INVESTIGATION AND DESIGN OF AN AUTOMOBILE TRANSMISSION AXLE

BY

WALTER JOHN HUGHES

THESIS FOR THE DEGREE OF BACHELOR OF SCIENCE

IN MECHANICAL ENGINEERING

IN THE

COLLEGE OF ENGINEERING

OF THE

UNIVERSITY OF ILLINOIS

JUNE, 1910

UNIVERSITY OF ILLINOIS

May 31 1910

THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

Walter John Hughes

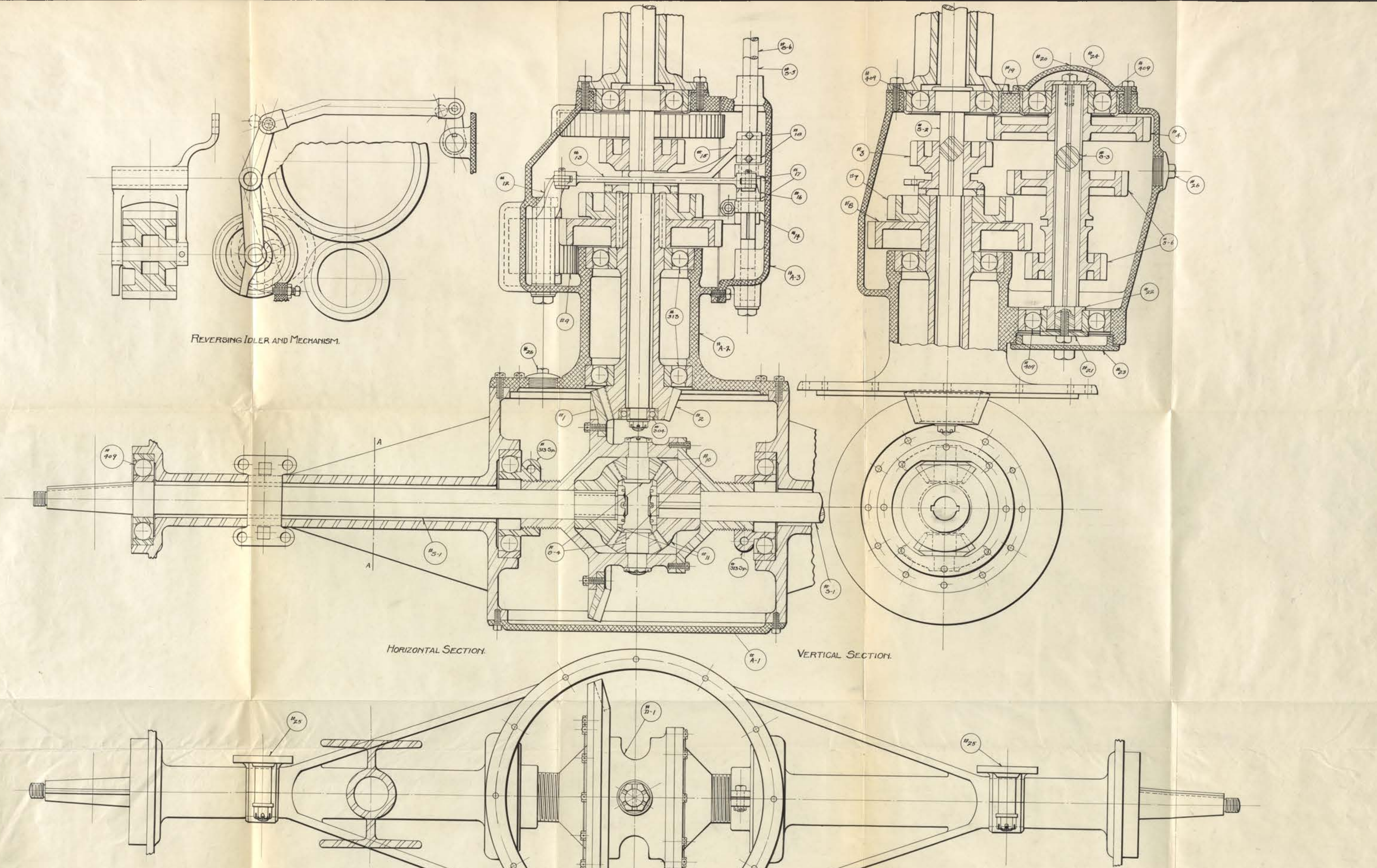
ENTITLED Investigation and Design of an Automobile Transission Axle.

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE DEGREE OF Bachelor of Science in Mechanical Engineering

C. A. Leutwiler
Instructor in Charge

APPROVED: G. A. Goodenough

HEAD OF DEPARTMENT OF Mechanical Engineering



MECH. ENG. DEPARTMENT
UNIV. OF ILL.
THESIS.
DESIGN OF AN AUTOMOBILE TRANSMISSION AXLE
Scale : Half Size
May, 1910. W. J. Hughes.
No. I.

Design of an Automobile Transmission Axle.

- Specifications -

Weight of Car in pounds .... 2500.
Tread in inches .... 56.
Diameter of Wheels in inches .... 36.
Horse Power of Motor 40
Revolutions per minute of Motor at 40 Horse Power .... 1680.

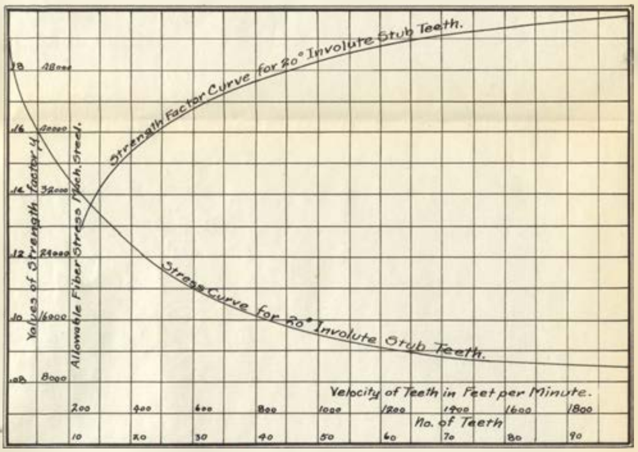
Design of Gearing

Let F = Force in pounds on a gear tooth.
" D = Diameter of gear in inches.
" H = Horse Power = 40.
" R = Revolutions per minute of Motor = 1680.

Then $H = \frac{F \times \pi \times D \times R}{12 \times 33000}$

or $F \times D = \frac{40 \times 12 \times 33000}{\pi \times 1680} = 3000$ in. lb. (1)

Since the bevel drive gears will be required to transmit 40 horse power at both high and low speeds, they will be investigated for strength under each condition. The Fellows "standard stub teeth" will be used throughout. Stresses and the factors y, for use in Lewis' gear formula, will be taken from the following chart. These factors, as given on this chart, are only approximately correct.



Since the gears and pinions are made of the same material, the maximum load transmitted by any pair will be determined by the pinion, hence gear calculations will not appear in what follows.

Calculations for Bevel Pinion No. 2 Running at High Speed

Substituting the proper values in equation (1), we have
$F = 750$ lb.

Now assuming the pinion has 16T 4P, we get $p' = .786$, $y = .143$, $n = .70$, $f = 1.75$." The velocity of the gear teeth at high speed is 1760 ft. per min., hence the allowable stress, from the above chart is 11000.

Substituting the above values in Lewis' formula for bevel gearing ($F' = Sp'fyn$), we have
$F' = 1500$ lb.

Calculations for Bevel Pinion No. 2, Running at Low Speed.

Substituting the proper values in equation (1), we have
$F = 3000$ lb.

Since the pinion has 16T 4P, we have, as before, $p' = .786$, $y = .143$, $n = .70$, $f = 1.75$." The velocity of the gear teeth at low speed is 440 ft. per min., hence the allowable stress from the preceding chart is 24000. Substituting the above values in Lewis' formula for bevel gearing ($F' = Sp'fyn$), we have
$F' = 3300$ lb.

Calculations for Pinion No. 3.

Substituting the proper values in equation (1), we have
$F = 750$ lb.

Now assuming the pinion has 24T 6P, we get $p' = .523$," $f = 1.25$" $y = .16$. The velocity of the teeth is 1760 ft. per min., hence the allowable stress by the preceding chart is 11000. Substituting the above values in Lewis' formula for spur gearing, we have from ($F' = Sp'fy$)
$F' = 1150$ lb.

Calculations for Sliding Combination, No. 5-6.

The Sliding Combination is made up of two pinions (see plate VI). Assuming that the smaller pinion has 24T 6P, we have $p' = .523$," $y = .16$, $f = 1.375$," and substituting the proper values in equation (1) we get
$F = 1500$ lb.

Substituting the above values in Lewis' formula, after obtaining the allowable stress, 16000, from the preceding chart, knowing the velocity of the teeth to be 880 ft. per min., we then have $F' = 1850$ lb.

Substituting the proper values in equation (1) we have, for the larger pinion of the Sliding Combination,
$F = 1000$ lb.

Assuming that the larger pinion has 36T 6P, we get $p' = .523$," $y = .173$, $f = 1.25$". The velocity of the teeth is 1320 ft. per min., hence the allowabe stress from the preceding chart is 12000. Substituting the above values in Lewis' formula for spur gearing ($F' = Sp'fy$), we have
$F' = 1360$ lb.

Calculations for Bevel Differential Pinion No. 11.

Since there are two bevel differential pinions each taking half the load coming on gear No. 1, and since there are two teeth in contact on each (see plate I), the actual load, coming on one tooth, is therefore equal to one fourth the maximum load on gear No. 1 X the ratio of the diameters of gears Nos. 1 and 10. From the above, $F = 2000$ lb.

Assuming the pinion has 12T 4P, we have $p' = .786$, $y = .13$, $n = .70$, $f = 1$." Now equating F' to F and substituting the above values in Lewis' formula for bevel gearing ($F' = Sp'fyn$) we get
$S = 28000$

By the preceding chart, the tooth velocity, corresponding to the above stress, is 340 ft. per min., a speed seldom, if ever, attained by differential gear teeth.

Design of Shafting.

Calculations for Direct Drive Shaft No. 5-2.

The bending moment at A (Fig. 1.) is caused by the force of 750 lb. acting on the teeth of pinion No. 3 (see calculations for pinion No. 3 above). The bending moment at A is therefore equal to $\frac{750 \times 15.625}{17} \times 1.375 = 950$ in. lb.

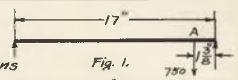


Fig. 1.

The twisting moment $= 750 \times 2 = 1500$ in. lb. Combining the bending and twisting moments, we have, by Guest's law $M_e = \sqrt{950^2 + 1500^2} = 1780$ in. lb., from which $\frac{M_e}{S} = .081$, which corresponds to a shaft 1 inch in diameter. The direct drive shaft is a $1\frac{1}{4}$" squared shaft.

Calculations for Sliding Gear Shaft No. 5-3.

The bending moment at A (Fig. 2.) is caused by the 1500 lb. force which acts on the teeth of the smaller pinion of the Sliding Combination No. 5-6. This moment is equal to $\frac{1500 \times 7}{11.25} \times 4\frac{1}{4} = 3960$ in. lb.

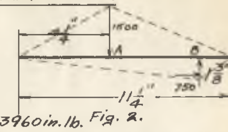


Fig. 2.

The bending moment at B (Fig. 2) is caused by 750 lb. force which acts on the teeth of End Bearing Gear No. 4. This moment at B is therefore $= \frac{750 \times 9.875}{11.25} \times 1.375 = 905$ in. lb. Combining, we have the maximum bending moment at A (Fig. 2) is equal to 3500 in. lb. The twisting moment $= 750 \times 4 = 3000$ in. lb. Combining the maximum bending and the twisting moments we have by Guest's law $M_e = \sqrt{3000^2 + 3500^2} = 4300$ in. lb., from which $\frac{M_e}{S} = .2$ which corresponds to a $1\frac{5}{16}$ inch shaft. Shaft No. 5-3 is milled from stock $1\frac{3}{8}$ inch in diameter. (See plate V.)

Calculations for Rear Axle - No. 5-1.

Assume the actual weight coming upon the rear axle is 50 per cent of the weight of the empty car plus the weight of three persons in the tonneau plus one half the weight of the two persons in the front seat, assuming each person weighs 150 lb. Hence $W = 1250 + 450 + 150 = 1850$ say 2000 lb. Fig. 3 shows the left half of the rear axle in diagram form. The reaction at A is due to one half weight, coming on the rear axle; this acts vertically. Also, A is partly caused by a force which acts in a horizontal plane and which gives the car motion. This horizontal component of A is equal to 500 lb. The reaction at the point A $= \sqrt{1000^2 + 500^2} = 1115$ lb., from which it follows that the bending moment at $A = \frac{1115 \times 19}{21.5} \times 2.5 = 2470$ in. lb. The twisting moment $= \frac{3000}{2} \times 6 = 9000$ in. lb. Combining by Guest's law, we have $M_e = \sqrt{9000^2 + 2470^2} = 9300$ in. lb. Therefore $\frac{M_e}{S} = .422$ which corresponds to a shaft $1\frac{5}{8}$ inch in diameter. Since the shaft is larger at A where the maximum moment comes, a $1\frac{1}{2}$ inch shaft may be used. At A the shaft is 1.7716 inch in diameter to permit the use of a commercial size of ball bearing.

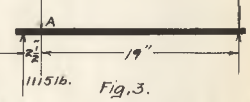


Fig. 3.

Calculations for Rear Axle Casting No. C-1.

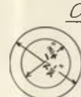


Fig. 4.

Taking the horizontal force E, (Fig. 3), which moves the car, the bending moment at $N = 1000 \times \frac{54\frac{1}{8}}{4} = 13500$ in. lb. Taking forces B and C due to weight on rear axle, the bending moment at M or O $= 1000 \times 7.31 = 7310$ in. lb. Fig. 3 shows graphically the bending moments at various points on the rear axle casting. The diagram AED represents the bending moment in a horizontal plane, while the diagram ABCD represents the bending moment in a vertical plane. Combining all moments at M, we get, for the maximum bending moment at that point $\sqrt{7310^2 + 4000^2} = 8800$ in. lb. Fig. 4 shows the smallest section as at M or O, where the stress $= \frac{Mc}{I} = 8800 \times \frac{1}{.098 \times 14.52}$ $= 6175$ lb. per sq. in.

Fig. 3.

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Thesis
Plate of Calculations
Scale : —
May, 1910
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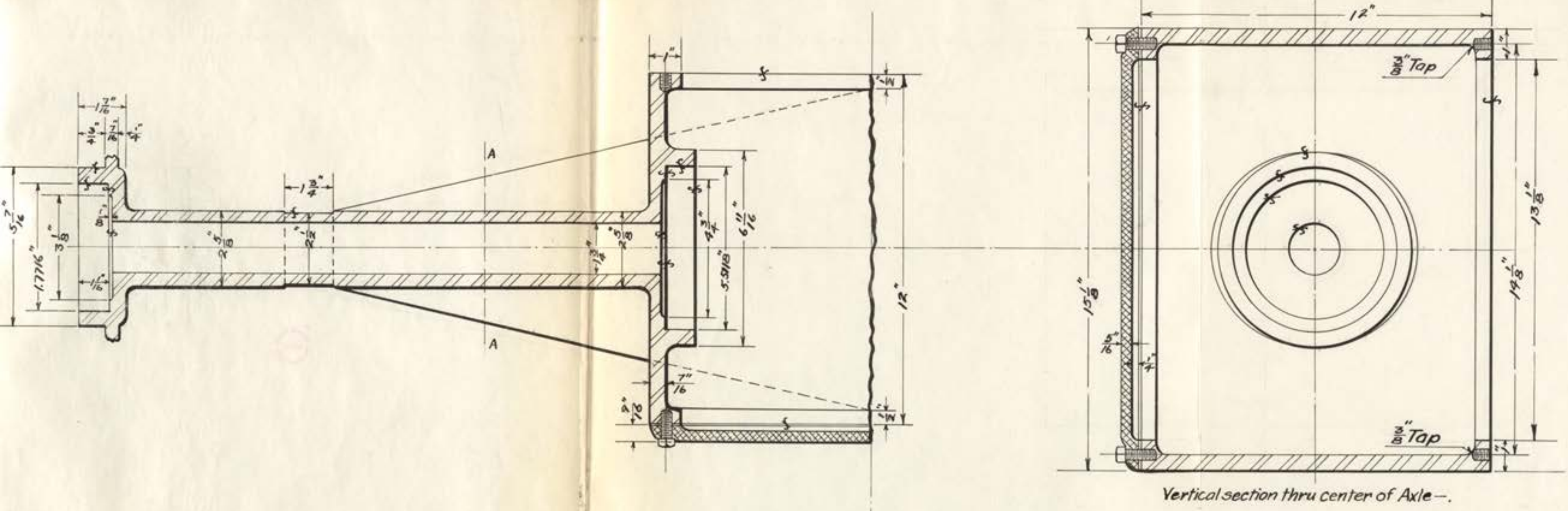
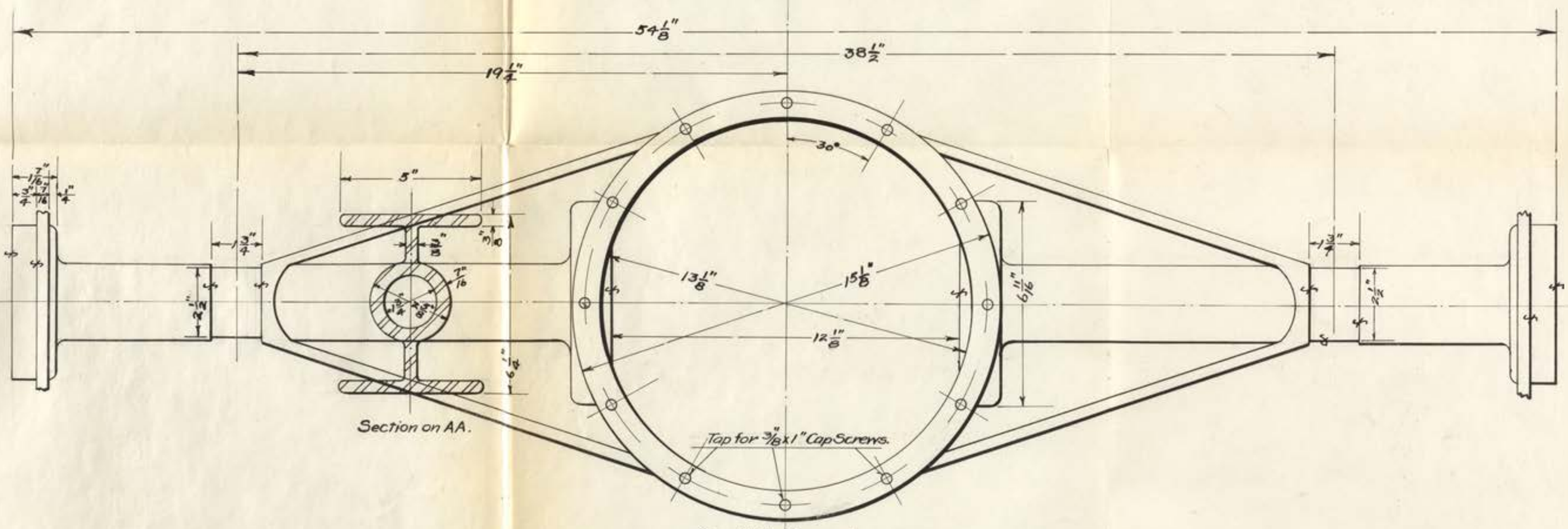


Vertical section thru center of Axle—.

SECTIONS THRU AXLE SHOWING COVER PLATE.



Section on AA.

Cover Plate Removed.

No. C-1, REAR AXLE, STEEL CASTING
AND
No. A-1 ALUMINUM COVER PLATE.

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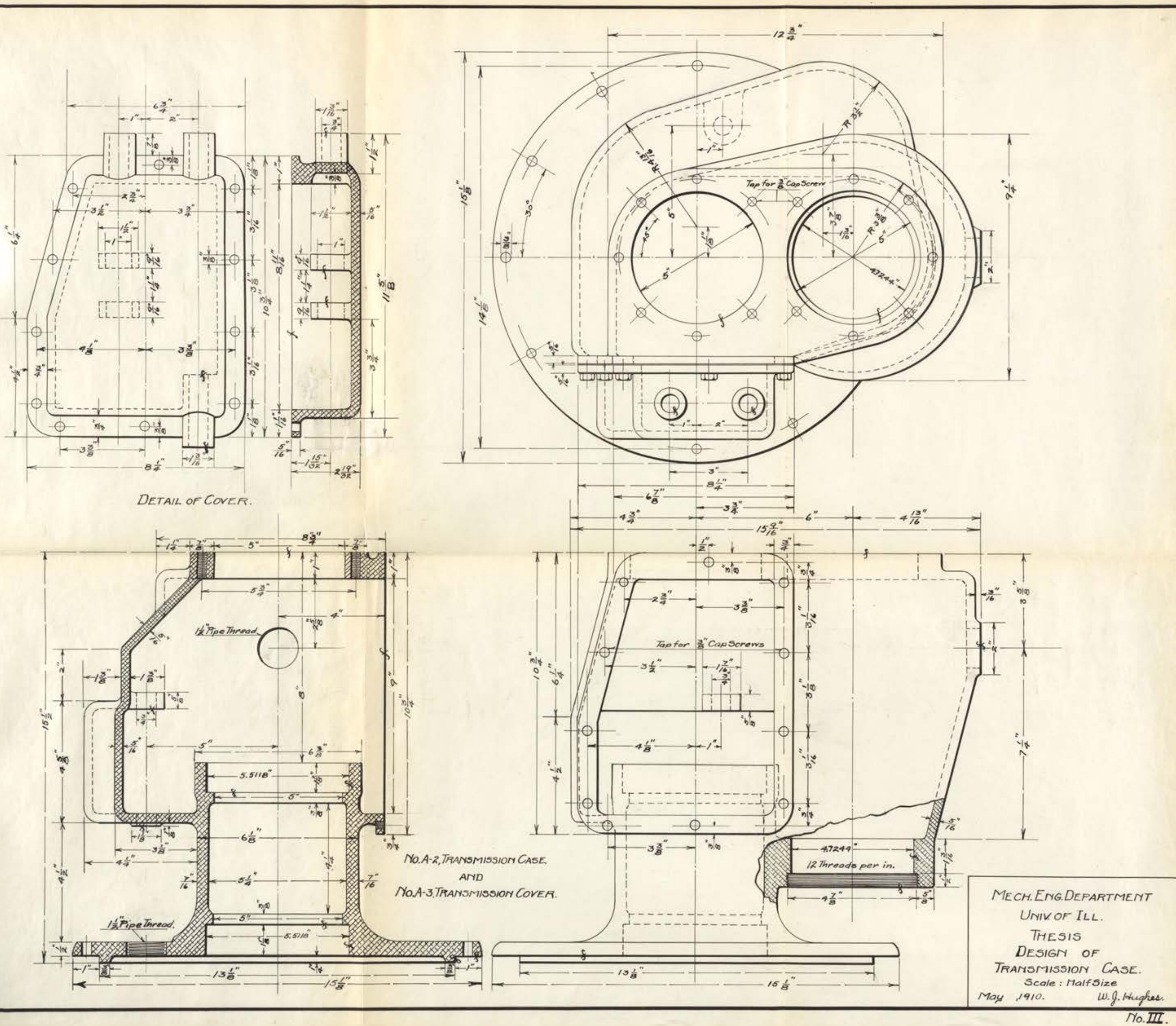
THESIS
DESIGN OF
REAR AXLE CASTING.
Scale : 3/8 Size
May, 1910 W. J. Hughes.

No. II.



DETAIL OF COVER.
Tap for 3/8" Cap Screw
Tap for 3/8" Cap Screws
1½" Pipe Thread
1½" Pipe Thread.
12 Threads per in.
No. A-2, TRANSMISSION CASE
AND
No. A-3, TRANSMISSION COVER.
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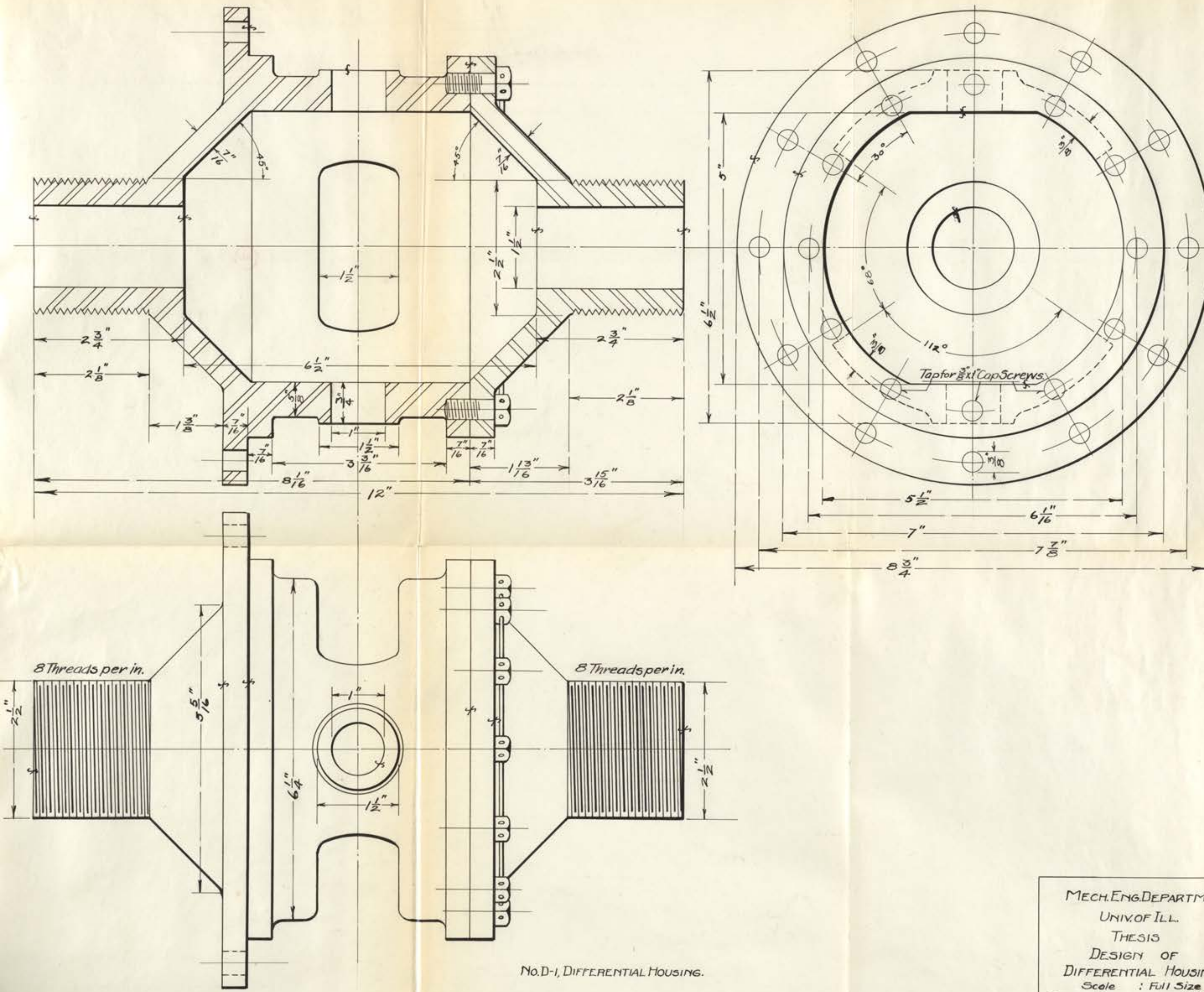
UNIV. OF ILL.
THESIS
DESIGN OF
TRANSMISSION CASE.
Scale: Half Size
May, 1910.
W. J. Hughes.
No. III.



No. D-1, Differential Housing.

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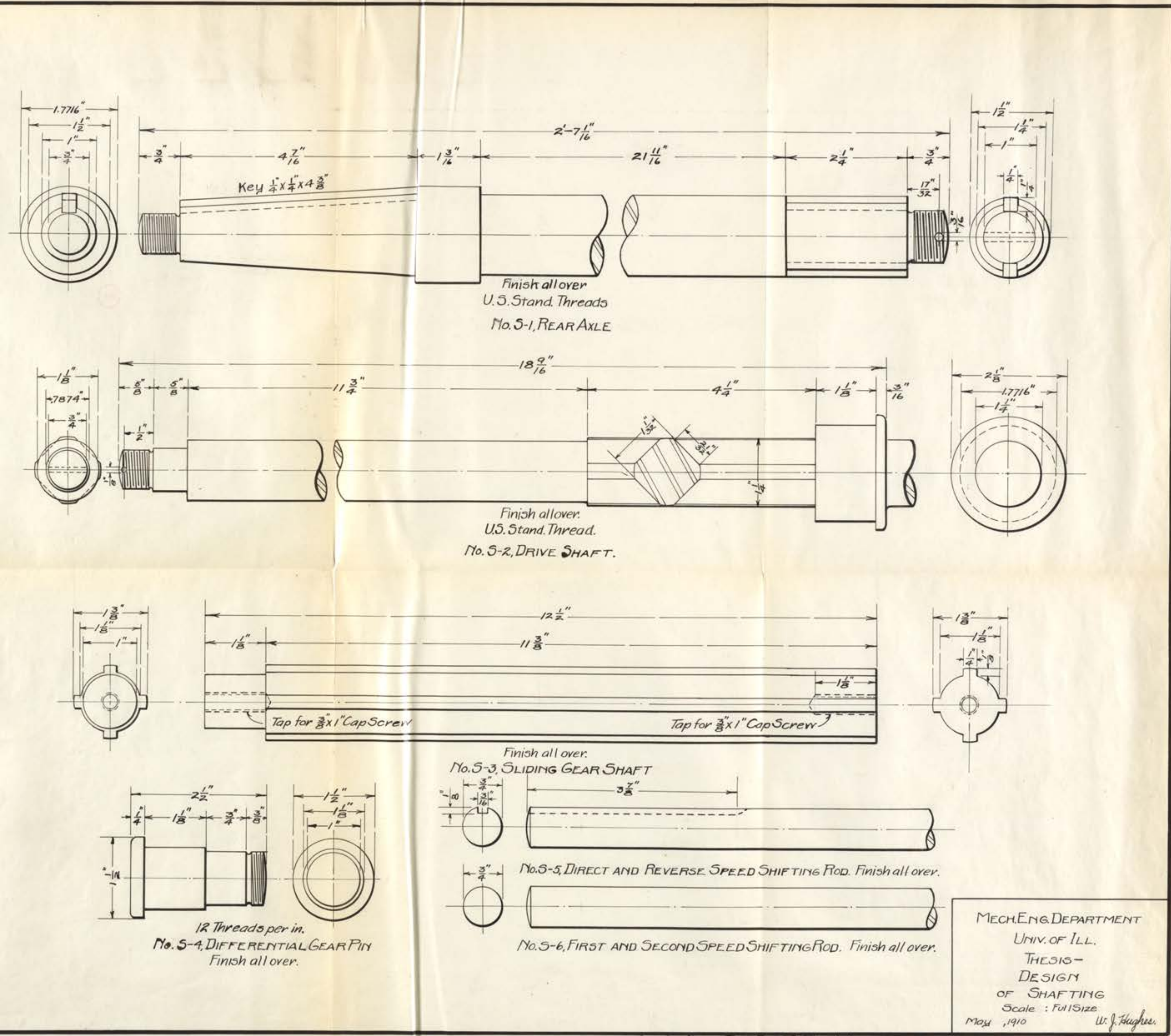
Thesis
Design of
Differential Housing
Scale : Full Size
May 1910
W. J. Hughes.


1.7716"
1½"
1"
¾"
2'-7 1/16"
¾"
4 7/16"
1 3/16"
21 11/16"
2¼"
¾"
Key ¼"x¼"x4⅜"
17/32"
1½"
1¼"
1"
¼"
Finish all over
U.S. Stand. Threads
No. S-1, Rear Axle
18 9/16"
1⅛"
.7874"
¾"
⅝"
⅝"
11¾"
4¼"
1⅛"
3/16"
½"
2⅛"
1.7716"
1¼"
1¼"
Finish all over.
U.S. Stand. Thread.
No. S-2, Drive Shaft.
1⅜"
1⅛"
1"
12½"
1⅛"
11⅜"
1⅛"
Tap for ⅜"x1" Cap Screw
Tap for ⅜"x1" Cap Screw
1⅜"
1⅛"
¼"
Finish all over.
No. S-3, Sliding Gear Shaft
2½"
¼"
1⅛"
¾"
⅜"
1"
1½"
1⅛"
1"
12 Threads per in.
No. S-4, Differential Gear Pin
Finish all over.
¾"
3/16"
3⅞"
No. S-5, Direct and Reverse Speed Shifting Rod. Finish all over.
¾"
No. S-6, First and Second Speed Shifting Rod. Finish all over.
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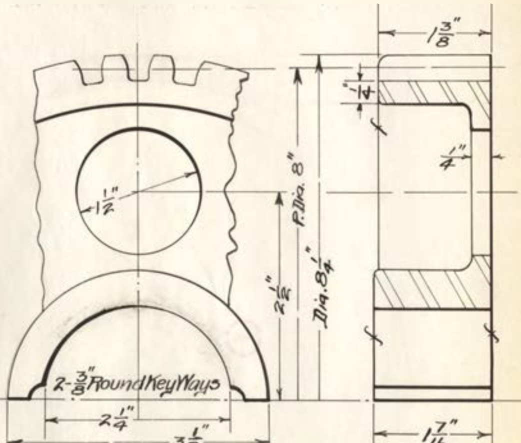
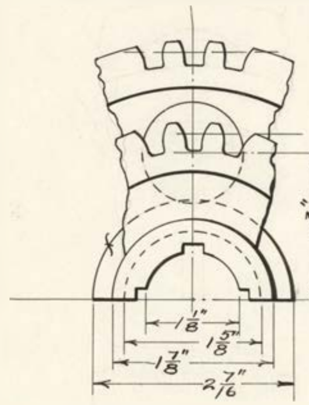
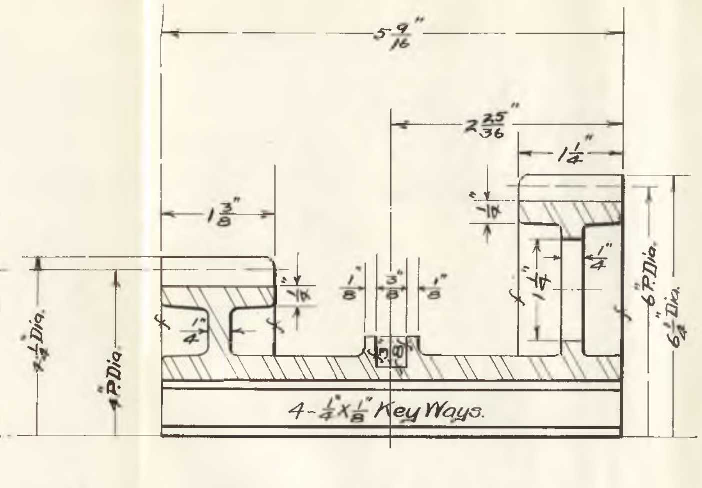
Thesis –
Design
of Shafting
Scale: Full Size
May, 1910
W. J. Hughes.

No. V.

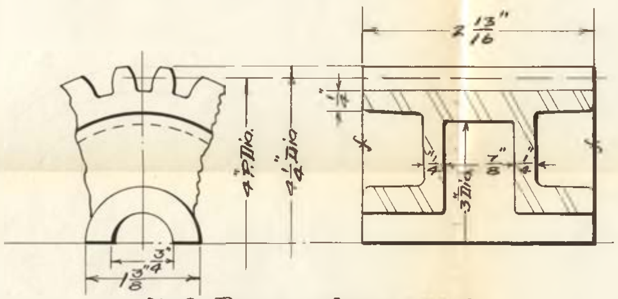


No. 8, 1st. Speed Gear. 48 T.



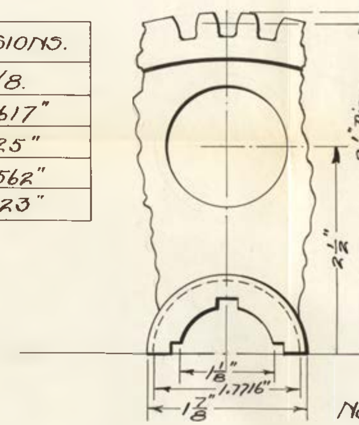


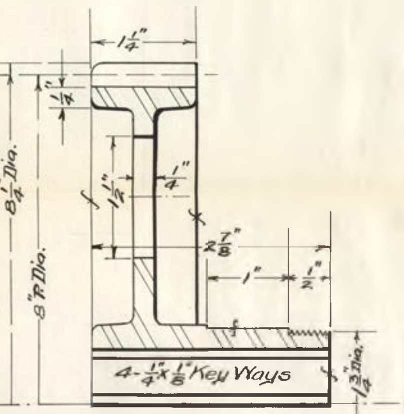
No. 5-6, Sliding Combination 24 and 36 T.



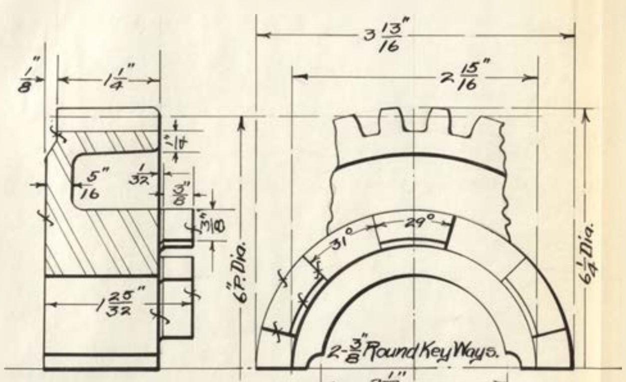
No. 9, Reverse Idler - 24 T.

Tooth Dimensions.	
Pitch	6/8.
Thickness	.2617"
Addendum	.125"
Dedendum	.1562"
Cir. Pitch.	.523"

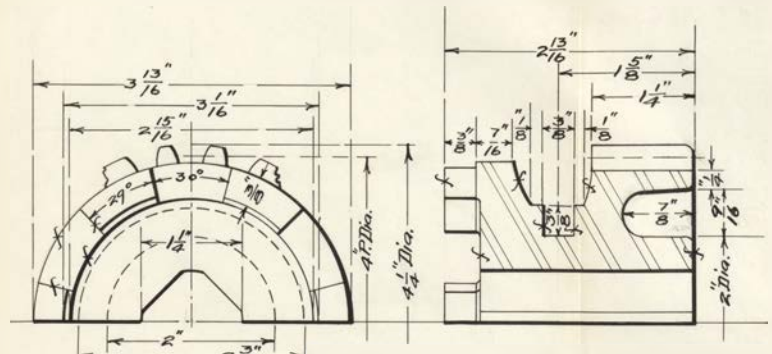




No. 4, End Bearing Gear. 48 T.



No. 7, 2nd. Speed Clutch Gear - 36 T.



No. 3, Sliding Clutch Pinion, 24 T.

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Transmission
Gearing.
Scale : Full Size
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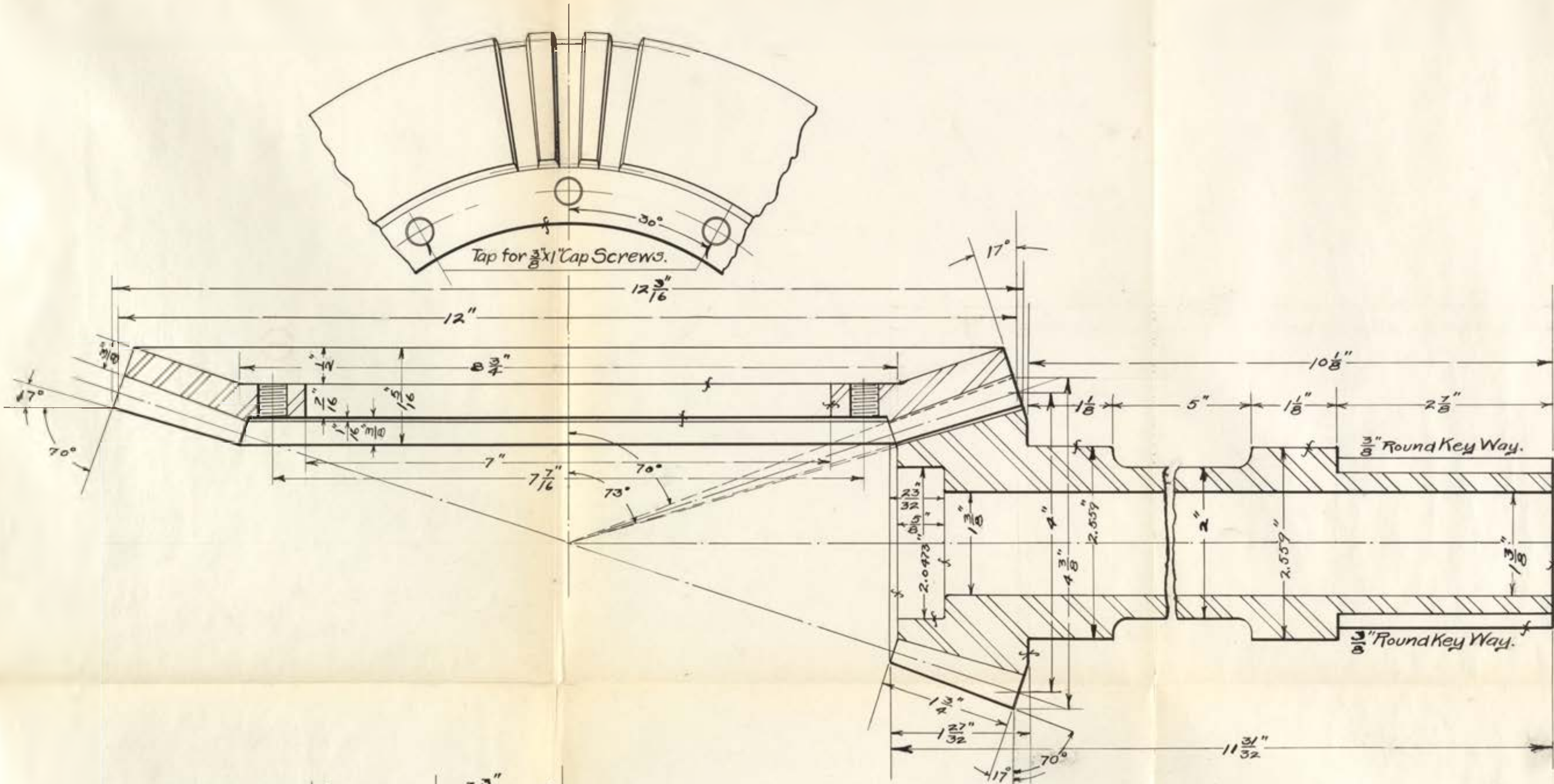
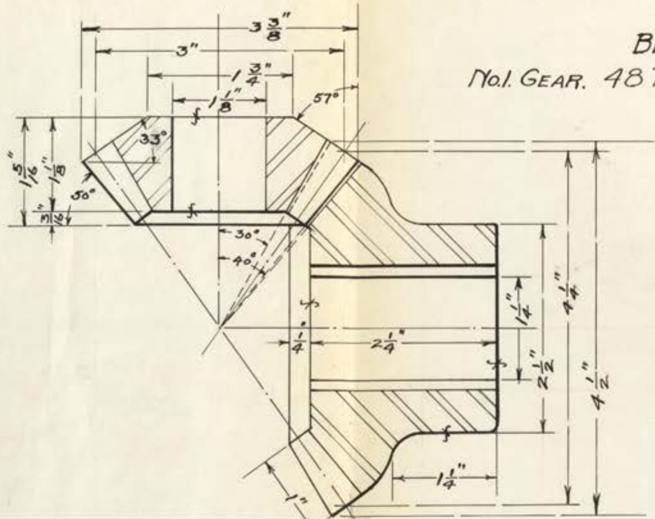
No. VI.



BEVEL DRIVE GEARING.

No.1. GEAR. 48 T. No.2. PINION. 16 T.



BEVEL DIFFERENTIAL GEARING.

No.10. PINION-12 T. No.11. GEAR-17 T.

TOOTH DIMENSIONS.	
PITCH	4/5
THICKNESS.	.3925"
ADDENDUM	.200"
DEDENDUM	.250"
CIR. PITCH	.785"

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THESIS -
BEVEL GEARING.
Scale : Full Size.
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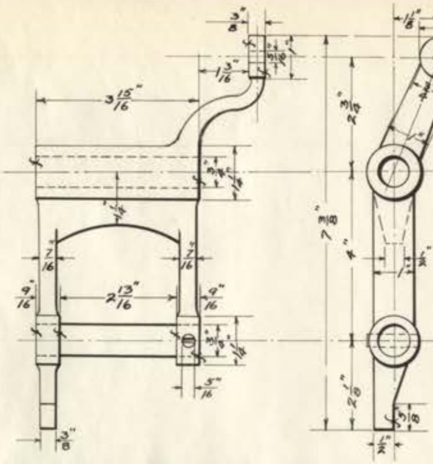
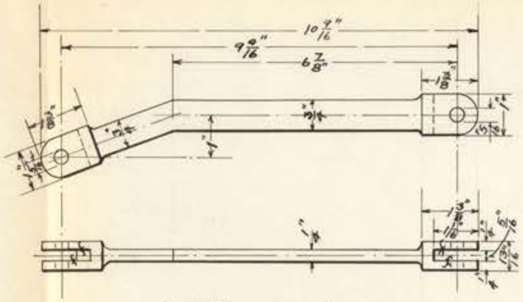
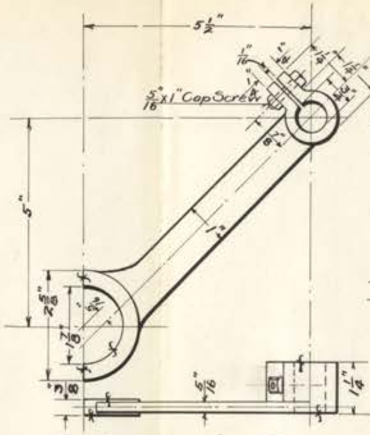


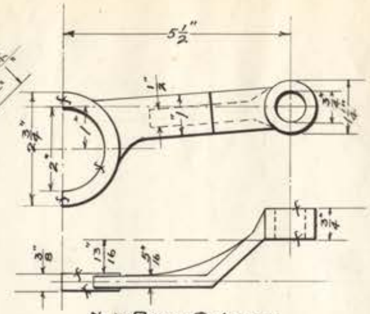
No. 12, Reverse Idler U Frame and Pin.



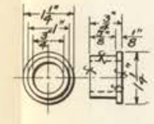
No. 13, Reversing Link.



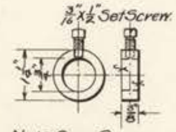
No. 14, 1st and 2nd Gear Shifter.



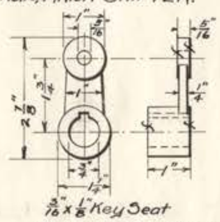
No. 15, Pinion Shifter.



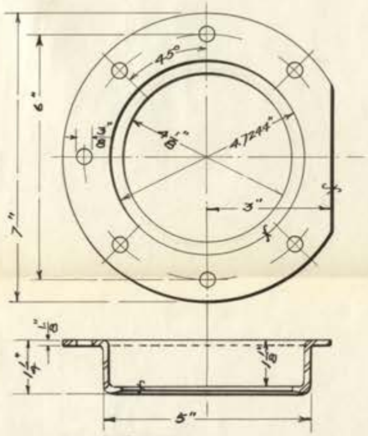
No. 17, Brass Bushing



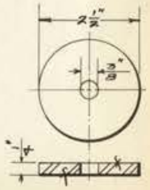
No. 18, Set Collar.



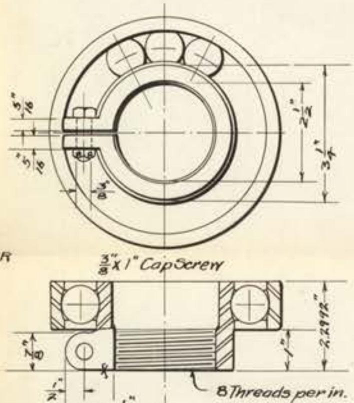
No. 16, Reversing Arm



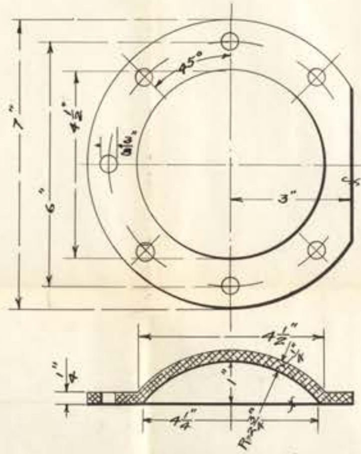
No. 19, Bearing Retainer



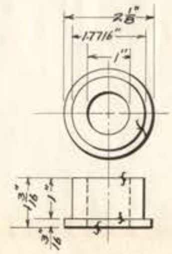
No. 21, Back Bearing Washer



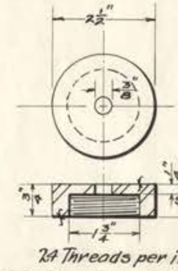
No. 313 H-B Special Differential Bearing



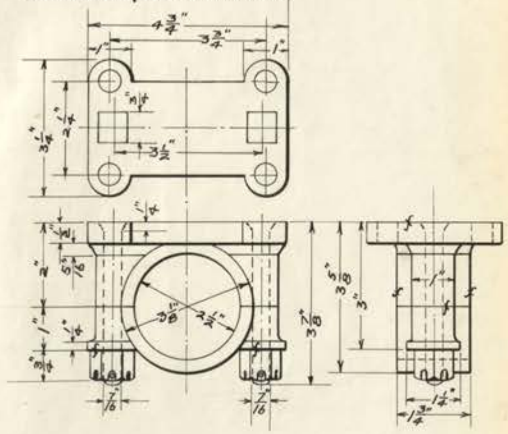
No. 20, Front Dust Cap.



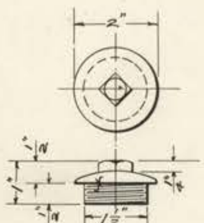
No. 22, Bearing Bush.



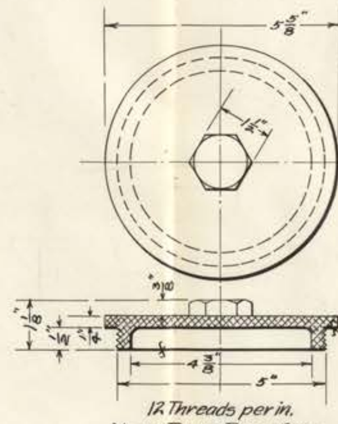
No. 24, Front Bearing Washer



No. 25, Axle Spring Saddle



No. 26, 1½ In. Pipe Plug



No. 23, Rear Dust Cap.

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Thesis
Design
of Details.
Scale : Half Size.
May, 1910. W. J. Hughes.

No. VIII.

Bill of Materials.

No.	Name	Material	Sheet No.	No. Wanted.
1.	Bevel Drive Gear	Drop Forging	VII.	1.
2.	Bevel Drive Pinion	" "	"	1.
3.	Sliding Clutch Pinion	" "	VIII.	1.
4.	End Bearing Gear	" "	"	1.
5-6.	Sliding Combination	Steel Casting	VI.	1.
7.	2nd. Speed Clutch Gear	Drop Forging	"	1.
8.	1st. Speed Gear	" "	"	1.
9.	Reverse Idler	" "	"	1.
10.	Bevel Differential Pinion	" "	VII.	1.
11.	Bevel Differential Gear	" "	"	1.
12.	Reverse Idler U Frame	" "	VIII.	1.
13.	Reversing Link	" "	"	1.
14.	1st and 2nd Gear Shifter	" "	"	1.
15.	Pinion Shifter	" "	"	1.
16.	Reversing Arm	" "	"	1.
17.	Bushing	Brass	"	2.
18.	Set Collar	Soft Steel	"	2.
19.	Bearing Retainer	Pressed Steel	"	3.
20.	Front Dust Cap	Aluminum	"	1.
21.	Back Bearing Washer	Soft Steel	"	1.
22.	Bearing Bushing	" "	"	1.
23.	Rear Dust Cap	Aluminum	"	1.
24.	Front Bearing Washer	Soft Steel	"	1.
25.	Axle Spring Saddle	Cast Iron	"	2.
26.	1 1/2 inch Pipe Plug	Soft Steel	"	2.
A-1.	Axle Cover Plate	Aluminum	II.	1.
A-2.	Transmission Case	"	III.	1.
A-3.	Transmission Case Cover	"	"	1.
S-1.	Rear Axle Shaft	Soft Steel	V	2.
S-2.	Drive Shaft	"	"	1.
S-3.	Sliding Gear Shaft	"	"	1.
S-4.	Differential Gear Pin	"	"	2.
S-5.	Direct and Reverse Speed Shifting Rod	"	"	1.
S-6.	1st and 2nd Speed Shifting Rod	"	"	1.
C-1.	Rear Axle	Steel Casting	II.	1.
D-1	Differential Housing	Drop Forging	IV.	1.
304	5/16 inch H-B Ball Bearing		I.	1.
409	7/8 inch H-B Ball Bearing		I.	5.
313	7/8 inch H-B Ball Bearing		I.	2.
313	7/8 inch Special H-B Ball Bearing		VIII.	2.

Stock List

No. Wanted	Description of Material
48	3/8" x 1" Cap Screws.
14	3/8" x 1 1/4" Cap Screws.
11	3/8" x 3/4" Cap Screws.
2	3/8" x 1 1/2" Cap Screws.
1	1/8" x 1" Cotter Pin
2	3/16" x 1/2" Cotter Pins
2	1/16" x 1" Cotter Pins
3	3/4" Hex. Nuts.
2	1" Hex Lock Nuts.
1	5/16" x 1" Mach. Bolt with Hex. Nut.
1	3/8" x 1" Mach. Bolt with Hex Nut.
2	5/16" x 1" Pins.
4	7/16" x 3 7/8" Square Taper Head Bolts.
2	3/16" x 1/2" Set Screws.
1	3/4" x 3 15/16" Idler Pin.

Mech. Eng. Department
Univ. of Ill.
Thesis
Stock List and
Bill of Materials
Scale : —
May, 1910. W.J. Hughes.