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ADVANCED CONTINUOUSLY VARIABLE TRANSMISSIONS FOR ELECTRIC AND HYBRID VEHICLES

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ABSTRACT

A brief survey of past and present continuously variable transmissions (CVT) which are potentially suitable for application with electric and hybrid vehicles is presented. Discussion of general transmission requirements and benefits attainable with a CVT for electric vehicle use is given. The arrangement and function of several specific CVT concepts are cited along with their current developmental status. Lastly, the results of preliminary design studies conducted under a NASA contract for DOE on four CVT concepts for use in advanced electric vehicles are reviewed.

HISTORICAL BACKGROUND

The advantages of an infinitely variable transmission for gasoline-powered automobiles over sliding-gear types was recognized by inventors at a very early date. As noted by P. M. Heldt in a comprehensive review of the evolution of the automatic transmission (ref. 1), the first type of automobile transmission which provided "infinite" transmission ratio selection was the friction-disc drive. Figure 1 illustrates the Cartercar, one of the earliest cars equipped with a traction CVT. The transmission consisted of an aluminum disc driven by the engine's crankshaft. This disc contacted a fiber-covered, roller follower mounted on a transverse counter shaft. To vary speed ratio, the roller follower would transverse the face of the disc; turntable fashion. Neutral was achieved when the roller follower was positioned in the center of the drive disc. By moving the follower past center, the follower would reverse its direction of rotation and allowed the vehicle to back up. Despite its catch slogan, "No clutch to slip - no gears to strip . . . . a thousand silent speeds and only one control lever, that is a Cartercar", the Cartercar Company experienced limited commercial success. From 1909 until 1912, Sears marketed a two-cylinder, 14-horsepower "Motor Buggy", equipped with a traction transmission of a design similar to that of the Cartercar (ref. 2). However, by about 1915, cars equipped with traction drives had virtually disappeared (ref. 2).

Despite these unsuccessful earlier attempts, the goal of designing a transmission which could operate the engine at its most efficient operating condition at any vehicle speed was not lost. In the late 1920's and early

1930's General Motors Research Laboratories did extensive testing on a double cavity, Toric traction CVT (ref. 3). This transmission was similar in principle to the toroidal drive patented by Charles Hunt back in 1877. Seventeen test vehicles equipped with this transmission accumulated over 300,000 miles of road testing (ref. 3). A 20 percent improvement in highway fuel mileage was reported. For a number of economic and engineering reasons, not the least of which was the risk of introducing a radically new transmission concept into the market place during the depression era of the early 1930's, the Toric transmission was never put into production.

Through the years there have been occasional attempts, both here and in Europe, to commercially introduce CVT's into passenger cars. Several of these efforts proved technically feasible, but they were never really serious contenders to replace the automatic, torque-convertor, gear-shift transmission which was widely adopted by the automotive industry since the early 1940's. Up until the 1970's, the primary emphasis for automatic transmissions was on transmission shift quality and cost while efficiency was basically a secondary consideration. In recent years the emphasis has changed to improving passenger car fuel economy by improving drive train efficiency. The shortage of petroleum has also stimulated research on alternate types of automotive powerplants, such as electric and gas turbine. These two factors, in turn, have triggered renewed CVT activity. The CVT's "infinite" number of shift points offers the engine designer the greatest possible latitude in optimizing his drive train, no matter what the powerplant.

It is the intent of this paper to highlight, with specific examples, some of the more recent automotive CVT developmental efforts, particularly those that may be suitable for electric and hybrid vehicle applications. Readers with a greater interest in the details of the specific CVT's mentioned or wanting to learn more about the historical development of these types of transmissions are advised to consult the references listed at the end of the paper.

ELECTRIC VEHICLE TRANSMISSION REQUIREMENTS

Figure 2 shows a schematic of a drive train for a parallel hybrid electric vehicle. In this arrangement either the heat engine or electric motor, or some combination thereof, powers the vehicle through the transmission. The role of the transmission is to match the torque and speed characteristics of the power plant to that of the vehicle in such a way as to maximize the efficiency of the overall system.

The operating efficiency of most heat engines, i.e., spark-ignition, diesel or gas turbine, is sufficiently sensitive to engine speed as to significantly benefit from the use of a CVT. However, the extent that CVT's benefit fuel economy relative to advanced automatic transmissions is often a point of controversy (for example see ref. 4). Reference 3 quotes a 25 to 35 percent fuel mileage improvement with the use of an "ideal" transmission for a S.I. engine. A test vehicle equipped with an Orshansky hydromechanical CVT demonstrated a 23 percent fuel economy improvement over the Federal composite driving cycle relative to the same vehicle with the production three-speed automatic transmission (ref. 5). These tests were conducted on a chassis dynamometer, and no adjustments for exhaust emissions were made (ref. 5). The above suggests that a CVT would be beneficial to a hybrid vehicle drive system because of the heat engine, but what about an electric vehicle without a heat engine?

Figure 3 shows the efficiency/speed characteristics of a typical D.C. shunt electric motor from a study performed in reference 6. It is clear from this figure that the D.C. motor is most efficient between 40 to 70 percent of rated power with a small drop of 5 to 10 efficiency points at power levels to 200 percent of rated power. It is also apparent that motor efficiency is relatively speed insensitive, varying at most 5 or 6 points from the highest efficiency at the base motor speed to the lowest efficiency at maximum speed. In view of this relatively small efficiency spread, other considerations such as transmission efficiency, ratio range, cost, or smoothness of operation would be needed to justify the use of a CVT over an automatically-shifted gear transmission for a vehicle powered only by a D.C. motor of today's design.

Figure 4 shows a typical D.C. motor speed map for a vehicle equipped with a 4-speed gearbox from reference 6. This figure illustrates that motor speed fluctuates between maximum and base speed or, in other words, between minimum and maximum motor efficiency with a 4-speed transmission. On the other hand, utilizing a wide ratio range CVT, motor speed can be maintained at base speed, except for start-up, so that motor efficiency can be maximized at all vehicle speeds.

In the case of an A.C. induction electric motor, the efficiency is fairly flat-rated with power, as shown in figure 5 from reference 6. Since A.C. machines do not have brushes, they can be operated at higher speeds than D.C. motors. This results in a significantly smaller frame size for the A.C. motor (ref. 6). For the A.C. motor of figure 5, the rated speed is 29 600 rpm. However, to take advantage of this small size, a means must be found to regulate speed. One approach would be to use a CVT. A second approach would be to use variable frequency control, as illustrated in figure 6. The penalty for frequency control is the attendant complexity and cost of the inverter. Figure 6, from reference 6, shows that relatively high motor efficiency can be maintained at low power levels, provided that motor speed can be kept at sufficiently high levels. Ideally, this could be accomplished with a CVT but a 2- or 3-speed gear box would probably suffice.

TRANSMISSION TYPES

Table I lists a number of transmission types that can provide variable speed ratio. The two multi-speed gear transmissions (one using a torque convertor and the other using a slipping clutch) would represent, respectively, a conventional automotive automatic transmission and a manually-shifted one. Two- or three-speed automatic transmissions are often adapted from small passenger cars to electric vehicles (ref. 7). While these transmissions provide sufficient "down-shifting" for passing and hill climbing, their average efficiencies, which typically range from 75 to 85 percent (refs. 8 to 10), are decidedly inferior to the 92 to 97 percent efficiencies of manual transmissions (ref. 11) and to the efficiencies of many CVT's. Also being investigated are microprocessor-shifted 2- or 3-speed gearboxes for electrical vehicle use. This type of transmission would, undoubtedly, provide excellent efficiency, but considerable attention to shift quality would be needed.

The remaining transmissions listed in table I are either continuously or infinitely variable, depending on whether the output shaft speed can be controlled down to zero rpm. Examples of each of these transmission types will be given in the next section, with the exception of the hydroviscous or wet slipping clutch drive, which is basically too inefficient to be suitable for automotive drive systems. Also, only passing reference will be made to chain-type variable-speed drives because of their basically industrial nature, although there is one notable attempt, that of GKN in England, to adapt them to passenger car use.

RECENT CVT DEVELOPMENTS

For the past 100 years there have been numerous attempts (far too many to discuss) to develop commercially acceptable automotive CVT's (e.g., see ref. 1 for early developments). With the exception of the DAF Variomatic twin, rubber V-belt drive on the Volvo-343 sedan, the author knows of no other automobile manufacturer producing CVT-equipped cars. However, recent CVT developments, particularly with metal V-belt and traction types, have received attention from several major car manufacturers. Both Fiat and British Leyland have done considerable testing with CVT prototypes whose arrangements will be described later.

It is generally accepted that flywheel energy storage can improve the performance of electric and hybrid vehicles (refs. 12 to 14). However, one of the limiting factors in applying flywheel energy storage to electric vehicles has been the necessity of an efficient, reliable and inexpensive CVT. Figure 7 illustrates the approach adopted by Garrett AiResearch under an ERDA program for their flywheel-equipped ETV-2 vehicle (ref. 12). The flywheel, together with two electrical machines, are mechanically coupled to a planetary gear differential. One of these two electrical machines is coupled to the output gear while both are electrically coupled together so that one can serve as a motor, the other as a generator, or vice versa, depending on the direction of the power flow. This combination of differential gearing in parallel with an electric motor and generator constitutes an electro-mechanical CVT. The objective is to split up the flywheel power so that most of it will be flowing through the highly efficient mechanical (gear) branch while minimizing the power handled electrically through the motor/generator circuit. This concept of power-splitting is quite commonly used to improve overall system efficiency of a CVT. The drawback is that the overall ratio range of the total transmission is always less than that of the variable speed drive element by itself. The greater the degree of power splitting, the smaller the total transmission ratio range.

The Orshansky hydromechanical CVT is another example of the use of a power-splitting gear differential in combination with a variable speed element which is, in this case, a hydrostatic (hydraulic pump/motor) drive (ref. 5). A photo of the first generation prototype appears in figure 8. This CVT has both low and high modes of operation initiated by the synchronous activation of low- and high-range clutches. As previously mentioned, this transmission demonstrated a fuel economy improvement of approximately 23 percent relative to the same model car with the stock 3 speed automatic transmission (ref. 5). A second generation prototype design with improved components was generated for a front-wheel-drive car, but funding was interrupted before this system was developed. However, the principal criticism of the first generation transmission for vehicle applications has been one of hydraulic pump/motor noise.

Figure 9 shows a cross-section of a hydrokinetic CVT. This CVT is a power-splitting, variable stator torque convertor type being jointly developed by Garrett Turbine Engine Co. and Ford Motor Co. for a single-shaft, gas turbine-powered automobile (ref. 15). This program is being funded under DOE's Advanced Gas Turbine-Powered Development Project and managed by the NASA Lewis Research Center. This CVT combines a variable stator torque convertor with a power-splitting planetary gear set to reduce the power flowing through the relatively inefficient torque convertor. A modified, Ford four-speed (integral overdrive) automatic gearbox is used to expand the overall transmission ratio. This program is in its early stages and a complete transmission has not yet been fabricated. Early developmental tests with the variable stator torque convertor alone has established its operational feasibility.

Under a parallel DOE program, Chrysler and Williams Research Corporations jointly propose to investigate the use of a compression-type metal V-belt CVT in the development of their single shaft, automotive gas turbine engine passenger vehicle (fig. 10). The driver pulley of a compression V-belt transmission "pushes" rather than "pulls" the driven pulley using a stack of V-shaped metal blocks. The principles of operation will be discussed in more detail later. The compression V-belt drive to be investigated is produced by van Doorne Transmissie B.V. of the Netherlands (ref. 16). Recently, Borg-Warner and Fiat have joined van Doorne to form a consortium to further the development of this concept and to prepare it for possible production in cars or light trucks. Several test vehicles, including a fleet of Fiat test cars, have been retrofitted with the van Doorne transmission and have apparently shown acceptable operation. The mechanical efficiency of the van Doorne Transmatic CVT is often reported to be somewhat better (ranging from 88 to 92 percent) than that of a conventional automatic transmission (ref. 16). Since the compression belt/pulley drive itself probably has efficiencies in the mid- to upper-90's, the power losses associated with the bearings, pump and other auxiliary components is an area for future improvement.

For automotive applications, the ratio range requirement is relatively broad. In one configuration being studied by Chrysler/Williams, the compression V-belt is connected to a "recirculating" planetary differential in order to expand the overall transmission's ratio range from the V-belt's ratio range which is limited to about 4- or 5-to-1. This arrangement is in contrast to that of "power-splitting", where the transmission ratio range is reduced from that of the variable-speed element, as discussed earlier. Here the variable-speed element's ratio range is increased at the expense of "recirculating" part of the power back to the input. The result is that the variable-speed element (in this case, the V-belt drive) carries more power (but no extra torque) than that either entering or leaving the transmission. The increase in the transmitted power through the variable-speed element increases its power loss. This is opposite to the power-splitting arrangement which reduces the variable-speed element's power loss. Although there is a penalty in terms of efficiency, particularly at high transmission reduction ratios, the "recirculating" power arrangement often eliminates extra gearing and clutches to achieve the required transmission ratio range.

A schematic of the flywheel-electric DOE/USPS postal vehicle drivetrain appears in figure 11 (ref. 17). A 36 000-rpm flywheel is working in tandem with an electric motor. The flywheel drives, through a reduction gear, a 7-to-1 ratio range, rubber V-belt drive to power the vehicle up to its 33 mph cruise speed. At speeds of 7 mph or less, the flywheel is decoupled from the vehicle by the fluid coupling, and the electric motor does all the driving. For small, relatively lower power vehicles, the rubber V-belt drive offers the combination of relatively low cost and efficient operation. However, for moderate- to high-power systems, the reliability and durability of the rubber V-belt drive has been questionable. DAF and Volvo got around this problem by combining two V-belt drives in parallel in their production cars at the expense of added cost and complexity. Improvements in the technology of rubber V-belt construction are continually being made, but current power limits, basically defined by the temperature limits of rubber, confine their current use to smaller systems such as small cars and snowmobiles.

TRACTION CVT's

Traction CVT's have been in existence for at least 100 years, the bulk of them performing speed matching for light duty industrial applications. Geometries of typical industrial traction drives appear in figure 12 (ref. 18). Many of these are commercially available and few have greater than 10 kW power capacity. The "wheel and single disk" drive typifies the Cartercar and Sears Motor Buggy transmissions. With the possible exception of the "toroidal" geometry, the commercial CVT's shown in figure 12 are generally too large, complex, or have insufficient power capacity to be suitable for advanced automotive applications.

In a traction drive, torque transfer is mainly accomplished by shear forces generated between smooth driving and driven rollers across an extremely thin lubricant film. Under high pressures and shear rates which exist in a typical traction contact, the lubricant's viscosity increases dramatically and the lubricant is thought to transform into a plastic-like material. This thin, stiff plastic film can tolerate relatively high amounts of torque transfer without rupturing while protecting the roller surfaces from appreciable metal-to-metal contact or wear. The failure mechanism of a well-designed traction drive is generally one of rolling-contact fatigue which is analagous to pitting failure in gears and spalling failure in ball and roller bearings. The torque capacity of a given traction drive is a function of its fatigue life and its construction; that is, its contact geometry and number of contacts working in parallel to share the load. With today's metallurgically-clean bearing steels that offer superior fatigue resistance and improved synthetic traction lubricants, whose higher coefficients of traction allow a reduction in contact pressure, modern traction drives have considerable more power capacity than their earlier counterparts (ref. 19). Coupling these advancements to the greater emphasis on improving fuel mileage and the down-sizing of both cars and engines, it is not surprising that there has been a resurgence in automotive CVT research and development (ref. 20).

One such effort is that being conducted by BL Technology, Ltd., formerly British Leyland, on a Perbury, double-cavity toroidal drive. This CVT appears in figure 13 (ref. 21). This concept is rather old (first patented in 1877). It is also rather well-explored, having been investigated by the G.M. Research Laboratory in the early 30's and late 50's, demonstrated in a 1934 Austin-Hayes, later in a 1957 Hillman-Minx and also in a 1973 Ford Pinto, but with offset rollers. The B.L./Perbury transmission was installed in a medium size test car having a 4-cylinder, 60 kW engine. The test car showed fuel mileage improvements of 15 to 20 percent for an average mix of European driving (ref. 21). Also, acceleration times were comparable to a manual transmission car having 10 percent higher power/weight ratio and driven by a skilled driver (ref. 21).

The double-toroidal drive units have a total of six tiltable transfer rollers between input and output toroids. A hydraulically controlled linkage system can tilt these rollers from one extreme position to another. Input and output toroid rolling radii and speed ratio can thus be changed from 1:2.7 to 2.0:1 or a total ratio range of about 5.4. By combining this toroidal drive with a two-range, output planetary gear system, the overall transmission ratio range is greatly expanded. In the low range, which covers reverse-neutral and forward speeds as low as about 8 or as high as about 69 km/h, depending on throttle level, the toroidal drive forms a power "recirculating" loop with the planetary differential. The toroidal unit always carries power that is greater than that produced by the engine. In the high range, actuated by a synchronous shift, the planetary differential is locked out and the toroidal unit is in "direct drive" with the output shaft. Thus, the drive carries only engine power. Rig and vehicle road tests with the B.L./Perbury CVT have been encouraging (ref. 21), with some additional control system work indicated for improved driveability. However, the future production picture for this transmission is not clear.

A promising traction CVT that is of a rather new vintage is the nutating drive being developed by Vadetec Corp., as shown in figure 14 (ref. 22). A double-conical-roller assembly, complete with an automatic loading mechanism, is mounted at an angle in a drive cylinder that is driven by the input shaft. As the input shaft rotates, the double-cones perform a nutating motion and at the same time are forced to rotate about their own axis as they make drive contact with a pair of movable control rings. These rings are grounded to the housing but can be axially moved together or apart. A gear pinion attached to the end of the cone shaft orbits the output shaft axis at input shaft speed while spinning about its own axis, due to cone rotation, in the opposite direction. By varying the axial position of the control rings, the rolling radius of the cones can be synchronously changed. This, in turn, causes a change in rotational speed of the cone shaft pinion but does not affect the pinion's rotating speed which occurs at input shaft frequency.

Since the cone shaft pinion is in driving engagement with the output shaft through one of several possible interchangeable gear arrangements, the variation in pinion rotational speed causes a corresponding variation in output shaft speed. By changing the output gearing, the pinion speed can either vectorally subtract from pinion/input shaft orbital speed allowing the output shaft speed to reach zero, or vectorally add to input orbital speed. The latter arrangement results in a transmission with only overdrive capability. Figure 14 illustrates an output gear of the first kind where pinion rotational and orbiting speeds are subtractive. For the Vadetec CVT, the amount of power flowing directly through the gearing and that carried by the ring-cone traction contact is a function of transmission ratio and ratio range. For output shaft speeds near input shaft speeds, the cones are turning slowly, carrying relatively little traction power. The transmission's efficiency is basically that of the gearing and bearings; that is, the low-to-mid-90's. On the other hand, at relatively low output shaft speeds, the cones are rotating quickly, carrying more traction power. For wide ratio range transmissions, particularly those that are required to operate both at forward and reverse speeds, the power caused by the traction contacts can equal or exceed the output power. When this occurs, the drive functions as a differential, like some of the previously described power "recirculating" transmissions, and efficiency is reduced.

According to reference 20, prototype tests by a Vadetec licensee of a 200-hp transmission installed in an off-the-road vehicle have been "very successful". The relatively low parts count and low operating contact stresses of the Vadetec design are definite advantages. However, high-speed, heavy-duty cone support bearings are required to react the sometimes large moment loads acting on the cones. These moment loads are generated by the frequently large, unbalanced, ring-cone contact loads that occur at each side of the drive. At higher input shaft speeds, centrifugal force loads from the cones tend to reduce this bearing load.

A planetary, cone roller type CVT under development by the Fafnir Bearing Division of Textron appears in figure 15 (ref. 23). This transmission is directed toward the mobile equipment market, particularly as a replacement to hydrostatic transmission in garden and light duty tractors up to about 37 kW (50 hp). A set of double-sided, conical traction rollers are trapped between a pair of inner races and a pair of outer races. The conical rollers are mounted in a carrier which drives the planet gears in the output gear differential. The gear differential serves the purpose of expanding the limited ratio range of the traction drive to cover output speeds from forward to reverse, if desired. This is another example of a power-recirculating transmission. The traction inner races are splined to the sun gear shaft of the planetary which is, in turn, keyed to the input shaft. To change ratio, the outer traction race halves which are grounded to the housing can be either manually or hydraulically, in a later version, pushed together or spread apart. This causes the cone rollers' rolling radii at the outer-race contact to increase or decrease while simultaneously causing the opposite to occur at the inner-race contact. The change in rolling radii causes a corresponding change in the orbit speed of the cones and, thus, output-shaft speed. Due to the inner-race load-springs, the transmission automatically "down-shifts" with increased load. The current design was specifically intended to be one of low cost with maximum flexibility so a single range output planetary was selected. Consequently, a high degree of power recirculation exists, limiting efficiency to about 85 percent maximum for the current design (ref. 20). A more efficient version would be needed for automotive applications.

CVT DESIGN STUDIES

DESIGN REQUIREMENTS - As part of the DOE Electric and Hybrid Vehicle Program, 9-month preliminary design studies were performed on four CVT concepts under contract to NASA Lewis Research Center by the contractors listed in table II. The CVT concepts were designed for a 1700 kg (3750 lb) vehicle equipped with a 14 000 to 28 000 rpm energy storage flywheel. The goal of NASA Lewis's CVT program is to provide enabling CVT technology to permit commercialization in the 1985-1990 time frame. It was expected that these four preliminary design studies would yield two or more promising CVT concepts which could be selected for detailed design, fabrication and testing. Another expected result of these studies would be the identification of new technology advancements needed to make each of the CVT's viable for electric vehicles.

The basic design requirements and general drive train arrangement used in these studies are shown in figure 16. These design requirements were not selected for any specific vehicle but were arbitrarily chosen to evaluate all CVT concepts on a common basis. A flywheel electric vehicle system is a logical choice for a CVT study not only from the standpoint that some form of CVT is an absolute necessity but, moreover, the selected CVT must have a particularly broad ratio range. This is because the flywheel speed may be at its maximum value when the vehicle's speed is at its lowest, or vice versa. In this particular study the speed ratio of the CVT was to be continuously controllable from flywheel speeds of 14 000 to 28 000 rpm to transmission output speeds of zero to 5000 rpm. However, as an option, the minimum CVT output speed could be 850 rpm and a slipping clutch element be used on the output side of the CVT to drop the speed from 850 rpm to zero. Using the 850 rpm minimum output speed, the CVT's ratio range is approximately 11.8 to 1. In addition to the requirements listed in figure 16, the CVT must be capable of bi-directional power flow for flywheel regeneration during braking; capable of handling normal driving shock loads; and have some clutching means to disengage and re-engage the flywheel from the rest of the drive train. The CVT did not have to have reverse rotation capability since the electric motor could be used to back-up the vehicle. The clutch between the motor and the differential permits flywheel charging when the vehicle is stopped.

The CVT designs were to be evaluated on the basis of the following criteria in order of overall importance:

1. Efficiency - particularly at operating conditions where the CVT spends most of its time.

2. Cost - on the basis of 100 000 units per year.

3. Size and weight - comparable to or less than present automatic transmissions.

4. High reliability - 2600 hours with 90-percent reliability at the weighted average conditions noted in figure 16.

5. Noise - minimize its generation and maximize its containment in the housing.

6. Controls - provide driver "feel" as close to current automobile transmissions as possible.

7. Maintainability - equal or better than current automobile transmissions.

DESCRIPTION OF CVT CONCEPTS - Preliminary design layouts of the four CVT studies listed in table II appear in figures 17 to 21. A general description of each CVT concept will be given in this section. For those interested, a more thorough discussion can be found in reference 24 and specific details in references 25 to 28.

Steel V-belt CVT - A preliminary layout of Batelle's steel V-belt CVT (ref. 25) appears in figure 17. This transmission uses two V-belts in series to achieve the minimum required ratio range of 11.8 to 1. An output modulating clutch is used to lower the output speed from 850 rpm down to zero and provide over-torque protection. A 2.8 to 1 spur gear set reduces the flywheel speed entering the high-speed belt. The belts are only used as reducers, varying from about 1:1 to 1:3.9 for the high-speed belt and from 1:1 to 1:3.3 for the low-speed belt. A microprocessor-controlled hydraulic system controls belt shifting and regulates the axial clamping force between pulleys to achieve the best compromise between drive efficiency and belt life.

Batelle's steel V-belt concept had received some early hardware development by Batelle Columbus Labs back in the 1960's. The construction of their belt as shown in figure 18 is similar to that of the van Doorne V-belt (ref. 16). The van Doorne V-belt was briefly discussed in a previous section. In brief, the belt is composed of a stack of solid cross-struts gathered together by a nested set of thin, steel bands. The bands lie freely on the top of these cross-struts allowing relative motion between the individual bands themselves and also between the band set and the struts. The ends of the struts contact the face of the V-shaped pulleys. This type of belt is termed a compression belt since the driver pulley pushes the driven pulley through the stack of cross-struts. The set of bands keep the struts from buckling. Consequently, the bands carry high-tensile forces and are subjected to high bending stresses as they travel around the pulleys. It is basically the bending fatigue strength of these bands that limits the torque capacity and minimum pulley diameters or, in other words, the ratio range of the drive. Also, since the bands slide over each other and over the struts as the belt moves, proper lubrication and material selection are important to prevent destructive surface galling.

Flat Rubber Belt CVT - Figure 19 illustrates the variable pulley diameter, flat rubber belt CVT concept of Kumm Industries (ref. 26). A pair of variable diameter pulleys are used in combination with differential gearing (not shown) to achieve the required speed variation. The differential gearing is used only in the "low" speed mode to attain zero output rpm while the 4-to-1 ratio range (2:1 to 1:2) of the pulleys is used in "direct" drive to achieve maximum vehicle speed in the "high" speed mode. The "low" and "high" speed modes are separated by synchronous clutching. Step-down gearing from the flywheel insures that the pulley speeds never exceed about 10 000 rpm.

The flat belt is in contact with a set of drive elements or cross-struts. The ends of these drive elements are contained in guideways or circular arcs which have been machined in a pair of inner discs and a pair of outer discs, but in opposite directions (see figure 19). The radial position of these drive elements, hence, pulley diameter, is determined by the intersection of the inner-disc guideways that are curved one way and the guideways of the outer discs that are curved in the opposite direction. As the inner and outer discs are rotated relative to one another by a hydraulic rotary actuator, the drive elements are moved radially in and out, changing drive ratio. A hydraulic control circuit is used to control the actuator which also provides sufficient belt tension to prevent slippage without overloading.

Toroidal Traction CVT - The preliminary design layout of AiResearch's toroidal traction CVT appears in figure 20 (ref. 27). The double-cavity toroidal drive, containing two power rollers per cavity, is permanently connected to differential gearing to form a single mode, power "recirculating" CVT. The differential gearing expands the 5.8 to 1 ratio range of toroidal drive section to cover the 0 to 5000 rpm output speed requirement of the study. The double toroidal drive elements are similar in many respects to the drive elements in the GM Toric drive, the Austin-Hayes transmission and the Perbury drive as discussed earlier.

The input shaft, through the input reduction gear set, drives the two outer toroids and the sun gear of the output differential. The tilting of the power rollers varies the speed of the inner toroids which is connected to the ring gear of the output differential via the transfer shaft. The power that is recirculating between the toroidal cavities and the output differential is always somewhat greater than the output power. A mechanical loading cam mechanism automatically increases the clamping force between the rollers and toroids in proportion to the transmitted torque. This insures that the traction contact will always have sufficient load to prevent slip while minimizing contact overloading under light torque conditions. The CVT ratio is controlled by a pressure-balanced hydraulic control system. This system "steers" the power rollers into a new "tilt" position when the command pressure acting on the roller's reaction piston is not exactly balanced by the traction forces acting on the roller. Commanding more torque increases hydraulic pressure, causing the rollers to upshift the transmission which, in turn, causes the vehicle to accelerate and load the transmission. This type of "torque controlled" ratio change system is ideally suited for flywheel drive systems where even small changes in speed can mean large surges of power.

Cone-Roller CVT - The mechanical arrangement of the Bales-McCoin cone-roller traction CVT appears in figure 21 (ref. 28). In this design, the variable ratio traction assembly is connected to output planetary differential through a set of bevel/helical idler gears. As in some of the prior examples, there is an output differential which expands the approximate 3.6 to 1 ratio range of the traction roller assembly to achieve output speeds from 5000 down to 850 rpm. A modulating clutch (not shown) is used to attain output speeds down to zero.

The traction assembly consists of a central traction roller surrounded by four cone rollers whose axes are inclined. By inclining the cones, their inner contact surface can be made parallel to axis of the roller. The worm-screw drive shown in figure 21 axially positions the central roller to change speed ratio. Bevel gears attached to the end of the cones drive the ring gear of the output planetary through the idler gears. The sun gear of the planetary is driven by the input shaft which is part of the traction roller ball spline. The output shaft is connected to the planet carrier.

The cones are loaded against the central roller with individual hydraulic pistons. The hydraulic pressure, hence contact load, is regulated by a novel microprocessor control system to attain the minimum load needed to prevent significant roller slip at any given operating conditions. The control system senses the instantaneous change in slip rate using speed measuring encoders attached to the roller and cone shafts and a linear transducer sensing the axial position of the roller. Based on these signal inputs, the microprocessor continuously computes whether to increase or decrease cone pressure, depending on the current slip rate. Ideally, with this type of "updating" loading control system, efficiency and fatigue life of the traction contacts can be improved. However, with the proposed design, control system reliability and response time needs to be established.

COMPARISONS OF PREDICTED PERFORMANCE - The efficiency, weight and size for each of the CVT preliminary designs were evaluated. The calculated efficiency was based on installed power losses which included bearing, gear, seal, traction contact or belt contact losses and the losses associated with the hydraulic system. The calculated efficiencies of the four CVT concepts are compared in figure 22 at an output power of 16 kW(22 hp) and speed of 3000 rpm for different flywheel speeds. This operating condition is the nominal, weighted averaged power condition used in the study to determine the reliability of the CVT. Predicted CVT efficiencies over the full range of speed and power conditions can be found in references 25 to 28. From figure 22 it is apparent that flywheel speed has little effect on efficiency. At this power level, the belt type CVT's appear to have about a 4-point efficiency advantage over the traction types. However, all predicted efficiencies are significantly higher than the nominal 75 to 85 percent values measured for current automatic transmissions (refs. 8 to 10). Predicted efficiencies at a low speed cruise condition where output power is only 5 kW (7 hp) are approximately 92, 90, 92 and 89 percent for the steel V-belt, flat belt, toroidal traction and cone-roller traction CVT's, respectively (ref. 24). Because the typical duty cycle of a car is, on the whole, quite modest, the relatively good predicted performance of these CVT's at low power levels is quite significant to drive train energy economy.

Estimated sizes and weights of the CVT concepts are given in reference 24. The calculated weights ranged from 32 kg for the cone-roller traction CVT to 70 kg for the steel V-belt drive. All of the units were expected to be as light or lighter than comparable automatic transmissions which range in weight from about 68 to 82 kg (ref. 24).

Based upon the encouraging results of this preliminary design study, a follow-up program is planned to detail design, fabricate and test at least two of the CVT concepts.

CONCLUDING REMARKS

CVT technology has shown slow but steady progress through the years. Increasing concern over our diminishing petroleum resources, coupled with recent technological advancements in materials and design techniques, has hastened the likelihood that a commercially viable, automotive CVT will be produced. Some of the CVT designs reviewed in this paper are already nearing the point of commercial acceptance and undoubtedly there will be improvements with time. The CVT field also offers plenty of opportunity for innovation.

In the case of electric and hybrid vehicles, CVT's are beneficial, provided they are efficient, which most are, possessing good reliability characteristics and are not too costly. In the case of flywheel-equipped electric vehicles, some form of CVT is a necessity.

In assessing the general level of CVT technology, it appears that the basic technology is in hand to make most CVT's functional. However, power limits, cost and reliability factors are largely unknown. Furthermore, identification of critical technology elements where improvements can be obtained is also warranted. In reviewing the various types of CVT's for electric and hybrid vehicles, belt and traction designs look like particularly promising candidates for advanced CVT applications.

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TABLE I - VARIABLE RATIO TRANSMISSION TYPES

Stepped, multispeed gear

- Torque converter/fluid coupling
- Slipping clutch

Electric/electro-mechanical

Hydrostatic/hydromechanical

Hydrokinetic

Hydroviscous

Belt and chain

- Metal or rubber V-belt
- Flat
- Pin or link chain

Traction

- Toroidal
- Cone/roller
- Nutating cone/ring
- Planetary cone roller

TABLE II - CVT CONCEPTS AND CONTRACTORS

FOR PRELIMINARY DESIGN STUDY

CVT CONCEPT	CONTRACTOR	REFERENCE
Variable Pulley, Steel V-belt	Battelle Columbus Labs Columbus, Ohio	25
Variable Pulley, Flat Rubber Belt	Kumm Industries, Inc. Tempe, Arizona	26
Toroidal Traction	Garrett Corp., AiResearch Mfg. Co. of California Torrance, California	27
Cone-roller Traction	Bales-McCoin Tractionmatic El Paso, Texas	28

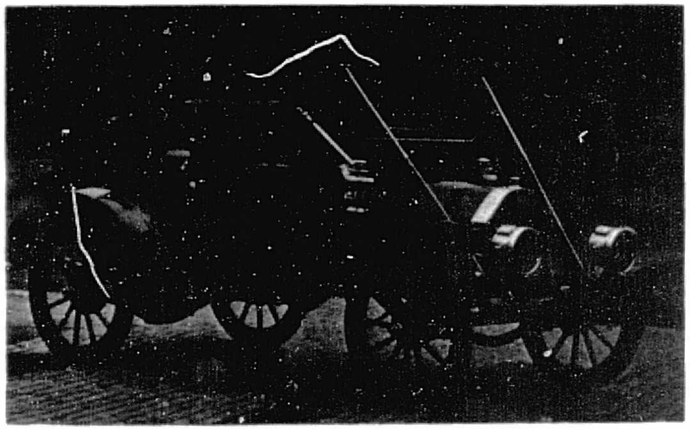

Figure 1. - 1909 Carter Car equipped with a traction CVT. (Courtesy of the Henry Ford Museum, Dearborn, MI.)

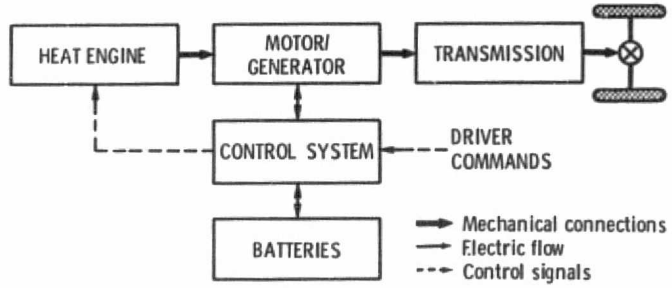

Figure 2. - Parallel hybrid electric vehicle drive train.

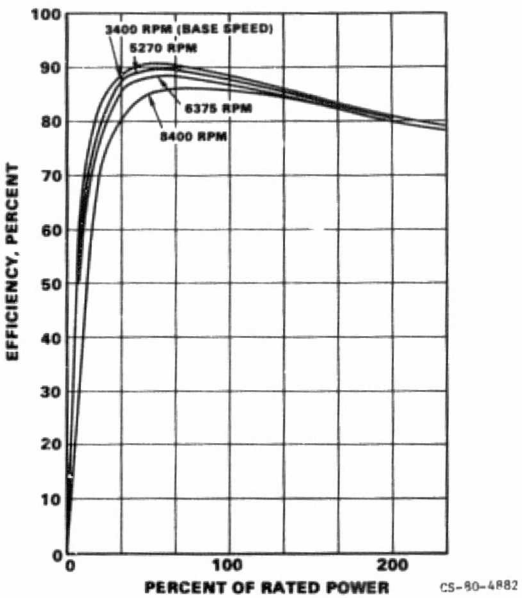

Figure 3. - Performance characteristics of a D.C. shunt electric motor (ref. 6).

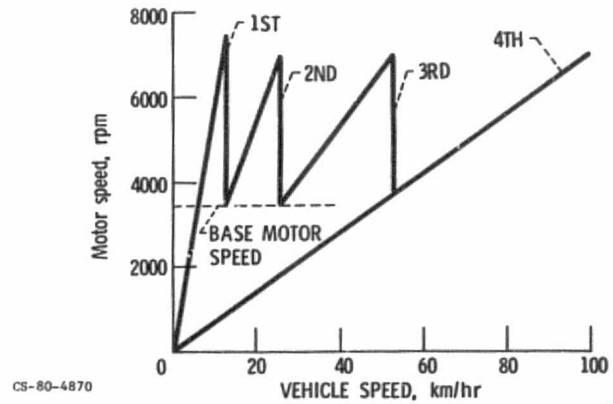

Figure 4. - Typical D.C. motor speed characteristics as a function of vehicle speed for a 4-speed gearbox (ref. 6).

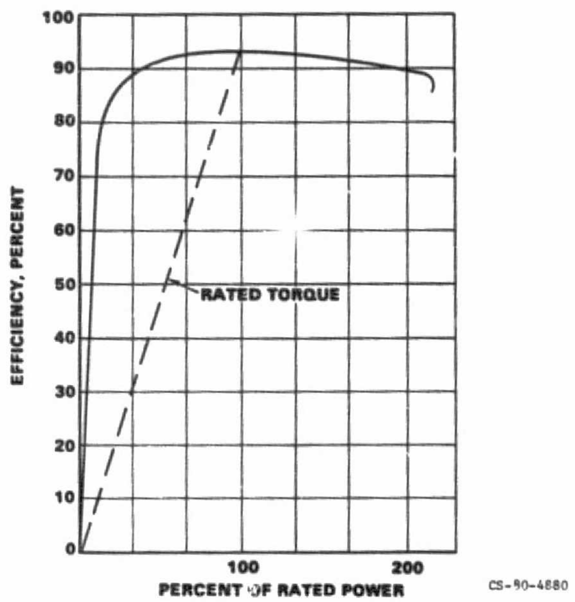

Figure 5. - Performance characteristics of a A. C. induction electric motor (ref. 6).

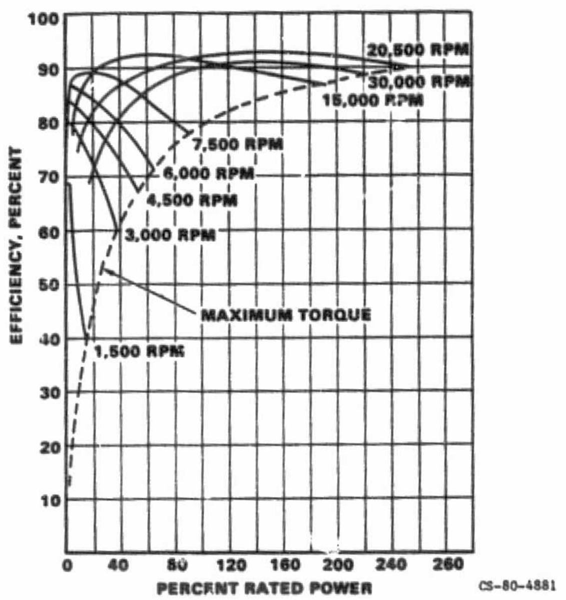

Figure 6. - Performance characteristics of a A. C. induction electric motor using variable frequency control (ref. 6).

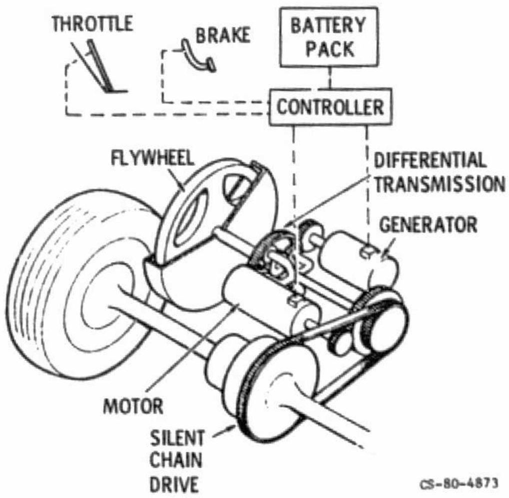

Figure 7. - Electro-mechanical CVT for DOE/AiResearch flywheel ETV-2 vehicle (ref. 12).

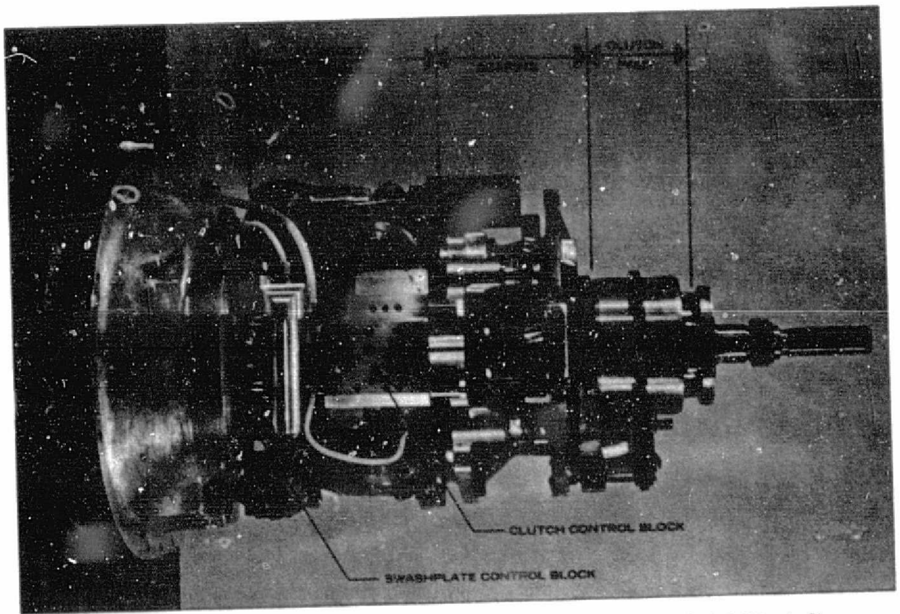

Figure 8. - Orshansky Transmission Corp. Dual mode hydromechanical CVT (ref. 5).

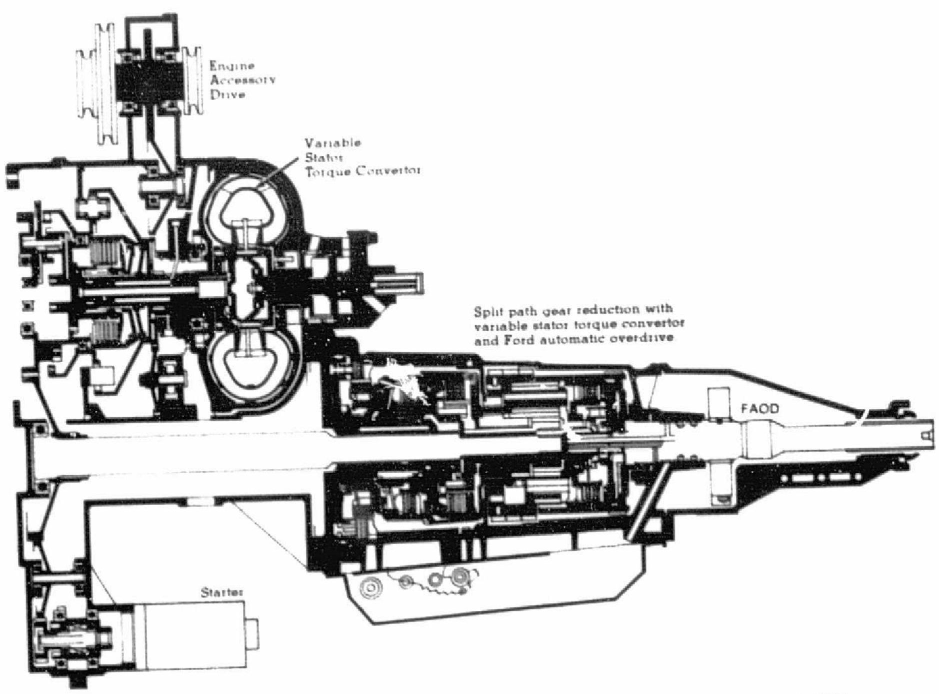

Figure 9. - DOE Garrett/Ford advanced gas turbine variable stator torque convertor CVT.

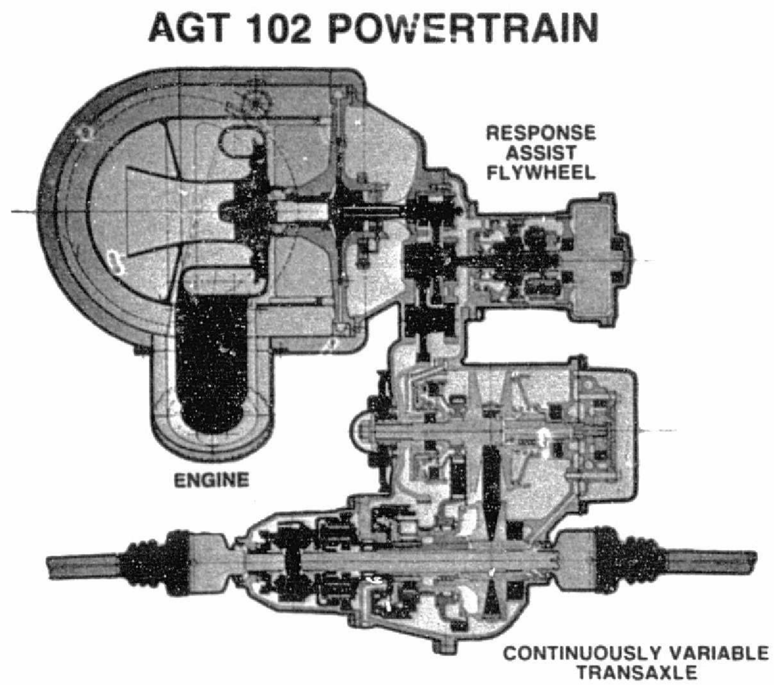

Figure 10. - DOE Chrysler/Williams advanced gas turbine drivetrain - preliminary layout.

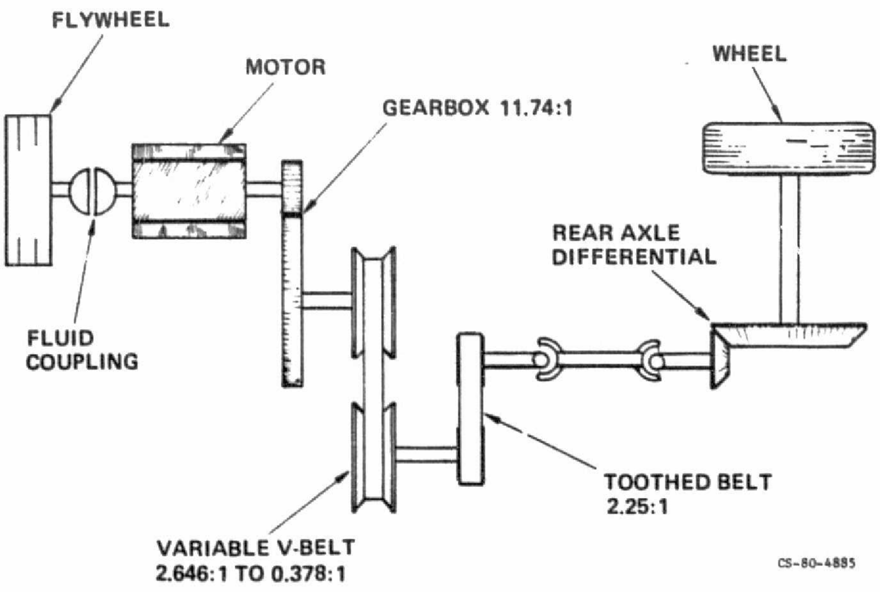

Figure 11. - Rubber V-Belt CVT for DOE/USPS delivery vehicle (ref. 17).

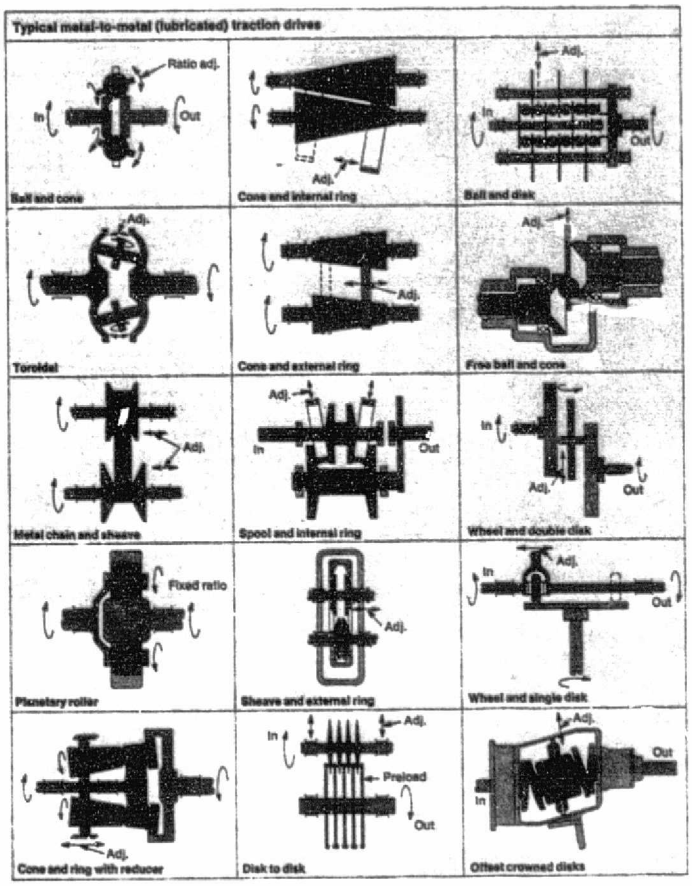

Figure 12. - Typical industrial traction drive geometries. (Courtesy of Design Engineering (ref. 18).)

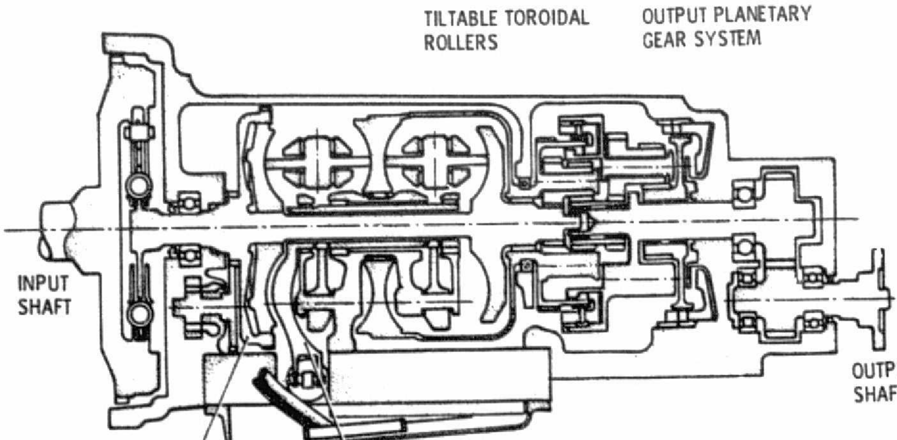

Figure 13. - B. L. Technology/Perbury traction CVT. 60 kW passenger car test installation (ref. 21).

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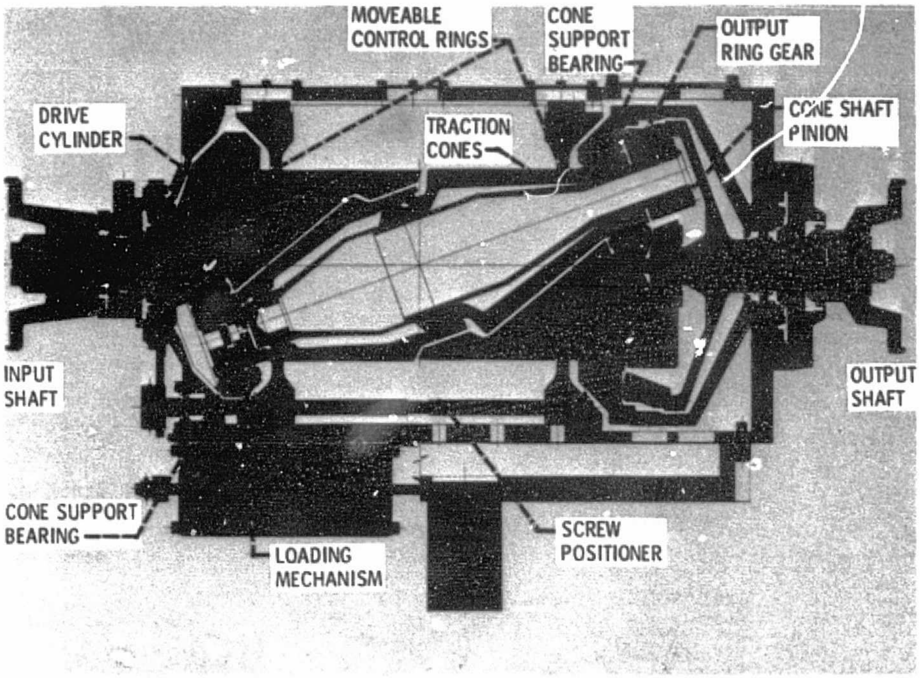
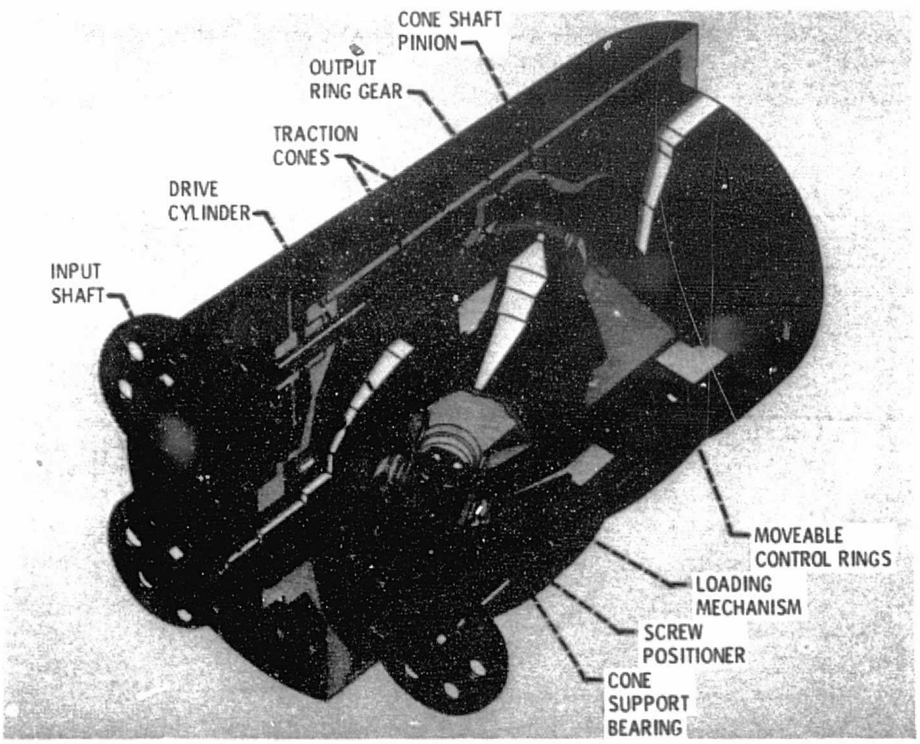


(a) CROSS- SECTION.



(b) ISOMETRIC VIEW.

Figure 14. - Vadetec nutating traction CVT. (Courtesy of Vadetec Corp., Troy, MI.)

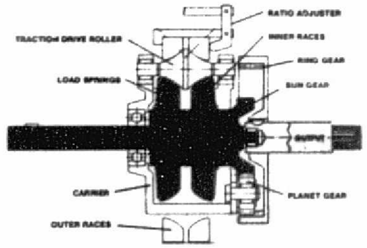


Figure 15. - Fafnir bearing division's planetary cone roller traction drive (ref. 23).

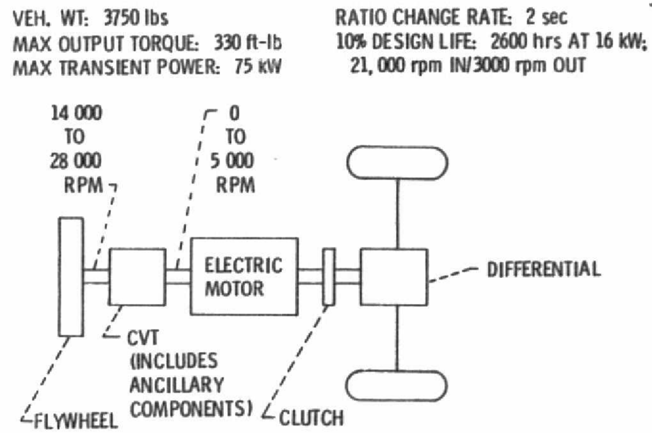


Figure 16. - CVT design study flywheel-electric vehicle drive train.

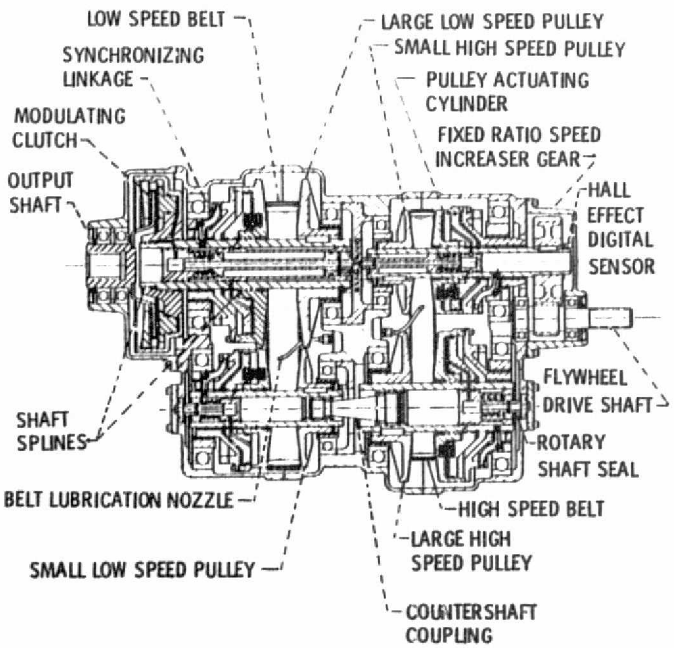


Figure 17. - Preliminary layout of Battelle's steel V-belt CVT (ref. 25).

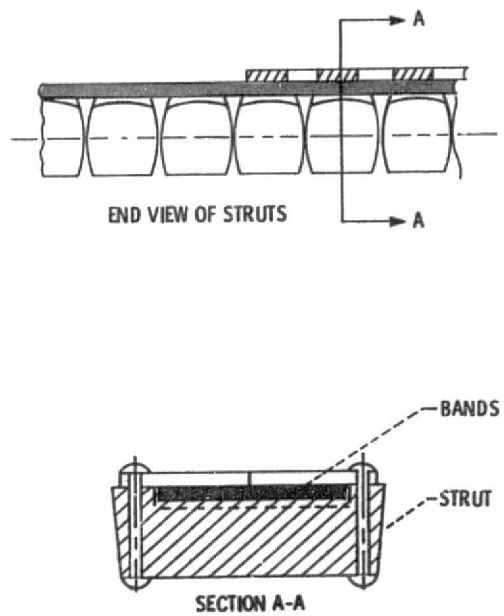


Figure 18. - Steel V-belt construction (ref. 25).

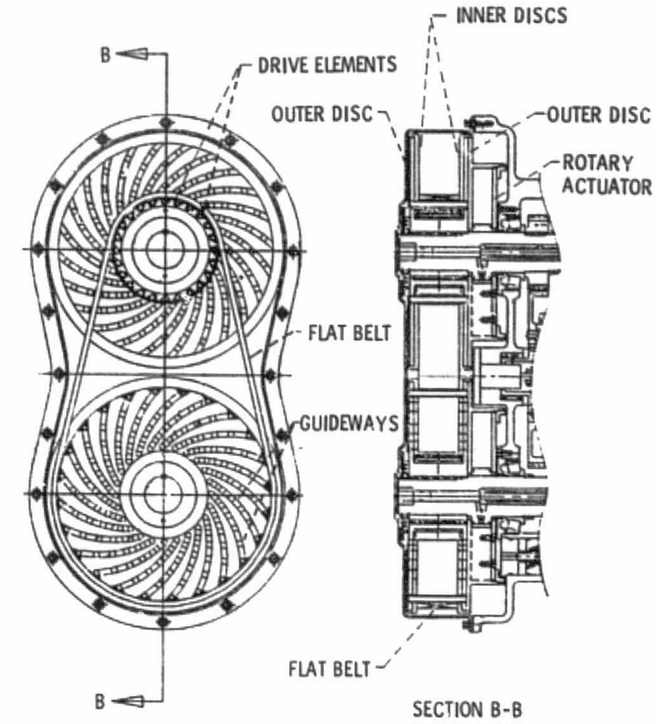


Figure 19. - Preliminary layout of variable-ratio pulleys for Kumm's flat belt CVT (ref. 26).

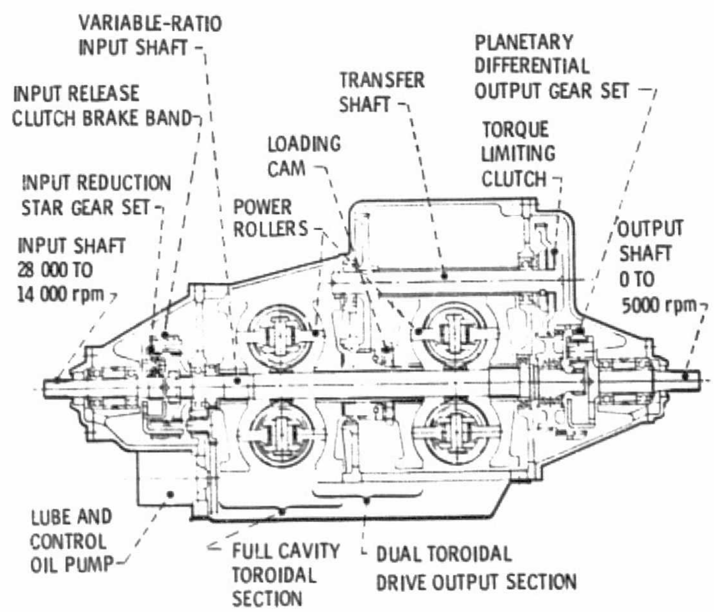


Figure 20. - Preliminary layout of AiResearch's torodial traction CVT (ref. 27).

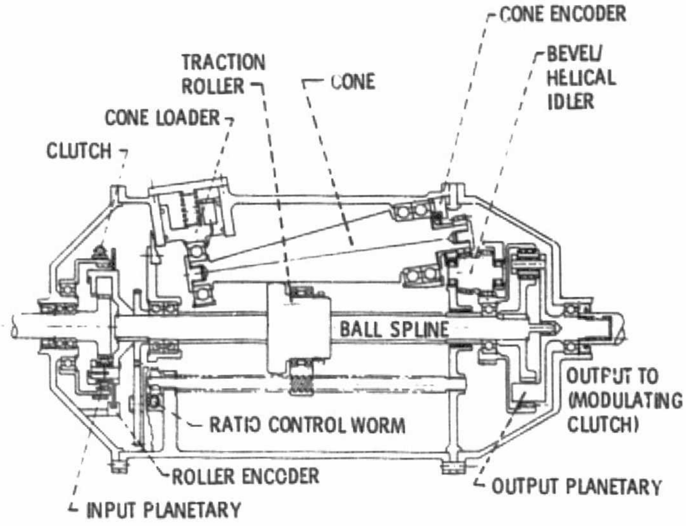


Figure 21. - Preliminary layout of Bales-McCoin's cone-roller traction CVT (ref. 28).

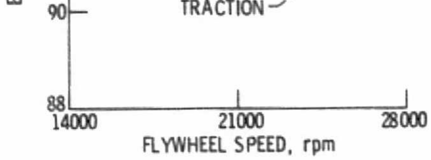


Figure 22. - Comparison of predicted CVT efficiencies at weighted average operating conditions (ref. 24).