International GE Company - Report on Railway (Streetcar - Tramway) Motors

Contents

1-10-2021

International General Electric Company

INCORPORATED

SECTION 1.

ENGINEERING REPORT ON RAILWAY MOTORS.

This book is issued for use within the Company only, and is intended to give comprehensive information regarding G. E. commutating pole motors in most general use.

Motors of other capacities, or having special characteristics, are also manufactured, but in order to reduce manufacturing costs, it is desirable, as far as is possible, to confine the sale of railway motors to those given in the list on Page 186.

STRICTLY CONFIDENTIAL

For the information of employees of the Company only.

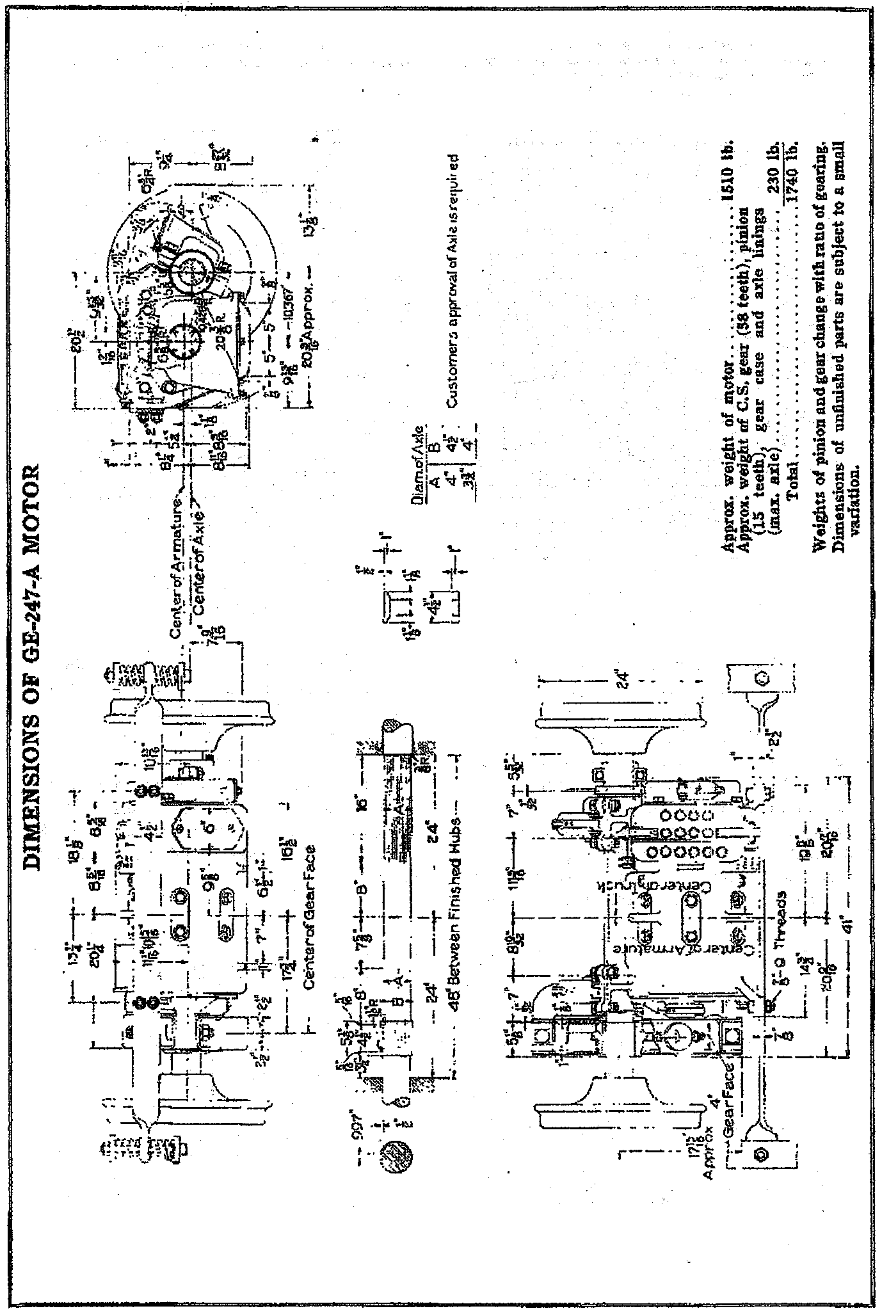
ENGINEERING REPORT

on

Railway Motors

Control Equipment &

Air Brakes



GENERAL ELECTRIC
GENERAL ELECTRIC COMPANY, SCHENECTADY, N. Y., SALES OFFICES IN PRINCIPAL CITIES
June, 1926 *Bulletin No. GEA-41 8*

GE-247-A, 600-VOLT RAILWAY MOTOR

For Light Weight Cars in City, Suburban or Interurban Service

Suspension Side

Axle Side

GE-247-A RAILWAY MOTOR

RATINGS

Hourly, for 75 deg. rise by thermometer;
40 h.p., 600 volts, 60 amperes, 715 r.p.m.

Continuous, for 65 deg. rise by thermometer;
36.6 amperes at 600 volts.
35.8 amperes at 450 volts.
34.4 amperes at 300 volts.

TRUCK DATA

	Form A 26-in. Wheel Motor
Maximum diameter of car axle in bearing linings	4 in.
Clearance under frame with 26-in. wheels (maximum reduction)	4 1/16 in.
Clearance under gear case with 26-in. wheels (max. red.)	4 5/32 in.
Max. gear reduction (3½ pitch)	58/15

WEIGHTS

	Approx. Wt. in Lb. Form A 26-in. Wheel Motor
Motor, including gear, pinion, gear case and axle linings	1740
Two-motor equipments, complete with two K-36 controllers	4590
Four-motor equipments, complete with two K-35 controllers	8520
Two-motor equipment, with Sprague G-E Type PC control complete	4680
Four-motor equipment, with Sprague G-E Type PC control complete	8400

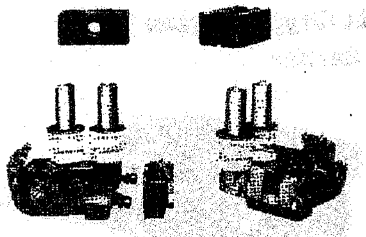
Supersedes 44406A

Frame

Cast steel, box type, bar suspension, large opening over commutator, covered hand hole suspension side, commutator end.

Axle Bearings

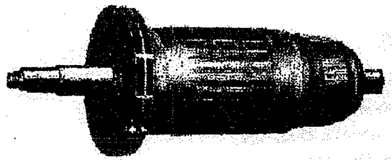
Bronze linings, standard size 4 in. diameter, 7 in. long, interchangeable, linings held by dowels.



Brush-holders

Brush-holders

Adjustable pressure, renewable carbon-way, 1 brush per holder, size 2¼ in. by 2½ in. by ½ in.



Armature

Armature

Hot banded, shaft removable without disturbing windings. Bearing surfaces rolled; thrust collars, drop forged, shrunk on shaft.

Armature Bearings

Linings held by keys, bronze lined with babbitt, pinion end 2¾ in. by 6⅝ in., commutator end 2⅜ in. by 4¾ in.

Field Coils

Wound with rectangular wire, compounded by vacuum process. Supported by spring flanges, terminals on coils.

Exciting and Commutating Field Coils

Pole Pieces

Exciting, laminated, mounted on steel key, held by tap bolts; commutating, drop forged, held by tap bolts.

Commutator

Hard drawn copper segments; selected mica insulation, mica grooved $\frac{3}{64}$ in. One-piece mica cones, moulded.

Lubrication

Oil and waste; large capacity wells; auxiliary wells.

Ventilation

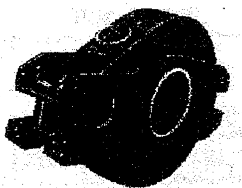
Multiple fan path through armature core, path around field coils, exhaust at pinion end.

Gearing

Short addendum gear teeth; long addendum pinion teeth, 3½ pitch, 4-" face.

Gear Case

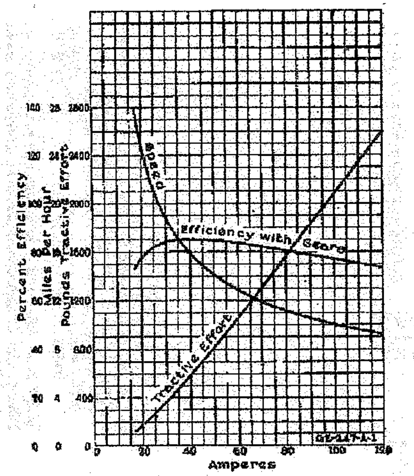
Pressed steel gear-case welded to supporting cradle.



Pressed Steel Gear Case

GE-247-A 26-IN. WHEEL MOTOR

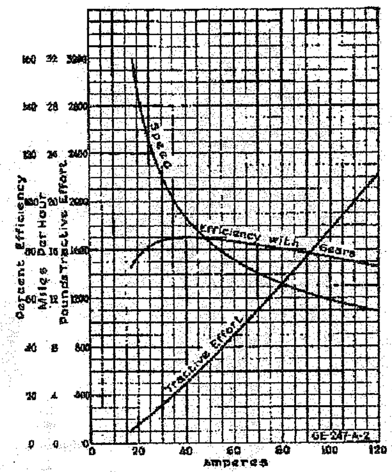
Characteristic Curves on 550 Volts, Diameter of Car Wheels, 26 Inches; Gear, 58 Teeth; Pinion, 15 Teeth; Ratio, 3.87; Maximum Gear Ratio



Characteristic Curve No. 388

GE-247-A 26-IN. WHEEL MOTOR

Characteristic Curves on 550 Volts, Diameter of Car Wheels, 26 Inches; Gear, 56 Teeth; Pinion, 17 Teeth; Ratio, 3.29



Characteristic Curve No. 399

SCHEDULE SPEED, GE-247-A 600-VOLT MOTOR

The following table indicates the capacity of the GE-247-A motor and will assist materially in determining whether this motor is suitable for the desired schedule. It is based on the following assumptions: Average trolley potential, 550 volts; acceleration and braking, 1.5 miles per hour per second; duration of stops, 10 seconds; coasting for 230 feet on all runs; straight level track, maximum temperature rise not exceeding 65 deg. C. Schedule speeds given are 10 per cent less than theoretical values, to allow for delays due to grades, curves, slow downs, or other factors that may affect the schedule.

It is strongly recommended that service data be supplied and the General Electric Company's engineers be consulted before the final selection of a motor and gear ratio, since co-operation has been found to be mutually beneficial.

Stops per Mile	Miles per Hour with Load in Tons per Motor											
	Gearing 58/15						Gearing 56/17					
	4.5	5	5.5	6	7	8	4.5	5	5.5	6	7	8
1	20.1	19.7	19.1	18.3	17.7	17.2	21.2	20.7	20.2	19.8	19.3	18.3
3	14.1	13.8	13.6	13.5	13.2	13.0	14.5	14.3	14.0	13.8	13.6	13.0
5	11.2	11.1	11.0	10.9	10.7	10.6	11.5	11.3	11.1	11.0	10.9	
6	10.1	10.0	9.9	9.9	9.8	9.7	10.4	10.2	10.1	10.0		
7	9.4	9.3	9.2	9.2	9.1	9.0	9.6	9.4	9.3	9.2		
8	8.7	8.6	8.5	8.5	8.5		8.9	8.7	8.6	8.5		
9	8.1	8.1	8.0	8.0	8.0		8.5	8.2	8.1			
Maximum free-running speed	29.5	28.4	27.8	25.5	24.7	23.3	32	31.5	30.4	29.3	27.2	26.0

Above free-running speed based on four-motor equipment.
Max. free-running speed of two-motor equipment approximately 90 per cent of above values.

TABLE OF CONTENTS

NOMENCLATURE

General Electric Railway Motors are identified as follows:

Type, Type numeral, Form letter, and Form numeral.

As an illustration:

Type Letters	Class Numeral	Form Letter	Form Numeral
GE	265	A	51

This indicates that the motor is a direct-current General Electric Railway Motor, belonging to class 265. For an alternating-current commutating type General Electric Railway Motor, the type designation is GEA and for alternating-current induction type GEI.

In general for a given class of motors the armatures will mechanically interchange in any frame although the electrical characteristics may not be the same in each case. Therefore, performance is independent of class.

The form letter changes with mechanical variations in a given class and type. The same form letter, however, does not indicate the same mechanical features for different classes of motors.

Any change in the motor may change the form numeral.

Motors having type designation form GE-50 to GE-199 are not fitted with commutating poles. Motors having type designation in the 200 and 300 series have commutating poles. The class numeral does not indicate whether the motor is of the ventilated or non-ventilated design.

Designs of motors that have been submitted on various propositions but have not been built are assigned numbers in a separate series and are known as follows: GEZ for D-C. series motors; GEAZ for A-C. commutating motors; and GEIZ for induction type A-C. motors. Some of the most recently designed motors have a model number stamped on them; for example, model number 5GE300A1 stamped on a motor completely identifies the motor as a G-E 300-A-1 motor.

APPARATUS SALES ADVICE No. Ry-2 . 158

Department: RAILWAY

Classification: GENERAL

Subject: RAILWAY MOTOR NOMENCLATURE AND STANDARD MOTORS THAT ARE CARRIED IN STOCK.

CONFIDENTIAL—(For Use of Employees of This Company Only).

Schenectady, December 13, 1928

New form numbers have been assigned to the standard railway motors now available with new types of bearings. Previous Sales Advices have described the bearings providing larger oil capacity, also G-E Constant Oil Level Bearings. Table given below specifies the form number of the standard motor with the old style lubrication and the corresponding new form number for motors with larger oil capacity bearing and constant oil level bearing.

MOTOR	OLD STYLE LUBRICATION	LARGE OIL CAPACITY	CONSTANT OIL LEVEL
GE-247	Form A	J	L
GE-247	D	..	..
GE-247	I	K	M
GE-258	C	M	..
GE-258	K	L	..
GE-264	A	D	F
GE-264	B	E*	..
GE-265	A	J	L
GE-265	C	H*	..
GE-265	..	F	K
GE-265	..	G	..
GE-275	B	J	K
GE-275	C	..	..
GE-275	F	L	..
GE-275	H	M	..

* NOTE: This form has large oil capacity frameheads and old style axle caps.

All of the standard motors now being manufactured for stock are provided with the large oil capacity bearing when this type of bearing is available as shown above.

CONSTANT OIL LEVEL BEARINGS

For the proper filling of G-E constant oil level bearings a supply of oil under pressure is required and the operator will have to provide such a system as will best suit his individual needs.

(Over)

A filler nozzle with shut off cock is supplied with each lot of motors and this nozzle should be attached to the hose used to supply oil to the bearings.

The following suggestions are offered towards a solution of operators problem.

Where only a few motors are operated a portable tank holding about five gallons may be used. This tank should be light and strong with suitable opening into which oil may be poured, which opening may be tightly closed. The tank should be equipped with hand pump and pressure gauge, also relief valve and pipe connection with shut off cock for hose connection. Probably 20 feet of 1/2 inch hose will be sufficient.

A tank, such as described above, is sold by the Binks Spray Equipment Company, Chicago, Ill. Order should read as follows:

> Furnish one five gallon tank style No. 8 suitable for filling railway motor bearings. Tank to be equipped with No. 50 lb. pressure gauge, shut off cock on tank and nipple for 1/2 in. hose connection with twenty feet of hose connected to same. Also provide relief valve on tank."

Where a large amount of oil is used in bearings, as in 200 hp. motors where several gallons of oil may be used per motor, the above mentioned tank would hardly prove satisfactory.

If arrangements can be made for the lubrication of all motors in a centralized location, a system of piping can be installed, oil being drawn by gravity from a tank located in a clean, warm place; ten or fifteen feet of head being sufficient to force the oil into the bearings.

An air line should not be connected to the tank of oil to obtain pressure, on account of the possibility of entrance of water.

E. P. WALLER

Manager Railway Dept.

Sales Office Mgrs.
Railway Representatives
225

RATINGS

Commercial ratings are slightly lower than the average obtained from tests. This allowance is necessary to take care of slight variations in individual motors.

All motors built before the GE-281 have their ratings based on the following:

Hourly ratings are given in horsepower delivered at the car axle on rated voltage for a 75 deg. C. rise by thermometer on any normally exposed surface of the windings, after one hour's run on stand with covers arranged to secure maximum ventilation.

Continuous ratings are based on a temperature rise of 65 deg. C. by thermometer, on any normally exposed surface of the windings, with the ventilation system as in service.

All motors built after the GE-280 have their ratings based on the A.I.E.E. Standards of 1925 which are as follows:

"The one-hour rating of a railway motor shall be the output at the motor shaft, measured in horsepower (or kilowatts), which the motor can carry for one hour on stand test, starting cold at its rated voltage and frequency (in case of alternating-current motor) with the ventilation system as in service without exceeding the temperature limits given in the table below."

"The continuous rating of a ventilated railway motor shall be the output at the motor shaft measured in horsepower (or kilowatts) which the motor can carry for an unlimited period on stand test, at its rated voltage and frequency (in the case of alternating-current motors) with the ventilating system as in service without exceeding the temperature limits given in the table below. Direct-current ventilated motors may also be given a continuous rating in amperes at full, three-quarters, and one half rated voltage."

"The continuous rating of totally enclosed direct-current railway motors shall be given in amperes at three-quarters, and one-half rated voltage."

The reason for the higher temperature rise allowed for totally enclosed motors is that on stand test cooling is inferior to that obtained in service.

The advantages to be gained from a self-ventilated motor depend upon the service in which it is to be used, since its ability to dissipate heat depends upon the fan speed, that is, armature speed.

Therefore, the self-ventilated motor when used in frequent stop service is of less advantage as compared to the totally-enclosed motor than when used in service with long distances between stops where a high average armature speed is obtained.

The hourly rating is arbitrary and should not be used to compare the service capacity of motors.

The continuous ampere ratings give an approximate indication of service capacity by which motors may be compared. When comparing motors, make certain that their ratings are determined on the same basis.

GENERAL ELECTRIC REVIEW

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IMPROVEMENTS IN DESIGN AND CONSTRUCTION OF RAILWAY MOTORS

By E. D. PRIEST

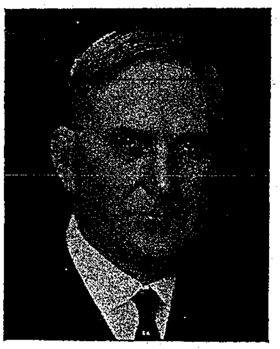
REPRINT

From Issue of April, 1920

Improvements in the Design and Construction of Railway Motors

By E. D. Priest

Engineer, Railway Motor Department, General Electric Company



E. D. Priest

In the November, 1913, number of the General Electric Review, the writer published a short article entitled: "The Development of the Modern Direct-current Railway Motor." This article was a brief review of the subject. Since its publication there have been many substantial improvements in the design and construction of railway motors; and it is the purpose of this article to supplement the earlier one in a measure and to describe briefly some of these improvements.

A marked advance has been effected in the design of railway motors. This has been accomplished by the use of higher grade materials, refinements in design, increased ventilation, higher armature speeds, increased gear ratio, and reduction in weight made possible by these changes. If it were not for these improvements, the present manufacturing cost of railway motors, to perform a given service, would be much higher.

Heat-treated alloy steel is now used for the armature shafts. The steel in the smaller motor shafts is substantially the same and, for like sizes, is equal to that used in the crank shafts of the "Liberty motors" designed for use in airplanes.

The quality of steel in gears and pinions has been improved and improved methods of heat treatment have been developed. The highest grade materials are now used for railway motor gears and pinions. New ways have been found of tempering cast-steel gears which produce qualities substantially equal to forged gears.

Bearing metals are now of the highest quality obtainable. All babbitt is genuine tin-base babbitt. This is the most expensive babbitt manufactured and long experience has shown it to be the best. The highest grade bronze is used in the linings.

In some instances key stock is heat treated to secure hardness and is ground to size to insure close fits and freedom from wear.

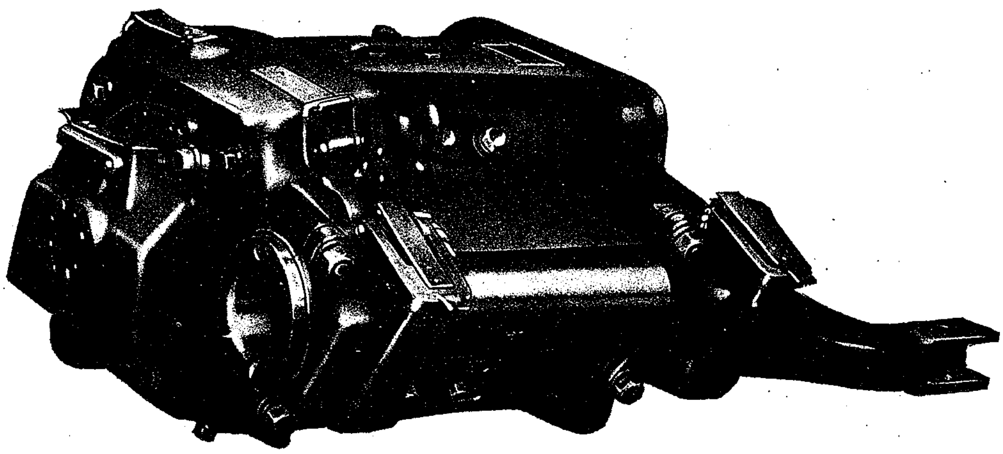


Fig. 1. A Modern Light-weight Railway Motor, showing Axle Side

Heat treated carbon steel bolts are quite generally used in the construction of motors and in some motors heat-treated alloy steel is used.

For brush-holders, expensive high-grade bronze castings are used exclusively and carbon brushes are of the highest grade obtainable.

All castings other than bronze are either malleable iron or steel, no cast-iron being employed in the construction of railway motors.

In general, the quality of materials now used is the best, and no inferior substitutes are employed. Operating conditions are so severe that maximum all round economy can be obtained only by the use of the best materials.

Much study and research has been devoted to producing higher grade varnishes employed for insulating purposes, and in the past few years there have been developed greatly superior varnishes which have higher insulating values and slower ageing qualities.

As with materials, so with workmanship; the best workmanship has been found to be the cheapest since reliability in service is of far more importance than first cost. While the rough exterior of a railway motor suggests quite ordinary workmanship, as a matter of fact it is doubtful if any other line of machinery manufactured has closer fits and more accurate workmanship.

Some of the tolerances in armature shaft fits are plus 0.00025 to minus .00000. For frame-head fits in box-frame motors, plus 0.002 to minus .000 are allowed. The fit must be so close as to require that the heads be driven into place. In boring the heads for armature linings and in turning the linings, very close tolerances are required in order to secure the proper pressing fit of the linings in the heads. A tolerance of plus 0.001 to minus 0.000 is used in the bore of solid gears.

The thread fit for frame-head bolts and for screws is made so close that special taps and dies are required to insure tight fitting threads. Throughout the whole construction of the motor, limits in workmanship are very close as it is found that imperfectly fitting parts rapidly loosen and wear in the abnormally hard service to which railway motors are subjected.

Armature shafts in bearings are ground to size and rolled, a process which produces a hard smooth polish having an ideal bearing surface. Equal care is taken to secure a hard smooth surface on the babbitt in the bearing linings.

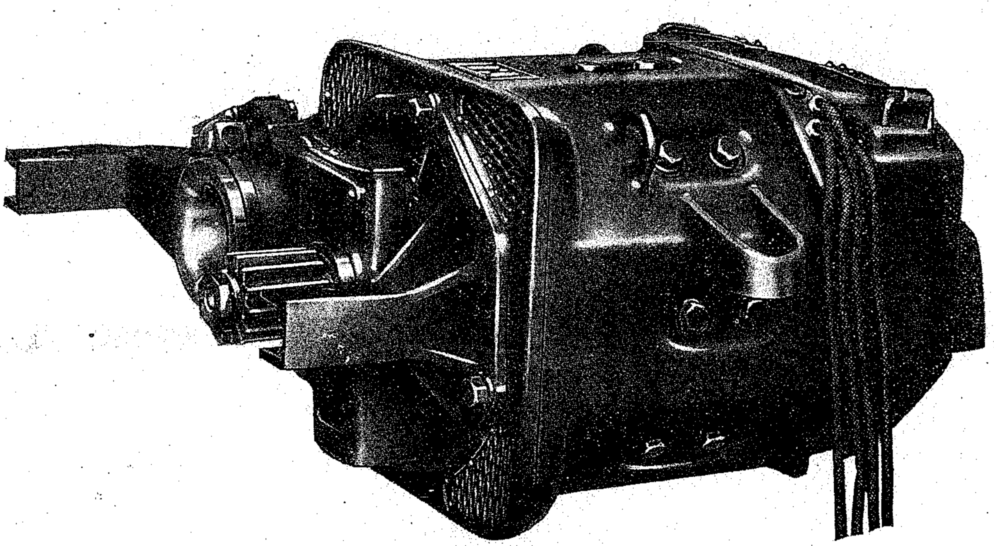
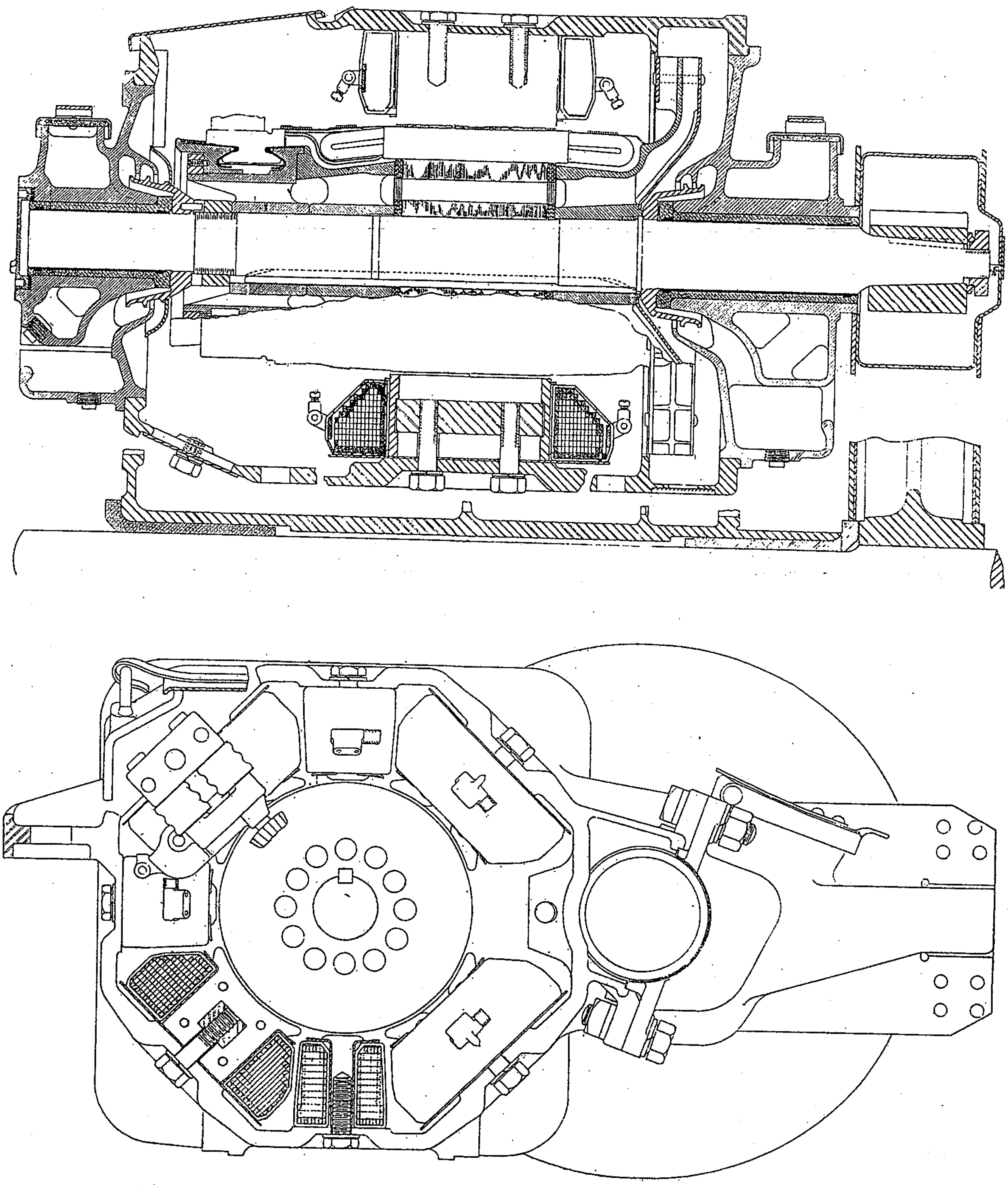


Fig. 2. Suspension Side of the Light-weight Motor Shown in Fig. 1

In order to prevent vibration due to armatures being out of balance, the detail parts of the armatures are balanced separately before being assembled on the shafts, and after assembly the completed core is balanced.

Aside from material and workmanship, substantial improvements have been made in the design of motors. Box-frame motors have come into almost universal use, this construction being greatly superior to the split-frame type in sturdiness and reliability of operation.

The ventilation of motors has been much improved so that multiple ventilated motors have largely increased service capacity. The continuous capacity in some instances is 70 per cent or more of the hourly rating. Ventilating fans have been strengthened so that trouble from breakage has been largely reduced.



Figs. 3 and 4. Sectional Drawings of the Light-weight Railway Motor shown in Figs. 1 and 2

A superior construction of armature bars applicable to large sizes of motors has been developed. This construction permits the use of thin folded crossed bars which make it possible to obtain greater capacity with a given size of armature core without increasing eddy current losses due to heavy copper sections.

A method of connecting bars at the back end of one-turn armatures has been devised which eliminates the use of soft solder that is liable to melt if motors are subjected to excessively heavy overloads which sometimes occur in locomotive service. The improvement consists in using electrically brazed joints in place of soldered joints.

In armature windings of more than one turn per coil, wire of rectangular section has come into more general use. The space factor with rectangular wire is materially higher than with round wire. This results in an increase in capacity of armatures for given core sections.

Taking greater advantage of the possibilities of employing commutating poles, the use of two turns per coil in armature construction has been extended to much larger motors than formerly thought possible, thereby decreasing the weight and cost of the motors.

Sheet steel gear cases have been developed to a higher point of perfection so that they are proving more reliable in service than sheet steel cases of earlier designs.

A much desired improvement has been brought about in the method used to prevent rotation of axle linings in large sizes of motors. The construction consists in the use of a long key set in the bore of the magnet frame for the lining, along the lower edge of the split in the lining. The lining is not materially weakened since it is at the point of separation of the two halves. This construction has been found to hold linings very securely.

Spring gears have been developed, the use of which in heavy work prevents excessive shocks on gear and pinion teeth, resulting from imperfections in the teeth or rough service conditions. When twin gears are used spring gears tend to equalize the work on the two sets of gearing.

Motors have been designed for largely increased potentials and 3000-volt direct-current railway motors have been in most successful operation for a number of years, handling the severest of service.

Higher armature speeds have been made possible by the use of stronger material in the shafts and in the pinions and gears, and by improved shape of gear and pinion teeth which permits the use of a finer pitch gearing, a smaller pitch diameter of pinion, and a smaller number of teeth in the pinion, without a reduction in the strength of the teeth as compared with coarser pitch gearing with inferior shaped teeth.

For many years the standard gear used in street railway service had three pitch 14½ deg. angle teeth. By changing the angle to 20 deg. approximately 25 per cent stronger teeth have been secured, and by lengthening the pinion teeth addendum and shortening the dedendum with a corresponding shortening of the gear teeth addendum and lengthening of the dedendum it is possible to change a three pitch to approximately a 3½ or 4 pitch without sacrificing strength, and with an incidental possibility of increasing the gear ratio. The shortening of the dedendum of pinion teeth and the use of a finer pitch permits a reduction in the number of teeth without a reduction in the thickness of the metal between the base of the teeth and the bore.

An increase in strength of the pinion and shaft has been effected by reducing the depth of the keyway in the pinion so that metal is not cut away at the large end of the bore and by shortening the keyway in the shaft so that it does not extend to the inner end of the pinion but is stopped inside the pinion fit at a point where the shaft is supported by the shrink fit of the pinion.

The maximum armature speed for a given car speed is of course fixed by the gear ratio. Consequently an increase in gear ratio makes it possible to design a lighter and cheaper motor for a given service. Increased armature speed not only reduces the size and cost of motors due to increase in speed, but also makes possible a further reduction because of increased ventilation resulting from increased speed.

The minimum number of teeth in pinions for a given pitch and tooth shape is limited by the diameter of pinion bore. Sufficient metal for ample strength being allowed between the base of the teeth and the bore, it is obvious that the higher the grade of armature shaft stock used the smaller the pinion fit and pinion bore can be made. Therefore the size of a motor is fundamentally affected by the grade of material used in the armature shaft and the grade of material used in the pinion and gear as well as by the pitch and shape of the pinion and gear teeth.

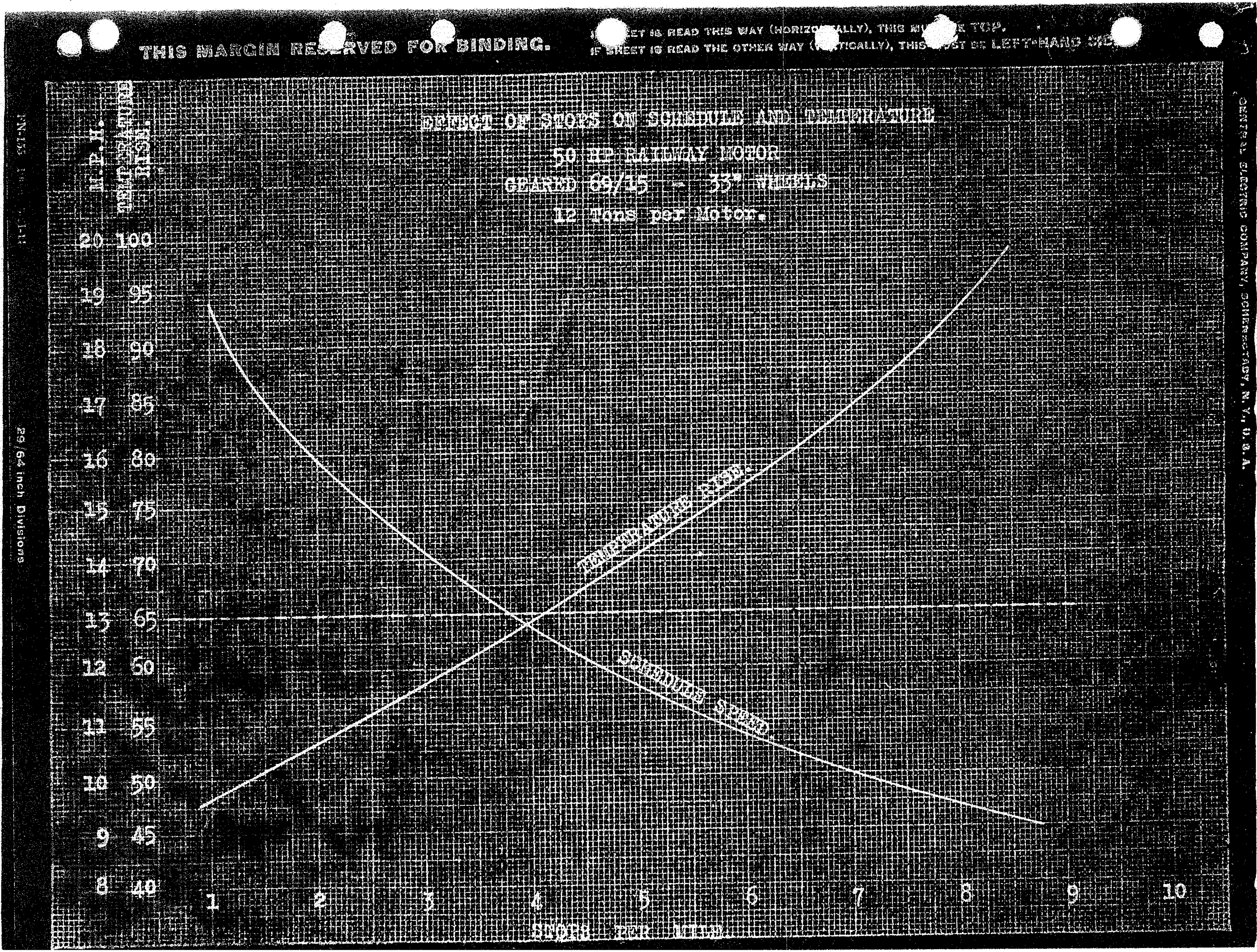
An improvement has been made in the design of pinions of small diameter by making them slightly bell-mouthed at the large end of the bore for a distance of 3/8 to 1/2 inch from the end of the pinion. By relieving the pinion in this way, so that for this distance it has no bearing on the shaft, the metal in the body of the pinion at the large end of the bore is stressed less when the pinion is driven and shrunk on the shaft. Consequently, there is less danger of failure from breakage both in the body of the pinion and in the teeth. Incidentally, this permits of a better design of shaft, since it is possible to use a fillet with a larger radius on the shaft between the pinion fit and the journal bearing.

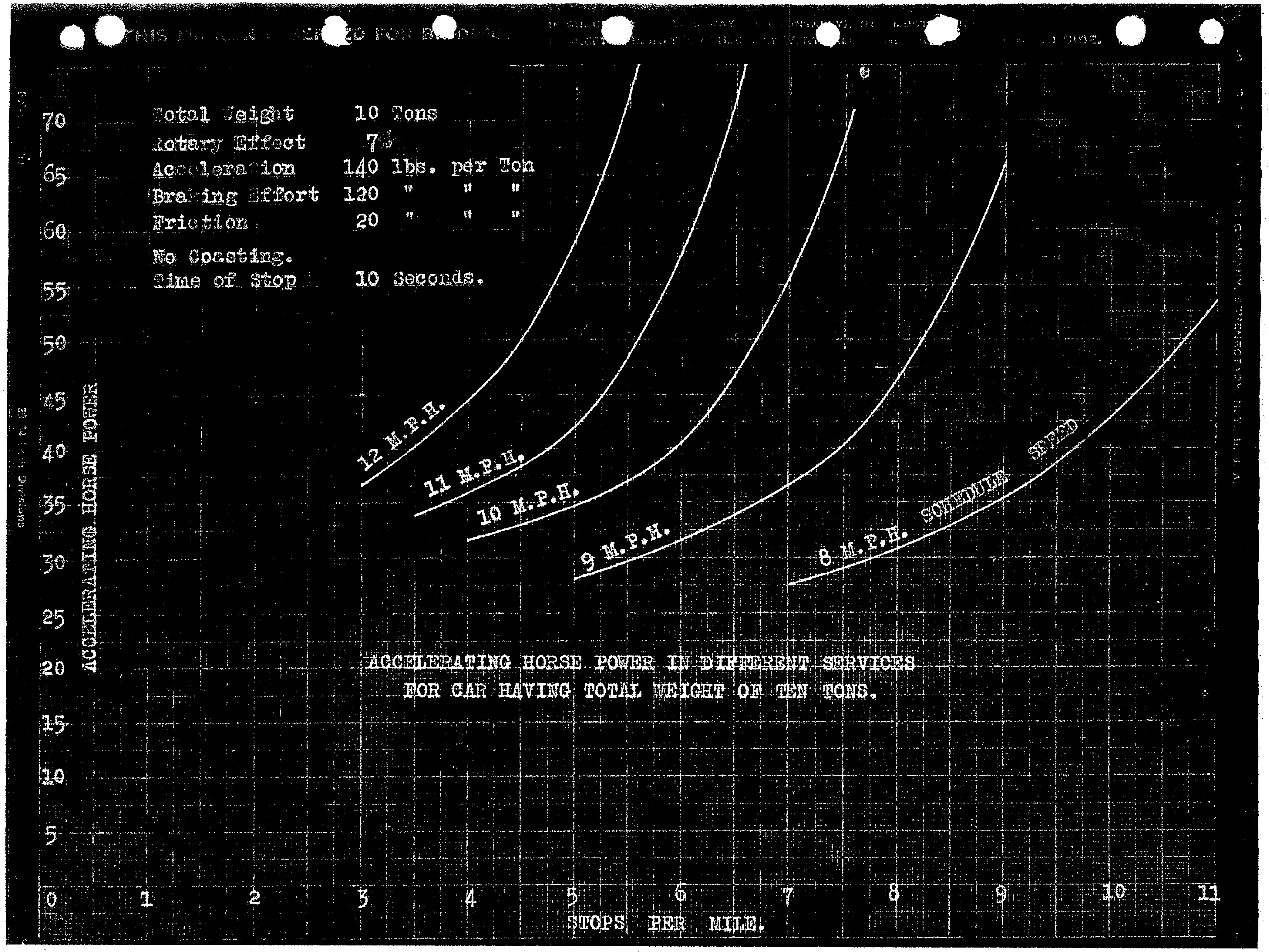
In the modern light weight motor, used on safety cars, very careful consideration has been given to the design of the motor with particular reference to armature shaft and gearing. The construction of the motor in other particulars is also worked out to secure maximum strength, reliability, effective ventilation, and lightness. This has resulted in the development of motors with continuous ratings equal to that of earlier types of non-ventilated motors of three to four times the weight.

In practical operation these light weight motors do not have increased service capacity in full proportion to their increased continuous rating. This is due to the fact that there is a larger short-time thermal capacity in heavy motors than in light motors, the heat generated being absorbed in the mass of material and slowly dissipated during periods of light load. However, the modern light weight motor is capable under ordinary operating conditions of handling a car about twice as heavy per pound of motor and of doing this with a much lower temperature rise. In fact, the service temperature of a modern safety car motor does not usually exceed 40 deg. rise as compared with older and heavier motors which are ordinarily run at a temperature of 60 to 65 deg. rise.

Street railway motors are now so efficiently ventilated that there is generally no substantial advantage in using heat proof insulation, since the losses are so effectively dissipated that it is questionable whether there is economy in operating at higher temperatures with increased losses and decreased power efficiency. Good ventilation has made it possible to use with economy non-heat-proof insulation which is cheaper in material cost and in application and is also more impervious to moisture.

Some of the railway motor improvements which have been briefly outlined are the most marked and far reaching that have been made during the past half dozen or more years. It would be possible to enumerate other improvements. Only the "high spots" can be touched in a short article and doubtless the writer has not mentioned all of these. Railway motor problems are being given constant study and further improvements will surely be made. However, the prediction of the writer, in the article referred to at the beginning, that "A pound of material will be made to do more and better work" has already been fulfilled in large measure.


THIS MARGIN RESERVED FOR BINDING.
EFFECT OF STOPS ON SCHEDULE AND TEMPERATURE
50 HP RAILWAY MOTOR
GEARED 69/15 - 33" WHEELS
12 Tons per Motor.
M.P.H.
TEMPERATURE RISE.
20 100
19 95
18 90
17 85
16 80
15 75
14 70
13 65
12 60
11 55
10 50
9 45
8 40
1
2
3
4
5
6
7
8
9
10
STOPS PER MILE.
TEMPERATURE RISE.
SCHEDULE SPEED.
29/64 inch Divisions
GENERAL ELECTRIC COMPANY, SCHENECTADY, N.Y., U.S.A.


Total Weight 10 Tons
Rotary Effect 7%
Acceleration 140 lbs. per Ton
Braking Effort 120 " " "
Friction 20 " " "
No Coasting.
Time of Stop 10 Seconds.
ACCELERATING HORSE POWER
70
65
60
55
50
45
40
35
30
25
20
15
10
5
0
1
2
3
4
5
6
7
8
9
10
11
STOPS PER MILE.
12 M.P.H.
11 M.P.H.
10 M.P.H.
9 M.P.H.
8 M.P.H. SCHEDULE SPEED
ACCELERATING HORSE POWER IN DIFFERENT SERVICES
FOR CAR HAVING TOTAL WEIGHT OF TEN TONS.

INTERNATIONAL GENERAL ELECTRIC COMPANY
INCORPORATED

SALES AND ENGINEERING DATA FILE 1041 - SERIAL 70

Schenectady, N.Y., July 23, 1930.

SUBJECT: RAILWAY MOTORS - DIRECTION OF ROTATION

The General Standardizing Committee has approved the following rules governing direction of rotation of railway motors, clockwise or couhter-clockwise. These rules will be the standards of the Company and should be followed in all cases.

In all cases, the observer will stand at the designated end of the machine facing toward the machine and in line with the rotating shaft.

Railway motors will be viewed from the commutator end; when built with two commutators, from the end opposite the pinion; when built with two commutators and two pinions the observer will face the end of the machine where the axle bearings are at his right.

A. Michaelis

Item	Type of Enclosure	Method or Temperature Determination to be Employed	Limiting Temperature Rise Deg. Cent. One-Hour Rating Class A Insulation	Limiting Temperature Rise Deg. Cent. One-Hour Rating Class B Insulation	Limiting Temperature Rise Deg. Cent. Continuous Rating Class A Insulation	Limiting Temperature Rise Deg. Cent. Continuous Rating Class B Insulation
1. Armature and field winding	Ventilated	Resistance	100	120	85	105
		Thermometer	80	95	65	80
	Totally Enclosed	Resistance	110	130	95	115
		Thermometer	90	105	75	90
2. Cores and Mechanical parts in contact with or adjacent to insulation	Ventilated	Thermometer	80	95	65	80
	Totally Enclosed	Thermometer	90	105	75	90
3. Commutators	Ventilated	Thermometer	95	110	80	95
	Totally Enclosed	Thermometer	105	120	90	105

4. Miscellaneous parts (such as brush holders, brushes, pole tips, etc.) other than those whose location is such that they may injuriously affect the adjacent insulation may attain such temperatures as will not be injurious in any other respect.

SERVICE CONDITIONS

Under the individual specifications of the more popular motors in Section II a tabulation of performance data is given, based on assumed service conditions outlined in each case. It is, therefore, important that this information be used judiciously because service conditions differing from the original hypothesis may change the results entirely.

A leeway of 10% over calculated results was allowed to care for slight variations from the assumed conditions. Where very heavy grades and numerous curves or other severe conditions are encountered, recommendations on the proper equipment should be obtained from the Engineering Department.

Tractive Resistance varies with different types of equipment and speed. A tractive resistance of 20 lb. per ton was used in making the calculations, which value is considered to be a fair average for frequent stop service.

A discussion of factors which affect schedule and heating follows:

VOLTAGE

Schedule speed varies with the voltage in approximately the following ratio:

Variation in Voltage	Stops per Mile	Variation in Schedule
1%	1	0.5%
1%	3	0.2%
1%	7	0.1%

It is evident from this that voltage variation is relatively unimportant in frequent stop service, since the greater portion of the time is spent in accelerating and braking. It may play an important part in influencing the schedule speed in infrequent stop service.

GEARING

By gear ratio or reduction is meant the actual number of gear teeth divided by the number of pinion teeth. It must first be remembered that a change in gear ratio affects the free running speed of the equipment. For a given change in gear ratio, although the free running speed is changed by a definite amount, the effect on schedule speed depends upon the service conditions, such as, stops, slow-downs per mile and duration of stops.

In general the maximum gear reduction which can be used and perform a desired schedule is the most economical as regards energy consumption. It also tends to keep the motor heating to a minimum.

One of the usual means of increasing the schedule speed is by decreasing the gear ratio but this must be done with discretion because the heating effect as well as the energy consumption increases faster than the schedule speed.

The effect on energy consumption and motor heating of decreasing the gear ratio is greater in frequent stop than in infrequent stop service.

WHEEL DIAMETER

The above, of course, applies for a given wheel diameter. Should the wheel diameter be changed with a given gear ratio, similar conditions would apply, that is, increasing the wheel diameter corresponds to decreasing the gear reduction and vice versa.

For a given current, car speed varies inversely with the gear reduction or directly with the wheel diameter; and tractive effort varies directly with gear reduction and inversely with wheel diameter.

STOPS AND SLOW-DOWNS

It is very evident that the number of stops per mile influence the schedule speed and as the number of stops increases it becomes more and more difficult to do anything that will improve the schedule. Under these conditions the schedule may be improved to a certain extent by shortening the duration of stops when possible, and increasing the accelerating and braking rates.

RATE OF ACCELERATING AND BRAKING

In long-run service the portion of time spent in acceleration and braking is a very small part of the total time; but in frequent stops or congested service acceleration and braking occupy the greater part of the time in motion. Therefore, it is evident that increasing the rates of acceleration and braking in frequent stop service would improve the schedule, but at the same time the temperature rise of the motors would be increased.

PROFILES

Grades tend to slow up schedule and increase the heating of motors.

Track curvature plays an important part in reducing schedule speeds due to the attendant slow-downs.

INFORMATION REQUIRED ON PROPOSITIONS

Sheets 1 and 2 of form 12745, shown on pages 15 and 17, should be carefully filled out and sent to the Engineering Department when motor recommendations are requested. The information contained on these sheets is needed to properly select the motor equipments required for the given service.

The sheets are self-explanatory and were prepared with the idea of simplifying the labor of making an adequate survey of service conditions.

GENERAL ELECTRIC COMPANY

CAR EQUIPMENT DATA—GENERAL

CUSTOMER

LOCATION

PROPOSITION NO. | REQUISITION NO.

INSTRUCTIONS: For Propositions or Requisitions,

Submit Form 12745

—Sheet ONE	—In all cases, and
—Sheet TWO	—When MOTORS are included,
—Sheet THREE	—When CONTROL is included,
—Sheet FOUR	—When AIR BRAKES are included.

A. Rolling Stock

	NEW CAR Motor	NEW CAR Trail	PRESENT CAR Motor	PRESENT CAR Trail
1. Weight in pounds......less air brake, motor,				
......and control equipments				
......complete, less load, ready to run				
2. Seating capacity......				
3. Number of axles per car......				
4. Wheel diameter—inches......				
5. Master Controllers on trail cars?......				
—Single or double end?......				
6. Operation—single or double end?......				
7. Number of motors per car......				
8. Type of motor } Preferred for new car				
9. Hp. rating } Preferred for new car				
10. Gear ratio } Preferred for new car				

B. System Data

1. Traffic classification

Congested......	
Non-congested......	
Interurban......	
Subway......	
Elevated......	
Baggage......	

2. General topography

—Level......	
—Rolling......	
—Hilly......	

3. Dual-voltage operation

Full Speed—1500 Volts only......	
—1200 Volts only......	
—1500 and 600 Volts......	
—1200 and 600 Volts......	
Half-speed operation on 600 Volts......	

4. Trolley voltages

Nominal system voltage......	
Maximum voltage at substation......	

* Service	Congested	Non-Congested	Inter-urban
Average voltage at car—power on......			
Minimum voltage at car—power on......			

*NOTE Congested service—7 or more stops per mile
Non-congested service—3 to 7 stops per mile
Interurban service—less than 3 stops per mile } See reverse side of Sheet TWO for details

5. Train make-up (use symbols; M = Motor Car, T = Trail Car)

Average train in

—Normal service......	
—Rush service......	

Maximum number of	M Car	T Car
—Cars in train (revenue service)......		
—Trail cars per motor car (revenue service)......		

Form 12745 (Supersedes 11273) 8m 4-11-28

GENERAL ELECTRIC COMPANY

CAR EQUIPMENT DATA—MOTORS

CUSTOMER
LOCATION
PROPOSITION NO. REQUISITION NO.

IMPORTANT—READ OTHER SIDE

C. Mechanical Details.

1. Position of motor leads—axle or suspension side?
2. Axle diameter in motor axle bearings—inches
3. Axle diameter in gear—inches
4. Gears—split or solid?
5. Track gauge—inches
6. Style of axle—A.E.R.A. classification
(If axle is not standard A.E.R.A. type, submit axle drawing.)

D. Route.

1. Distance—round trip—* Congested service—miles
—* Non-congested service—miles
—* Interurban service—miles
2. Grades—if profile of route is not submitted, substitute tabulation of grades, including percentages, lengths in feet, directions (up or down), sequence, and location on route for round trip.

E. Service.

	Normal			Rush		
*Service	Congested	Non-congested	Inter-urban	Congested	Non-congested	Inter-urban
1. Round trip time (no layovers)—minutes						
2. Total layover time (round trip)—minutes						
3. Total number of stops (round trip)						
4. Average duration of stops—seconds						
5. Total number of slowdowns (round trip)						
6. Average number of passengers (motor car)						
7. Average number of passengers (trail car)						
8. Number of trail cars per motor car						
9. Number of consecutive trips in rush service						

10. Do present cars successfully perform service given in Item E, 1-9?
11. Limiting conditions—describe below in detail the factors which limit the speed or capacity of equipments on any single trip, if such exist.

NOTE: **Customer's signature approves the use of data given under Items A, B, C, D and E, Form 12745, Sheets ONE and TWO, for determining the suitability of the motors under consideration for the proposed service.**

SIGNED
BY
DATE

*See reverse side of this sheet for details of Service Classification.

Form 12745-1 (Supersedes 11273) 8m 4-11-28

SERVICE CLASSIFICATION

An accurate description of service conditions involves the division of the route into a number of sections, selected either for convenience in taking data, or because of obvious changes in the character of the service. Classify all sections thus chosen as "Congested," "Non-congested", or "Interurban" according to the following table:

SERVICE	AVERAGE NUMBER OF STOPS PER MILE	USUALLY FOUND IN
"Congested"	7 or more	Congested city districts; heavy traffic sections.
"Non-congested"	3 to 7	Non-congested city districts; residential sections.
"Interurban"	less than 3	Sparsely settled districts; open country.

Summarize the data for all sections in each class and enter results under Item E, 1-9. It should not be assumed that the service changes from "Congested" to "Non-congested" or from "Non-congested to "Interurban" at such places as the city limits.

SERVICE DATA

Under Item E, 1-11, describe the proposed service in detail. Where this service is to be operated over an existing route, the data, descriptive of such service conditions as will remain unchanged, should consist of the averages of a number of actual observations made according to the following notations.

General

Observations should be made for a number of round trips representative of daily conditions. Determine the averages of such observations for each class and type of service in terms of one round trip.

Number of Stops—Item E-3

Where possible such data should be determined from observations. It will usually differ from the number of timetable scheduled stops, or the number of possible stopping places.

Slowdowns—Item E-5

Slowdowns may be defined as temporary reductions in car speed to approximately half the maximum speed normally reached in service. Any slowdown to more or less than half speed should be evaluated. For instance, one slowdown to three-quarters speed and another to one-quarter speed may be considered as approximately equal to two standardized slowdowns.

Number of Passengers—Item E-6, 7

The average numbers of passengers in all services should be determined by uniformly frequent counts of all the passengers on the car, and the division of the total number counted in each service by the number of counts made in such service.

TROLLEY VOLTAGES—Item B-4

Where possible these data should be determined from actual observations. Readings taken at the car should be made while *power is on the motors.* The average values in all services are the averages of such readings made over a period of time for each service. They should not be taken as the mean of the maximum and minimum values.

NOTE

The careful entry on this form of the specified data, whether compiled by engineering means or taken from actual observations, will enable the GENERAL ELECTRIC COMPANY to determine the most suitable motor equipment for the particular service involved. Accuracy in the making of tabulations and in the taking of observations will avoid the mis-application of motor equipments. It is assumed that the motor equipments will be handled in a careful and proper manner when performing the specified service. Brakes will be adjusted to avoid dragging and will be released at all times while power is on.

GENERAL ELECTRIC COMPANY

CAR EQUIPMENT DATA—CONTROL

CUSTOMER
LOCATION
PROPOSITION NO. REQUISITION NO.

F. General Information.

1. Type of Control—Drum
   —Multiple Unit

2. Motor Resistors—BG
   —EW
   —RG
   —CG

3. Circuit—Ground Return
   —Metallic Return

4. Headlight

	Fixed	Portable
—Arc		
—Incandescent		

5. Current Collector
   —Wheel Trolley
   —Pantograph
   —3rd Rail Shoes

6. Lighting Equipment
   —Special
   —G-E Standard

7. Lightning Arrester
   —Air Gap Type
   —Aluminum Cell

G. Drum Control.

1. Type—K Controller
   —B Controller

2. (Overload—Circuit Breaker
   Protection)—Line Breaker
   —Main Fuse

H. Multiple Unit Control.

1. Is operation required in train with existing equipments?
2. If required, state type of existing equipments

   Note: If above existing equipments are not of G.E. manufacture, a wiring diagram of the equipments is required.

3. Acceleration—Hand Control
   —Automatic Control

4. Couplers—Dashboard
   —Platform
   —Automatic

5. Master controllers—without dead man's handle
   —with dead man's handle
   —with dead man's handle and emergency air brake

6. Bus Line for Motor Circuit?—Yes
   —No

7. Bus Line for Heaters and Lights?—Yes
   —No

8. Control Voltage—600 Volts.
   —Generator
   —Battery

9. Car Panel?—Yes
   —No

10. Auxiliaries Motor Generator
    —Dynamotor

Form 12745-2 (Supersedes 11273) 8m 4-11-28

GENERAL ELECTRIC COMPANY

CAR EQUIPMENT DATA—AIR BRAKES

CUSTOMER
LOCATION
PROPOSITION NO. | REQUISITION NO.

J. General Information.

1. Type of Equipment
   - —Straight Air
   - —Straight Air with Emergency Feature
   - —Automatic Variable Release
   - —Combined Straight and Automatic
   - —Air Brake and Safety Car Control Equipment

2. Compressed Air Required for Operation Of:
   - —Sanders
   - —Whistles
   - —Bell Ringers
   - —Snow Scrapers
   - —Other Devices

3. Type of Air Couplings
   - —Standard Hose Couplings
   - —Automatic Couplers with Air Connections

4. Place of Attachment for Standard Hose Couplings
   - —Car Body
   - —Radial Draw Bars

K. Air Brake and Safety Car Control Equipment Details.

1. Type and Form of Car Controller

2. Doors
   - —Folding—
   - —Sliding —

3. Door Engine Mounting
   - —Overhead
   - —Floor

4. Will cars be equipped with a Line Breaker which has a pneumatic tripping cylinder as an inherent part?

5. Will cars be equipped with
   - —LB-4 Control Device?
   - —Westinghouse TA Control Device?

6. If door control is other than standard, submit a full description of operation.

Form 12745-3 (Supersedes 11273) 8m 4-11-

INFORMATION SHEET

Read the other side of this sheet carefully

DATA REQUIRED BY THE GENERAL ELECTRIC COMPANY FOR RAILWAY MOTORS

on the .... Railway

MOTOR CARS: Seating capacity.......... Capacity with standing load.......... Length of car over all..........

Weight of empty cars and trucks not including electrical equipment.............................. lb.

Have cars single or double trucks?.......... No. of motors per car.......... Diam. of car wheel.......... in.

TRAIL CARS: Seating capacity.......... Capacity with standing load.......... Length of car over all..........

Weight of empty cars and trucks.............................. lb.

LINE POTENTIAL: Maximum voltage is.......... Minimum voltage is.......... Average voltage is..........

TOTAL DISTANCE round trip.............. miles.

Distance round trip in city service.......... miles. Suburban.......... miles. Interurban.......... miles.

AVERAGE SERVICE: Time, excluding layovers required to make round trip.............. minutes.

Time excluding layovers, to make round trip, city service.......... minutes. Suburban service.......... minutes. Interurban service.......... minutes.

No. of stops in round trip, city service.......... Suburban.......... Interurban..........

No. of slow downs, round trip, to approx. half speed, city service.......... Suburban service.......... Interurban service..........

No. of trail cars handled by motor car.......... No. of consecutive round trips with trail cars..........

Average passengers during round trip, motor car.......... Trail cars..........

RUSH SERVICE: Time excluding layovers required to make round trip.............. minutes.

Time required to make round trip, city service.......... minutes. Suburban service.......... minutes. Interurban service.......... minutes.

No. of stops in round trip, city service.......... Suburban.......... Interurban..........

No. of slow downs, round trip, to approx. half speed, city service.......... Suburban service.......... Interurban service..........

No. of trail cars handled by motor car..........

No. of consecutive round trips per car, motor car only.......... Motor car and trail cars..........

Average passengers during round trip, motor car.......... trail cars..........

DURATION OF STOPS: For average city service.......... seconds. Suburban.......... seconds. Interurban.......... seconds.

For rush city service.......... seconds. Suburban.......... seconds. Interurban.......... seconds.

LAYOVERS: Round trip, number and duration..............................

GRADES

Underscore grades which cars both ascend and descend in round trip.

Length in ft.	%	Length in ft.	%	Length in ft.	%	Length in ft.	%	Length in ft.	%

MOTOR: Motor frame box or split?.......... MOTOR LEADS: Axle or suspension side?..........

AXLE DIAMETER: In motor axle bearings.......... inches. In gear.......... inches. In axle collar .... inches. Should gear be split or solid?.......... If solid gear give *exact* bore.......... inches.

Distance between car wheel hubs.........., Style of axle..........

Track gauge.......... inches.

REMARKS:

Dated.............................. 191....... Signed..............................

Form 11273 (Supersedes 11018) 5m 4-2-17 **READ OTHER SIDE** By..............................

READ CAREFULLY BEFORE FILLING IN FORM

By filling in this Information Sheet accurately, the General Electric Company is in a better position to determine the most suitable size of motor, ratio of gearing, winding, etc., which affect not only the energy consumption, but also maintenance costs.

In order that the temperatures of motors in service may not be excessive, it is important that the conditions under which motors are to operate be carefully studied and correct data as regard running time, stops, distances, slow downs, etc., be given on this data sheet.

Care should be taken to give the number of stops as counted in actual service in the distances and corresponding running times designated and not the number of possible stopping places.

The "average service" may generally be taken as occurring between one and five p.m., and the "rush service" from five to seven p.m.

The average passengers per car during a round trip may be determined by counting all the passengers in the car at 5 minute intervals and dividing the sum of the passengers counted each time by the number of times the count is made.

It frequently happens that during the "rush service," the schedule speed, number of stops and slow downs, as well as passenger load is greatly different from the average conditions, and is of such a character as to be limiting as regards motor capacity.

The average voltage should not be given as an average of the maximum and minimum, but the average voltage maintained at the car only during the time power is applied to the motors and not including the time of coasting, braking, or standing.

Unsatisfactory operation of motor equipment has resulted, owing to incorrect service data having been given, and customer and agents are requested to give as accurate and complete information as possible.

The General Electric Company will assume in making engineering recommendations that the motor equipment will be handled in a careful and proper manner when performing the service specified. Brakes will be adjusted so that they will not drag and will be in the released position while power is on.

APPARATUS SALES ADVICE No. RY-2 . 145

Department: RAILWAY

Classification: GENERAL

Subject: CAR EQUIPMENT SERVICE DATA

Schenectady, August 14, 1928

We believe all the district and local offices realize the importance of furnishing complete service data, either when a recommendation is requested or with the requisition if no recommendation has been made.

Our Service Data Sheet (form 11273) has been found inadequate, and in an effort to obtain complete data, on which an intelligent recommendation can be made, we have prepared a new set of forms, as follows:

Form 12745	Car Equipment Data	General
Form 12745.1	" " "	Motors
Form 12745.2	" " "	Control
Form 12745.3	" " "	Air Brakes

Copies of each are attached.

Please remember that the careful and intelligent use of these forms with every request will assure the correct recommendation being made for each proposition and will result in more prompt shipment and more satisfied customers. We request your cooperation.

E. P. WALLER

Manager Railway Department

Distribution:

Sales Office Managers
Railway Representatives
225

International General Electric Co.,
Incorporated.

2122

SALES AND ENGINEERING DATA - File in Folder 1041.

Schenectady, N.Y., October 11, 1922.

SUBJECT: Railway Car Equipment - Replacement of Obsolete Motors.

Below is quoted a letter recently sent out by the Manager of the Railway Department to all General Electric Railway Specialists, calling their attention to sales possibilities in the matter of replacing obsolete railway motors on an economic basis. This letter should apply equally well to other countries and should receive your earnest consideration.

"There are still in active service in this country thousands of motors so old and of such obsolete design that it would be a measure of distinct economy to replace them with modern machines. Better material, more skilful assembly, and improvements in design combine to produce motors that for a given service are not only lighter in weight, and lower in power consumption, but also more reliable, subject to fewer failures in service, and which have a materially lower maintenance cost.

For instance, the records of a large Eastern property show during 1920 armature failures as follows on cars in active service:

Type of Motors	No. in Service	Armatures rewound	%
GE-67-GE-80-W68-W-101	3785	1004	25.5
GE-258-264-203- W-506	609	53	8.7

The average age of the first group was about 12 years; of the second group about 5 years. In other words, this is not a record of brand new motors; only about 15% of the second group had been running less than two years, and 40% had been in service about 7 years.

On another large Eastern property, during 1920, 1885 GE-58 motors, used mainly in tripper service, and therefore worked less severely than on all day runs, averaged $7.30 per 1000 motor miles for inspection and repairs, whereas 1412 GE-203 and GE-247 motors in regular service averaged only $1.19 per 1000 miles in their fourth year of operation.

On a property in the Southwest, 220 GE-54 motors in 1921 averaged $4.49 per 1000 motor miles, and 104 GE-200 motors in their ninth year of service, cost only $1.42. A large part of the older motors were on single truck cars, used only for tripper service, while the GE-200's were all kept on the road to the maximum extent.

Another Southern property reports costs on their older motors, GE-800, GE-1000, GE-57, GE-58, GE-67, GE-80, W-12, W-56, W-101 (about 1000 of which were in operation in 1920); ranging from about $2.50 per 1000 miles on the best to about $5.40 for the worst, and averaging about $3.75, as compared to approximately $.50 per 1000 motor miles for 100 GE-201 motors that had been in service five years.

In general, experience shows that modern motors properly applied should not cost more than $1.00 per 1000 motor miles, or from $30.00 to $40.00 per motor annually to keep them in first class operating condition. On most of those sold prior to 1912, costs will average four to five times as great, or from $120.00 to $200.00 per motor annually on the same mileage basis.

The direct saving in maintenance will frequently show 20% to 25% return on the cost of replacement, while the other savings may add considerably to this. For instance, in New Orleans it was proved that by replacing GE-800 and GE-1000 motors with the GE-247, a materially faster schedule could be operated, the new motors being capable of more rapid acceleration and higher free running speeds, and that a smaller number of cars with the new motors could provide the same service, with material economies in platform wages.

The reduction in weight that can be obtained by motor replacement, with consequent savings in power cost and track maintenance, is considerable and should be considered. For instance, the GE-265 motor which will usually be found suitable for use on the older single truck two motor cars, or double truck four motor equipments in city service weighs 1500 lb. The weights of some of the motors for which it might be substituted are as follows:

GE- 800	- 1880 lb.	GE-54	- 1900 lb.	GE- 70	- 2750 lb.
" 1000	- 2180 "	" 58	2225 "	" 80	2850 "
" 52	1890 "	" 67	2450 "	" 81	2020 "

It is evident, therefore, that on a two motor equipment from 760 to 2700 lb. can be saved, and on a four motor car from 1500 to 5400 lb. A still further reduction in weights can be secured by dropping the cars down on to low wheel (24" to 26") trucks, and utilizing the low wheel motors. A saving of about 2000 lb. per truck and an additional saving of 100 lb. per motor can be made in this manner. The Winnipeg Electric

Railway have done this with nearly 200 old cars, equipping them with 26" wheel trucks and GE-258 motors, and more recently the Dayton City Railway have re-equipped 15 cars that had maximum traction trucks and two 65 HP motors with 26" wheel trucks with four GE-264 motors.

Summarized, there are many places where motor replacement can be justified by the reduction in maintenance expense and increased reliability of service. In other cases, as in New Orleans and in Los Angeles, the ability to make a better schedule may be the determining factor. In others, the necessity of pulling trailers may force the use of a larger motor; this has recently led to several purchases in New Orleans. In other cases, as in Dayton and Winnipeg, the desirability of lowering step heights and reducing weight has led to the Motor replacement. All of these points should be in regular service as factors that might justify an investment in new equipment where the maintenance saving by itself might be considered insufficient.

The information given above has been prepared by Mr. J. C. Thirlwall and we hope you will make use of these facts to prove to your customers that their old motors should be replaced."

W. A. FALLON

COMMERCIAL ENGINEERING DEPARTMENT.

WAF:SSM

TYPICAL COSTS OF MODERN MOTORS

Type	GE-76	GE-212	W-312	GE-213	GE-216	GE-227	GE-86
Road	I	E	F	A	A	F	F
Number in service	346	1600	100	382	1028	300	1000
Annual mileage	38,000	55,000	15,000	39,000	36,000	34,000	20,000
Cost per 1000 M. miles							
Inspection	.03	.03	.04	.03	.03	.04	.04
Repairs	.02	.25	.24	.23	.26	.29	.22
Total	.05	.28	.28	.26	.29	.33	.26
Annual cost 40,000-mile basis	2.00	11.20	11.20	10.40	11.60	13.20	10.40

Type	GE-200			GE-203					W-307	GE-201	W-306
Road	D	B	Total	C	G	B	D	Total	C	D	K
Number in service	40	300	340	120	8	224	26	378	136	122	408
Annual mileage	23,000	30,000	29,000	40,000	70,000	40,000	30,000	40,000	40,000	30,000	44,000
Cost per 1000 M. miles											
Inspection	.03	.04	.04	.03	.03	.10	.04	.07	.03	.03	.03
Repairs	.065	.20	.18	.05	.09	.20	.42	.15	.11	.035	.30
Total	.095	.24	.22	.08	.12	.30	.46	.22	.14	.065	.33
Annual cost 40,000-mile basis	3.80	9.60	8.80	3.20	4.80	12.00	18.40	8.80	5.60	2.60	13.20

TOTAL FOR MODERN MOTORS—12 TYPES

Total number of motors	6140
Average miles per year	39,000
Average inspection cost per 1000 miles	.035
Average repair cost per 1000 miles	.18
Average total cost per 1000 miles	.215
Average cost per motor annually, dollars	8.60

The above motors have been in service 2 to 6 years—average age about 3 to 4 years.

*MOTORS WHICH COULD ORDINARILY BE REPLACED BY THE GE-203-P MOTOR

* MOTORS WHICH COULD ORDINARILY BE REPLACED BY THE GE-240

Type	GE-57				W-56	GE-74			GE-90			GE-73			GE-68
Road	A	J	B	Total	A	A	F	Total	H	D	Total	A	F	Total	F
Number in service	584	20	82	686	16	100	92	192	12	12	24	188	90	278	138
Miles per motor per year	38,000	30,000	20,000	35,000	50,000	33,000	25,000	30,000	55,000	15,000	35,000	35,000	31,000	34,000	29,000
Cost per 1000 M. miles, dollars															
Inspection	.12	.10	.55	.17	.15	.16	.19	.17	.07	.25	.16	.05	.13	.08	.25
Repairs	.99	1.73	3.18	1.28	1.30	1.39	2.61	1.98	.71	5.08	2.89	.43	2.12	1.00	2.70
Total	1.11	1.83	3.73	1.45	1.45	1.55	2.80	2.15	.78	5.33	3.05	.48	2.25	1.08	2.95
Annual cost per motor 40,000-mile basis	44.40	73.20	149.20	58.00	58.00	62.00	112.00	86.00	31.20	213.20	122.00	19.20	90.00	47.20	118.00
Estimated annual cost GE-203-P	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	12.00	12.00	12.00	12.00
Weight reduction per motor, lb.	850	850	850	850	800	1400	1400	1400	700	700	700	300	300	300	150
Saving per motor per year maintenance	34.40	63.20	139.20	48.00	48.00	52.00	102.00	76.00	21.20	203.20	112.00	7.20	78.00	35.20	106.00
Ditto, acct. weight	42.50	42.50	42.50	42.50	40.00	70.00	70.00	70.00	35.00	35.00	35.00	15.00	15.00	15.00	7.50
Total saving, dollars	76.90	105.70	181.70	90.50	88.00	122.00	172.00	146.00	56.20	238.20	137.00	22.20	93.00	50.20	113.50
Approx. interest on cost of new motor	18 %	25 %	42%	21 %	20 %	29 %	40 %	34 %	13 %	56 %	34 %	3 %	13 %	7 %	16 %
Mileage to pay 15 per cent	33,000	24,000	14,000	29,000	30,000	20,000	15,000	18,000	46,000	11,000	18,000	200,000	46,000	85,000	32,000

* Specific recommendations for the motors to replace these types should be secured from the engineers and must be based on service data.

OBSOLETE MOTORS WHICH CAN BE REPLACED BY THE GE-247 MOTOR

Type	WP-30				SRG			GE-800				
Road	A	F	B	Total	C	A	Total	C	A	F	K	Total
Number in service	272	1442	437	2151	35	36	71	35	82	188	180	485
Miles per motor per year	5600	5600	1500	4800	15,000	3300	9000	15,000	32,000	11,000	5000	13,000
Cost per 1000 M. miles, dollars												
Inspection	.24	.30	.72	.38	.20	.32	.26	.16	.14	.18	.24	.19
Repairs	2.04	3.95	12.50	5.42	.74	2.66	1.70	.74	1.15	1.67	4.20	2.46
Total	2.28	4.25	13.22	5.80	.94	2.98	1.96	.90	1.29	1.85	4.44	2.65
Annual cost per motor 40,000-mile basis	91.20	170.00	528.80	232.00	37.60	119.20	78.40	36.00	51.60	74.00	177.60	106.00
Estimated annual cost GE-247	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00
Weight reduction per motor, lb.	600	600	600	600	500	500	500	......	......	......	......	......
Saving per motor per year maintenance	83.20	162.00	520.80	224.00	29.60	111.20	70.40	28.00	43.60	66.00	169.60	98.00
Ditto, acct. weight	30.00	30.00	30.00	30.00	25.00	25.00	25.00	......	......	......	......	......
Total saving, dollars	113.20	192.00	550.80	254.00	54.60	136.20	95.40	28.00	43.60	66.00	169.60	98.00
Approx. inte est on cost of new motor	33 %	55 %	157 %	73 %	16 %	39 %	28 %	8 %	12 %	19 %	48 %	28 %
Mileage to pay 15 per cent	18,000	11,000	4000	8000	38,000	15,000	21,000	75,000	50,000	31,000	13,000	22,000

Type	GE-1000					W-12-A				JL-27	W-69
Road	C	A	B	H	Total	A	B	D	Total	G	C
Number in service	210	686	370	84	1350	22	40	44	108	4	200
Miles per motor per year	30,000	29,000	22,000	13,000	26,000	13,000	12,000	2000	8000	20,000	38,000
Cost per 1000 M. miles, dollars											
Inspection	.15	.17	.20	.20	.18	.16	.24	.20	.20	.18	.15
Repairs	.85	1.42	2.25	3.80	1.71	1.36	1.50	1.97	1.64	1.78	.92
Total	1.00	1.59	2.45	4.00	1.89	1.52	1.74	2.17	1.84	1.96	1.07
Annual cost per motor 40,000-mile basis	40.00	63.60	98.00	160.00	75.60	60.80	69.60	86.80	73.60	78.40	42.80
Estimated annual cost GE-247	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00
Weight reduction per motor, lb.	250	250	250	250	250	300	300	300	300	250	......
Saving per motor per year maintenance	32.00	55.60	90.00	152.00	67.60	52.80	61.60	78.80	65.60	70.40	34.80
Ditto, acct. weight	12.50	12.50	12.50	12.50	12.50	15.00	15.00	15.00	15.00	12.50	......
Total saving, dollars	44.50	68.10	102.50	164.50	80.10	67.80	76.60	93.80	80.60	82.90	34.80
Approx. interest on cost of new motor	13 %	19 %	29 %	47 %	23 %	20 %	22 %	27 %	23 %	24 %	10 %
Mileage to pay 15 per cent	46,000	32,000	21,000	13,000	26,000	30,000	27,000	22,000	26,000	25,000	60,000

OBSOLETE MOTORS WHICH CAN BE REPLACED BY THE GE-247 MOTOR

Type	GE-52—54					GE-58			W-92	GE-81
Road	D	A	B	K	Total	G	F	Total	G	D
Number in service	58	148	100	24	330	52	1950	2002	22	44
Miles per motor per year	20,000	38,000	15,000	7000	26,000	28,000	19,000	20,000	26,000	18,000
Cost per 1000 M. miles, dollars										
Inspection	.15	.17	.91	.30	.40	.08	.11	.11	.16	.18
Repairs	1.53	1.95	3.20	3.87	2.40	.67	1.49	1.47	1.24	1.72
Total	1.68	2.12	4.11	4.17	2.80	.75	1.60	1.58	1.40	1.90
Annual cost per motor 40,000-mile basis	67.20	84.80	164.40	166.80	112.00	30.00	64.00	63.20	56.00	76.00
Estimated annual cost GE-247	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00
Weight reduction per motor, lb.	....	....	....	....	....	300	300	300	300	100
Saving per motor per year maintenance	59.20	76.80	156.40	158.80	104.00	22.00	56.00	55.20	48.00	68.00
Ditto, acct. weight	....	....	....	....	....	15.00	15.00	15.00	15.00	5.00
Total saving, dollars	59.20	76.80	156.40	158.80	104.00	37.00	71.00	70.20	63.00	73.00
Approx. interest on cost of new motor	17 %	22 %	45 %	45 %	30 %	11 %	20 %	20 %	18 %	21 %
Mileage to pay 15 per cent	34,000	27,000	13,000	13,000	20,000	54,000	30,000	30,000	33,000	29,000

Type	GE-67								GE-70			
Road	K	C	A	G	D	B	H	Total	A	B	F	Total
Number in service	700	154	1390	20	80	306	88	2738	1640	176	132	1948
Miles per motor per year	37,000	40,000	31,000	12,000	20,000	21,000	35,000	32,000	33,000	43,000	26,000	34,000
Cost per 1000 M. miles, dollars												
Inspection	.07	.15	.12	.16	.24	.80	.15	.11	.07	.08	.13	.08
Repairs	.80	.85	1.00	1.64	2.26	1.84	2.72	1.14	.61	.89	1.37	.69
Total	.87	1.00	1.12	1.80	2.50	2.64	2.87	1.25	.68	.97	1.50	.77
Annual cost per motor 40,000-mile basis	34.80	40.00	44.80	72.00	100.00	105.60	114.80	50.00	27.20	38.80	60.00	30.80
Estimated annual cost GE-247	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00
Weight reduction per motor, lb.	500	500	500	500	500	500	500	500	800	800	800	800
Saving per motor per year maintenance	26.80	32.00	36.80	64.00	92.00	97.60	106.80	42.00	19.20	30.80	52.00	22.80
Ditto, acct. weight	25.00	25.00	25.00	25.00	25.00	25.00	25.00	25.00	40.00	40.00	40.00	40.00
Total saving, dollars	51.80	57.00	61.80	89.00	117.00	122.60	131.80	67.00	59.20	70.80	92.00	62.80
Approx. interest on cost of new motor	15 %	16 %	18 %	25 %	33 %	35 %	38 %	19 %	17 %	20 %	26 %	18 %
Mileage to pay 15 per cent	40,000	38,000	33,000	24,000	18,000	17,000	16,000	32,000	35,000	30,000	23,000	33,000

Type	GE-80						
Road	G	C	A	K	B	D	Total
Number in service	8	340	720	220	332	36	1656
Miles per motor per year	57,000	40,000	41,000	40,000	35,000	22,000	39,000
Cost per 1000 M. miles, dollars							
Inspection	.06	.08	.09	.07	.33	.17	.14
Repairs	.40	.40	.78	.88	1.50	1.83	.88
Total	.46	.48	.87	.95	1.83	2.00	1.02
Annual cost per motor 40,000-mile basis	18.40	19.20	34.80	38.00	73.20	80.00	40.80
Estimated annual cost GE-247	8.00	8.00	8.00	8.00	8.00	8.00	8.00
Weight reduction per motor, lb.	900	900	900	900	900	900	900
Saving per motor per year maintenance	10.40	11.20	26.80	30.00	65.20	72.00	32.80
Ditto, acct. weight	45.00	45.00	45.00	45.00	45.00	45.00	45.00
Total saving, dollars	55.40	56.20	71.80	75.00	110.20	117.00	77.80
Approx. interest on cost of new motor	16 %	16 %	20 %	21 %	32 %	33 %	22 %
Mileage to pay 15 per cent	37,000	37,000	30,000	29,000	19,000	18,000	27,000

FIELD CONTROL

The application of motors having field control involves an increase in the amount and cost of the control apparatus and under some conditions increases the weight and cost of the motor. Whether or not this extra expense is justified in any application is a matter to be decided only after an accurate study of the service conditions.

REPLACEMENT OF OBSOLETE MOTORS

There are in service on some properties, motors which are so old that it is uneconomical to run them because of excessive maintenance.

Following is a list of motors considered obsolete and a corresponding list of motors which replace them. By referring to the features of the various motors it will be seen wherein the newer designs are superior.

OBSOLETE G-E MOTORS	MODERN G-E MOTORS
52-54-81-800-1000	264-265
67-70-78-80-86-88-202) 213-216-219-226)	203-247
75-87-90-217-227	203-275
73-74-204-210-214	263-240
211-225	240-254-259

MECHANICAL DETAILS

FRAMES

Box frames have been adopted as the standard for G-E Railway Motors although split frames can be supplied for some of the motors.

G-E motor frames are made of cast steel with accurately machined frame head openings.

Machined lugs are provided on the bottom of the frame on which to set up the motor.

Adequate drain holes are provided in the bottom of the frames.

Lifting bails are cast integral with the frame.

Openings are provided either in the frame or frame heads for the entrance and exhaust of cooling air.

Location of openings depends upon the type of ventilation and design of individual motors.

Protection for the intake openings is provided by shields or by expanded metal grids, bolted over the intake openings.

Inspection is facilitated by a large opening over the commutator and wherever else necessary.

Commutator covers of punched sheet steel with ventilating holes and baffle plates are regularly furnished on the small multiple ventilated motors.

Solid winter covers are made which can be used over intake openings on multiple ventilated motors to keep out snow and water. On pages 28 and 29 is a list of winter covers for various motors.

WINTER COVERS

Motors	Winter Cover Cat. No.	Description
GE-239	1666194P1	Sheet steel used between magnet frame and ventilating frame cover at commutator end suspension side.
GE-240	1995441G1	Sheet steel cover, axle side, for (C.E.) frame head P-1155351 Pt. 1.
	1995441G2	Sheet steel cover, suspension side, for (C.E.) frame head P-1155351 Pt. 1.)
	1992717G1	Sheet steel cover for frame head (C. E.) P-1889952G1 (Standard form A and B frame head since 1920).
GE-247	2168316G1	Sheet steel for use with long skirt (C.E.) frame head.
	1666198P1	Sheet steel for use with G. E. frame head, P-1604051, cast prior to Dec. 1916.
*	2136993G1	Solid type frame cover used in place of punched type commutator cover.
GE-253	210010	Cover used on pinion end of frame. Ventilated but has hood to deflect snow from air opening.
GE-254	1666194P1	Sheet steel used between magnet frame and ventilating frame cover at commutator end suspension side.
GE-258	1666192P1	Sheet steel used between ventilating shield, M-1677590, and magnet frame commutator end.
	2168303G1	Sheet steel for use with long ventilating shield P-2136976.
GE-258-H	2168302G2	Sheet steel for use with ventilating shield M-1677590.
*GE-258	2136962G1	Solid top frame cover used in place of punched type commutator cover.
GE-263	1995443G1	Sheet steel for closing suspension side opening in bottom of skirt on commutator end frame head.
	1995443G2	For axle side, otherwise like 1995443G1.
*	2136992G1	Solid top frame cover used in place of punched type commutator cover.
GE-264	1817799P1	Sheet steel for use with short-skirt framehead
	2162491G1	Sheet steel for use with long-skirt framehead.

*With the new punched-cover ventilation, winter covers are unnecessary in city service, because of the slow speed, also in interurban service in milder climates. For the sections usually having severe winters the solid top frame covers may be used as protection on interurban lines; in city service, even in these sections, winter covers have not been found necessary with the punched cover ventilation.

WINTER COVERS (Cont'd)

Motors	Winter Cover Cat. No.	Description
*	2136962G1	Solid top frame cover used in place of punched type commutator cover.
GE-265	2101854P1	Sheet steel for use with short-skirt frame head.
	2168316G1	Sheet steel for use with long-skirt frame head.
	2807943G1	Sheet steel for use with short-skirt frame head (Axle side).
	2807943G2	Sheet steel for use with short-skirt frame head (Suspension side).
*	2136993G1	Solid top frame cover used in place of punched type commutator cover.
*GE-275	2136992G1	Solid top frame cover used in place of ventilated type commutator cover.
	2162484G3	Solid bottom cover for closing openings in bottom of frame commutator end (2 required).
GE-282-E	2697393G1	Sheet steel for commutator end suspension side of magnet frame.
GE-284	1666194	Sheet steel for commutator end suspension side of frame.
GE-285	2840397P1	Steel for commutator end suspension side of magnet frame.
*GE-288	2136993G1	Solid top frame cover to take the place of punched type commutator cover.
GE-290	2697389G1	Sheet steel, used on frame head opposite commutator end. Has hood to deflect snow.
GE-292	3627289P1	Sheet steel for commutator end suspension side of magnet frame.
*GE-294	2136993G1	Solid top frame cover to take the place of punched type commutator cover.
GE-297	3616596G1	Sheet steel for commutator end suspension side of magnet frame.
*GE-299	2136992G1	Solid top frame cover to take the place of punched type commutator cover.

Winter covers are to be used only during the winter season in localities where snow in quantity may be expected. Since they restrict the passage of air there is danger of motor burn-outs if used in mild climates or after the "snow season" has ended.

* With the new punched-cover ventilation, winter covers are unnecessary in city service, because of the slow speed, also in interurban service in milder climates. For the sections usually having severe winters the solid top frame cover may be used as protection on interurban lines; in city service, even in these sections, winter covers have not been found necessary with the punched cover ventilation.

SUSPENSION

Support of the motors is taken care of in the following way: One side is hung on the axle while the other is supported from the truck frame by means of lugs cast integral with the motor frame.

Modern motor suspension is divided into three types: Cast Nose, Bar, and Spring Nose. An idea of these various types may best be gained by consulting the following prints:

Spring Nose	Print No. 245605 Motor GE-265-A—Page 31
Cast Nose	Print No. 248576 Motor GE-282-D—Page No. 32
Bar	Print No. 440915 Motor GE-247-A—Page No. 33.
Spring Nose	Print No. 244764 Motor GE-286-A—Page No. 34.

The motor is suspended on the axle by means of axle brackets cast integral with the frame and by the axle caps. The frame and axle caps are bored with shims between the axle brackets and axle caps in order that the bearing linings will be firmly clamped in position.

Dust guards of sheet steel pressed into shape are assembled around the axle between axle caps. The assembly is shown on photograph 232417 page 35.

Axle caps besides clamping the bearing lining contain the oil wells and waste chambers.

FRAME HEADS—AXLE CAPS

Frame heads held by bolts have a drive fit in the frame. Two jack holes are provided in each frame head at diametrically opposite points.

Oil box covers are built with deep lips and are lined with felt to prevent dust and water from entering.

Drain pockets cast in the frame heads receive oil thrown off by the deflectors.

Drain plugs are fitted in the bottom of all oil wells.

Axle caps of cast steel hold the axle bearing linings in place and in addition contain the oil and waste chambers for lubricating the bearings. With two point supported gear cases the pinion end axle cap carries one of the gear case supporting lugs.

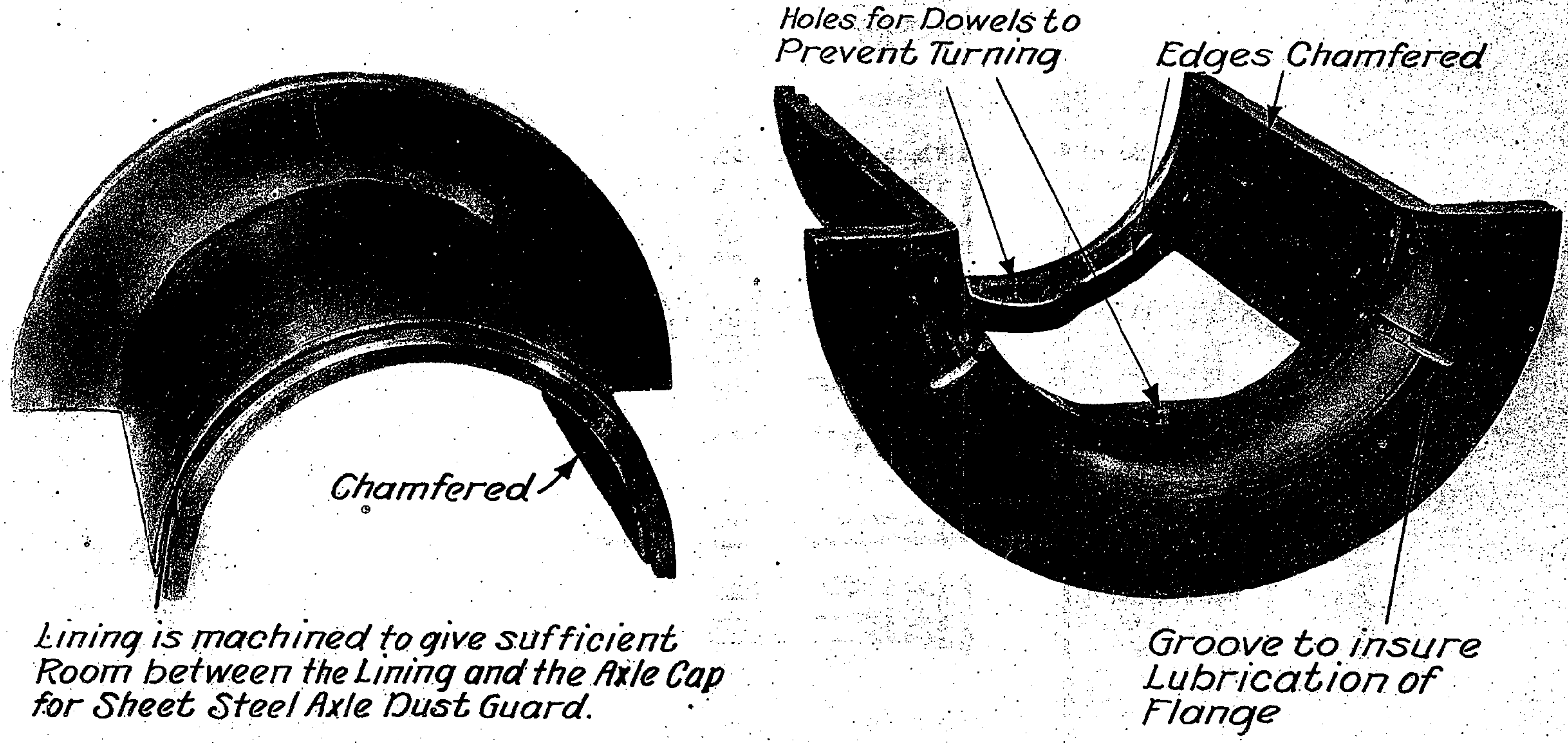
Oil box covers similar in construction to those on the frame heads prevent the entrance of dirt and water.

Frame heads and axle caps may be divided into three general types, namely:

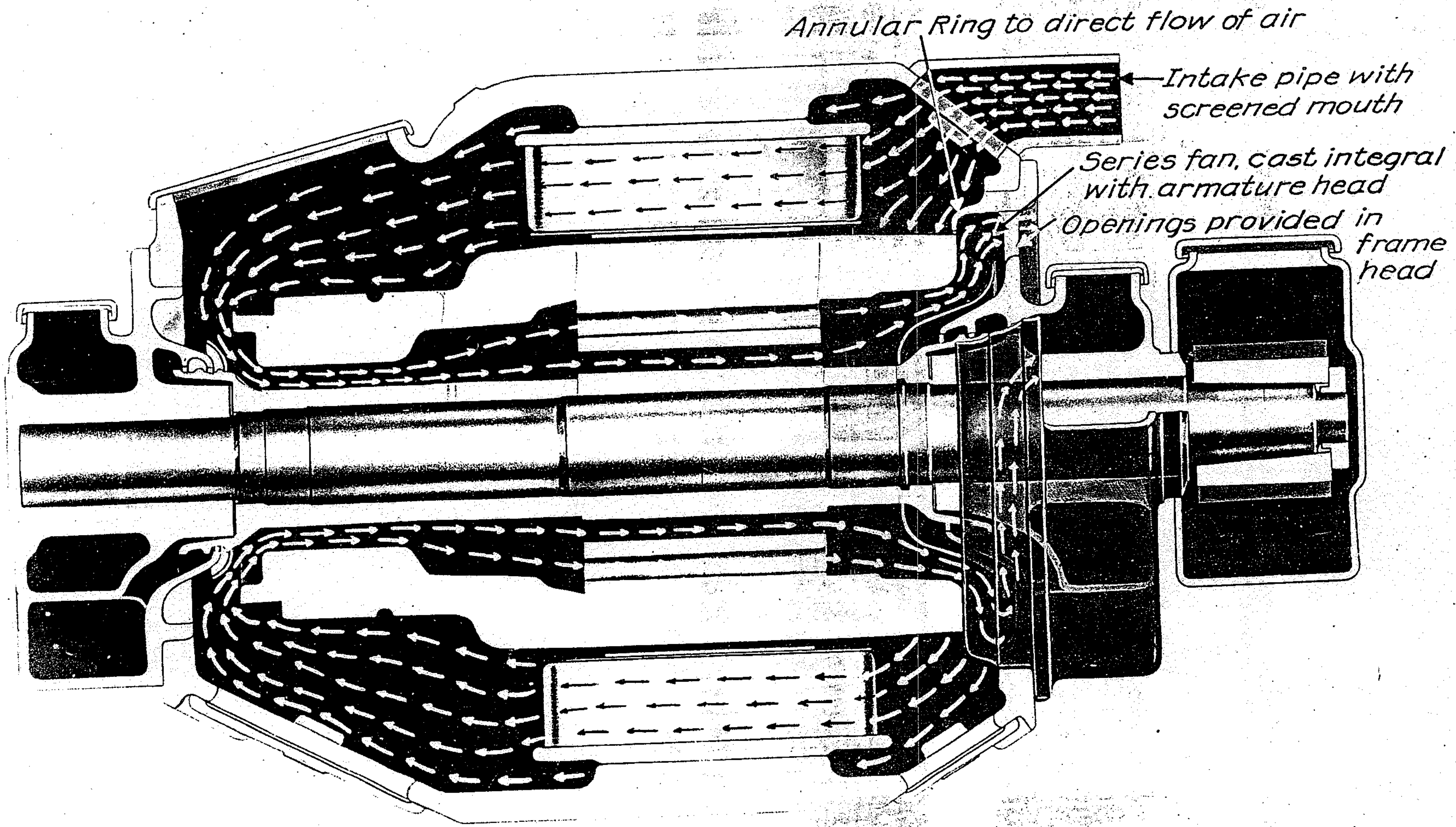
1. Without auxiliary oil well.
2. With auxiliary oil well.
3. Constant oil level.

All of these depend on capillary action for carrying oil from the oil well to the bearing surface.

1. Frame heads and axle caps without auxiliary oil wells have a common chamber to both oil and waste. They are filled by pouring oil directly on the waste. This style is used with old type motors only.

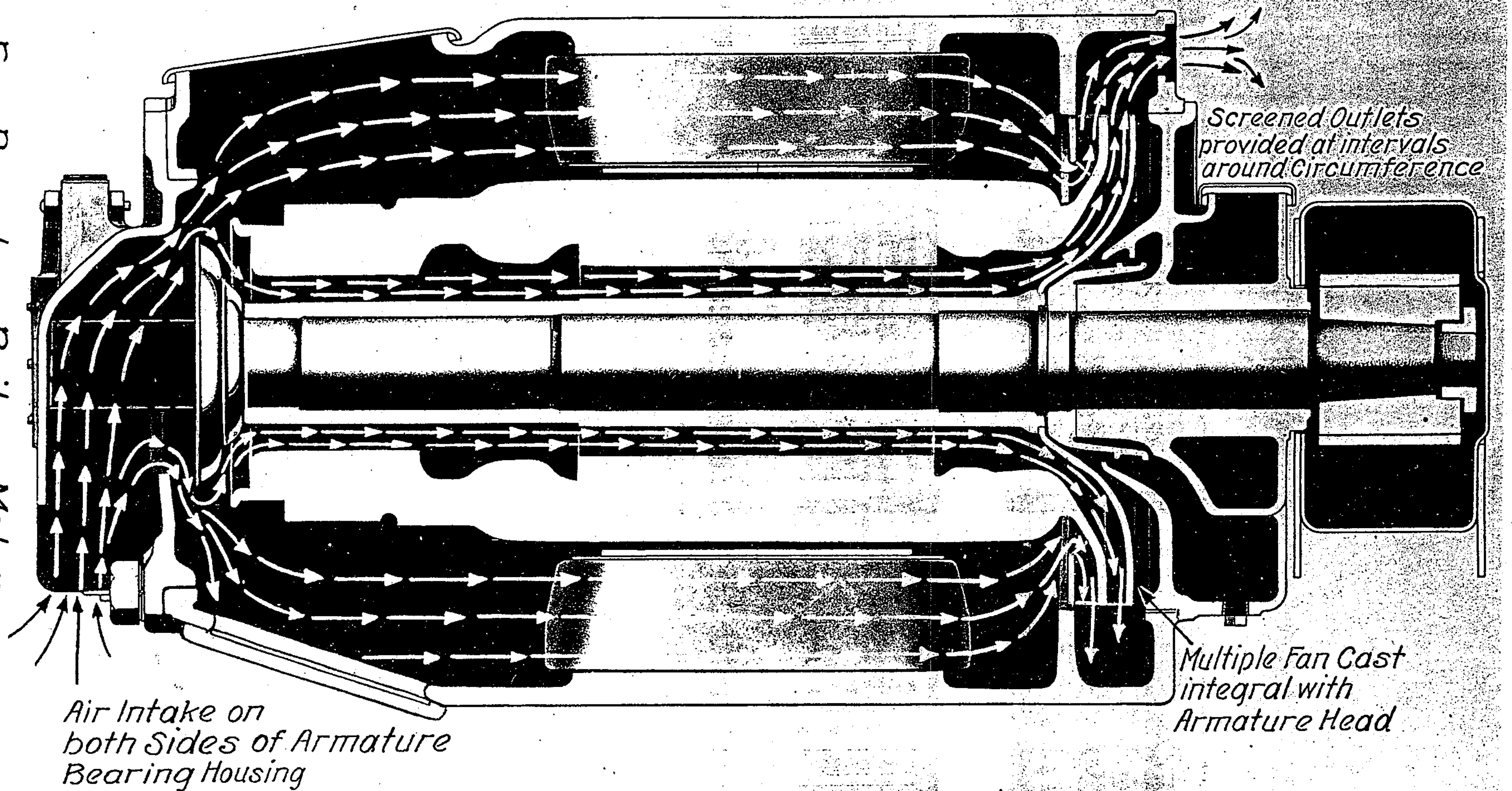


GE Railway Motor Axle Lining with Dowels



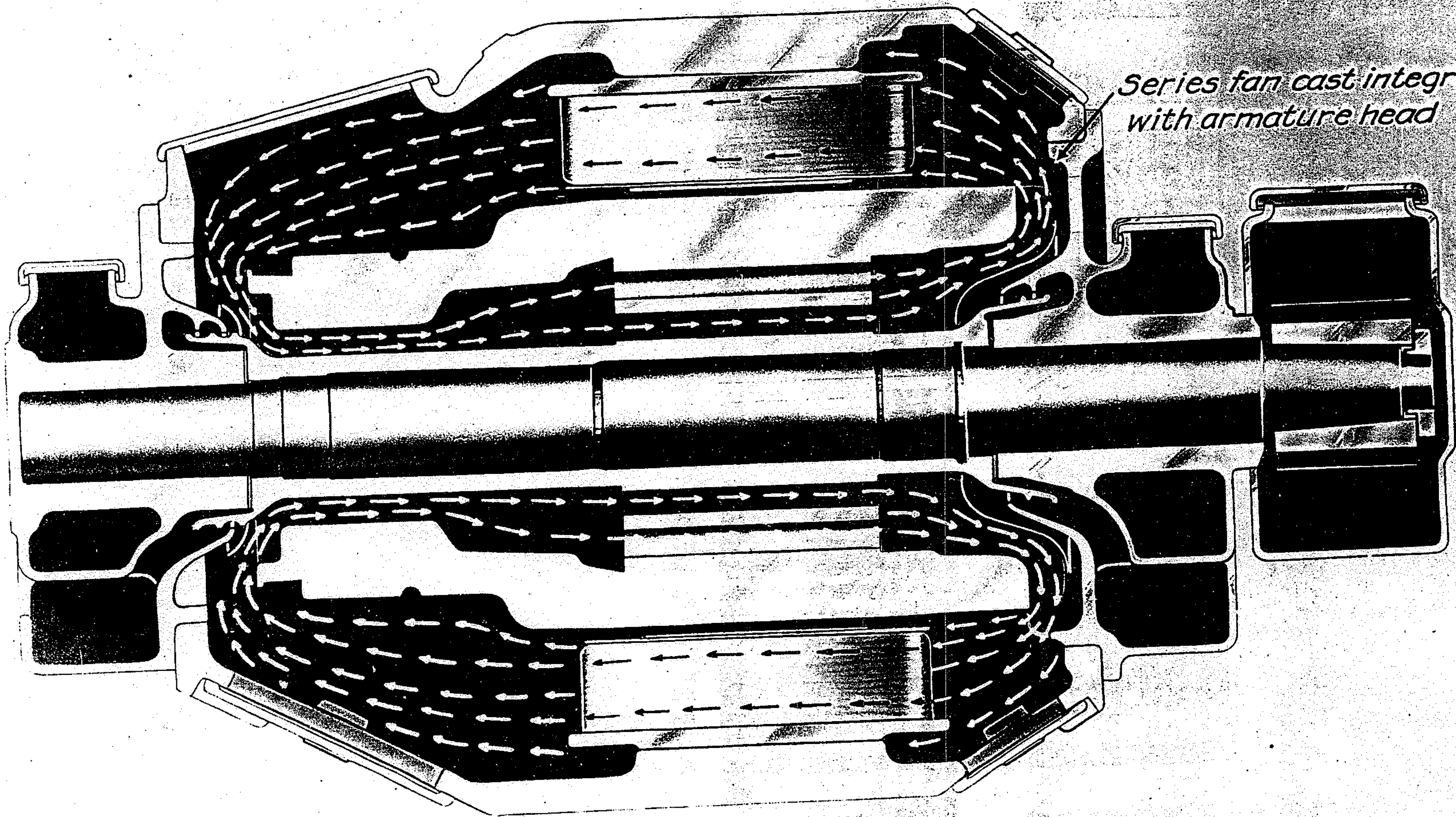
Ventilated. - From 20 to 30 per cent more service capacity is obtained than with closed motors of the same hourly rating. All covers are solid, with the exception of the intake pipe which is fitted over the opening at the pinion end.

GE Railway Motor, Longitudinal Section, Ventilated



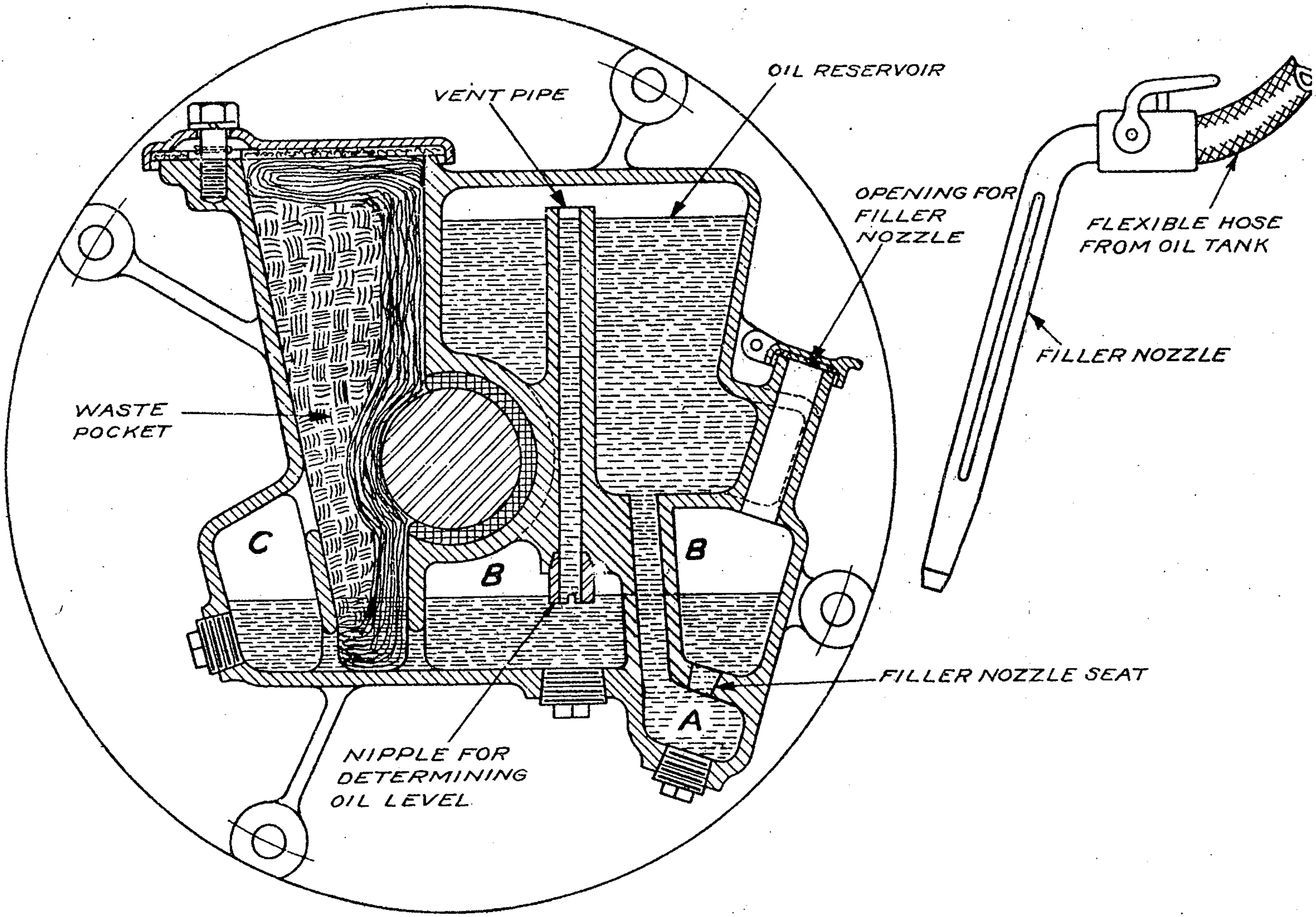
Multiple Ventilation. The Air on entering the Frame divides into two streams one passing over the Armature and around the Field Coils to the Fan, The other through the Commutator Shell and the longitudinal ducts in the Armature Core. After Passing though the Fan. The Streams of Air unite and are exhausted to the Atmosphere though screened outlets in the end of the Frame.

GE Railway Motor. Longitudinal Section. Ventilated with Multiple Fan

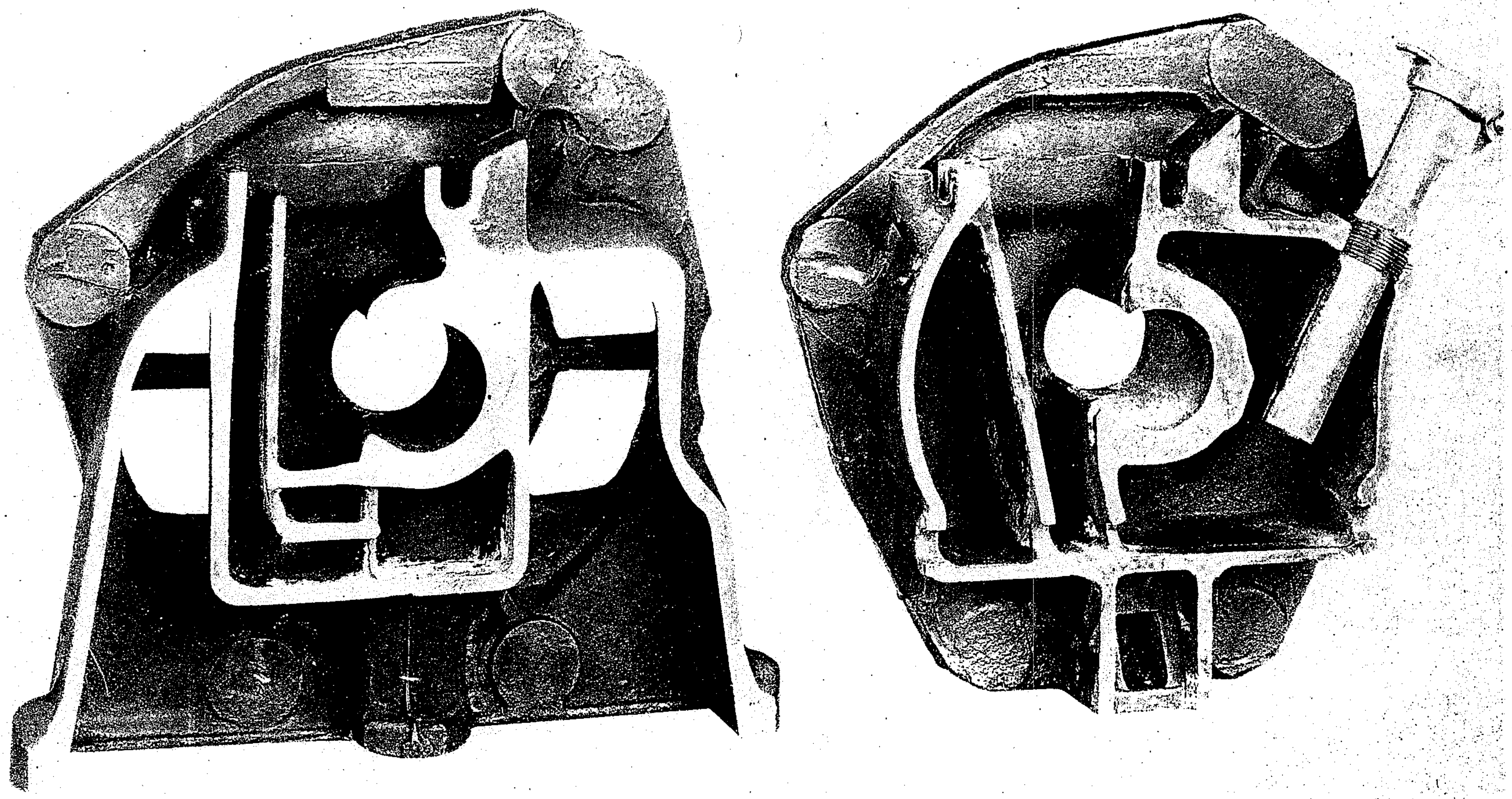


Closed Motor. - Solid covers are used throughout, and there are no openings to the exterior. The internal features of the ventilated motor are retained and a positive circulation of air within the frame is maintained by the fan. Heat is more effectually conveyed from the interior of the armature to the radiating surface of the frame, and an increase of about 5 per cent in service capacity obtained when compared with the older style motors of the same hourly rating.

GE Railway Motor Section Showing Longitudinal Ventilation, Closed

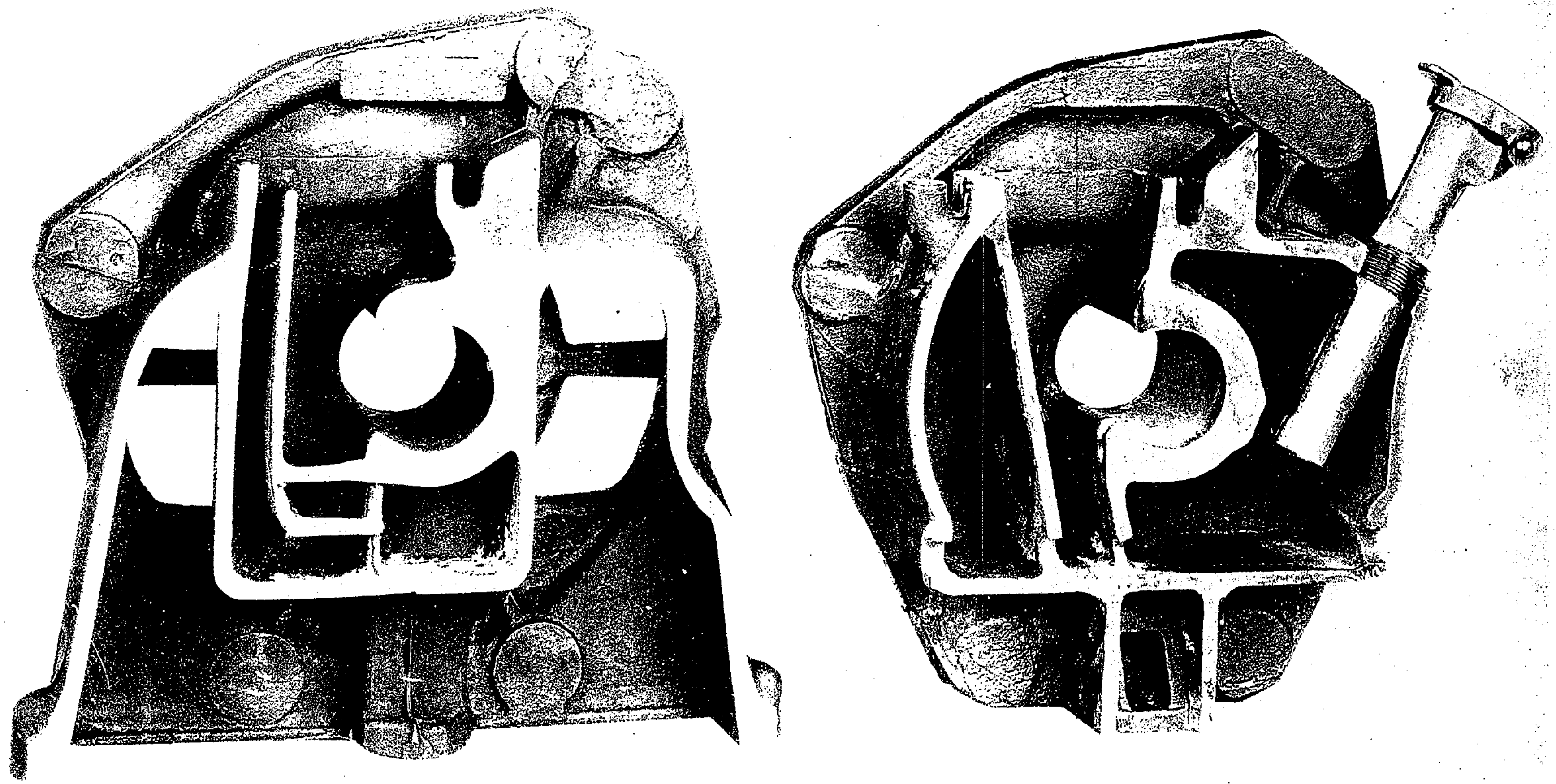


K-761051 G-E. CONSTANT OIL LEVEL BEARING (ARMATURE)



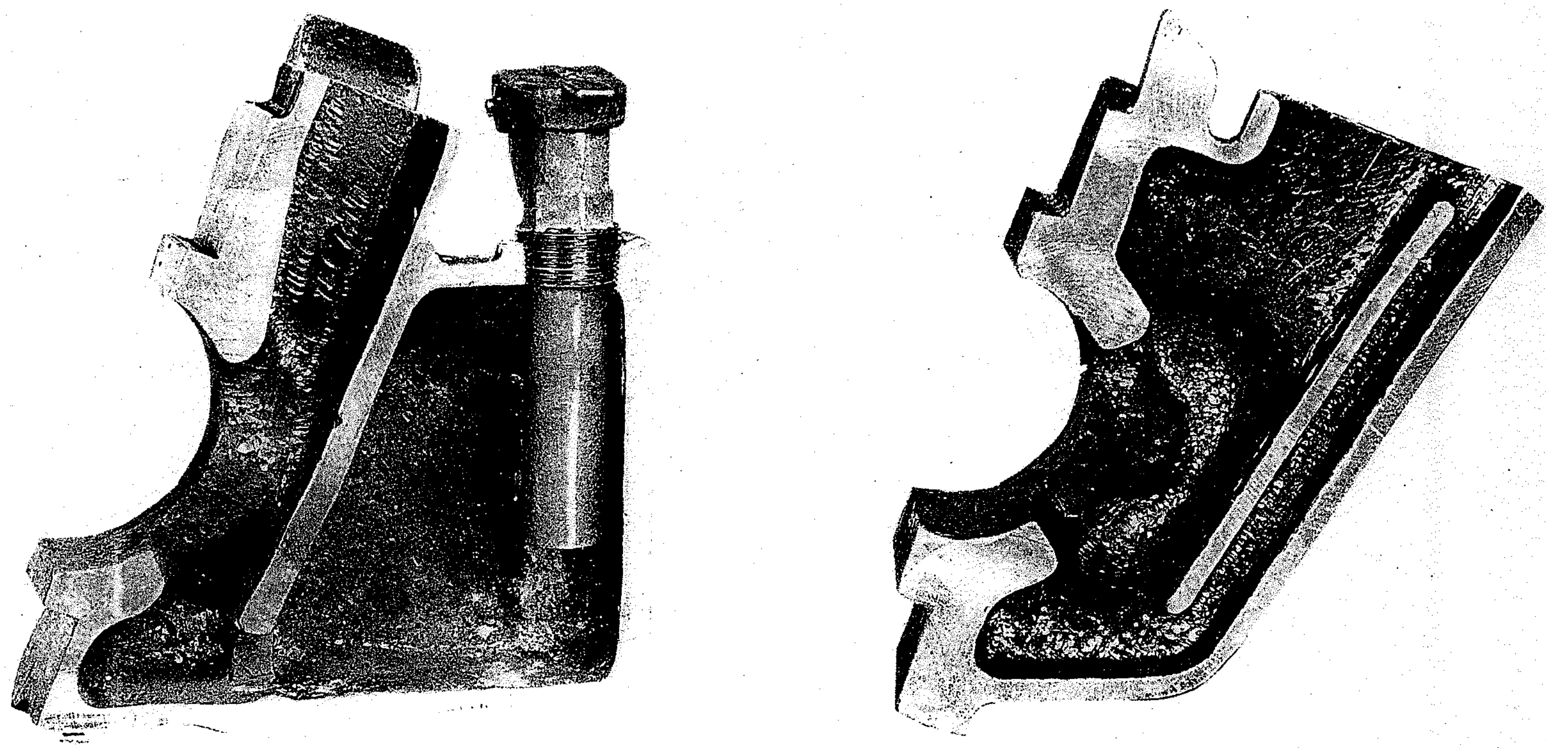
247085 CROSS SECTION OF FRAME HEADS FOR RAILWAY MOTORS. RIGHT, NEW DESIGN, WITH INCREASED OIL CAPACITY AND FILLER PIPE. LEFT, OLD DESIGN.

E-315.1 6-16-27



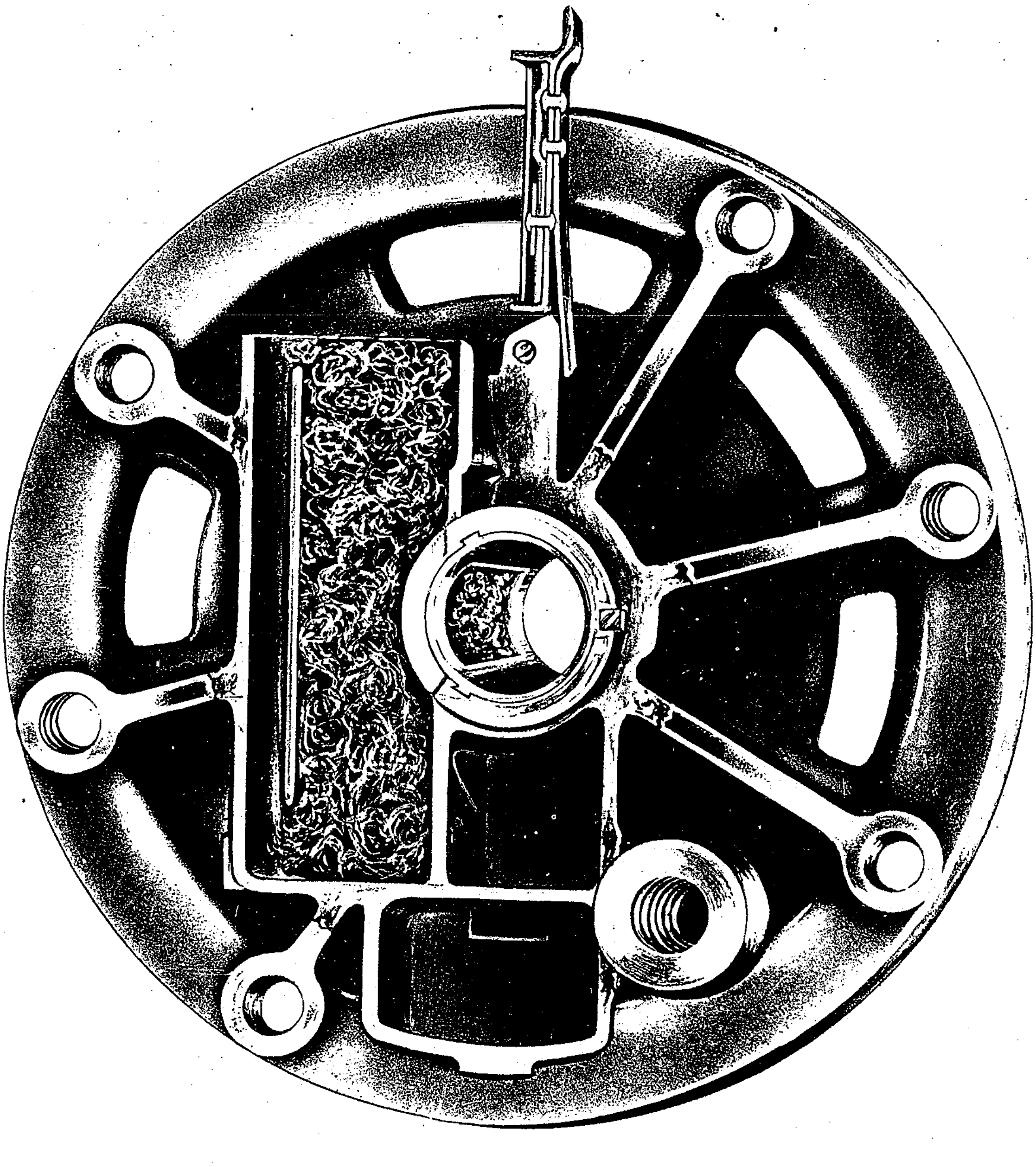
247085 CROSS SECTION OF FRAME HEADS FOR RAILWAY MOTORS. RIGHT, NEW DESIGN, WITH INCREASED OIL CAPACITY AND FILLER PIPE. LEFT, OLD DESIGN.

E-315.1 6-16-27

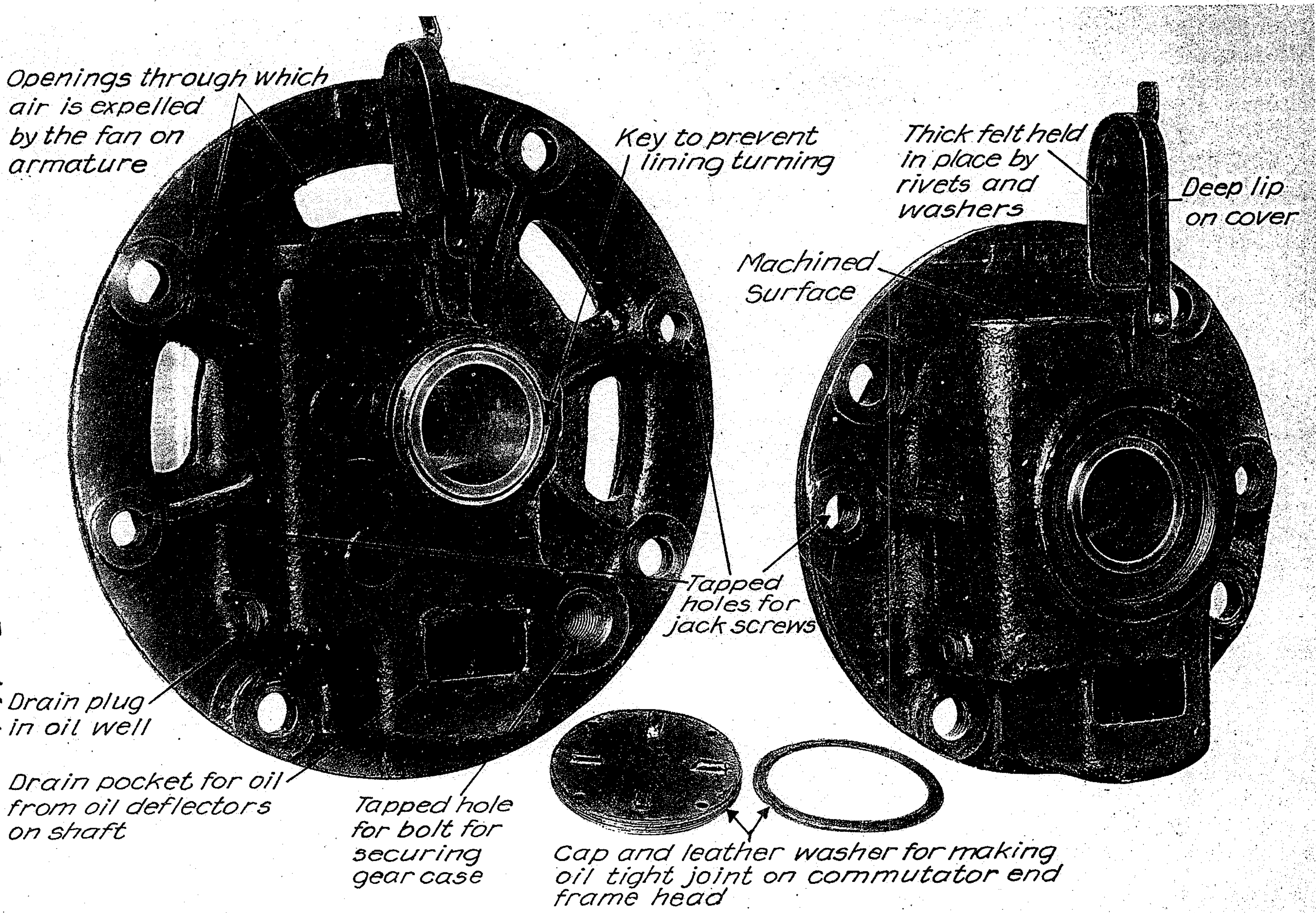


247084 CROSS SECTION OF AXLE CAPS FOR RAILWAY MOTORS. LEFT, NEW DESIGN, WITH INCREASED OIL CAPACITY & FILLER PIPE. RIGHT, OLD DESIGN.

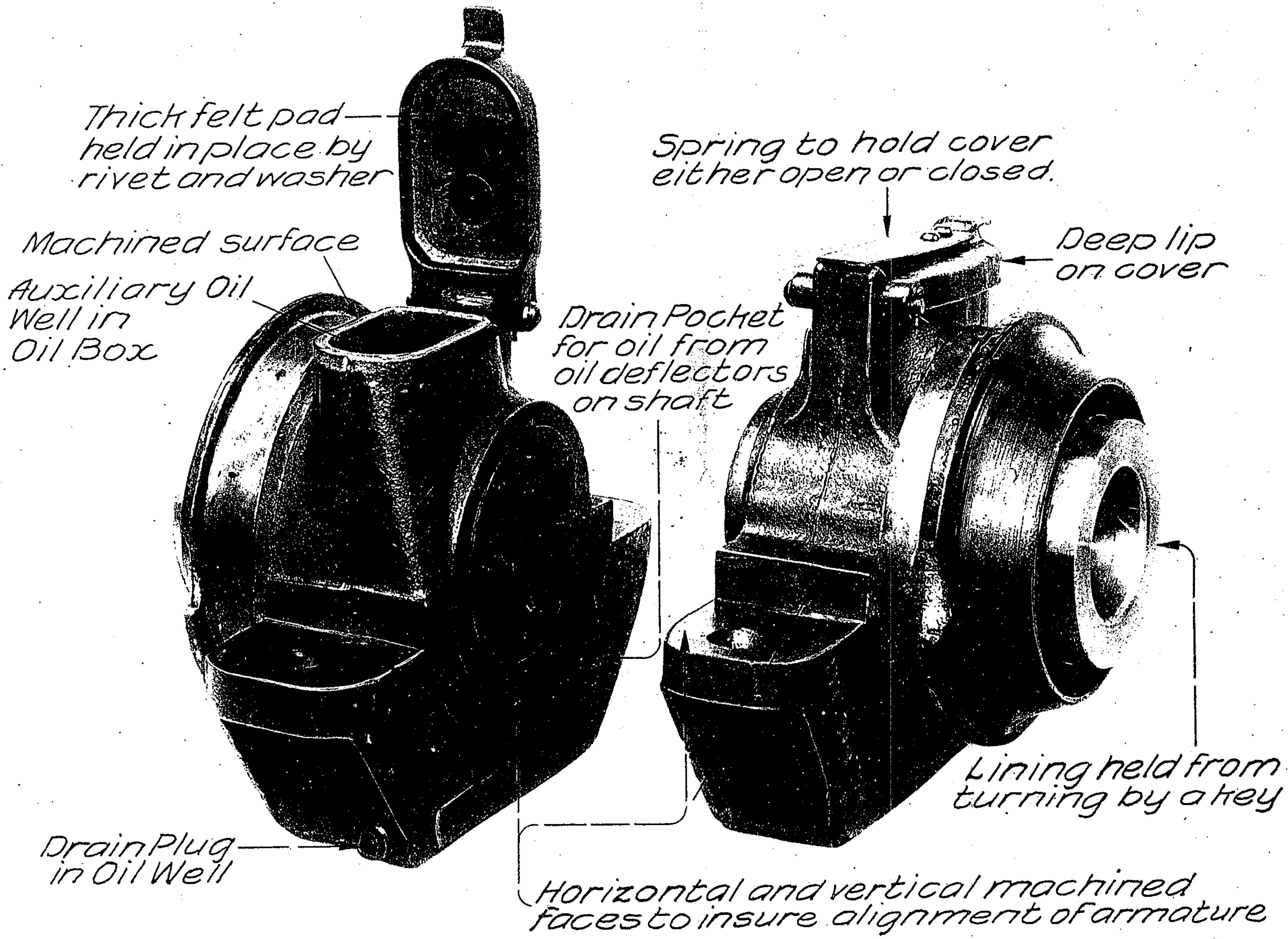
E-315.1 6-16-27



GE Ventilated Railway Motor, View of Frame
Showing Oil Box In Section
Sept. 1, 1914 Eng. Report on Ry. Motors



Frame Heads For Box Frame GE Railway Motor Ventilated



Frame Heads For G.E. Split Frame Motor.

2. Frame heads and axle caps with auxiliary oil wells are of different styles but all have separate oil and waste chambers. New oil is inserted in a separate channel leading directly to the oil well instead of being poured on top of the waste. In the original style both oil and waste chambers were closed by a common cover. This allowed the dirt accumulated on the cover to drop on the waste when replenishing the oil. The next step was to provide separate covers for each of the oil and waste chambers so that in adding oil the waste cover is not necessarily disturbed. In a later type the size of oil chamber was increased allowing a somewhat longer period between oilings. The difference between these two types is shown in the cross sections on page 37 (photo. 247085).

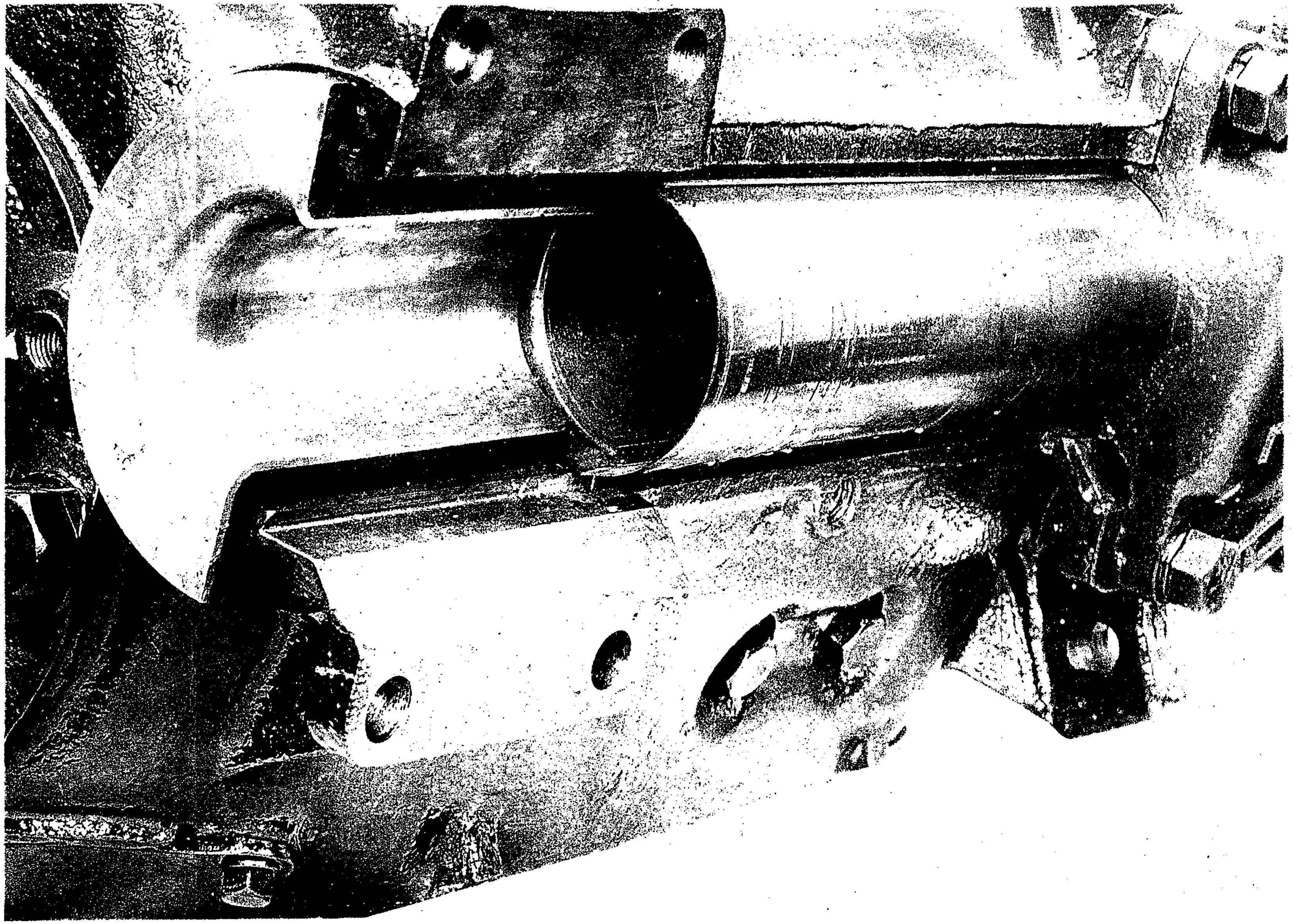
3. The "Constant Oil Level Bearing," as shown on drawings K-761051 and K-761058, is designed to provide a constant oil level in the waste chamber.

To add oil to the bearing, the filler nozzle is seated in the opening above "A" which is connected to the oil reservoir by an unrestricted channel.

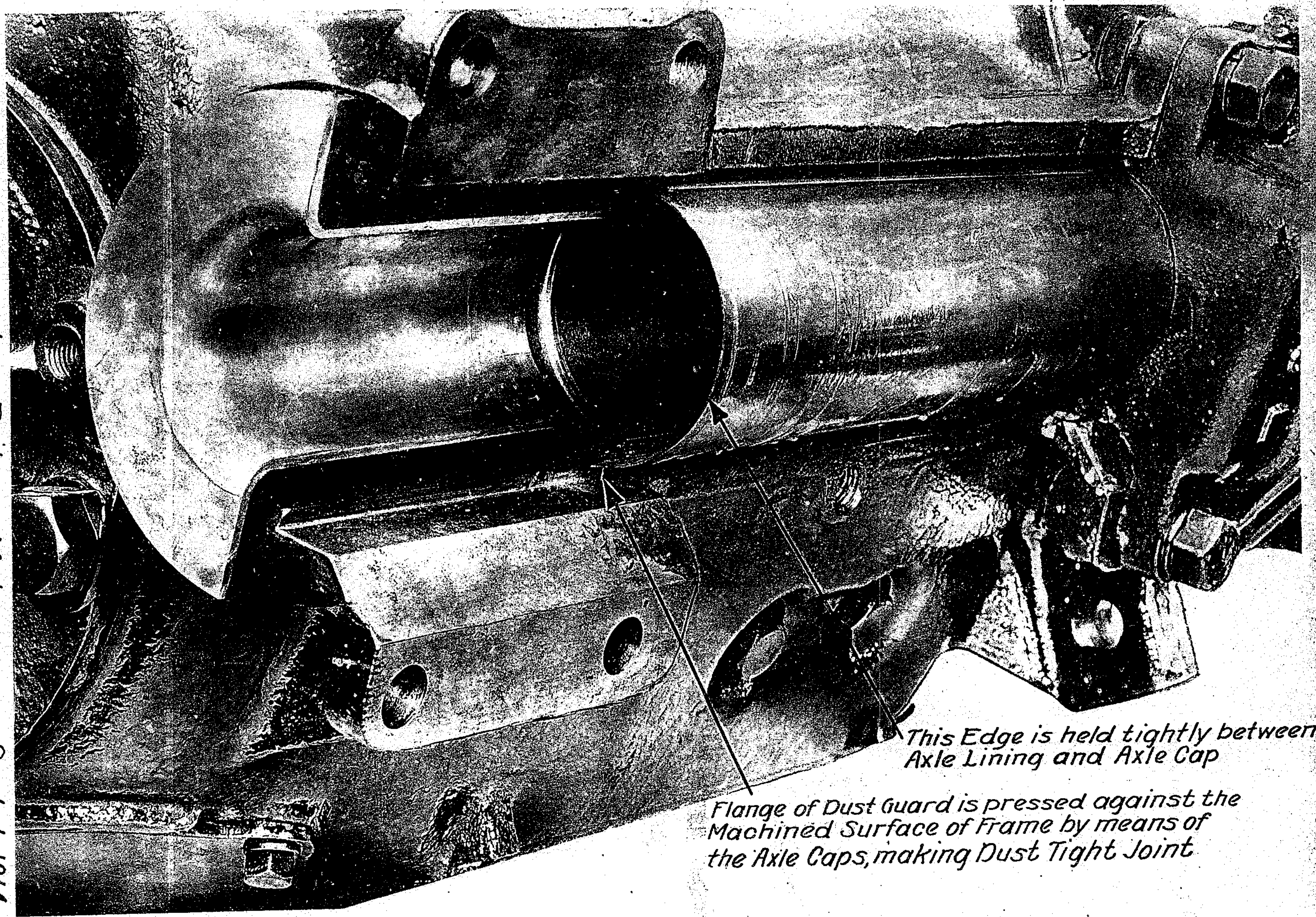
As oil is forced into the reservoir, air escapes to the atmosphere by way of the vent pipe, chamber "B," and the groove in the filler nozzle. When the level in the reservoir rises to the top of the vent pipe, oil passes into chamber "B" until it overflows through the groove in the filler nozzle. This indicates that sufficient oil has been added. After the filling operation the oil level in "B" is higher than in the expansion chamber "C" (in frame heads only). These levels are equalized by the slow passage of oil through the waste at the bottom of the waste pocket. As oil is used the level in "B" falls below the nipple at the bottom of the vent pipe, admitting air to the reservoir and releasing oil which flows through "A" to "B" raising the level to the proper point. The height of the oil level which it is desired to maintain can be fixed by the length of the nipple used.

Reference to the following photographs will aid in understanding the operation of these bearings.

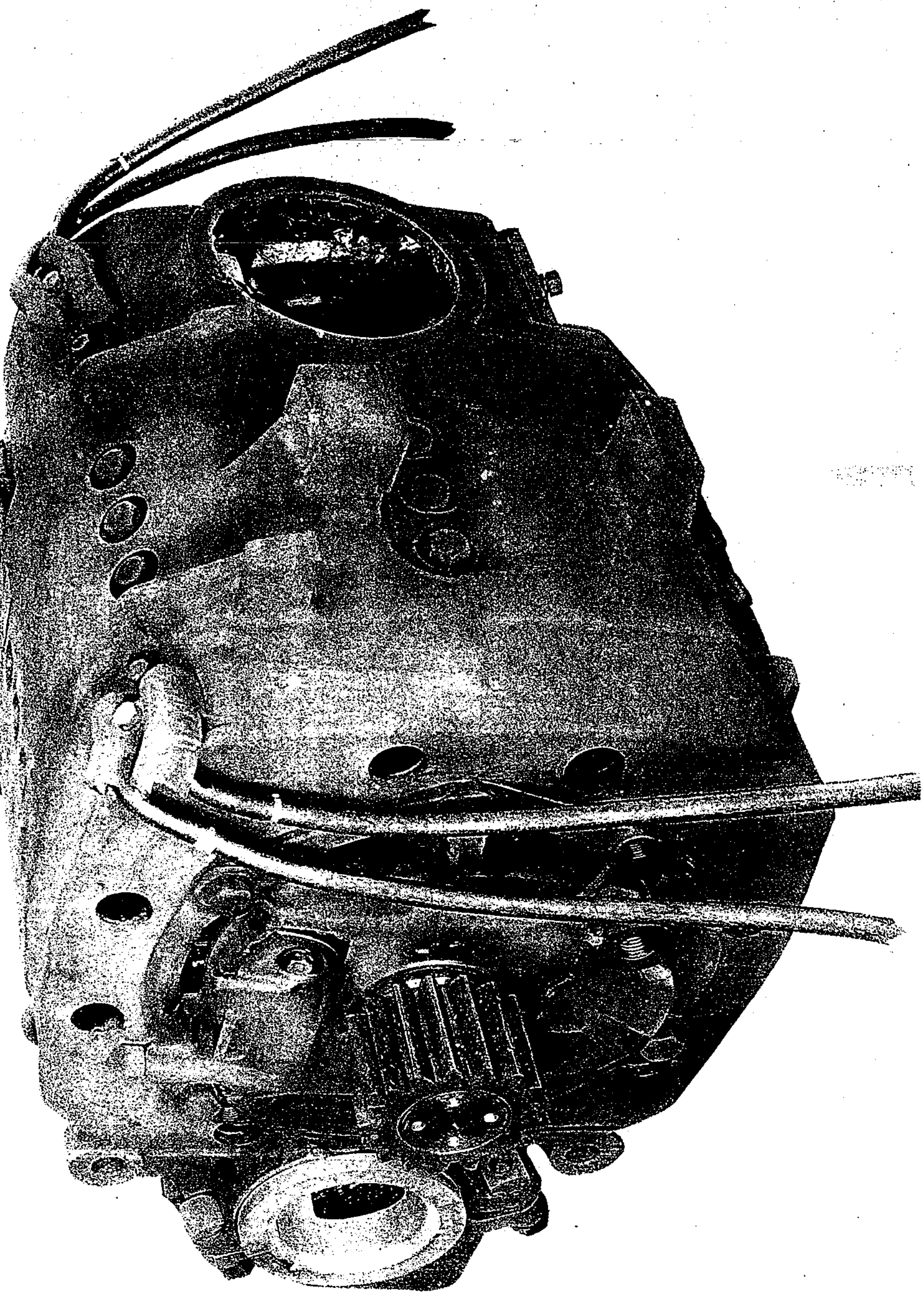
248672	Page 40
248673	Page 41
248674	Page 42
248675	Page 43
248676	Page 44
248677	Page 45



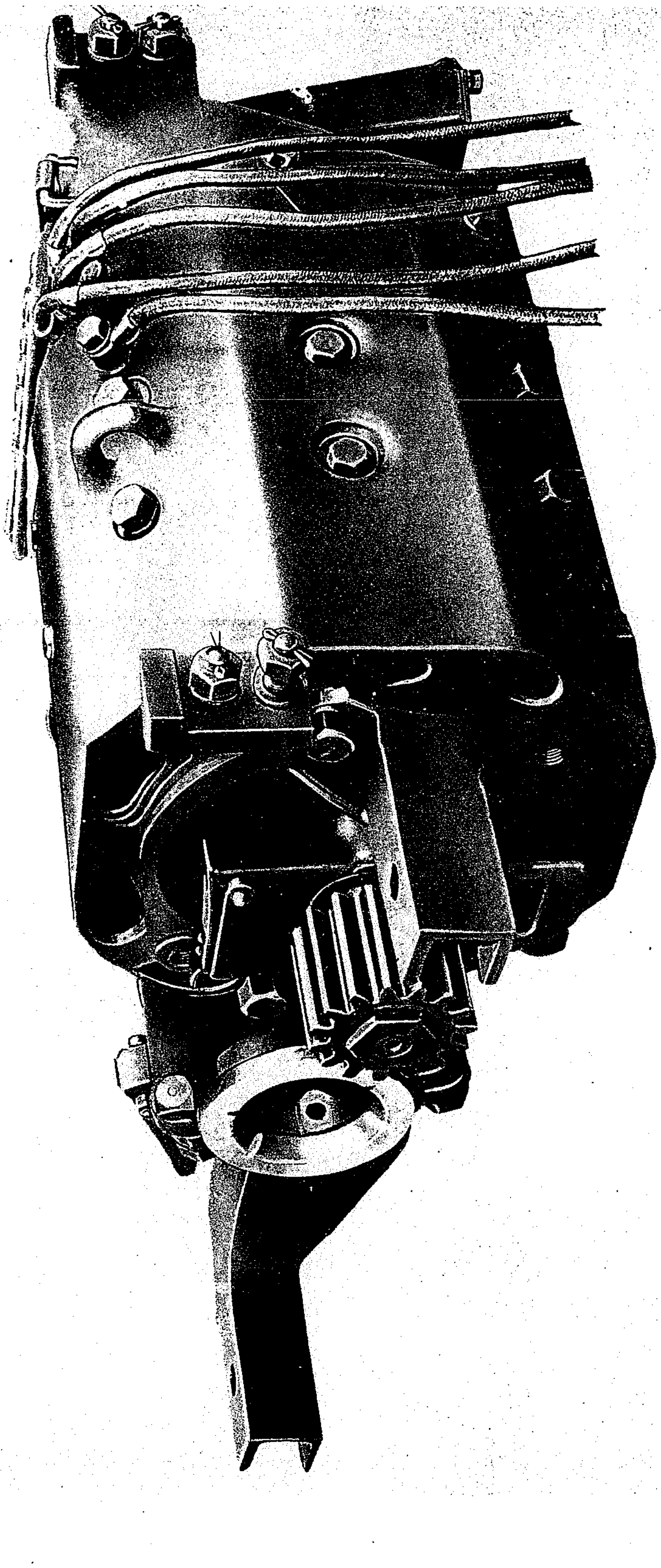
232417 GE RAILWAY MOTOR. AXLE DUST GUARD.



GE-Railway Motor Axle Dust Guard

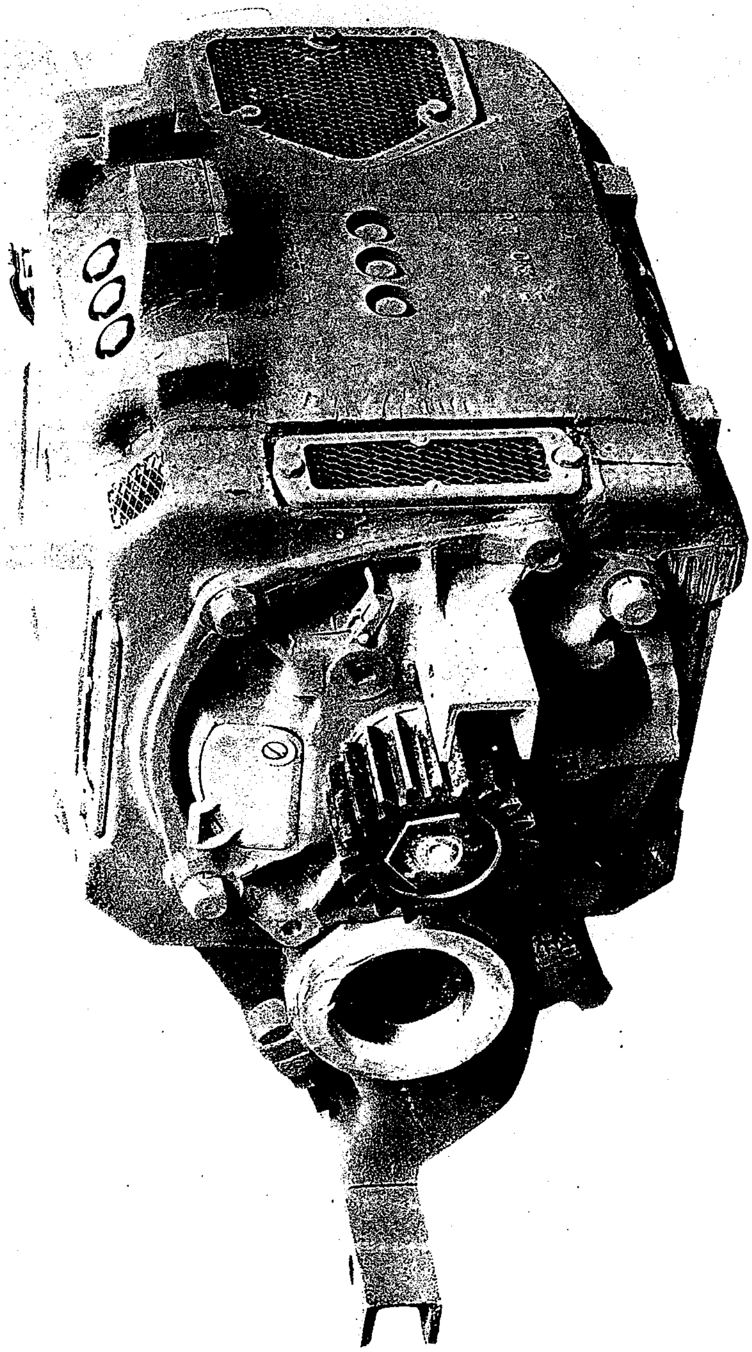


244764 G-E. SPRING NOSE SUSPENSION RAILWAY MOTOR

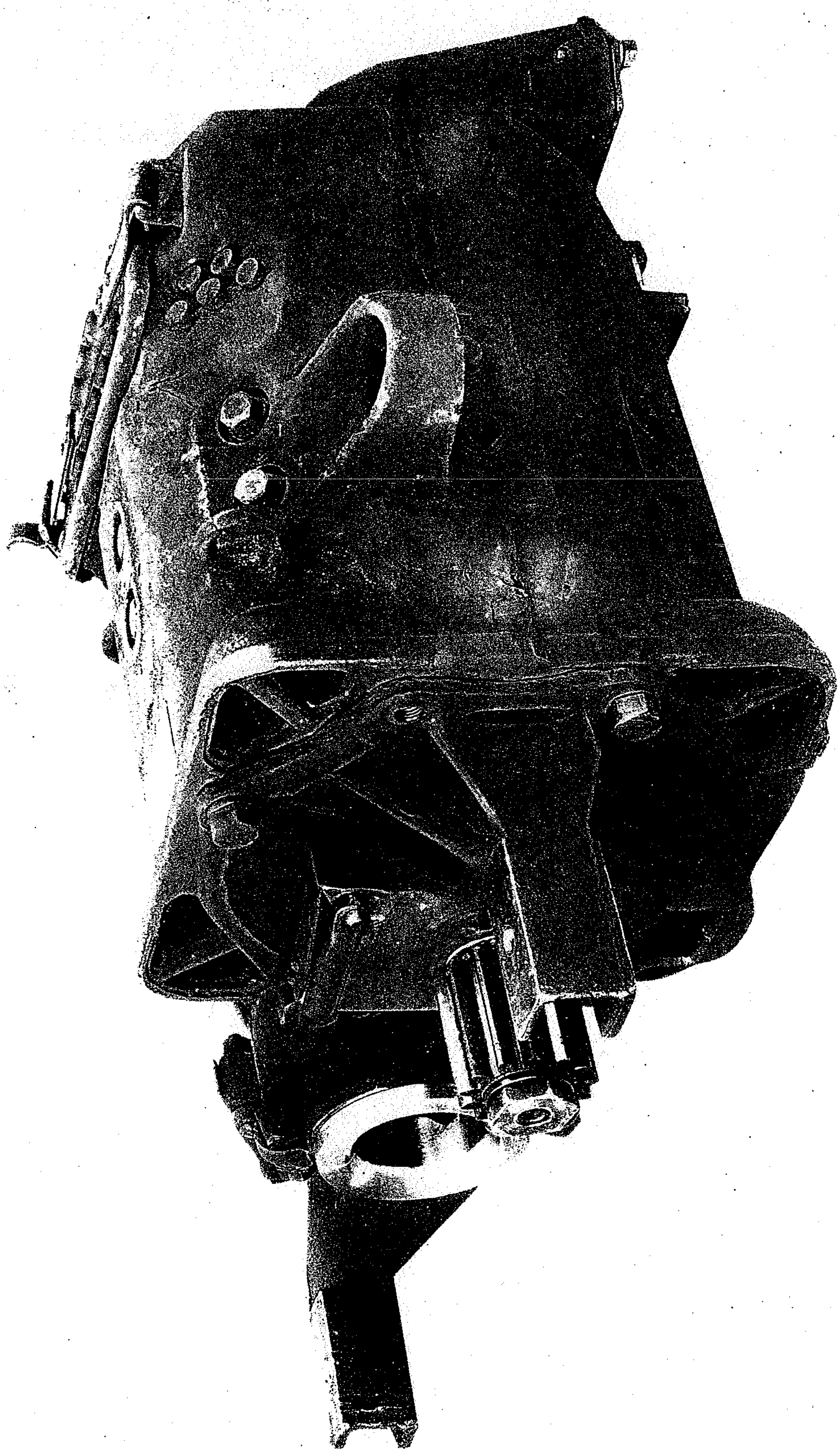


G-E BAR SUSPENSION RAILWAY MOTOR

440915 INDEX E-315.1 8 13 26



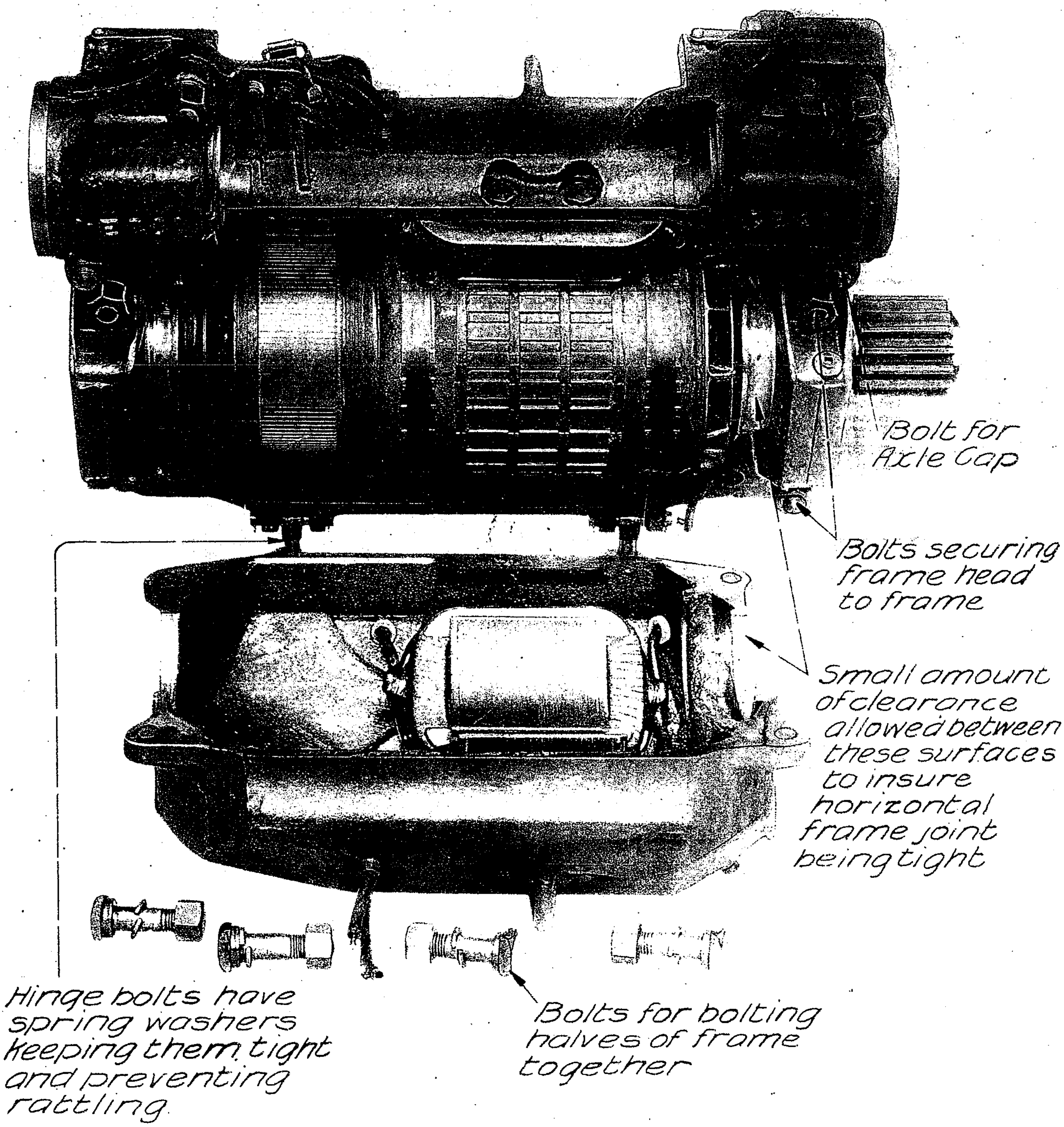
248576 G-E. LARGE NOSE SUSPENSION RAILWAY MOTOR E315.1 5-31-28



G-E. SMALL NOSE SUSPENSION RAILWAY MOTOR

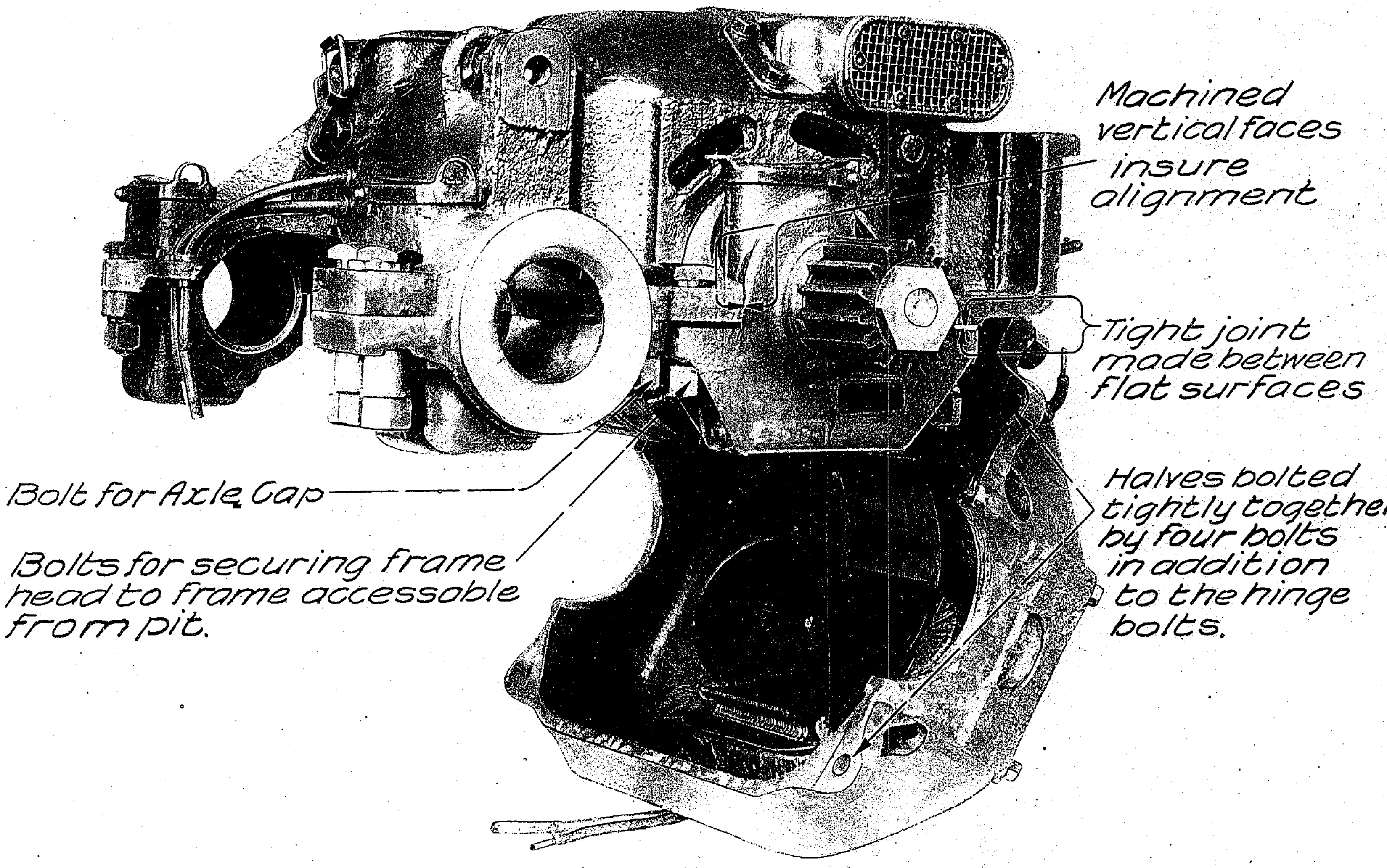
INDEX E-315.1

245605

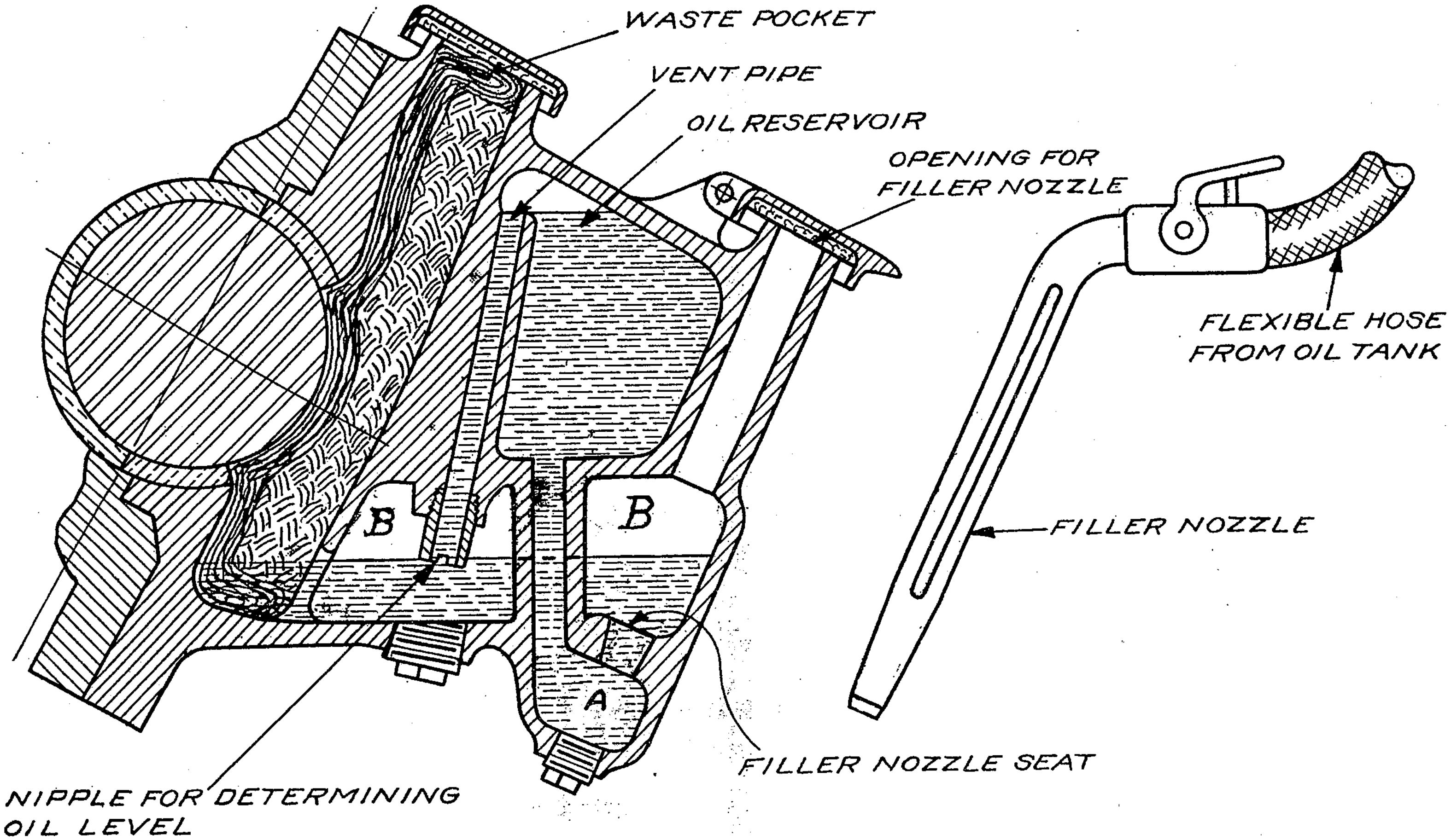


G. E. Type of Split Frame Motor.

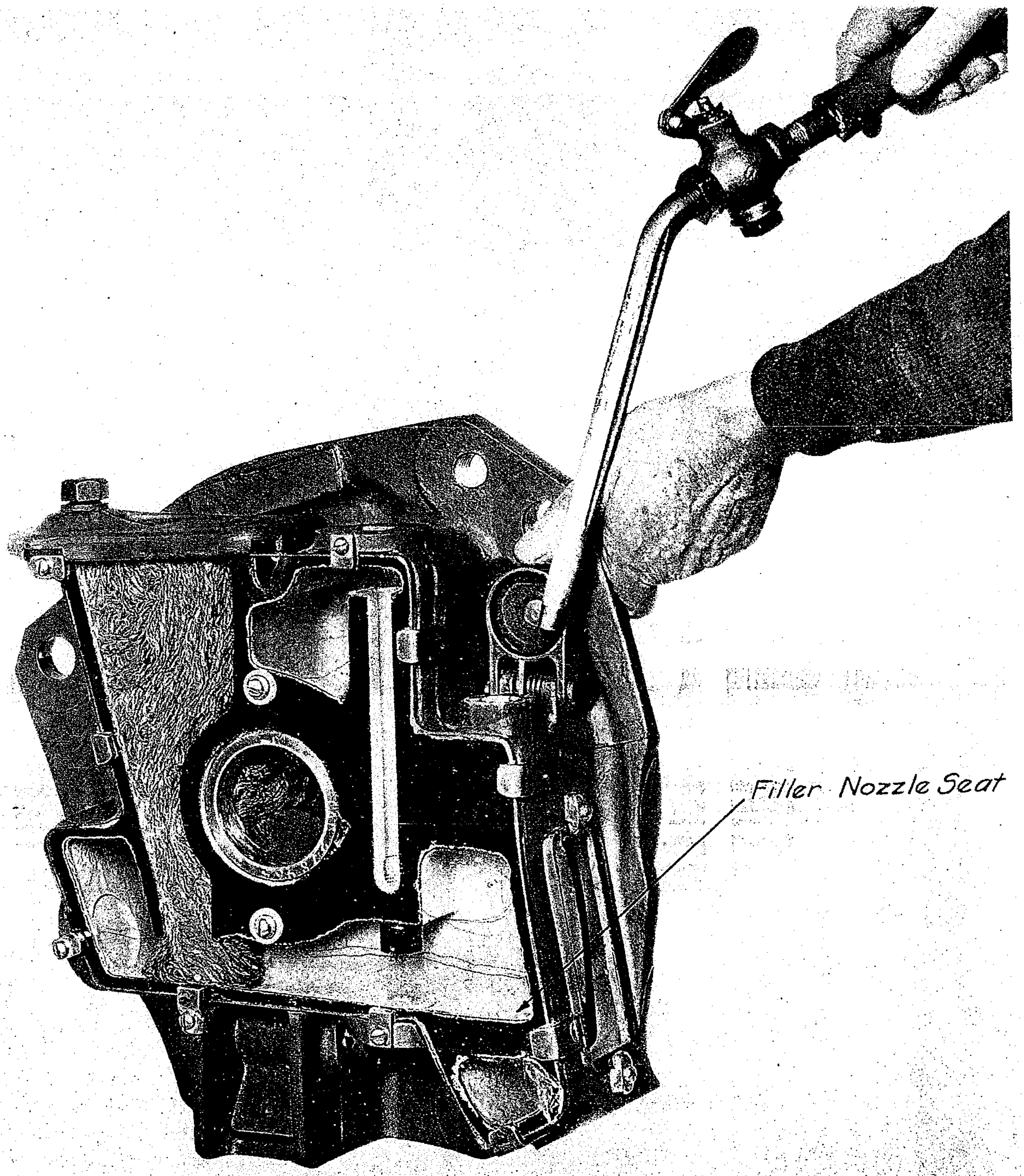
Eng. Report on Railway Motors. June 1915



G.E. Type of Split Frame Motor.

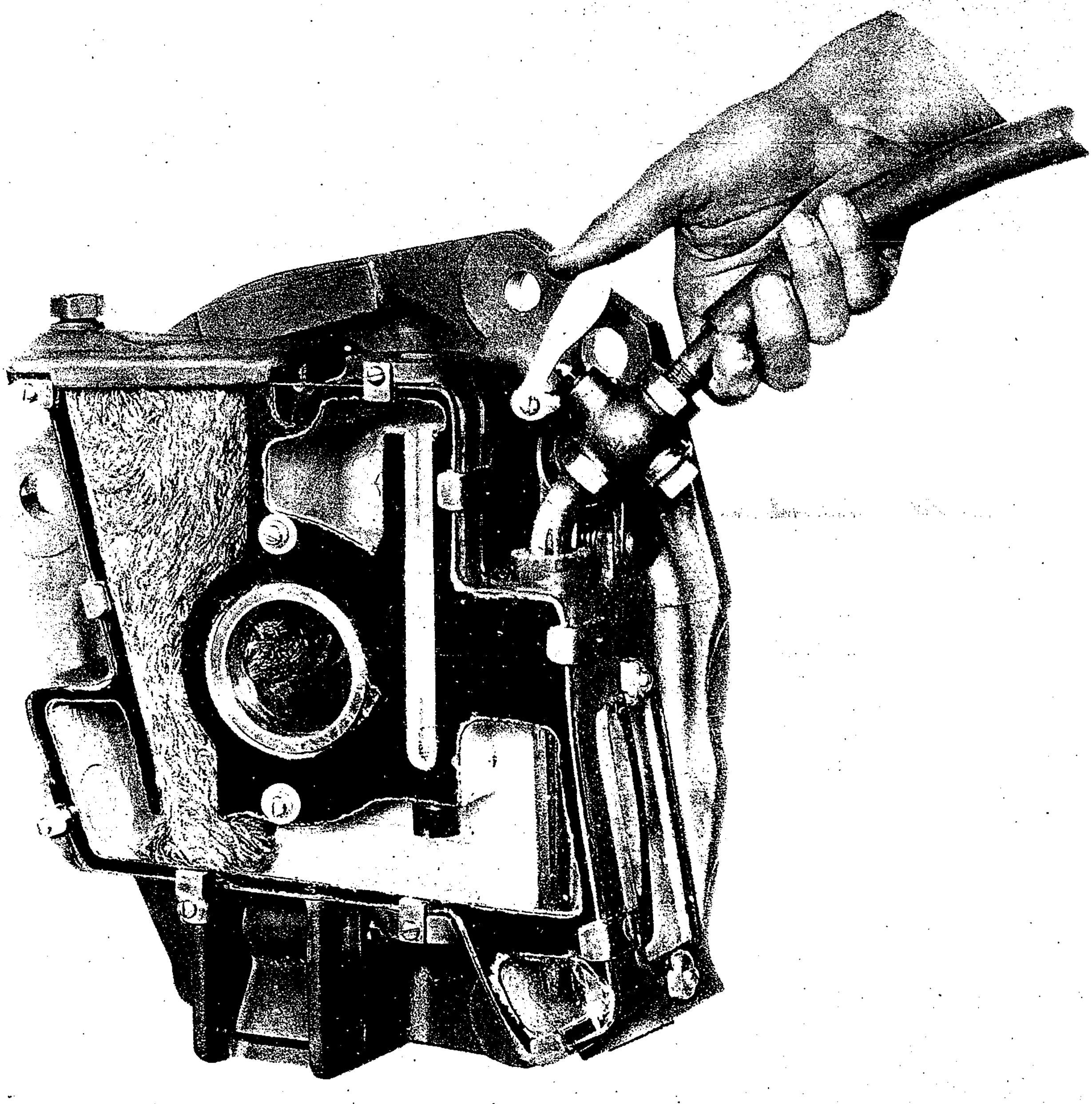


K-761058 G-E. CONSTANT OIL LEVEL BEARING (AXLE)



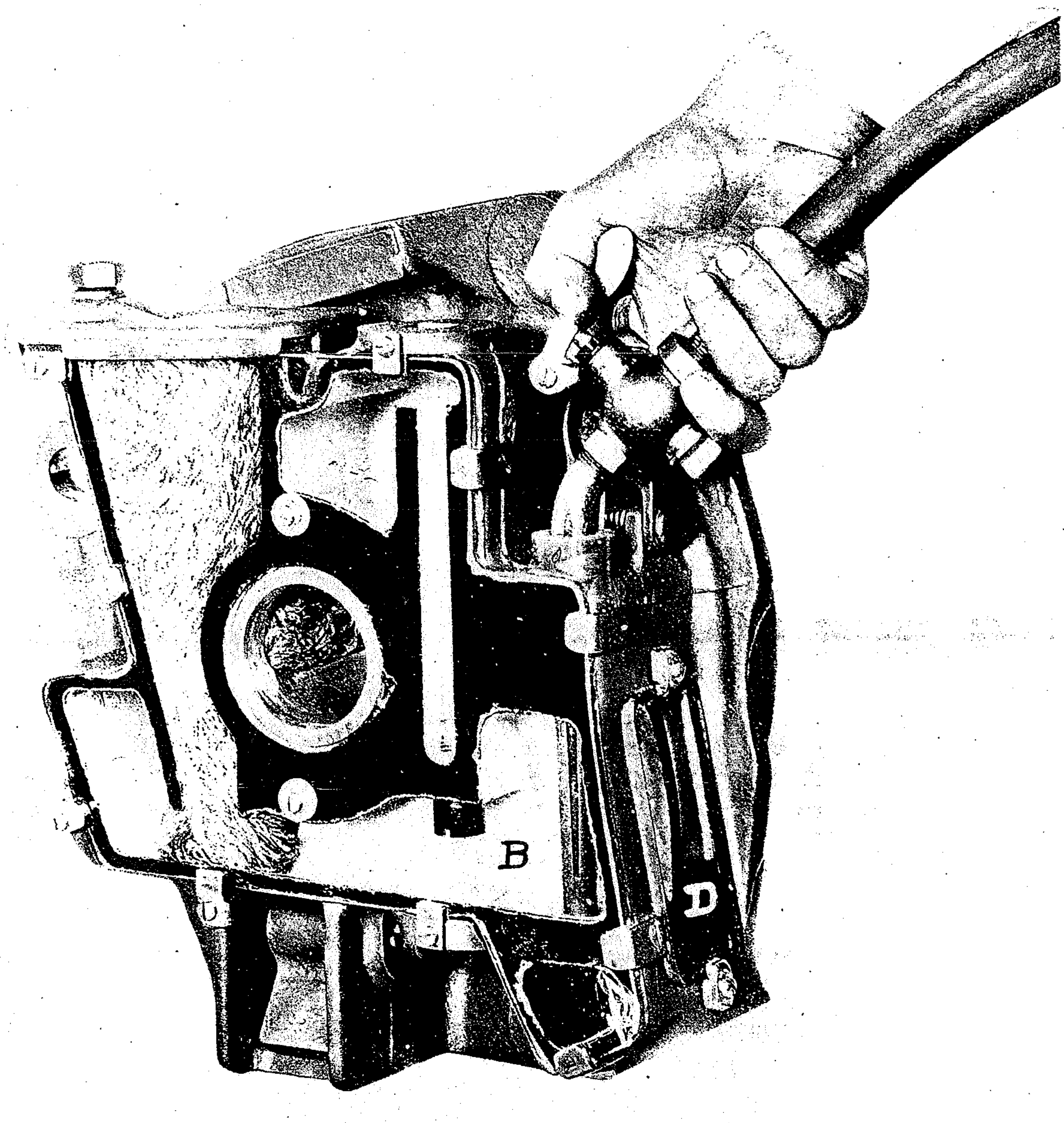
#I
Constant Oil Level Bearing
Ready to Insert Filler Nozzle

248672 GE RAILWAY MOTOR
BEARING. CONSTANT OIL LEVEL. FIG.1
E315.1 6-30-28



248673 GE RAILWAY MOTOR
BEARING. CONSTANT OIL LEVEL. FIG.2

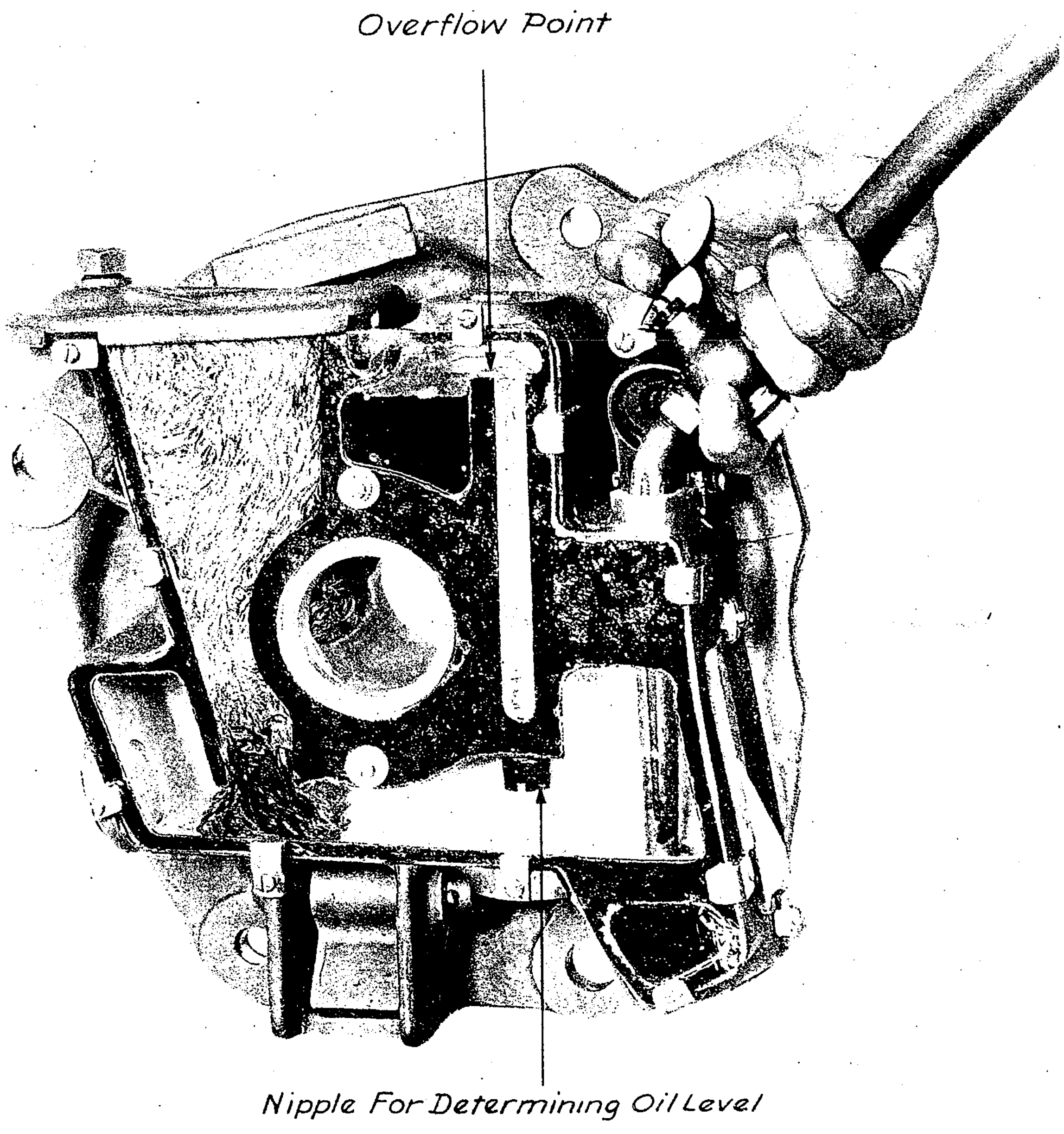
E315.1 6-30-28



#3
Constant Oil Level Bearing
Filler Nozzle Valve Is Now Open And Oil Is Rising In Reservoir As Seen At "D". The Displaced Air Passes Out Thru The Vent Pipe To Chamber "B" Where It Escapes To Atmosphere Around The Filler Nozzle

248674 GE RAILWAY MOTOR
BEARING. CONSTANT OIL LEVEL. FIG.3

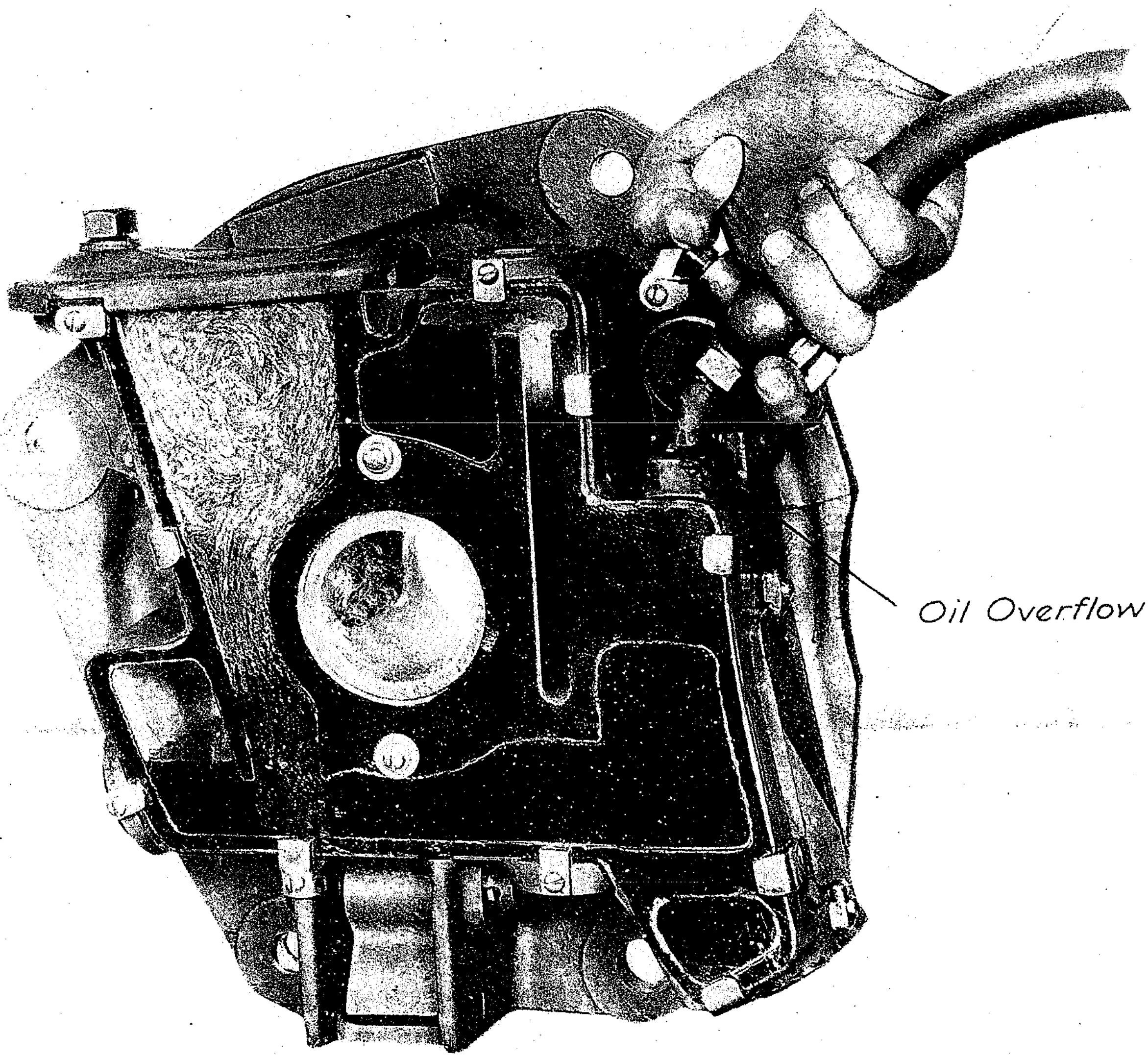
E315.1 6-30-28 -



\# 4
Constant Oil Level Bearing
Oil Reservoir Filled To Overflow Point. Up To This Time No Oil Has Entered Waste Compartment.

248675 GE RAILWAY MOTOR
BEARING. CONSTANT OIL LEVEL. FIG.4

E315.1 6-30-28



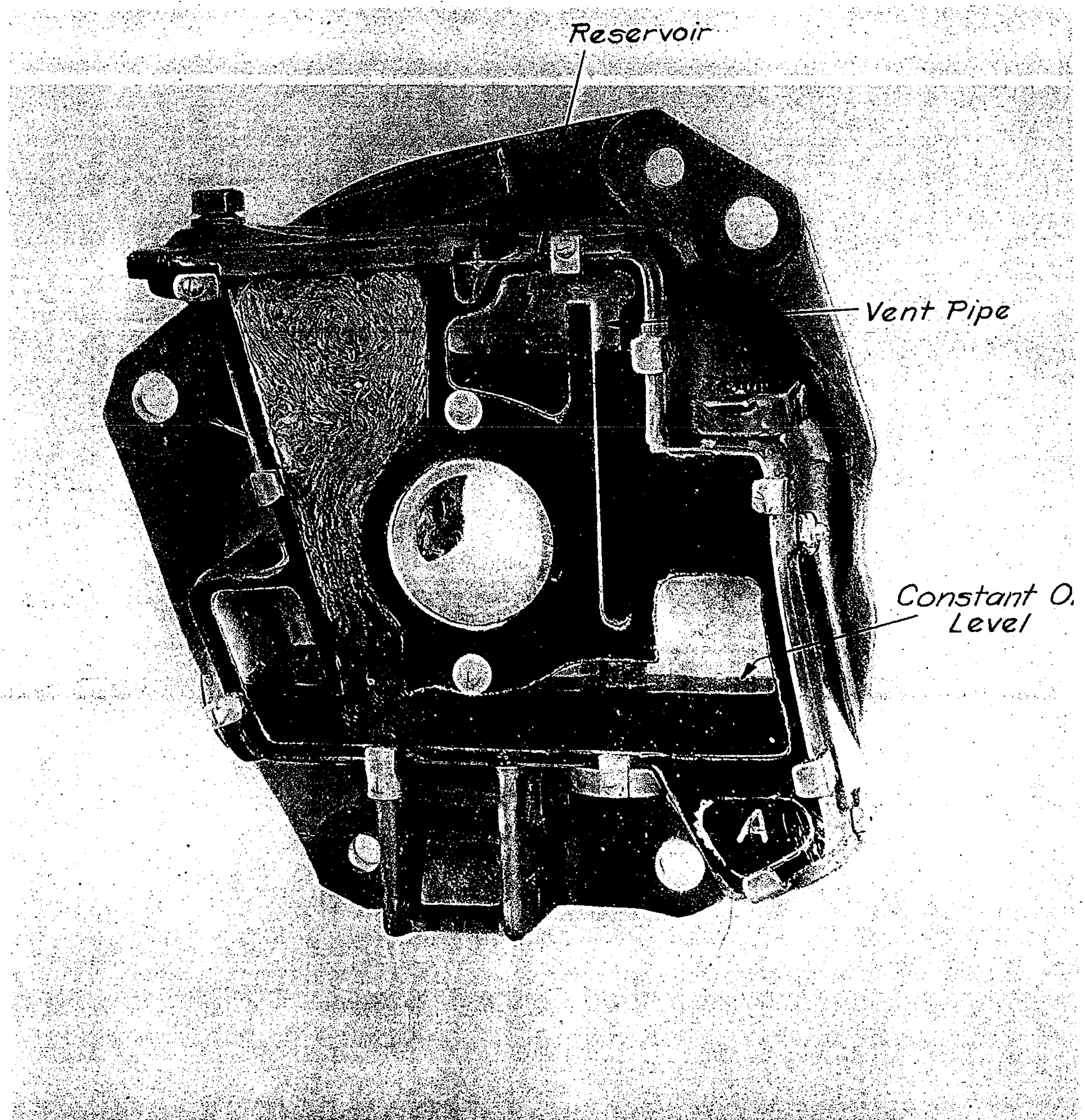
#5

Constant Oil Level Bearing

Bearing Completely Filled As Indicated By Oil Coming Out Around Filler Nozzle.

248676 GE RAILWAY MOTOR
BEARING. CONSTANT OIL LEVEL. FIG.5

E315.1 6-30-28



#6
Constant Oil Level Bearing
Operating Height Of Oil Level. Oil Is Held In The Reservoir By Vacuum. When The Oil Level Has Been Reduced Below The Predetermined Point Air Is Admitted Through Vent Pipe Allowing Oil To Flow By Way Of "A" And The Filler Nozzle Seat Opening To Restore The Level.

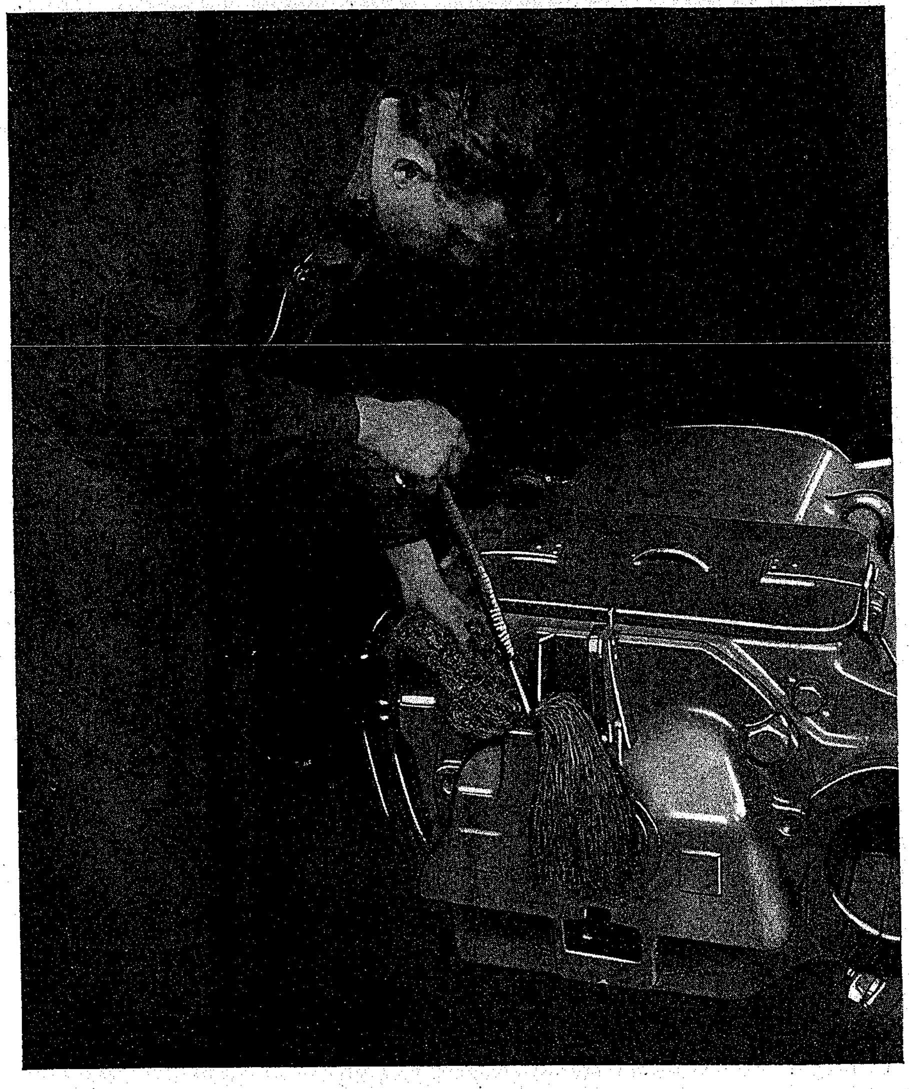
248677 GE RAILWAY MOTOR
BEARING CONSTANT OIL LEVEL. FIG.6
E315.1 6-30-28

SAMPLE COPY

GEA-60A

The Care of

RAILWAY MOTOR BEARINGS



Packing a Railway Motor Bearing

Railway motor bearings are all of the sleeve type with the exception of a few small motors which have armature bearings of the ball or roller type. Lubrication is provided by means of waste and oil; *proper lubrication and care of bearings is something which cannot be over emphasized.* In addition to serving their usual function, axle bearings must also support a considerable part of the weight of the motor.

GENERAL ELECTRIC COMPANY
SCHENECTADY, N. Y.
Sales Offices in Principal Cities

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METHOD OF PACKING BEARINGS

Auxiliary oil wells are provided in most modern railway motors for replenishing the supply of oil and gauging its depth. Use these wells when renewing the oil supply since they conduct the oil to the bottom from whence it is fed up through the waste, thus avoiding the introduction of dirt to the bearing surface. If no auxiliary oil wells are provided, press the waste toward the opening in the lining and pour the oil down the wall of the pocket. Oil poured on top of the waste will wash some of the dirt in the waste down to the bearing.

The lubrication of the bearings depends on capillary attraction for feeding the oil from the reservoir to the journal surface of the shaft. Experience has shown that cotton waste is not suitable for packing material owing to the difficulty of holding it against the shaft and the fact that it glazes quickly. The springy nature of long-fiber, all wool yarn, even after it is saturated with hot oil, makes it the most desirable bearing packing now obtainable. The individual pieces of yarn should be at least 12 inches long.

To insure initial lubrication of the bearings after packing, the packing should previously be thoroughly saturated with the oil. Unless special provision is made for saturating the packing, it should be soaked in the oil for at least 48 hours and drained for about 24 hours, at about 60 deg. F.

Since oil will climb up straight strands of the packing yarn faster than it will pass through a number of balls or wads of the packing, it is desirable that the packing be in the form of a wick in the housing. The larger the wick the easier it will be for the necessary oil to reach the shaft.

The pieces of packing required for the wicks, for the armature bearings, should be made into skein form of sufficient length to reach from the bottom of the waste chamber up to about 6 inches above the seat for the waste chamber cover. The skeins should then be twisted about one complete turn in order to hold all of the strands of yarn in place and produce a more springy wick. As many skeins as possible should be packed in the waste chamber, starting at the inner end of the bearing and working outward, and allowing the upper ends of the skeins to hang out over the seat of the cover about 6 inches or enough to hold them in place.

After the wick is formed in the chamber it should be pressed over horizontally by means of a suitable packing iron, tight against the shaft, and held in place by forcing the necessary additional packing behind it. The loose upper ends of the skeins of the wick should then be folded over the other waste and tamped down tight.

The waste chambers of the axle caps should be packed in the same way as those for the armature bearings.

A pad of the saturated waste, large enough to fill the remainder of the waste chamber, should be placed on top of the wick to catch and hold dirt which might fall in when the bearing housing cover is opened.

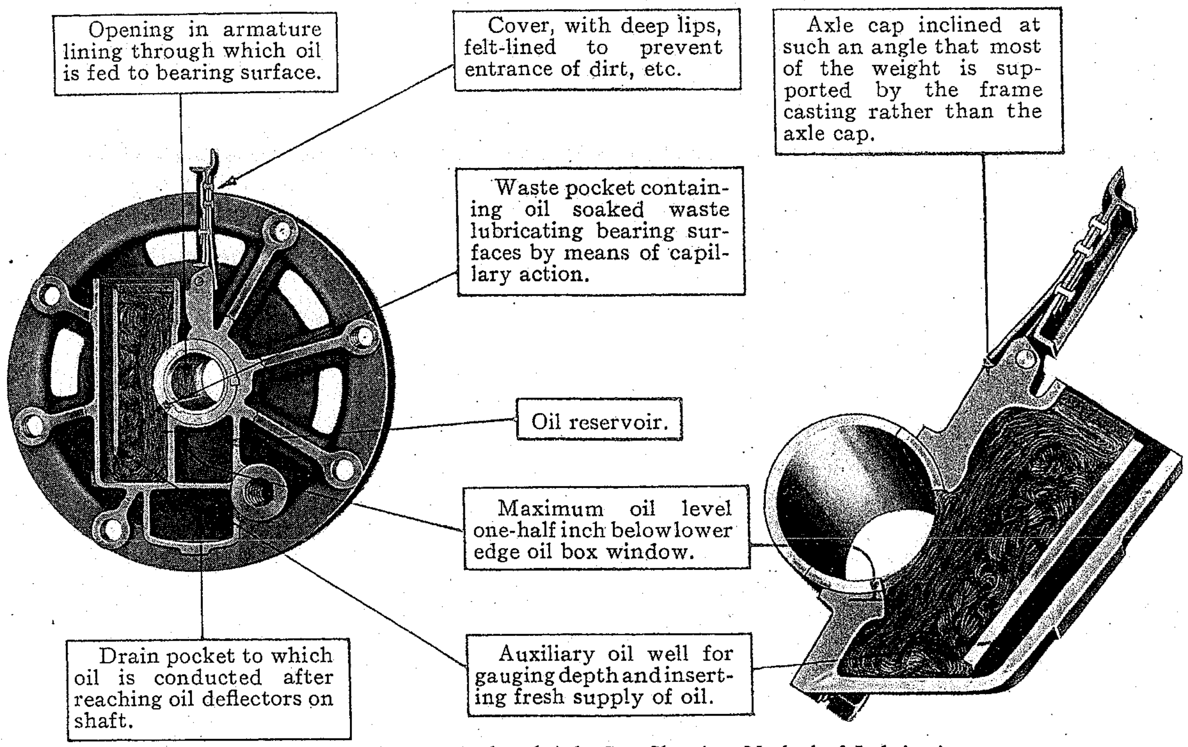
The equivalent of the Galena Signal Oil Company's or the Texas Oil Company's Electric Car Oil (summer and winter grades) should be used for armature and axle bearings.

PROPER OIL LEVELS

The maximum oil level in the bearing pocket should not be above a point one-half inch below the lower edge of the opening in the oil box casting. The minimum depth of oil should not be less than one inch depending on the shape and size of the waste pocket. The accompanying table indicates the maximum and minimum depths of oil to be carried in armature and axle bearings of some modern G-E railway motors. The bearings should not be filled above the maximum depth or the bearings may be flooded and the oil wasted. The depths indicated are as measured in the auxiliary oil wells. In most cases axle caps are inclined at an angle of about 60–70 deg. with the horizontal, consequently the oil thus measured would not be a true depth. The depths as listed below are as they would appear on a measuring stick placed in the well.

PROPER DEPTHS OF OIL IN INCHES—G-E STANDARD RAILWAY MOTORS

Motor	Axle Bearings Axle Caps		Armature Bearings Pinion End		Armature Bearings Commutator End	
	Min.	Max.	Min.	Max.	Min.	Max.
GE-201-G	1½	3	2	3¾	1	2½
GE-203-P	1	2½	2	3½	1	2½
GE-240-A	1½	3½	1½	3½	1½	3½
GE-247-A-D	1	2½	1½	3⅛	1½	3⅛
GE-254-A	1	3	1½	3½	1	3
GE-258-C	1	2½				
GE-263-A	1½	3¼	2¼	4¼	1½	3¼
GE-264-A-B	1	2½	1½	3⅜	1	2½
GE-265-A-C	1	3	1½	3¼	1½	2⅞
GE-275-A-D	2½	4½	1½	3⅜	1	2½
GE-282-C	2½	4¾	2	4"	2	4



Sections Through Armature Head and Axle Cap Showing Method of Lubrication

INSPECTION

The consumption of oil depends on the mileage made, the condition of the bearings and the severity of the service. The frequency of inspection can best be determined by experience on a particular line keeping in mind that new equipment should always have careful attention. It has been found to be necessary at intervals of from ten days to three weeks. A few drops of oil may save ten times their cost in babbitt at a critical time. The packing should be turned over occasionally and particular care taken to see that the lining flanges are being properly lubricated.

ARMATURE LININGS

The armature linings are carried in motor frameheads which have waste pockets and drain pockets. The standard armature lining consists of a bronze shell lined with babbitt and keyed in the framehead. The babbitt is of such thickness that should the metal melt and run, due to accidental overheating, the armature is prevented from striking the pole pieces by the bronze shell and the armature shaft bearing surfaces are protected from injury. The linings are provided with an opening on one side to allow the oily waste in the pockets to come in contact with the bearing surface of the shaft.

AXLE LININGS

The axle linings of a railway motor are generally of bronze for maximum axle sizes, but malleable iron lined with babbitt is sometimes used for smaller diameters of axle. The two halves are prevented from turning by a clamp fit and are either keyed or doweled. An opening provided in the lining on the low pressure side allows the oily waste in the bearing pockets to come in contact with the axle.

The edges of the openings in the linings are chamfered to assist oil in entering the bearing, and grooves are provided to insure the passage of oil from the bearing surface to the flange which must take the thrust. Every precaution is taken to exclude dirt from the bearings. The bearings are of ample size and with proper care and lubrication should give long life.

BRASS AND BABBITT LININGS

The material used for lining shells must offer the best bearing surface with the least amount of friction and have sufficient strength and ductility to withstand the severe operating conditions of railway service. The alloy used for brass or bronze shells is composed of copper, tin, zinc, and lead, the proportions of which have been determined after many years of exacting and exhaustive tests.

Alloy No. 4 is the equipment standard preferred by many railways for replacements on account of its general recognition as the highest grade bearing alloy obtainable. Alloy No. 80 has been standardized for replacements by many large operating companies and gives nearly as good service as Alloy No. 4 at a considerable reduction in cost.

The composition of these alloys is as follows:

MATERIAL	ALLOY NO. 4	ALLOY NO. 80
Copper	84.0%	78.0%
Lead	0.5	16.0
Tin	12.0	4.0
Zinc	3.5	2.0

Brass and babbitt linings are now almost universally used for armature bearings. This type is preferred to the straight bronze type for the following reasons:

1. A rough or uneven surface due to imperfect machining which might be barely detected will quickly wear out of the babbitt presenting a perfectly smooth bearing surface.
2. When grit or any cutting substance becomes lodged between the shaft and bearings, it imbeds in the babbitt without injury to the shaft, whereas in the bronze bearing, the grit powders and acts as an abrasive lap on both shaft and bearing.
3. All types of linings will run hot on either armature shaft or axle if not properly lubricated. When this trouble develops, the shaft is seldom damaged by the babbitt bearing. In fact, many cases have been noted where the babbitt after running hot and melting or becoming scored on the bearing surface took on a smooth glazed surface by the application of sufficient lubricant without causing the least damage to the shaft. Under similar conditions, the bronze unlined bearings were ruined and invariably damaged the bearing surface of the armature shaft.
4. The babbitt can be held more securely in a brass shell than in an iron shell, for the babbitt can be sweated in the brass in addition to being anchored by the usual dovetailed grooves. This permits the use of a very thin liner of babbitt or a thickness slightly less than the air gap between the armature and pole pieces. Therefore, in case the lining runs hot and the babbitt melts, the armature cannot rub on the pole pieces without first wearing into the brass shell.

IRON AND BABBITT LININGS

Cast or malleable iron shells lined with babbitt have been used quite extensively in old type motors. Where provision is made in a bearing for various sizes of shafts, it is customary to use bronze linings either babbitt lined or merely tinned for maximum shaft sizes. For the smaller shaft sizes where more bearing metal is required, malleable iron shells with a thick layer of babbitt cost much less than a straight bronze shell. It is standard practice to use bronze linings for axle bearings except where the thickness of the shell would exceed 5/8 of an inch, in which case malleable iron and babbitt are used.

REBABBITTING BEARINGS

First, rough bore the bearing shell, slot the dovetailed anchored grooves, and turn the grooves in each end. Then, clean the bearing lining thoroughly, removing all of the old babbitt and any foreign matter in order to obtain a clean bright surface to which the babbitt will adhere. After cleaning, brass linings should be tinned in a bath of half and half solder (melting point 178 deg. Centigrade) and pour the babbitt while still hot from the tin bath. The best results are obtained by preheating the shell and jig to a high temperature nearly equal to that of the babbitt. The latter will then flow freely into anchored grooves and adhere firmly to the tin surface. Then bore and ream the babbitt shell to size and finish the outside.

Heat the babbitt to a temperature of 500 to 550 deg. Centigrade (but not more than 550 deg. or decomposition of the alloy will take place). The dross which rises to the top of the metal should be cleaned off with a ladle before dipping the babbitt from the kettle. Be sure none of this dross is allowed to get into the bearing.

When pouring armature linings, use nothing but new metal. Under no circumstances mix babbitt from old linings in the pot with the new metal. Babbitt melted from old linings may be used for journal or axle bearings.

A rough estimate of the temperature of the babbitt may be made by dipping a pine stick in the pot. If the metal is hot enough to be poured, it will quickly set the stick on fire. However, the most reliable method is to determine the temperature with a thermometer or with a pyrometer. When babbitting iron shells, it is essential that all of the inner surfaces to which the babbitt must adhere are thoroughly clean. Dovetailed grooves are provided in the flange as well as in the body of the shell to anchor the babbitt. Heat the shells to the temperature of the babbitt, place them in jigs and pour the babbitt. Then bore the lining ream to size and finish the outside and ends.

BABBITT

The babbitt used in General Electric railway motor bearings has a tin base composition, the proportions being 83⅓ per cent tin, 8⅓ per cent copper and 8⅓ per cent antimony. This is known as Alloy No. 17. Not only in regard to the proportions of its ingredients, but also in regard to methods of mixing, handling, and the temperature employed, this alloy is the result of exhaustive tests over a period of many years.

SAMPLE COPY

INSTRUCTIONS GEH-783

ARMATURE AND AXLE BEARING LININGS

FOR RAILWAY MOTORS

Armature Bearing Linings

The standard armature bearing lining consists of a bronze shell, lined with babbitt and held in the framehead by a press fit and also a key to secure the proper location of the bearing window. The babbitt is of such thickness that, should the metal melt and run due to over-

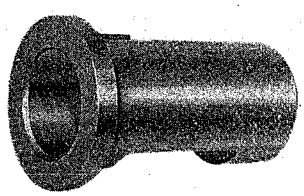


Fig. 1. Railway Motor Armature Bearing Lining

heating, the armature is prevented from striking the pole pieces by the bronze shell, and the armature shaft bearing surfaces are protected from injury. The lining is provided with an opening on one side to allow the oily waste in the pocket to come in contact with the bearing surface of the shaft. A typical armature bearing lining is shown in Fig. 1.

Axle Bearing Linings

Axle bearing linings on most modern motors are of bronze alloy. For some of the larger motors the bearing surfaces and flanges are babbitted.

The linings are made in two halves which are held tightly in place by the clamp fit of the axle caps, and are either keyed or doweled. An opening on one side of the lining allows the oily waste in the bearing pocket to come in contact with the axle. The edges of the openings in the linings are chamfered to assist oil in entering the bearing, and small grooves are provided at the flange end to aid the passage of oil from the bearing surface to the flange, which takes the end-thrust.

Axle bearing linings of the keyed and doweled type are shown in Fig. 2 and 3 respectively.

Limits of Wear

The bearing wear should be measured at regular intervals by gauging the clearance between the shaft or axle and the lining bore. The American Electric Railway Engineering Association Manual, on page 50 of the 1927 Supplement, recommends the following clearances:

Lining	Size of Motor	Total Clearance on Dia. in Inches
Axle	Up to 50 hp.	$\frac{1}{16}$
Axle	50 hp. and larger	$\frac{1}{8}$
Armature	Up to 50 hp.	$\frac{3}{64}$
Armature	50 hp. and larger	$\frac{1}{16}$

The wear of armature bearing linings can also be checked by measuring the air gap between the armature and the bottom field pole.

The end play in armature and axle bearings should also be checked regularly and should not be allowed to become excessive. The allowance for end play will vary with different motors, de-

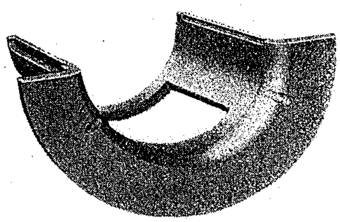


Fig. 2. Railway Motor Axle Bearing Lining, Keyed

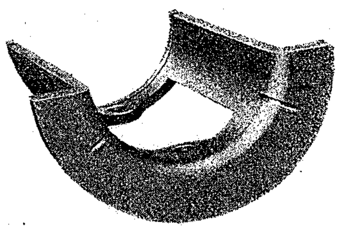


Fig. 3. Railway Motor Axle Bearing Lining, Doweled

pending on the clearance of the gear and pinion with respect to the sides of the gear case, the position of the armature with respect to sta-

GENERAL ELECTRIC COMPANY
SCHENECTADY, N. Y.

December, 1929 (2M)

Superseding GEA-60A in part

B-9120

tionary parts, the adjustment of the axle collar, and other conditions which can be best determined by the operator.

When armature bearing linings are worn to the limit, they should be rebabbitted or replaced with new linings. Axle bearing linings which are not babbitted should be replaced with new ones when worn to the limit.

Axle caps should be kept tightly bolted in place, and it is important to see that both armature and axle bearing linings fit tightly in their respective housings at all times.

Rebabbitting Bearings

Babbitt metal having a tin base is recommended for railway motor bearings. The babbitt used by the General Electric Company has the following composition:

Tin	83⅓ per cent
Copper	8⅓ per cent
Antimony	8⅓ per cent

The babbitt should be heated to a temperature of about 500 degrees C. but should not be allowed to become hot enough to decompose. Use a thermometer or pyrometer to determine the temperature. Skim off all dross from the molten metal before pouring.

Use nothing but new babbitt metal for armature bearing linings, and under no circumstances mix babbitt from old linings in the pot with new metal.

Before rebabbitting, rough bore the bearing shell, slot the dovetailed anchor grooves and turn the grooves in each end. These grooves are shown in Fig. 4.

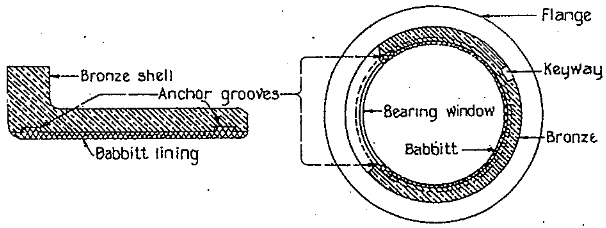


Fig. 4. Sectional Views of Armature Bearing Lining

Bronze bearing linings should be tinned on the surfaces which are to be babbitted. Clean the bearing shell thoroughly, removing all of the old babbitt and any foreign matter. Paint with chalk whiting the surfaces which are not to be tinned. Warm the lining and dip it into molten tin or half-and-half solder, then put it into a jig or babbitting machine immediately and pour the babbitt at about 500 degrees C. When the bearing is cool, bore and ream to size and finish the outside.

Although this type of bearing requires a nozzle and source of pressure for proper filling, if it becomes necessary to add oil when these parts are not available, it is possible to do so by pouring it in the filler opening under which condition the bearing will operate as one with an ordinary auxiliary oil well.

BEARING LININGS

Axle linings are made in two halves and most modern ones are of a bronze alloy. These, to a great extent, supersede the malleable or cast iron and babbitt axle linings formerly used. For some of the larger motors, especially of the locomotive type, the bearing surface is tinned.

Armature linings are made of a bronze shell with a babbitt bearing surface.

ANTI-FRICTION BEARINGS

Several motors such as the GE-258, GE-288, and GE-296 have been equipped with anti-friction armature bearings. Results to date have shown that when kept free from dirt and properly lubricated these bearings will operate successfully.

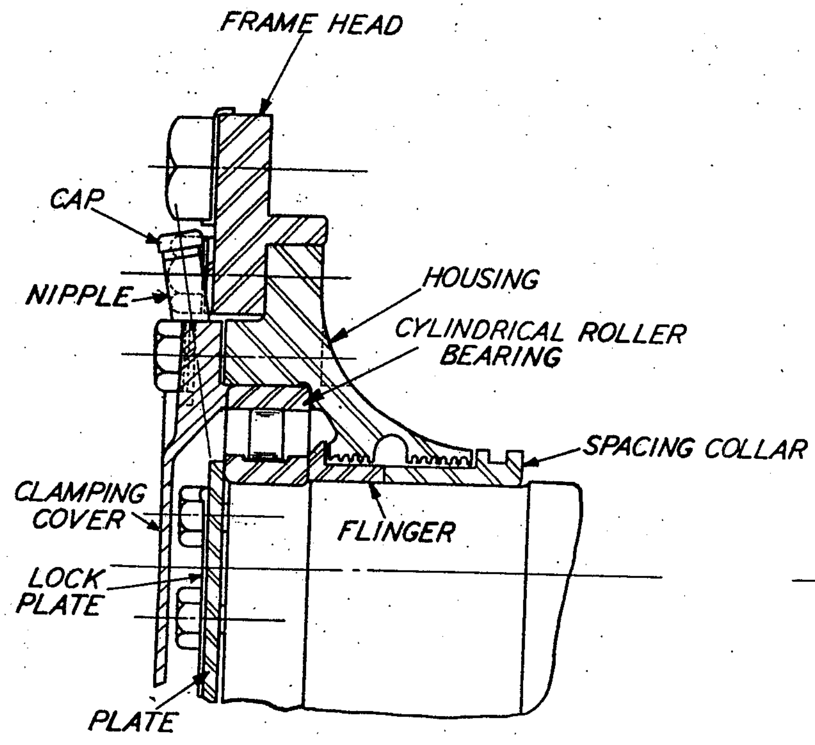
Different constructions have been used, such as; ball bearings on both ends; Cylindrical roller bearing on the pinion end, with a ball bearing at the commutator end for the thrust load; and spherical roller bearing at the pinion end with a plain cylindrical roller bearing at the commutator end.

In some of the early applications, felt was used as a grease and dirt seal. The felt proving unsuccessful, the later installations depend on a complicated labyrinthian path. This is obtained by using oil flinger rings shrunk on the shaft on each side of the inner race. These are corresponding grooves out in the housing which catch the lubricant and form a seal. This new seal has, to date, been very successful in excluding dirt and foreign matter from the bearings and in retaining the lubricant in the bearing.

Bulletin GEH-728, giving information on the maintenance and operation of bearings, may be obtained from the Publicity Department.

For general photographs of anti-friction bearing assemblies refer to:

Photo. 249081—Page 47.
Photo. 249080—Page 48.



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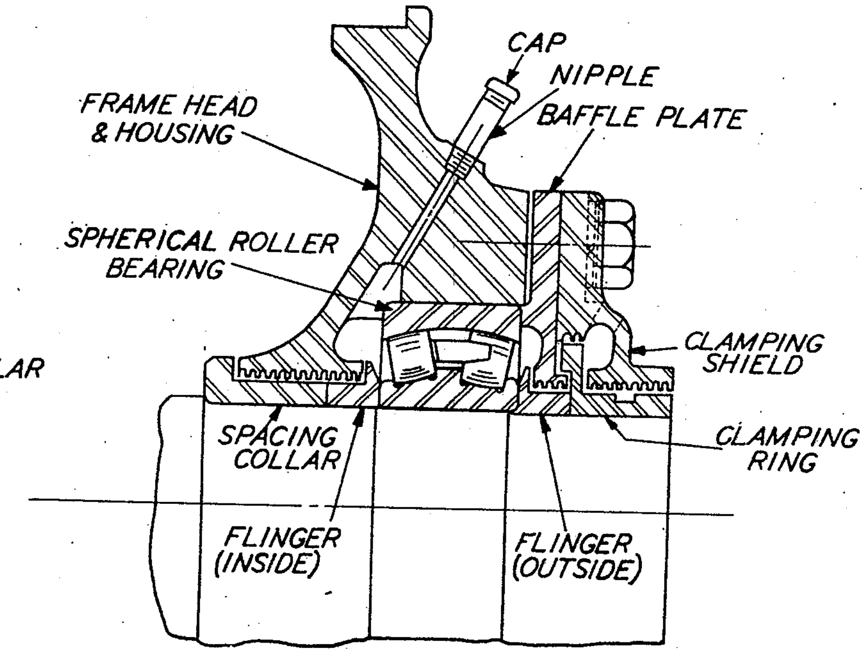


PINION END

249081 GE RAILWAY MOTOR. ROLLER BEARING ASSEMBLY. SPHERICAL ROLLER BEARING, PINION END; CYLINDRICAL ROLLER BEARING, COMMUTATOR END.

E315.1 10-26-28

FIELDS

Seats or bosses for the pole pieces are machined on the inside of the frame.

Pole pieces for the exciting field are built up of punched steel laminations riveted together and enclosing a steel key into which the supporting bolt holes are drilled and tapped.

Laminations are dipped in varnish and baked before assembly. This varnish prevents high eddy current losses in the pole piece.

Commutating pole pieces are made of solid steel. In some cases special tips of non-magnetic material are employed which aid in making a narrower flux band, thereby improving commutation. Non-magnetic shims are used between commutating pole pieces and the magnet frame in some cases to improve the motor's resistance to arcing which might be caused by a circuit interruption.

Field coils for the smaller motors are usually wound with cotton-insulated, round or rectangular copper.

Exciting field coils of the larger motors are usually flatwise wound with strip copper in decks with asbestos insulation between turns. The decks are separated by mica collars.

Commutating field coils are either flatwise or edgewise wound with strip copper insulated between turns with asbestos.

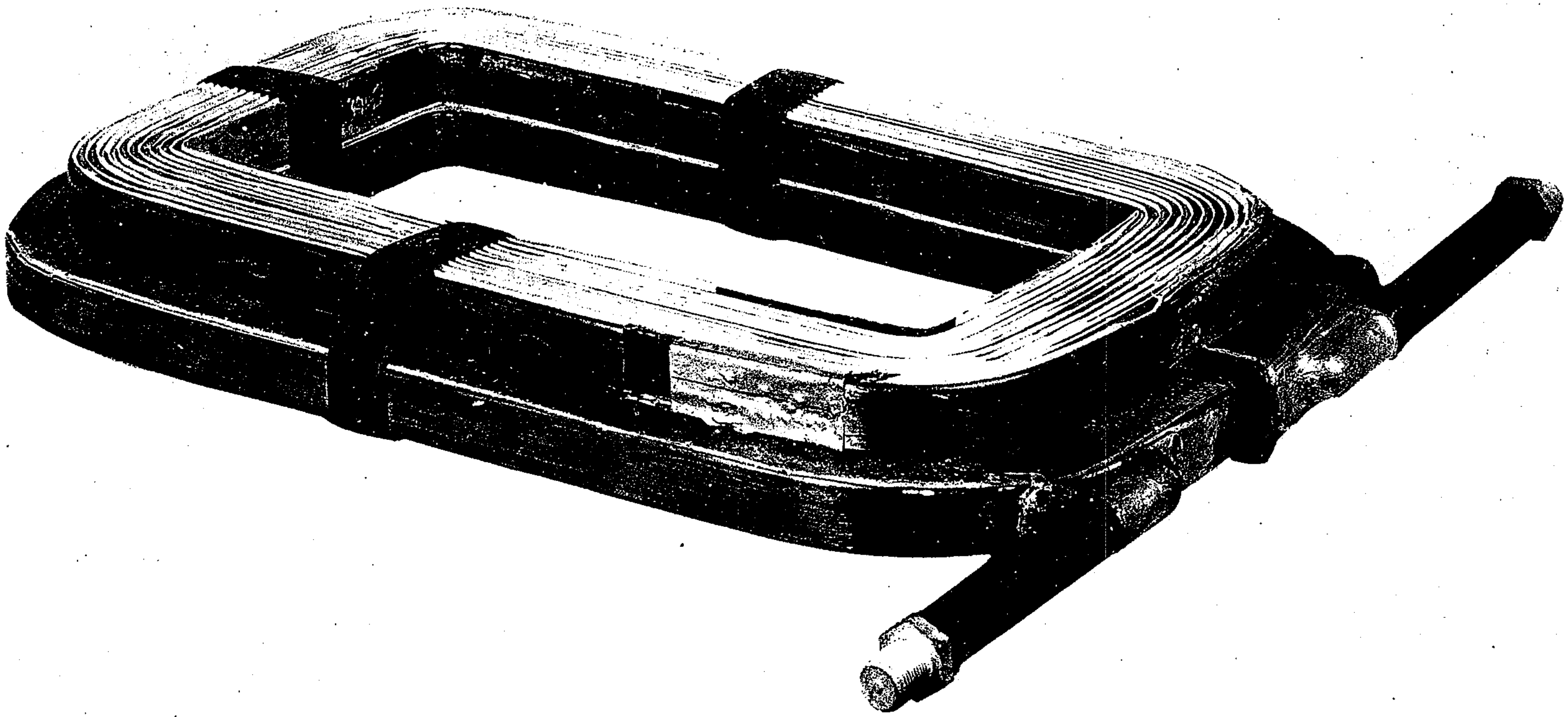
Insulating compound is applied by the vacuum process. This compound thoroughly impregnates the coils and makes a strong, moisture-proof, mechanical and electrical structure. After compounding, the coils are taped with the proper insulation, given a final dipping in insulating varnish, and baked.

Terminals are riveted and soldered to the copper and are taped firmly in place under the coil insulation. Photograph 249044, on page 50, represents an uninsulated coil showing how the clips and terminals are secured to the copper conductor.

Supports for the field coils include: spring pads, flat pads, spring flanges and spool flanges. The pads fit between the frame and the coil; the flanges between the coil and pole piece. They hold the coil tightly in place to prevent chafing and provide a smooth surface against which to clamp the coil. In cases where no pad is used, a machined seat is provided on the magnet frame.

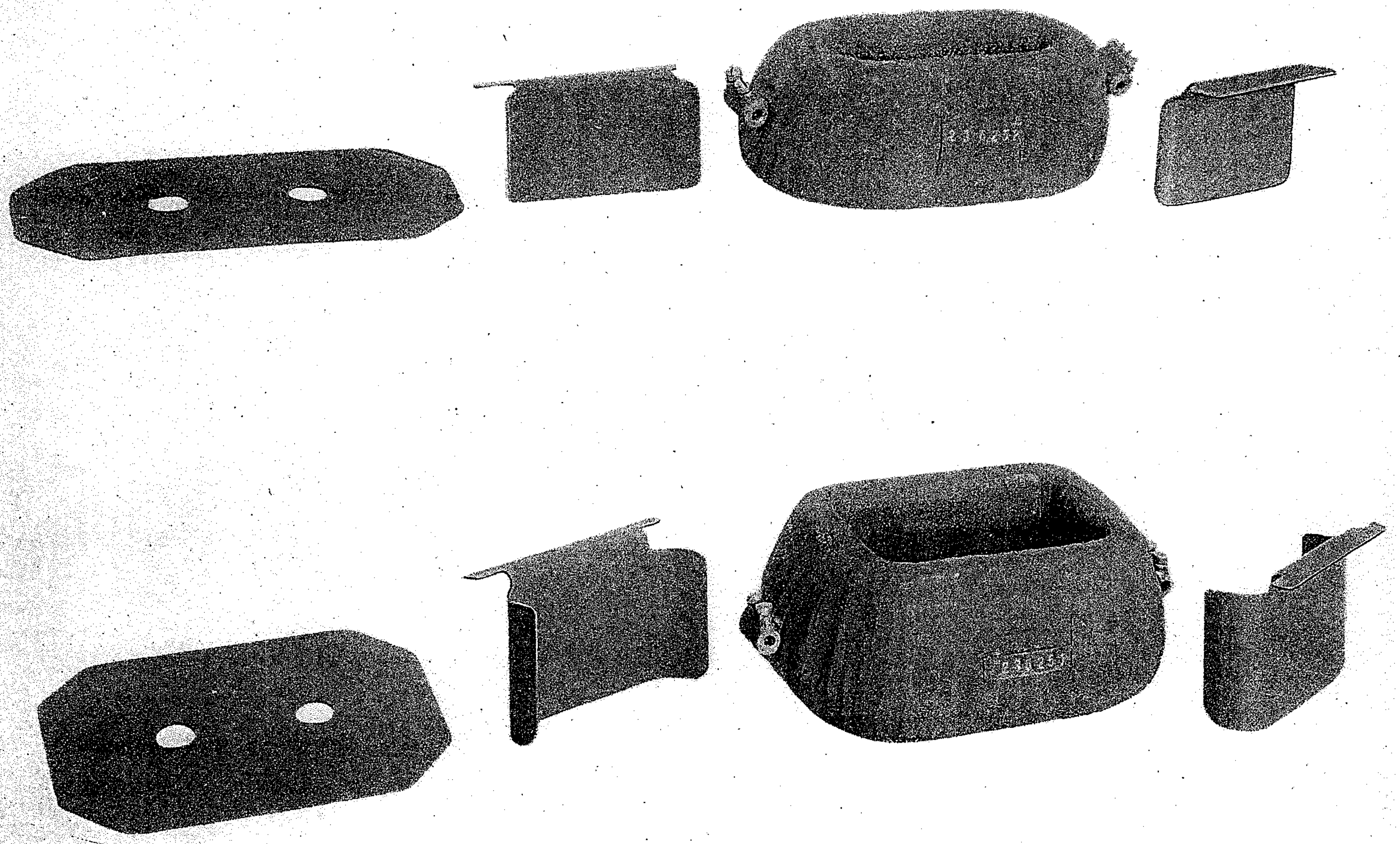
Finished coils and supports are shown by:

Photograph 242405 Page 51
Photograph 244092 Page 52
Photograph 244106 Page 53
Photograph 246886 Page 54
Photograph 247764 Page 55
Photograph 247807 Page 56



249044 RAILWAY-MOTOR FIELD COIL BEFORE INSULATING. APPROX. 1/4 SIZE.

E315.1 10-17-28

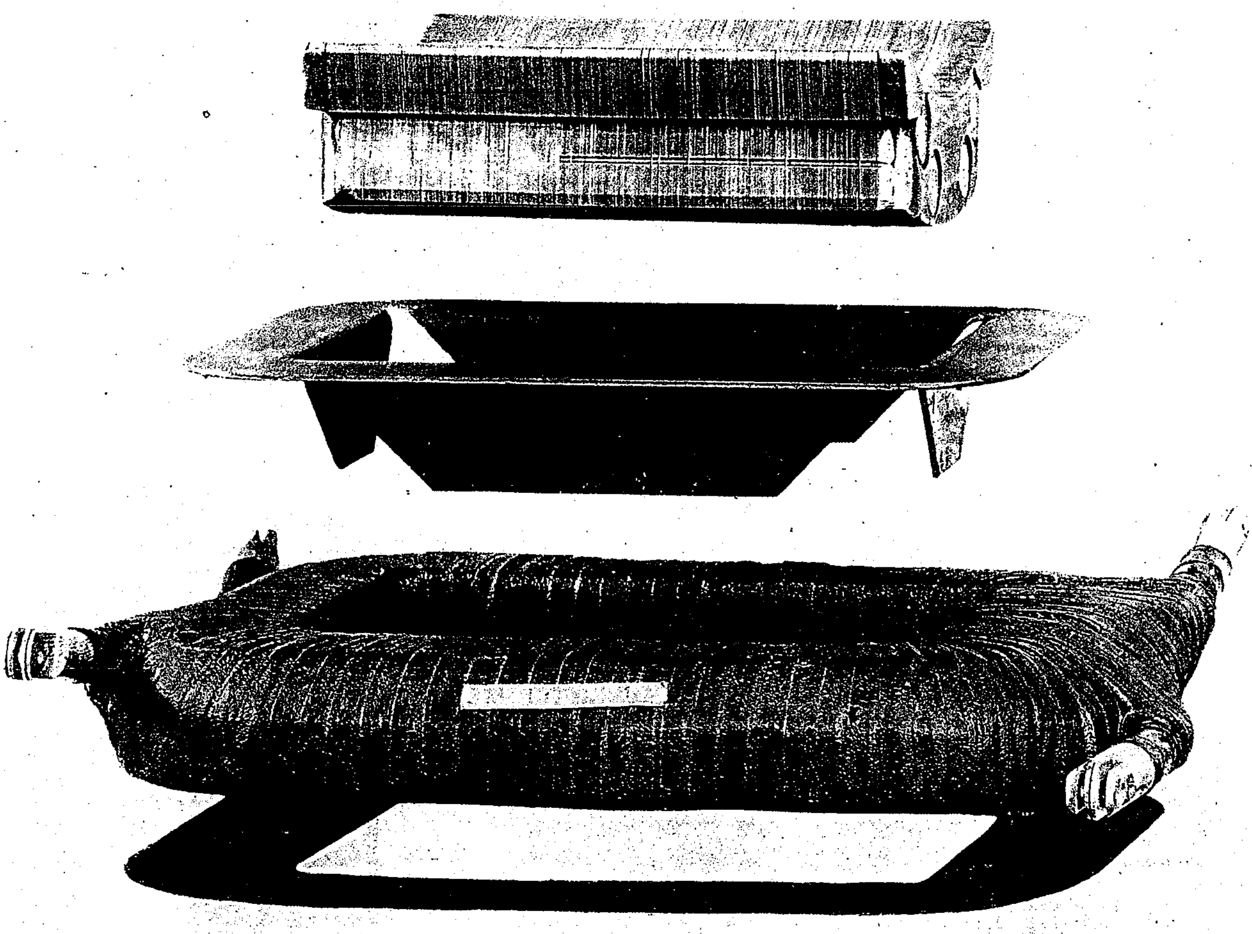


FIELD COILS, SPRING FLANGES, AND STEEL PADS
FOR G-E. RAILWAY MOTORS

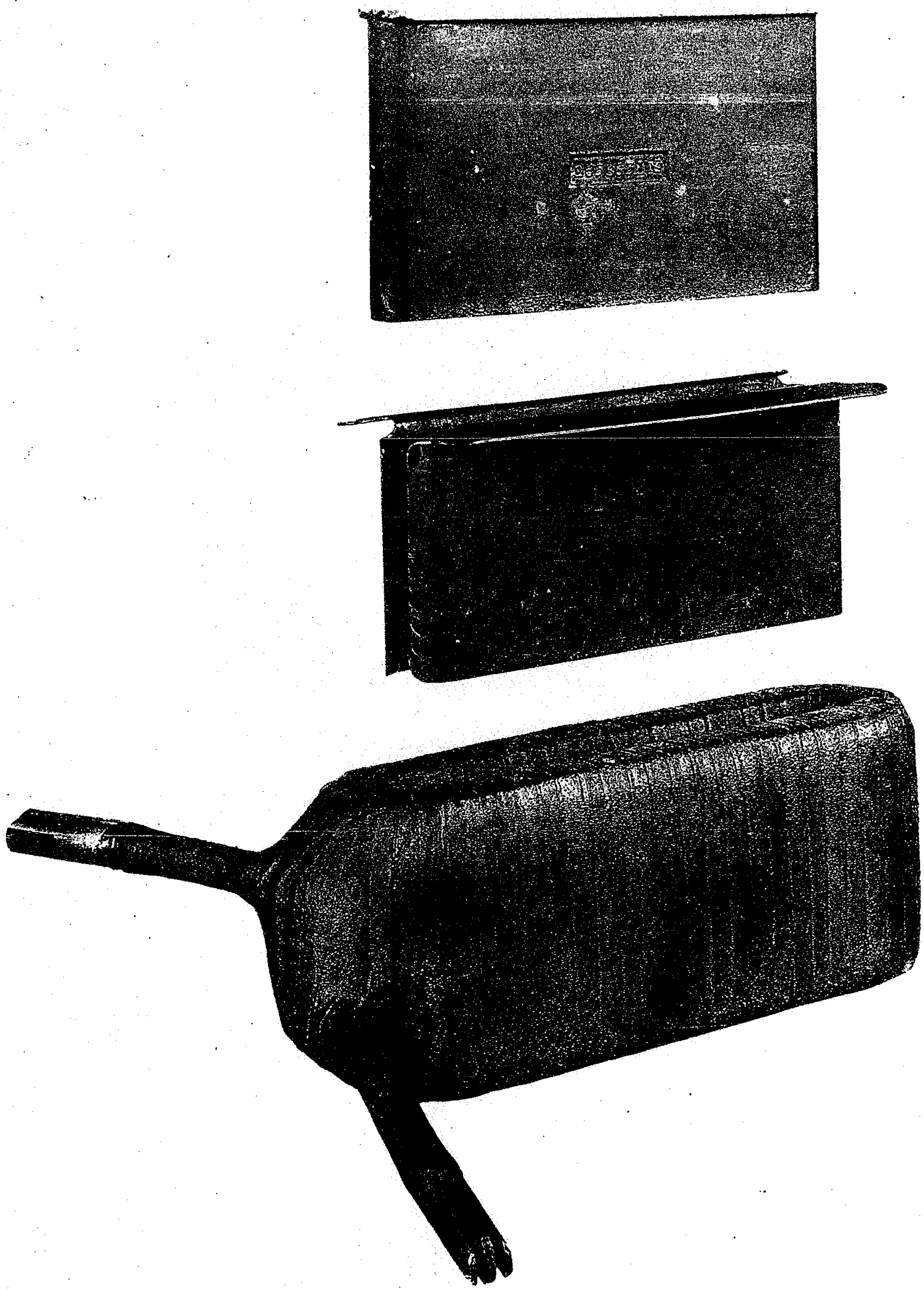
242405

INDEX E-315.1

8 14 22

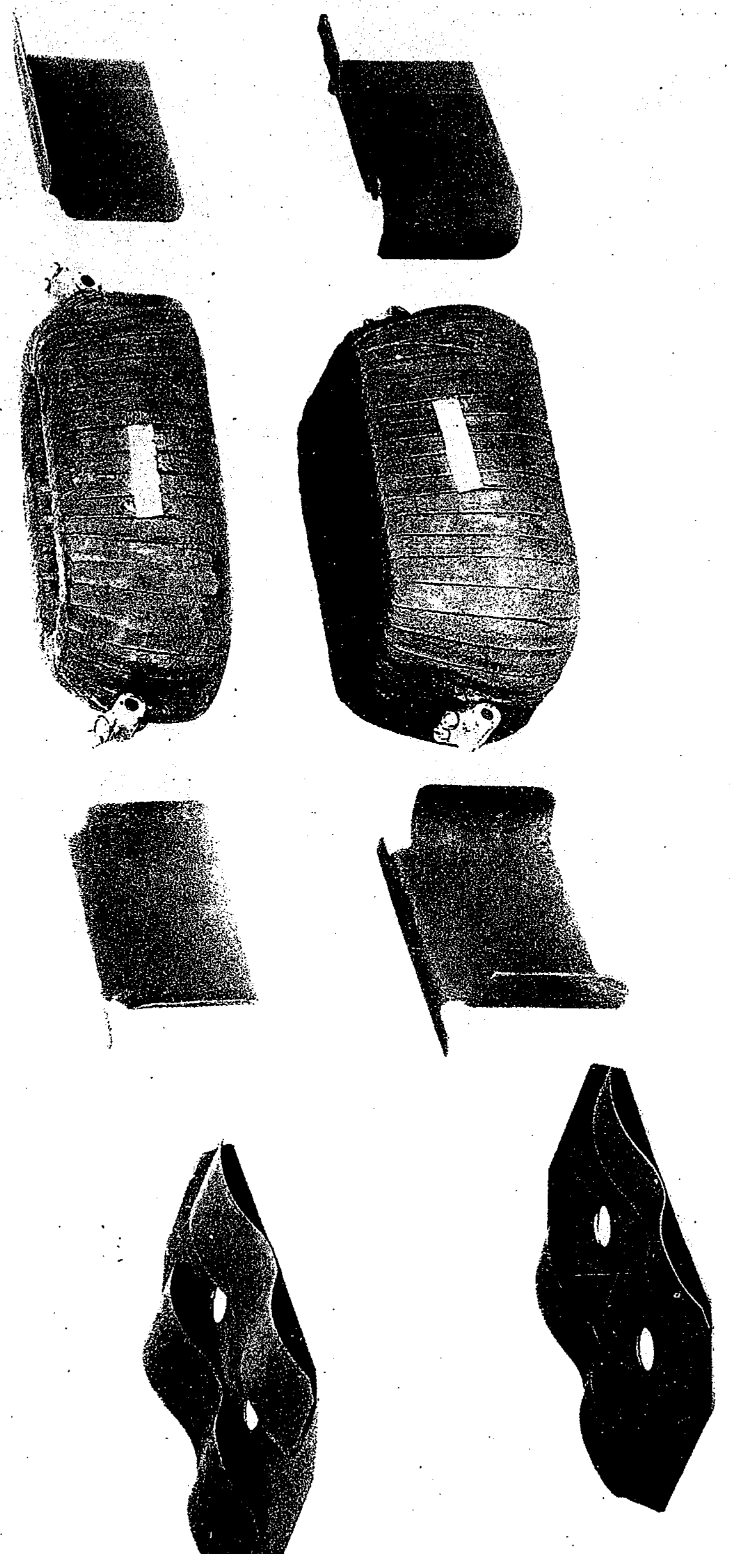


244092 G-E. RAILWAY MOTOR EXCITING FIELD COIL, POLE PIECE, SPOOL FLANGE AND STEEL PAD
INDEX E-315.1



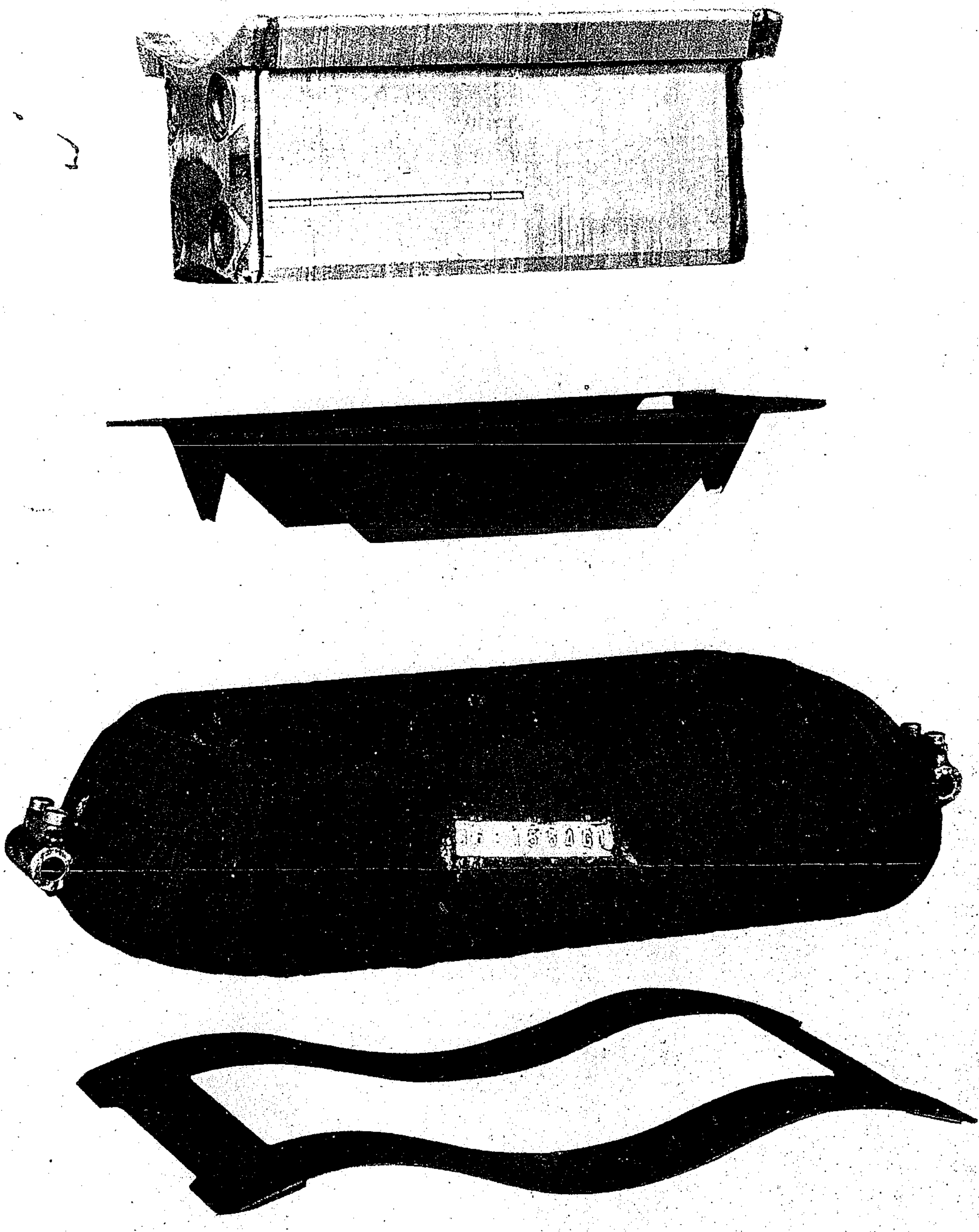
244106 G-E. RAILWAY MOTOR COMMUTATING FIELD COIL, POLE PIECE AND SPRING FLANGES

INDEX E-315.1



G-E. RAILWAY MOTOR FIELD COILS, SPRING FLANGES AND SPRING PADS
APPROX.1/4 SIZE E-315.1 5-7-27

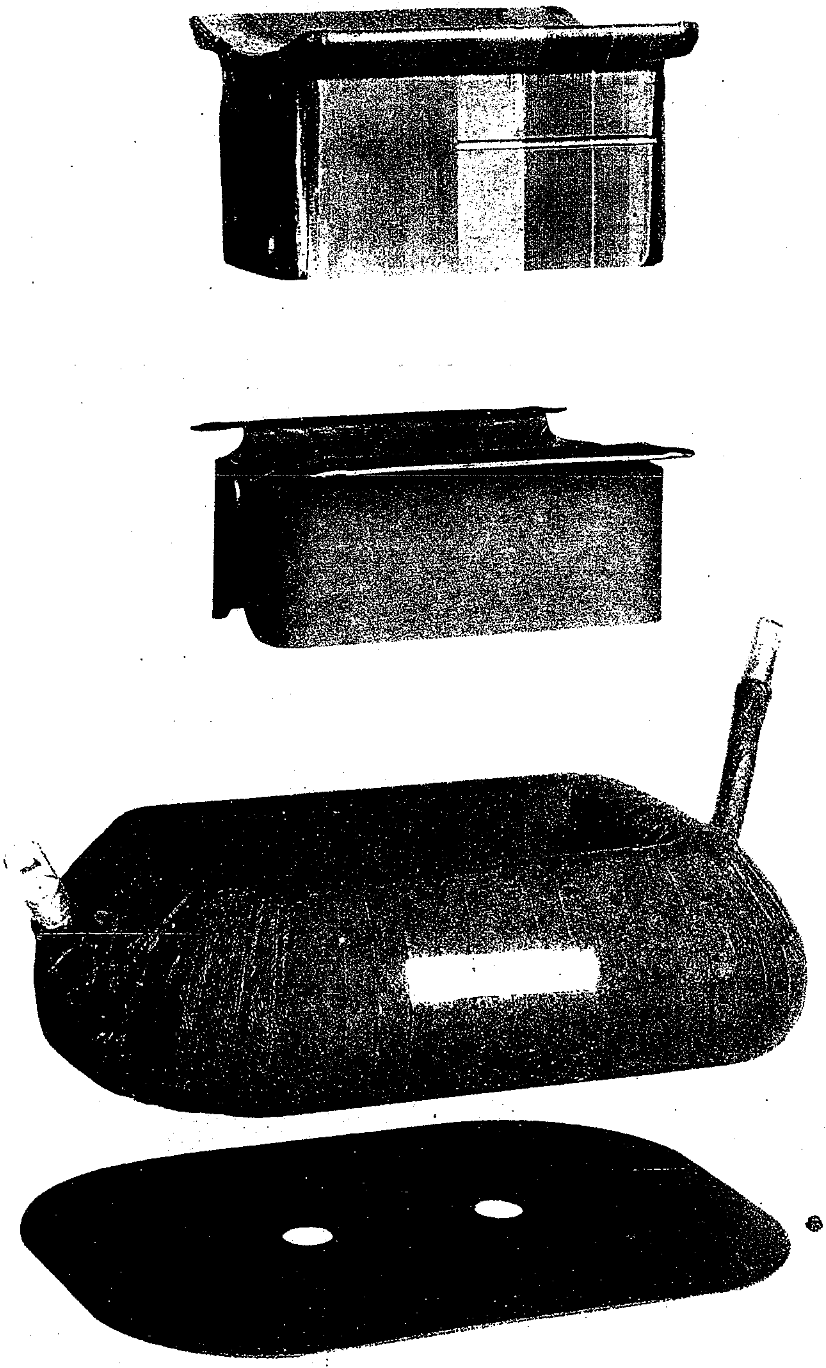
246886



247764

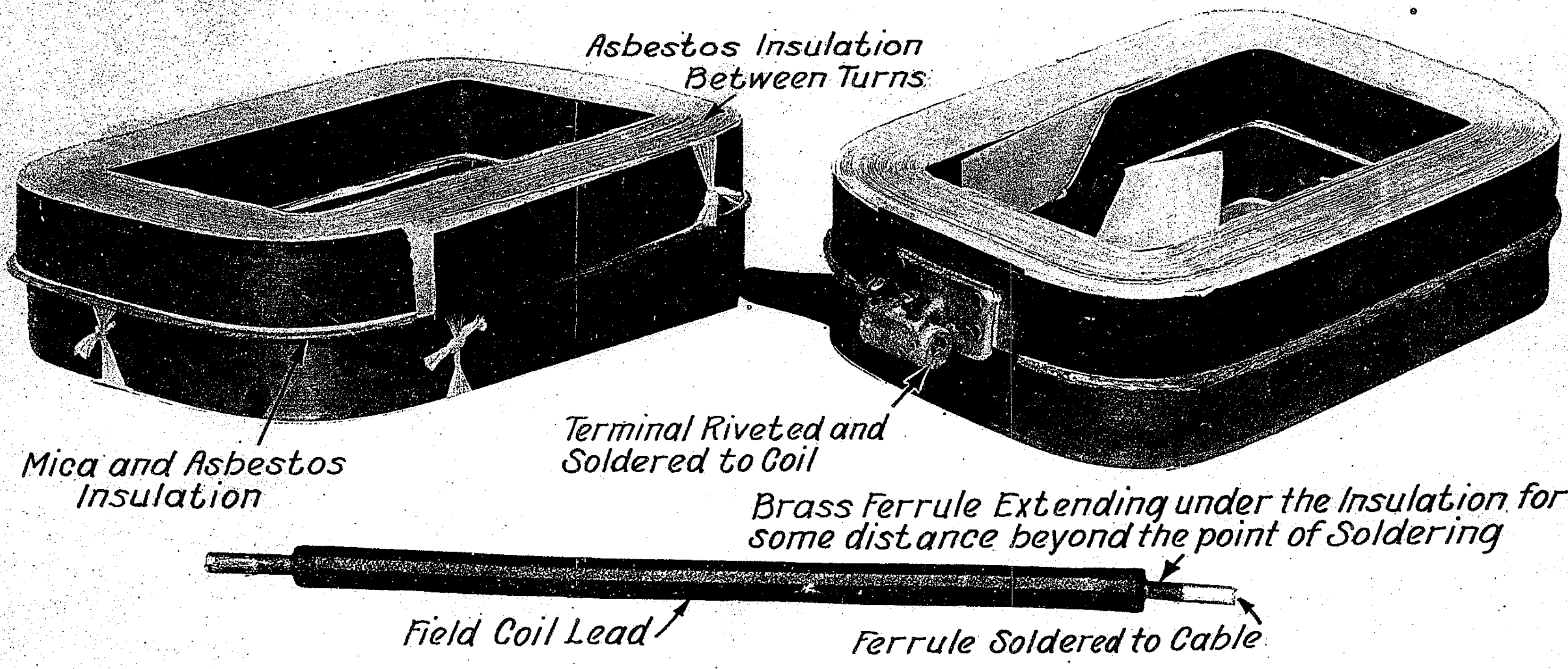
G-E. Railway Motor Exciting Field Coil, Pole Piece, Spool Flange, and Spring Pads

E-315.1 11-15-27

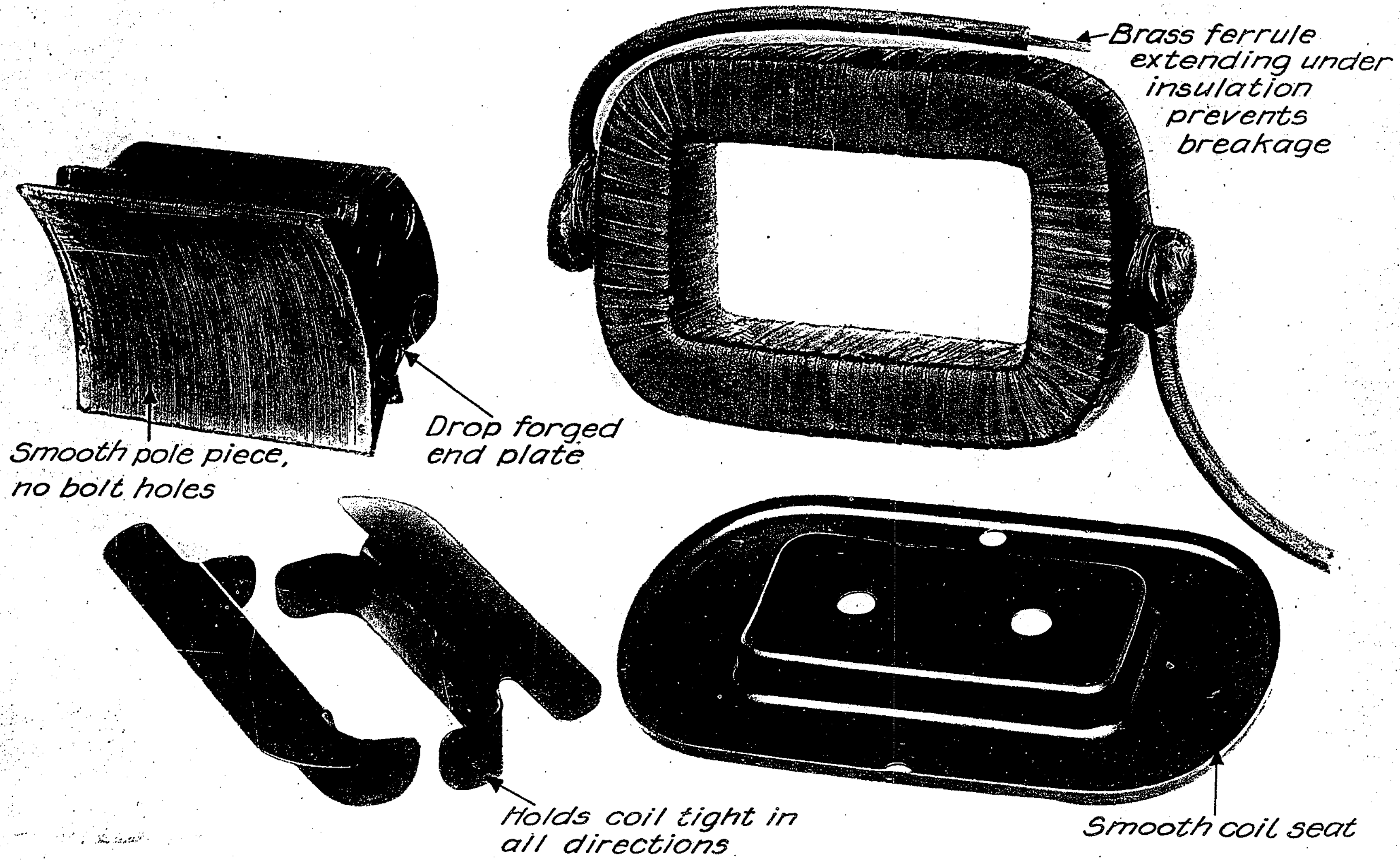


247807 G-E. RAILWAY MOTOR EXCITING FIELD COIL, POLE PIECE, SPRING FLANGES, AND STEEL PAD

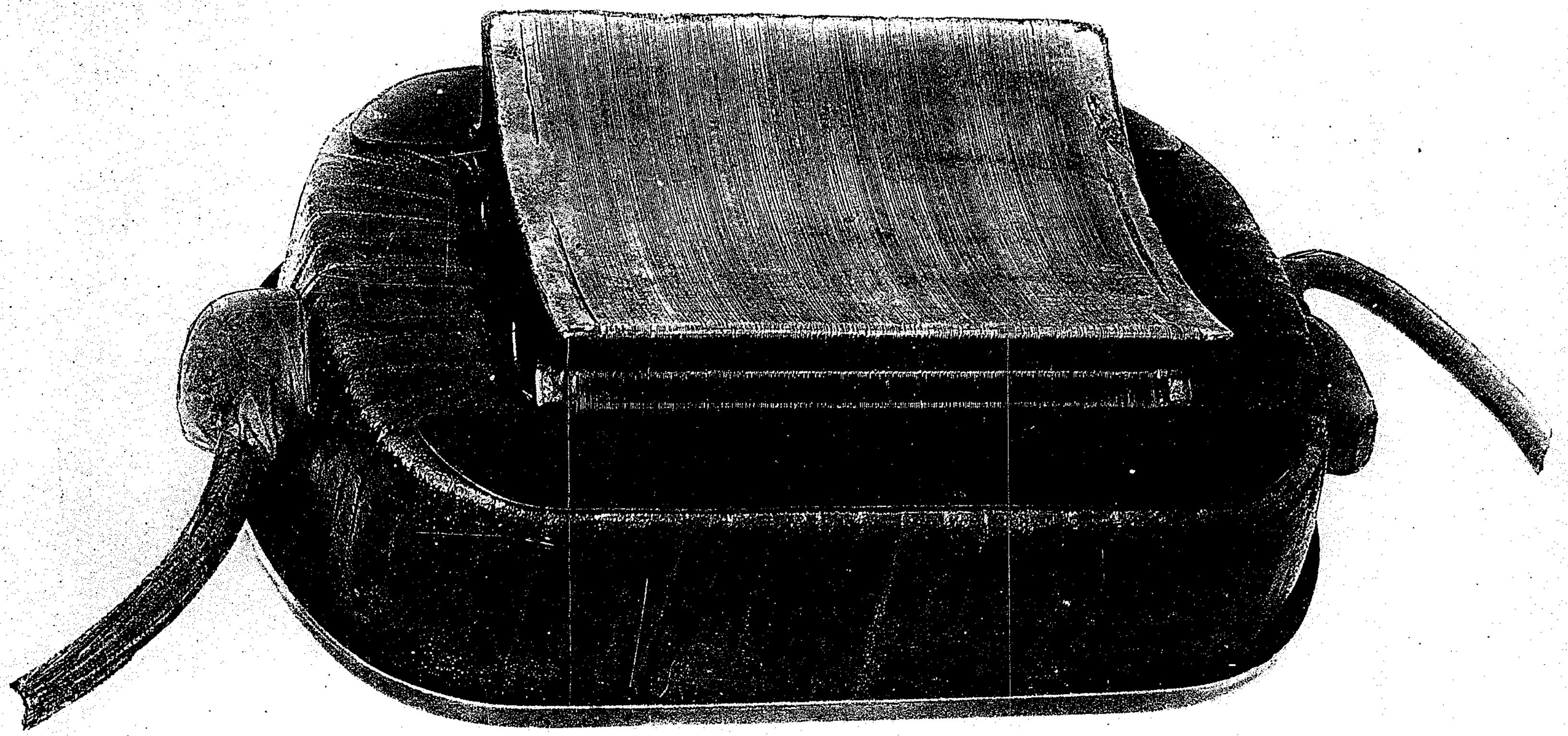
E-315.1 11-23-27



GE Railway Motor Field Coils. Showing Construction and Method of Attaching Leads



Exciting Field Coil and Coil Support With Spring Steel Flanges, G E Railway Motor



Exciting Field Coil and Support Assembled on Pole Piece, GE Railway Motor.

ARMATURE

SHAFT

Shafts are made of special alloy steel machined to size, threaded, and splined for keyways. Bearing surfaces are accurately ground and rolled.

Steel keys of the proper size and shape are used.

SPIDERS

Steel spiders are used on some of the larger motors. They tend to stiffen the shaft and form a more rigid core.

LAMINATIONS

Laminations having slots, ventilating ducts and keyways are punched from annealed steel.

Laminations are dipped in enamel and baked to reduce core loss.

ARMATURE HEADS

Armature heads are of cast steel and for self-ventilated motors the fan is either mounted on or cast integral with the pinion end head.

COMMUTATORS

Commutators are of either the bolted or ring nut type.

Cap and shell are made of steel with longitudinal ventilating ducts. These ducts are also used for the bolts that hold the core together when pressing out the shaft.

Moulded mica cones insulate the segments from the cap and shell.

Sheet mica is used for segment insulation.

Firmness of the commutator is assured by tightening the retaining bolts or nut while the commutator is hot.

Exposed mica between the segments and bead ring is covered with varnished cloth which in turn is bound with cord. It is then painted with insulating compound.

Ring Nut Commutator, Photograph 229393, Page 58.
Bolted Commutator, Photograph 233638, Page 59.

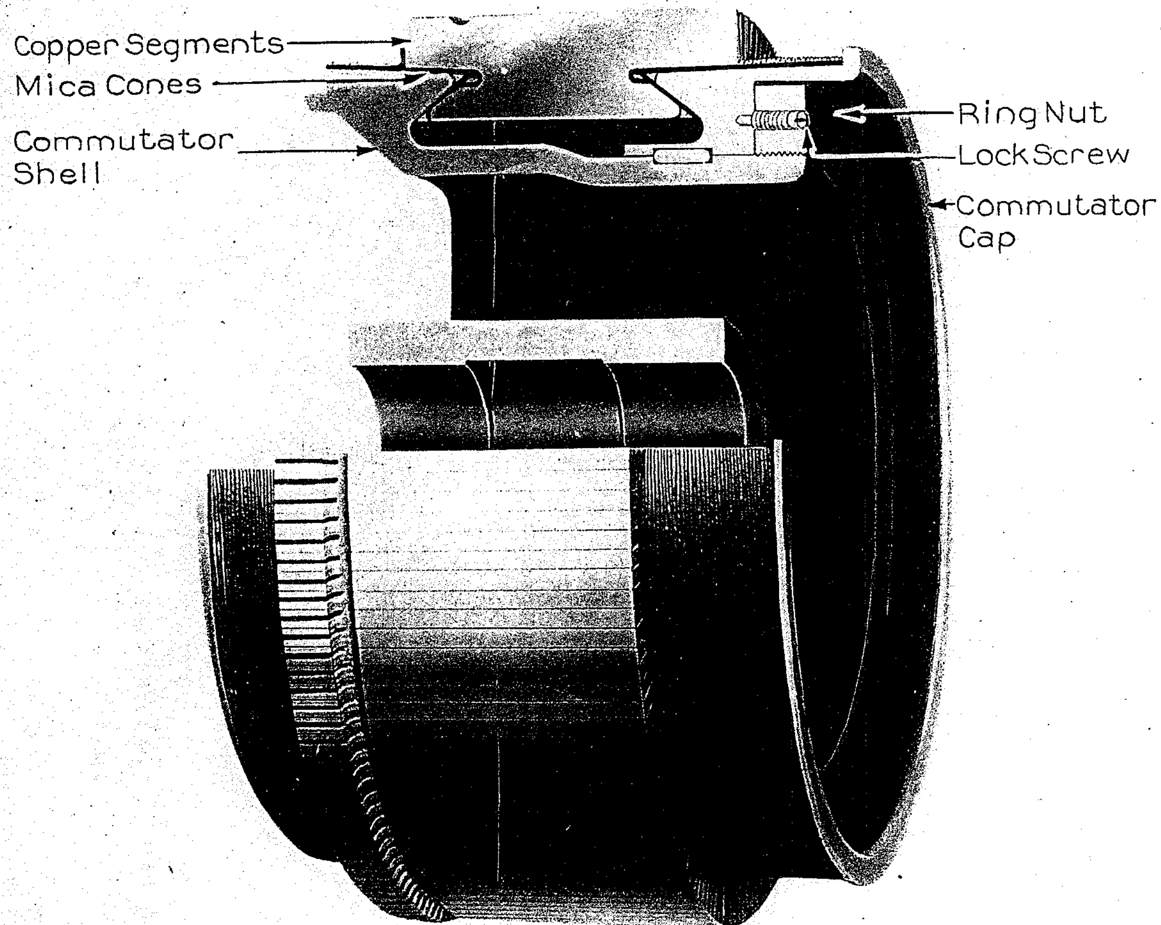
THRUST COLLARS

Thrust collars are of forged steel or malleable iron with oil deflectors assembled on them or made a part of them.

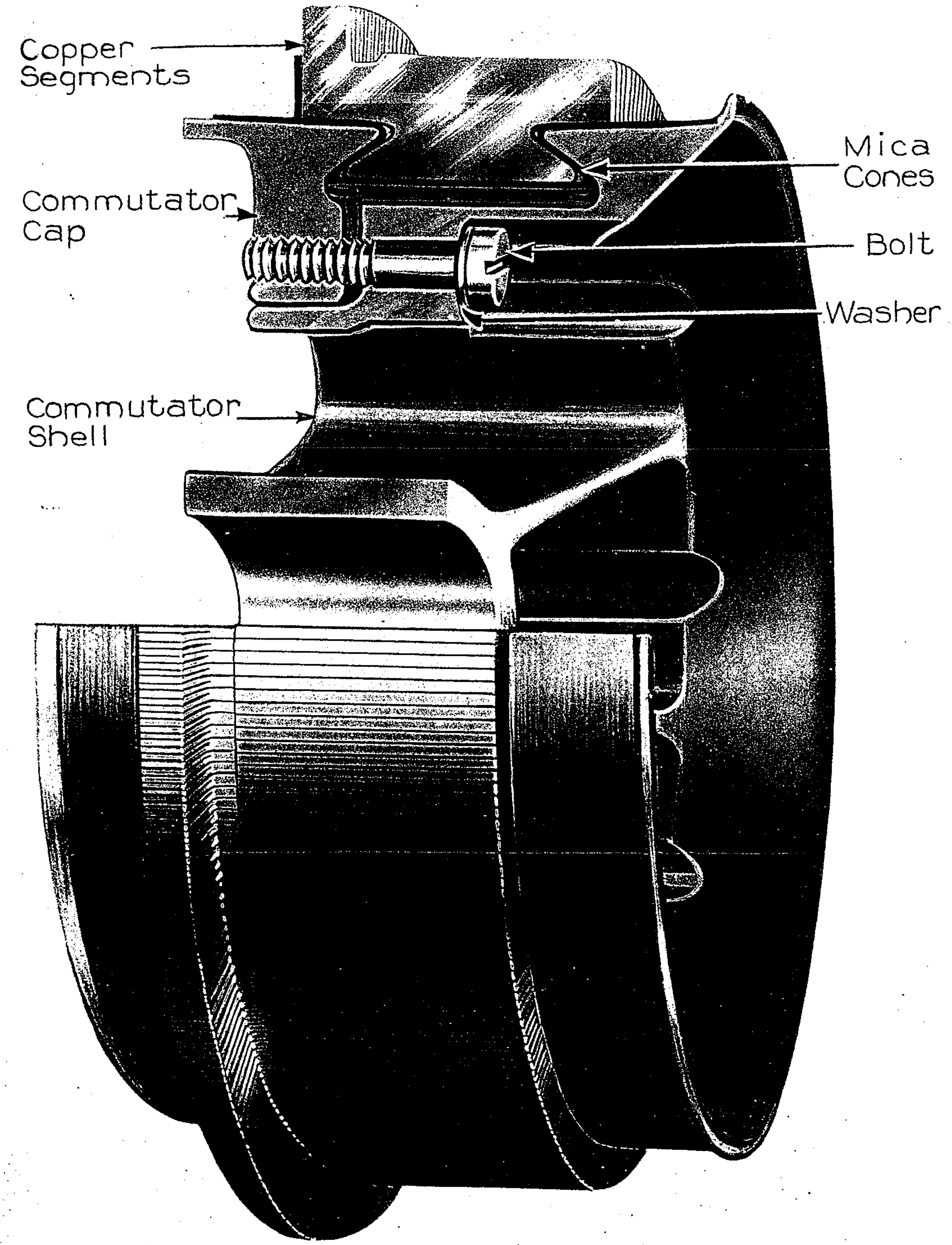
WINDINGS

Coils of more than one turn are wound with either round or rectangular copper; one turn coils are made of rectangular copper.

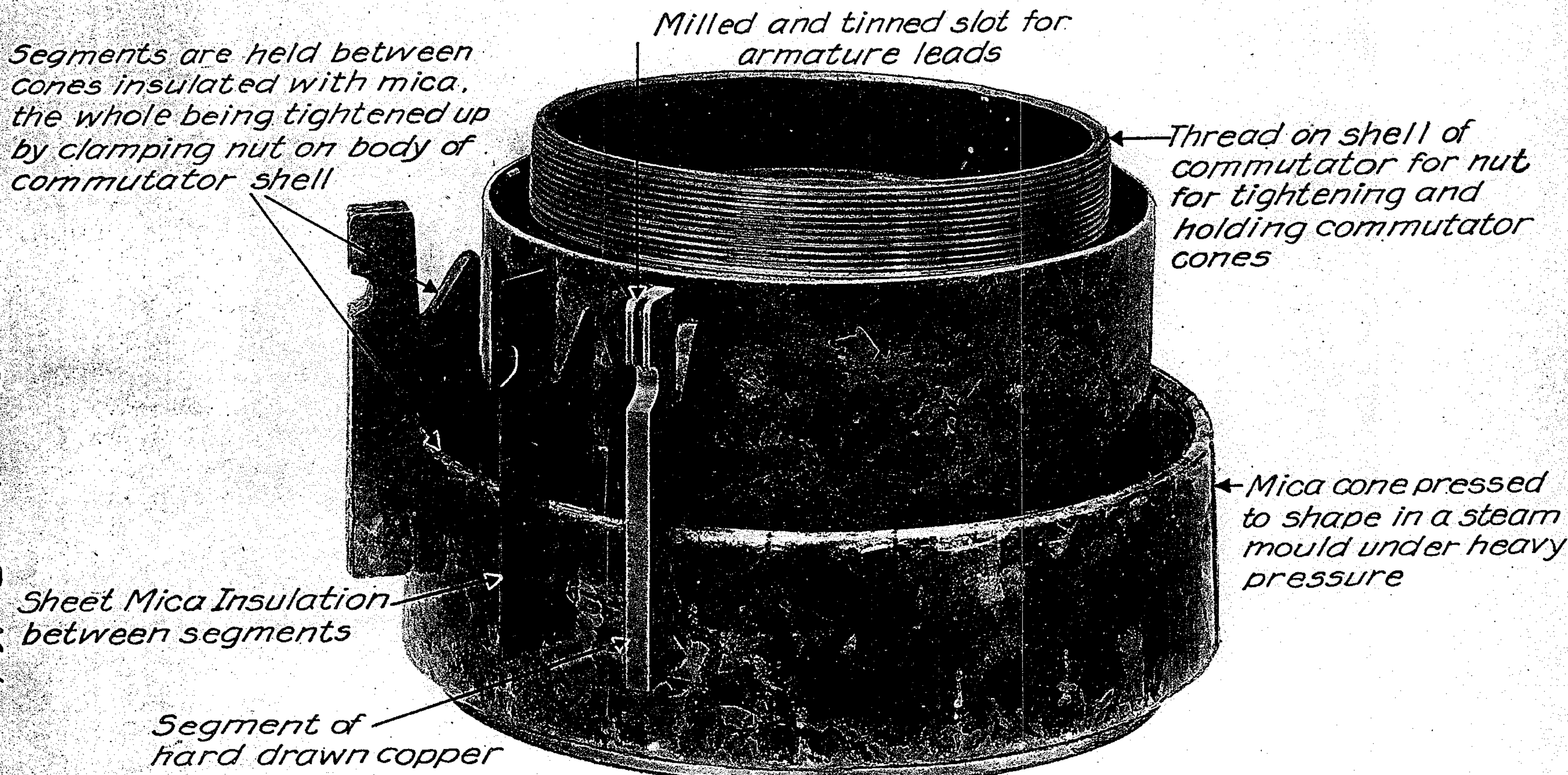
After forming and insulating, the individual coils are assembled into poly coils which are baked, hot moulded, varnished, taped, and, after final taping, are dipped in insulating compound. The ends of the leads are tinned.



229393 Commutator for G-E Railway Motor (Ring Nut Type)



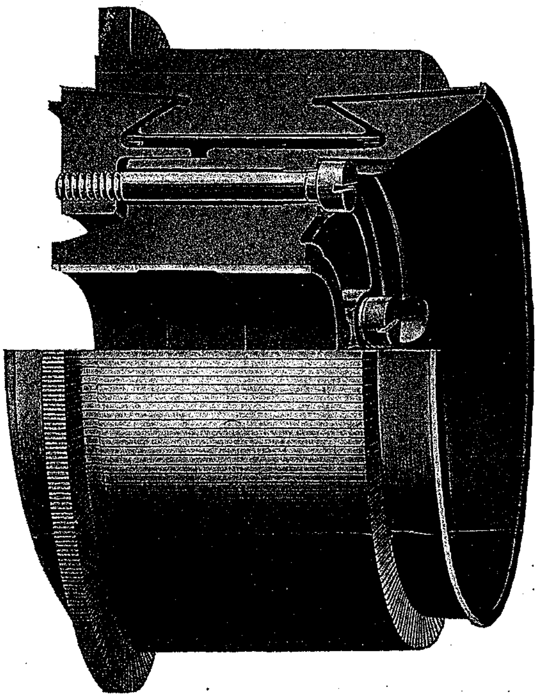
232638 Commutator For G-E Railway Motors (Bolted Type)



Parts of Commutator GE-203 Railway Motor

THE REPAIR OF COMMUTATORS OF RAILWAY AND INDUSTRIAL-HAULAGE MOTORS

The commutators of G-E railway and industrial-haulage motors are of either the *bolted* or *ring nut* construction. The bolted type usually has the cap at the back and the shell which is pressed on the shaft or spider at the front; the two members are held together by bolts. The ring-nut type of commutator has the cap in front and the shell at the back, the two members being held together by a ring nut threading on the shell which extends through to the front end of the commutator. The commutator ring nut is locked in place by a set screw. When it is necessary to repair either type, the process is very similar after the shell or cap, as the case may be, is removed.



Bolted Type Commutator

REPLACEMENT OF SEGMENTS

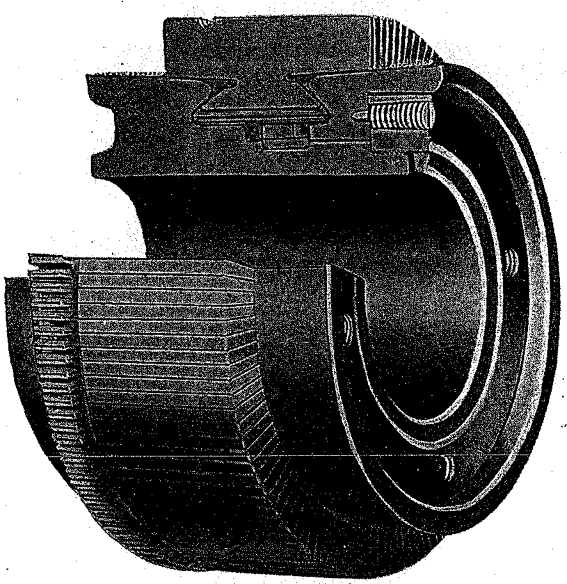
BOLTED TYPE COMMUTATOR

When replacing the copper segments in bolted type commutators, operations should proceed as follows: Remove thrust collar from shaft and wind a few turns of wire tightly around the commutator to prevent segments from separating during the removal of the shell which frequently entails more or less pounding and jarring. Remove the leads directly connected to the segments to be replaced and remove all bolts, then pull out the shell; next remove the mica cone; then take off the wire band, drive from the back and take out *one* of the segments to be replaced. A new segment should be made using the old one as a template. This should be cut from solid copper since commutator segments are not interchangeable and must be of the same bar gauge or taper as the old segment. Place the two segments together with the bottom edges or thin side even, then lay out and form the new segment from the old one, taking care that the 30-degree and 3-degree angles are exact. Insert new side mica and place the new segment in the commutator. If necessary to replace several segments, proceed, *one segment at a time*, as described above. The mica cone, if not damaged while being removed, should be put back. If it is damaged, insert a new cone. Then press the shell back on the shaft until it is approximately one inch from its original position. Insert the bolts and take them up all around a little at a time to insure that the cap at the back of the commutator is drawn up evenly as the shell is being pressed home. The commutator should next be heated with a gas ring to a temperature of 150 to 160 deg. C. and the bolts tightened while it is still hot. It is important that the segments be clamped as tightly as possible so they will not loosen in service; test for this by tapping them with a light hammer. After cooling, turn the face of the commutator and regroove if necessary.

RING-NUT TYPE COMMUTATOR

In order to replace the segments on a commutator of the ring-nut type, remove the thrust collar, band some wire around the segments, and disconnect the coil leads from the segments to be replaced. Take out the set screw and unscrew the commutator nut. Remove the cap and mica cone. Next replace the copper segments as described in the preceding paragraph; reassemble the mica cone and cap; and thread in the nut as far as possible while the commutator is cold. Heat the commutator as described above and tighten the ring nut. Turn the face and regroove if necessary.

THE REPAIR OF COMMUTATORS OF RAILWAY AND INDUSTRIAL-HAULAGE MOTORS



Ring-nut Type Commutator

TURNING THE COMMUTATOR

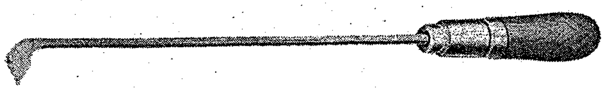
Before turning a commutator, a suitable head covering should be made to prevent chips or dust from working into the armature. This is best accomplished as follows: Take a strip of cotton several inches wider than the length of the end connections and long enough to encircle the commutator; wrap it around the commutator, binding the inside edge with cord as closely to the end connections as possible; then turn the cloth up over the latter and bind with cord to the outside of the armature. Make sure that the turning post is so set that the ways are absolutely parallel to the commutator and are fastened and braced securely. Use a side-cutting tool with point ground to about a $\frac{1}{16}$-in. radius. The cutting side and point should be given considerably more rake than is customary for working iron or steel. The tool must be sharp enough to make a clean, smooth cut without dragging copper over the mica.

While turning, the commutator surface should be run at a speed of approximately 300 feet per minute. This is about as fast as a tool will cut without burning. It is important to round off the ends of the copper segments to at least a $\frac{1}{16}$-in. radius with a file while the commutator is in the lathe. If this is not done and sharp corners are left at the ends of the copper segments the mica is easily broken out and a short circuit may be established by oil and dust at these points.

GROOVING THE COMMUTATOR

After turning the commutator, the side mica should be grooved to a depth of approximately $\frac{3}{64}$ of an inch. Refer to G-E Railway Supplies Catalog No. 6002, pages 303 and 304, for commutator grooving machines, and to Descriptive Sheet No. 64407; also to GEA-774 for care of commutators.

The finishing of the slots left by the grooving saw is an important operation because good commutation and brush wear depend very much upon the condition in which the commutator goes into service. The hand scraper, Cat. No. 775854G1, illustrated below is used for removing mica fins which are left in the slot by the grooving saw. The grooving saw is usually 0.005 in. less in thickness than the mica between the commutator segments. The grooving saw generally cuts into the copper and leaves projections which must be removed. A curved triangular file is sometimes used for removing these copper projections, but the removal of a very thin portion of the commutator surface



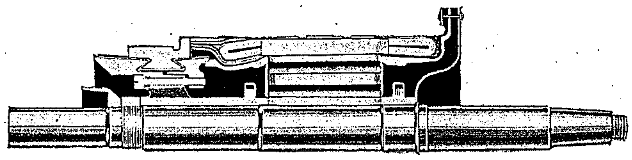
Commutator Hand Scraper

by turning in a lathe is recommended. For this final turning, a special high-speed steel tool (No. 3 Stellite or equivalent) will give good results. The remaining copper burr which projects into the slot on the trailing edge of each commutator segment can be removed by the hand scraper. Final polishing with sandpaper will make the smooth surface necessary for good commutation and long brush life.

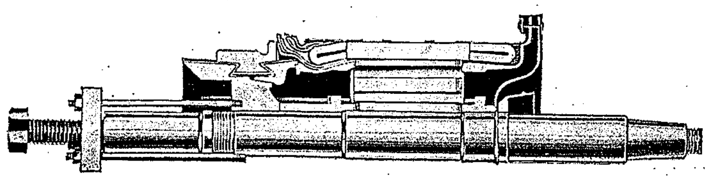
THE REPAIR OF COMMUTATORS OF RAILWAY AND INDUSTRIAL-HAULAGE MOTORS

In making repairs to the commutator, care should be taken to keep all parts clean and free from dust and foreign material. Careful work is essential for the best results.

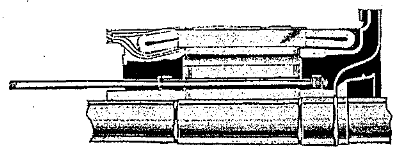
VIEWS OF ARMATURE WITH BOLTED TYPE COMMUTATOR



Normal Operating Condition

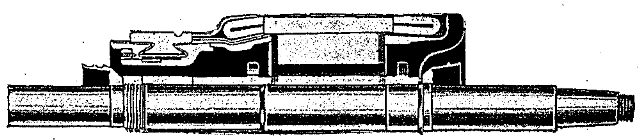


Showing Bolts for Removing Commutator

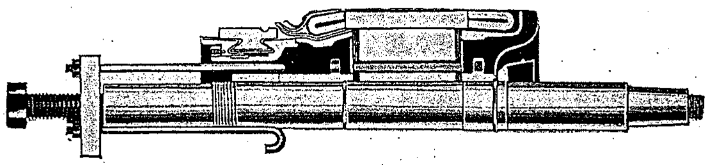


Commutator Removed

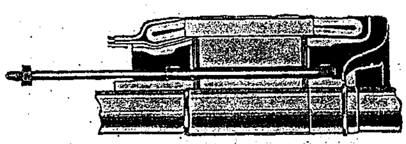
VIEWS OF ARMATURE WITH RING-NUT TYPE COMMUTATOR



Normal Operating Condition



Showing Hook Bolts for Removing Commutator



Commutator Removed

REMOVAL AND REPLACEMENT OF COMPLETE COMMUTATORS

In case the commutator as a whole must be replaced, the above illustrations show methods of removing it from the shaft for both the bolted and the ring-nut types of commutators. Note the bolts used for clamping the core laminations together while the armature nut is removed.

ARMATURE (Cont'd)

WINDINGS (Cont'd)

Large motors which usually are wound with one turn coils use bars, consisting of strip copper annealed, slotted, and pressed to proper shape.

There are two styles of bars used:—the cross-over bar and folded bar. In the first the sections of the bar are crossed over one another and a strip of mica inserted at the cross-over. The folded bar consists of two separate bars each folded and the two fitted together to form one conductor. The purpose of the slotting and shaping of the bars in this way is to reduce to a minimum the eddy current loss in the copper.

Ears are tinned, clips fitted on, bent to the correct form, taped and bound together in poly-coils.

Mica and asbestos or varnished cambric and cotton are used as insulating materials depending upon whether class A or class B material is required.

ASSEMBLY

Pinion end armature heads are shrunk on the shaft.

Laminations are pressed in position on the shaft, the slots being aligned by means of the keyway punched in the laminations. See Photograph 249046, Page 61.

Commutator end armature heads are pressed tight against the laminations.

In some cases a nut is screwed on the shaft tight against this head.

In other cases the commutator is pressed against the head and a nut screwed tightly against it.

In some of the larger motors these parts are mounted on a spider which is shrunk and keyed to the shaft. See Photograph 249049, Page 62.

Before assembling the armature coils the armature heads are thoroughly insulated.

The leads are soldered to the proper commutator ears, and insulation wrapped around the end connections of the armature coils. This insulation is bound tightly with cord and painted with insulating compound over which the canvas head dressing is placed.

After assembly the armature is heated and, while hot, temporary binding bands are applied. This forces the armature coils into proper position.

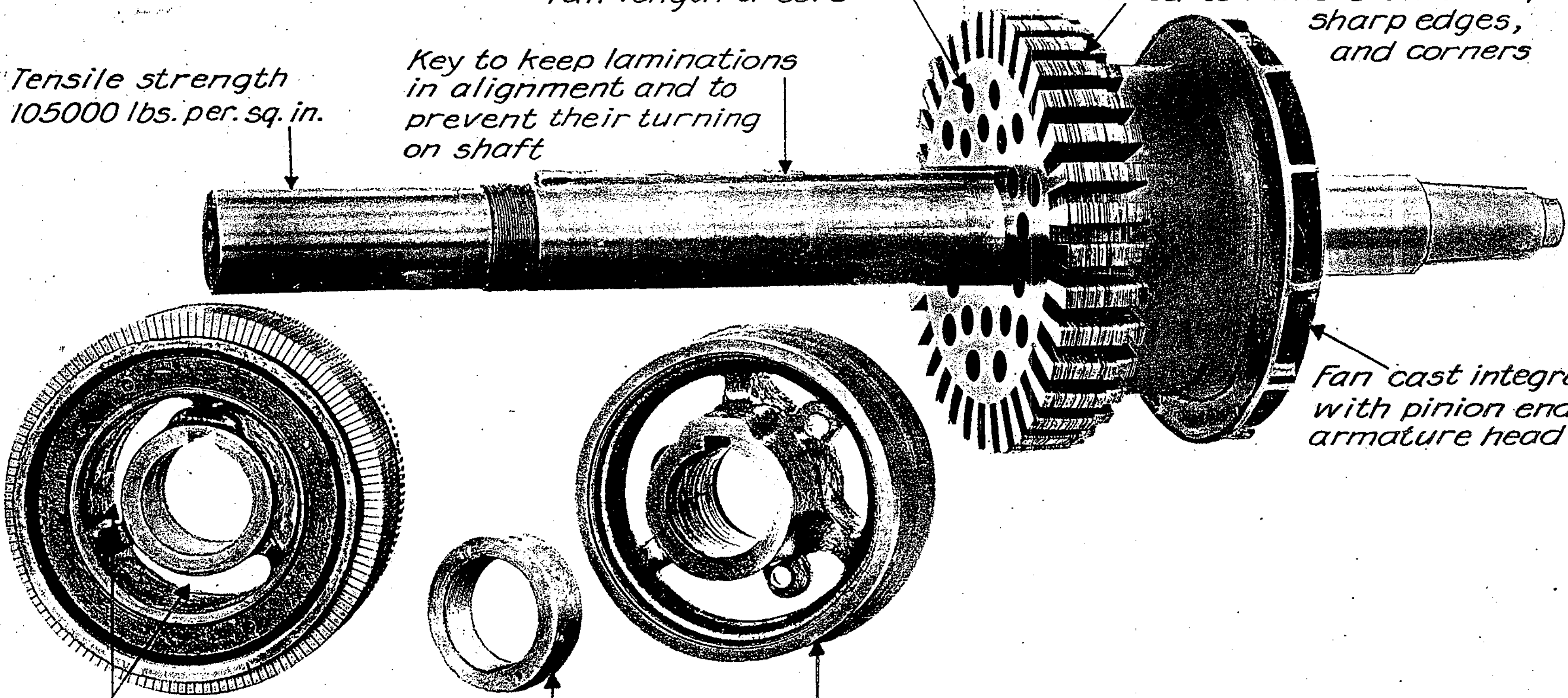
When cold, these bands are removed and permanent bands of tinned steel wire are wound into the recesses provided for them. Wire bands also secure the head dressing. They are clipped at intervals about the circumference and are completely soldered to the bands.

The armature is dipped in insulating varnish and baked after which the brush surface is trued and the commutator is slotted.

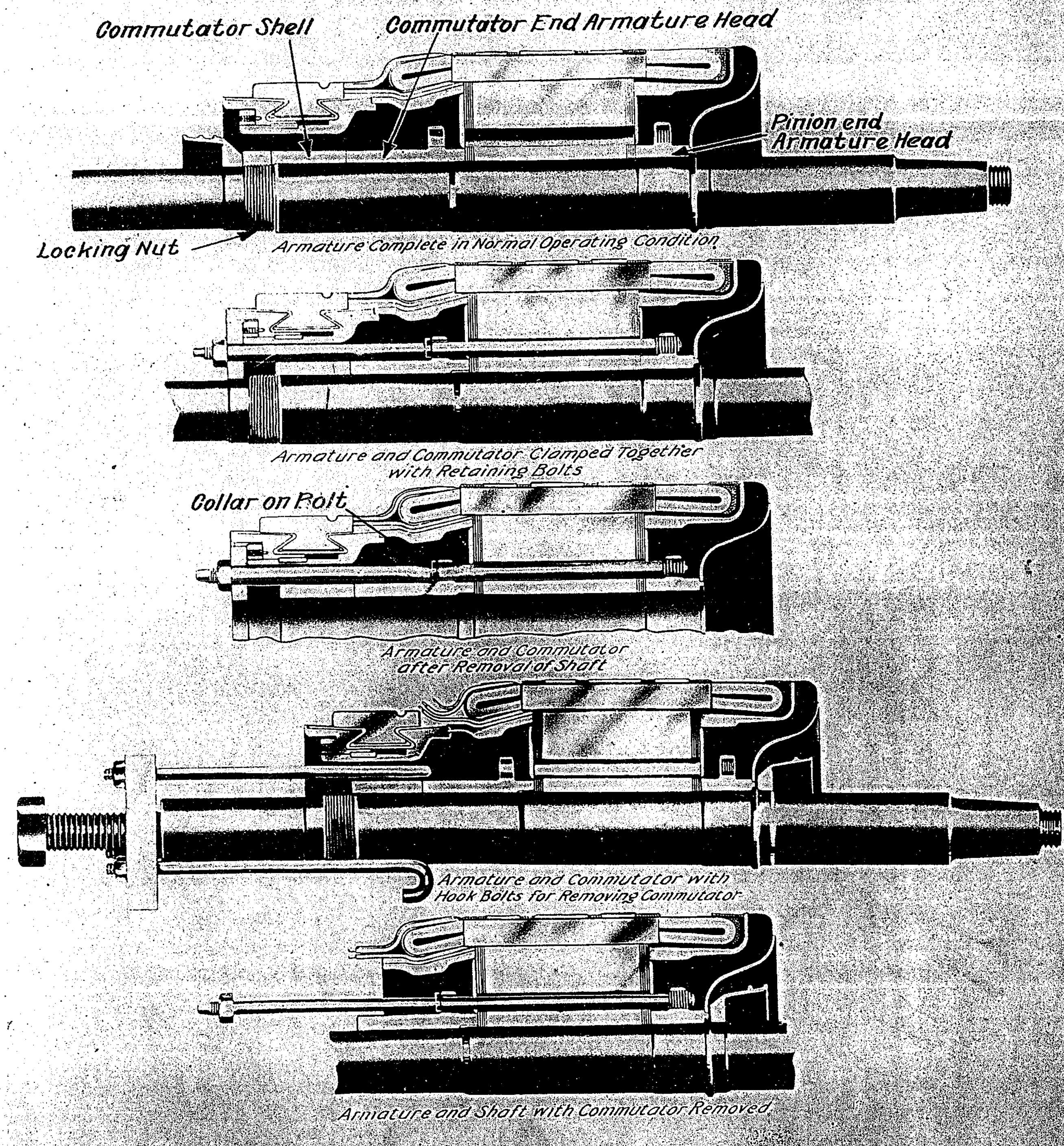
In the case of bar windings usually the top bars and bottom bars with clips attached are assembled on the core after which the top bars are soldered, brazed or riveted to the clips, and insulated.

General construction of railway motor armatures is shown on the following photographs:

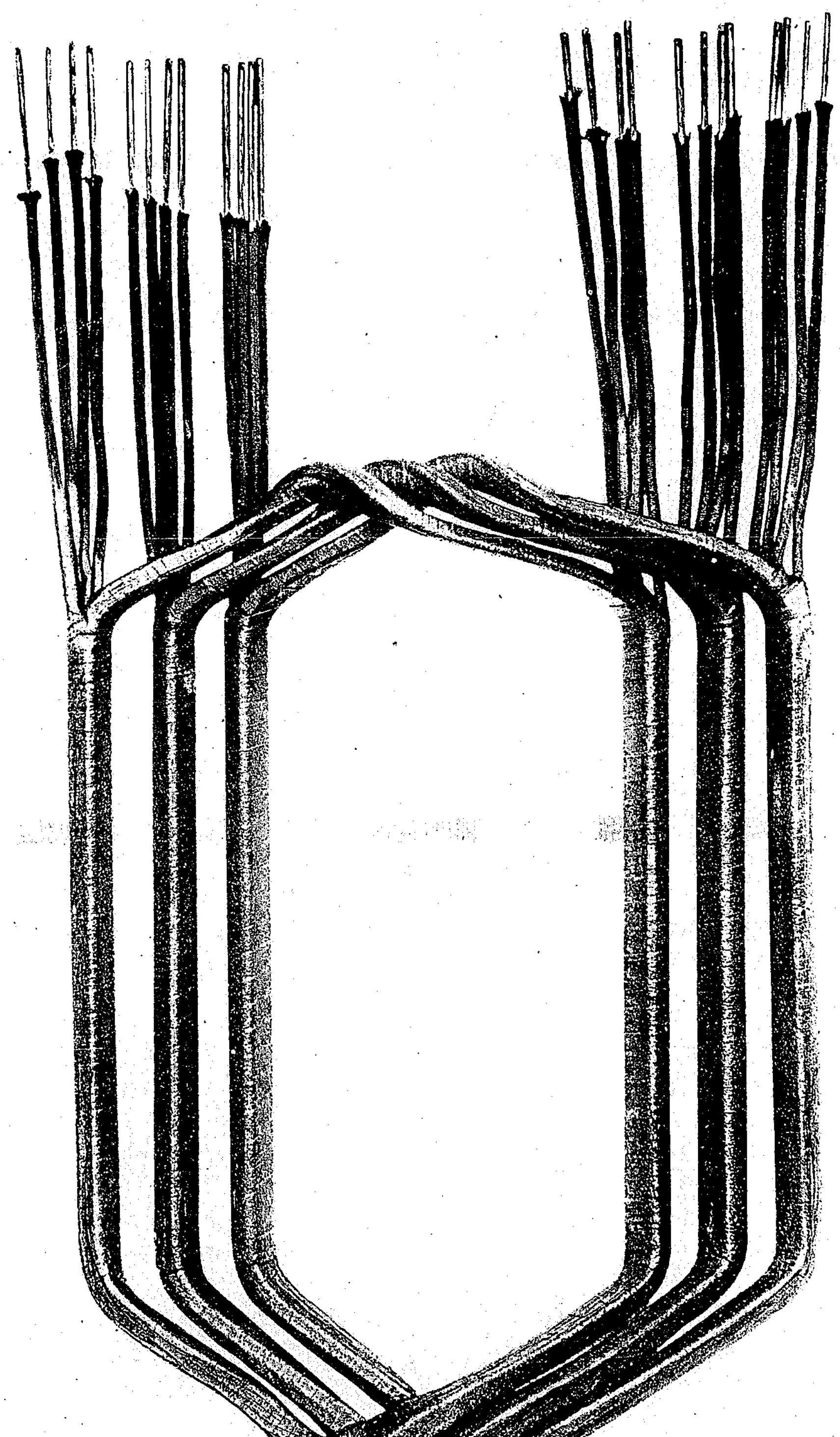
249078—Page 63
249079—Page 64
247238—Page 65
244085—Page 66
246737—Page 67



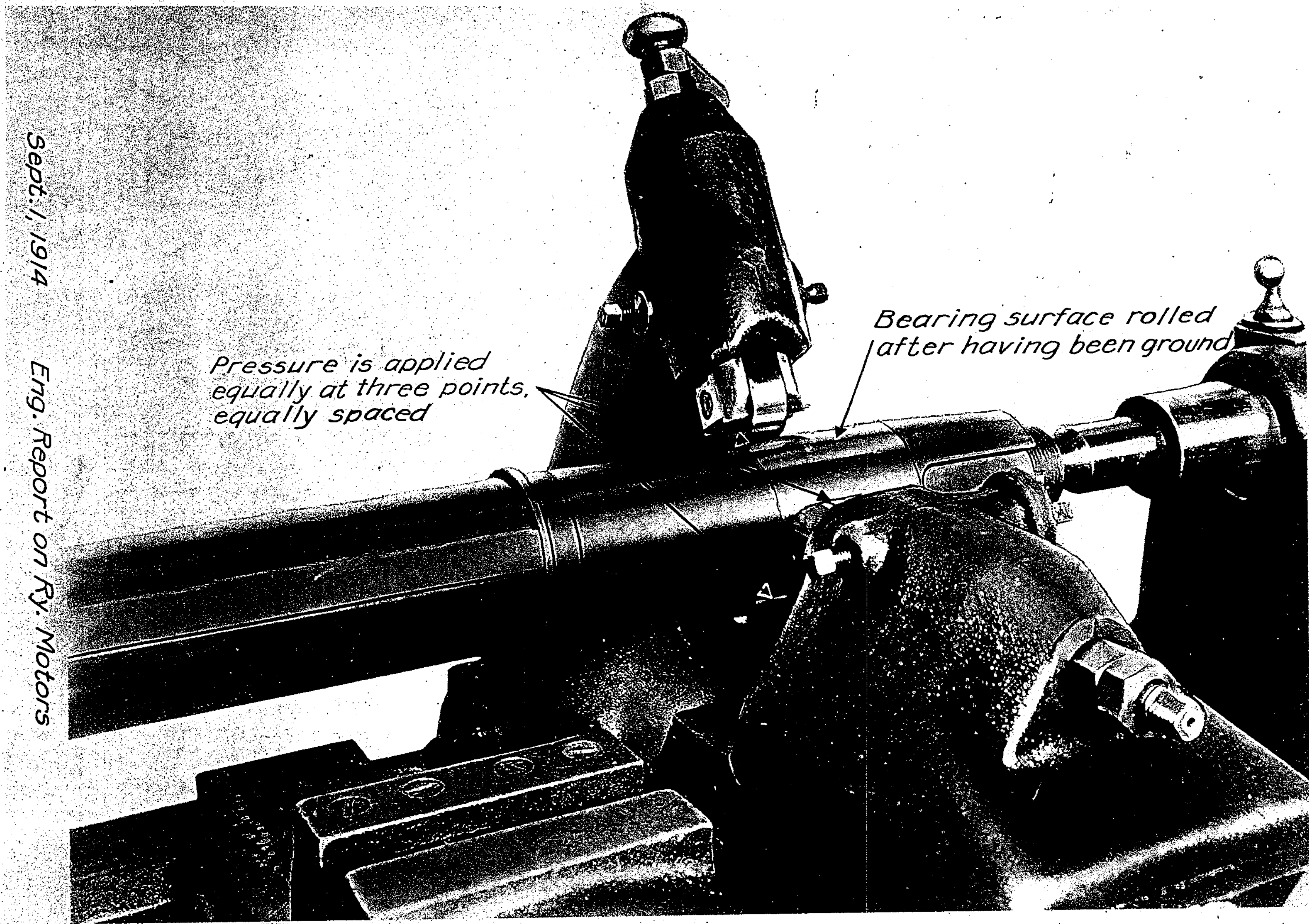
Armature Construction GE Railway Motor



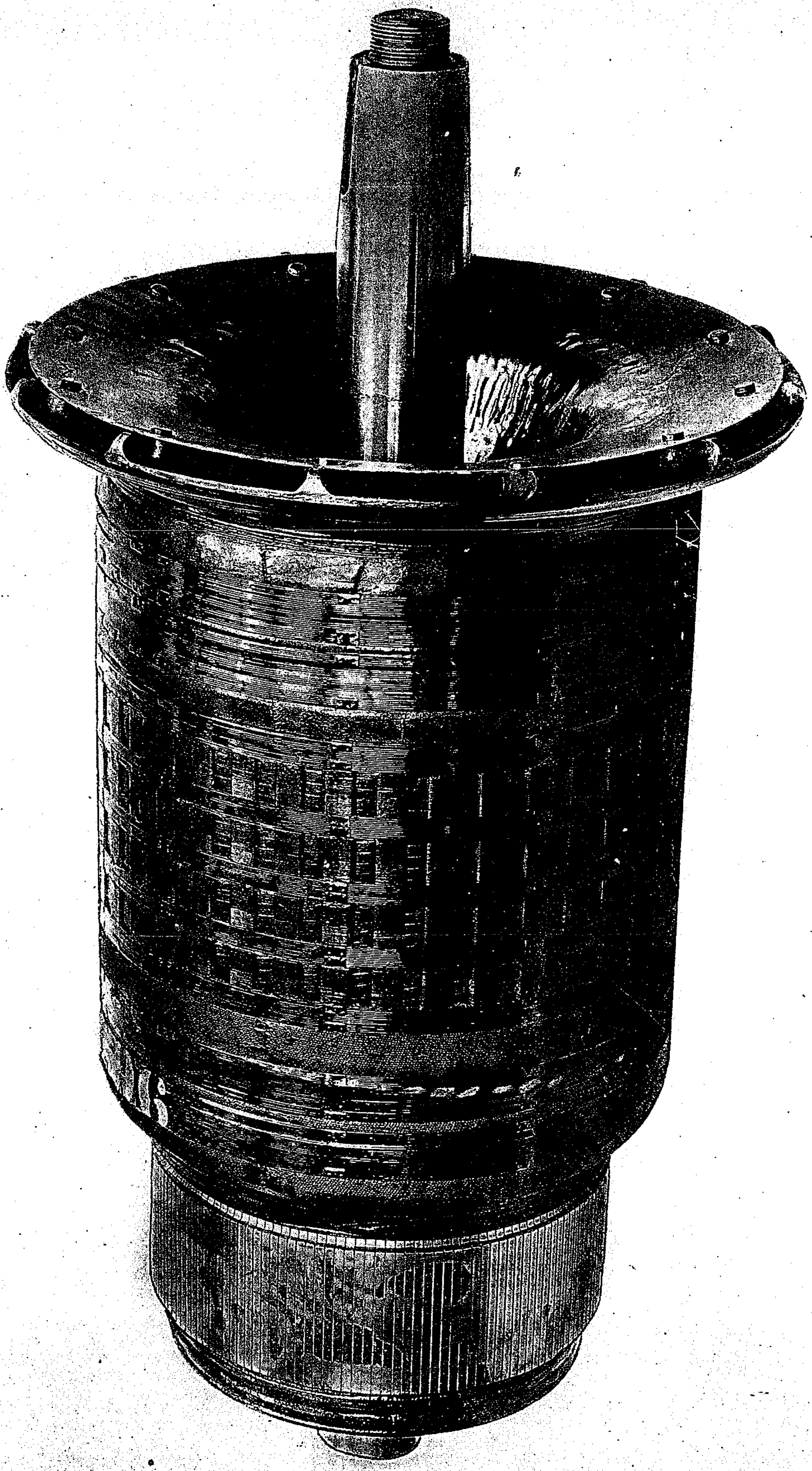
Armature Cross Section of G E Railway Motor showing method of removing Shaft and Commutator without Disturbing Windings



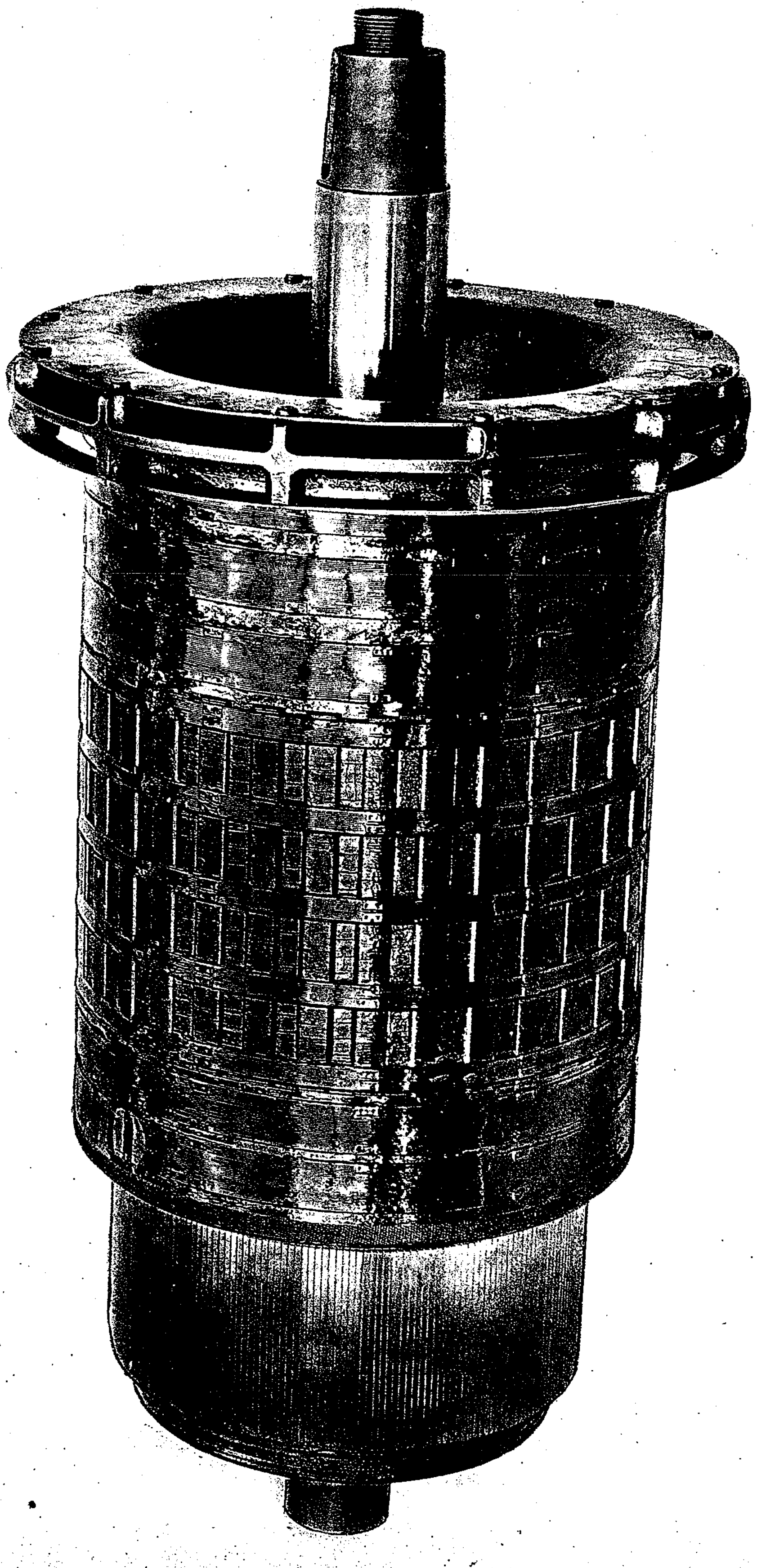
Armature Coil of GE Railway Motor
Sept. 1, 1914 Eng. Report on Ry. Motors



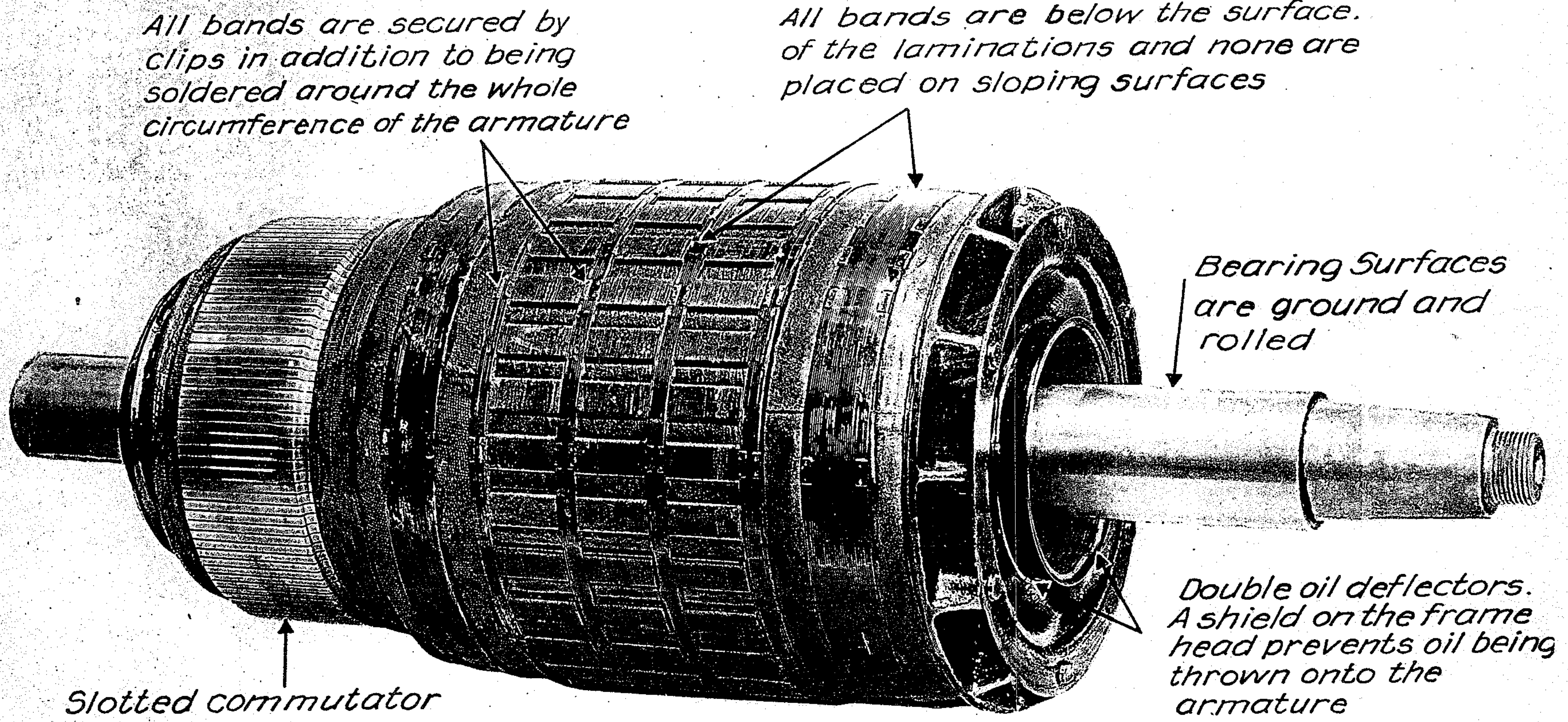
Armature Construction, Rolling The Bearing Surfaces of Armature Shafts For Railway Motors



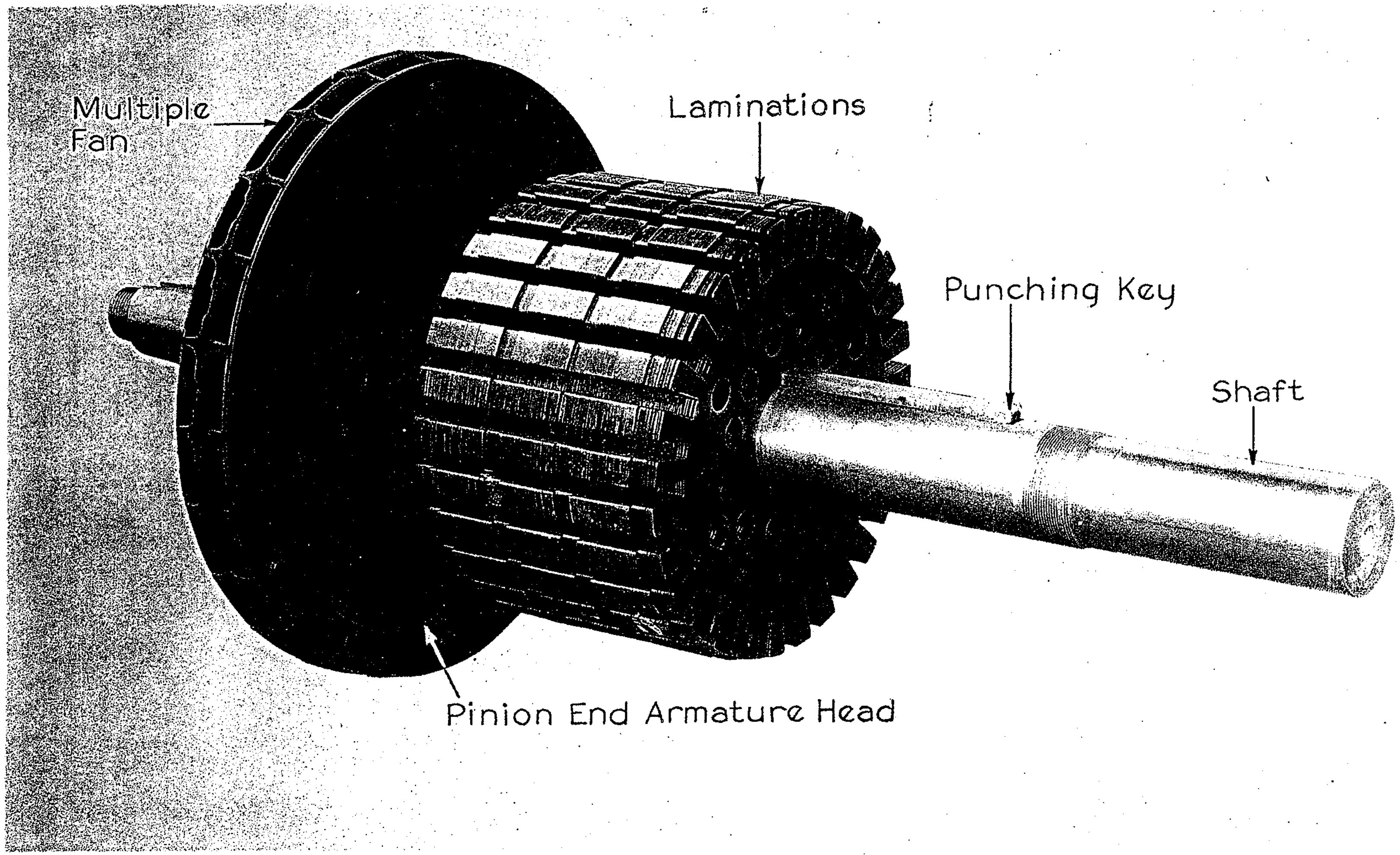
GE-237 Railway Motor Armature with Large Series Fan



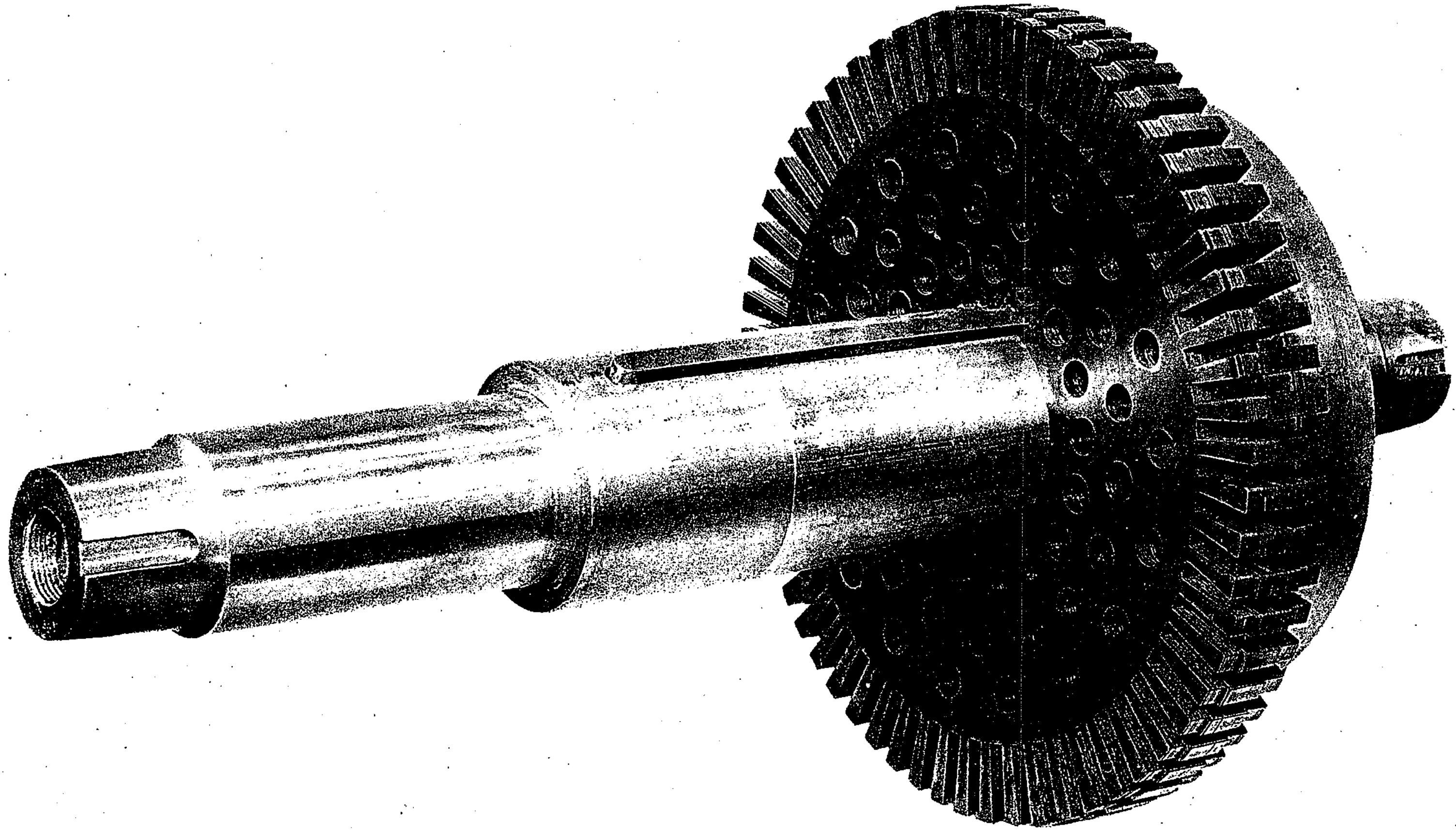
GE-248 Railway Motor Armature with Multiple Fan



GE-201 Railway Motor Armature

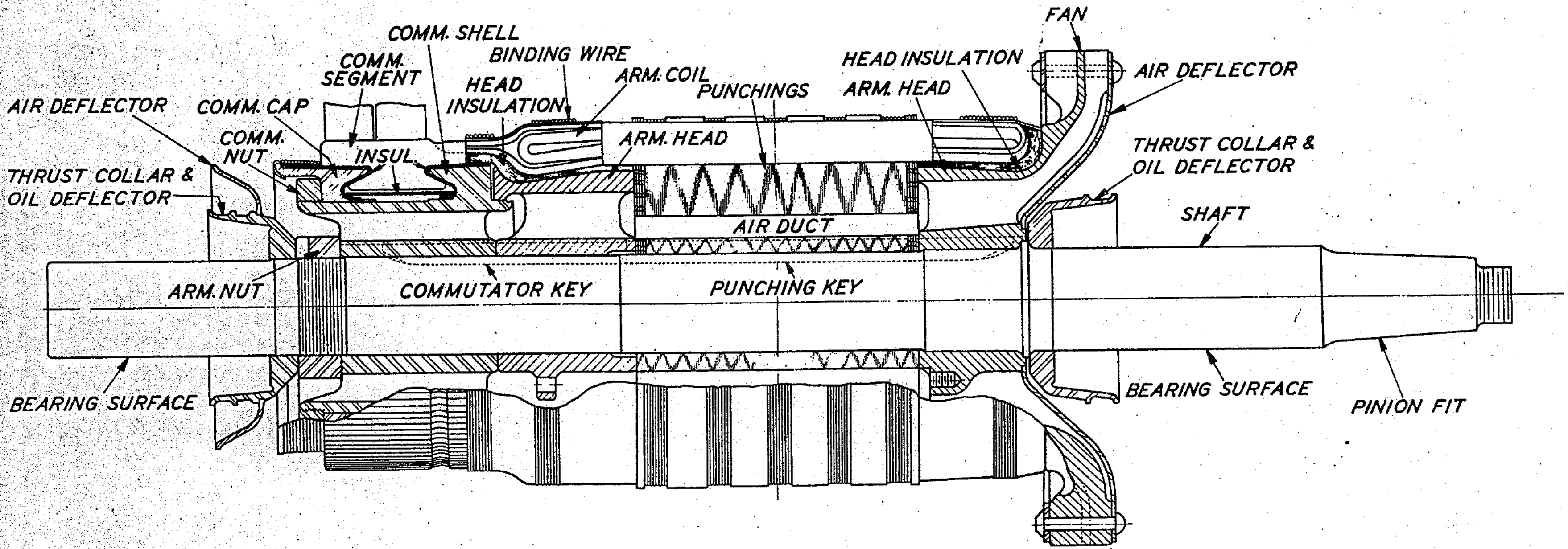


249046 RAILWAY-MOTOR ARMATURE CORE PARTIALLY ASSEMBLED. APPROX. 1/6 SIZE.
E315.1 10-17-28



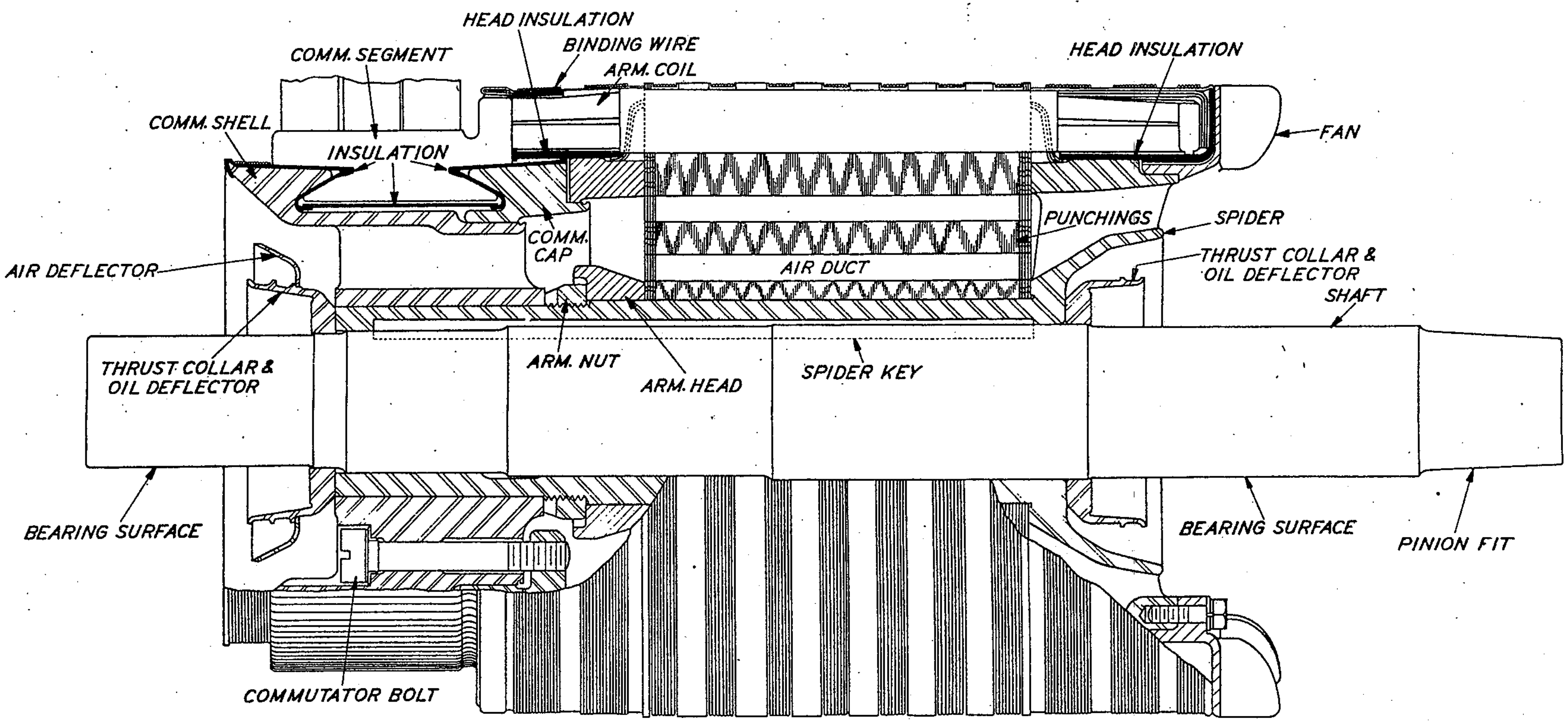
249049 GE RAILWAY MOTOR. ARMATURE CORE, PARTIALLY ASSEMBLED.
(SPIDER TYPE) APPROX. 1/6 SIZE.

E315.1 10-24-28



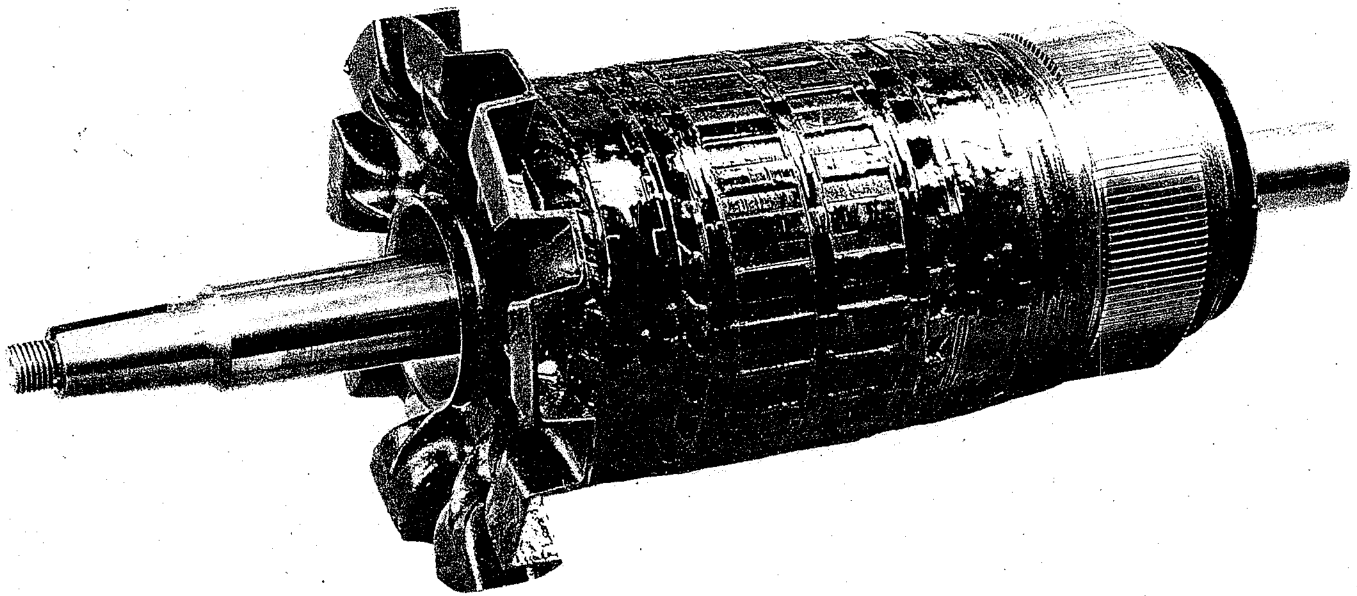
249078 GE RAILWAY MOTOR. ARMATURE ASSEMBLY. PUNCHINGS MOUNTED DIRECTLY ON SHAFT.

E315.1 10-26-28



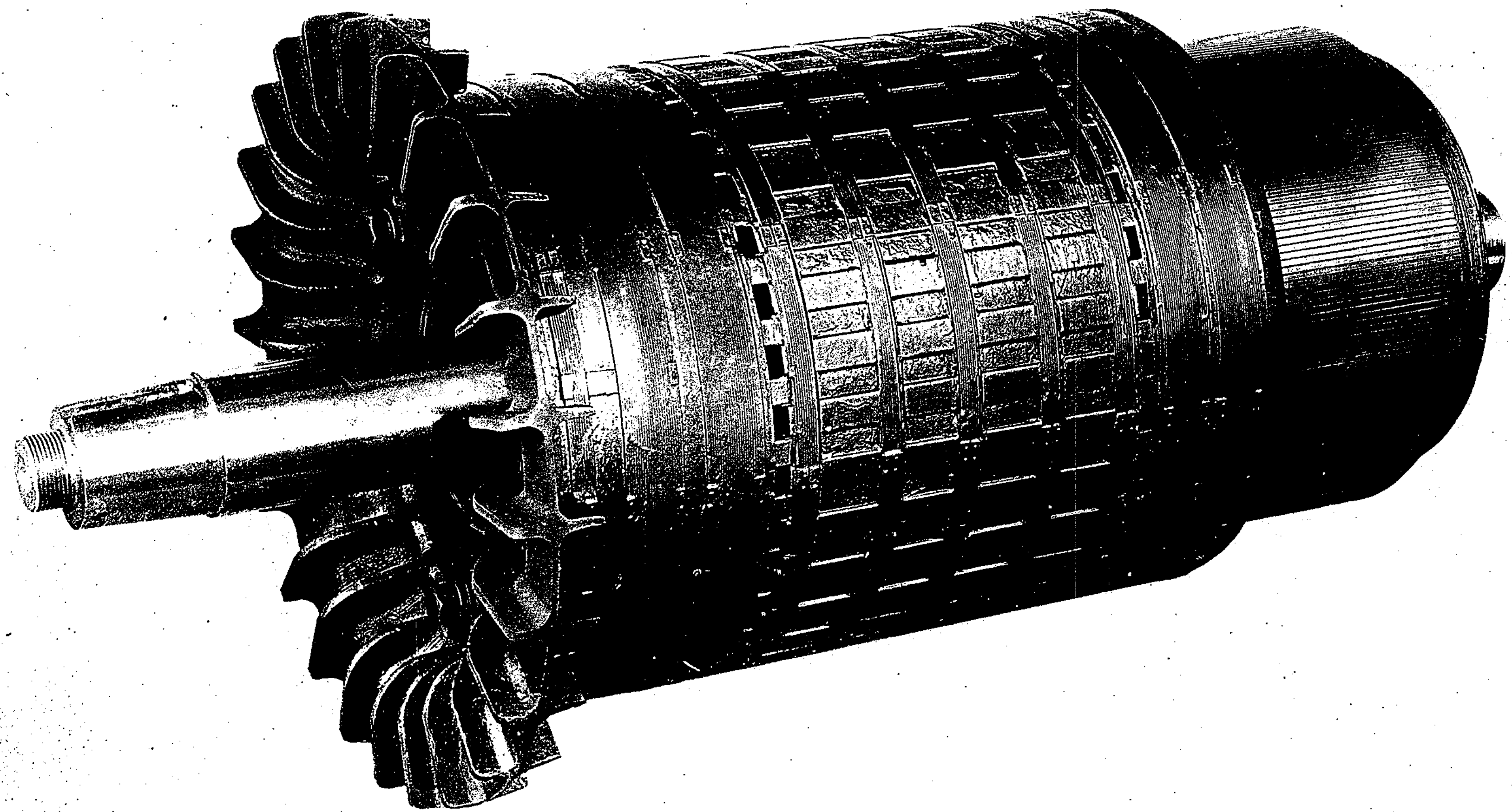
249079 GE RAILWAY MOTOR. ARMATURE ASSEMBLY. PUNCHINGS MOUNTED ON SPIDER.

E315.1 10-26-28



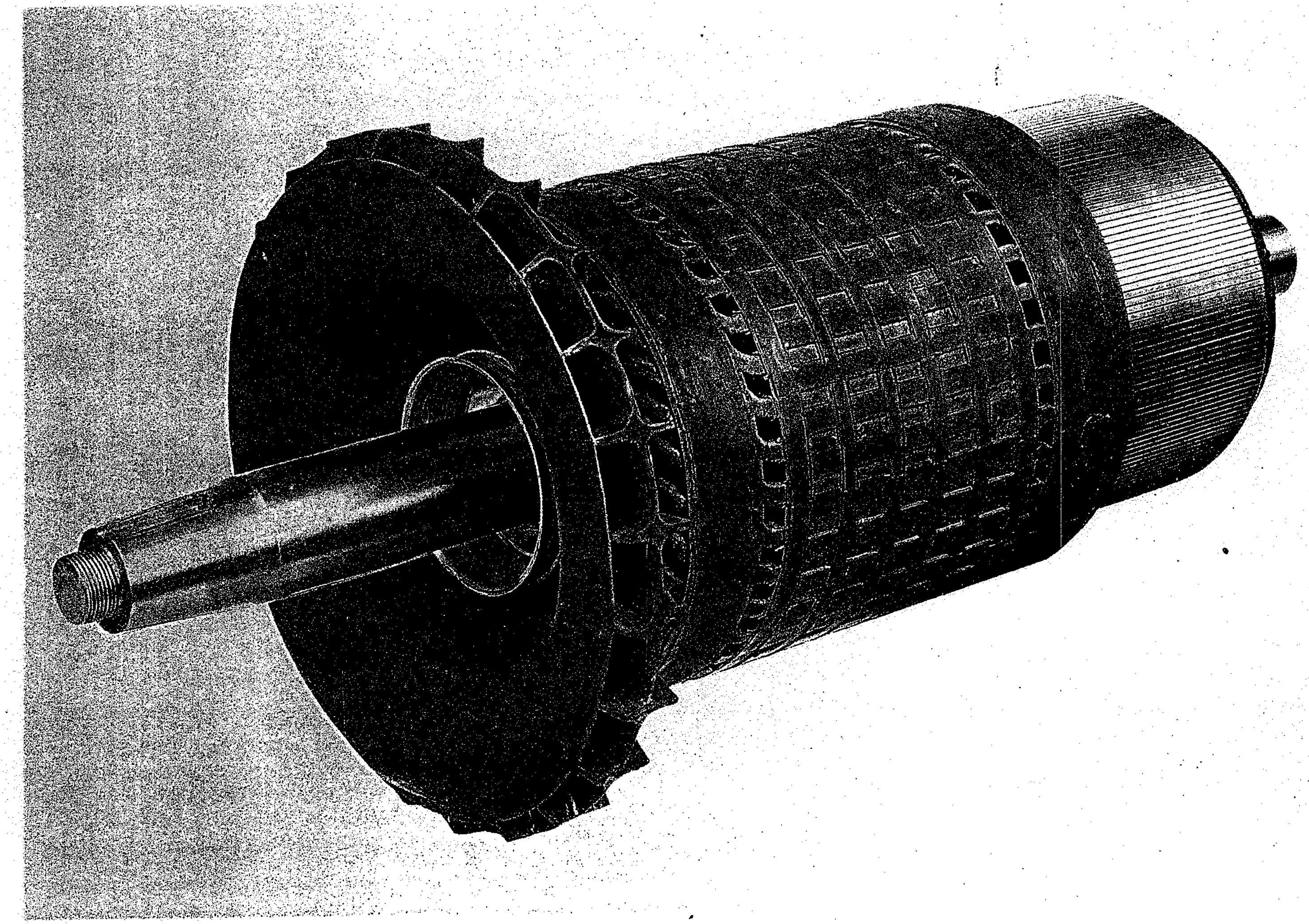
247238 G-E. RAILWAY MOTOR ARMATURE

E-315.1 8-2-27

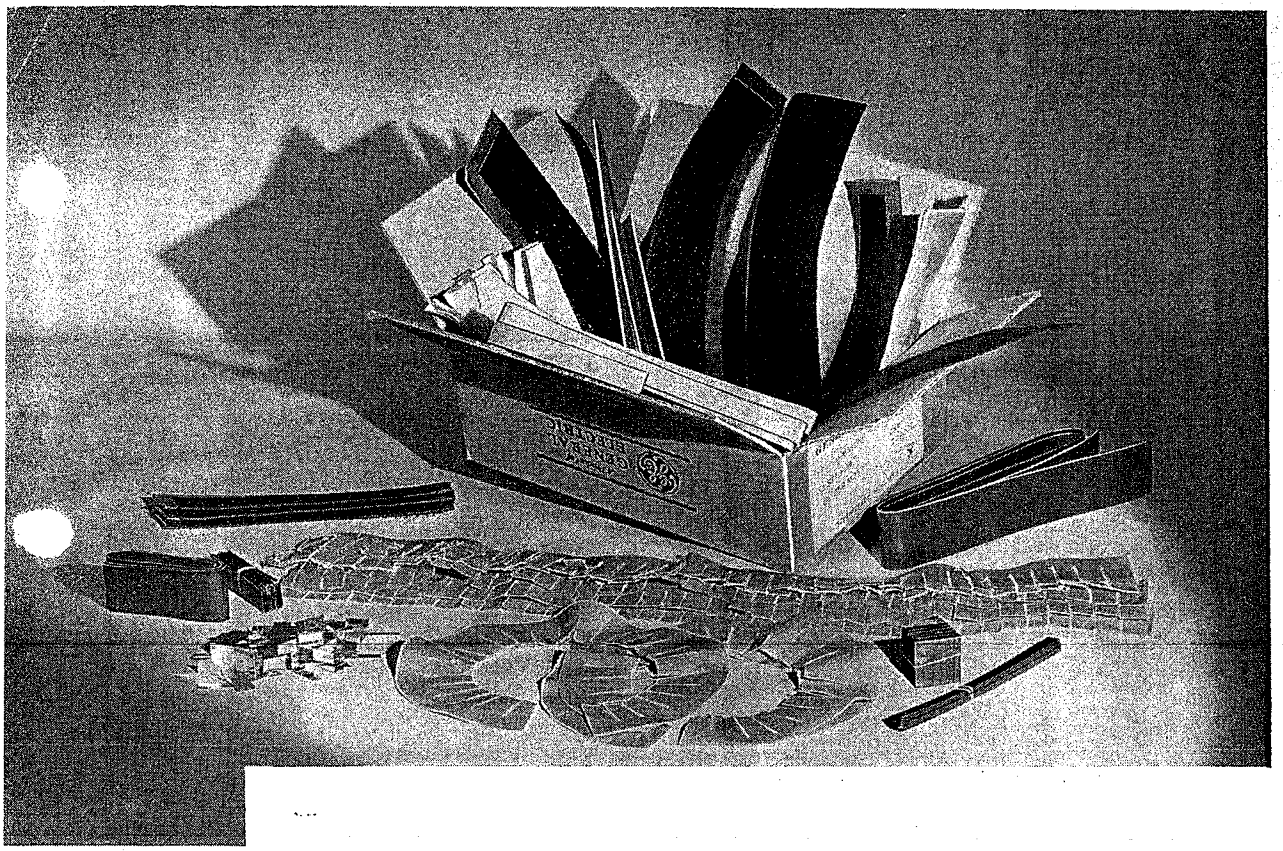


244085 G-E. RAILWAY MOTOR ARMATURE

INDEX E-315.1



246737 G-E. RAILWAY MOTOR ARMATURE
APPROX. 1/5 SIZE INDEX E-315.1 3 22 27



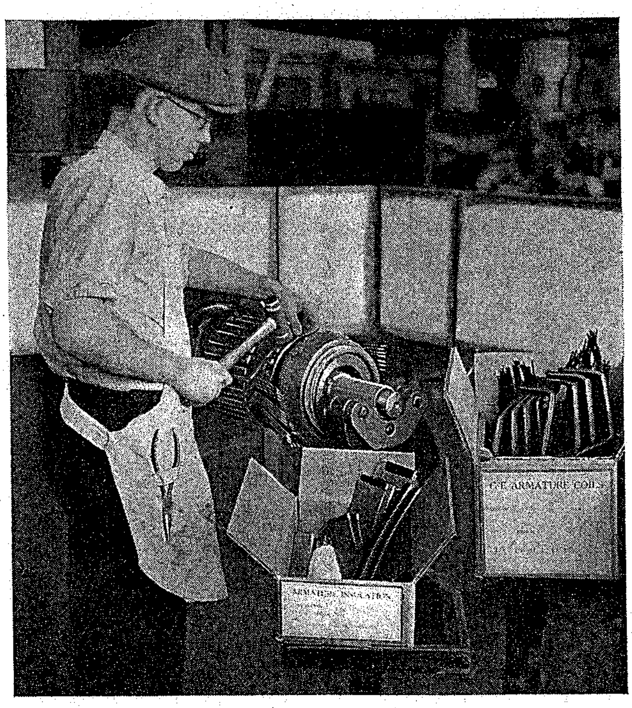
READY-TO-USE ARMATURE INSULATIONS for Car and Locomotive Motors

G-E packaged armature insulations, cut to shape and ready to use, are available for all G-E transportation motors. Each package contains all the insulating materials and clips necessary for completely rewinding one armature. Users have found that this modern method of armature repair enables them to make substantial savings over the use of bulk insulations.

ADVANTAGES
Of Ready-to-use Armature Insulation

1. Waste of materials is prevented—no cutting is required.
2. Proper fit is assured—the materials are cut at the factory exactly as they would be for a new armature.
3. Omission of essential parts is prevented—*all* the necessary materials are on hand when needed.
4. Time is saved both in the stock room and at the winder's bench—no detailed stock orders, no cutting of materials.
5. Longer-life armature rewinds are assured. From every standpoint—quality of materials, accuracy of fit, and completeness—a factory job can be done in your repair shop. Thus, you give your rewound armatures new-equipment standards of insulation reliability.
6. Sealed in cartons, the insulation can be stocked without appreciable deterioration.
7. Your repair men can apply G-E ready-to-use insulations properly. Individual instruction charts are available for most of the types of G-E motors in service. Each includes a list of materials and an illustration showing where each piece of insulation is applied.

READY



TO USE

Before you buy insulating materials, consider this important fact: a great many rewound armatures fail because the rewinding insulations are of inferior quality. Forty years of research have enabled General Electric to develop insulating materials of the highest reliability in the industry. It is real economy to buy the best.

(5M)
No. 9240

CLASSES OF INSULATION

From the A.I.E.E. standards the various classes of insulation are defined as follows:

Class A Insulation: Class A insulation consists of cotton, silk, paper, and similar organic materials when impregnated*; also enamel as applied to conductors.

Class B Insulation: Class B insulation consists of inorganic materials such as mica and asbestos in built-up form combined with binding substances. If Class A material is used in small quantities in conjunction for structural purposes only, the combined material may be considered as Class B, provided the electrical and mechanical properties of the insulated winding are not impaired by the application of the temperature permitted for Class B material. (The word "impair" is here used in the sense of causing any change which could disqualify the insulating material for continuous service).

BRUSH HOLDERS AND BRUSHES

Brush holders are designated as either end- or side-supported, according to their position on the motor frame.

Brush holder studs for smaller motors are clamped tightly by cleat supports bolted to machined bosses on the frame. See Photograph 247349, Page 69.

The studs are bolted directly to the frame in the larger motors. See Photograph 248092, Page 70.

Studs are mica insulated. In the cleat type of holder mica cylinders protected by brass sleeves are used. The bolted type studs have a press fit in mica insulated holes in the support. In either case smooth sided porcelain insulators are used to obtain proper creepage between the brush holder body or support and frame.

In the cleat type the cleat itself is the support, but in the bolted stud type the support is a separate malleable iron casting or drop forging, to which the body is bolted.

Modern brush holder bodies are of brass alloy, and have either renewable or non-renewable carbonways. A renewable carbonway is a separate brass casting bolted to the body casting itself. See Photograph 247349, Page 69, and 248092, Page 70.

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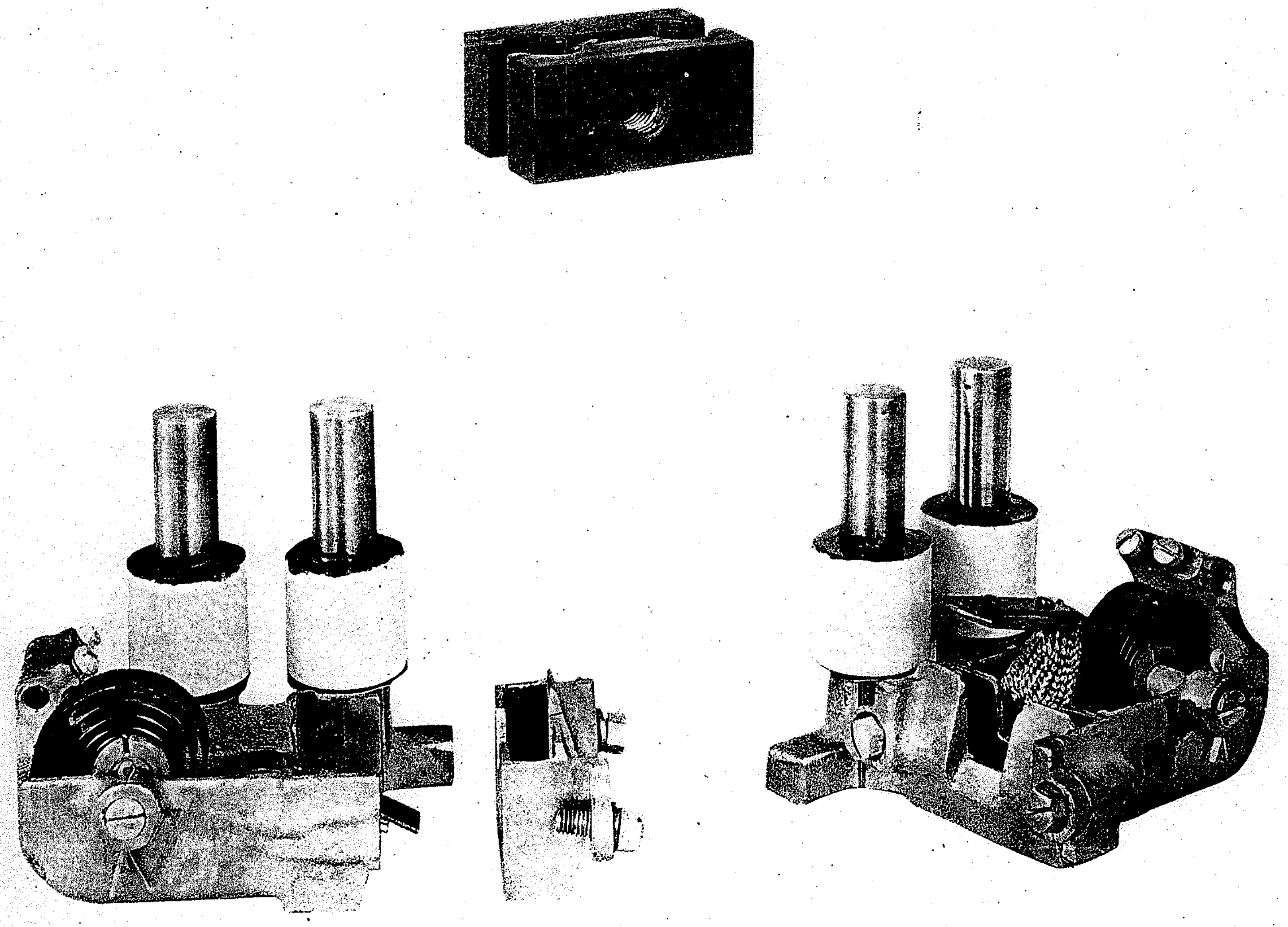
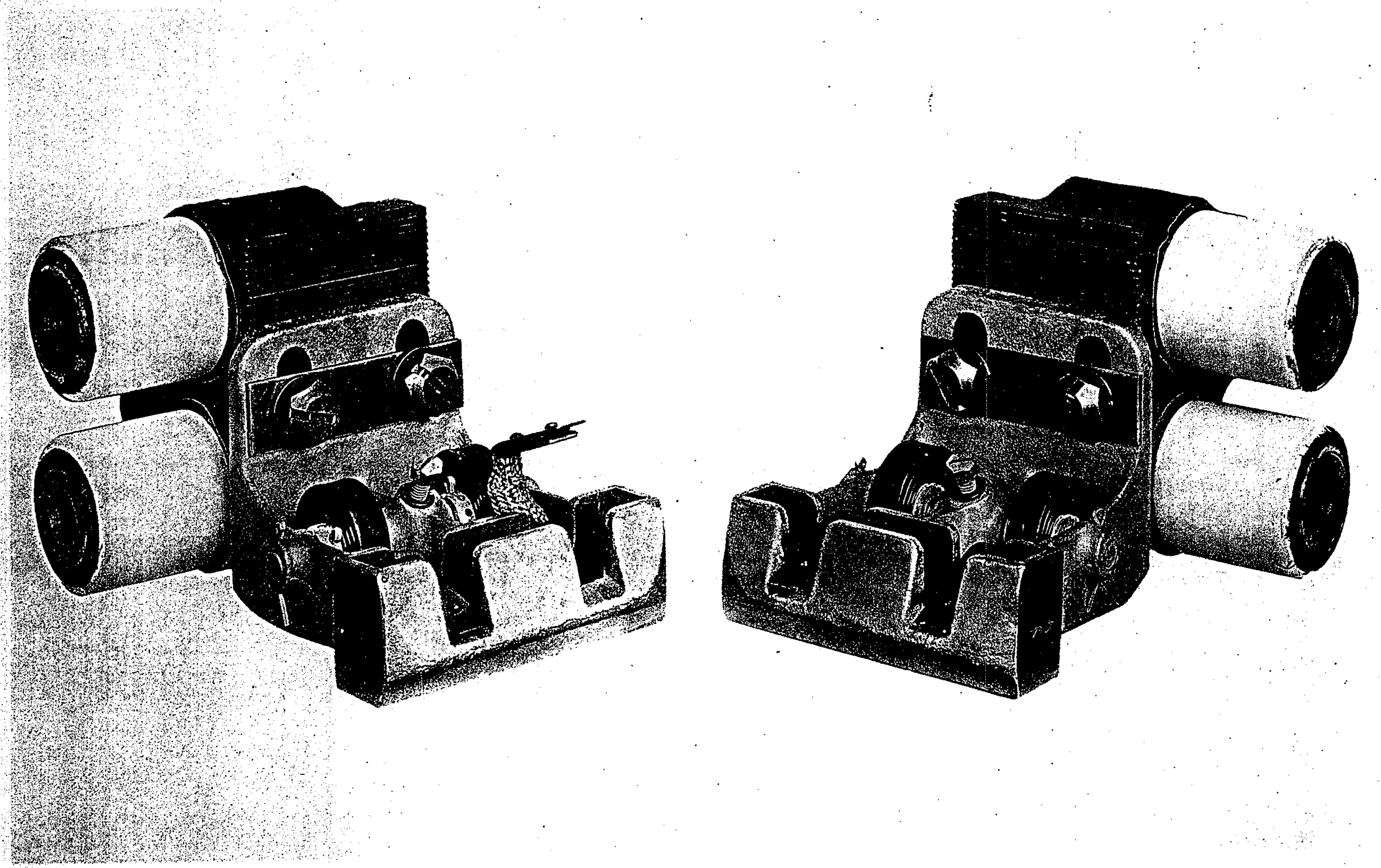
*Impregnated Cotton, Paper, or Silk: An insulation is considered to be "impregnated" when a suitable substance replaces the air between its fibres, even if this substance does not completely fill the spaces between the insulated conductors. The impregnating substance, in order to be considered suitable, must have good insulating properties; must cover the fibers and render them adherent to each other and to the conductor.



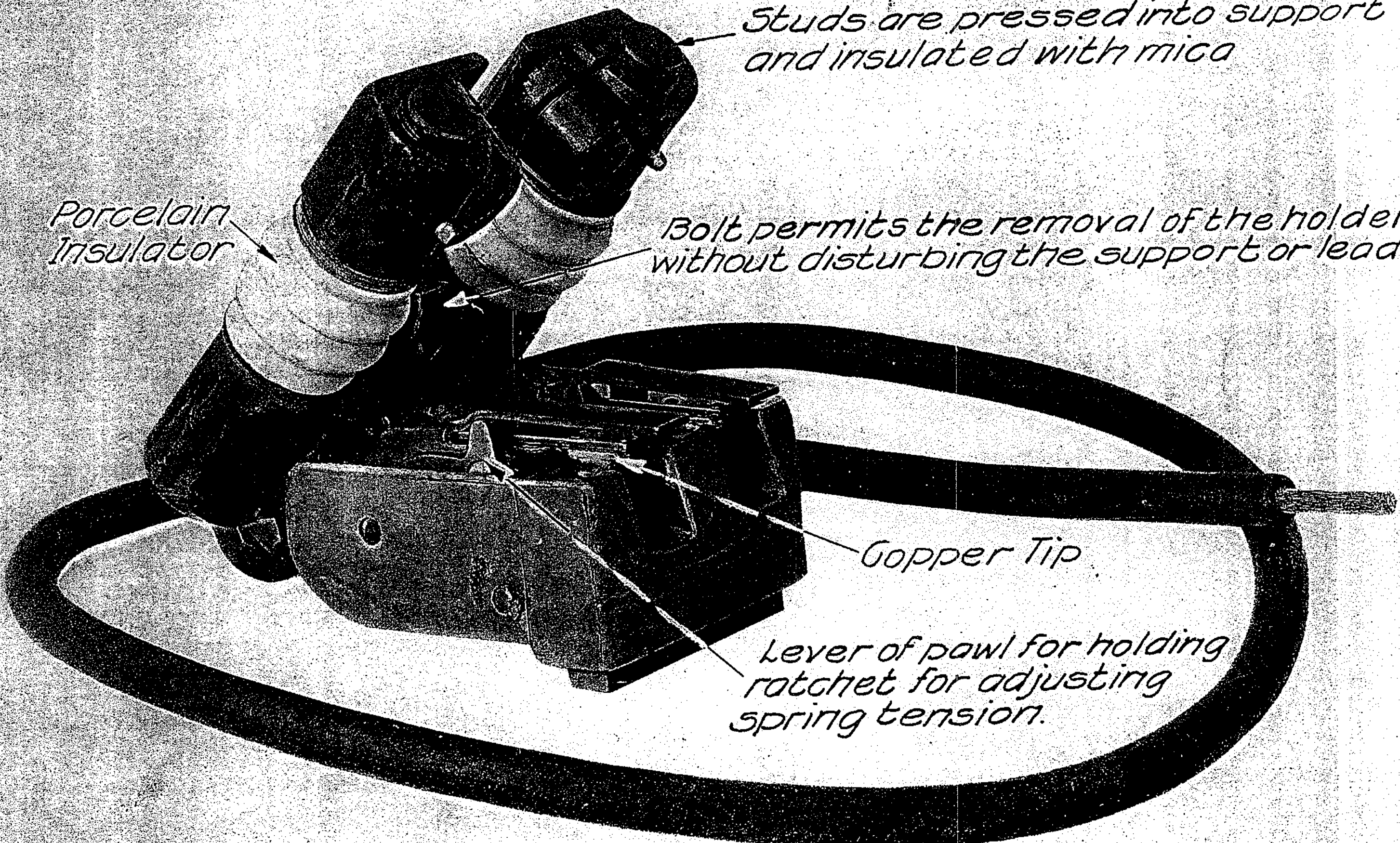
247349 G-E. RAILWAY MOTOR BRUSH-HOLDERS - CLEAT TYPE SUPPORT

E-315.1 9-1-27

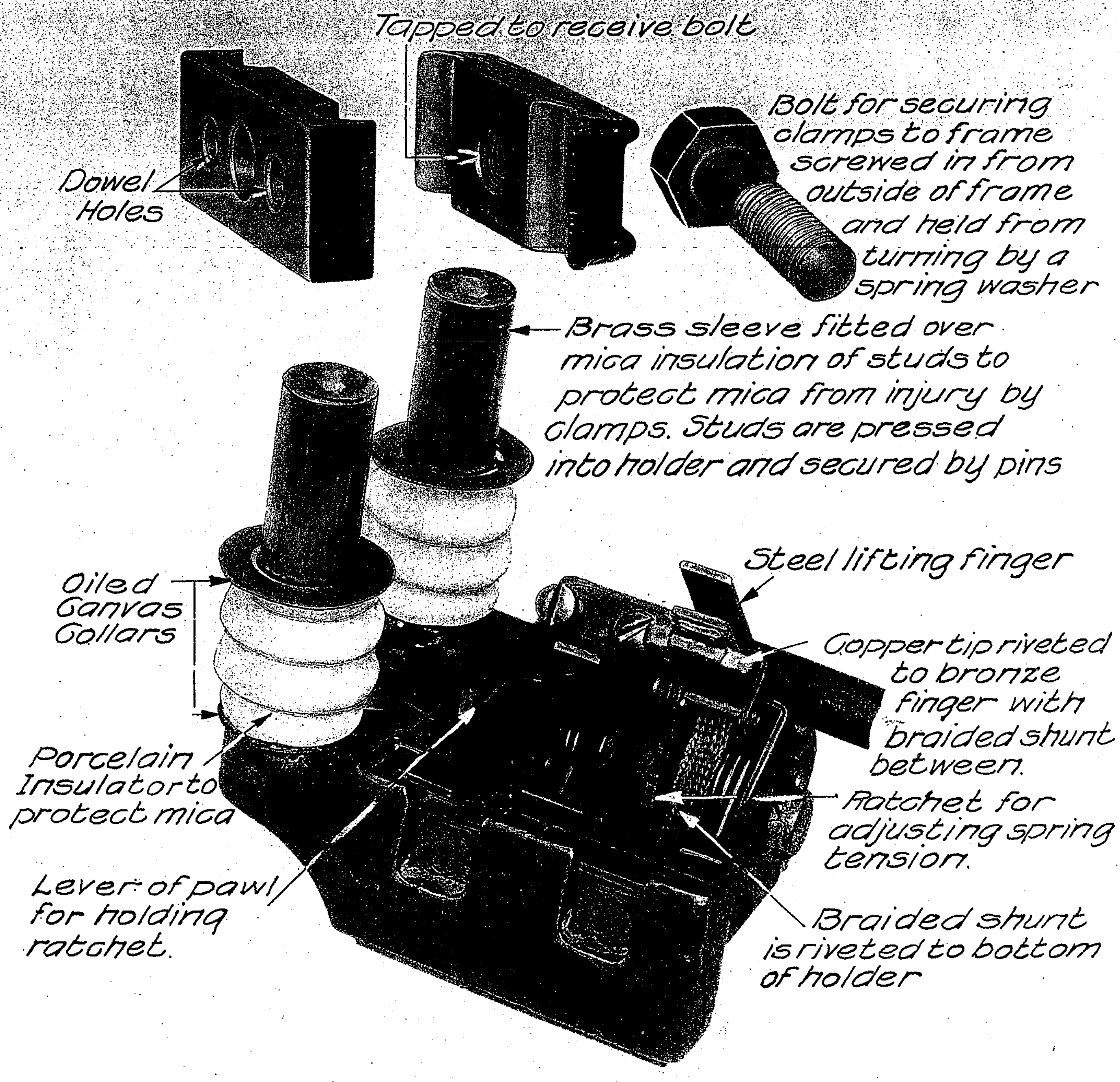


248092 G-E. RAILWAY MOTOR BRUSH-HOLDERS - STUD TYPE SUPPORT

E-315.1 1-24-28

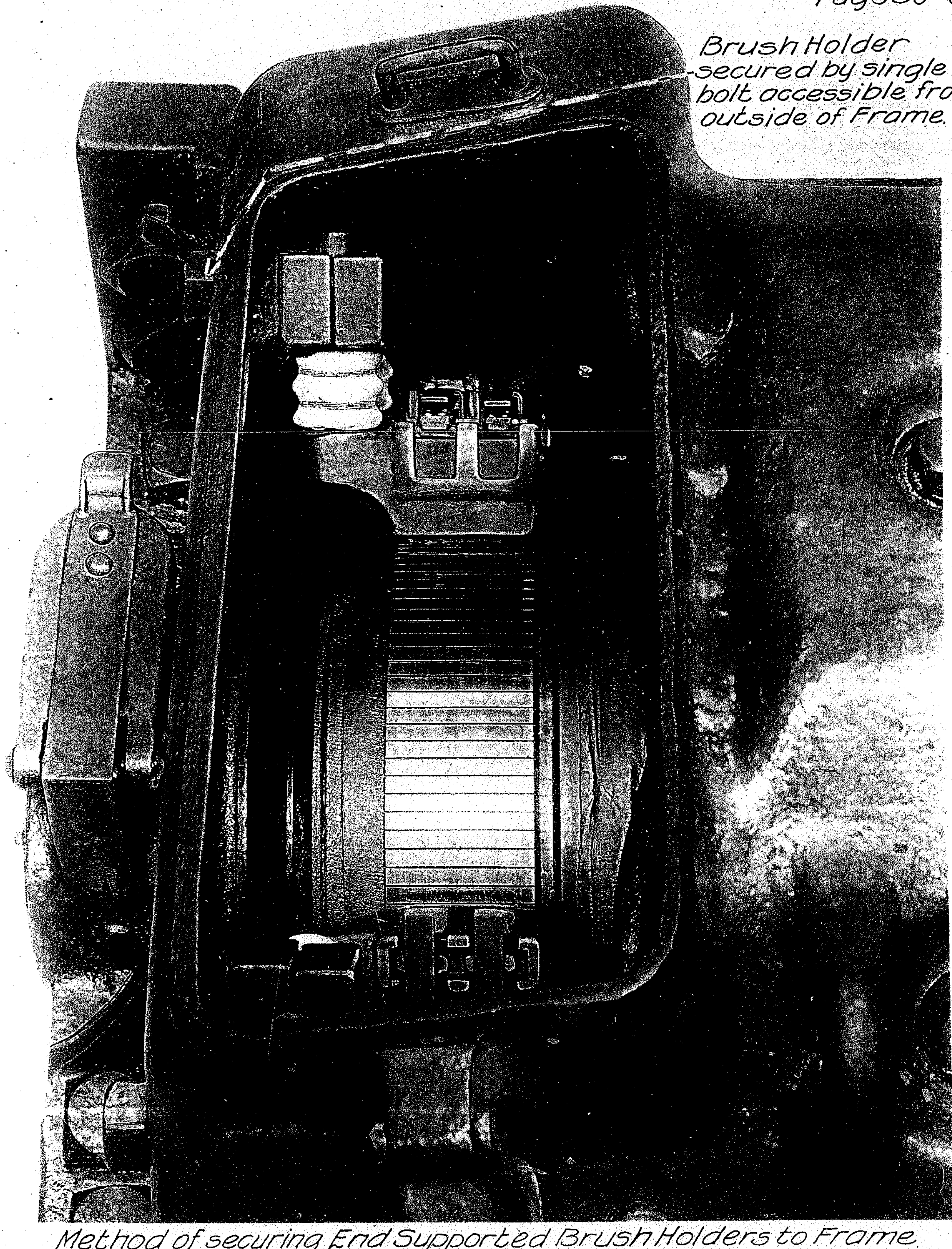


Side Supported Brush Holder for G E Railway Motors

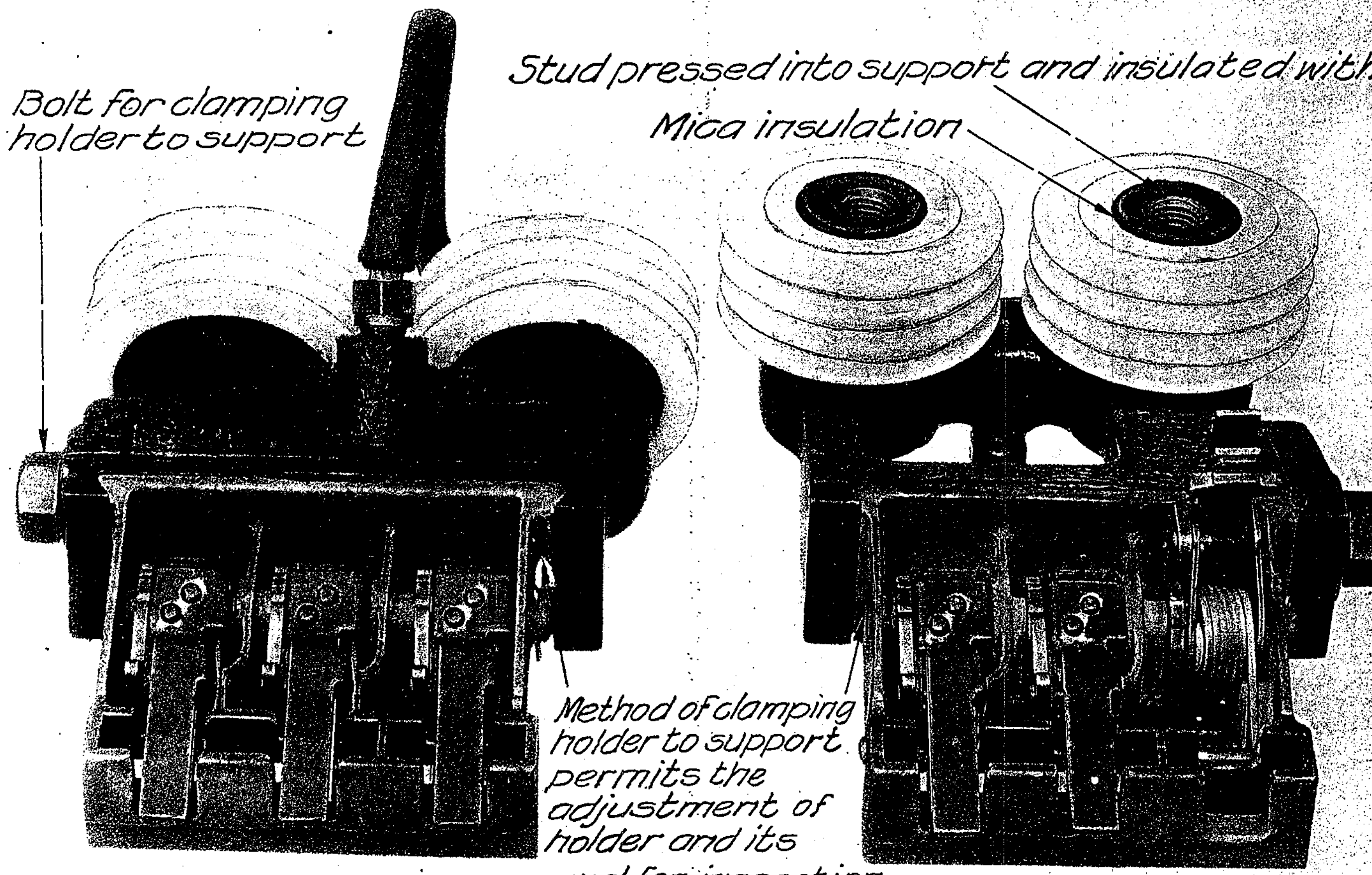


End Supported Brush Holder for GE Railway Motors.

For list of motors fitted with this type of brush holder see "Design of Holder Supports".



Method of securing End Supported Brush Holders to Frame.



Three Brush Holder for G.E Railway Motors.

Most modern brush holders use steel clock springs and recessed or collared seats similar to photograph 248722 Page 73.

Steel clock springs are used to maintain constant pressure on the brush. The seat or collar on which the spring is mounted is recessed and the first turn of the spring is wound tightly on it to insure correct alignment of the spring. Adjustment is obtained by changing the location of cotter pins which fit in holes drilled about the circumference of the collar.

In some cases cast levers with coiled steel springs are used, similar to photograph 249047 Page 74.

In this type of holder spring adjustment is obtained by a ratchet and pawl.

All brush holders are shipped from the factory with the recommended spring tension.

Brushes must be selected according to service conditions and motor characteristics. A tabulation of brush recommendations is found on pages 75–89.

GEAR CASES

Gear cases are made of pressed steel or malleable iron and are provided with either two or three points of support.

Pressed steel is furnished with some of the smaller motors such as GE-247, GE-258, GE-261, GE-262, GE-264, GE-266, GE-288, etc.

VENTILATION

Modern GE Railway Motors are either self-ventilated or blown by an external fan.

Street railway motors are usually self-ventilated while locomotive motors are of the blown type.

Ventilation is of two types, series and multiple.

In the series type air is taken in at the pinion end past the fields, over the commutator, through the commutator shell and armature core, and out the pinion end.

In the multiple type, air is taken in at the commutator end where it divides into two paths, one over the commutator, past the fields and the other through the commutator shell and armature core. The air is exhausted at the pinion end.

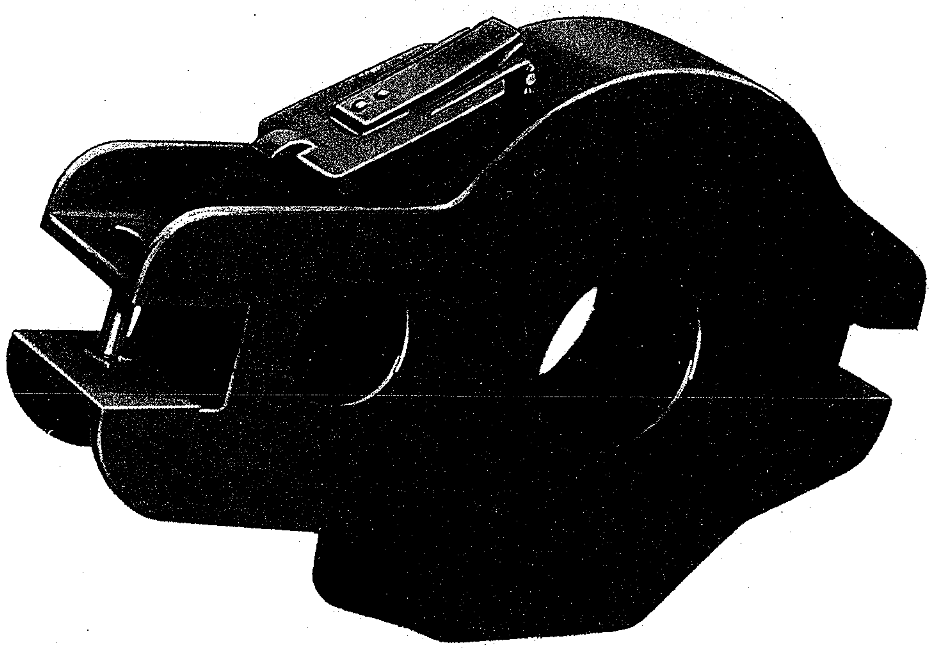
9146

MALLEABLE IRON GEAR CASES

FOR

RAILWAY MOTORS

TO REPLACE PRESSED OR SHEET STEEL GEAR CASES



Malleable Iron Gear Case for GE-265 Railway Motor

Malleable iron gear cases are now available for certain motors which formerly used sheet steel or pressed steel cases. The malleable iron gear case has the advantage of rigidity and strength not possessed by pressed or sheet steel, yet there is no appreciable increase in the weight.

The new gear case has an overlapping joint which keeps out dirt and moisture and affords a means of making adjustments to compensate for wear of the supporting brackets.

The following table lists former standard sheet steel or pressed steel gear cases and malleable iron cases which are interchangeable with them.

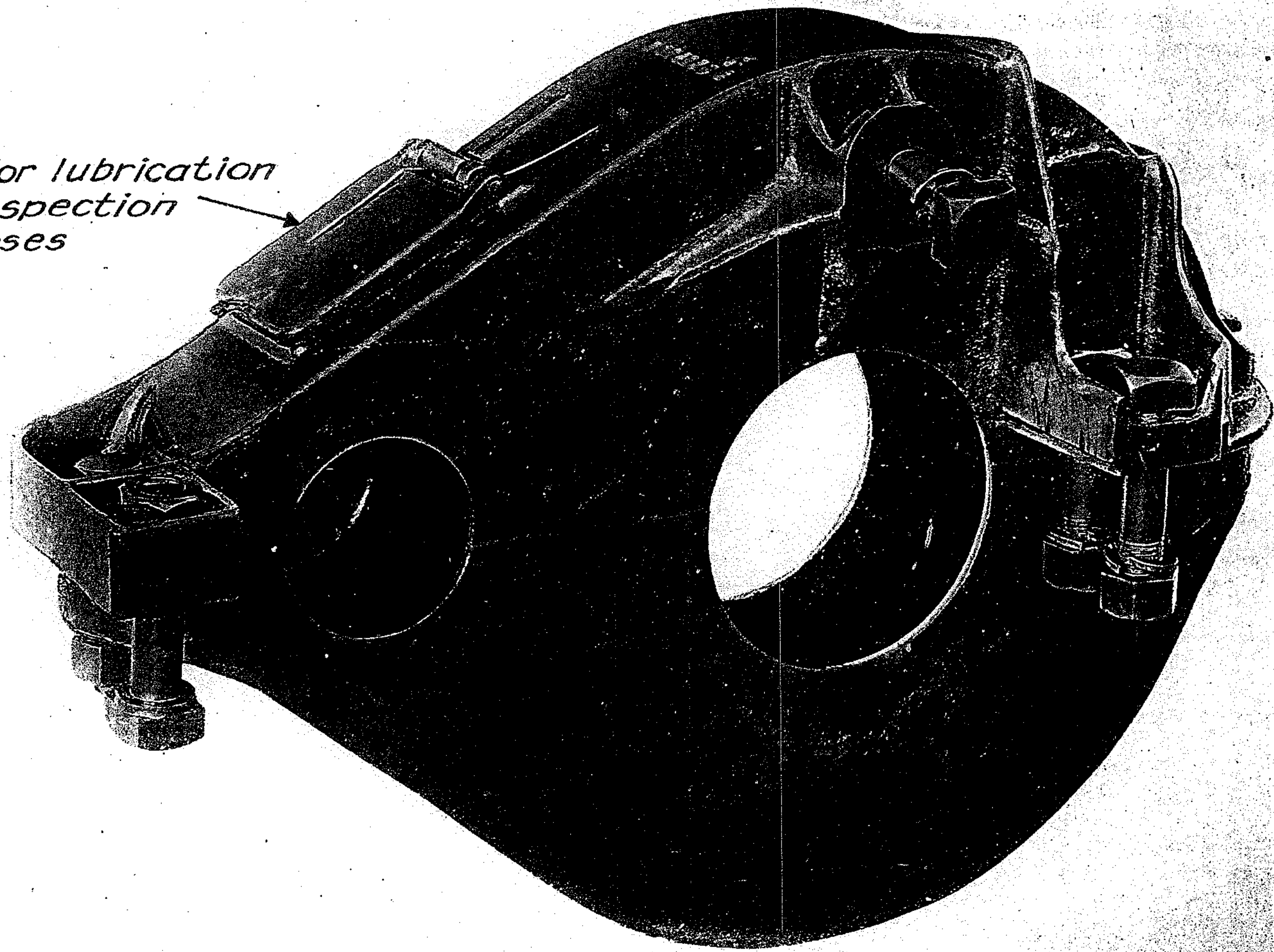
Motor	Form	Gear Ratio		Pitch	Cat. No. Pressed or Sheet Steel Gear Case	Cat. No. Malleable Iron Gear Case
		Max.	Min.			
GE-248	A	* 62/21	58/25	2½	169531	2669626
GE-260	A and D	* 62/21	58/25	2½	2662457	2669610
GE-265	A	69/14	61/22	4	1951505 2662448	2669685
GE-282	A	* 62/21	58/25	2½	2662457	2662499

* Maximum tooth gear indicated must have short addendum teeth, otherwise it will be too large for the gear case.

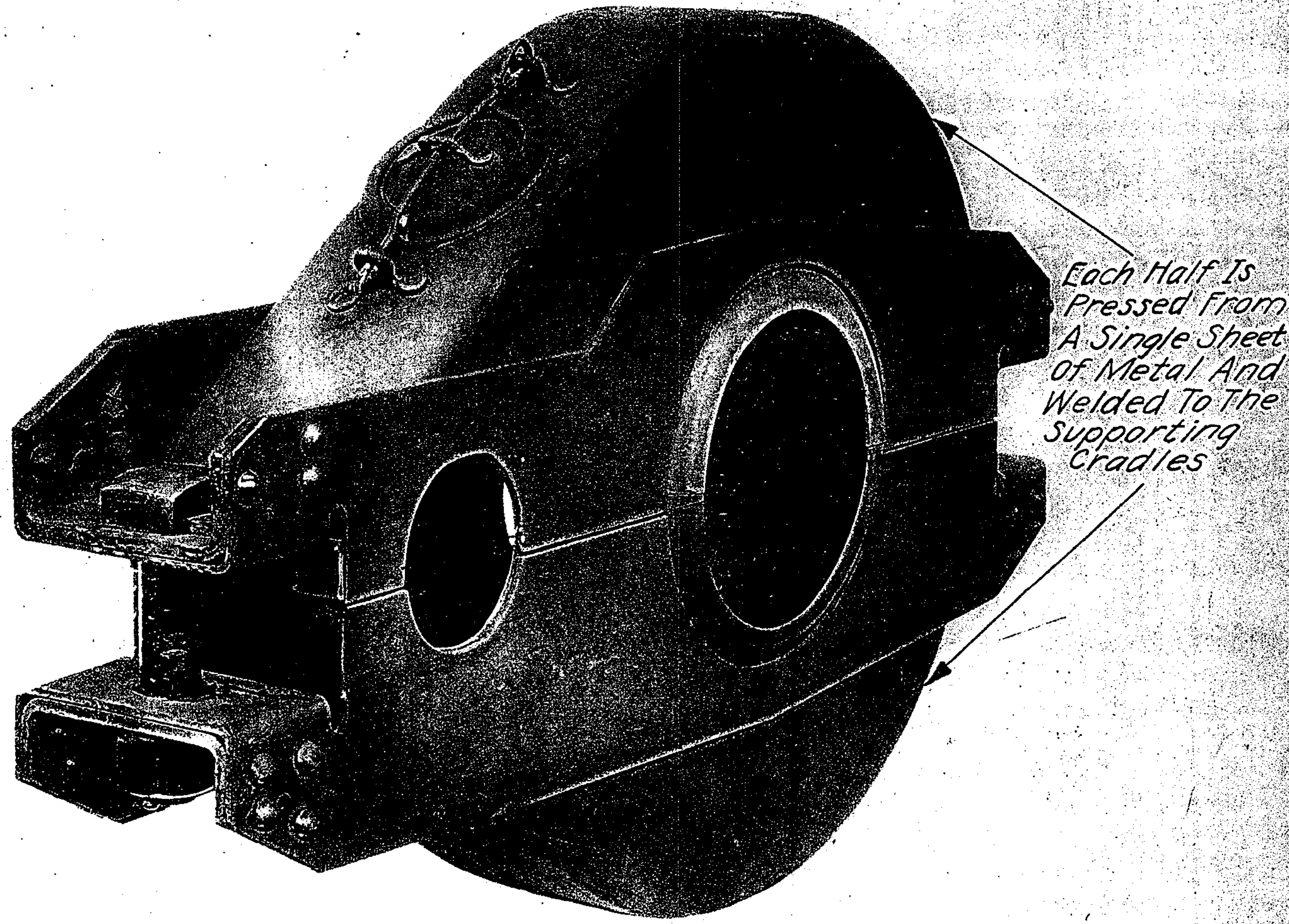
GENERAL ELECTRIC COMPANY

SCHENECTADY, N. Y.

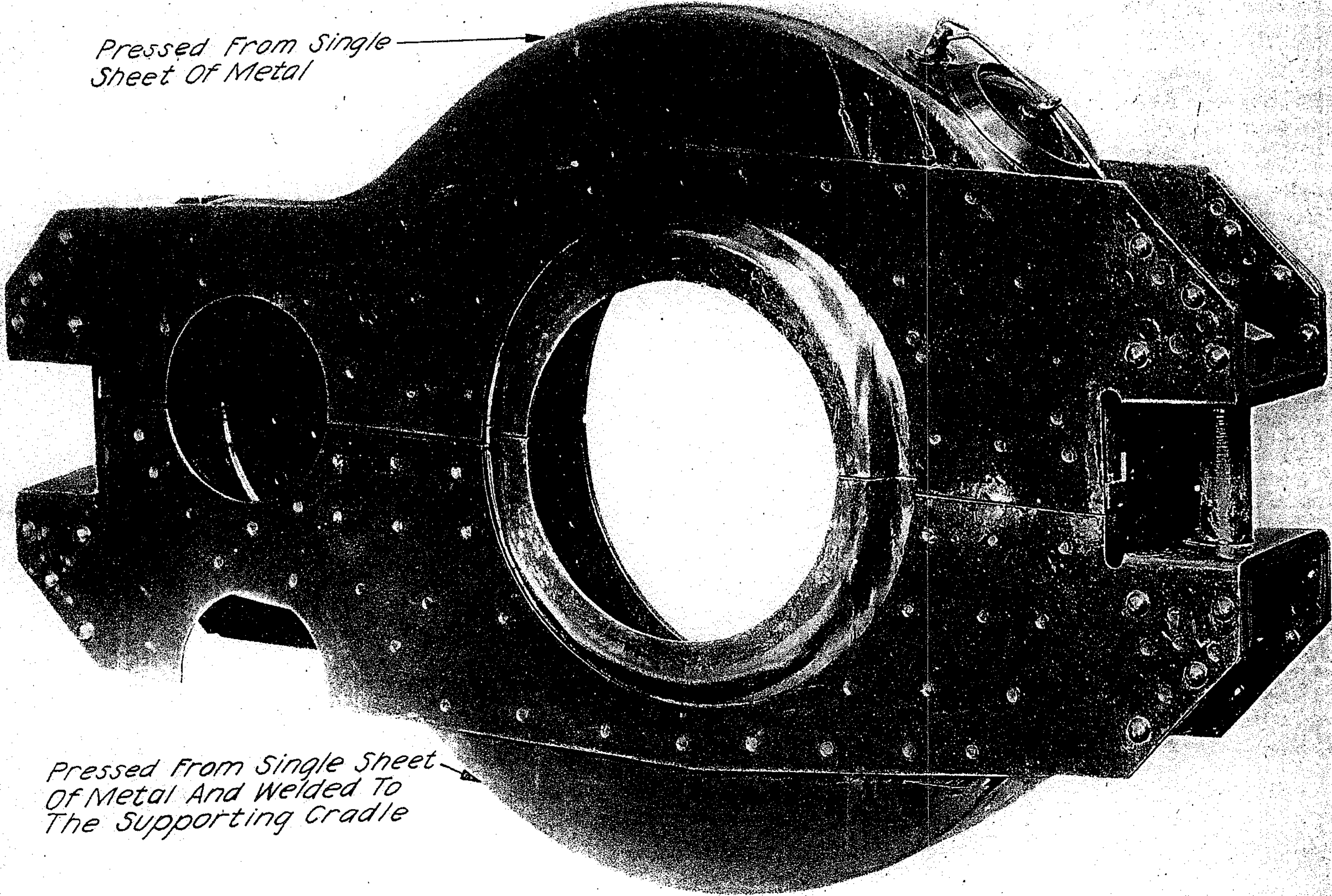
Sales Offices in Principal Cities



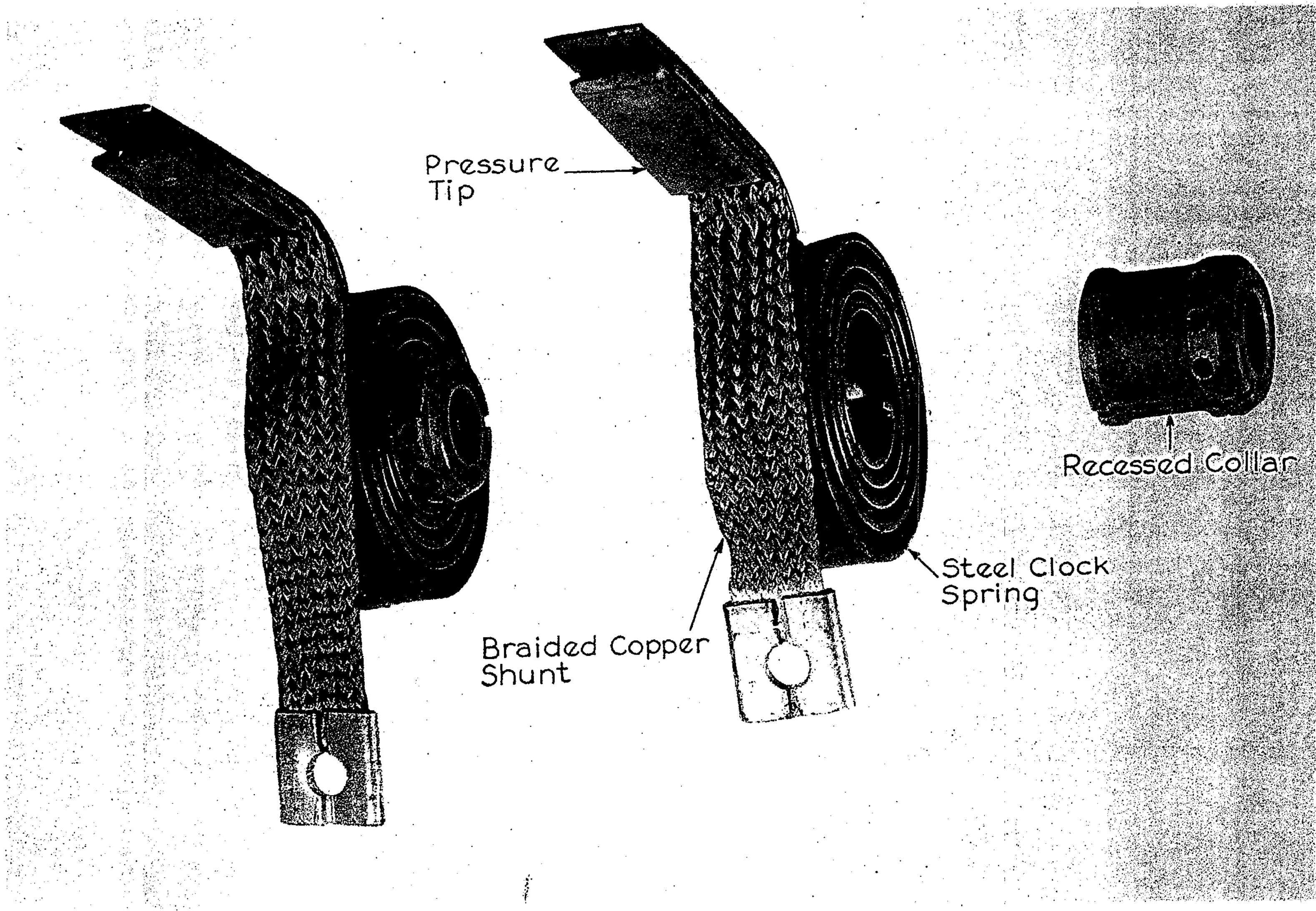
Malleable Iron Gear Case GE Split Frame Motor



Pressed Steel Gear Case For GE Railway Motors

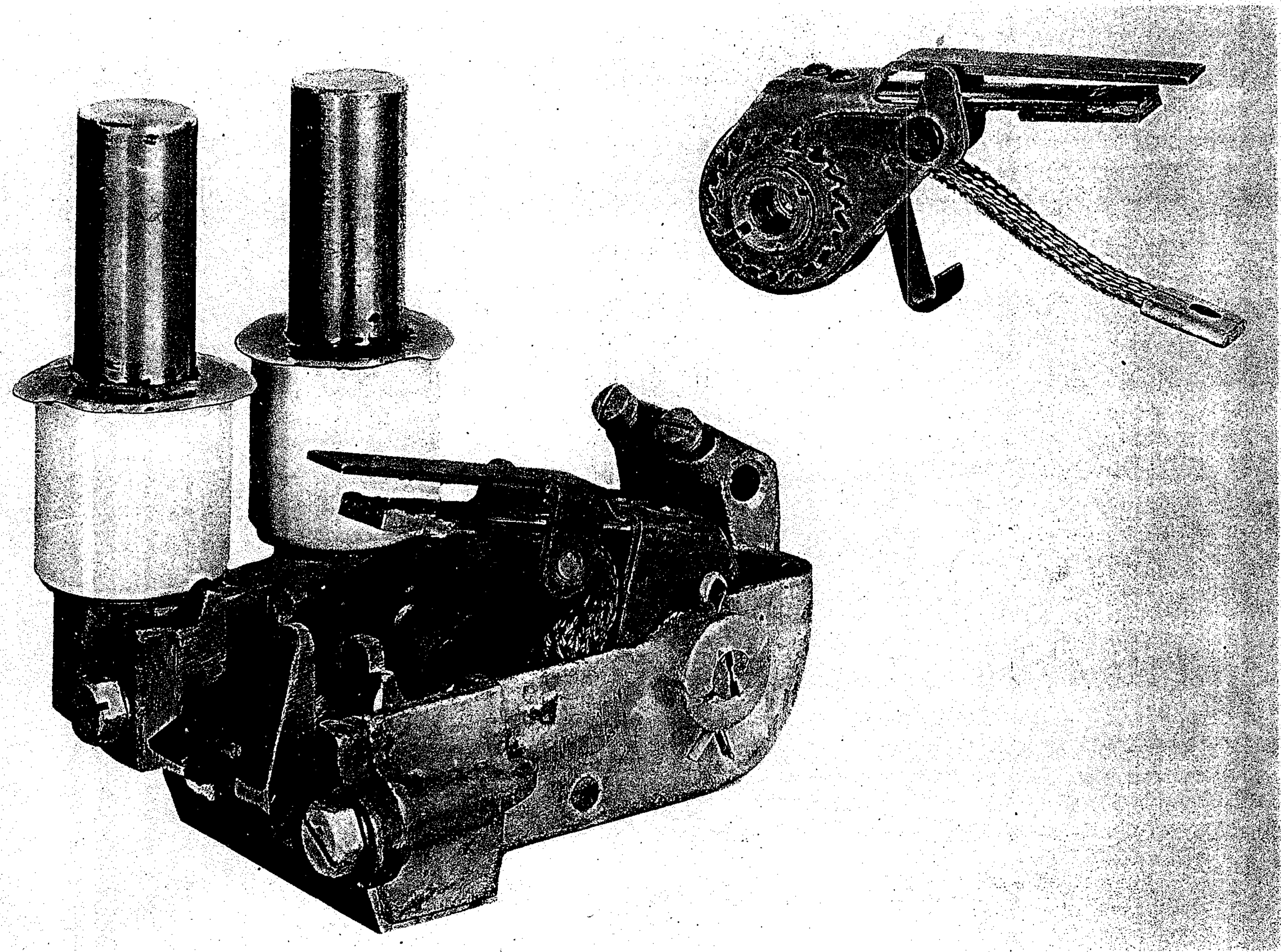


Pressed Steel Gear Case For GE Railway Motors



248722 RAILWAY MOTOR BRUSH-HOLDER CLOCK SPRING LEVER.

E315.1 7-20-28



249047 G-E. RAILWAY MOTOR "CAST OR PUNCHED FINGER" TYPE
BRUSH-HOLDER 10-17-28

CARBON BRUSHES FOR COMMUTATING POLE RAILWAY MOTORS

Motor	Volts	Arm. Turns	Pressure Per Spring Lb.	Brush Dimensions Long	Wide	Thick	Brush Holders Per Mtr.	Springs per Brush	Brushes per Holder	Grade of Brush	Drawing Number	Catalog Number
200	600	4	6-7	2¼ x	1-1/8 x	9/16	2	1	2	D	P-80541-135	134997
200	600	4	6-7	2¼ x	2¼	x 9/16*	2	2	1	D	K-1817710-G1	1817710G1
200	600	4	6-7	2¼ x	2¼	x 9/16	2	2	1	D	P-80541-114	128420
200	600	4	6-7	2¼ x	2¼	x 9/16	2	2	1	D	P-80541-114	128420
201	600	3	7-8	2¼ x	1¾	x 9/16	2	1	2	D	P-80541-77	100376
201	600	3	7-8	2¼ x	1¾	x 9/16	2	1	2	D	P-80541-77	100376
201	600/1200	3	6-7	2¼ x	1-3/8 x	½	2	1	2	D	P-80541-11	14764
202	600	3	5-6	2¼ x	2½	x ½	2	2	1	D	P-80541-29	50395
203	600	4	7-8	2¼ x	1-23/64x	9/16	2	1	2	D	P-80541-132	133324
203	600	3	6-7	2¼ x	1-3/8 x	½	2	1	2	D	P-80541-11	14764
203	600	4	7-8	2¼ x	1-3/8 x	9/16	2	1	2	D	P-80541-156	151393
204	600	2	7-8	2½ x	2	x7/16	2	1	2	B-2	P-80541-94	59987

*Pigtail brushes

CARBON BRUSHES FOR COMMUTATING POLE RAILWAY MOTORS

Motor GE	Volts	Arm. Turns	Pressure Per Spring Lb.	Brush Dimensions Long Wide Thick	Brush Holders per Mtr.	Springs per Brush	Brushes per Holder	Grade of Brush	Drawing Number	Catalog Number
205	600	2	8-9	$2\frac{1}{4}$ x $2\frac{1}{4}$ x 5/8	2	1	2	D	P-80541-52	100663
205	600	2	8-9	$2\frac{1}{4}$ x $2\frac{1}{4}$ x 5/8	2	1	2	D	P-80541-52	100663
205	600/1200	2	7-8	$2\frac{1}{4}$ x $1\frac{3}{4}$ x $\frac{1}{2}$	2	1	2	D	P-80541-86	49743
205	600/1200	2	8-9	$2\frac{1}{4}$ x 2 x 5/8	2	1	2	D	P-80541-1	36063
205	1200	3	6-7	$2\frac{1}{4}$ x $1\frac{1}{2}$ x 7/16	2	1	2	D	P-80541-100	62509
205	750/1500	2	7-8	$2\frac{1}{4}$ x 1-7/8 x 9/16	2	1	2	D	P-80541-145	143442
206	600	2	9-10	$2\frac{1}{4}$ x 3 x 5/8	2	1	2	D	P-80541-92	107579
206	600	2	8-9	2 x $2\frac{3}{4}$ x 5/8	2	1	2	D	P-80541-126	133668
206	600/1200	2	7-8	$2\frac{1}{4}$ x $2\frac{1}{4}$ x 9/16	2	1	2	D	P-80541-114	128420
207	600	1	7-8	2 x $2\frac{1}{4}$ x 5/8	2	1	3	D	P-80541-149	36321
207	600/1200	1	7-8	2 x 2 x 5/8	2	1	3	D	P-80541-79	59578
				(2 x $2\frac{1}{4}$ x 5/8)	2	1	1		(P-80541-49	36321)
207	600/1200	2	8-9	(2 x $2\frac{3}{4}$ x 5/8)	2	1	1	D	(P-80541-126	133668)
207	600/1200	3	$6\frac{1}{2}$-$7\frac{1}{2}$	2 x1-5/8x5/8	2	1	2	D	P-80541-149	144466

CARBON BRUSHES FOR COMMUTATING POLE RAILWAY MOTORS

Motor GE	Volts	Arm. Turns	Pressure Per Spring Lb.	Brush Dimensions Long Wide Thick	Brush Holders Per Mtr.	Springs per Brush	Brushes per Holder	Grade of Brush	Drawing Number	Catalog Number
209	600	1	7-8	2 x 2½ x 11/16	2	1	3	D	P-80541-95	104952
209	600	1	7-8	2 x 2½ x 11/16*	4	1	3	D	K-3611252G1	3611252G1
209	600	1	7-8	2 x 2½ x 11/16	4	1	3	D	P-80541-95	104952
210	600	3	6-7	2¼ x 1-5/8 x ½	2	1	2	D	P-80541-98	59889
211	600	1	9-10	2¼ x 3¼ x 9/16	2	1	2	D	P-80541-50	24843
212	600	1	6-7	2 x 2 x 5/8	2	1	4	D	P-80541-79	59578
212	600	1	6-7	2 x 2 x 5/8*	2	1	4	D	K-1666106G1	178480
212	600/1200	1	7-8	2 x 2-3/8 x 5/8	2	1	3	D	P-80541-128	133666
213	600	3	6-7	2¼ x 2½ x ½	2	2	1	B-2	P-80541-29	50395
214	600	2	7+8	2½ x 2 x 7/16	2	1	2	D	P-80541-94	59987
214	600	2	7-8	2¼ x 2 x ½	2	1	2	D	P-80541-105	129370
215	600	3	6-7	2¼ x 2½ x ½	2	2	1	D	P-80541-29	50395
216	600	3	5½-6½	2¼ x 3 x ½	2	2	1	B-2	P-80541-102	61176

*Pigtail brushes

CARBON BRUSHES FOR COMMUTATING POLE RAILWAY MOTORS

Motor GE	Volts	Arm. Turns	Pressure Per Spring Lb.	Brush Dimensions Long Wide Thick	Brush Holders Per Mtr.	Springs per Brush	Brushes per Holder	Grade of Brush	Drawing Number	Catalog Number
216	750	3	$5\frac{1}{2}$-$6\frac{1}{2}$	2 x 2-3/8 x 5/8	2	2	1	B-2	P-80541-128	133666
217	600/1200	3	6-7	$2\frac{1}{4}$ x $1\frac{1}{4}$ x $\frac{1}{2}$	2	1	2	B-2	P-80541-3	15698
217	750/1500	4	6-7	$2\frac{1}{4}$ x $1\frac{1}{4}$ x $\frac{1}{2}$	2	1	2	B-2	P-80541-3	15698
218	600	3	6-7	$2\frac{1}{4}$ x 3 x 5/8	2	2	1	D	P-80541-92	107579
219	600	3	6-7	$2\frac{1}{4}$ x 3 x $\frac{1}{2}$	2	2	1	B-2	P-80541-102	61176
219	750	3	6-7	$2\frac{1}{4}$ x $2\frac{3}{4}$ x $\frac{1}{2}$	2	2	1	B-2	P-80541-28	50396
222	600	2	7-8	2 x 1-13/16 x 5/8	2	1	3	D	P-80541-141	140954
222	600	2	(8-9 (8-9	(2 x 3 x 5/8) (2 x $2\frac{1}{2}$x 5/8)	2 2	1 1	1 1	D D	(P-80541-123 (P-80541-122	143303) 133667)
222	600/1200	2	(8-9 (8-9	(2 x $2\frac{3}{4}$ x 5/8) (2 x $2\frac{1}{4}$ x 5/8)	2 2	1 1	1 1	D D	(P-80541-126 (P-80541-49	133668) 36321)
222	600/1200	2	7-8	2 x 1-5/8 x 5/8	2	1	3	D	P-80541-149	144466
225	600/1200	2	7-8	2 x 2 x 9/16	2	1	2	D	P-80541-69	44616
225	600/1200	2	7-8	2 x 2 x 5/8	2	1	2	D	P-80541-79	59578

CARBON BRUSHES FOR COMMUTATING POLE RAILWAY MOTORS

Motor GE	Volts	Arm. Turns	Pressure Per Spring Lb.	Brush Dimensions Long Wide Thick	Brush Holders Per Mtr.	Springs per Brush	Brushes per Holder	Grade of Brush	Drawing Number	Catalog Number
226	600	4	6-7	2¼ x 1-23/64 x 9/16	2	1	2	D	P-80541-132	133324
227	600	3	7-8	2¼ x 1¾ x 9/16	2	1	2	D	P-80541-77	100376
229	1200/2400	1	6½-7½	2¼ x 2 x 5/8	4	1	2	D	P-80541-12	36070
230	600	3	7-8	2¼ x 1¾ x 9/16	2	1	2	D	P-80541-77	100376
231	600	4	6-7	2¼ x 1-23/64 x 9/16	2	1	2	D	P-80541-132	133324
232	600/1200	3	6-7	2¼ x 1½ x 3/8	2	1	2	D	P-80541-136	135752
233	600	2	7-8	2¼ x 2 x ½	2	1	2	D	P-80541-105	129370
233	600/1200	2	6-7	2¼ x 1¾ x ½	2	1	2	D	P-80541-86	49745
234	600	3	6-7	2¼ x 1-3/8 x 7/16	2	1	2	D	P-80541-137	136271
235	775	1	6-7	2¼ x 1¾ x 5/8	2	1	4	D	P-80541-9	18167
236	600	3	6-7	2 x 1¼ x 7/16	2	1	2	D	P-80541-144	143441
237	750/1500	2	6-7	2¼ x 1-5/8 x 5/8	2	1	3	D	P-80541-155	150391

CARBON BRUSHES FOR COMMUTATOR POLE RAILWAY MOTORS

Motor GE	Volts	Arm. Turns	Pressure Per Spring Lb.	Brush Dimensions Long Wide Thick	Brush Holders Per Mtr.	Springs per Brush	Brushes per Holder	Grade of Brush	Drawing Number	Catalog Number
238	600	3	6-7	$2\frac{1}{4}$ x $1\frac{1}{4}$ x 7/16	2	1	2	D	P-80541-152	145823
239	1200/2400	2	6-7	2 x 2 x $\frac{1}{2}$	2	1	2	"G"	P-80541-99	59988
240	600	2	8-9	$2\frac{1}{4}$ x $2\frac{1}{4}$ x 5/8	2	1	2	"G"**	P-80541-52	100663
240	600/1200	2	8-9	$2\frac{1}{4}$ x 2 x 5/8	2	1	2	"G"	P-80541-1	36063
241	600	3	7-8	$2\frac{1}{4}$ x 1-3/8 x 9/16	2	1	2	D	P-80541-156	151393
241	600	3	7-8	$2\frac{1}{4}$ x 1-3/8 x 9/16*	2	1	2	D	K-1666112G1	207271
242	600	3	7-8	$2\frac{1}{4}$ x $1\frac{3}{4}$ x 9/16	2	1	2	D	P-80541-77	100376
243	600	1	7-8	2 x2-1/8 x 9/16	2	1	3	D	P-80541-159	153436
244	600/1200	2	6-7	2 x1-3/8 x 9/16	4	1	2	D	P-80541-161	154373
245	600	3	6-7	$2\frac{1}{4}$ x 1-3/8 x 9/16	2	1	2	D	P-80541-156	151393
246	600	4	6-7	$2\frac{1}{4}$ x 1-3/8 x 9/16	2	1	2	D	P-80541-156	151393
246	600	4	6-7	$2\frac{1}{4}$ x 1-3/8 x 9/16	2	1	2	D	P-80541-131	133669

*Pigtail brushes

** For Gas Electric Cars Use NCC-9406

CARBON BRUSHES FOR COMMUTATOR POLE RAILWAY MOTORS

Motor GE	Volts	Arm. Turns	Pressure Per Spring Lb.	Brush Dimensions Long Wide Thick	Brush Holders Per Mtr.	Springs per Brush	Brushes per Holder	Grade of Brush	Drawing Number	Catalog Number
247	600	3	7-8	2 x $1\frac{1}{4}$ x $\frac{1}{2}$	2	1	2	"G"	P-80541-163	157023
247	600	3	10-11	$2\frac{1}{4}$ x $2\frac{1}{2}$ x $\frac{1}{2}$	2	1	1	"G"	P-80541-29	50395
247	600	3	6-7	$2\frac{1}{4}$ x $2\frac{1}{2}$ x $\frac{1}{2}$	2	2	1	"G"	P-80541-29	50395
247	600	4	7-8	2 x $1\frac{1}{4}$ x $\frac{1}{2}$	2	1	2	"G"	P-80541-163	157023
247	600	3	10-11	2-1/8 x $2\frac{1}{2}$ x 9/16	2	1	1	"G"	V-1493230	149230
248	600	1	$8\frac{1}{2}$-$9\frac{1}{2}$	2 x 2-1/8 x 11/16	2	1	3	D	P-80541-160	153771
249	600	4	6-7	$2\frac{1}{4}$ x 1-1/8 x $\frac{1}{2}$	2	1	2	G	P-80541-162	157022
249	600	4	8-9	$2\frac{1}{4}$ x $2\frac{1}{4}$ x 5/8	2	1	1	G	P-80541-52	100663
249	600	4	6-7	$2\frac{1}{4}$ x 1-1/8 x 5/8	2	1	2	G	P-80541-165	168148
249	600	5	6-7	$2\frac{1}{4}$ x 1-1/8 x 5/8	2	1	2	G	P-80541-165	168148
250	600	2	7-8	$2\frac{1}{4}$ x 1-7/8 x 5/8	4	1	2	D	P-80541-34	122860
251	600/1200	1	7-8	2 x 2-3/8 x 5/8	2	1	3	D	P-80541-128	133666
251	600/1200	1	7-8	2 x 2-3/8 x 5/8*	2	1	3	D	K-3611255G1	3611255G1

* Pigtail brushes

CARBON BRUSHES FOR COMMUTATOR POLE RAILWAY MOTORS

Motor GE	Volts	Arm. Turns	Pressure Per Spring Lb.	Brush Dimensions Long Wide Thick	Brush Holders Per Mtr.	Springs per Brush	Brushes per Holder	Grade of Brush	Drawing Number	Catalog Number
253	1500/3000	1	6-7	$2\frac{1}{4}$ x $1\frac{3}{4}$ x 11/16	4	1	2	D	P-80541-164	161200
254	600	2	7-8	2 x 1-13/16 x 5/8	2	1	3	"G"**	P-80541-141	140954
254	600/1200	2	7-8	2 x 1-5/8 x 5/8	2	1	3	"G"	P-80541-I49	144466
254	600/1200	2	7-8	2 x 1-5/8 x 5/8*	2	1	3	"G"	K-3611235G1	3611235G1
255	600/1200	1	6-7	2 x 1-5/8 x 5/8	4	1	3	D	P-80541-149	144466
255	1000/3000	1	7-8	$2\frac{1}{4}$ x $2\frac{1}{4}$ x 5/8	4	1	2	D	P-80541-52	100663
255	1500/3000	2	6-7	$2\frac{1}{4}$ x 2 x $\frac{1}{2}$	4	1	1	D	P-80541-105	129370
257	600	1	7-8	2 x $2\frac{1}{4}$ x 5/8	2	1	3	D	P-80541-49	36321
257	600/1200	1	7-8	2 x 2 x 5/8	2	1	3	D	P-80541-79	59578
258	600	5	8-9	2 x 1-7/8 x 9/16	2	1	1	"G"	P-80541-166	176199
258	600	4	8-9	2 x 1-7/8 x 9/16	2	1	1	"G"	P-80541-166	176199
258	600	3	8-9	2 x 1-7/8 x 9/16	2	1	1	"G"	P-80541-166	176199
259	600	2	7-8	2 x $1\frac{3}{4}$ x 5/8	2	1	3	"G"	P-80541-167	178574
259	600/1200	2	7-8	2 x $1\frac{3}{4}$ x 5/8	2	1	3	"G"	P-80541-167	178574

* Pigtail brushes

** For Gas Electric Cars Use NCC-9406

CARBON BRUSHES FOR COMMUTATING POLE RAILWAY MOTORS

Motor GE	Volts	Arm. Turns	Pressure Per Spring Lb.	Brush Dimensions Long Wide Thick	Brush Holders Per Mtr.	Springs per Brush	Brushes per Holder	Grade of Brush	Drawing Number	Catalog Number
260	600	1	8-9	2 x 2-1/8 x 11/16	2	1	3	D	P-80541-160	153771
260	600	1	8-9	2 x 2-1/8 x 11/16*	2	1	3	D	K-1877982Pt1	224960
261	250	2	7-8	2 x 1-1/8 x 9/16	2	1	2	G	P-80541-168	179725
262	600	4	8-9	2 x 1-7/8 x 9/16	2	1	1	G	P-80541-166	176199
263	600	3	7-8	$2\frac{1}{4}$ x $1\frac{3}{4}$ x 9/16	2	1	2	G	P-80541-77	100376
263	600/1200	3	6-7	$2\frac{1}{4}$ x 1-3/8 x $\frac{1}{2}$	2	1	2	G	P-80541-11	14764
264	600	4	8-9	$2\frac{1}{4}$ x 1-7/8 x 9/16	2	1	1	G	P-80541-145	143442
264	600	5	8-9	$2\frac{1}{4}$ x 1-7/8 x 9/16	2	1	1	G	P-80541-145	143442
264	600	4	8-9	2 x 1-7/8 x 9/16	2	1	1	G	P-80541-166	176199
265	600	4	10-11	2 - 1/8 x$2\frac{1}{2}$ x 9/16	2	1	1	G	V-1493230	1493230
265										
265	600	4	7-8	$2\frac{1}{4}$ x $1\frac{1}{4}$x 9/16	2	1	2	G	V-1493241	1493241
265	600	5	10-11	2-1/8 x $2\frac{1}{2}$ x 9/16	2	1	1	G	V-1493230	1493230
265	600	4	7-8	2-1/8 x $1\frac{1}{4}$ x 9/16	2	1	2	G	P-80541-62	100372
265	600	4	6-7	2-1/8 x $2\frac{1}{2}$ x 9/16	2	2	1	G	V-1493230	1493230

*Pigtail Brushes

CARBON BRUSHES FOR COMMUTATOR POLE RAILWAY MOTORS

Motor GE	Volts	Arm. Turns	Pressure Per Spring Lb.	Brush Dimensions Long Wide Thick	Brush Holders Per Mtr.	Springs per Brush	Brushes per Holder	Grade of Brush	Drawing Number	Catalog Number
266	125	1	6-7	2 x 1-15/16x11/16*	2	1	2	G	K-1877976G1	1877976G1
267	1500/3000	1	9-11	2¼ x 2 x ½	4	1	2	NCC-AX	P-80541-105	129370
269	600	3	7-8	2¼ x 1½ x 5/8	2	1	2	G	P-80541-6	52546
269	750/1500	3	8-9	2¼ x 2¼ x 5/8	2	1	1	G	P-80541-52	100663
270	600	3	7-8	2¼ x 1½ x 5/8	2	1	2	G	P-80541-6	52546
272	600	3	7-8	2 x 2¼ x 5/8	2	1	2	D	P-80541-49	36321
273	600	4	7-8	2¼ x 1¼ x 9/16	2	1	2	G	1493241	1493241
274	750/1500	1	7-8	2¼ x 2¼ x 11/16	4	1	2	D	1492299	1492299
275	600	3	7-8	2¼ x 3¼ x 5/8"	2	2	1	G	1496086	1496086
275	600	3	8-9	2¼ x 1-5/8 x 5/8	2	1	2	G	P-80541-155	150391
275	600/1200	3	8-9	2¼ x 1-5/8 x 5/8	2	1	2	G	P-80541-155	150391
276	1500	1	6-7	2.560x1.772 x .630*	4	1	2	D	1496023	1496023

* Pigtail Brushes

CARBON BRUSHES FOR COMMUTATOR POLE RAILWAY MOTORS

Motor GE	Volts	Arm. Turns	Pressure Per Spring Lb.	Brush Dimensions Long Wide Thick	Brush Holders Per Mtr.	Springs per Brush	Brushes per Holder	Grade of Brush	Drawing Number	Catalog Number
277	750/1500	1	7-8	2.244x1.890x.689*	4	1	3	D	1496009	1496009
278	1500/3000	1	7-8	$2\frac{1}{4}$ x 2 x 11/16	4	1	2	G	1493263	1493263
278	1500/3000	1	9-11	$2\frac{1}{4}$ x 2 x $\frac{1}{2}$	4	1	2	NCC-AX	P-80541-105	129370
279	750/1500	2	6-7	$2\frac{1}{4}$ x 1-3/8 x 9/16	4	1	2	D	P-80541-156	151393
281	600	2	7-8	$2\frac{1}{4}$ x 1-5/8 x 11/16	2	1	2	G	2440647	2440647
281	600/1200	2	7-8	$2\frac{1}{4}$ x 1-5/8 x 11/16	2	1	2	G	2440647	2440647
281	750/1500	2	7-8	$2\frac{1}{4}$ x 1-5/8 x 11/16	2	1	2	G	2440647	2440647
282	600	1	8-9	2 x 2-1/8 x 11/16	2	1	3	D	P-80541-160	153771
282	600	1	8-9	2 x 2-1/8 x 11/16*	2	1	3	NCC-AX	2440652	2440652
282	600	1	8-9	2 x 2-1/8 x 11/16*	2	1	3	NCC-AX	2444566	2444566
284	750/1500	2	7-8	2 x 1-5/8 x 5/8	2	1	3	G	P-80541-149	144466

*Pigtail brushes.

CARBON BRUSHES FOR COMMUTATOR POLE RAILWAY MOTORS

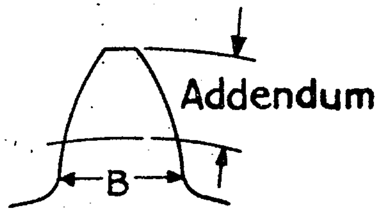
Motor GE	Volts	Arm. Turns	Pressure Per Spring Lb.	Brush Dimensions Long Wide Thick	Brush Holders Per Mtr.	Springs per Brush	Brushes per Holder	Grade of Brush	Drawing Number	Catalog Number
285	750/1500	1	8-9	$2\frac{1}{4}$ x $2\frac{1}{2}$ x 5/8	2	1	3	G	P-80541-76	62508
286	600/1200	1	7-8	$2\frac{1}{4}$ x 2 x $\frac{3}{4}$*	4	1	3	G	2440666	2440666
286	1500/3000	1	7-8	$2\frac{1}{4}$ x 2 x $\frac{3}{4}$*	4	1	2	G	2440666	2440666
287	225	1	8-9	$2\frac{1}{4}$ x $1\frac{3}{4}$ x 1*	4	1	3	NCC-AX	2440680	2440680
287	300/600	1	10-11	$2\frac{1}{4}$ x 2-5/8 x 1	4	1	2	NCC-AX	2444557	2444557
288	600	3	6-7	$2\frac{1}{4}$ x $2\frac{1}{2}$ x $\frac{1}{2}$	2	2	1	G	P-80541-29	50395
288	600	3	10-11	$2\frac{1}{4}$ x $2\frac{1}{2}$ x 9/16	2	1	1	G	1493236	1493236
288	600	3	10-11	$2\frac{1}{4}$ x $2\frac{1}{2}$ x $\frac{1}{2}$	2	1	1	G	P-80541-29	50395
288	600	3	7-8	2 x $1\frac{1}{4}$ x $\frac{1}{2}$	2	1	2	G	P-80541-163	157023
289	750/1500	1	6-7	$2\frac{1}{4}$ x $1\frac{3}{4}$ x 11/16*	4	1	2	G	2444519	2444519
290	750/1500	1	9-10	$2\frac{1}{2}$ x $2\frac{3}{4}$ x 7/8*	4	1	2	NCC-AX	2444529	2444529
291	600	4	10-11	2-1/8 x $2\frac{1}{2}$ x 9/16	2	1	1	G	1493230	1493230
292	450	1	8-9	2 x 2-1/8 x 5/8	2	1	3	NCC-AX	2444551	2444551

*Pigtail brushes
*Pigtail brushes

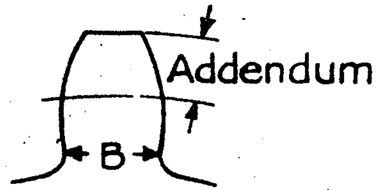
CARBON BRUSHES FOR COMMUTATOR POLE RAILWAY MOTORS

Motor GE	Volts	Arm. Turns	Pressure Per Spring Lb.	Brush Dimensions Long Wide Thick	Brush Holders Per Mtr.	Springs per Brush	Brushes per Holder	Grade of Brush	Drawing Number	Catalog Number
293	600	4	8-9	$2\frac{1}{4}$ x $2\frac{1}{4}$ x 5/8	2	1	1	G	P-80541-52	100663
294	600	3	7-8	2 x $1\frac{1}{4}$ x $\frac{1}{2}$	2	1	2	G	P-80541-163	157023
295	600	3	7-8	$2\frac{1}{4}$ x $1\frac{3}{4}$ x 5/8	4	1	1	G	V-713220	713220
296	600	1	8-9	2 x 2-1/8 x 11/16	2	1	3	D	P-80541-160	153771
297	600	1	8-9	2 x 2-1/8 x 11/16	4	1	3	NCC-AX	P-80541-160	153771
298	600	2	7-8	1-7/8 x 1-1/2 x 5/8	4	1	1	D	V-2707178	2707178
299	600	3	7-8	$2\frac{1}{4}$ x $3\frac{1}{4}$ x 5/8	2	2	1	G	1496086	1496086

15 Tooth Pinions

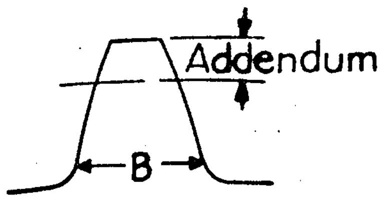


Long Addendum Tooth
20° Pressure Angle.

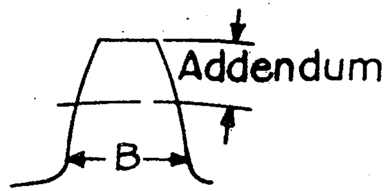


B & S Tooth
$14\frac{1}{2}°$ Pressure Angle

69 Tooth Gears

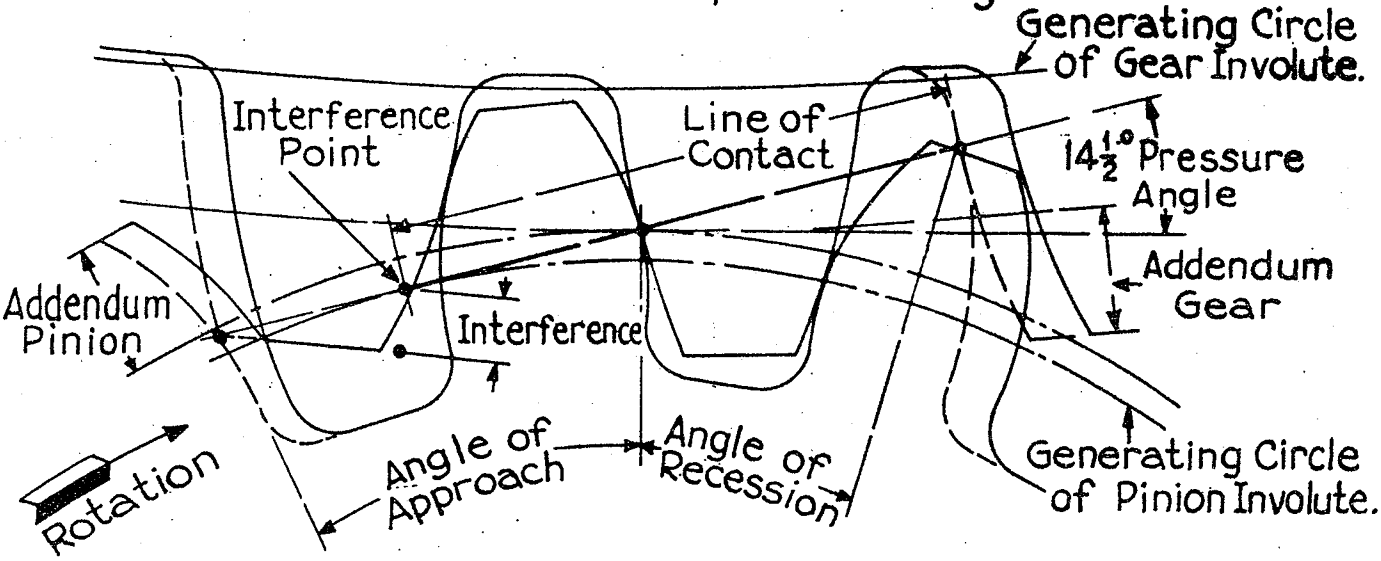


Short Addendum Tooth
20° Pressure Angle.



B & S Tooth
$14\frac{1}{2}°$ Pressure Angle

Brown and Sharpe Gearing



McLaury

International General Electric Company
Incorporated

SALES AND ENGINEERING DATA - File 1041 - Serial 54

Schenectady, N.Y., Feb. 12, 1926

SUBJECT: Advantages of Long and Short Addendum Type Railway Gears.

The following article on the advantages of Long and Short Addendum Type Railway Gears, written by Mr. George I. Kotz of our Engineering Department for the G. E. News Bureau, contains much information that would be useful in the exploitation of G. E. railway motors and gears. Although many of you may have read the article in a recent issue of the RAILWAY JOURNAL or have seen the copies which come to you through the News Bureau service, we consider the matter covered of such importance that it is being sent out as Sales and Engineering Data in order that it can be properly filed for ready reference.

"During the past few years much has been accomplished by the design of improved gear teeth for use in railway motor work. Teeth of the old 14-1/2° Brown and Sharpe proportions would not, in many instances, meet the requirements of present day demands due to improper tooth shapes of pinions under thirty teeth. Either the teeth had to be undercut, or both the flank of the pinion tooth and the face of the gear tooth relieved in order to avoid interference. This condition of undercutting or relieving produced a tooth thin at its base, where strength is required.

As a majority of the pinions used in railway service have between thirteen and thirty teeth, it was realized that undercutting could not be avoided. There was a demand for a design of something better than the old Brown and Sharpe tooth shape. This demand resulted in the General Electric Company introducing the present efficient type of long and short addendum gearing.

Long and short addendum gearing is exactly what its name implies; the pinion has a longer addendum than a pinion of the Brown and Sharpe proportions and the gear a correspondingly shorter addendum. (Refer to Figures 1 and 2 which show respectively a long addendum pinion tooth compared with the same tooth of Brown and Sharpe proportions and a short addendum gear tooth compared with the same Brown & Sharpe tooth.)

The lengthening of the pinion addendum, while maintaining the standard pitch diameter, increases its outside diameter. This would result in teeth with very narrow ends if the thickness of the tooth at the pitch line were not made greater than one-half the circular pitch, which has for years been standard with the Brown and Sharpe proportions. The increase in thickness at the pitch line results in a corresponding increase in thickness at the base of the tooth and greatly increases the strength of the tooth at this point. For example, a single tooth considered as a beam fixed at one end with a load applied at the other would be approximately 60% stronger in the case of a 15-tooth, 3 pitch, 20° angle, long addendum pinion as compared with a 15-tooth, 3-pitch, 14 1/2° angle Brown and Sharpe. (Refer again to Figure 1 and note the increased thickness in section at the base of the tooth in the long addendum type compared with the same tooth of Brown and Sharpe proportions as indicated by Line B).

The increase in outside diameter of the pinion, while maintaining the standard length of tooth, increases the amount of metal between the bore and the bottom of the teeth, thereby reducing the stresses in the core of the pinion which result from mounting it on the armature shaft. The magnitude of these stresses has a direct bearing on the life of the pinion teeth when placed in service.

The thickness of metal between the bottom of the keyway and the bottom of the teeth has been further increased in many instances by using what is termed a "flush key". In this case the depth of the keyway in the pinion at the large end of the tapered bore is zero with the bottom of the keyway parallel to the center line of shaft.

The use of long addendum pinions, with the resulting increase in tooth strength, makes possible a higher gear reduction by the use of finer pitch gearing. This enables the use of a higher speed armature and consequently a lighter motor.

The thickness of the gear tooth, while less than one-half the circular pitch at the pitch line, is approximately 10% stronger than a Brown and Sharpe tooth at the base due to the increased angularity of the tooth profile. (Refer again to Figure 2 and note the increased thickness in section at the base of the tooth in the short addendum type compared with the same tooth of Brown and Sharpe proportions as indicated by Line B). By shortening the addendum of the gear it is possible to obtain a greater clearance between the gear case and the track. Another advantage of the shorter gear addendum is that it decreases the arc of approach. As the rate of sliding between the tooth surfaces is greater during this portion of the contact, it follows that a shorter gear addendum reduces the sliding at the maximum end and consequently the rate of sliding between the tooth surfaces.

This is due to the smaller and more sensitive portion of the pinion involute being in mesh with the larger and less sensitive portion of the gear curve. The shorter gear addendum does not make contact as soon as the addendum of a Brown and Sharpe gear, thereby decreasing the arc of approach and reducing the time o excessive sliding contact. Moreover, the length of true involute profile on the addendum of a long addendum pinion tooth more nearly equals the length of profile on the addendum of its mating gear tooth, resulting in a greater percentage of rolling contact than can be obtained with Brown and Sharpe teeth in which the pinion, if unde thirty teeth, has this portion of its profile relieved or undercut. This relieved portion of the pinion tooth profile cannot be expected to deliver other than sliding contact. Therefore, the short length of involute profile on the flank of a relieved pinion tooth when operating against a much longer profile on the face of its mating gear tooth indicates a large percentage of sliding contact.

Inasmuch as the shorter gear addendum decreases the arc of approach, it is natural to expect that the increased pinion addendum should increase the arc of recession and the sliding action. This is true, but the percentage of sliding contact is less than in a Brown and Sharpe combination as the action is taking place on a portion of the involutes further distant from their origin and more nearly equal to each other. Moreover the sliding in the arc of recession is of a smooth and wiping and not a chattering action. The action of the sliding can be nicely illustrated by holding a pencil in a vertical position with the point resting on a desk or table. Then push the pencil forward and at the same time gradually lower the upper end toward you until it has assumed a horizontal position. This phase of the illustration represents the angle of approach or the action of the pinion tooth from the time it enters mesh until contact has been made at its pitch line. In carrying out this illustration with the pencil it will be noted that at the start there is a stubbing or chattering action which gradually disappears as the pencil approaches the horizontal position. With this illustration in mind it is easy to realize why the teeth produce a chattering noise in this part of the mesh. Now continue the forward motion of the pencil and at the same time gradually raise the point toward a vertical position and note that the action is smooth and of a wiping nature. This illustrates the action of a pinion tooth from the time its pitch line has made contact until its outside diameter crosses the line of pressure and it ceases to be in mesh.

The total arc of action is greater than can be obtained with teeth of the Brown and Sharpe proportions. This is due to the elimination of interference and the increased pinion addendum. Referring to Figures 3 and 4, the true arc of action is indicated between the points connected by the line of contact. Both figures show the same combination, which is a 69-tooth, 3-pitch gear meshed with a 15-tooth, 3-pitch pinion drawn to an enlarged scale. The amount of interference present in gearing of the Brown and Sharpe proportions is shown in Figure 3. This interference results from using a tooth form in which a part of the gear tooth extends below the interference point, which is the point of tangency between the pressure line and the pinion generating circle. For this reason it is either necessary to relieve both the portion of the pinion tooth profile which is below its generating circle and the portion of the gear tooth profile which extends beyond the generating circle of the pinion, or undercut the pinion tooth, which greatly impairs its strength. As the portion of the pinion tooth below its generating circle is not an involute curve, we cannot expect this part of the profile to transmit uniform angular velocity, which is one of the main essentials of quiet gearing.

Referring to Figure 4, note that the outside diameter of the gear does not extend beyond the interference point, thus making undercutting or a relief of any sort unnecessary. As there is no interference in the long and short addendum type of gearing we have true involute curves on the entire working length of the tooth. This point alone is highly desirable because the involute is a curve which can be duplicated by any manufacturer of cutters whereas the modified involute has never been standardized by cutter manufacturers, and it is a well known fact that gearing produced by cutters from one manufacturer may not interchange satisfactorily with gearing produced by cutters of another make.

From the foregoing discussion it is reasonable that we may summarize the advantages of long and short addendum gearing as following:

(1) Increased life due to increased strength and true involute profiles. True involute profile transmit uniform angular velocity.

(2) Smoother operating and, therefore, quieter gearing due to the increased arc of action.

(3) Greater flexibility in the design of motors due to the ability to use finer pitch gearing or a smaller number of teeth in the pinions.

(4) Ease of cutter duplication.

The above advantages indicate a marked improvement in the field of gearing. This type of long and short addendum gearing has been in service about four years and is adequately meeting the increased demands of service.

As this type of gearing is not interchangeable with gearing of the Brown and Sharpe or any other tooth proportions neither gear or pinion should be meshed with the gear or pinion of other tooth shapes. In order to avoid this possibility two grooves, each 1/8" wide and 1/16" deep, have been machined in the outside diameter of the teeth at a distance of approximately 1/2" from the wheel side. These grooves are cut in both gears and pinions so that it is only necessary to make sure that the grooves are in both members of the combination in order to obtain proper meshing conditions".

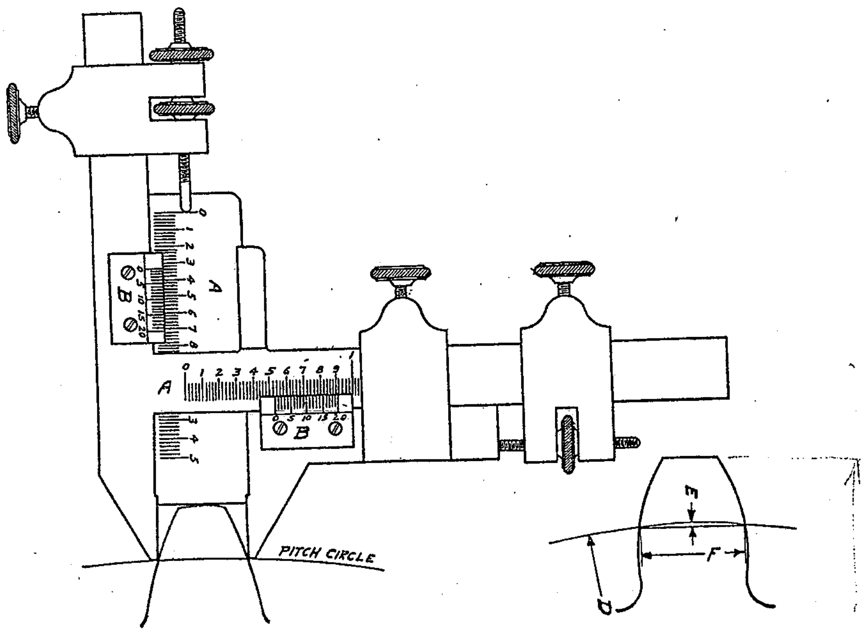
All long and short addendum gears and pinions can be identified by two grooves cut on face of the teeth near the wheel side, each groove being about 1/8" wide and 1/16" deep.

W. A. FALLON

WAF:SSM

COMMERCIAL ENGINEERING DEPT.

MEASURING GEAR AND PINION TEETH



It is now the practice of many operating companies to keep records of mileage, wear of teeth and operating conditions of the gears and pinions they have in service in order to obtain accurate information on the relative wear and life of different grades. The measurement of teeth requires considerable care, and the use of a suitable instrument for doing so, otherwise incorrect and misleading data may be obtained. The following method of measuring teeth and the use of the tooth Vernier referred to may be recommended by agents to operators desiring information on the best method of measuring wear.

The tooth Vernier made by the Brown and Sharpe Mfg. Company and shown in the above cut is a satisfactory one to use. The sliding jaw moves upon a bar graduated to read, by means of a Vernier, to thousandths of an inch. A tongue moving at right angles to the jaws, is graduated in the same manner. Scales shown at A are graduated as follows: .02", .04", .06", .08", .1", .12", etc. while those at B .001", .002", .003", etc. To set either the tongue or jaws at say .333 move scale A until 0 of scale B is opposite the first division to the right of 3 or .3. The distance between jaws will then be equal to .3" plus .02" or .32". Now count off thirteen divisions on scale B (which will be equal to .013") and move scale A until this thirteenth division registers with the first division next on its right of scale A which will equal .32" plus .013" or .333".

Gears and pinions should be measured before bein placed in service as follows: Number or with prick punch mark each gear and pinion for future identification. With the tongue of the gauge set equal to the addendum (theoretical distance between top of tooth and pitch circle .333" for 3 pitch and .4" for 2-1/2 pitch) measure a tooth at three points across its face; viz. next to motor, center of face and outside.

Measure two other teeth at approximately 120° apart in the same manner, making a total of nine measurements on each gear and pinion, the average of which will give the average tooth thickness. The wear is invariably the greatest on the side next motor. After one year's service take another reading in the same manner, not necessarily the same teeth but the tongue must be set at the original depth. The average thickness of the teeth at the second reading can then be deducted from the average original thickness (of the same gear or pinion) and the wear per 1,000 car miles estimated.

While the above method of measuring teeth is satisfactory for comparative wear, it is not suitable for checking the thickness for the teeth at the pitch line. All gear and pinion manufactures allow their workman considerable variation on the diameter over top of teeth, therefore the outside diameter must first be determined (difficult if odd number of teeth) and the variation added to or deducted from the addendum for setting the tongue of the gauge.

In addition to this variation the E dimension (shown on accompanying sketch)

N= Number of teeth
D= Pitch diameter
$E= D/2\ (1-\cos 90°/N)$

should be added to the addendum. The theoretical thickness of the tooth is a circumferential distance while that shown by the gauge is a chordal dimension. The length of the chord can be obtained by the following formula:

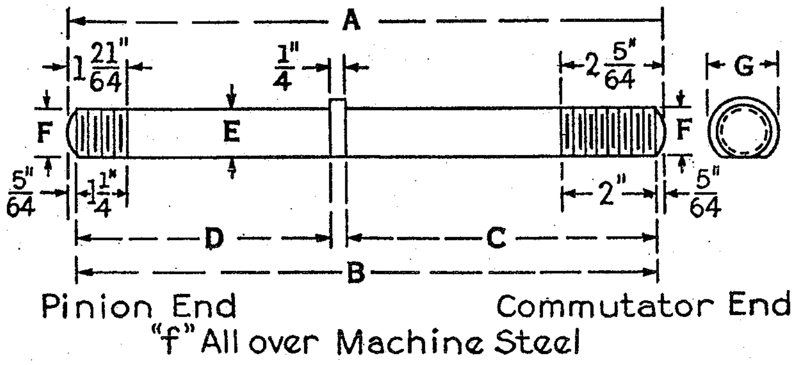
N= Number of teeth
D= Pitch diameter
$F= D \sin 90°/N$

This difference is, however, very slight and no account of it is taken in checking comparative wear.

ARMATURE REMOVING TOOLS

Bolts for Clamping Armatures While Removing Armature Shaft

Three bolts with standard hexagon nuts, and one clamping ring are required for clamping an armature.



	DIMENSION IN INCHES							
MOTOR	A	B	C	D	E	F	THREADS PER INCH	G
GE-200 A-B-C-D-E-F-G-H I-J-K-L-M-N-	19-3/32	18-15/16	9	9-11/16	5/8	5/8	11	13/16
GE-201 A-B-L	22-29/32	22-3/4	9-3/8	13-1/8	3/4	3/4	10	15/16
GE-201 D-E-F-G-H-I-J-K-	22-17/32	22-3/8	9-3/8	12-3/4	5/8	5/8	11	13/16
GE-203 A-B-P-Q-R	19-21/32	19-1/2	9-1/2	9-3/4	3/4	3/4	10	15/16
GE-203 C-D-E-G-H-I-K-L-M-N-O	21-27/32	21-11/16	10-1/16	11-3/8	5/8	5/8	11	13/16
GE-221 A-B	26-1/32	25-7/8	11-7/16	14-3/16	3/4	3/4	10	15/16
GE-222 A-C-D-F-G	26-19/32	26-7/16	11-11/16	14-1/2	3/4	3/4	10	15/16
GE-225 B-C	24-7/32	24-1/16	11-9/16	12-1/4	3/4	3/4	10	15/16

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DIMENSION IN INCHES

MOTOR	A	B	C	D	E	F	THREADS PER INCH	G
GE-227 A	22-25/32	22-5/8	9-5/8	12-3/4	3/4	3/4	10	15/16
GE-230 B	24-5/32	24	9-1/2	14-1/4	5/8	5/8	11	13/16
GE-233 A	26-5/32	26	11-1/8	14-5/8	3/4	3/4	10	15/16
GE-235 A	36-25/32	36-5/8	14-7/8	21-1/2	3/4	3/4	10 Whit.	15/16
GE-237 A	29-5/32	29	16-1/2	12-1/4	3/4	3/4	10 Whit.	15/16
GE-239 A-B-C	27-7/32	27-1/16	12-9/16	14-1/4	7/8	7/8	9	1-1/8
			thread back 2-1/4 in. on "D" end.					
GE-240 A-B-C-D-E-F-G	25-13/32	25-1/4	12-1/2	12-1/2	3/4	3/4	10	15/16
GE-241 A-B-C	24-1/32	23-7/8	15-3/4	7-7/8	1/2	1/2	13	11/16
GE-242 A-B-C	23-11/32	23-3/16	10-3/16	12-3/4	5/8	5/8	11	13/16
GE-245 A	21-31/32	21-13/16	10-3/16	11-3/8	5/8	5/8	11	13/16
GE-246 A-B	21-43/64	21-1/2	10-1/4	11	5/8	"C" END 5/8	11	1
						"D" END 3/4	10	
GE-247 A-B-C-D E-F-G-H I-J-K	22-13/32	22-1/4	14-1/2	7-1/2	1/2	1/2	13	11/16
GE-248 A-B	25-29/32	25-3/4	10-1/2	15	3/4	"C" END 7/8	9	1-1/8
						"D" END 3/4	10	
GE-249 A	17-29/32	17-3/4	10-1/8	7-3/8	5/8	5/8	11	13/16
GE-252 A	17-9/32	17-1/8	7/1/8	9-3/4	1/2	1/2	13	11/16
GE-254 A-B-C	27-21/32	27-1/2	12-1/2	14-3/4	3/4	3/4	10	15/16
GE-258 A-B-C-D-E-F G-H-I-J-K-L-M	16-15/32	16-5/16	7-1/16	9	1/2	1/2	13	11/16

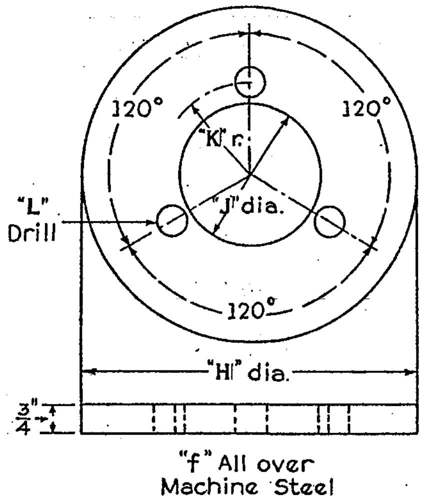
(CONTINUED FROM PREVIOUS PAGE)

DIMENSION IN INCHES

MOTOR	A	B	C	D	E	F	THREADS PER INCH	G
GE-259 A-B-C	24-29/32	24-3/4	11-1/4	13-1/4	5/8	5/8	11	13/16
GE-260 A-B-C-D	25-29/32	25-3/4	10-1/2	15	3/4	"C" END 7/8	9	1-1/8
						"D" END 3/4	10	
GE-261 A-C	16-15/32	16-15/16	7-1/16	9	1/2	1/2	13	11/16
GE-262 A	16-15/32	16-15/16	7-1/16	9	1/2	1/2	13	11/16
GE-263 A	22-25/32	22-5/8	9-5/8	12-3/4	3/4	3/4	10	15/16
GE-264 A-B-C-D-E	16-15/32	16-15/16	7-1/16	9	1/2	1/2	13	11/16
GE-265 A-B-C-D-E-F-G-H-J-K-L	17-31/32	17-13/16	7-3/4	9-13/16	1/2	1/2	13	11/16
GE-266 A	16-13/32	16-1/4	7-7/8	8-1/8	1/2	1/2	13	11/16
GE-269 A-B	18-9/32	18-1/8	8-3/4	9-1/8	5/8	5/8	11	13/16
C	18-9/32	18-1/8	8-3/8	9-1/2	5/8	5/8	11	13/16
270-A	18-9/32	18-1/8	8-3/4	9-1/8	5/8	5/8	11	13/16
E-272-A	26-25/32	26-5/8	11	15-3/8	3/4	3/4	10	15/16
GE-273 A-B	17-31/32	17-13/16	7-3/4	9-13/16	1/2	1/2	13	11/16
GE-275 A-B-C-D, E-F-G-H, I-J-K; GE-299	22-19/32	22-7/16	9-1/8	13-7/16	1/2	1/2	13	11/16
GE-282 A-B-C-D-E	22-29/32	22-3/4	9	13-1/2	5/8	(5/8 N (7/8 M	11) 9)	1-1/8
GE-292 A	2′ 3-1/2″	2′ 2-7/8″	13	13-5/8	3/4	3/4	10	15/16
GE-288 A; GE-294 A	22-13/32	22-1/4	14 1/2	7-1/2	1/2	1/2 W. 1/2 W.	13	11/16
GE-296 A; GE-297 A	22-29/32	22-3/4	9	13-1/2	5/8	(5/8 N (7/8 M	11) 9)	1-1/8

ARMATURE REMOVING TOOLS (Cont'd)

Clamping Rings for Clamping Armatures While Removing Armature Shaft



DIMENSIONS IN INCHES

MOTOR	H	J	K	L
GE-200 A-B-C-D E-F-G-H I-J-K-L	8-1/8	4-5/16	2-17/32	21/32
GE-201 A-B	9-1/4	4-3/8	2-13/16	25/32
GE-201 D-E-F-G H-I-J-K	9-1/4	4-7/8	2-29/32	21/32
GE-203 A-B	8	4-1/4	2-11/16	25/32
GE-203 C-D-E-G H-I-K-L M-N-O	8-1/2	4-5/8	2-3/4	21/32
GE-221 A-B	10-1/2	5-1/4	3-1/4	25/32

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MOTOR	DIMENSION IN INCHES H	J	K	L
GE-222 A-C-D F-G	10-1/2	5-1/4	3-1/4	25/32
GE-225 B-C	10-1/2	5-1/4	3-1/4	25/32
GE-227 A	9-1/4	4-7/8	2-29/32	25/32
GE-230 B	9-1/4	4-7/8	2-29/32	21/32
GE-233 A	9-1/4	4-15/16	3	25/32
GE-235 A	16	6-3/4	4-1/4	25/32
GE-237 A	14-5/8	6	3-3/4	25/32
GE-239 A-B-C	11-1/4	6	3-3/4	29/32
GE-240 A-B-C	10	5-1/4	3-1/4	25/32
GE-241 A	8-1/2	4-5/8	2-5/8	17/32
GE-242 A-B	9-1/2	5	2-29/32	21/32
GE-245 A	8-1/2	4-5/8	2-3/4	21/32
GE-246 A-B	8-1/4	4-1/16	2-11/16	25/32
GE-247 A-B-C-D	6-7/8	3-1/2	2-1/8	17/32

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MOTOR	H	J	K	L
	DIMENSIONS IN INCHES			
GE-248 A-B	9-1/4	5-5/8	3-11/16	25/32
GE-249 A	9-5/8	4-3/8	2-9/16	21/32
GE-252 A	6-1/2	3-3/4	2-1/8	17/32
GE-254 A-B-C	10-1/2	5-1/4	3-1/4	25/32
GE-258 A-B-C-D E-F-G-H I-J-K-L M	5-5/8	3	1-7/8	17/32
GE-259 A-B-C	7-7/8	4-3/4	3-3/8	21/32
GE-260 A-B-C-D	9-1/4	5-5/8	3-11/16	25/32
GE-261 A-C	5-5/8	3	1-7/8	17/32
GE-262 A	5-5/8	3	1-7/8	17/32
GE-263 A	9-1/4	4-7/8	2-29/32	25/32
GE-264 A-B-C-D E	5-5/8	3	1-7/8	17/32
GE-265 A-B-C-D E-F-G-H J-K-L	6-3/4	3-1/4	2-1/8	17/32

(CONTINUED FROM PREVIOUS PAGE)

MOTOR	DIMENSIONS IN INCHES H	J	K	L
GE-266 A	5-5/8	3	1-7/8	17/32
GE-269 A-B-C	9-5/8	4-3/8	2-9/16	21/32
GE-270 A	9-5/8	4-3/8	2-9/16	21/32
GE-272 A	10-1/2	5-1/4	3-1/4	25/32
GE-273 A-B	6-3/4	3-1/4	2-1/8	17/32
GE-275 A-B-C-D E-F-G-H I-J-K	8	4	2-9/16	17/32
GE-282 A-B-C-D E	13-3/4	6	5-3/4	21/32
GE-288 A-B	6-7/8	3-1/2	2-1/8	17/32
GE-292 A	12-1/2	5-5/8	3-7/16	25/32
GE-294 A	6-7/8	3-1/2	2-1/8	17/32
GE-296 A	13-3/4	6	5-3/4	21/32
GE-297 A	13-3/4	6	5-3/4	21/32
GE-299 A	8	4	2-9/16	17/32

GEARING

The General Electric Company manufactures several grades of gearing to meet the varied requirements of present day railway operating conditions.

Grade "F" gears and pinions are used in heavy traction service because of their great toughness.

Grade "M" gears and pinions which are considerably harder but not quite so tough as grade "F" are used in lighter service.

Grade "A-1" pinions are a recent development and will replace grade "M" pinions on light weight equipments. This grade provides a wearing surface harder than grade "M" and with a toughness comparable with grade "F". This grade costs slightly more than grade "M" but its longer life should more than justify its use.

Solid gears are recommended in all cases although split gears can be supplied if required.

Successful operation of gearing is largely dependent upon its proper installation and the maintenance of equipment parts affecting the gear center distance. Instructions on pinion mounting and removal will be found in descriptive sheet GEA-776-A.

The removal of pinions should not be accomplished by the application of heat or wedges between the pinion and bearing housing, but by means of a pinion puller which grips all the teeth evenly.

Gears should be pressed on the axle. The use of a key for solid gears is unnecessary as the press fit is sufficient to deliver the torque of the motor.

For a press fit the gear bore should be less than the axle diameter by the following amounts.
Grade........"F"........1 mil per inch axle diam.

Grade......B, M, L, and K........1 1/2 mils per in. axle diameter.

The gear bore and axle should be carefully cleaned and white leaded in order to prevent abrasion of surfaces.

It is recommended that customer purchase gears finish bored as this operation is accomplished at the factory by centering the gear from the pitch line, insuring concentricity of bore and pitch line, which is one of the essentials of quiet gearing. Many customers who order rough bored gears, center the gear for reboring from this rough bore, which may not be concentric with the pitch line. It is, therefore, quite possible that these customers are not obtaining the best service even when gearing is first put in operation.

As bearings wear allowing the centers to spread, contact between gear and pinion teeth does not occur on the pitch line and causes uneven wear on the tooth surface. This condition also tends to concentrate the load at the ends of the teeth resulting in high stresses which are not equally distributed throughout the tooth section.

The combined wear of armature and axle bearings should not exceed 3/16 in. for gearing having diametral pitches up to 3, and correspondingly less for the finer pitches.

Proper lubrication is a very important item in the life of gearing. A good grade of grease of a leveling back consistency should be maintained at a sufficient level in the gear case so that the gear teeth will be immersed. Every possible precaution should be exercised to keep dirt from entering the case.

The General Electric Company manufactures gearing of various tooth forms; but a full-depth involute tooth with pinion of long addendum proportions and gear of short addendum proportions, is recommended for quietness and strength.

This type is standard for new equipments and is suggested for replacement where both gear and pinion are required.

Helical gearing can be furnished but is not recommended because of objectional end thrust on linings and thrust collars. In reducing end thrust to a reasonable degree the angle of helix must be decreased to such an extent that the advantage of quiet operation, obtained with a large helix angle, is lost.

In railway work the long and short addendum tooth is comparable in quietness with the helical gearing and has the advantage of not causing excessive end thrust.

Several bulletins containing valuable information on the above subject are:

List of grades of gearing, qualities, and functions; meshing combinations; construction of various types. Some details of high-grade finish tooth form and operating data GEA-194-A

Advantages of solid over split gears GEA-853

Removing and replacing pinions GEA-776-A

Stress distribution in pinions GEA-95

Advantages of long and short addendum teeth GEA-713

Gears and pinions for railway and industrial haulage motors GEA-1054

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Stress Distribution in Electric Railway Motor Pinions as Determined by the Photo-elastic Method

By PAUL HEYMANS and A. L. KIMBALL, Jr.

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Stress Distribution in Electric Railway Motor Pinions as Determined by the Photo-elastic Method

By Paul Heymans, Cambridge, Mass.
and A. L. Kimball, Jr., Research Laboratory, General Electric Company

I. DESCRIPTION OF THE METHOD

How to Define the State of Stress at any Point of a Solid Body

The state of stress at any point in a solid body is determined when the traction across every plane through the point is known. There exist at any point three orthogonal planes across which the traction is purely normal and which are called the planes of principal stress. The normal tractions across those planes are called the principal stresses. The state of stress at any point is completely determined by the direction and the magnitude of the principal stresses at the point under consideration. The principal stresses, given in direction and in magnitude, express in the most general and complete way the elastic state at any given point. The bending moment, the shearing forces, etc., are readily deduced from the direction and the magnitude of the principal stresses. Furthermore, one of the principal stresses always expresses the maximum stress.

2. The notion of principal stress may be illustrated as follows:

3. Consider a spherical element in a solid body. External applied loads will deform this spherical element into an ellipsoidal element (Fig. 1). The axes of this ellipsoid will correspond in direction and in magnitude to the direction and the magnitude of the principal stresses.

4. The orientation and the form of the ellipsoid, and therefore the direction and the magnitude of the principal stresses, will define the state of stress at the point under consideration.

[1] A complete theory of stress and strain may be found in the "Treatise on the Mathematical Theory of Elasticity," by A. E. H. Love, 3d ed., chapters i-iv.

5. The axes of the ellipsoid represent the largest and the smallest deformation at the point under examination. Correspondingly, the principal stresses give the direction and the magnitude of the maximum and the minimum stress.

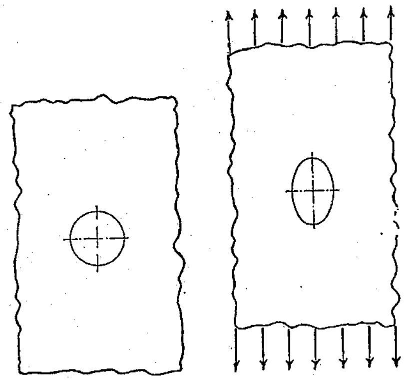

Fig. 1. Ellipsoidal Element Resulting from Subjecting a Spherical Element to Stress

6. If the three principal stresses vary from point to point in the structure, the problem to be dealt with is a three-dimensional elastic one. If one of the three principal stresses vanishes throughout, it is a two-dimensional elastic or plane-stress problem.

7. Corresponding to the three- and two-dimensional elastic-stress problems there are also the three- and two-dimensional elastic-strain problems, when the deformations corresponding to the principal stresses are considered.[1]

8. A great number of structural problems (bridge, ship, airplane, plate, dam, etc., construction) are, or their stress analysis may be reduced to, two-dimensional elastic problems.

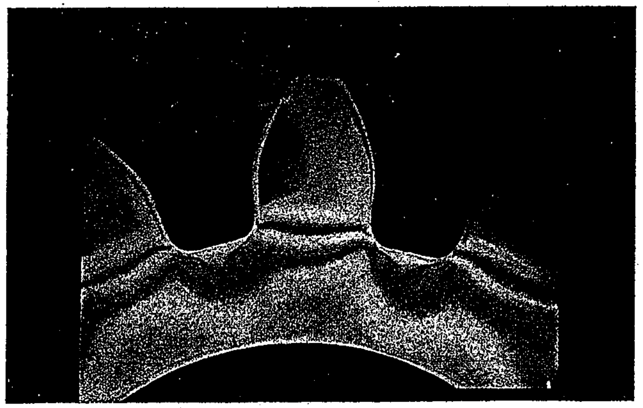

FIG. 2. COLORED IMAGE WHEN NORMAL INSIDE PRESSURE ALONE IS APPLIED

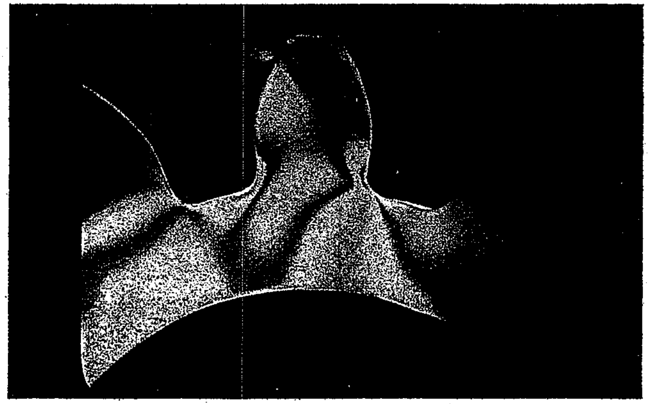

FIG. 5. COLORED IMAGE OBTAINED FOR NORMAL INSIDE PRESSURE AND REDUCED TORQUE

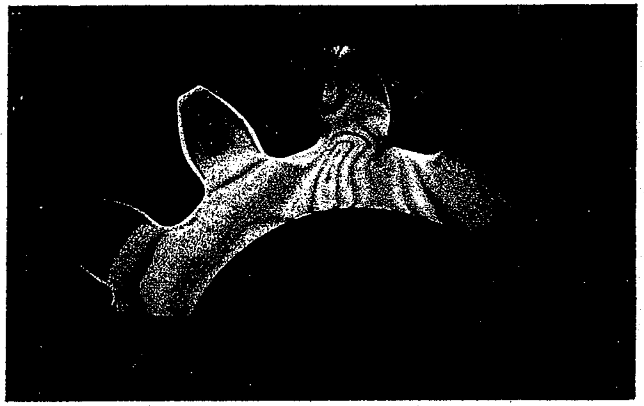

FIG. 3. COLORED IMAGE WHEN BOTH NORMAL INSIDE PRESSURE AND MAXIMUM TORQUE ARE APPLIED

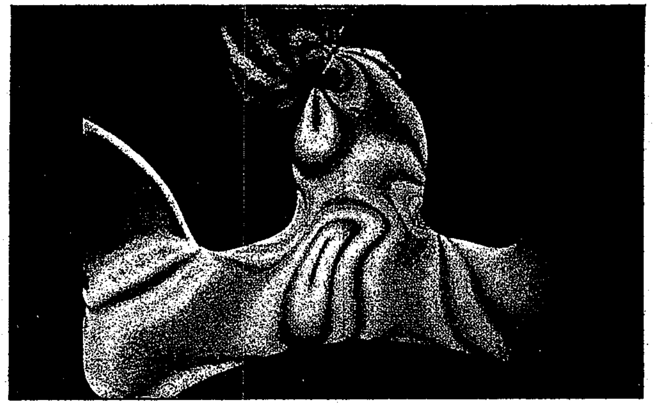

FIG. 4. COLORED IMAGE WHEN BOTH NORMAL INSIDE PRESSURE AND MAXIMUM TORQUE ARE APPLIED

The Photo-elastic Method of Stress Determination

9. As set forth in Par. 1, the state of stress at any point is most completely defined by the direction and the magnitude of the principal stresses. These are, therefore, the elements which we wish to determine for a complete analysis.

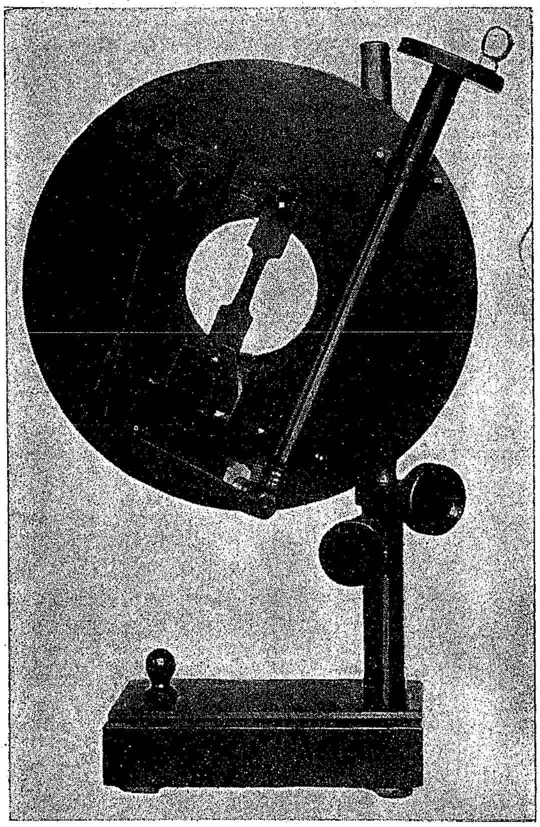

Fig. 6. Frame for Comparison Member Designed by E. G. Coker and A. L. Kimball, Jr.

10. The photo-elastic method solves the two-dimensional elastic problems. It primarily takes advantage of the double refracting properties shown by isotropic transparent substances when put under stress. The stresses in the structure may therefore be determined from models made of a homogeneous transparent material, and ordinarily on a reduced scale. The stresses in a steel, cement, or any other structure, homogeneous throughout and obeying Hooke's law of linear proportionality between stress and strain, may be readily deduced from the values obtained by the analysis of the corresponding transparent model for the case of two-dimensional elastic problems.

11. If plane polarized light is passed through a stressed specimen of celluloid and afterward through a second nicol prism whose principal section is parallel to the plane of polarization of the original beam of light, only the points where the principal stresses are respectively parallel and perpendicular to the principal sections of the crossed nicols remain dark. This result makes it possible to determine the directions of the principal stresses at any given point. Morever, this information is needed for the measurements which will be described later.

12. If now circularly polarized light be passed through the specimen, by interference of the two component rays, which in the double-refracting specimen have suffered a relative retardation at each point proportional to the difference in magnitude of the two principal stresses, a colored image is obtained. (Figs. 2, 3, 4 and 5.)*

13. By a comparison method, based upon the interposition in the proper direction of a comparison member of constant cross-section, put under uniform tension in a suitable frame (Fig. 6), the value of the difference of the principal stresses at any given point may be read on the dynamometer of the frame.

14. Now, in the two-dimensional elastic problems the transverse deformation, i.e., the deformation along a normal to the plane of the two principal stresses, is proportional to the sum of those two stresses. By means of a lateral extensometer (Fig. 7), we measure this transverse deformation.

15. From the values of the differences and the sums of the principal stresses, the separate values of each of them are computed, thus determining completely the state of stress.

16. A question naturally arising is whether the results obtained on a transparent body such as celluloid hold for structural materials.

17. It is shown by the general discussion of the equations of elastic equilibrium that in the case of strain or plane stress in an isotropic body obeying Hooke's law of linear proportionality between stress and strain, the stress distribution is independent of the moduli of elasticity and consequently of the material of which the body is made. Thus the stress distribution experimentally determined in the case of a celluloid body is the same as it is when the body is made of any other isotropic substances such as iron, steel, etc., obeying Hooke's law, in distribution,

* See Frontispiece of this issue of the REVIEW.

direction, and magnitude.[1] Moreover these conclusions derived from the general theory of elasticity have been checked by experiment.[2]

18. The photo-elastic method can be applied to the great majority of structural problems, not only in taking the place of mathematical computation, but particularly in solving those structural problems where mathematics becomes too involved to be of help. Moreover it has the great advantage of giving the maximum stress at each point throughout the whole structure, and it therefore offers an effective means of increasing safety and reducing superfluous material.

II. A STUDY OF THE STRESS DISTRIBUTION IN GEAR PINIONS

19. When accidents occur with gear wheels, besides the metallurgical question, three possible causes of failure suggest themselves:

a. The gear wheel may not have been properly designed.

b. It may have failed under an excessive load.

c. When the pinion was shrunk hot or forced on to a tapered shaft, an excessive inside radial pressure may have been set up.

20. It is easy to see that the ordinary methods of resistance calculations of gear wheels, based on considering the tooth as a cantilever loaded at its end, would not be expected to give reliable and complete information as to stress distribution, not even for the root section of the tooth which is under consideration.

21. Indeed, the shape of the tooth, the curvature at the root, the ratio of the diameter of the pinion bore to the root and outside diameter, the permanent stresses introduced by the placing of the pinion on the shaft, etc., all affect the stress distribution and the maximum stress. Photo-elastic analysis shows that these factors affect the stresses considerably more than would be expected from present methods of estimating. For standardized pinions the correction coefficients can only partially take account of these factors. For special pinions or for pinions of which more efficient running is required, a photo-elastic analysis seems to be the best if not the only effective way to determine the stress distribution and to locate the maximum stress.

22. A detailed analysis of the stress distribution determined for different gear pinions

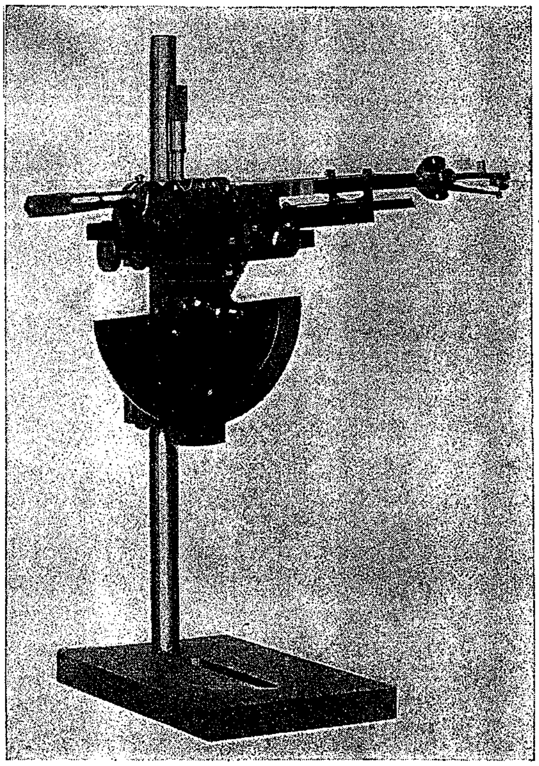

Fig. 7. Lateral Extensometer Designed by P. Heymans

and under different loading conditions is given below.

23. The authors wish first to call attention to certain interesting points brought out by photo-elastic analysis, which have been checked by tests carried out on steel sections. These are particularly interesting because they are unexpected.

24. Besides the stress distribution in the different sections of the pinions represented by Fig. 8, the photo-elastic analysis has given as maximum stress under normal inside radial pressure and normal torque:

80,000 lb. per sq. in. for tooth form *A*
70,350 lb. per sq. in. for tooth form *B*
60,900 lb. per sq. in. for tooth form *C*.

Moreover the 12-tooth pinion shows, besides a smaller maximum stress, a better stress distribution.

[1] Except, however, if the body is multiply connected and the resultant applied forces do not vanish separately over each boundary. In this particular case the correction coefficients for passing from one isotropic substance to another may be experimentally determined. ("On Stresses in Multiply-connected Plates," by L. N. G. Filon, British Assn. Report, 1921.)

[2] "Photo-elastic Measurements of the Stress Distribution in Tension Members Used in the Testing of Materials," by E. G. Coker, Excerpt Proc. Inst. C. E. (London),—vol. ccvii, part II, p. 8.

"Photo-elastic and Strain Measurements of the Effects of Circular Holes on the Distribution of Stress in Tension Members," by E. G. Coker, Trans. Inst. Engrs. & Shipbuilders in Scotland, vol. lxiii, part I, p. 33.

"La Photo-elasticimétrie, ses principes, ses méthodes et ses applications," by Paul Heymans. Bull. Soc. Belge Ing. et Ind., Aug., 1921, pp. 147-154, 165-167, 189-199.

25. For steel pinions the maximum stress attained under normal conditions, although high, appears not to be excessive. *Tooth C appeared to be a better design* under normal conditions.

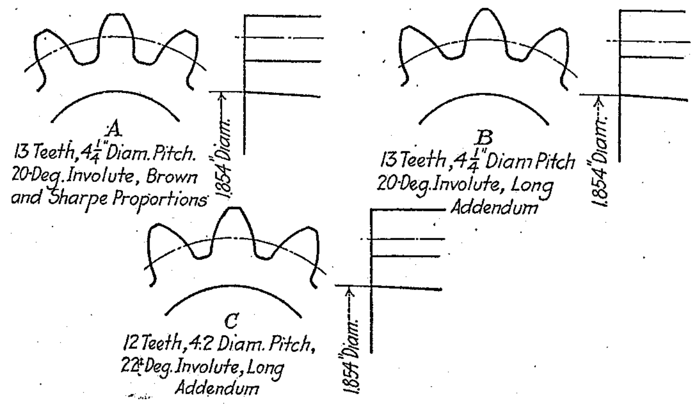

Fig. 8. Tooth Forms of Pinions Subjected to Photo-Elastic Analysis

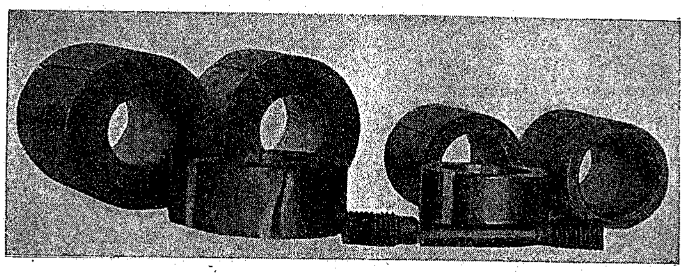

Fig. 9. Steel Rings Ruptured by Being Forced onto a Tapered Plug

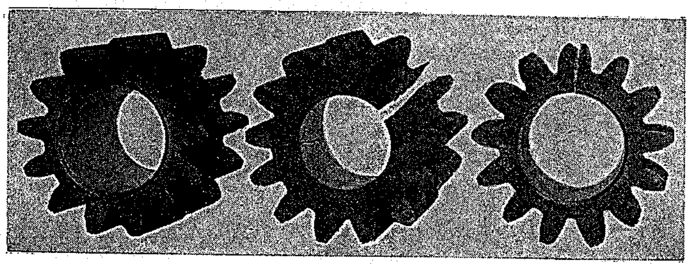

Fig. 10. Steel Pinions Ruptured by Being Forced onto a Tapered Plug

26. The stresses due to shrinking or forcing the pinion on the shaft can only be estimated. The pinion may be assumed to be a plain circular ring, for which case the stresses may be mathematically computed. The stress at any point of the ring as well as the maximum stress in the ring depends upon the lengths of the inside and outside radii. The opinion generally expressed is that for the case of the pinion the maximum stress will be intermediate between the maximum values obtained for rings of which the outside diameters are respectively equal to the root diameter of the tooth and to the outside diameter of the pinion, the inside bore being the same.

27. Photo-elastic analysis shows that *the gear pinion is even weaker than the plain circular ring whose outside diameter is equal to the root diameter of the tooth.* The change of external profile, due to the presence of the teeth, although requiring an addition of material, weakens the structure.

28. Figs. 9 and 10 show the steel specimens after having been tested by forcing a tapered plug into the bore; and Table I gives the rupture load applied to the tapered arbor forced into the bore for the different specimens. These confirm the photo-elastic results.

29. Previous to the photo-elastic investigation of the stresses due to radial inside pressure in pinion sections, fracture due to pure radial inside pressure would have been expected to occur through the minimum radial cross-section.

30. From Fig. 2,* representing the color image obtained in the photo-elastic analysis, it appears that *the regions under the teeth are under higher stress* and that the points at the inside boundary right under the teeth are points of maximum stress.

31. Fig. 10 gives the fracture obtained on steel sections. Two of the sections show fractures right through the thickest layer of material, while all of them started at points where the photo-elastic analysis had revealed maximum stress. The unevenness of the material must account for the deviation of the fracture in one of the cases.

32. Can any statement be made as to the causes of the failure by inspection of the shape of the fracture? In the case in which the authors were interested, the photo-elastic analysis determined the best design. As before said, either the placing of the pinion on the shaft, if carelessly done, for instance by pounding the pinion heavily on the tapered

* See Frontispiece of this issue of the REVIEW.

shaft, or excessive torque and blows due to sudden meshing or the taking on of a heavy load, will set up dangerous stresses.

TABLE I

RUPTURE LOAD ON ARBOR FORCED INTO SPECIMENS TESTED

	Inside Diam., In.	Outside Diam., In.	Root Diam., In.	Rupture Load, Lb.
Ring	1.854	3.5	....	85,000
Ring	1.854	2.5	....	51,000
Pinion	1.854	3.5	2.5	47,000

33. The authors' photo-elastic analysis has shown that *the sections of dangerous stresses are different for different values of inside radial pressure and applied torque load.*

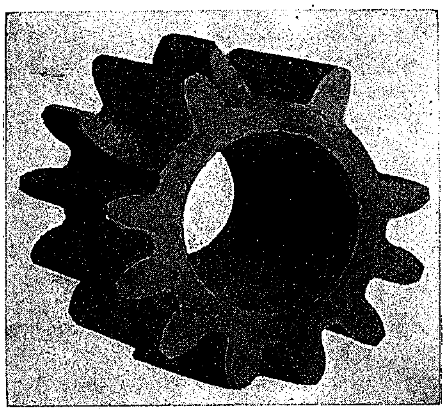

Fig. 11. Fatigue Failures of Teeth Produced by Experiment (without Radial Pressure in Bore)

34. The fracture shown in Fig. 11 is of an *open* V-shape. Photo-elastic analysis shows that *the higher the inside radial pressure becomes*, for a given torque load, *the sharper becomes the V-shape of the section of dangerous stresses.* (Fig. 12.) If the fracture is due to too high a torque load, the angle of the V will approach 180 deg. Tests on steel sections have been made with a specially built impact machine.

35. Without inside radial pressure the fracture obtained is a straight line through the root section of the tooth. With increasing pressures the V-shaped fracture becomes sharper. For an inside radial pressure exceeding the elastic limit, however, the observation does not hold. The reason for this departure from what the photo-elastic method had predicted is to be found in the fact that beyond the elastic limit the stress-and-strain relation no longer follows Hooke's law. Therefore the stresses set up in the steel pinions by the shrinking process no longer correspond with those set up in the celluloid model.

36. While the flat shape of the break in Fig. 11 is one limiting case (torque without radial shrinking pressure), Fig. 10 may be considered as the other limiting case (radial shrinking pressure without torque), showing a V-shaped fracture for which the angle of the V has become equal to zero.

37. It may be concluded, then, that the inspection of the fracture may be a means of determining the cause of the failure. In this way, possibly, the responsibility may be established between builder and customer as regards pinion mounting.

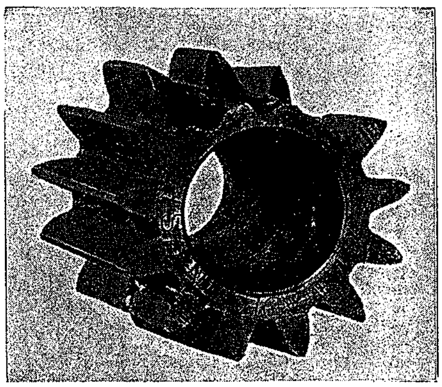

Fig. 12. Fatigue Failures of Teeth Produced by Experiment (with Heavy Radial Pressure in Bore)

The Detailed Stress Analysis

38. *External Forces Applied to the Pinion When in Service.* The pinion is shrunk onto the shaft after having been bored so as to fit the shaft at a temperature of 160 deg. F. above normal room temperature.

39. In normal working conditions, the torque load to which the pinion is subjected corresponds to a tractive force of 500 lb. per inch of face of the tooth, tangent in direction to the pitch circle. The whole torque is supposed to be transmitted by a single contact.

40. Calling respectively $\widehat{rr}$ and $\widehat{\theta\theta}$ the radial and the tangential principal stress in a circular ring, of which the outside diameter equals the root diameter of the teeth, the inside bore being the same as the pinion bore, $(\widehat{rr}-\widehat{\theta\theta}) = 28{,}800$ lb. per sq. in. for $\Delta t = 160$ deg. F. This value of $(\widehat{rr}-\widehat{\theta\theta})$ is the stress value of the color bands obtained in polarized light (isochromatic bands), and will therefore be used

in the stress analysis of the celluloid model to secure the right expansion pressure before the torque is applied. For radial pressures higher than this normal shrinking pressure, the same characteristic of the $(\widehat{rr}-\widehat{\theta\theta})$ value will be used.

41. The tangential tractive force is applied at varying distances from the root of the tooth, depending upon the point of contact. The most unfavorable conditions arise when this force is applied at the top of a single tooth. Moreover the starting torque load being higher than that realized under normal running conditions, the applied tractive force was brought up from 500 lb. to 1500 lb. per inch of face.

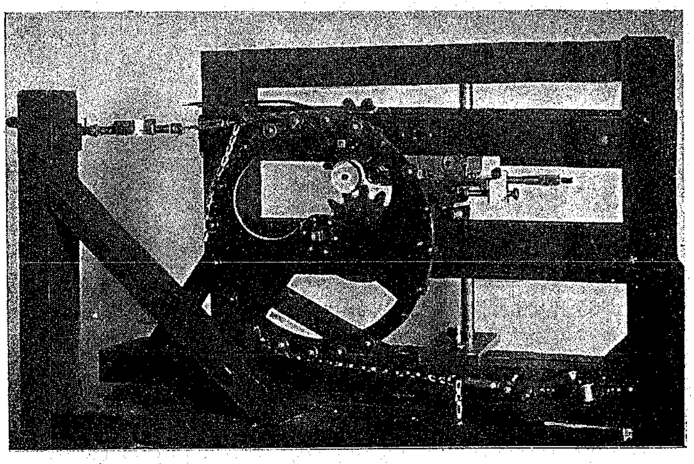

Fig. 13. Frame Used for Applying Loads to Celluloid Models of Pinions

42. Let us for convenience call:

a. The *normal* inside pressure, the value of 28,800 lb. per sq. in. for $(\widehat{rr}-\widehat{\theta\theta})$, corresponding to a shrinking pressure due to a temperature variation of 160 deg. F.

b. The *maximum* torque, the torque corresponding to a tractive load F of 1500 lb. applied normally to the contour of the tooth (condition of contact) at the top of one pinion tooth.

c. The *normal* torque, the torque corresponding to a tractive load F of 500 lb. applied under the same conditions as above.

d. Increased inside pressures, the values of $(\widehat{rr}-\widehat{\theta\theta})$ exceeding the normal inside pressure, as defined above.

43. *The Photo-Elastic Analysis.* Fig. 13 represents the frame used for the loading of the models. A tapered expansion ring is used to produce the radial inside pressure. The torque is measured by properly mounted dynamometers.

* See Frontispiece of this issue of the REVIEW.

44. The first sets of measurements were made under normal inside pressure and maximum torque load. Fig. 14 represents the lines of principal stress, deduced from the isoclinic bands. The tangents to these lines represent at each point the directions of the principal stresses.

45. Fig. 2* gives the colored image when the normal inside pressure alone is applied, whereas Figs. 3* and 4* give the image obtained when both the normal inside pressure and the maximum torque are applied. An optical measurement on the image shown in Fig. 2* allows one to adjust properly the amount of inside pressure before the torque is applied.

46. The determination of the values of the difference $(p-q)$ of the principal stresses is made on the image shown in Fig. 4.* One of the two principal stresses vanishes at a boundary where no external forces are applied. In this case the optical measurements of the values of $(p-q)$ give directly the values of the tangential stress.

47. Inside of the body the optical measurements are supplemented by measuring the transverse change of thickness, which gives the values of the sum $(p+q)$ of the principal stresses.

TABLE II

VALUES OF THE PRINCIPAL STRESSES ACROSS THE MINIMUM CROSS-SECTION OF THE LOADED TOOTH

Tenths of Distance AB (Fig. 15) Measured from A	p Lb. per Sq. In.	q Lb. per Sq. In.
0	0	72,600
0.1	13,850	57,300
0.2	10,450	49,000
0.3	3,710	41,700
0.4	—10,620	25,800
0.5	—20,300	18,700
0.6	—29,000	11,900
0.7	—40,000	9,000
0.8	—51,900	....
0.9	—65,700	5,320
B	—80,000	0

48. From the values of the principal stresses at a given point it is easy to obtain the stress on a section in any given direction at that point. Moreover, as said before, the two principal stresses represent respectively the maximum and the minimum stress. Thus the larger of the principal stresses will always

give at each point the maximum stress in direction and magnitude.

49. At the edges where one of the principal stresses has vanished the values of $(p-q)$ and $(p+q)$ must correspond, i.e., the optical determination of $(p-q)$ and the determination of $(p+q)$ must check.

50. Also if we know the total force acting normally to a given section, the graphical integral of the curve, obtained by plotting the resultant stresses acting normally to this section, must correspond to the total force. In the case of the pinions the data for such a check are not available.

51. Table II gives the values of the principal stresses through the minimum cross-section of the pinion tooth, to which the load is applied. The results given in this table have been plotted in Fig. 15. At each point where measurements have been made the two principal stresses have been plotted in direction and in magnitude, the arrows serving to distinguish between tension and compression. At the points A and B, $(p-q)$ and $(p+q)$ must check: they differ for A by 0.9 per cent and for B by 0.8 per cent.

52. The maximum tension occurs at A and is equal to 72,600 lb. per sq. in. The maximum compression occurs at B and is equal to 80,000 lb. per sq. in. This difference between the absolute values of these stresses is of course due to the pressure on the inside of the pinion, which affects the tension and the compression stresses differently.

53. Figs. 16 and 17 give the values of the tangential stresses along the edge of the tooth on which the load is applied. The numerical results of Table III have been plotted in Fig. 16, this table giving the tangential stresses at the tension side. Also the numerical results of

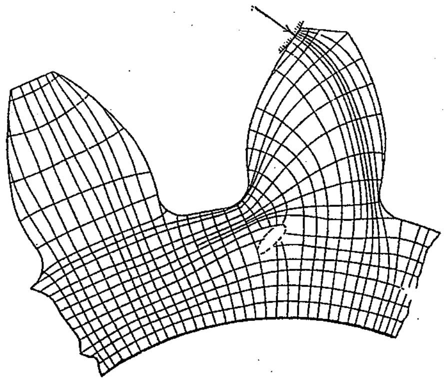

Fig. 14. Lines of Principal Stress Determined by Polarized Light—Normal Inside Pressure and Maximum Torque Load

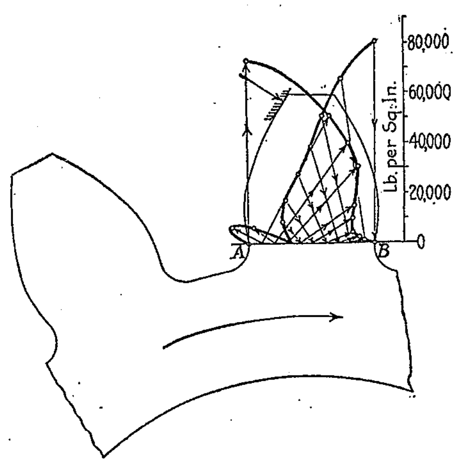

Fig. 15. Curves showing the Two Principal Stresses in Direction and Magnitude for Points along the Section AB

TABLE III

VALUES OF THE TANGENTIAL STRESS AT THE BOUNDARY OF THE LOADED TOOTH—TENSION SIDE

No. of Point in Fig. 16	q Lb. per Sq. In.	
1	41,000	
2	54,100	
3(A)	72,300	72,750[1]
4	73,200	
5	64,800	
6	57,600	
7	54,100	
8	41,000	
9	....	

[1] Value obtained by taking ½ $[(p+q)+(p-q)]$, the other values being $(p-q)$ measurements.

TABLE IV

VALUES OF THE TANGENTIAL STRESS AT THE BOUNDARY OF THE LOADED TOOTH—COMPRESSION SIDE

No. of Point in Fig. 17	p Lb. per Sq. In.	
1	20,500	
2	41,000	
3(B)	79,500	80,000[1]
4	80,000	
5	82,200	
6	60,000	
7	29,000	
8	0	

[1] Value obtained by taking ½ $[(p+q)+(p-q)]$, the other values being $(p-q)$ measurements.

Table IV have been plotted in Fig. 17, this table giving these stresses on the compression side. Since no external load is applied at this side, the optical measurements give the values of the tangential stresses up to the top of the tooth.

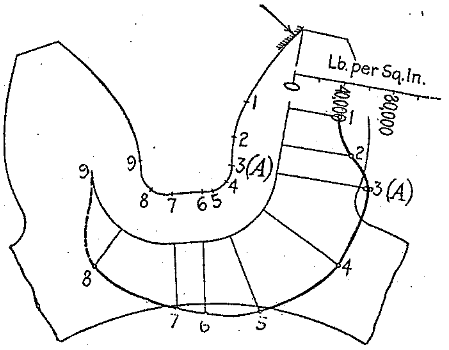

Fig. 16. Tangential Stress at Tension Side—Normal Inside Pressure and Maximum Torque

54. Table V and Fig. 18 give the numerical and plotted values of the stress difference $(\widehat{rr}-\widehat{\theta\theta})$ along the inside boundary of the pinion, the normal inside pressure and the torque load being applied. A circular ring to which a uniform inside pressure is applied will show concentric isochromatic bands. The deflections of those bands (Fig. 2)* in the case of the pinion show the disturbance due to the presence of the teeth.

55. When the maximum torque is applied, the values obtained for $(\widehat{rr}-\widehat{\theta\theta})$ give the curve of Fig. 18. The colored images as well as the diagrams show that the load applied at the top of one tooth extends its influence as far as the inside boundary of the pinion. The combination of the inside uniform pressure, already disturbed by an irregular outside boundary, with irregularly distributed stresses—tensions in certain parts and compressions in others—due to the torque load, do not of course give a resultant stress distribution which shows any symmetry with respect to the point of contact. The upper pinion being the driving pinion, it may be seen on the colored image (Fig. 3)* that the stresses vanish rather rapidly in the withdrawing part, but that the penetration extends much farther into the approaching part.

56. It may also be interesting to point out that there is a zone of zero stress inside of the pinion under the root of the tooth when the torque load is applied. This is shown on the diagram of the lines of principal stress (Fig. 14) by the converging of the lines of principal stress. Where several lines of principal stress intersect, the principal stresses usually vanish.

57. The question of engineering interest was to find the relative influence of the factors which affect the maximum stress, and the authors therefore varied the values of:

a. The inside normal pressure

b. The torque load.

TABLE V

VALUES OF $(\widehat{rr}-\widehat{\theta\theta})$ ALONG THE BOUNDARY OF THE BORE

No. of Point in Fig. 18	$(\widehat{rr}-\widehat{\theta\theta})$ Lb. per Sq. In.
1	36,600
2	54,100
3	36,600
4	18,100
5	41,000
6	61,500
7	43,500
8	38,700

58. The values of $(\widehat{rr}-\widehat{\theta\theta})$ along the inside boundary when the maximum torque load is applied are given in Table VI and have been plotted in Fig. 19 for the case of reduced inside pressure. The colored image did not show noticeable variation across the minimum cross-section AB and along the outside edges of the main tooth. The influence of the inside pressure on the above-mentioned limit does not affect materially the regions of maximum stress, due in this case to the torque load.

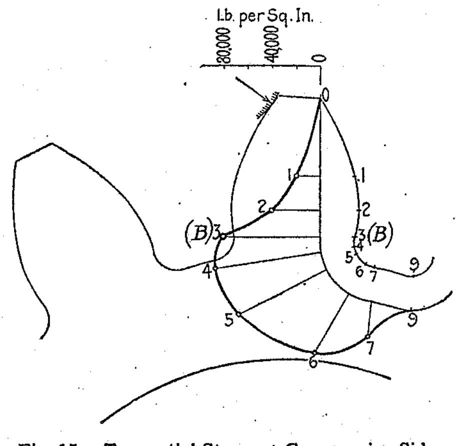

Fig. 17. Tangential Stress at Compression Side—Normal Inside Pressure and Maximum Torque

59. Fig. 5* shows the image obtained for normal pressure and reduced torque. Having applied 0.7 of the maximum torque value, the

* See Frontispiece of this issue of the REVIEW.

Table IV have been plotted in Fig. 17, this table giving these stresses on the compression side. Since no external load is applied at this side, the optical measurements give the values of the tangential stresses up to the top of the tooth.

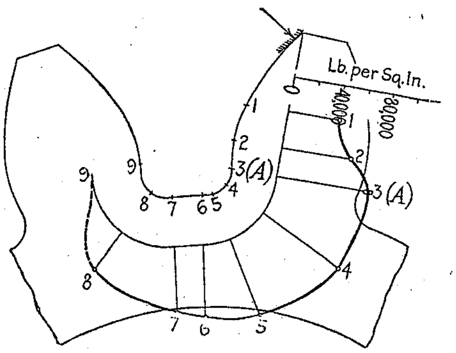

Fig. 16. Tangential Stress at Tension Side—Normal Inside Pressure and Maximum Torque

54. Table V and Fig. 18 give the numerical and plotted values of the stress difference $(\widehat{rr}-\widehat{\theta\theta})$ along the inside boundary of the pinion, the normal inside pressure and the torque load being applied. A circular ring to which a uniform inside pressure is applied will show concentric isochromatic bands. The deflections of those bands (Fig. 2)* in the case of the pinion show the disturbance due to the presence of the teeth.

55. When the maximum torque is applied, the values obtained for $(\widehat{rr}-\widehat{\theta\theta})$ give the curve of Fig. 18. The colored images as well as the diagrams show that the load applied at the top of one tooth extends its influence as far as the inside boundary of the pinion. The combination of the inside uniform pressure, already disturbed by an irregular outside boundary, with irregularly distributed stresses—tensions in certain parts and compressions in others—due to the torque load, do not of course give a resultant stress distribution which shows any symmetry with respect to the point of contact. The upper pinion being the driving pinion, it may be seen on the colored image (Fig. 3)* that the stresses vanish rather rapidly in the withdrawing part, but that the penetration extends much farther into the approaching part.

56. It may also be interesting to point out that there is a zone of zero stress inside of the pinion under the root of the tooth when the torque load is applied. This is shown on the diagram of the lines of principal stress (Fig. 14) by the converging of the lines of principal stress. Where several lines of principal stress intersect, the principal stresses usually vanish.

57. The question of engineering interest was to find the relative influence of the factors which affect the maximum stress, and the authors therefore varied the values of:

a. The inside normal pressure

b. The torque load.

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VALUES OF $(\widehat{rr}-\widehat{\theta\theta})$ ALONG THE BOUNDARY OF THE BORE

No. of Point in Fig. 18	$(\widehat{rr}-\widehat{\theta\theta})$ Lb. per Sq. In.
1	36,600
2	54,100
3	36,600
4	18,100
5	41,000
6	61,500
7	43,500
8	38,700

58. The values of $(\widehat{rr}-\widehat{\theta\theta})$ along the inside boundary when the maximum torque load is applied are given in Table VI and have been plotted in Fig. 19 for the case of reduced inside pressure. The colored image did not show noticeable variation across the minimum cross-section AB and along the outside edges of the main tooth. The influence of the inside pressure on the above-mentioned limit does not affect materially the regions of maximum stress, due in this case to the torque load.

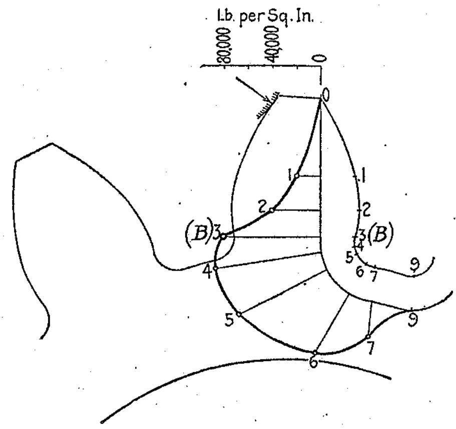

Fig. 17. Tangential Stress at Compression Side—Normal Inside Pressure and Maximum Torque

59. Fig. 5* shows the image obtained for normal pressure and reduced torque. Having applied 0.7 of the maximum torque value, the

* See Frontispiece of this issue of the REVIEW.

stresses showed a general reduction in the region of high stress. The values of the tangential stresses along the tension side of the boundary of the main tooth are given in Table VII and are plotted in Fig. 20. This should be compared with the same diagram (Fig. 16) for

TABLE VI

VALUES OF $(\widehat{rr}-\widehat{\theta\theta})$ ALONG THE BOUNDARY OF THE BORE

(Maximum torque—reduced radial pressure)

No. of Point in Fig. 19	$(\widehat{rr}-\widehat{\theta\theta})$ Lb. per Sq. In.
1	37,600
2	36,600
3	20,500
4	14,550
5	20,500
6	36,600
7	54,100
8	54,100
9	41,000
10	20,500

the case where the full load is applied. The maximum tension has dropped from 73,200 lb. per sq. in. (Table II) to 57,700 lb. per sq. in. (Table VII); i.e., it has been reduced to 0.8 of its previous value. The fact that it has dropped only to 0.8, whereas the torque was reduced to 0.7, is explained by the permanent stress due to the inside radial pressure which had been maintained at its previous value. A reduction of the torque load has as a result a reduction of the maximum stress. We shall see later that this is not always the case.

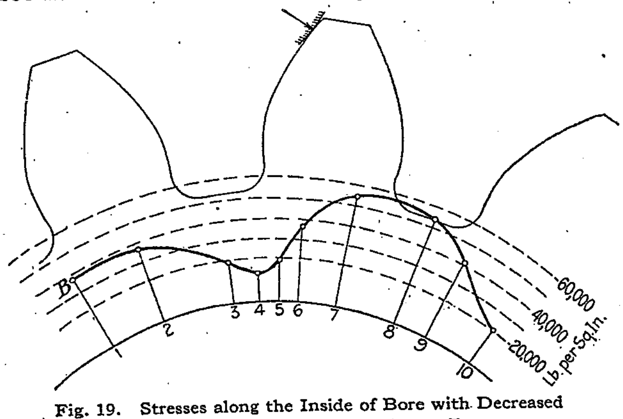

Fig. 19. Stresses along the Inside of Bore with Decreased Pressure (18,100 lb.) and Maximum Torque

60. When the inside radial pressure is increased in such proportion that without any torque being applied it produces stresses at the outside boundary of the gears of a magnitude approaching that due to the torque load, it will be this internal pressure which will have a preponderant influence.

61. Pinions have been examined with maximum values for $(\widehat{rr}-\widehat{\theta\theta})$ of 60,000 and 81,500

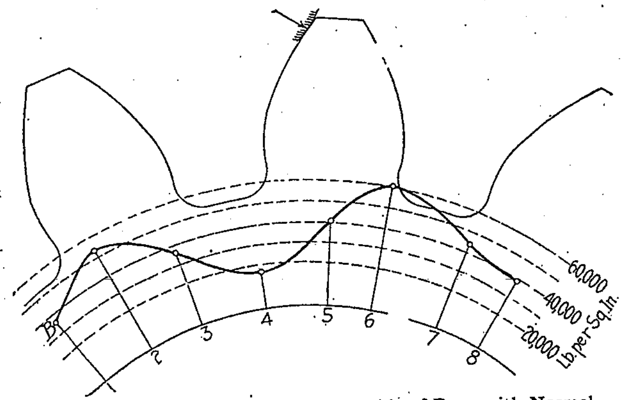

Fig. 18. Stresses along the Inside of Bore, with Normal Pressure (28,820 lb.) and Maximum Torque

TABLE VII

VALUES OF THE TANGENTIAL STRESS ALONG THE BOUNDARY OF THE LOADED TOOTH—TENSION SIDE

(Normal inside pressure—reduced torque)

No. of Point in Fig. 20	q Lb. per Sq. In.
1	39,700
2	51,500
3	58,500
4	57,700
5	56,600
6	51,500
7	38,000
8	19,500

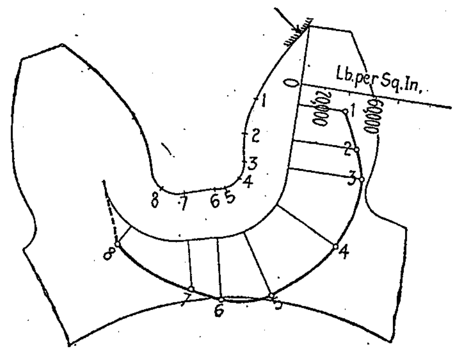

Fig. 20. Tangential Stress at Tension Side—Normal Inside Pressure and Reduced Torque

lb. per sq. in. at the inside boundary with the torque load at its normal value of 500 lb. tractive force per inch of face. The tractive force was afterward brought up to its maximum value of 1500 lb.

62. These tests showed that the torque load, when applied to the pinion subjected to those increased radial pressures, affects only the distribution of the stresses. It makes the high stresses extend over a larger area, but it does not increase materially the maximum stress. In these cases the dangerous section is no longer a straight section through the root of the tooth but it follows a V-shaped line, the lower point of which lies toward the inside bore. The sharpness of the angle of the V-shaped fracture at the base of the tooth appears to be due to an excess of radial shrinking pressure. In practice this excess is due to the pounding of the pinion onto the tapered shaft past its normal position.

63. In this connection a study was made of the stress distribution through two radial sections, passing respectively through the points A and B of the minimum cross-section of the main tooth, the points of maximum tension and compression.

64. The values of $(p-q)$ were deduced from the colored image of Fig. 4.* Extensometer measurements of $(p+q)$ were made. As before, the scales of both measurements were determined so that the stresses in the models should represent the stresses in the steel pinion.

65. The maximum torque and the normal inside pressure were applied. Table VIII and Figs. 21 and 22 give the values obtained. Fig. 21 gives the magnitude of the principal stresses along the two sections AA' and BB'. Fig. 22 gives a portion of the lines of principal stress taken from Fig. 14, and for the same sections AA' and BB' shows the two principal stresses plotted in direction and in magnitude.

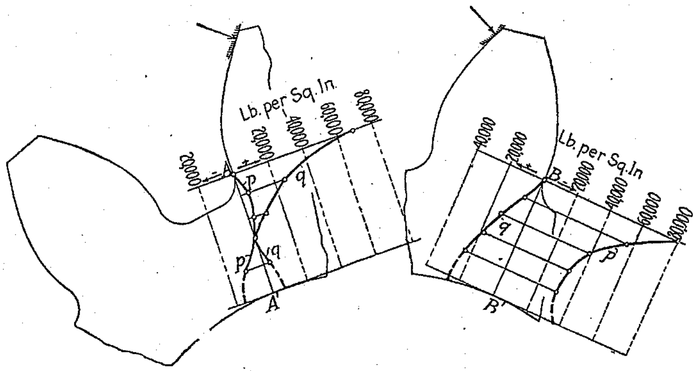

Fig. 21. Principal Stresses Across Radial Sections of Tooth—Normal Inside Pressure and Maximum Torque

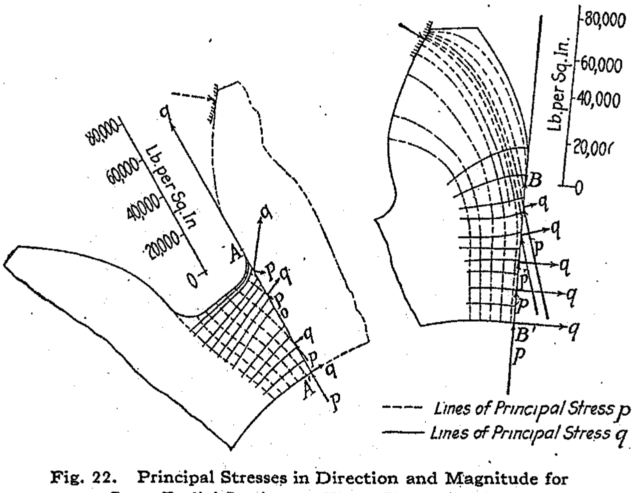

Fig. 22. Principal Stresses in Direction and Magnitude for Same Radial Sections as Those Shown in Fig. 21

*See Frontispiece of this issue of the REVIEW.

TABLE VIII

VALUES OF THE PRINCIPAL STRESSES ACROSS THE RADIAL SECTIONS PASSING RESPECTIVELY THROUGH THE POINTS *A* AND *B* OF THE MINIMUM CROSS-SECTION OF THE LOADED TOOTH

Cross-Section *BB'* Fig. 21; Distance in Inches from Point B'	p Lb. per Sq. In.	q Lb. per Sq. In.
0.410 (*B*)	—79,900	0
0.334	—55,800	6,000
0.256	—39,000	15,200
0.179	—32,600	18,400
0.102	—27,100	23,500
Cross-Section *AA'* Fig. 21; Distance in Inches from Point A'		
0.410 (*A*)	0	69,350
0.334	4,350	25,400
0.256	2,700	10,300
0.179	0	0
0.102	—11,350	3,350

66. A good way to visualize the state of stress at a given point is to consider a rectangular element with its sides parallel to the two principal stress directions at that point. By considering such elements along the sections *AA'* and *BB'* (Fig. 22) from this viewpoint, one can form a mental picture of how the section is acted upon by the elastic forces.

67. It would require too much space to include in this article a full discussion and to make a complete report of the results summarized here. The authors trust that the material they have presented will stimulate those interested in this subject to further efforts in the development and use of the photo-elastic method.

68. It seems, finally, almost superfluous to call attention to the comparative ease with which such a stress problem as this can be handled by the photo-elastic method, whereas the use of ordinary engineering methods gives untrustworthy results and the exact mathematical solution based upon the theory of elasticity is impossible.

69. Acknowledgment is due to the Massachusetts Institute of Technology for permission to use in this article certain of the results included in the thesis submitted by Dr. Paul Heymans, University of Ghent, Belgium, as partial fulfillment of the requirements for the degree of Doctor of Science from the Institute.

WHEN SERVICE IS REQUIRED

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Cincinnati, Ohio	215 West Third Street
Cleveland, Ohio	1133 East 152nd Street
Dallas, Tex.	1801 North Lamar Street
Detroit, Mich.	700 Antoinette Street
Kansas City, Mo.	819 East Nineteenth Street
Los Angeles, Calif.	5203 Santa Fe Avenue
Minneapolis, Minn.	410 Third Avenue, North
New York, N. Y.	414 West Thirteenth Street
Philadelphia, Pa.	1223 Washington Avenue
Pittsburgh, Pa.	16 Terminal Way
St. Louis, Mo.	1009 Spruce Street
Salt Lake City, Utah	360 West Second South Street
Seattle, Wash.	1508 Fourth Avenue, South

Special service divisions are also maintained at the following works of the Company: Erie Pa.; Ft. Wayne Ind.; Oakland, Calif.; Pittsfield Mass.; Schenectady, N. Y.; and West Lynn, Mass.—River Works and West Lynn Works

SALES OFFICES (Address nearest Office)

Akron, Ohio	159 South Main Street
Amarillo, Tex.	806 South Grant Street
Atlanta, Ga.	187 Spring Street, Northwest
Baltimore, Md.	39 West Lexington Street
Binghamton, N. Y.	19 Chenango Street
Birmingham, Ala.	602 North Eighteenth Street
Bluefield, W. Va.	307 Federal Street
Boston, Mass.	84 State Street
Buffalo, N. Y.	39 East Genesee Street
Butte, Mont.	20 West Granite Street
Canton, Ohio	700 Tuscarawas Street, West
Charleston, W. Va.	201 Capitol Street
Charlotte, N. C.	200 South Tryon Street
Chattanooga, Tenn.	536 Market Street
Chicago, Ill.	230 South Clark Street
Cincinnati, Ohio	215 West Third Street
Cleveland, Ohio	925 Euclid Avenue
Columbus, Ohio	17 South High Street
Dallas, Tex.	1801 North Lamar Street
Davenport, Iowa	111 East Third Street
Dayton, Ohio	25 North Main Street
Denver, Colo.	650 Seventeenth Street
Des Moines, Iowa	418 West Sixth Avenue
Detroit, Mich.	700 Antoinette Street
Duluth, Minn.	14 West Superior Street
El Paso, Tex.	109 North Oregon Street
Erie, Pa.	10 East Twelfth Street
Fort Wayne, Ind.	1635 Broadway
Fort Worth, Tex.	410 West Seventh Street
Grand Rapids, Mich.	148 Monroe Avenue
Hartford, Conn.	18 Asylum Street
Houston, Tex.	1016 Walker Avenue
Indianapolis, Ind.	106 North Illinois Street
Jackson, Mich.	212 Michigan Avenue, West
Jacksonville, Fla.	11 East Forsyth Street
Kansas City, Mo.	1004 Baltimore Avenue
Knoxville, Tenn.	602 South Gay Street
Little Rock, Ark.	223 West Second Street
Los Angeles, Calif.	5201 Santa Fe Avenue
Louisville, Ky.	455 South Fourth Street
Memphis, Tenn.	130 Madison Avenue
Miami, Fla.	25 Southeast Second Avenue
Milwaukee, Wis.	425 East Water Street
Minneapolis, Minn.	107 South Fifth Street
Nashville, Tenn.	234 Third Avenue, North
Newark, N. J.	20 Washington Place
New Haven, Conn.	129 Church Street
New Orleans, La.	837 Gravier Street
New York, N. Y.	120 Broadway
Niagara Falls, N. Y.	201 Falls Street
Oklahoma City, Okla.	15 North Robinson Street
Omaha, Nebr.	409 South Seventeenth Street
Philadelphia, Pa.	1321 Walnut Street
Phoenix, Ariz.	11 West Jefferson Street
Pittsburgh, Pa.	535 Smithfield Street
Portland, Oreg.	329 Alder Street
Providence, R. I.	76 Westminster Street
Richmond, Va.	700 East Franklin Street
Roanoke, Va.	202 South Jefferson Street
Rochester, N. Y.	89 East Avenue
St. Louis, Mo.	112 North Fourth Street
Salt Lake City, Utah	200 South Main Street
San Antonio, Tex.	201 Villita Street
San Francisco, Calif.	235 Montgomery Street
Schenectady, N. Y.	1 River Road
Seattle, Wash.	811 First Avenue
Spokane, Wash.	421 Riverside Avenue
Springfield, Ill.	504 East Monroe Street
Springfield, Mass.	1387 Main Street
Syracuse, N. Y.	113 South Salina Street
Tacoma, Wash.	1019 Pacific Avenue
Tampa, Fla.	112 Cass Street
Terre Haute, Ind.	701 Wabash Avenue
Toledo, Ohio	520 Madison Avenue
Tulsa, Okla.	409 South Boston Street
Utica, N. Y.	258 Genesee Street
Washington, D. C.	1405 G Street, Northwest
Waterbury, Conn.	195 Grand Street
Worcester, Mass.	340 Main Street
Youngstown, Ohio	16 Central Square

Canada; Canadian General Electric Company, Ltd., Toronto — Hawaii: W. A. Ramsay, Ltd., Honolulu

Motor Dealers and Lamp Agencies in all large cities and towns

INSTRUCTIONS GEH-779
GEARS AND PINIONS FOR RAILWAY MOTORS

These instructions apply equally well to all kinds of gears and pinions except where special features are noted.

The chief points to be observed in the care and maintenance of gearing are as follows:

1. Proper combination with respect to grade, tooth form and gear ratio.
2. Proper mounting and dismounting.
3. Lubrication.
4. Wear of bearing linings.
5. Replacement.
6. Inspection.

GRADE, TOOTH FORM AND GEAR RATIO

Gearing furnished by the General Electric Company is made in several grades, each of which is suitable for certain particular service. The grade is indicated by a letter enclosed in a circle which is stamped on the gear and pinion.

Besides the grade, other information in the following order will be found stamped on the motor side of the gear rim and on the wheel side of the pinion (except on very small pinions, when all except the catalog number and grade may be stamped on the motor side):

Catalog number; Grade letter; Month and Year; Serial number; G-E Co., U.S.A.

The most economical results can be obtained only by meshing gears and pinions of similar characteristics. Recommended combinations are as follows:

Grade A-1 pinion with grade B or M gear.
Grade M pinion with grade B or M gear.
Grade K pinion with grade K or L gear.
Grade F pinion with grade F, D or E gear.

Gears and pinions of the long and short addendum tooth form furnished by the General Electric Company are not interchangeable with other tooth forms. In this type the pinion has a longer addendum than the Brown and Sharpe type and the gear a correspondingly shorter addendum. A ready means of identification is provided by means of two grooves each 1/16 in. deep and ⅛ in. wide machined in the ends of the teeth at the wheel side of each short addendum gear and long addendum pinion, as shown in Fig. 1. Always make sure that if either the gear or the pinion has these grooves the other member is similarly marked.

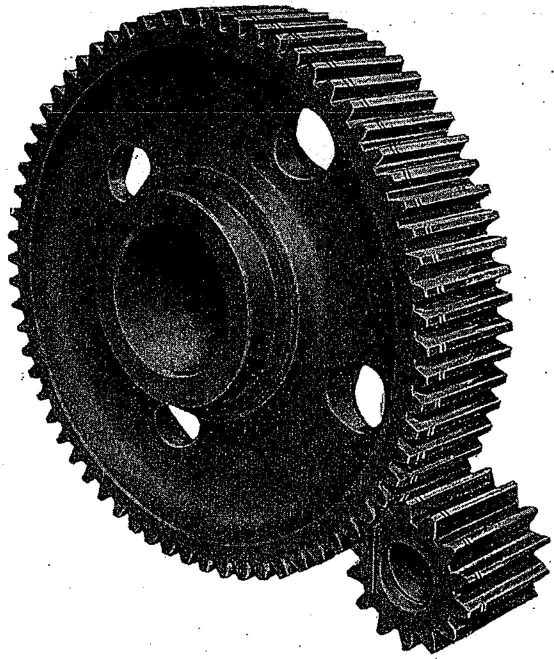


Fig. 1. Short-addendum Gear with Long-addendum Pinion, Showing Method of Marking

In using any type of gearing, care should be taken to see that the proper tooth form of both gear and pinion is used.

Be sure that the gearing used with any particular motor is of the proper pitch and ratio for the service required, and see that the gear and pinion line up properly on the gear centers. The total number of teeth in the gear and pinion divided by twice the diametral pitch should give the exact distance between the gear centers.

GENERAL ELECTRIC COMPANY
SCHENECTADY, N. Y.

September, 1929 (5m)

B-9146

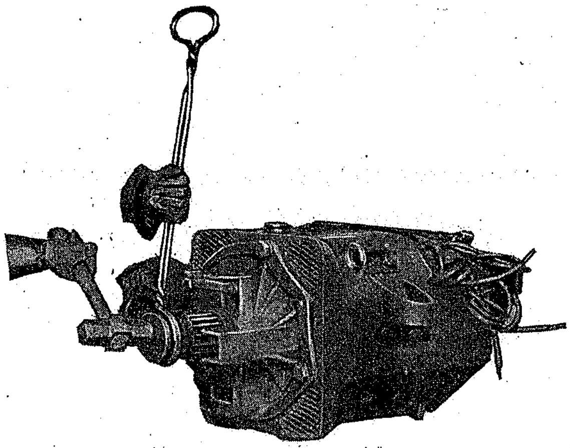


Fig. 2. Pinion Being Replaced

MOUNTING PINIONS

Before mounting a pinion, clean the pinion bore, keyway, shaft, and key and make sure that they are free from burrs and foreign matter. Round off the sharp corners on the top of the key.

Carefully check the fit of the pinion to make sure that it bears evenly around the shaft and makes contact with at least 75 per cent of the tapered surface of the shaft. This can be checked easily by rubbing Prussian blue or red lead and oil on the pinion bore and seating the cold pinion on the shaft by hand.

If the pinion meets this condition, insert the key and see that it fits properly in the keyway. With the cold pinion in place, the side nearest the motor should be within 1/16 in. of its final position and there should be sufficient clearance between the top of the key and the bottom of the keyway to allow a 0.015-in. feeler gauge to slide freely the length of the key. Take care not to use too wide a key in the shaft, as this will bulge the metal on each side and localize the mounting stress in or near the keyway.

Wipe the pinion clean and place it in boiling water until it is thoroughly heated, which will require about one hour. To prevent rusting and to insure a clean bore add washing soda to the water in the proportion of ¼ lb. of soda to each 5 gal. of water.

When the pinion is heated sufficiently, place it on the shaft immediately and seat it firmly by hand. Hold a round cup-shaped metal block firmly against the pinion and drive it home with one or two squarely-struck blows from a light-weight, short-handled sledge.

Put the lock washer on and tighten the nut while the pinion is still hot. Give the pinion another blow and tighten the nut firmly. Bend up a portion of the pinion washer which projects beyond the nut and firmly seat it against the flat side of the nut.

The complete operation of installing the pinion should be performed as quickly as possible before the pinion begins to cool off and contract.

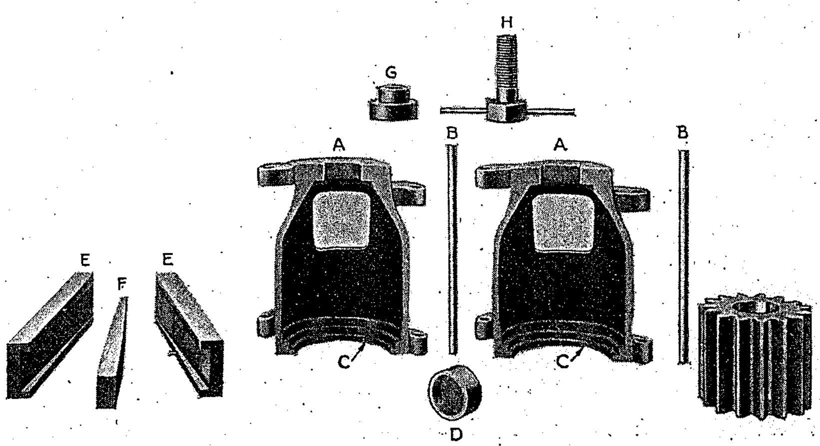


Fig. 3. Disassembled View of Pinion Puller

Caution must be exercised in mounting pinions in order that the metal will not be overstressed. Blows with the sledge should not be too heavy and the pinion should not be driven too far up on the shaft. The intensity of the blows should be in proportion to the size of the pinion.

REMOVING PINION

The proper method of removing pinions is to use a pinion puller, in which the pulling member grips all the teeth firmly and evenly. Heating the pinion or driving it off by means of a wedge are methods which should never be employed, as they injure not only the pinion, but also other parts of the equipment.

A pinion puller developed by the General Electric Company is illustrated in Fig. 3.

The puller casting is in halves A-A, which are held together by two removable pins B-B, one acting as a hinge and the other as a lock. When in place, projections C on the bottom of the casting grip the teeth at the heel of the pinion. A cap D is provided which fits over the end of the shaft in order to protect the threads. The other members of the pinion puller are a split guide E-E, wedge F, flange G and clamping screw H.

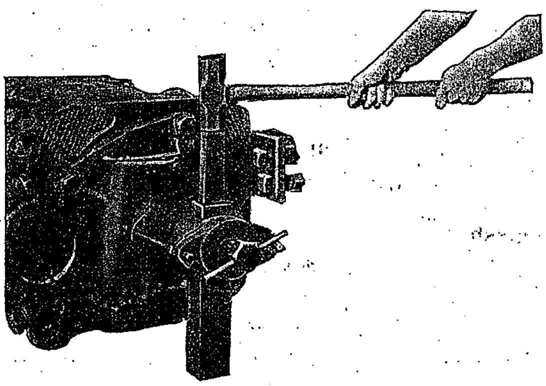


Fig. 4. Method of Using Pinion Puller

The method of using the pinion puller is illustrated in Fig. 4. Set the motor on a solid, level floor or iron block, and remove the pinion nut and lock washer. Place the puller around the pinion and insert the locking pin. Grease the guides and insert them through the opening in the puller casting. Allow the ends of the guides to rest on the floor or block, one half of the guide bearing against the pressure cap on the end of the shaft and the other half against the clamping screw in the puller casting. Turn up the end screw to hold the guides in place. Grease the wedge, insert it in the guides, and drive it with a sledge until the pinion comes off.

MOUNTING GEARS

Solid gears should be assembled by pressing on the axle. In manufacturing, the gear bore is made slightly smaller than the diameter of the axle to provide the proper press fit. The use of a key is unnecessary, as the press fit is sufficient to deliver the torque of the motor.

The exact pressure required for mounting cannot be given in a general way as this varies with the diameter of the bore, the length and diameter of the hub, and the condition of the bore and axle.

It is important that the diameter of the gear fit and the gear bore be held to close limits and that smooth finished surfaces be obtained. Great care should be taken to obtain the proper fit, sufficient to cause the gear to remain tight on the axle and not enough to stretch the gear hub permanently and distort the gear rim. In no case should the seat on the axle exceed the gear bore by more than 1½ mils per inch of axle diameter.

The gear bore and the seat on the axle should be carefully cleaned and white-leaded to prevent abrasion of surfaces.

LUBRICATION

The best gear lubricant is a high grade grease of such consistency that it will slowly level back to the bottom of the gear case instead of clinging or caking on the sides.

Where there is considerable variation in climatic conditions, it may be advisable to use a heavy grade of grease in warm weather and change to a lighter grade in winter, in order to maintain the leveling back feature.

The grease should be maintained at a sufficient level in the bottom of the gear case to reach the pitch line of the gear so that the gear teeth will be immersed and carry lubricant to the mesh. The

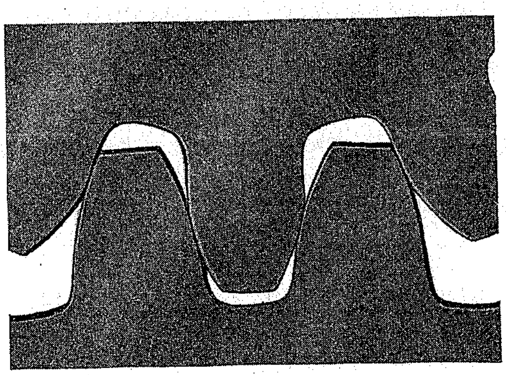


Fig. 5. Correct Mesh
Armature and Axle Linings Not Worn

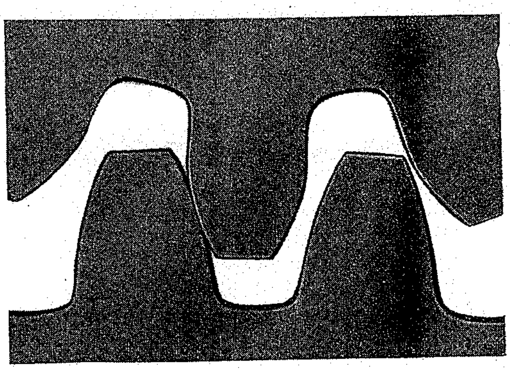


Fig. 6. Improper Mesh
Armature and Axle Linings Badly Worn

exact amount necessary cannot be given as this varies with different motors and different combinations of gear cases and gearing. However, the proper amount for each motor can be readily determined by trial.

It is essential that the lubricant be kept as free from grit as possible as this combined with the grease forms an abrasive which results in rapid wear of the teeth.

At regular inspection periods lubricant should be added if necessary. A small quantity frequently applied is more economical than a large amount applied at long intervals.

BEARING WEAR

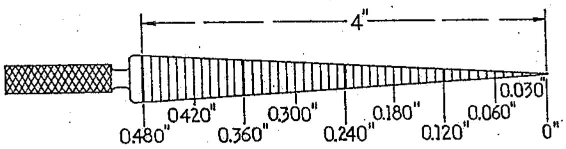
In order to obtain efficient and quiet operation and maximum life of gears and pinions the armature and axle lining wear should be kept within as small a limit as practical.

The effect of bearing wear on gearing is shown in Fig. 5 and 6. Fig. 5 shows the correct mesh when the armature and axle linings are new, and Fig. 6 shows the conditions when the armature and axle linings are badly worn. Note the improper mesh and extreme back lash. This causes noise, broken teeth and a great reduction in life.

Wear of the linings can be measured by inserting a gauge between the shaft or axle and the lining bore and measuring the total clearance on the diameter. The A.E.R.A. Committee on Limits of Wear recommends the following maximum allowances:

Linings	Size of Motors	Total Clearance on Diameter Inches
Axle	Up to 50 hp.	1/16
Axle	50 hp. and larger	1/8
Armature	Up to 50 hp.	3/64
Armature	50 hp. and larger	1/16

The armature lining wear can also be checked by measuring the air gap between the armature and the bottom field pole.



Each Graduation is $\frac{1}{12}$ of the total length and increases the diameter 0.010".

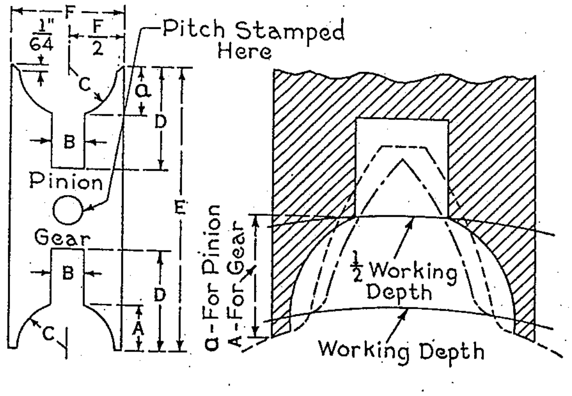
Fig. 7. Wear Gauge for Axle Linings

Fig. 7 shows a handy gauge for checking axle lining wear, which can easily be made by the operator.

REPLACEMENT

Many operators using Brown and Sharpe gearing have become accustomed to scrap gears and pinions when the teeth are worn to a knife edge. The long addendum pinion has less surface on the top of the tooth when new and can, therefore, be worn beyond this point, provided the service is not too severe.

For city or light service a discard gauge similar to that shown in Fig. 8 may be used to determine the wear-out point of gears and pinions.



Pitch	Dimensions in Inches						
	A	a	B	C	D	E	F
4¼	0.281	0.321	0.247	5/16	¾	2½	¾
4	0.299	0.339	0.262	11/32	¾	2½	13/16
3½	0.341	0.379	0.299	⅜	⅞	2½	⅞
3	0.398	0.448	0.348	15/32	1	2¾	1 1/16

Fig. 8. Discard Gauge for Railway Spur Gearing

In heavy traction service with proper lubrication and relative freedom from grit, gears and pinions will often have a very long life and show comparatively little wear. Under these conditions it may be necessary to scrap the gearing because of reaching the fatigue limit, rather than on account of wear of the teeth. This limit may be set up on a mileage basis determined from actual experience in the particular service involved.

INSPECTION

It is important that gearing be inspected perodically to make sure that it is operating properly. The frequency of inspection can be best determined from operating experience, but it is recommended that the following points be observed every 1500 to 2000 miles.

1. Remove dirt from around the inspection door, taking care that none of it falls into the gear case. If the bottom half of the case is removed, the inside of the case should be protected so that dirt will not get into it.

2. See that there is sufficient lubricant in the gear case and add more if necessary.

3. Examine the gear and pinion for mesh and wear, and replace if worn excessively.

4. Check the wear of the bearing linings and replace them if necessary.

5. Inspect the gear case carefully to see that it is in good repair, that the bolts are tight, that the inspection door is properly fitted, and there is not excessive leakage of the lubricant.

At longer intervals, or at least whenever the motor is overhauled, remove the old lubricant and clean the inside of the gear case thoroughly by washing with kerosene or gasoline.

SPECIAL TYPES OF GEARING

Composite Gears

A composite gear consists of a heat-treated forged rim having teeth on the outer surface, shrunk on an untreated cast steel hub. To allow for a suitable shrink fit, rims are bored slightly smaller than the hubs. Rims and hubs are assembled at the factory unless otherwise ordered.

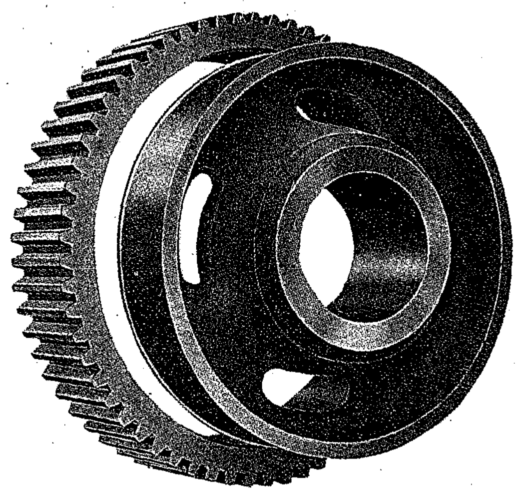


Fig. 9. Composite Gear

When composite gears are assembled by the operators, care must be taken not to overheat the rim when shrinking it on. There are three methods of heating the gear rim: by means of a transformer, an oven, or a gas ring.

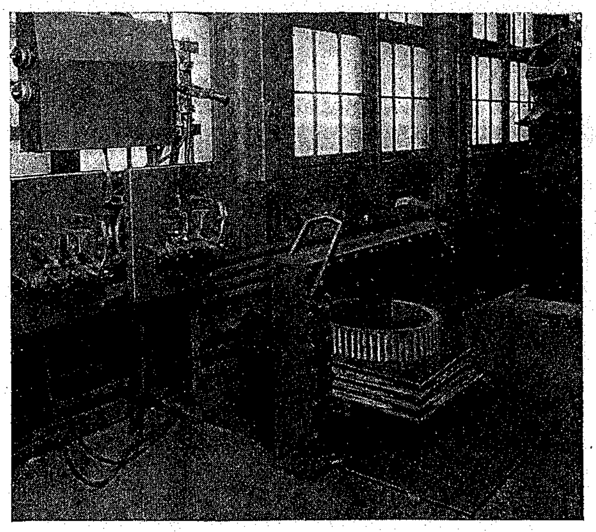


Fig. 10. Transformer Method for Heating Gear Rims Uniformly

Fig. 10 shows the best method for heating gear rims. Coils connected to a power circuit are placed within an iron core and act as the primary winding of a transformer. The gear rim is placed on top of the coils, surrounding the central leg of the core, and acts as a closed secondary. When current flows through the coils induced currents are set up in the gear rim and cause it to become heated. By this method the rim is uniformly heated, and the necessary expansion can be readily measured while the rim is heating.

If the rims are heated in an oven, pyrometers should be provided and the temperature should not be allowed to exceed 400 deg. F.

If neither of the above methods is available, the rim may be heated by a gas ring. This should be of such diameter that when placed in the center of the rim the flame just touches the bore. A piece of sheet iron laid on top of the rim will retain the heat and force the flame against the bore. The correct temperature may be judged with sufficient accuracy by dropping water on the rim. When the water snaps, the expansion has progressed sufficiently to allow the hub to be slipped in place. The temperature of the rim should not exceed 400 deg. F.; otherwise the temper will be drawn and the value of the heat treatment lost. This method is not as good as the others mentioned above, because when a flame is used, expansion may be uneven and it is generally necessary to shut off the flame while checking the amount of expansion. There is also the danger of localizing the heat over a small area sufficiently to draw the temper at that point.

The General Electric Company has also furnished gear rims for mounting on special wheel hubs which take the place of the separate hubs used for composite gears. The preceding instructions for heating gear rims apply in this case also.

Cushion Gears

The cushion gear, designed for heavy traction duty, contains springs to absorb shocks resulting from acceleration and braking, uneven rail joints, and other conditions.

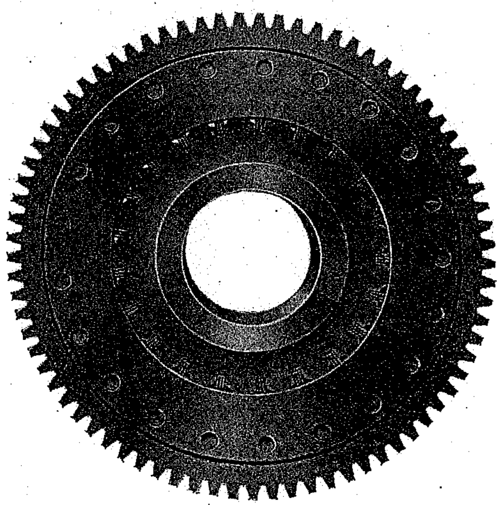


Fig. 11. Cushion Gear

The cushion gear consists of a forged rim with teeth on the outer surface and pockets for assembling springs on the inner surface. The rim has a sliding fit on a cast steel center which is provided with spring pockets to match those in the rim. Pairs of flat steel springs are assembled in the spring pockets under pressure. Side plates are riveted to either the center or the rim, and practically cover the assembled springs, thus restraining them from endwise movement. There are suitable oil holes to provide proper lubrication of the moving parts.

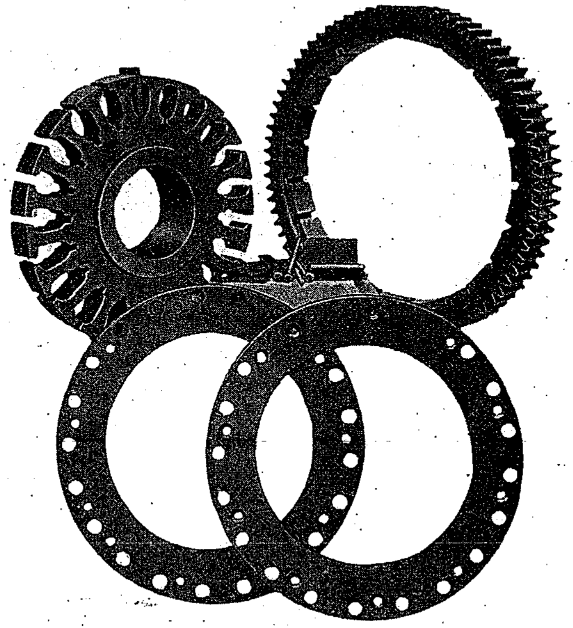


Fig. 12. Cushion Gear—Disassembled View

For protection during shipment, the oil holes in the gear are plugged up with round pasteboard disks and a large paper ring placed between the retaining ring and the hub to keep out dirt. These should be removed before the gear is mounted.

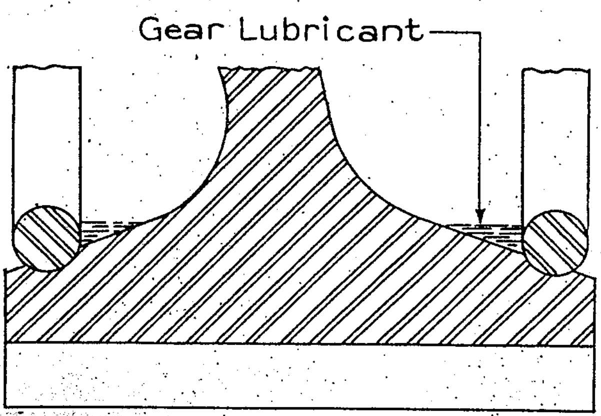


Fig. 13. Section of Non-resonant Gear, Showing Annular Rings and Grooves Filled with Gear Lubricant

Care should be taken to see that none of the small disks are overlooked, as these would interfere with lubrication.

Non-resonant Gears

Non-resonant gears provide quieter operation by eliminating the ringing noise common to most metal gear sections.

This type of gear has annular pockets at each side of the web, formed by rings snapped into machined grooves cut on the under side of the rim. These rings have a different period of vibration than the gear section and are in themselves an efficient deadening medium. Further quieting effect is obtained by filling the annular pockets with gear lubricant. Only a small quantity of grease on the under side of the rim is required. The best procedure is to place the usual amount of grease in the case for suitable lubrication of the teeth and a slight additional amount in each annular pocket.

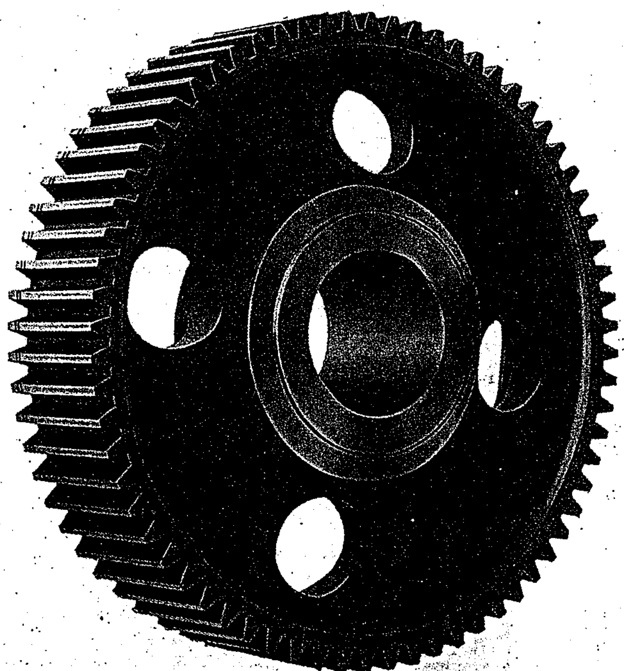


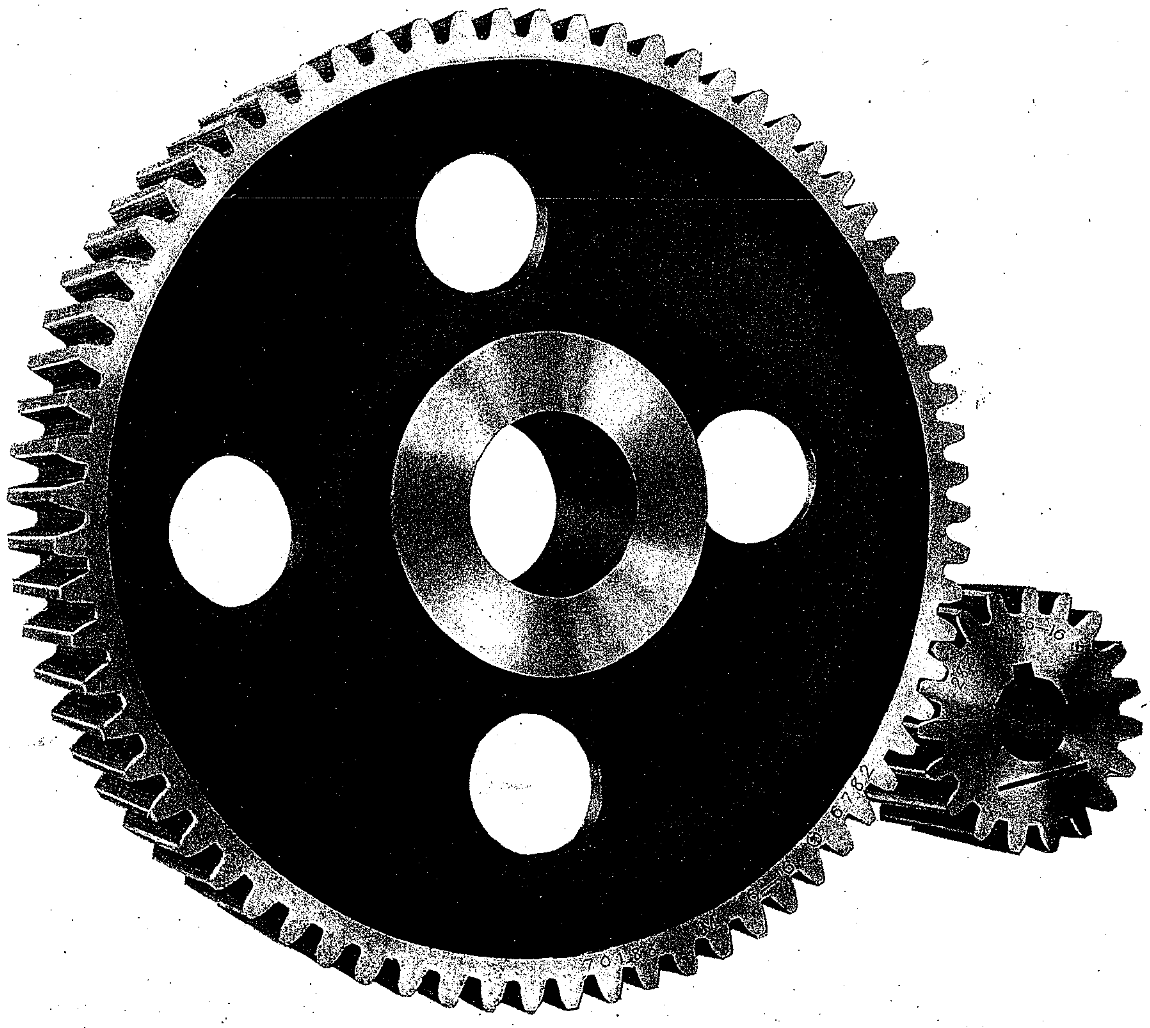
Fig. 14. Non-resonant Gear

General Electric Company
Schenectady, N.Y.

SUPPLY DEPARTMENT

September, 1916 | Bulletin No. 44419

RAILWAY MOTOR GEARS AND PINIONS



GRADE M GEAR AND PINION SHOWING A.E.R.A. STANDARD METHOD OF MARKING. SEE PAGE 10.

The General Electric Company manufactures several types of Railway Motor Gears and Pinions, suitable for various classes of service and differing in chemical characteristics, physical characteristics and price. Each type is designated by a Grade Letter as follows:

NOTE.—The data in this publication are subject to change without notice.

GEARS

Grade A

Cast Steel Untreated. Furnished in the split or solid types. Operates with grades H or F pinion.

Grade E

High Quality Cast Steel Untreated. Furnished in the split or solid types. Operates with grades H or F pinion.

Grade K

Forged Steel Case Hardened. Tooth structure consists of low carbon core with high carbon, hard wearing surface. Depth of case approximately one-sixth of tooth thickness at pitch line. Furnished in the solid type only. Operates with grade K pinion.

Grade L

Cast Steel Case Hardened. Tooth structure consists of low carbon core with high carbon, hard wearing surface. Depth of case approximately one-sixth of tooth thickness at pitch line. Furnished in split or solid types. Operates with grade K pinion.

STOCK OF FORGED GEAR BLANKS

Grade D

Forged Steel Untreated. Furnished only in the solid type. Operates with grades H or F pinion.

Grade F

Special Forged Steel, High Carbon Content, Heat Treated. Furnished in the solid or composite types only. Operates with grade F pinion.

Grade M

Forged Steel Medium Carbon Content. Homogeneous tooth structure, specially hardened and tempered throughout, combining the strength of grade F and a hardness similar to grade K. Furnished in the solid type only. Operates with grade M pinion.

Grade B

Cast Steel, specially hardened and tempered. Furnished in the split or solid types. Operates with grade M pinion.

PINIONS

Grade H

Forged Steel Medium Carbon Content, Heat Treated. Operates with grades A, D or E gear.

Grade F

Special Forged Steel High Carbon Content, Heat Treated. Operates with grades A, D, E or F gear.

Grade K

Forged Steel Case Hardened. Tooth structure consists of low carbon core with high carbon, hard wearing surface. Depth of case approximately one-sixth of tooth thickness at pitch line. Operates with grades K or L gear.

Grade M

Forged Steel Medium Carbon Content. Homogeneous tooth structure, specially hardened and tempered throughout, combining the strength of grade F and a hardness similar to grade K. Operates with grades M or B gear.

Selection of Grades

Of the various grades of gears and pinions described, each has its particular field of usefulness, and selection must be governed to a great extent by local operating conditions.

Except under special conditions, combinations consisting of Grade M pinions, Grades M or B gears, and Grade K pinions, Grades K or L gears, will afford the greatest economy, for their increased life is much greater than their increased price as compared with combinations consisting of Grades A, D, E, F and H.

There are, however, many cases where the service is extremely light and many where the life of Grades M, B, K and L gears will be much greater than the life of the old motors which will be retired before the gears reach their life limit; in such cases an untreated gear combination will be the most economical.

It is impossible to give definite mileage limits for the various grades as the average life of any combination on different roads is governed by the conditions under which the gears and pinions are operated; such as gear ratio, pitch, weight and radius of gyration of armature, limits of armature and axle lining wear, quality and quantity of lubrication, amount of grit allowed to accumulate in the gear pan, schedule speed, stops per mile, weight of car, rate of acceleration and breaking, and conditions of track; such as low rail joints, switches and crossovers.

An estimate of the life factors of different grades is also difficult to determine accurately but there is every reason to expect that the resistance to wear will be about proportional to the relative hardness of the different grades, and service records closely confirm this estimate. On this basis and on the basis of placing the grades in mesh as previously recommended, the following conservative estimate of life factors may be used

Knowing the average life of Grades A or E gears and H or F pinions in any particular service, application of the life factor for any other grade will determine its probable life in the same service.

Grade	Estimated Life Factor (Conservative)	Grade	Estimated Life Factor (Conservative)
A	1	H	1
D	1.13	F	1.5
E	1.13	M	2.5
F	1.8	K	2.8
B	2.5		
M	2.8		
K	3		

Grade M gears have been operated satisfactorily in mesh with Grade K pinions and Grade M pinions with A, D, E, F, K and L gears but actual service tests indicate that the most economical combinations are those as recommended under each particular grade.

How to Order Gears and Pinions

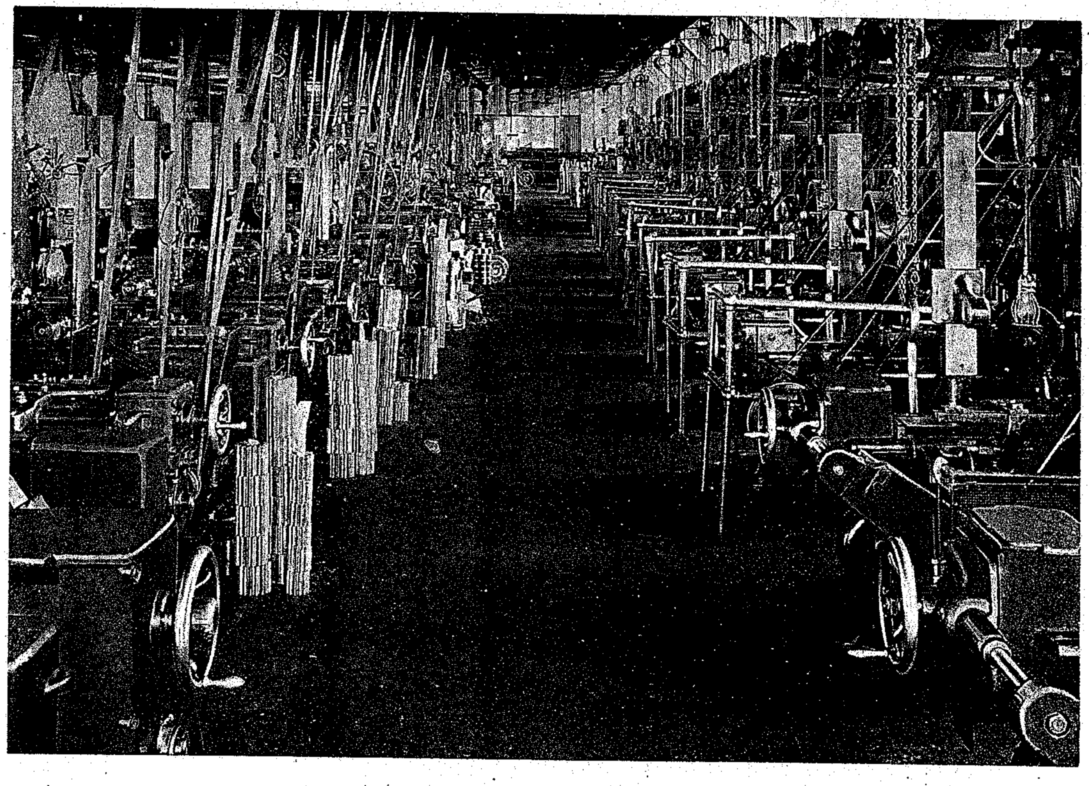
Orders for gears and pinions should specify the motor with which they are to be used, number of teeth, and grade of material desired. Gear orders should also specify the bore, width and depth of keyway and whether the gear is split or solid. If the gear or pinion is to be a duplicate of one already in use, specify the catalog number which will be found on the rim of the gear or outer end of the pinion.

If gears are to be rebored by the purchaser, orders should specify the minimum bore

FORGED GEARS

Forged gears can be furnished in the solid type only, for all standard G-E and Westinghouse motors with standard gear ratios.

The steel blocks from which the gear blanks are made are sheared from a round bloom rolled from the ingot after sufficient metal has been discarded to insure freedom from piping and segregation.



BATTERY OF CUTTERS

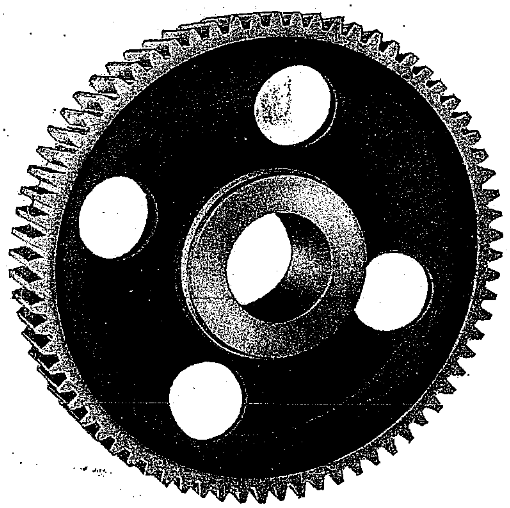
desired, depth of keyway, if any, and also the maximum bore. The gears will then be finished with exact minimum bore and hub suitable to accommodate the maximum bore. Never use the misleading term "Rough Bore." All gears are bored to a finish.

If gears and pinions are desired for motors other than those listed on pages 15 and 16. fill in the dimensions called for on data sheet, page 24; extra copies can be obtained from any district or local office.

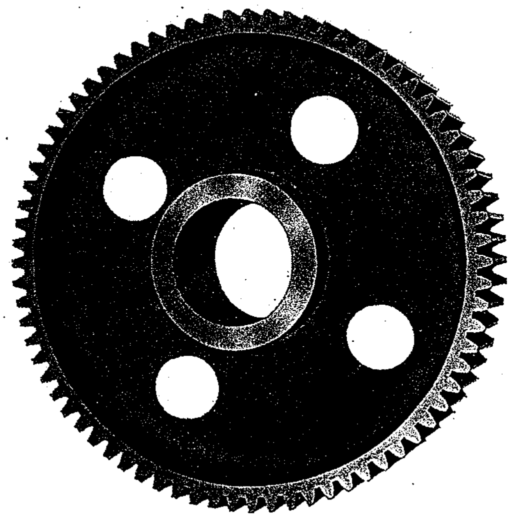
In order to accommodate the wheel press jig when pressing off the car wheel, four holes are punched in the web of the gear 90 deg. apart, 3½ in. in diameter, and on 7¼ in. radius. This size and location can be furnished on all gears except in cases where the outside diameter of the hub exceeds 9 in. or the rim is smaller than that determined by 64 teeth, 3 pitch, or 57 teeth, 2½ pitch. Gears outside these limits will be furnished with the largest hole possible in the thinnest portion of the web.

SOLID CAST STEEL GEARS

In order to obtain uniformity of hardness and homogeneity of structure, steel gear castings are made in moulds with provisions for large "risers." All heats are prepared under the direct supervision of our factory chemists, and in no case is a heat poured until analyzed and approved.

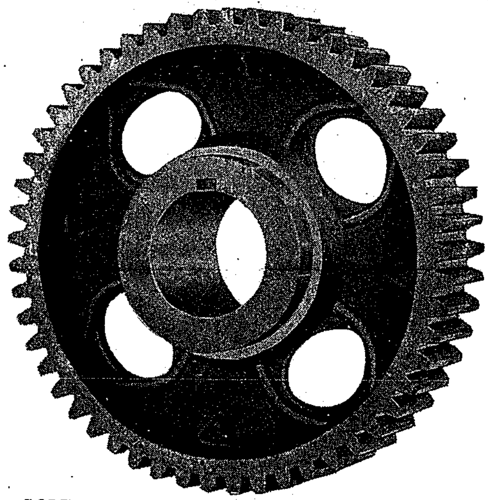


FORGED GEAR
GRADES D, F, K AND M

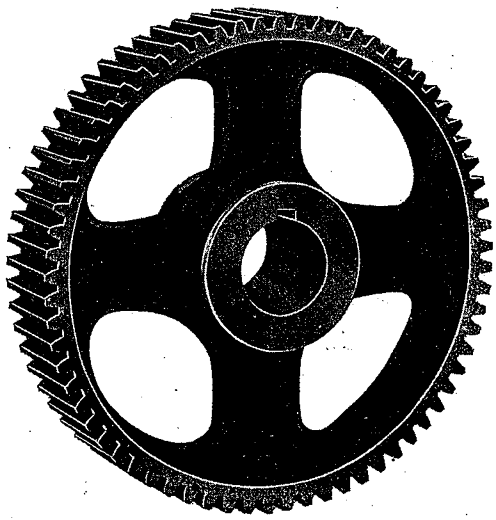


SOLID CAST STEEL GEAR
GRADE B

The design of the solid type Grade B gear corresponds to the forged gear; that is, solid web with four jack holes. The thickness of the web, the rim and radii between the rim, web and hub is such as to eliminate shrink strains in the castings and prevent undue warpage during treatment.

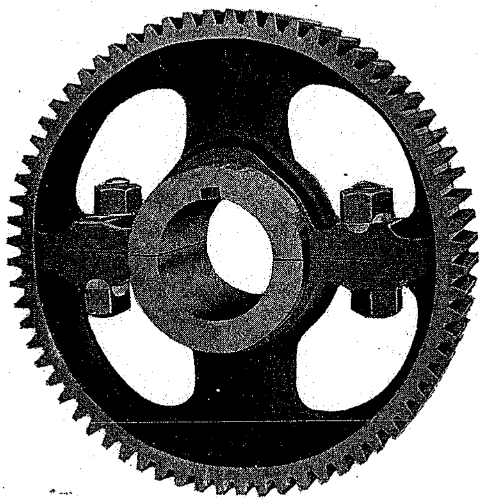


SOLID CAST STEEL GEAR, HEAVY SERVICE
GRADE E



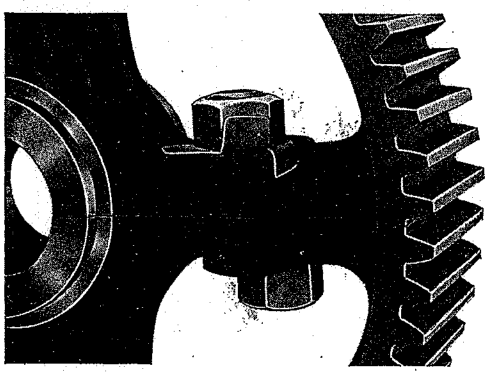
SOLID CAST STEEL GEAR, CITY SERVICE
GRADES A, E AND L

Grades A, E and L gears in the solid type are of the four spoke design. The cross sections of the metal in the rim spoke and hub are equalized. The spokes are elliptical in form, thick and wide, the rib or bead in the center of the rim between spokes is arched and joins the spokes in curves of large radii.



SPLIT CAST STEEL GEAR, HEAVY SERVICE
GRADE E AND B

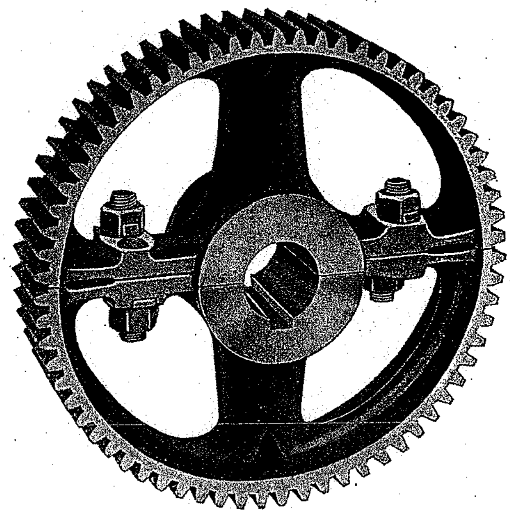
The spokes at this point are also reinforced by small ribs located on each side of the spoke and following the contour of the radius. All these features practically eliminate the possibility of cracks from shrink strains, thereby giving a much stronger and lighter casting.



METHOD OF LOCKING NUTS

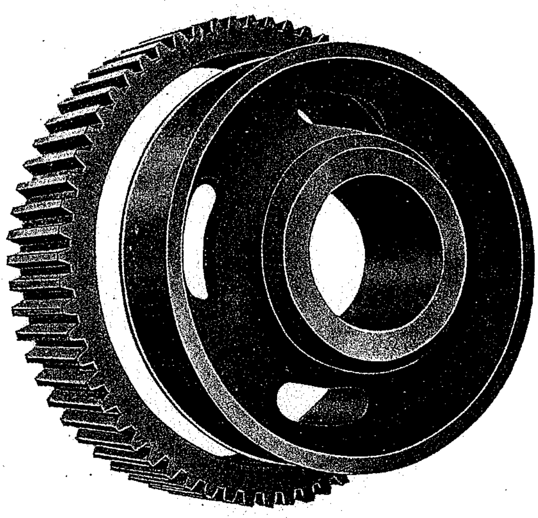
SPLIT CAST STEEL GEARS

With the exception of the small reinforcing ribs on each side of the spoke, the design of split gears embraces all the special features of the solid four spoke gear. The reinforcing of the bolt spokes by arched ribs prevents the halves opening at rim or bore when loaded to the maximum strength of the tooth.



SPLIT CAST STEEL GEAR, CITY SERVICE
GRADES A, B, E AND L

The halves are held together by four studs, two on each side of the hub placed side by

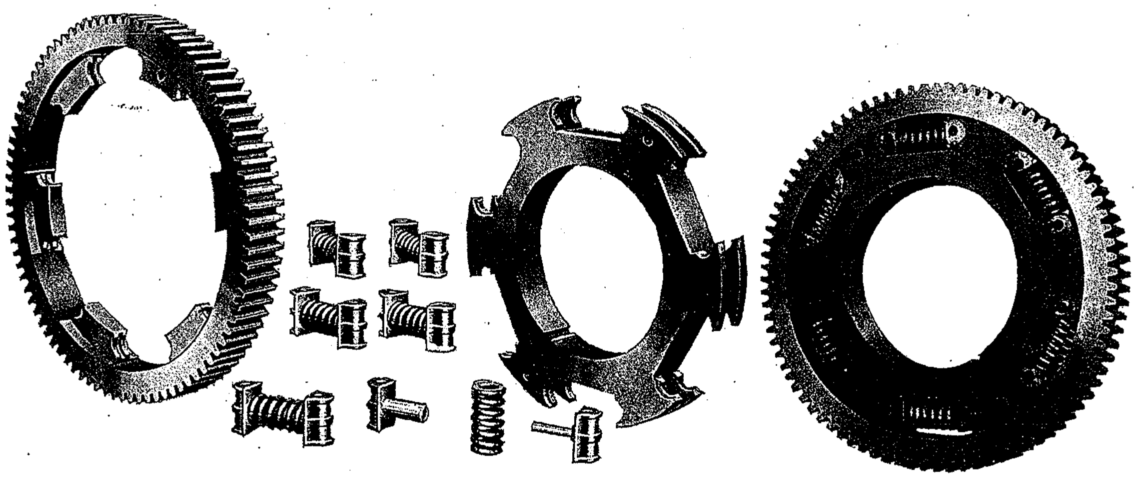


COMPOSITE GEAR
GRADE F RIM AND CAST STEEL UNTREATED CENTRE

side and about half way between the hub and the rim. Two of the studs are screwed into each casting, securely seated, and welded to prevent turning; the hex nuts on the other ends are positively locked with threewing, sheet steel washers $\frac{1}{16}$ in. thick, the wings of which are bent to bear on the flat sides of the nuts and on the face of the castings as illustrated. This permits the use of studs of equal length and of such diameter as will give an aggregate strength equal to or greater than that of the eight bolts in the older designs. For city service, the studs are 1¼ in. in diameter and for heavier service 1½ in. in diameter. Such sizes prevent the injurious stresses to which the smaller bolts of the eight-bolt type were liable at the hands of the shopman.

of sheet iron laid on top of the rim will be sufficient to retain the heat and force the flame against the bore. The required temperature may be determined with sufficient accuracy by dropping water on the rim. When the water snaps the expansion has progressed sufficiently to allow the hub to be slipped in place. Under no circumstances should the rim be placed in an oven or the flame allowed to touch the teeth.

These composite gears are now furnished only for special applications when blanks for solid forged gears are not available.



THE SPRING GEAR— ASSEMBLED AND DISASSEMBLED

COMPOSITE GEAR

A composite gear consists of a Grade "F" forged rim, shrunk on a cast steel hub. The design of the hub embraces all the special features of the solid cast gear. To allow for a suitable shrink fit, rims are bored smaller than the hubs by 0.8 mil per inch of their inside diameter. Rims and hubs will not be assembled before shipment unless specifically so ordered.

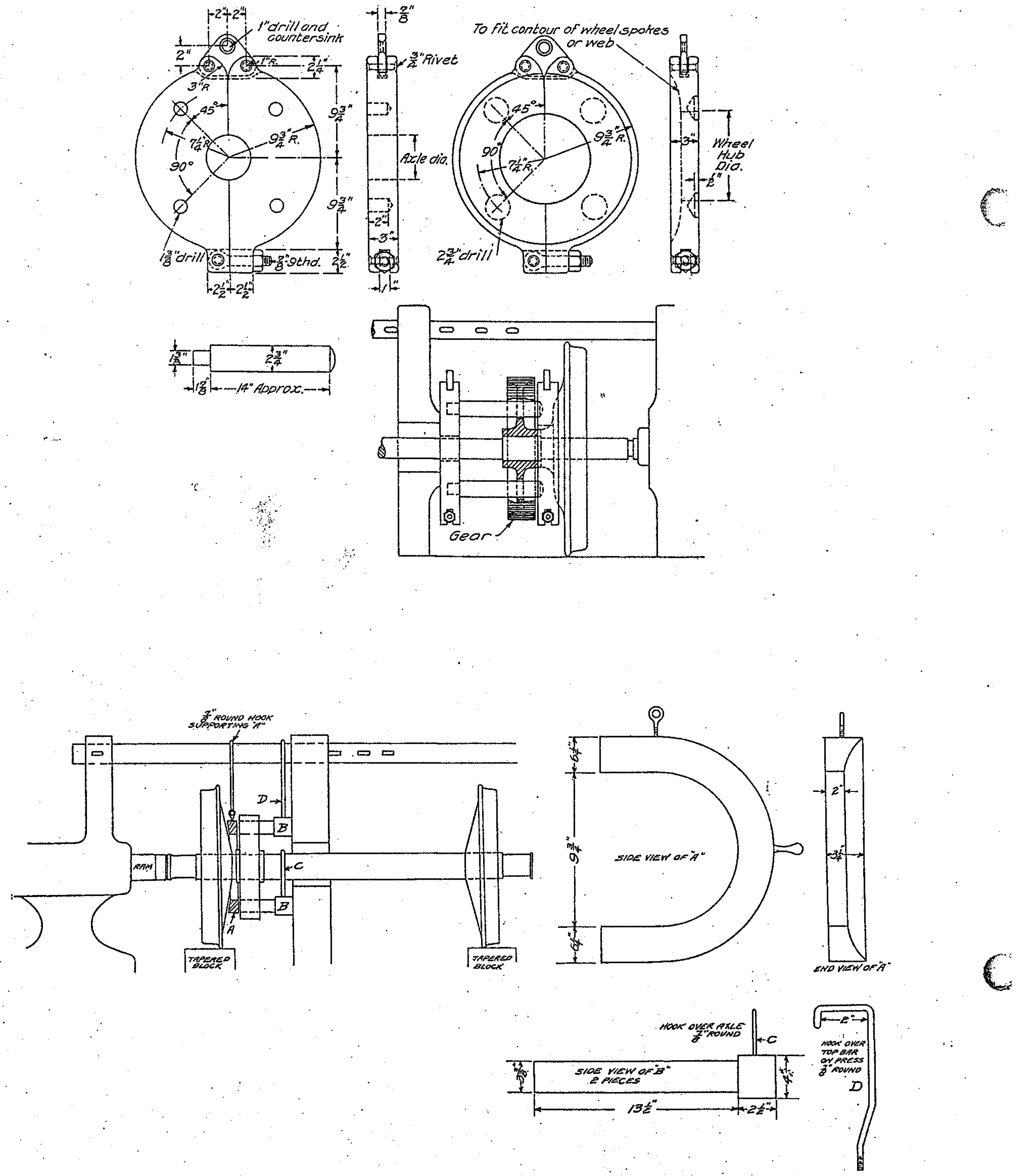
It is very important when heating the rims that the temperature does not exceed 500 deg. F., otherwise the temper will be drawn and the value of the treatment lost. The rims may be heated with a gas ring of such diameter that when placed in the center of the rim the flame will play on the bore. A piece

SPRING GEAR

The Spring Gear was designed for heavy traction duty and has been in service since the early part of 1914. Its operation has been so satisfactory and its cushioning effect so beneficial to the entire equipment, that it is considered a valuable feature on locomotives equipped with twin geared motors.

The gear consists of a cast or forged steel center or hub with six or more twined projecting arms and a forged rim with internal arms corresponding to the number of arms on the hub. The width and depth of the grooves between the twined arms on the hub are made large enough (sliding fit.) to accommodate the rim arms.

44419-8 Railway Motor Gears and Pinions



TYPES OF POPULAR WHEEL PRESS JIGS

To assemble the gear, the rim is placed over the hub, the six rim arms passing through the spaces in the hub until the rim arms are opposite the grooves in the hub arms. The rim is then turned on the hub, the rim arms passing into the grooves in the hub arms until the rim arms register with the hub arms. The springs are now assembled on their spring seats, clamped together, placed in position and the clamps removed. The three semicircular flanges, which will be noticed on the ends of the spring seats, fit into corresponding grooves in the rim and hub arms; these prevent the spring seats from working out.

Gear Bore

The bore of split gears should always be given as equal to the diameter of the axle. Gears are bored in the factory with shims fourteen mils thick between the halves. When mounting on the axle, these shims should be removed from the gear to insure a clamping fit.

It is standard practice to make the diameter of the bore of solid gears smaller than the axle by (.001 in.), one mil per inch diameter, for grades A, D, E, F and L, and (.0015 in.), one and one half mils, for grades M and K. On axles ranging from 4 to 6 inches in diameter, with a smoothly finished surface on the axle gear seat and the bore of the gear, this allowance will give a sufficient mounting pressure to prevent the gear from turning when mounted without a key.

Solid Versus Split Gears

Except for old type axles on which the wheel and gear fit are the same diameter, solid gears have practically replaced the split type, due to their superior operation, freedom from broken bolts, loose gears, scored axles, and lower first cost. In most cases it is impossible to obtain the advantage of modern high grade gearing except by the adoption of the solid type.

In the past, the most serious objection to the adoption of the solid gear has been the question of pressing the wheel and gear on and off the axle but with the advent of the long life gear, the life of which will equal or outlast that of the wheel, this objection has entirely disappeared.

Practically all general repair shops are equipped with a wheel press and, with the proper jig, the wheels can be removed without disturbing the gear. The drawings on the page opposite show the details of two popular forms of jigs which are used on large systems.

PINIONS

All grades of pinions are made from forged carbon steel billets and differ in physical characteristics according to the manner of forging, carbon content and method of heat treatment after machining.

All billets are forged from cast steel ingots cropped of sufficient metal to insure freedom from piping and segregation, thereby affording uniformity of strength and texture throughout.

Finish

The standard allowable finish limits on gears and pinions come well within the finish limits adopted as standard by the American Electric Railway Association in 1915.

The ends of all gear hubs which bear against the axle lining flanges are given a Journal Finish to insure minimum wear on the flanges.

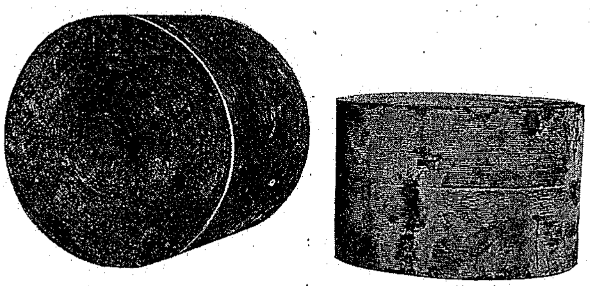
The bores of all pinions are reamed after treatment to insure a uniform bearing on the armature shaft.

Combination roughing and finishing cutters are employed in cutting the teeth of all gears; the teeth of Grades F, K and M pinions are given a second cutting to insure the maximum accuracy in tooth dimensions.

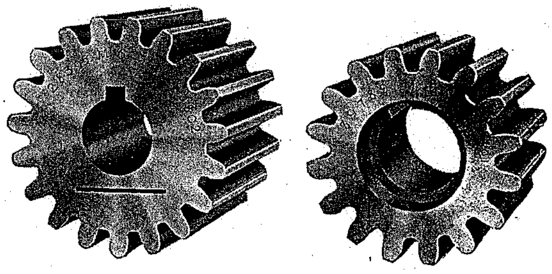
The excessive scale which forms on the teeth during treatment is removed before shipment to prevent the possibility of its becoming mixed with the lubrication and acting as cutting substance after the gears or pinions are placed in service.

Each gear and pinion is marked in accordance with the standard method adopted by the American Electric Railway Association except that the General Electric Company monogram is placed between the year and serial number instead of following the serial number; this arrangement is less confusing.

The stamping is placed on the rim of the gear and, when space will permit, on the outer end of the pinion near the base of the teeth. The first number and letter is the catalog number and grade letter. These combined are a positive identification and, if inserted on orders, will assist in making prompt delivery and insure the shipment of material duplicate of that from which they were obtained. The grade letter is followed in succession by numerical number of the month, the year, the General Electric Company monogram and a serial number (consecutive for each month). The serial number combined with the date will aid the purchaser in keeping mileage records.



PINION BLANKS



FINISHED PINIONS

Inspection

All gears and pinions are carefully inspected by a competent corps of inspectors.

It is impossible to determine the meshing qualities of a gear or pinion, especially after treatment, by the ordinary method of checking with scales and gauges; therefore, each gear or pinion, before being passed for shipment, is given a running test on a specially designed Meshing Test Machine.

On this machine are two shafts which correspond to the axle and armature shaft, the centers of which are adjustable and can be accurately spaced by a vernier scale. If gears are being inspected, each gear is placed on the axle and a master pinion, having the same number of teeth as the one with which the gear will mesh in service, is mounted on the armature shaft. The centers are then adjusted to correspond to the gear centers of the motor on which the gear will operate. The gear is then run at high speed in both directions, both light and loaded.

Defects, which might cause excessive noise or impair successful operation of the gear or its companion pinion, are easily detected and either rectified or the gear is scrapped. By placing a master gear on the axle, this same method is followed when inspecting pinions.

Lubricant

There are many good gearing lubricants on the market and the operator should determine from actual service tests on his own equipments which is best suited for his conditions.

The lubricant should be of such consistency that it will level back into the bottom of the pan and be used in such a quantity that it will be picked up by the gear teeth.

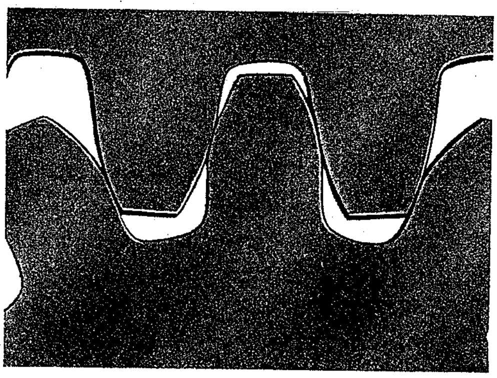
No effort should be spared to keep the lubricant free from dust or sand which is either carried into the pan by the wheel wash or enters through carelessness of the pit-men when removing the gear pan or when adding lubricant through the inspection door.

Maximum Lining Wear

If maximum life is expected from gears and pinions the maximum wear limits of the armature and axle linings should be set at a minimum.

For 3 and 2½ pitch it is good practice to set the maximum limits at $\frac{1}{16}$ for the armature lining and ⅛ for the axle lining; for smaller teeth these limits should be reduced.

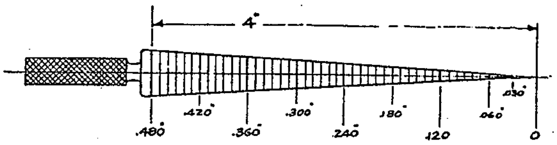
In the illustrations which show the actual size of a new GE-80, 3 pitch, 71-tooth gear and 15-tooth pinion in mesh, one shows the correct mesh when the armature and axle linings are in perfect condition; the other, actual conditions when armature linings are worn $\frac{1}{16}$ and axle linings ⅛ in. Note the



CORRECT MESH

improper mesh and extreme back lash. This causes noise, broken teeth, and a surprising reduction in life.

The illustration shows a handy gauge for checking axle lining wear which can be easily made by the operator. If the axle is



EACH GRADUATION IS $\frac{1}{12}$ OF THE TOTAL LENGTH AND INCREASES THE DIAMETER .010"

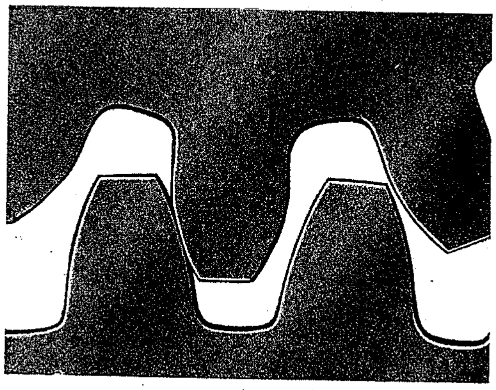
AXLE LINING GAUGE

enclosed in a dust guard which is not provided with an inspection door, a fair estimate of the axle lining wear can be made without removing the guard by jacking up the bearing housing with a block and pinch bar.

Mounting Pinions on Shaft

In pinions of small diameter, the thin section of metal between the root of the teeth and the bore at the large end of the taper is not only subjected to the repeated tooth load stresses but, in addition, the mounting stresses or the stresses set up in the metal when the pinion is mounted on the armature shaft.

It is not uncommon practice to drive the pinion on the shaft with a sledge hammer; the mounting stresses resulting from such a practice often approach the elastic limit of the metal, for the maximum stresses depend on the weight of the sledge hammer used,



INCORRECT MESH

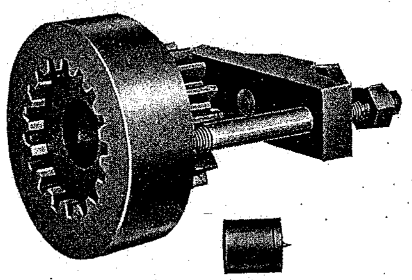
the number of blows delivered and the proficiency of the man who swings the sledge. Such excessive initial mounting stresses, plus the repeated tooth load stresses which are added when the pinion is placed in service, causes the metal in the body of the pinion to fatigue.

Fatigue invariably begins in the metal on the motor end of the pinion, due to the thin section at this point and to the localization of the tooth load on the motor ends of the teeth, which takes place when the axle and armature linings become worn. The fractures occur either in the metal over the keyway and extend through the body of the pinion, or at the base of one or more teeth, beginning in a V shaped fracture and progressing irregularly to a point at the top of the tooth about one third across its face.

The General Electric Company has adopted the boiling water method of mounting pinions on all equipment motors. It has been found by shrinking pinions on the shaft that all parts of the metal contract together resulting in a uniform stress throughout which insures the maximum possible protection against loose pinions and fatigue failures. This method, which has been uniformly adopted by the operators, is described in detail in the following.

The pinion is first slid onto the shaft to make sure that it fits properly and that it will slide easily into place without binding on the SIDES or TOP of the key. It is then placed in boiling water until it is heated clear through, about 45 minutes or more depending on the size. It is then seated quickly on the shaft; a hard wood block one foot long, about the diameter of the pinion, and cupped at one end to clear the armature shaft, is held against the end of the pinion and struck one blow with a ten pound sledge hammer. The pinion nut is quickly set up, the wooden block is replaced, given one more blow and the nut again set up.

PINION PULLER



PINION PULLER

The usual method of driving wedges between the pinion and the bearing housing to remove the pinion, not only damages the bearing housing and the ends of the armature linings, but springs the armature shaft, and subjects the ends of one or more teeth, which bear against the wedges, to injurious shocks, especially so with the case hardened material which is easily damaged and which is removed several times during its life.

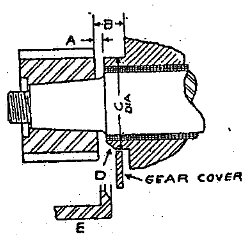
The pinion puller manufactured by the General Electric Company is giving perfect satisfaction and, as it grips all the teeth, the possibility of localizing the removing stresses on two or three teeth is reduced to a minimum.

The puller consists of a ring or puller, designed in the form of an internal gear, and connected to a yoke containing the jack screw by two adjustable studs.

A pressure cap protects the end of the shaft. To remove the pinion, the puller ring is slipped onto the pinion, the internal teeth passing between the teeth of the pinion. When the puller ring teeth are clear of the pinion teeth, the puller is turned on the pinion until the puller teeth register with the pinion teeth. The pressure cap is then inserted and the jack screw set up. If the pinion cannot be removed when the jack screw has been set up to the limit which one man can exert with a 4 ft. wrench, a good rap with a ball-peen hammer on the end of the jack screw or top of the pinion teeth, will readily start the pinion.

To prevent injury to the top of the teeth, a suitable metal protection should be provided and, to transmit the desired effect of the blow to the body of the pinion, the bottom of the latter should be blocked up rigidly to the floor.

This puller can be assembled on all types of split frame motors and all modern box frame G-E motors but, on some of the old types of box frames and all Westinghouse box frame motors, it cannot be assembled unless the outside diameter of the pinion teeth is at least ¾ in. greater than the projection shown as C diameter in the following diagram.



SECTION OF PINION AND FRAME HEAD

Space B will permit a puller jaw thick enough to withstand the stresses. On the latter types of G-E box frame motors the frame heads are chamfered as shown at D. This, plus the space A, which is usually ¼ in., will accommodate a puller jaw similar to E and permit the use of a satisfactory puller on the minimum pinions.

Rather than use wedges and damage the pinions, it would be more economical for the operator to chamfer his frame heads in the same manner when the motors are being overhauled.

Heat Limits

As all car barns have facilities for boiling water, this method of heating pinions before mounting is strongly recommended. The use of a gas oven or gas flame is extremely dangerous for the temper can be easily drawn and the virtue of the treatment lost.

The gas flame is frequently used to expand the hub when removing the gear from the axle. In such cases keep the flame away from the rim and teeth. The GENERAL ELECTRIC COMPANY WILL NOT ACCEPT THE RESPONSIBILITY OF A GEAR OR PINION AFTER A FLAME HAS BEEN APPLIED TO THE TEETH.

All gears and pinions are stamped with the following heat limits which should not be exceeded.

Grade "F" gears and "F" and "H" pinions—"DO NOT HEAT OVER 500° F." Grades "B," "K," "L" and "M" gears and "K" and "M" pinions—"DO NOT HEAT OVER 212° F."

TOOTH DIMENSIONS

Diametral Pitch (P)	Thickness of Tooth on Pitch Line (t)	ADDENDUM (S)		WHOLE DEPTH OF TOOTH (W')	
		Standard	Stub	Standard	Stub
1	1.5708	1.0000	0.6666	2.1571	1.4381
1¼	1.2566	0.8000	0.5714	1.7257	1.2326
1½	1.0472	0.6666	0.5000	1.4381	1.0785
1¾	0.8976	0.5714	0.4444	1.2326	0.9587
2	0.7854	0.5000	0.4000	1.0785	0.8628
2¼	0.6981	0.4444	0.3636	0.9587	0.7844
2½	0.6283	0.4000	0.3333	0.8628	0.7190
2¾	0.5712	0.3636	0.3077	0.7844	0.6637
3	0.5236	0.3333	0.2857	0.7190	0.6163
3¼	0.4833	0.3077	0.2667	0.6637	0.5752
3½	0.4488	0.2857	0.2500	0.6163	0.5393
3¾	0.4189	0.2667	0.2353	0.5752	0.5075
4	0.3927	0.2500	0.2222	0.5393	0.4793
4¼	0.3696	0.2353	0.2105	0.5075	0.4541
4½	0.3490	0.2222	0.2000	0.4793	0.4314
4¾	0.3307	0.2105	0.1905	0.4541	0.4108
5	0.3142	0.2000	0.1818	0.4314	0.3922

GEAR FORMULAE

Factor Given		FACTORS TO BE FOUND: Diametral Pitch	Circular Pitch	Number Teeth	Outside Diameter	Pitch Diameter	Working Depth	Whole Depth	Thickness at P. Line	Root	Addendum	Clearance
		P	P′	N	D	D′	W	W′	t	r	S	f
Diametral pitch	P		$\frac{3.1416}{P}$				$\frac{2}{P}$	$\frac{2.157}{P}$	$\frac{1.5708}{P}$	$\frac{1.157}{P}$	$\frac{1}{P}$	$\frac{.157}{P}$
Circular pitch	P′	$\frac{3.1416}{P_7}$					.6366P′	.6866P′	$\frac{P'}{2}$	.3683P′	.3183P′	.05P′
Teeth and pitch diameter	N D′	$\frac{N}{D'}$	$\frac{D'}{.3183N}$		$\frac{N+2}{\frac{N}{D'}}$							
Teeth and outside diameter	N D	$\frac{N+2}{D'}$	$\frac{D}{.3183(N+2)}$			$\frac{ND}{N+2}$						
Teeth and diametral pitch	N P				$\frac{N+2}{P}$	$\frac{N}{P}$						
Teeth and circular pitch	N P				(N+2) .3183P′	.3183 NP′						
Teeth and addendum	N S				S(N+2)	NS						
Circular pitch and outside diameter	P′ D					D−(.6366P′)						
Circular pitch and pitch diameter	P′ D′			$\frac{3.1416D'}{P'}$	D_7+(.6366P′)							
Diametral pitch and pitch diameter	P D′			D′P	$D'+\frac{2}{P}$							
Diametral pitch and outside diameter	P D			DP–2		$D-\frac{2}{P}$						

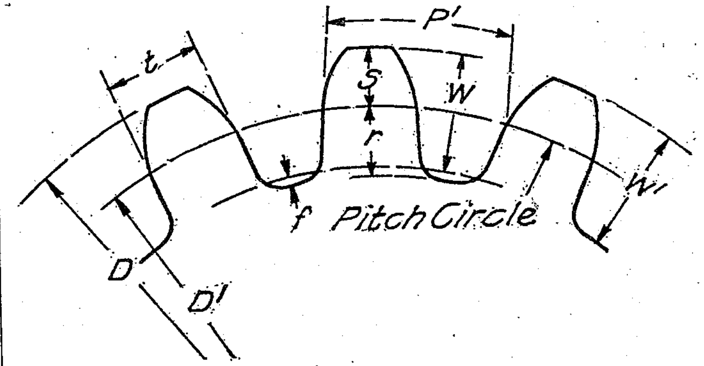
DIAMETRAL PITCH (P) is the standard of measurement, and is equal to the number of teeth for every inch of pitch diameter.

CIRCULAR PITCH (P′) is the distance from the center of one tooth to the center of the next tooth, measured along the pitch circle

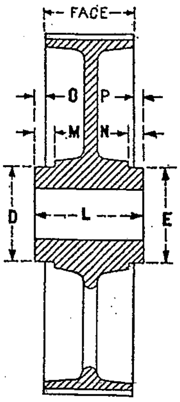
3 P. OR 3 PITCH means 3 diametral pitch.

3″ P. OR 3 INCH PITCH means 3 inch circular pitch.

The distance between centers of two gears is found by adding the number of teeth together and dividing half the sum by the diametral pitch.



CLASSIFICATION AND PRINCIPAL DIMENSIONS



GENERAL ELECTRIC AND WESTINGHOUSE GEARS

Class	Dimensions in Inches										Motors Used For
	Face	D	E	L	M	N	O	P	Keyway* Width	Keyway* Depth	
1	4 1/2	5	5	6 3/4	1	1	1 1/8	1 1/8	3/4	21/64	GE800B
2	4 1/2	6	6	6 3/4	1	1	1 1/8	1 1/8	3/4	21/64	GE800B, GE1000A-C-D, GE 52A-B-H, GE54A-B-H, GE62A, GE63A, GE67A, GE81A, GE85A
3	4 1/2	6 3/4	6 3/4	6 3/4	1	1	1 1/8	1 1/8	3/4	21/64	GE1000A-C-D, GE52A-B-H, GE54A-B-H, GE67A-B, GE78A, GE81A, GE85A
4	5	6 3/4	6 3/4	6 19/32	7/8	1 7/32	1/2	1 3/32	1	33/64	GE51B-C-D-H-J, GE57A, GE72A, GE75A
5	5	8	8	6 19/32	7/8	1 7/32	1/2	1 3/32	1	33/64	GE51B-C-D-H-J, GE57H, GE70A-B-C, GE80A-B-C GE87A-B, GE90A, GE202 A, GE210A-B, GE213A, GE217A-C, GE246A
6	4 1/2	6	6	6 5/32	17/32	1 1/8	17/32	1 1/8	3/4	21/64	GE53A, GE58A-C, GE60A-B GE82A
7	4 1/2	6 3/4	6 3/4	6 5/32	17/32	1 1/8	17/32	1 1/8	3/4	21/64	GE53A
8	5 1/4	8	8	7 13/32	31/32	1 7/8	5/8	1 17/32	1	33/64	GE55A-B-D-F
9	4	6	6	5 7/32	3/32	1	3/32	1 1/8	3/4	21/64	GE59A
10	3 1/2	6	6	4 27/32	21/32	13/16	19/32	3/4	1	33/64	GE61A-B
11	5	8	8	7	1	1 1/2	5/8	1 3/8	1	33/64	GE64A
12	6	12	12	8 7/16	1 5/32	1 7/32	29/32	1 3/16	1 1/4	41/64	GE65A
13	5	8 3/4	8 3/4	7 5/16	7/8	1 3/4	9/16	1 3/4	1	33/64	GE66A, GE211A-C
14	5 1/4	9 3/4	9 3/4	7 5/16	0	1 13/16	1/4	1 13/16	1	33/64	GE68B
15	5 1/4	10 1/2	10 1/2	6 11/16	3/16	1 1/4	3/16	1 1/4	1 1/4	41/64	GE69B, GE207A, GE212A, GE251A, GE257A
16	4 1/4	8 3/4	8 3/4	6	3/4	1 1/8	3/4	1	1	33/64	GE71A
17	5	8 3/4	8 3/4	7 3/4	1 5/16	1 3/4	1	1 3/4	1	33/64	GE73A-C-E, GE205A-B-C-E
18	5 1/2	8 3/4	8 3/4	6 15/16	13/16	1 1/4	5/8	13/16	1	33/64	GE74A
19	3 1/2	6	6	4 3/4	11/32	1 1/32	9/32	31/32	1	33/64	GE77A
20	5 1/4	12	12 1/2	6 11/16	3/8	1 1/2	3/16	1 1/4	1 1/4	41/64	GE76A, GE207A-C
21	3	6	6	4 1/4	1/2	3/4	1/2	3/4	3/4	21/64	GE79A
22	5	6	6	6 31/32	7/8	1 3/32	7/8	1 3/32	1	33/64	GE1200B
24	5	9 3/4	9 3/4	7 5/16	7/8	1 3/4	9/16	1 3/4	1	33/64	GE66B-C, GE211B, GE22F

* Orders should specify width and depth of keyway.

CLASSIFICATION AND PRINCIPAL DIMENSIONS (Continued)

GENERAL ELECTRIC AND WESTINGHOUSE GEARS

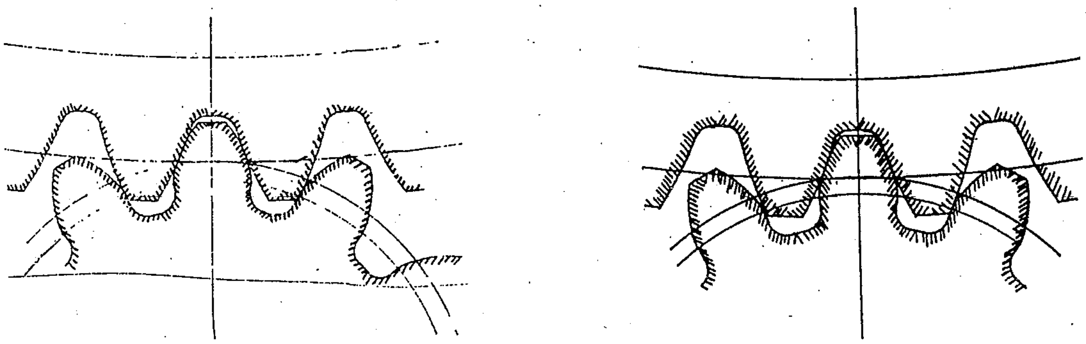
Class	Dimensions in Inches										Motors Used For
	Face	D	E	L	M	N	O	P	Keyway* Width	Keyway* Depth	
25	2	4¼	4¼	3⅜	11/16	11/16	11/16	11/16	½	17/64	GE95A
26	3	5¼	5¼	4¼	⅝	⅝	⅝	⅝	½	17/64	GE96A
27	4½	8	8	6 3/16	9/16	1⅛	9/16	1⅛	1	33/64	GE97A-B
28	5	9	9	6 19/32	⅞	1 7/32	½	1 3/32	1	33/64	GE204A
29	5	9	9	6⅛	⅛	1	⅛	1	1	33/64	GE88A-B-C-D, GE98A-B, GE201A-B-D-E-F-G-H-I-JK GE216A-D-E-F, GE219A-B-C GE226A-B, GE227A, GE230 A-B, GE242-A, B
30	5	10	10	6⅛	⅞	¾	½	⅝	1	33/64	GE210C-D-E-F-G-H, GE217 B-D-E
31	4½	9	9	6⅛	⅜	1¼	⅜	1¼	1	33/64	GE203A-B-G-H-I-J-K-L-M-N-O, GE218A-B-C, GE231, A, GE245-A
32	5	10	10	6⅛	⅛	1	⅛	1	1	33/64	GE214A-B-C-D, GE224A, GE233A
33	5	9½	9½	7¾	1 5/16	1¾	1	1¾	1	33/64	GE205B-D-E
34	5¼	11	11	6⅛	⅛	¾	⅛	¾	1	33/64	GE206A, GE239A
35	5	11	11	6⅛	9/16	⅞	¼	⅞	1	33/64	GE222A-B-C-D-E-G, GE-225A-B-C, GE240-A, GE245-A
36	4½	8	8	6⅛	⅜	1¼	⅜	1¼	1	33/64	GE200A-B-C-D-E-F-G-I-J-K-L
38	5	8	8	6⅛	⅛	1	⅛	1	1	33/64	GE234A, GE241A
44	4	6½	6½	5⅛	1	⅛	1	⅛	1	33/64	GE247A-B-E
45	4	8	8	6⅛	1½	⅝	1½	⅝	1	33/64	GE247C-D
101	5	5 15/16	6¾	6⅛	1⅛	0	1⅛	0	1 / ¾	½ / 21/64	Westinghouse 3W-12W-38W-47W-48W-49W-51W-52W-56W-68W-69W
104	5	10⅞	10⅞	5	⅜	⅜	0	0	1¼ / 1¼	½ / ½	W50-L / W76
106	5	7 15/16	8¾	6⅛	1⅛	0	1⅛	0	1	½	W93-W112
110	5	6¾	6¾	6¾	⅞	⅞	⅞	⅞	1 / ¾	½ / 21/64	W92- W101
111	5	8⅞	10	6⅛	1⅛	0	1⅛	0	1	½	W112
112	5¼	10½	10½	6 11/16	¼	¼	¼	1 3/16	1¼	½	W113
113	5	8	8	6⅛	1	⅛	1	⅛	1	½	W304-W305-W306-W307-W310-W317
114	5¼	9½	9½	6⅛	¾	0	¾	⅛	1	33/64	W303-W321-W322
115	5	9½	9½	7¾	1 5/16	1¾	1	1¾			W12-W38-W47-W48-W49-W51-W52-W56-W68-W69
116	5	6 15/16	7¾	6⅛	1⅛	0	1⅛	0	1 / ¾	½ / 21/64	
117	5	8	8	4 9/32	¼	0	⅛	minus 31/32	1	33/64	W337

* Orders should specify width and depth of keyway.

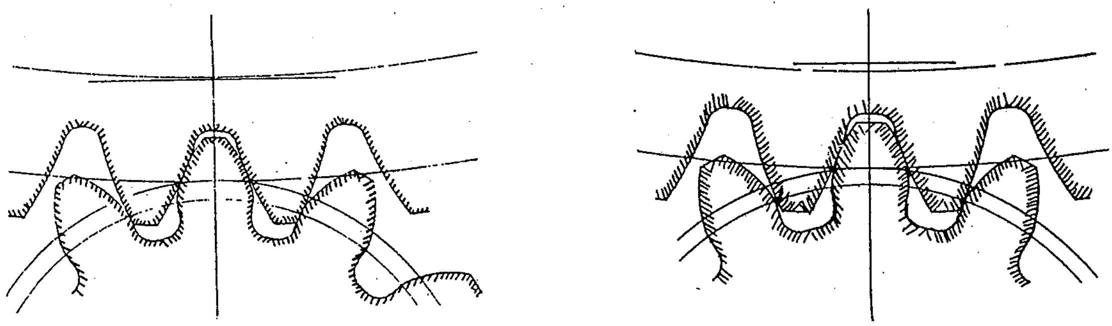
COMPARATIVE SIZES OF GEAR AND PINION TEETH

While in railway practise there is but infrequent occasion to employ other than 2½ and three pitch teeth, several other sizes are shown here for the benefit of engineers who may be interested in making comparisons.

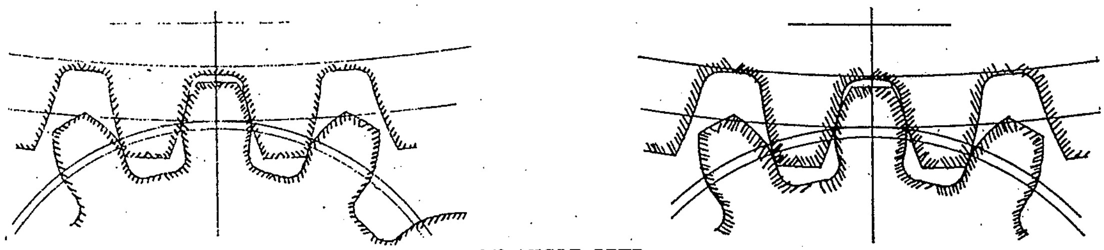
4½ PITCH 12–80 RATIO | **4¼ PITCH 13–74 RATIO**



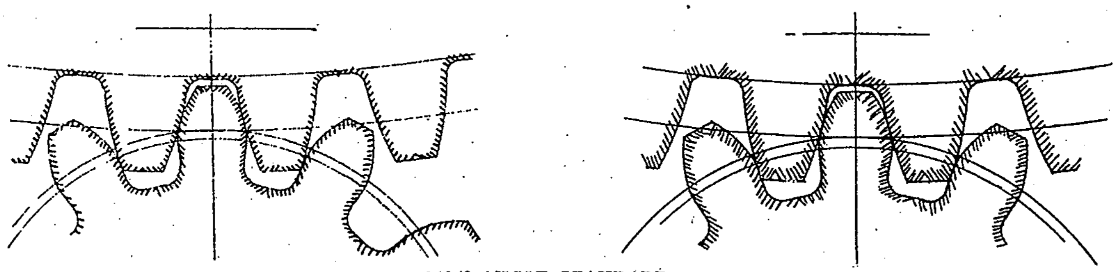
20° ANGLE STUB



20° ANGLE FULL DEPTH



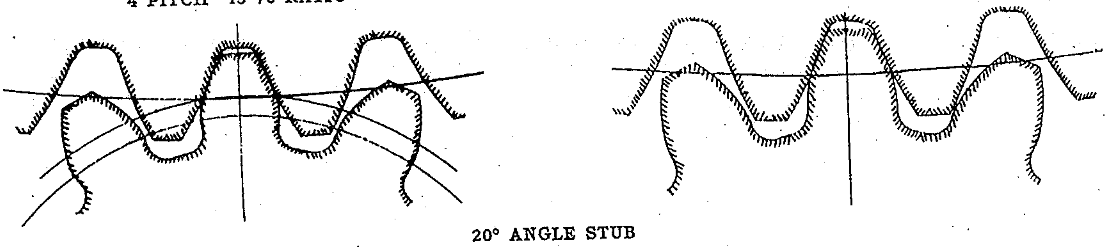
14½° ANGLE STUB



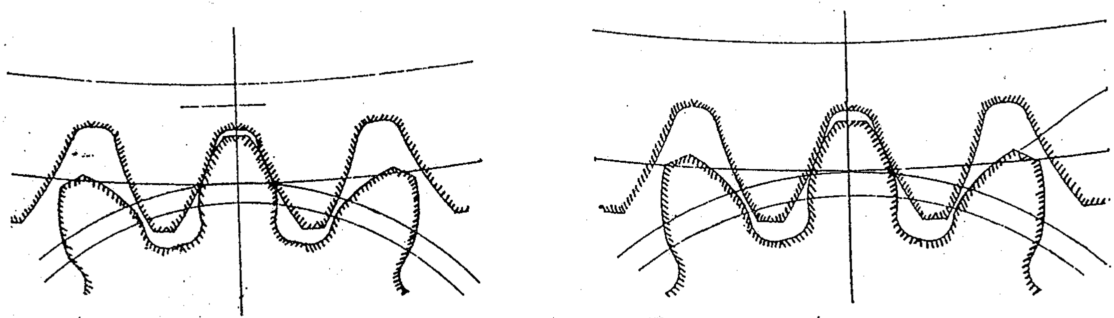
14½° ANGLE STANDARD

COMPARATIVE SIZES OF GEAR AND PINION TEETH

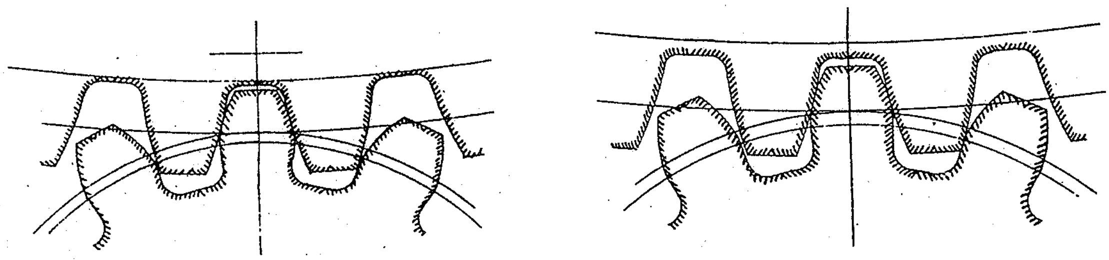
4 PITCH 13–70 RATIO | 3¾ PITCH 15–84 RATIO



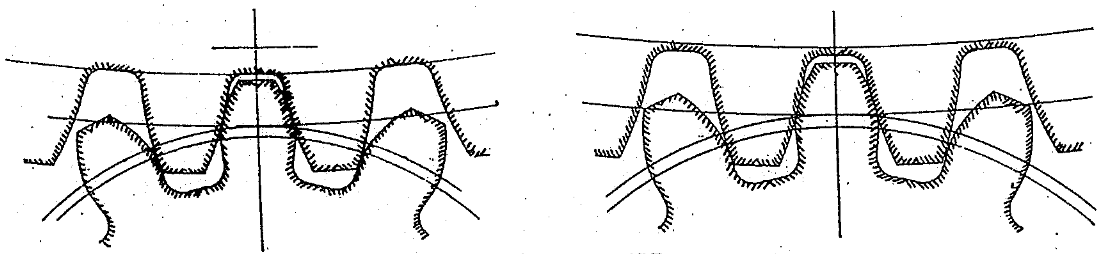
20° ANGLE STUB



20° ANGLE FULL DEPTH



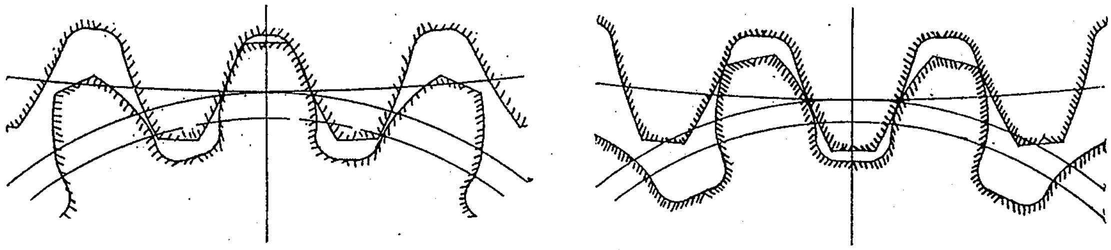
14½° ANGLE STUB



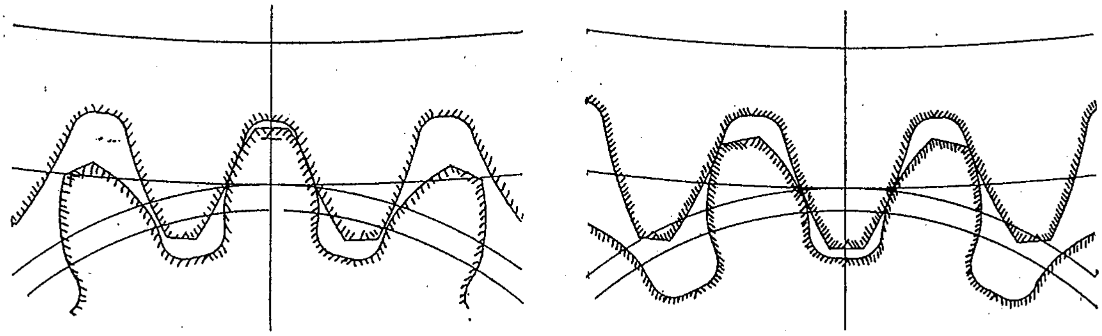
14½° ANGLE STANDARD

COMPARATIVE SIZES OF GEAR AND PINION TEETH

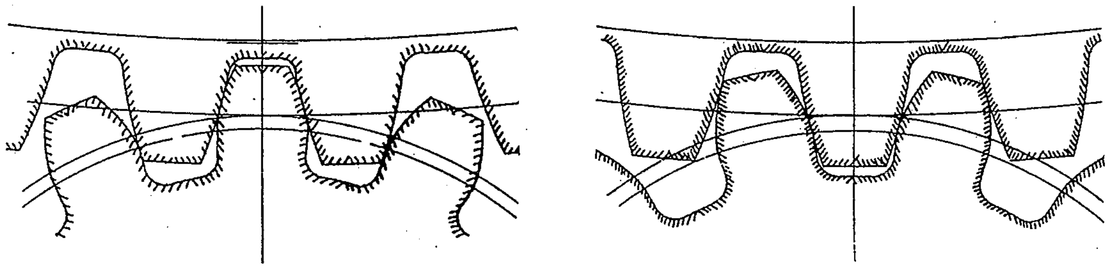
3½ PITCH 15–83 RATIO | 3¼ PITCH 14–77 RATIO



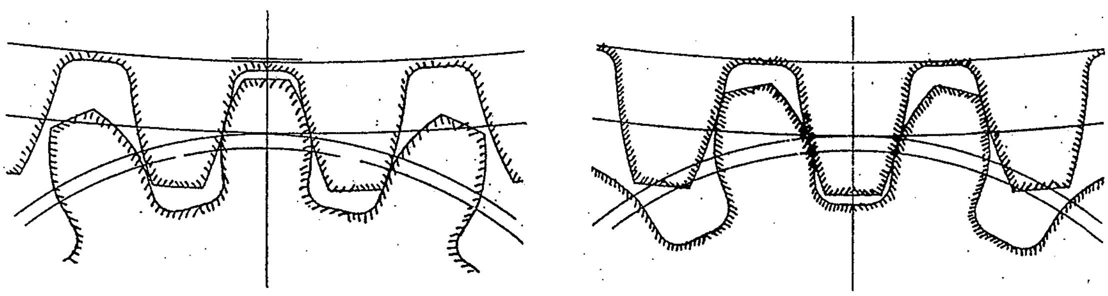
20° ANGLE STUB



20° ANGLE FULL DEPTH



14½° ANGLE STUB

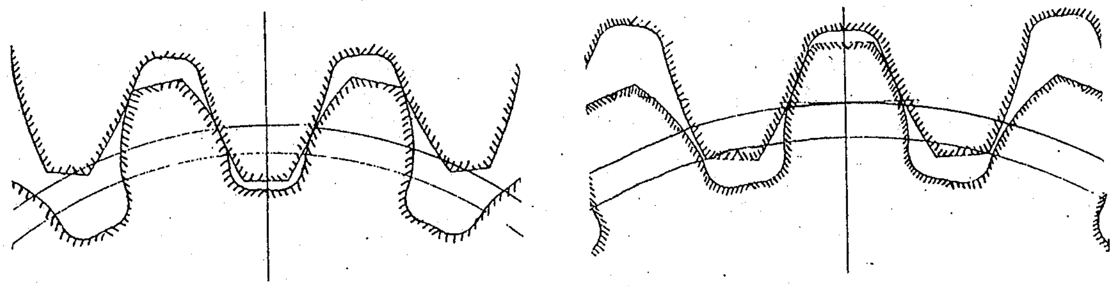


14½° ANGLE STANDARD

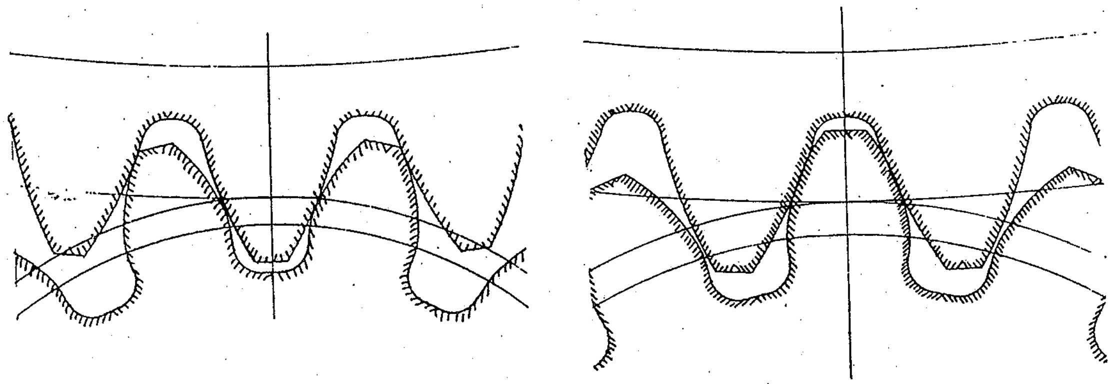
COMPARATIVE SIZES OF GEAR AND PINION TEETH

44419-20 Railway Motor Gears and Pinions

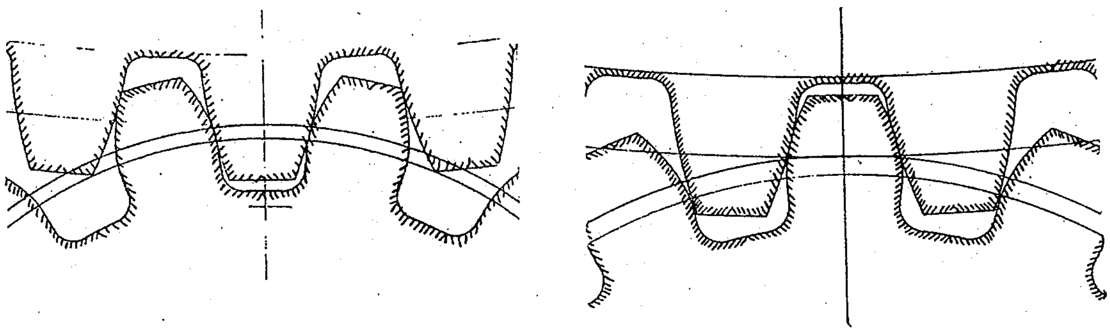
3 PITCH 14–67 RATIO | 2¾ PITCH 16–70 RATIO



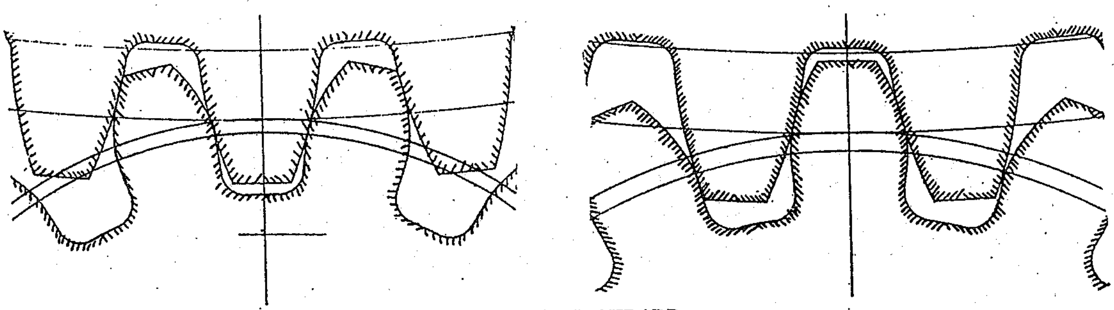
20° ANGLE STUB



20° ANGLE FULL DEPTH



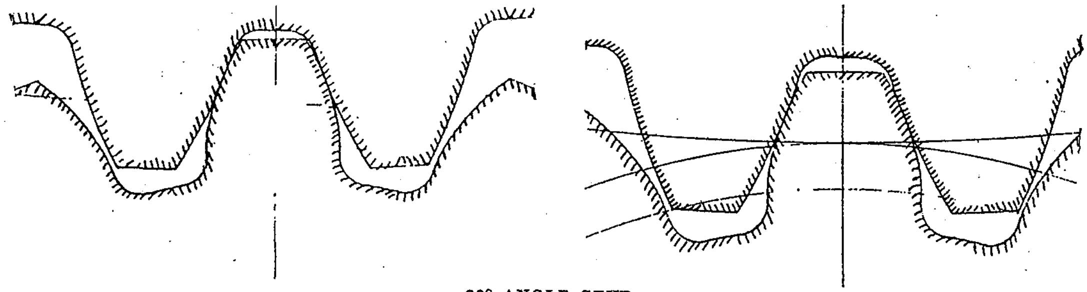
14½° ANGLE STUB



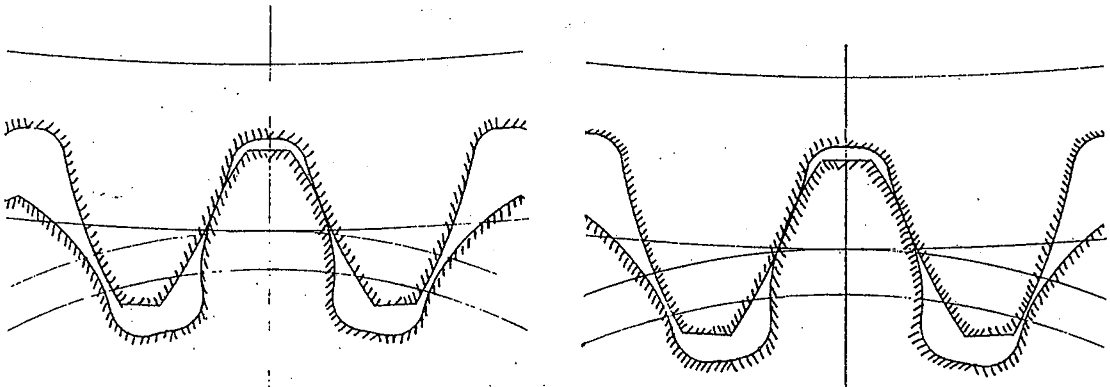
14½° ANGLE STANDARD

COMPARATIVE SIZES OF GEAR AND PINION TEETH

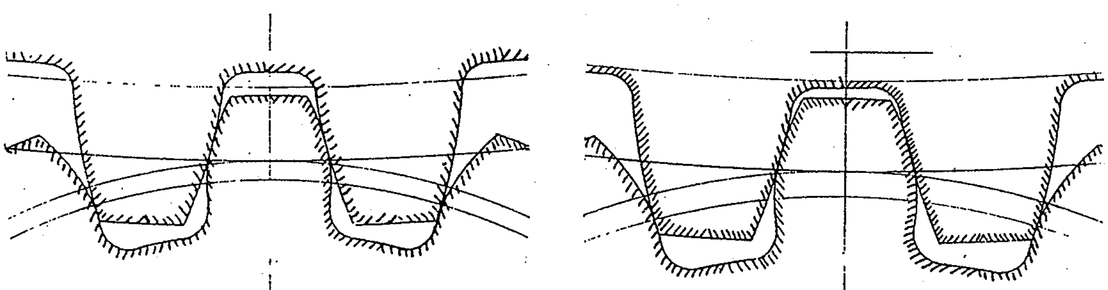
2½ PITCH 16–70 RATIO — 2¼ PITCH 17–64 RATIO



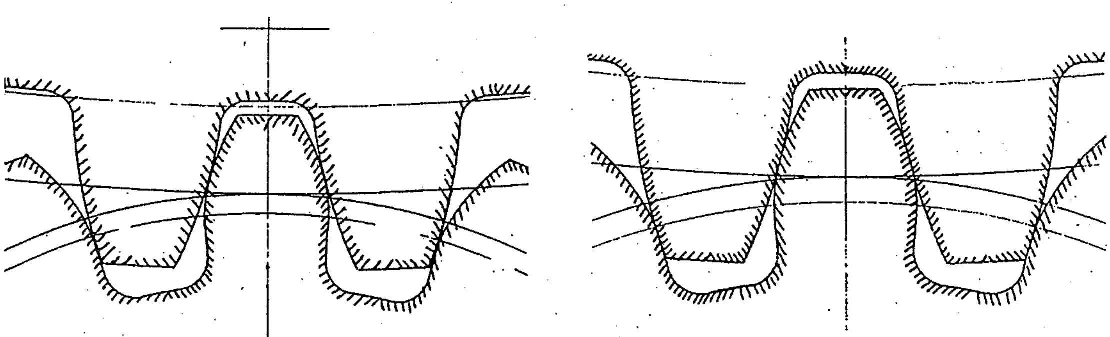
20° ANGLE STUB



20° ANGLE FULL DEPTH



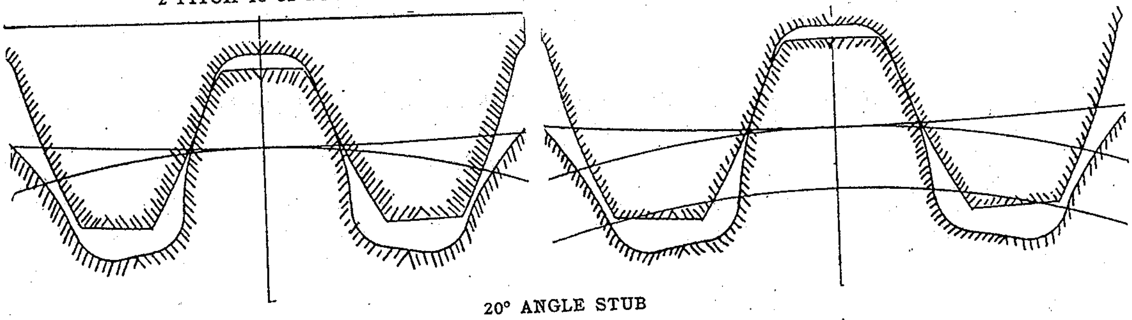
14½° ANGLE STUB



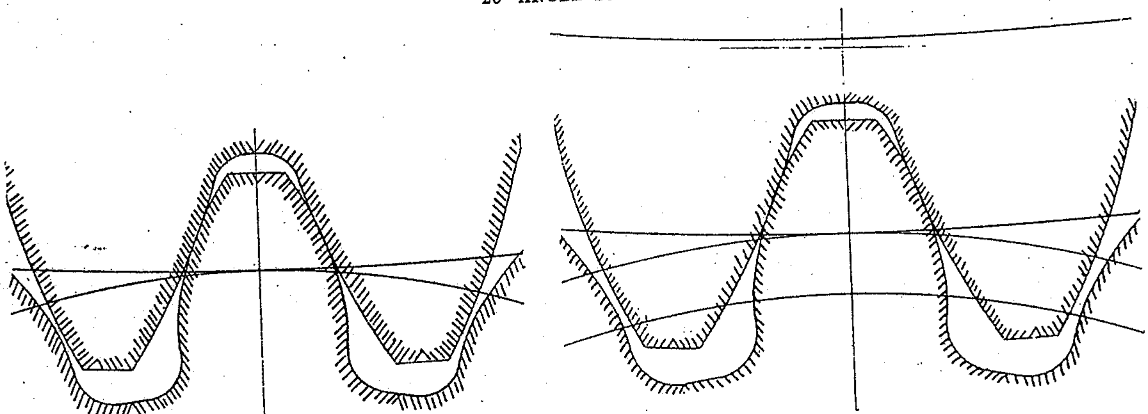
14½° ANGLE STANDARD

COMPARATIVE SIZES OF GEAR AND PINION TEETH

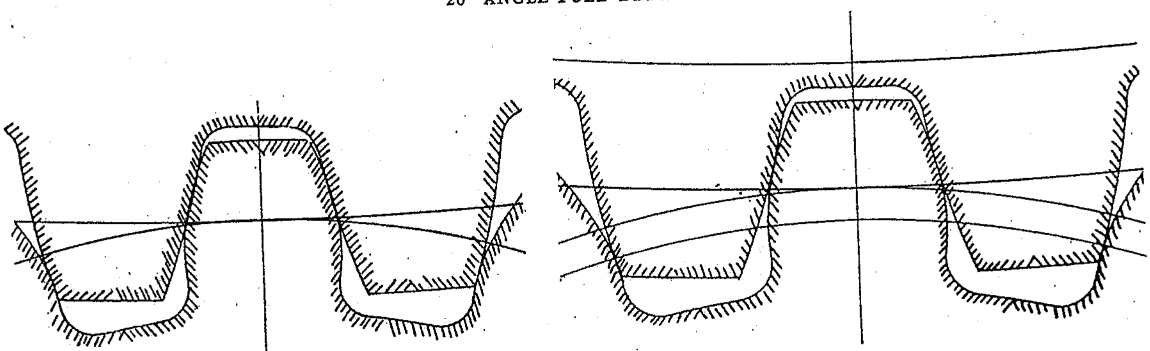
2 PITCH 18–82 RATIO — 1¾ PITCH 18–70 RATIO



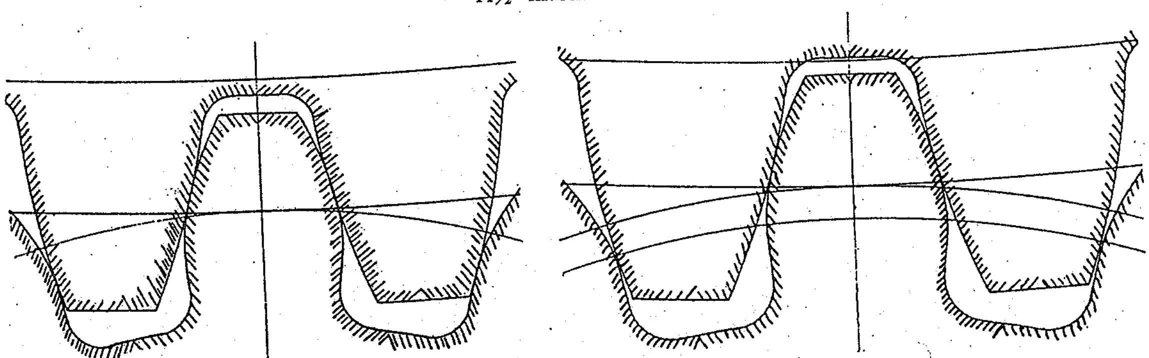
20° ANGLE STUB



20° ANGLE FULL DEPTH



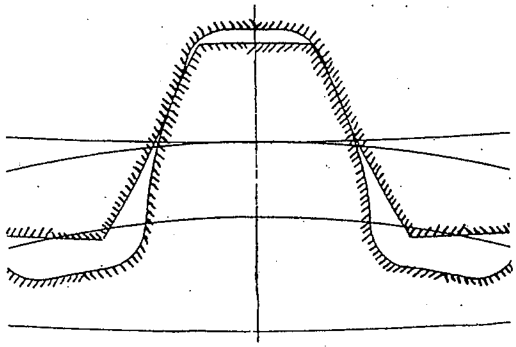
14½° ANGLE STUB



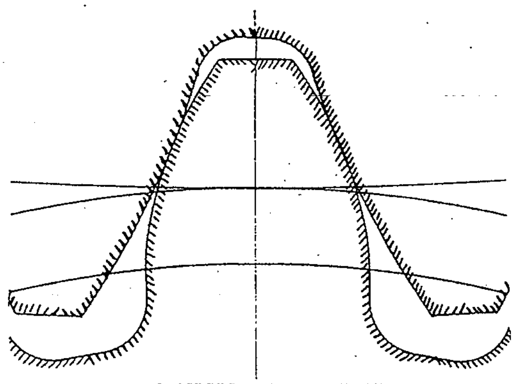
14½° ANGLE STANDARD

COMPARATIVE SIZES OF GEAR AND PINION TEETH

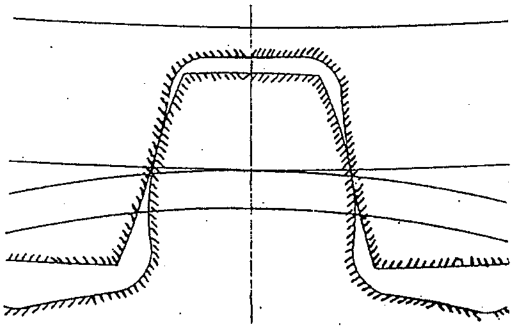
1½ PITCH 18-69 RATIO



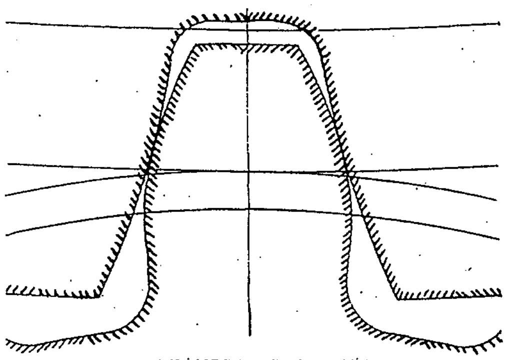
20° ANGLE STUB



20° ANGLE FULL DEPTH



14½° ANGLE STUB

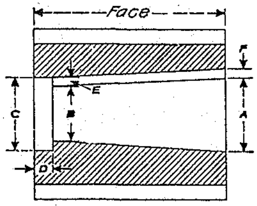


14½° ANGLE STANDARD

DATA SHEET

This sheet, properly filled out, should accompany inquiries or orders for gears or pinions for other than General Electric motors.

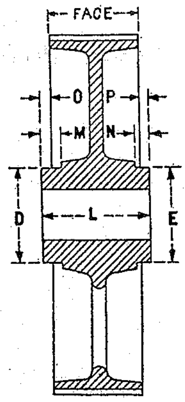
PINIONS



QUANTITY	MOTOR	TEETH	PITCH	FACE	GRADE	DIMENSIONS						
						A	B	C	D	‡E	‡F	WIDTH OF KEYWAY

‡ Give depth of keyway at both ends of pinion.

GEARS



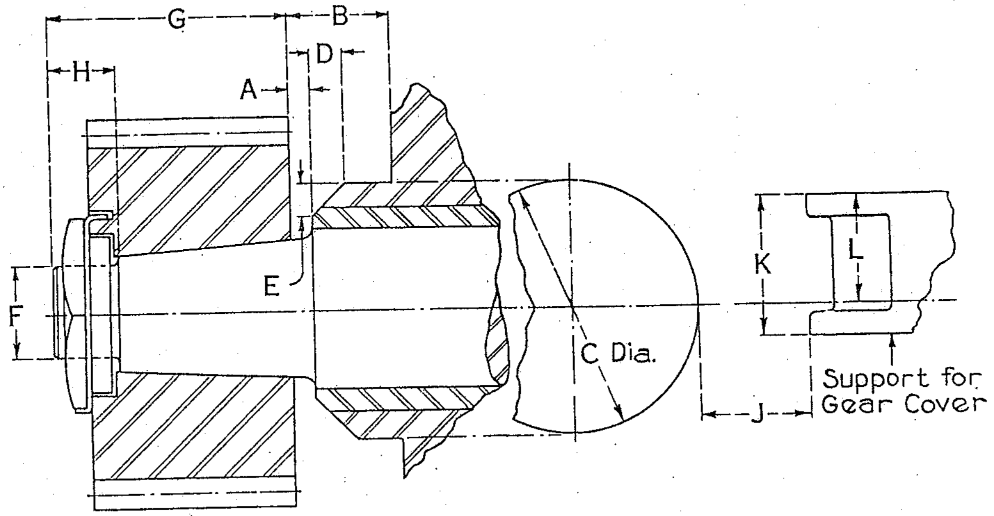
QUANTITY	MOTOR	TEETH	PITCH	FACE	†BORE	SPLIT OR SOLID	KEYWAY		GRADE	DIMENSIONS						
							WIDTH	DEPTH		D	E	L	M	*N	O	*P

† If solid gears are to be bored at factory for pressing fit, exact diameter in thousandths of an inch should be stated; otherwise allowance desired for reboring.
* In case there is no hub extension on the motor side, dimensions N and P become zero and should be so given.

Pinion Pullers for Railway Motors

Information for Ordering

Motor	No. of Teeth	Diam. Pitch	Outside Dia.	A	B	C	D	E	F	G	H	J	K	L



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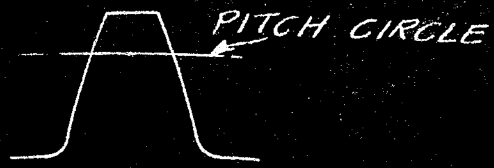
INTERNATIONAL GENERAL ELECTRIC CO., INC.
120 Broadway, New York City, and Schenectady, N. Y.
REPRESENTATIVES AND AGENTS IN ALL COUNTRIES

July, 1926 (3m)



70 TEETH 3 DIAM. PITCH
SHORT ADDENDUM.

14 TEETH 3 DIAM. PITCH
LONG ADDENDUM.

(1) REVISED MAR-25-24.

GEAR TEETH

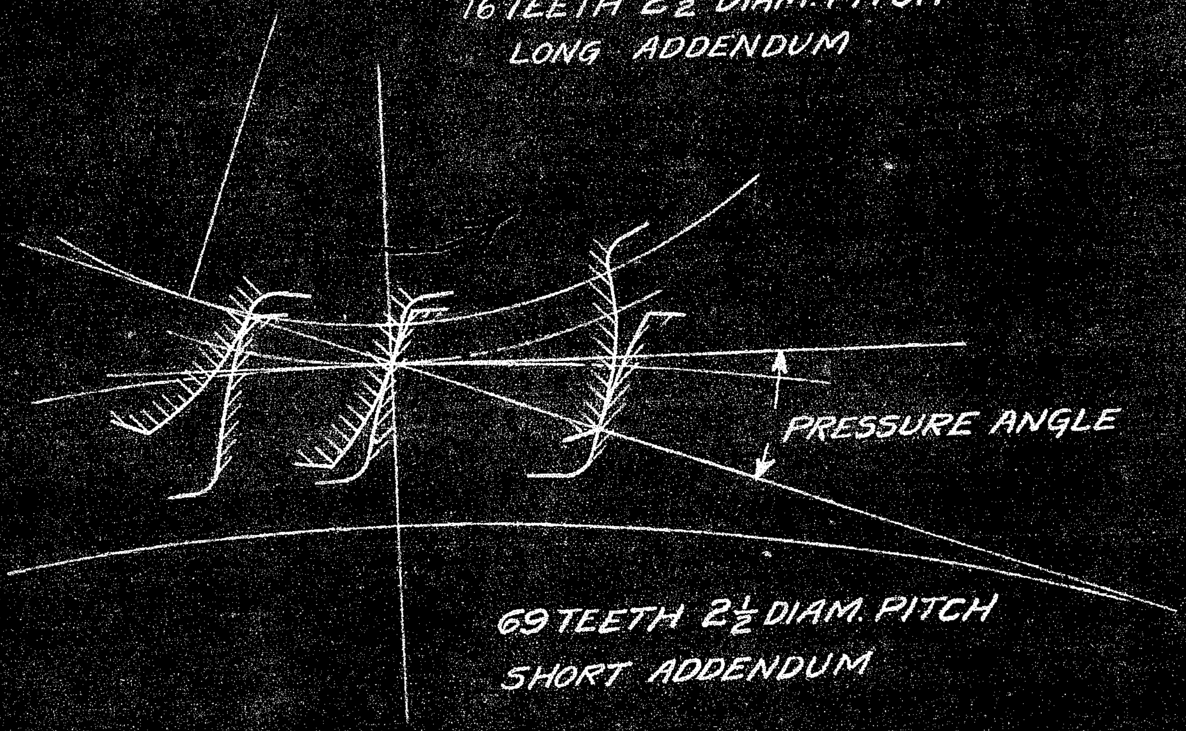
FIRST MADE FOR REQ. BAL 53628

DRAWN BY G. KOTZ, MCH. 18, '24 INSPECTED BY Mar 19 24

K-2675525

GENERAL ELECTRIC COMPANY, SCHENECTADY N.Y.

[illegible] 6-15-22



GEAR TEETH
FOR RY. MOTORS

GENERAL ELECTRIC CO., SCHENECTADY N.Y.

K-267552

DRAWN BY G. KOTZ MCH. 29 '24 INSPECTED Mar-31-24

RY.

G.E. Motors

Motor	Turns	Volts	Rating: Amps	Rating: H.P.	Rating: R.P.M.	Resistances 75°C: Arm.	Resistances 75°C: Exc.	Resistances 75°C: Com.	Resistances 75°C: Total	Coreloss 500V	Coreloss 600V
200	5	600									
201	3	600	104	72	703	·192	·141	·0744	·428	980	13[illegible]
202	3	600	81·5	54	690	·322	·226	·140	·722	850	1090
203	4	600	78	52·5	750	·281	·214	·0905	·6166	780	1040
204	2	600	125	84·5	675	·174	·09	·061	·349	1150	1570
205	2	600	170	117	610	·106	·108	·0412	·2685	1350	1900
205	2	6-1200	127	85	470	·242	·112	·108	·480	750	980
205	3	1200	61	83	565	·800	·519	·322	1·68	960	1300
206	2	600	—								
206	2	6-1200	160	112	475	·126	·110	·053	·308	1300	1900
207	1	600	250	173	595	·0602	·052	·029	·1499	2600	370
207	1	6-1200	220	152	610	·0725	·0585	·0345	·1754	2350	3180
208	1	600	350	245	560	·046	·0322	·0162	·1022	2400	3200
209	1	600	400	275	360	·045	·0310	·0145	·0982	3800	5700
210	3	600	115	77	518	·224	·163	·105	·578	770	1091
211	1	600	272	188	665	·0585	·0526	·0274	·1499	2400	3450
212	1	600	350	245	608	·046	·0322	·0163	·1022	2250	320
213	3	600	81·5	54	690	·322	·226	·140	—	850	1090
214	2	600	125	85	590	·181	·123	·072	·4007	1000	1300
216	3	600	80	53·5	615	·343	·226	·155	·759	710	99
217	3	6-1200	80	53	485	·405	·256	·173	·849	640	79
218	3	600	114	77	530	·223	·158	·077	·4693	960	130
219	3	600	80	53·5	615	·343	·226	·155	·759	710	99
220	1	600	260	180	600	·0481	·0505	·0266	·1406		390
221	1	600	210	145	650	·063	·067	·0295	·1733	2360	32
222	2	600	210	140	683	·068	·0637	·0288	·18	1940	265
223	1	600									
224	2	600									
225	2	600									
226	4	600	70	47	740	·283	·313	·112	·738	730	100
227											
228											

Max axle diam	Motor	HP	Wt. with Gear + Case	Clearance above Rail 33" Wh. Motor	Clearance above Rail 33" Wh. Gear Case	Sum of Gear + Pinion	Max Gear Ratio	Min Gear Ratio	No. of Slots in Arm.	No. Com Bars	Com diam Original	Com dia[m] Worn
4	GE 1200	38	5900			77	60/17	50/27	105	105	8 1/2	7 5/8
4	1000	35	2200	4 13/16	4 5/8	84	67/17	62/22	93	93	8 3/4	6 5/8
4	800	25	1800	5 1/16	4 5/8	81	67/14	67/14	105	105	9 1/8	7 5/8
5 1/2	90	50	2490	4 3/8	4	86	71/15	64/22	29	145	9 5/8	
5 1/2	87	60	3430	3 1/2	4	87	71/16	64/23	43	129	11	9 1/8
5 1/2	80	40	2830	4 1/8	4	86	71/15	64/22	37	111	9 5/8	7 3/4
6	74	65	3580	3 1/2	4 1/8	89	73/16	61/28	39	117	11 1/4	
6	73	75	4150	3 1/2	3 5/8	75	58/17	57/24	39	117	13 1/4	
5 1/2	70	40	2790	4	4 1/8	86	71/15	64/22	37	111	9 5/8	
4 1/4	67	40	2430	4 1/2	4 5/8	84	69/15	60/24	37	111	9 1/8	7 1/2
5 3/4	66	125	4430	3 1/2	3 3/4	89	73/16	54/35	39	195	14 3/4	
4 1/2	57	50	3000	3 1/2	4 1/8	85	67/18	59/26	33	99	10 1/4	9 1/4
4	54	25	1860	5 7/8	4 5/8	81	67/14	67/14	29	115 [illegible]	8 1/2	6 7/8
4	52	25	1500			84	69/15	60/24	29	87	8	6 7/8
5	Wh 101B	40	2730	4 1/8	4	84	69/15	58/26	37	111	10 1/4	8 3/[illegible]
5 1/2	93A	60	3360	3 3/8	3 5/8	87	71/16	52/35	45	135	10 1/4	8 1/2
5	92A	35	2270	4 5/8	4 3/16	84	69/15	58/26	41	123	9	7 3/[illegible]
4	49	35	1920	4 3/4	4 1/8	82	68/14	58/24	59	117 [illegible]	8 3/8	6 5/[illegible]
4	12A	25	2270	4 3/4	4 3/8	82	68/14	68/14	47	93	8 3/8	6 7/[illegible]
4	12A	30	2200	4 3/4	4 3/8	82	68/14	68/14	47	93	8 3/8	6 7/[illegible]
4	12A	30	2270	4 3/4	4 3/8	82	68/14	68/14	47	93	8 3/8	6 7/8
4	12	25	2640	4	3 7/8	84	68/14	68/14	47	93	8 3/8	6 7/[illegible]
4	3	25	5600	6 1/2	5	80	62/18	62/18	95	95	8 3/4	6 7/[illegible]
6	112	65	3410	3 3/8	3 7/16	89	73/16	54/35	45	225	12 1/2	11

	Turns in Fields	Size Wire
1500	270	#6
1000	143	#4
800	203	#6
90	90½	#2
87	87½	2x#4
80	110½	#5BWG.
74	70½	1·531x·053
73	120	1·125x·08 + 1x·08
70	110	~~1x·08~~ #5BWG
67	110	~~#5B~~WG. ·22
66	85	~~·22~~ 1⅛x·12 + 1x·12
57	110	·035x 15/16
54	178½	#4
52	153½	# 5
101B	110½	·078x·655
93A	78½	·043 x1·25
92A	125½	·078x·5
49	114	#4
12A	187	#6
12A	187	# 6
12A	187	# 6
12	187	# 6
3	180	# 8
114	60½	·062x 1·25

G.S. 73 has 2½" pitch all others 3" pitch.

DATA ON GENERAL ELECTRIC RAILWAY MOTORS
INTERPOLE TYPE

ELECTRICAL DATA

TYPE OF MOTOR	GE-200	GE-201A 3 turn	GE-202A 3 turn	GE-203A 4 turn	GE-204A	600-V GE-205 2 turn B	6-1200 GE-205 2 turn A	1200 GE-205 3-turn
ARMATURE								
No. of conductors								2 mult.
Coils per slot		3		3		5	5	5
Width conductor		.168"		.120"		0.128 di	2 #10 B&S in M.	.102"
Breadth conductor		.100"		.136"		---		--
Pitch of Coil		S1 to S10		S1 to S9		---		
Specification No.		A-52064		A-52063		A015883	A-15897	A-15899
esistance 75 Deg. C.		.19 ohms		.2830		0.107	.2420	.800
urns per coil		3		4	2	2	2	3
EXCITING FIELD								
Turns per coil		76		93½	46	40	52	100
Total turns		304		374		160	208	400
Width of conductor		101/32"		1"		1"	1"	.182"
Thickness of conductor		.057"		.043"		0.075"	0.06"	--
Specification No.		114635		F-114621		F-43338	F-44471	F-44475-6
Resistance 75 Deg. C.		.15		.2830		0.076	.1115	.519
COMMUTATING FIELD								
Turns per coil		54		63½		40	58	89
Total turns		216		254		160	232	356
dth of conductor		1-11/32"		1.381"		2"	2"	.182" d
hickness conductor		.042"		.032"		0.040"	.025"	-
Specification No.		114636		F-114622		F-43340	F-44473	F-44477-8
Resistance 75 Deg. C		.081		.6160		0.050	0.1078	.322
TOTAL RESISTANCE MOTOR (75°)		.444				0.250	0.4859	1.680
INPUT OUTPUT BASED 75° RISE 1 HOUR ACTUAL								
Volts input		600		600		600	600	1200
Amps. input		113.5		80		175	127	61
Horse power output		77		53.9			85	82
MPH 33" wheels				13.10		20	14	16.7
Tractive effort				1550		2250	2300	1845 lb.
Percent efficiency				83.8		86	83.8	84.8
Percent C.R. loss				8.48		7	10	8.4
Percent core loss				2.23		2	1.2	1.78
ercent gear & friction				5.5		5	5	5.0
mature speed								
Gear ratio				5.53		3.35	3.35	3.35
COMMERCIAL RATING INPUT - OUTPUT. SPEED ETC.								
Volts input		600	600	600	600	600	6-1200	1200
Amps. input		88	75	75	108	144	110	54.5
H.P. output		60	50	50	75	100	75	75
RPM armature		745	710	750	725	645	500	595

DATA ON GENERAL ELECTRIC RAILWAY MOTORS
INTERPOLE TYPE
MECHANICAL DATA

TYPE MOTOR	GE 200	GE-201A 3 turn	GE-202A 3 turn	GE-203A 4 turn	GE-204 A	600V GE-205 2 turn B	6-1200 GE-205 2 turn A	1200 GE-205 3 turn
ARMATURE								
Diameter	14"	14"	13.75"	13.75"	16"	16"	16"	16"
Core length		8½		6.25		11"	11"	11"
No. of ducts				12longt		2	2	2
Width of ducts				15/16 di		3/8"	3/8"	3/8"
No. of slots		39		35		25	37	41
Depth of slots		1.529"		1.453"		1.527	1.675"	1.675
Width of slots		.443"		.513		0.847	0.59	0.504
[illegible]n under slot		~~1/8&5/32~~				3.09"		2.95
[illegible]r gap		1/8&5/32				1/8&5/32		
Com. diam.		11½		10.5		13.25	13.25"	13.25"
No. com. segments		117		105		125	185	205
Arm.bearing pin.end		3-1/8x9"	3¼x8½	2-7/8x8½	3¾x9¼	3¾x10	3¾x10"	3¾x10"
Arm.bearing com.end		2-7/8x7½"	2¾x7½	2-5/8x7	3-3/8x7	3¼x7½	3¼x7½	3¼x7½
EXCITING FIELD								
No. of poles		4		4		4	4	4
Width of core		4½"		4.75"		5.5"	5.5"	11"
Length of core		8½"		6-1/8"		11"	11"	5.5"
Percent pole arc		62½		60				
Length pole cord		6-5/8"		6¼"				
Area magnet frame		20½ sq.in.		14 sq.in.				
COMMUTATING FIELD								
Width core		1-1/8"		1"		1-1/8"	7/8"	1-1/8"
Length core		8¼"		5½"		11"	11"7/8	11"
[illegible]dth pole face		1-7/8"				1-¼"	1¼"	1¼"
[illegible]USH HOLDERS								
Brushes per stud		2		2		2	2	2
Brush length						2¼"	2¼"	2¼"
Brush width		1¾"		1-3/8"		2¼"	1¾"	1½"
Brush thickness		9/16"		9/16"		5/8"	1/2"	3/8"
Coef. brush friction		.378						
WEIGHTS								
Motor complete		2735 lb.		2100 lb.	3425	3940	3940	3650
Mtr.less gear&case						3230	3230	3230
Exciting field coils		212 "		169				
Com.field coils		116 "						
Exciting poles		164 "						
Commutating poles		34 "						
A[illegible]ature complete		609 "		476		985	985	985
G[illegible]r case		130 "						
GEARING								
Total teeth		86	86	84	77	74	74	74
Pitch		3	3	3	2½	2½	2½	2½
Maximum ratio		71/15	71/15	70/14	60/17	58/16	58/16	58/16
MISCELLANEOUS								
Min.dist.bet. hubs		48"	48"	43"	48"	48"	48"	48"
Max. axle diam.		5"	5"	5"	6"	6"	6"	6"
Lgth.of axle bearing		9"	9"	8¼"	10¾"	10¾"	10¾"	10¾"
Ventilated		Fully		Yes		No	No	No
T[illegible]e of Frame			Split	Box		Box	Box	Solid

Fell

GENERAL DATA.

Motor No.	Frame	Rated H.P.	Volts	Max. Gear	Mot. comp. with M.I. gear case.	Mot. without gear case, pinion or axle lining	R.P.M. at rated H.P.
200	Split	34-40	600	67/14	2078	1700	730
200	Box	34-40	600	67/14	2047	1669	730
201	"	55-65	600	71/15	2842	2385	750
201	Split	55-65	600	71/15	2920	2396	750
202	"	40-50	600	71/15	2757	2300	710
203	Box	40-50	600	83/15	2224	1805	750
203	"	40-50	600	70/14	2542	2107	625
203	Split	40-50	600	69/15	2585	2150	625
204	"	62-75	600	60/17	2465	2855	725
205	Box	110	600	58/16	3911	3345	645
205	"	80	6/1200	53/21	3911	3275	500
205	"	80	1200	58/16	3911	3181	595
206	"	125	600	65/17	4650	4082	580
206	"	100	6/1200	65/17	4650	4082	510
207	""	165	600	63/18	5200	4456	614
207	"	140	6/1200	61/20	5200	4500	630
207	"	110	6/1200	67/19	5200	4500	395
208	"	225	600	67/19	6538	5868	580
209	"	275	600	83/19	9646	8686	360
210	"	70	600	71/16	3369	2860	537
210	"	70	600	69/16	3272	2849	537
211	"	135-160	600	72/17	4607	4081	705
211	"	135-160	600	58/17	4674	4185	705
211	"	135-160	600	72/17	4650	4130	705
212	"	235	600	64/19	6000	5343	630
212	"	200	6/1200	65/18	6207	5459	630
213	"	40-50	600	71/15	2586	2119	710
214	Split	80	600	60/17	3909	3291	635
214	Box	80	600	60/17	3804	3186	635
216	"	40-50	600	71/15	2863	2370	635
216	"	40-50	600	71/15	2800	2300	635
217	"	50	6/1200	69/16	3249	2743	500
217	"	50	6/1200	71/16	3440	2945	500
218	"	70	600	71/16	3146	2654	550
219	Split	40-50	600	71/15	2875	2390	635
219	"	50	750	71/15	2875	-	660
220	Box	160	600	61/16	5000	4521	635
221	"	140	600	63/26	3950	3519	650
222	"	140	600	59/19	4100	3484	683
222	"	130	6/1200	55/23	4100	3572	620
224	"	90	600	60/17	3825	-	730
225	"	115	600	61/17	3860	3269	635
225	"	105	6/1200	61/17	3860	3269	635

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CONSTRUCTION DETAILS.

Motor.	ARMATURE BEARINGS Pin.End	Com.End	Max. Bore Axle Bear.	Type Axle	Max. Ht. above Axle C.L.	Clearance with Std. Wheels Frame To rail	Gear Case To rail	Min. Dist. Between Std. Wheel Hubs.	Wheel Diam.
200-ABC	3 X 8	2¾ X 6½	4 1/2	E.A.	13 1/8	5 7/8	4 1/2	48	33
201-AB	3 1/8X 9	2 7/8X 7½	5	E.B.	14	5	3 13/16	48	33
201-FG	3½ X 9	3¼ X 7½	5	E.B.	12 3/4	4 1/4	3 15/16	48	33
201-HI	3½ X 9	3¼ X 7½	5	E.B.	14	5 1/4	3 3/4	48	33
202-A	3¼ X 8½	2¾ X 7½	5	Spl.	12 15/16	5	3 15/16	48	33
203-AB	2 7/8X 8½	2 5/8X 7	5	E.B.	13 5/8	5 3/8	3 7/8	45	33
203-GIKL	3¼ X 8½	3 X 7	5	EA-EB	13 3/4	5 1/4	3 7/8	45	33
203-HMN	3¼ X 8½	3 X 7	5	EA-EB	13 3/4	5 1/4	3 15/16	48	33
204-A	3¼ X 9¼	3 3/8 X 7	6	Spl.	13 11/16	5	3 5/8	48	33
205-BE	3¾ X 10	3¼ X 7½	6	"	13 3/4	3 3/4	4	48	33
205-C	3¾ X 10	3¼ X 7½	5 1/2	"	13 3/4	3 3/4	4	48	33
205-D	3¾ X 10	3¼ X 7½	6 1/2	"	13 3/4	3 1/2	3 7/8	48	33
205-F	3¾ X 10	3¼ X 7½	6 1/2	"	13 3/4	3 1/2	3 7/8	50	33
206-A	4 1/8X 10	3¾ X 7	6	E.C.	15 3/4	4 3/4	4 1/16	50	36
207-ABCDE	4¼ X 10	3¾ X7 9/16	6 1/2	Spl.	15	4 15/16	4 5/16	50	36
209-AB	5½ X 10	5½ X 10	8	"	21	5 3/8	4 1/2	51	48
210-AB	3¾ X 9¼	3 3/8 X 7	4 1/2	"	13 1/8	4 9/16	4 1/8	48	33
210-CDE	3¾ X 9¼	3 3/8 X 7	5 1/2	EB-1	13 1/2	4 9/16	3 15/16	48	33
210-FGH	3¾ X 9¼	3 3/8 X 7	5	E.B.	13 1/2	4 9/16	3 15/16	48	33
211-A	4 X 10	3¾ X 6 7/16	5 1/4	Spl.	16 11/16	3 3/8	3 15/16	50	33
211-B	4 X 10	3¾ X 6 7/16	5 5/8	"	16 11/16	3 1/4	4 3/16	50	33
211-C	4 X 10	3¾ X 6 7/16	5 1/2	"	16 11/16	3 1/4	3 11/16	50	33
212-AB	4¼ X 10	3¾ X 7 9/16	6 1/2	"	17 3/16	3 3/8	3 1/16	50 1/8	33
212-CG	4¼ X 10	3¾ X 7 9/16	6 1/2	"	17 1/8	3 3/8	3 3/4	50 1/8	33
212-I	4¼ X 10	3¾ X 7 9/16	6 1/2	"	17 3/16	3 3/8	3 1/16	51 1/8	33
213-A	3¼ X 8½	2¾ X 7½	5	"	14 1/4	5 1/2	3 15/16	48	33
214-ACD	3¾ X 9¼	3 3/8 x 7	5 1/2	EB-1	14 3/16	4 9/16	3 5/8	48	33
214-B	3¾ X 9¼	3 3/8 X 7	5 1/2	"	13 15/16	3 11/16	3 9/16	48	33
216-A	3¼ X 8½	2¾ X 7½	5	E.B.	13	5 1/4	3 15/16	48	33
216-BC	3¼ X 8½	2¾ X 7½	4¾ -5	Spl.	13	5 1/4	3 15/16	48	33
216-DEF	3¼ X 8½	2¾ X 7½	5	E.B.	13	5 1/4	3 15/16	48	33
217-AC	3¾ X 9¼	3 3/8 X 7	4 3/4	Spl.	13 1/2	4 9/16	4 1/8	48	33
217-BDE	3¾ X 9¼	3 3/8 X 7	5 1/2	EB-1	13 1/2	4 7/16	3 15/16	48	33
218-AB	3½ X 7¾	3¼ X 5¾	5	Spl.	13 1/4	3 1/2	3 3/4	38 1/2	33
218-C	3½ X 7¾	3¼ X 5¾	5	"	13 3/4	3 1/2	3 3/4	38 1/2	33
218-D	3½ X 7¾	3¼ X 5¾	5 1/2	"	13 3/4	3 1/2	3 3/4	38 1/2	33
219-ABC	3¼ X 8½	2¾ X 7½	5	E.B.	13	5 1/4	3 15/16	48	33
220-A	4¼ X 10	3¾ X 7 9/16	5 1/4	Spl.	16 1/4	3 1/2	3 3/4	50	33
221-AB	3¾ X 10	3½ X 6 7/8	5 1/4	"	15 15/16	3 5/16	3 3/4	50	33
222-AB	3¾ X 10	3½ X 6 7/8	5 1/4	"	15 7/8	3 1/4	4 9/16	50	33
222-DE	3¾ X 10	3½ X 6 7/8	6	E.C.	15 7/8	3 1/4	4 9/16	50	33
222-F	3¾ X 10	3½ X 6 7/8	5 1/2	Spl.	15 7/8	3 1/4	4 9/16	50	33
224-A	3¾ X 9¼	3 3/8 X 7	5 1/2	EB-1	14 3/16	4 9/16	3 5/8	48	33
225-BC	3 5/8 X10	3 5/8 X 6 7/8	6	E.C.	16	3 1/4	2 15/16	50	33
226-AB	2 7/8 X8½	2 5/8 X 7	4 1/2	E.A.	13 5/8	5 3/8	3 7/8	48	33
227-A	3½ X 9	3¼ X 7½	5	E.B.	13 3/8	3 3/4	4 1/8	48	33
229-A	5 X 9 5/8	5 X 9 5/8	7	Spl.	20	5	3 5/8	51	44

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TEST DATA.

Mot. No.	Volts	Test H.P.	Test Amps.	Resistance 75 degs. Arm.	Exc. Fld.	Com. Fld.	Total	Core loss at 600 V.	Watts radiated 65°	Armature Winding Coils	Turns	Slots
200	500	37	64½	.415	.334	.142	.930	-	-	3	4	39
201	600	72	104	.190	.150	.081	.444	1320	-	3	3	39
201	600	69	102	.231	.165	.090	.508	-	1490	4	3	29
202	600	54	81.5	.322	.226	.140	.722	1090	1300	5	3	25
203	600	52.5	78	.281	.214	.091	.617	1040	-	3	4	35
203	600	55	82	.300	.266	.108	.706	-	-	5	4	33
204	600	84.5	125	.174	.090	.061	.349	1510	1650	5	2	29
205	600	117	170	.106	.108	.041	.268	1900	1800	3	2	41
205	6/1200	114	165	.110	.110	.044	.281	1900	1700	3	2	41
205	6/1200	85	127	.242	.112	.108	.486	980	1700	5	2	37
205	1200	83	61	.800	.519	.322	1.68	1300	1700	5	3	41
206	6/1200	112	160	.125	.112	.055	.309	1900	2000	3	2	45
207	600	173	250	.060	.054	.029	.150	3700	2300	7	1	29
207	6/1200	117	175	.073	.059	.034	.175	1340	2300	7	1	29
207	6/1200	152	220	.0715	.058	.034	.172	3180	2300	7	1	29
209	600	275	400	.045	.031	.015	.098	5100	3800	5	1	41
210	600	77	115	.234	.163	.105	.528	1090	1650	5	3	25
211	600	188	272	.058	.053	.027	.150	3450	2300	5	1	39
212	600	245	350	.046	.032	.016	.102	3200	2400	5	1	33
212	6/1200	205	290	.058	.037	.023	.107	2450	-	5	1	39
213	600	54	81.5	.322	.226	.140	.722	1090	1300	5	3	25
214	600	85	125	.181	.123	.072	.401	1300	1650	5	2	29
216	600	53.5	80	.343	.226	.155	.759	990	1300	5	3	25
217	6/1200	53	80	.405	.256	.173	.849	790	1600	5	3	29
218	600	77	114	.223	.158	.077	.469	1300	1470	3	3	41
219	600	53.5	80	.343	.226	.155	.759	990	1300	5	3	25
219	750	51.5	60	.477	.451	.235	1.203	920	1300	5	3	29
220	600	180	260	.048	.051	.027	.141	3900	2000	5	1	39
221	600	145	210	.063	.067	.029	.178	3230	-	7	1	33
222	600	145	210	.068	.064	.029	.180	2650	-	3	2	39
222	6/1200	139	200	.082	.072	.031	.200	2260	-	3	2	43
225	6/1200	110	160	.119	.099	.045	.263	1900	3600	4	2	39
225	600	125	180	.085	.076	.036	.216	-	3650	3	2	47
227	600	69	102	.231	.165	.090	.487	-	1490	4	3	29
229	12/2400	-	-	.106	.069	.028	.204	-	-	7	1	45
230	600	-	-	.244	.220	.093	.580	-	-	-	-	-
232	6/1200	44	64	.416	.382	.188	.987	680	-	4	3	31
246	600	41	75	.249	.246	.087	.136	830	-	3	4	35

JAN.11,1913.

JCT:MLG

A-6559

Motor No.	Frame	Rated H.P.	Volts	Max. Gear	Mot. comp. with M.I. gear case	Mot. without gear case, pinion or axle lining.	R.P.M. at rated H.P.
227	Split	50-60	600	71/15	2960	2490	705
229	Box	275	12/2400	87/18	7100	-	- 575
230	"	50-60	600	-	-	-	570
230	Split	50-60	600	-	-	-	570
231	"	40-50	600	69/15	2500	2003	640
232	"	40	6/1200	67/14	-	1930	730
233	Box	62-75	600	-	-	-	-
234	"	50-60	600	61/14	-	2298	585
235	"	250	600	-	-	-	-
246	Split	30-40	600	69/15	2500	2003	630
W-300-B	Box	200-220	600	64/19	6380	-	620
301-B	"	160 -175	600	59/18	5510	4860	725
302	"	125-140	600	61/16	4600	-	670
303	"	100-110	600	61/16	4000	-	680
310	"	60-75	600	69/15	3440	-	605
304	Split	75-90	600	71/16	3550	-	760
305	"	60-75	600	71/16	3550	-	605
306	"	50-60	600	70/14	2850	2473	710
307	"	40-50	600	70/14	2850	-	630

A-6560

MOTOR FORMS & CHARACTERISTICS.

Name & Form	Split or Box	Std. or Spl.	Ventila-tion	Aux. Oil Wells	Fld. Wind.	Arm. Wind.	Brush Holders	Air Circ.	Remarks.
200-A	Split	Std.	Semi	Yes	Strap	Wire	Wood Yokes Adj.	Long.	
200-C	"	"	full	"	"	"	" "	"	Otherwise same as A.
200-B	Box	"	semi	"	"	"	" "	"	" " " "
201-A	"	"	"	"	"	"	Mica Stud Adj.	"	
201-B	"	"	closed	"	"	"	Wood Yokes Adj.	"	
201-F	"	"	semi	"	"	"	Mica Adj.	"	Used 227 Arm. & fld. Dimen. different from A-B.
201-G	"	"	full	"	"	"	" "	"	Otherwise same as "F" "
201-H	Split	"	semi	"	"	"	" "	"	Dimensions differ from "G"
201-I	"	"	full	"	"	"	" "	"	Otherwise like "H"
202-A	"	Ob.	none	Yes	Wire	"	Wood Yoke	Radial	
202-B	"	"	"	"	"	"	" "	"	Pole pc.seat diff. from "A"
203-A	Box	S.S.	full	"	Strap	"	Mica Adj.	Long.	Light weight
203-B	"	"	none	"	"	"	" "	"	" "
203-G	"	Std.	semi	"	"	"	" "	"	Slow speed & heavier than A
203-H	Split	"	"	"	"	"	" "	"	Otherwise same as G.
203-I	Box	"	none	"	"	"	" "	"	" " " "
203-J	"	"	"	"	"	"	" "	"	Spl.Axle Brackets & Gear Ca
203-K	"	"	semi	"	"	"	" "	"	Otherwise same as G.
203-L	"	"	full	"	"	"	" "	"	" " " "
203-M	Split	"	semi	"	"	"	" "	"	Same as H except susp.bolts
203-N	"	"	full	"	"	"	" "	"	" " N " ventilatio
204-A	"	Ob.	closed	"	Wire	"	Wd.Non.Adj.	Radial	
205-B	Box	Std.	"	No	Strap	"	Mica " "	"	
205-C	"	"		"	"	"	" " "	"	Gr.Cen. & Ax. Diam. diff. B
205-D	"	"		"	"	"	" " "	"	" " " " " " B & C
205-E	"	"		"	"	"	" " "	"	Same as B except keyed ax.B
205-F	"	"		"	"	"	" " "	"	Overall dimen. diff.fromB,C D & E.
205-G	"	"		No.Arm. Yes Axle	"	"	" " "	"	Otherwise like B.
205-I	"	"		Yes	"	"	" " "	"	" " "
206-A	"	"	closed	"	"	"	" " "	"	
207-A	"	"	"	No	"	"	" " "	"	
207-B	"	"		No			" " "		Same as A except in bails
207-C	"	"		Yes Pin.End			" " "		Otherwise like "A".
20?-	"	"		No.Arm.End					
207-D	"	"		Yes Arm. No Axle			" " "		" " "
207-E	"	"		Yes			" " "		" " "
208-A	Box	Ob.	closed	No			" " "		
209-A	"	Splitblown		"			" " "		Twin Gears lin.
209-B	"	"	"	"					Same as A except keyed arm.
210-A	Box	Std.	Closed	No	Wire	Wire	Wd.Non.Adj.	Rad.	
210-B				"			Mica " "		Otherwise like A.
210-C				Yes			" " "		Larger ax. & gear.

A-6562

RAILWAY MOTORS AND CAR EQUIPMENTS

Owing to the many types and forms of railway motors and also to the great variation in the detail parts, requisitions covering railway motors and equipments should specify in detail the following:

1.—Type, Class and Form.

It is necessary to state the type, class and form of the motor. For example, GE-203-L.

2.—Voltage.

3.—Motor Leads.

The position of motor leads must be stated in every instance. That is, indicate whether they are on the axle or suspension side of the motor frame.

4.—Armature.

Give the number of turns in the armature coil when known, otherwise the Railway Motor Engineering Department will decide this point on receipt of data covering the work to be done by the motors.

5.—Size of Axle.

Size of axle in the motor bearings, gear fit and axle collars should be given. These should be stated separately for each.

6.—Gear and Pinion.

State whether gears are to be split or solid. If solid, the exact bore should be given to the third decimal point. If exact bore is not specified but size of axle in gear fit is given as called for in 5th clause, the factory will proceed on the basis of boring gears for the usual pressing fit for the diameter of axle specified. Unless otherwise stated, standard keyways will be cut in both solid and split gears. State whether or not axle keys are to be furnished.

Gear ratio, if known, to be given; otherwise to be determined by the Railway Motor Engineering Dept. on receipt of complete service data.

7.—Service Data.

Service data sheet, original signed by the customer, should be sent forward in duplicate attached to the requisition when possible. If not, it should be sent forward in duplicate as soon as possible attached to an instruction sheet.

Service data sheet may be omitted when the requisition states that the motors are to be duplicates of and to be used in the same service as ones already operating to the customer's satisfaction. In this case refer to the requisition by number and data on which the previous motors were furnished.

If the customer assumes entire responsibility for the successful operation of the motors in the proposed service, it should be so stated on the requisition, and service data sheet may be omitted.

8.—Types M and MK Control.

It is unnecessary to give complete detail of the material to be furnished. The following information should be given.

(a) Automatic or non-automatic.
(b) Dashboard or platform type of coupler.
(c) Whether or not lighting material is required; if standard lighting equipment is sufficient, it should be so stated. If it is not sufficient, car lighting data sheet should accompany the requisition or follow as soon as possible attached to an instruction sheet.
(d) Double end or single end operation.
(e) One or two trolleys.
(f) Length of trolley.
The length of the pole equipment is the distance from the butt of the pole to the center of the wheel, and the length of the pole is the corresponding dimension minus the length of the harp and rope eye.
The various lengths for standard high and low speed equipments are given on DS sketch No. 19845.
(g) In case control equipment is furnished without motors, it will be necessary to have the form and type of motor, gear ratio, maximum and minimum line voltage and weight of car, in order to figure the rheostats.
(h) In case any of the details of a standard equipment are to be modified or omitted, such detail should be noted.
(i) Lightning arresters MD or aluminum cell type.

9.—Cylinder control, either Types B or K.

Type and form of controller, cable data sheet and items c to i inclusive under the 8th clause.

G-E NON-COMMUTATING POLE RAILWAY MOTORS

Class and Form of Motor	Armature Turns	Rated H.P.	At Rated H.P.: Armature R.P.M.	At Rated H.P.: Volts	At Rated H.P.: Amperes	° Net Wt. in Lb.	†† Type of Frame	Max. Gear Reduction	General Dimensions in Inches (Approx.): Distance from Center Line of Axle to Bottom of Gear Cover	Distance from Center Line of Axle to Bottom of Motor	Distance from Center Line of Axle to Top of Motor	Overall Length of Motor Along Armature Shaft	Bearings Dimensions in Inches: Axle Linings Max. Dia.	Axle Linings Length	Armature Linings Pinion End Dia.	Pinion End Length	Commutator End Dia.	Commutator End Length	† Characteristic Curve for Max. Gear Ratio See Sec.	Page	No. of Dimension Diagrams
51B	2	80	640	500	140	4000	SOD	69/16	12½	13½	15¼	51⅝	5	9	2¾	7 1/16	2½	5¼	...	...	13433
52A	4	25	650	500	48	1890	SOD	67/14	11⅞	10⅝	11⅞	45⅝	4½	8	2¾	7¾	2½	6⅛	...	...	13492
52A	5	22	590	500	45	1890	SOD	67/14	11⅞	10⅝	11⅞	45⅝	4½	8	2¾	7¾	2½	6⅛	...	...	13492
52A	6	20	453	500	39	1890	SOD	67/14	11⅞	10⅝	11⅞	45⅝	4½	8	2¾	7¾	2½	6⅛	...	...	13492
54A	3	25	670	500	45	1900	SOD	67/14	11⅞	10⅝	11⅞	45⅝	4½	8	2¾	7¾	2½	6⅛	213	1	13612
54H	3	25	670	500	46	1900	SOU	67/14	11⅞	10⅝	11⅞	45⅝	4½	8	2¾	7¾	2½	6⅛	C-2839		*200263
55A	1	160	550	500	275	5415	BOX	56/17	11 29/32	13 3/16	16 3/16	52⅜	5¼	10¾	3¾	11	3¼	7½	239	1	13532
55A	2	90	265	500	160	5415	BOX	56/17	11 29/32	13 3/16	16 3/16	52⅜	5¼	10¾	3¾	11	3¼	7½	239	5	13532
55H	1	160	550	500	275	5500	BOX	60/17	12 13/16	13 3/16	16 3/16	52⅜	6.7	10¾	3¾	11	3¼	7½	239	10	*200333
55H	2	90	265	500	160	5500	BOX	60/17	12 13/16	13 3/16	16 3/16	52⅜	6.7	10¾	3¾	11	3¼	7½	239	9	*200333
57A	4	40	346	500	76	3025	SOD	69/16	12⅜	13	13⅛	51 9/16	4½	9	3¼	8¾	2⅞	6⅛	C-1776		13467
57A	3	50	470	500	93	3025	SOD	69/16	12⅜	13	13⅛	51 9/16	4½	9	3¼	8¾	2⅞	6⅛	227	1	13467
57A	2	50	640	500	92	3025	SOD	69/16	12⅜	13	13⅛	51 9/16	4½	9	3¼	8¾	2⅞	6⅛	227	7	13467
57H	4	40	346	500	76	3050	SOD	71/16	12⅝	13	13⅛	51 9/16	5¼	9	3¼	8¾	2⅞	6⅛	C-2485		*200481
57H	3	50	470	500	92	3050	SOD	71/16	12⅝	13	13⅛	51 9/16	5¼	9	3¼	8¾	2⅞	6⅛	227	20	*200481
57H	2	50	572	500	88	3050	SOD	71/16	12⅝	13	13⅛	51 9/16	5¼	9	3¼	8¾	2⅞	6⅛	227	19	*200481
58A	6	27	359	500	51	2230	SOD	69/15	12¼	12¼	13⅝	36 17/32	4	8	3	7⅞	2⅝	6	221	5	13458
58A	4	37	505	500	68	2230	SOD	69/15	12¼	12¼	13⅝	36 17/32	4	8	3	7⅞	2⅝	6	221	2	13458
58C	4	37	505	500	68	2230	SOU	69/15	12¼	12¼	13⅝	36 17/32	4	8	3	7⅞	2⅝	6	221	2	*200560
58C	6	27	359	500	51	2230	SOU	69/15	12¼	12¼	13⅝	36 17/32	4	8	3	7⅞	2⅝	6	221	5	*200560
60A	6	19	500	500	35	1665	SOD	67/14	11⅞	11½	12¾	33¼	4	8	2¾	7½	2½	5⅝	214	2	13506
60A	4	27	700	500	50	1665	SOD	67/14	11⅞	11½	12¾	33¼	4	8	2¾	7½	2½	5⅝	214	1	13506
60B	6	19	500	500	35	1665	SOU	67/14	11⅞	11½	12¾	33¼	4	8	2¾	7½	2½	5⅝	214	2	*200592
60B	4	27	700	500	50	1665	SOU	67/14	11⅞	11½	12¾	33¼	4	8	2¾	7½	2½	5⅝	214	1	*200592
62A	4	25	650	500	48	1725	SOD	67/14	11⅞	10⅝	11⅞	41 7/16	4½	8	2¾	7 1/16	2½	5⅜	...	...	DS730
62A	5	22	590	500	45	1725	SOD	67/14	11⅞	10⅝	11⅞	41 7/16	4½	8	2¾	7 1/16	2½	5⅜	...	...	DS730
62A	6	20	453	500	39	1725	SOD	67/14	11⅞	10⅝	11⅞	41 7/16	4½	8	2¾	7 1/16	2½	5⅜	...	...	DS730
64A	3	60	365	500	110	3624	SOU	71/16	12 11/16	13	14⅞	46 15/16	4½	10¾	3½	8	3⅛	6⅝	...	...	*200598
66A	1	125	548	500	215	4375	BOX	71/18	12¾	13¼	16 11/16	51 9/16	5¼	10¾	4	10	3¾	6 7/16	C-10564		13663
66B	1	125	548	500	215	4375	BOX	77/17	13½	13¼	16 11/16	51 9/16	6½	10¾	4	10	3¾	6 7/16	...	...	*200601
66C	1	125	548	500	215	4375	BOX	77/17	13 9/16	13¼	17 3/16	51 9/16	6½	10¾	4	10	3¾	6 7/16	...	...	.....
67A	3	40	505	500	71	2478	SOD	69/15	12 3/16	12	13¼	45	4½	8	3	8	2⅝	6⅛	222	2	13593
67A	4	35	340	500	64	2478	SOD	69/15	12 3/16	12	13¼	45	4½	8	3	8	2⅝	6⅛	222	5	13593
69B	1	200	570	500	340	6178	BOX	64/19	13⅝	13⅛	17⅛	51 13/16	6½	11 9/16	4¼	10	3¾	7 9/16	240	2	14214
69C	1	200	470	600	280	6230	BOX	68/18	14⅜	13⅛	17⅛	51 25/32	7½	11 9/16	4¼	10	3¾	7 9/16	C-4666		14339
70A	3	40	510	500	72	2845	SOU	71/15	12 9/16	12 1/16	13¾	49⅝	5	9	3¼	8½	2¾	7½	...	...	14050
73C	2	75	520	500	130	4137	BOX	73/17	12 15/16	13	16	50⅜	5½	10¾	3½	10	3⅛	7 11/16	233	1	13749
74A	2	65	575	500	116	3534	SOU	73/16	12⅞	13	14¼	50 9/16	5½	10¾	3⅝	8¾	3⅛	6¾	...	...	14051
75A	2	50	580	500	89	2845	SOU	70/15	12⅜	12 1/16	13 7/16	49⅝	4¼	9	3¼	8½	2¾	7½	...	...	*208233
76B	1	160	480	550	250	5166	BOX	64/17	13⅝	13⅛	15	51¾	6½	11½	4¼	10	3¾	7 9/16	C-10725		*208229
78A	3	35	575	500	62	2560	SOD	69/14	12 5/16	11⅞	12⅞	47⅝	4¾	9	3⅛	8¼	2¾	6¾	219	1	14255
80A	3	40	510	500	72	2910	SOD	71/15	12 9/16	12⅛	13⅜	49 11/16	5	9	3¼	8½	2¾	7½	223	2	14182
80B	3	40	510	500	72	2890	SOU	71/15	12 9/16	12⅛	13⅜	49 11/16	5	9	3¼	8½	2¾	7½	223	2	14281
80C	3	40	510	500	72	2755	SOU	71/15	12 9/16	12⅛	13⅜	49 11/16	5	9	3¼	8½	2¾	7½	223	2	14340
81A	3	30	610	500	55	2010	SOD	67/14	11⅞	10⅝	10 13/16	45 7/16	4½	8	2¾	7¾	2½	6½	216	1	14274
86A	3	40	500	500	72	2870	SOD	69/15	12⅜	12¼	13⅝	41 17/32	4	9	3¼	8	2¾	6¼	...	...	*214393
87A	2	60	500	500	105	3475	SOD	71/16	12⅝	13	13	51 9/16	5½	10¾	3½	8¾	3⅛	7	229	1	14231
87B	2	60	495	500	105	3510	SOU	71/16	12⅝	12⅜	13⅝	51 9/16	5½	10¾	3½	8¾	3⅛	7	229	1	14253
88C	3	40	480	500	72	3060	BOX	71/15	12 9/16	12 5/16	14	49	5	9	3¼	8½	2¾	7½	224	2	14500
88D	3	40	480	500	72	3075	SOD	71/15	12 11/16	12 5/16	14 1/16	48 5/16	5	9	3¼	8½	2¾	7½	224	2	14508
90A	2	50	590	500	85	2831	SOD	71/15	12 9/16	12⅛	13⅜	49 11/16	5	9	3¼	8½	2¾	7½	C-3227		14254
98A	2	50	510	500	88	3310	BOX	71/16	12 9/16	12¾	13	50¾	5	9	3½	9	3	7½	C-10203		14553
98B	2	50	510	500	88	3276	SOD	71/16	12 11/16	12¾	13	49 9/16	5	9	3½	9	3	7½	C-10203		14554

° Includes gear, gear cover, (for max. gear ratio) pinion, axle linings.
† Section numbers and pages refer to the Railway Equipment book.
†† SOU = Split, opening up.
SOD = Split, opening down.
BOX = Box.
* Photograph numbers, other numbers are for white prints.

G-E 600-VOLT COMMUTATING POLE RAILWAY MOTORS

NOTE.—"Active" standard motors are included in this list as well as detailed information being given in Part II, in order that information may be available in the event of a motor being removed from the active list and detailed information withdrawn. For list of "Active" motors see Page 186, Part II.

Class and Form of Motor	Armature Turns	Rated H.P. at 600 Volts	At Rated H.P. 600 Volts: Amperes	At Rated H.P. 600 Volts: R.P.M.	Rated H.P. at 500 Volts	At Rated H.P. 500 Volts: Amperes	At Rated H.P. 500 Volts: R.P.M.	Cont. Ampere Capacity for Temp. Rise of 65 Deg. C. by Thermometer at Voltages Given Below: 300	450	600	Type of Frame, Ventilation and Type of Fan	Approx. Weight: ‡Motor Alone	Motor, Gear, Malleable ‡Iron Gear Case, Pinion, Axle Lining and Axle Collar	Maximum Gear Reduction	General Dimensions in Inches (Approx.) Distance from Center Line of Axle to: Bottom of Gear Cover	Bottom of Motor	Top of Motor	§Swing up Dimension	Overall Length of Motor Along Armature Shaft	*A.E.R.A. Axle	Bearings Dimensions in Inches: Axle Linings: Max. Bore	Length	Armature Linings: Pinion End: Bore	Length	Commutator End: Bore	Length	No. of Dimen. Diagram or Outline Drawing
200-F	4	40	59	745	33	60	605	26.8	25.7	23.3	B.C.S.F.	1720	2115	67/14	12	10 5/8	13 1/8	25 5/8	41 5/8	EA	4 1/2	8 1/4	3	8	2 3/4	6 1/2	P-1155372
200-G	4	40	59	745	33	60	605	26.8	25.7	23.3	SOD.C.S.F.	1800	2195	67/14	12	10 5/8	13 3/8	....	41 9/16	EA	4 1/2	8 1/4	3	8	2 3/4	6 1/2	P-1155368
200J	4	40	59	745	33	60	605	32.5	33.8	34.0	B.V.S.F.	1720	2115	67/14	12	10 5/8	13 1/8	25 3/8	41 5/8	EA	4 1/2	8 1/4	3	8	2 3/4	6 1/2	14830-A
200K	4	40	59	745	33	60	605	31.7	32.7	32.8	SOD.V.S.F.	1800	2195	67/14	12	10 5/8	13 1/8	....	41 5/8	EA	4 1/2	8 1/4	3	8	2 3/4	6 1/2	14832-A
201-J	3	65	95	710	55	99	570	35.7	30.3	24.1	B.C.S.F.	2370	2845	71/15	12 9/16	12 1/4	12 3/4	27 3/8	48 1/4	EB	5	9	3 1/2	9	3 1/4	7 1/4	P-1155364
201-K	3	65	95	710	55	99	570	35.7	30.3	24.1	SOD.C.S.F.	2465	2935	71/15	12 3/4	11 1/4	14	....	47 1/4	EB	5	9	3 1/2	9	3 1/4	7 1/4	P-1155363
201G	3	65	95	710	55	99	570	45.6	45.3	44.1	B.V.S.F.	2370	2845	71/15	12 9/16	12 1/4	12 3/4	27 5/8	48 1/8	EB	5	9	3 1/2	9	3 1/4	7 1/4	14839-A
201I	3	65	95	710	55	99	570	42.8	41.4	38.8	SOD.V.S.F.	2465	2935	71/15	12 3/4	11 1/4	14	....	47 1/4	EB	5	9	3 1/2	9	3 1/4	7 1/4	14829-A
202A	3	50	75	710	40	72	570	....	....	....	SOD.C.R.D.	2210	2700	71/15	12 9/16	11 1/2	12 1/4	29	49	Spec.	5	9	3 1/4	8 1/2	2 3/4	7 1/2	P-194444
203-C	3	50	73	635	40	72	525	31.5	27.5	21.0	B.C.S.F.	2190	2630	69/15	12 9/16	11 1/4	13 3/4	27	44 11/16	EB	5	8 1/4	3 1/4	8 1/2	3	7	P-1112123
203-D	3	50	73	635	40	72	525	31.5	27.5	21.0	SOD.C.S.F.	2210	2655	69/15	12 9/16	11 1/4	14	....	44 11/16	EB	5	8 1/4	3 1/4	8 1/2	3	7	P-1155365
203L	3	50	73	635	40	72	525	39.9	40.7	40.3	B.V.S.F.	2190	2630	69/15	12 9/16	11 1/4	13 3/4	27	44 11/16	EB	5	8 1/4	3 1/4	8 1/2	3	7	14838-A
203N	3	50	73	635	40	72	525	37.0	37.2	36.6	SOD.V.S.F.	2210	2655	69/15	12 9/16	11 1/4	14	....	44 11/16	EB	5	8 1/4	3 1/4	8 1/2	3	7	14843-A
203P	4	50	74	760	40	72	630	42.0	44.8	46.0	B.V.S.F.	1865	2280	69/15	12 5/8	11 1/8	13 5/8	26 3/4	42 15/16	EB	5	8 1/4	2 7/8	8 1/2	2 5/8	7	14889-A
204A	2	75	108	725	62	104	605	....	....	....	SOD.C.R.D.	2855	3425	60/17	12 7/8	11 1/2	12 3/4	30 5/8	51 1/16	Spec.	6	10 3/4	3 3/4	9 1/4	3 3/8	7	P-194448
205E	2	110	159	625	..	...	...	54.6	45.7	32.6	B.C.R.D.	3345	3920	58/16	12 1/2	12 3/4	13 3/4	28 1/4	50	Spec.	6	10 3/4	3 3/4	10	3 1/4	7 1/2	P-194422
207E	1	165	238	605	..	...	...	....	....	....	B.C.R.D.	4500	5200	64/17	13 11/16	13 1/16	15	30 1/4	51 1/8	Spec.	6 1/2	11 9/16	4 1/4	10	3 3/4	7 9/16	14836
210E	3	70	104	535	60	111	430	44.6	38.8	31.0	B.C.R.D.	2860	3360	71/16	12 9/16	11 15/16	12 15/16	28 3/8	51 3/16	EB-1	5 1/2	9 1/2	3 3/4	9 1/4	3 3/8	7	P-850133
212B	1	235	333	620	..	...	...	119.1	96.0	69.8	B.C.R.D.	5340	6050	64/19	13 7/16	13 1/16	17 3/16	32	51 3/4	Spec	6 1/2	11 1/2	4 1/4	10	3 3/4	7 9/16	P-893161
213A	3	50	75	710	40	72	570	....	....	....	B.C.R.D.	2135	2650	71/15	12 9/16	11	14 3/4	26 3/16	48 3/4	Spec.	5	9	3 1/4	8 1/2	2 3/4	7 1/2	P-801122
214C	2	75	108	635	60	104	534	....	....	....	B.C.R.D.	3185	3760	60/17	12 7/8	11 15/16	13 3/16	28 11/16	50 1/2	EB-1	5 1/2	10 3/4	3 3/4	9 1/4	3 3/8	7	P-855120
216E	3	50	75	635	40	73	525	....	....	....	B.C.R.D.	2390	2880	71/15	12 11/16	11 1/4	13	26 1/4	50 3/16	EB	5	9	3 1/4	8 1/2	2 3/4	7 5/8	P-854173
218C	3	70	104	550	55	97	473	....	....	....	B.C.R.D.	2730	3210	71/16	12 3/4	12 3/8	13 1/4	28 1/2	39 1/8	Spec.	5	9	3 1/2	7 3/4	3 1/4	5 3/4	P-1117159
219B	3	50	75	635	40	71	530	....	....	....	SOD.C.R.D.	2390	2890	71/15	12 11/16	11 1/4	13	....	49 7/16	EB	5	9	3 1/4	8 1/2	2 3/4	7 1/2	P-854171
222G	2	140	199	700	..	...	...	92.1	98.8	101.0	B.V.S.F.	3570	4260	61/17	13 1/8	13 1/4	15 7/8	29 5/16	50 11/16	EC	6	10 3/4	3 7/8	10	3 5/8	6 7/8	14834
226A	4	50	75	750	40	73	610	....	....	....	B.C.S.F.	1810	2200	70/14	12 5/8	11 1/8	13 5/8	26 3/4	43 3/8	EB	5	8 1/4	2 7/8	8 1/2	2 5/8	7	P-1110113
226B	4	50	75	750	40	73	610	....	....	....	B.V.S.F.	1810	2200	70/14	12 5/8	11 1/8	13 15/16	26 3/4	43 3/8	EB	5	8 1/4	2 7/8	8 1/2	2 5/8	7	P-1117104
227A	3	60	89	745	50	90	600	....	....	....	SOD.C.S.F.	2520	2990	71/15	12 3/8	12 3/4	13 1/4	27 3/4	45 1/16	EB	5	9	3 1/2	9	3 1/4	7 1/4	P-1112146
231A	4	50	72.5	685	40	70	570	....	....	....	SOD.C.S.F.	2000	2450	70/14	12 11/16	11 1/8	13 5/8	29 1/16	43 15/16	EB	5	8 3/8	2 7/8	8 1/2	2 5/8	7	P-1116135
233A	2	80	117	640	65	115	537	51.1	49.4	47.5	B.V.S.F.	2850	3350	71/16	12 3/4	11 3/4	14 1/4	27 5/8	50 3/16	EB-1	5 1/2	10 3/4	3 5/8	9 1/4	3 3/8	7	14828
234A	3	55	81	610	45	82	500	....	....	....	B.C.R.D.	2300	2630	61/14	10 7/8	11	11 1/2	24 5/8	48 11/16	EA	4 1/2	10	3 1/4	8 1/2	3	7	P-1155367
236C	3	35	52	770	30	54	620	....	....	....	B.SPEC.V.	1725	2000	57/14	8 13/16	8 3/4	10 1/4	19 15/16	46 5/16	Spec.	4	8 1/4	3	8	2 5/8	6 1/4	P-1158426
†240A	2	110	157	812	..	...	...	93.0	99.0	102.0	B.V.M.F.	3170	3840	59/16	13	12 13/16	13 5/16	28 1/4	50 13/16	EC	6	10 3/4	3 3/4	10	3 1/4	7	14833
241A	3	55	82	600	45	82	490	45.0	49.0	52.0	B.V.M.F.	2355	2730	61/16	11 1/64	11	12	24 1/2	48 9/16	EB	5	8 3/4	3 1/4	8 1/2	3	7	14841
†242B	3	65	96	678	55	98	550	51.4	52.9	54.3	B.V.L.S.F.	2505	2985	71/15	12 11/16	12 1/4	12 3/4	27 5/8	49 5/16	EB	5	9	3 1/2	9	3 1/4	7 1/4	14845
†243A	1	155	223	685	..	...	...	143.0	151.0	155.0	B.V.M.F.	4550	5210	60/17	12 15/16	13 3/8	16 1/16	29	51 13/16	EC	6	10 7/8	4	10	3 15/16	6 1/2	P-1158481
245A	3	60	87	705	50	88	575	42.5	43.3	42.5	B.V.S.F.	2200	2640	69/15	12 9/16	11 1/4	13 3/4	27	44 11/16	EB	5	8 1/4	3 1/4	8 1/2	3	7	14842
246B	4	50	74	635	40	72	528	....	....	....	SOD.S.V.S.F.	2000	2450	69/15	12 5/16	12 1/2	12 7/8	28 7/16	44 1/8	Spec.	4 1/2	9	2 7/8	8 1/2	2 5/8	7	P-1155390
247A	3	40	60	715	33	60	580	34.4	35.8	36.6	B.V.M.F.	1510	1740	58/15	8 31/32	8 11/16	11 1/16	20 3/8	41	Spec.	4	7	2 3/4	6 5/8	2 3/8	4 3/4	14835-A
247D	3	40	60	715	33	60	580	34.4	35.8	36.6	B.V.M.F.	1550	1870	63/15	11 3/16	10 7/16	9 5/16	23 1/4	41 3/8	EA	4 1/2	7 1/2	2 3/4	6 5/8	2 3/8	4 3/4	14979
†248A	1	160	227	659	..	...	...	132.0	138.5	143.0	B.V.M.F.	5050	5720	61/22	13 1/64	13	16 1/16	32	51 11/16	EC	6 1/2	12 5/16	4 1/4	10 1/16	3 3/4	7 1/8	14846
**249A	4	43	64	655	35	63	535	36.0	39.4	42.3	SOD.V.L.S.F.	1910	2320	69/15	12 3/8	11 3/8	13 7/8	....	36 5/16	Spec.	4 1/2	7 3/4	3	7	2 3/4	5 3/8	14887
254A	2	140	198	740	..	...	...	118.2	125.0	128.2	B.V.M.F.	3855	4515	62/17	12 1/2	13 1/4	16	29 1/2	51 1/16	EC-1	6 1/2	10 3/4	3 7/8	10	3 5/8	6 7/8	P-1608637
258A	4	25	37	1236	21	37.8	1010	32.4	34.9	35.2	B.V.M.F.	745	885	74/13	9 5/32	8 11/16	8 1/2	19 7/8	29 31/32	Spec.	4	6	Ball		Ball		P-1654971
258B	4	25	37	1236	21	37.8	1010	32.4	34.9	35.2	B.V.M.F.	776	1000	100/13	12 7/32	8 11/16	8 1/2	22 3/4	29 31/32	Spec.	4 1/2	6	Ball		Ball		P-1654972
259A	2	120	171	842	..	...	...	120.0	127.0	132.3	B.V.M.F.	3400	4000	61/16	12 57/64	12	13 3/4	28 1/2	50 33/64	Spec.	6 3/4	12	3 7/8	10	3 1/2	6 3/4	P-1654939
260A	1	195	274	632	..	...	...	160.0	171.6	177.5	B.V.M.F.	5125	5745	62/21	13 1/8	13	16 1/8	32	51 1/8	Spec.	6 3/4	12	4 1/4	10	3 3/4	7 1/16	P-1654940

G-E 600/1200-VOLT COMMUTATING POLE MOTORS

Class and Form of Motor	Armature Turns	Rated H.P.	At Rated H.P.: Arm. R.P.M.	Volts	Amperes	Cont. Ampere Capacity for Temp. Rise of 65 Deg. C. by Thermometer at Voltages Given Below: 300	450	600	Type of Frame, Ventilation and Type of Fan	Approx. Weight: ‡Motor Alone	‡Motor, Gear, Malleable Iron Gear Case, Pinion, Axle Lining and Axle Collar	Maximum Gear Reduction	General Dimensions in Inches (Approx.) Distance from Center Line of Axle to: Bottom of Gear Cover	Bottom of Motor	Top of Motor	§Swing up Dimension	Overall Length of Motor Along Armature Shaft	*A.E.R.A. Axle	Bearings Dimensions in Inches: Axle Linings: Max. Bore	Length	Armature Linings: Pinion End: Bore	Length	Commutator End: Bore	Length	No. of Dimen. Diagram or Outline Drawing
201G	3	55	655	600/1200	81	38.5	37.0	34.7	B.V.S.F.	2370	2845	71/15	12 9/16	12 1/4	12 3/4	27 3/8	48 1/4	E.B.	5	9	3 1/2	9	3 1/4	7 1/4	14839-A
205E	2	80	490	600/1200	118	....	....	....	B.C.R.D.	3275	3850	58/16	12 1/2	12 3/4	13 1/2	28 3/8	51 5/16	Spec.	6	10 3/4	3 3/4	10	3 1/4	7 1/2	P-194422
205E	2	100	620	600/1200	144	....	....	....	B.C.R.D.	3345	3920	58/16	12 1/2	12 3/4	13 1/2	28 3/8	51 5/16	Spec.	6	10 3/4	3 3/4	10	3 1/4	7 1/2	P-194422
205E	3	80	575	1200	59	....	....	....	B.C.R.D.	3200	3775	58/16	12 1/2	12 3/4	13 1/2	28 3/8	51 5/16	Spec.	6	10 3/4	3 3/4	10	3 1/4	7 1/2	P-194422
206C	2	100	500	600/1200	146	....	....	....	B.C.R.D.	4080	4650	65/17	13 15/16	13 1/4	14 3/4	31 9/16	51 5/8	E.C.	6	10 3/4	4	10	3 1/2	7	P-856137
207E	2	110	390	600/1200	161	....	....	....	B.C.R.D.	4500	5200	64/17	13 11/16	13 1/16	15	30 1/4	51 1/8	Spec.	6 1/2	11 9/16	4 1/4	10	3 3/4	7 9/16	14836
207E	1	145	620	600/1200	210	....	....	....	B.C.R.D.	4500	5200	64/17	13 11/16	13 1/16	15	30 1/4	51 1/8	Spec.	6 1/2	11 9/16	4 1/4	10	3 3/4	7 9/16	14836
212G	1	190	560	600/1200	269	....	....	....	B.C.R.D.	5340	6050	64/19	13 11/16	13 1/8	17 1/2	32	51 3/4	Spec.	6 1/2	12 5/16	4 1/4	10	3 3/4	7 9/16	P-893161
217D	3	50	500	600/1200	76	36.8	33.0	28.1	B.C.R.D.	2820	3315	71/16	12 9/16	12 1/16	13 1/2	28 3/8	51 5/16	EB-1	5 1/2	9 1/2	3 3/4	9 1/4	3 5/8	7	P-851125
222G	2	130	635	600/1200	186	89.6	88.0	84.4	B.V.S.F.	3570	4260	61/17	13 1/8	13 1/4	15 7/8	29 5/16	50 11/16	E.C.	6	10 3/4	3 7/8	10	3 5/8	6 7/8	14834
225B	2	100	660	600/1200	144	79.6	84.1	85.3	B.V.S.F.	3270	4000	61/17	13 9/16	13 1/4	16	29 1/4	48 15/16	E.C.	6	10 3/4	3 5/8	10	3 3/8	6 7/8	14840
232A	3	40	725	600/1200	60	....	....	....	SOD.C.R.D.	1930	2355	67/14	12	11 3/4	10 7/8	....	45 15/16	Spec.	4 1/2	8	3	7 1/4	2 3/4	6 1/4	14837
233A	2	75	625	600/1200	110	46.3	46.5	46.2	B.V.S.F.	2850	3350	71/16	12 3/4	11 3/4	14 1/4	27 5/8	50 3/16	EB-1	5 1/2	10 3/4	3 5/8	9 1/4	3 3/8	7	14828
240A	2	90	674	600/1200	130	72.6	74.6	76.6	B.V.M.F.	3170	3840	59/16	13	12 13/16	13 5/16	28 1/4	50 13/16	E.C.	6	10 3/4	3 3/4	10	3 1/4	7	14833
†254A	2	145	760	600/1200	205	128.0	131.0	128.0	B.V.M.F.	3820	4480	62/17	12 1/2	13 1/4	16	29 1/2	51 11/16	EC-1	6 1/2	10 3/4	3 7/8	10	3 5/8	6 7/8	P-1608637

† Ratings given on tapped field. ‡Weights of gears and pinions are for maximum ratio and linings for maximum axle. Weights of motors of 75 h.p. and under on 600 volts, based on C.S. gears, and do not include axle collar. "Motor alone" is the motor without gear, gear case, pinion, axle linings and axle collar. §Swing up dimension is distance from center of axle to furthest point on suspension side of motor below support. All ratings given here are commercial ratings. **Narrow gauge motor. B = Box. SOD = Split opening down. C = closed. V = ventilated. S.V. = semi-ventilated. S.F. = Series fan. L.S.F. = Large series fan. M.F. = Multiple Fan. R.D. = Radial ducts in armature.

* See page 43 for standard axle preparations.

NOTE.—No data on this page is to be used for guarantees or for construction purposes.

G-E 600-VOLT COMMUTATING POLE RAILWAY MOTORS

NOTE.—"Active" standard motors are included in this list as well as detailed information being given in Part II, in order that information may be available in the event of a motor being removed from the active list and detailed information withdrawn. For list of "Active" motors see Part II.

Class and Form of Motor	Armature Turns	Rated H.P. at 600 Volts	At Rated H.P. 600 Volts: Amperes	R.P.M.	Rated H.P. at 500 Volts	At Rated H.P. 500 Volts: Amperes	R.P.M.	Cont. Ampere Capacity for Temp. Rise of 65 Deg. C. by Thermometer at Voltages Given Below: 300	450	600	Type of Frame, Ventilation and Type of Fan	‡Approx. Weight: Motor Alone	Motor, Gear, Gear Case, Pinion, Axle Lining and Axle Collar	Maximum Gear Reduction	General Dimensions in Inches (Approx.) Distance from Center Line of Axle to: Bottom of Gear Cover	Bottom of Motor	Top of Motor	§Swing up Dimension	Overall Length of Motor Along Armature Shaft	*A.E.R.A. Standard Axle	Bearings Dimensions in Inches, Axle Linings: Max. Axle	Length	Armature Linings, Pinion End: Diam.	Length	Com. End: Diam.	Length	No. of Dimen. Diagram or Outline Drawing
200J	4	40	59.0	745	33.0	60.0	605	32.5	33.8	34.0	B.V.S.F.	1720	2115	67/14	12	10 5/8	13 1/8	25 3/8	41 5/8	E-3	4 1/2	8 1/4	3	8	2 3/4	6 1/2	14830-A
200K	4	40	59.0	745	33.0	60.0	605	31.7	32.7	32.8	SOD.V.S.F.	1800	2195	67/14	12	10 5/8	13 1/8	....	41 5/8	E-3	4 1/2	8 1/4	3	8	2 3/4	6 1/2	14832-A
201G	3	65	95.0	710	55.0	99.0	570	45.6	45.3	44.1	B.V.S.F.	2370	2845	71/15	12 9/16	12 1/4	12 3/4	27 5/8	48 1/8	E-6	5	9	3 1/2	9	3 1/4	7 1/4	14839-A
201I	3	65	95.0	710	55.0	99.0	570	42.8	41.4	38.8	SOD.V.S.F.	2465	2935	71/15	12 3/4	11 1/4	14	....	47 1/4	E-6	5	9	3 1/2	9	3 1/4	7 1/4	14829-A
203L	3	50	73.0	635	40.0	72.0	525	40.0	40.5	40.5	B.V.S.F.	2190	2630	69/15	12 9/16	11 1/4	13 3/4	27	44 11/16	E-6	5	8 1/4	3 1/4	8 1/2	3	7	14838-A
203N	3	50	73.0	635	40.0	72.0	525	37.0	37.2	36.6	SOD.V.S.F.	2210	2655	69/15	12 9/16	11 1/4	14	....	44 11/16	E-6	5	8 1/4	3 1/4	8 1/2	3	7	14843-A
203P	4	50	74.0	760	40.0	72.0	630	42.0	44.8	46.0	B.V.S.F.	1865	2280	69/15	12 5/8	11 1/8	13 5/8	26 3/4	42 11/16	E-6	5	8 1/4	2 7/8	8 1/2	2 5/8	7	14889-A
205E	2	110	159.0	625	...	...	...	54.6	45.7	32.6	B.C.R.D.	3345	3920	58/16	12 1/2	12 3/4	13 3/4	28 1/4	50	Spec.	6	10 3/4	3 3/4	10	3 1/4	7 1/2	14355
ϕ 207E	1	165	238.0	605	...	...	...	77.5	...	...	B.C.R.D.	4500	5200	64/17	13 11/16	13 1/16	15	30 1/4	51 1/8	Spec.	6 1/2	11 9/16	4 1/4	10	3 3/4	7 9/16	14836
212B	1	235	333.0	620	...	...	...	119.1	96.0	69.8	B.C.R.D.	5340	6050	66/17	13 7/16	13 1/16	16 5/8	32	51 3/4	Spec.	6 1/2	11 1/2	4 1/4	10	3 3/4	7 9/16	14571
**218C	3	70	104.0	550	55.0	97.0	473	42.0	30.0	...	B.C.R.D.	2730	3210	71/16	12 3/4	12 3/8	13 1/4	28 1/2	39 1/8	Spec.	5	9	3 1/2	7 3/4	3 1/4	5 3/4	P 1117159
240A	2	105	150.0	655	85.0	147.5	540	84.8	88.0	88.5	B.V.M.F.	3170	3840	59/16	13	12 13/16	13 5/16	28 1/4	50 13/16	E-8	6	10 3/4	3 3/4	10	3 1/4	7	14833
247A	3	40	60.0	715	33.0	60.0	580	34.4	35.8	36.6	B.V.M.F.	1510	1740	58/15	8 27/32	8 11/16	11 1/16	20 3/8	41	Spec.	4	7	2 3/4	6 5/8	2 3/8	4 3/4	14835-A
247D	3	40	60.0	715	33.0	60.0	580	34.4	35.8	36.6	B.V.M.F.	1550	1870	63/15	11 3/16	10 7/16	9 5/16	23 1/4	41 3/8	E-3	4 1/2	7 1/2	2 3/4	6 5/8	2 3/8	4 3/4	14979
247I	3	40	60.0	715	33.0	60.0	580	34.4	35.8	36.6	B.V.M.F.	1510	1740	58/15	8 15/16	8 11/16	10 15/16	20 3/8	41 1/8	E-3	4 1/2	7 1/2	2 3/4	6 5/8	2 3/8	4 3/4	P 2136963
†248A	1	160	227.0	659	135.0	233.0	534	132.0	138.5	143.0	B.V.M.F.	5050	5720	66/17	13 1/64	13	16 1/16	32	51 11/16	E-9	6 1/2	12 5/16	4 1/4	10 1/16	3 3/4	7 1/8	14846
**249A	4	40	60.0	670	33.0	59.5	548	35.0	37.0	38.0	SOD.V.L.S.F.	1910	2320	69/15	12 3/8	11 3/8	13 7/8	....	36 5/16	Spec.	4 1/2	7 3/4	3	7	2 3/4	5 3/8	14887
ϕ 251A	1	235	333.0	620	...	...	...	119.1	96.0	69.8	B.C.R.D.	5340	6100	69/17	14 13/16	12 3/4	16 1/2	32 11/16	51 3/4	Spec.	7	12	4 1/4	10	3 3/4	7 9/16	P 1608640
254A	2	140	198.0	740	115.0	197.0	610	118.2	125.0	128.2	B.V.M.F.	3855	4515	62/17	11 13/16	13 1/4	16	29 1/2	51 11/16	E-9	6 1/2	11 1/4	3 7/8	10	3 5/8	6 7/8	P 1654988
ϕ 257A	1	165	238.0	605	...	...	...	77.5	...	...	B.C.R.D.	4440	5150	64/17	13 11/16	13 1/16	14 7/16	30 1/2	51 1/8	Spec.	6 1/2	12 1/16	4 1/4	10	3 3/4	7 9/16	P 1608672
258C	4	25	37.0	1236	21.0	37.8	1010	32.4	34.9	35.2	B.V.M.F.	746	900	74/13	9 9/32	8 9/16	8 1/2	19 7/8	30 1/16	E-2	4	6 1/2	Ball Bearing				15217
258D	4	25	37.0	1236	21.0	37.8	1010	32.4	34.9	35.2	B.V.M.F.	776	1000	100/13	12 3/8	8 9/16	8 1/2	22 3/4	29 31/32	E-3	4 1/2	6 1/2	Ball Bearing				15073
†259C	2	120	171.0	842	100.0	173.0	680	120.0	127.0	132.3	B.V.M.F.	3400	4000	61/16	12 51/64	12	13 3/4	28 1/2	50 33/64	Spec.	6	12 1/4	3 7/8	10	3 1/2	6 3/4	P 1979228
†260C	1	195	274.0	632	160.0	272.0	522	160.0	171.6	177.5	B.V.M.F.	5125	5745	66/17	13 11/16	13 1/16	16 1/8	32	51 11/16	Spec.	6 1/2	12 1/2	4 1/4	10	3 3/4	7 1/16	P 1889904
263A	3	65	93.5	725	55.0	97.5	585	63.5	68.0	70.5	B.V.M.F.	2550	3014	71/15	12 11/16	12 1/4	12 3/4	27 7/8	48 3/4	E-6	5	9 1/2	3 1/2	9	3 1/4	7 1/4	P 1822962
263D	3	65	93.5	725	55.0	97.5	585	63.5	68.0	70.5	B.V.M.F.	2550	3014	83/15	12 7/16	12 1/4	12 3/4	27 11/16	47 15/16	E-5	4.725	9 9/16	3 1/2	9	3 1/4	7 1/4	P 1889928
264A	4	25	37.0	1236	21.0	37.8	1010	32.4	34.9	35.2	B.V.M.F.	844	1005	74/13	9 9/32	8 9/16	8 3/8	19 7/8	34 7/8	E-2	4	6 1/2	2 3/8	7	2	5	15216
264B	4	25	37.0	1236	21.0	37.8	1010	32.4	34.9	35.2	B.V.M.F.	880	1130	100/13	12 3/8	8 9/16	8 3/8	22 3/4	35 3/32	E-2	5	6 1/2	2 3/8	7	2	5	15231
265A	4	35	59.6	1055	30.0	53.5	900	36.5	39.0	40.5	B.V.M.F.	1134	1415	69/14	9 5/16	9	10 1/8	20 1/8	37 5/16	E-2	4	7 1/2	2 5/8	7	2 3/8	5 1/4	15288
265C	4	35	59.6	1055	30.0	53.5	900	36.5	39.0	40.5	B.V.M.F.	1185	1500	86/14	11 7/16	9	10	22 5/16	37 1/16	E-3	4 1/2	8	2 5/8	7	2 3/8	5 1/4	15289
**269C	3	55	80.0	905	45.0	79.5	750	54.0	58.0	59.0	B.V.L.S.F.	2045	2490	69/15	12 5/8	11 3/8	13 7/8	26 7/8	36 1/8	Spec.	5	8 1/2	3	7	2 3/4	5 3/8	P 2136944
**270A	3	55	80.0	840	45.0	79.5	700	51.0	54.5	56.0	SOD.V.L.S.F.	1975	2420	69/15	12 5/8	11 3/8	13 7/8	....	36 5/16	Spec.	4 1/2	8 1/4	3	7	2 3/4	5 3/8	P 1979224
272A	3	80	117.5	365	65.0	116.0	298	61.0	64.5	66.5	B.V.M.F.	4040	4700	64/15	13 1/2	13 1/4	16	29 1/2	51 11/16	E-9	6 1/2	11 1/4	3 7/8	10	3 5/8	6 7/8	P 1979299
**273B	4	35	59.6	1055	30.0	53.5	900	36.5	39.0	40.5	B.V.M.F.	1134	1500	69/14	9 5/16	9	10 1/8	20 1/8	36 3/8	Spec.	4 1/2	7 1/2	2 5/8	7	2 3/8	5 1/4	P 2136918
275A	3	60	87.5	727	50.0	88.5	595	58.0	54.0	52.0	B.V.M.F.	2175	2410	57/15	10 3/16	10 1/16	11 7/8	24 1/8	47 5/16	E-6	5	9 1/4	3 1/4	8 9/16	3	7 1/16	15290

G-E 600/1200- AND 750/1500-VOLT COMMUTATING POLE MOTORS

Class and Form of Motor	Armature Turns	Rated H.P.	At Rated H.P.: Arm. R.P.M.	Volts	Amperes	Cont. Ampere Capacity for Temp. Rise of 65 Deg. C. by Thermometer at Voltages Given Below: 300	450	600	Type of Frame, Ventilation and Type of Fan	‡ Approx. Weight: Motor Alone	Motor Gear, Malleable Iron Gear Case, Pinion, Axle Lining and Axle Collar	Maximum Gear Reduction	General Dimensions in Inches (Approx.) Distance from Center Line of Axle to: Bottom of Gear Cover	Bottom of Motor	Top of Motor	§Swing Up Dimension	Overall Length of Motor Along Armature Shaft	*A.E.R.A. Axle	Bearings Dimensions in Inches, Axle Linings: Max. Bore	Length	Armature Linings, Pinion End: Bore	Length	Com. End: Bore	Length	No. of Dimen. Diagram or Outline Drawing
201G	3	55	655	600/1200	81.0	38.5	37.0	34.7	B.V.S.F.	2370	2845	71/15	12 9/16	12 1/4	12 3/4	27 3/8	48 1/4	E-6	5	9	3 1/2	9	3 1/4	7 1/4	14839-A
205E	2	80	490	600/1200	118.0	48.5	39.7	29.1	B.C.R.D.	3275	3850	58/16	12 1/2	12 3/4	13 1/2	28 3/8	51 5/16	Spec.	6	10 3/4	3 3/4	10	3 1/4	7 1/2	14355
205E	2	100	620	600/1200	144.0	48.3	32.5	......	B.C.R.D.	3345	3920	58/16	12 1/2	12 3/4	13 1/2	28 3/8	51 5/16	Spec.	6	10 3/4	3 3/4	10	3 1/4	7 1/2	14355
ϕ 207E	1	145	620	600/1200	210.0	68.0	......	......	B.C.R.D.	4500	5200	64/17	13 11/16	13 1/16	15	30 1/4	51 1/8	Spec.	6 1/2	11 9/16	4 1/4	10	3 3/4	7 9/16	14836
ϕ 207E	2	110	390	600/1200	161.0	52.0	34.0	......	B.C.R.D.	4500	5200	64/17	13 11/16	13 1/16	15	30 1/4	51 1/8	Spec.	6 1/2	11 9/16	4 1/4	10	3 3/4	7 9/16	14836
ϕ 212B	1	190	560	600/1200	269.0	89.1	34.0	......	B.C.R.D.	5340	6050	66/17	13 1/16	13 1/16	16 5/8	32	52 1/2	Spec.	6 1/2	11 1/2	4 1/4	10	3 3/4	7 9/16	14571
240A	2	95	662	600/1200	137.0	77.6	81.0	83.5	B.V.M.F.	3170	3840	59/16	13	12 13/16	13 5/16	28 1/4	50 13/16	E-8	6	11 1/4	3 3/4	10	3 1/4	7	14833
**244A	2	100	700	600/1200	145.0	77.0	80.0	80.5	B.V.L.S.F.	3285	3920	67/17	13 1/8	13	16	32 1/8	38 1/8	Spec.	5 1/2	10 1/2	3 3/4	9	3 1/2	5 3/4	14844
254A	2	135	665	600/1200	195.0	111.5	114.0	115.3	B.V.M.F.	3820	4480	62/17	11 13/16	13 1/4	16	29 1/2	51 11/16	E-9	6 1/2	11 1/4	3 7/8	10	3 5/8	6 7/8	P 1654988
ϕ 257A	1	145	620	600/1200	210.0	68.0	......	......	B.C.R.D.	4440	5150	64/17	13 11/16	13 1/16	14 7/16	30 1/2	51 1/8	Spec.	6 1/2	12 1/16	4 1/4	10	3 3/4	7 9/16	P 1603672
263A	3	55	660	600/1200	80.5	56.0	60.0	63.0	B.V.M.F.	2550	3014	71/15	12 11/16	12 1/4	12 3/4	27 7/8	48 3/4	E-6	5	9 1/2	3 1/2	9	3 1/4	7 5/16	P 1822962
						375	560	750																	
**269C	3	50	892	750/1500	58.0	40.0	42.0	40.0	B.V.L.S.F.	2045	2490	69/15	12 3/8	11 3/8	13 7/8	26 7/8	36 1/8	Spec.	5	8 1/2	3	7	2 3/4	5 3/8	P 2136944
						600	900	1200																	
205E	3	80	575	1200	59.0	25.2	19.9	14.5	B.C.R.D.	3200	3775	58/16	12 1/2	12 3/4	13 1/2	28 3/8	51 5/16	Spec.	6	10 3/4	3 3/4	10	3 1/4	7 1/2	14355

† Ratings given on tapped field. ‡ Weights of gears and pinions are for maximum ratio and linings for maximum axle. Weights of motors of 75 h.p. and under on 600 volts, based on C.S. gears, and do not include axle collar. "Motor alone" is the motor without gear, gear case, pinion, axle linings and axle collar. § Swing up dimension is distance from center of axle to furthest point on suspension side of motor below support. All ratings given here are commercial ratings. ϕ Motors also used for locomotives with forced ventilation although ratings given are without forced ventilation. ** Narrow gauge motor. Dimension. "Distance from center line of axle to bottom of gear case" is for maximum gear reduction.

B = Box. SOD = Split opening down. C = Closed. V = Ventilated. S.V. = Semi-ventilated. S.F. = Series fan. L.S.F. = Large series fan. M.F. = Multiple fan. R.D. = Radial ducts in armature.

* See next page for standard axle preparations.

NOTE.—No data on this page is to be used for guarantees or for construction purposes.

STANDARD AMERICAN ELECTRIC RAILWAY ASSOCIATION AXLES

As adopted in 1916

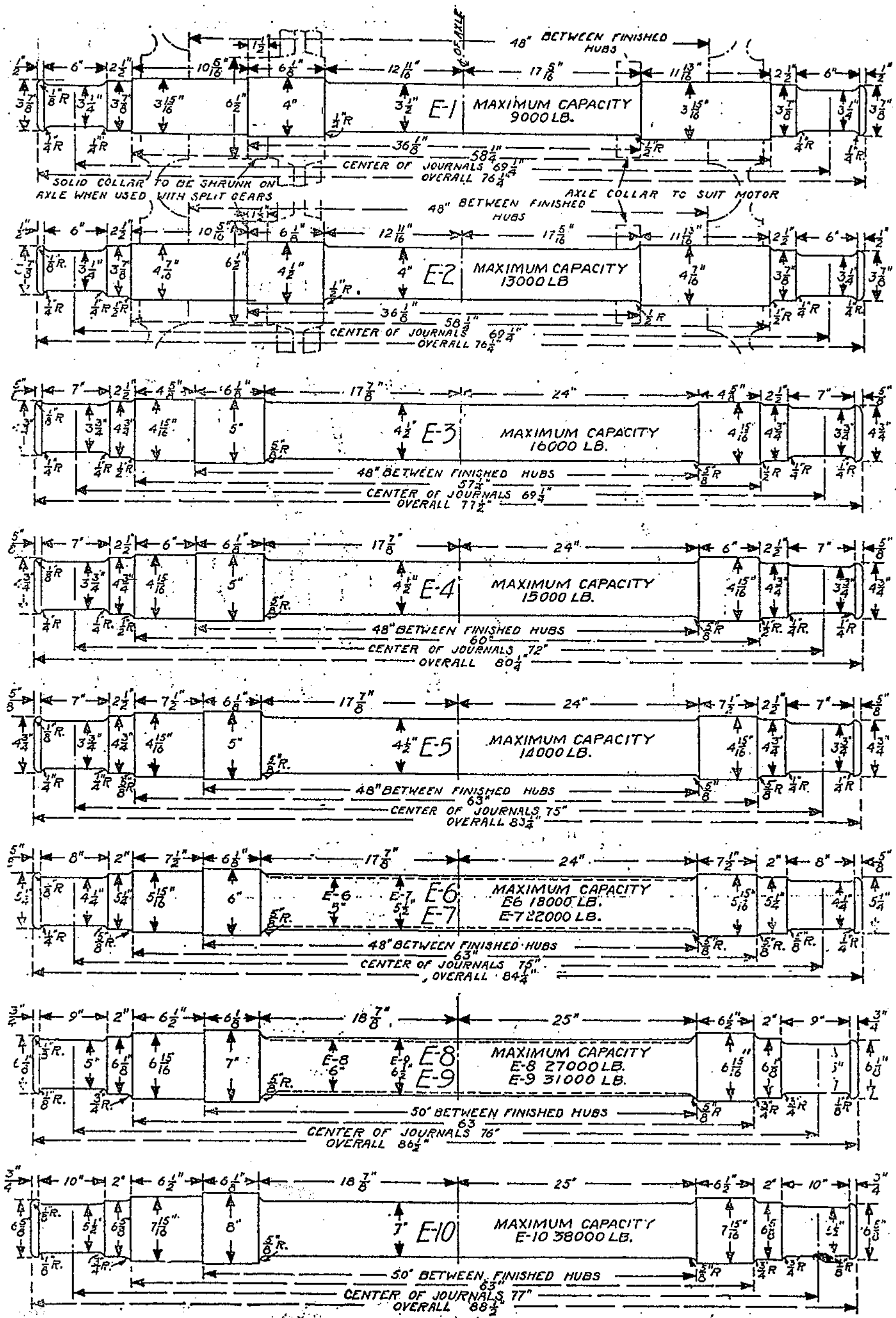
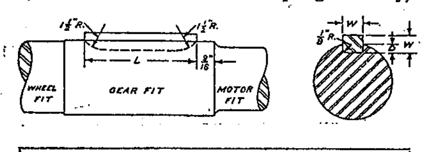


DIAGRAM SHOWING KEYS
(Recommended for use with split gears only)



AXLE	DIMEN. IN IN.		
	W	D	L
E-1, E-2	¾	⅜	4
E-3 to E-10 (inclusive)	1	½	5

W. CO. NON-COMMUTATING POLE RAILWAY MOTORS

† Type of Frame	Type No.	NOMINAL RATING H.P. at 500 Volts	Amp.	Speed	Complete Wt. in Lb. °	Max. Gear Reduction	33 IN. WHEEL CLEARANCE IN IN. Under Motor	Under Gear Cover	Minimum Distance Between Wheel Hubs	Max. Dia. in In. of Axle
B	3	25	50	345						
S	12A	25	47	525	2200	68/14	$4\frac{3}{4}$	$4\frac{3}{8}$	3 ft. $5\frac{3}{4}$ in.	4
S	12A	30	54	700	2200	68/14	$4\frac{3}{4}$	$4\frac{3}{8}$	3 ft. $5\frac{3}{4}$ in.	4
S	12A	30	54	550	2270	68/14	$4\frac{3}{4}$	$4\frac{3}{8}$	3 ft. $5\frac{3}{4}$ in.	4
S	38B	45	80	550	2390	68/14	$4\frac{1}{2}$	$4\frac{3}{8}$	3 ft. $7\frac{3}{8}$ in.	4
S	46	25			Foreign Motor Trade					
S	49	35	63	600	1920	68/14	$4\frac{3}{4}$	$4\frac{1}{8}$	3 ft. $2\frac{1}{2}$ in.	4
S	49B	30		410			Foreign Motor Trade			
S	50C	150	250	565		52/19	$2\frac{5}{16}$	5	4 ft. $2\frac{15}{16}$ in.	6
S	56	55	93	500	3000	68/14	$4\frac{7}{16}$	$4\frac{1}{8}$	3 ft. $10\frac{5}{16}$ in.	5
S	68	40	71	550	2280	68/14	$4\frac{13}{16}$	$4\frac{1}{8}$	3 ft. $4\frac{3}{8}$ in.	$4\frac{3}{4}$
S	69	30	54	550	1950	68/14	$5\frac{3}{8}$	$4\frac{1}{8}$	3 ft. $1\frac{7}{8}$ in.	$4\frac{1}{2}$
S	70	70				70/18				
S	76	75	128	500	3840	66/16	$2\frac{1}{2}$	$2\frac{1}{4}$	3 ft. $8\frac{1}{2}$ in.	6
S	81	55	96	535	Similar to W-56 Made only for Brooklyn R.T. Co.					
S	83	110	184	535	4830	56/20	$4\frac{3}{8}$	$4\frac{3}{8}$	4 ft. $3\frac{1}{16}$ in.	6
S	85	75	128	520	4500		$4\frac{1}{4}$	$3\frac{5}{8}$	4 ft. $2\frac{3}{4}$ in.	$6\frac{1}{2}$
S	86	175	290	650	6600	63/19	$3\frac{1}{8}$	3		$6\frac{1}{2}$
S	89	50								
B	91	100		Single-Phase						
S	92A	35	61	540	2265	69/15	$4\frac{5}{8}$	$4\frac{3}{16}$	3 ft. $7\frac{1}{32}$ in.	5
S	93A	60	105	510	3490	71/16	$3\frac{3}{8}$	$3\frac{5}{8}$	4 ft.	$5\frac{1}{2}$
S	101B	40	73	520	2730	70/14	$4\frac{1}{8}$	$4\frac{3}{16}$	3 ft. $7\frac{31}{32}$ in.	5
S	101D	55	96	660	2730	70/14	$4\frac{1}{8}$	$4\frac{3}{16}$	3 ft. $7\frac{31}{32}$ in.	5
S	101K	35	64	361	2830	79/14	$4\frac{1}{8}$	$2\frac{1}{2}$	3 ft. $3\frac{7}{8}$ in.	7
B	106	100	Single-Phase							
B	107	75	Single-Phase							
B	108	50	Single-Phase							
S	109	150		Modified 50 Type Motor for Met. W.S. Elev. R.R. Co., Chicago						
S	112	75	127	660	3490	73/16	$3\frac{3}{8}$	$3\frac{7}{16}$	4 ft.	6
DS	113	* 200	300	580	6554	64/19	‡ $4\frac{7}{8}$	‡ $4\frac{1}{16}$	4 ft. $2\frac{1}{8}$ in.	6
DS	114	* 160	250	638	5300	58/19	$3\frac{7}{16}$	$3\frac{3}{4}$	4 ft. $2\frac{11}{16}$ in.	6
DS	119	* 125	195	650	4600	58/17	$3\frac{15}{16}$	$3\frac{1}{4}$	4 ft. 2 in.	6
DS	121	* 90	140	650	4300	58/17	$5\frac{1}{4}$	$3\frac{3}{4}$	4 ft.	6
B	132A	100	Single-Phase		5400					
B	134	* 160	250	638			$4\frac{5}{16}$	$3\frac{3}{4}$	4 ft. $2\frac{1}{8}$ in.	6
B	135	75	Single-Phase							
B	148	150	Single-Phase							

° Weights complete do not include axle collar and axle linings.
* 550 volts.
‡ 36 in. wheels.
† S = Split frame.
B = Box frame.
DS = Diagonal split frame.

600-500 VOLT W. CO. COMMUTATING POLE RAILWAY MOTORS

Motor	Armature Turns	Nominal Rating at 600 Volts	At Nominal Rating 600 Volts		Nominal Rating at 500 Volts	At Nominal Rating 500 Volts		Type of Frame	*Wt. of Motor, Gear, Gear Case and Pinion	Max. Gear Reduction	Clearance in In. with 33 In. Wheels		Min. Distance in In. Between Wheel Hubs	Maximum Axle Diameter in In.	Bearing Dimensions in In. Armature Linings Pinion End		Commutator End	
			Amp.	RPM		Amp.	RPM				Under Gear Case	Under Motor			Dia.	Length	Dia.	Length
301 D-2	2	100	146	303				Box	5510	60/17	3 5/8	3 5/8	50		4 1/2	10	3 3/4	7 1/4
304	2	90	130	760	75	130	640	Split	3550	71/16	3 5/8	3 1/2	48	5 1/2	3 3/4	8 15/32	3 1/2	6 25/32
305	2	75	109	605	60	107	505	Split	3550	69/15	3 5/8	3 1/2	48	5 1/2	3 3/4	8 15/32	3 1/2	6 25/32
306 C	3	60	87	704	50	89	564	Box	2715	70/14	4	4 5/8	40 7/16	5	3 1/4	8 1/2	3 1/4	7 1/8
306 D	3	60	87	704	50	89	564	Box	2715	70/14	4	4 5/8	40 7/16	5	3 1/4	8 1/2	3 1/4	7 1/8
307	3	50	73	630	40	72	525	Split	2850	70/14	4	4 1/8	40 7/16	5	3 1/4	8 1/2	3 1/4	6 3/4
307 F	3	50	73	630	40	72	525	Split	2850	70/14	4	4 1/8	40 7/16	5	3 1/4	8 1/2	3 1/4	6 3/4
308 B-2	2	120	185	285				Box	6530	57/16	2 3/4	‡4 7/8	50	7	4 3/4	10	4	7
° 308 B-5		180	216	662				Box	6560	57/16	2 3/4	‡4 7/8	50	7	4 3/4	10	4	7
† 308 B-6		120	185	282				Box	6560	57/16	2 3/4	‡4 7/8	50	7	4 3/4	10	4	7
† 308 D-3		250	375	582				Box	6780	57/16	2 3/4	‡4 7/8	50	7	4 3/4	10	4	7
310 C	2	75	109	605		107	505	Box	3510	71/16	4 1/8	4 9/16	48	5 1/2	3 1/2	8 7/16	3 1/2	6 3/4
312					40	72.5	485			69/15								
316					60	105	535											
317		90	130	770	75	130	640	Box	3660	73/16	3 7/16	3 1/2	48	5 1/2				
317 A	2	90	130	775				Box	3660	73/16	3 7/16	3 1/2	48	5 1/2				
317 A	2	90	130	882				Box	3660	73/16	3 7/16	3 1/2	48	5 1/2				
§ 318		75	109	555		107	462	Box	3510	71/16	4 1/8	4 1/8	48	§				
321		90	132	540				Box	4150	61/16		3 3/8	50	6	4	8 1/2	3 3/4	7
323 A		40	55	703	33	58	566	Split	1890	81/16		4	37 1/32	4	2 3/4	6 3/8	2 1/2	5
328					30	54	772	Box	1700					4	2 3/4	6	2 3/8	4 1/2
333 A	2				100	175	616	Box	3900									
334 E-6	2	**95						Box	3870					6	4	8 1/2	3 1/2	6 3/4
337		53																
340					40	73	556	Box	2400	57/15	3°°	3°°		4 1/2	3 1/4	8 1/2	3 1/4	6 7/8

* Does not include axle linings and axle collars.
† 600/1200 volts.
‡ With 36 in. wheel.
° 750/1500 volts.
§ Same as 310 C except adapted for larger axle.
°° With 26-in. wheels. Armature center to bottom of motor, 10 1/8 in.
** 115 h.p. on 750 volts.

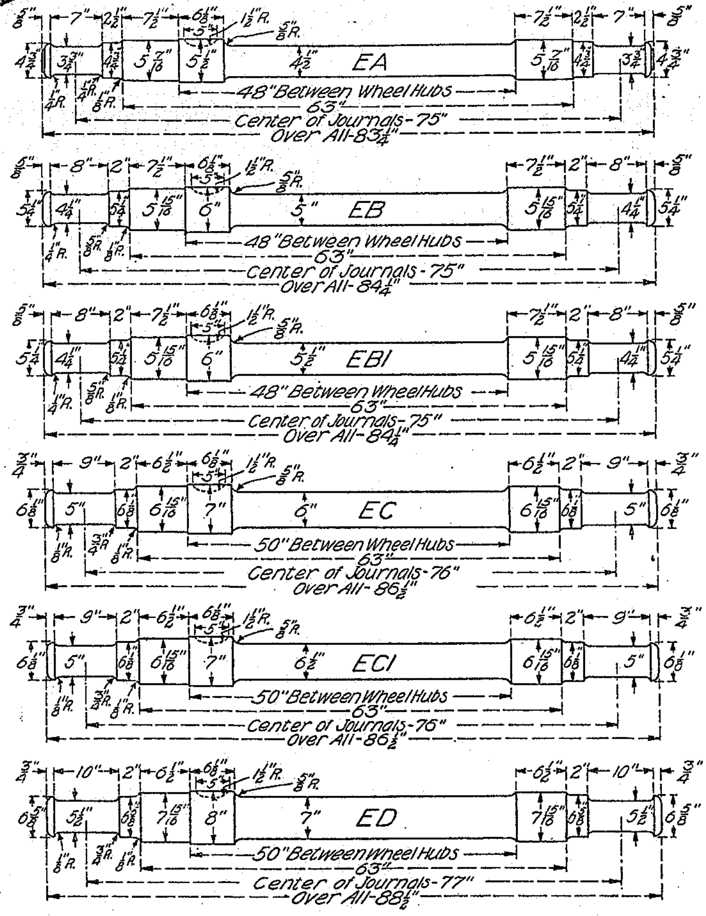
600-550 VOLT W. CO. COMMUTATING POLE RAILWAY MOTORS

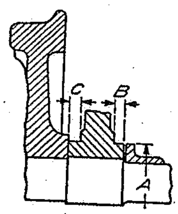
Motor	Armature Turns	Nominal Rating at 600 Volts	At Nominal Rating 600 Volts		Nominal Rating at 550 Volts	At Nominal Rating 550 Volts		Type of Frame	*Wt. of Motor, Gear, Gear Case and Pinion	Max. Gear Reduction	Clearance with 33 In. Wheels		Min. Distance in In. Between Wheel Hubs	Maximum Axle Diameter in In.	Bearing Dimensions in In. Armature Linings Pinion End		Commutator End	
			Amp.	RPM		Amp.	RPM				Under Gear Case	Under Motor			Dia.	Length	Dia.	Length
300 B	1	220	310	620	200	310	565	Box	6380	64/19	2 3/4	3 1/4	50 1/8	6 1/2	4 3/4	10	4	7
301 D		175	246	725	160	246	660	Box	5510	60/17	3 5/8	3 5/8	50	6 1/2	4 1/2	10	3 3/4	7 1/4
301 B	2	175	246	725	160	246	660	Box	5510	60/17	3 5/8	3 5/8	50	6 1/2	4 1/2	10	3 3/4	7 1/4
302	2	140	195	670	125	195	610	Box	4600	61/16	3 3/4	5 1/4	50	6	4	10	3 3/4	7
303 A	2	115	165	633	107	170	580	Box	4150	61/16	3 3/8	5 3/8	50	6	4	8 1/2	3 3/4	7

* Does not include axle linings and axle collars.

used prior to 1916

STANDARD AMERICAN ELECTRIC RAILWAY ASSOCIATION AXLES





Part	DIMEN. IN IN.			Max. Load on Axle	Horse Power	GEAR	
	A	B	C			Pitch	Face
EA	8	⅛	1	15000	Up to 45	3	5
EB	8	⅛	1	19000	45 to 65	3	5
EB1	8	⅛	1	22000	65 to 100	3	5
EC	9½	¾	⅛	27000	100 to 150	2½	5¼
EC1	9½	¾	⅛	31000	150 to 200	2½	5¼
ED	10½	¾	⅛	38000	200 to 250	2½	5¼

W. COMPANY'S RAILWAY MOTOR NOMENCLATURE

In connection with the various railway motors manufactured by the W. Company, the 300 series are usually non-ventilated and the 500 series ventilated, commutating pole motors.

The letter following the number usually denotes the mechanical characteristics. A change in this letter indicates a change in the mechanical design.

The number following the letter refers to the electrical construction and a change in this number indicates a change in electrical design.

If, for example a motor is rated 306-C-4, changing the letter "C" to "D" would indicate a change in mechanical design; changing the figure "4" to "5" would indicate a change in electrical design.

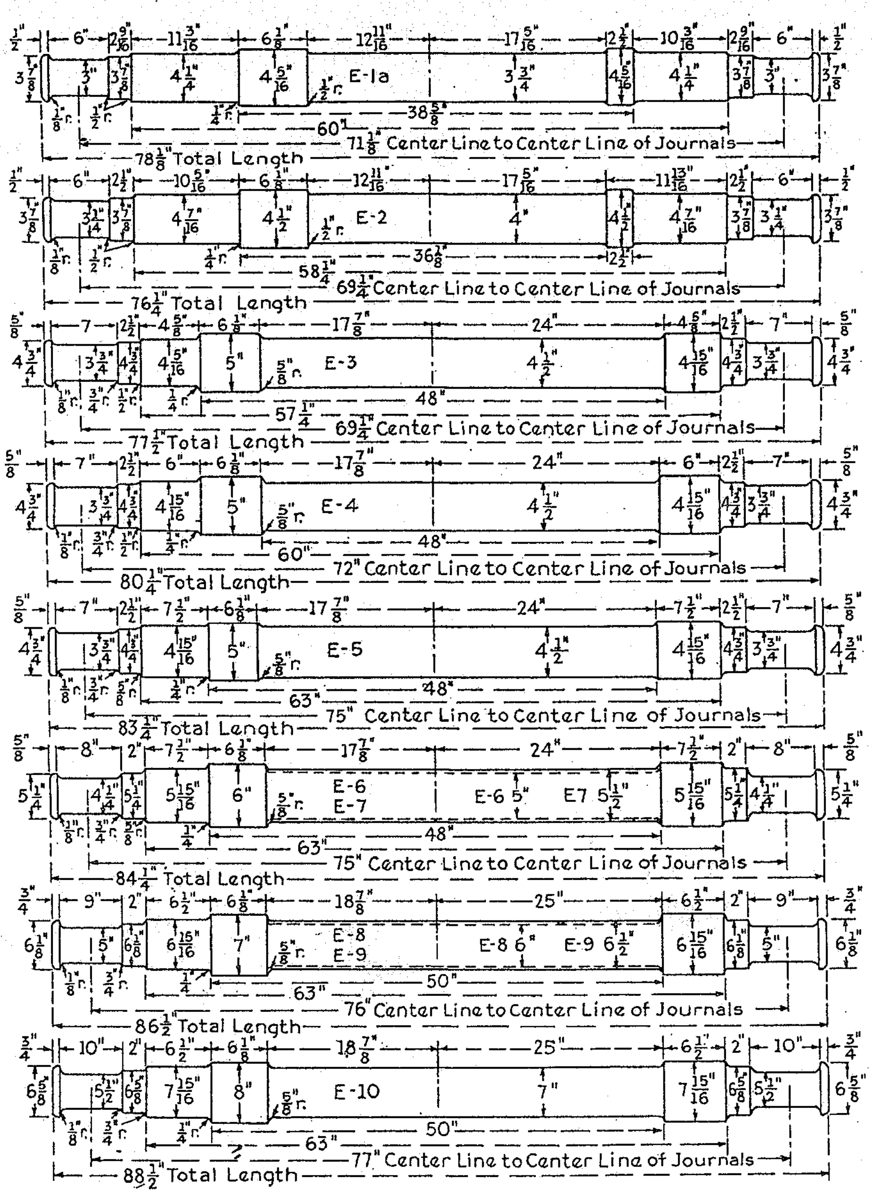
It is the practice of the W. Company to assign numbers to motors that have been merely designed and submitted on different propositions but not yet built. These motors are known by Delta numbers and are of the 1000 series; for example, Delta 1157 is a 90 hp. motor, 600 volts, and was originally submitted for an interurban proposition. These numbers are later superseded by a regular number.

The following symbols sometimes appear as a portion of the motor name:

X—indicates a motor with standard distance between gear and pinion center.

Y—indicates a motor with distance between gear and pinion center less than standard.

STANDARD AMERICAN ELECTRIC RAILWAY ASSOCIATION AXLES



Tolerances - The gear and motor bearing seats shall be of smooth finish and of the size specified with the following tolerances for machining

Gear Seat .... Plus 0.001" minus 0.000"
Motor Bearing Seat .... Plus 0.002" minus 0.002"

Type	DIMENSIONS IN INCHES												Maximum Capacity in Lb.
	Journal	Dia. Motor Fit	Dia. Gear Fit	Dia. Wheel Fit	Wheel Hub		Distance Between Hubs	Centers of Journals	Length of Gear Hub	Dia. of Gear Hub and Motor Bearing Flange	Extension of Gear Hubs		
					Length	Rough Dia.					Wheel Side	Motor Side	
E-1a	3x6	3¾	4 5/16	4¼	4½	7¼	48	71⅛	4⅝	6½	1	⅛	13000
E-2	3¼x6	4	4½	4 7/16	4½	7¼	48	69¼	4⅝	6½	1	⅛	13000
E-3	3¾x7	4½	5	4 15/16	5 or 5½	8¼	48	69¼	6⅛	7	1	⅛	16000
E-4	3¾x7	4½	5	4 15/16	5 or 5½	8¼	48	72	6⅛	7	1	⅛	15000
E-5	3¾x7	4½	5	4 15/16	5 or 5½	8¼	48	75	6⅛	7	1	⅛	14000
E-6	4¼x8	5	6	5 15/16	6	10¼	48	75	6⅛	8	1	⅛	18000
E-7	4¼x8	5½	6	5 15/16	6	10¼	48	75	6⅛	9	1	⅛	22000
E-8	5x9	6	7	6 15/16	6	10¼	50	76	6⅛	9½	⅛	1	27000
E-9	5x9	6½	7	6 15/16	6	10¼	50	76	6⅛	10	⅛	1	31000
E-10	5½x10	7	8	7 15/16	6	11¼	50	77	6⅛	10½	⅛	1	39000

ENGINEERING REPORT

on

Railway Motors

Part II

DESCRIPTIONS OF INDIVIDUAL MOTORS

G-E 600/1200- AND 750/1500-VOLT COMMUTATING POLE MOTORS

Class and Form of Motor	Armature Turns	Rated Hp.	At Rated Horsepower: Amps.	Volts	R.P.M.	Cont. Ampere Capacity for 65 Deg. C. Rise by Thermometer at Voltages Given Below: 300	450	600	Comm. Diam. in In.	Min. Depth of New Comm. Segment Above Mica Cone	No. of Brush-holders per Motor	No. of Brushes per Brush-holder	Type of Frame, Ventilation and Type of Fan	‡ Approx. Weig: Motor Alone	Moto Gear Gear C Axl Colla Pinic and A Lini
201-G	3	55	81	600/1200	655	38	37	35	11⅝	0.7208	2	2	BVSF	2465	294
205-E	2	80	118	600/1200	490	48	40	29	13¼	1.2177	2	2	BCRD	3275	385
205-E	2	100	144	600/1200	620	48	22	...	13¼	1.2177	2	2	BCRD	3440	400
207-D	2	110	161	600/1200	390	52	...	...	14¾	1.1840	2	2	BCRD	4500	520
207-D	3	65	96	600/1200	270	..	...	...	14¾	1.1840	2	2	BCRD	4500	520
ϕ 207-E	1	145	210	600/1200	620	68	...	...	14¾	1.1840	2	3	BCRD	4500	520
ϕ 207-E	2	110	161	600/1200	390	52	34	...	14¾	1.1840	2	3	BCRD	4500	520
207-G	2	110	161	600/1200	390	52	34	...	14¾	1.1059	2	2	BCRD	4740	544
212-G	1	190	261	600/1200	565	89	34	...	16⅛	1.1684	2	3	BCRD	5340	605
240-A	2	95	137	600/1200	662	78	81	84	13	0.9083	2	2	BVMF	3365	403
* 244-A	2	100	145	600/1200	700	77	80	80	15¼	0.86215	4	2	BVLSF	3285	392
ϕ 251-A	1	205	292	600/1200	543	89	34	...	16⅛	1.1684	2	3	BCRD	5340	610
254-A	2	135	195	600/1200	665	112	114	115	14½	0.8878	2	3	BVMF	3940	460
† 254-A	2	145	205	600/1200	760	126	130	135	14½	0.8878	2	3	BVMF	3940	46[illegible]
ϕ 257-A	1	145	210	600/1200	620	68	...	...	14¾	1.1218	2	3	BCRD	4440	51[illegible]
† 259-C	2	105	155	600/1200	890	96	99	101	13½	0.9874	2	3	BVMF	3400	40[illegible]
263-A	3	55	80	600/1200	660	56	60	63	11⅝	0.7208	2	2	BVMF	2550	311
281-A	3	72	102	600/1200	581	68	71	73	13⅞	0.8952	2	2	BVMF	2845	331
† 281-A	3	64	89	600/1200	705	66	70	74	13⅞	0.8952	2	2	BVMF	2845	331
						375	560	750							
* 269-C	3	50	58	750/1500	892	40	42	40	12½	0.8029	2	2	BVLSF	2045	24[illegible]
† 279-A	2	115	128	750/1500	1040	86	85	81	15¼	0.8622	4	2	BVSF	3687	40[illegible]
281-A	3	91	102	750/1500	741	70	73	75	13⅞	0.8952	2	2	BVMF	2845	331
† 281-A	3	80	89	750/1500	895	68	73	74	13⅞	0.8952	2	2	BVMF	2845	331
† 284-A	2	130	142	750/1500	810	97	98	101	14½	0.8878	2	3	BVMF	3940	46[illegible]
† 285-A	1	235	255	750/1500	840	200	205	207	16⅝	1.0746	2	3	BVMF	6480	72[illegible]
						600	900	1200							
205-E	3	80	59	1200	575	25	20	14	13¼	1.2177	2	2	BCRD	3200	37[illegible]

(Cont'd from above)

Class and Form of Motor	Max. Gear Reduction	Diametral Pitch	General Dimensions in Inches (Approx.): Distance from Center Line of Axle to: Bottom of Gear Case	Bottom of Motor	Top of Motor	¶ Swing-up-Dimension	Overall Length of Motor Along Armature Shaft	A.E.R.E.A. Standard Axle	Bearing Dimensions in Inches: Axle Linings: Max. Axle	Length	Armature Linings: Pinion End: Diam.	Length	Comm. End: Diam.	Length	Out: Drav Nun
201-G	71/15	3	12$\frac{11}{16}$	12¼	12¾	27⅜	48⅜	E-6	5	9	½	9	3¼	7¼	P-11
205-E	58/16	2½	12$\frac{11}{16}$	12¾	13	28¼	50	Spec.	6	10¾	3¾	10	3¼	7½	P-19
205-E	58/16	2½	12$\frac{11}{16}$	12¾	13	28¼	50	Spec.	6	10¾	3¾	10	3¼	7½	P-19
207-D	64/17	2½	13$\frac{11}{16}$	13$\frac{1}{16}$	15	30½	51⅛	Spec.	6½	11$\frac{9}{16}$	4¼	10	3¾	7$\frac{9}{16}$	P-19
207-D	64/17	2½	13$\frac{11}{16}$	13$\frac{1}{16}$	15	30½	51⅛	Spec.	6½	11$\frac{9}{16}$	4¼	10	3¾	7$\frac{9}{16}$	P-19
ϕ 207-E	64/17	2½	13$\frac{11}{16}$	13$\frac{1}{16}$	15	30½	51⅛	Spec.	6½	11$\frac{9}{16}$	4¼	10	3¾	7$\frac{9}{16}$	P-19
ϕ 207-E	64/17	2½	13$\frac{11}{16}$	13$\frac{1}{16}$	15	30½	51⅛	Spec.	6½	11$\frac{9}{16}$	4¼	10	3¾	7$\frac{9}{16}$	P-19
207-G	67/16	2½	14$\frac{5}{16}$	13$\frac{1}{16}$	21⅛	30¾	51⅛	Spec.	6½	37$\frac{11}{16}$	4¼	10	3¾	7$\frac{9}{16}$	T-36
212-G	65/18	2½	13$\frac{11}{16}$	13⅛	17$\frac{3}{16}$	32	51¾	Spec.	6½	12$\frac{5}{16}$	4¼	10	3¾	7$\frac{9}{16}$	P-89
240-A	59/16	2½	13	12$\frac{13}{16}$	13$\frac{5}{16}$	28¼	50$\frac{13}{16}$	E-8	6	10¾	3¾	10	3¼	7	P-11
* 244-A	67/17	2½	13⅛	13	16	32⅛	38⅛	Spec.	5½	10	3¾	9	3½	5¾	P-11
ϕ 251-A	69/17	2½	14$\frac{13}{16}$	12¾	17$\frac{1}{16}$	32$\frac{11}{16}$	51¾	Spec.	7	11½	4¼	10	3¾	7$\frac{9}{16}$	P-16
254-A	62/17	2½	13½	13¼	16	29½	51$\frac{11}{16}$	E-9	6½	10¾	3⅞	10	3⅝	6⅞	P-28
† 254-A	62/17	2½	13½	13¼	16	29½	51$\frac{11}{16}$	E-9	6½	10¾	3⅞	10	3⅝	6⅞	P-28
ϕ 257-A	64/17	2½	13$\frac{11}{16}$	13$\frac{1}{16}$	15¼	30½	51¼	Spec.	6½	11$\frac{9}{16}$	4¼	10	3¾	7$\frac{9}{16}$	P-16
† 259-C	61/16	2½	12⅞	12⅛	13¾	28½	50½	Spec.	6	11¼	3⅞	10	3½	6¾	P-28
263-A	71/15	3	12$\frac{11}{16}$	12¼	12¾	27⅞	48¾	E-6	5	9	3½	9	3¼	7$\frac{5}{16}$	P-18
281-A	71/15	3	12⅝	12	16¼	28¾	36⅛	Spec.	5½	10	3½	8	3	5½	P-28
† 281-A	71/15	3	12⅝	12	16¼	28¾	36⅛	Spec.	5½	10	3½	8	3	5½	P-28
* 269-C	69/15	3	12⅜	11⅜	13⅞	26⅞	36⅛	Spec.	5	8	3	7	2¾	5⅜	P-21
† 279-A	67/20	2½	14⅜	13$\frac{11}{32}$	16$\frac{11}{32}$	33	26⅛	Spec.	6	10	3¾	9	3½	5¾	T-21
281-A	71/15	3	12⅝	12	16¼	28¾	36⅛	Spec.	5½	10	3½	8	3	5½	P-28
† 281-A	71/15	3	12⅝	12	16¼	28¾	36⅛	Spec.	5½	10	3½	8	3	5½	P-28
† 284-A	62/17	2½	13½	13¼	16	29½	51$\frac{11}{16}$	E-9	6½	10¾	3⅞	10	3⅝	6⅞	P-26
† 285-A	60/21	2¼	14⅛	14	18	34	51$\frac{9}{16}$	Spec.	7	12	4¾	10$\frac{15}{16}$	3¾	7	P-26
205-E	58/16	2½	12$\frac{11}{16}$	12¾	13	28¼	50	Spec.	6	10¾	3¾	10	3¼	7½	P-19

* Narrow-gauge motor. † Ratings given on tapped field. ‡ Weights of gear and pinions are for maximum ratio and linings for maximun "Motor alone" is the motor without gear, gear case, pinion, axle linings, or axle collar. ¶ Swing-up dimension is distance from center of axle to fa point on suspension side of motor below support. ϕ Motors also used for locomotives with forced ventilation although ratings given are without ventilation. Dimension "Distance from center line of axle to bottom of gear case" is for maximum gear reduction.

B = Box. V = Ventilated. C = Closed. RD = Radial ducts in armature. SF = Series fan. LSF = Large series fan. MF = Multiple fan.

See page 103 for A.E.R.A. standard axle preparations.

NOTE.—No data on this page are to be used for guarantees or for construction purposes.

WESTINGHOUSE NON-COMMUTATING POLE RAILWAY MOTORS

* Type of Frame	Type No.	Nominal Rating			Complete Wt. in Lb. §	Max. Gear Reduction	33-in. Wheel Clearance in Inches		Minimum Distance Between Wheel Hubs	Maximum Diameter in In. of Axle
		Hp. at 500 Volts	Amps.	Speed			Under Motor	Under Gear Cover		
B	3	25	50	345	....	......	....	....	....	...
S	12A	25	47	525	2200	68/14	4¾	4 9/16	3 ft. 5¾ in.	4
S	12A	30	54	700	2200	68/14	4¾	4 9/16	3 ft. 5¾ in.	4
S	12A	30	56	545	2270	68/14	4¾	4 9/16	3 ft. 5¾ in.	4
S	38B	50	80	525	2400	68/14	4⅛	4⅜	3 ft. 7⅜ in.	4
S	¶46	25	...	...	....	......	....	....	....	...
S	49	35	63	600	1900	68/14	4¾	4⅛	3 ft. 2½ in.	4
S	¶49B	30	...	410	....	......	....	....	....	...
S	50C	150	250	580	5000	52/19	2 5/16	5	4 ft. 2 15/16 in.	6
S	56	55	93	500	3000	68/14	4 7/16	4⅛	3 ft. 10 5/16 in.	5
S	68	40	71	550	2280	68/14	4 13/16	4⅛	3 ft. 4⅜ in.	4¾
S	69	30	54	550	1950	68/14	5⅜	4⅛	3 ft. 1⅞ in.	4½
S	70	100	175	450	5285	70/18	....	....	....	...
S	76	75	128	500	3840	66/16	2½	2¼	3 ft. 8½ in.	6
S	△81	55	93	490	3125	......	....	....	....	...
S	83	110	184	535	4830	56/20	4⅜	4⅜	4 ft. 3 1/16 in.	6
S	85	75	128	520	4500	59/15	4¼	3⅝	4 ft. 2¾ in.	6½
S	86	175	290	650	6600	63/19	3⅛	3	....	6½
S	89	60	100	580	2700	66/20	3	4⅜	3 ft. 3½ in.	5
B	ϕ91	100	...	...	....	......	....	....	....	...
S	92A	35	65	500	2265	69/15	4⅝	4 3/16	3 ft. 7 1/32 in.	5
S	93A-2	60	105	510	3440	71/16	3⅜	3⅝	4 ft.	5½
S	101B-2	40	72	520	2780	69/15	4⅛	4	3 ft. 7 31/32 in.	5
S	101D-2	50	86	690	2780	69/15	4⅛	4	3 ft. 7 31/32 in.	5
S	101K	35	64	350	2830	79/14	4⅛	2½	3 ft. 3⅞ in.	7
B	ϕ106	100	...	...	....	......	....	....	....	...
B	ϕ107	75	...	...	....	......	....	....	....	...
B	ϕ108	50	...	...	....	......	....	....	....	...
S	π109	150	...	...	....	......	....	....	....	...
S	112	75	129	650	3485	73/16	3⅜	3 7/16	4 ft.	6
DS	113	† 200	300	580	6554	64/19	‡ 4⅞	‡ 4 1/16	4 ft. 2⅛ in.	6
DS	114	† 160	250	638	5300	58/19	3 7/16	3¾	4 ft. 2 11/16 in.	6
DS	119	† 125	195	650	4600	58/17	3 15/16	3¼	4 ft. 2 in.	6
DS	121	† 90	140	650	4300	58/17	5¼	3¾	4 ft.	6
B	ϕ132A	100	...	...	5400	......	....	....	....	...
B	134	† 160	250	638	....	......	4 5/16	3¾	4 ft. 2⅛ in.	6
B	ϕ135	75	...	...	....	......	....	....	....	...
B	ϕ148	150	...	...	....	......	....	....	....	...

* S = Split frame. B = Box frame. DS = Diagonal split frame. With exception of "Heavy Traction" types, all motors in above list are obsolete designs.
† 550 volts. ‡ 36-in. wheels.
§ Weights complete do not include axle collar and axle linings.
¶ Foreign motor trade.
△Similar to W-56 made only for Brooklyn R. T. Co.
ϕ Single-phase.
π Modified 50 type motor for Met. W. S. Elev. R. R. Co., Chicago.

COMPARISON OF G-E AND WESTINGHOUSE COMPANY'S STANDARD RAILWAY MOTORS

Type	Hp. at 600 Volts	At Rated Hp. 600 Volts		Complete Weight	Continuous Rating Based on 65 Deg. C. Rise			Max. Gear Ratio	Std. Wheel	Gearing Pitch	Max. Axle	A.E.R.A. Axle
		Amps.	R.P.M.		300 Volts	450 Volts	600 Volts					
W-508-E	25	35	1285	1035	31	34	35	74/13	24–26	4¼	4	E-2
GE-264-A	25	37	1236	1005	30	33	34	74/13	24–26	4¼	4	E-2
W-508-C	25	35	1285	1100	31	34	35	97/13	30–33	4¼	4½	E-2
GE-264-B	25	37	1236	1100	30	33	34	100/13	30–33	4¼	4¼	E-2
W-510-E	35	51	1085	1475	37	39	41	69/14	24–26	4	4¼	E-2
GE-265-A	35	51	1125	1415	37	39	41	69/14	24–26	4	4	E-2
W-510-C	35	51	1085	1590	37	39	41	89/14	30–33	4	5	E-6
GE-265-C	35	51	1125	1500	37	39	41	86/14	30–33	4	4½	E-5
W-514-A	40	58	767	1650	35	36	37	58/15	24–26	3½	4½	E-5
GE-247-A	40	60	715	1740	34	37	38	58/15	24–26	3½	*4	....
W-514-C	40	58	767	1770	35	36	37	76/15	30–33	3½	5	E-6
GE-247-D	40	60	715	1870	34	37	38	63/15	30–33	3	4½	E-5
W-532-A	50	72	670	2300	44	45	46	57/15	26–28	3	4½	E-5
W-532-B	50	72	670	2325	44	45	46	69/15	30–33	3	5	E-6
GE-203-P	50	74	760	2280	42	44	46	69/15	30–33	3	5	E-6
W-535-A	60	85	797	2400	52	54	58	57/15	26–28	3	5	E-6
GE-275-J	60	87	720	2410	52	54	58	57/15	26–28	3	5	E-6
W-535-B	60	85	797	2475	52	54	58	69/15	33	3	5½	E-7
GE-275-L	60	87	720	2600	52	54	58	71/15	33	3	5	E-6
W-306-CV-4	65	92	697	2700	58	60	60	69/15	30–33	3	5	E-6
GE-263-A	65	94	725	3050	63	67	69	71/15	30–33	3	5½	E-6

* These motors may be rebored for 4½-in. axle.

WESTINGHOUSE COMMUTATING POLE RAILWAY MOTORS

Motor	Arm. Turns	Nominal Rating at 600 Volts	At Nominal Rating 600 Volts		Nominal Rating at 500 Volts	At Nominal Rating 500 Volts		Type of Frame	Wt. in Lb. of Motor Gear-pin and Case	Max. Gear Re-duc-tion	Clearance with 33-in. Wheel Under		Min. Distance in In. Between Wheel Hubs	Max. Axle Dia. in In.	Armature Bea… Dimensions in I…		
			Amps.	R.P.M.		Amps.	R.P.M.				Gear Case	Motor			Pinion End Diam.	Pinion End Length	Com… Dian…
300-B	1	220	310	610	200	310	565	Box	*6380	64/19	2¾	3¼	50⅛	6½	4¾	10	4
301-B	2	175	246	725	160	246	660	Box	*5510	60/17	3⅝	3⅝	50	6½	4½	10	3¾
301-D	...	175	246	670	160	246	660	Box	*5510	60/17	3⅝	3⅝	50	6½	4½	10	3¾
301-D2	2	100	150	300	...	...	...	Box	*5510	60/17	3⅝	3⅝	50	...	4½	10	3¾
302	2	140	200	665	125	195	610	Box	*4685	61/16	3¾	5¼	50	6	4	10	3¾
303-A	2	115	165	640	107	170	580	Box	*4150	61/16	3⅝	5⅞	50	6	4	8½	3½
304	2	90	130	760	75	130	640	Split	*3550	71/16	3⅝	3½	48	5½	3¾	8 15/32	3½
305	2	75	109	595	60	109	505	Split	*3550	71/16	3⅝	3½	48	5½	3¾	8 11/16	3½
306-CV-4	3	65	92	697	50	89	564	Box	*2700	69/15	4	4⅝	40 7/16	5	3¼	8½	3¼
306-D	3	60	88	695	50	89	564	Box	*2630	69/15	4	4⅝	40 7/16	5	3¼	8½	3¼
307	3	50	73	615	40	72	525	Split	*2700	69/15	4	4⅛	40 7/16	5	3¼	8½	3¼
308-B2	...	120	185	285	...	...	...	Box	*6740	57/16	2¾	‡4⅞	50	7	4¾	10	4
§ 308-B5	1	180	216	662	...	...	...	Box	*6740	57/16	2¾	‡4⅞	50	7	4¾	10	4
† 308-B6	...	120	185	285	...	...	...	Box	*6740	57/16	2¾	‡4⅞	50	7	4¾	10	4
308-D	...	220	310	595	...	...	...	Box	*6740	57/16	2¾	‡4⅞	50	7	4¾	10	4
† 308-D3	...	220	310	595	...	...	...	Box	*6740	57/16	2¾	‡4⅞	50	7	4¾	10	4
308-D5	...	120	175	335	...	...	...	Box	*6740	57/16	...	...	...	...	...	...	...
310-C	2	75	109	595	60	107	505	Box	*3510	71/16	3⅝	4 13/16	48	5½	3¾	8 7/16	3½
312	...	50	72	657	40	72.5	485	...	*2630	69/15	...	...	...	...	...	...	...
316	...	75	109	580	60	105	535	...	*3050	...	...	...	...	...	...	...	...
317	2	90	130	740	75	130	630	Box	*3660	73/16	3 7/16	3½	48	5½	3¾	8 7/16	3½
317-A	2	90	130	770	75	130	640	Box	*3660	73/16	3 7/16	3½	48	5½	3¾	8 7/16	3½
317-A2	2	90	130	885	75	130	730	Box	*3660	73/16	3 7/16	3½	48	5½	3¾	8 7/16	3½
¶ 318	2	75	109	595	60	107	505	Box	*3660	73/16	3 7/16	3½	48	¶	3¾	8 7/16	3½
319-B	...	50	73	666	...	...	...	Split	*2865	69/17	...	...	...	...	3¼	...	3
† 321	...	90	132	530	...	...	...	Box	*4150	61/16	...	3⅜	50	6	4	8½	3⅜
323-V	...	40	60	660	33	...	...	Split	*1990	81/16	4	5⅛	...	4½	3	7	2½
323-A	4	40	59	690	33	58	565	Split	*1890	81/16	4	4¾	37 1/32	4	2¾	6⅜	2½
† 324	...	75	109	560	60	107	475	Box	*3660	73/16	3 7/16	3½	48	5½	3¾	8 7/16	3½
328	3	35	52	740	30	...	...	Box	*1680	58/15	π3	π3¼	...	4	2¾	6	2¾
333-V	...	125	180	735	...	...	...	Box	*3850	61/16	3½	4 13/16	...	6	4	8½	3
333-A	2	115	165	...	100	175	616	Box	*3900	...	...	...	...	...	...	...	
334-E6	2	** 90	130	590	...	...	...	Box	*3870	...	...	...	...	6	4	8½	3
337-C	...	50	73	...	...	...	...	Box	...	62/14	5⅜	5⅞	48	4½	...	...	
340	...	48	72	660	40	72	553	Box	*2400	57/15	φ3	φ3	...	4½	3¼	8½	3
505-X	...	19	29	825	...	...	...	Box	890	68/14	π2⅞	π3¾	...	3½	2½	4½	2
506-A2	4	25	35	1285	...	...	...	Box	900	68/14	π3½	π3 11/16	48	4	2½	4¾	2
508-A	3	25	35	1285	...	...	...	Box	1100	74/13	π2¾	π3 11/16	48	4	2½	6¾	2
508-C	...	25	35	1285	...	...	...	...	1100	97/13	...	...	...	4½	...	...	
508-E	...	25	35	1285	...	...	...	...	1035	74/13	...	...	...	4	...	...	
510-A	...	35	52	1070	...	...	...	...	1475	69/13	π2½	π2¾	...	4¼	2¾	6½	2
510-C	...	35	51	1085	...	...	...	...	1590	89/14	...	...	...	5	...	...	
510-E	...	35	51	1085	...	...	...	...	1475	69/14	...	...	...	4¼	...	...	
512-B	...	35	52	780	...	...	...	Box	1575	66/15	...	...	...	5	2¾	6	2
514-A	...	40	57	767	...	...	...	...	1650	58/15	...	...	...	4½	...	...	
514-C	...	40	58	760	...	...	...	Box	1770	76/15	4⅞	7 3/16	...	5	2¾	6	2
515-A	...	40	59	720	...	...	...	...	1650	58/15	π3⅛	π3⅛	...	...	2¾	6	2
526-L	...	50	74	900	...	...	...	...	2180	66/14	...	...	...	4½	3	7	2
532-A	3	50	74	670	40	...	...	Box	2300	57/15	△2 13/16	△2 13/16	...	5	3¼	8½	3
532-B	...	50	72	670	...	...	...	...	2325	69/15	...	...	...	5	...	...	
533-T4	3	53	80	590	40	...	480	Box	2545	...	...	...	...	...	...	...	
534-Y1	3	58	85	735	50	...	435	...	2690	69/15	...	...	...	5	3¼	8½	3
544-J	...	50	74	586	...	...	...	...	2575	72/15	...	...	...	5½	3¾	8½	
547-A	...	80	118	650	...	...	...	...	3070	71/16	...	...	...				
548-C8	2	100	145	810	...	...	...	Box	3175	73/16	3½	3½	...	6	3¾	8½	
557-A5	1	140	202	900	...	...	...	Box	4050	61/16	3½	4 9/16	50	6	4	8½	
577-A1	...	200	280	670	...	...	...	Box	5650	67/16	...	...	...	6½	4½	10	
632-B	...	50	74	665	40	...	...	Split	2375	69/15	4	5 9/16	...	5	3¼	8½	
535-A	3	60	85	797	...	...	...	Box	2400	57/15	△2 13/16	△3	...	5½	3¼	7¾	
535-B	...	60	87	797	...	...	...	...	2475	69/15	...	...	...	...	...	...	
567-A6	1	165	234	640	...	...	...	Box	4900	61/16	3½	4 9/16	...	...	4¼	10	

* Does not include axle linings and axle collars.
† 600/1200 volts. ‡ With 36-in. wheels. § 750/1500 volts.
¶ Same as 310-C except adapted for larger axle.
φ With 26-in. wheels. Armature center to bottom of motor 10⅞ in.
** With 115 hp. on 750 volts. π With 24-in. wheels.
△ Clearance with 26-in. wheels.

G-E 600-VOLT COMMUTATING POLE MOTORS (INACTIVE LIST)

Class and Form of Motor	Armature Turns	Rated Hp. at 600 Volts	At Rated Hp. 600 Volts: Amps.	At Rated Hp. 600 Volts: R.P.M.	Cont. Ampere Capacity for 65 Deg. C. Rise by Thermometer at Voltages Given Below: 300	450	600	Rated Hp. at 500 Volts	At Rated Hp. 500 Volts: Amps.	At Rated Hp. 500 Volts: R.P.M.	Comm. Diam. in In.	Min. Depth of New Comm. Segment Above Mica Cone	No. of Brush-holders per Motor	No. of Brushes per Brush-holder	Type of Frame, Ventilation and Type of Fan	‡ Approx. Wt.: Motor Alone	‡ Approx. Wt.: Motor Gears, Gear Case, Axle Collar, Pinion and Axle Lining
200-J	4	40	59.0	745	32	34	34	33	60.0	605	10½	0.7339	2	2	BVSF	1720	2120
200-K	4	40	59.0	745	32	33	33	33	60.0	605	10½	0.7339	2	2	SODVSF	1800	2200
201-G	3	65	95.0	710	46	45	44	55	99.0	570	11⅝	0.7069	2	2	BVSF	2370	2850
201-I	3	65	95.0	710	43	41	39	55	99.0	570	11⅝	0.7069	2	2	SODVSF	2465	2940
205-E	2	110	159.0	625	55	46	33	..	...	...	13¼	1.2177	2	2	BC	3440	4000
φ 207-E	1	165	238.0	605	78	..	..	..	...	...	14¾	1.0923	2	2	BCRD	4500	5200
212-B	1	235	333.0	620	119	96	70	..	...	...	16⅛	1.1400	2	4	BCRD	5340	6050
* 218-C	3	70	104.0	550	42	30	..	55	97.0	473	13¼	1.2150	2	1	BCRD	2730	3210
† 248-A	1	160	227.0	659	132	138	143	135	233.0	534	16	1.1026	2	3	BVMF	5050	5720
φ 251-A	1	235	333.0	620	119	96	70	..	...	...	16⅛	1.1684	2	3	BCRD	5340	6100
φ 257-A	1	165	238.0	605	78	..	..	..	...	...	14¾	1.0923	2	3	BCRD	4440	5150
265-C	4	35	51.0	1125	37	39	41	30	53.5	900	9¼	0.7486	2	1	BVMF	1280	1570
272-A	3	80	118.0	365	61	64	66	65	116.0	298	14½	0.8780	2	2	BVMF	4040	4740
* 273-B	4	35	52.0	1125	36	39	40	30	54.0	900	9¼	0.7486	2	1	BVMF	1190	1400
288-A	3	37	53.0	757	34	36	37	..	...	...	9¼	0.7076	2	1	BVMF	1525	1820
288-B	3	37	53.0	757	34	36	37	..	...	...	9¼	0.7076	2	1	BVMF	1570	1890
* 293-A	4	40	58.0	680	35	37	38	33	58.0	545	12	0.7964	2	1	SODVLSF	1980	2390
294-A	3	37	53.0	757	34	36	37	..	...	...	9¼	0.7076	2	2	BVMF	1730	2080
† 295	3	60	87.0	910	66	68	72	50	88.0	755	14½	0.7568	4	1	BVMF	2320	2760
† 296	1	200	271.0	629	179	181	183	167	274.0	516	16	1.1026	2	3	BVMF	5550	6240
299	3	60	84.0	750	52	54	58	50	85.0	625	10⅝	0.7760	2	1	BVMF	2080	2490

(Cont'd from above)

Class and Form of Motor	Max. Gear Reduction	Diametral Pitch	General Dimensions in Inches (Approx.) — Distance from Center Line of Axle to: Bottom of Gear Case	Bottom of Motor	Top of Motor	¶ Swing-up Dimension	Overall Length of Motor Along Armature Shaft	A.E.R.E.A. Standard Axle	Bearing Dimensions in Inches — Axle Linings: Max. Axle	Axle Linings: Length	Armature Linings, Pinion End: Diam.	Pinion End: Length	Comm. End: Diam.	Comm. End: Length	Outline Drawing Number
200-J	67/14	3	12	10⅝	13⅛	25⅝	41⅝	E-3	4½	8¼	3	8	2¾	6½	P-1604071
200-K	67/14	3	12	10⅝	13⅜	....	41⅝	E-3	4½	8¼	3	8	2¾	6½	P-1604061
201-G	71/15	3	12 11/16	12¼	12¾	27⅜	48⅜	E-6	5	9	3½	9	3¼	7¼	P-1151246
201-I	71/15	3	12¾	11¼	14	....	47½	E-6	5	9	3½	9	3¼	7¼	P-1155345
205-E	58/16	2½	12½	12 11/16	13	28¼	50	Spec.	6	10¾	3¾	10	3¼	7½	P-194422
φ 207-E	64/17	2½	13 11/16	13 1/16	15	30½	51⅛	Spec.	6½	11 9/16	4¼	10	3¾	7 9/16	P-194429
212-B	64/19	2½	13⅝	13⅛	17 3/16	32	51¾	Spec.	6½	11 9/16	4¼	10	3¾	7 9/16	P-1158479
* 218-C	71/16	3	12¾	13	13⅛	32½	40¼	Spec.	5	9	3½	7¾	3¼	5¾	P-1117159
† 248-A	66/17	2½	14	13	16¼	32	51 11/16	E-9	6½	12 5/16	4¼	10	3¾	7 1/16	P-1158475
φ 251-A	69/17	2½	14 13/16	12¾	16½	32¾	51¾	Spec.	7	11½	4¼	10	3¾	7 9/16	P-1608640
φ 257-A	64/17	2½	13 11/16	13 1/16	15¼	30½	51¼	Spec.	6½	11 9/16	4¼	10	3¾	7 9/16	P-1608672
265-C	86/14	4	11 7/16	9	10⅛	22 5/16	37 1/16	E-3	4¾	7½	2⅝	6 15/16	2⅜	5¼	P-2809813
272-A	64/15	2½	13 11/16	13¼	16	29⅝	51 13/16	E-9	6½	10¾	3⅞	10	3⅝	6⅞	P-3643540
* 273-B	69/14	4	9 5/16	9	10⅛	20⅛	36⅜	Spec.	4½	7	2⅝	6 15/16	2⅜	4¾	P-2740134
288-A	59/14	3½	9⅛	9 11/16	11 1/16	20⅜	40 3/16	Spec.	4	7	Anti-friction				P-2740447
288-B	76/14	3½	11⅜	10 7/16	9 5/16	23¼	40 9/16	E-3	4½	7½	Anti-friction				P-2740448
* 293-A	83/15	3½	12⅝	11⅜	14	....	35 15/16	Spec.	4½	7¾	Anti-friction				P-2740117
294-A	76/15	3½	11⅜	10 7/16	9 5/16	23¼	46¼	E-3	4½	7½	Anti-friction				P-2740094
† 295	71/15	3	12⅝	11 23/32	13 9/32	27⅛	32	Spec.	5½	9¼	3¼	8	3	6	T-755510
† 296	62/21	2½	13 3/16	13	16¼	32 1/16	51 5/16	E-10	7	12 7/16	Anti-friction				P-2740248
299	58/14	3	10 3/16	10 1/16	11⅞	24¼	44⅛	E-6	4½	8¾	Anti-friction				P-2740385

* Narrow-gauge motor. † Ratings given on tapped field. ‡ Weights of gears and pinions are for maximum ratio and linings for maximum axle. "Motor alone" is the motor without gear, gear case, pinion, axle linings, or axle collar. ¶ Swing-up dimension is distance from center of axle to farthest point on suspension side of motor below support. φ Motors also used for locomotives with forced ventilation although ratings given are without forced ventilation. Dimension "Distance from center line of axle to bottom of gear case" is for maximum gear reduction.

B = Box. SOD = Split opening down. V = Ventilated. C = Closed. SF = Series fan. LSF = Large series fan. MF = Multiple fan. RD = Radial ducts in armature. See page 103 for A.E.R.A. standard axle preparations.

NOTE.—No data on this page are to be used for guarantees or for construction purposes.

Improved Brush-holders

For GE-70 and GE-80 Railway Motors

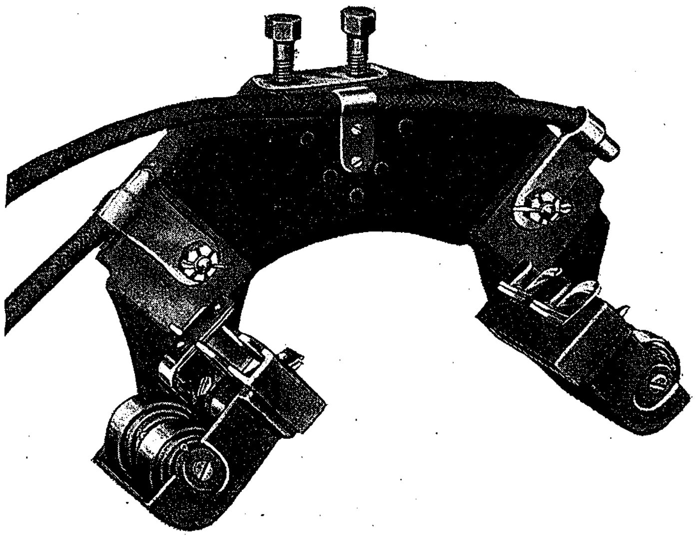
Modern brush-holders are generally equipped with renewable carbon-way boxes, an improvement which cannot be overestimated, as these boxes can be replaced at very little expense, thus eliminating the necessity of discarding the complete brush-holder when the brush ways become too badly worn for continued use. Furthermore, service tests show a marked improvement in commutation and in the life of brushes when brush-holders are fitted with flat-steel clock-spring pressure arms and recessed adjusting sleeves, as the extreme flexibility of the spring insures practically uniform pressure under all conditions of brush wear.

As brush-holders equipped with these modern features, interchangeable with the older types, are available for the GE-70 and GE-80 motors in service, it is recommended that they be specified for replacements, as the substitution will mean more efficient and economical operation due to reduced maintenance costs.

The following table itemizes the catalog numbers of older brush-holders complete which are interchangeable with the steel clock-spring pressure-arm type with recessed adjusting sleeves and renewable carbon-way boxes.

Old Brush-holders Cat. Nos.	Characteristics	Side Used on	Interchangeable Modern Brush-holders Cat. Nos.
34060	Lever and spiral-spring type	Axle	5X410
34061	Lever and spiral-spring type	Susp.	5X409
* 222233	Phosphor-bronze clock-spring type and renewable carbon-way boxes.	Axle	5X410
* 222234	Phosphor-bronze clock-spring type and renewable carbon-way boxes.	Susp.	5X409

* These brush-holders can be modernized by substituting steel spring-pressure arm Cat. No. 2840372G1 and recessed adjusting sleeve Cat. No. 2462153 for phosphor-bronze spring-pressure arm Cat. No. 222235 and adjusting sleeve Cat. No. 1416842. It is recommended therefore that this change be made at the first opportunity as spring steel is greatly superior to phosphor bronze, giving more uniform brush pressure and will not set or lose its tension.



Modern Brush-holders Equipped with Renewable Carbon-way Boxes and Steel Clock-spring Pressure Arms

9120