MECHANISM FOR DRIVING PROPELLERS

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Fig. 1

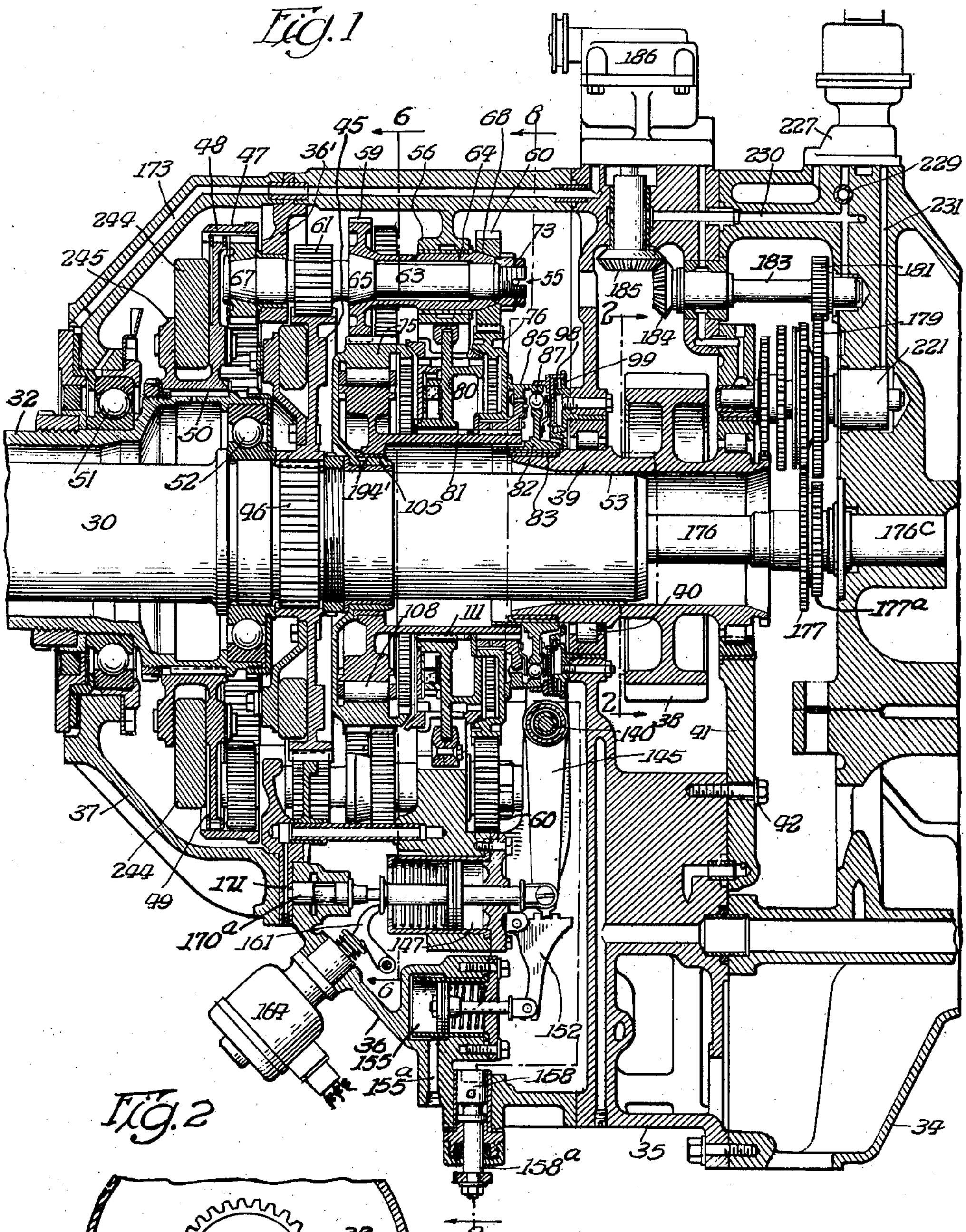
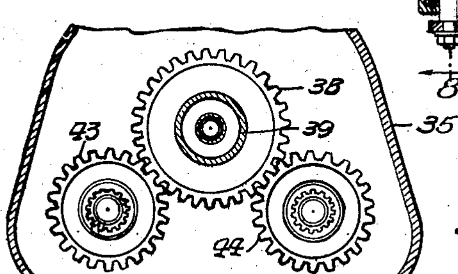


Fig. 2



INVENTORS
Samuel K. Hoffman
and Harold Caminez, Dec'd.
By Hazel Caminez, Extrix.
By Fred Gerlach
Their Atty.

Fig. 5

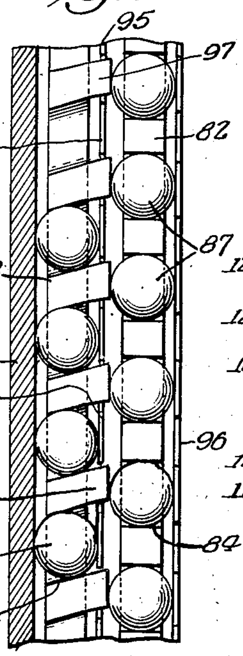


Fig. 3

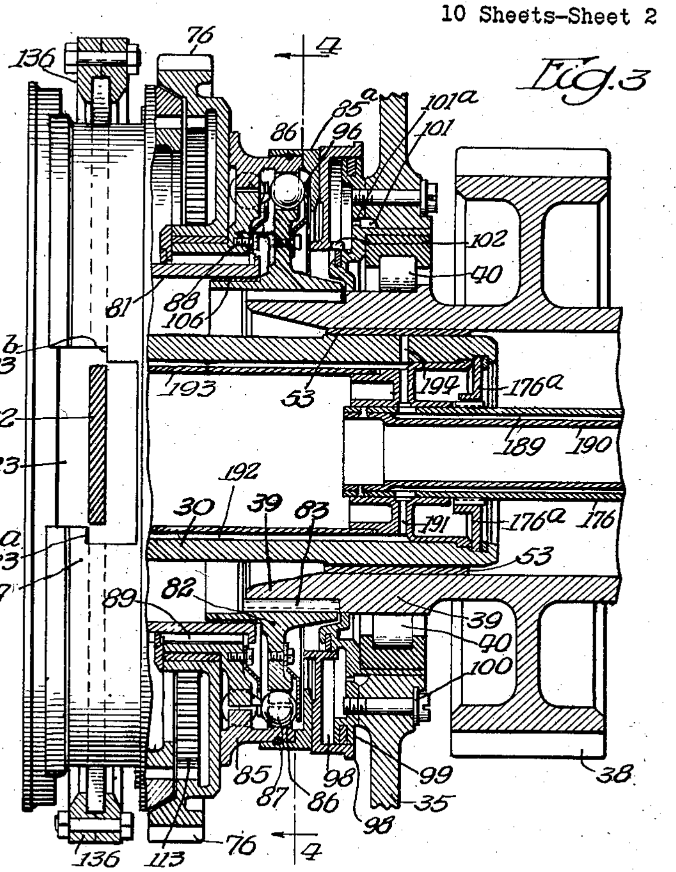


Fig. 5a

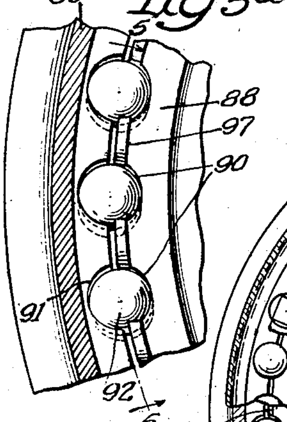
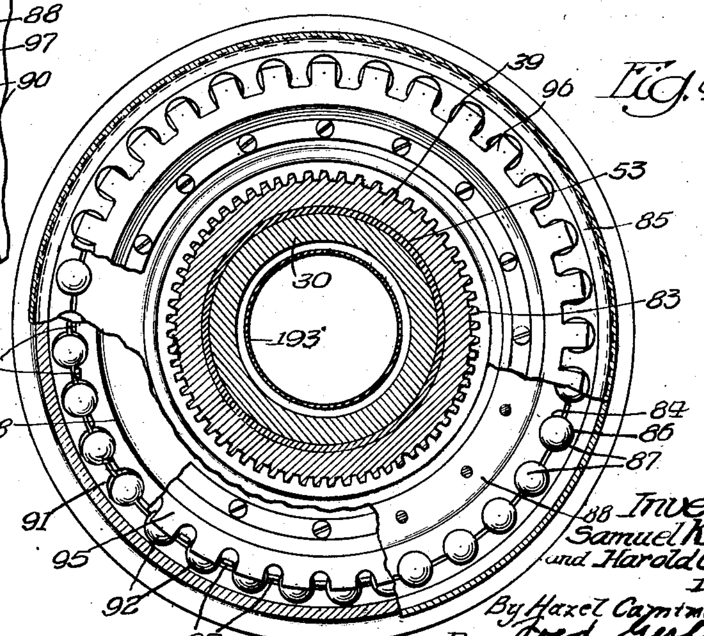


Fig. 4



Inventors.
Samuel K. Hoffman
and Harold Caminez
Deceased
By Hazel Caminez, Ex'trx.
By Fred Gerlach
Their Att'y.

MECHANISM FOR DRIVING PROPELLERS

Fig. 6

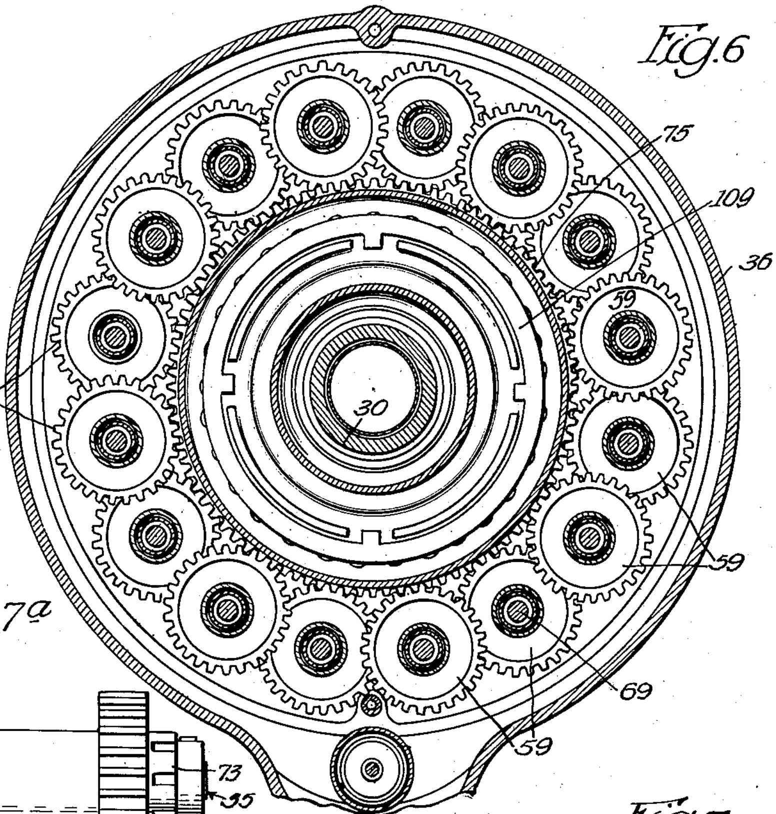


Fig. 7a

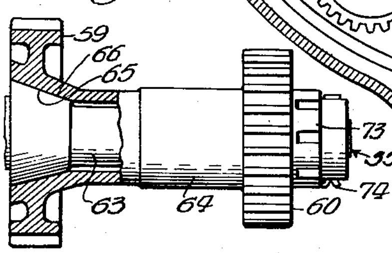
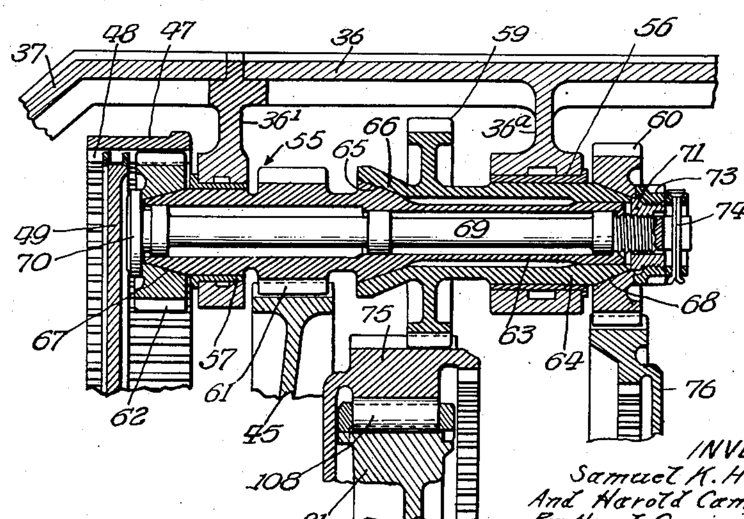


Fig. 7



INVENTORS
Samuel K. Hoffman
And Harold Caminez, Dec'd.
By Hazel Caminez, Extrix.

By Fred Gerlach
Their Atty.

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Fig. 8

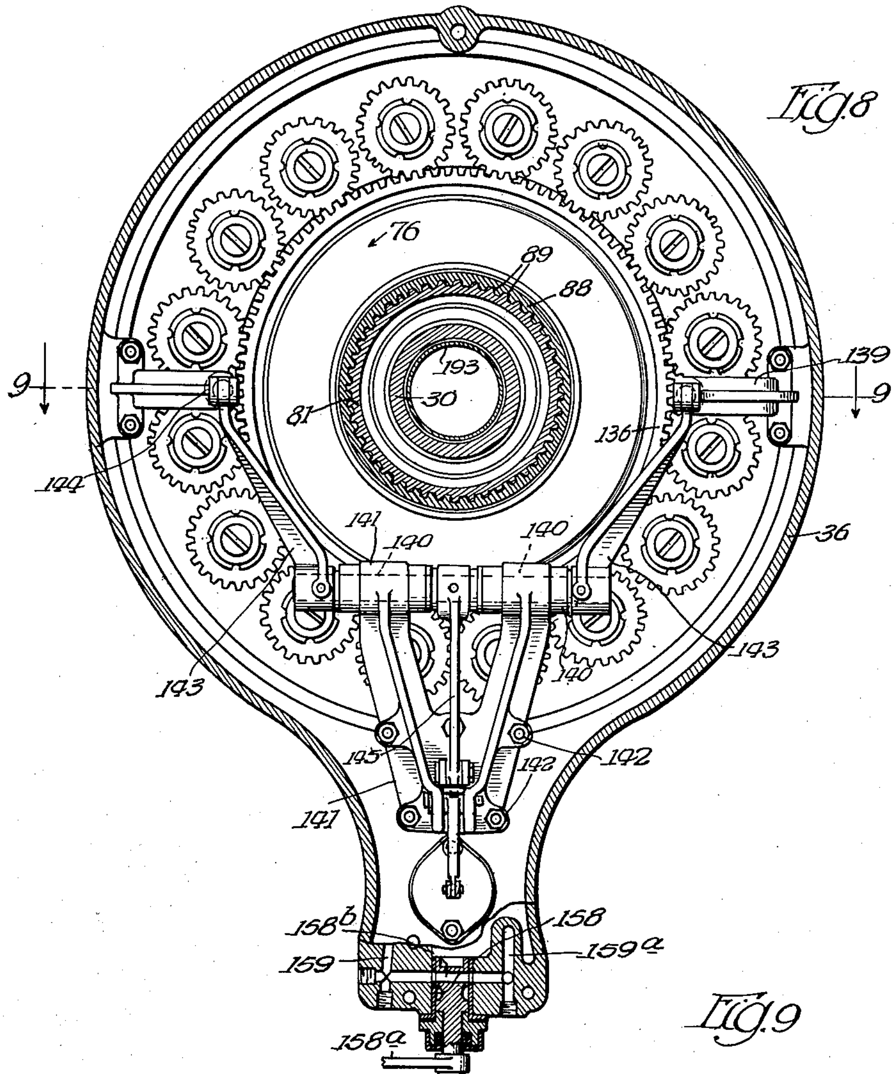
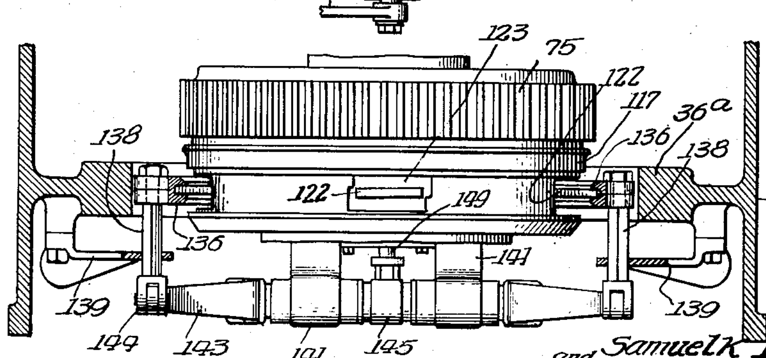


Fig. 9



Inventors
and Samuel K. Hoffman,
Harold Caminez, Dec'd
By Hazel Caminez, Extrix.

By Fred Gerlach Att'y

June 20, 1950 S. K. HOFFMAN ET AL 2,512,103

MECHANISM FOR DRIVING PROPELLERS

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Fig. 10

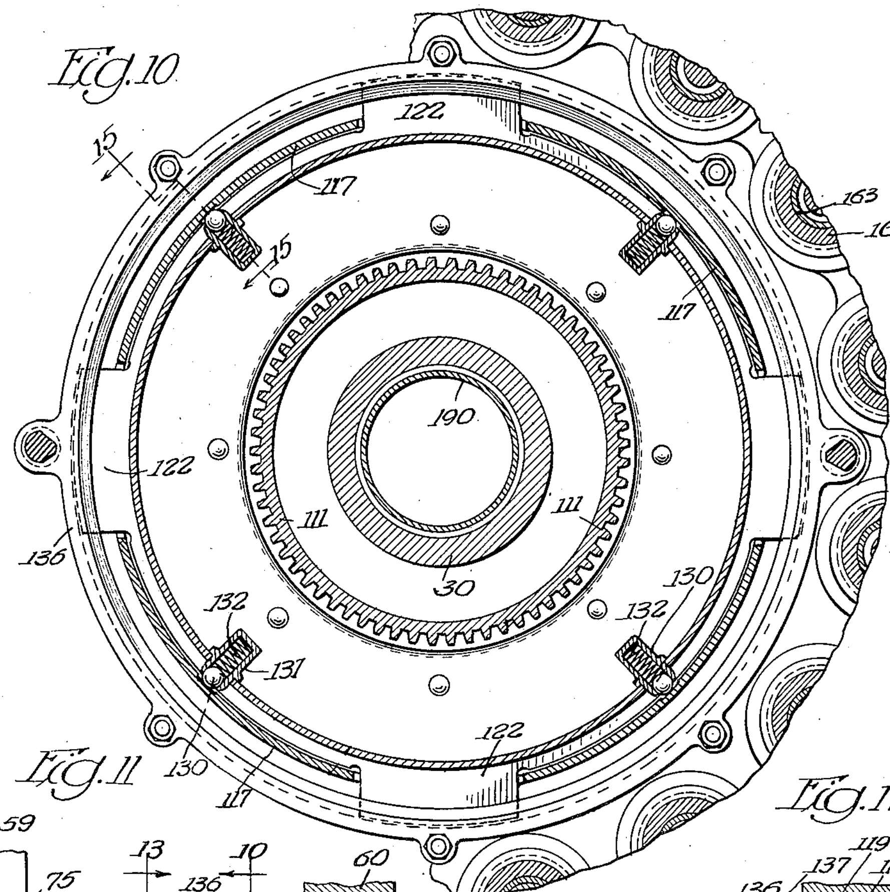


Fig. 11

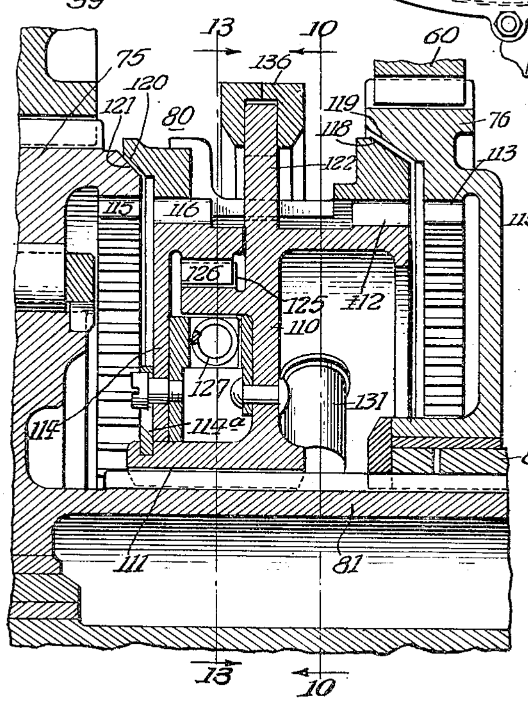
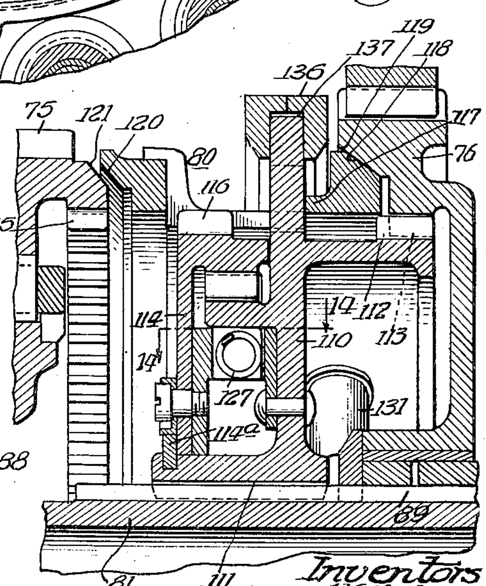


Fig. 12



Inventors
Samuel K. Hoffman
And Harold Caminez, Dec'd.
By Hazel Caminez, Extrix.
By Fred Gerlach Atty.

June 20, 1950 S. K. HOFFMAN ET AL 2,512,103

MECHANISM FOR DRIVING PROPELLERS

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Fig. 13

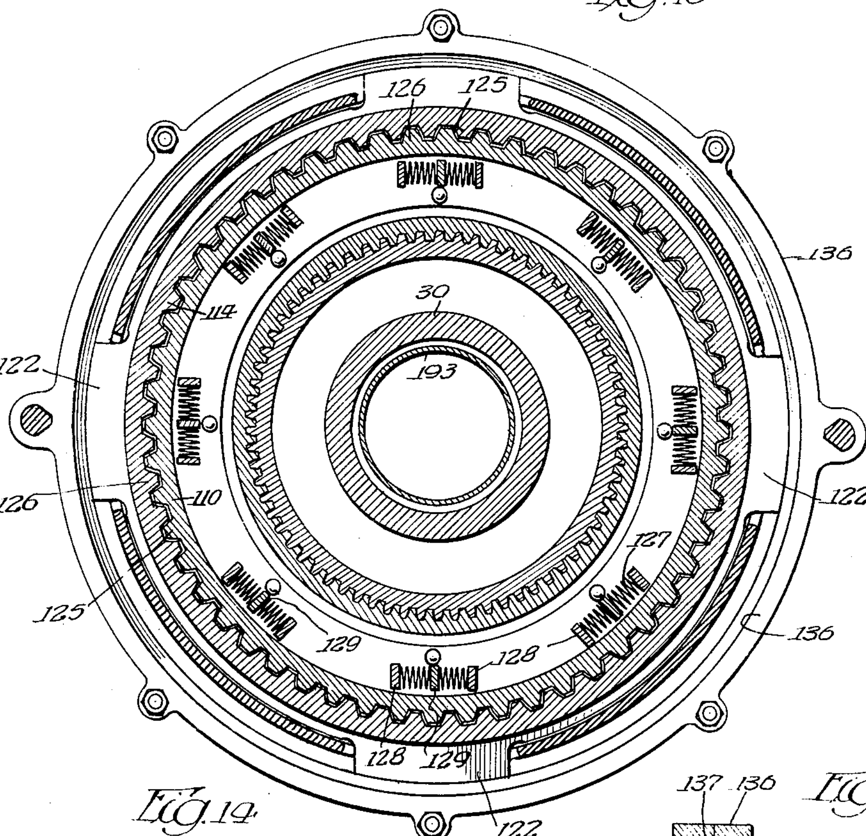


Fig. 14

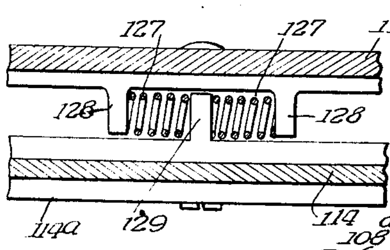


Fig. 15

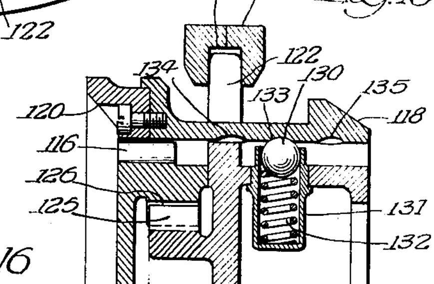
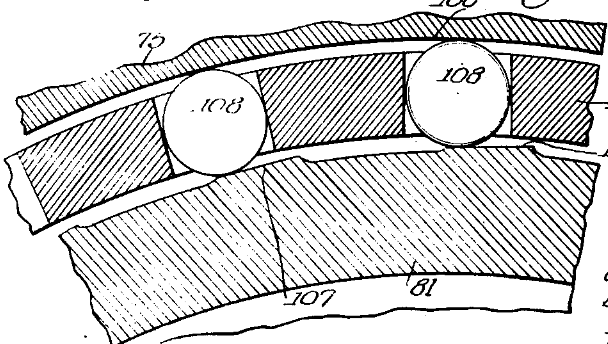


Fig. 16



INVENTOR
Samuel K. Hoffman
and Harold Caminez, Dec'd.
By Hazel Caminez, Ex'trix.
By Fred Gerlach
Atty.

June 20, 1950 S. K. HOFFMAN ET AL 2,512,103

MECHANISM FOR DRIVING PROPELLERS

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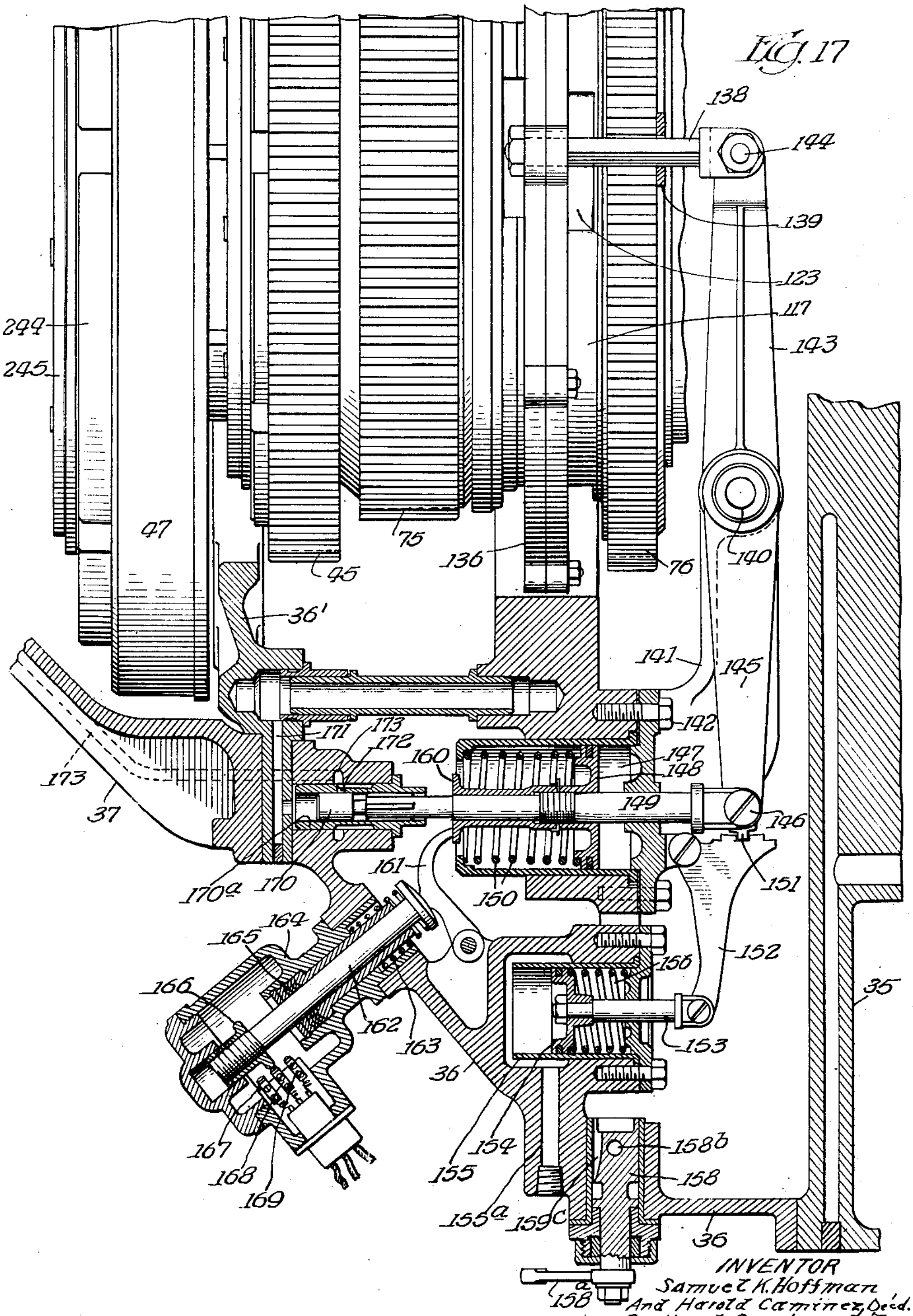
Fig. 17



INVENTOR
Samuel K. Hoffman
And Harold Caminez, Dec'd.
By Hazel Caminez, Ex'trix.
By Fred Gerlach Atty.

June 20, 1950 S. K. HOFFMAN ET AL 2,512,103

MECHANISM FOR DRIVING PROPELLERS

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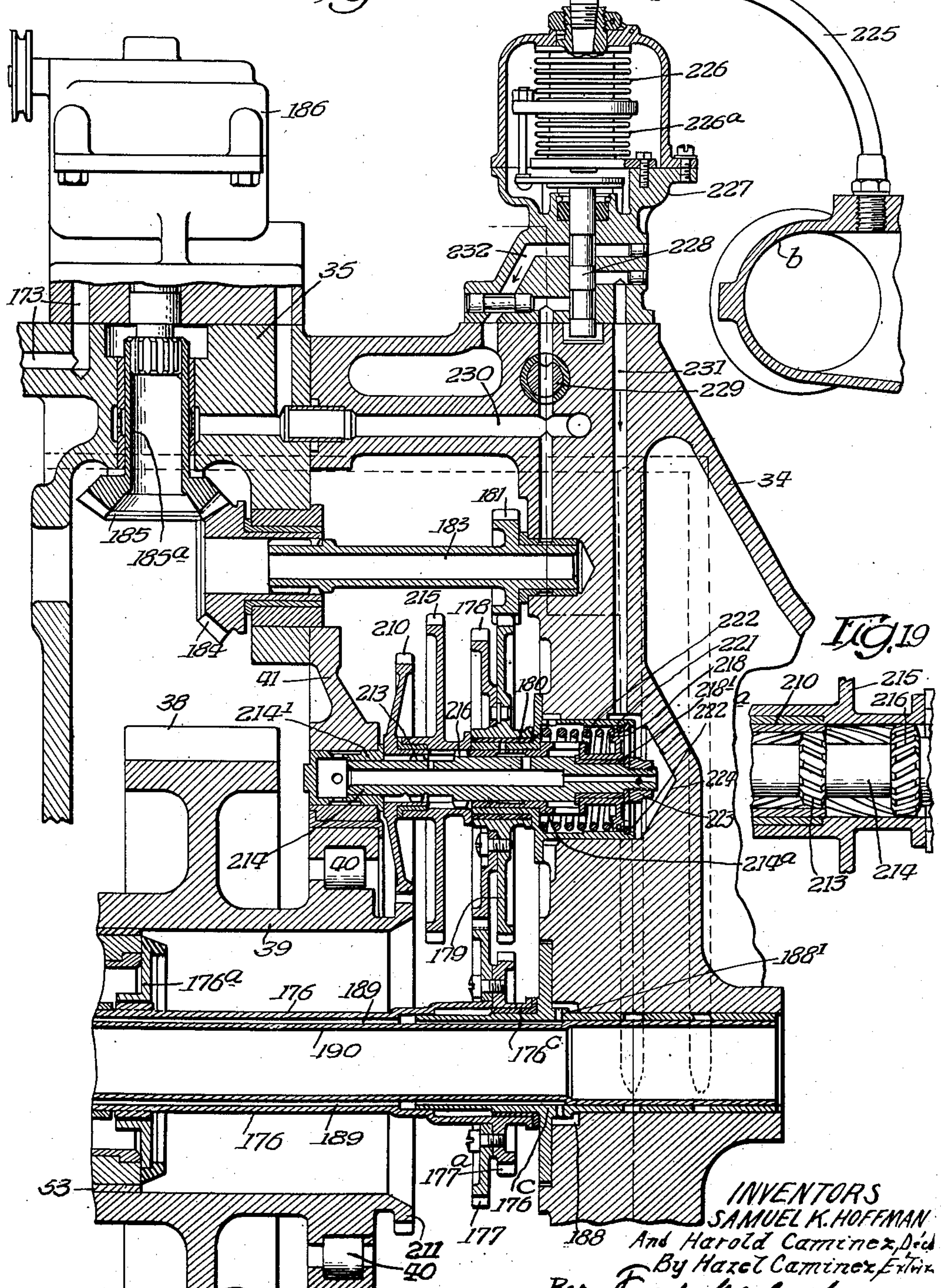
Fig. 18



INVENTORS
SAMUEL K. HOFFMAN
And Harold Caminez, Dec'd
By Hazel Caminez, Extrix.
By Ford Gerlach Atty

MECHANISM FOR DRIVING PROPELLERS

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Fig. 20

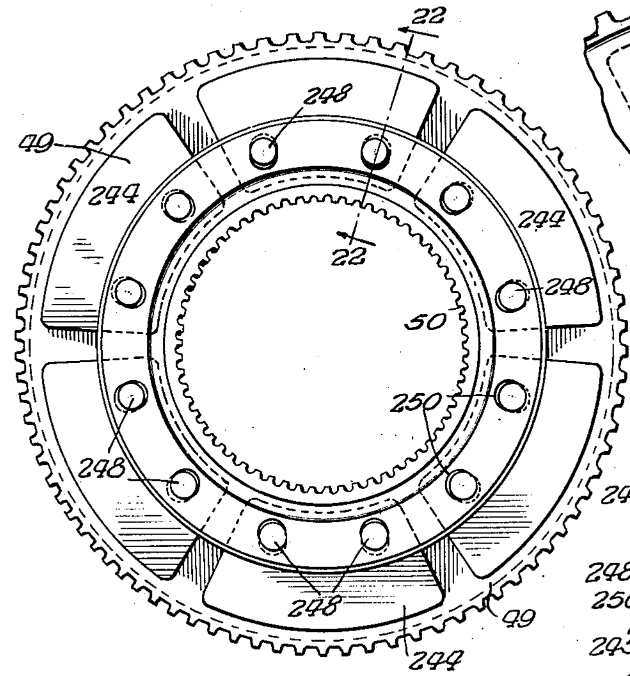


Fig. 21

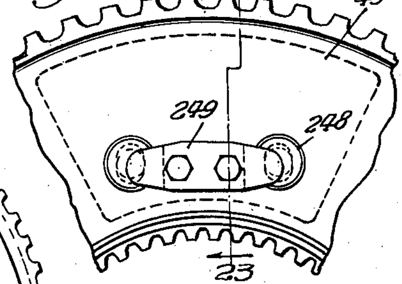


Fig. 22

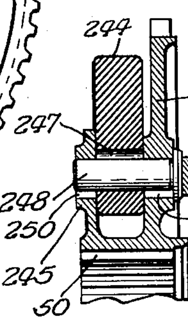


Fig. 23

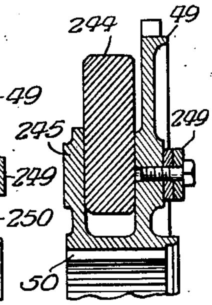


Fig. 24

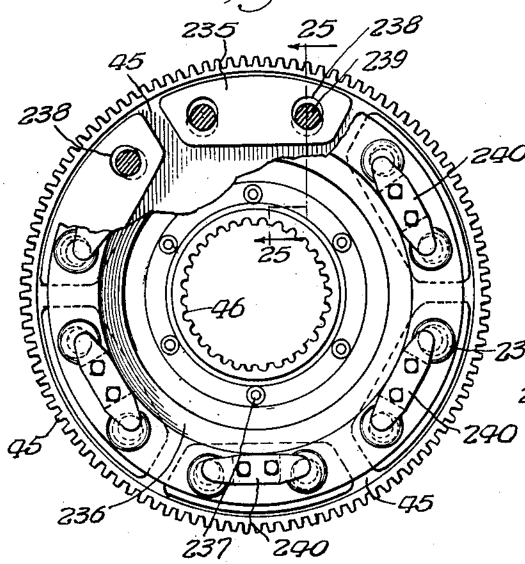


Fig. 26

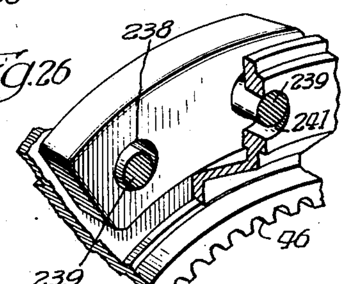
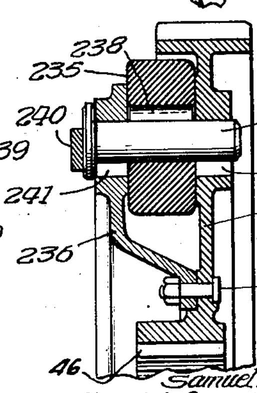
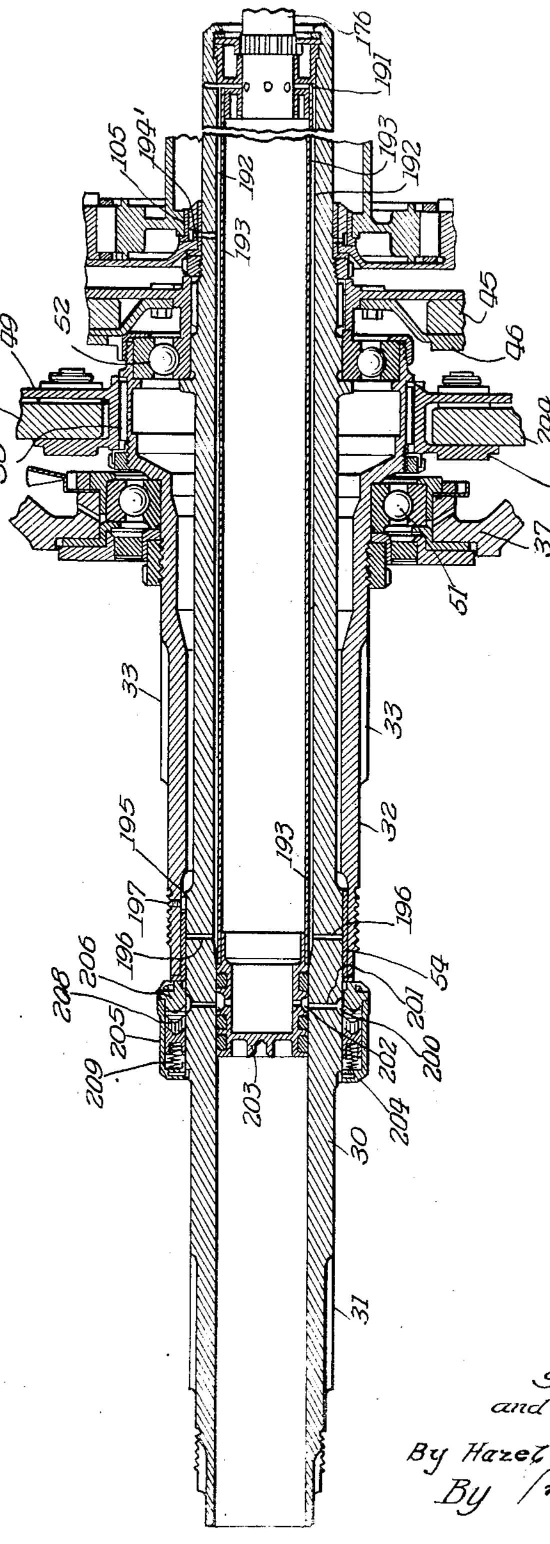


Fig. 25



Inventors
Samuel K. Hoffman
and Harold Caminez, Decd.
By Hazel Caminez, Extrix.
By Fred Gerlach Atty.

Fig. 27



Inventors
Samuel K. Hoffman
and Harold Caminez
Deceased
By Hazel Caminez, Ex'trix.
By Fred Gerlach
Their Atty.

Patented June 20, 1950 2,512,103

UNITED STATES PATENT OFFICE

2,512,103

MECHANISM FOR DRIVING PROPELLERS

Samuel K. Hoffman, Williamsport, Pa., and Harold Caminez, deceased, late of Williamsport, Pa., by Hazel Caminez, executrix, Williamsport, Pa., assignors to Avco Manufacturing Corporation, a corporation of Delaware

Application February 18, 1944, Serial No. 522,910

14 Claims. (Cl. 192—48)

1

The invention relates to mechanism for driving aircraft propellers.

The objects of the invention are to provide: improved mechanism for driving a pair of contra-rotating propellers; improved means for driving contra-rotating propellers at a plurality of speed ratios relative to the engine speed; improved means whereby the pilot may selectively control the driving mechanism for operation at different speed ratios from the engine; variable speed driving mechanism for the propellers which includes toothed slidably engageable clutching elements and means for controlling the engagement of such elements to prevent knocking of the teeth in effecting speed changes; a unit of improved construction for equalizing the tooth loads on a plurality of gears on a common shaft used for variable speed driving; improved torque controlled means for controlling the clutch device for speed changes; improved hydraulic and torque controlled means selectively controllable by the pilot for controlling the speed changes; driving mechanism for contra-rotating propellers with vibration dampers for eliminating the interference effect of blade-passing and to isolate the vibrations from the remainder of the driving mechanism and the engine; improved means for lubricating an outboard bearing between the shafts of contra-rotating propellers; a toothed clutching device for controlling the variable speed gearing and associated means which requires elimination of appreciable torque in order to effect engagement of the clutch teeth; means for preventing engagement of the clutch teeth with the driving gears for different speeds while appreciable torque is being transmitted through the overdrive clutch used in one of the driving gears; and other objects which will appear from the detailed description.

The invention consists in the several novel features which are hereinafter set forth and more particularly defined by claims at the conclusion hereof.

In the drawings:

Fig. 1 is a vertical central longitudinal section of driving mechanism embodying the invention, parts being shown in elevation.

Fig. 2 is a transverse section taken on line 2—2 of Fig. 1.

Fig. 3 is a central longitudinal section through

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the main driving gear, the torque meter and associated parts, a portion of the balk ring for the clutch device being shown in elevation.

Fig. 4 is a section taken on line 4—4 of Fig. 3, parts being broken away for illustrative purposes.

Fig. 5 is a section taken on line 5—5 of Fig. 5^a.

Fig. 5^a is a detail illustrating the torque responsive ring, the driven element, and the driving balls between them.

Fig. 6 is a section taken on line 6—6 of Fig. 1.

Fig. 7 is a longitudinal section through one of the lay shaft units and its associated gears.

Fig. 7^a is a view of a portion of one of the lay shaft units, the low speed driving gear and a portion of the sleeve member being shown in section.

Fig. 8 is a section taken on line 8—8 of Fig. 1 illustrating more particularly the mechanism for shifting the clutch device for variable speed driving.

Fig. 9 is a section taken on line 9—9 of Fig. 8, parts being broken away.

Fig. 10 is a section taken on line 10—10 of Fig. 11 and illustrating more particularly the spring pressed balls carried by one of the clutch members for shifting the balk ring longitudinally.

Fig. 11 is a longitudinal section through the clutch device and the high speed and low speed driving gears with which the clutch device is engageable, the clutch member being illustrated in neutral position.

Fig. 12 is a view similar to Fig. 11 illustrating the clutch device for engagement with the high speed driving gear.

Fig. 13 is a section taken on line 13—13 of Fig. 11.

Fig. 14 is a section taken on line 14—14 of Fig. 12.

Fig. 15 is a section taken on line 15—15 of Fig. 10.

Fig. 16 is a section illustrating the overrunning clutch between the low speed driving gear and the driving sleeve for the clutch device.

Fig. 17 is a vertical longitudinal section illustrating the hydraulic device for shifting the clutch, the torque controlled device for locking the clutch shifting mechanism, the manually controlled valve for controlling the operation of said hydraulic device, the switch mechanism for indicating the clutch settings, and the valve for controlling the flow of oil to the pitch governor for the blades.

Fig. 18 is a central longitudinal section illustrating the gearing for driving the blade pitch governor, the gearing for driving the engine distributor and magneto, and the automatic pressure responsive device for timing the gear for driving the distributor.

Fig. 19 is a detail illustrating the spline connections through which the gear for driving the timer is driven.

Fig. 20 is a side elevation of the gear which drives the sleeve on which one of the propellers is mounted and the vibration dampers thereon.

Fig. 21 is an elevation of a portion of the opposite side of the gear illustrated in Fig. 20.

Fig. 22 is a section taken on line **22—22** of Fig. 20.

Fig. 23 is a section taken on line **23—23** of Fig. 21.

Fig. 24 is an elevation of the gear which is fixed to drive the shaft for one of the propellers, parts being broken away.

Fig. 25 is a section taken on line **25—25** of Fig. 24.

Fig. 26 is a perspective illustrating one of the vibration dampers shown in Figs. 24 and 25.

Fig. 27 is a longitudinal section through the propeller shafts and the bearings therefor, and parts thereon.

The invention is exemplified in a two-speed driving mechanism for a shaft **30**, the front end of which is provided with splines **31** (Fig. 27) for driving the hub of an outboard propeller and a sleeve **32** around shaft **30** which is provided with splines **33** and functions as a shaft for driving the inboard propeller of a contra-rotation propeller unit. The propellers may be of any suitable construction and are usually provided with variable pitch blades which are adapted to be hydraulically controlled in any manner well understood in the art. The shaft **30** and sleeve **32** are mounted in and project forwardly from a gear-case which is formed of sections and supported from the engine which drives the propellers through the variable speed gearing enclosed in the casing.

This gear-casing comprises a section **34** (Fig. 1) which is secured to and forms a part of the casing of the engine for driving the propellers; a section **35** secured to the front of section **34**; a section **36** secured to the front of section **35**; a front section **37** secured to the front of section **36**; and a wall **36'** secured between sections **36** and **37**.

The variable speed gearing for driving the shafts **30** and **32** is driven from a gear **38** which is provided with a hollow hub **39** which is journaled in roller bearings **40** in the front wall of casing section **35** and a wall **41** secured by screws **42** to the rear face of said casing section. The gear **38** is driven by gears **43** and **44** (Fig. 2) each of which is rotated with and driven by one of the crankshafts of a two crankshaft engine (not shown).

Propeller shaft **30** is driven through an externally toothed gear **45** which is splined as at **46** to the shaft **30**. Propeller shaft **32** is driven through an internally toothed ring gear **47** which is splined at **48** to a wheel body **49**, the hub of which is splined at **50** to shaft **32**. Hollow shaft **32** is journaled in a ball bearing **51** which is mounted in the casing-section **37** and on a ball bearing **52** around the shaft **30**. The inner end of propeller shaft **30** is journaled in a bushing **53** which is held in the hub **39** of the main drive gear **38**. Forwardly of the gear-case, shaft **30** is journaled in a plain bearing **54** which is carried in the front end of the hollow propeller shaft **32**. Gear **45**, which drives the propeller shaft **30**, and gear **47** which drives the hollow propeller shaft **32**, are adapted to be driven by an annular series of longitudinally extending lay shaft units generally designated **55**, which are adapted to be driven at two different speeds relatively to engine-speed.

Each of the lay shaft units **55** (Figs. 1 and 7) is journaled in a bearing **56** in a wall **36**ᵃ which is integral with the casing-section **36** and a bearing **57** in the plate **36'** which is secured between the contiguous faces of the casing-sections **36** and **37**. Each lay shaft unit **55** (Fig. 7) comprises a low speed driving gear **59**, a high speed driving gear **60**, a pinion **61** which meshes with and drives the gear **45** which drives the propeller shaft **30**, and a gear **62** which meshes with and drives the ring gear **47** and wheel body **49** for driving the hollow propeller shaft **32**. The pitch diameters of pinions **61** and gears **62** on the lay shafts and gears **45** and **47** are proportioned to drive gears **45** and **47** at the same speed in opposite directions for the contra-rotation of the propeller shafts **30** and **32**. Each lay-shaft unit **55** is composed of sections which are adapted to be conveniently secured together for equal tooth-loading of all of the gears on each shaft. The gears **59** on circumferentially alternating lay-shaft units **55** are staggered and all mesh with a gear **75** through which the lay-shaft units are driven for low speed driving. Except for this staggered relation of the gears **59**, the lay-shaft units are alike in construction. Each unit **55** comprises a hollow inner shaft-section **63**, and a surrounding sleeve-section **64**. A pinion **61** is integral with each shaft-section **63**. A gear **59** is integral with each sleeve-section **64**. Each shaft-section **63** is provided with a conical portion **65** which fits in a mating conical socket **66** in the sleeve-section **64**. The front end of each shaft-section **63** is provided with a conical portion **67** which fits in a mating conical socket in the gear **62**. The rear end of each shaft-section **64** is provided with a conical portion **68** which fits into a mating conical socket in gear **60**. A bolt **69** extends completely through the hollow shaft-section **63** and is provided with a head **70** which abuts against the front face of the gear **62**. A nut **71** is screw-threaded to the rear end of bolt **69** and abuts against the rear end of the sleeve-section **64** and is adapted to force together longitudinally the gear **62** into jammed relation with conical portion **67** on shaft-section **63**, and the sleeve-section **64** into jammed relation with the conical portion **65** on shaft-section **63**. A second nut **73** is screw-threaded to the nut **71** and is adapted to abut against the rear face of gear **60**, and to force the gear **60** in jammed relation with conical portion **68** on shaft **64**. A lock pin **74** extends through the nut **73** and nut **71** and a slot in the inner end of bolt **69** for preventing relative rotation of nuts **71** and **73** and bolt **69**.

In assembling the gearing, the lay shafts are all assembled in their respective bearings **56** and **57** and the nuts **71** are turned sufficiently to move the conical engaging portions of each shaft **63** into loose relation with the conical sockets in the gears **62** and the sleeve sections **64**. Gear **45** is prevented from rotating by locking its associated propeller against rotation.

Gear **75**, which meshes with gears **59** on the sleeve sections **64** of all the lay shaft units, is then rotated in the direction of normal operation while slippage is permitted between conical sections 65 on shafts 63 and the conical sockets 66 of the sleeve sections 64. Rotation of gear 75 is continued until the tooth loads of all of the pinions 61 on gear 45 are equalized. Gear 45 and its associated propeller are then released for rotation and the propeller associated with gear 47 is then locked against rotation thus preventing its rotation. Gear 75 is again rotated as before and slippage of conical portions 67 in the conical sockets of gears 62 is then permitted until the tooth loads of all of the gears 62 on gear 47 are equalized. No slippage occurs between conical portions 65 and sockets 66 while slippage is occurring between gears 62 and conical portions 67 because gear 45 and its associated propeller are free to rotate and because of the large frictional bearing areas of conical portions 65 compared to the relatively smaller areas of conical portions 67. The nuts 71 are then tightened on bolts 69 which jams the gears 62 on the conical portions 67 of shafts 63 and the conical portions 65 in the sockets 66 of the sleeve sections 64 which are integral with gears 59, respectively, so that these elements are fixedly secured together for conjoint rotation with equal tooth loading. During the setting of the nut 71, a tool is inserted in the slot in the rear end of bolt 69 to prevent its rotation.

Next, the nuts 73 of the lay shaft units are set to hold the gears 60 for slippage on the conical portions 68 of the sleeve sections 64. The gear 76 which meshes with all of the gears 60 is then rotated in the direction of normal operation while slippage is permitted which results in uniform distribution of tooth loads on the gears 60. The propellers are locked against rotation while this adjustment is made. The nuts 73 are then tightened to jam and lock the gears 60 on the conical portions 68 of sleeves 64.

This construction renders it unnecessary to accurately index the teeth of each of the gears of each lay shaft unit relative to one another and provides for ready interchangeability of the lay shafts and gears thereon. By holding the bolts 69 against rotation while nuts 71 and 73 are being tightened, no torsional stresses are applied to the bolts which eliminates any spring or twist which might tend to disturb the adjustment of the gear setting.

The gears 60 on all of the lay-shaft units 55 mesh with and are driven by a gear 76 for the high speed operation of the propellers and the gears 59 on said lay-shafts are driven by gear 75 for low speed operation of the propellers. Gears 75 and 76 are controlled for operating the propellers at two speeds relatively to the engine or the main drive gear 38 through a clutch device generally designated 80 (Figs. 1, 11 and 12) which is controlled as hereinafter described. This clutch device is driven by a sleeve 81 which is driven from hollow hub 39 of the main driving gear 38 by mechanism which comprises a torque responsive member which is utilized in controlling the coupling of the clutch device for driving gears 75 and 76, as hereinafter described.

The driving mechanism for the clutch-device 80 comprises (Figs. 4, 5 and 16) a driving member 82 which is splined and keyed at 83 to the hub 39 of gear 38 in front of the bearing 40, and is provided on its outer periphery with a series of semi-cylindrical longitudinally extending recesses 84 having their axes parallel to the axis of the gear 38 and the propeller shafts; a ring 85 extending circumferentially around the member 82 and provided with semicylindrical recesses 86 mating with recesses 84 and having their axes parallel to the axis of gear 38 and member 82; a series of balls 87 in the semi-cylindrical recesses 84 and 86; a driven member 88 which is provided with a hub which is splined and keyed at 89 to the rear portion of sleeve 81 on which the clutching device 80 is slidable and with a series of semi-cylindrical longitudinally extending recesses 90 (Figs. 4, 5 and 5[a]) in the outer periphery of driven member 88 on axes which are inclined relatively to the axis around which members 88 and 82 rotates helically; a series of semi-cylindrical recesses 91 mating with recesses 90 in the inner periphery of ring 85 and which are helical and parallel to the recesses 90 in member 88; and balls 92 confined in the mating recesses 90 and 91 in the ring 85 and the driven member 88. The balls 87 in recesses 84 and 86, which form cylindrical holes therefor, positively drive ring 85 and member 82 while permitting axial movement of said ring. The mating helical recesses 91 in ring 85 and recesses 90 in the driven member 88 and balls 92, cause the ring 85 to move longitudinally on balls 87 relatively to the driving member 82 according to the torque being transmitted from said member 82 to the driven member 88, and thus render the ring 85 torque responsive.

As the torque transmitted from driving member 82 to driven member 88 increases, the helical recesses 90, 91 and balls 92 will force the ring 85 rearwardly and as the torque decreases the ring 85 will be shifted forwardly. The balls 87 and 92 act as anti-friction members during the longitudinal movement of the ring 85. The cylindrical holes for balls 87 are offset or staggered with respect to the holes for the balls 92. Balls 87 are held in the recesses in the driving member 82 and the ring 85 by a spring disk 96 and abutments 97 on the driven member 88. Balls 92 are held in their recesses in ring 85 and driven member 88 between a spring disk 95 and the rear face of gear wheel 76. The disks 95 and 96 exert only sufficient force to retain the balls in their respective recesses when no torque is being transmitted through the ring 85. The sleeve 81 to which the driven member 88 is splined and keyed, is journaled on a bearing 106 on the hub of driving member 82 and a bearing 105 on the hub of gear 75.

The axial movement of the torque responsive ring 85 controls the pressure of oil in a hydraulic cylinder 98 (Fig. 3) which is slidably mounted on a piston 99 which is secured by screws 100 to the front wall of casing-section 35. A thrust ring 85[a] is secured to the ring 85 and is adapted to abut against the cylinder 98 for axially shifting cylinder 98 rearwardly when the ring 85 is correspondingly shifted. Oil from the engine pressure lubrication system is delivered by a suitable duct in the gear casing to the duct 101, usually at 100 lbs. p. s. i. pressure, and flows from said duct through an orifice 101[a] into the cylinder 98. The cylinder 98 is provided with a restricted discharge notch or orifice 102. The orifice 102 in cylinder 98 by a valve action over piston 99 reduces the discharge of oil from cylinder 98 as the torque transmitted through the ring 85 increases. The shifting of the ring 85 rearwardly, due to increase in transmitted torque, causes a corresponding increase of pressure in said cylinder. Decrease of transmitted torque permits the oil pressure in cylinder 98 to move the piston 99 and ring 85 forwardly. This movement of piston 99 increases the exposed area of orifice 102 and allows a higher rate of discharge of oil from orifice 102. This high rate of discharge correspondingly reduces the pressure in said cylinder. The port 102 in cylinder 98 and piston 99 function as a by-pass valve for controlling the discharge of oil from the engine pressure oil system back to the gear casing to vary the pressure in said cylinder. The flow of oil permitted by the orifice 102 may be such that when the maximum thrust is transmitted through ring 85, the pressure drop in the cylinder will not exceed 20 lbs. p. s. i. and as said orifice is uncovered by the forward movement of piston 98ᵃ, there will be an increase in the outflow of oil which results in a greater pressure drop with a resulting decrease of pressure in the cylinder. Decrease in the outflow of oil through orifice 102 causes the oil pressure in cylinder 98 to increase until the thrust of ring 85 is balanced by the pressure in the cylinder. As the torque transmitted through ring 85 is decreased, said ring is shifted forwardly by the pressure of oil in the cylinder 98 to increase the exposed area of orifice 102, which increases the outflow of oil from cylinder 98 and reduces the pressure therein. It will thus be seen that the pressure of oil in the cylinder 98 is proportionate to the torque transmitted to the driven member 88. The variation of pressure in cylinder 98 is utilized in controlling the shift of the clutch device 80 to selectively complete said device to drive the low-speed gear 75 and the high-speed gear 76, as hereinafter described.

The clutching device 80 (Figs. 11 and 12) is adapted to alternately and positively drive the low-speed driving gear 75 and the high-speed driving gear 76 and comprises a pair of members 110 and 114 which are connected together for conjoint longitudinal movement by a ring 114ᵃ. Member 110 is provided with a hub which is slidably splined at 111 to the sleeve 81 so that it will be driven from sleeve 81 and the driven member 88. Clutch member 110 is provided with a series of external clutch teeth 112 which are longitudinally slidable into engagement with internal clutch-teeth 113 on the high-speed driving gear 76. Clutch member 114 is provided with external clutch teeth 116 which are slidable into engagement with internal clutch teeth 115 in the low-speed driving gear 75. The clutching device in its neutral position, as illustrated in Fig. 11, is uncoupled from both of the gears 75 and 76. Clutch members 110 and 114 (Figs. 13 and 14) are connected for limited relative rotative movement by external teeth 125 on clutch member 110 and internal teeth 126 on clutch member 114. Opposed springs 127 (Figs. 13 and 14) engage abutments 128 and 129 on rings fixedly secured to members 110 and 114, respectively, to yieldingly permit this limited relative rotative movement of said clutch members.

The clutch member 110 is provided with arms 122 which extend radially into an annular space in a shifter ring 136 by which the clutching-device is shiftable longitudinally between neutral and low speed and high speed driving positions. The non-rotatable clutch-shifter ring 136, formed of sections, is held against rotation and is provided with an annular groove 137 between the sections in which the fingers 122 are rotatable.

A balk ring 117 (Figs. 3, 9, 11 and 12) extends around and is adapted to slide axially on the teeth 112 and 116 of the clutch members 110 and 114, and is provided at its rear end with a conical face 118 for frictionally engaging the face of a conical socket 119 in the gear 76 and at its front end with a conical socket 120 for frictionally engaging a conical face 121 on gear 75. The arms 122 on the clutch member 110 extend through slots 123 in the balk ring (Fig. 3). A series of spring pressed balls 130 (Figs. 10 and 15) are individually confined in tubular guides 131 which are fixedly mounted in the rim of clutch member 110 and pressed against the inner periphery of the balk ring by springs 132. Balls 130 are adapted to successively seat in curved recesses 133 in the inner periphery of the balk ring when the clutch device is in its neutral position, in similar recesses 134 when the clutch device is in its low speed driving position, and in similar recesses 135 when the clutch device is in high speed driving position. When the clutch device 80 is in its neutral position, as illustrated in Fig. 11, and the shifter ring 136 and the clutch member are shifted forwardly, the spring pressed balls 130 will shift the balk ring 117 forwardly and initially cause the conical face 120 on the balk ring to frictionally engage the conical surface 121 on the gear 75 in advance of the engagement of clutch teeth 116 on clutch member 114 with the clutch teeth 115 in the gear 75. The continued forward movement of the clutch device 80 will cause the balls 130 to move forwardly into the recesses 134 so that the balls will, during the rearward movement of the clutch device, retract the balk ring. The recesses 133, 134, and 135 are elongated circumferentially sufficiently to maintain engagement with the balls 130 when slight relative rotative movement occurs between clutch member 110 and the balk ring. When the clutch device 80 is engaged with the gear 75 and the clutch members 110 and 114 are shifted rearwardly away from low-speed gear 75 and to engage the high-speed gear 76, the balls 130 seated in recesses 134, will yieldingly shift the balk ring 117 rearwardly and cause the conical face 118 on the balk ring to frictionally engage the conical surface 119 on the high-speed driving gear 76 before the clutch teeth 112 on clutch member 110 engage the clutch teeth 113 on the gear 76. The slots 123 in the balk ring through which the arms 122 on clutch member 110 extend are of sufficient width to permit independent longitudinal shift of the clutch members after the balk ring has engaged gears 75 and 76, respectively. The balk ring 117 is formed of sections joined at the front side of slots 123 for assembly around the clutch-members with the arms 122 extending through slots 123.

When the ring 136 is shifted forwardly to effect low speed driving, the balk ring will be shifted by spring-pressed balls 130 to cause its friction face 120 to engage the friction face 121 on low-speed gear 75 before the clutch-teeth 116 on clutch member 114 meet the clutch-teeth 115 on gear 75. When the ring 136 is shifted rearwardly to effect high-speed driving, the friction face 118 on the balk ring will engage the friction face 119 on the high-speed gear 76 before the clutch-teeth 112 on clutch-member 110 meet the clutch teeth 113 on gear 76.

Shoulder 123ᵃ (Fig. 3) are provided in the balk ring at one end of the slots 123, which are engaged by arms 122, to prevent the shift of coupling member 114 to bring its clutch teeth into mesh with the mating clutch teeth 115 in the gear 75 until relative rotation occurs between the arms 122 and the balk ring to release said shoulders. Shoulders 123[b] are provided in the balk ring in the opposite end of slots 123, which are engaged by the arms 122, to prevent the clutch member 110 from being shifted to slide clutch teeth 112 into engagement with the mating clutch teeth 113 in high-speed gear 76 until relative rotation occurs between the balk ring and said coupling member to release said shoulders. This relative rotation is produced by throttle control of the engine, as hereinafter more fully described, to permit shift of the arms 122 for delayed engagement of the clutch teeth on members 114 and 110 and the gears 75 and 76, respectively. After the clutch teeth 116 and 115 are engaged, the gear 75 will be positively driven from the clutch device, and after clutch teeth 112 and 113 are engaged, gear 76 will be positively driven from the clutch device.

The lay-shaft units 55 are driven at high and low speeds in relation to engine speed for driving the gears 45 and 47 which drive the propeller shafts in opposite directions, and gears 59 of the lay-shaft units are kept in constant mesh with the gear 75 for low speed driving, and gears 60 are kept in constant mesh with the gear 76 for high-speed driving, and as a result both of the gears 75 and 76 are rotated during low and high speed driving.

An overrunning clutch (Fig. 16) is provided between the sleeve 81 and the gear 75 and comprises peripheral cam surfaces 107 on the sleeve, and rollers 108 held in rings 109, and this makes it possible for the gear 75 to rotate at a higher speed than the sleeve 81 when the lay-shaft units 55 are being driven at high speed from the gear 76 and also makes it impossible to drive the sleeve 81 at a higher speed than that at which the gear 75 is driven.

Shifter-ring 136 is slidably supported by a pair of diametrically opposite studs 138 (Figs. 8, 9 and 17) which are slidably guided in brackets 139 which are fixed to the cross-wall 36[a] of casing section 36. A rock-shaft 140 is pivotally supported in a bracket 141 which is secured by bolts 142 to a cross-wall of the casing section 36. Arms 143 are fixedly secured to shaft 140 and are pivotally connected at 144 to the studs 138, respectively. An arm 145 is fixedly secured to the central portion of rock shaft 140 and is adapted to rock shaft 140 and arms 143 to slidably shift studs 138 and the ring 136 for shifting the clutching device between neutral and high-speed and low-speed driving positions. The ring 136 is shiftable rearwardly by arms 143 and 145 and fluid under pressure in a hydraulic motor which comprises a cylinder 147 (Fig. 17) and a piston 148 in said cylinder which has secured thereto a stem 149 to which is pivoted at 146 the lower end of arm 145. A spring 150 between piston 148 and the front end of cylinder 147 urges the piston rearwardly to shift arm 145 in counterclockwise direction for shifting the clutch device forwardly. Fluid under pressure in cylinder 147 is adapted to shift arm 145 in clockwise direction to shift shifter ring 136 rearwardly. Fluid to the cylinder 147 for shifting the piston 148 forwardly for shifting the shifter ring 136 and the clutch device 80 is controlled by a rotatable valve 158 (Figs. 8 and 17) which is provided with an arm 158[a] which is operable by the pilot through a suitable connection. A cross-port 158[b] in valve 158, when it is in one position, is adapted to supply oil from a duct 159 in the casing from the oil pressure system of the engine to a conduit 159[a] in the gear-casing, which is communicatively connected to the cylinder 147. When the valve 158 is rotated to an alternative position, oil will drain from cylinder 147 through conduit 159[a] and a channel 159[c] (Fig. 17) in valve 158 back into the gear case for drainage back to the oil reservoir. This exemplifies manually controlled combined hydraulic and spring means for shifting the arms 145 and 143 and shifter ring 136 to shift the clutch device 80.

Torque-controlled mechanism forming a control device is provided for locking the shifter-ring 136 and arms 143 and 145 to hold the clutch device 80 in neutral, high-speed and low-speed positions. This mechanism comprises a locking device or latch-lever 152 which is fulcrumed on the rear wall of cylinder 147, a finger or tooth 151 on the lower end of arm 145 which is adapted to be locked in three positions by notches in the latch-lever 152, a piston 154 slidable in a cylinder 155, a stem 153 on the piston, the outer end of which is pivotally connected to the lower end of the latch lever 152, and a spring 156 for shifting piston 154 in one direction. Tooth 151, when held in the central notch in latch-lever 152, will lock the arm 145 and the shifter-ring 136 in position to hold the clutch device 80 in its neutral position. Tooth 151, when held in the rear notch, will lock arm 145, ring 136 and clutch device 80 in low speed position and when held in the forward notch will hold arm 145, ring 136 and clutch device 80 in high speed driving position. A duct 155[a] in the gear-casing is communicatively connected with cylinder 155 and by a duct extending through any suitable portion of the casing to the cylinder 98 in which the pressure of oil is proportionate to the torque being transmitted through ring 85 to the driven member 88 and sleeve 81. The pressure of oil in the cylinder 155 forces the piston 154 rearwardly to hold latch-lever 152 in locking relation with arm 145 when appreciable torque is being transmitted by the ring 85 to the driven member 88, at which time the orifice 102 restricts the discharge of oil from cylinder 98 and maintains sufficient pressure therein to force piston 154 rearwardly and the lever 152 in its locking position. When no appreciable torque is being transmitted through ring 85, the pressure drop which results from the increased outflow of oil from cylinder 98 through orifice 102, reduces the pressure in cylinders 98 and 155 and permits spring 156 to shift the piston 154 forwardly and disengage lever 152 from the tooth 151 on arm 145. As a result, the shifting mechanism for clutch device 80 will be locked by the latch-lever 152 when substantial torque is being transmitted through the ring 85 and said mechanism will be automatically released when no substantial torque is being transmitted through the ring 85. This makes it possible for the operator, by slowing down the engine through control of the throttle, to release the lever 152 to unlock the shifting mechanism for shifting the clutch device 80 for driving the propeller shafts 30, 32 at high or low speeds. This exemplifies mechanism for locking the clutch device 80 against axial shift between neutral and high speed and low speed positions which is reasonably under control of the pilot by closing the throttle for the engine so that shifting can be accomplished only when substantially no torque is being delivered from the engine to the gearing for driving the propellers.

A pressure gauge (not shown), usually located in a convenient place, such as in the cockpit of the airplane, is communicatively connected to the cylinder 98 and may be calibrated to indicate the torque being transmitted through the ring 85 as reflected by the pressure of the oil in cylinder 98 which pressure is proportionate to the torque.

A hydraulically controlled governor 186 (Fig. 18) for varying the pitch of the propeller blades, is mounted on the top of the casing section 35 and may be of a construction well understood in the art. This governor is controlled by a valve 170 on the stem 149 of piston 148 which shifts the arm 145 and the shifter-ring 136 for the clutch device 80. Valve 170 is slidable in a cylinder 170^a fixed in the casing-section 36. The front end of the cylinder 170^a communicates with a conduit 171 in plate 36' and is connected to the oil pressure system of the engine. When the shifter-ring 136 and the clutch device 80 are in low speed driving position, at which time the piston stem 149 is in its rearward position, the valve 170 will uncover ports 172 in cylinder 170^a and cause oil under pressure to flow through a conduit 173 in the gear casing to the governor 186 for automatically controlling its operation to set the propeller blades at the desired pitch for low speed operation, as well understood in the art. When the clutch device 80 is in high speed driving position, piston-stem 149 will be in its forward position and valve 170 will be positioned forwardly of the ports 172 so that oil from the governor 186 and conduit 173 can drain through ports 172 and escape from the rear end of cylinder 170^a around the stem of the valve 170 which will eliminate the pressure of the fluid in governor 186.

The stem 149 of piston 148 in cylinder 147 is provided with an abutment 160 which is engaged by an arm 161 which is pivotally supported in the casing-section 36, and is held against abutment 160 by a rod 162 and a spring 163 applied to said rod. Rod 162 is slidably guided in a housing 164 which is secured in the casing-section 36. Rod 162 passes through packing 165 to prevent the escape of oil from the gear casing. Rod 162 carries a contacter 166 adapted to close stationary electric switches 167, 168 and 169 for controlling circuits for devices (not shown) for indicating low-speed, neutral, and high-speed positions of the clutching device 80.

A hollow drive shaft 176 (Figs. 3 and 18) is secured in the head 176^a which is fixed to the rear end of and is driven from the propeller shaft 30 and its rear end is journaled on a bearing 176^c in the casing-section 34. A gear 177 is secured on shaft 176 and meshes with a gear 178 to which is fixedly secured a gear 179 which is journaled on a bearing 180 supported from the casing-section 34. A pinion 181 which is integral with a hollow stub shaft 183, meshes with gear 179. Shaft 183 is splined to and drives a bevel gear 184 which meshes with a bevel gear 185 on a hollow vertical shaft 185^a which is coupled to drive the governor 186, as well understood in the art. A gear 177^a is secured to gear 177 for driving one of the accessories for the engine.

Gearing for driving the distributor and magneto for the engine and automatic hydraulic means controlled by pressure in the intake manifold *b* of the engine for timing said gearing are provided (Fig. 18). This gearing comprises a gear 211 which is integral with the main drive gear 38; a gear 210 meshing with gear 211; a shaft 214 provided with helical splines 213 which engage corresponding splines in the hub of gear 210; a gear 215 journaled on a bushing on the hub of gear 210; and opposite helical splines 216 on the shaft 214 which interfit with corresponding splines in the hub of gear 215. The distributor and magneto are driven by any suitable gearing from the gear 215. Axial movement of the shaft 214 through oppositely inclined helical splines 213 and 216 is adapted to rotate gear 215 relatively to gear 210 for timing the operation of the distributor and magneto. The front end of shaft 214 is journaled and slidable in a bearing 214' which is held in the wall 41 in the gear-casing. Shaft 214 is also slidably and rotatably mounted in a bearing sleeve 214^a which is fixedly secured in a hub on the cylinder 221 which is fixed to the casing-section 34.

The automatic means for shifting the shaft 214 to vary the timing of the distributor and magneto, comprises a piston 218 which is slidably mounted in a cylinder 221. The piston 218 has a hub in which is journaled a bushing 218' which is longitudinally movable with shaft 214. The piston 218 is slidably guided axially and held against rotation by the fixed bearing 214^a. This permits the piston 218 to slide in, and causes it to longitudinally shift shaft 214, while permitting the shaft to rotate. A spring 222 in cylinder 221, urges the piston 218 and the shaft 214 rearwardly toward a stop ring 222^a in the rear end of the cylinder 221. A nut 223 on the rear end of shaft 214 holds piston 218 on the bushing 218' in which the shaft 214 rotates. A chamber 224 for fluid is formed in the casing-section 34 and communicates with the rear end of the cylinder 221. The fluid under pressure in chamber 224 is adapted to shift shaft 214 forwardly against the force of spring 222 for causing the splines 213, 216 to rotate shaft 214 and gear 215 relatively to gear 210 for varying the timing of gear 215, and this pressure is controlled by the pressure in the intake manifold *b* of the engine, a Sylphon bellows 226 which is communicatively connected by a tube 225 with the manifold *b*, and a slide-valve 228 actuated by said bellows. The bellows 226 is mounted in a casing 227 which is fixed on the top of casing-section 34. The bellows 226 is in opposed relation with a bellows 226^a which compensates for altitude. The bellows 226 is operatively connected to the valve 228. The valve 228 controls the flow of oil in the engine pressure system in a duct 230 to a duct 231 which communicates with the chamber 224. The valve 228 also controls the flow of oil from conduit 231 to a conduit 232 to permit the oil from chamber 224 to drain back into the case responsively to the variation in the pressure in the manifold. A manually shiftable rotary valve 229 is included in the conduit 230 to cut off the oil pressure to the conduit 231 and chamber 224 when automatic time regulation is not desired. The construction described exemplifies hydraulic timing control for the gear 215 which is controlled by the pressure in the intake manifold of the engine.

In the operation of contra-rotating propellers, the passing of the blades results in an aerodynamic interference effect which tends to produce torsional vibration in the propeller shafts. The invention provides for isolating and damping such effects on the gear 45 which is fixed to the propeller shaft 30, is driven by the body 49 of the internal gear 47 which is fixed to shaft 32 (Figs. 20–26) by series of inertia members for damping torsional vibrations of said gears, respectively. These dampers are tuned to the frequency at which the blades of the contra-rotation propellers pass one another and have a fixed value depending upon the number of propeller blades used, for example, with propellers provided with three blades sixth order of dampers are used. These inertia members rotate with the propellers and act to isolate from the engine and the gearing between the engine and gears 47 and 45 any torsional vibration caused by the passing blades irrespective of the speed of rotation of the propellers after they are once properly tuned.

Inertia members (Figs. 24 to 26) are carried by the body of the gear 45 which is splined to the propeller shaft 30 and by the body 49 of gear 47 (Figs. 20 to 23).

A series of six segmentally shaped inertia members 235 is confined between the body of gear 45 and a ring 236 which is secured by bolts 237 to said body. Each member 235 is provided adjacent each end with a cylindrical hole 238 which extends therethrough. A roller 239 extends through each of the holes 238 and is provided with a head which abuts against the outer face of ring 236, and a plate 240 bolted to ring 236 engages the heads of the two rollers in each member 235 to hold the rollers against axial movement. Each roller 239 also extends through longitudinally aligned cylindrical holes 241 in the body of wheel 45 and a cylindrical hole 238 in the inertia member 235. The diameter of the holes 238 and 241 is equal and greater than the diameter of rollers 239. The axes of the rollers 239, holes 238 and 241 for each member 235 are equidistantly spaced apart. During rotation of gear 45, the inertia members 235 will be subjected to centrifugal force which will cause engagement between rollers 239 and holes 238 in said members at the points nearest the axis of rotation of said gear, and the engagement of rollers 239 with holes 241 at the points farthest from the axis of rotation of said gear. The forces required to move said members toward said axis against centrifugal force and inertia and the rolling action between the opposite portions of rollers 239 and the oppositely curved portions of the holes 238 and 241, damp the torsional vibrations in the gear 45.

A series of six segmentally shaped inertia members 244 is confined between the body 49 of gear 47 and a flange 245 which is integral with said body. Each member 244 is provided adjacent each end with a cylindrical hole 247 which extends therethrough. A roller 248 extends through each of holes 247 and is provided with a head which abuts against the outer face of flange 245, and a plate 249 bolted to flange 245 engages the head of the two rollers in each member 244 to hold the rollers against axial movement. Each roller 248 extends through longitudinally aligned cylindrical holes 250 in the body of wheel 49 and flange 245 and through a cylindrical hole 247 in the inertia member 244. The diameter of the holes 247 and 250 is equal and greater than the diameter of roller 248. The axes of the rollers 248, holes 247 and 250 for each member 244 are equidistantly spaced apart. During rotation of the body of gear 47, which is secured to the propeller shaft 32, inertia members 244 will be subjected to centrifugal force which will cause engagement between rollers 248 and the holes 247 in said members at points nearest the axis of rotation of body 49 and the engagement of rollers 248 with holes 250 in body 49 and flange 245 at points farthest from the axis of rotation of said body. Members 244 move against centrifugal force and inertia and by the rolling action between the opposite portions of rollers 248 and the opposite portions of holes 247 and 250, damp the torsional vibration in the body of gear 47.

These vibration dampers which are applied to the gears fixed to the contra-rotation propeller shafts, dampen the torsional vibration which results from the aerodynamic interference effect of the oppositely rotating propeller blades and isolates them from the engine and the gearing between the engine and the gearing on the propeller shafts.

Lubricating oil is supplied from the engine pressure system through a duct 188 (Fig. 18) and ports 188' to an annular conduit 189 (Figs. 3 and 18) which is formed between a tube 190 and the hollow shaft 176. From annular conduit 189 oil flows through radial ports 191 to an annular conduit 192 which is formed between the inner periphery of propeller shaft 30 and a tube 193 which is concentric with and secured in said shaft. Some of this oil flows through a port 194 (Fig. 3) to the bearing 53 between shaft 30 and the hub of gear 38 and a port 194' to the bearing 186 for the sleeve 81. Oil also flows (Fig. 27) forwardly to radial ducts 196 to the bearing 54 between the contra-rotating shafts 32 and 30. The bearing 54 is secured in the shaft 32 by a pin 197. Leakage oil can flow from the rear end of bearing 54 through the annular space between shafts 30 and 32 to the ball-bearing 52 from which it drains into the gear casing. Leakage oil from the front of bearing 54 can flow to an annular groove 200 in the periphery of shaft 30 which is connected by radial ports 201 to an annular groove 202 in a dam or head 203 which is fixed in shaft 30, into the interior of tube 193 from which it returns to the gear-casing through tubes 193 and 190. An oil seal is provided at the front of bearing 54 for preventing the escape of oil outwardly from said bearing, and comprises a sleeve 204 around shaft 30 and movable therewith; a carbon sealing ring 206 slidably confined in sleeve 204 and engaging the front end of bearing 54; a plurality of elastic gaskets 208 which are V-shaped in cross-section and extend around shaft 30 and in sleeve 204; a follower ring 205 engaging the foremost gasket 208; and springs 209 between the front end of sleeve 204 and the follower which press the follower rearwardly to compress the gaskets against the sealing ring 206. The vertex of rings 208 and the contiguous surfaces on sealing ring 206 and follower ring 205 extend forwardly so that the gaskets 208 are tightly pressed against shaft 30 to prevent the escape or loss of oil from groove 200 in said shaft.

The operation will be as follows: assuming the engine is supplying power to the main drive gear 38 through gears 43 and 44 which are driven by the crank shaft of the engine, and the clutching device 80 is in its forward position, the driving member 82 which is splined to the hub of gear 38 will be driven; the driving member 82 will drive the torque responsive ring through balls 87 between said ring and said driving member; the ring 85 will drive the driven member 88 through the balls 92; the driven member 88 will drive sleeve 81 and the clutch members 110 and 114 of the slidable clutching device 80; clutch teeth 116 on clutch member 114 will be engaged with clutch teeth 115 to positively drive the low speed gear 75; the propeller shafts 30 and 32 will be driven through the lay-shaft units 55 and gear 75; gear 215 (Fig. 18) will be driven to drive the distributor and magneto from gears 210 and 211; the gearing for driving the governor 186 will be driven from shaft 176 which is driven by propeller shaft 30; valve 158 will be set in position to drain the oil from cylinder 147 so that spring 150 will hold piston 148 and the clutch shifting device in its low speed driving position; the torque transmitted through ring 85 will control the pressure in cylinders 98 and 155 to hold latch-lever 152 so the arm 145 of the clutch shifting mechanism will be locked in the rear notch in said lever; friction face 120 of balk ring 117 will be engaged with the friction surface 121 on gear 75; valve 170 will be in its forward position to deliver oil through conduit 173 to the governor 186; and the abutment 160 on piston stem 149 will control the spring pressed movement of arm 161 and rod 162 to cause contactor 166 to close switch 169 for indicating to the pilot that the clutch device is positioned for driving the propeller shafts at the low speed ratio. The gear 75 will be positively driven from sleeve 81 through clutch teeth 115 and 116 to drive the lay-shaft units 55. Propeller shaft 30 will then be driven in one direction through gear 45 and gears 59 of the lay-shaft units 55 and propeller shaft 32 will be driven through gear 47 and the gears 62 on said lay-shaft units. The propellers will then be driven in opposite directions from the engine at the low speed ratio.

When the pilot desires to shift the clutch device 80 for driving the propeller shafts at the high speed ratio, he will first rotate valve 158 to deliver oil under pressure from conduit 159 through valve port 158^b and conduit 159^a to the rear end of cylinder 147 which will tend to shift the piston 148 forwardly. The arm 145 then remains locked in the rear notch of latch-lever 152 so that the clutch device 80 remains in low speed driving position. Next, the pilot will close the engine throttle until no appreciable torque is transmitted through the ring 85 which will cause the pressure in communicating cylinders 98 and 155 to be reduced so that the spring 156 will shift piston 154 forwardly to swing lever 152 to release arm 145 of the clutch shifting mechanism. The pressure in cylinder 147 will then act to move the piston 148 forwardly and shift arm 145 to urge the shifter-ring 136 rearwardly, and the friction face 118 on the balk ring will engage the friction face 119 on the gear 76. The arms 122 will move rearwardly until they are arrested by shoulders 123^b on the balk ring to prevent the clutch teeth 112 on clutch member 110 from engaging the clutch teeth 113 on gear 76. The friction faces 118 and 119 will urge gear 76 toward synchronous speed with the clutch members 110 and 114. The clutch members will then be held in neutral position while this occurs, and the contactor 166 will close switch 168 (Fig. 17) to indicate the neutral position of the clutch device. The gear 76 and the balk ring 117 are then usually rotated slower than the clutch members. As the engine slows down the clutch members will slow down until they attain a speed slightly lower than the speed of the gear member 76. When this occurs, the arms 122 will rotate slightly relatively to the balk ring 117 in a direction opposite to the rotation of said arms. The shoulders 123^b will then be moved out of the path of arms 122 and permit the rearward movement of the clutch member 110 and its clutch teeth 112 to slide into engagement with the teeth 113 on gear 76 as the result of the pressure of fluid in cylinder 147 and the forward movement of piston 148, for positively driving the propeller shafts at high speed ratio from the engine through high speed gear 76. The rearward movement of piston 148, on the completion of the shift of the clutch and the abutment 160, will cause contactor 166 to be shifted to close switch 167 for indicating to the operator that the shift to high speed ratio has been completed. The clutch will be held in high speed position while valve 158 remains set to supply fluid under pressure to cylinder 147. Valve 170 will then permit fluid to be drained through conduit 173 from the governor 186. The throttle for the engine will then be opened to drive the propellers at high speed which will increase the torque transmitted through ring 85 and cause an increase of pressure in cylinders 98 and 155. This increase of pressure will shift piston 154 to rock latch-lever 152 into position to lock arm 145 of the clutch shifting mechanism in high speed driving relation. While the gear 76 is positively driving the propellers through the lay-shaft units 55, the overrunning clutch will permit the low-speed driving gear 75 to rotate faster than the clutch member and sleeve 81.

When the pilot desires to shift from high speed ratio to low speed ratio, the valve 158 is rotated into position to drain the oil from cylinder 147 to permit the spring 150 to be released for shifting piston 148 rearwardly, but the arm 145 is then locked in the front notch of latch lever 152 because of the torque being transmitted through the ring 85. The pilot will next, by the throttle control, slow down the engine so that there will be no appreciable torque transmitted through the ring 85 and oil from the engine pressure system will then flow from the cylinder 98 to reduce the pressure therein and in the cylinder 155, which is communicatively connected to cylinder 98, sufficiently to permit the spring 156 to shift piston 154 forwardly and disengage latch-lever 152 from the arm 145 of the shifting mechanism for the clutch device 80. When the arm 145 is released by latch-lever 152, the force of spring 150 in cylinder 147 will be applied to shift arm 145 rearwardly. The balk ring 117 will then be shifted forwardly by the spring pressed balls 130 which are initially seated in recesses 134 until the conical face 120 on clutch member 114 engages the conical face 121 on the gear 75 which will tend to drive said gear and the balk-ring at synchronous speed. The engagement of friction faces 120 and 121 will cause gear 75 to be rotated faster than the clutch members 110 and 114 and cause the shoulders 123^a to arrest arms 122 and the clutch members in neutral position. The contactor 166 will then close switch 168 for indicating the shift of the clutch device to its neutral position. If the clutch teeth 115 and 116 were permitted to engage at this time, they would tend to wedge together upon decrease of the torque being supplied to the propellers due to the "springing" effect of the rollers 108 and the cam surfaces 107. The pilot will next momentarily open the engine throttle to increase its speed which will normally tend to drive the propellers through the overrunning clutch rollers 108, and then rotation of the clutch members necessary to cause the rollers 108 to drive gear 75 causes arms 122 to rotate out of the path of shoulders 123^a. At this time the clutch members and gear 75 will be rotated at substantially the same speed and the contactor

166 will close switch 168 to indicate to the pilot that the clutch device is in its neutral position. To complete the shift for low speed driving, the pilot again closes the throttle so no appreciable torque will be transmitted through the ring 85 and the pressure in cylinders 98 and 155 will be reduced to permit spring 156 to shift latch-lever 152 to release arm 145, whereupon the spring 150 in cylinder 147 will shift arm 145 and the clutch members to slide clutch teeth 116 and 115 into engagement for driving the propellers at low speed ratio. This shift will be made without substantial torque on the rollers 108 and the gearing will then be operated at the low-speed ratio as before described. The contactor 166 will then close the switch 169 for indicating the clutch device has been shifted to low speed driving position. As the engine throttle is opened to drive the propellers, the torque transmitted through ring 85 will cause an increase of pressure in cylinders 98 and 155 to lock the clutch device in its low-speed driving position.

After the low speed shift is completed, the pilot will then open the throttle the desired amount for engine operation. As torque is again supplied to the propellers, one set of springs 127 will compress and relative movement will occur between clutch member 110 and 114 until teeth 125 engage against teeth 126 to form a positive drive between members 110 and 114. We prefer to have the torque in low-speed driving transmitted through the positive engagement of teeth 125 and 126 and 115 and 116 and in such event it is necessary that the back lash or play between the teeth 125 and 126 be less (in terms of angular travel) than the angular travel necessary to cause the rollers 108 of the overdrive clutch to lock sleeve member 81 with gear 75. In some driving instances, however, it may be desirable to have the torque transmitted through the rollers 108 and to utilize the teeth 115, 116, 125 and 126 as a safety device in event of failure of the overdrive clutch. In the case of the latter operation the angular travel to engage rollers 108 would be less than that required for engagement of teeth 125 and 126.

If the propellers should start to speed up so that the shift to low-speed could not be completed, the pilot can again shift the clutch to high-speed position as before described. The friction driving faces 118 and 119 have sufficient holding force to drive the engine from the propeller so that the clutch teeth 115, 116 will be driven sufficiently close to synchronous speed to permit said clutch teeth to mate and slide into engagement.

The friction surfaces between the high speed drive gear 76 and the friction surface 118 on the balk ring are preferably of sufficient capacity to prevent any "windmilling" of the propeller blades during the shifting period from increasing the speed of said gear with such rapidity, that the clutch teeth 112 and 113 cannot be engaged for positive drive of the engine by the propellers.

The invention exemplifies two speed transmission gearing for driving contra-rotational propellers through a series of lay-shafts which remain in mesh with high and low speed gears and a clutch for selectively controlling the high speed and low speed operation of the propellers relatively to the engine.

The invention also exemplifies a slidable toothed clutch for drivably engaging the high and low speed driving gears and means for controlling the clutch to prevent knocking of the clutch teeth in effecting speed changes; the invention also exemplifies a clutch which is torque controlled to control the engagement of the clutch with the high and low speed gears.

The invention also exemplifies a clutch device with a balk ring which retards the engagement of the clutch with the gears and is controlled by a relative speed between the clutch and the gear; the invention also exemplifies hydraulic and torque controlled mechanism for controlling the shifting of the clutch.

The invention also exemplifies a lay-shaft unit which can readily be assembled with the high and low speed gears for uniformly loading the teeth on the gears. An application filed by us November 2, 1949, Serial No. 125,058 describes and claims the Lay Shaft Unit as divisional subject matter of this application.

The invention also exemplifies high and low speed gearing which is provided with vibration dampers for isolating the vibrations due to blade passing from the variable speed gearing and the engine.

The invention also exemplifies improved automatic means for controlling the timing of the distributor and generator.

The invention also exemplifies an outboard bearing between the contra-rotating propeller shaft and means for supplying oil through the inner shaft to said bearing.

The invention is not to be understood as restricted to the details set forth since these may be modified within the scope of the appended claims without departing from the spirit and scope of the invention.

Having thus described the invention, what we claim as new and desire to secure by Letters Patent is:

1. Engine driven transmission gearing for driving contra-rotational coaxial propeller shafts comprising coaxial gears, each gear being provided with clutch teeth, an engine driven slidable clutch between and coaxial with said gears and provided with clutch teeth for slidably engaging the clutch teeth on the gears, respectively, a balk-ring axially slidable on the clutch and provided with friction faces at its ends for respectively engaging the coaxial gears, and with slots, the clutch comprising a pair of members movable together longitudinally, spring and stop means between the members for permitting limited relative rotation of the members, and means for axially shifting the clutch and the balk-ring, an engine-driven overrunning clutch for driving one of said gears, and means controlled by the relative speed of the clutch and the gears for controlling the engagement of the clutch-teeth.

2. Engine driven transmission gearing for driving contra-rotational coaxial propeller shafts, comprising coaxial gears, each gear being provided with clutch teeth, an engine driven slidable clutch between, coaxial with said gears and provided with clutch teeth for slidably engaging the clutch teeth on the gears, respectively, a balk-ring axially slidable on the clutch and provided with friction faces at its ends for respectively engaging the coaxial gears, and with slots, the clutch comprising a pair of members movable together longitudinally, spring and stop means between the members for permitting limited relative rotation of the members, and means for axially shifting the clutch and the balk-ring, an engine-driven overrunning clutch

for driving one of said gears, means controlled by the relative speed of the clutch and the gears for controlling the engagement of the clutch teeth, and torque controlled means for locking the clutch in low-speed, neutral and high-speed driving positions.

3. Engine-driven transmission gearing for driving contra-rotational propeller shafts, comprising coaxial gears, each provided with clutch teeth, an engine driven slidable clutch provided with teeth for slidably engaging the clutch teeth on the gears, respectively, combined spring and fluid pressure means for slidably shifting the clutch to alternately engage the clutch teeth on said gears, and hydraulic means responsive to variations in the torque of the engine for locking said spring and fluid pressure means to hold the clutch into engagement with said gears, respectively, and in neutral position.

4. Engine driven transmission gearing for driving contra-rotational propeller shafts, comprising coaxial gears, each provided with clutch teeth, an engine driven slidable clutch provided with teeth for slidably engaging the clutch teeth on the gears, respectively, combined spring and fluid pressure means for slidably shifting the clutch to alternately engage the clutch teeth on said gears, torque controlled hydraulic means for locking the spring and fluid pressure means to hold the clutch in engagement with said gears, respectively, and in a neutral position, and means controlled by said spring and fluid pressure means for indicating the positions of the clutch.

5. Engine driven transmission gearing for driving contra-rotational coaxial propeller shafts, comprising coaxial gears of different pitch diameters, each provided with clutch teeth, an axially slidable clutch between said gears, coaxial with said gears and provided with teeth for slidably engaging the clutch teeth on the gears, respectively, a balk-ring around and slidable on the clutch and provided with friction faces at its ends for respectively engaging the high and low speed gears, means for slidably shifting the clutch and for shifting the balk-ring for engaging its friction faces with said gears in advance of the engagement of the clutch teeth, the balk-ring being limitedly rotatable relatively to the clutch, and means on the balk-ring for holding the clutch in a neutral position and preventing engagement of the clutch with the gears, an engine-driven overrunning clutch for driving one of said gears, the balk-ring being releasable by varying its relative rotation to the clutch to permit the clutch teeth to engage the clutch teeth on the gears, respectively.

6. Engine driven transmission gearing for driving contra-rotational propeller shafts, comprising coaxial gears, each provided with clutch teeth, a driving sleeve, a clutch axially slidable on the sleeve coaxial with said shafts and including relatively rotatable members provided with teeth for slidably engaging the clutch teeth on the gears, respectively, a balk-ring around and slidable on the clutch and provided with friction faces at its ends for respectively engaging the gears, means for slidably shifting the clutch, means for axially shifting the balk-ring by the clutch-shifting means for engaging its friction faces with said gears in advance of the engagement of the clutch teeth, the balk-ring being slidable and rotatable relatively to the clutch, stop-means for the clutch on the balk-ring for preventing engagement of the clutch with the gears, an overrunning clutch between the sleeve and one of the coaxial gears, the balk-ring being rotatable to release the clutch and permit its clutch teeth to engage the clutch teeth on the gears by relative movement of the clutch and the gears, respectively.

7. Engine driven transmission gearing for driving contra-rotational coaxial propeller shafts, comprising coaxial gears, each gear being provided with clutch teeth, an engine driven slidable clutch between and coaxial with said gears and provided with clutch teeth for slidably engaging the clutch teeth on the gears, respectively, a balk-ring around and axially slidable on the clutch and provided with friction faces at its ends for respectively engaging the coaxial gears, and with slots, means on the clutch extending through and movable relatively to the ring axially and rotatively in the slots, means in the slots for arresting the axial movement of the clutch after the balk-ring is fixedly engaged with the gears and before the teeth on the clutch engage the gears, means for axially shifting the means extending through the ring to shift the clutch, the balk ring being controlled by the relative speed of the clutch and the gears when the ring is frictionally engaged with the gears for disengaging the arresting means to permit the teeth on the clutch to engage the clutch teeth on the gears, and torque controlled means for locking the clutch engaged with the coaxial gears respectively.

8. Engine driven transmission gearing for driving contra-rotational coaxial propeller shafts, comprising coaxial gears, each gear being provided with clutch teeth, an engine driven slidable clutch between and coaxial with said gears and provided with clutch teeth for slidably engaging the clutch teeth on the gears, respectively, a balk-ring around and axially slidable on the clutch and provided with friction faces at its ends for respectively engaging the coaxial gears and movable into a neutral position, and with slots, means on the clutch extending through and movable relatively to the ring axially and rotatively in the slots, means in the slots for arresting the axial movement of the clutch after the balk-ring is frictionally engaged with the gears and before the teeth on the clutch engage the gears, means for axially shifting the means extending through the ring to shift the clutch, the balk-ring being controlled by the relative speed of the clutch and the gears when the ring is frictionally engaged with the gears for disengaging the arresting means to permit the teeth on the clutch to engage the clutch teeth on the gears, and torque controlled means for locking the clutch engaged with the coaxial gears respectively and in an intermediate neutral position.

9. Transmission gearing comprising coaxial driving and driven members, a ring longitudinally movable relatively to and surrounding said members, driving means between the ring and one of said members which permits longitudinal sliding movement of the ring, the inner periphery of the ring and the outer periphery of the other member being provided with mating longitudinally extending helically inclined recesses, balls in said mating recesses connecting said other member and the ring for longitudinal movement of the ring produced by torque, hydraulic means for shifting the ring longitudinally, and valve-means for the hydraulic means, responsive to the longitudinal movement of the ring.

10. Transmission gearing comprising coaxial driving and driven members disposed side-by-side, a ring longitudinally movable relatively to and surrounding said members, driving means between the ring and one of said members which permits longitudinal sliding movement of the ring, the inner periphery of the ring and the outer periphery of the other member being provided with mating annular series of extending open-ended helical recesses, an annular series of balls in said recesses connecting said other member and the ring for torque responsive longitudinal movement of the ring, yieldable means for holding the balls in the helical recesses, hydraulic means for shifting the ring longitudinally, and valve-means for the hydraulic means, responsive to the longitudinal movement of the ring.

11. Transmission gearing comprising coaxial driving and driven members disposed side-by-side, a ring longitudinally movable relatively to and surrounding said members, the inner periphery of the ring and the outer periphery of one of said members being provided with an annular series of longitudinally extending mating recesses, an annular series of balls in said recesses for slidably connecting the ring and said last named member, the inner periphery of the ring and the outer periphery of the other member being provided with mating helically extending recesses, an annular series of balls in said mating helical recesses connecting said other member and the ring for longitudinal movement of the ring produced by torque, and a control device for the gearing and means operable responsive to the torque transmitted to the driven member for operating said control device.

12. Transmission gearing comprising coaxial driving and driven members disposed side-by-side, a ring longitudinally movable relatively to and surrounding said members, the inner periphery of the ring and the outer periphery of one of said members being provided with an annular series of longitudinally extending mating recesses parallel to the axis of said members, an annular series of balls in said recesses for slidably connecting the ring and said last named member, the inner periphery of the ring and the outer periphery of the other member being provided with mating helically extending recesses, an annular series of balls in said mating helical recesses connecting said other member and the ring for longitudinal movement of the ring produced by torque, and a control device for the gearing, and valve means operable by the longitudinal movement of the ring, and responsive to the torque transmitted to the driven member for operating the control device.

13. Transmission gearing comprising coaxial driving and driven members, a ring longitudinally movable relatively to and surrounding said members, the inner periphery of the ring and the outer periphery of one of said members being provided with a series of longitudinally extending mating recesses, a series of balls in said recesses for slidably connecting the ring and said last named member, the inner periphery of the ring and the outer periphery of the other member being provided with mating annular series of helically extending open-ended recesses, balls in said mating helical recesses connecting said other member and the ring for longitudinal movement of the ring produced by torque, yieldable means for holding the balls in the helical recesses, and a hydraulic device operable by oil under pressure, a locking device for the gearing controlled by the hydraulic device, and valve means operable responsive to the torque transmitted to the driven member for varying the pressure in said hydraulic device for operating said locking device.

14. Transmission gearing comprising coaxial driving and driven members, a ring longitudinally movable relatively to and surrounding said members, the inner periphery of the ring and the outer periphery of one of said members being provided with a series of longitudinally extending open-ended mating recesses, a series of balls in said recesses for slidably connecting the ring and said last named member, means for holding the balls longitudinally in said recesses, the inner periphery of the ring and the outer periphery of the other member being provided with mating helically extending open-ended recesses, balls in said mating helical recesses connecting said other member and the ring for longitudinal movement of the ring produced by torque, yieldable means for holding the balls in the helical recesses, and a hydraulic device operable by oil under pressure, a locking device for the gearing controlled by the hydraulic device, and valve means operable by the longitudinal movement of the ring responsive to the torque transmitted to the driven member for varying the pressure in said hydraulic device for operating said locking device.

SAMUEL K. HOFFMAN.
HAZEL CAMINEZ,
Executrix of the Estate of Harold Caminez, Deceased.

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