the high spots of the surface to hit the passing air, and as the friction between air and air is less than between air and a smooth solid body the resistance of rough surfaces is smaller. It would be worth while to put this theory to a practical test on railways, but it is a question if it could be used to advantage, since a rough surface offers better opportunities for dust and soot to settle and cling to the car, and is harder to dislodge in the cleaning yards, than from smooth surfaces, some advantages could be derived in the general appearance of the car, since it is impossible to prevent slight buckles with even the very best of care and the use of so-called "patent level" material, which are shown up by a glossy coat of varnish, which defect could be effectively covered by the use of a dull finished surface.

Bearing friction, one of the main factors of train resistance, is one of the points that was apparently neglected until quite recently. We must concede, of course, that a lot of improvements have been made in the use of antifriction metals, but the main issue, the conversion of gliding friction into rolling friction, which would signify a considerable reduction in draft resistance, has been woefully neglected. I am sure that this neglect, or rather the lack of interest shown in taking this matter up for high class passenger rolling stock has had many reasons that could serve as an excuse. Excluding ball bearings entirely, the reasons held against the application of roller bearings were the high pressures to which rollers were subjected to, the complication of design and the high initial cost of the bearings, the possibility of an absolute locking of axle, if one of the rollers should break, and the danger resulting therefrom to the passengers, the difficulty of repairs, the end thrust that has to be taken care of, and a lot of other objections of lesser importance.

Several leading railways of the country carried on experiments in years gone by, but the result was in many instances not encouraging. The pressures on the rollers, made of impure and not properly hardened steel, were excessive, the rollers broke, the wear was excessive where pressures had exceeded a certain limit, so that the application of roller bearings was confined to light rolling stock only, and although it was conceded that the results derived from roller bearings were favorable when the cars were run at low speed, the draft resistance at higher speeds was increasing rapidly and exceeded in some cases that of gliding friction bearings. As to these objections much can be said that will put the roller bearing in a more favorable light for consideration.

Heat treatment of special high grades of steel, and the use of vanadium in the manufacture of steel, has made the rollers tougher and harder, so that they can be subjected to higher pressures with impunity.

End thrust bearings were designed to take care of the end thrust of axles, and it can be only a question of time and experiments until a roller bearing for cars will be evolved that will meet all requirements, so that the saving in drawbar pull will more than counterbalance the extra expense of roller bearings.

Experiments were made by Prof. W. P. Graham, of Syracuse university, in 1905, in which the merits of roller bearings were clearly demonstrated by the power consumption of electric street cars equipped with roller bearings and without roller bearings. These experiments showed a saving of power of about 50% by the use of roller bearings, compared to gliding friction bearings of the old type. As the testing facilities must have been of a crude kind, no definite information was given regarding the power consumption per ton mile at varying speeds, the speeds being in every case below 25 miles an hour, so that not many deductions can be drawn from this test. No special test was made to show why the resistance of roller bearings should increase more rapidly after attaining a certain speed limit than gliding friction bearings, and although we can surmise that this is caused by the crowding of rollers, which thus set up a gliding friction among themselves, taking place at double the circumferencial speed of one roller, we have no confirmation by test. The use of. and experiments with, roller cages would be advisable, because they are designed to prevent such a state of affairs. Further provisions will have to be made to prevent accidents in case of breakage of rollers, and special care will have to be taken to make the journal as dust tight as possible. That the roller bearing has been a success in all those places where the power is limited has been proven by the use of these roller bearings in harvesting machinery, and this is still more surprising since we know that the working conditions of agricultural machinery are worse than those of a truck in motion.

One other factor in making up our drawbar resistance is slip. The percentage of slip is varying and increases with the difference in diameters of mated wheels on one axle. Although all our railway shops pay special attention to this fact, when they are turning new or repaired passenger equipment out of their shops, it is impossible to maintain equal diameters for any reasonable length of time, since the difference in finish of cut, material, carbon percentage, hardness, etc., are different in every wheel and in every brakeshoe, so that the best diameter balance will be upset in the shortest time. To investigate the amount of slip in actual service and the increase in drawbar pull necessitated by these gliding friction losses at high pressure would be highly interesting, the more so since the drawbar pull alone is only one factor that is mentioned here. The wear on tires and rails must be excessive, since the pressure per square inch of contact between wheel and rail is very high.

To prove this assertion the following figures, based on a theoretical modulus of gliding friction of 0.75 between steel and steel for exceptionally high pressures per square inch and without lubrication, are set down.

The assumption is a train, consisting of 6 cars of 60 tons weight each, the diameter of wheels differing ¼ in., and the train being carried a distance of one mile. The foot pounds of drawbar pull to move a train at a speed of 45 miles an hour, with a pull of 17 lbs. per ton, is equal to:

$17 \times 6 \times 60 \times 5280 = 32{,}313{,}600$ ft. lbs.

Revolutions of axle with a standard 36 in. wheel per mile=

$$\frac{5280}{3.1415 \times 3} = 559 \text{ revolutions per mile.}$$

Slip per revolution in inches $= 0.25 \times 3.14159 = 0.785$ inches.

$$\text{Total slip in ft. per mile} = \frac{0.785 \times 559}{12} = 36.56 \text{ ft.}$$

Friction loss in ft. lbs.=

$$\frac{36.56 \times 6 \times 120000 \times 0.75}{2}$$

=9, 875,000 with load of train.

The friction loss in ft. lbs. with load of train is, therefore, equal to 30.5% of the total drawbar pull.

These figures are, of course, only based on theoretical assumptions, and are used to point a way to investigate one of our heaviest power losses. There is no doubt that the percentage of this resistance will change materially with the speed of trains and weight of cars, and further we maintain that the bulk of flange wear is caused by this difference in diameter of tires, since there is always a lot of play between journal box and journal brass, which permits a constant twisting of the axle, caused by the afore-mentioned forces, which results in the tendency to climbing of flange on rail.

The trouble of slip could be overcome by the application of individual wheels to the truck, or simpler still by the use of sleeve shafts. It would be going too far to advocate the application of this principle, for which a practical solution will have to be found to our present day trucks, but the time will come some day, when the reduction of draft resistance will become imperative, and if we could find means of reducing our draft resistance some 50 to 75% it would mean that we could haul

FREIGHT ENGINES

Railroad	Size of cylinder inches	Boiler pressure pounds	Length of fire box inches	Width of fire box inches	Depth of fire box front – inches	Depth of fire box back – inches	No. of flues	Sizes of flues inches	Length of flues	Dia. of boiler at first course – inches	Grate area sq. ft.	Heating surface of fire box – sq. ft.	Heating surface of tubes – sq. ft.	Flue heating surface / Fire box heating surface	Total heating surface / Grate area	Flue heating surface / Grate area	Fire box heating surface / Grate area	Sq. ft. of H.S. per each cu. ft. of total cyl. vol.	Fuel	Heating surface of small superheater tubes	Combustion chamber heating surface	Water bar heating surface	
A	20x28	200	72	84	64	54	238	2	14-8½	61	42	143.23	1971	13.7	50.3	47	3.41	207	Bit. coal		55	22.82	o Estimated
G	20x28	200	113	41	79	79	342	2	13-2⅜	68	31.5	203.4	2342.8	11.5	80.9	74.4	6.45	249.5			None		
G	20x28	200	113	41	79	79	24 / 212	5 / 2	13-2⅜	68	31.5	203.4	1866.9	9.2	65.7	59.2	6.46	203	·	430			
D	21x28	200	108⅛	63¼	77	62	390	2	13-6	75	52	180.4	2754.6	15.2	56.4	52.9	3.47	261			·		
K	21x32	200	107⅛	67¼	77	63	398	2	14-6	78⅛	49.5	168	3021	18.2	64.3	61.1	3.3	249					
P	21x32	200	122	96¼	55	47	408	2	14-3⅜	78	81.5	195	3064	15.7	40	37.6	2.39	254					
B	22x30	200	108	66	73½	63¾	425	2	15-0	80	49.5	180	3319	18.4	70.7	67	3.64	264					
E	22x30	200	108	66	74	63¾	413	2	15-0	78½	49.5	210	3250	15.25	70	65.3	4.3	262	Oil				o
E	22x30	200	108	66	74	63¾	413	2	15-0	80	49.5	210	3250	15.25	70	65.3	4.3	262	Bit. coal				o
L	22x28	205	107	66	72	61	373	2	13-6	69½	49.4	166.5	2677	16	57.5	54.1	3.37	230	"				
Q	22x30	205	107½	75	70	57	324	2	16-0	74¼	56.2	179	2595	14.5	49.4	46.2	3.19	209.8					
M	22x28	205	107	66	65 5/16	51 11/16	373	2	13-8½	69½	49.1	166.5	2675	16.1	57.6	54.5	3.4	229					
O	22½&35x32	210	96⅞	75¼	72¾	72¾	353	2	15-0	67	50.62	168.2	2757.1	16.4	57.8	54.5	3.3	58.2	Bit. Mallet		None		
S	23x30	185	108⅛	72¼	77¾	69¾	328	2	20-0	74	54.2	190	3420	18	66.7	63.2	3.5	251	Bit. coal				
F&N	23x32	200	108⅛	75¼	76¾	58⅛	438	2	15-0½	80	56.47	210	3428	16.3	62.7	60.8	3.7	236					
N	23x32	200	108⅛	75¼	76¼	58⅛	34 / 268	5⅜ / 2	15-0½	80	56.47	210	2813.1	13.4	53.7	49.8	3.73	196.5	"				
E	23¾x30	200	120⅛	84	87½	73½	495	2	20-6	80½	70.4	286.7	5320	18.55	79.7	75.5	4.08	366	Oil Mikado				o
D	24x28	205	110¼	72	72	58¾	465	2	15-0	76½	55	187	3653	19.5	69.8	66.4	3.4	281	Bit. coal				
L	24x28	205	110¼	72	70	58	465	2	15-0	76¾	55.13	187	3652	19.5	69.7	66.4	.34	260.9					
Q	24x32	205	120	84	81	71½	389	2¼	21-0	78	70	228	4789	21	71.5	68.5	3.26	299					
Q	24x32	205	120	84	81	71½	34 / 218	5½ / 2¼	21-0	78	70	228	3702	16.2	56.2	52.8	3.26	235	·	998	·		
M	24x28	205	110⅛	72	67¼	54⅜	465	2	15-0	76¾	55.1	187	3632	19.5	69.7	66.4	3.4	260.9			·		
N	25x32	180	108⅛	75¼	76¾	58⅛	34 / 262	5⅜ / 2	15-0½	80	56.47	210	2770	13.2	52.8	49.2	3.73	161	·	895			·
F	26x28	200	108⅛	75¼	84	67	34 / 248	5⅜ / 2	20-0	85	56.5	235.6	3552	15.08	67.1	62.9	4.17	236.5	"				o
G	26&40x30	200	126	78¼	75½	70½	401	2¼	21-0	84	68.4	232	6162	26.5	93.5	90	3.4	103	Oil Mallet		None		
E	26&40x30	200	126	78¼	75½	70½	401	2¼	21-0	84	68.4	232	6162	26.5	93.5	90	3.4	103	Bit. Mallet		None		
Q	26&41x32	210	126	114	89½	67	38 / 277	5⅜ / 2¼	24-0	90	67	321	5205	16.2	82.5	77.7	4.8	80.7	"	1002	63		
C	27x30	170	108⅛	72¼	84⅜	72⅝	24 / 221	5⅜ / 2¼	21-0	78	54.2	223	3457	15.5	67.8	63.7	4.12	185	Bit. coal		None		
B			96	42	65½	65½					28	DATA	NOT	FURNISHED					·		-		
B			120	42	65½	65½					35	"	"	"	"	"			·	·	·		
T	25x32	170	108⅛	70¼	83	63	438	2	15-2	81½	53	186	3458	18.5	68.8	65.2	3.5	197	·		·		
T	25x32	170	108⅛	70¼	83	63	36 / 263	5⅜ / 2	15-2	81½	53	188	2838.7	15.2	57.2	53.5	3.5	163.5	"	610	·		
T	21x26	200	102⅛	65¼	57½	56½	280	2	16-0	68⅜	46.27	152	2333.2	15.3	53.6	50.3	3.3	239	"		"		

o Estimated

NOTE

· Heating surface of small superheater tubes not included in total heating surface

· Heating surface combustion chamber included in heating surface of fire box

Freight Locomotives.—Dimensions of Fireboxes Reported as Giving Good Service.