Roller-Bearing Service in Locomotive, Passenger, and Freight Equipment

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Modern civilization is based on transportation, and the fundamental prime mover in transportation is the steam locomotive. Improvements in the steam locomotive affecting its efficiency and reliability are reflected in a corresponding manner in the entire transportation industry. The loads and stresses developed in locomotive service on bearings are exceptionally severe, and the consequences of failure are far reaching, and these conditions together have militated against rapid introduction of the roller bearing in locomotive service. The application of the roller bearing in passenger service has progressed in an encouraging manner over a period of years, but efforts to interest railroad men in application of roller bearings to a complete locomotive were unsuccessful, and finally convinced the Timken company of the desirability of building a locomotive equipped on all wheels with roller bearings and loaning it for an extended period of service to the railroads of the United States. The Timken locomotive was designed for application of Timken bearings on all of the drivers, engine-truck, trailer, and tender-truck wheels, on the Franklin booster, and on various elements of the control mechanism. The introduction of roller bearings on the drivers permits of higher rotative speeds, as the bearings surround the drivers completely, and eliminates pounds within the bearing boxes. Heating is eliminated, as the temperature rise does not exceed 25 deg above atmosphere. The wheel diameter was therefore selected between that prevailing for modern high-speed freight locomotives, averaging 70 in., and high-speed passenger locomotives, with 80-in. drivers. The economy in friction, estimated at 12 to 15 per cent, was utilized in increasing the diameter of the drivers over that of the modern freight locomotive, and developing through the saving in friction a drawbar capacity equivalent to the latter. The 73-in. wheel was therefore selected. The 4-8-4 wheel arrangement with four-wheel trailer truck was favored.

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The 73-in. wheel was therefore selected. Careful proportioning of reciprocating parts and rods permits of operating the 73-in. driver at speeds sufficiently high to handle the existing American passenger-train schedules.

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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.

The weight was held within the limits imposed on certain American roads, namely, 61,000 lb per driving axle, but in order to compare reasonably with much heavier freight power on other roads a duplex steam pressure was utilized in connection with the weight transfer between the drivers and trucks, making available a weight of 66,000 lb per driver and a steam pressure of 250 lb on roads permitting of the heavier axle loads. The supporting of the large boiler capable of developing maximum power at high speeds favored the adoption of the 4-8-4 wheel arrangement with four-wheel trailer truck.

Locomotive Specifications

The locomotive companies, the motive-power departments of the railroad companies, and the specialty companies assisted wholeheartedly in the selection of specifications for the locomotive and made available their enormous funds of information and data on locomotive design. The locomotive is a composite of specifications of a number of trunk-line railroads and was built as large and powerful as the clearance limitations of the principal railroads of the United States would permit. Specifications are as follows:

Owner, Timken Roller Bearing Company	
Builder, American Locomotive Company	
Type of locomotive, 4-8-4	
Service, freight and passenger	
Maximum rated tractive force (boiler pressure, 235 lb), lb	59,900
Rated tractive force of booster (boiler pressure, 235 lb), lb	12,000
Tractive force at starting (boiler pressure, 235 lb), lb	71,900
Maximum rated tractive force (boiler pressure, 250 lb), lb	63,700
Rated tractive force of booster (boiler pressure, 250 lb), lb	12,800
Tractive force at starting (boiler pressure, 250 lb), lb	76,500
Weight on drivers ÷ tractive force (boiler pressure, 235 lb)	4.10
Weight on drivers ÷ tractive force (boiler pressure, 250 lb)	4.14
Cylinders, diameter and stroke, in	27 × 30
Valve gear, Walschaert type; valves, piston type, size, in	12
Maximum travel, in	8 1/2
Steam lap, in	1 1/2
Exhaust clearance, in	1/4
Lead, in	1/4
Cut-off in full gear, per cent	85
Weights in working order (boiler pressure, 235 lb):	
On drivers, lb	246,000
On trailing truck, front, lb	48,500
On trailing truck, rear, lb	55,500
On front truck, lb	67,500
Total engine, lb	417,500

Weights in working order (boiler pressure, 250 lb):	
On drivers, lb	264,000
On trailing truck, front, lb	34,500
On trailing truck, rear, lb	59,000
On front truck, lb	60,000
Total engine, lb	417,500
Total tender, lb	294,000
Total engine and tender, lb	711,500
Wheelbases:	
Driving, ft	19 3/12
Driving, rigid, ft	12 10/12
Total engine, ft	45 10/12
Total engine and tender, ft	89 9/12
Wheels, diameter outside tires:	
Driving, in	73
Trailing truck, front, in	36
Trailing truck, rear, in	44
Front truck, in	33
Journals, nominal diameter:	
Driving, main, in	11 1/2
Driving, others, in	11 1/2
Trailing truck, front, in	7 × 14
Trailing truck, rear, in	9 × 14
Front truck, in	7 × 12
Boiler (extended wagon-top type):	
Steam pressure (weight on drivers, 246,000 lb), lb	235
Steam pressure (weight on drivers, 264,000 lb), lb	250
Diameter, first ring, inside, in	84 1/4
Tubes, 66 in number, diameter, in	2 1/4
Flues, 194 in number, diameter, in	3 1/2
Length over tube sheets, ft	21 1/2
Grate area, sq ft	88.3
Heating surfaces:	
Firebox and combustion chamber, sq ft	360
Arch tubes, sq ft	18
Thermic siphons, sq ft	105
Tubes and flues, sq ft	4637
Total evaporation, sq ft	5120
Superheating, sq ft	2157
Combined evaporation and superheating, sq ft	7277
Tender:	
Water capacity, gal	14,200
Fuel capacity, tons	21
Wheels, diameter, in	33
Journals, normal diameter and length, in	6 × 11
Weight proportions (boiler pressure, 235 lb):	
Weight on drivers ÷ total engine weight per cent	59
Weight on drivers ÷ tractive force	4.10
Total weight engine ÷ comb. heating surface	57.4
Boiler proportions (boiler pressure, 235 lb):	
Tractive force ÷ comb. heating surface	8.24
Tractive force × diam. drivers ÷ comb. heating surface	602
Firebox heating surface ÷ grate area	5.47
Firebox heating surface per cent of evap. heating surface	9.44
Combined heating surface ÷ grate area	82.4
Weight proportions (boiler pressure, 250 lb):	
Weight on drivers ÷ total engine weight per cent	63.4
Weight on drivers ÷ tractive force	4.14
Boiler proportions:	
Tractive force ÷ comb. heating surface	8.77
Tractive force × diam. drivers ÷ comb. heating surface	639

Roller-Bearing Applications

The driver application was made without adjustable mechanism. Hardened steel trunnion guides are mounted on the bearing housings centrally pivoted to permit of the housing following track irregularities while maintaining full surface contact with the hardened steel liners on locomotive frame.

The forces due to piston thrust, therefore, are transmitted and absorbed in a complete train of moving parts composed of hardened steel. These comprise the pedestal liner, trunnion guide, hardened wear plates on the bearing housing, and the inner and outer races of the bearing, together with the rolls. The mounting construction is shown in greater detail on the elevations and cross-sections of the locomotive (Figs. 1 and 2). The accessibility of the bearings for inspection at major shopping periods is illustrated by a view of the axle assembly (Fig. 3).

The complete housing of the driver axles and the use of bearings restraining the axle on a complete circle of 360 deg eliminate pounding while under steam and while coasting, and, together with careful proportioning of reciprocating parts, permit of operation of 73-in. drivers at speeds of 85 mph.

The engine truck follows in general the construction of the driver. It utilizes the integral split housing, the trunnion guides, and one bearing for each wheel. The trailer trucks are designed for direct replacement of plain bearings, making no change in trucks, pedestals, or springs. The trailer is an outboard application and requires the double-bearing construction. The tender truck is likewise designed for direct replacement, using tender trucks, pedestals, equalizers, and springs designed for A.R.A. plain bearings.

The booster was applied on the locomotive to increase the

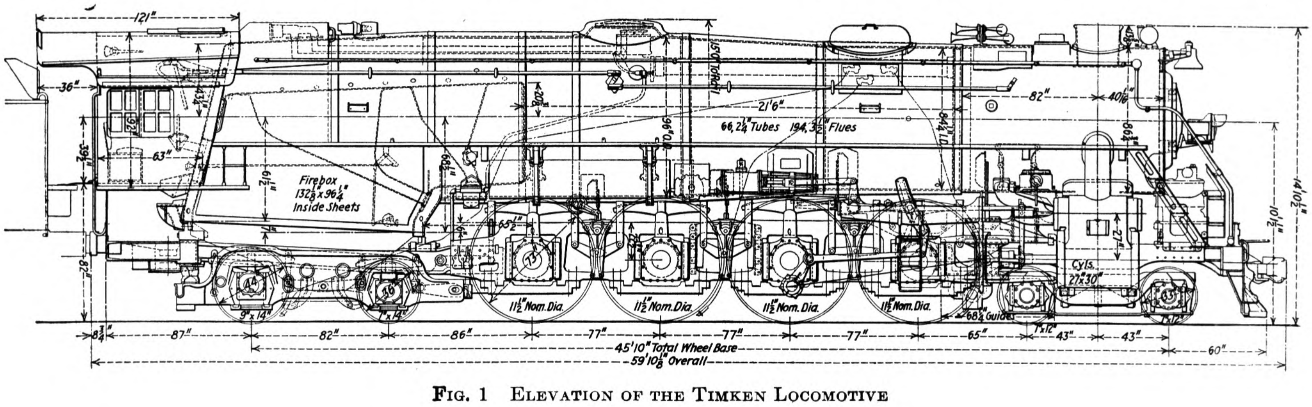

Fig. 1 Elevation of the Timken Locomotive

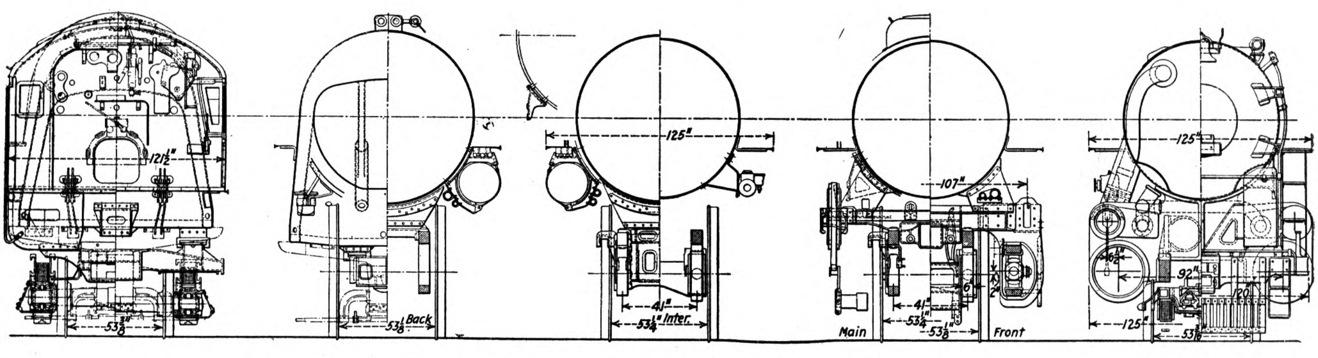

Fig. 2 Cross-Sections of the Timken Locomotive

acceleration, to increase the starting power and utilize the full boiler capacity in starting, and to handle heavy trains without helpers over ruling grades, especially where stops must be made on these grades. The booster has been found especially valuable in handling heavy passenger trains in mountainous territory without helper service. The booster is equipped on the crankshaft with two double bearings, and the idler gear is equipped with the Timken quad bearing. The application of roller bearings to the booster permits of its operation at higher speeds, and experience has confirmed the practise of dropping in the booster on adverse grades at speeds of 22 mph, holding approximately this speed over the top. The booster bearings have been examined periodically when trailer wheels were removed for attention to tires and have been found in perfect condition.

The lateral motion of Alco design and construction is applied on No. 1 driver. The lateral motion involves no change in bearing housing, but the trunnion guide is provided with $1^1/_4$-in. total lateral freedom, with respect to the pedestal liner, whereas the lateral freedom on drivers 2, 3, and 4 is $^1/_4$ to $^5/_{16}$ in. The lateral motion device on the front driver permits of free operation over 20-deg curves and limits the rigid wheelbase to 12 ft 10 in. It is of interest to note that the lateral freedom on all drivers after two years' operation measures the same (within 0.0020 to 0.0037 in.) as when the locomotive left the shop. This is an indication that the desired lateral freedom in drivers having been ascertained, this freedom can be built into roller-bearing housing construction with the expectation that it will not be subject to change over a period of years of service.

Crosshead guides and valve link operate in open atmosphere and are subject to the lapping action of dust and grit. An effort was made to reduce the wear of these parts by making the crosshead guides and the valve links of Timken bearing steel. This steel has a tough core with approximately 6 per cent alloy and is cased approximately $^1/_4$ in. deep to eutectoid to develop a surface material of exceptional wear-resistant characteristics. The crosshead slipper is lined with tin, as this permits atmospheric dust to imbed in the soft tin and avoid cutting the crosshead guide, and in addition the guide is of sufficiently hard material so that atmospheric dust and grit, composed largely of silica, does not wear and abrade the guide. The experience of two years indicates the soundness of this selection of materials, as the crosshead guides show very little evidence of wear and have taken on an exceedingly high polish of great hardness.

The link and link block of the same material and treatment have about 0.010 in. looseness after two years and have not received any attention and do not now require any. It is of interest to note that the crosshead slipper operates at a temperature of 20 to 30 deg above atmosphere.

The side-rod bearings are of the floating bushing type. The main and main side-rod floating bushings are of the conventional type. Pins 1, 3, and 4 have hardened bearing-steel bushings pressed in the rods. These bushings have a spherical bore of a radius approximately equal to the distance between wheels. A floating bronze bushing operates between the pins and the fixed steel bushing and has parallel bore and o.d. crowned approximately the same as the spherical bore of the steel bushings. This results in an equal distribution of wear on the pins and may have assisted in prolonging the life of the pin bearings.

Reciprocating parts were given special thought and were reduced in weight in accordance with the best American practise. The specifications given the builder, the American Locomotive Company, was a speed of 85 mph with a dynamic augment not exceeding 10,000 lb. This is 12 miles in excess of diameter speed and was made available by the use of low-carbon-nickel steel, as per specifications of the International Nickel Company in the main rod, side rods and pins, and the use of the hollow piston-rod, heat-treated steel castings for crossheads, of composition developed by Union Steel Castings Company. The total reduction in reciprocating parts per side under conventional practise of locomotives of similar capacity is 460 lb.

The balancing system is of the cross-balance type, involving distribution of approximately one-half the overbalance on the main drivers and the other half of the overbalance distributed on pins 1, 3, and 4.

The detailed attention given to reciprocating parts and balancing system, together with the complete housing of the axles of the roller bearings, has produced a locomotive of exceptionally smooth operating characteristics. The operation in the 70's is exceptionally smooth, and vibration is not excessive at 85 mph.

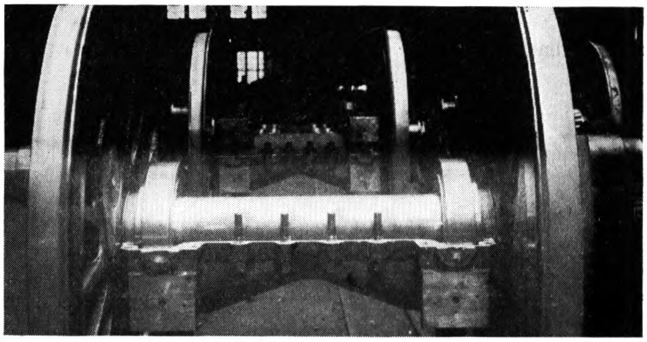

FIG. 3 DRIVER MOUNTING TAPER BEARINGS AND LOWER HALF HOUSING IN POSITION
(Far view of assembled housing.)

Lubrication of the drivers and of truck wheels is accomplished by immersing the bearings in a bath of oil. The oil level is approximately $^1/_2$ in. below the enclosure level, permitting the rolls to dip into lubricant at each revolution. The straight-line elements of the taper roller bearing permit of close running clearances between the axle sleeves and the enclosure. These close clearances fill up with heavy ends of the lubricant and form an effective seal. The lubricant has been changed but three times in two years' operation, and the added lubricant between changes has been slight.

ROLLER-BEARING INFLUENCE ON LOCOMOTIVE DESIGN

The introduction of the roller bearing modifies in a number of ways the characteristics of the locomotive, which can be briefly enumerated.

Thrust plates are eliminated entirely. The thrust reactions due to curvature or flange thrust from any cause are taken on the roller-bearing surfaces of the tapered bearings, and no provision need be made anywhere in the locomotive for thrust plates. Maintenance due to the presence of thrust plates is eliminated entirely.

The engine-truck wheels do not require the large hub surface on the inside, and wheels of symmetrical design, having hub diameters the same on both sides of the web similar to those used in tender service, provide the best construction for roller, bearing engine trucks. This method provides a stronger wheel inasmuch as the heat treatment is more uniform, the cost is reduced, and likewise the unsprung weight. The four years' experience with the engine-truck wheels indicates an increase in life. The life of the wheel is limited by tread wear only. The heat rise of the bearings is about 15 deg, and consequently checking of the thrust surfaces of plain bearings is entirely eliminated. A number of engine trucks have operated in excess of 400,000 miles without wheel replacements.

Axles can be selected for stress and deflection only; no provi-

TABLE 1 STARTING EFFORT OVER 72,000 POUNDS FROM ERIE TEST

Test No.	Mile post	Drawbar dyn. car roll	Grade correction	Actual effort	Factor adhesion	Ton in train
134-A......	167	80,640	+6072	86,812	3.72	2348
134-B......	64	75,000	−2508	72,492	4.46	2603
135-B......	255	77,200	−1650	75,550	4.27	2137
136-A......	206	82,520	+5428	87,948	3.68	2697
137-B......	201.75	76,100	− 264	75,836	4.26	2107
132-A......	238.2	71,500	+4554	76,054	4.25	....
132-A......	284	73,200	+ 990	74,190	4.35	....
132-A......	239.1	72,500	+3432	75,732	4.27	....
129-A......	164.75	72,500	......	72,500	4.45	2602

Note: Total weight on driver plus weight on rear booster axle = 264,000 + 59,000 = 323,000 lb. Factor of adhesion equals total weight divided by starting effort. Booster used in all of the starting efforts.

sion need be made for wear. The temperature rise varies from 15 deg at the front end of the engine to 40 or 50 deg under the tender. No provision need be made in the axle as regards increased diameter to provide for wear. Use of the roller bearing will permit of the axle to be made of alloy steel with treatment to develop the superior physical properties of alloy steel, the reason being that the axles are not subject to heat checks due to daily cycles of high temperature and ultimate cooling. The axles on the Timken engine are composed of carbon-vanadium normalized steel. The driver axles were of uniform diameter, $11^1/_2$ in. This reduced the diameter of the main from 1 to $1^1/_2$ in. under corresponding plain-bearing original equipment.

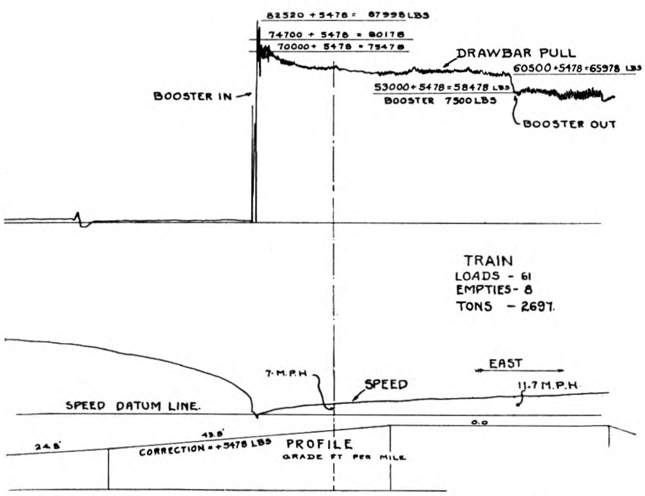

Fig. 4 Timken Locomotive, Starting Drawbar Pull, Curve 1
(Erie test 136-A, Kent to Meadville east.)

The bearing housings are provided with separate oil pockets for each bearing on the engine truck and drivers. This avoids the flow of lubricant from one side to the other on tracks of high super-elevation.

The reduction in vibration following the use of roller bearings indicates a reduced maintenance expense on account of the noticeable absence of loose bolts and loosened clamps and broken pipe connections, these features being frequently commented upon by railroad men servicing the locomotive.

The factor of adhesion is modified in certain respects with the introduction of the roller bearing on drivers. A higher percentage of the piston thrust is communicated to the drivers at point of contact with the rail, and consequently the power "input" to the locomotive for given "drawbar pull" is reduced. A saving in friction can be conservatively estimated at 12 per cent. The reductions following this modification would be that the adhesion factor should be slightly higher if the locomotive was not modified in any other respect, and on the other hand, the experience to date would indicate that the cylinder capacity could be reduced in accordance with the saving in friction. Probably the best compromise would be to split the saving in power and put one-half of it in reduced cylinder capacity and the other half in reduced factor of adhesion. The adhesion factor of the Timken locomotive is 4.14. This gives uniformly good service, but comment has been made of slipperiness under bad rail conditions.

The subject has been carefully studied, and the opinions of experts on the subject have been sought. The opinion of one locomotive expert was that locomotives have always slipped and always will slip under certain conditions, even though the factor be made 100 to 1. The fact that the roller-bearing engine is new has made it a target for criticism on any point that is even slightly different from conventional practise, and if a slip would occur it would be blamed on the roller bearings even though plain-bearing engines had slipped over identical track for one-hundred years under certain atmospheric conditions. On an observation on one railroad over a distance of more than 450 miles, with full tonnage train, there was noted only a slip of half a

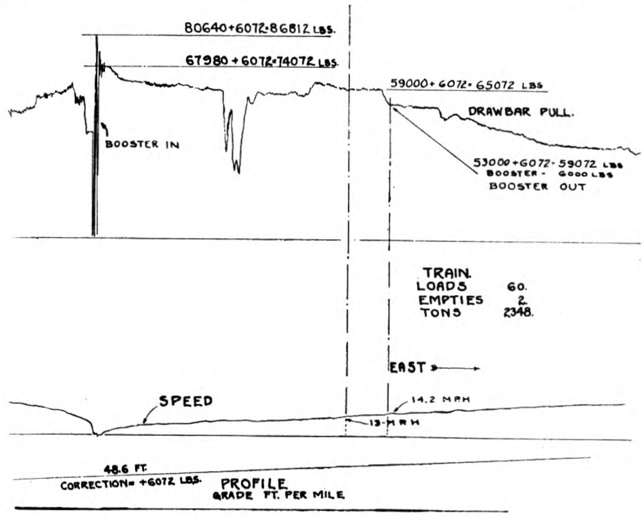

Fig. 5 Timken Locomotive, Starting Drawbar Pull, Curve 2
(Erie test 134-A, Marion to Kent east.)

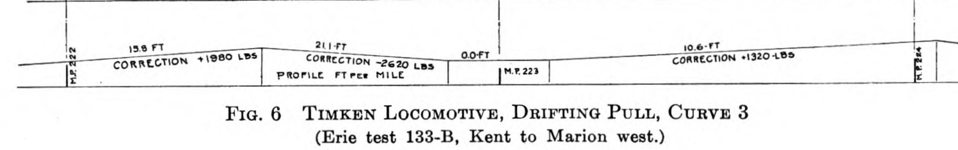

Fig. 6 Timken Locomotive, Drifting Pull, Curve 3
(Erie test 133-B, Kent to Marion west.)

dozen revolutions on poorly maintained curved track at the ladder of a yard.

The starting characteristics of the locomotive on roller bearings is materially changed. The Timken engine has a normal starting drawbar pull of 63,700 lb, based on conventional formula. It has been frequently observed to develop starting power of 68,000 to 69,000 lb. Table 1 from Erie tests gives the data on a number of starts in excess of 72,000 lb with booster; some of these starts are in excess of 80,000 lb, the maximum being 87,998 lb. Two starts, curve 1 (Fig. 4) and curve 2 (Fig. 5), on Erie test at m.p. 206 and m.p. 267 indicate speed, grade, and test conditions and illustrate the high momentary surge at starting. The sustained power in starting is over 80,000 lb and illustrates also the effect of drawbar pull at the point of cutting off the booster. The added starting power contributed by the roller bearing is of particular value and adds to the capacity of the locomotive in handling heavy trains.

The free coasting is another of the outstanding features of the roller-bearing locomotive. Free coasting is of particular value in eliminating surges in passenger trains when the throttle is closed. The coasting characteristics of the engine are about equivalent to the free running of the train, and closing the throttle does not result in the bunching of the train on the engine and the consequent taking up of slack when the throttle is opened. The freedom from train surging with the roller-bearing locomotive has been frequently commented upon in passenger service. The curve 3 (Fig. 6) illustrates a drifting test on the Erie. The locomotive in drifting on a 21-ft grade at a speed of 33 mph developed a drawbar pull of 2900 lb on the dynamometer car. The pull of the locomotive on the train continued on an ascending grade of 10.6 ft to the mile, with a gradual decrease in train speed from 37 to 29 mph, at which point the throttle was opened. The train had 6 loads and 68 empties, a total of 2128 tons. The reduction in wear and tear of couplers and draft equipment on account of the continuous stretching of either freight or passenger trains is a valuable feature of the roller-bearing locomotive.

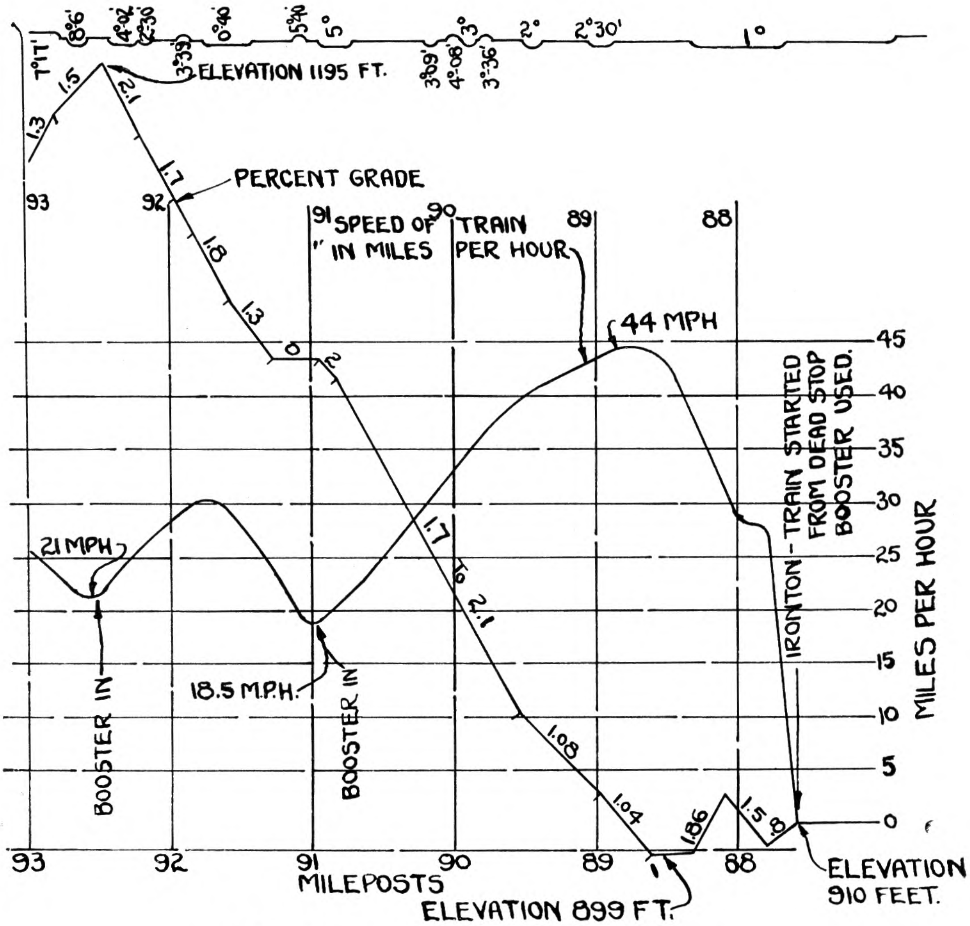

Fig. 7 Curve of Actual Speed of Timken Locomotive

(Pulling 18 passenger cars, of which 13 were Pullmans; total 1260 tons; passenger train No. 17, over critical Ozark Mountain grade between Ironton and Tiptop on run between St. Louis and Poplar Bluff, over Missouri Pacific Railroad.)

Performance Record

The performance record has been ably presented in the trade publications. The locomotive performed with equal efficiency in passenger and freight service, and of the 119,586 miles, 51,655 miles were in freight service (43 per cent) and 67,931 miles were in passenger service (57 per cent).

Passenger service includes operation on some of the fastest American trains, among which are the 20-hour mail trains on the Pennsylvania in New York–Chicago service, the mail trains on the same road in New York–St. Louis service, the "Sportsman" on the Chesapeake & Ohio, the "Erie Limited" the "Merchants Limited" on the New Haven, 109 runs on the Lackawanna in passenger service between Scranton, Pa., and Hoboken, N. J., the "Lehigh Limited," the "Sunshine Special" on the Missouri Pacific, the "Aristocrat" and "Overland Limited" on the Chicago, Burlington & Quincy between Lincoln, Neb., and Denver, Colo., and four months of hauling the "North Coast Limited" on the 906-mile run over three ranges of the Rocky Mountains between Missoula, Mont., and Jamestown, N. D.

The locomotive has handled all varieties of freight service, the heaviest being a 9960-ton coal train on the Chesapeake & Ohio. The outstanding freight service is the handling of heavy fast-freight trains at high speed on the New York Central, Pennsylvania, Erie, Boston & Maine, Lehigh Valley, Nickel Plate, Missouri Pacific, Chicago, Burlington & Quincy, and Northern Pacific.

Outstanding Runs

A non-stop run, Altoona to Enola on the Pennsylvania, 124 miles, with 115 cars weighing 8625 tons, was at an average speed of 25.7 and evaporated 8.33 lb of water per pound of coal. This trip was at a work rate of 221,427 gtm per train-hour, exclusive of locomotive.

A merchandise train was hauled between Crestline and Ft.

Wayne, on the Pennsylvania, consisting of 102 cars of 3986 tons, a distance of 131 miles, at a rate of 41.5 mph, at an evaporative rate of 8.2 and work rate of 165,457 gtm per train-hour.

Twelve passenger-equipment cars were handled unassisted on the east slope of the Alleghanies over a uniform mountain grade of 1.9 per cent, compensated.

Pennsylvania train No. 52, "New Yorker," Chicago to Crestline, made the 280 miles in 274 min running time.

The heaviest train hauled was on the Chesapeake & Ohio and comprised 134 coal cars, weighing with locomotive 10,219 tons. The run was made at the rate of 212,000 gtm per train hour.

The "Erie Limited" was handled between Hornell, N. Y., and Salamanca, N. Y., making up 25 minutes in a distance of 82 miles at an average speed of 59 mph.

The assignment on the Boston & Maine, a total of 35 runs being made, involved operation at temperatures of 20 deg below zero. An outstanding run was a train of 86 cars, 2040 tons, hauled 112 miles at a rate of 33.7 mph in sub-zero weather.

The passenger service on the Lackawanna was outstanding, involving 109 runs between Scranton, Pa., and Hoboken, N. J. This includes the Pocono mountain grade and long grades over the hills of northern New Jersey. A Lackawanna run working the engine to the limit was train No. 12, March 11, 1931, 11 cars, which left Scranton 56 min late, was detained 13 min at Stroudsburg, and made up the 69 min lost time in 112 miles running. The run over the division was made with 9 tons of coal. Speeds of 78 mph were attained on ascending 0.4 per cent grades.

The average speed of 54.8 mph in passenger service on the Lehigh Valley is outstanding. This involves running for long distances at speeds in excess of 70 mph.

On a Lehigh Valley freight run, Manchester to Sayre, 2410 tons, 67 cars, 89 miles were made in 1 hour and 52 min.

An interesting passenger run, which was repeated twice, was that of hauling of 18 passenger-equipment cars, including 13 Pullmans, over the Ozark Mountain grades on the Missouri Pacific between St. Louis and Poplar Bluff. These runs encountered maximum grades of 2.1, and the practise followed was to approach the grades at speeds of 50 mph, and as the speed fell the booster was dropped in at 22 mph, and in the three runs the train was handled over Tip Top grade at speeds of 15, 17, and 19 mph. These trains average 1260 to 1290 tons, and indicate with booster a tractive capacity approximating 80,000 lb. The chart (Fig. 7) shows the profile and speed which this train made, as taken from the valve pilot record.

An outstanding passenger-service record was on the Northern Pacific, where the "North Coast Limited" was hauled between Jamestown, N. D., and Missoula, Mont., a distance of 906 miles, which includes three Rocky Mountain ranges with grades of 2.2 and several grades of the Black Hills between Glendive, Mont., and Mandan, N. D. There were 19 trips made on this run, which developed and showed the adaptability of the roller-bearing equipment for long continuous service.

Operating Temperatures

The normal temperature rise of the bearings varies from 15-deg rise on the engine truck to 40-deg to 50-deg rise on the tender, the interesting feature being that the temperature rise of the drivers is only 15 to 20 deg above atmosphere. The normal condition of bearing housings in zero weather is with frost adhering to the bearing housings and the end of the axles. The photograph (Fig. 8) illustrates the condition of frost adhering to the engine truck and the driver axles and to trailer and tender boxes.

Maintenance

The maintenance work consists of the normal engine-house attention, including replacement of rod bearings, brake shoes, etc. The entire demonstration of two years' time, 120,000 miles, over every condition of topography and temperature, with mid-winter assignments on the Boston & Maine and Northern Pacific and

Fig. 8 Engine Truck, Driver, and Trailer at Completion of 900-Mile Run

(*A*, engine truck; *B*, driver; *C*, trailer.)

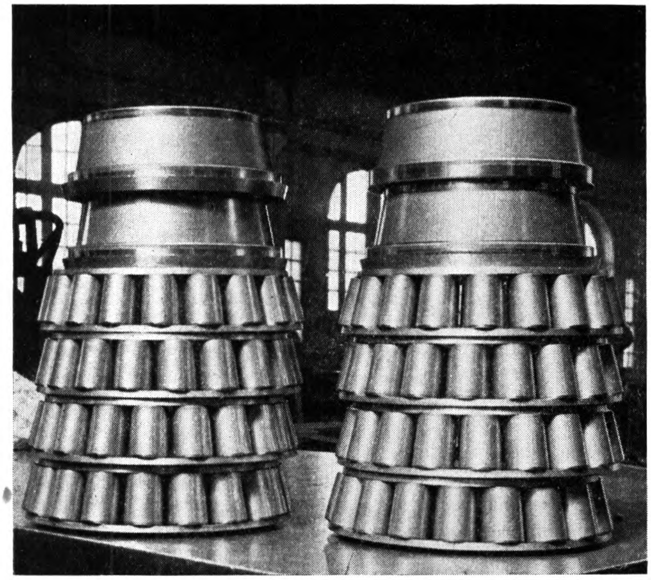

FIG. 9 TIMKEN LOCOMOTIVE DRIVER BEARINGS
(In perfect condition at 119,600 miles, or two years' work.)

mid-summer on the Missouri Pacific, was made without development of roller-bearing troubles on any of the wheels.

A looseness developed on the main driver on the Burlington, which was taken up by pressing on each wheel center an additional $^{1}/_{32}$ of an inch. The looseness was due to seating of the bearing in the housing; this has been corrected on subsequent designs by increasing the backing of the bearings.

TABLE 2 PERFORMANCE OF MAIN AND SIDE-ROD BUSHINGS

	Back end main		Main side rod		Front wheel		Back int.		Back wheel	
Mileage	Right	Left	Right	Left	Right	Left	Right	Left	Right	Left
146	..	..	..	..	(1)	..	..	..	(1)	..
1,462	(2)	..	..	..	..	..	..	..	..	..
2,098	..	..	..	..	(3)	..	..	..	..	..
6,582	..	..	..	..	(4)	(4)	(4)	(4)	(4)	(4)
42,010	..	..	..	..	..	(5)	..	..	..	..
64,653	(5B)	(5A)	(5B)	(5A)	..	(6)	(5)	..	..	..
64,740	(7)	..	(7X)	..	..	..	..	..	..	..
66,112	(8)	..	..	..	..	..	..	..	..	..
69,129	(9)	..	(5)	..	(5)	(6)	(6)	(5)	(5)	(6)
69,651	..	..	(9)	..	..	..	..	..	..	..
73,516	(9)	..	(10)	..	..	..	..	..	..	..
80,146	(12)	(12)	(11A)	(11A)	..	..	(11A)	..	..	..
83,145	..	(11)	..	(13)	..	(5)	..	..	..	..
109,963	..	..	..	..	..	..	(5)	..	(6)	(6)
110,867	..	..	..	..	(5)	(5)	..	(9)	(14)	(14)
116,291	..	..	(9)	..	..	(15)	..	(15)	..	..
119,586	..	..	(16)	(16)	(16)	(16)	(16)	(16)	(16)	(16)

NOTE: Maximum mileages for any one bushing application: 63,191, 64,653, 69,042, 64,653, 62,547, 35,428, 58,071, 62,547, 62,547, 62,547.
Reasons for renewing rod bushings at the mileages indicated in the table:

(1) Rods were out of tram.
(2) Bushing was thought tight on o.d.; was rebored and then found loose.
(3) Not enough lateral.
(4) Changed from solid to three-piece floating bushing with crown.
(5) Diameter wear.
(5A) Diameter wear and changed from floating to three-piece bushing with crown.
(5B) Diameter wear and changed from floating to three-piece bushing with crown.
(6) Broken bushing.
(7) Experimental bushing.
(7X) Experimental bushing defective, but replaced; removed at 64,653 miles.
(8) Defective casting.
(9) Cracked bushing.
(10) Crankpin rough and was reground.
(11) Replaced with solid floating bushing.
(11A) Replaced with solid floating bushing, as it was damaged when main crankpin broke.
(12) Rebored to fit new pin after right main crankpin broke.
(13) Bushing rough.
(14) Bushings applied at 109,963 miles; did not have sufficient stock to finish o.d.
(15) Steel bushings were loose; new steel and brass bushings were applied.
(16) Bushings damaged account of accident.

TABLE 3 SUMMARY OF ALL RUNS IN FREIGHT SERVICE

		New York Central	Pennsylvania	Chesapeake & Ohio	Erie	New York, New Haven & Hartford	Boston & Maine	Lehigh Valley	New York, Chicago & St. Louis	Missouri Pacific	Chicago, Burlington & Quincy	Northern Pacific	Totals and averages
Number of runs		44	45	8	28	19	35	50	12	15	50	22	328
Average running speed, mph		28.26	29.64	24.78	27.12	22.99	28.79	36.93	31.70	33.64	34.27	28.0	29.80
Locomotive-miles	Trip	149.5	237.0	117.0	104.3	158.0	120.0	92.0	175.8	261.4	123.0	122.0	152.0
	Total	6,582	10,667	930	2,922	3,007	4,193	5,535	2,110	3,921	7,137	3,659	51,655
Train-miles	Trip	146.7	220.0	114.0	102.0	149.0	110.0	89.9	174.1	232.8	122.0	110.1	126.9
	Total	6,458	9,911	915	2,856	2,836	3,856	4,495	2,090	3,493	6,094.8	2,422.08	45,427
No. of cars	Trip	103.6	101.0	123	71.5	107.4	80.3	79.8	76.0	79.8	77.8	60.0	83.4
	Total	4,561	4,558	984	2,003	2,042	2,813	3,981	913	1,198	3,893	1,323	27,356
Tons per trip	Excl. loco	3,500	3,999	6,391	2,411	3,507	3,067	2,755	2,592	2,885.7	2,828	2,679	3,160
	Incl. loco	3,860	4,354	6,743	2,798	3,844	3,417	3,054	2,947	3,240.7	3,246	3,034	3,516
Total tons	Excl. loco	154,130	179,962	51,126	67,506	66,640	107,354	137,764	31,114	43,286	138,577	58,934	1,036,393
	Incl. loco	169,840	195,937	53,946	78,356	73,030	119,611	152,730	35,374	48,611	159,078	66,744	1,153,257
Coal, tons	Trip	13.8	22.4	14.3	11.3	16.5	9.9	9.2	17.3	23.9	13.8	15.0	14.56
	Total	608	1,009	115	316	315	347.5	460.5	227.3	358.2	689.75	330.75	4,777
Water, gal	Trip	22,788	39,596	24,380	20,083	29,966	17,580	14,365	31,964	38,014	20,592	18,294	23,874
	Total	1,002,660	1,781,820	195,038	562,334	569,363	615,294	718,256	383,570	570,220	1,029,640	402,461	7,830,656
Car miles	Trip	14,381	20,696	14,741	6,864	14,200	8,868	6,945	15,141	17,677	8,872	6,590	11,699
	Total	632,753	931,342	117,925	192,194	269,992	310,385	347,274	181,690	265,155	443,634	144,987	3,837,331
G.t.m. per trip	Excl. loco	514,486	807,185	746,069	246,559	512,683	341,984	238,224	470,401	710,087	324,464	283,355	440,321
	Incl. loco	566,865	886,896	786,390	282,808	564,198	381,095	269,479	594,865	795,610	374,193	322,438	491,077
G.t.m. total	Excl. loco	22,637,383	36,323,360	5,968,551	6,903,645	9,740,975	11,969,434	12,808,527	5,644,821	10,296,268	15,898,743	6,233,826	144,425,533
	Incl. loco	24,942,033	39,910,360	6,291,116	7,918,626	10,719,755	13,338,313	14,600,799	6,387,031	11,536,351	18,335,466	7,093,662	161,073,512
Aver. lb coal per 1000 gtm	Excl. loco	53.64	55.50	38.33	91.66	64.36	57.89	77.32	88.43	69.57	86.76	106.11	66.15
	Incl. loco	48.68	50.51	36.37	71.91	58.49	51.95	68.35	76.21	62.09	75.23	93.25	59.31
Running time, hr	Trip	5.19	7.42	4.60	3.76	6.48	3.82	2.52	5.68	6.92	3.56	3.93	4.65
	Total	228.39	334.20	36.76	105.26	123.19	133.93	126.09	68.19	103.78	177.97	86.48	1,524.2
Train-hours	Trip	6.52	10.08	5.50	4.56	10.94	4.58	3.08	7.97	9.34	4.43	5.28	6.12
	Total	286.80	453.60	43.98	127.60	207.87	160.28	154.39	95.65	140.23	221.86	116.15	2,008.4
Gross ton-miles per train-hour		78,931	80,078	135,649	54,070	46,686	74,668	78,678	61,032	73,424	71,661	53,670	71,911
Evaporation, lb water per lb of coal		6.88	7.36	7.10	7.41	7.56	7.39	6.50	7.02	6.63	6.22	5.07	6.83

NOTE: The Delaware, Lackawanna & Western, with 834 locomotive-miles, and the Chicago & Alton with 158 locomotive-miles in freight service, are omitted from the table, as performance details are not available. The mileages, however, are included in the total.

TABLE 4 SUMMARY OF ALL RUNS IN PASSENGER SERVICE

		Pennsylvania	Chesapeake & Ohio	Erie	New York, New Haven & Hartford	Delaware, Lackawanna & Western	Lehigh Valley	Missouri Pacific	Chicago, Burlington & Quincy	Northern Pacific	Totals and averages
Number of runs		23	20	28	2	109	11	14	3	52	262
Average running speed, mph		45.89	38.73	41.56	48.01	39.39	54.83	41.88	45.16	43.6	42.5
Locomotive-miles	Trip	342	339	105	180	138.9	258.1	253	323	518	267
	Total	7,871	7,128	3,058	360	15,140	3,017	3,448	970	26,933	67,931
Train miles	Trip	341	343	98.5	180	136.7	253.5	243	322	516	250.9
	Total	7,846	6,872	2,760	360	14,901	2,788.6	3,403.5	967	26,815	66,713
Number of Cars	Trip	12.5	8.8	9.8	14	10	11.5	14.5	13.3	11	10.7
	Total	289	177	275	28	1,121	127	204	40	520	2,781
Coal, tons	Trip	16	14	5.7	7.5	8.7	12.7	15.7	20.5	30.7	15.67
	Total	370	283.5	160.5	15	952	139.5	219.25	61.5	1,901.9	41,037
Water, gal.	Trip	31,651	26,891	11,218	15,550	13,500.6	22,707	24,180	31,100	49,088	24,109
	Total	727,993	537,830	314,120	31,100	1,471,567	249,776	338,515	93,300	2,552,576	6,316,377
Car miles	Trip	4,119	2,890	997	2,198	1,384.8	2,887	3,328	4,135	5,003	2,621
	Total	94,744	57,795	27,934	4,396	150,945	31,757	46,586	12,405	260,169	686,731
Average lb coal per car-mile		7.77	9.68	11.43	6.82	12.57	8.78	9.41	9.91	14.62	11.9
Running time, hr	Trip	7.43	8.88	2.37	3.20	3.47	4.63	5.71	7.13	9.89	5.98
	Total	170.92	177.62	66.55	6.40	379.18	51.00	79.91	21.39	614.6	1567.57
Train hours	Trip	8.61	9.57	....	3.56	3.78	....	6.42	8.45	13.86	6.28
	Total	198.10	191.53	....	7.13	412.59	....	89.95	25.37	720.83	1645.5
Average lb water per lb coal		8.24	8.00	8.20	8.63	6.46	7.46	6.43	6.32	5.59	6.4

Bearing Life Expectancy

The trailer and tender bearings have been examined from time to time when servicing wheels. A few hairlines have developed on one roll of the trailer bearing and which has continued without change for six months. It does not require replacement or renewal.

The driver bearings were removed and minutely examined at the conclusion of the demonstration period. The cups (outer races) and rolls showed no evidence whatever of wear. The driver cones (inner races) showed very slight wear, visible only under a microscope. As a matter of experiment the removal of 0.0005 in. of metal in regrinding removed the microscopic evidences of wear. The cones have been returned to service without other modifications (some of the cones were not touched) to study further the development of wear. The indications from the study of the bearings at the conclusion of the demonstration period, visualized against the accumulative available information on service of roller bearings in all kinds of industries, would indicate that the wheel bearings, with the exception of driver cones, should have a normal life expectancy of 1,000,000 miles, the driver cones having a life expectancy of one-half that figure. This presumption is further borne out by the study of the condition of the engine-truck bearings after 400,000 miles' service, which show no appreciable evidence of wear, and is supplemented further by the study of the original roller-bearing applications on the Milwaukee Road, many of which have now crossed 1,000,000 miles, and which have been studied in detail over the past seven years.

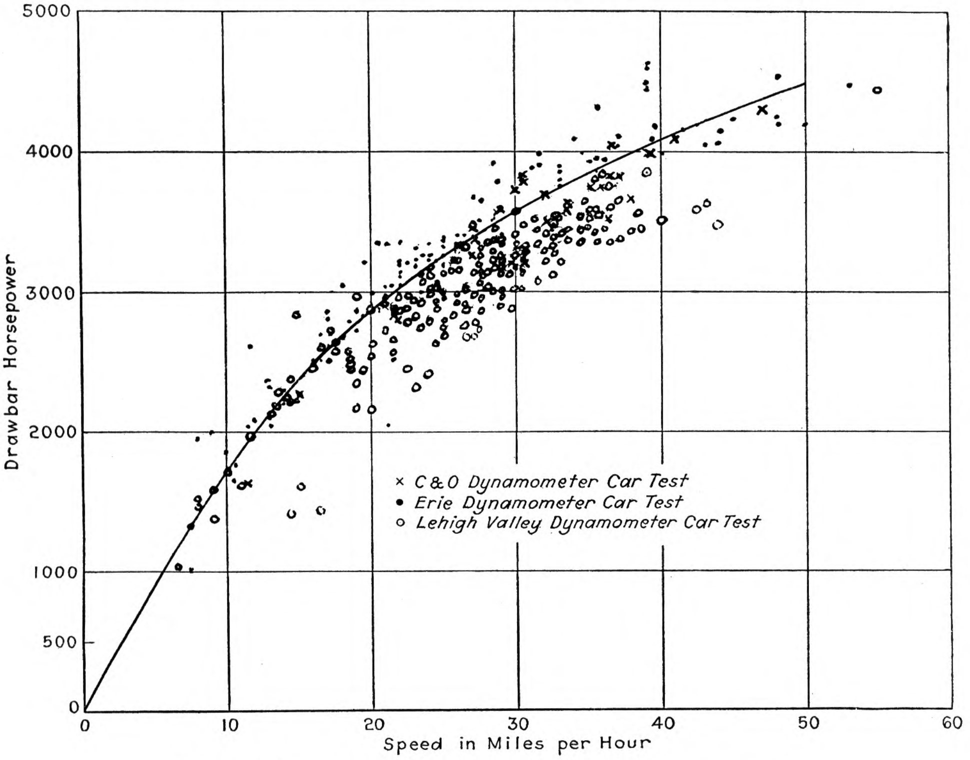

Fig. 10 Timken Locomotive Drawbar Horsepower, Curve 4

An axle assembly of engine-truck bearings removed from a Michigan Central engine, having run 303,756 miles, is illustrated in Fig. 9. The disassembly was made on account of wheel change and developed the perfect condition of the bearings after two complete shopping periods, approximating 153,000 miles each. These bearings did not show sufficient wear to indicate any reasonable percentage of life expectancy, having been given up in service. A number of engine-truck bearings have been examined with similar results at mileages between 400,000 to 500,000.

The rod-bearing life is apparently increased while operating on roller-bearing drivers, as is evidenced by Table 2, which shows the replacement of rod bearings. A number of the bushings on each of the main-rod and side-rod applications have operated in excess of 60,000 miles, which is about double the average life expectancy of these bearings. The probability is that rod bearing life is increased on account of the greater accuracy of alignment of driver axles in the roller bearings.

The availability is definitely increased by the roller bearings. The operation for two years in a wide variety of services without any bearing trouble whatever, and which for long periods of time involved the dispatchment of four divisional trips each 24 hours, is an indication that the use of the roller bearing increases the availability of the locomotive by 50 per cent.

Fuel Economy

The record of fuel and water consumption is indicated in the summaries of freight runs (Table 3) and passenger runs (Table 4). The lowest average was on the Chesapeake & Ohio, of 38.3 lb of coal per 1000 gtm, but individual runs on this road were as low as 26 lb of coal per 1000 gtm.

Favorable loads and favorable grades favor low coal consumption. A fair average is that of 53.6 and 55.5 lb of coal per 1000 gtm on the New York Central and the Pennsylvania, which average covers a wide variety of service over typical trunk-line roads. A tendency is noted for fuel consumption to increase on the Western roads. Much of this is due to the lower fuel value of the coals available to the Western roads, this being particularly true of the Montana coals and some of the coals available on the Burlington and the Missouri Pacific. The coal consumption on the Erie of 91.6 is on account of operating in the rolling country between Meadville, Pa., and Marion, Ohio, against frequent ruling grades of 1 per cent.

An exact comparison with comparable locomotives in identical service was not made available during the demonstration, but the study of such records as are available would indicate that the roller-bearing locomotive operated at rates of 10 to 30 lb per 1000 gtm below that of the plain-bearing locomotives. The fuel consumption is in general one-half of that published by the Interstate Commerce Commission, which, however, is not strictly comparable on account of the latter records including all service and all power.

The locomotive was, in general, operated at full capacity and frequently ran for long distances at cut-offs longer than 60 per cent. This was made possible on account of the free-steaming characteristics of the Timken locomotive boiler and which encouraged road foremen and enginemen to work the locomotive at full capacity. The fuel consumption could have been greatly improved provided effort had been made to operate the cut-offs not in excess of 50 per cent, with the particular object in view of fuel economy records. The attitude of the owners of the locomotive, however, was to encourage the railroad people to work the locomotive to the limit so as to develop the capabilities of roller bearings, and therefore the only efforts at fuel economy were voluntary on the part of engine crews and road foremen.

In consideration of the wide variety of services encountered and generally with strange crews, totaling approximately 840, unfamiliar with the locomotive, and the prevailing attitude of attempting to work the new machine to the limit, taken all together the performance record of fuel and water is truly remarkable and is an indication of the economy records possible with completely equipped roller-bearing locomotives, where economy in fuel rather than demonstration of bearings is the prime object of the test.

The tender is built on a General Steel Casting tender bed and has a capacity of 21 tons of coal and 14,000 gal of water. It is equipped with a cupola for the accommodation of the owners' observer. The operation of the locomotive and the fuel performance have been modified adversely by the water capacity of the tender, particularly in view of the high steaming capacity of the boiler. The survey of American roads indicated the presence of a number of 90-ft turntables at critical points, over which it was desired to operate the locomotive, which by limiting the overall wheelbase to slightly less than 90 ft, placed decided limitations on tender capacity. The performance of the locomotive as regards increased average speed and fuel economy would have been materially improved had the tender capacity for water been increased to 21,000 gal.

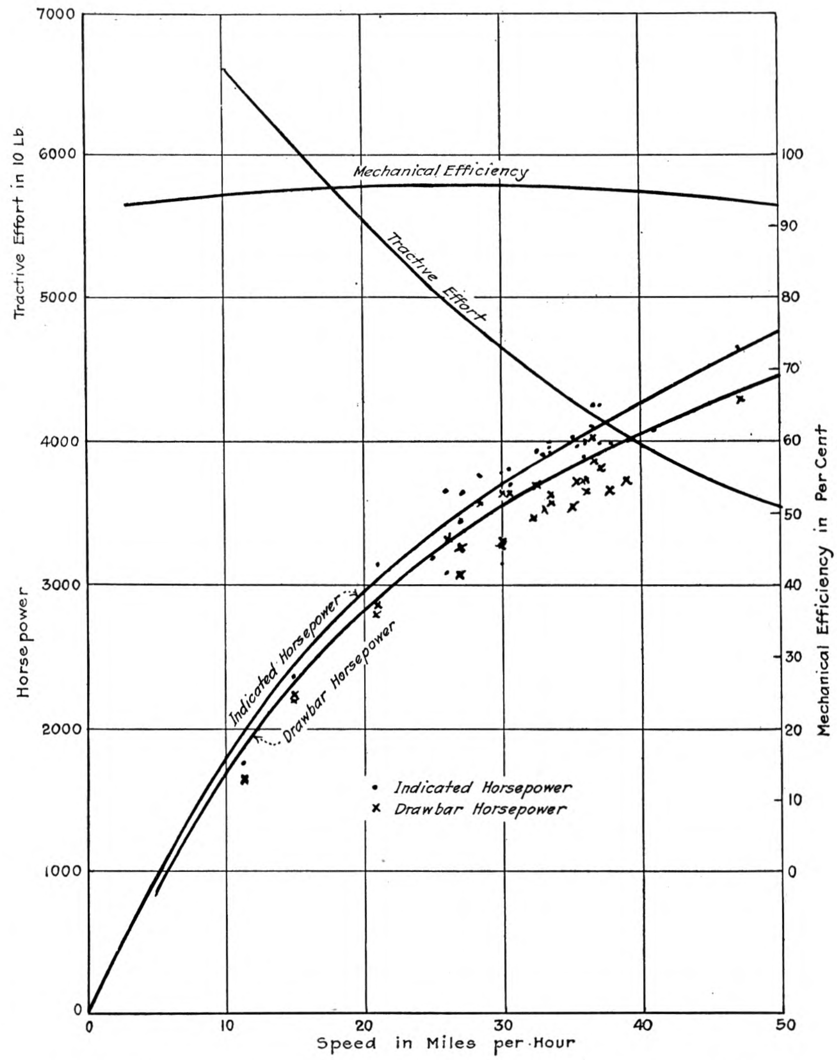

Fig. 11 Power Performance, Timken Locomotive, Lehigh Test, Curve 5

Mechanical Efficiency

Dynamometer-car tests were made on the Chesapeake & Ohio, Erie, Lehigh Valley, Nickel Plate, and Northern Pacific. A number of readings of the Chesapeake & Ohio, Erie, and Lehigh Valley tests have been plotted. In addition to the dynamometer-car record, a limited number of indicator-card tests were available—the Lehigh Valley from both cylinders and the Northern Pacific on one side.

The drawbar horsepower readings, taken from the Chesapeake & Ohio, Erie, and Lehigh Valley tests, with the locomotive working to capacity or nearly so, are indicated on curve 4, Fig. 10. High readings were available on the Erie on account of the frequent 1 per cent grades with tonnage trains. The C. & O. readings were made with heavy coal trains on a water-level division. The Lehigh Valley tests were made on the Seneca division over rolling country. The outstanding feature of this diagram is the

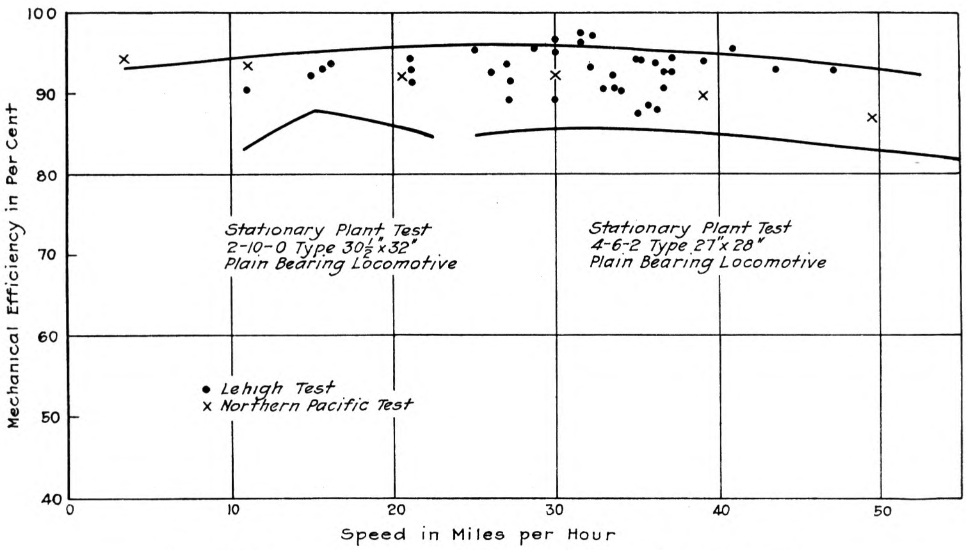

FIG. 12 TIMKEN LOCOMOTIVE DYNAMOMETER CAR TEST, CURVE 6
(Plotted points indicate efficiency from Lehigh and Northern Pacific test.)

development of 4000 drawbar horsepower at speeds in excess of 36 mph under favorable train and grade conditions.

The power performance showing mechanical efficiency is indicated on curve 5, Fig. 11. This is compiled from readings of the Lehigh Valley and the Northern Pacific. The mechanical efficiency varies from 90 to 96 per cent, although a reading is available at $97^1/_2$ per cent. The high efficiency, as indicated on the tests, is confirmed by the low temperature rise of the wheel bearings, particularly the driver bearings, this average being 15 to 20 deg above atmosphere.

The mechanical efficiency of roller-bearing and plain-bearing locomotives is shown on curve 6, Fig. 12. The roller-bearing curve is transferred from curve 5, Fig. 11. The plain-bearing curve is taken from the stationary plant test of a 2-10-0 freight locomotive for the lower speed range and a 4-6-2 passenger locomotive for the higher speed range. The stationary plant readings for the plain-bearing locomotive omits the effect of windage and track resistance. Strictly comparable tests would tend to increase the spread between the roller-bearing and plain-bearing curves.

The increased capacity of the roller bearings, as applied to the locomotive, is indicated on curve 7, Fig. 13, which is combined from curves 5 and 6 and is plotted to show the increased locomotive performance resulting from the roller bearing. This increased capacity for work varies from 10 to 13 per cent, averaging 12 per cent. A conservative assumption is that the roller-bearing locomotive can be reduced in size 10 per cent to perform equally with plain-bearing locomotives, or, conversely, locomotives identical with a roller-bearing locomotive could be expected to produce an increase of 10 to 12 per cent with roller bearings.

The service performance record confirms the test data in that the roller-bearing locomotive consistently exceeded the performance of similar-sized plain-bearing locomotives with identical cylinders and pressure and equaled, and at times exceeded, the performance of engines having 1 in. to $1^1/_2$ in. increased cylinder diameter. The plain-bearing cylinder capacity in some cases was 20 per cent in excess of the Timken engine 1111.

ECONOMIC VALUE OF ROLLER BEARINGS

The value of the roller bearing in locomotive construction is reflected in a number of ways. Some of these advantages can be evaluated, and while others are present and recognized, the definition of value is more difficult to determine. The advantages are as follows:

1 Reduction in maintenance
2 Reduction in consumption of lubricants
3 Reduction in consumption of fuel
4 Reduction in consumption of water
5 Increased development of power, a conservative figure being 10 per cent
6 Increased availability, about 50 per cent.

These advantages are subject to evaluation and can be capitalized; and in addition the following conditions are present, but are more difficult to evaluate:

1 Increased permissible speed
2 Engine-house force reduction
3 Elimination of axle failure due to heat checks
4 Reduction in rod maintenance

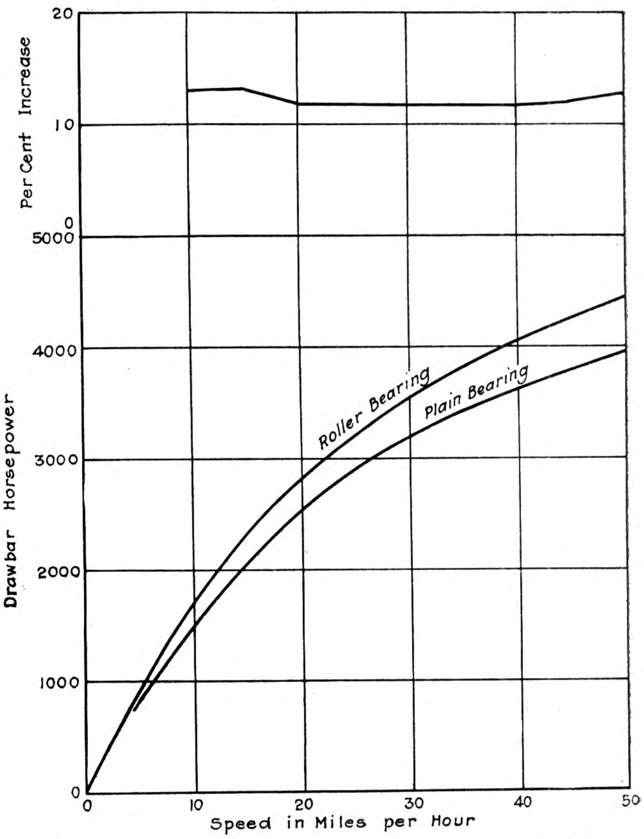

FIG. 13 DRAWBAR HORSEPOWER, ROLLER-BEARING AND PLAIN-BEARING LOCOMOTIVE, CURVE 7
(Based on mechanical efficiency, for roller-bearing locomotive, Lehigh test, and plain-bearing stationary test.)

The approximate cost of roller-bearing equipment, including bearings and housings and application parts for the 4-8-4 locomotive, not including tender, is $8000.

The bearing-replacement charges on the basis of an annual mileage of 168,000 and a life expectancy of wheel bearings, except driver cones, of 1,000,000 miles, and driver cones 500,000 miles, would be $1500 per year.

Saving in maintenance and lubrication, based on data accumulated during the roller-bearing-locomotive demonstration period, would vary from $3900 per locomotive year for territory with light grades to $6700 per locomotive year in heavy service in mountainous territory. This is on the basis of:

1 Attention during running
2 Maintenance at terminals
3 Semi-monthly inspection
4 Replacement of bearing brasses
5 Replacement of hub liners
6 Saving in lubrication;

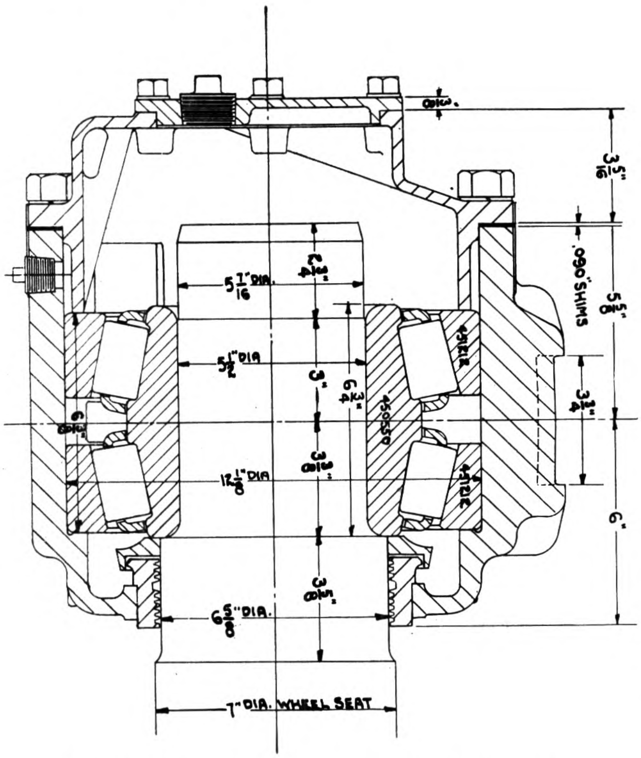

Fig. 14 Passenger-Equipment Roller-Bearing Layout
(Wide-pedestal type, $5^1/_2$ by 10 in.)

The net maintenance saving, after deducting the bearing-replacement charges, would vary from $2400 to $5200 per year.

Fuel economy, based on average machine efficiency of 86 per cent for the plain-bearing locomotive and 93 per cent for the roller-bearing locomotive, would vary from 750 tons to 1200 tons per year, this representing an annual saving of $1500 in the lighter service to $2000 in the heavier service.

The locomotive operating cost would therefore be reduced in amounts from $3900 annually in light service to $7200 in mountainous service, corresponding with direct returns on investment of 49 per cent and 90 per cent, respectively, the high return corresponding with the greater severity of service.

The return of bearing investment, in addition to economy in operation due to savings in maintenance, fuel, and lubrication would also include a factor on account of increased availability of the roller-bearing locomotive and an account of increase in power, averaging 12 per cent—in conservative figures, 10 per cent.

An increase in availability of 50 per cent on an investment of $80,000 is equivalent to $40,000 to be written off in the life of, say, 20 years, or $2000 a year.

An increase in power of 10 per cent on an investment of $80,000 can be evaluated at $8000, which in a life of 20 years is worth $400 annually.

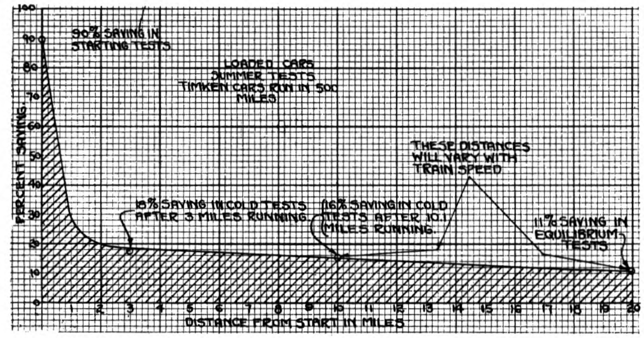

Fig. 15 Tests in Summer of 1930 on Loaded Cars
(Per cent of saving, with distance operated, Timken cars run in 500 miles.)

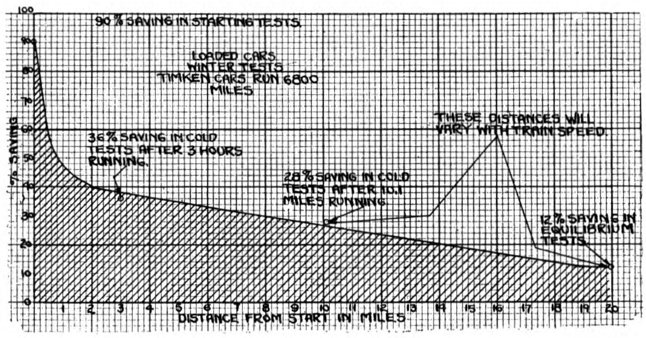

Fig. 16 Tests in Winter of 1931 on Loaded Cars
(Per cent of saving, with distance operated, Timken cars run in 6800 miles.)

The increased value of the roller bearings in locomotives on account of increased power development and increased availability would be worth approximately $10,000 or considerably in excess of the purchase price of the roller-bearing equipment.

Roller-Bearing Rod Applications

Considerable study has been devoted to the application of roller bearings to main and side rods, and methods have been developed that will permit of the application of roller bearings to these parts.

A unique device avoids the transmission of twisting and extraneous loads, incident to locomotive operation, to the bearings. The application avoids an increase in rotating parts carried on the pins, as compared to plain bearings, on the basis of using carbon steels in the rods. The advantages of the roller bearing, involving long continuous runs, reduction in maintenance, and economy in lubrication, are available without sacrificing maximum speed performance or increasing dynamic blow on rails.

An interesting feature of the roller-bearing study is that the main and side rods have practically uniform sections and permit and facilitate the use of alloy steels, developing higher physical properties. The roller-bearing application on the basis of using alloy steels effects reductions of 25 per cent in reciprocating and rotating parts and permits of increase of speed of 12 to 15 per cent with the development of dynamic augment corresponding with plain bearings with diameter speed.

This increase in permissible speed has a high economic value and improves the position of the railroads as regards meeting competition of other forms of transportation. The higher speeds will permit of reducing train schedules to the extent of 10 to 15 per cent and will place the railroads in a position more in keeping with present-day economic demands for high speed.

Passenger-Equipment Applications

The improved service and advantages resulting from the use of roller bearings in locomotives apply with equal force in passenger equipment, and as a consequence the tapered bearings are generally used in this service. A distinguishing feature of the passenger mountings is the outboard construction, using double bearings in pedestal mounting with increased width of pedestal opening.

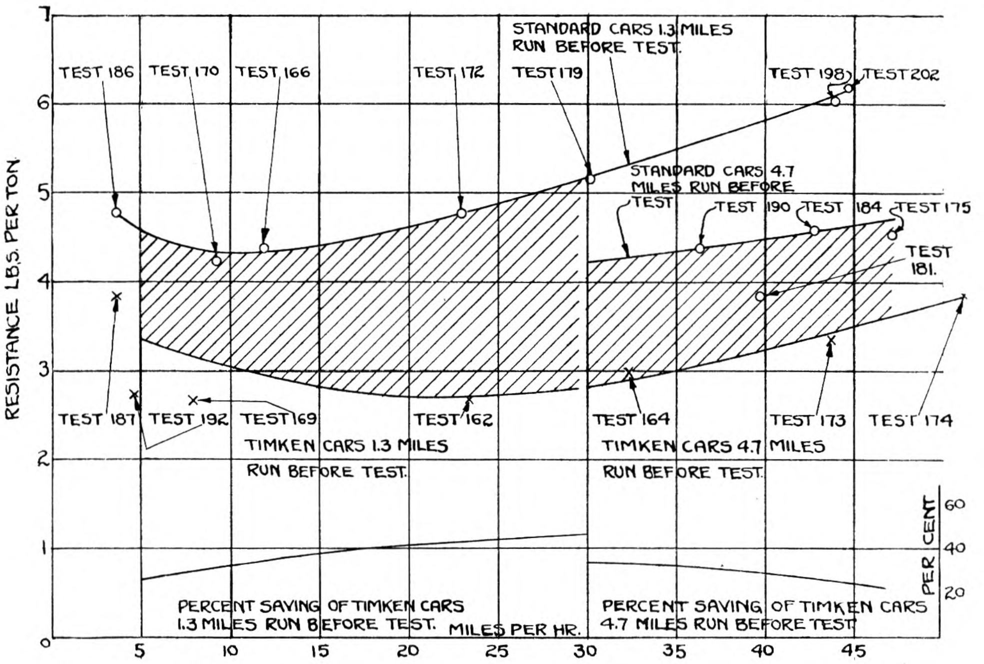

Fig. 17 Cold Test, Loaded Cars, Winter of 1931
(Timken cars run in 6800 miles.)

A typical installation is shown in Fig. 14—a $5^1/_2$-in. by 10-in. application as adapted to General Steel Casting four- or six-wheel trucks.

Milwaukee Installation

The Milwaukee Road made the first large passenger installation on Timken bearings. This involved the complete equipment of the "Pioneer Limited," for service between Chicago and Twin Cities, and the "Olympian," operating between Chicago and Seattle. The original equipment, numbering 127 cars, was placed in service in 1927. The Milwaukee Road has since extended the installation of roller bearings to 153 cars, which include coaches, Pullmans, and diners. The accumulative mileage of these cars to June, 1932, is 116,000,000 miles, the average mileage per car being in excess of 1,000,000 miles.

Pennsylvania Timken Passenger Equipment

The Pennsylvania has 304 passenger-equipment cars in main-line coaches, dining cars, gas-mechanical, and multiple-unit electric cars, the original installations being made in 1927. The total mileage in Pennsylvania service is in excess of 47,000,000 miles, and it is an interesting example of durability and reliability that in this mileage a train detention has never been charged to Timken bearings.

Freedom from hot boxes is a general characteristic of the tapered roller bearing in passenger service.

Greater availability follows the use of the roller bearing, inasmuch as inspection is only necessary at monthly intervals, and equipment can therefore be dispatched with no delays at terminals on account of bearing inspection and maintenance.

An absence of surging is particularly noticeable on complete roller-bearing trains, this resulting from a uniformly low rolling resistance of the roller bearing, and particularly the low resistance at starting.

Reduced cost of maintenance on equipment generally follows the use of the roller bearing on account of the absence of surging in service, and in particularly the reduced blow to draft gears, couplers, and sub-frames in starting.

Freight-Equipment Application

Freight-train equipment presents a field for the application of roller bearings which in value to the railroads should rank next to the successful application to the locomotive. The development of the freight-car application has been slow. The great surplus of freight equipment has practically stopped the purchase of new freight rolling stock, but notwithstanding these conditions, the Timken Company has continued the development of the application of roller bearings to the freight car in the belief that ultimately the value of the development will be recognized, and with the return to more nearly normal business conditions utilizing the reserve of rolling stock, the purchase of new freight cars would initiate the gradual introduction of roller bearings in freight service.

Freight-car roller-bearing development has involved the construction, testing, and arrangement for operation of equipment in capacities of 40, 50, 70, and 100 tons. These cars were built singly or in groups of two or three to develop the characteristics of specific constructions.

The 100-car train has been tested as regards service for a total of 3,000,000 car miles, and in complete trains or in single cars has been tested under the following conditions, in comparison with plain-bearing cars of identical capacity. These tests comprise running resistance tests, both in summer and winter conditions, empty and loaded. The tests were made starting cold and after obtaining equilibrium temperature. There were starting tests for complete trains, starting tests for single cars, acceleration tests for complete trains, and tonnage rating tests for complete trains.

The data, as regards running tests and starting test of single cars in addition to the accumulated service of the 100 cars over a period of three years, permit of drawing reasonable deductions as to the value of roller-bearing equipment in railroad service.

The freight service improvement by roller bearings, from the data derived from the tests, supplemented by service records, can be listed under the following headings:

The break-away resistance of the roller-bearing cars, as developed in 115 tests, was slightly in excess of the low-speed rolling resistance, as compared with the break-away resistance of the plain-bearing cars, which is ten or more times the lowest rolling resistance. This characteristic should be reflected in reduction of wear and tear in couplers, draft gear, and car bodies, and in reducing the strain on locomotives in starting.

The running-resistance tests indicate a reduction in rolling resistance of the roller bearing throughout the entire speed range. The reduction is considerable in starting cold trains amounting to approximately 40 per cent.

A reduction of 16 to 28 per cent, varying with weather conditions, is indicated after ten miles of operation.

The reduction at equilibrium point, attained after approximately 20 miles of running where the temperature of the plain bearing ceases to rise, averages 11 per cent. The reduction in rolling friction with loaded cars under summer and winter conditions, at starting, after running 3 miles, 10 miles, and 20 miles, is indicated on two curves shown in Figs. 15 and 16. The four points are a general average of 170 running tests and 115 starting tests. The shape of the curve between the points, particularly between 0 and 3 miles operation, is conjectural. The normal resistance as controlled by load and temperature conditions, and which control the shape of the curve between the 0 and 3-mile points, is probably attained in the first few hundred feet of operation.

The winter tests are more nearly representative of the comparison in rolling friction of the plain and roller bearings, as these tests were made with plain-bearing cars well run-in after several years of service and with roller-bearing cars after one year of service, averaging 6800 miles.

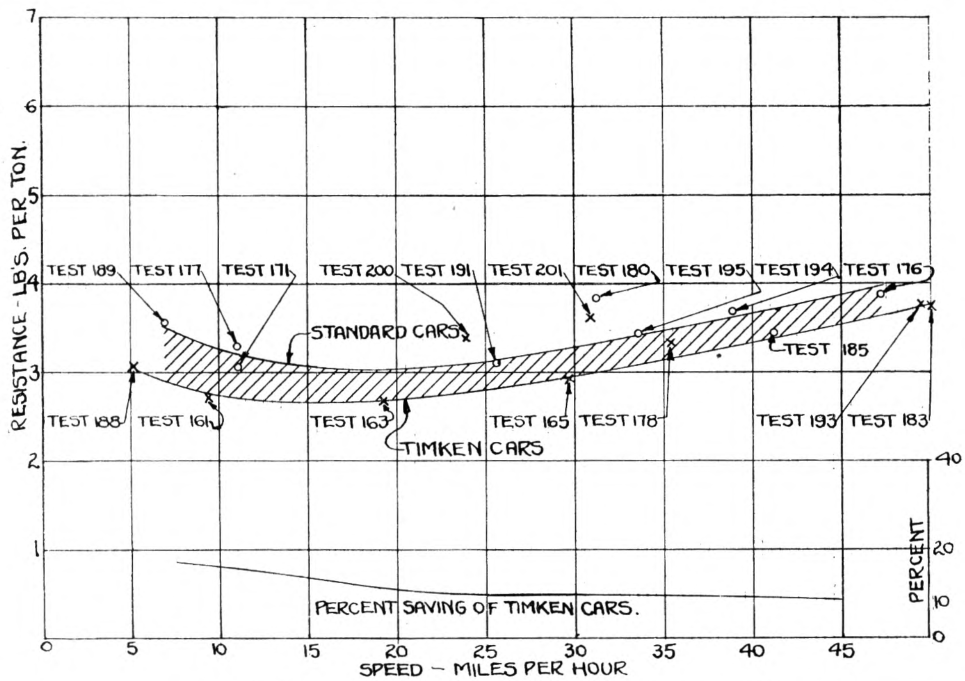

FIG. 18 EQUILIBRIUM TESTS, LOADED CARS, WINTER OF 1931
(Timken cars run in 6800 miles.)

The comparative frictional resistance of loaded cars under winter conditions, after running 3 to 10 miles, is indicated on the curve in Fig. 17.

The equilibrium test on a loaded car is the best indication available of the comparative rolling resistance of cars with the two types of bearings. A general reduction of the roller bearing, ranging from 18 per cent at 10 mph to 9 per cent at 45 mph, is indicated in Fig. 18. It will be noted that on account of the wide varying of conditions affecting rolling resistance, in addition to the roller bearing friction, a number of the tests show results slightly above or below the average. This is indicative of the need for a much greater number of tests before definite conclusions can be drawn. The 170 tests, spread over a wide variety of conditions, do not give any points to locate a curve representing specific conditions; however, it is the best data available. A fair check is provided by plotting the curve through points at various speeds from 10 to 50 mph, and where nearly all of these points show a logical curve, the influence of the erratic tests in locating the curve can be considerably reduced.

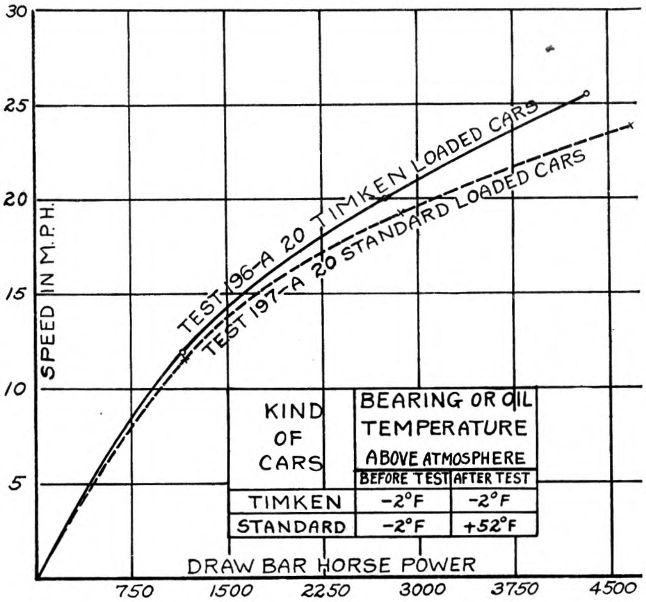

FIG. 19 FIRST ACCELERATION TEST
(Relation between drawbar horsepower and speed.)

The test data and the experience available with roller bearings indicate a wide divergence at starting, and the comparable rolling resistance of the plain and roller bearings converge at approximately 25 mph with a parallelism of the curves above that speed. The data available in the comparison of plain and roller bearings in steel-mill service indicate greater divergence and more important economies, while other data available for higher speeds and heavy loads, available in the copper-rolling industry, indicate still further economies in favor of the roller bearing.

Acceleration tests, while limited in number, furnished indications that roller-bearing trains can be accelerated to predetermined speeds in less time or with less power, and given the same power, will attain a certain speed in a shorter distance. Curves showing acceleration tests are shown in Figs. 19 to 22. The curve in Fig. 20, representing relation between drawbar horsepower and speed, is of interest in showing a temperature rise in attaining a speed of $25^1/_2$ mph for the roller bearing of zero, whereas the temperature rise of the plain bearing in attaining a speed of 24 mph is 54 deg.

Starting tests on complete trains, made in limited numbers, gave indications that trains of equal weight can be started with plain or roller bearings, with trains having full slack, and 0.3 per cent grades.

The increase in the size of the stretched train that could be started with identical locomotive on roller bearings was 76 per cent. A total of 76 cars were started with either type of bearing in a slack train, but with a stretched train 65 roller-bearing cars

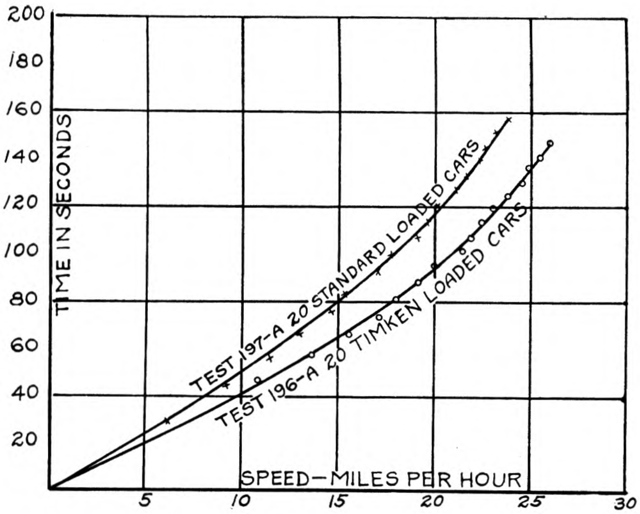

Fig. 20 First Acceleration Test
(Relation between time and speed.)

and 37 plain-bearing cars developed the full capacity of the identical locomotive.

Hot boxes, based on experience from October, 1925, on the first cars built and including the experience of the 100-car roller-bearing train, indicate that hot boxes are eliminated as a factor in railroad operation by the roller bearing. There have been no hot boxes to date.

Reduction in car maintenance is indicated by the experience with the 100 cars, with over 3,000,000 car miles without repairs to bearings or related parts and without a recorded repair to car bodies. While the experience is limited, a reduction in maintenance with roller bearings is indicated.

Increased speed of transportation should result with use of roller bearings. Speed limitation, as imposed by plain bearings, is eliminated entirely. It was found that 50 mph was the maximum permitted speed of the loaded 70-ton equipment as used in the running tests. The roller-bearing cars were operated at the top speed of the freight locomotive, namely 65 mph, without appreciable heat rise, but the plain-bearing cars developed hot boxes with such frequency at speeds of 50 mph as to cause the abandonment of the test program with plain bearings at the higher speeds. Higher speeds than 50 mph are operated on plain bearings in passenger service and in general freight commodity service, but with reduced axle loads. It is expected that increased truck competition will force higher railroad speeds.

Economy in fuel should follow the reduction in rolling friction throughout the entire speed range from 5 to 50 mph. An economy of 10 per cent on level track would be equivalent to a 3 per cent saving on a normal heavy tonnage grade of 0.3 per cent and would be equivalent to $1^1/_2$ per cent on a 1 per cent grade.

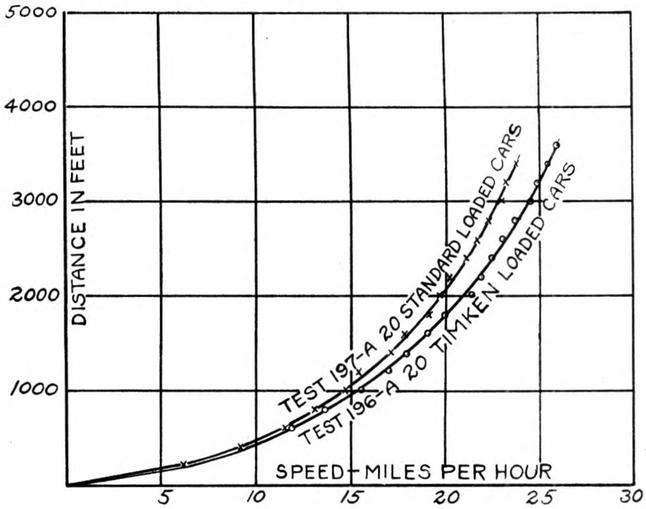

Fig. 21 First Acceleration Test
(Relation between distance and speed.)

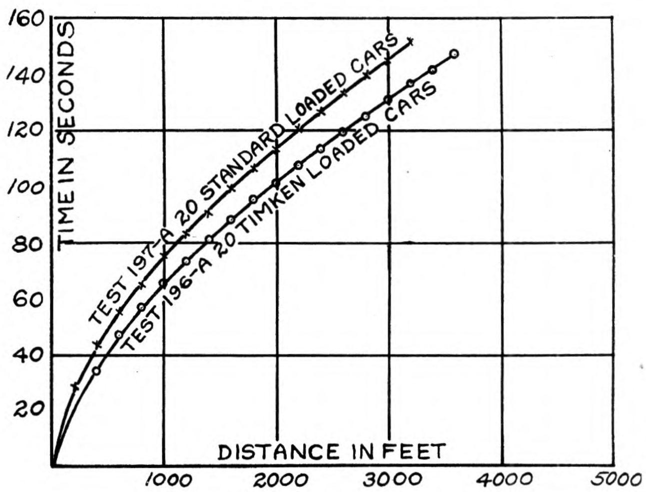

Fig. 22 First Aceleration Test
(Relation between time and distance.)

Fuel saving is affected by many other variables, but a general average could be expected.

Improvement in coupler and draft-gear service should follow the use of roller bearings on account of the 90 per cent reduction in effort required in break-away and the 75 per cent reduction in starting a stretched train.

An increased capacity of locomotives to haul larger trains or to haul equivalent trains at faster speeds is a natural corollary in the reduction of rolling friction.

The weight reduction is a factor of importance, amounting to 4400 lb per 70-ton car, which, with the identical rail load, increases the value of the car for hauling commodities.