Editor "American Engineer:

I am glad of the opportunity to answer the question Mr. Henderson asks in the above letter discussing the strength of driving axles.

It is often better to reason from known facts and base our conclusions upon them than to assume certain conditions which are not so clearly established. It seems more desirable to assume normal conditions for the working stresses, and keep the fiber stress at such a low figure as to give an ample margin of strength to resist the abnormal and extraordinary stresses which may occur from time to time.

In a locomotive having but one pair of drivers the entire piston thrust must evidently be resisted by the one crank pin and axle. The bending moment is the thrust of the piston multiplied by the lever arm; the maximum fiber stress caused by

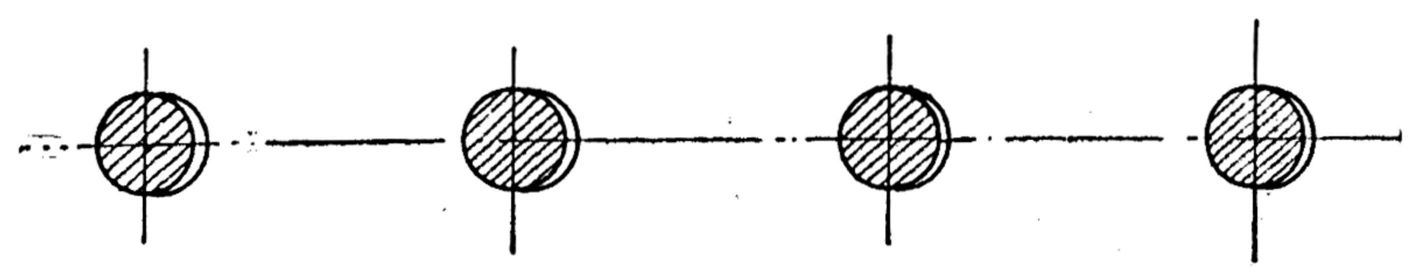

Fig. 1.

this force is the bending moment divided by the modulus of section of the axle when worn out. If the tractive force exceeds the resistance caused by the adhesion of the wheels, slipping will take place. In ordinary types of engines having two or more pairs of drivers, the parallel rods transmit to the other pairs of wheels a proportional amount, so that each pair has to resist an equal turning moment. It is a generally accepted proposition that an equal force is transmitted to each pair, or that the maximum turning moment on any pair is equal to the piston thrust divided by the number of pairs of drivers. This fact being established, it follows that the parallel rod prevents the entire thrust being borne by the main axle and transmits it equally to all the drivers. If the lost motion is excessive, or if a lack of parallelism exists between the axles and the crank pins, slipping will take place at every revolution to adjust the irregularities. The stress on the main axle under these conditions will be increased up to the limit of adhesion, an excess amount probably not exceeding 25 or 30 per cent. of the normal stress.

When the crank is on the dead center the effect of lost motion in the rods or driving boxes is felt at its maximum. It is probable that ordinarily when there is much lost motion in the parallel rod bearings, the main axle will be also loose enough in its box to allow the load to be distributed among the drivers as at other portions of the stroke. There is also a certain amount of spring or deflection in the axle and crank pin, which occurs before the maximum stress is reached, and assists materially in equalizing the pressure between the drivers, taking up in a de-

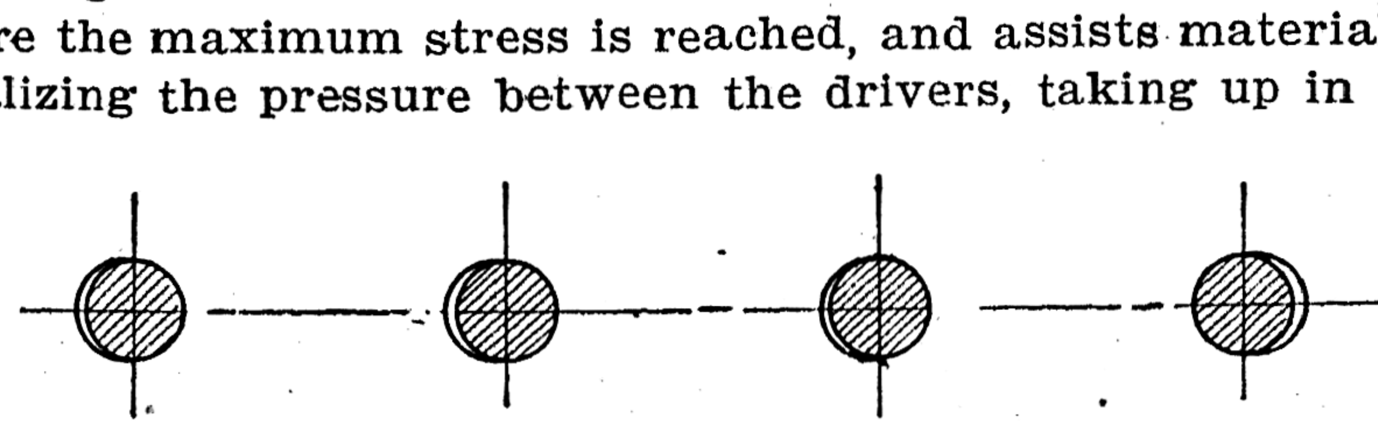

Fig. 2.

gree the lost motion in the rod bearing. The possibility of overstrain in the main axle due to excessive slack in the parallel rod bearings, was carefully considered in the suggestion that the fiber stress should be decreased as the number of axles increases.

Mr. Henderson assumes that only the main axle is subjected to increased stresses due to excessive slack or improper adjustment, whereas, under certain conditions any of the other axles may be required to resist considerably more than their normal loading. Fig. 1 shows the crank pins of a consolidation engine at the commencement of the stroke, with an equal amount of slack in front of all the pins. This is the condition assumed by Mr. Henderson. It is evident, however, that a slight looseness in the main driving-box and spring in the pin and axle will in most cases prevent the entire piston thrust from being resisted by the main axle.

Fig. 2 shows the slack on the back of the main, second and fourth, and in front of the first pair. Here the first axle has to withstand an overload, unless relieved by looseness in the boxes or springing.

In Fig. 3 the slack is back of the main, first and second, and in front of the fourth pair, the latter axle having to withstand the overload.

Were it a fact that the working stress on the main axle of a consolidation engine should be taken as the entire piston thrust, irrespective of the number of axles, it could be shown by a number of instances of engines which have been in service for 15 or 20 years, during which time comparatively few breakages have occurred, in spite of the fact that the assumed stresses would figure up to 30,000 to 32,000 pounds per square inch for hammered iron axles, undergoing, roughly speaking, about 12 millions of repetitions of alternating loads per year.

The ability of wrought iron to resist repeated changes of loading is now so well known that it is unnecessary to go into details, but merely to state that breakages may always be expected to occur after a few million changes of load at from 16,000 to 25,000 pounds stress per square inch, the former being for reversing or alternating tension and compression, and the latter a load applied and entirely removed, but always acting in the same direction.

An axle under a freight locomotive is subjected to an alternating stress in which the reversals, although they do not reach the maximum yet when the engine is moving slowly and cutting off steam at, say, 80 per cent., do reach 50 or 60 per cent. of the maximum stress acting in the opposite direction. The breaking stress after two or three million repetitions could, therefore, reasonably be taken at about 20,000 pounds per square inch for good wrought iron. This is the breaking load for this range of stress as opposed to the static breaking load once applied, as seen in the testing machine, and known as the ultimate strength of the material, neither having any margin of strength or factor of safety, the latter breaking quickly with a single load steadily applied and not removed until after fracture, and the other breaking equally as surely, although not so quickly, after the required number of changes of load have taken place.

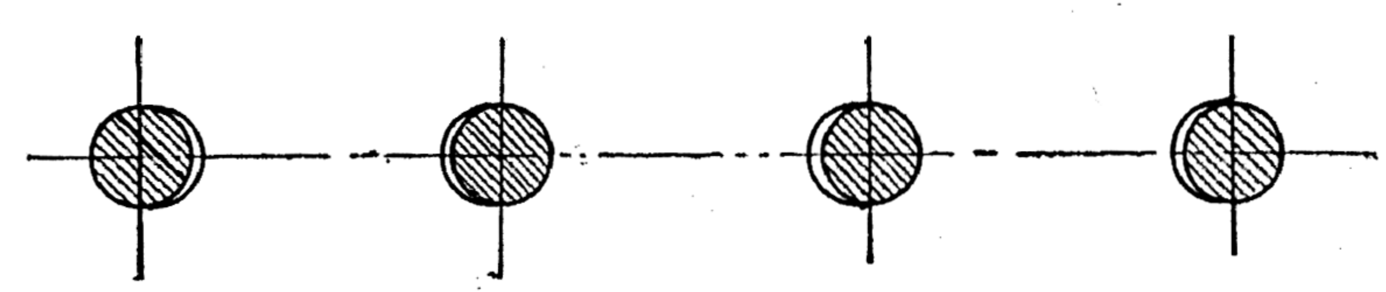

Fig. 3.

Again, the working stress of an axle can be approximately determined by the amount the rear axle of an eight-wheel engine can withstand, the parallel rod transmitting the force and the pin and axle being, therefore, entirely unsupported. Here instances can be cited where fractures usually occurred with iron axles when the stress exceeded 12,000 or 13,000 pounds. It seems unreasonable, therefore, to consider it necessary to design the main axle of a consolidation engine to resist the entire piston thrust unaided by the parallel rod, and keep the fiber stress down to, say, 8,000 to 10,000 pounds per square inch, figures which would usually be considered the maximum safe stress for this manner of loading. Taking the higher figure of 10,000 pounds, a mogul, consolidation or 10-wheel engine having 20-inch cylinders, steam pressure 200 pounds and a lever arm of 21 inches (center of main rod to center of frame), would require an axle of 11 inches diameter when worn down to the limit to resist the piston thrust alone.

For crank pins the support of the parallel rod was duly considered. On engines having the main rod outside it is comparatively small, so that the simplest method is to disregard it and use a higher working stress, as suggested in page 125 of the April number, and shown graphically in Fig. 6, page 154, of the June number. The distance from the wheel face to the center of the parallel rod is usually from 2 to 3 inches, making the counter-moment small in proportion to the bending moment, so that, after all, the real question is whether it is more desirable to use the higher stress without and the support of the parallel rod, or the lower stress, when it is taken into consideration. The latter seems to me the more rational, although more complicated, method.

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Pneumatic tubes for the transmission of mail between the New York and Brooklyn Postoffices were put into service Aug. 1. The tubes pass over the Brooklyn Bridge, and are provided with three expansion joints on the structure. The cost of the plant, including the double-pipe system, was $60,000, and the annual rental paid to the New York Newspaper and Transportation Co. for carrying first-class matter is to be $14,000 for a three-years' lease. The system is similar to that illustrated in our issue of November, 1897, page 379.