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STUDY AND TEST TO CONFIRM AUTOMOBILE DRIVETRAIN COMPONENTS TO IMPROVE FUEL ECONOMY

VOLUME 1

HISTORY OF THE AUTOMOBILE TRANSMISSION IN THE UNITED STATES

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16. Abstract

Since the earliest days of the motor car, engineers have tinkered together various means of connecting the engine to the ground. While the wheel rapidly became the norm for the ground/vehicle interface, many different engine/wheel coupling techniques have been, and still are, being tried. The history of the automotive transmission provides an interesting insight into the development of the automobile and its drivetrain, and illustrates how the marketplace has and will govern the design of the motor car. This report explores the design from the earliest manual sliding gear transmission up through the various forms of automotive gear boxes with emphasis on the development of the automatic transmission in the United States.

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PREFACE

This interim report, prepared by Arthur D. Little, Inc., for the U. S. Department of Transportation, presents a study of U. S. Automatic Transmissions - their history and design philosophy. This report consists of two volumes.

Volume I presents a history of the automobile transmission in the United States, with particular emphasis on the family tree of the U. S. automatic transmissions. Volume I also contains, in tabular form, a description of the 1970-1975 engine/transmission/rear axle ratio/vehicle size and weight combinations available in all U. S. cars of this time period.

Volume II is a handbook-like narrative on the selection of transmissions and rear axle gear ratios, shift points, shift quality, and other pertinent drivetrain design parameters.

Arthur D. Little, Inc. wishes to acknowledge and thank Mr. H. Gould and Mr. R. Colello of the Department of Transportation, Transportation Systems Center for their guidance and assistance in the preparation of these volumes. ADL also wishes to gratefully acknowledge Chilton Company for their permission to use copyrighted material from the magazine Automotive Industries from which much of Volume I, has been drawn.

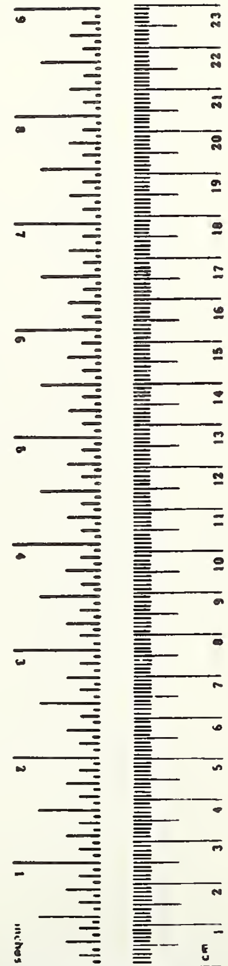
We also wish to thank Mr. John Ivey of Borg-Warner for his helpful suggestions in refining this report.

The work on this project was completed under the sponsorship of the U.S. Department of Transportation, National Highway Traffic Safety Administration's Automotive Fuel Economy Research and Analysis Program of the Transportation Systems Center.

METRIC CONVERSION FACTORS

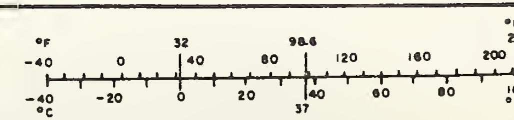
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	36	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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1. EARLY DEVELOPMENTS

1.1 THE GASOLINE VEHICLE'S LIMITATIONS

The history of the automobile transmission in the United States is a record of perpetual movement toward an instrument that would be silent, easy to control, inexpensive to produce, reliable, and achieve a maximum efficiency while matching engine output to vehicle and driver demands.

Many successes and failures, (mostly the latter) contributed to the trial and error process of developing a technology that would place a measure of control in the driver's hands.

The modern day automatic transmission, with its planetary gear set and complex array of controls, must be considered one of the feats of modern technology. But throughout the history of the industry in the United States and Europe a progression of devices has been invented, and improved upon, to make the once balky, gear crashing automobile more operable. The gradual improvements in the transmission broadened the appeal of the automobile and along with the self-starter, helped prepare a mass market.

The producers of steam and electrical vehicles around the turn of the century were able to tout important advantages over the gasoline engine in regard to transmission of torque. By being directly connected to the driving wheels these power plants could in effect accomplish a ratio change by a simple change in engine output. Actually, there was no true 'ratio change' but a boost in engine output.

In a steam car, for example, as speed fell the power output could be maintained by increasing the average steam pressure throughout the stroke. Constant power was thus available throughout a wide range of engine speeds, and car speed could be determined for any grade on the basis of constant power.

In comparison, grade climbing for a gasoline car could be accomplished through gear reductions, and the necessary ratio calculated by figuring the degree of grade that must be ascended. It was found early on that each ratio was exactly suited to only one particular grade for a particular set of vehicle capabilities.

Level ground top speed also had to be compromised in early vehicles if any degree of grade climbing was to be provided.

The gasoline engine's major drawback was, and still is, limited flexibility, though the improvement has been monumental since the turn of the century and the advent of electronic controls in the coming years will add the last possible increments. Early researchers found no way to make the engine conform with the overall requirements of the car's operation. No engine could be made flexible enough to develop high torque over the whole speed range, as could the steam engine.

The speed ratios in the transmission still are dictated on overall engine performance, and take into account engine speed, car weight and desired performance on a grade and over level ground. What has changed is that the mechanics of the transmission itself, gear shapes and materials, optimum gear ratios, etc., have become optimized through experience.

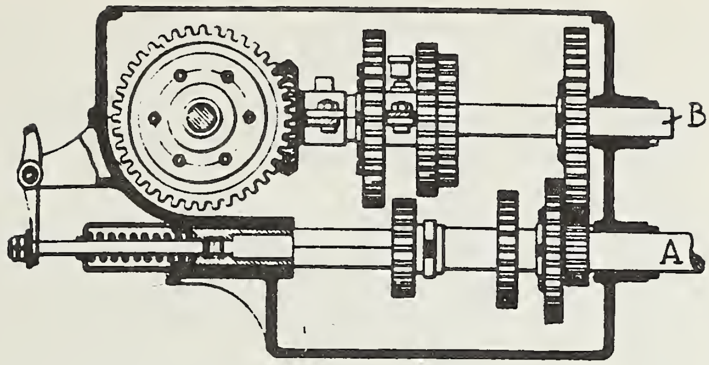
1.2 SLIDING GEAR TRANSMISSIONS

The first successful innovation in a transmission specifically aimed at solving some of the problems of the inflexible gasoline engine was the sliding gear transmission invented by Levassor, engineer for the firm Panhard and Levassor, of France (Figure 1). It came into nearly universal use in the United States by the 1920's for cars, trucks and buses. It consisted of two parallel shafts mounted on bearings. The primary shaft, connected to the clutch, was squared and carried three gears. Three other gears were mounted on the secondary shaft. The driver could shift the position of the primary shaft to bring the corresponding gears into mesh.

The disadvantage with this configuration was unncessary power loss, noise and wear at high speed. Improvements were subsequently made by Louis Renault. First, the gears were rolled into mesh, rather than slid. Also, Renault divided the primary shaft into two parts, the rear, driven member piloted within the forward section. High became direct drive by having the two halves locked together by means of clutch jaws formed integral with a gear on each of the halves (Figure 2). These gears slid into engagement to create direct drive. The progressive three-speed transmission with direct drive, as the Renault device was called, became obsolete in the early part of the century.

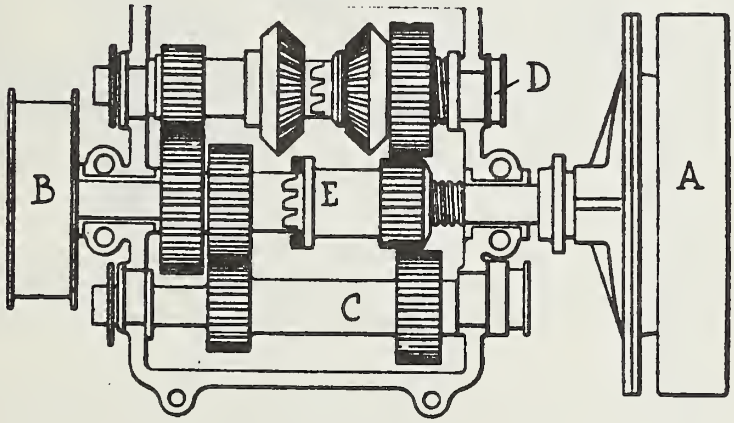
The next improvements in the sliding gear transmission proceeded from the necessity to add a fourth gear due to increases in vehicle speeds. The addition of the fourth speed was indicated in order to enable the driver to operate the engine nearer its peak efficiency, and to reduce the large gaps in three-speed gear sets.

To add a fourth gear to a progressive-type transmission would result in an inordinately long shaft to prevent the gears on each shaft from conflicting. A long shaft would flex under high tooth loads, resulting in noisy operation.



Source: *Automotive Industries,* February 15, 1917, p. 387.

FIGURE 1 DIAGRAM OF ORIGINAL PANHARD TRANSMISSION IN WHICH ALL SPEEDS WERE INDIRECT. THIS IS GENERALLY KNOWN AS THE STRAIGHT-THROUGH, OR THE PROGRESSIVE TYPE OF GEAR



Source: *Automotive Industries,* February 15, 1917, p. 387.

FIGURE 2 DIAGRAM OF ORIGINAL RENAULT DIRECT DRIVE TRANSMISSION. IN THIS THE COUNTERSHAFTS ARE ENGAGED IN THE SAME WAY AS THE BACK GEAR TO A LATHE

Wilhelm Maybach, of Daimler Motor Company, solved this problem by placing two sliding sets in non-direct drive transmission. This selective control was later applied to direct drive. The selective sliding gear principle had the advantage that a change from one speed to another could be accomplished without passing through intermediary gears (Figure 3).

While these developments improved the operation of the transmission or 'change speed gear', improvements continued in the engine and elsewhere to provide more flexibility and ease of shifting. The automatic carburetor awarded the engine a new degree of flexibility. Changes in gear detail allowed gears to mesh more easily. Increased vehicle speeds mandated a fourth gear and broader power curves partly eliminated the task of downshifting on moderate grades.

1.3 THE PLANETARY GEAR TRANSMISSION

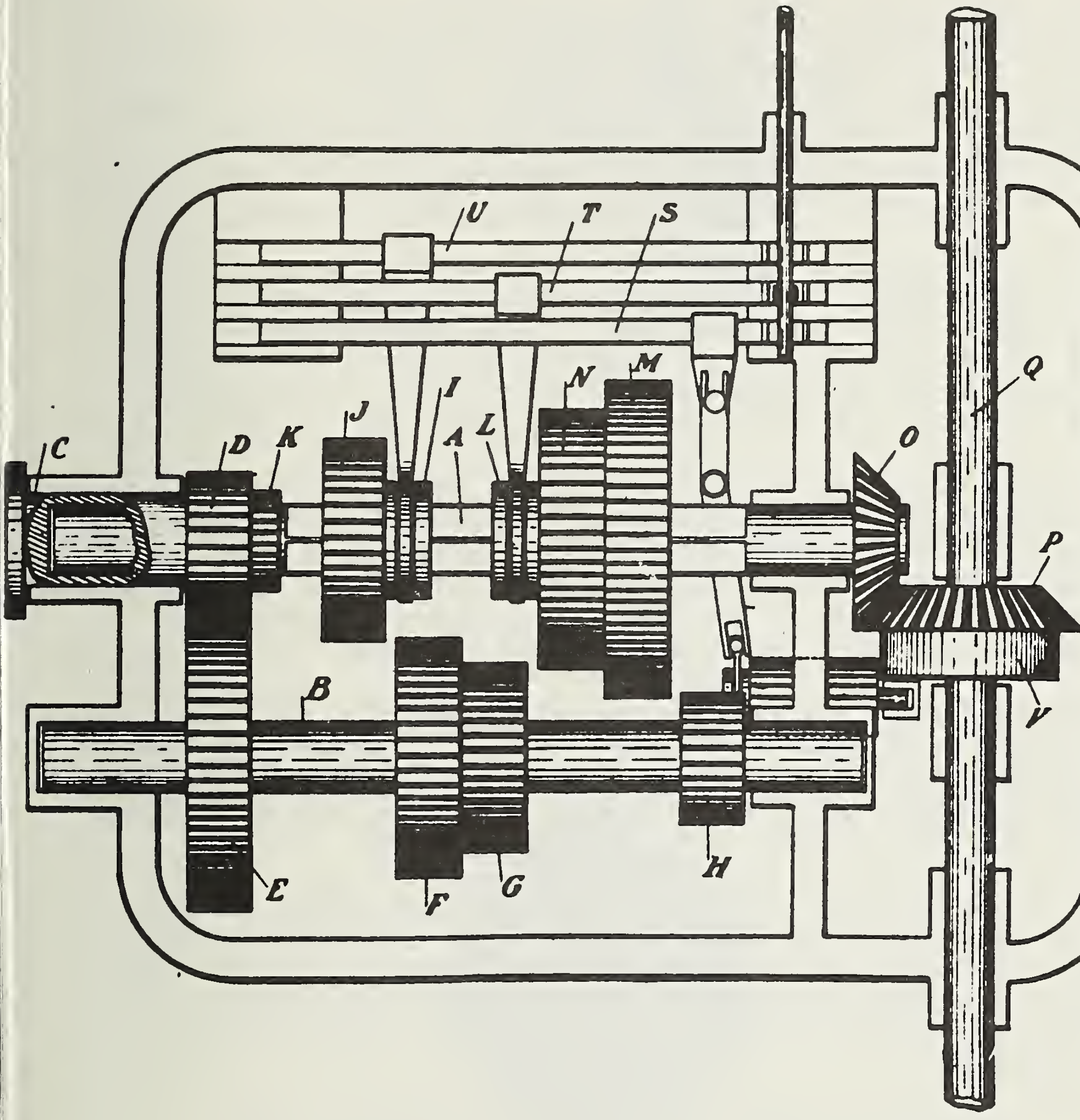
The planetary gear transmission had some popularity in the early days of the gasoline-powered vehicle since it dispensed with the crudity of crashing gears to initiate a shift on a sliding gear transmission. Early American planetary gearing practice involved the use of only two forward speeds, with one being direct from the engine shaft to the live rear axle via a chain. This eliminated the intermediate transmission of power through the sliding gears that were still being used on many French cars before the total victory of the direct drive.

However, the fatal flaw in the first generation of planetary transmission was the difficulty of acquiring a producible, quiet three-speed configuration. At least, that was the technical limitation of the time. The engine had to be geared down to conform to the two-speed set-up to enable it to carry moderate grades on high gear. This limited top speed.

For many years Ford was the sole existing adherent of the planetary gearset in the United States. The Model-T carried a spur gear type planetary treansmission until the model was dropped by Ford in 1928.

The planetary transmission forms existed in the early days, the internal gear type and the more popular spur gear type. In 1909 twenty-four automobile models out of the 292 made that year used a planetary gearing system. By 1913 Ford stood alone. In 1909 three-speed selective transmissions held nearly 61% of the models, with four-speed selective about 17%. By 1914 selective transmissions were used in about 95% of U.S. models.

At one time the four-speed progressive type had been vastly prevalent in the U.S. chassis but by 1909 had become nearly obsolete due to its excessive weight and size.



Source: *Automotive Industries,* July 19, 1966, p. 77.

FIGURE 3 MAYBACK-TYPE FOUR-SPEED SELECTIVE TRANSMISSION WITH BEVEL GEAR DRIVE TO COUNTERSHAFT

1.4 VARIOUS INFINITELY VARIABLE TRANSMISSIONS

Several other types of transmissions were experimented with through the first decades of this century. Some never made it past the scale model stage, and others made it into short-lived limited production. But all extended knowledge of what was practical, if only to eliminate a strategy as being impractical. All attempted to add flexibility to the still relatively inflexible gasoline engine. Most also strived to relieve the driver of part of the onerous gear changing task.

The first to be considered, the belt drive, was a defeated rival of the sliding gear and planetary, or epicyclic, transmissions. By 1903, it was fast disappearing. When Benz abandoned it that year, Fouillaron of France was the sole representative of a belt-driven automobile. Its belt, with a triangular section, ran over two expandable pulleys to produce a range of speed variations determined by relative dimensional change of the pulleys.

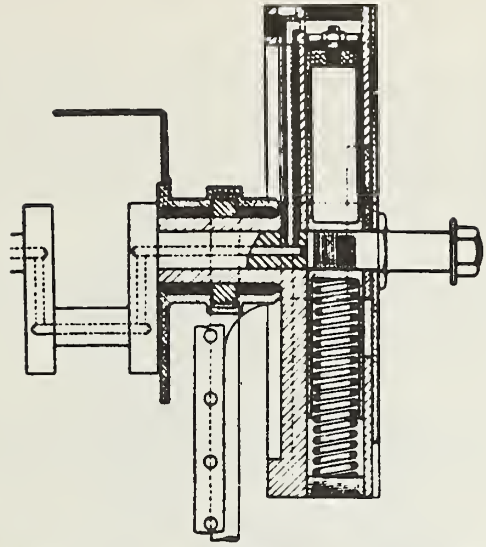
The demise of belt drive was caused by the susceptibility of the leather belt to stretch. It was also found to be almost impossible to house the belt in a manner that would prevent moisture from reaching the belt and cause excessive slippage. Early advantages, lower noise and greater efficiency, were erased as the art of gear cutting improved.

It was only recently that improvements in belt materials, combining steel or high strength inorganic fibers such as aromatic polamid (kevlar) with a tough plastic, has the belt made a comeback on the DAF vehicle. It still appears, however, to suffer the limitation in the amount of torque it can transmit, and thus is limited to very small vehicles.

The failure of the belt-type infinitely variable transmission did not thwart inventors from trying new methods. One of the first noted by the trade press was a "variable throw" transmission. In Europe, a small truck made by R. Hagen of Cologne was equipped with a link motion by which varying throw was imparted to rods which drove the rear axle through reciprocating clutches.

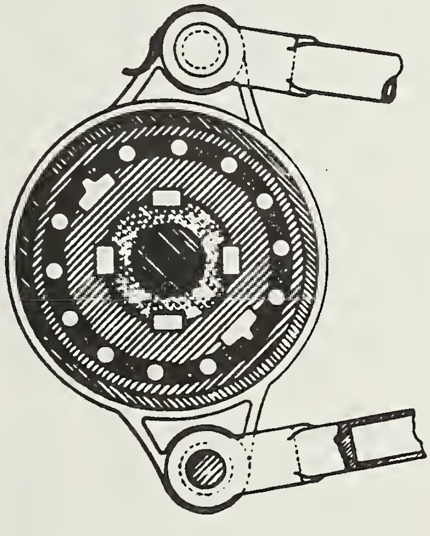
In Hartford, Connecticut, around 1900m a variable eccentric drive mechanism, the 'Dietrich', was demonstrated on a bicycle, but apparently was never adopted for automotive use. This consisted of plural toggle arms which yielded to torque and thus varied the effective stroke. The device contained two free-wheel clutch drive units.

Such variable stroke torque converters had appeared previously. But the development of 'automatic' stroke controls marked an important improvement. This invention, at the hands of George S. Strong, was incorporated into a group of trucks which participated in a number of contests in 1903 and 1904 (Figures 4-5). His Union trucks were built in Philadelphia, and



Source: *Automotive Industries,* July 3, 1924, p. 5.

FIGURE 4 "STRONG" VARIABLE THROW CRANK, WHICH IS FORCED BY OIL PRESSURE AGAINST THE PRESSURE OF A COILED SPRING TO A GREATER OR SMALLER DISTANCE FROM THE AXIS OF ROTATION



Source: *Automotive Industries,* July 3, 1924, p. 5.

FIGURE 5 SILENT REVERSIBLE ROLLER RATCHET USED IN CONNECTION WITH THE VARIABLE THROW CRANK SHOWN IN FIGURE 4 one continued to run there for a number of years. The crankpin was attached to a piston located in a radial bore in the flywheel. The piston was moved against the spring by oil forced into the cylinder by an engine driven pump. The crankpin was attached to the rear axle through a reversible roller ratchet.

In Great Britain around the same time a variable stroke transmission invented by T. W. Barber was applied to the Hutton automobile. It also employed a hydraulic control system.

In 1905, a Newman variable stroke device, utilizing a manually shiftable eccentric which changed the drive ratio stroke rates, was demonstrated on a Daimler car at the British Olympia show by the firm of Johnson & Phillips, of London.

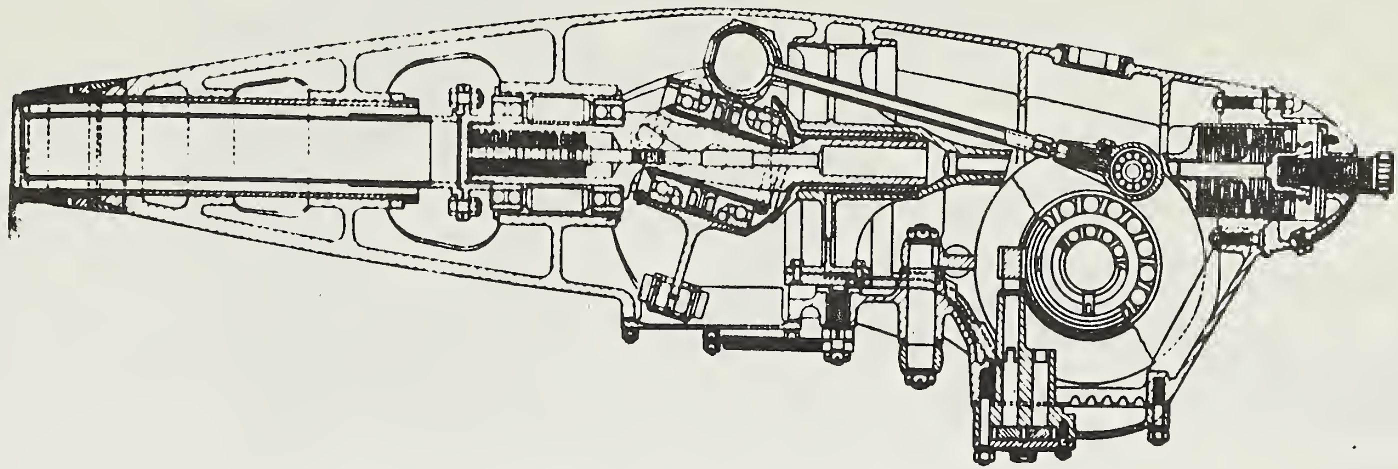
All these devices possessed one-way clutches operated intermittently by the stroke mechanisms in order to transmit a series of torque impulses to the output shaft. These one-way ratchet clutches engaged once per engine revolution and were, when engaged, wedged between surfaces of hardened steel and made an acute angle with each other. The strain during the transmittal of torque to the rear axle was very great. The fact that the torque to the rear axle was not uniform added to the stress on the transmission members.

The variable stroke transmissions developed during the 1920's, such as the deLevaud (Figure 6), and the Constantinesco and Spontan (Figure 7) infinitely variable units, were descendants of the Strong, Barber and Newman, and for automotive use the same one-way clutch difficulties persisted. These designs have survivied today as small variable drives used in laboratories and some machine tools.

The one infinitely variable transmission to reach production in the United States was, of course, the friction drive used by the Cartercar, Sears, Lambert and others. The interest aroused by the friction drive is exemplified by the fact that General Motors bought Cartercar soon after the corporation was organized in 1908. In 1909 thirteen models carried a friction driven transmission of one sort or another.

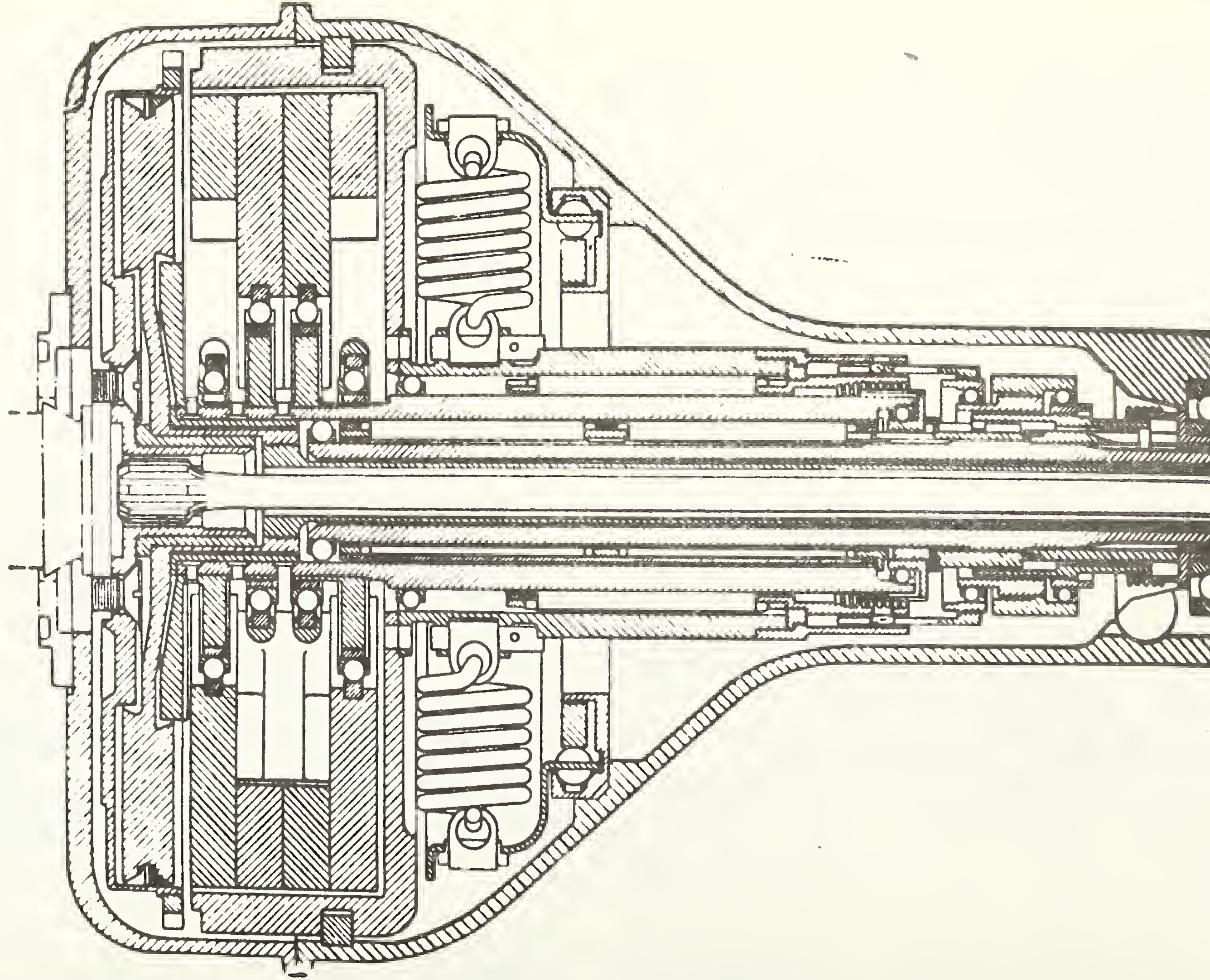
Friction transmission consisted essentially of a driven disk riding on its edge on a driving disk (Figure 8). The nearer the driven disk moved to the center of the driving disk the slower it rotated. When it moved across the center point the direction of rotation reversed. As the driven disk moved toward the periphery of the driving disk, the speed rose in proportion to the increased diameter of the driving disk.

One drawback of the disk-and-roller friction drive was that to effect a change in ratio it was necessary to hold the disk and roller apart during the shifting of the roller to a new ratio.



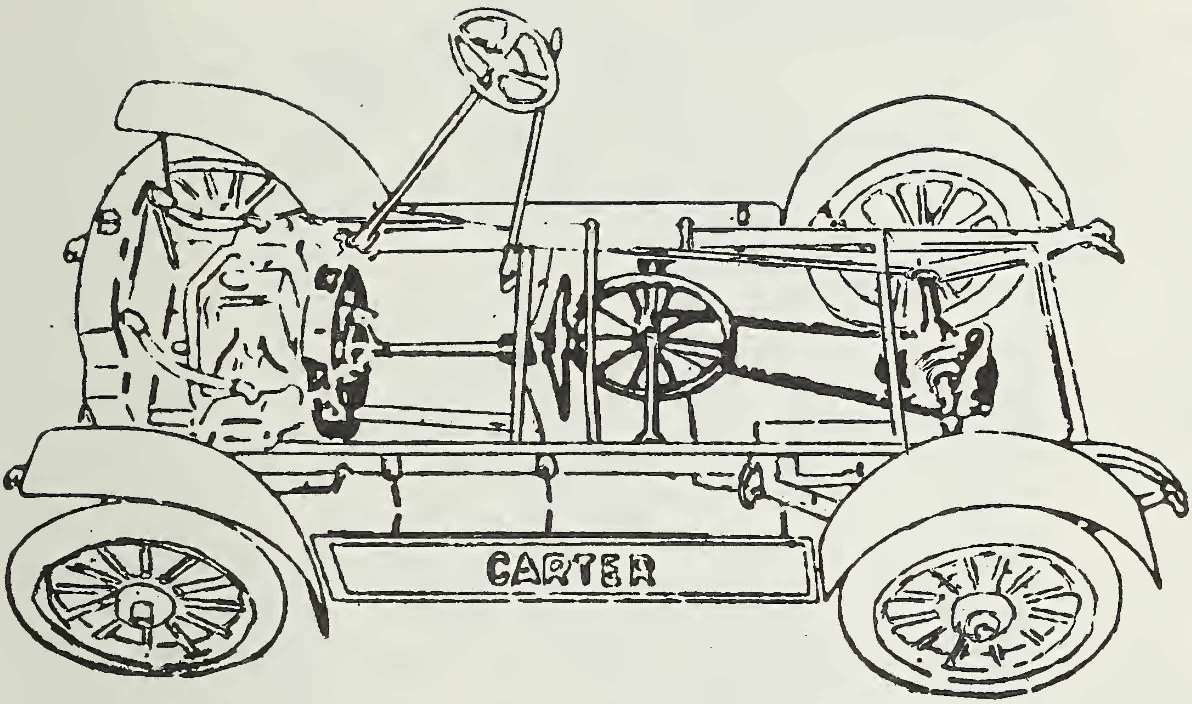
Source: *Automotive Industries,* June 18, 1925, p. 1059.

FIGURE 6 SECTIONAL SIDE VIEW OF LAVAUD TRANSMISSION



Source: *Automotive Industries*, April 12, 1930, p. 590.

FIGURE 7 LONGITUDINAL SECTION THROUGH SPONTAN TRANSMISSION



Source: ADL Archives

FIGURE 8 CONTINUOUSLY VARIABLE FRICTION DRIVE
USED IN THE CARTER CAR

In a double drive configuration, in which the transmitting disk was driven from both sides and consisted of four disks in all, the disks parallel to the main shaft had to be moved back and the driven disk or roller shifted to the desired position. The number of speeds were actually limited in most friction transmissions by the number of detent positions that could be formed in the quadrant that contained the shift level.

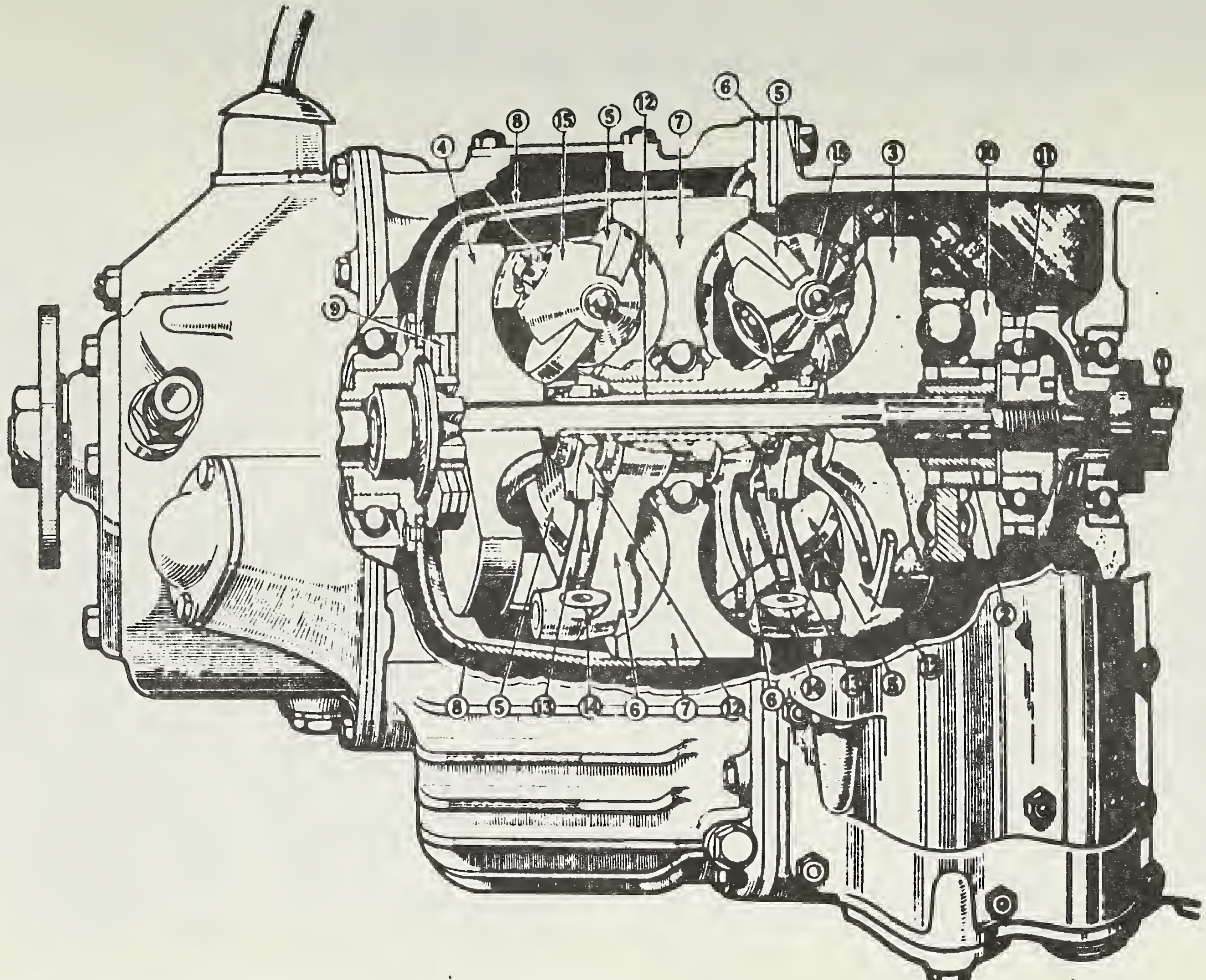
Numerous operational problems hounded the friction drive out of existence. These included the difficulty in maintaining a suitable contact surface between the disk and roller. This surface would erode due to the shocks of reobtaining torque after shifting and due to drivers' mistake of allowing the disk and roller to rest together while idling. Higher torque engines probably also spelled the doom for friction driven cars. Another problem involved the development of tilt as the roller splines wore. This would create a reaction forcing the roller to wander along its shaft to a position in which this force would be neutralized.

In 1933, this wandering tendency was harnessed in the Austin-Hayes IV transmission (Figure 9). The roller was tilted a predetermined degree and when the desired shift point was reached the roller was restored to the non-tilt position. However, this infinitely variable transmission, suffered penalties of weight, cost and complexity and did not become extensively produced.

1.5 THE FIRST "AUTOMATIC"

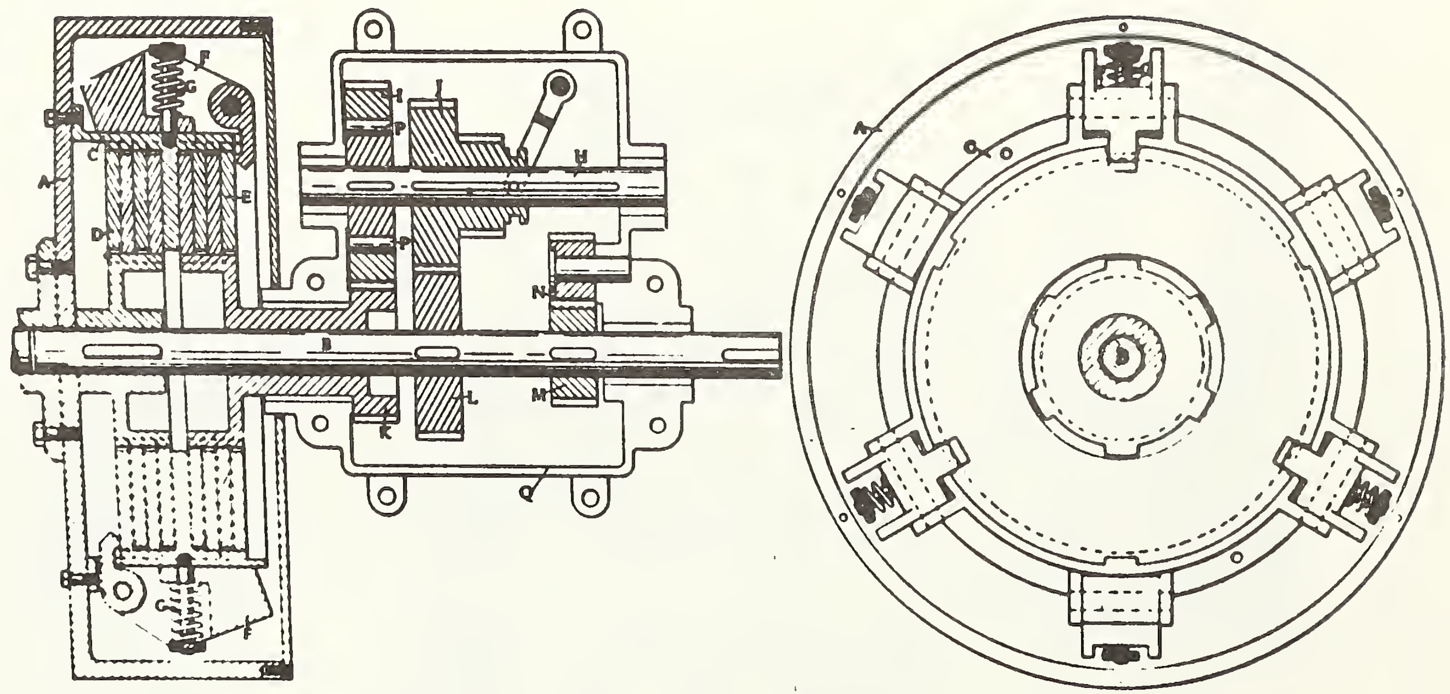
Development of the variable stroke and friction transmissions was paralleled by the invention and commercialization of the first "automatic" transmission, built by the Sturtevant Brothers, Boston, Massachusets in 1904 (Figure 10). The Sturtevant underwent a number of modifications before it was discontinued in 1908. Initially, the Sturtevant possessed a double, centrigugal-weight disk clutch. One clutch was used for high gear and the other for low gear. The centrifugal weights engaged the clutches in sequence with the rise in engine speed. When the stack of clutch disks was fully loaded, the low speed gear train was released by a one-way clutch.

A major fault with the Sturtevant was its inability to maintain high gear drive down to lower car speeds. It would drop out of drive completely when engine braking was most needed. A related deficiency was the lack of overlapping range of speeds between the low and high gears. When the transmission shifted into high, the high speed clutch would engage automatically. But entering high gear slowed the engine, producing a tendency for the clutch to disengage again. The clutch would "hunt' and slip continuously and overheat if the driver did not leave this speed range immediately.



Source: *Automotive Industries*, November 18, 1933, p. 604.

FIGURE 9 CUTAWAY VIEW OF THE AUSTIN-HAYES TRANSMISSION



Source: *Automotive Industries*, July 11, 1933, p. 85.

FIGURE 10 SECTION AND ELEVATION OF STURTEVANT AUTOMATIC TRANSMISSION DEVELOPED IN 1904.

1.6 OTHER EARLY AUTOMATIC DEVELOPMENTS

While the Sturtevant brothers could claim precedence in U.S. automatic transmission development, a couple of Europeans also experimented with automatic gear ratio changing, but following different concepts.

In 1897, a French inventor, Johabert L. H. Maugras, invented an "infinitely and continuously variable" device based on a split-torque principle. One portion was transmitted through a friction disk and roller and the other portion of the power through a fixed gear train. Differential crown gear elements were each connected to the two elements respectively, so the gear train provided a combining-gear action.

The friction roller was shifted by air pressure from an air servo system, and variably controlled by a valve from a fly-ball governor controlled by the engine speed. Maugras' concept was far too early, however, for the technology of the day to elevate the design to hardware.

The second French inventor, Leander Megy, developed a constant mesh gear train free to rotate on their own shafts. Individual friction clutches which rotated with the shafts were engaged and disengaged with the gears by a mechanism which was powered from the engine. The shift mechanism was automatically controlled by a fly-ball governor, or could be operated manually.

The Megy units were never produced in the quantity achieved by the Sturtevant.

The next "automatic" development, this from England, was the result of a wager by Captain F. W. Stanley, youngest son of the Earl of Darby, that he could produce an automatically shifting automobile within six months that would improve upon the Sturtevant's performance. Though he wasn't completely accurate, since he finished the device in about nine months, the Stanley transmission did have an improved control mechanism. The shifting device was not unlike the Megy mechanism. However, it was arranged to operate the sliding gear transmission and open and close the main clutch at the proper interval controlled by a flyball governor. However, the Stanley transmission was probably too expensive to put into production.

While inventors continued to experiment on advanced concepts, some of which belong on the modern automatic's family tree, the production sliding gear transmission also continued to improve. Innovations included pre-selectors, improved gear materials, heat treatments and overall design, and improvements of the clutch.

1.7 THE CLUTCH PROBLEM

1.7.1 Early Designs - Expanding, Contracting and Cone Clutches

The bulky cone clutch was blamed for some of the difficulty in shifting the early transmissions. Faulty gear construction, poor steel choices, and improper gear tooth engagement angles also led to gear crashing. Development of more efficient designs and friction materials was part of the continuing search for the easy-to-handle, silent, trouble-free transmission.

If the obvious solution to the gear shifting problem, making it automatic, proved unsuccessful, the best option within the available technology was to make the engagement and disengagement of the engine to transmission as shock-proof as possible.

For many years the cone-type clutch was used almost exclusively for both passenger cars and trucks in the U.S. About 1910 the multiple dry disk type clutch was introduced and around 1915 the single disk clutch. In the U.S. the cone clutch fell from favor until in 1921 it was used on less than 8% of all models. It remained popular in Britain for a longer period, used on approximately 60% of British cars in 1920.

The earlier clutches on American gasoline vehicles were bands that either expanded or contracted against a metal drum by means of a sliding collar. They differed from modern practice by being normally disengaged, the driver shifting a lever to bring them into engagement. In 1915 these types were quickly being dropped. That year, Peerless was the last manufacturer using the expanding band clutch, and Haynes-Apperson the sole adherent of the contracting band.

The cone clutch was first used in conjunction with sliding gear transmissions and shunned the then current practice by being normally held in engagement by a spring. The clutch was disengaged by the driver depressing a pedal which compressed the spring while drawing the two members apart.

The friction surface of the cone normally made a 15 deg angle with the axis of the cone and was faced originally with leather, and later with asbestos lining. Gear shifting was difficult for a number of reasons. First, the driven member had too much inertia to become engaged without producing a jolt. Many cone clutches suffered from the fact that full spring pressure would be applied immediately upon engagement causing the clutch to grab. This could be overcome by placing springs beneath the facing, which also solved the early problems associated with loss of alignment. Another method was the provision of two springs which would be applied progressively. Sharper cone angles also allowed lower spring pressures.

1.7.2 Multiple Disk Clutch -- Wet and Dry

The cone clutch was partially displaced by the lubricated multiple disk type with alternating disks of steel and bronze. This type also had considerable inertia, and oils of the day did not have adequate viscosity control. In cold weather substantial drag would develop, producing gear clashing when a shift was ventured.

The multiple disk dry clutch was the next development, eliminating the oil and having friction surfaces on both faces of each disk. It required fewer disks than the wet type, since its coefficient of friction was not reduced by the lubricant, and could also be lighter and less expensive. There were two types used in the U.S. In one the torque was picked up by three or more pins attached to the driven or driving members which corresponded with holes in the opposite member. In the other, internal splines picked up the drive. The latter was superior since the force could be divided among the teeth. The pins in the first variety were loose fitting in order to compensate for expansion due to heat. The hole-and-pin variety therefore was frequently noisy.

Overall, the multiple disk clutches had problems with excessive inertia that could be alleviated only through the addition of more mechanism, which again increased weight and complexity. Also, the disks in the center of the clutch would become quite hot in severe service which would lead to heat distortion and subsequent drag. The choice of placing friction facings on the driven members, contrary to later practice with double plate clutches, can be blamed for the creation of excess heat on the thin driven disks. By the early 1930s multiple disk clutches had almost entirely disappeared from automobiles, but are still being used in some motocycles and racing cars.

Gradually, the merits of the single plate clutch became clear, until in 1930 81% of all U.S. passenger cars were so equipped.

1.7.3 Single-Plate Clutches

The single plate clutch first appeared about 1910, and these used metal disks in direct contact with no intervening friction lining. Also, some early versions used the two heavier pressure plates as the driven member, instead of the lighter central disk.

The correct (modern) configuration was soon settled upon. The development of asbestos-based friction meaterial, and design advances for lighter inertia, provided greater flexibility in engagement and improved heat dissipation. The requirement for ruggedness competed with the desire for smooth engagement over a wide range of speeds. Smooth engagement depended upon facing compressibility which is inversely proportional to facing density. Durability was dependent on facing tensile strength, density, a consistently high coefficient of friction, and high temperature stability.

Asbestos linings were originally woven on a loom from long asbestos fibers with a small proportion of cotton, around 10% by weight, and twisted around a brass or copper wire to form a thread. However, the curing temperature had to be set below the charring point of cotton, which lowered the degree of heat resistance achieved by the compound. Molded facings possessed a higher degree of asbestos but were not unreasonable in cost since the facing rings were punched from a sheet formed from short asbestos fibers and some resinous binder. These were then impregnated with an asphaltic material and cured at high temperature.

Later forms of production, also referred to as molded facings, used either spiral winding, in which a tape or strip was wound over a V-shaped mandrel, or winding a yarn over a mandrel between two disks. Cords were zigzagged across the facings as they were wound, creating radial strength to resist cracking.

Clutch developments between 1915 and 1930 tended to reduce the effort required to complete a shift and kept alive the notion that the shift could one day be eliminated completely. Improvements included the provision of a cushioning element to allow travel of the clutch pedal and pressure plates before full pressure was brought to bear. This helped eliminate grab and chatter. Cork inserts, crimped rims and cushion springs were all used to perform this function.

Other improvements included the development of means to apply pressure more uniformly over the entire friction surface, the use of self-aligning parts to compensate for wear, the decrease of the pressure needed for disengagement, and redesign of clutch construction to eliminate warpage.

1.7.4 Automatic Clutches

Starting in the late 1920s, automatic clutches began to appear, often in combination with a pre-selecting shift mechanism. These clutches provided better control of the transmission by drawing on engine power for engagement. Reduced shock was a further advantage, since torque could be transmitted gradually, the rate of engagement being determined by engine speed. The automatic clutch provided one big step towards automating the shift since a car could be started in motion via the accelerator pedal and stopped by the brake without resorting to operation of an additional control under most conditions.

Probably the first automatic clutch furnished on a production automobile was produced for an N.A.G.-Protos by National Automobile

Gesellschaft, Berlin, around 1928. It was similar in configuration to the Powerflo clutch (Fig. 11), made by the Automotic Clutch and Transmission Co., Gloucester, NJ, during the early '30s for the Continental Motor Co. The rights to the design were later acquired by the Spicer Manufacturing Co.

The Powerflo clutch consisted of three centrifugal masses carried on the pressure plate and backing plate by boot-shaped members. When the centrifugal masses spun outward due to the increase in engine speed, the "toe" of the supporting member took hold on the backing plate, while the "heel" pressed against the pressure plate and forced the three friction members into contact.

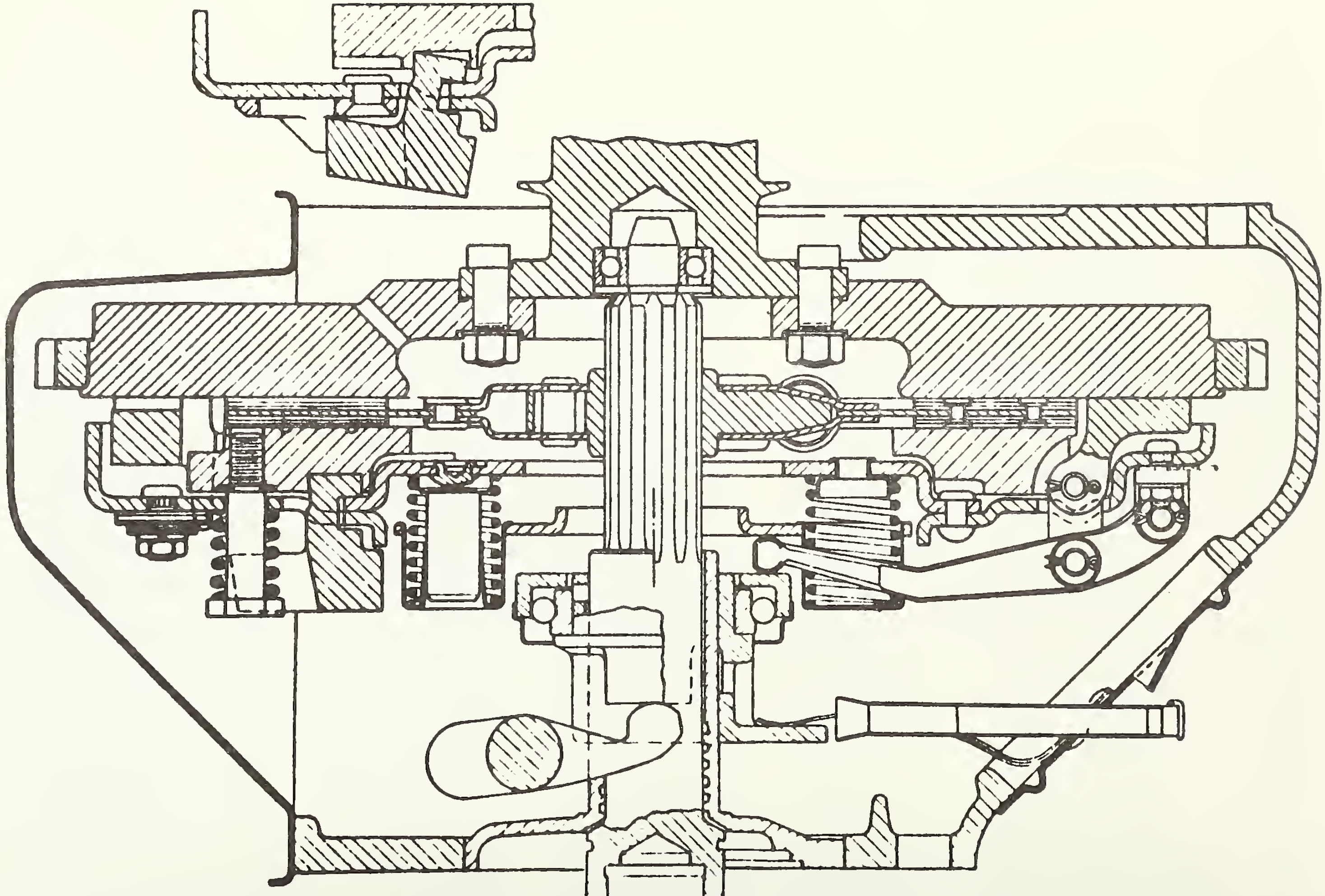
In this type automatic clutch, the mode of operation consisted of the force of the centrifugal weights gradually overcoming the pressure of the clutch spring. By increasing engine speed from idle it was possible to start the car without resorting to the clutch pedal. When used in conjunction with a free-wheeling unit, it was not necessary to use the clutch at all , even for shifting.

The main drawback of this unit was that the clutch was more likely to slip than a driver-operated unit. This was particularly true when the car was ascending a grade at part throttle. Also, the centrifugal clutch was larger and clearly more complex than the conventional clutch and the operation of the clutch, particularly when used in combination with free-wheeling, was more difficult to learn to operate properly.

Vacuum control of the clutch, and later of the shifting itself, evolved from inventions from the workshops of Bendix Aviation Corp., and was included usually as optional equipment. Like the normal foot-controlled clutch, a clutch equipped with vacuum control was held in the engaged position by springs. When the driver desired to make a shift, he would release the accelerator which closed the throttle and resulted in a rise in intake manifold vacuum. Movement of a piston in a cylinder communicating with the manifold would disengage the clutch. The clutch was re-engaged by gradual depression of the accelerator.

1.8 EARLY GEARBOX LOCATION AND GEAR RATIOS

During the first decade-and-a-half of the twentieth century, engineers debated the best location of the gearbox: central, rear, or in combination with the engine. Some engineers argued that the rear axle location was superior since less noise from the gearbox would intrude into the passenger compartment. More importantly, torque magnification could be transmitted directly to the bevel pinion gear instead of through a universal joint.



Source: *Automotive Industries*, June 3, 1973, p. 674.

FIGURE 11 SECTIONAL VIEW OF THE POWERFLO CLUTCH

However, it was argued, disadvantages of such a placement included direct transmission of road shocks to the gear box and the necessity for long control arms from the shift lever. A rear axle location was never popular, reaching its zenith in 1912 with 78 models, or 21.3% of the total number of models available that year. (More recently, Pontiac, Alfa Romeo and Porsche have resurrected this design).

The amidships location was used extensively in the early days, equipping 75% of all models in 1910. Advantages included a contribution towards car balancing, along with easier servicing and lubrication than the gear-box-engine unit allowed. Less strain on bearings also resulted, due to the use of universal joints both fore and aft of the transmission.

But 1914 was the turning point in gearbox location. It was already acknowledged that attachment of the gearbox to the crankcase had advantages in size and cost. It also eliminated the material needed for rods, shafting, etc., formerly required in adapting standard gear boxes to the particular model's requirements. The continued trend toward the unit engine-transmission led to less complexity in construction, attachment and mounting of the clutch. Changes in clutch and gearset access removed the serviceability objection. From 1911 when the engine-transmission unit was carried by only 42 models, this configuration more than doubled its adherents in two years, and by 1921, 88% of all U.S. models followed this route.

Rear axle gear ratios, and the method of their determination, has also changed radically since the early production years. In the first decade, cars were ordinarily geared around 2.5:1, while by 1913 most models were between 3.5 and 4.1, with only the high-powered models as low numerically as 3.1:1. By 1915 the average was 3.88:1.

Intermediate gear ratios were determined by each car manufacturer after years of experimentation, leading to a low degree of commonality among the several dozen carmakers. Warner Gear in 1916 complained that too much variation existed in the range covered by standard models. This prevented the establishment of ideal gear reduction steps. By then Warner had instituted the now universal practice of arranging gear ratios in a nearly geometric progression.

2. EVOLUTION OF THE SELF-SHIFT

2.1 THE PRE-SELECTOR

Elimination of the gear shifting function, or even of the gearbox itself was a prime topic for conversation from the very earliest days of automotive design. Ideas began finding their way into hardware as early as 1903 when Megy introduced his device at a number of automotive shows. This transmission, previously described, could be either driver operated or automatically controlled.

As can be seen, the movement to develop a power shifting method was coincident with the desire to have the car alter ratio in direct response to a change in road condition or driver command through the accelerator pedal.

The most important development before 1920 in the evolution of the automatic transmission was the introduction of pre-selective gear changers. With this device the driver would move a lever into a position corresponding to the gear speed he would next require. When the change in ratio was actually needed, the driver would either ease up on the accelerator or push the clutch pedal to have the shift effected by mechanical, electrical, hydraulic, or vacuum means.

A mechanical device invented in 1912 by a W. D. Hughes of Atlanta, GA, using a system of rods and a transverse arm hooked to pull the different gear combinations into mesh, is the first notice the author has found of a pre-selector in the U.S.

One of the first pre-selective devices to reach production in the U.S. was on the Haynes-Apperson in 1923. This mechanical system also relied on the driver's physical effort through the clutch pedal to effect gear changes.

Instead of relieving the driver, the mechanical pre-selector actually made the change of ratio more burdensome, since the effort on the clutch pedal was usually greater than normally required, and a degree of expertise was required for accurate shift timing.

Electro-magnetic gear shifting was used on a few high-priced cars for a number of years but was gradually abandoned as too expensive, erratic, and a power drain on the battery.

In 1918, the U.S. Automatic Shift Co., Madison, WI, developed a hydraulic gear shift mechanism. Ratio changing occurred when the driver depressed the clutch pedal. However, the gear mesh during downshift was unsatisfactory. Seven years later the firm introduced a system of automatic shifting combined with a synchronization device. Four sliding rails, two upper and two lower, were cut with slots. A shaft carrying a rotary gear selector was mounted centrally between the rails. For any position of the selector lever, an upper and lower rail were locked. Rotary action of the selector, induced by movement of the clutch pedal, occurred when the notches were brought into a common plane. As the pedal returned the motion of the upper rail was transmitted to the lower through the selector, effecting the change.

When second was in mesh, the high gear position was already determined. Thus the shift from second to high could be made without movement of the selector. Another full depression of the clutch pedal brought the second gear back into engagement.

The sliding gear synchronizer consisted of an auxiliary shaft driven by the output shaft carrying a beveled gear which meshed with the appropriate gear on the input shaft as the clutch pedal neared the end of its stroke.

This system was one of the many mechanical pre-selective systems that never survived beyond the experimental stage due to mechanics that would quickly get out of order or the extra effort demanded of the operator.

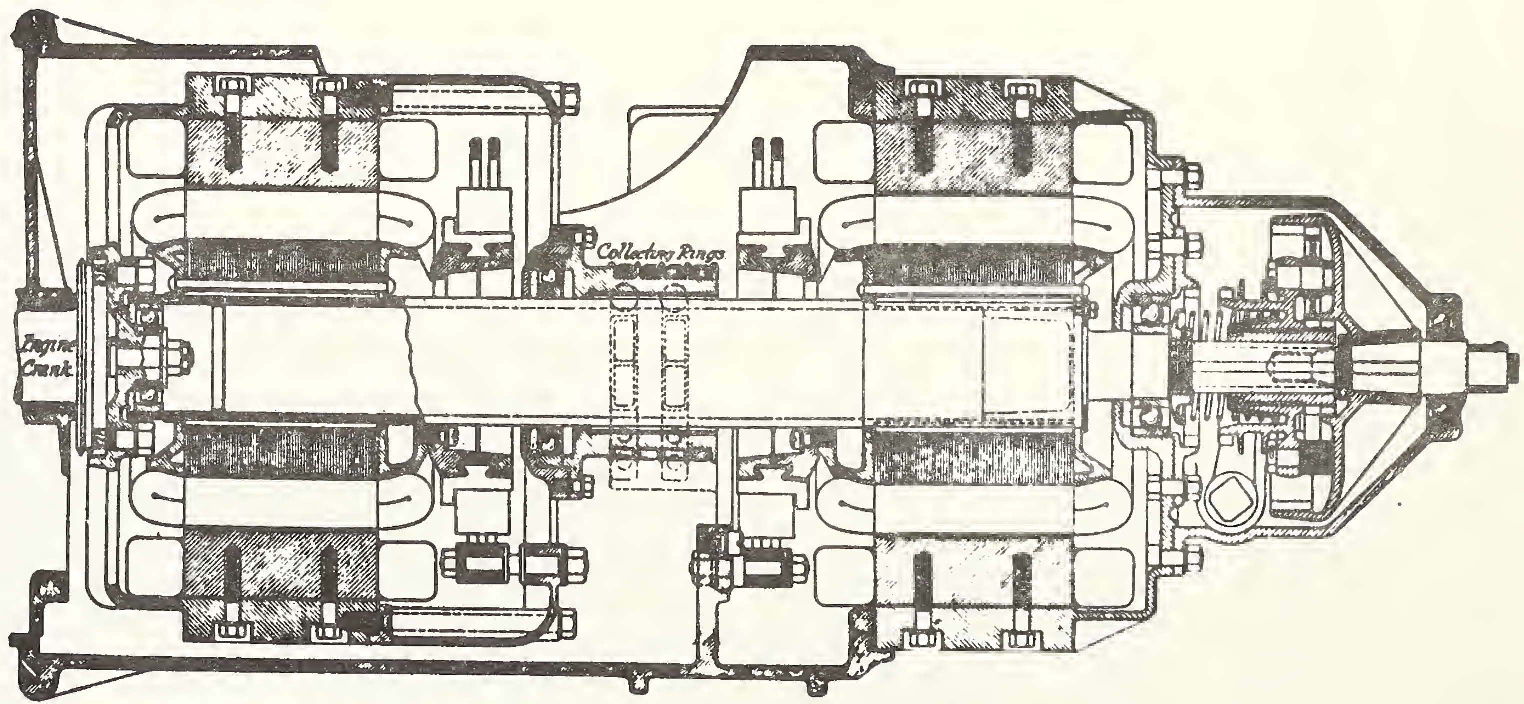
Vacuum shifting was introduced about 1930 when many manufacturers were also installing synchronization devices. At first, transmissions using vacuum control were not as smooth as a hand-shifted unit, but improvements were made, at the cost of some complexity. Vacuum shift will be covered in more detail later in conjunction with the progression to the automatic transmission.

2.2 ELECTRIC DRIVES

In 1917, with the introduction of the Owens Magnetic (Fig. 12) and Woods Dual-Power, most factories were investigating electric and hydraulic substitutes for geared transmissions. Some engineers had drawn the hasty conclusion that power plant flexibility had progressed as far as possible and that the only avenue left for improved performance was through a radical departure in transmissions. While the two new concepts produced a few imitators, the geared transmission continued to improve while the electric and hydraulic transmissions, with added weight and cost penalties could not compete.

For one year, 1917, the Vesta electric drive was used on the McFarland automobile. A crude test unit had been built in 1898 by William Morrison of Chicago. As speed increased, the centrifugal force of the Vesta contact brushes created mechanical drag. This centrifugal clutch effect augmented the electrical torque capacity and decreased the slip. The Vesta, like the Carter car, suffered from friction drive problems, and lasted only one year in production.

Other electric drives, such as the Columbus and Owens Magnetic, were manually operated. They were also heavy, costly, and not flexible enough to attract many customers.



Source: *Automotive Industries,* April 24, 1937, p. 636.

FIGURE 12 ENTZ ELECTRIC TRANSMISSION OF OWEN MAGNETIC CAR

2.3 EARLY AUTOMATIC CONTROL EXPERIMENTS

During this period, a railroad transmission having an automatic or self-shift control was developed by Raymond Zeitler, a University of Illinois student. It was tested on several railcars but never entered general use.

Another automatic or self-shift mechanism, designed by Louis Biava, West Hoboken, NJ, during 1917-18, utilized engine power through screw-shaft servo connections, controlled by differential gears, small energizing brakes and a flyball governor. The unit was inserted into a Pierce Arrow truck for demonstration but never produced in quantity. The Biava transmission, like the Stanley 'automatic' of 1907, required declutching and clutching for each shift.

The Flexo transmission was introduced on a few cars around 1921, including the Peerless. A secondary jaw clutch was added to the gearbox by which the gearset could be disconnected from the propeller shaft. The driver would make a slight motion of the lever at the steering post which caused this rear or secondary clutch to operate. The gears could be shifted while relieved of load and positive drive since the gearset was unloaded at both ends.

The overwhelming problem with early transmissions based on a discontinuance of torque during shifts, as well as with early preselectors, was that they were often guilty of 'hunting', or not being able to restore a smooth drive after executing a shift. Also, the preselector could not defeat the sliding gear transmission's propensity for noise and gearcrashing and had to await the advant of improved automatic clutches and the constant-mesh transmission before widespread acceptance could be obtained.

3. DEVELOPMENT DURING THE 1920's

3.1 CONSTANT-MESH TRANSMISSIONS

The most important innovations during the 1920's were based upon the constant me transmission, the debate over the advantages of the three-speed versus the four-speed transmission, renewed investigation of the planetary gearset, the fluid coupling, (still outside automobile production) and the use of synchronization.

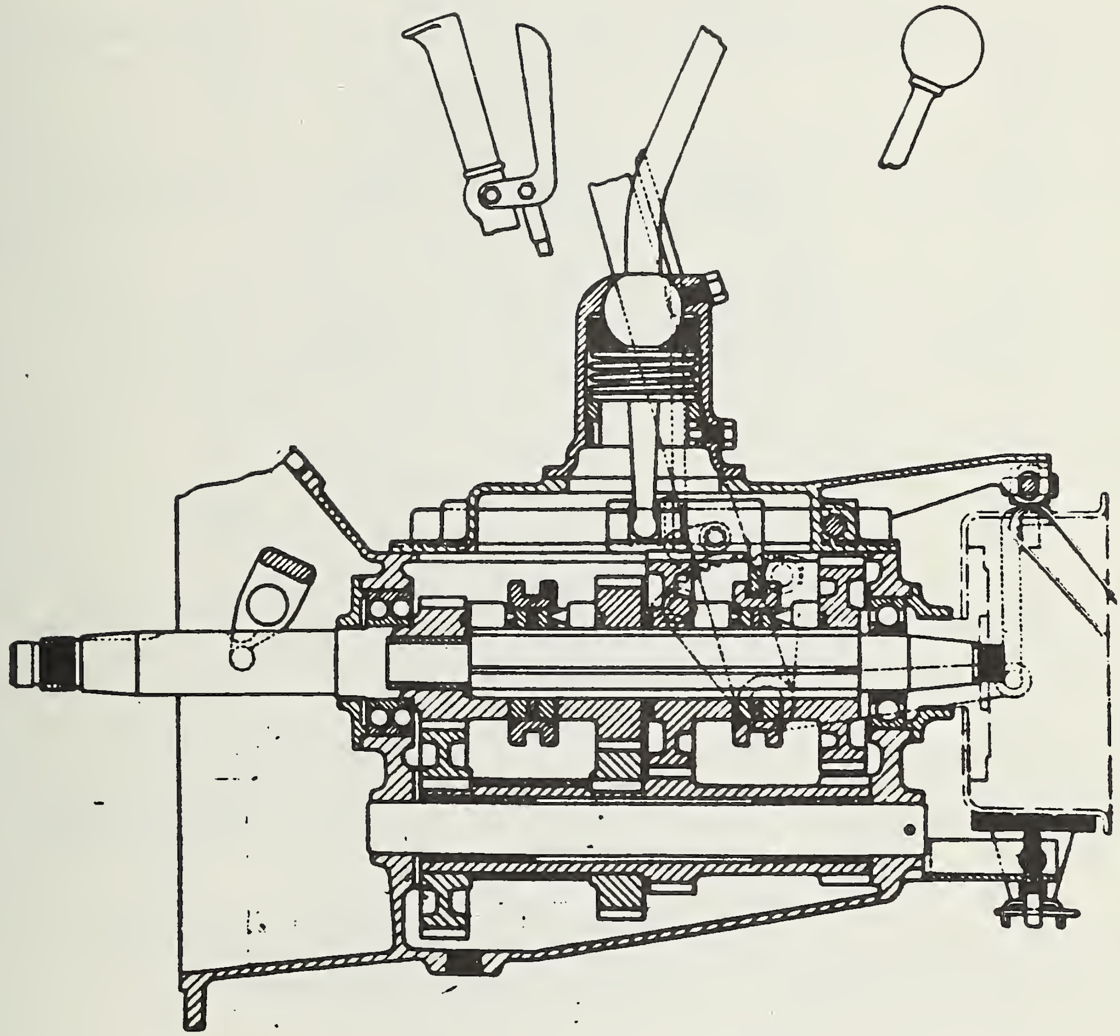
Leander Megy's attempt at automation in 1903, previously referred to, was the first recorded constant mesh gearbox. Cotta unveiled a constant mesh in 1913. Another of the first displayed in the United States was invented by W. A. McCarrell in the early 1920's (Figure 13). The helical gear transmission contained collars that were shifted over roller clutches integrally formed on the ends of each gear. Previous constant mesh automotive designs had used jaw clutches. The roller clutch was lifted from marine experience. When a gear was not engaged, and the shaft was in rotation, centrifugal force threw the rollers into the outer position in their cage. To engage a gear, the intermediate sliding collar was shifted over a cage projection by the hand lever. As the collar moved toward the desired gear it first engaged three longer rollers, allowing about 90 degrees to pick up before three short rollers also engaged and the entire roller arrangement was forced into a splined shaft.

While constant mesh transmissions of the early 1920's were quieter, they suffered from weight and cost penalties. Also, most used a 'positive' clutch. Danger arose from these clutches as the forces generated between the meshed gears while downshifting at a high rate of speed could kick back the shift lever, and possibly harm the driver's arm.

3.2 THREE VERSUS FOUR SPEEDS

The debate over the benefits and drawbacks of equipping a car with a three-speed or a four-speed transmission was most vocal during the period 1925-30. Manufacturers' choice was dictated by the type and size of engine, rear axle ratio, tire size, vehicle weight and desired performance, and the design of contiguous elements such as the clutch, propeller shaft and universal joints, as well as the market demands for greater acceleration and reduced frequency of gear shifting.

Through the late '20s many models increased rear axle ratio along with engine speeds to take slight grades on high gear. It was argued, however, that such a combination would produce noise and vibration problems. Also, the engine would not be able to develop its full torque on the level at moderate speeds. Further, the greater axle ratio increased fuel consumption, and could lead to the engine operating beyond its peaking speed if the rear axle reduction was very large and the car was operated on level ground at wide open throttle.



Source: *Automotive Industries,* February 7, 1924, p. 294.

FIGURE 13 McCARRELL CONSTANT MESH TRANSMISSION IN NEUTRAL

With the four-speed transmission this problem was alleviated. Its advantages were lower engine speeds with a decrease in vibration and noise, with attendant lower propeller shaft speeds, better fuel economy, and an ability to accelerate more rapidly without excessive gear noise in next-to-top speed with a lower rear axle ratio. Advocates also claimed easier shifting and generally quieter operation.

Supporters of the three-speed transmission countered with the argument that it could be quieter than the four-speed and was easier and cheaper to manufacture.

Ultimately, it was acknowledged that the four-speed possessed a higher degree of flexibility. It was also possible to make the lower gears as silent as direct drive using internal gears as easily as in the three-speed.

Though the four-speed unit provided a wider range of ratios to insure that the engine could be adequately loaded when operating at moderate speeds, it had the negative marketing aspect of demanding more shifts. It was claimed that the public never learned to operate the four-speed correctly. The automatic overdrive, introduced by Chrysler in 1934, lifted the onus of the extra shift.

3.3 THE INTERNAL GEAR

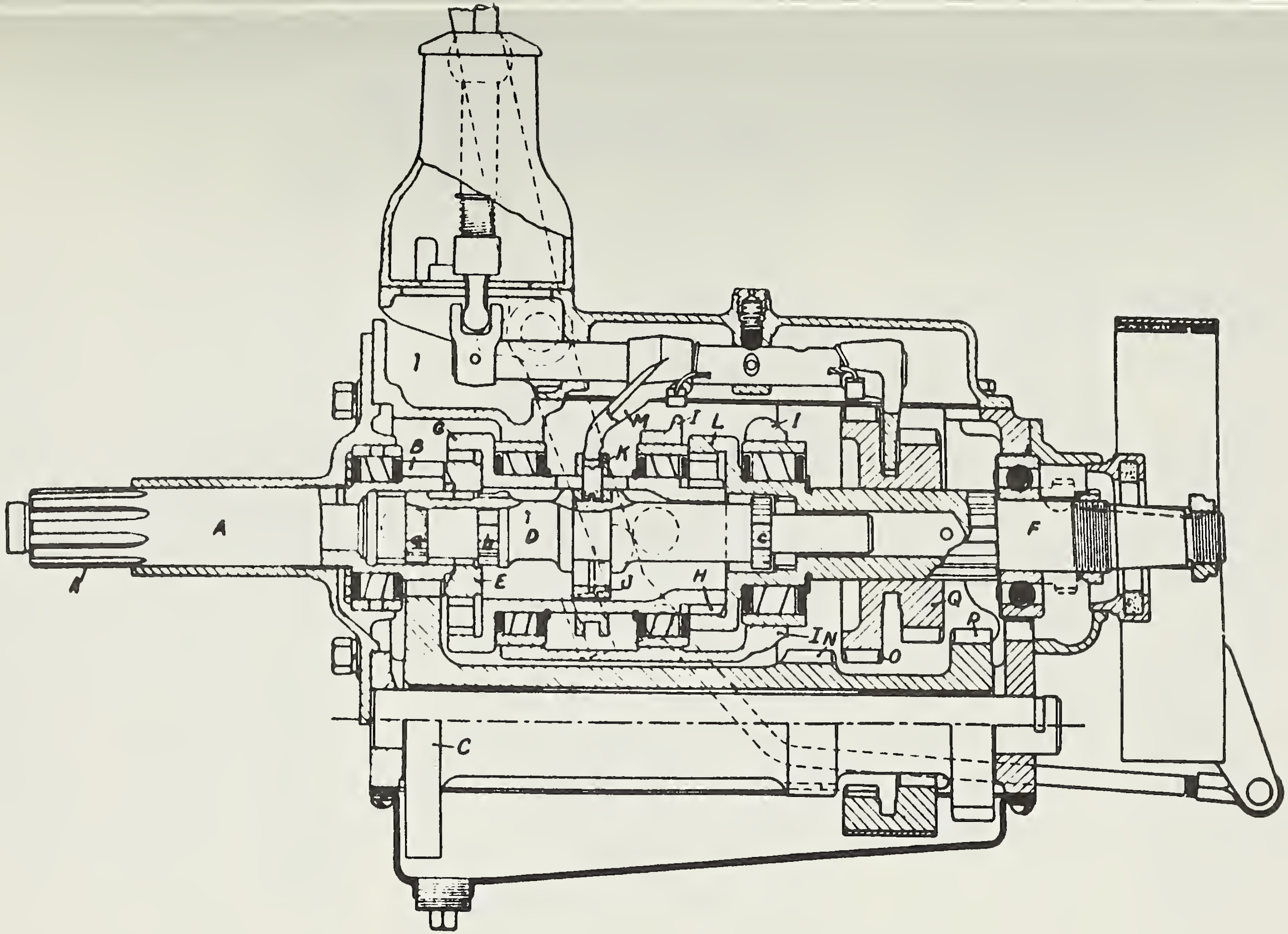
Toward the end of the '20s, Warner Gear introduced a four-speed transmission, the Hi-Flex (Fig. 14), which incorporated internal gears. While it was not a commercial success, it paved the way for other solutions to gear noise.

The internal gear provided a next-to-top speed almost as silent as direct, with a sure shift between the two speeds. The internally geared transmissions also allowed a faster rear axle ratio, with a slower engine speed resulting.

Reasons for the more silent internal gear operation were credited primarily to the fact that the gear web and rim could muffle noise, and that tooth contact took place within an oil bath and extended over a greater arc of circumference.

The internal gear train resulted in improved oiling systems for transmissions and gave added impetus to the research for lubricating fluids that would not "squish" or change viscosity excessively due to changes in ambient conditions.

Many combinations of transmission elements were attempted in the third decade. Both the sliding gear and constant mesh were applied to internal elements. They were placed both ahead and to the rear of the general gear train and between the constant mesh front driving pair and the lower speed gears.



Source: *Automotive Industries,* March 26, 1927, p. 471.

FIGURE 14 LONGITUDINAL SECTION OF WARNER HI-FLEX FOUR-SPEED TRANSMISSION

An example of the unusual combinations possible was made by Detroit Gear in 1928. The internal gear reduction in both three and four-speed units were segregated from the spur gears for ease of lubrication and to insure adequate oil for the internal gears. The lower speeds, spur gears, were intended for emergency use only, for starting up grades or while driving in sand, snow, etc. When the oil supply fell, the rear compartment for the spur gears emptied first. The driver would be warned of the problem by an increase in noise from that section while the more sensitive internal gears remained well-lubricated.

The ordinary three-speed sliding gear transmission with plain spur gears, dominant in 1928, was clearly doomed three years later. The original four-speed units with silent third depended on internal gearing. By 1931 more than a dozen models carried transmissions with the "silent" internal third speed. Some of these, as well as a number of others with the conventional spur gear arrangement, utilized jaw clutches to engage intermediate and high. The two members were brought into synchronization by friction members.

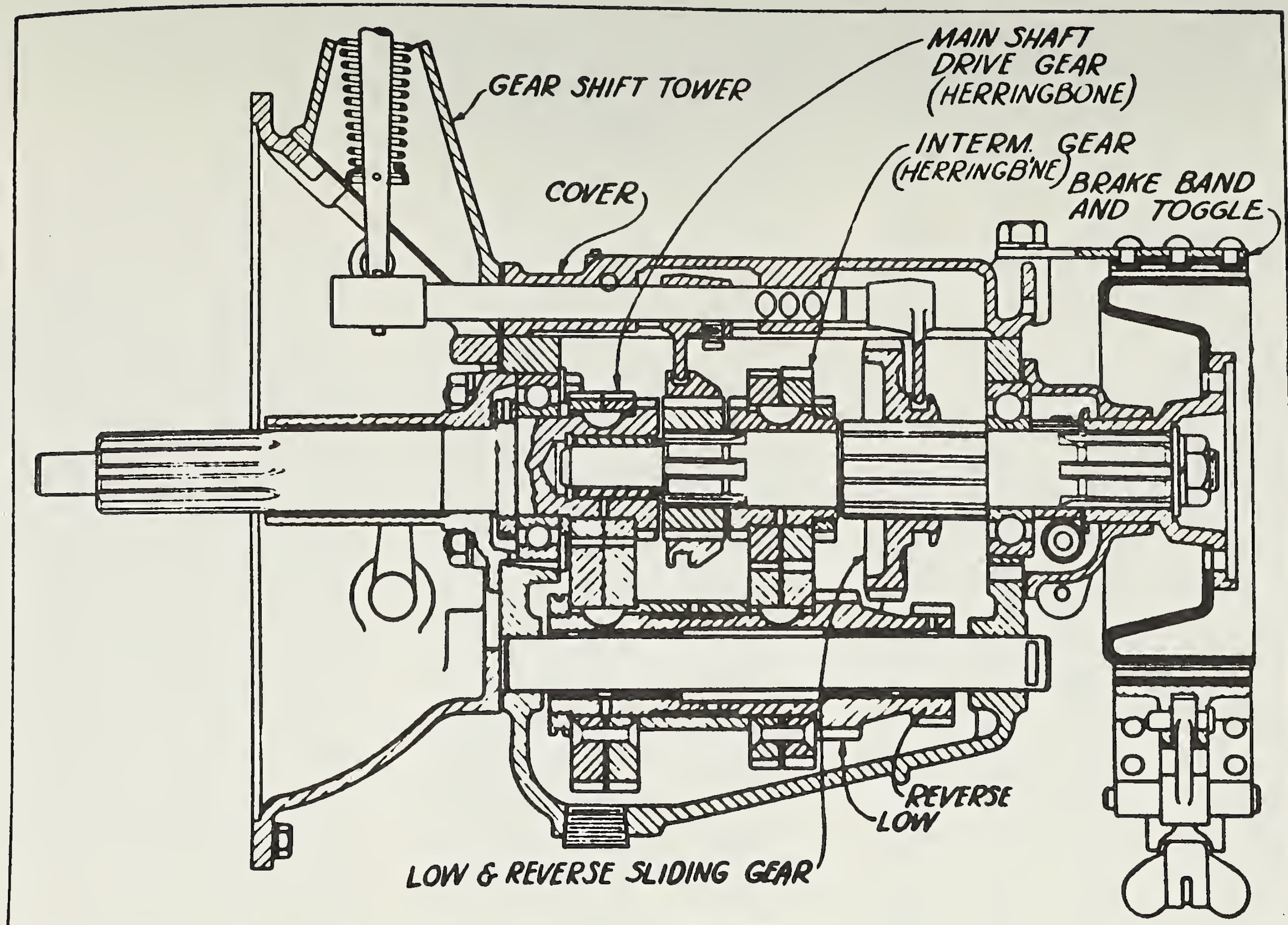
Various forms were used for the internal gear around 1930. Herringbone gearing was attempted, particularly by Studebaker, but did not lend itself to lateral shifting. More long-lasting was the appearance of constant mesh internal helical gears. Engagement of the different combinations was most often through positive clutches. In late 1929 Reo introduced a model that reportedly achieved a greater degree of silence by cutting the two halves of the paired three-speed set with teeth of different pitches (Fig. 15).

By 1934, Nash, Buick and Oldsmobile were using helical gears for all speeds. The Hudson, to prevent noise from idling low speed and reverse gears while driving in intermediate or high gear, provided means for unmeshing these gears. The most important innovation that year, however, was Chrysler's incorporation of a planetary overdrive unit controlled by a vacuum-operated clutch as the car reached a preset speed. The clutch was interlocked with the free-wheeling unit so that overdrive could be engaged only when the free-wheeling device was actuated.

Other industry developments incorporated into the new gearsets included more uniform steels, better forgings, improved heat treatments and gear cutting methods and increased accuracy. In addition, the stub tooth form lost ground for the first constant mesh and intermediate pairs in favor of full height teeth. Better heat treatment control reduced distortion and permitted the use of higher and thinner tops. The capability to produce gears which kept at least two teeth in contact to divide the load thus won currency during the late 1920's and early '30s.

3.4 EXPERIMENTAL TRANSMISSIONS

Through the 1920s a number of experimental automatic and infinitely variable transmissions were displayed, but very few were seriously considered for production.



Source: *Automotive Industries,* August 31, 1929, p. 305.

FIGURE 15 PLAN DRAWING OF THE NEW THREE-SPEED TRANSMISSION USED ON THE REO MASTER FLYING CLOUD

One of the more important from the standpoint of precursors of the modern automatic transmission was introduced by the Automatic Clutch and Transmission Co., Philadelphia, Pa. Its special hydraulic displacement-pump type transmission was installed in a Buick and shown at the 1924 Summer SAE meeting at Spring Lake, New Jersey. The system consisted of a friction clutch located in the flywheel which engaged through centrifugal weights when the vehicle speed passed 20 mph. The weights were located in the clutch drum. A low speed and reverse clutch was also set in the clutch drum and was operated by a second set of centrifugal weights. This clutch became disengaged as speed increased. The two clutch governors were arranged to provide a slight overlap in the action of the two clutches. The idea was to avoid interruption in drive as the change was made from indirect drive to direct.

The low speed centrifugal disk clutch connected a rotary oil pump to the engine, the ratio in that range being controlled by a hand-selected valve regulating the drive pump-to-motor feed when needed.

The movement of a cylindrical control valve which controlled the rate and direction of oil flow caused the oil to pass through all four of the hydraulic motors provided, causing their motors to rotate. Power was transmitted to the propeller shaft by a roller clutch. Further movement of the control valve shut off the oil from one motor after another resulting in a torque decrease and an increase in speed. In the high speed position, the motor and propeller shaft rotated at about the same speed as the engine. After the control valve had been placed in the high speed position, the car would soon attain the speed in which the centrifugal weights acted to disconnect the oil pump from the engine and simultaneously engage the direct drive clutch. The Buick was run over 30,000 miles with this transmission and a redesigned model was made, but the project was abandoned due to lack of funds.

Also in 1924, a demonstration was made of an automatic planetary transmission developed by Captain Alexander Dow. The American Car & Foundry Co. had acquired the rights to the unit. This control consisted of a lever that provided shifts from forward to reverse. Other shifts were handled automatically by a reversal of torque accompanying throttle closing when the foot was lifted from the accelerator pedal. In this respect it resembled a later form of the Banker transmission, which was designed for both railroad and automotive use. The Dow automatic had been designed to shift under torque, but its helical steel spring clutches and weights, energized by centrifugal weights, made for a severe control problem. After three years of test and development, the project was discontinued.

Renault also publicized an automatic planetary unit in 1924. The semi-automatic, two-speed planetary and shift gear combination contained two centrifugally operated friction disks and a separate centrifugal main clutch for elimination of the clutch pedal. This transmission never entered production for the same reasons that many previous automatics failed -- an uncertainty over acceptance of a new device that would require a large tooling investment.

During the middle and late '20s serious attempts were made to build automatic controls for motor driven railcars. Remote control was a major difficulty.

A multi-unit planetary transmission was built by the William Beardsmore Co., Glasgow, Soctland, and tested by the Canadian Railway for a number of years starting around 1925. The unit, designed by A.E.L. Chorlton, was hydraulically controlled and had multiple disk reaction brakes. Reportedly, the transmission components could not withstand the high friction loads.

In 1926, a Sykes railcar equipped with a Campbell key-shift transmission having the step gears constantly meshed took the rails at East St. Louis. The Campbell was air-servo actuated and comprised six forward speeds selected by operator lever and control valving. A flyball governor acted as a "gate" to permit the desired shift to be executed at the proper time.

Another railcar, equipped with the Banker planetary transmission was publicized in 1932-33 when it was demonstrated in the Pullman "Railplane." The transmission used triple-gear planets and sun gears for the drive combinations only, all without ring gears. The railcar unit could shift without discontinuing the torque but the passenger car version, to be covered more fully under planetary transmission developments, could not. A revised design appeared commercially in the GMC Yellow Coach of 1936.

4. NEW TECHNIQUES -- LATE '20s AND '30s

4.1 THE FRICTION ELEMENT PROBLEM

The central hindrance in the pursuit of a self-changing transmission was the problem of the change-speed friction elements. This friction problem had been an obstacle for practically every proposed automatic, except for one type that reportedly imitated the hand and foot motions of the driver using a standard transmission. These devices, however, were slow and required an inordinate amount of control machinery. They were also unable to shift under torque.

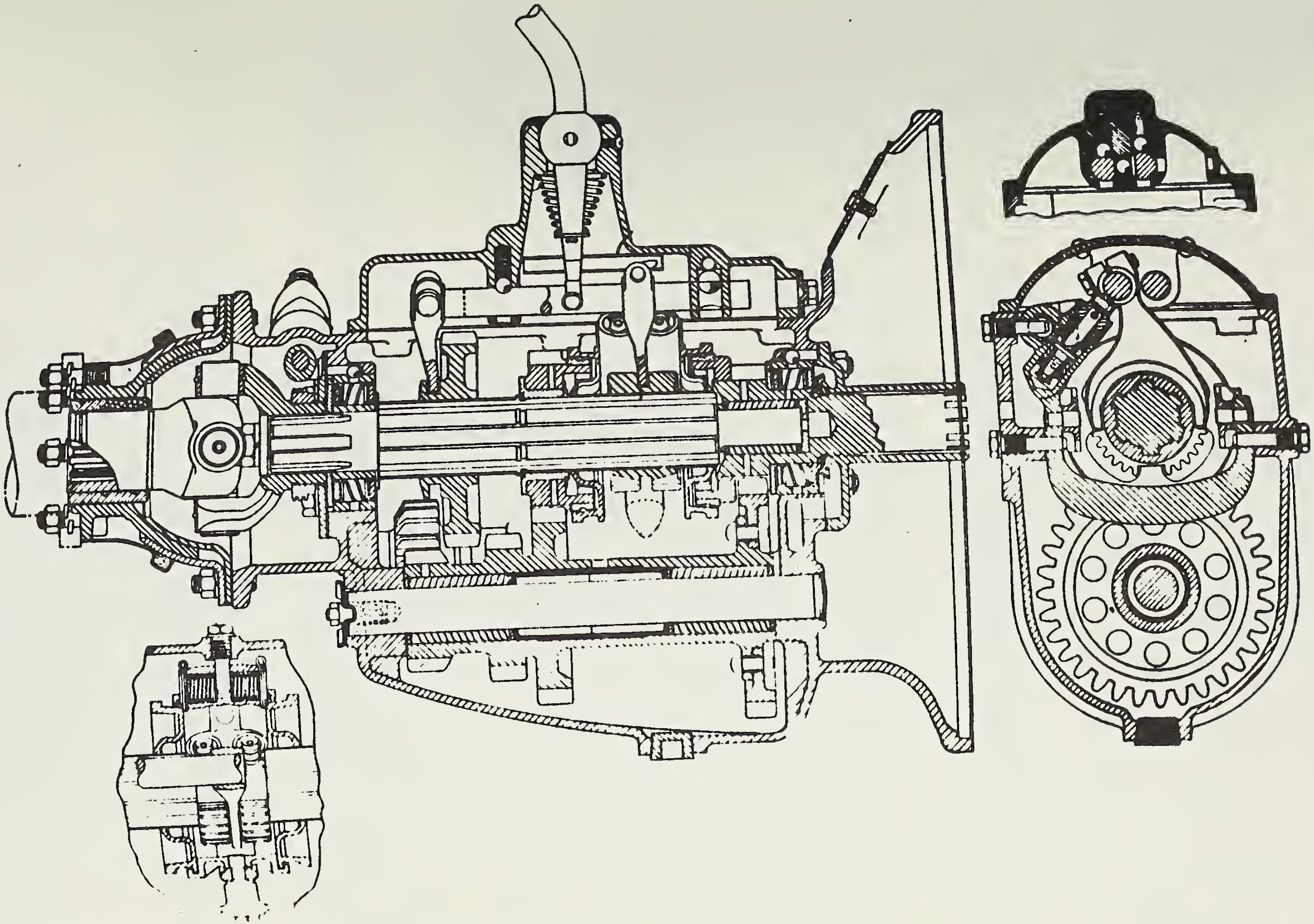
Two early types of self-shifting transmissions can be discerned in U.S. practice. One required main clutch operation with each shift effected by jaw clutches, and the other used a set of friction clutches, reaction brakes, or both. The Syncromesh design, introduced on Cadillac and LaSalle models late in 1928, is representative of the former category. It was found, however, that the synchronization machinery was too complex and prone to get out of order. Its principal obstacle, however, was a no-drive interval between gear shift positions. If the mechanism did not fully complete a shift, the drive could be lost, and some extra means had to be devised to restore operator control.

Opposed to the jaw clutch variety, the friction clutch type, of which the 1933 Reo semi-automatic was an example, theoretically allowed a progression from one ratio to another without shock while maintaining torque. The constant mesh and planetary gearsets could both be manufactured with friction clutches to be engaged either individually or in combinations to provide a change in speed ratios. However, jaw clutches were normally less expensive to produce than friction clutches for handling the same torques. As a result, the friction clutch-equipped constant mesh gearbox and the planetary unit were both more expensive than "standard" shift gear-boxes. It wasn't until 1937 that technological improvements permitted manufacturers to produce a friction clutch type transmission in quantity and in the same price range as the sliding gear transmission.

4.2 SYNCHRONIZATION

The 1929 models of Cadillac and LaSalle using the "non-clashing" Syncro-Mesh (Fig. 16) eliminated the requirement that the driver attend to the relative engine and propeller shaft speeds when either shifting up or down between intermediate and high gears. The occasional necessity for double clutching to effect a downshift was also eliminated. For proper synchronization lever action had to be smooth and deliberate, maintaining the prerequisite of driver skill that had always plagued the industry. Synchronization was furnished only on constant mesh transmissions.

The Syncro-Mesh design differed from "standard" gearsets in that the second speed gear, instead of being splined to the main shaft, rode on a bronze bushing splined to the shaft, being free to turn and in constant mesh with its corresponding gear on the countershaft. In synchronized transmissions, the gears on the main shaft had to be clutched to the



Source: *Automotive Industries,* August 25, 1928, p. 256.

FIGURE 16 SECTIONS OF NEW "SYNCRO-MESH" TRANSMISSION ADOPTED ON BOTH CADILLAC AND LaSALLE CHASSIS shaft by positive clutches. In direct drive the clutch shaft had to be connected to the main drive shaft. The positive clutches consisted of members with internal and external teeth, one integral with the gear and the other a collar splined to the driving shaft, and able to slide upon it. A bronze-faced male cone formed part of or rotated with the gear to be engaged. A corresponding female cone formed part of the sliding collar. The movement of the collar towards the gear to be engaged brought the corresponding pair of male and female friction cones into contact. Friction between the cones equalized the speeds of the gear and the shaft. When speed synchronization had been attained the toothed clutch members were pushed into engagement which coupled the gear to the shaft.

The engagement of the clutches had to precede engagement of the toothed clutch by a certain length of time. Two plunger operated dash pots mounted on the clutch operating collar insured the proper time interval.

The 1930 version used on the Cadillac and LaSalle was generally heavier, with wider gear faces and larger bearings. The most important change, however, was the moving of the bronze lining from the male to the female member of the cone clutches. The bronze lining was moved since the greater heat expansion of the bronze tended to tighten it in the drum rather than loosen it from the steel cone. A greater reduction was provided for second gear, with a lower reduction in low gear and reverse, being 1.79 in second, 3:1 in low and 3.5:1 in reverse.

The weak point of the early designs was a demand for excessive pressure in order for them to synchronize quickly. It was also difficult to shift gears while the car was at a standstill because of the necessity of resorting to stronger detent springs to insure quick, quiet shifts. Lighter springs would not create a strong enough force for a fash shift, leading to more gear crashing.

A later GM design consisted of three spokes terminating in teeth radiating from a drum. The teeth engaged the grooves of the splined shaft. Three humped springs were located in the grooves of the shaft. As a sliding member was moved toward the intermediate gear, it came into contact with the hump on each spring. The sliding member moved the spring into flat contact with the spokes of the drum which was pushed into contact with the friction cone of the gear.

As the gear approached synchronization, the resistance from the secondary shaft, clutch shaft, and various parts attached to them, vanished as the force on the sliding member, through the drum spokes, was converted into force on the drum to overcome the resistance of the springs' inclined surfaces.

The clutching of the intermediate gear to the splined shaft was performed by the teeth on the sliding member meshing with the internal teeth on the intermediate-speed gear extension. Pressure on the drum and the friction between the drum and cone was relieved when the chamfered surface of the sliding member passed the drum spoke, allowing the positive clutch members to become engaged.

Another early synchronizer, used by the 1931 Graham-Paige, and produced by Warner Gear, was used only between third and fourth speeds. The forward end of the mainshaft was splined and carried a sliding head on which were mounted two female conical surfaces provided with bronze liners. These corresponded with male cones carried on extensions of the main drive gear and of the third speed main shaft gear. Sliding on the clutch head was a toothed sleeve. The internal teeth fit the external teeth of the clutch member. Carried within the clutch member was a series of six spring-pressed plunger ball poppets. A groove on the inside of the sliding sleeve matched the contour of the ends of the poppets.

During a shift into second, the sleeve and head moved together until the female cone of the head contacted the male cone of the second speed gear. The resistance created by the poppets caused the clutch head to slide along the splines on the main shaft into engagement with the mating cone.

A smaller series of Cadillac and LaSalles introduced for the 1937 model year contained a number of transmission and synchronizer modifications. The new transmission, for Series 50 and 60, had all helical gears, with low and reverse of the sliding type mounted on helical splines while the second speed was constant mesh. Gear diameter was reduced further which decreased tooth speeds providing easier shifting and reducing wear. A pin-type synchronizer was used. Three tapered pins passed through holes in a flange on the sliding clutch member. The insertion of the pins through the holes, provided the inertia-overcoming force to equalize speeds of gears and shaft.

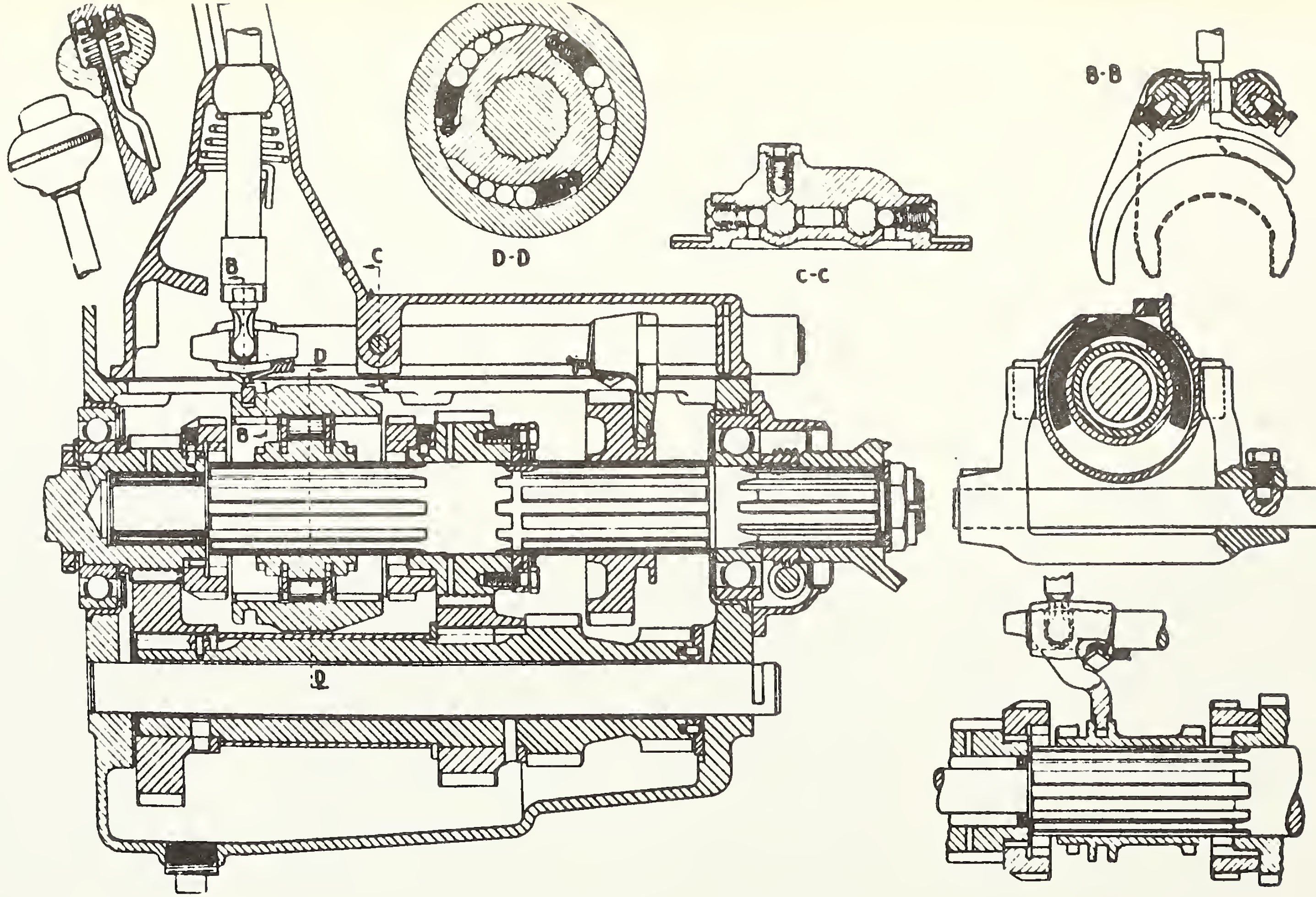
4.3 FREE WHEELING

Free-wheeling, introduced by Studebaker (Fig. 17) in the middle of 1930, also produced benefits in terms of ease of operation in some situations. By September 1931 free-wheeling was available on almost half of all U.S. models (38 of 78), though it was standard on only fourteen.

Free-wheeling allowed a car to drift or glide whenever car momentum on a downgrade or otherwise tended to make it travel at a higher speed than that corresponding to engine speed by automatically disconnecting the engine from the transmission.

Free-wheeling had already been introduced on some British autos by the time Studebaker's device was available. However, Studebaker improved on the British design by incorporating it into the transmission proper rather than just connecting it to the back.

The Studebaker transmission consisted of two pairs of constant mesh helical gears. It also possessed a unique arrangement for the engagement of the intermediate and high gears which permitted free-wheeling in either gear whenever the accelerator was released. The device allowed easy shifting between high and intermediate without the necessity for releasing the clutch and without bringing the car and engine speeds to synchronization as ordinarily required.



Source: *Automotive Industries*, July 12, 1930, p. 41.

FIGURE 17 STUDEBAKER FREE-WHEELING TRANSMISSION AND CLUTCH – 1930

The "pineapple" free-wheeling unit consisted of an inner cam member, splined to the main driving shaft, that supported an outer sleeve having inner teeth at either end arranged to engage corresponding tooth members on the forward pinion gear of the rearward second speed gear. Twelve rollers arranged in three groups were interposed between the inner cam member and the outer shell. When the unit was shifted forward so that the outer member only was in engagement with the main driving gear, the torque was transmitted through the roller clutch to the main drive shaft.

Free-wheeling was credited with awarding the driver an easier task of shifting even if the speeds of the two gears were considerably out of "sync" since there was only the lower inertia of the clutch sleeve to overcome.

Free-wheeling and the introduction of the "silent" second (or third) gear worked considerable change in transmission design. The conventional sliding gear transmission had a monopoly of the U.S. market in 1927. By 1932, its share had fallen to 26 of the 78 models, while 52 models used constant mech transmissions. Thirty-two free-wheeling transmissions used helical gears, with four equipped with herringbone gears for the "silent" second speed.

Consumer reaction to free-wheeling was most positive in regard to the reduction of operations required in shifting, and least favorable to the "coasting" element which often felt unsalfe to the average driver.

4.4 AUTOMATIC CLUTCHES AND THE FLUID COUPLING

The automatic clutch was quickly perceived as offering the same advantages of the free-wheeling roller clutch with a superiority in some areas, provided good synchronizers were applied. In a great number of cases free-wheeling accompanied the automatic clutch.

Automatic clutches developed in the late 1920's made the constant mesh transmission feasible since the gears did not have to be shifted into and out of neutral. The automatic clutch, contrary to ordinary clutch practice, was disengaged until engine speed increased. It mimicked true automatic operation since the car could be started by just pressing the accelerator and stopped by brake with no intervening operation.

With the mechanical variety, engagement had to be positive above the engine speed at which the maximum torque was developed. With the "fluid flywheel" the clutch retained its flexibility and shock absorbing characteristics by maintaining some slippage through the whole speed range.

The turbine-bladed fluid coupling held a considerable advantage over vacuum and centrifugal clutches since it could be shifted under torque in a continuous fashion.

The invention of the fluid coupling is credited to Hermann Föttinger, an electrical engineer employed by the Vulcan Ship Yards in Hamburg. Around 1905 it occurred to him that a hydraulic reduction gear could be used between ship turbines and the propeller. It was his concept to place both the impeller of the centrifugal pump and the runner of the turbine in the same housing, obtaining a compact unit and eliminating losses due to the flow of liquid through the connecting piping. Föttinger also arrived at the idea of including a reaction member consisting of a set of vanes fixed in the housing to increase torque. His torque converter gave a speed reduction of about 5:1 and showed peak efficiency around 85%.

In 1919, a fluid coupling-equipped Radcliffe automobile was put on display at the New York auto show (Fig. 18).

Harold Sinclair, a British engineer, was instrumental in applying the hydraulic coupling to motor vehicles. Around 1926, while studying the use of such devices in ship hoists, he perceived that a hydraulic coupling could improve bus operation, particularly on routes run at low speeds and having frequent stops.

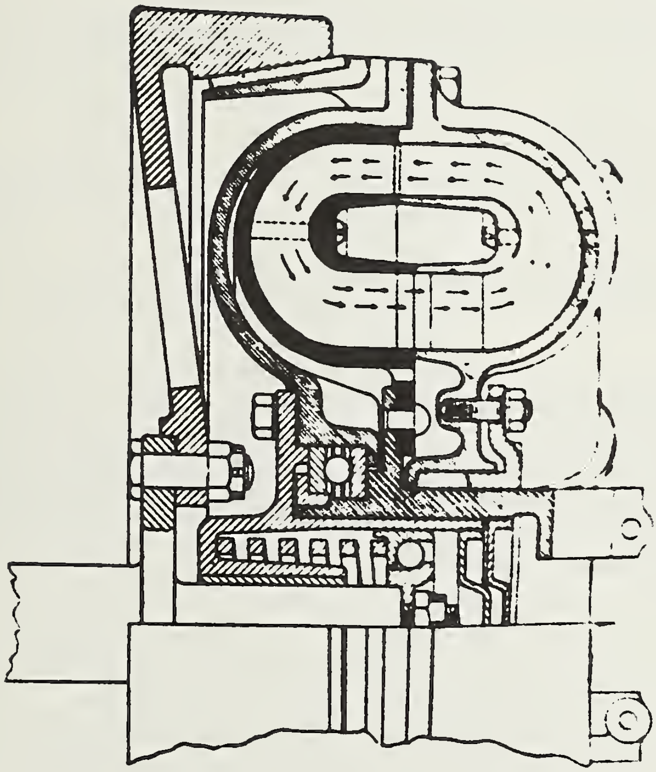
Another German engineer, Rieseler, was one of the pioneers in applying the fluid coupling (his experimental models actually employed torque converters) to automobiles. His work was reported between 1923 and 1928. He equipped several Buicks and Mercedes with hydraulic torque converters in combination with two-speed-and-reverse planetary tranmsissions of the all-spur type (as on Ford's Model T).

Power was first transmitted through the hydraulic unit and the low gear. Next a multiple disk friction clutch would lock the torque converter by fixing its housing to the engine flywheel. For direct drive, the planetary unit was locked together by its friction clutch. There were actually three friction clutches, one for low forward speed, one for reverse, and the other as a vehicle brake, a not uncommon practice in early transmissions.

The Daimler Motor Co., Coventry, England, adopted a fluid coupling with a planetary gearset, in conjunction with a pre-selective feature, during 1930.

In the fluid coupling, the driving member (impeller) in most conditions rotated faster than the driven member (runner, or, as it is known today, the turbine), respectively. At start-up, large differences existed between the kinetic energy of the fluid in the driving and driven members. This difference is given up to the turbine during each fluid circuit.

The fluid in the impeller, under the influence of the centrifugal force, enters the turbine near its outer circumference and forces liquid from the runner into the impeller near the inner circumference. The circulation



Source: *Automotive Industries,* June 18, 1925, p. 1055.

FIGURE 18 RADCLIFFE HYDRAULIC DRAG TYPE OF TRANSMISSION

continues as long as there is a difference in speed between the impeller and turbine.

Originally, Daimler had used a friction clutch with the fluid flywheel because the driven member of the fluid flywheel had too high an inertia to permit easy shifting of a sliding gear transmission. By 1931, Daimler had adopted the Wilson "self-changing" gear, a planetary transmission controlled by a preselecting device. A pedal was depressed to effect a shift to the selected gear. A hydraulic device was thus combined with the Wilson gearset which had been used in Britain as early as 1898.

Subsequent European applications followed the example set by the Wilson pre-selective transmissions as used in buses and the Daimler vehicle. There, the majority of fluid couplings were used with planetary sets that incorporated friction members so that no separate friction clutch was required to separate the engine output from the gear box input. In the U.S., through the 1940's and early 1950's, fluid couplings were used mostly with geared transmissions whose individual trains were engaged by positive clutches.

From 1930-35, the fluid coupling alos appeared in the experimental Tri-Lok transmission, the Vickers-Coats, and the Salerni device.

The Salerni transmission, invented by Commendatore Piero Salerni, London, England, combined a fluid flywheel with a sliding gear or constant mesh transmission. The goal of the device was to utilize the fluid coupling through the whole speed range to eliminate dependency upon friction for the transmission of torque. However, shifts were accomplished when the transmission was "unloaded," that is, when torque was not transmitted, creating the danger of losing the drive.

When the clutch pedal was depressed prior to a speed change, a sliding valve within the fluid coupling would close, preventing fluid circulation and torque transmission. Further pedal movement disengaged a mechanical coupling to the rear of the transmission. The final degree of movement applied a friction brake and brought the turbine to a fast stop. Since the transmission was completely disconnected and stationary, a shift could be made into any position without gear crashing.

When the fluid flywheel was used, it replaced the engine flywheel and formed a flexible link which prevented the transmission of road shocks to the engine. Because of its high inertia, it could not replace the friction clutch used in conjuction with a sliding gear or a constant mesh transmission in which the different speeds were engaged by sliding gears into mesh or by engaging positive clutches. For this reason its most successful applications were with planetary transmissions which required no separate friction clutch.

4.5 OTHER HYDRAULIC CONTROLS

The use of hydraulics was, of course, not limited to clutch-type applications. In the first few decades of the automotive industry, hydraulic transmissions were experimented with. The concept was to pass fluid under pressure through a hydraulic motor to generate mechanical power.

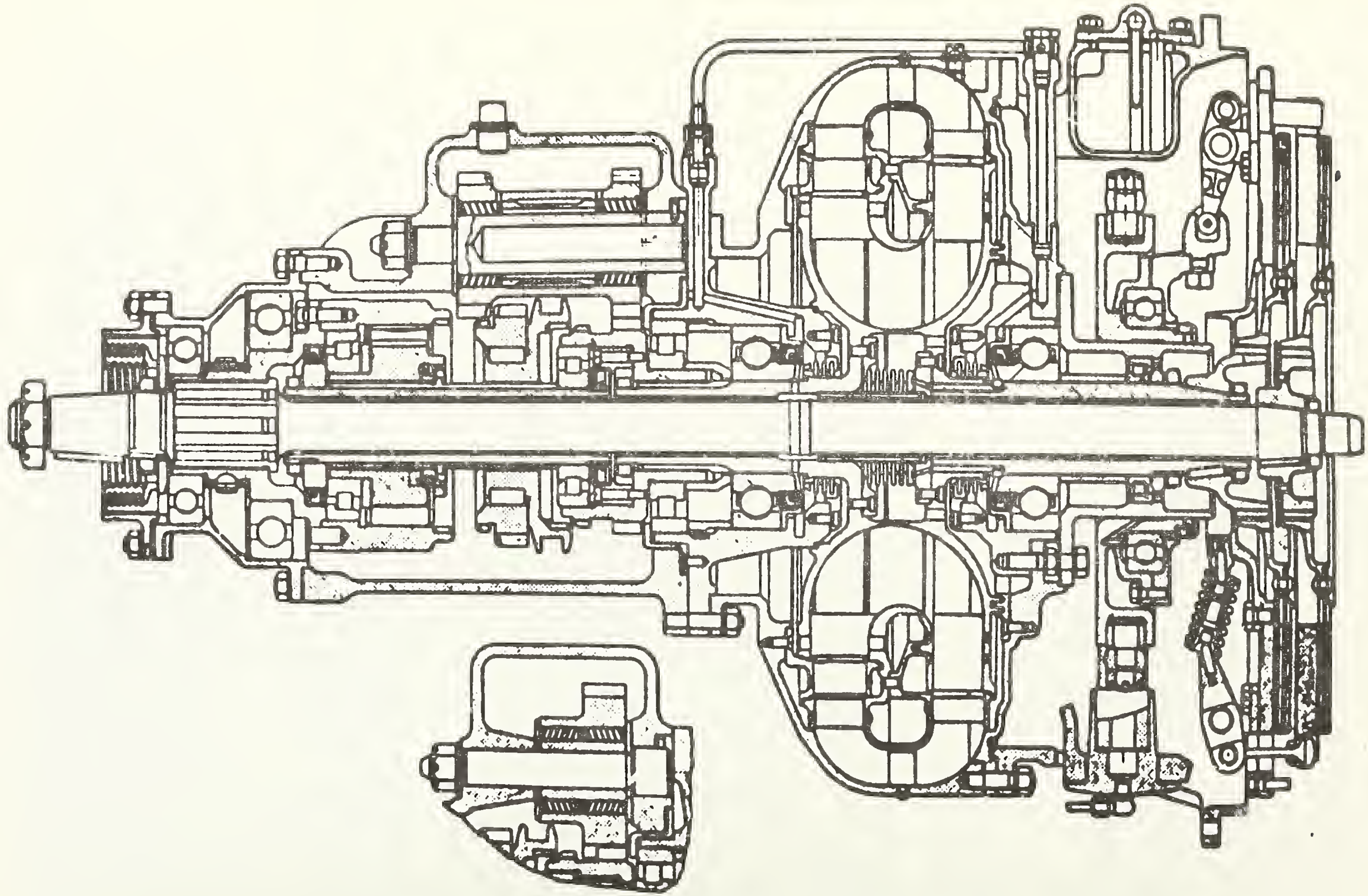
In this hydrostatic-type transmission, mechanical power was generated by sustained pressure from the fluid acting on moving pistons or vanes. It was attractive because a continuous variation in ratio could be achieved. To change ratio, the stroke and working pressure of the pump could be varied. For instance, if the pump stroke was reduced by one-half and pressure doubled, and the rotational speed of the pump and the horsepower input remained the same, twice the fluid pressure would double the torque on the shaft of the hydraulic motor, while the speed was halved, because of the lower rate from the pump.

As engine power increased, the hydrostatic transmission becaome more unsuitable for automotive use. The conversion of rotational into hydraulic energy then back again into mechanical energy entailed unacceptable efficiency losses. Its peak efficiency, between 75% and 80%, was reached only at low speeds. Efficiency fell rapidly as the velocity of the flow increased, since the losses varied with the cube of the flow velocity while power varied only directly with flow velocity.

By comparison, in the hydrodynamic or hydrokinetic transmission, a fluid is set in motion by an impeller provided with a series of vanes, just as in the "fluid flywheel" or hydraulic coupling. But a third member was added to distinguish the "torque converter" from the fluid coupling. The stator performs as the reaction member and redirects the fluid flow from the turbine back against the impeller until all available energy is extracted from the fluid. This helps to turn the impeller (engine) which in turn helps to push the turbine.

The blades of the torque converter differed from the design required by the hydraulic coupling by being more "stream-lined," set at an angle with the trailing edge sloping backward relative to the direction of motion. Multiple stage runners and impellers were also used but added to the cost, and were therefore normally limited to the turbine. Variable stator vane angles were developed later for torque converters in the 1958 Buick.

One of the earliest commercialized transmissions combining an hydraulic torque converter with a gear train was perfected by the Ljungstram Steam Turbine Co., Stockholm, Sweden, around 1928 (Fig. 19). The transmission, known as the Lyshom-Smith, was produced under license by Leland, Spicer and Krupp Works. As first proposed, the unit consisted of a direct-drive clutch, torque converter with a multi-stage turbine, a disconnecting



Source: *Automotive Industries*, April 10, 1937, p. 565.

FIGURE 19 LYSHOLM – SMITH HYDRAULIC TORQUE CONVERTER FEATURING DIRECT-DRIVE

mechanism for the impeller and turbine while in direct drive, and a reversing gear. A friction clutch was included to disconnect the impeller. The runner was designed to drive the output shaft through a roller clutch so that it became isolated when the transmission was in direct drive.

A planetary transmission with hydraulic control, the de Normanville safety gear (Fig. 20), was offered as an option on three models of the Humber line in 1935. A compound plunger oil pump was driven by a worm gear at the front end of the gearbox. The pump forced oil into an accumulator which contained powerful springs which forced a piston forward. At a certain point in its stroke, the piston uncovered small ports. Any oil entering the reservoir was passed out into the axial hole through the main shaft from which the whole transmission was lubricated.

The brake drums working on the planetary gear set were operated by an hydraulic cylinder. When fluid under pressure from the accumulator was admitted between the two pistons which controlled the lower end of the brake shoes, the shoes were applied to the brake drums.

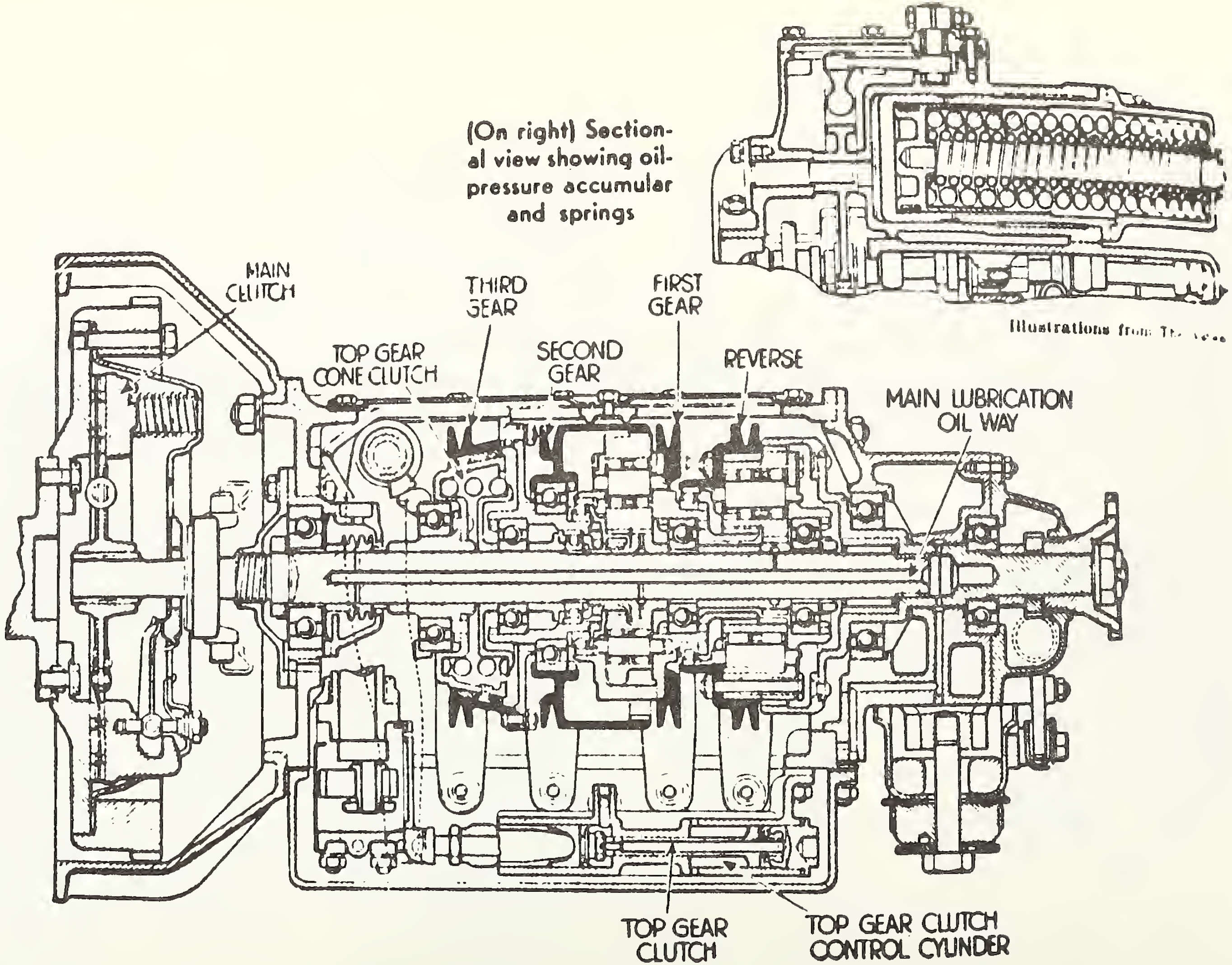
The de Normanville system is still employed in the Laycock overdrive, supplied by the British firm GKN.

4.6 VACUUM CLUTCHES AND SHIFT CONTROL

Coincident with the early evolution of the hydraulic torque converter and fluid coupling in the automotive transmission was the use of vacuum clutches, and the control of the gear shift through a vacuum power unit making use of intake manifold vacuum. The vacuum shift control was introduced around 1930 by the Bendix Aviation Corp. at the same time as synchronizing devices were being employed. For the same reason that synchronization was not initially popular, a demand for more than ordinary skill for smooth operation, the vacuum clutch and control were not immediately satisfactory.

A vacuum-controlled clutch was normally held engaged by springs. To make a shift, the driver momentarily released the accelerator pedal which closed the throttle valve resulting in a great increase in manifold vacuum. The vacuum drew a piston in a power cylinder, as previously described.

By 1932, Buick, Reo, Cadillac, LaSalle, Dodge, DeSoto and Chrysler had adopted this scheme of clutch control. The last three incorporated a roller-clutch type of free-wheeling device to accompany vacuum control. When the accelerator was released, the transmission became disconnected from the high inertia elements at both ends with the result that gears could be shifted up or down with ease. An interlocking device was always provided so that when the transmission was free-wheeling, the automatic control of the clutch was rendered inoperative, allowing the engine to idle as the car coasted along.



Source: *Automotive Industries,* April 6, 1935, p. 430.

FIGURE 20 SECTIONAL VIEW OF DeNORMANVILLE SAFETY GEAR TRANSMISSION

In the Dodge transmission of 1932, two control valves were located in a housing at the intake manifold, the lower being interconnected with the free-wheeling lockout, while an upper valve was connected to the accelerator pedal. Since a free-wheeling capability was included, in addition to the automatic clutch, it was not necessary to provide metering valves to vary the rate of clutch engagement to accord with gear ratio. Instead, the rate of engagement varied according to throttle position by air bleed from the operating cylinder back to the main control valve. Therefore with the throttle wide open the clutch engaged faster than at lesser throttle positions.

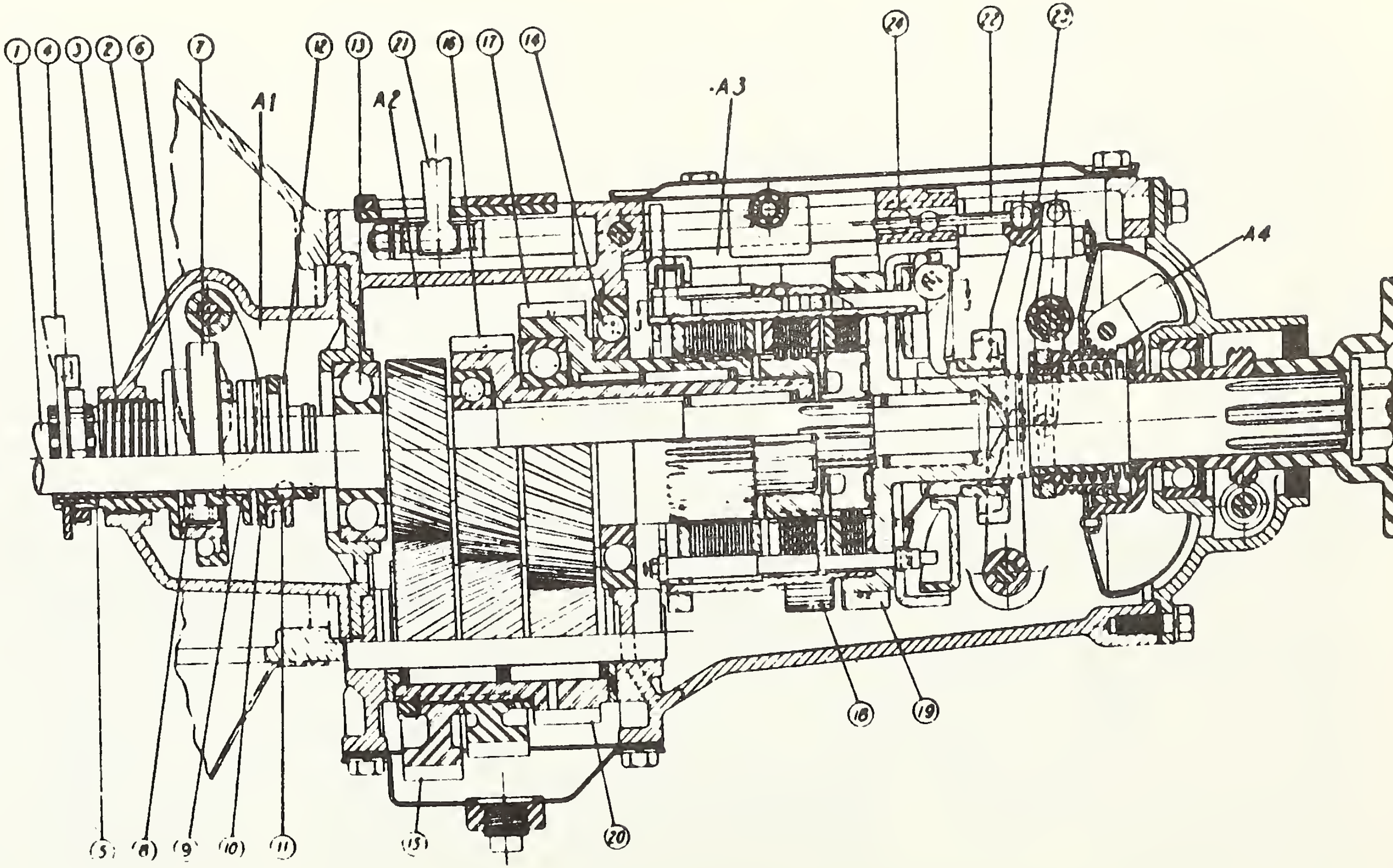
Around the same time the Tyler Unimatic transmission was developed (Fig. 21). It was an all-helical constant mesh unit utilizing three small multiple disk friction clutches engaged by springs to make the normally free turning gear of each combination fast to the main shaft.

Vacuum was selectively admitted to one of three diaphragms, by means of a slide (shuttle) valve which was activated by a centrifugal governor on the main transmission shaft. A master control valve, controlled by a button within or adjacent to the accelerator pedal, admitted vacuum to the main clutch actuator chamber and the valve chamber for the shifting compartment. Power actuation of the clutch and automatic gear shifting were possible only when the master control button was fully retracted. The button projected slightly above the accelerator pedal. By pressing the button to the level of the accelerator when the pedal was in the idling position the car was able to free-wheel.

In operation, the Tyler automatic shifted into low as soon as the engine was started. The car did not move immediately since the high vacuum in the inlet manifold kept the clutch disengaged. At higher engine speeds, the inlet manifold vacuum decreased which allowed first the transmission gear clutch to engage and then the main clutch. As the car accelerated in low gear, the selector was moved to second or intermediate position. The shift to this position was completed when the driver released the accelerator pedal.

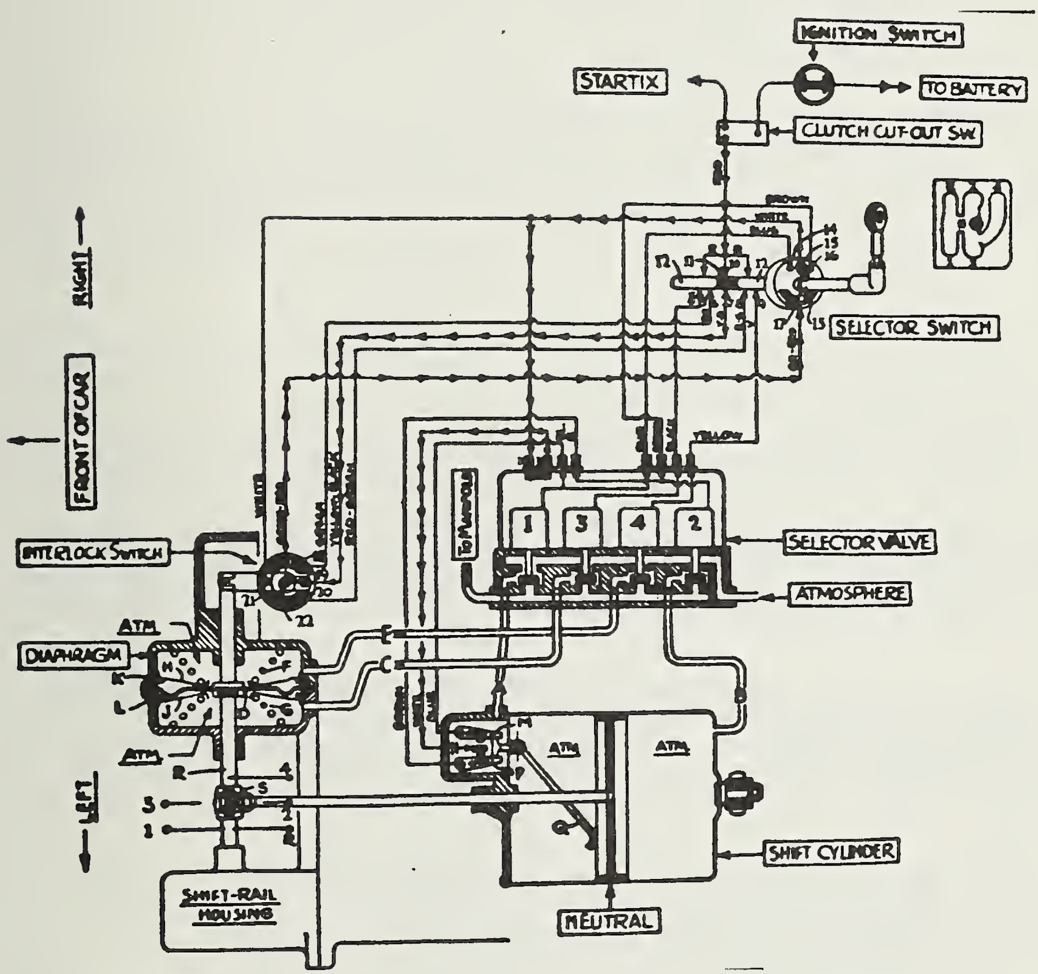
The 1936 Cord applied a Bendix vacuum gear shift control (Fig. 22), and made use of atmospheric pressure. In the Bendix system, if a selector lever was moved into the first speed position, with the engine running and the clutch disengaged, battery current would flow to a solenoid which lifted a valve to connect a tube to the vacuum line from the intake manifold. A diaphragm in a cylinder was then exposed to atmospheric pressure on its top and manifold vacuum at its bottom and moved a cross shaft and completed the shift. It was possible to move the selector into any position at any time but a shift occurred only when the clutch pedal was depressed to complete the circuit.

Hudson utilized vacuum-power for clutch operation and gear shifting for a number of years, but retained an option for the driver to revert to



Source: *Automotive Industries*, April 8, 1933, p. 424.

FIGURE 21 TYLER UNIMATIC TRANSMISSION



Source: *Automotive Industries,* November 9, 1935, p. 635.

FIGURE 22 WIRING DIAGRAM OF THE BENDIX VACUUM CONTROLLED GEAR SHIFT

hand shifting and foot-controlled clutching. Improvements in 1937 included an automatic return of the selector lever to neutral if the constant mesh gears would not entirely engage.

An interesting hydrid device was developed and marketed by the Long Manufacturing Co., of Borg-Warner Co. The Tri-Vac was essentially centrifugally actuated, but was also provided with a vacuum control to override the centrifugal action under certain conditions. The vacuum cylinder piston was connected indirectly to the clutch reaction plate on which the centrifugal mechanism was mounted.

The Tri-Vac was developed in an effort to overcome some of the major complaints about the centrifugal clutch, which were that it was not possible to start a car by pushing or towing since the clutch was not engaged, linings wore quickly under full throttle acceleration from a standing start because of excessive slip, and difficulty in pulling long grades at low speed without excessive slippage.

4.7 RETURN OF THE PLANETARY TRANSMISSION

Even as the planetary transmission was disappearing from the U.S. with the demise of the Model T, the seeds were being sown for its resurrection. Focus on the reduction of noise, improvements in efficiency and ease of operation through the constant mesh and silent second and third gears raised the possibility in some minds that it would be profitable to give the planetary transmission another chance.

It was recognized from previous experience that the planetary gear train elminated gear crash and required no special skill in operation. Also, losses in direct drive were lower than a constant mesh transmission with a countershaft since there was no excessive churning of the heavy lubricant by revolving gears.

The first evidence that a planetary gearset more complex than the early two-speeds could be successfully produced was provided by British experience with the Wilson transmission which replaced the sliding gear on Daimler's hydraulic clutch-controlled car. The main problem with the Daimler was that with the friction clutch in combination with the hydraulic clutch only added cost and weight without reducing system complexity.

A later transmission based on the Wilson design, intended for railroad application, was constructed by Improved Gears Ltd., London, and constituted five forward speeds through four planetary trains. A pneumatic servo system controlled the transmission through two air chambers provided on the gearbox which engaged either a forward or reverse drive bevel gear in the transmission shaft, and a single air cylinder which engaged the forward speeds.

Among the earlier deficiencies of planetary transmissions which were corrected in these later Wilson units was the support of the planet gears only at one end and with plain bearings, which proved difficult to lubricate. These planet gears tended to deflect under high tooth loads and would become misaligned. Eventually needle bearings were installed in place of the plain bearings.

Also in the early American planetary gearsets the brakes for locking the various gear trains were usually applied by direct mechanical action from a gear change lever or pedal. In one group of Wilson transmissions, once a gear was selected through the pre-selector lever, the brake was contracted on the drum of the gear by the force of a spring transmitted through a toggle mechanism.

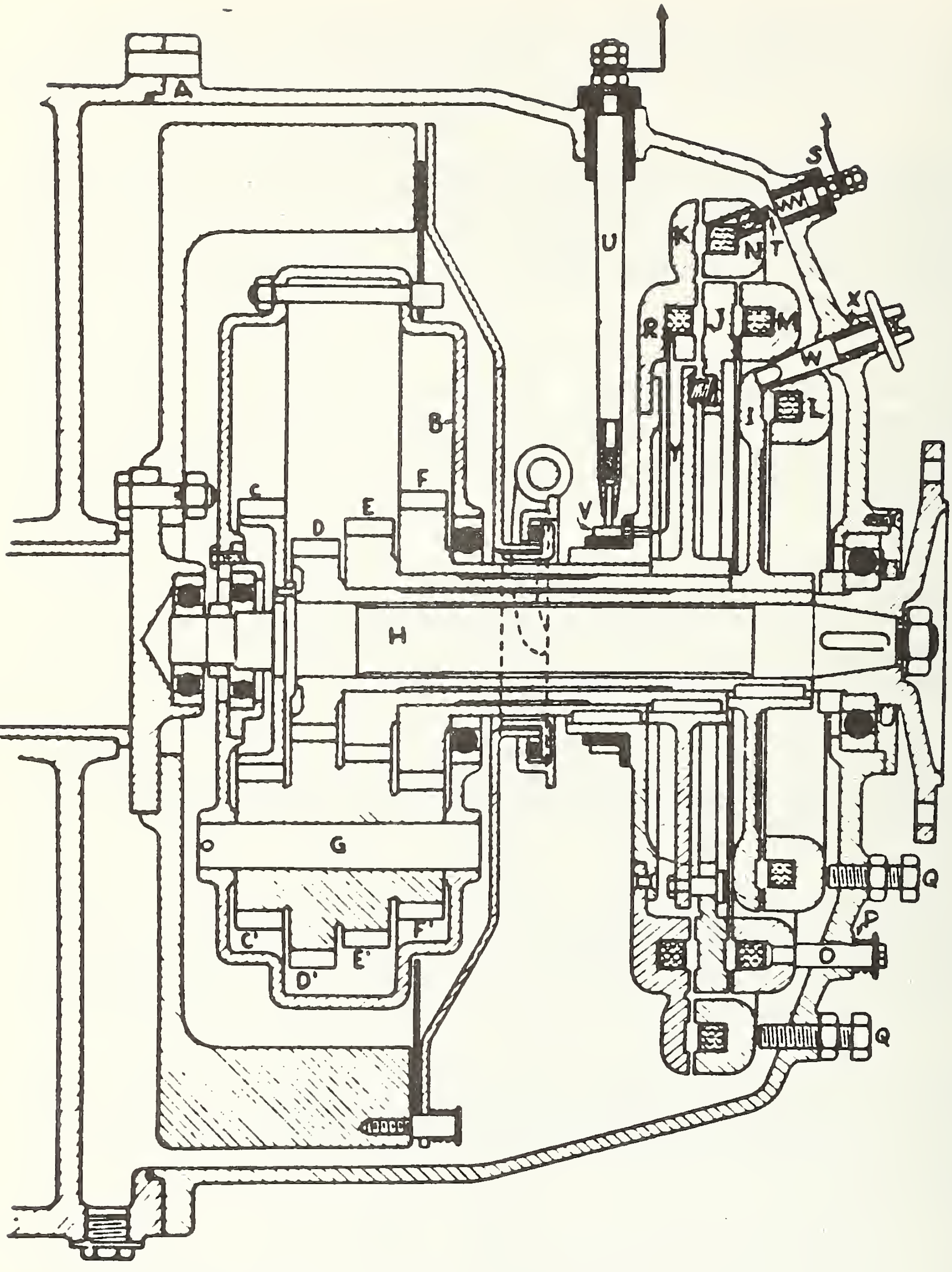
Larger versions were remotely controlled by pneumatics or through an equivalent hydraulic servo. With friction bands controlled in these ways, the release of the bands could be accomplished gradually while the next ratio was simultaneously brought into operation by the other friction clutch. Therefore, the problem of maintaining torque during a gear change was cured.

However, a fool-proof control of the brake clutches to provide a smooth transition remained one of the most stubborn problems associated with the planetary transmission, despite the experience acquired with the U.S. and European models.

An electro-magnetic effort at brake band control was made around 1925 in France with the Cotal transmission (Fig. 23). It was an all-spur gear unit and applied magnetic clutch control. A casing containing four sun gears surrounded the flywheel. The sun gears engaged with three sets of planetaries. The first sun gear was keyed to the main shaft while the other three had sleeves of decreasing length mounted concentrically. A steel disk keyed to the sleeve formed part of the second sun gear. Plates were also keyed to the last sun gear. Opposite the plates were circular magnets. Electric current could be sent to the desired magnet for the desired gear change.

Later Cotal transmissions were in a different configuration. Of the four magnetic clutches, the first was secured to the driving gear and the last to the driven shaft while the middle two were secured to the housing. An armature located between the first two clutches had a ring gear with dual planets secured to it.

Another developmental planetary transmission, the Banker, was first tested for railroad use and later for automotive. Torque reversal was the



Source: *Automotive Industries*, June 17, 1926, p. 1045.

FIGURE 23 SECTION THROUGH THREE-SPEED COTAL PLANETARY GEAR WITH MAGNETIC CLUTCH CONTROL

means of accomplishing the locking of the brake clutches. In a version displayed around 1933, the car equipped with the transmission shifted from low into second at a speed about 5 mph while a shift from second to high could take place at any speed above 9 or 10 mph. A shift could also be made directly from low into high at a speed above 10 mph by releasing the accelerator for a longer period of time.

The Mono-Drive (Fig. 24), as the Banker was called in this later version as used in the GM Yellow Coach, engaged its brake clutches in a centrifugal manner. The planet carrier was mounted on an overrunning clutch so that when torque was applied it remained stationary with the reaction taken by the case. When torque reversed the carrier would rotate. The GM buses acquired forward drive by shifting a sliding clutch member to engage the outer member of the roller clutch, locking the clutch to the housing. First speed was secured by having the sliding clutch in the forward position, which held the planet carrier against rotation by the roller clutch.

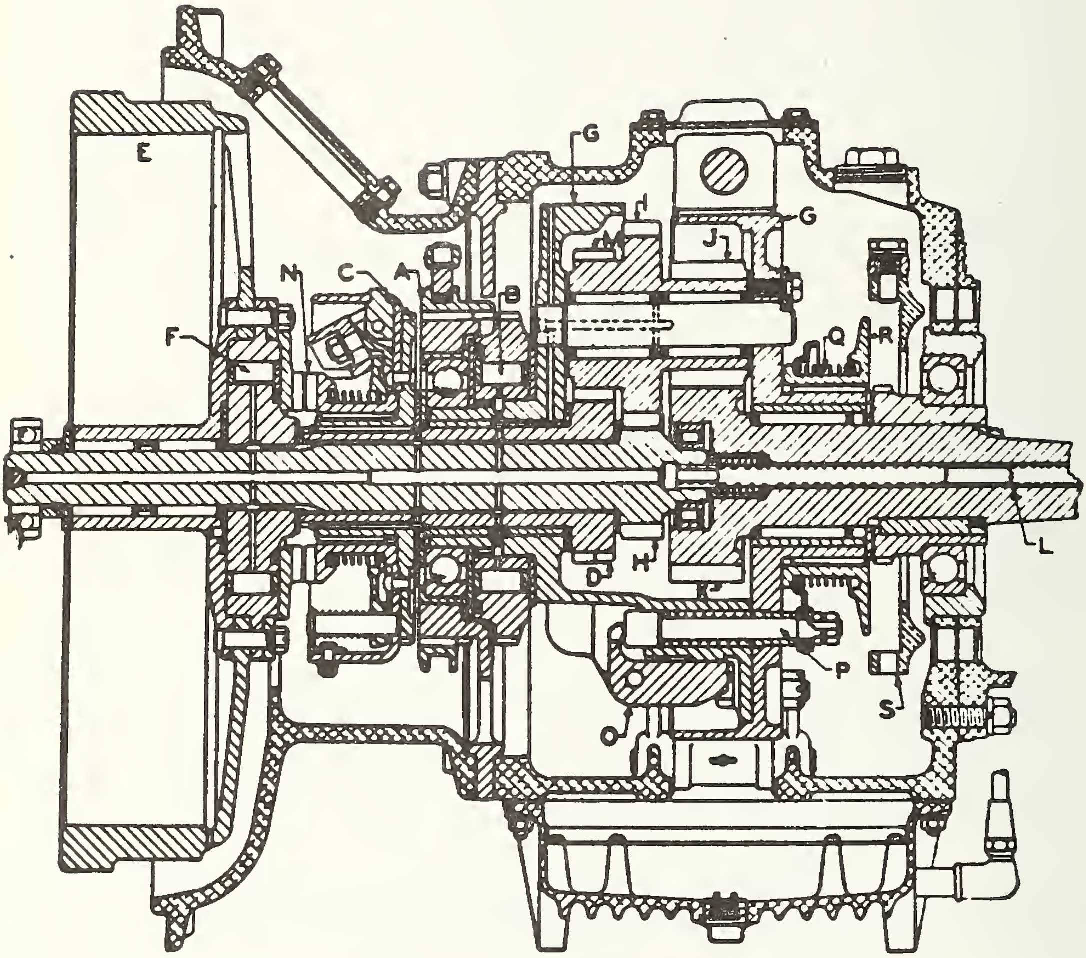
One of the earliest revivals of the planetary gear in a production automobile was by the Auburn in 1931 that utilized a planetary gear in the axle. A sun, ring and five planets were mounted directly on the axle shafts with the sun gear splined to the shaft to provide overdrive.

The next significant episodes in the revival of the planetary transmission were the introduction of the Reo Self-Shifter in 1933 and the Warner Gear automatic overdrive on Chrysler vehicles in 1934.

The Warner overdrive was essentially a two-speed transmission added onto a three-speed manual unit. In an early version direct drive was achieved through a roller-type overrunning clutch which disconnected the planetary member from the output shaft.

The major problem with the early Warner overdrive units was that due to the use of a speed-sensitive device known as the Keller clutch, the change from direct to overdrive occurred always about the same speed, around 45 mph, and back again at a slightly lower speed. This meant that the driver could not downshift at will, nor were the upshifts load responsive.

Solenoid control later supplanted the Keller clutch, by which the driver would release the sun gear, which was prevented from rotating during overdrive operation, and downshifts were possible at any speed.



Source: *Automotive Industries,* April 24, 1937, p. 626.

FIGURE 24 **LONGITUDINAL SECTION OF YELLOW COACH AUTOMATIC SHIFTING TRANSMISSION (MONODRIVE PATENTS)**

5. EVOLUTION OF THE MODERN AUTOMATIC TRANSMISSION

5.1 PRE-WORLD WAR II DEVELOPMENTS

The Reo Self-Shifter (Fig. 25) introduced as standard equipment on the Royale and an option on the Flying Cloud in 1933, was the first U.S. attempt at an automatically shifted transmission based on planetary gearing.

This device was actually a combination of two reduction units: a two-speed sliding gear transmission operated manually, combined with an automatically-actuated planetary gearset.

The internal-external gear train was unique in American practice, though this type had been used abroad since the turn of the century.

The Reo car was put into motion in the conventional manner of clutch, shift and declutch. When the accelerator was first depressed, the transmission was in "low-high" having a reduction of 2.07:1. When the car reached between 10 and 15 mph, a centrifugal multiple disk clutch running in oil gradually engaged the direct drive. The transmission remained in high until car speed dropped below 10 mph. When additional power was required, or when it was desired to accelerate rapidly in low gear to speeds up to 50 mph, the control handle had to be pulled all the way back to engage the low range. The centrifugal clutch was effective in this range as well, so that there was an automatic shift from low to a "high gear" in the low range, and vice versa, providing flexibility.

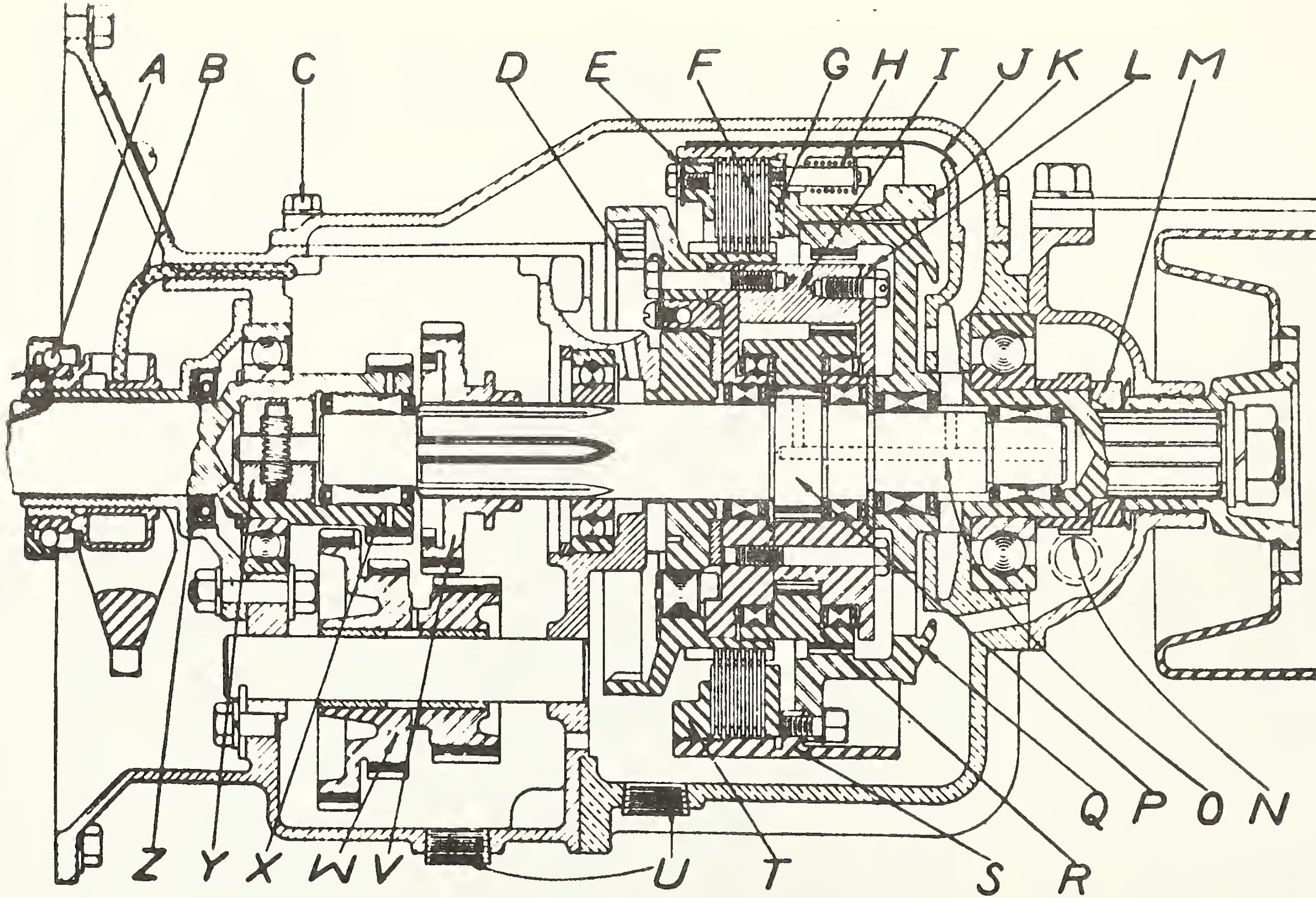
The unit was shifted by centrifugally operated weights which acted through a connecting clutch to lock the gears in the planetary set, resulting in direct drive at a predetermined speed. The unit automatically shifted from a low to high ratio in each of the two ranges.

The four gear ratios were:

Lo-Lo	2.73:1
Hi-Lo	2.07:1
Lo-Hi	1.33:1
Hi-Hi	1.00:1

The direct drive clutch was engaged by gradual squeezing of the stack of multiple disks by a set of seven governor weights. The friction clutch suffered from the traditional slippage problems of centrifugal clutches. In addition, the self-shift control was improperly designed and tended to hunt between ratios, causing slippage and overheating. A successful automatic control would have to be able to make the transmission complete a shift, and not hunt afterwards, as well as instruct the unit when to shift.

The Reo had an additional handicap in the lack of optimization for shift points, which were speed sensitive rather than load sensitive, resulting in poor acceleration. Because of the expensive tooling and the service problems of the new transmission, in conjunction with low public acceptance, Reo appears never to have made a success of its semi-automatic transmission.



Source: *Automotive Industries*, April 29, 1933, p. 530.

FIGURE 25 REO SELF-SHIFTER TRANSMISSION

Although the Reo did not live up to expectations, it suggested that automatic control was within reach and that the market interest was strong.

In 1937 General Motors introduced their Oldsmobile "Safety" transmission (Fig. 26), so called since it allowed the driver to keep both hands on the wheel. It was also produced as an option for the Buick. The Safety transmissions had many of the ingredients of the Hydramatic that went into production in 1939 for the 1940 Oldsmobiles, and a little later for the 1941 Cadillac.

The Olds transmission was made up of three distinct geartrains--a conventional sliding gear assembly at the forward end which determined forward or reverse, a single-reduction planetary unit in the center and at the rear a double-reduction planetary set. A manually operated friction clutch was provided for starting and stopping.

The Olds Safety unit was the first transmission to apply the hydraulic servo principle to control the clutch loading of the friction bands. The hydraulic technique paved the way for commercialization of the automatic, since it was possible to use less than one-third of the space required for a mechanical automatic shift system. All four gears were shifted automatically. The Olds unit's shift points were more load responsive than the Reo's and its performance was on a par with other vehicles of the day.

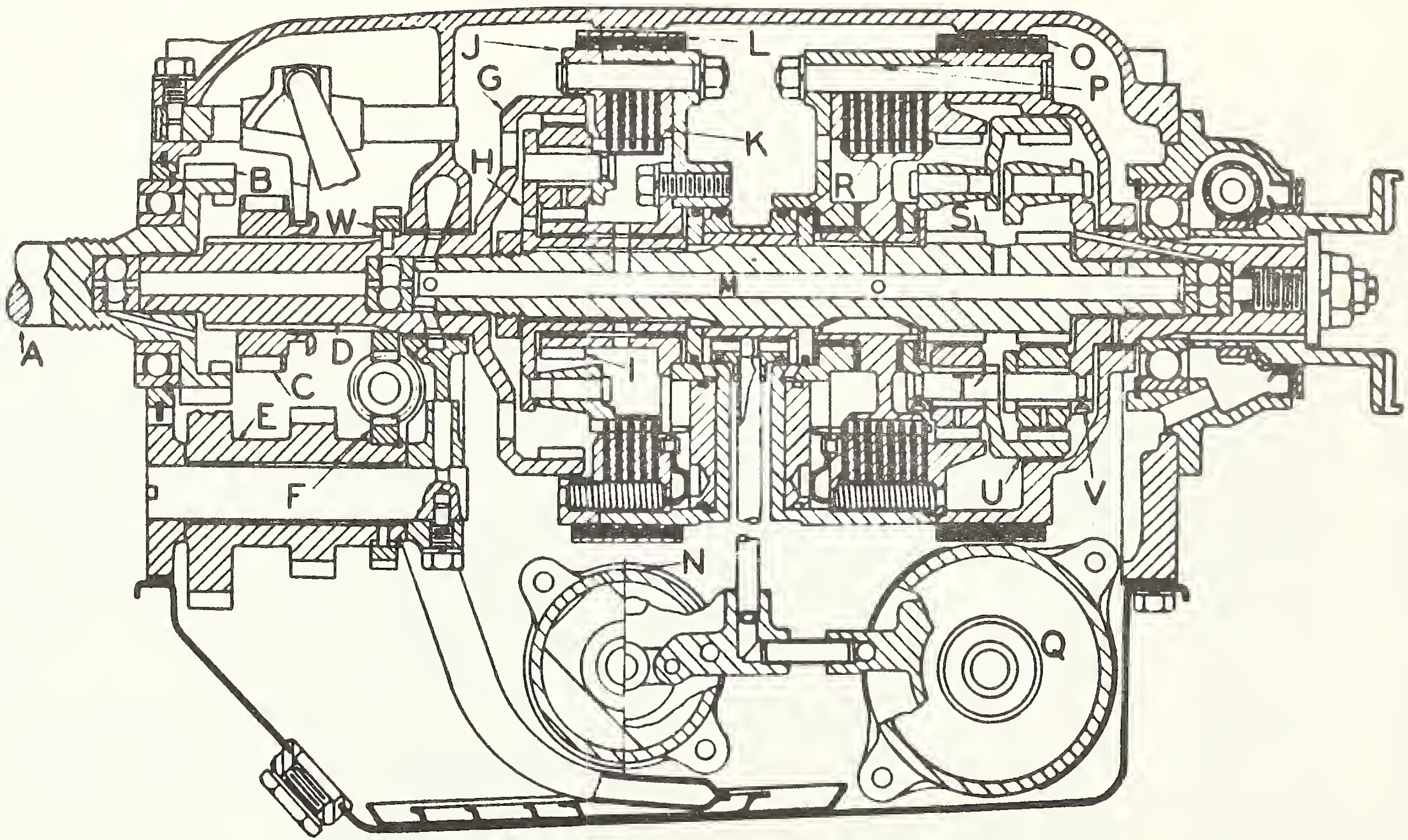
Variable automatic shifts were accomplished by shuttle valves. These valves controlled the flow of hydraulic fluid to pistons which in turn actuated brake bands and clutches within the planetary train. The position of the shuttle valve resulted from a balance between the throttle valve pressure (a function of accelerator position) and an opposing governor pressure (a function of road speed). A double gear pump delivered oil for both the pressure lubrication and for the servo controls.

This transmission was the first automatic unit which responded to driver demands and upshifted or downshifted according to driver demand (throttle position) and road speed.

The Automatic Safety Transmission yielded the following ratios:

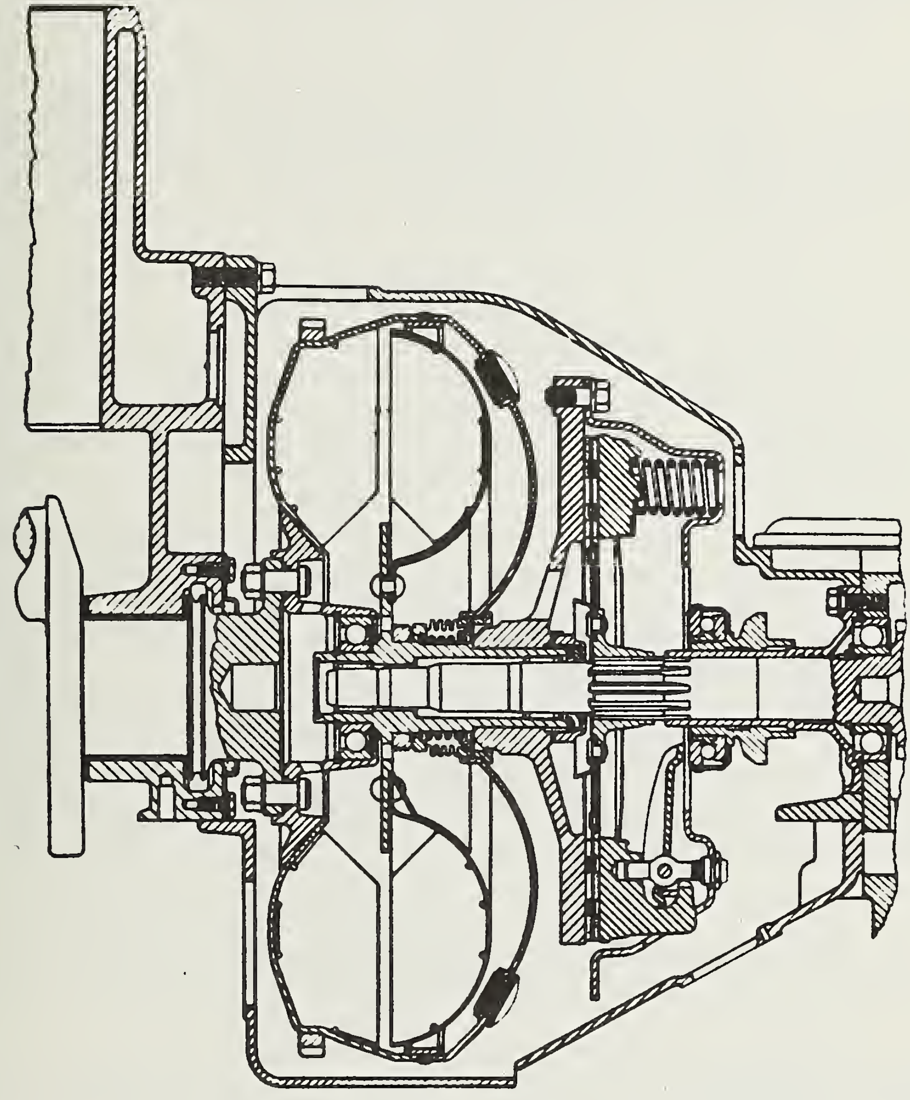
First:	3.16:1
Second:	2.23:1
Third:	1.45:1
Fourth:	1.00:1

In 1938 Chrysler introduced the Fluid Drive, the first time an American vehicle adopted the fluid clutch. It was standard equipment on the Custom Imperial (Fig. 27). The unit consisted of a fluid coupling for starting, a friction clutch for gear range selection, and two-speed sliding gear synchronmesh units in series. A kickdown overdrive in conjunction with the fluid coupling gave an automatic two-speed transmission in high gear.



Source: *Automotive Industries,* May 29, 1937, p. 806.

FIGURE 26 OLDSMOBILE AUTOMATIC "SAFETY" TRANSMISSION



Source: *Automotive Industries,* December 24, 1938, p. 319.

FIGURE 27 SECTIONAL VIEW OF THE "FLUID-DRIVE" ASSEMBLY WITH THE DRIVER AND RUNNER OF CHRYSLER FLUID DRIVE SHOWN AT THE RIGHT

Like the Reo Self-Shifter, automatic shifts were possible between first and second gear and between third and fourth gear. Unlike the Reo, however, driver manipulation of the clutch was not necessary in normal driving. The automatic upshift was driver-controlled and accomplished by vacuum servos connected to the selector forks. Return springs effected the downshift. An ignition interruptor switch was incorporated into the design to eliminate engine torque when down shifting.

This transmission, with minor modifications such as electro-hydraulic controls and, later, a torque converter, became the basis for the Chrysler Prestomatic, DeSoto Tip Toe, Dodge Gyromatic, and the three-speed Plymouth Hydrive. This basic transmission design was used by Chrysler until the development of the Powerflite in 1954.

In 1940 the Hydra-Matic was introduced by Oldsmobile as the first fully automatic transmission to be mass-produced in the U.S. It combined a fluid coupling for starting and stopping with the automatically shifting gearbox of the Safety Transmission. This design enabled Oldsmobile to eliminate the clutch pedal altogether, and any manual shifting other than forward and reverse. To operate the vehicle the driver just put the transmission into "drive" and depressed the accelerator pedal.

The hydraulic control system of 1940 was very similar to the system Olds and Buick employed with the Safety Transmission in 1937. The major difference was that reverse gear on the Safety was effected by conventional sliding gears whereas reverse on the Hydra-Matic was achieved by the addition of a third planetary gear set.

Another problem that the Hydra-Matic solved was that of torque fight between the two friction members involved in a shift, one being released while the other was engaged. The Hydra-Matic solved this by hydraulically applying the direct coupling clutch while relieving the reaction brake.

The gear ratios were slightly altered from the Automatic Safety Transmission. They were:

First:	3.81:1
Second:	2.63:1
Third:	1.45:1
Fourth:	1.00:1

This four-speed hydramatic transmission has evolved into the present three-speed Turbo Hydra-Matic transmission. Although many changes have since taken place, the underlying principle of all GM's Hydra-Matic designs (and all modern three-speed automatics) have been the use of two or more hydraulically controlled planetary gearsets to obtain different ratios.

5.2 POST WORLD WAR II TO 1950

During World War II, when no automobiles were being produced for public consumption, government-sponsored research into the fluid coupling and torque converter for tanks and other military vehicles made great strides toward the maturation of these components. Research-proven systems stimulated the auto manufacturers to get these improvements into automotive production units after the end of the war.

During this period, General Motors with its considerable resources was able to support a two-pronged approach to the development of an improved automatic transmission. Buick and Chevrolet proceded along the lines of a completely hydrodynamic drive. Oldsmobile, Pontiac and Cadillac continued to develop the fully automatically shifting planetary gearbox in the form of the Hydromatic transmission.

The problem during the late '40s and early '50s was to raise the efficiency of the torque converter. Losses from a torque converter are of two kinds: flow and shock. Flow losses result from molecular shearing of the fluid molecules flowing over solid surfaces. Shock losses occur at the "cell" entrances. Flow losses were reduced through the development efforts of hydraulic fluid suppliers. Shock losses were reduced by resorting to two or more stages in runner blading with intermediate stages of reactor blading.

In 1948 Buick introduced the Dynaflow, the first modern American-made transmission to utilize a torque converter. In normal operation all the required torque multiplication was carried out by the mechanical gear-box. The Dynaflow combined a torque converter with a relatively low stalling torque with a planetary emergency low and reverse gear.

The basic design of the initial Dynaflow consisted of a five-element, multi-phase torque converter including a primary impeller, a primary turbine, a free-wheeling pump and two free-wheeling stators. The primary turbine in normal operation was directly connected to the output shaft. Torque multiplication was accomplished by changing the direction of the fluid flow with a series of differently shaped stators.

An impeller and two stators were mounted on one-way clutches between the primary impeller and the turbine. Each of these reaction members had a different blade angle. At low vehicle speeds, the direction of fluid flow from the impeller was such that the one-way clutches of all three reaction members were actuated holding all three members stationary. This produced a certain composite blade shape with certain torque multiplication characteristics.

As the vehicle moved faster, the relative speed of the impeller and turbine altered direction of fluid flow. As the flow direction changed, the fluid would strike the back face of the first reaction member and cause it to revolve on its one-way clutch. The two remaining stationary reaction members resulted in a blade of a different composite shape having different torque multiplication characteristics. As vehicle speed increased further, the other reaction members would progressively begin to rotate until the turbine approached the speed of the impeller, at which time all the reaction members would be free-wheeling with no torque multiplication. At this point, the unit would act as a fluid coupling.

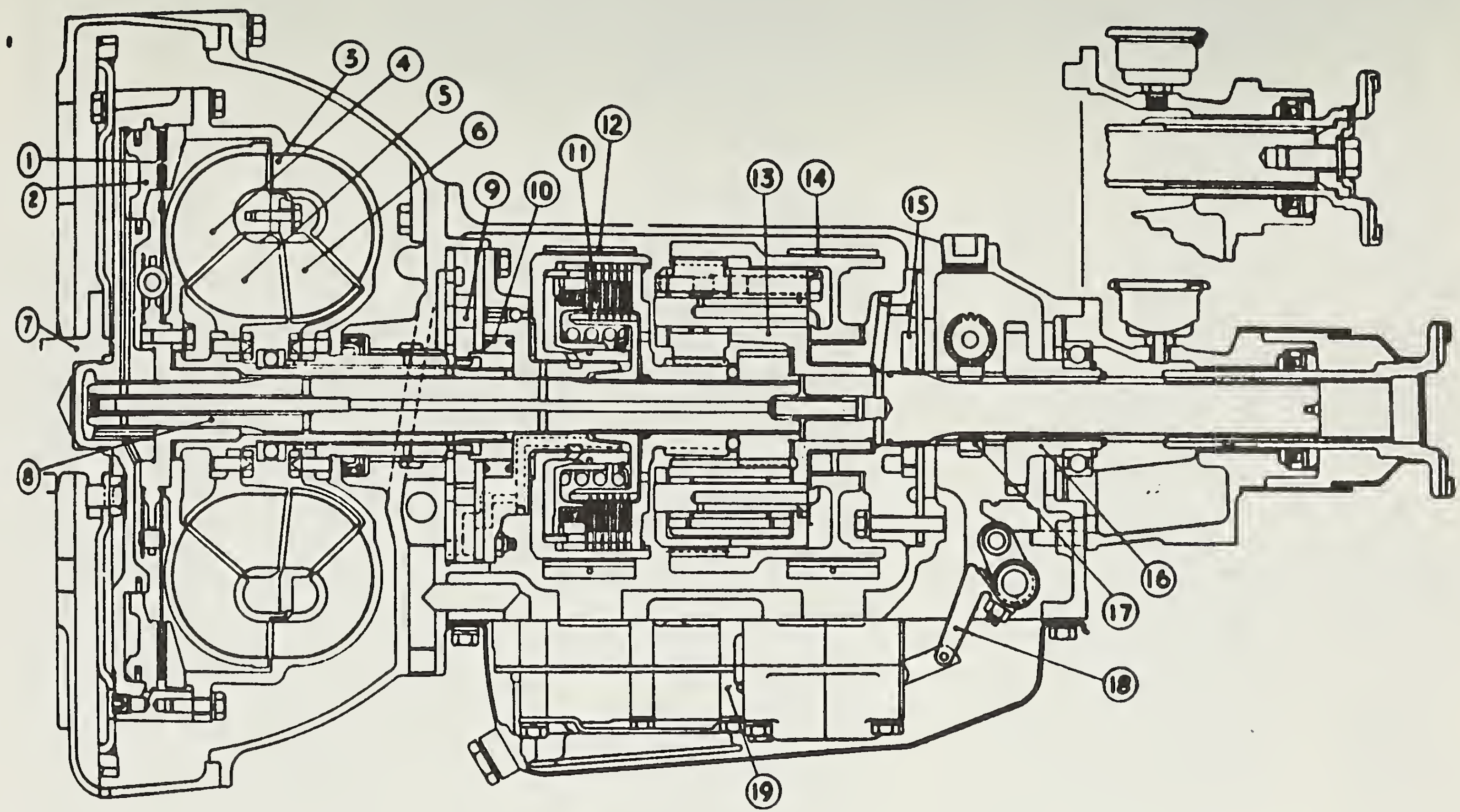
The one major drawback of the Dynaflow was the lack of low-speed torque multiplication. Under normal acceleration the unit would roughtly equal other cars of the day. However, the maximum torque multiplication for the Dynaflow was only 2.25 versus almost 6 to 1 for the Hydramatic. This resulted in a maximum acceleration capability that was far less than other transmissions. The Dynaflow continued to evolve in a number of forms, all of which eliminated the need for automatic shifts in an additional gearbox. But the concept could never simultaneously match the coupling efficiency and torque multiplication of the Hydramatic units.

In 1949 Packard introduced the Ultramatic transmission (Fig. 28) and with it a new type of torque converter, a three-element unit with only one reactor stage. Also new was the provision of a lock-up friction clutch on the torque converter. An auxiliary planetary gearset was provided for emergency low and reverse.

The torque converter was designed to operate only during acceleration, providing a maximum torque multiplication on the order of 2.4:1, and was a good torque multiplier over a wide range of speed ratios with little regard for coupling characteristics. In fact the converter reached the coupling point at a speed ratio of only .82 which resulted in very poor efficiency at high speed. To overcome this, selective action of a special centrifugal governor moved a friction clutch to lock-up the converter to provide a positive direct drive between engine and driving axle at high speeds. The automaic shift from converter to direct operation was achieved by means of a hydraulic control system. This change was determined by a shuttle valve in which governor pressure was opposed to throttle pressure in a manner similar to the Hydramatic.

Under ordinary part throttle conditions the direct drive clutch engaged near 15 mph. This varied with throttle position to provide a range of shift points up to 50 mph with maximum throttle.

The governor, driven from the speedometer gear, consisted of two weight arms. Reportedly, the governor hydraulic pressure curve was in the form of a reasonably straight line in respect to road speed. One weight gave the characteristics of the output pressure curve while the second served as a vent valve when speed was about 15 mph and cut out when speed dropped between 12-13 mph.



Source: *Automotive Industries,* May 1, 1949, p. 29.

FIGURE 28 PACKARD ULTROMATIC TRANSMISSION WITH 3-ELEMENT TORQUE CONVERTER AND LOCK-UP CLUTCH

The Ultramatic had a number of drawbacks including harsh shifting, poor economy in city traffic during non-lock up operation, and acceleration little better than the Dynaflow. Its major advantage was improved fuel economy at highway speeds, due to the lock-up feature.

The Ultramatic was continuously produced until 1955 when it was replaced by the Twin-Ultramatic in the Nash, Packard and Hudson. The Twin-Ultramatic was an attempt to remedy the problems of its predecessor with an additional automatically-controlled planetary gearset. However, the engagement of the lock-up friction clutch could not be made smooth enough with the available technology. This transmission was finally dropped in 1957 when the non lock-up Flight-o-Matic was introduced.

The efficiency of the Packard Ultramatic torque converter, according to an SAE paper in 1949 by Packard's chief engineer, never appears to have surpassed about 85%.

5.3 1950-55

The period of 1950 to 1955 was a period of further development on the torque converter and planetary gearing. The main avenue of attack centered around augmenting the torque converter with an automatically shifting gearbox. The other approach, followed solely by Chevrolet and Buick, centered on having all the torque multiplication performed by the converter without any positive shifting in the gearbox. The latter designs were either dropped completely, as in the case of the Buick Dynaflow, or modified extensively, as in the case of Chevrolet's versions, to conform to the primary design strategy.

In 1950 Chevrolet introduced the Powerglide transmission. This unit was essentially the same as the Buick Dynaflow and suffered from the same lack of acceleration. In 1953, therefore, Chevrolet introduced a new Power glide, a three element torque converter augmented by an automatically shifted planetary gearset. The torque converter had a maximum stall torque of 2.2:1 combined with a two-speed box with ratios of 1.82 and 1.00:1 for low and high, respectively. This transmission was more in line with the general trend of a more efficient low torque multiplication converter augmented with additional gearing. The problem with the transmission was that it had only two speeds which sometimes resulted in poor driveability.

In 1953 Buick announced a modified version of the Dynaflow, the Twin Turbine Dynaflow. The new transmission offered a new four-element torque converter instead of the five-element unit. The four elements were: an impeller, a stator, and two turbine members.

Each of the two turbines were connected by planetary gearing to the output shaft. One turbine was connected to the ring gear, the other to the planet carrier. The sun gear was attached by a one-way clutch to the stator. The two turbines would operate under either of two conditions. Both turbines were always connected together by the gearing such that the second turbine rotated at 5/8ths the speed of the first. At low road speeds the first turbine, which rotated at a speed faster than the second, supplied the torque for initial acceleration.

At higher speeds the second turbine developed a larger torque and would accelerate to exceed the 5/8ths speed ratio, at which time the stator would no longer hold the sun gear stationary against the one-way clutch. The sun gear would then free-wheel and the output torque would be obtained from the second turbine. In actual operation, all the torque conversion at low speed was obtained through the gearset and the first turbine. The torque for this combination gradually diminished while the torque output of the second turbine increased until it carried the full load at high speed.

Although the Twin Turbine Dynaflow offered better initial acceleration performance, it did not appear to be a significant improvement over earlier Dynaflows.

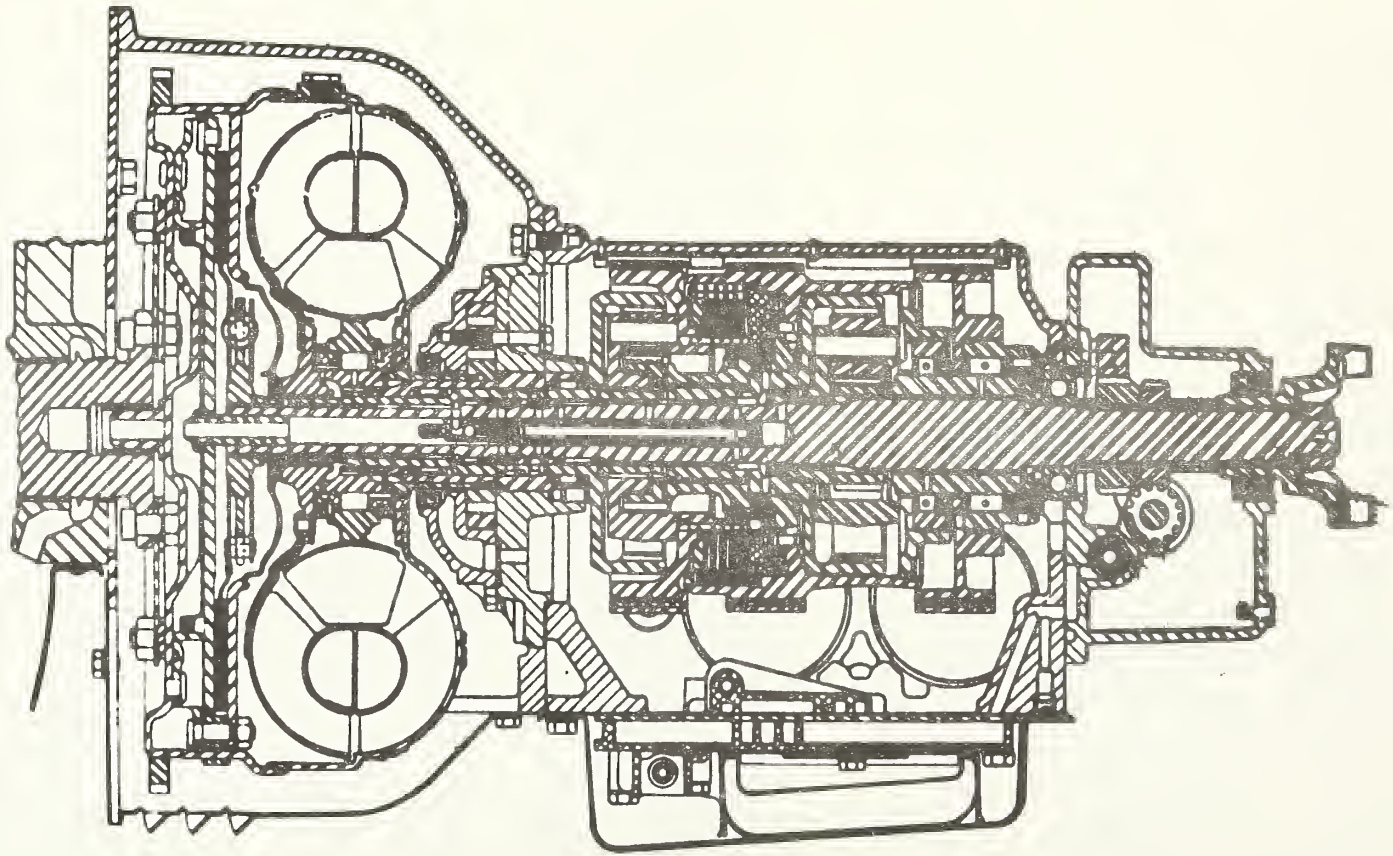
In 1955 Buick introduced the Variable Pitch Dynaflow, basically the Twin Turbine design in which a variable pitch stator was inserted. Twenty separate stator blades were mounted on a central hub such that their angle could be changed by hydraulic pressure as a function of throttle position. The main object of the variable pitch stator was to permit higher fluid flow conditions in the high angle position at low road speeds which would result in improved torque output for high performance or improved acceleration. The low angle position corresponded to normal cruise conditions.

The stall ratio was increased from 2.1:1 in the Twin Turbine design to 3.2:1 for low stator blade angle and from 2.4:1 to 3.5:1 for high stator blade angle.

Although this modification did improve acceleration, it was still unable to provide performance comparable to other designs.

In 1950 Studebaker introduced the Studebaker Automatic Transmission, developed jointly by Studebaker and Borg-Warner (Fig. 29). It consisted essentially of a three-element torque converter augmented by two single-planet epicyclic gearsets with helical gears providing three forward and one reverse speed. The system included two gear-type oil pumps, a multiple disk clutch with four friction plates and a single plate cork-faced disk clutch running in oil for direct drive.

A transmission lock-up device was included as a Hill-Holder to prevent vehicle movement when a stop was made on a grade while in direct drive. It was released by accelerator pressure.



Source: *Automotive Industries,* December 15, 1949, p. 29.

FIGURE 29 LONGITUDINAL SECTION OF STUDEBAKER'S FULLY AUTOMATIC TRANSMISSION SHOWING DETAILS OF THE TORQUE CONVERTER, SINGLE PLATE CLUTCH FOR DIRECT DRIVE, TWO PLANETARY GEAR SETS, ETC.

Maximum torque multiplication at stall was on the order of 2.15:1 at 1500 rpm. The torque converter was among the first to substitute stamping for castings and was air cooled. Total reduction through the two planetary gearsets was 2.3:1. This combined with torque multiplication to produce a maximum ratio of 4.96:1. The rear set had a ratio of 1.44:1. When combined with the torque multiplication of the converter this resulted in a total reduction of 3.08:1. Third gear was direct drive. Coupling occurred at a speed ratio of .87.

In top gear a hydraulically operated single plate friction clutch was used to provide positive lock-up between the engine and drive axle. In this way fuel savings were obtained by eliminating converter losses during cruise operation.

The auxiliary gear box on this transmission is of interest in that the three forward ratios and a reverse ratio were obtained from only two sets of planetary gears. The planetary gearbox incorporated the use of one-way overrunning clutches. This eliminated the multiple bands used to hold particular members stationary at certain times in transmissions like the Hydramatic. The elimination of the simultaneous engagement of more than one hydraulically-controlled clutch, and/or band resulted in very smooth shifts. This was offset, however, by a somewhat rougher-than-normal shift during direct drive lock-up clutch engagements. Because of the overrunning clutches, the hydraulic controls were somewhat simpler than in other designs, but operated on the principle of a balance between speed and load (governor and throttle pressure), similar to the Hydramatic.

The transmission apparently performed satisfactorily, but was later abandoned due to a change in manufacturing policy when Packard merged with Studebaker in 1955.

In 1951, Ford and Borg-Warner introduced the Fordomatic type transmission, known as the Lincoln Turbodrive and Mercomatic. This unit was similar in concept to the Studebaker transmission. The Fordomatic had a three-element torque converter combined with a three-ratio planetary gearbox, but did not employ a lock-up clutch.

Ford's emphasis was on smooth shifting characteristics, and on eliminating the rough ratio characteristics of the direct drive engagement of the Studebaker transmission. This was accomplished by designing a high efficiency torque converter with a coupling speed ratio of .90. This caused the maximum stall torque ratio to be reduced to only 2.00:1. To compensate for this, the gear ratios were increased to provide acceleration performance comparable to that of less efficient, higher ratio torque converters:

First:	2.44:1
Second:	1.49:1
Third:	1.00:1

The car was normally started in the 1.49:1 intermediate range assisted by the torque converter for a 3:1 total starting torque. There was thus only one automatic shift, to high, at speeds from 17 to 63 mph. A unique feature was the provision of intermediate gear braking below 40 mph when the control lever was shifted into low for negotiating grades at medium speeds.

The now universal shifting pattern was also introduced by Ford, by moving "reverse" to the left of "neutral" on the quadrant in order to simplify the process of rocking the car in mud or snow.

The Ford transmission relied on the normal coupling operation of the torque converter for efficiency in top gear cruising. This resulted in smoother top gear engagement at the cost of increased fuel consumption over the locked-up converter.

The Fordomatic was an acceptable automatic transmission. In order to meet market demand, it was joined by the Cruiso-o-Matic. The Fordomatic pioneered the concept of a torque converter/three-speed planetary gearset which relied solely on fluid coupling efficiency of the torque converter for normal cruising operation. This concept continues to dominate present day transmissions, in which there are inherent losses in the torque converter during all driving phases.

In 1951 Chrysler introduced the Fluid Torque transmission which represented Chrysler's first use of the torque converter. The transmission consisted of a four-element torque converter combined with the semi-automatic sliding gear train used by the Fluid Drive unit. The converter contained two stators for torque multiplication. The manually operated friction clutch was retained, behind the torque converter, to allow manual speed range selection.

The Fluid Torque was followed several years later by the Powerflite in 1954. This unit consisted of a four-element torque converter augmented by an automatically shifted planetary geartrain. The torque converter was the two stator type used by the Fluid Torque. This was now combined with an automatically shifted planetary set giving two forward ratios: 1.72:1 and 1.00:1. The gearset was shifted by shift valves that opposed throttle pressure with governor pressure. Since the four-element torque converter was capable of torque multiplication over a wide range, only two forward speeds were necessary. The four-element torque converter suffered from a relative lack of initial acceleration that plagued all multi-reaction member converters.

In 1955 Chrysler introduced the Torqueflite. This brought Chrysler into the mainstream of transmission design philosophy with a low torque multiplication converter accompanied by an automatically shifted planetary gear set. The Torqueflite consisted of a simple three-element torque converter and two compound planetary gearsets offering three forward ratios. The available ratios were:

First:	2.45:1
Second:	1.45:1
Third:	1.00:1
Reverse:	2.20:1

The wider range of gear ratios permitted the use of a lower rear axle ratio for improved engine life and fuel economy. An axle ratio of 3.18:1 was attainable compared to the previous 3.36 rear axle ratio. Overall torque ratio at the rear wheels at a start was 21 to 1 compared to 15 to 1 for the Powerflite. This reportedly provided over 15% more acceleration for 0-15 mph and over 12% more acceleration than the Powerflite from 0 to 30 mph.

Torqueflite also had two multiple disk clutches instead of the one that Powerflite had used, and added an overrunning clutch. It possessed 19 hydraulic valves compared to the 10 previously required to take care of the added functions of the control scheme.

The stall torque ratio for the Torqueflite was improved over the ratio achieved by the Powerflite only after a brief introductory period. The Powerflite had a 2.6 stall torque ratio, while the Torqueflite for the 1956 Imperial was rated at 2.7. This was later decreased to 2.2.

5.4 1955-64

During this period, the many transmission designs being produced evolved to the modern common point -- a three-element low-multiplication torque converter and a three-speed planetary gear set. The manufacturers also standardized on the Simpson gear train as the most desirable gear set.

The Simpson gear train is unique in the automobile industry in that the license to manufacture it has been obtained by all the major manufacturers -- General Motors, Ford and Chrysler. Like Henry Ford, Howard W. Simpson, a consultant for Ford Motor Co., was fascinated by the possibilities of the planetary gear set. Shortly after World War II, Simpson was told he was dying of cancer. He spent the final months of his life studying all of the possible planetary gear arrangements. The optimum design which he settled on used a common sun gear for both the front and rear planetary sets, and identical planet and ring gears. This commonality of parts produced a major savings in the production of the automatic transmission.

Although Simpson's patents were in effect in 1950, they were not incorporated into production of automatic transmissions until 1957, when Chrysler built them into the Torqueflite for the new Imperial. The gearsets worked so well that they were incorporated into all Chrysler transmissions. This adaptation, with some minor changes such as an aluminum case, is the basic design for the present Chrysler Torque Flite A-904 and A-727 automatic transmissions.

Although Ford took out a license to manufacture Simpson gearsets in 1953, the design was not used until 1964 when the present C-series transmissions went into production. In 1958, Ford modified the successful Fordomatic slightly and changed the name to the Cruise-o-Matic. The modification consisted of a one-way clutch which was introduced into the gearing to relieve some of the load on the rear friction band. This change combined with some additional hydraulic modifications resulted in a smoother shifting transmission. The good performance of this transmission was responsible for the delay in adopting the Simpson sets until the introduction of the C-series in 1964.

From 1962-64 Ford also introduced the Fordomatic and Merc-o-Matic two-speed automatics for the smaller Falcon and Mercury Comet. They were similar in design to the larger transmissions, but with one forward ratio eliminated, and were considered adequate for lighter vehicles.

Until 1955, the General Motors Hydramatic was still basically the same design used in 1940. In 1956, General Motors introduced the Stratoflight Hydramatic, using a fluid coupling and four-speed gearbox with no torque converter. The method of control of the planetary set was changed, with one-way clutches replacing the original brake bands.

The new arrangement upgraded shift smoothness and eliminated most of the shift sensations. The mechanism was considerably simplified by the use of the one-way sprag-type clutches to replace the bands and associated servo mechanism for the first two gear sets, reducing the overall number of parts.

The key to the smoothness of the Stratoflight was the introduction of a small coupling, termed a dump and fill type clutch, mounted on a one-way clutch. The coupling filled with fluid whenever the shift signal called for a lock-up of the front unit giving a direct drive. It dumped the fluid when torque multiplication was required. The small coupling replaced the front unit clutch and made it unnecessary to employ a complicated serve mechanism for timing. It also prevented mistimed shifts and eliminated the possibility of fight between communicating members and prevented rapid changes in output torque.

Total reduction in each speed ran as follows:

First:	3.97:1
Second:	2.55:1
Third:	1.55:1
Fourth:	1.00:1

Developments in 1961 brought the Hydramatic more in line with present designs, when GM introduced the Hydramatic three-speed for Oldsmobile and Pontiac. The transmission consisted of a low multiplication three-element torque converter combined with a three forward ratio planetary set. The gear train was a modification of the old Hydramatic with two planetary sets and not a true Simpson set. The following ratios were incorporated:

First:	2.97:1
Second:	1.56:1
Third:	1.00:1

In 1964, General Motors introduced the Turbo Hydramatic, which used a new three-element torque converter combined with a Simpson gearset. Except for minor modifications the 1964 design is the same as the present day Turbo Hydramatic.

While work continued on the Hydramatic, Chevrolet was still pursuing the alternate program of Powerglide development. Not satisfied with the performance of their two-speed Powerglide, Chevrolet in 1957 introduced the Turboglide, similar in concept to the Twin Turbine Dynaflow. It had a five-element converter consisting of an impeller, a stator, and three turbine members. It contained two sets of planetary gearing arranged so that the output of the three turbines could be connected in series. The two planetary sets of the Twin Turbine Dynaflow, and the later Turboglide, did not require the sequenced gear shift capability of other transmissions.

In the Turboglide design, the first turbine was connected by a one-way clutch to the output shaft by the rear planetary set. The second turbine was connected through a one-way clutch and the front planetary set to the output shaft. The third turbine drove directly on the output shaft by a cone clutch. The second turbine rotated at 5/8ths the speed of the first, the third turbine rotated at 3/8ths the speed of the first.

The transmission operated in a mode similar to the Twin Turbine Dynaflow. As the vehicle accelerated, the first turbine was transmitting most of the power to the driving wheels, the second meanwhile provided proportionally less torque while the third turbine contributed very little to starting torque. As engine and vehicle speed increased to the desired level, the first turbine was cut out by a one-way clutch and essentially free-wheeled. In a similar manner, the second turbine would reach its limit and free-wheel leaving the third turbine to transmit all the power to the drive axle.

The Turboglide was very smooth in its operation but again did not allow adequate initial acceleration. General Motors' final attempt to produce a non-shifting transmission came in 1958 when Buick introduced the Flight Pitch Dynaflow. The Flight Pitch was the same design as the Turboglide except that it incorporated the variable pitch stator from the Variable Pitch Dynaflow. Even with this modification, the Flight Pitch did not have performance comparable to other automatic transmission designs.

A third generation Powerglide was therefore introduced by Chevrolet in 1958 with an aluminum case version produced in 1959. The Powerglide had a three-element torque converter combined with a two-speed planetary set. The torque converter had a maximum 2.10:1 torque multiplication at stall, and ratios were 1.82 and 1.00:1.

This transmission was considered acceptable for Chevrolet's lighter cars and continued in production until 1972, when the GM product line was standardized on Turbo Hydramatics. The Flight Pitch Dynaflow was discontinued in 1961 in favor of the new Hydramatic. However, for its smaller cars, Buick produced the 300 Super Turbine transmission from 1962-68. This transmission attempted to obtain three-speed performance out of a two-speed gearset by using the Buick variable pitch stator. The transmission had two forward ratios, 1.76:1 and 1.00:1. The variable pitch stator was intended to cover the larger ratio step when the two-speed was used in larger vehicles. The transmission was eventually replaced by the Turbo Hydramatic 350.

Two other automatics were produced by Borg-Warner during this period the Studebaker Flightomatic and the American Motors Flash-o-Matic. The Flightomatic was a conventional three-element converter combined with a three-speed box. It was produced initially in 1957 and continued in production until the demise of the manufacturer.

The Flash-o-Matic went into production in 1961 and used a three-element torque converter with a maximum stall ratio of 2.12. The Flash-o-Matic did not utilize a Simpson set, but had a Revenaux planetary gearset with the following ratios:

First:	2.40:1
Second:	1.47:1
Third:	1.00:1

This transmission was discontinued by AMC in 1972 when they adopted Chrysler's Torqueflite. It continues to be built overseas by Borg-Warner as the Model 35 and 65.

5.5 1965-1976

The years 1945-64 represented a period in which tremendous time and effort was expended on the perfection of the automatic transmission. The period from 1965 to the present has been a period of relative stability in which a standardized transmission concept has been continuously produced. Ford in its C-series, Chrysler with the Torqueflite, and General Motors with the Turbo Hydramatic series, have continually produced transmissions with the same design concepts developed prior to 1964. Minor modifications, such as the use of electrically-operated hydraulic switches, one-way clutches, new friction materials, and oil seals as well as weight reducing designs.

Gear ratios and torque ratios of torque converters have changed to a small degree in response to changing vehicle requirements, but the basic principle of operation remains the same. The major changes that are affecting the automobile industry have not yet resulted in a new transmission design, though new concepts are waiting on the workbench at all the major manufacturers for durability testing and tooling capitalization.

Although a variety of different models of automatic transmissions are produced by U.S. car manufacturers, the underlying design principles are identical for all current models. The specific transmissions vary in configuration, gear ratio, hydraulic controls, and capacity, depending on manufacturer and application.

However, all present U.S.-made automatics are based on a three-element torque converter having a relatively low torque multiplication, combined with a three-ratio automatically operated planetary gear train.

6. ELEMENTS OF THE MODERN AUTOMATIC TRANSMISSION

The following components and operating modes of all current production automatic transmissions are based upon the following three major components which have been developed during a trial and error evolution period of some fifteen years:

1. Torque Converter
2. Planetary Gear Train
3. Hydraulic Controls

Each of the major components is discussed below.

6.1 TORQUE CONVERTER

The present torque converter is a three-element unit consisting of an impeller, turbine and stator. The stator is mounted on a one-way clutch and serves to redirect the fluid flow from the driven turbine to the driving impeller for torque multiplication. When torque multiplication is not needed, the stator free-wheels with the turbine and the torque converter acts as a fluid coupling. (For further discussion refer to Volume II, Section 3.2.) At vehicle standstill, the torque converter allows the engine to continue to run, and provides starting capability.

6.2 GEAR TRAIN

The modern day torque converter alone has insufficient torque multiplication characteristics to accelerate a vehicle with the performance requirements of today's automobiles. The torque converter must be augmented by an automatically operated gearbox in order to yield acceptable vehicle performance through torque multiplication. The almost universally utilized type of planetary gear train used in the automatic gearboxes of U.S. automobiles is commonly called the Simpson Gear Train, after the inventor, Howard W. Simpson.

The Simpson gear train consists of two identical planetary gearsets configured around a common sun gear. This set has the production advantage of a commonality of parts and the planetary set offers the advantage of having all the gears constantly in mesh. The gears in the planetary gear train are arranged so that power can be applied to one of two members of the set and taken off only one member of the set by a series of hydraulically-controlled wet multiple disk clutches and externally contracting brake bands.

6.3 HYDRAULIC CONTROLS

The hydraulic controls effect the automatic operation of the gearbox insuring the smooth execution of shifts at the appropriate conditions of vehicle speed and load. The greatest variety in automatic transmission design occurs in the hydraulic control system; however, all the arrangements of hydraulic controls have these common features:

1. Oil pump
2. Shift valves
3. Governor
4. Throttle valve

An oil pump driven by the engine supplies the necessary hydraulic pressure and fluid circulation through a series of valves and connecting plumbing to actuate servos which engage or disengage bands, apply or relieve pressure to activate clutches, and, of course, to supply lubrication.

The specific routing of hydraulic fluid to effect a shift and select the proper gear ratio is controlled by a shift valve. The shuttle or spool-type valve alters the direction of fluid flow according to its position. The position of the shift valve and the timing of its movement are functions of vehicle speed, load and driver demands.

The governor senses driveshaft and hence vehicle speed, while the throttle valve determines engine load and/or driver demands by sensing either manifold vacuum or throttle position. These generate opposing hydraulic pressure signals across the shift valve(s). The shift valve(s) move until their position against a spring load creates a balance between the two signals. Hence, engine load and vehicle speed are used to determine the position of the shift valve and hence the appropriate gear ratio.

7. FAMILY TREE OF THE MODERN AUTOMATIC TRANSMISSION

Figure 30 depicts the family tree of the contemporary automatic transmission.

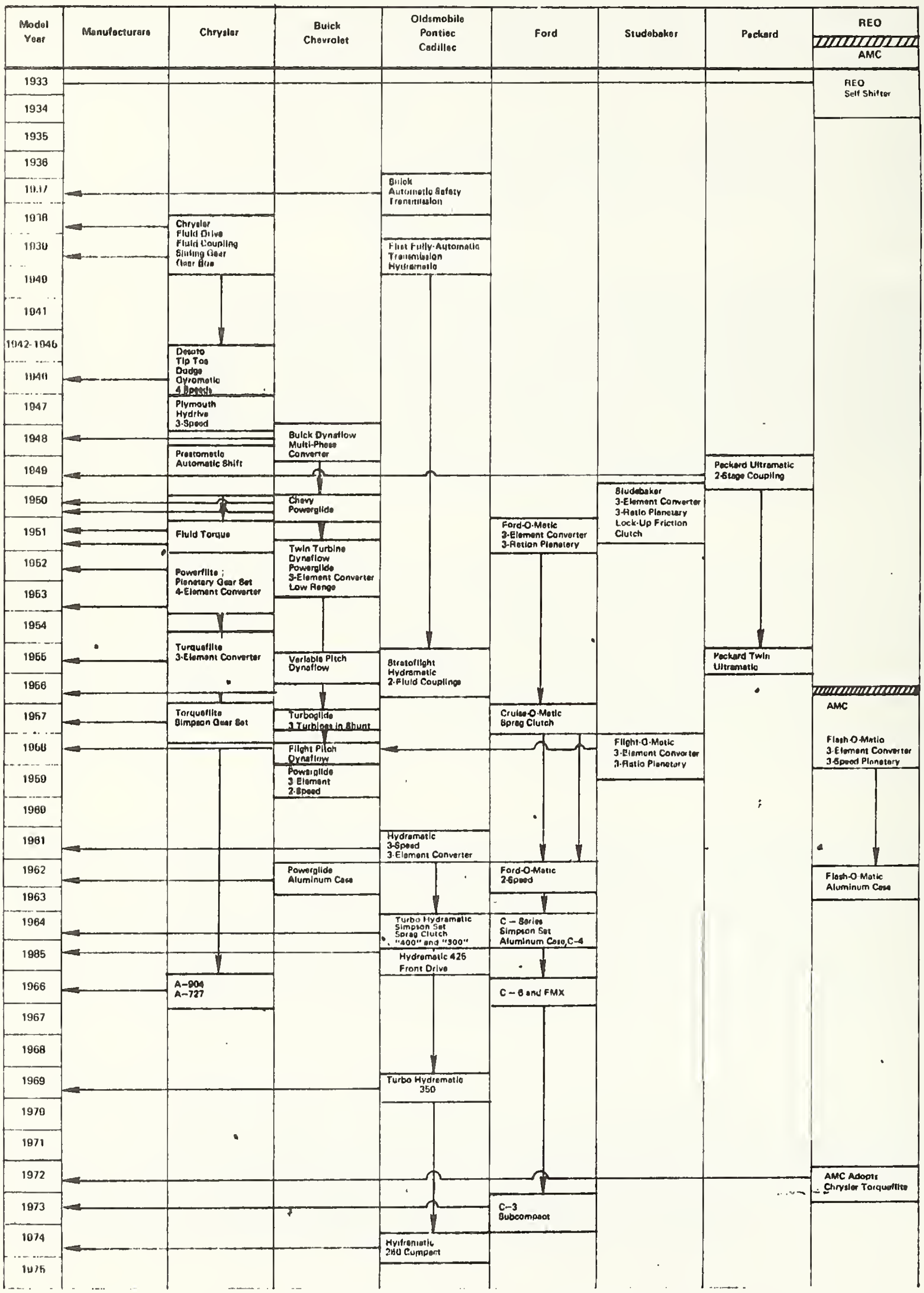


FIGURE 30. FAMILY TREE OF THE MODERN AUTOMATIC TRANSMISSION

8. DESCRIPTION OF AMERICAN AUTOMATIC TRANSMISSION DRIVE TRAINS 1970-75

The information in these tables details automatic transmissions as a function of engine displacement. The power teams are arranged by manufacturer and model year. The engine information includes cubic inch displacement, number of barrels in the carburetor, compression ratio and rated horsepower and torque. By selecting an engine of specific displacement and rated horsepower one can determine the possible transmissions and torque converters that are matched with that engine.

Information on the transmission includes the trade name and all possible gear ratios. The information on torque converters includes the nominal diameter and maximum ratio at stall.

The selection of an engine, a transmission and a matching torque converter represents a power team. Once the power team has been selected the weight range of vehicles and range of rear axle ratio available to this power team are shown. The availability of these power teams in subcompact, compact, intermediate and full size cars is shown at the bottom of the table.

The source of this information is the Motor Vehicle Manufacturers Association Passenger Car specification forms. The power team table does not show all possible combinations since analysis indicates an unworkable number of possible engines due in part to the phasing out of old engine lines and the development of new ones. Data has therefore been reduced to the apparent dominant engine configurations of a given manufacturer, retaining enough engines to demonstrate the range of variability within a manufacturers' range.

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: American Motors **Year:** 1970

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	199 L-6	232 L-6	232 L-6	304 V-8	360 V-8	360 V-8	390 V-8	390 V-8			
Carburetor	1	1	2	2	2	4	4	4			
Compression Ratio	8.5:1	8.5:1	8.5:1	9.0:1	9.0:1	10.0:1	10.0:1	10.0:1			
BHP	128	145	155	210	245	290	325	340			
RPM	4400	4300	4400	4400	4400	4800	5000	5100			
Torque	182	215	222	305	365	395	420	420			
RPM	1600	1600	1600	2800	2400	3200	3200	3600			
Transmission Trade Name	Shift Command	Shift Command	Shift Command	Shift Command	Shift Command	Shift Command	Shift Command	Shift Command			
Rear Axle Ratios	3.08:1 3.31:1	2.37:1 2.73:1 3.08:1 3.31:1 3.15:1	3.08:1 3.31:1 3.15:1	2.87:1 3.15:1	2.87:1 3.15:1	2.87:1 3.15:1 3.54:1	2.87:1 3.15:1 3.54:1	3.54:1 3.15:1			
Gear Ratio: 1	2.39:1	2.39:1	2.39:1	2.39:1	2.40:1	2.40:1	2.40:1	2.40:1			
Gear Ratio: 2	1.45:1	1.45:1	1.45:1	1.45:1	1.47:1	1.47:1	1.47:1	1.47:1			
Gear Ratio: 3	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1			
Gear Ratio: Reverse	2.09:1	2.09:1	2.09:1	2.09:1	2.00:1	2.00:1	2.00:1	2.00:1			
Weight Range of Vehicles	2761-3032	2765-3666	2769-3770	3008-4118	3192-4178	3214-4185	3117-4241	3117-4241			
Torque Converter Max. Ratio at Stall	2.00:1	2.00:1	2.00:1	2.00:1	2.18:1	2.18:1	2.18:1	2.18:1			
Converter Diameter	11"	11"	11"	11"	12"	12"	12"	12"			

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: American Motors

Year: 1971

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	232 L-6	258 L-6	304 V-8	360 V-8	360 V-8	401 V-8					
Carburetor	1	1	2	2	4	4					
Compression Ratio	8.0:1	8.0:1	8.4:1	8.5:1	8.5:1	9.5:1					
BHP	135	150	210	245	285	330					
RPM	4000	3800	4400	4400	4800	5000					
Torque	210	240	300	365	330	430					
RPM	1600	1800	2600	2600	5000	3400					
Transmission Trade Name	Shift Command 3-S	3-S	2-S	3-S	3-S	3-S					
Rear Axle Ratios	2.37:1 2.73:1 3.08:1 3.31:1 3.15:1 3.54:1	2.73:1 3.08:1 3.31:1 3.15:1 3.54:1	2.87:1 3.15:1	2.87:1 3.15:1	3.15:1 3.54:1 2.87:1	2.87:1 3.15:1 3.54:1					
Gear Ratio: 1	2.39:1	2.39:1	2.39:1	2.40:1	2.40:1	2.40:1					
Gear Ratio: 2	1.45:1	1.45:1	1.45:1	1.47:1	1.47:1	1.47:1					
Gear Ratio: 3	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1					
Gear Ratio: Reverse	2.09:1	2.09:1	2.09:1	2.00:1	2.00:1	2.00:1					
Weight Range of Vehicles	2581-3706	2680-3806	2818-4116	3105-4176	3130-4184	3250-4274					
Torque Converter Max. Ratio at Stall	2.00:1	2.00:1	2.00:1	2.00:1	2.00:1	2.00:1					
Converter Diameter	11"	11"	11"	12"	12"	12"					

Vehicle Class

Subcompact ⩽ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ⩾ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: American Motors Year: 1972

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	232 L-6	258 L-6	304 V-8	360 V-8	360 V-8	360 V-8	401 V-8				
Carburetor	1	1	2	2	4	4	4				
Compression Ratio	8.0:1	8.0:1	8.4:1	8.5:1	8.5:1	8.5:1	8.5:1				
BHP	100	110	150	175	195	220	255				
RPM	3600	3500	4200	4000	4400	4400	4600				
Torque	185	195	245	285	295	315	345				
RPM	1800	2000	2500	2400	2900	3100	3300				
Transmission Trade Name	Torque Command 3-S	3-S	3-S	3-S	3-S	3-S	3-S				
Rear Axle Ratios	2.37:1 2.73:1 3.08:1 3.31:1 3.15:1 3.54:1	2.73:1 3.08:1 3.31:1 3.15:1 3.54:1	2.87:1 3.15:1	2.87:1 3.15:1	2.87:1 3.15:1	2.87:1 3.15:1	3.15:1 2.87:1 3.54:1				
Gear Ratio: 1	2.45:1	2.45:1	2.45:1	2.45:1	2.45:1	2.45:1	2.45:1				
Gear Ratio: 2	1.45:1	1.45:1	1.45:1	1.45:1	1.45:1	1.45:1	1.45:1				
Gear Ratio: 3	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1				
Gear Ratio: Reverse	2.20:1	2.20:1	2.20:1	2.20:1	2.20:1	2.20:1	2.20:1				
Weight Range of Vehicles	2606-3499	2597-3769	2826-4129	2947-4165	3214-4175	3214-4175	3266-4230				
Torque Converter Max. Ratio at Stall	2.10:1	2.10:1	2.00:1	2.00:1	2.00:1	2.00:1	2.00:1				
Converter Diameter	10.75"	10.75"	10.75"	11.75"	11.75"	11.75"	11.75"				

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: American Motors **Year:** 1973

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	232 L-6	258 L-6	304 V-8	360 V-8	360 V-8	360 V-8	401 V-8				
Carburetor	1	1	2	2	4	4	4				
Compression Ratio	8.0:1	8.0:1	8.4:1	8.5:1	8.5:1	8.5:1	8.5:1				
BHP	100	110	150	175	220	195	255				
RPM	3600	3500	4200	4000	4400	4400	4600				
Torque	185	195	245	285	315	295	345				
RPM	1800	2000	2500	2400	3100	2900	3300				
Transmission Trade Name	Torque Command 3-S	3-S	3-S	3-S	3-S	3-S	3-S				
Rear Axle Ratios	2.73:1 3.08:1 3.31:1 3.15:1 3.54:1	2.73:1 3.08:1 3.31:1 3.15:1 3.54:1	2.87:1 3.15:1 3.54:1	2.87:1 3.15:1 3.54:1	2.87:1 3.15:1 3.54:1	3.15:1 3.54:1	2.87:1 3.15:1 3.54:1				
Gear Ratio: 1	2.54:1	2.54:1	2.54:1	2.54:1	2.54:1	2.54:1	2.54:1				
Gear Ratio: 2	1.45:1	1.45:1	1.45:1	1.45:1	1.45:1	1.45:1	1.45:1				
Gear Ratio: 3	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1				
Gear Ratio: Reverse	2.20:1	2.20:1	2.20:1	2.20:1	2.20:1	2.20:1	2.20:1				
Weight Range of Vehicles	2734-3600	2745-3881	2959-4308	3109-4357	3119-4367	3119-4367	3358-4443				
Torque Converter Max. Ratio at Stall	2.10:1	2.10:1	2.00:1	2.00:1	2.00:1	2.00:1	2.00:1				
Converter Diameter	10.75"	10.75"	10.75"	11.75"	11.75"	11.75"	11.75"				

Vehicle Class

Subcompact ⩽ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ⩾ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: American Motors **Year:** 1974

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	232 L-6	258 L-6	304 V-8	360 V-8	360 V-8	401 V-8					
Carburetor	1	1	2	2	4	4					
Compression Ratio	8.0:1	8.0:1	8.4:1	8.25:1	8.25:1	8.25:1					
BHP	100	110	150	175	220	235					
RPM	3600	3500	4200	4000	4400	4600					
Torque	185	195	245	285	315	335					
RPM	1800	2000	2500	2400	3100	3200					
Transmission Trade Name	Torque Command 3-S	3-S	3-S	3-S	3-S	3-S					
Rear Axle Ratios	2.73:1 3.08:1 3.15:1 3.54:1	2.73:1 3.08:1 3.15:1 3.54:1	3.15:1 3.54:1	3.15:1 3.54:1	3.15:1 3.54:1	3.15:1 3.54:1					
Gear Ratio: 1	2.45:1	2.45:1	2.45:1	2.45:1	2.45:1	2.45:1					
Gear Ratio: 2	1.45:1	1.45:1	1.45:1	1.45:1	1.45:1	1.45:1					
Gear Ratio: 3	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1					
Gear Ratio: Reverse	2.20:1	2.20:1	2.20:1	2.20:1	2.20:1	2.20:1					
Weight Range of Vehicles	2761-3768	2772-4051	2980-4439	3153-4448	3249-4458	3398-4513					
Torque Converter Max. Ratio at Stall	2.10:1	2.10:1	2.00:1	2.00:1	2.00:1	2.00:1					
Converter Diameter	10.75"	10.75"	10.75"	11.75"	11.75"	11.75"					

Vehicle Class

Subcompact ≤ 100" ____________

Compact 101 – 111" ____________

Intermediate 112 – 119" ____________

Full Size ≥ 120" ____________

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: American Motors **Year:** 1975

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	232 L-6	258 L-6	304 V-8	360 V-8	360 V-8	401 V-8					
Carburetor	1	1	2	2	4	4					
Compression Ratio	8.0:1	8.0:1	8.4:1	8.25:1	8.25:1	8.25:1					
BHP	90	95	120	140	180	N.A.					
RPM	3050	3050	3200	3300	3600	N.A.					
Torque	163	179	220	251	280	N.A.					
RPM	2200	2100	2000	1600	2800	N.A.					
Transmission Trade Name	Torque Command 3-S	Torque Command 3-S	Torque Command 3-S	Torque Command 3-S	Torque Command 3-S	Torque Command 3-S					
Rear Axle Ratios	2.73:1 3.08:1	2.73:1 3.08:1 3.15:1 3.54:1	2.87:1 3.15:1 3.54:1	3.15:1 3.54:1	3.15:1 3.54:1	3.15					
Gear Ratio: 1	2.45:1	2.45:1	2.45:1	2.45:1	2.45:1	2.45:1					
Gear Ratio: 2	1.45:1	1.45:1	1.45:1	1.45:1	1.45:1	1.45:1					
Gear Ratio: 3	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1	1.00:1					
Gear Ratio: Reverse	2.20:1	2.20:1	2.20:1	2.20:1	2.20:1	2.20:1					
Weight Range of Vehicles	2778-3315	2789-4128	3036-4287	3854-4326	3912-4386	3948-4416					
Torque Converter Max. Ratio at Stall	2.10:1	2.10:1	2.00:1	2.00:1	2.00:1	2.00:1					
Converter Diameter	10.75"	10.75"	10.75"	11.75"	11.75"	11.75"					

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Chrysler Corp. **Year:** 1970

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	198 L-6	225 L-6	318 V-8	340 V-8	383 V-8	383 V-8	383 V-8	426 V-8	440 V-8	440 V-8	440 V-8
Carburetor	1V	1V	2V	4V	2V	4V	4V	2-4V	4V	4V	3-2V
Compression Ratio	8.4	8.4	8.8	10.5	8.7	9.5	9.5	10.2	9.7	9.7	10.5
BHP	125	145	230	275	290	330	335	425	350	375	390
RPM	4400	4000	4400	5000	4400	5000	5200	5000	4400	4600	4700
Torque	180	215	320	340	390	425	425	490	480	480	490
RPM	2000	2400	2000	3200	2800	3200	3400	4000	2800	3200	3200
Transmission Trade Name	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite
Rear Axle Ratios	2.76 3.23 3.55	2.76 2.93 2.94 3.23 3.55	2.71 2.76 3.23 3.55 3.91 2.94	3.23 3.55 3.91	2.76 3.23 3.55 3.91	2.76 3.23 3.55 3.91	3.23 3.55 3.91	3.23 3.55 4.10	2.76 3.23	3.23 3.55 4.10	2.76 3.23 3.55 4.10
Gear Ratio: 1	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45
Gear Ratio: 2	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20
Weight Range of Vehicles	2930-3215	2930-4485	3015-4650	3275-3485	3305-4940	3305-4940	3320-4110	3610-3980	3015-4940	3680-4940	3015-4315
Torque Converter Max. Ratio at Stall	2.10	2.10	2.10	2.10	2.00	2.10	2.10	2.10	2.00	2.00	2.00
Converter Diameter	10.75"	10.75"	10.75"	10.75"	11.75"	10.75"	10.75"	10.75"	11.75"	11.75"	11.75"

Vehicle Class

Subcompact ⩽ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ⩾ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Chrysler Corp. **Year:** 1971

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	198 L-6	225 L-6	318 V-8	340 V-8	360 V-8	383 V-8	383 V-8	426 V-8	440 V-8	440 V-8	440 V-8
Carburetor	1V	1V	2V	4V	2V	2V	4V	2-4V	4V	4V	3-2V
Compression Ratio	8.4	8.4	8.6	10.3	8.7	8.5	8.5	10.2	8.5	9.5	10.3
BHP	125	145	230	275	255	275	300	425	335	370	385
RPM	4400	4000	4400	5000	4000	4400	4800	5000	4400	4600	4700
Torque	180	215	320	340	360	375	410	490	460	480	490
RPM	2000	2400	2000	3200	2400	2800	3400	4000	3200	3200	3200
Transmission Trade Name	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite
Rear Axle Ratios	2.76 3.23 3.55	2.76 2.94 3.21 3.23 3.55	2.71 3.91 2.76 2.94 3.21 3.23 3.55	3.23 3.55 3.91	2.76 3.23	2.45 2.76 2.94 3.23	2.76 3.23 3.55 3.91	3.23 3.55 4.10	2.76 3.23	2.76 3.23 3.55 4.10	3.23 3.55 4.10
Gear Ratio: 1	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45
Gear Ratio: 2	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20
Weight Range of Vehicles	2910-3175	2910-4085	3010-4695	3200	3825-4695	3825-4695	3390-4695	3680-3760	3825-4695	4075-4665	3680-3760
Torque Converter Max. Ratio at Stall	2.16	2.16	2.16	2.16	2.16	2.02	2.16	2.16	2.02	2.02	2.02
Converter Diameter	10.75"	10.75"	10.75"	10.75"	10.75"	11.75"	10.75"	10.75"	11.75"	11.75"	11.75"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Chrysler Corp. **Year:** 1972

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	198 V6	225 V6	318 V8	340 V8	360 V8	400 V8	400 V8	440 V8	440 V8	440 V8	440 V8
Carburetor	1V	1V	2V	4V	2V	2V	4V	4V	4V	4V	4V
Compression Ratio	8.4	8.4	8.6	8.5	8.8	8.2	8.2	8.2	8.2	8.2	10.3
BHP	100 | 94	110 | 97	150	240	175	190 | 181	255 | 246 | 265	225 | 216	230 | 221	280 | 271 | 290	330
RPM	4400	4000	4000	4800	4000	4400	4800	4400	4400	4800	4800
Torque	160 | 158	185 | 180	260	290	285	310 | 305	340 | 335 | 345	345 | 340	355 | 350	375 | 370 | 380	410
RPM	2400	2000	1600	3600	2400	2400	3200	3200	2800	3200	3600
Transmission Trade Name	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite
Rear Axle Ratios	2.76 3.23 3.55	2.76 3.23 3.55	2.71 2.76 3.21 3.23 3.55	2.76 3.23 3.55	2.71 2.76 3.23	2.76 3.23	2.76 3.23 3.55	2.76 3.23 | 2.76 3.23 3.55	2.76 3.23	3.23 3.55	3.23 3.55
Gear Ratio: 1	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45
Gear Ratio: 2	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20
Weight Range of Vehicles	2845–3140	2845–3550	2950–4670	3195–3880	3880–4670	3400–4990	3400–4105	4195–4970	3885–4670	Road Runner 3680	Road Runner 3680
Torque Converter Max. Ratio at Stall	2.16	2.16	2.16	2.16	2.16	2.02	2.16	2.02	2.02	2.02	2.02
Converter Diameter	10.75"	10.75"	10.75"	10.75"	10.75"	11.75"	10.75"	11.75"	11.75"	11.75"	11.75"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Chrysler Corp. Year: 1973

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	198 L-6	225 L-6	225 L-6	318 V-8	318 V-8	340 V-8	360 V-8	360 V-8	400 V-8	400 V-8	400 V-8
Carburetor	1V	1V	1V	2V	2V	4V	2V	2V	2V	2V	4V
Compression Ratio	8.4	8.4	8.4	8.6	8.6	8.5	8.4	8.4	8.2	8.2	8.2
BHP	95	98	105	150	170	240	163	170	175	185	220
RPM	4000	4000	4000	3600	4000	4800	4000	4000	3600	3600	4000
Torque	150	178	185	265	270	295	280	285	305	310	310
RPM	1600	1600	1600	2000	2000	3600	2400	2400	2400	2400	3200
Transmission Trade Name	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite
Rear Axle Ratios	2.76 3.21 3.23 3.55	2.76 2.94 3.21 3.23 3.55	2.76 2.94 3.21 3.23 3.55	2.71 2.76 2.94 3.21 3.23 3.55	2.71 3.21 3.23 3.55	2.76 3.21 3.23 3.55	2.71 2.76 3.23	2.71 2.76 3.23	2.71 2.76 3.23	2.71 2.76 3.23	2.76 3.23
Gear Ratio: 1	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45
Gear Ratio: 2	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20
Weight Range of Vehicles	2830-3110	2830-3650	2830-3650	2980-4695	3525-3725	3275-3900	3900-4815	3900-4815	3600-4400	3965-4875	3965-4875
Torque Converter Max. Ratio at Stall	2.16	2.16	2.16	2.16	2.16	2.16	2.16	2.16	2.02	2.02	2.02
Converter Diameter	10.75"	10.75"	10.75"	10.75"	10.75"	10.75"	10.75"	10.75"	11.75"	11.75"	11.75"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Chrysler Corp. Year: 1973 Cont'd.

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	400 V8	400 V8	440 V8	440 V8	440 V8	440 V8	440 V8				
Carburetor	4V	4V	4V	4V	4V	4V	4V				
Compression Ratio	8.2	8.2	8.2	8.2	8.2	8.2	8.2				
BHP	245	260	208	213	215	220	280				
RPM	4800	4800	3600	3600	3600	3600	4800				
Torque	325	335	340	345	345	350	380				
RPM	3200	3600	2000	2000	2000	2400	3200				
Transmission Trade Name	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite				
Rear Axle Ratios	2.76 3.23	2.76 3.23 3.55	2.76 3.23	2.76 3.23	2.76 3.23	2.76 3.23	2.76 3.23 3.55				
Gear Ratio: 1	2.45	2.45	2.45	2.45	2.45	2.45	2.45				
Gear Ratio: 2	1.45	1.45	1.45	1.45	1.45	1.45	1.45				
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00				
Gear Ratio: Reverse	2.20	2.20	2.20	2.20	2.20	2.20	2.20				
Weight Range of Vehicles	3965-4875	3600-4400	4260-5135	4035-4865	4260-5135	4035-4865	3725-4865				
Torque Converter Max. Ratio at Stall	2.02	2.16	2.02	2.02	2.02	2.02	2.02 | 2.16				
Converter Diameter	11.75"	10.75"	11.75"	11.75"	11.75"	11.75"	11.75 | 10.75 "				

Vehicle Class

Subcompact ⩽ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ⩾ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Chrysler Corp. **Year:** 1974

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	198 L-6	225 L-6	318 V-8	318 V-8	360 V-8	360 V-8	360 V-8	400 V-8	400 V-8	400 V-8	400 V-8
Carburetor	1V	1V	2V	2V	2V	4V	4V	2V	4V	4V	4V
Compression Ratio	8.4	8.4	8.6	8.6	8.4	8.4	8.4	8.4	8.2	8.2	8.2
BHP	95	105	150	170	180	200	245	185	205	240	250
RPM	4000	3600	5000	4000	4000	4000	4800	4000	4000	4800	4800
Torque	145	180	255	265	290	290	320	315	310	320	330
RPM	2000	1600	2200	2600	2400	3200	3200	2400	2400	3200	3400
Transmission Trade Name	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite
Rear Axle Ratios	2.76 3.21 3.23 3.55	2.76 2.94 3.21 3.23 3.55	2.45 3.55 2.71 2.76 2.94 3.21 3.23	2.71 3.21 3.23 3.55	2.71 2.94 3.23	2.71 3.23	2.76 2.94 3.21 3.23 3.55	2.71 3.23	2.71 3.23	2.71 3.23	2.76 3.23 3.55
Gear Ratio: 1	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45
Gear Ratio: 2	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20
Weight Range of Vehicles	2975-3400	2975-3700	3075-4330	3510-3815	3560-5050	3560-4750	3260-4380	4350-5270	3710-5305	4265-5305	3710-4535
Torque Converter Max. Ratio at Stall	2.16	2.16	2.16	2.16	2.02 | 2.16	2.02 | 2.16	2.02 | 2.16	2.02	2.02	2.02	2.16
Converter Diameter	10.75"	10.75"	10.75"	10.75"	11.75" | 10.75	11.75" | 10.75	11.75" | 10.75	11.75	11.75	11.75	10.75

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Chrysler Corp. Year: 1974 Cont'd.

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	440 V-8	440 V-8	440 V-8	440 V-8							
Carburetor	4V	4V	4V	4V							
Compression Ratio	8.2	8.2	8.2	8.2							
BHP	220	230	250	275							
RPM	4000	4000	4000	4400							
Torque	345	350	350	375							
RPM	3200	3200	3200	3200							
Transmission Trade Name	Torque Flite	Torque Flite	Torque Flite	Torque Flite							
Rear Axle Ratios	2.71 3.23	2.71 3.23	2.71 3.23	2.71 3.23 3.55							
Gear Ratio: 1	2.45	2.45	2.45	2.45							
Gear Ratio: 2	1.45	1.45	1.45	1.45							
Gear Ratio: 3	1.00	1.00	1.00	1.00							
Gear Ratio: Reverse	2.20	2.20	2.20	2.20							
Weight Range of Vehicles	4380-4230	4300-5230	4300-5205	3850-4660							
Torque Converter Max. Ratio at Stall	2.02	2.02	2.02	2.02 | 2.16							
Converter Diameter	11.75"	11.75"	11.75"	11.75" | 10.75							

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Chrysler Corp. **Year:** 1975

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	225 L-6	225 L-6	318 V-8	318 V-8	318 V-8	318 V-8	318 V-8	360 V-8	360 V-8	360 V-8	400 V-8
Carburetor	1V	1V	2V	2V	2V	2V	2V	2V	4V	4V	2V
Compression Ratio	8.4	8.4	8.5	8.5	8.5	8.5	8.4	8.4	8.4	8.4	8.2
BHP	90	95	135	140	145	150	150	180	190	230	165
RPM	3600	3600	3600	4000	4000	4000	4000	4000	4000	4400	4000
Torque	165	170	245	255	255	255	260	290	270	300	295
RPM	1600	1600	1600	1600	1600	1600	1600	2400	3200	3600	3200
Transmission Trade Name	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite
Rear Axle Ratios	2.45 2.71 2.76 3.21 3.23	2.45 2.71 2.76 2.94 3.21 3.23	2.45 2.71 3.21	2.45 2.71 3.21	2.45 2.71 3.21	2.45 2.71 3.21	2.45 2.71 3.21	2.45 2.71 3.21	2.45 2.71 3.21	2.94 3.21	2.45 2.71 3.21
Gear Ratio: 1	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45
Gear Ratio: 2	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20
Weight Range of Vehicles	2972-4493	2972-4493	3555-4493	2992-3572	2972-3572	3555-4493	4224-5224	3605-5274	3600-5150	2972-4493	3755-4693
Torque Converter Max. Ratio at Stall	2.16	2.16	2.02	2.16	2.16	2.02	2.02	2.02	2.02 | 2.16	2.16	2.02
Converter Diameter	10.75"	10.75"	11.75"	10.75"	10.75"	11.75"	11.75"	11.75"	11.75" | 10.75"	10.75"	11.75"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Chrysler Corp. Year: 1975 Cont'd.

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	400 V-8	400 V-8	400 V-8	400 V-8	400 V-8	400 V-8	440 V-8	400 V-8	440 V-8		
Carburetor	2V	4V	4V	4V	4V	4V	4V	4V	4V		
Compression Ratio	8.2	8.2	8.2	8.2	8.2	8.2	8.2	8.2	8.2		
BHP	175	185	190	195	235	240	215	250	260		
RPM	4000	4000	4000	4000	4200	4400	4000	4000	4400		
Torque	300	285	290	285	320	325	330	360	355		
RPM	2400	3200	3200	3200	3200	3200	3200	3200	3200		
Transmission Trade Name	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite	Torque Flite		
Rear Axle Ratios	2.45 2.71 3.21	2.45 2.71 3.21	2.45 2.71 3.21	2.71 3.21	3.21	2.71 3.21	2.71 3.21	2.71 3.21	2.71 3.21		
Gear Ratio: 1	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45	2.45		
Gear Ratio: 2	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45	1.45		
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00		
Gear Ratio: Reverse	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20	2.20		
Weight Range of Vehicles	4304-5424	3755-4493	3755-4493	4435-5300	3755-4493	4400-5400	4404-5424	4404-5329	4404-5424		
Torque Converter Max. Ratio at Stall	2.02	2.02	2.02	2.02	2.02	2.02	2.02	2.02	2.02		
Converter Diameter	11.75"	11.75"	11.75"	11.75"	11.75"	11.75"	11.75"	11.75"	11.75"		

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford **Year:** 1970

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	170 L-6	200 L-6	240 L-6	250 L-6	302 V-8	351 V-8	351 V-8	390 V-8	428 V-8	429 V-8	429 V-8
Carburetor	1V	1V	1V	1V	2V	2V	2V	2V	4V	2V	4V
Compression Ratio	8.7	9.7	9.2	9.0	9.5	9.5	11.0	9.5	10.6	10.5	10.5
BHP	105	120	150	155	220	250	300	265	335	320	360
RPM	4200	4000	4000	4000	4600	4600	5400	4400	5200	4400	4600
Torque	156	190	234	240	300	355	380	390	440	460	480
RPM	2200	2200	2200	1600	2600	2600	3400	2600	3400	2200	2800
Transmission Trade Name	Select Shift Cruise-o-matic	Select Shift Cruise-o-matic	SelectShift Cruise-o-matic	SelectShift Cruise-o-matic	Select Shift Cruise-o-matic	Select Shift Cruise-o-matic	Select Shift Cruise-o-matic	Select Shift Cruise-o-matic	Select Shift Cruise-o-matic	Select Shift Cruise-o-matic	Select Shift Cruise-o-matic
Rear Axle Ratios	2.83 3.08 3.20	2.83 3.08 3.20	2.75 3.07 3.25	2.79 3.00 2.83	2.79 3.00	2.75 3.00 3.25 3.07	3.00 3.25 3.50	2.75 3.25	3.25 3.50 3.25 3.91 4.30	2.75 3.25	2.80 3.25
Gear Ratio: 1	2.46	2.46	2.46	2.46	2.46	2.40	2.40	2.46	2.46	2.46	2.46
Gear Ratio: 2	1.46	1.46	1.46	1.46	1.46	1.47	1.47	1.46	1.46	1.46	1.46
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.20	2.20	2.20	2.20	2.20	2.00	2.00	2.18	2.18	2.18	2.18
Weight Range of Vehicles	2548-2748	2548-3218	3755-4030	2687-3367	3091-4619	3241-4705	3326-3668	4011-4420	3529-3963	4048-4880	4121-4953
Torque Converter Max. Ratio at Stall	2.10	2.10	2.05	2.02	2.02	2.05	2.05	2.05	2.05	2.05	2.05
Converter Diameter	10.25"	10.25"	12.00"	11.25"	11.25"	12.00"	12.00"	12.00"	12.00"	12.00"	12.00"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford

Year: 1970 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	460 V-8										
Carburetor	4V										
Compression Ratio	10.5										
BHP	365										
RPM	4600										
Torque	500										
RPM	2800										
Transmission Trade Name	Select Shift Cruise-o-matic										
Rear Axle Ratios	2.80 3.00										
Gear Ratio: 1	2.46										
Gear Ratio: 2	1.46										
Gear Ratio: 3	1.00										
Gear Ratio: Reverse	2.17										
Weight Range of Vehicles	4860-5110										
Torque Converter Max. Ratio at Stall	2.05										
Converter Diameter	12.00"										

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford **Year:** 1971

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	122 L-4	200 L-6	240 L-6	250 L-6	302 V-8	351 V-8	351 V-8	351 V-8	390 V-8	400 V-8	429 V-8
Carburetor	2V	1V	1V	1V	2V	2V	4V	4V	2V	2V	2V
Compression Ratio	9.0	8.7	8.9	9.0	9.0	9.0	10.7	9.0	8.6	9.0	10.5
BHP	100	115	140	145	210	240	285	280	225	260	320
RPM	5600	4000	4000	4000	4600	4600	5400	5800	4400	4400	4400
Torque	120	180	230	232	296	350	370	345	376	400	460
RPM	3600	2200	220	1600	2600	2600	3400	3800	2600	2200	2200
Transmission Trade Name	C-4 Cruise-o-matic	C-4	C-4	C-4	C-4	FMX	C-6	C-6	Cruise-o-matic	Cruise-o-matic	Cruise-o-matic
Rear Axle Ratios	3.18 3.55	2.79 3.00	3.00 3.25	2.79 3.00 3.25	2.79 2.75 3.00 3.25	2.75 3.00 3.25	2.75 3.00 3.25 3.50	2.75 3.00 3.25 3.50	2.75 3.25	2.75 3.25	2.75 3.25
Gear Ratio: 1	2.46	2.46	2.46	2.46	2.46	2.40	2.46	2.46	2.46	2.46	2.46
Gear Ratio: 2	1.46	1.46	1.46	1.46	1.46	1.47	1.46	1.46	1.46	1.46	1.46
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.20	2.20	2.20	2.20	2.20	2.00	2.18	2.18	2.18	2.18	2.18
Weight Range of Vehicles	2064-2308	2630-2925	3840-4256	3029-4045	3121-4095	3284-4821	3341-4118	3402-3554	4102-4786	4102-4786	4186-4875
Torque Converter Max. Ratio at Stall	2.60	2.10	2.05	2.02	2.10	2.05	2.05	2.16	2.05	2.05	2.05
Converter Diameter	10.25"	10.25"	12.00"	11.25"	11.25"	12.00"	12.00"	10.25"	12.00"	12.00"	12.00"

Vehicle Class

Subcompact ⩽ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ⩾ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford Year: 1971 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	429 V-8	429 V-8	429 V-8	460 V-8							
Carburetor	4V	4V	4V	4V							
Compression Ratio	10.5	11.3	11.3	10.0							
BHP	360	370	375	365							
RPM	4600	5400	5600	4600							
Torque	480	450	450	500							
RPM	2200	3400	3400	2800							
Transmission Trade Name	Cruise-o-matic	Cruise-o-matic	Cruise-o-matic	Cruise-o-matic							
Rear Axle Ratios	2.75 3.25	3.25 3.50	3.91 4.11	2.80 3.00							
Gear Ratio: 1	2.46	2.46	2.46	2.46							
Gear Ratio: 2	1.46	1.46	1.46	1.46							
Gear Ratio: 3	1.00	1.00	1.00	1.00							
Gear Ratio: Reverse	2.18	2.18	2.18	2.18							
Weight Range of Vehicles	4192-4881	3542-3914	3577-3949	5073-5313							
Torque Converter Max. Ratio at Stall	2.05	2.05	2.05	2.05							
Converter Diameter	12.00"	12.00"	12.00"	12.00"							

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford

Year: 1972

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	122 L-4 (2000 cc)	200 L-6	240 L-6	250 L-6	250 L-6	250 L-6	302 V-8	302 V-8	351W V-8	351C V-8	351C V-8
Carburetor	2V	1V	1V	1V	1V	1V	2V	2V	2V	2V	2V
Compression Ratio	8.2	8.3	8.5	8.0	8.0	8.0	8.5	8.5	8.3	8.6	8.6
BHP	86	91	103	95	98	99	140	143 | 141	153	163	177
RPM	5400	4000	3800	3600	3600	3600	4000	4000	3800	3800	4000
Torque	103	154	170	181	183	184	230 | 239	242	266	277	284
RPM	3200	2200	2200	1600	1600	1600	2200 | 2000	2000	2000	2000	2000
Transmission Trade Name	Select Shift Cruise-o-matic C-4	C-4	C-4	C-4	C-4	C-4 | FMX	C-4	C-4 | FMX	FMX	FMX	C-6 | FMX
Rear Axle Ratios	3.18 3.55	2.79 3.00	2.75 3.00 3.25	2.75 2.79 | 3.25 3.00	2.79 3.00	2.79 3.00	3.00 | 2.75 3.25	2.79 3.00	2.75 3.25	2.75 3.00 3.25	2.75 3.25
Gear Ratio: 1	2.46	2.46	2.46	2.46	2.46	2.46 | 2.40	2.46	2.46 | 2.40	2.40	2.40	2.46 | 2.40
Gear Ratio: 2	1.46	1.46	1.46	1.46	1.46	1.46 | 1.47	1.46	1.46 | 1.47	1.47	1.47	1.46 | 1.47
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00 | 1.00	1.00	1.00 | 1.00	1.00	1.00	1.00 | 1.00
Gear Ratio: Reverse	2.20	2.20	2.20	2.20	2.20	2.20 | 2.00	2.20	2.20 | 2.00	2.00	2.00	2.18 | 2.00
Weight Range of Vehicles	2140-2625	2681-2940	3888-4190	3540 4300 | 3600 4300	2770-3030	3040-3370	3623 4404 | 3850 4150	2880 3150 | 3140 3477	4088-4780	3750-4780	3300-3630
Torque Converter Max. Ratio at Stall	2.60	2.14	2.10	2.10	2.01	2.10 | 2.09	2.10	2.01 | 2.10	2.05	2.05	2.16 | 2.09
Converter Diameter	10.25"	10.25"	11.25"	11.25"	11.25"	11.25 | 12.00	11.25"	11.25"	12.00"	12.00"	10.25 | 12.00

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford

Year: 1972 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	351 V-8	351 V-8	400 V-8	400 V-8	429 V-8	429 V-8	460 V-8	460 V-8			
Carburetor	4V	4V	2V	2V	2V	4V	4V	4V			
Compression Ratio	8.6	8.6	8.4	8.4	8.5	8.5	8.5	8.5			
BHP	248	266	168	172	205 / 208	212	212	224			
RPM	5400	5400	4200	4000	4400	4400	4400	4400			
Torque	299	301	297	298	322	327	342	357			
RPM	3600	3600	2200	2200	2600 / 2800	2600	2800	2800			
Transmission Trade Name	Select Shift Cruise-o-matic C-6	C-6	C-6	C-6	C-6	C-6	C-6 / Select Shift	Select Shift			
Rear Axle Ratios	3.25 3.50	3.25 3.50	3.00 3.25	2.75 3.25	3.00 3.25 / 2.75	2.75 3.00 3.25	2.75 3.00 / 2.75 2.80 3.00	2.75 2.80 3.00			
Gear Ratio: 1	2.46	2.46	2.46	2.46	2.46	2.46	2.46	2.46			
Gear Ratio: 2	1.46	1.46	1.46	1.46	1.40	1.46	1.46	1.46			
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00			
Gear Ratio: Reverse	2.18	2.18	2.18	2.18	2.18	2.18	2.18 / 2.17	2.17			
Weight Range of Vehicles	4040-4150	3420-3750	3875-4650	4160-4850	3960, 4740 / 4250 4940	4582-4782	5084-5336	5084-5336			
Torque Converter Max. Ratio at Stall	2.05	2.16	2.05	2.05	2.05	2.05	2.05	2.05			
Converter Diameter	12.00"	10.25"	12.00"	12.00"	12.00"	12.00"	12.00"	12.00"			

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford **Year:** 1973

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	122 L-4	200 L-6	250 L-6	250 L-6	302 V-8	302 V-8	302 V-8	302 V-8	351W V-8	351W V-8	351W V-8
Carburetor	2V	1V	1V	1V	2V	2V	2V	2V	2V	2V	2V
Compression Ratio	8.2	8.3	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0
BHP	85	84	93	93	135	137	138	141	154	156	158
RPM	5600	3600	3200	3200	4200	4200	4200	4000	3800	3800	3800
Torque	98	151	200	200	228	230	234	242	256	260	260
RPM	3800	1800	1400	1400	2200	2200	2200	2000	2400	2400	2400
Transmission Trade Name	Select Shift Cruise-o-matic C-4	C-4	C-4	FMX	C-4 / FMX	C-4 / FMX	C-4	C-4 / FMX	FMX	C-4 / FMX	C-6
Rear Axle Ratios	3.40	2.79 3.00	2.79 3.00 / 3.25	2.79 3.00	3.00	2.75 2.79 3.00	2.79 3.00	2.79 3.00	2.75 3.25	2.75 3.25	2.75 3.25
Gear Ratio: 1	2.46	2.46	2.46	2.40	2.46 / 2.40	2.46 / 2.40	2.46	2.46 / 2.40	2.40	2.46 / 2.40	2.46
Gear Ratio: 2	1.46	1.46	1.46	1.47	1.46 / 1.47	1.46 / 1.47	1.46	1.46 / 1.47	1.47	1.46 / 1.47	1.46
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00 / 1.00	1.00 / 1.00	1.00	1.00 / 1.00	1.00	1.00 / 1.00	1.00
Gear Ratio: Reverse	2.20	2.20	2.20	2.00	2.20 / 2.00	2.20 / 2.00	2.20	2.20 / 2.00	2.00	2.20 / 2.00	2.18
Weight Range of Vehicles	2250-2740	2761-3055	2850-3370 / 2850-3980	2850-3370	3750-4470	3743-4075	2940-3185	3239-3570	4695-4950	3870-4240	3870-4240
Torque Converter Max. Ratio at Stall	2.60	2.14	2.01 / 2.10	2.09	2.02 / 2.09	2.02 / 2.09	2.01	2.10 / 2.09	2.05	2.02 / 2.09	2.05
Converter Diameter	10.25"	10.25"	11.25"	12.00"	11.25 / 12.00	11.25 / 12.00	11.25"	11.25 / 12.00	12.00"	11.25 / 12.00	12.00"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford

Year: 1973 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	351W V-8	351C V-8	351C V-8	351C V-8	351C V-8	351C V-8	351C V-8	400 V-8	400 V-8	429 V-8	429 V-8
Carburetor	2V	2V	2V	2V	2V	2V	2V	2V	2V	4V	4V
Compression Ratio	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.4	8.0	8.5	8.0
BHP	158	154	154	159	159	161	177	167 / 171	163 / 168	198 / 202	197 / 201
RPM	3800	4000	4000	4000	4000	4000	4000	3600	3800	4400	4400
Torque	264	246	246	250	250	254	284	312 / 314	310 / 300	320	320 / 322
RPM	2400	2400	2400	2400	2400	2400	2000	2200 / 2000	2000	2800	2600
Transmission Trade Name	FMX	C-4 / FMX	C-6	C-4 / FMX	C-6	FMX	FMX	C-6	C-6	C-4	C-6
Rear Axle Ratios	2.75 3.25	3.00 3.25	3.00 3.25	2.75 3.25	2.75 3.25	2.75 3.00 3.25	2.75 3.25	3.00 / 2.75 3.25	2.75	3.00 / 2.75 3.25	3.00 3.25 / 2.75 3.25
Gear Ratio: 1	2.40	2.46 / 2.40	2.46	2.46 / 2.40	2.46	2.40	2.40	2.46	2.46	2.46	2.46
Gear Ratio: 2	1.47	1.46 / 1.47	1.47	1.46 / 1.47	1.46	1.47	1.47	1.46	1.46	1.46	1.46
Gear Ratio: 3	1.00	1.00 / 1.00	1.00	1.00 / 1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.00	2.20 / 2.00	2.18	2.20 / 2.00	2.18	2.00	2.00	2.18	2.18	2.20	2.18
Weight Range of Vehicles	4205-4500	3756-4470	3756-4470	3743-4075	3743-4075	4205-4950	3440-3770	4750 5000 / 4250 4560	4010 4500 / 4000 4250	4975 5050 / 4375 4700	4080 4800 / 4000 4400
Torque Converter Max. Ratio at Stall	2.05	2.02 / 2.09	2.05	2.02 / 2.09	2.05	2.05	2.09	2.05	2.05	2.05	2.05
Converter Diameter	12.00"	11.25 / 12.00	12.00"	11.25 / 12.00	12.00"	12.00"	12.00"	12.00"	12.00"	12.00"	12.00"
Vehicle Class											
Subcompact ≤ 100"											
Compact 101 – 111"							—				
Intermediate 112 – 119"		—	—	—	—				—		—
Full Size ≥ 120"	—					—		—		—	

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford

Year: 1973 (Cont.)

Power Team Number	12		13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	460 V-8											
Carburetor	4V											
Compression Ratio	8.0	8.6										
BHP	219	207										
RPM	4400	4600										
Torque	360	383										
RPM	2800	2800										
Transmission Trade Name	Select Shift	C-4										
Rear Axle Ratios	2.75 3.00	3.25										
Gear Ratio: 1	2.46	2.46										
Gear Ratio: 2	1.46	1.46										
Gear Ratio: 3	1.00	1.00										
Gear Ratio: Reverse	2.17	2.20										
Weight Range of Vehicles	5214 5450	4405 5150										
Torque Converter Max. Ratio at Stall	2.05											
Converter Diameter	12.00"											

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119" ____________

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford   Year: 1974

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	122 L-4 (2000 cc)	140 L-4 (2300 cc)	140 L-4 (2300 cc)	170.8 V-6 (2800 cc)	200 L-6	250 L-6	302 V-8	302 V-8	351W V-8	351W V-8	351C V-8
Carburetor	2V	2V	2V	2V	1V	1V	2V	2V	2V	2V	2V
Compression Ratio	8.2	8.4	8.4	8.2	8.3	8.0	8.0	8.0	8.0	8.0	8.0
BHP	80	82	88	105	84	91	140	140	162	162	163
RPM	5400	4600	5000	4600	3800	3200	3800	3800	4000	4000	4200
Torque	98	113	116	140	150	190	230	230	275	275	278
RPM	3000	2600	2600	3200	1800	1600	2600	2600	2200	2200	2000
Transmission Trade Name	Cruise-o-matic C-4	C-4	C-3 / C-4	C-3 / C-4	C-4	C-4	C-4	FMX	C-4 / FMX	C-6	C-4 / FMX
Rear Axle Ratios	3.40 3.55	3.40	3.55	3.55	2.79 3.00	2.79 3.00	2.79 3.00	3.00	2.75 / 3.00 3.07; 3.25	2.75; 3.25	2.75; 3.25
Gear Ratio: 1	2.46	2.46	2.47 / 2.46	2.47 / 2.46	2.46	2.46	2.46	2.40	2.46 / 2.40	2.46	2.46 / 2.40
Gear Ratio: 2	1.46	1.46	1.47 / 1.46	1.47 / 1.46	1.46	1.46	1.46	1.47	1.46 / 1.47	1.46	1.46 / 1.47
Gear Ratio: 3	1.00	1.00	1.00 / 1.00	1.00 / 1.00	1.00	1.00	1.00	1.00	1.00 / 1.00	1.00	1.00 / 1.00
Gear Ratio: Reverse	2.20	2.20	2.11 / 2.20	2.11 / 2.20	2.20	2.20	2.20	2.20	2.20 / 2.00	2.18	2.20 / 2.00
Weight Range of Vehicles	2440-2843	2643-2843	2773-3123	2880-3230	2840-5140	2900-4300	3000-4601	3881-4601	3788 4680 / 3788 5050	3788-4680	3788-4680
Torque Converter Max. Ratio at Stall	2.60	2.60	2.60	2.60	2.14	2.10	2.10	2.01 / 2.09	2.14	2.05	2.14
Converter Diameter	10.25"	10.25"	10.25"	10.25"	10.25"	11.25"	11.25"	11.25"	12.00"	12.00"	12.00"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford

Year: 1974 (Cont.)

Power Team Number	12	13	14		15		16	17	18	19	20	21	22
Engine Displacement and Type	351C V-8	351 V-8	400 V-8		460 V-8		460 V-8	460 V-8	460 V-8	460 V-8			
Carburetor	2V	4V	2V		4V		4V	4V	4V	4V			
Compression Ratio	8.0	8.0	8.0		8.0		8.0	8.0	8.8	8.8			
BHP	163	255	170		195		215	220	260	275			
RPM	4200	5600	3400		3800		4000	4000	4400	4400			
Torque	278	290	330		355		350	355	380	395			
RPM	2000	3400	2000		2600		2600	2600	2700	2800			
Transmission Trade Name	C-6	C-6	C-4	C-6	C-6	CW	Select Shift	C-6	C-6	CW			
Rear Axle Ratios	3.00 3.25	3.25	3.00	2.75 3.25	2.75 3.25	3.00	2.75 3.00	2.75 3.25	2.75 3.25	3.00			
Gear Ratio: 1	2.46	2.46	2.46	2.46	2.46	2.40	2.46	2.46	2.46	2.40			
Gear Ratio: 2	1.46	1.46	1.46	1.46	1.46	1.47	1.46	1.46	1.46	1.47			
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00			
Gear Ratio: Reverse	2.18	2.18	2.20	2.18	2.18	2.00	2.175	2.18	2.18	2.00			
Weight Range of Vehicles	3788-4940	4090-4420	4480 5270	3990 4801	4609 4901	4480 5270	5195-5410	4180-4600	4180-4901	4480-4830			
Torque Converter Max. Ratio at Stall	2.05	2.16	2.14	2.05	2.05	2.14	2.05	2.05	2.05	2.14			
Converter Diameter	12.00"	10.25"	12.00"		12.00"		12.00"	12.00"	12.00"	12.00"			

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford Year: 1975

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	140 L-4 (2300 cc)	140 L-4 (2300 cc)	140 L-4 (2300 cc)	171 V-6 (2800 cc)	171 V-6 (2800 cc)	171 V-6 (2800 cc)	200 L-6	250 L-6	250 L-6	302-V-8	302 V-8
Carburator	2V	2V	2V	2V	2V	2V	1V	1V	1V	2V	2V
Compression Ratio	8.2	8.2	8.2	8.2	8.2	8.2	NA	8.0	8.0	8.0	8.0
BHP	87	87	87	97	97	97	NA	70	72	115	122
RPM	4600	4600	4600	4400	4400	4400	NA	2800	2900	3600	3800
Torque	114	114	114	138	138	138	NA	175	180	203	208
RPM	2800	2800	2800	3200	3200	3200	NA	1400	1400	1400	1800
Transmission Trade Name	C-3	C-3	C-4	C-3	C-4	C-4	C-4	C-4	C-4	C-4	C-4
Rear Axle Ratios	3.40 3.55	3.40 3.55	3.40 3.55	3.40	3.00 3.40	3.00 3.40	2.79	3.00	3.00	3.00	3.00
Gear Ratio: 1	2.47	2.47	2.46	2.47	2.46	2.46	2.46	2.46	2.46	2.46	2.46
Gear Ratio: 2	1.47	1.47	1.46	1.47	1.46	1.46	1.46	1.46	1.46	1.46	1.46
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.11	2.11	2.20	2.11	2.20	2.20	2.20	2.20	2.20	2.20	2.20
Weight Range of Vehicles	2574-3157	2754-3157	2574-2981	2725-3130	2725-3260	2725-3260	2820-3150	3020-3420	3020-3430	3170-3580	3170-3580
Torque Converter Max. Ratio at Stall	2.49	2.45	2.45	2.45	2.35	2.45	2.04	2.04	2.04	2.04	2.04
Converter Diameter	10.25"	10.25"	10.25"	10.25"	10.25"	10.25"	10.25"	11.25"	11.25"	11.24"	11.25"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford **Year:** 1975 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	302 V-8	302 V-8	351M V-8	351M V-8	351M V-8	351M V-8	351M V-8	351W V-8	351W V-8	351W-V-8	351M V-8
Carburetor	2V	2V	2V	2V	2V	2V	2V	2V	2V	2V	2V
Compression Ratio	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.2	8.2	8.2	8.0
BHP	129	133	148	148	148	148	148	154	154	154	150
RPM	4000	3600	3800	3800	3800	3800	3800	3800	3800	3800	3800
Torque	213	223	243	243	243	243	243	268	268	268	244
RPM	1800	2000	2400	2400	2400	2400	2400	2200	2200	2200	2800
Transmission Trade Name	C-4	C-4	C-4	FMX	FMX	FMX	C-6	C-4	FMX	C-6	FMX
Rear Axle Ratios	3.00	3.00	2.75 3.00 3.25	2.75 3.00 3.25	2.75 3.00 3.25	2.75 3.00 3.25	2.75 3.00 3.25	3.00	3.00	3.00	2.75 3.25
Gear Ratio: 1	2.46	2.46	2.46	2.40	2.40	2.40	2.46	2.46	2.40	2.46	2.40
Gear Ratio: 2	1.46	1.46	1.46	1.47	1.47	1.47	1.46	1.46	1.47	1.46	1.47
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.20	2.20	2.20	2.00	2.00	2.00	2.18	2.20	2.00	2.18	2.00
Weight Range of Vehicles	3160-3557	3160-3557	4065-4809	4065-4809	4065-4809	4065-4809	4065-4809	4141-4472	4141-4472	4141-4472	4275-4606
Torque Converter Max. Ratio at Stall	2.04	2.04	2.14	2.11	2.12	2.14	2.05	2.14	2.14	2.05	2.11
Converter Diameter	10.25"	10.25"	12.00"	12.00"	12.00"	12.00"	12.00"	12.00"	12.00"	12.00"	12.00"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: Ford Year: 1975 (Con't)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	400 V-8	400 V-8	400 V-8	400 V-8	460 V-8	460 V-8	460 V-8	460 V-8	460 V-8		
Carburetor	2V	2V	2V	2V	4V	4V	4V	4V	4V		
Compression Ratio	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0		
BHP	144	144	158	158	206	216	218	223	226		
RPM	3600	3600	3800	3800	3800	4000	4000	4000	4000		
Torque	255	255	276	276	357	366	369	266	374		
RPM	2200	2200	2000	2000	2600	2600	2600	2600	2600		
Transmission Trade Name	FMX	C-6	FMX	C-6	Select Shift	C-6	C-6	Select Shift	C-6		
Rear Axle Ratios	2.75 3.25	2.75 3.25	2.75 3.00 3.25	2.75 3.00 3.25	2.75 3.00	2.75 3.00	2.75 3.00	3.00	3.00		
Gear Ratio: 1	2.40	2.46	2.47	2.46	2.46	2.46	2.46	2.46	2.46		
Gear Ratio: 2	1.47	1.46	1.47	1.46	1.46	1.46	1.46	1.46	1.46		
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00		
Gear Ratio: Reverse	2.00	2.18	2.00	2.18	2.175	2.18	2.18	2.175	2.18		
Weight Range of Vehicles	4500-5250	4500-5250	4120-5250	4120-5250	5235-5477	4400-5000	4670-5350	5235-5477	4670-5350		
Torque Converter Max. Ratio at Stall	2.12	2.08	2.12	2.08	2.05	2.05	2.03	2.05	2.03		
Converter Diameter	12.00"	12.00"	12.00"	12.00"	12.00"	12.00"	12.00"	12.00"	12.00"		

Vehicle Class

Subcompact ⩽ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ⩾ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: General Motors **Year:** 1970

Power Team Number	1	2		4		5	6		7		8	9		10	11	
Engine Displacement and Type	153 L-4	230 L-6	230 L-6	250 L-6		250 L-6	307 V-8		350 V-8		350 V-8	350 V-8		350 V-8	350 V-8	
Carburetor	1V	1V	1V	1V		1V	2V		2V		2V	2V		2V	4V	
Compression Ratio	8.5	8.5	8.5	8.5		8.5	9.00		9.00		9.00	8.8		8.8	10.25	
BHP	90	140	140	155		155	200		250		250	255		255	300	
RPM	4000	4400	4400	4200		4200	4600		4800		4400	4600		4600	4800	
Torque	152	220	220	235		235	300		345		355	355		355	380	
RPM	2400	1600	1600	1600		1600	2400		2800		2600	2800		2800	3700	
Transmission Trade Name	Torque-Drive	Torque Drive / Power Glide	Turbo Hydramatic	Torque Drive	Power Glide	Turbo Hydramatic	Power Glide	Turbo Hydra-matic	Power Glide	Turbo Hydra-matic	Turbo Hydramatic	Pontiac 2-Speed		Pontiac 2-Speed	Power Glide	Turbo Hydra-matic
Rear Axle Ratios	3.08	2.73	3.56	2.73	3.36	2.73 3.08 2.56 2.78 3.08 3.23	2.73 3.08 3.06	2.56	2.56 2.73 3.36		3.08 3.23	3.36	3.23	3.23	2.73 3.07 3.08	
Gear Ratio: 1	1.82	1.82	2.52	1.82		2.52	1.82	2.52	1.76	2.52	2.52	1.88	1.76	1.76	1.76	2.52
Gear Ratio: 2	1.00	1.00	1.52	1.00		1.52	1.00	1.52	1.00	1.52	1.52	1.00	1.00	1.00	1.00	1.52
Gear Ratio: 3			1.00			1.00		1.00		1.00	1.00					1.00
Gear Ratio: Reverse	1.82	1.82	1.93	1.82		1.93	1.82	1.93	1.76	1.93	1.93	1.82	1.76	1.76	1.76	1.93
Weight Range of Vehicles	2914-3140	3013-3210	3013-3210	2914 3370	2914 4000	2914-4000	3143-3991		3200 4740	3225 4770	3572 3800 4000 4400	3534 3750	3622 4240	4167-4400	3200 4740	3225 4770
Torque Converter Max. Ratio at Stall	2.40	2.10	2.10	2.10		2.10	2.10		2.10							
Converter Diameter	11.00"	11.75"	11.75"	11.75"		11.75"	11.75"		11.75"							

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: General Motors   Year: 1970 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	350 V-8	350 V-8	350 V-8	400 V-8	400 V-8	400 V-8	400 V-8	400 V-8	400 V-8	400 V-8	400 V-8
Carburetor	4V	4V	4V	4V	2V	4V	4V	4V	4V	4V	4V
Compression Ratio	10.25	10.25	11.00	10.25	8.8	10.0	10.25	10.0	10.25	10.25	10.5
BHP	310	325	360	350	265	290	330	330	345	350	366 / 370
RPM	4800	5400	6000	5200	4600	4600	4800	4800	5000	5000	5100 / 5500
Torque	390	360	380	415	397	428	410	445	430	445	445
RPM	3200	3600	4000	3400	2400	2500	3200	2900	3400	3000	3900 / 3600
Transmission Trade Name	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic
Rear Axle Ratios	3.08 3.23	3.08	3.73 4.10	3.31	2.78 3.23	3.23	3.31	3.23	3.08 3.36	2.93 3.23 3.55	3.55
Gear Ratio: 1	2.48	2.52	2.48	2.48	2.48	2.48	2.48	2.48	2.48	2.48	2.48
Gear Ratio: 2	1.48	1.52	1.48	1.48	1.48	1.48	1.48	1.48	1.48	1.48	1.48
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.08	1.93	2.08	2.08	2.08	2.08	2.08	2.08	2.08	2.08	2.08
Weight Range of Vehicles	3572-4400	3600-4000	3430-3670	3500-3875	3620-4150	4167-4400	3546-4480	3530-4650	3550-3800	3780-4240	3781-4000
Torque Converter Max. Ratio at Stall			2.10	2.10			2.10			2.30	
Converter Diameter			12.20"	12.20"			12.20"			12.50"	

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: General Motors **Year:** 1970 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	454 V-8	454 V-8	455 V-8	455 V-8	455 V-8	455 V-8	455 V-8	455 V-8	455 V-8	455 V-8	455 V-8
Carburetor	4V	4V	4V	4V	4V	4V	4V	4V	4V	4V	4V
Compression Ratio	10.25	10.25	9.0	10.25	10.0	10.0	10.5	10.25	10.0	10.25	10.25
BHP	345 / 360	390	310	320	350 / 360	360	365 / 370	370	370	375 / 400	390
RPM	4400 / 4400	4800	4200	4200	4600	4600	5000 / 5200	4600	4600	4600 / 4800	5000
Torque	500		490	500	510	500	500	500	510	510 / 500	500
RPM	3000 / 3200	3400	2400	2400	2800	3100	3600	3100	2800	3000 / 3200	3200
Transmission Trade Name	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic
Rear Axle Ratios	2.56 / 3.31	2.73	2.93 3.08 3.23	3.08	3.42	3.23 3.55	3.08 2.56	3.07 3.31 3.23 3.42	2.56 2.78	3.07	2.93 3.23
Gear Ratio: 1	2.48	2.48	2.48	2.48	2.48	2.48	2.48	2.48	2.48	2.48	2.48
Gear Ratio: 2	1.48	1.48	1.48	1.48	1.48	1.48	1.48	1.48	1.48	1.48	1.48
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.08	2.08	2.08	2.08	2.08	2.08	2.08	2.08	2.08	2.08	2.08
Weight Range of Vehicles	3980 4830 / 3700 4000	4030-4880	3572-4380	3572-3770	3769-3970	3781-4850	3771 4597 / 3771 3970	3970-4170	4310-5130	4498-4700	4078-4370
Torque Converter Max. Ratio at Stall	2.10	2.10						2.30	2.05	2.05	
Converter Diameter	12.20"	12.20"						12.5"	11.75"	11.75"	

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: General Motors

Year: 1970 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	472 V-8	500 V-8 (8200 cc)									
Carburetor	4V	4V									
Compression Ratio	10.0	10.0									
BHP	375	400									
RPM	4400	4400									
Torque	525	550									
RPM	3000	3000									
Transmission Trade Name	Hydramatic	Turbo Hydramatic									
Rear Axle Ratios	2.93 3.15	3.07									
Gear Ratio: 1	2.48	2.48									
Gear Ratio: 2	1.48	1.48									
Gear Ratio: 3	1.00	1.00									
Gear Ratio: Reverse	2.09	2.09									
Weight Range of Vehicles	4758-5780	4721									
Torque Converter Max. Ratio at Stall	2.03	2.03									
Converter Diameter	13.04"	13.04"									

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: General Motors **Year:** 1971

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	140 L-4	140 L-4	250 L-6	307 V-8	307 V-8	350 V-8	350 V-8	350 V-8	350 V-8	350 V-8	350 V-8
Carburetor	1V	2V	1V	2V	2V	2V	2V	2V / 4V	4V		4V
Compression Ratio	8.00	8.00	8.5	8.5	8.5	8.5	8.5	8.5	8.5	8.0	8.5
BHP (NET)	80	93	110	140	140	165	165	(Gross) 230 / 260	(Gross) 240 / 260	250 (Gross)	175
RPM	4400	4800	4200	4400	4400	4800	4800	4400 / 4600	4200 / 4600	4400	4800
Torque (NET)	121,	121	185	235	235	280	280	(Gross) 350 / 360	(Gross) 350 / 360	350 (Gross)	290
RPM	2800	3200	1600	2400	2400	2800	2800	2400 / 3000	2400 / 3200	2400	3200
Transmission Trade Name	Torque Drive / Power Glide	Torque Drive / Power Glide	Torque Drive / Power Glide	Power Glide	Turbo Hydramatic	Power Glide	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic
Rear Axle Ratios	2.92	3.36	3.08; 2.73 3.23 / 3.36	3.08 / 3.36	2.56 2.73 3.36 / 3.08 3.31	2.73	3.08 / 2.56 3.31; 3.42	3.08 3.42	3.08 / 3.23	3.23 3.42	2.73 3.07 3.08 3.31 3.42
Gear Ratio: 1	1.82	1.82	1.82	1.82	2.52	1.76	2.52	2.52	2.52	2.52	2.52
Gear Ratio: 2	1.00	1.00	1.00	1.00	1.52	1.00	1.52	1.52	1.52	1.52	1.52
Gear Ratio: 3					1.00		1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	1.82	1.82	1.82	1.82	1.93	1.76	1.93	1.93	1.93	1.93	1.93
Weight Range of Vehicles	2250–2530	2250–2530	3040 3400 / 3325 4060	3036 3510 / 3264 4040	3026, 3510 / 3264 4040	4014–5042	4014 5042 / 3050 4240	3580–4452	3542 4490 / 3645 3850	3580–4800	3100–5040
Torque Converter Max. Ratio at Stall	2.40 / 2.10	2.40 / 2.10	2.10	2.10	2.10	2.10	2.10				2.10
Converter Diameter	11.0 / 11.75	11.0 / 11.75	11.75 / 11.0	11.75 / 11.0	11.75"	11.00"	11.75"				12.20"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: General Motors **Year:** 1971 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	350 V-8	390(402)V-8	400 V-8	400 V-8	400 V-8	400 V-8	400 V-8	454 V-8	454 V-8	455 V-8	455 V-8
Carburetor	4V	4V	2V			4V	4V	4V	4V		
Compression Ratio	8.5	8.5	8.5	8.2	8.2	8.5	8.2	8.5	9.0	8.5	8.2
BHP	330 (Gross)	260	170	265 (Gross)	300 (Gross)	206	255	285	325	(Gross) 280 | 370	280 (Gross)
RPM	4600	4400	3400	4400	4800	4400	4400	4000	5600	4000 | 4400	4400
Torque	360 (Gross)	345	325	400 (Gross)	400 (Gross)	323	340	390	390	(Gross) 445 | 460	455 (Gross)
RPM	4000	3200	2000	2400	2400	2400	3200	3200	3600	2000 | 2800	2000
Transmission Trade Name	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic
Rear Axle Ratios	3.73 4.10	2.73 3.31 3.42	2.73 3.08 3.42	2.73 3.23 3.42	3.23 3.55	2.73 3.42	3.08 3.23	2.73 3.08 | 3.31	3.31 4.10	3.08 3.23 2.93 | 2.73	2.73 3.08
Gear Ratio: 1	2.48	2.48	2.52	2.48	2.48	2.48	2.48	2.48	2.48	2.48	2.48
Gear Ratio: 2	1.48	1.48	1.52	1.48	1.48	1.48	1.48	1.48	1.48	1.48	1.48
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.08	2.08	1.93	2.08	2.08	2.08	2.08	2.08	2.08	2.08	2.08
Weight Range of Vehicles	3425-3630	3500-4470	4014-4842	3500-4430	3730-4175	4125-5170	3966-4170	4165-5000 | 3500-3700	3300-3700	4285-5000	4433-5170
Torque Converter Max. Ratio at Stall	2.10	2.10	2.10			2.10	2.30	2.10	2.10		
Converter Diameter	12.20"	12.20"	11.75"			12.20"	12.5"	12.20"	12.20"		

Vehicle Class

Subcompact ⩽ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ⩾ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: General Motors **Year:** 1971 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	455 V-8	455 V-8	455 V-8	455 V-8	455 V-8	455 V-8	472 V-8	500 V-8			
Carburetor	4V	4V	4V				4V	4V			
Compression Ratio	8.5	8.5	8.2	8.4	8.5	8.5	8.5	8.5			
BHP	230 | 255	265	260	335 (Gross)	340 (Gross)	350 (Gross)	220	235			
RPM	4000 | 4200	4400	4000	4800	4600	440	4000	3800			
Torque	365 | 380	385	380	480 (Gross)	460 (Gross)	465 (Gross)	380	410			
RPM	2600 | 2800	2800	2800	3600	3200	2800	2400	2400			
Transmission Trade Name	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Hydramatic			
Rear Axle Ratios	2.73 2.93 3.42	2.93 3.42	3.07 3.08 3.23 3.42	3.23 3.42	3.23	3.07 3.23	2.93 3.15	3.07			
Gear Ratio: 1	2.48	2.48	2.48	2.48	2.48	2.48	2.48	2.48			
Gear Ratio: 2	1.48	1.48	1.48	1.48	1.48	1.48	1.48	1.48			
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00			
Gear Ratio: Reverse	2.08	2.08	2.08	2.08	2.08	2.08	2.09	2.09			
Weight Range of Vehicles	4391-4980	4391-4980	3579-4413	3579-3864	3792-4000	3792-4870	4769-6160	4822-5300			
Torque Converter Max. Ratio at Stall	2.20	2.20	2.30			2.05	2.03	2.03			
Converter Diameter	11.75"	11.75"	12.50"			11.75"	13.04"	13.04"			

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: GMC   Year: 1972

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	140 L-4	140 L-4	140 L-4	140 L-4	250 L-6	307 V-8	307 V-8	350 V-8	350 V-8	350 V-8	350 V-8
Carburetor	1	1	2	3	1	2	2	2	4	4	4
Compression Ratio	8.00	8.00	8.00	8.00	8.51	8.5	8.5	8.5	8.5	8.5	9.00
BHP	80	80	90	90	110	130	130	165	175	200	255
RPM	4400	4400	4800	4800	3800	4000	4000	4000	4000	4400	5600
Torque	121	121	124	124	185	230	230	280	280	300	280
RPM	24-2800	24-2800	28-3200	28-3200	1600	2400	2400	2400	2400	2800	4000
Transmission Trade Name	Powerglide	Turbo Hydramatic	Powerglide	Turbo Hydramatic	Powerglide	Powerglide	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic
Rear Axle Ratios	2.92 3.36	2.92 3.36	3.36	3.36	3.08 3.36	3.08 3.36	2.73 3.08 3.31 3.42	2.73 2.56 3.08 3.31 3.42	2.73 3.31	3.08 3.42	3.73 4.10
Gear Ratio: 1	1.82	2.52	1.82	2.52	1.82	1.82	2.52	2.52	2.52	2.52	2.48
Gear Ratio: 2	1.00	1.52	1.00	1.52	1.00	1.00	1.52	1.52	1.52	1.52	1.48
Gear Ratio: 3		1.00		1.00			1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	1.82	1.93	1.92	1.93	1.82	1.82	1.93	1.93	1.93	1.93	2.08
Weight Range of Vehicles	2213-2588	2213-2588	2220-2590	2220-2590	3032-5205	3199-4162	3199-4162	3911-5008	3199-5008	3219-3560	3340-3540
Torque Converter Max. Ratio at Stall	2.10	2.10	2.10	2.10	2.10	2.10	2.10	2.10	2.10	2.10	2.10
Converter Diameter	11.00"	11.75"	11.00"	11.75"	11.75"	11.75"	11.76"	11.76"	11.75"	11.76"	12.20"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: GMC **Year:** 1972 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	400 V-8	400 V-8	402 V-8	402 V-8	454 V-8	454 V-8	455 V-8	455 V-8	455 V-8	472 V-8	500 V-8
Carburetor	2	4	4	4	4	4	4	4	4	4	4
Compression Ratio	8.5	8.2	8.5	8.5	8.5	8.5	8.5	8.2	8.5	8.5	8.5
BHP	170	250	210	240	230	270	225	250	260	220	235
RPM	3400	4400	4400	4400	4000	4000	4000	3600	4400	4000	4000
Torque	325	325	320	345	360	390	360	375	380	365	385
RPM	200	3200	2400	3200	3200	3200	2600	2400	2800	2400	2400
Transmission Trade Name	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic 400	Turbo Hydramatic	Turbo Hydramatic 400	Turbo Hydramatic	Turbo Hydramatic
Rear Axle Ratios	2.73 3.08 3.42	3.08 3.23	2.73 3.08 3.42	2.73 3.31 3.42	2.73 3.08	2.73 3.08 3.31	2.73 2.93 3.42	2.73 2.93 3.07 3.31 3.42	2.93 3.42	2.93 3.15	3.07
Gear Ratio: 1	2.52	2.48	2.48	2.48	2.48	2.48	2.48	2.48	2.48	2.48	2.48
Gear Ratio: 2	1.52	1.48	1.48	1.48	1.48	1.48	1.48	1.48	1.48	1.48	1.48
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	1.93	2.08	2.08	2.08	2.08	2.08	2.08	2.08	2.08	2.08	2.09
Weight Range of Vehicles	4171-5008	3962-4162	4171-5008	3518-4584	4808-5008	3510-4746	4490-4884	3962-4884	4490-4760	4189-5984	4829-5119
Torque Converter Max. Ratio at Stall	2.1	2.3	2.1	2.1	2.1	2.1	2.2	2.3	2.2	2.03	2.03
Converter Diameter	11.75"	12.5"	12.20"	12.20"	12.20"	12.20"	11.75"	12.5"	11.75"	13.038"	13.038"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: GMC Year: 1973

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	140 L-4	140 L-4	140 L-4	140 L-4	250 L-6	250 L-6	307 V-8	350 V-8	350 V-8	350 V-8	400 V-8
Carburetor	1	1	2	2	1	1	2	2	4	4	2
Compression Ratio	8.00	8.00	8.00	8.00	8.25	8.25	8.5	8.5	8.5	9.00	8.5
BHP	72	72	85	85	100	100	115	145	175 | 190	245 | 250	150
RPM	4400	4400	4800	4800	3600	3600	3600	4000	4000 | 4400	5200	3200
Torque	100	100	115	115	175	175	205	225	260 | 270	280 | 285	295
RPM	2000	2000	2400	2400	1600	1600	2000	2400	2800	4000	2000
Transmission Trade Name	Powerglide	Turbo Hydramatic	Powerglide	Turbo Hydramatic	Powerglide	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic
Rear Axle Ratios	2.92 3.36	2.92 3.36	3.36	2.92 3.36	3.08	3.42 3.08	3.08 3.42 2.73	2.73 3.08 3.42	2.73 3.42 3.08	3.73 3.36 3.42	2.73 3.08 3.42
Gear Ratio: 1	1.82	2.52	1.82	2.52	1.82	2.52	2.52	2.52	2.52 | 2.48	2.48	2.52
Gear Ratio: 2	1.00	1.52	1.00	1.52	1.00	1.52	1.52	1.52	1.52 | 1.48	1.48	1.52
Gear Ratio: 3		1.00		1.00		1.00	1.00	1.00	1.00 | 1.00	1.00	1.00
Gear Ratio: Reverse	1.82	1.93	1.82	1.93	1.82	1.93	1.93	1.93	1.93 | 2.08	2.08	1.93
Weight Range of Vehicles	2263-2584	2303-2624	2273-2594	2313-2634	3134-4238	3164-4268	3266-4343	3220-4448	3280 | 3407 4555 | 3610	3238 | 3407 3635 | 3610	4256-5197
Torque Converter Max. Ratio at Stall	2.10	2.10	2.10	2.10	2.10	2.10	2.10	2.10	2.10	2.10	2.10
Converter Diameter	11.00"	11.00"	11.00"	11.00"	11.00"	11.75"	11.75"	11.75"	11.75"	12.20"	11.75"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: GMC

Year: 1973 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	400 V-8	454 V-8	454 V-8	455 V-8	455 V-8	455 V-8	455 V-8	472 V-8	500 V-8		
Carburetor	4V	4V	4V	4	4	4	4	4	4		
Compression Ratio	8.0	8.25	8.25	8.5	8.0	8.5	8.4	8.5	8.5		
BHP	230	245	275	225	250	260	310	220	235		
RPM	4400	4000	4400	4000	4000	4400	4000	4000	3800		
Torque	325	375	395	360	370 / 375	380	390	365	385		
RPM	3200	2800	2800	2600	2800 / 4000	2800	3600	2400	2400		
Transmission Trade Name	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic 400	Turbo Hydramatic 400	Turbo Hydramatic 400	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic		
Rear Axle Ratios	2.93 3.08	2.73 3.42	3.36	2.93 3.23 2.73	3.08 3.23 / 2.73 3.23 2.93	3.23	3.42	2.93 3.15	3.07		
Gear Ratio: 1	2.48	2.48	2.48	2.48	2.48	2.48	2.48	2.48	2.48		
Gear Ratio: 2	1.48	1.48	1.48	1.48	1.48	1.48	1.48	1.48	1.48		
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00		
Gear Ratio: Reverse	2.08	2.08	2.08	2.08	2.08	2.08	2.08	2.09	2.09		
Weight Range of Vehicles	4124-4324	3932-5197	3407-3650	4634-5379	4124-4847	4647-4847	4131-4331	4943-6105	4777-5204		
Torque Converter Max. Ratio at Stall	2.20	2.10		2.20	2.20	2.20	2.20	2.03	2.03		
Converter Diameter	12.5"	12.20"		11.75"	12.5 / 11.75	11.75"	12.5"	13.038"	13.038"		

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: GMC   Year: 1974

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	140 L-4	140 L-4	250 L-6	350 V-8	350 V-8	350 V-8	350 V-8	350 V-8	350 V-8	350 V-8	350 V-8
Carburetor	1	2	1	2	2	2				4	4
Compression Ratio	8.0	8.0	8.25	8.5	8.5	7.6	7.6	8.5	8.5	8.5	8.5
BHP	75	85	100	145	150	155	170	175	180	185	195
RPM	4400	4400	3600	3800	3600	4000	4000	3800	3800	4000	4400
Torque	115	122	175	250	270	275	280 / 290	260	275	270	275
RPM	2400	2400	1800	2200	2000	2400	2000 / 2400	2000	2800	2400	2800
Transmission Trade Name	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic
Rear Axle Ratios	2.53 2.92	2.92 3.36	3.08	2.73 3.42	2.73 3.08	3.08 2.73 2.93	3.08 2.73 / 2.93	2.73	2.73 3.23 3.08	3.08 3.42	3.36
Gear Ratio: 1	2.52	2.52	2.52 / 2.48	2.52	2.48	2.48	2.48	2.48		2.52	2.52
Gear Ratio: 2	1.52	1.52	1.52 / 1.48	1.52	1.48	1.48	1.48	1.48		1.52	1.52
Gear Ratio: 3	1.00	1.00	1.00 / 1.00	1.00	1.00	1.00	1.00	1.00		1.00	1.00
Gear Ratio: Reverse	1.93	1.93	1.94 / 2.08	1.94	2.08	2.08	2.08	2.08		1.94	1.94
Weight Range of Vehicles	2446-2787	2471-3012	3413-3613 / 3390-3590	3554-3754	3449-4701	3505-4650	3555-4207	3499-4246	3524-4715	3412-3874	3390-3590
Torque Converter Max. Ratio at Stall	2.00	2.00	2.00 / 2.20	2.00	2.20	2.20	2.20	2.20	?	2.00	2.00
Converter Diameter	10.00"	10.00"	11.75 / 12.5	11.75"	12.20"	12.5"	12.5"	12.20"	?	11.75"	11.75"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: GMC **Year:** 1974 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	350 V-8	350 V-8	350 V-8	350 V-8	400 V-8	400 V-8	400 V-8	400 V-8	454 V-8	454 V-8	455
Carburetor	?	?	4	4	2	4	4	4	4	4	4
Compression Ratio	7.6	8.5	9.0	9.0	8.5	8.0	8.5	8.0	8.25	8.25	8.5
BHP	200	200	245	250	150	175	180	225	235	270	175
RPM	4400	4200	5200	5200	3200	3600	3800	4000	4000	4400	3400
Torque	295	300	280	285	295	315	290	330	360	380	355
RPM	2800	3200	4000	4000	2000	2000	2400	2800	2800	2800	2000
Transmission Trade Name	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic
Rear Axle Ratios	3.23 2.73	2.73	3.42 3.73	3.36	2.73 3.08 3.42	2.93 3.08 2.73	3.08 3.42	2.93 3.08	2.73 3.42	3.36	3.08
Gear Ratio: 1	2.48		2.52	2.52	2.52	2.48	2.52	2.48	2.52	2.52	2.48
Gear Ratio: 2	1.48		1.52	1.52	1.52	1.48	1.52	1.48	1.52	1.52	1.48
Gear Ratio: 3	1.00		1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.08		1.94	1.94	1.93	2.08	1.93	2.08	1.93	1.93	2.08
Weight Range of Vehicles		3898-4290	3604-3804	3440-3640	4427-4627	2711-5289	3683-5144	3762-4431	3783-5244	3490-3690	4351-4551
Torque Converter Max. Ratio at Stall	2.20	?	2.00	2.00	2.00	2.20	2.00	2.20	2.00	2.00	2.20
Converter Diameter	12.5"	?	11.75"	11.75"	11.75"	12.20"	11.75"	12.5"	11.75"	11.75"	12.20"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: GMC

Year: 1974 (Cont.)

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	455	455	455	455	455	455	455	455	455	472	500
Carburetor	4	4	4	4	4	4	4	4	4	4	4
Compression Ratio	8.5	8.5	8.5	8.0	8.5	8.5	8.5	8.0	8.4	8.25	8.25
BHP	190	210	210	215	230	230	245	250	290	205	210
RPM	3600	3600	3600	3600	3800	3800	4000	4000	4000	3600	3600
Torque	320	335	350	355	355	370	360	380	395	365	380
RPM	2000	2200	2400	2400	2200	2800	2400	2800	3200	2000	2000
Transmission Trade Name	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic
Rear Axle Ratios	2.73	2.93 3.23	2.93 2.73 3.08	2.93	2.93 3.23	2.73	2.73 2.93 3.23	2.73 2.93 3.08	3.08	2.93 3.15	3.07 2.73
Gear Ratio: 1	2.48	2.48		2.48	2.48		2.48	2.48	2.48	2.48	2.48
Gear Ratio: 2	1.48	1.48		1.48	1.48		1.48	1.48	1.48	1.48	1.48
Gear Ratio: 3	1.00	1.00		1.00	1.00		1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	2.08	2.08		2.08	2.08		2.08	2.08	2.08	2.09	2.09
Weight Range of Vehicles	4290-4470	4705-5390	4615-5436	4655-4855	4747-5408	4838-5038	4747-5408	3865-4631	3911-4111	5043-6232	5105-5309
Torque Converter Max. Ratio at Stall	2.20	2.20	?	2.20	2.20	?	2.20	2.20	2.20	2.20	2.20
Converter Diameter	12.20"	12.20"	?	12.50"	12.20"	?	12.20"	12.50"	12.50"	13.038"	13.038"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: GMC **Year:** 1975

Power Team Number	1	2	3	4	5	6	7	8	9	10	11
Engine Displacement and Type	140 L-4	140 L-4	231 V-6	250 L-6	260 V-8	262 V-8	350 V-8	350 V-8	350 V-8	350 V-8	350 V-8
Carburetor	1	2	2	1	2	2	2	4	4	4	4
Compression Ratio	8.0	8.0	8.0	8.25	8.0 8.5	8.5	8.5	8.5 | 7.6	8.5 | 8.0	8.5	7.6
BHP	78	87	110	105	110	110	145	155 | 155	165	170	175
RPM	4200	4400	4000	3800	3400	3600	3800	3800 | 4000	3800	3800	4000
Torque	120	122	175	185	205	200	250	250 | 280	255 | 260	275	280
RPM	2000	2800	2000	1200	1600	2000	2200	2400 | 2000	2400 | 2200	2400	2000
Transmission Trade Name	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic
Rear Axle Ratios	2.92	2.92 3.42	2.56 2.73	2.73 3.08	2.73 2.56	2.73 2.56	2.73 2.56	3.08 2.56 | 2.56	3.36 | 3.08 2.73	2.73	2.56 3.08
Gear Ratio: 1	2.52	2.52	2.52	2.52	2.52	2.52	2.52	2.52	2.48	2.48	2.48
Gear Ratio: 2	1.52	1.52	1.52	1.52	1.52	1.52	1.52	1.52	1.48	1.48	1.48
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Gear Ratio: Reverse	1.94	1.94	1.93	1.93	1.93	1.93	1.93	1.93	2.08	2.08	2.08
Weight Range of Vehicles	2495-2875	2494-2953	3007-3959	3416-3973	3445-3846	3043-3728	3445-4520	3496-4045	3529 3730 | 3445 4712	3845-4744	3596-4145
Torque Converter Max. Ratio at Stall	2.60	2.00	2.25	2.00		2.00	2.00	2.00	2.00		2.00
Converter Diameter	10.00"	11.75"	12.5"	11.75"		11.75"	11.75"	11.75"	12.20"		12.20"

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

POWER TEAMS AS A FUNCTION OF ENGINE DISPLACEMENT

Make: GMC

Year: 1975 (Cont.)

Power Team Number	12	13	14	15	16	17	18	19	20	21	22
Engine Displacement and Type	350 V-8	350 V-8	400 V-8	400 V-8	400 V-8	454 V-8	455 V-8	455 V-8	455 V-8	500 V-8	
Carburetor	F.I.	4	2	4	4	4	4	4	4	4	
Compression Ratio	8.1	9.0	7.6	8.5	7.6	8.15	8.5	7.6	7.9	8.5	
BHP	180	205	170	175	185	215	190	200	205	190	
RPM	4400	4800	4000	3600	3600	4000	3400	3500	3800	3600	
Torque	275	255	305	305	310	350	350	330	345	360	
RPM	2000	3600	2000	2000	1600	2400	2000	2000	2000	2000	
Transmission Trade Name	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	Turbo Hydramatic	
Rear Axle Ratios	2.56	3.36	2.56	2.73 2.56	2.56	2.73 3.08	2.56 2.73	2.56	2.93 2.73	2.73	
Gear Ratio: 1	2.48	2.48	2.48	2.52	2.48	2.48	2.48	2.48	2.48	2.48	
Gear Ratio: 2	1.48	1.48	1.48	1.52	1.48	1.48	1.48	1.48	1.48	1.48	
Gear Ratio: 3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	
Gear Ratio: Reverse	2.08	2.08	2.08	1.93	2.08	2.08	2.08	2.08	2.08	2.09	
Weight Range of Vehicles	4341-4541	3529-3730	4190-4843	3496-4045	3696-5311	4149-4718	4045-5411	3696-5311	4612-5083	5149-6066	
Torque Converter Max. Ratio at Stall				2.00	2.00	2.10	2.10	2.00	2.20	2.00	
Converter Diameter				11.75"	12.20"	12.20"	12.20"	12.20"	12.20"	13.038"	

Vehicle Class

Subcompact ≤ 100"

Compact 101 – 111"

Intermediate 112 – 119"

Full Size ≥ 120"

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