of opposite counterbalance
 $M_t = \text{items } (4 + 7 + 10)$
 $= M_1 + M_2 + M_3 = 177,400$ in-lb
13. F = distance in inches between center of gravity of all rotating parts, except eccentric cranks, and center plane of opposite counterbalance
 $F = \text{item } 12/\text{item } 11$
 $= M_t/W_t = 71.3$ in.

Notes on Section 1:

Item 2. To find the equivalent weight of the crankpin hub, together with the part of the pin encircled by the hub, it is necessary to find the actual weight of these parts and their center of gravity. The weight to be taken into account is that of the cross-hatched area in Fig. 6 which lies outside the circumference of the axle hub. Then if:

- W = actual weight of these parts in pounds
- R_1 = distance in inches of their center of gravity from longitudinal axis of axle
- R = radius of rotation of crankpin in inches
- W_1 = equivalent weight of these parts

$$W_1 = W \times R_1/R$$

⁴ See Notes on Section 1.

“Equivalent weight” of any part rotating about the longitudinal center line of the axle is defined as the weight of that mass which rotating at the crank radius would produce the same inertia force as the actual mass under consideration. The above example shows how the equivalent weight is found when its actual weight and the radius of rotation of its center of gravity are known.

Item 5. The equivalent weight of the side rod carried on the

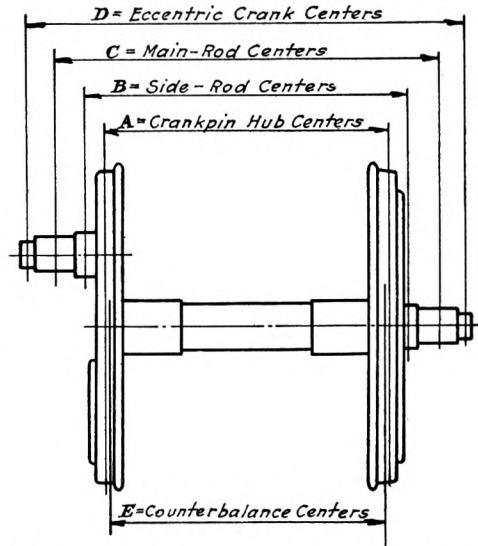


FIG. 5 DATA FOR COMPUTATION OF EFFECT OF ROTATING PARTS

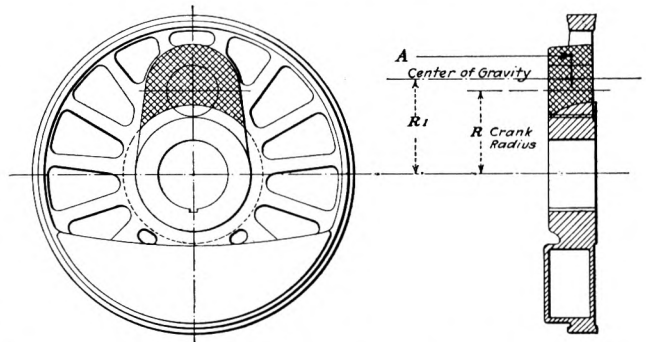


FIG. 6 PORTION OF CRANKPIN HUB SHOWN CROSS-HATCHED TO BE TAKEN INTO ACCOUNT AS ITEM 2, SECTION 1, APPENDIX

crankpin is the actual weight supported by the pin. For the front and back drivers this is the weight of one end of one side rod. For the main and intermediate drivers it is the weight of one end of each of two rods. As the center of gravity of the side-rod end rotates with the same radius as the center of the crankpin, the equivalent weight of these parts is the same as their actual weight.

Item 8. The equivalent weight of the back end of the main rod to be used in this item can be taken as approximately six-tenths of the total weight of the main rod. If strict accuracy is required, the weight to be used can be found from the equation:

$$W_{m2} = W_m \times k^2/l^2$$

where

- W_m = total weight of main rod in pounds
- l = length of rod in inches between wristpin and main-pin centers

k = radius of gyration in inches of the main rod about the wristpin center
 W_{m2} = weight of main rod to be considered as rotating at the crankpin.

The radius of gyration k can be found by suspending the main rod to swing as a pendulum about the wristpin center. Then if

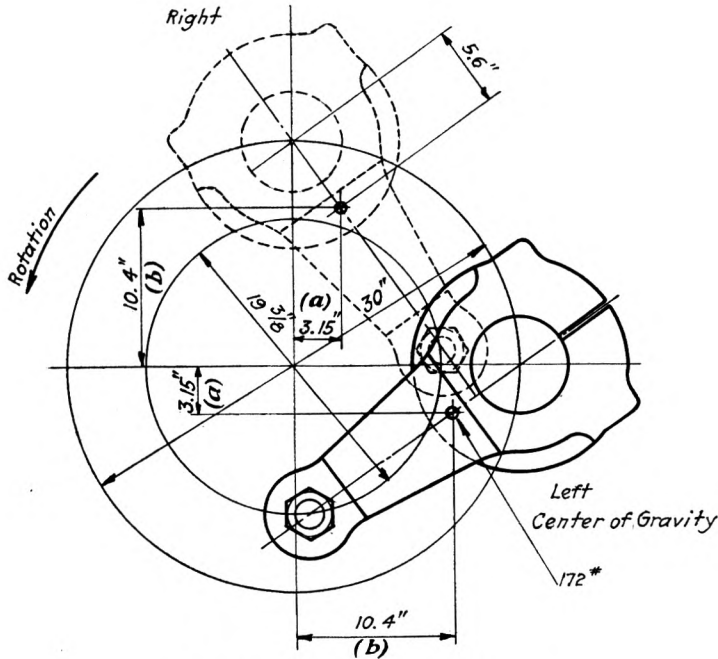


FIG. 7 POSITION OF ECCENTRIC CRANKS

x_0 = distance in inches of the center of gravity of the rod from the wristpin center, and
 t = time of one swing in seconds
 $k^2 = 3.26 t^2 x_0$

If the radius of gyration k cannot be found experimentally, and if the main rod is of normal design, a reasonably close approximation can be obtained by assuming a value of 0.6 for k^2/l^2 . The equivalent weight of the back end of the main rod is then six-tenths of the total weight.

$$W_{m2} = 0.6 \times W_m$$

SECTION 2—TO RESOLVE THE EQUIVALENT WEIGHT OF THE ROTATING PARTS ON ONE SIDE OF A PAIR OF WHEELS INTO TWO COMPONENTS, ONE IN THE WHEEL CARRYING THE ROTATING PARTS AND THE OTHER IN THE OPPOSITE WHEEL OF THE PAIR

This section is illustrated by Fig. 1 (a) and (b). Items which were taken from section 1 are given the same number here.

- 11. W_t = total equivalent weight of rotating parts (section 1), 2490 lb
- 13. F = distance in inches from center of gravity of rotating parts to center plane of opposite counterbalance (section 1) 71.3 in.
- 1. E = distance between center planes of counterbalances (section 1) 62.0 in.

- 14. W_l = component of left-hand rotating parts acting in center plane of left counterbalance

$$W_l = W_t \times F/E = 2870 \text{ lb}$$

- 15. W_r = component of left-hand rotating parts acting in center plane of right counterbalance

$$W_r = W_l - W_t = 380 \text{ lb}$$

SECTION 3—TO DETERMINE THE INERTIA EFFECT OF THE ECCENTRIC CRANKS

Fig. 7* shows the relative positions of the eccentric cranks. The center of gravity of each crank does not lie on the main crank radius. The crank cannot, therefore, be accurately represented by a single equivalent weight acting along the main crank radius. Each crank must be represented by two equivalent weights acting, one along the crank radius, the other along a radius 90 deg ahead of this. This is illustrated by the isometric diagram in Fig. 8. In this, each eccentric crank is represented by two equivalent weights in the plane of rotation of the center of gravity of the eccentric crank. These four equivalent weights must be replaced by four others, two acting in each counterbalance plane. These are computed as follows:

- 16. W_e = actual weight in pounds of one eccentric crank, together with weight of part of crank pin encircled by eccentric crank. . . . 172 lb
- 17. b = length in inches of intercept of center of gravity of eccentric crank on crank radius (see Fig. 7). 10.4 in.
- 18. a = length in inches of intercept of center of gravity of eccentric crank on radius 90 deg ahead of crank. 3.15 in.

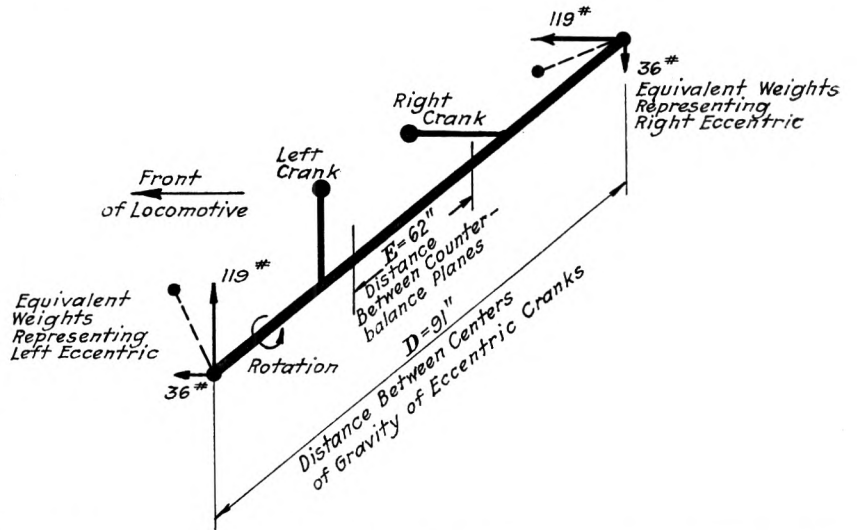


FIG. 8 ISOMETRIC DIAGRAM ILLUSTRATING REPLACEMENT OF EACH ECCENTRIC CRANK BY TWO EQUIVALENT WEIGHTS ACTING IN THE PLANE OF ROTATION OF THE CRANK

- 19. D = distance in inches between centers of gravity of eccentric cranks (Fig. 8) 91 in.
- 1. E = distance in inches between the center planes of counterbalances. 62 in.
- R = crank radius in inches. 15 in.

From these data are computed the component equivalent

* The cranks are shown trailing. In Fig. 8 and the text, however, they are assumed to be leading the crank pins.

weights which, acting in the planes' left- and right-hand counterbalances, represent completely the inertia effects of the eccentric cranks.

In Left-Hand Counterbalance Plane:

20. L_{ob} = component equivalent weight acting along crank radius
 $= \frac{W}{2RE} [b(D + E) + a(D - E)] = 156 \text{ lb}$
21. L_{ea} = component equivalent weight acting along radius 90 deg ahead of crank
 $= \frac{W}{2RE} [a(D + E) - b(D - E)] = 17 \text{ lb}$

In Right-Hand Counterbalance Plane:

22. R_{ob} = component equivalent weight acting along crank radius
 $= \frac{W}{2RE} [b(D + E) - a(D - E)] = 138 \text{ lb}$
23. R_{ea} = component equivalent weight acting along radius 90 deg ahead of crank
 $= \frac{W}{2RE} [a(D + E) + b(D - E)] = 73 \text{ lb}$

Note: The equivalent weights are not the same in the two counterbalance planes, although the resultant is the same in both planes. The reason for the lack of symmetry can be seen in Fig. 8. In the horizontal plane through the main axle the 36-lb component of the left eccentric and the 119-lb component of the right eccentric both act in the forward direction. In the vertical plane, on the other hand, the 119-lb component of the left eccentric acts upward, while the 36-lb component of the right eccentric acts downward.

SECTION 4—TO COMBINE INERTIA EFFECT OF ECCENTRIC CRANKS AND OTHER ROTATING PARTS

In sections 2 and 3 the inertia effect of the other rotating parts and of the eccentric cranks have been resolved into equivalent weights acting in the center planes of the two counterbalances. The net effect of all the rotating parts is found by adding the values found in the two preceding sections. The operation is illustrated by Fig. 9. The quarters of each wheel are numbered for convenience of reference.

In Left-Hand Counterbalance Plane, Diameter Through Left Crankpin:

14. W_1 = equivalent weight representing other rotating parts at quarter (1) 2870 lb
20. L_{ob} = equivalent weight representing eccentric cranks at quarter (1) 156 lb
24. W_1' = net equivalent weight at quarter (1) 3026 lb

Diameter Perpendicular to Left Crank:

15. W_r = equivalent weight representing other rotating parts at quarter (4) 380 lb
21. L_{ea} = equivalent weight representing eccentric cranks at quarter (2) 17 lb
25. W_4' = net equivalent weight at quarter (4) 363 lb

In Right-Hand Counterbalance Plane, Diameter Through Right Crankpin:

14. W_1 = equivalent weight representing other rotating parts at quarter (2) 2870 lb
22. R_{ob} = equivalent weight representing eccentric cranks at quarter (2) 138 lb
26. W_2' = net equivalent weight at quarter (2) 3008 lb

Diameter Perpendicular to Right Crank:

15. W_r = equivalent weight representing other rotating parts at quarter (3) 380 lb
23. R_{ea} = equivalent weight representing eccentric cranks at quarter (3) 73 lb
27. W_3' = net equivalent weight at quarter (3) 453 lb

Fig. 9 illustrates these operations. The figures are the same as in Fig. 1 (c). Items 24, 25, 26, and 27 give the four equivalent weights which, acting two in each counterbalance plane, completely represent the inertia effect of all rotating parts,

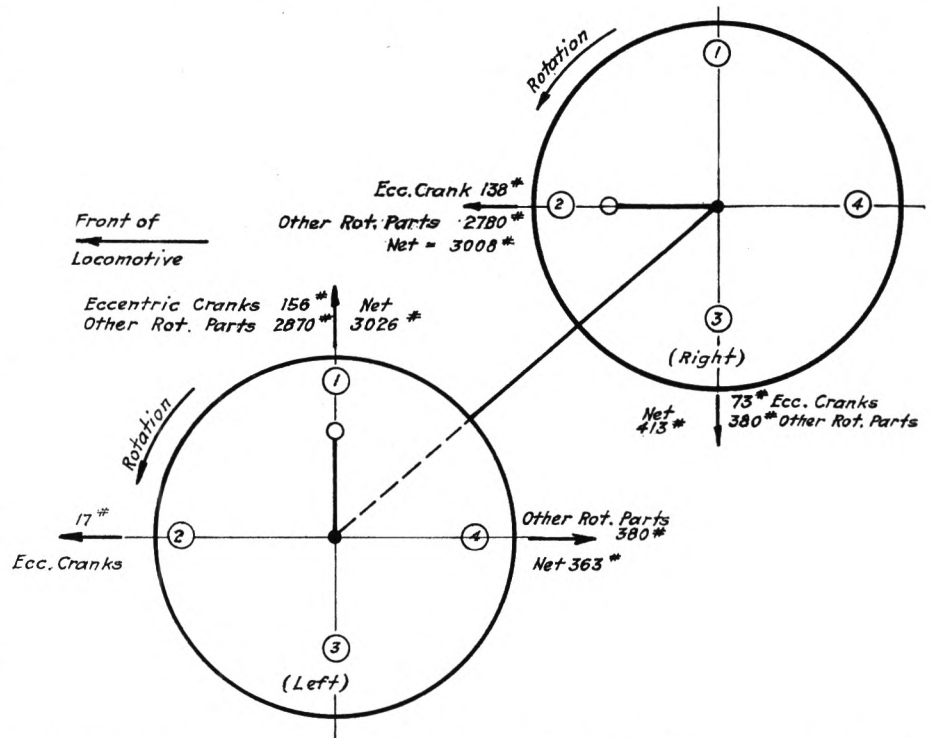


FIG. 9 EQUIVALENT WEIGHTS ACTING IN CENTER PLANES OF COUNTERBALANCES REPRESENTING ECCENTRIC CRANKS AND OTHER ROTATING PARTS

including the eccentric cranks. Having these, it is a simple matter to complete the analysis of the wheels by adding the effect of the counterbalance as in Fig. 1 (f). The net equivalent weights representing counterbalances and all rotating parts are shown in Fig. 1 (g). Details of the operation are given in section 7.

SECTION 5—TO FIND THE POSITION AND VALUE OF THE MAXIMUM DYNAMIC AXLE LOAD WHEN THE RESULTANT EQUIVALENT WEIGHTS PRODUCING DYNAMIC AUGMENT IN THE WHEELS ARE NOT THE SAME

This section is illustrated by Fig. 10.

- 28. W_l = resultant equivalent weight in pounds producing dynamic augment in left-hand wheel. 138 lb
- 29. W_r = resultant equivalent weight in pounds producing dynamic augment in right-hand wheel. 132.5 lb
- 30. Z = angle in degrees between planes in which the above equivalent weights act. 119 deg 15 min

In the position of maximum axle dynamic augment, assume

- 31. X = angle in degrees between direction of left-hand resultant and the vertical
- then $Z - X$ = angle in degrees between the direction of right-hand resultant and the vertical

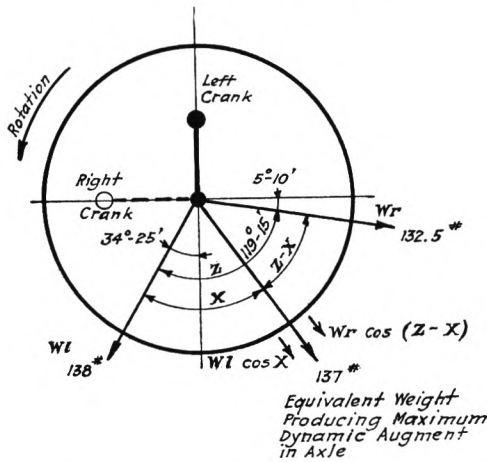


FIG. 10 COMPUTATION OF MAXIMUM DYNAMIC AUGMENT IN AXLE (When equivalent weight producing maximum dynamic augments in wheels have values W_l and W_r and stand at an angle of Z from each other.)

It follows that

- $W_l \cos X$ = vertical component contributed by left-hand resultant
- $W_r \cos (Z - X)$ = vertical component contributed by right-hand resultant

Then $W_l \cos X + W_r \cos (Z - X)$ is the total equivalent weight producing the maximum dynamic augment in the axle.

This has its maximum value when

$$\tan X = \frac{\sin Z}{W_l/W_r + \cos Z}$$

With the values given above for W_l , W_r , and Z , the value of X is found to be 57 deg 40 min. Then the total equivalent weight producing the maximum dynamic augment in the axle:

$$\begin{aligned} W_l \cos X + W_r \cos (Z - X) &= 138 \times 0.535 + 132.5 \times 0.476 \\ &= 74 + 63 \\ &= 137 \text{ lb} \end{aligned}$$

SECTION 6—DETERMINATION OF EQUIVALENT WEIGHT AND POSITION OF COUNTERBALANCE TO BALANCE EXACTLY THE ROTATING PARTS AND PROVIDE A GIVEN OVERBALANCE FOR THE RECIPROCATING PARTS

This section is illustrated by Fig. 3. A blank diagram similar to Fig. 3 should be used, and the values should be filled in as described. In the following description the diagram is assumed

to represent the left-hand wheel of the pair and the quarters are numbered 1, 2, 3, and 4, quarter No. 1 being that of the left-hand crankpin.

- 24. W_1' = component of rotating parts acting in plane of left counterbalance at quarter (1); see section 4. . 3026 lb
 - 25. W_4' = component of rotating parts acting in plane of left-hand counterbalance at quarter (4); see section 4 363 lb
 - 32. W_0 = equivalent weight of overbalance to be added to oppose the reciprocating parts. 66 lb
- The value to be given this item may be chosen arbitrarily. The better plan is to determine it to hold the dynamic augment to a definite limiting value.
- 33. W_{c3} = component of counterbalance to act in quarter (3), opposite the crankpin:
Item 29 = Item 24 + Item 32 = 3092 lb
 - 34. W_{c2} = component of counterbalance to act in quarter (2):

$$W_{c2} = W_{r'} = 363 \text{ lb}$$

- 35. W_c = resultant equivalent weight of counterbalance to produce the components shown in quarters 3 and 4 in Fig. 8 (a):

$$W_c = \sqrt{W_{c3}^2 + W_{c2}^2} = 3113 \text{ lb}$$

- 36. α = angle which radius through center of gravity of counterbalance makes with the diameter through quarters 1 and 3:

$$\begin{aligned} \tan \alpha &= W_{c2}/W_{c3} = 363/3092 \\ &= 0.1275 \\ \alpha &= 6 \text{ deg } 42 \text{ min} \end{aligned}$$

The equivalent weight and position of the counterbalance thus determined are shown in Fig. 3 (c).

Note: In determining the actual counterbalance to have this equivalent weight proper allowance must be made for the equivalent weight of the spokes and rim adjacent to the crankpin hub which occupy the space corresponding to the space covered by the counterbalance in the opposite quarter of the wheel.

The counterbalance determined by Items 35 and 36 will exactly cross-balance the reciprocating parts, and the only unbalanced mass producing dynamic augment will be the overbalance of equivalent weight W_0 acting in quarter (3) directly opposite the crankpin.

It may be noted that it is important to cross-balance when the value of W_r , the component due to the rotating parts of the opposite wheel, is large compared with W_0 , the overbalance.

If W_r does not exceed one-quarter of W_0 , it is unnecessary to cross-balance. If $W_r = 0.25 W_0$, a counterbalance of equivalent weight $W_{c4} = W_1 - W_0$ can be placed directly opposite the pin at quarter (3) and the dynamic augment will not exceed that of the cross-balanced wheel by more than 3 per cent. This is practically negligible.

SECTION 7—ANALYSIS OF A GIVEN COUNTERBALANCE TO DETERMINE THE OVERBALANCE AND DYNAMIC AUGMENT

This section is illustrated by Fig. 1 (d) and (e).

- 37. W_c = equivalent weight of counterbalance in pounds 3170 lb
- 38. β = angle made by radius through center of gravity of counterbalance with diameter through quarters (3) and (1) Fig. 1 (c). 8 deg
- 39. W_{c3} = component of counterbalance acting at quarter (3)
 $W_{c3} = \cos \beta \times W_c = 3140 \text{ lb}$
- 40. W_{c2} = component of counterbalance acting at quarter (2)
 $W_{c2} = \sin \beta \times W_c = 441 \text{ lb}$

Discussion

A. I. LIPETZ.⁵ I quite agree with the author that "it is not easy to see why American civil and mechanical railway engineers have neglected for so long the proper balancing of locomotives," especially in view of the fact that "in Europe correct balancing of locomotives has been common practise." This situation has been always a puzzle to me, because at least one American engineer contributed a great deal to the question of counterbalancing in the early days of locomotive construction. Thomas Rogers, founder of the Rogers Locomotive Works, patented as early as 1837 the application of counterbalance opposite the crank with sufficient weight to counterbalance the crank and connecting rods, thus introducing counterbalancing of revolving parts.

Up to 1845 it was customary to balance only revolving weights, and the balancing of reciprocating weights was not recognized until later, when some European engineers started investigations of the problem of balancing. The disturbances caused by the unbalanced inertia forces first became apparent when Nollau⁶ in Germany made tests with a locomotive suspended by chains from a roof, although previously to that W. Fernihough in England, in October, 1845, suggested and apparently tried⁷ the use of weights for counterbalancing main rods, pistons, and other reciprocating parts.

Le Chatelier in France, in 1848, made tests similar to Nollau's and enunciated a very complete theory of balancing which is now known as cross-balancing. He published his theory in 1849,⁸ and shortly afterward his theory was further developed by various investigators, mostly French.⁹ Later the correct method of cross-balancing was popularized for the English-reading public by Daniel Kinnear Clark in his classic work book "Railway Machinery,"¹⁰ published simultaneously in Glasgow, Edinburgh, London, and New York.

Since then, all textbooks appearing in French,¹¹ German,¹¹ Russian,¹² Hungarian,¹³ Japanese,¹⁴ and other languages made use of the Le Chatelier-Clark method as the only correct way of balancing locomotives. Locomotive builders all over the world, except the United States, adopted this method of counterbalancing. Some American books¹⁵ also expounded the cross-balancing theory, although they did not recommend it for practical use, and in England Professor Dalby, in his well-known book, "The Balancing of Engines," developed very con-

venient graphical methods of cross-balancing. Incidentally the tables given in his book are almost identical with the schedule given in the Appendix to Mr. Fry's paper.¹⁶

I have been looking for a long time for an explanation of the fact that cross-balancing of locomotives was not common practise in the United States. On the basis of the theory that nothing in engineering, like in nature, can exist for a long period of time without a justifiable reason, I tried to reach some explanation in my discussion of Dr. R. Eksergian's paper "The Balancing and Dynamic Rail Pressure."¹⁷ To a great extent such a condition was probably due to the attitude of the American Railway Association, Mechanical Division, which followed for many years certain rules of what is called "static balancing" and did not recommend cross-balancing until 1931.¹⁸ Locomotive builders nevertheless, were applying the cross-balancing method to locomotives, when specified, as it is evidenced by applying the correct method in locomotives built for use abroad, like France in 1908, Russia during the war, and Japan after the war, and some isolated cases for different experimental locomotives, in the United States since 1905.

Cases are known when American railroads recently converted statically balanced high-speed passenger locomotives into cross-balanced. That the question of cross-balancing is attracting attention now more than before is probably due to the increase in speed of our present-day locomotives, both passenger and freight, the limitations of weight, and the necessity of refinements. I should therefore say that Mr. Fry's paper is very timely, notwithstanding the fact that a long time has elapsed since the correct method of balancing was made known to the world by Le Chatelier.

In my opinion, of all the questions of locomotive engineering, counterbalancing is probably the only complete, well-founded, and clear-cut exposition of a theory which permits the establishing of indisputable formulas and laws. Even in its general algebraic form it is very simple and does not call for "simplifications." The whole theory of counterbalancing can be represented by the following five formulas:

$$W_{cs} = \Sigma W \frac{D + E}{2E} + k W_{rc} \frac{C + E}{2E} \dots\dots\dots [1]$$

$$W_{c2} = W_{cs} - \Sigma W - k W_{rc} \dots\dots\dots [2]$$

$$W_c = \sqrt{W_{c2}^2 + W_{rc}^2} \dots\dots\dots [3]$$

$$\tan \alpha = \frac{W_{c2}}{W_{rc}} \dots\dots\dots [4]$$

$$K = \Sigma k \dots\dots\dots [5]$$

The designations are mostly those used by the author. Symbol Σ stands for the sum of products given by him in Appendix; W are the various weights W_1, W_2, W_3, W_4 of revolving parts, and D stands for the corresponding distances A, B, C, D of the Appendix. Coefficient k is the percentage of balancing of reciprocating weights in the particular wheel, K is the total percentage of balancing of reciprocating weights of the locomotive. W_{rc} is the value of these weights on one side. Other symbols are the same as used by Mr. Fry.

The author is actually following these formulas. Although he asserts that the analysis which he made is stated in a simple, non-mathematical manner, it is hardly so. Arithmetic is also part of mathematics, and the author's arithmetic in the Appendix

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⁶ *Journal des Chemins de Fer Allemands*, October, 1848.

⁷ Report of the Gage Commissioners, 1846.

⁸ *Etudes sur la Stabilité des Machines Locomotives en Mouvement*, by Le Chatelier, Paris, 1849. (This and references 6 and 7 are quoted from "Railway Machinery," D. K. Clark, 1855, p. 166.)

⁹ *Voie, Matériel Roulant et Exploitation Technique des Chemins de Fer*, by M. Ch. Couche, Paris, 1873, vol. 2, p. 389.

¹⁰ *Ibid.*, p. 411, and the subsequent French literature on locomotives; for instance, "La Machine Locomotive," by E. Sauvage, Paris, 1918, pp. 211-212.

¹¹ *Die Gesetze des Lokomotivbaues*, by F. Redtenbacher, Mannheim, 1855, pp. 132-134, and subsequent German literature on Locomotives, for instance, *Kurzes Lehrbuch des Dampflokotivbaues*, by F. Meineke, Berlin, 1931, pp. 108-109.

¹² "Parovozhy" (Steam Locomotives), by Romanoff, St. Petersburg, 1903.

¹³ "Lokomotivok," by Dr. Szábó Gusztáv, Kapus László, Budapest, 1919, pp. 114-119.

¹⁴ "Locomotive Engineering," by Mori and Matsuno, vol. 2, pp. 275-277; published by the Okura Book Co., Inc., Tokyo, Nitronbashi, 1926, eighth edition. Also *Locomotive Designers' Handbook* Nos. 232-2, 232-3, issued and printed by Rolling Stock Section, Department of Mechanical Engineering, Japanese Government Railways, 1931.

¹⁵ "Locomotive Operation," by G. R. Henderson, published by *Railway Age*, Chicago, 1904, pp. 70-72.

¹⁶ "The Balancing of Engines," by W. E. Dalby, London, 1907, p. 82.

¹⁷ A.S.M.E. Trans., 1929, paper no. RR-51-5.

¹⁸ Proc. Am. Ry. Assn., Division V, Mechanical, 1930, pp. 805-828, and 1931, p. 99.

is actually carrying out a numerical example by using the foregoing formulas, at least in so far as the revolving weights are concerned. As regards the reciprocating weights, there is a slight difference; namely, in the example worked out by the author, the excess balance is added to the large component of the rotating balance (Appendix 1, section 6, item 33), while in the cross-balancing method that part of the reciprocating weights which is to be balanced in this particular wheel would be added to the part of the revolving weights W , which is balanced in the plane of the main rod (Appendix 1, section 1, item 8), thus requiring no additional calculation.

The difference between the results of these two procedures is this: In the ordinary method the percentage of balancing of the fore-and-aft vibrations is different from that of the nosing vibrations of the locomotive; or, in other words, the balancing of the forces is different from the balancing of the couples due to forces of inertia of reciprocating parts, whereas by the cross-balancing method forces and couples are balanced to the same degree, and the meaning of the term "percentage of balancing" assumes definiteness which is lacking in the ordinary method. So, for instance, the author states that the weight of the balanced reciprocating parts is 465 lb on the left-hand and 483 lb on the right-hand side, an average of 474 lb, which in reference to the total reciprocating weights on one side (2241 lb) may seem to amount to 21.1 per cent, but the actual percentage of balancing the couples will be, using the author's designations, represented by:

$$\frac{474}{2241} \times \frac{2E}{C + E} = 21.1 \times \frac{124}{147} = 17.8 \text{ per cent}$$

or less than the percentage of balancing the reciprocating forces (21.1 per cent). It is true that while the difference between these two figures may seem large, these figures do not represent the actual conditions, because the unbalanced, not the balanced, forces and couples are what actually matters, and they are 78.9 per cent in the case of forces and 82.2 per cent in the case of couples.

Thus the difference between the two methods is slight. But there is no real balancing of forces, unless the moments (couples) are balanced too. Moreover, the percentage of balancing has no real meaning, if the couples are neglected. The writer is, therefore, of the opinion that cross-balancing is also preferable with respect to reciprocating weights.

As to the dynamic augment, it is immaterial which method is followed, as long as the excess balance in the wheel is the same. If there should be a slight numerical difference, it would be due to the fact that the smaller figure might be an approximate one while the other is the correct one.

It is interesting to note that the author, in section 3 of the Appendix, resorted to a rather involved algebraic method in determining the components of inertia forces of the eccentric crank. Everybody who can master these formulas (items 20 to 23) will be able to follow the formulas of the cross-balancing method in a general way, and need not be shown a numerical example in the belief that he is avoiding mathematics. It is further interesting to note that in this particular point a simplification is possible when, as the case normally is, the right and left main wheels are made from one pattern. In this case the author recommends to consider the averages of the components acting along the crank radius and at 90 deg to it. But from items 20 to 23, using Fig. 9, it can be seen that

$$\frac{L_{ob} + R_{ob}}{2} = \frac{Wb}{R} \frac{D + E}{2E}$$

$$\frac{R_{oa} - L_{oa}}{2} = \frac{Wb}{R} \frac{D - E}{2E} = \frac{L_{ob} + R_{ob}}{2} - \frac{Wb}{R}$$

In other words, we may consider the equivalent weight of the eccentric crank (Wb/R ; see Fig. 7) as a revolving weight attached to the crankpin in the plane of the eccentric crank, and need not be bothered with intercept a or any eccentric crank calculations.

By the way, I wish to mention that the Russian Decapods built in 1916-1917 in this country had counterbalances figured by the cross-balancing method with taking into account the equivalent weights of the eccentric crank (and also half-weights of the eccentric rod), and averaging the counterbalances and angles of the right and left wheels resulting from calculations in accordance with the author's suggestions.¹⁹

I agree with the author that the revolving parts should be completely balanced, so that no parasitic forces should take place. It is very essential that the revolving weights should be balanced completely, because otherwise the balance of the reciprocating weights is unfavorably affected. As the balances for both revolving and reciprocating weights are combined into one balance in the wheel, the action of the counterbalance for balancing the reciprocating weights does not start if and until the complete balancing of the revolving weights takes place. Consequently, if there is a deficiency in the balances of the revolving weights, the balancing of the reciprocating weights is impaired.

This is borne out by some of the author's numerical examples and because of that I do not understand the author's attempt to introduce simplifications in the balancing of revolving weights. I have in mind his statement "that unless the parasitic effect to be eliminated by cross-balancing amounts to more than 30 per cent of the overbalance, cross-balancing hardly justifies the additional complication." The writer cannot agree with this statement, as it should be remembered that no real simplification will be achieved by neglecting the small component and the angle of the balance. The calculation has to be made anyway, in view of the differences in planes, and in order to determine the major component; and the increase in the amount of work involved in the calculation by figuring the other component and the proper angle, as well as in preparing the pattern of the counterbalance, of the proper size and direction, is so insignificant that it is hardly possible to consider it a complication. I could never appreciate the "simplification" of saving a calculator several hours of work in a design of a locomotive, the building of which requires tens of thousands of man-hours, especially as very often only one calculation is made for a great number of duplicate locomotives.

I think cross-balancing should be made on all coupled wheels, irrespective of the ratio of the components or of the angle, and this for the sake of correct balancing of the reciprocating weights.

Another point I wish to touch upon is the distribution of the excess balance between coupled wheels. The author's example proves that very little is gained by placing an excess balance in the main wheel. The main axle is usually the one with the heaviest load. In addition, there is a piston-thrust component increasing the load on the main wheel during the forward movement of the locomotive, which the author did not take into consideration. At long cut-offs and low speeds this component amounts to a considerable force, sometimes 12,000 to 15,000 lb per wheel. At high speeds, when the counterbalancing effect gets into play, this component is much smaller—probably not over several thousand pounds, due to the shorter cut-off used at the higher speed. Nevertheless, this should not be neglected in considering the maximum permissible load. In view of this, as suggested by me in my discussion of Dr. Eksbergian's paper, it would be just as well to leave the main wheel without any excess balance, and have it properly cross-balanced for revolving weights only. The balance for the reciprocating weights could be distributed among the coupled axles.

¹⁹ A. I. Lipetz, "Russian Decapod Locomotives Built in the United States" (in Russian), New York, 1920, pp. 148-149.

Summarizing my remarks, I should say that I am in favor of applying the correct cross-balancing method for all revolving weights (both for main and coupled axles), as well as for balancing reciprocating weights, and for omitting the overbalance for reciprocating weights on the main wheels.

S. S. RIEGEL.²⁰ The author has long been known as one of the pioneers in improving locomotive counterbalancing and has done much very valuable work on the subject. I am in accord with all of his points and recommendations.

As this subject was placed before the A.R.A. Locomotive Construction Committee, of which I am a member, Mr. Fry brought the present information to our attention early in the year, as he contemplated, if possible, reading it there, so that our committee had opportunity then to benefit by his constructive criticism, and a supplementary recommendation was made by the Locomotive Counterbalancing Committee in June, 1932, and is now a matter of record there.

In view of this fact it may not be amiss to review this part of our supplementary report as it, to a considerable extent, fits as a discussion of Mr. Fry's recommendations now advanced. I believe it will be as interesting here as it was there. In substance this is, namely:

In the modern superpower locomotives with large firebox overhang and weight, it seems permissible to regard considerations of nosing and swaying as less important when judging effects of the reciprocating balance on the smoothness of operation or riding qualities. This leaves only the fore-and-aft oscillations whose magnitude is in direct proportion to the total mass of the locomotive and the force causing the oscillation, which in turn is a function of the mass of the unbalanced reciprocating parts and its frequency of vibration.

The design of a locomotive modifies according to individual designers of separate railroads and builders, and is influenced by changes in transportation developments and demands. In some regions the traffic demands require high tractive effort with low total weight, concentrated on drivers and large reciprocating parts, while in others, even on the same road, a unit of equal or lower tractive force capable of higher sustained horsepower is needed.

The recent trend in locomotive building is toward the latter type; so that when we compare this with the older lighter units, on the basis of ratio of percentage of reciprocating weight balanced, we obtain from the modern locomotive a higher figure for pounds of total weight per pound of unbalanced reciprocating weight and have smoother operating units, or what may be more logical we can balance a lower percentage of the reciprocating weight and thus by increasing the unbalanced portion, retain the figure for pounds of total weight per pound of unbalanced reciprocating weight, and the newer locomotive will be as smooth in operation and cause lower track stresses. This will compensate in some measure for the increased weight per driver of the modern high-speed superpower locomotive.

The real question to determine in all this is: Which is to be permitted to suffer more, the rail and track structures or the locomotive? For, certainly if the recurring load is lifted from the track, it must be borne by the boxes, frames, and other parts of the running gear of the locomotive, whether considered as a percentage of the reciprocating weight, the ratio of the unbalanced weight to the total weight, or something else. A series of tests might be made with instruments to record the track stresses and the vibrations of the locomotive, to definitely determine the magnitude of these forces and movements and thereby increase the total sum of human knowledge. It is doubtful, however,

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if a compromise could be made even then that would be perfectly satisfactory to both bridge and right-of-way interests and to those operating and caring for the locomotive.

There appears to be, however, one aid left which has long been recognized but not given the utmost consideration it deserves; namely, the lightening of reciprocating parts by use of higher strength alloy steels and non-ferrous metals. Lately many metallurgical advances have been made that were eagerly taken up by other industries. Perhaps our present steels of 60,000 to 80,000 lb per sq in. tensile strength should be replaced by those of 100,000 to 120,000 lb tensile strength, effecting saving of 30 to 40 per cent in weight with advantageous results.

It is believed that by the use of such alloy steels for main rods, crossheads, pistons and piston rods, and possibly aluminum-alloy crosshead shoes and piston bull rings, the unbalanced reciprocating weight can be reduced several hundred pounds per side. When we realized that any addition or reduction in the reciprocating weight increases or decreases the force on the driving boxes by 45 to 55 times its amount, at diameter speed, or by 64 to 78 times the force tending to shake the whole locomotive, it can be appreciated that even slight reductions of weight of these parts is worth while. Equally, then, some of this reduction can apply to reduce the reciprocating balance, which also is multiplied by the greater figure above stated to include both sides, and this also may be amplified by as much as 20 for the bridge stresses when the recurring load synchronizes with the natural frequency of the bridge span.

Considering the second, it seems, as may be expected from its simplified nature, that several refinements to the method outlined in the A.R.A. Committee Report of 1930 have been suggested. One, the weight to be added to the main wheel for part of the reciprocating balance should be added to the main revolving balance, designated W_0 in the report, before it is combined with the weight added to offset the cross-effect of the overhanging parts. This point is well taken, as the reciprocating balance should naturally be placed directly opposite the crank-pin and not at an angle with it as previously obtained. By doing this, the main-wheel balance can be reduced in weight and a new slightly less angle for the balance be obtained. This refinement is to be recommended.

If desired to introduce further refinements, another suggestion is to cross-balance the intermediate driver, as an appreciable reduction of rail blow from this driver can in some cases be effected thereby. This is consistent, and while not stated, it was implied in the report and can also consistently be definitely recommended.

It is most important especially that greater exactness and care be observed in securing the weights we need in the balances and to see that like weights are applied in opposite wheels, as carelessness in allowing dissimilar weights to be placed in opposite wheels will have very disturbing effects, since we are now operating at much higher speeds.

A. H. FETTERS.²¹ I notice that the author uses the A.R.A. method as a basis. I have been practising this method verbatim for several years with satisfactory results. I have ridden many engines before and after having been cross-counterbalanced at the main wheel, and I find that the riding qualities of a locomotive are not always a safe guide to a perfect balance, especially if the main axle happens to be under or near the virtual center of the locomotive, as in this case the vertical component due to overbalance or underbalance does not exert its effect in teetering the engine, and therefore the effect is not felt in the cab. I have ridden a 4-8-2 with a cross-overbalance of 300 lb, and while the dynamic augment was 26,000 lb, it did not show up in the cab.

²¹ General Mechanical Engineer, Union Pacific System, Omaha, Neb.

Had any other axle on this locomotive been off-balance to that or less extent it would have shown up in the cab as a very rough-riding engine. Again, I have found many other physical things that caused rough riding that it is difficult to determine by trial if the counterbalance or some other factor is responsible for rough riding. I have put some engines in ideal counterbalance, and they would ride smooth one trip and very rough the next, due to stuck wedges, fouled equalizers, slack between engine and tender, and similar causes. We can depend, however, on proper calculations and applications of the cross-counterbalance to the main drivers to reduce dynamic augment, thereby reducing damage to track and wear of machinery of the locomotive.

In view of the gradually increasing speeds of locomotives in freight service the last few years, the subject of cross-counterbalancing main wheels becomes of still greater importance than formerly, and the mechanical organization of any road which overlooks this fact and fails to take advantage of this very obvious improvement is remiss in its duties.

D. J. SHEEHAN.²² Since the subject of cross-counterbalance of locomotives became a subject of common discussion about four years ago, there have been many queer ideas advanced concerning counterbalance in general. To the busy mechanical engineer on the average railroad, who accepted the discussion of out-of-plane forces and dynamic balance as the work of the theorist and the master mind of locomotive design, it was merely another method of counterbalance. It was probably all right and would give results as satisfactory as the method that he had been using for the past fifteen or twenty years. But he was not having a lot of trouble with counterbalance considered from the standpoint of the old static balance method, and it was much easier to handle. Some day when he had time he would study this new method at least so that he could talk about it.

One day this mechanical engineer had a report that one of his passenger engines was riding rough and jumping up and down. He rode this engine and confirmed the report. When the counterbalance was checked by the old method of static balance, the figures indicated that 65 per cent of the weight of the reciprocating parts were balanced, the balance equally distributed among all the wheels. However, an additional weight equal to 175 lb at crankpin radius was applied. The subsequent reports indicated that the engine rode considerably better, but still vibrated up and down.

Imagine the astonishment, when the counterbalance was properly checked, with due consideration to the forces acting outside of the plane of the balance at their proper moment arms, and the results indicated that the balance in the main wheel still lacked approximately 100 lb at crankpin radius to balance the out-of-plane revolving forces.

Needless to say, when this engine was finally returned to service, properly counterbalanced for the forces acting in various planes outside the balance, both in the near and in the far wheel, the reports indicated that the locomotive rode like a "Pullman."

This little story, while possibly a little exaggerated, indicates the urgent need for a simple method of consideration of the subject of cross-counterbalance of locomotives. The engineer who spends a little time studying this subject soon discovers that primarily it contains only the simple fundamentals of mechanics, but he has thus far been somewhat frightened by complicated discussions and intricate mathematical analysis of the subject. Several unfamiliar terms such as out-of-plane forces, dynamic balance, dynamic augment, rail load, track load, and others were not to be found in his vocabulary of common usage.

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This subject is most important, and a simple method of analysis will greatly assist the busy railroad mechanical man to grasp the true significance.

JOHN A. PILCHER.²³ The paper outlines the principles involved in counterbalancing steam locomotives. The author has gone into a refinement of the counterbalance that is often very much neglected. The counterbalancing of a locomotive is entirely a compromise as between the horizontal and vertical forces. He has outlined the subject in a very intelligible way, showing the significance and importance of cross-balancing.

Reference is made to what are called "parasitic forces." These are forces introduced by improper location of the counterbalance. In other words, the component of the counterbalancing forces may be in a direction which is not available for balancing the reciprocating forces, but which would tend to increase the dynamic augment. In this connection it is significant to realize that in the case of non-cross-balanced engines the entire force of overbalance is not available for balancing the reciprocating weights.

The author points out the fact that by properly placing the counterbalance—that is, by shifting it the proper amount from the position directly opposite the crankpin—its effectiveness can be materially increased without increasing the weight of the counterbalance itself. This may be particularly valuable in counterbalancing engines in which the room in the main wheel is so limited as to make it impossible to secure as much balance as is desired. The importance of the cross-balancing is continually increasing on modern locomotives having heavy rotating parts and wide cylinder spacing, thus placing the plane of the rotating and reciprocating parts a very considerable distance outside of the plane of the counterbalance.

A. GIESL-GIESLINGEN.²⁴ It might be interesting to note that the author's figures for the weight of the unbalanced reciprocating masses compared with the weight of the locomotive correspond exactly to those for the 2-8-4 type passenger locomotive of the Austrian Federal Railways²⁵ where said unbalanced reciprocating masses are 1/231 of the engine weight exclusive of the tender. The writer is glad to acknowledge from his experience that this and similar relations proved entirely satisfactory. He would like to add some information which he found to be a good guide in quantitatively answering the questions connected with counterbalancing of locomotives.

One of the primary questions is: How great a dynamic augment may be permitted as a result of balancing the reciprocating masses (or, generally speaking, as a result of any free centrifugal force influencing the wheel pressure)? Many European railroads limit this dynamic augment, for the maximum operating speed, to 15 per cent of the static wheel load. This is a very conservative figure; it is often being exceeded in other countries on perfectly satisfactory locomotives. Two distinctly different considerations enter here: First, the limit imposed by the track structure and, second, the fluctuations of the wheel pressure that may be consistent with safe riding. Some light is thrown on the former by results obtained on a test track supported by helical springs, installed by Dr. Wirth of the Austrian Federal Railways in 1928.²⁶ Under the above-mentioned 2-8-4 type locomotive, having an axle load of 40,000 lb on the drivers, the maximum depression of the rails was 0.146 in. at very low speeds where

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²⁴ Mechanical Engineer, New York, N. Y.

²⁵ See *Railway Mechanical Engineer*, May, 1930.

²⁶ See the Journal of the Austrian Society of Engineers and Architects, "Zeitschrift des Oesterr. Ingenieur- & Architekten-Vereines," 1930, p. 353.

the dynamic augment is zero. A dynamic augment of 12 per cent of the static load depressed the rail only 3 per cent more, corresponding to one-quarter of the theoretical excess load. This may be explained by the short time available for the process of depressing the rail in the neighborhood of the diameter speed—namely, about 1/10 of a second—yet it is remarkable that this effect was noticed with so stiff a track structure. Thus we may conclude that even with our highly stressed rails, a dynamic augment of 15 to 20 per cent of the static wheel load will impose upon the rail only a fractional additional stress. Similar results will probably be apparent from the otheograph records of the General Electric Company.

The other factor—namely, the fluctuations of wheel pressures as related to safe riding—is illustrated by the fact that a dynamic augment of 30 per cent will set up a theoretical fluctuation of the wheel pressure between the limits of 70 per cent and 130 per cent of the static load, but for the same reasons as just explained the actual fluctuations are much smaller. Therefore, much higher figures on certain locomotives have not led to apparent inconveniences, although they may be objectionable, especially when it is considered that locomotives often exceed regular speed limits.

As a result of experience and comparative studies, the writer submits the following recommendations for smooth and safe riding, leaving a good margin for occasional excess speeds:

Recommended for the maximum speed at which the locomotive is expected to operate regularly (80 to 100 per cent of the diameter speed for conventional engines, 115 to 125 per cent of the diameter speed for special high speed designs).

(1) Maximum dynamic augment $A = 25$ to 30 per cent of the static wheel load W for the wheel in question, but $(A + W) = 115$ to 120 per cent of the maximum static wheel load as permitted by the track structure; whichever of the two figures for the dynamic augment A thus obtained may be lower.

(2) Maximum weight of unbalanced reciprocating masses: 1/400 to 1/300 of the combined weights of the engine with 50 per cent loaded tender.

The latter condition limits the oscillating movement of the locomotive, resulting from the unbalanced reciprocating masses, to theoretically around $1/8$ in., but this amount is further reduced by frictional influences. The stiff connection between engine and tender makes it allowable to regard both as a unit. It appears that, if the reciprocating masses are light enough to fall within the foregoing limits, no counterbalancing of any part of them would be required for fairly smooth riding.

These recommendations are open to discussion. Cross-counterbalanced locomotives corresponding to them will be found satisfactory.

AUTHOR'S CLOSURE

It is gratifying to find that Mr. Lipetz has no fault to find with the general principles of the paper. The references that he gives to earlier work on the subject are interesting and valuable. The author had no intention of claiming any originality for the principles advocated. In fact, thirty years ago his first approach to cross-balancing was guided by von Borries' account of the subject.

The author appreciates the comments by Messrs. Riegel, Feters, Sheehan, and Pilcher.

Mr. Giesl's remarks are not entirely clear and do not seem to be applicable to American practise. The suggestion that a dynamic augment of 15 to 20 per cent of the static load will impose on the rail only a fractional additional stress is not supported by Professor Talbot's experiments. The results of these as reported to the A.R.A. by Mr. Ripley show that as the locomotive speeds were increased, there was a very considerable increase in rail stress due to dynamic augment.

The two recommendations made by Mr. Giesl as to permissible dynamic augment and unbalanced reciprocating mass beg the whole question. He recommends that the dynamic augment should not exceed 25 to 30 per cent of the static wheel load. Surely the maximum combined dynamic and static wheel load should be determined by the civil engineer after due consideration of the particular track structures involved. When a limit has been set, the mechanical engineer will probably be interested in obtaining the maximum possible static load and consequently will aim at the minimum possible dynamic augment. Whether the dynamic augment is 10 per cent or 50 per cent of the static load is in itself immaterial. The important thing is that the combined loads shall not exceed the limiting value proper for the permanent way.

Mr. Giesl also suggests that the mass of the unbalanced reciprocating parts should not exceed 1/300 to 1/400 of the masses of the locomotive and of the half-loaded tender. The paper shows that locomotives with very much greater unbalanced masses are running satisfactorily in this country. The author feels that further study of this question should be carried out before any attempt is made to set up definite limits for the unbalanced reciprocating mass.