

SAE Design Guideline: Metal Belt Drive Continuously Variable Ratio (CVT) Automatic Transmissions

Foreword—The purpose of this guideline is to provide a basic understanding of the design considerations that are essential to the development of metal belt drive continuously variable transmissions. The specific focus of the information presented is to provide recommended practices for the design of the variator section of the transmission. Information is provided that enables engineers to understand the function and design methodology of the major components within the variator system. The information presented was assembled from SAE technical publications and papers and discussions with transmission and component manufacturers. The information represents the current practices utilized in variator system design.

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1. Scope—The purpose of this guideline is to provide the essential design considerations for the metal “V” belt variator used in continuously variable transmissions. Information from SAE papers, transmission manufacturers, and component manufacturers is provided to aid engineers in understanding the function and design methodology of the major components within the variator system. Recommended design practices are given based on current practices.

2. References

2.1 Applicable Publications—The following publications form a part of this specification to the extent specified herein. Unless otherwise indicated, the latest version of SAE Publications shall apply.

2.1.1 SAE PUBLICATIONS—Available from SAE, 400 Commonwealth Drive, Warrendale, PA 15096-0001.

- L. R. Oliver, K. G. Hornung, J. E. Swenson and H. N. Shapiro, “Design Equations for a Speed and Torque Controlled Variable Ratio V-Belt Transmission,” SAE Paper 730003, 1973
- C. Chan, T. Volz, D. Breitweiser, A. Frank, F. S. Jamzadeh and T. Omitsu, “System Design and Control Considerations of Automotive Continuously Variable Transmissions,” SAE Paper 840048, 1984
- Y. Sakai, “The ‘ECVT’ Electro Continuously Variable Transmission,” SAE Paper 880481, 1988
- W. Seidel, J. Petersmann, U. Hickmann and W. Mollers, “Adaptive Control System for Continuously Variable Transmission (CVT),” SAE Paper 925062, 1992
- K. Abo, M. Kobayashi and M. Kurosawa, “Development of a Metal Belt-Drive CVT Incorporating a Torque Converter for Use with 2-liter Class Engines,” SAE Paper 980823, 1998

2.1.2 ASTM PUBLICATIONS—Available from ASTM, 100 Barr Harbor Drive, West Conshohocken, PA 19428-2959.

- ASTM D 130—Test Method for Detection of Copper Corrosion from Petroleum Products by the Copper Strip Tarnish Test
- ASTM D 445—Test Method for Kinematic Viscosity of Transparent and Opaque Liquids (the Calculation of Dynamic Viscosity)
- ASTM D 665—Test Method for Rust-Preventing Characteristics of Inhibited Mineral Oil in the Presence of Water
- ASTM D 892—Test Method for Foaming Characteristics of Lubricating Oils
- ASTM D 1748—Test Method for Rust Protection by Metal Preservatives in the Humidity Cabinet
- ASTM D 2882—Test Method for Indicating the Wear Characteristics of Petroleum and Non-Petroleum Hydraulic Fluids in a Constant Volume Vane Pump
- ASTM D 2983—Test Method for Low-Temperature Viscosity of Automotive Fluid Lubricants measured by Brookfield Viscometer
- ASTM D 3233—Test Methods for Measurement of Extreme Pressure Properties of Fluid Lubricants (Falex Pin and Vee Block Methods)
- ASTM D 3945—Test Methods for Shear Stability of Polymer-Containing Fluids Using a Diesel Injector Nozzle
- ASTM D 4172—Test Method for Wear Preventive Characteristics of Lubricating Fluid (Four-Ball Method)
- ASTM D 4683—Test Method for Measuring Viscosity at High Shear rate and High Temperature by Tapered Bearing Simulator
- ASTM D 6082—Test Method for High Temperature Foaming Characteristics of Lubricating Oils

2.1.3 CEC PUBLICATIONS—Available from the Coordinating European Council, Madou Plaza - 25th Floor, Place Madou 1, B-1030 Brussels, Belgium.

- CEC L45-T-93
- CEC L-48-A
- CEC L-070-A85

- 2.1.4 DIN PUBLICATION—Available from Deutsches Institut Fur Normung E V, Burggratestr. 6, D-10787, Berlin, Germany.
- 2.1.5 ISO PUBLICATIONS—Available from ANSI, 11 West 42nd Street, New York, NY 10036-8002.
- ISO 3104/ASTM D 445
ISO 2160/ASTM D 130
- 2.1.6 JASO PUBLICATION—Available from Society of Automotive Engineers of Japan, Inc., 10-2, Goban-cho, Chiyoda-ku, Tokyo 102, Japan.
- JASO M 347-95
- 2.1.7 JSAE PUBLICATIONS—Available from JSAE 10-2, Goban-Cho, Chiyoda-Ku, Tokyo 102 Japan
- M. Fuchino and K. Ohsono, "Development of Fully Electronic Control Metal Belt CVT," JSAE Paper 9636286, 1996
K. Sato, R. Sakakiyama and H. Nakamura, "Development of Electronically Controlled CVT System Equipped with CVTip," JSAE Paper 9636321, 1996
T. Ide, H. Uchiyama and R. Kataoka, "Experimental Investigation on Shift Speed Characteristics of a Metal V-Belt CVT," JSAE Paper 9636330, 1996
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 2. "CVT Lubrication: The Effect of Lubricants on the Frictional Characteristics of the Belt - Pulley Interface," R.F. Watts and K.M. Richard 1997 Tribology Colloquium, Technische Akademie Esslingen
 3. "The Effect of Belt-Drive CVT Fluid on the Friction Coefficient Between Metal Components," T. Ishikawa et al., SAE Paper 972921
 4. "A Comparison of Methods for Evaluating Automatic Transmission Fluid Effects on Friction Torque Capacity - A Study by the International Lubricant Standardization and Approval Committee (ILSAC) ATF Committee," J. L. Linden et al., SAE International Fall Fuels and Lubricants Meeting and Exposition, San Francisco CA, October 19-22, 1998, SAE Paper 982672
 5. "Development of a Vehicle Test for Determining ATF Performance in an Electronically Controlled Capacity Clutch," C. Jones, J.L. Linden, SAE International Fall Fuels and Lubricants Meeting and Exposition, San Francisco CA, October 19-22, 1998, SAE Paper 982667
 6. SPEZIFIKATION - GETRIEBEÖL FÜR STUFENLOSE PKW-AUTOMATGETRIEBE, Erstellt unter der Mitarbeit der Firmen Audi AG, Volkswagen AG, AZF Friedrichshafen AG, Draft 2, January 1997
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 8. Chrysler MS-9602
 9. Dexron™ III
 10. "Determination of the Air Entrainment Characteristics of Automatic Transmission Fluids Using a Dynamic Flow Apparatus," P.J. Colby et al., SAE International Fall Fuels and Lubricants Meeting and Exposition, San Francisco CA, OCTO 19-22, 1998, SAE Paper 982669
 11. "Shear Stability of Automatic Transmission Fluids - Methods and Analysis. A Study by the International Lubricants Standardization and Approval Committee (ILSAC) ATF Subcommittee, J.W. Sprys et al., SAE International Fall Fuels and Lubricants Meeting and Exposition, San Francisco CA, October 19-22, 1998, Paper 982673

2.2 Related Publications

2.2.1 ASME PUBLICATION—Available from ASME, 22 Law Drive, Box 2900, Fairfield, NJ 07007-2900.

ASME 72-PTG-21, "Mechanics of the Flat Belt Drive," T. C. Firbank, 1972

2.2.2 OTHER PUBLICATIONS

1. W. S. Worley, "Designing Adjustable-Speed V-Belt Drives for Farm Implements," SAE Transactions Vol. 63, pp 321-333, 1955
2. K. G. Hornung, "Factors Influencing the Fatigue Characteristics of Rubber-Textile Machine Elements," Dissertation for Doctor of Philosophy at Ohio State University, 1959
3. B. G. Gerbert, "Force and Slip Behaviour in V-Belt Drives," Acta Polytechnica Scandinavica, Mech. Eng. Series, No. 67, Helsinki, 1972
4. T. C. Firbank, "Mechanics of the Flat Belt Drive," ASME 72-PTG-21, 1972
5. D. L. Keller and R. E. Wilson, "Design and Development of a High Horsepower Torque Sensing Variable Speed Drive," SAE Paper 720709, 1972
6. B. G. Gerbert, "Adjustable Speed V-Belt Drives - Mechanical Properties and Design," SAE Paper 740747, 1974
7. D. J. Brents, "Axial Force and Efficiency Tests of Fixed Center Variable Speed Belt Drive," SAE Paper 810103, 1981
8. R. R. Radtke, L. E. Unnewehr and R. J. Freedman, "Optimization of a Continuously Variable Transmission with Emission Constraints," SAE Paper 810107, 1981
9. R. F. Stieg and W. S. Worley, "A Rubber Belt CVT for Front-Wheel-Drive Cars," SAE Paper 820746, 1982
10. J. P. Dolan and W. S. Worley, "Closed-Form Approximations to the Solution of V-Belt Force and Slip Equations," ASME Transactions, Journal of Mechanisms, Transmission, and Automation in Design, Vol 107, pp 292-300, June 1985
11. E. L. Kumm and T. C. Kraver, "Development of a Flat-Belt, Continuously-Variable Transmission," SAE Paper 850488, 1985
12. S. Abrate, "Rubber CVT Belts," SAE Paper 850571, 1985
13. E. L. Kumm and T. C. Kraver, "Enhancement of the Flat-Belt, Continuously-Variable Transmission Technology," SAE Paper 861214, 1986
14. S. Avramidis, "A 'State of the Art' Chain Driven CVT Design," SAE Paper 861354, 1986
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21. JSAE Automotive Engineers' Handbook, Part 2 Design, pp 232-237, March 1991
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32. H. Asayama, J. Kawai, A. Tonohata and M. Adachi, "Mechanism of Metal Pushing Belt," 1994 JSAE Spring Convention Proceedings, JSAE Paper 9432804, 1994
33. T. Ide, A. Udagawa and R. Kataoka, "A Dynamic Response Analysis of a Vehicle Containing a Metal V-Belt CVT," JSAE Conference Proceedings 994 1994-10, pp 213-216, 1994
34. J. D. Micklem, D. K. Longmore and C. R. Burrows, "Belt Torque Loss in a Steel V-Belt Continuously Variable Transmission," Proceedings of Institution of Mechanical Engineers, Vol. 208, pp 91-97, 1994
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36. P. Mott and B. Martin, "The Gemini Phased Chain System: A New Tool in Automotive Power Transmission," SAE Paper 950667, 1995
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42. G. Gerbert, "Skew V-Belt Pulleys," JSAE Paper 9636259, 1996
43. F. Sorge, "Influence of Pulley Bending on Metal V-Belt Mechanics," JSAE Paper 9636268, 1996
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45. M. Boos and H. Mozer, "ECOTRONIC - The Continuously Variable AF Transmission (CVT)," SAE Paper 970685, 1997
46. H. Yoshida, "A Study of Forces Acting on Rings for Metal Pushing V-Belt Type CVT," SAE Paper 970686, 1997
47. D. Kobayashi, Y. Mabuchi and Y. Katoh, "A Study on the Torque Capacity of a Metal Pushing V-Belt for CVTs," SAE Paper 980822, 1998
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49. K. Ohya and H. Suzuki, "Development of CVT Pulley Piston Featuring Variable Thickness and Work-Hardening Technologies," SAE Paper 980826, 1998
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55. A. Brandsma, J. van Lith and E. Hendriks, "Push Belt CVT Developments for High Power Applications," Proceedings of the International Congress on Continuously Variable Power Transmission, pp 142 - 147, Sept. 16 & 17, 1999

3. Definitions—The Nomenclature in parentheses and italics is the Alternate Nomenclature.

- 3.1 Variator Belt**—A flexible, endless member that transmits torque between two shafts through frictional contact with conical sheaves.
- 3.2 Variator Ratio (CVT Ratio, Sheave Ratio, Pulley Ratio, Belt Ratio)**—Speed Ratio: primary sheave speed/secondary sheave speed
- 3.3 Variator Ratio Range (Overall Ratio, Ratio Coverage)**—Underdrive ratio/overdrive ratio
- 3.4 Underdrive Ratio (Low Ratio)**—Highest (numerical) variator ratio. Typically equivalent to low (first) gear in an A/T or M/T
- 3.5 Overdrive Ratio (High Ratio)**—Lowest (numerical) variator ratio. Typically equivalent to the highest gear in an A/T or M/T
- 3.6 Primary Sheave (Pulley) Set (Input Sheave/Shaft, Pulley, Sheave Set)**—Sheave set that is coupled to the engine and represents the input to the CVT variator unit. (May have gearing between engine and primary sheave.)
- 3.7 Secondary Sheave (Pulley) Set (Output Sheave/Shaft Pulley, Sheave Set)**—Sheave set that is coupled to the wheels and represents the output of the CVT variator unit. (Typically has gearing between the secondary sheave and the wheels.)
- 3.8 Variator Belt Width (Nominal)**—The width of the belt at an effective pitch line. Belt width and length combine to define the operating geometry.
- 3.9 Belt Length**—The length of the belt at the effective pitch line. Belt width and length combine to define the operating geometry.
- 3.10 Wrap Angle**—The angle on the sheave through which the belt contacts the sheaves. Can be divided into active and inactive angles.
- 3.11 Active Wrap Angle (Arc)**—That portion of the wrap angle where the belt tension is increasing (or decreasing).
- 3.12 Passive Wrap Angle (Arc) (Inactive Wrap Angle)**—That portion of the wrap angle where the belt tension is constant.
- 3.13 Shaft Diameter**—The diameter of the primary or secondary shafts underneath the belt.

- 3.14 Fixed Sheave (*Fixed Shaft*)**—The sheave component that is rigidly attached to a shaft. Typically the fixed sheave is an integral part of a shaft.
- 3.15 Moveable Sheave (*Variable Sheave*)**—The sheave component that moves axially relative to the fixed sheave to change belt ratio.
- 3.16 Center Distance**—The distance between the centers of the primary and secondary shafts.
- 3.17 Sheave Alignment**—The relative axial position of the primary fixed sheave to the secondary fixed sheave.
- 3.18 Sheave Offset (*Gage Width*)**—The dimension in the axial direction between the primary and secondary fixed sheaves at a predetermined reference diameter. For example, sheave offset may be 24.6 mm at a reference diameter of 100 mm dia.
- 3.19 Sheave Angle**—The angle of the sheave surface relative to the operating plane of the belt. Typically 11 degrees.
- 3.20 Pitch Diameter**—The effective diameter at which the torque is transmitted. Also equal to the diameter at which the sheave opening is equal to the nominal belt width.
- 3.21 Pitch Distance**—The tangential distance between belt segments that contact the sheaves.
- 3.22 Variator Input Torque**—Input torque to the primary sheave set.
- 3.23 Top Speed Ratio**—The variator ratio at the conditions that define maximum vehicle speed.
- 3.24 Coefficient of Friction (*Traction*)**—The effective coefficient of friction between the belt and sheaves.
- 3.25 Balance Dam**—A feature that can be used in combination with a piston or cylinder to reduce the centrifugal force component.
- 3.26 Stops Internal/External**—The feature in the sheave design that controls axial position of the moveable sheave at the extremes of travel.
- 3.27 Sheave Stroke**—The axial travel of the moveable sheave to allow the variator to cover the overall ratio range.
- 3.28 Clamping Force**—The axial force exerted on the belt by the sheave.
- 3.29 Clamping Force Ratio**—The ratio of the primary sheave axial force to the secondary sheave axial force.
- 3.30 C.V.U. (*Variator*)**—The Continuously Variable Unit: The combination of the primary sheave set, secondary sheave set, and belt.
- 4. Application Considerations**—The first automotive application of continuously variable transmission (CVT) technology occurred in one of the original gasoline powered cars developed in 1886. This application employed a rubber V-belt as part of the variator system. From this initial installation, continuously variable technology has evolved over the decades as major automobile and component manufacturers continue to pursue variable ratio as an alternative to step ratio design transmissions. The metal belt is now used to increase the torque carrying capability of CVT belt drives. Advances in electronic controls have overcome many of the design issues and have significantly improved the driveability and efficiency of CVT applications. The focus on CVT technology has been intensified as a result of more pressure on the automobile industry to reduce exhaust emissions and improve fuel economy. This section will discuss the application considerations for automotive metal belt CVT applications.

4.1 Vehicle Requirements—There are several factors that must be considered in selecting a transmission configuration for a particular vehicle design. The most significant vehicle characteristics are determined by the particular market segment that the vehicle is intended to serve. Powertrain requirements are then established for the market segment that can further be reduced to engine - transmission configurations. Factors considered in powertrain selection include vehicle acceleration, economy of operation, driveability, emissions, noise (NVH), weight, packaging, and cost targets.

The previous considerations apply to any configuration of automotive application. However, each market segment and vehicle type will have relatively well defined operating characteristics that will establish the powertrain configuration. Market segment and demographics also establish transmission selection. A typical European economy automobile will be equipped with a manual transmission, whereas, the same vehicle class in the United States will be equipped with an automatic transmission.

New transmission designs must incorporate wider overall ratio range to provide good vehicle throttle response while lowering engine speeds during steady-state driving conditions for improved fuel economy and lower emissions. CVTs offer a wide overall ratio range without the concerns of shift quality and shift busyness.

4.2 Engine Characteristics—Engine developments of the past decade have focused on lowering engine speeds to reduce emissions. Output torque operating ranges and fuel efficiency islands have also been shifted to lower engine speeds. The overall effect of engine developments has reduced engine operating range in order to achieve fuel efficiency and lower emissions. Transmission design must focus on maintaining engine operation within the effective power band and fuel efficiency islands in order to provide optimum overall vehicle operation.

4.3 Transmission Requirements—As discussed previously, the transmission must provide the optimum performance in both steady-state and non-steady-state vehicle operations. The major areas for design consideration are power flow, ratio range, packaging, cost, duty cycle, durability, and manufacturing. During steady-state conditions such as interstate highway operation, the transmission must provide high efficiency. The efficiency of the metal belt CVT transmission is typically lower than that of stepped gear transmissions. The lower transmission efficiency is overcome by the CVT, by allowing the overall powertrain to be controlled closer to the optimum conditions resulting in an overall increase in powertrain efficiency. The powertrain efficiency is also improved by the ability to achieve an earlier coupling of the starting device in the CVT design, which improves powertrain efficiency. Non-steady-state operation, such as accelerating from rest, passing, deceleration, throttle response, and shift quality are the most difficult considerations of transmission design because the requirements need to be evaluated over a wide range of operating conditions.

4.3.1 POWER FLOW—The power flow requirements for a CVT depend on many different considerations. The choice of starting device is one of the primary considerations in establishing CVT design. The types of starting devices for current production CVT include torque converters, input starting clutches, or output starting clutches. The torque converter will provide torque multiplication from the engine to the transmission during vehicle launch conditions. Torque multiplication must be included in the design of the rest of the transmission. The design of the power flow is often influenced by the transmission packaging space or cost targets for the vehicle application. Typical CVT transmission cross sections are shown in Appendix A.

With an input starting device, such as a torque converter or input starting clutch, the location of the starting device is usually between the engine and the CVT sheave set. When in gear, the CVT pulleys are at zero speed when the wheels are at rest. During this vehicle condition, changing the ratio of the sheave set is very difficult.

When an output starting clutch design is used, the sheave set is located between the engine and the starting clutch. The output side of the start clutch is coupled to the output end of the transmission. When the vehicle is at rest, the starting clutch de-couples the sheaves from the wheels and allows the sheaves to rotate with the engine. This condition allows ratio change with the sheave set when the output of the transmission is at zero speed.

Another consideration in the power flow is the location of the range clutches for forward and reverse. With the start clutch between the engine and the sheave set, the pulleys are de-coupled from the engine in a neutral state. This is an advantage for shifting into gear because the inertia of the sheaves does not need to be a design consideration when the vehicle is at rest. The disadvantage of this configuration becomes apparent during a vehicle panic stop when the wheels stop suddenly. During this event, the sheaves would also stop if another clutch is not placed between the wheels and the sheave set to provide ratio change opportunity. The range clutches could also be located on the output side of the sheaves between the output sheave and the transmission output. In this configuration, the sheaves are typically coupled directly to the transmission input, allowing the sheaves to rotate when the vehicle is at rest. When shifting into gear, the inertia of the sheave set needs to be a design consideration.

- 4.3.2 **RATIO RANGE**—As with any transmission, the selection of ratios is a critical area of the transmission design. With the continuously variable transmission design, ratio selection is slightly different than a step ratio transmission. Because of the availability of all ratios between the maximum underdrive and overdrive ratios, these two conditions are the key points for ratio selection. These points can be plotted as a function of vehicle speed and primary sheave on a graph commonly referred to as a variogram (Figure 1). The underdrive ratio needs to be established to provide the desired launch performance of the vehicle in combination with the starting device that was selected. The overdrive ratio should be selected to provide the ideal or most effective engine speeds to provide optimum fuel economy. The ratio spread between the underdrive and overdrive ratios is referred to as the ratio range. The typical ratio range for a CVT is between 5 to 6 compared to 3.8 to 4.3 for a four-speed step transmission. Depending on the vehicle application, typically, the CVT will have better performance and fuel economy than a step transmission. The transmission design considerations for ratio selection will be discussed in a later section.

**Example Variogram: $V_{max}=210\text{ kph}$ $R_c=5.44$; $V_{1000\text{ OD}}=55.0\text{ kph}$; $V_{1000\text{ LOW}}=10.1\text{ kph}$;
 $i_{OD}=0.450$; $i_{TOP}=0.550$; $i_{LOW}=2.450$**

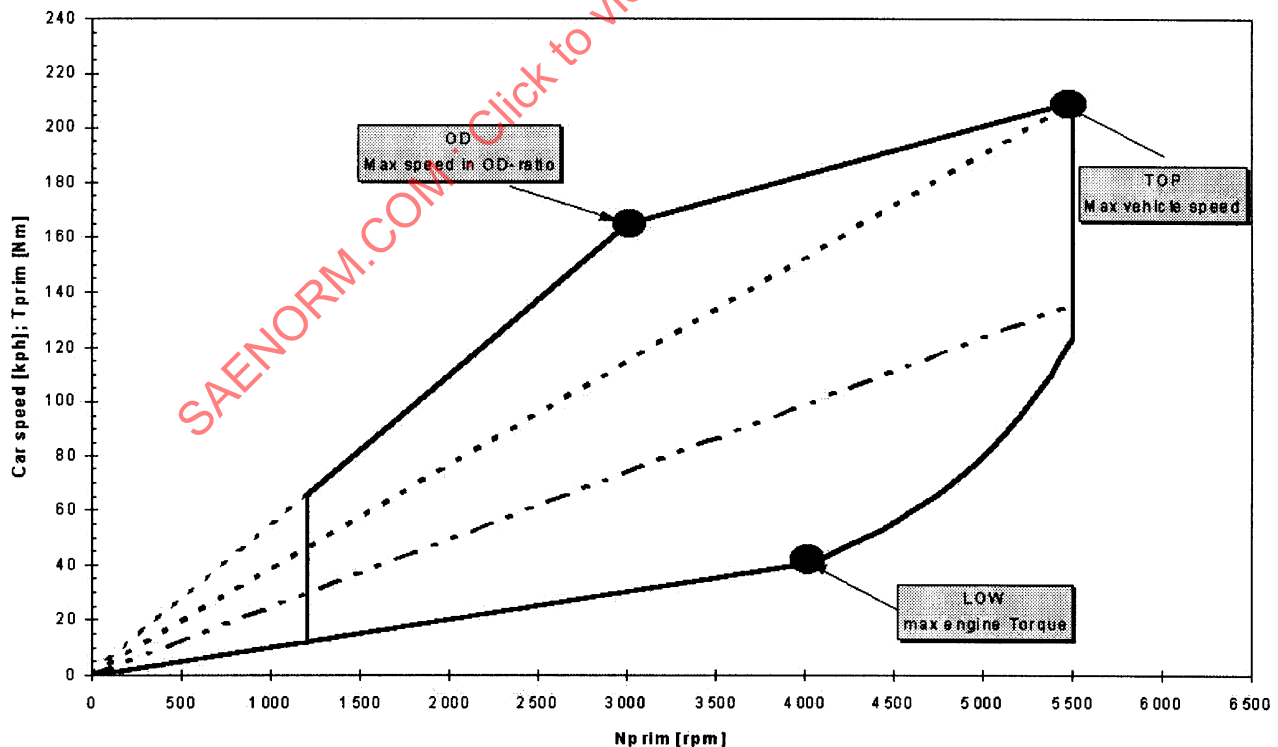


FIGURE 1—VARIOGRAM

- 4.3.3 **PACKAGING**—The packaging requirements of the vehicle need to be considered when evaluating transmission requirements. The packaging environment may influence the power flow, the choice of starting device, the ratio range and several of the other transmission requirements. The packaging environment required for the sheaves is a unique requirement for CVTs. By the nature of the design, the sheave set requires two axes for the variator. For this reason, the belt drive CVT design is inherently more suited for multi-axis transmission designs, such as front wheel drive, than a single axis design like many rear wheel drive transmissions.
- 4.3.4 **DUTY CYCLE/DURABILITY**—The duty cycle and durability requirements of a CVT are comparable to those of a stepped gear transmission. These requirements may vary by application and region. There are some operating characteristics of the CVT which differ from stepped gear transmissions. Due to the fact that this CVT is a metal-to-metal friction drive device, the belt and sheaves will not tolerate slip at any time. The control system must be robust to provide adequate clamping of the belt under all conditions.
- 4.3.5 **COMPONENTS**—While many components of the CVT are similar to those in stepped gear transmissions, there are some unique characteristics of the CVT that should be considered. The CVT has fewer parts than a stepped gear automatic transmission, but the unique CVT components, such as the sheaves, require high precision. The pump in the CVT also has a different duty cycle than that of a stepped gear transmission in that the pressure requirements are generally higher and the flow demand for moving the sheaves is an additional consideration.
5. **Belt Design**—This section will describe the construction and operation of metal push belts (compression type) and metal chain belts (tension type). There have been several papers published relating to the operation of the CVT belt which are listed in the reference section.

5.1 Compression Type Belts

- 5.1.1 **CONSTRUCTION**—The metal belt consists of two sets of metal bands and a number of wedge-shaped steel blocks (elements). The compression type belt is designated as a push belt since the compressed blocks act as a solid column to transfer torque from one sheave to the other. The thin steel bands form the main structure of the belt. The bands are fitted closely together to form a nested set with no play. There are two sets of bands, one on each side of the belt assembly. Each band is approximately 0.2 mm thick, and a band set typically consists of 9 or 12 bands (Figure 2).

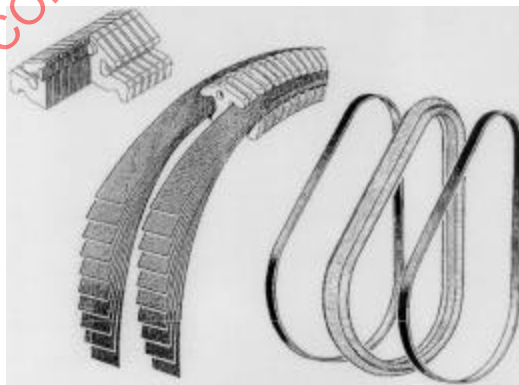


FIGURE 2—COMPRESSION BELT CONSTRUCTION (VAN DOORNE TYPE)

These band sets give the push belt assembly high tensile strength with a maximum of flexibility. A special profile on the inner side of each band optimizes the lubrication area and minimizes the friction losses between the sliding bands. This design, combined with the material selection used, results in a compact system with a high power density and fatigue strength.

Typical block thickness varies from 1.5 mm for the 24 mm wide belts, to 1.8 mm for the 30 mm wide belts (Figure 3). The blocks have a wedge shape, and the sides of the block that contact the sheaves are profiled. The power transfer occurs over the full height of the profiled side. The 30 mm blocks have a larger contact area than the 24 mm blocks in order to decrease the contact pressure experienced at the higher torque loading.

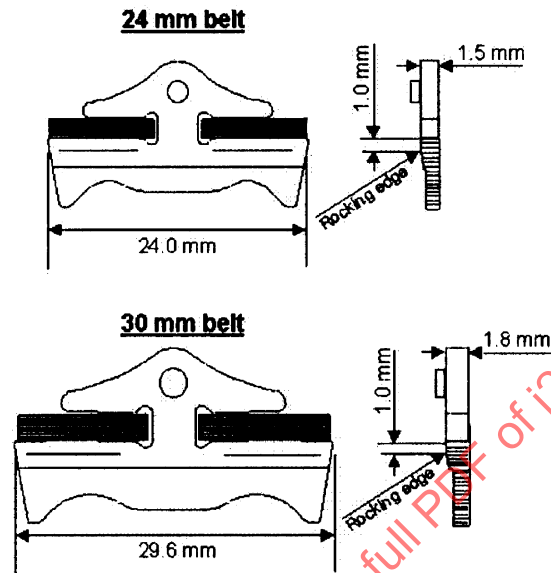


FIGURE 3—BELT ELEMENT DETAIL (VAN DOORNE)

The steel blocks are clamped axially between the sheave halves. The blocks have tendency to move outward, along the radius of the sheave, due to their wedge shape (Figure 4). The outward movement of the blocks is countered by the bands, thereby placing the bands in tension and holding the blocks in the V of the sheave. This balance between the radial forces between the blocks on the bands and the tensile forces in the bands, is the basis for establishing a frictional force between the blocks and pulleys. This friction force and the accompanying contact area are what allow torque transfer to occur within the sheave/push belt assembly.

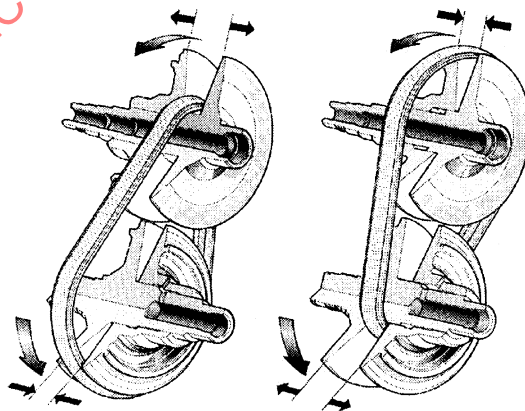


FIGURE 4—BELT OPERATION

The Van Doorne belt has been designed to ensure that the tensile force in the band assemblies is higher than the “push” force between the blocks. This allows the bands to push the blocks down into the sheaves, enabling them to transmit torque, and at the same time prevents the stacked column of blocks from buckling and pushing out of the sheave sets (Figure 5). The Van Doorne design defines the push force transmission axis (belt pitch line) at the height of the rocking edge of the block. The rocking edge is located approximately 1 mm below the block saddles (the crowned portion of the blocks where the innermost band contacts the block over almost the full width of the band). This dimension is used to determine the geometrical running radius on both the primary and secondary pulley, and ultimately the overall ratio.

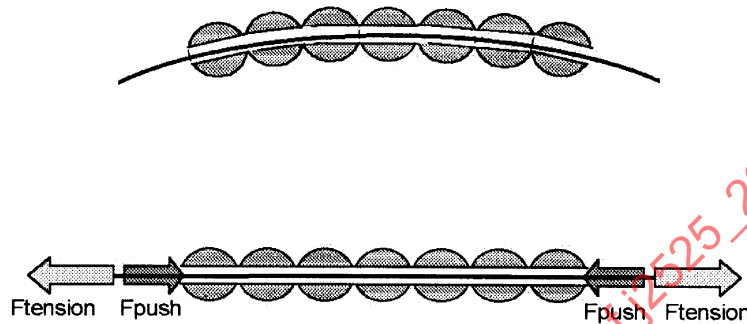


FIGURE 5—BELT FORCE DIAGRAM

The bands and the rocking edge of the blocks travel along different radii through the sheave sets. This causes a relative motion between the bands and the blocks when the belt is not in 1:1 ratio (Medium ratio). This relative motion increases towards the extreme ratio settings. Because of this, the push belt typically shows it's highest efficiency in Medium ratio.

- 5.1.2 **BELT LENGTH AND PITCH LINE DEFINITION**—The length of the Van Doorne push belt varies for each specific transmission layout. Unlike chains, the length of the van Doorne push belt can vary non-incremental, since this belt length is dependent on the diameter of the band sets, not on the thickness of the blocks. Once the diameter of the bands has been determined, a quantity of blocks with slightly different thickness is selected to “fill” the belt.

The official product code (<block type/width>/<number of bands in each set of bands>/<thickness of blocks>/<diameter of the innermost ring>: for example 24/9/1.5/208.8) calls out the diameter of the innermost band, not the diameter of the belt itself. However, using the definition of the belt pitch line (the pitch line radius or rocking edge position is the radius of innermost band minus 1 mm), the total length of the belt can be calculated from this belt code.

- 5.1.3 **BELT CAPACITY**—The maximum torque capacity of the belt is primarily determined by the stress levels in the bands. These stresses are a combination of the tensile stresses (due to the axial clamping forces from the sheaves and centrifugal forces generated by the blocks), and bending stresses (due to bending of the band sets that occurs in extreme ratio settings at small wrap angles) experienced by the belt at different ratio and load settings.

A variety of belt sizes, varying in width and number bands within the band sets is available for various applications. Typically, the wider the belt, and the more bands in each set, the higher the belt capacity (Figure 6).

Application range of standard VDT pushbelts in standard automotive applications

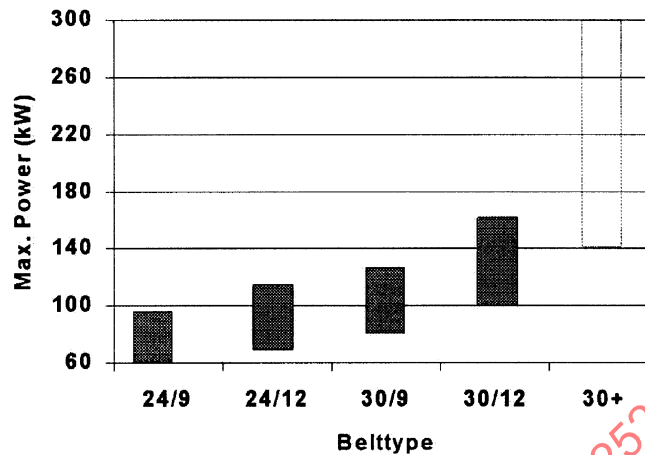


FIGURE 6—BELT CAPACITY

The combinations of ratio coverage, input torque and belt speed, and other related parameters determine the belt capacity and type (Figures 7 and 8).

The belt loads need to be calculated at three distinct ratio points in the variogram (belt operating curve): TOP, OD, and LOW, in order to select the correct belt for an application. The highest centrifugal belt forces are found at maximum vehicle speed (TOP ratio). The highest bending stresses are found in the OD ratio setting, along with the high centrifugal forces. The axial sheave forces resulting in the highest contact pressures across the blocks, along with the highest tensile and bending band stresses are found in the LOW ratio setting.

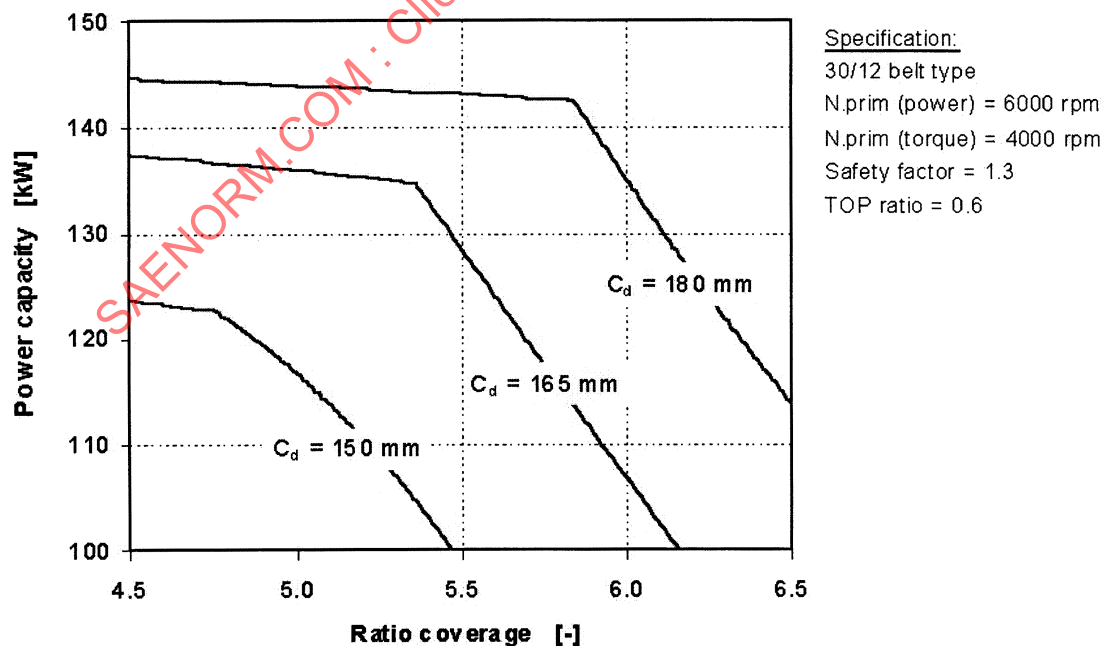


FIGURE 7—POWER CAPACITY VERSUS RATIO COVERAGE FOR REPRESENTATIVE SHEAVE CENTER DISTANCES

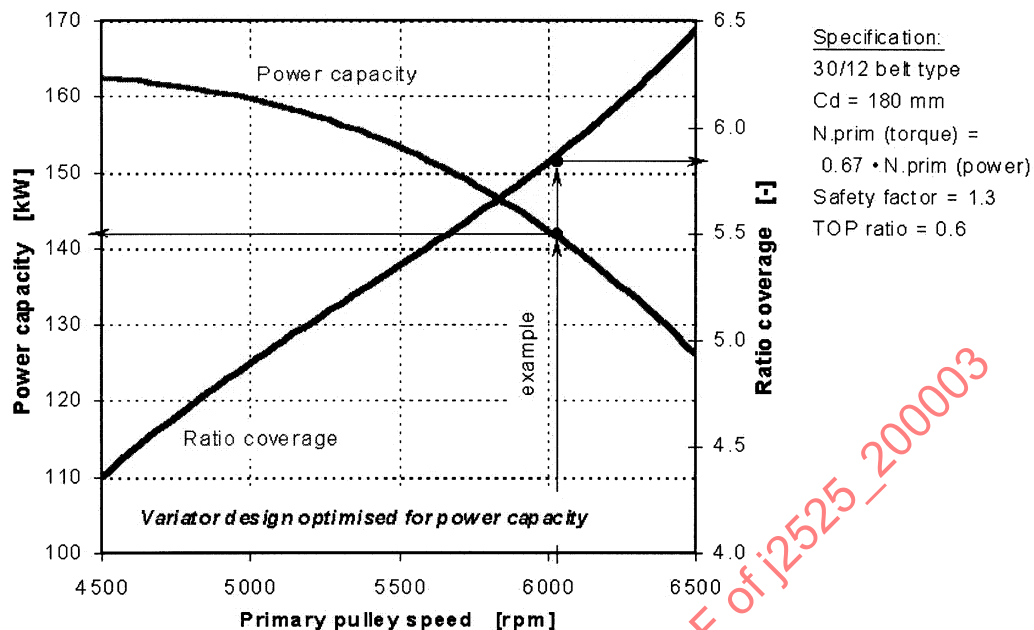


FIGURE 8—POWER CAPACITY AND RATIO COVERAGE VERSUS PRIMARY SPEED

- 5.1.4 POWER LOSS MECHANISMS—The efficiency of the push belt is ratio dependent, due to the varying internal friction losses of the bands at different ratio settings. Since the sliding speeds of the bands increase towards the extreme ratios, the efficiency decreases approximately 1 to 2% at these ratio settings (Figure 9). Belt losses are at a minimum in the medium ratio setting, where the differential sliding speed of the bands is minimized.

Axial clamping forces exerted by the sheaves has a greater impact on the belt efficiency. Typically the clamping force is set at least 30% higher than the minimum required clamping force to prevent the belt from slipping under torque loading. Excessive clamping forces above this 30% safety margin leads to higher belt stresses and unnecessary friction losses. The belt efficiency decreases on the order of 1 to 2% due to overclamping (Figure 10).

The amount of overclamping is expressed as the safety factor. The safety factor is defined as the actual applied clamping force divided by the minimum required clamping force to prevent belt slip. Therefore, a 30% overclamping force equates to a SF of 1.3.

- 5.1.5 FAILURE MECHANISMS—Failure modes for the belt include overloading (belt slip), overspeeding, misalignment, and overheating due to insufficient lubrication.

Insufficient clamping force in relation to the applied torque level will lead the belt to slip over the sheaves, damaging the sides of the block and the sheaves and eventually lead to premature failure.

Overspeeding occurs when the belt exceeds the maximum rated speeds, thereby exceeding the maximum allowable band stresses, and causing premature belt failure.

Misalignment of the sheaves can cause the solid column of blocks to exert excessive forces on the sides of the bands. When the stresses resulting from these side forces exceed the maximum allowable band stresses, premature failure will result.

Sufficient belt lubrication is needed to keep the operating temperature of the belt within operating range.

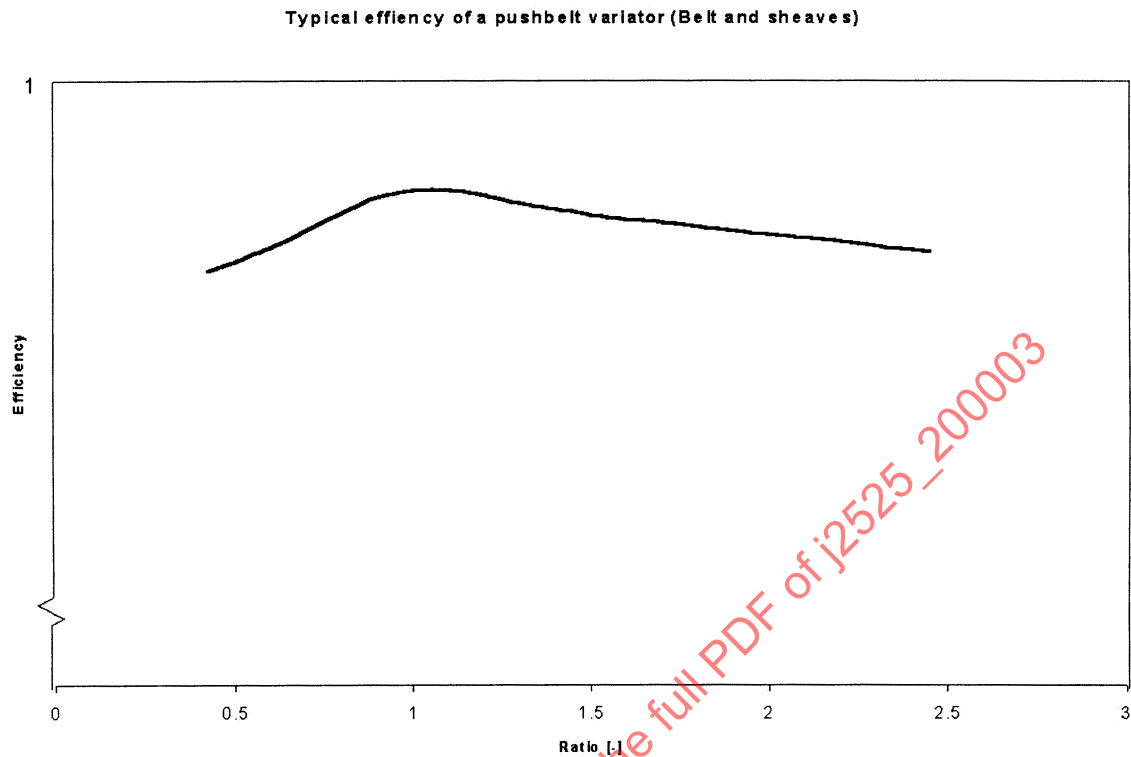


FIGURE 9—COMPRESSION BELT EFFICIENCY VERSUS RATIO

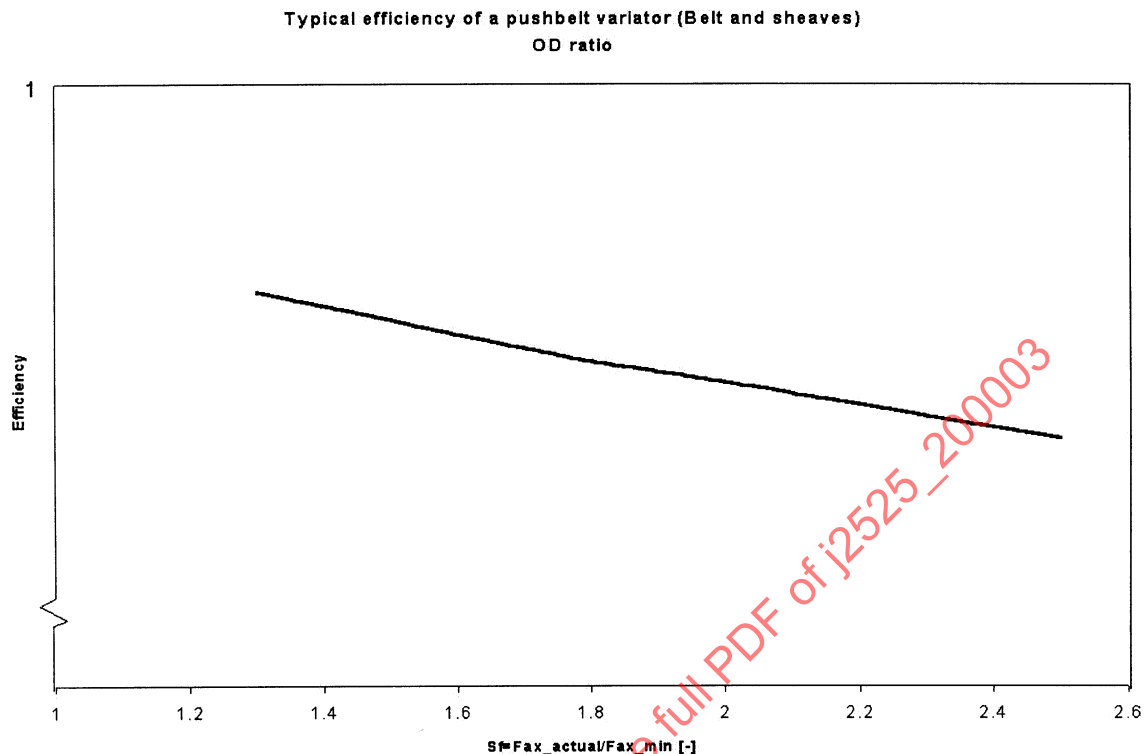


FIGURE 10—COMPRESSION BELT EFFICIENCY VERSUS CLAMP LOAD

- 5.1.6 NOISE MECHANISMS—The belt generates noise as each of the blocks contacts the sheave. The thinner the blocks, the smaller the pitch and the higher the frequency at a given belt speed. With the relative small pitch of the Van Doorne belt (1.5 or 1.8 mm), the audible belt noise only occurs at very low belt speeds.

5.2 Tension Type Belts

- 5.2.1 CONSTRUCTION—Metal tension belts are derived from fixed ratio power transmission chains. All chain type metal belts transfer the torque through a tension difference between the slack and tight sides (similar to a power transmission chain). Torque is transferred from a tension difference between two chain strands. Links and rocker joint pins, similar to those used in fixed ratio chains, are used to transmit the chain tension. The main difference between classes of metal chain belts is the manner in which they engage the sheaves. One class of chains contacts the sheaves through extended rocker joints. The other contacts the sheaves through a separate member (strut) which either envelops the link stack or is placed below the link stack. In both cases, the contacting member transfers the radial component of the sheave apply force to the links, developing a sufficient tension in the chain strands for torque transmittal. Figure 11 shows the Borg Warner type of tension type belt construction and Figure 12 shows the belt construction from Luk.

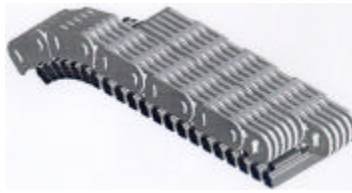


FIGURE 11—BORG WARNER TYPE BELT

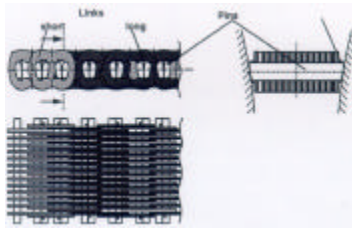


FIGURE 12—LUK TYPE BELT

- 5.2.2 **BELT LENGTH AND PITCH LINE DEFINITION**—Belt length is measured at the articulation points of the rocker joint belt. Belt pitch line changes slightly with articulation (due to the chordal compensation effect of rocker joint belts) and must be considered for small pitch radii.

Similarly the pitch line of the chain is the center of the articulation axis of the rocker joint. This is often displaced from the point of contact of the sheave-contacting member.

- 5.2.3 **BELT CAPACITY**—The tension belt capacity is limited by the maximum tension allowed in the chain and, unlike the compression belt, is relatively unaffected by articulation radius. Tension in the chain is created by the apply pressure, torque, and centrifugal effects. Therefore, the torque carrying capability is determined by this combination of variables.
- 5.2.4 **POWER LOSS MECHANISMS**—Power losses in chain belt CVT's originate from two sources; chain articulation losses and strut/sheave relative motion. Typically, chain articulation losses are quite small (rocker joint chains have been shown to have efficiencies of 99.0 to 99.5%) over the full range of articulations, speeds, and loads.

The predominant source of losses is belt sheave relative motion. This motion occurs due to deflection of the sheaves (and to a lesser degree compression of the pins/struts) and entrance and exit scrubbing due to the pitch length of the belt. These losses can be minimized through the design of sufficiently stiff sheaves and minimizing over-clamping.

- 5.2.5 **FAILURE MECHANISMS**—The tension belt is designed such that the primary failure mode is fatigue of the link plates. The elements that contact the sheaves (pin or strut) are a secondary failure site. Fatigue in the link plates is generally well understood from the widespread implementation of chains for fixed ratio drives. In general, the fatigue stress in the link plates is primarily due to the tension of the belt and to a lesser degree, due to articulation. Belt tension arises from the apply pressure (required to transmit torque), the transmitted torque, and the tension arising from centrifugal effects. Thus the maximum rating of the belt is dependent upon the induced stresses arising from sheave apply pressure, torque and speed. Given the fatigue strength of the belt, an appropriate operating envelope can be constructed.

Most chain belts utilize a rocker type chain that typically does not limit the maximum belt speed. Instead, centrifugally-generated tension and the desired transmitted power primarily limit maximum belt speed. In reality, belt speeds are practically limited at about 50 m/s, since speeds above this significantly reduce the torque carrying capability of the belt.

- 5.2.6 NOISE MECHANISMS—Noise is a major issue with tension belts of all types. The noise produced is a pure tone (or whistle) at the pitch length passing frequency. The primary source of noise is believed to be forced vibration resulting from chordal action due to the two tight strands and the pitch of the belt. The noise produced by the belt (without some modification) is sufficient to render it commercially unacceptable for most automotive applications.

Several techniques are utilized to reduce the noise of the belt. Randomization can be added to the belt to reduce the pure tone of the belt. Randomization can be achieved through variation of the pitch length or by variation of the strut/pin width. Randomization is reasonably effective in reducing the pitch noise of the belt, however most applications require further noise reduction techniques.

Belt pitch frequency can also be reduced through belt phasing. This technique, originally developed for fixed ratio chains has been applied to CVT belts as well. This technique utilizes two chains offset by 1/2 pitch. The pitch offset significantly reduces the pitch passing frequency while not increasing other harmonics.

Belt transverse free strand resonance can also be an issue at low speeds and high torques. These resonances can be controlled by modulation of the secondary piston supply pressure.

6. **Sheave Design**—This section will cover the materials and construction of the secondary and primary fixed and active sheave sets which are used to clamp the belt and provide ratio changing.

6.1 Sheave Construction

- 6.1.1 SHEAVE/PISTON CONSTRUCTION—Correct design of the sheave/piston components is critical for optimum performance of a metal belt CVT. The function of the sheave/piston assembly is to provide appropriate axial force under a large operating range of the application, proper metallurgical surface to provide the frictional transfer of torque and to minimize deviations from a tangential path.

- 6.1.2 PISTON SIZING—At steady-state conditions throughout the operating range, the equilibrium force requirements of the primary sheave are nearly always higher than that of the secondary sheave; typically, the secondary force requirements are monitored and controlled to prevent belt slip. The primary piston force requirements, which in overdrive ratios can be nearly twice that of the secondary servo, are typically sized to be approximately twice that of the secondary servo. This allows the use of a common line pressure system with the primary pressure being less than the secondary servo pressure. Some systems use independent pressure control which allows the use of similarly sized piston areas on both primary and secondary at the expense of controls system complexity.

Although it is desirable to use large diameter servos to keep system pressures low, the centrifugal effects of the oil at high speed (particularly associated with the secondary servo) can be significant and must be considered. Typically, a balance dam on the secondary servo is required to compensate for the centrifugally generated forces. Inadequate balance can result in higher forces than required or the inability to maintain desired ratios at high speed.

- 6.1.3 SHEAVE SPLINE—The movable pulley must transfer torque to the shaft assembly. The requirements of the connection between the moveable sheave and shaft are low backlash, high torsional stiffness and low friction. This is usually done through a ball spline, however, straight splines and other methods have been tested.

- 6.1.4 **SHEAVE STIFFNESS**—For maximum efficiency, it is desirable for the belt/sheave relative motion (while the belt is engaged in the sheave) to be minimized. In actual practice, due to sheave and shaft deflection, the belt travels radially while engaged in the sheaves. The motion of the sheave deflection requires work and thus in an improperly designed system can produce measurable losses. Therefore, it is desirable to keep radial position variation, and thus the losses, to a minimum.

Four factors contribute to the stiffness of the belt/sheave mechanism. First, belt stiffness, unlike elastomeric belts, is usually quite high and belt stretching can be neglected. Second, movable sheave clearance between the sheave and shaft causes a nutating motion of the moveable sheave and shaft. This can significantly contribute to the imposed radial motion of the belt on the sheave and must be closely controlled. Third, shaft bending can also be a significant contributor to belt/sheave radial motion. Shaft deflection is usually highest on the secondary sheave since the belt tensions are highest in low ratio when the belt is at the maximum diameter on this sheave. The sheaves act as lever arms on the shafts, causing the shafts to bend in the opposite direction of the belt center pull force. The fourth factor causing deflection of the sheave flanges is the rotating inertia and any lack of stiffness of the sheaves themselves. Ideally, the sheaves are tapered to minimize rotating inertia associated with mass at the outside diameter of the sheaves and to maximize stiffness.

- 6.1.5 **SHEAVE STOPS**—Extremes of ratio are controlled by mechanical axial stops on the sheaves. Since the secondary sheave is normally designated for controlling axial force, the system is designed so that both extremes of ratio are controlled by stops on the primary sheave.
- 6.1.6 **HYDRAULIC OIL SUPPLY**—Typically, oil is supplied to the sheave servos via center drilled passages of sufficient cross section as to not restrict the flow. No special seal arrangement between the sheave and shaft is required due to the small clearance required between the sheave and shaft for control of the radial movement of the belt. Seals are generally required at the servo outside diameter.
- 6.1.7 **ROTATIONAL BALANCE**—Due to the high speeds and large rotational inertia, the dynamic balance of the sheave assemblies should be considered.

6.2 Material Selection

- 6.2.1 **SHEAVE METALLURGY**—Sheaves used for metal belt CVT's are typically SAE 8620, Rc 60-62 with a 1.0 to 1.5 mm case depth. The choice of material for the sheaves must strike a balance between formability, machinability, durability, and cost. The stress level at which the sheaves operate and their shape have led to the forging of the current production sheaves. Depending on the selection of material and level of dimensional accuracy required, the forgings can be hot-, warm-, or cold-formed.

The type of forging used to form the sheaves is determined by the material, cost, and function. Basically, all carbon levels of steel can be hot forged, whereas warm and cold forgings are limited by the carbon level in the steel. Because of its large diameter, the sheave pulley requires either a hot or warm upset. The level of precision is better for cold forgings than for hot forgings. Therefore, it is possible to get more nearly net surfaces with cold forgings. Another design aspect to consider is to leave surfaces "as forged" wherever possible.

Alloying elements are used to increase the hardenability, toughness, and strength of steel. In order to minimize wear on the sheave face, the surface should be hardened to approximately Rc 60 and the case depth should be a minimum of 1.3 mm. The general design of the sheaves can influence the heat treatment response of the sheave throughout the cross-section. Because the sheaves typically have a lot of mass, it is difficult to maintain a consistent surface hardness pattern and still have the desired core hardness.

Factors affecting the fatigue life characteristics have their origins in the steel-making practice, forging process, heat treatment, and final surface finishing procedures. In steel making, the use of clean steel practices with limits on allowable oxygen content is important. Similarly, the inclusion content and their morphology need to be carefully controlled. Forging design needs to be reviewed for proper grain flow. When carburizing is used as the heat-treating process, the amount of retained austenite needs to be specified. Finally, surface finish on the finished sheaves has a significant impact on wear and has to be controlled through process monitoring, tool change frequencies, and measurements of surface texture.

- 6.3 Transmission Case Design**—CVT transmission case design is very important for proper function and durability of the variator system. The transmission case must support and locate the sheave and shaft assemblies with a high degree of accuracy during all loading conditions. Transmission case materials, optimized case structure, and location and tolerancing of the bearing supports in the transmission case will assure appropriate belt alignment for proper belt loading and durability.

Variator support bearings must be selected to minimize axial and radial freeplay. The amount of freeplay in the bearings under loaded and unloaded conditions will directly affect the positioning of the sheaves and belt misalignment. The belt manufacturer will specify the allowable misalignment in the variator system under all loading conditions.

- 7. Variator System Considerations**—This section will describe the considerations which must be taken into account when designing and packaging the variator system for a metal compression or metal tension type CVT belt. The variator system is the combination of the primary sheave set, the secondary sheave set, and the belt.

- 7.1 Packaging of the Variator System in the Transmission**—The major factors to be considered when packaging the CVT variator system in the transmission are:

- Ratio Range
- Method of Oil Feed
- Belt Type
- Input Torque and Speed
- Manufacturing and Service
- Control System

The factors mentioned previously affect many other packaging details including: sheave travel and length, centrifugal balance chambers, applied piston area and diameter, the use of single or double applied pistons, belt alignment, inside and outside sheave diameters, shaft diameters, center distance between the primary and secondary shafts, and packaging clearance to the belt. Because these items are interrelated, they must be studied as a system. Tradeoffs may be necessary, depending on the packaging constraints and/or desired performance.

- 7.1.1 RATIO RANGE**—The ratio range will affect both the radial and axial packaging of the variator system. Because of the working angle/profile of the sheave assemblies, at least one of the sheave sides must move axially to achieve the desired ratio. A larger ratio range or larger minimum wrap radius will increase the total amount of sheave travel. The location and method of the travel stops for the variators must be considered. The sheave stops will ultimately affect the axial package of the sheave assembly and the overall transmission length.

An increase in sheave travel will result in an increase in belt misalignment by geometry. When using a fixed length belt and flat angled sheaves, misalignment will occur and can be tolerated but should be minimized. The geometric belt misalignment can be minimized by placing the movable sheave halves on opposite sides of their respective shafts. Geometric sheave misalignment can be minimized by changing the sheave geometry (crowned sheaves). Control of axial tolerances in the variator system will minimize the extent of misalignment beyond the geometric sheave misalignment range.

A larger ratio range will increase the radial dimension of the variator system as the outside diameter of the sheave becomes larger (for a given belt type, torque, and speed level). A larger ratio range can also be achieved by decreasing the minimum sheave radius and increasing the shaft outside diameter within constraints of the belt. For a given torque and speed level, the belt manufacturer will specify a minimum belt wrap radius and the required clearance to the shaft from the belt pitch radius. A higher torque and speed level will typically require an increase in the minimum pitch radius of the belt to decrease the stress on the belt and increase the strength and stiffness of the shaft. Therefore, if a specific ratio range is required, the center distance may be increased within the constraints recommended by the belt supplier for belt speeds and loads.

7.1.2 METHOD OF OIL FEED—The method and number of oil feeds to the applied pistons, balance dams, bearings, etc., will affect both the radial and axial packaging in the variator system. The volume and amount of fluid flow to the variator applied pistons and balance chambers will depend on their size and the requirements for the rate of ratio change. The length of the sheave pilot diameters will also affect axial packaging. The sheave pilot length will depend on the shaft diameter and applied chamber maximum pressures. Variators with larger shafts and higher pressures will typically require longer pilot diameters and must be considered when routing the oil passages.

7.1.3 BELT TYPE—Belt types vary depending on manufacturer and application. The belt types considered in automotive applications are the metal compression belt and the metal tension belt. Allowable belt misalignment may vary depending on the belt type used. The belt type used will also dictate the sheave angle/profile and overall belt width, which will affect axial packaging of the variator system. Sheave deflection will also contribute to belt misalignment. Sheave deflections under all operating conditions should be kept to a minimum. Refer to the belt manufacturer's guidelines for misalignment, sheave deflection allowances, and sheave angle/profile information.

The belt manufacturer will specify an outside radial running clearance from the belt to transmission components outside of the variator system (transmission case, fluid feed tubes, etc).

In most cases, the sheave may be designed such that the top of the belt runs outside of the sheave when the belt is at the maximum pitch radius. The minimum sheave diameter will be specified by the belt manufacturer.

The dimensional tolerances and wear of the variator system must be considered to allow adequate radial clearance between the belt and other components over the life of the transmission.

7.1.4 INPUT TORQUE AND SPEED—The clamping forces required must take into account the maximum input torques and speeds, including the type of launch device to be used, i.e., torque converter, starting clutch, etc.

As the torque level increases, the force required to clamp the belt must also increase to avoid slippage. A smaller belt running radius will also require an increase in the clamp force on the belt. For a given line pressure, the variator applied piston area must become larger to increase the clamp force on the belt. The larger applied piston area will make the overall diameter of the piston cylinder larger, increasing the overall radial package of the variator system. As the applied piston area becomes larger, centrifugal effects of the fluid must be taken into account or compensated for. If a balance chamber is utilized to compensate for the centrifugal effects of the fluid, axial packaging of the variator will be negatively affected. The radial package of the applied chamber can be decreased by utilizing a dual cylinder design. A dual cylinder design uses two applied chamber areas in tandem to provide adequate clamp force on the belt. However, the dual piston design will require an increase in axial packaging space.

- 7.1.5 **MANUFACTURING AND SERVICE**—Transmission design must always take into consideration the method of manufacturing and the service requirements of the transmission. Ultimately, these requirements will affect the packaging of the variator system. Typically, the variator system must be pre-assembled prior to installation into the transmission.

Servicing of the transmission in the field must also be considered. Radial and axial clearance envelopes may need to be made larger than required for transmission function due to considerations for adequate tool clearance or general serviceability.

- 7.1.6 **CONTROL SYSTEM**—The type of control system and basic architecture of the variator system will affect both radial and axial packaging. Refer to Section 8: Controls Design for detailed information on the different types of controls and variator construction.

- 7.2 **Torque Capacity**—To date, automotive v-belt CVT's have been limited to small displacement engines (2.5 L and below). The reason is that current belt technologies have limited power ranges at the durability cycles currently required for automotive applications, 150 000 miles and beyond. There are examples of v-belt CVT's used successfully in high-power application, such as the Van Doorne equipped formula-one race car in Europe, but this had a durability cycle of only a few hours. A proper understanding of the forces and duty cycles were is required to select a v-belt.

The torque capacity of the variator involves belt/sheave geometry, coefficient of traction between the belt and sheave and sheave axial clamping forces. Torque is transferred from the one sheave assembly to another via forces within the belt. These belt forces are also related to the variator torque capacity. The torque capacity factors are outlined as follows.

- 7.2.1 **BELT LENGTH**—The effective pitch length of a variator belt is a geometric function (Figure 13) of the center distance, a particular variator ratio, and the pitch diameters at a particular variator ratio. Once these parameters are chosen, the effective length is as follows:

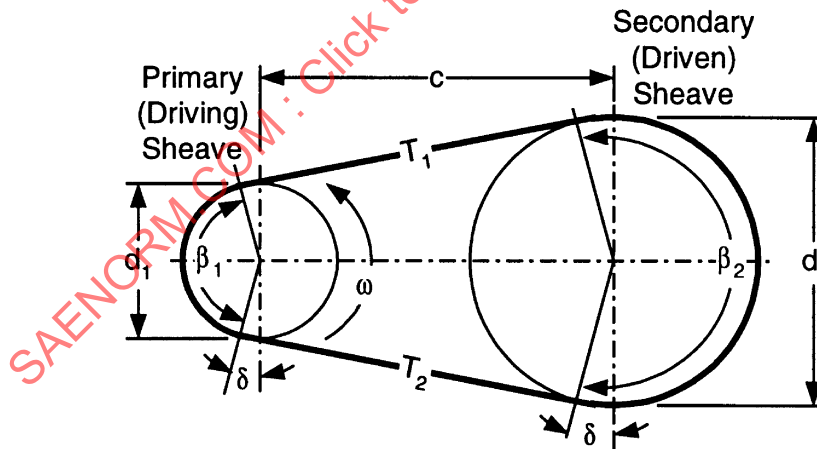


FIGURE 13—BELT LENGTH DIAGRAM

$$\text{Variator Ratio: } i = \frac{d_2}{d_1} \quad (\text{Eq. 1})$$

$$\text{Included Angle: } \sin(\delta) = \frac{d_1}{2c}(i - 1) \quad (\text{Eq. 2})$$

$$\begin{aligned}\text{Wrap Angles: } \beta_1 &= \pi - 2\delta \\ \beta_2 &= \pi + 2\delta\end{aligned}\quad (\text{Eq. 3})$$

$$\text{Effective Length: } l = 2c \cos(\delta) + \frac{d_1}{2} [\pi - 2\delta + i(\pi + 2\delta)] \quad (\text{Eq. 4})$$

7.2.2 COEFFICIENT OF TRACTION—Since the v-belt CVT is a traction device, there must be proper understanding of the relationship between geometry and the coefficient of traction; the coefficient of traction is a function of material (both sheave and belt) and fluid. To prevent self-locking, the sheave angle (Figure 14) must always satisfy the following relation:

$$\tan(\alpha) > \mu \quad (\text{Eq. 5})$$

where:

μ = coefficient of traction of belt to sheave

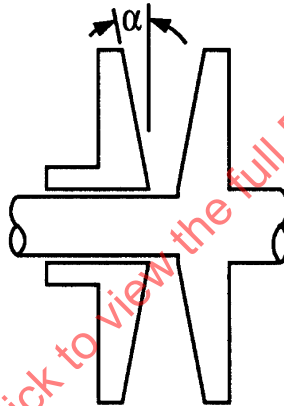


FIGURE 14—SHEAVE ANGLE

For a given belt design, care must be taken to know the actual coefficient of traction for each individual fluid being used. Refer to the belt manufacturer for recommended fluids and required sheave angles/profile. Typical angles for steel belts are 11 degrees.

7.2.3 AXIAL CLAMPING FORCE

7.2.3.1 *Driven Pulley*—The approximate clamping force required to hold the secondary (driven) sheave in equilibrium is derived by assuming that the frictional force between the variator belt and the sheave only acts tangentially to the pitch circle. With this simplifying assumption, and remembering that the movable sheave only sees half the torque, then, from Figure 15:

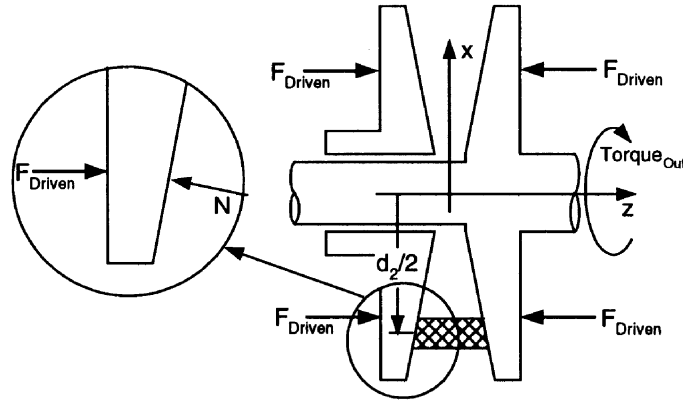


FIGURE 15—CLAMPING FORCES

$$\sum F_z = F_{\text{Driven}} - N \cos(\alpha) = 0 \quad (\text{Eq. 6})$$

$$\sum M_z = \mu N \frac{d_2}{2} - \frac{\text{Torque}_{\text{Out}}}{\mu d_2} [\cos(\alpha)] \quad (\text{Eq. 7})$$

Since N is not constant around the wrap angle, we eliminate this by solving Equation 7 for N and substituting into Equation 6. Thus;

$$F_{\text{Driven}} \approx \frac{\text{Torque}_{\text{Out}}}{\mu d_2} [\cos(\alpha)] \quad (\text{Eq. 8})$$

Equation 8 now predicts the clamping force required on the secondary movable sheave at the threshold of slip between the variator belt and the sheave. It is recommended that a Safety-Factor be applied to this equation, since under-clamping would result in gross belt slippage and failure. A typical safety-factor for metal v-belts is in the range of 1.3. Consult the belt manufacturer for their recommended Safety-Factor.

7.2.3.2 Drive Pulley—It would seem that Equation 8 could be applied to the behavior of the primary (driving) sheave, however all previous experimental work has shown this to be untrue. A good analytical equation for predicting the behavior of the primary sheave has yet to be formulated; thus today empirical methods are often used.

The clamping force required on the primary sheave to hold a given ratio is a function of the ratio, the actual clamping force applied to the secondary sheave, the input torque, and the actual safety-factor.

Figure 16 is an example of a typical graph obtained from belt manufacturers test data. As can be seen, over-clamping of the secondary pulley with different safety-factors changes the equilibrium force required on the primary sheave.

The Force Ratio in the graph is defined as shown in Equation 9:

$$\text{Force Ratio} = \frac{\text{Force}_{\text{Drive}}}{\text{Force}_{\text{Driven}}} \quad (\text{Eq. 9})$$

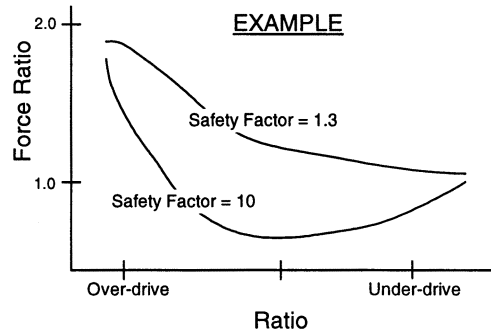


FIGURE 16—SAFETY FACTOR

7.2.4 BELT TENSION FOR PULL BELTS—Torque is transferred from the sheaves to the variator belt (and visa versa) through frictional forces developed between the belt and the contacting surfaces. These frictional forces are generated through axial clamping forces on the sheaves and, in general, are balanced through tension forces in the belt. It is these frictional forces that cause a tension gradient to occur as the variator belt travels around the sheave.

The established theory of belt drive mechanics separates the belt's wrap angle into two zones; the active arc and the inactive arc. It is within the active arc that the frictional forces act, causing belt radial contraction, or expansion. From this understanding, it is possible to formulate an equation to predict the tensions within in the variator belt.

For an infinitesimal belt element on the secondary sheave of Figure 15 and shown in Figure 17, assume that the belt is operating under equilibrium conditions (constant speed and torque), that the active arc is equal to the wrap angle, and that within the wrap angle, the pitch radius of the belt remains constant. Thus, to solve for the equilibrium belt tension ratio, it is necessary to solve the following vector equation (Equation 10) (which for convenience will be in cylindrical coordinates);

$$\sum \vec{F} = m\vec{a} \quad \text{or} \quad \begin{cases} \sum F_r = ma_r \\ \sum F_\theta = ma_\theta \\ \sum F_z = ma_z \end{cases} \quad \text{with} \quad \begin{cases} a_r = \ddot{r} - r\omega^2 = -r\omega^2 \\ a_\theta = r\dot{\omega} + 2\dot{r}\omega = 0 \\ a_z = \ddot{z} = 0 \end{cases} \quad (\text{Eq. 10})$$

Applying Equation 10 to the free body diagram (Figure 17) of the infinitesimal belt segment of length ds and mass dm : with $dm = \rho ds = \rho r d\theta$ (ρ = mass/unit length); and assuming that the resultant friction angle remains constant over the pulley arc:

$$\sum F_r = 2dN \sin(\alpha) + 2dF_f \sin(\gamma) \cos(\alpha) - T \sin\left(\frac{d\theta}{2}\right) - (T + dT) \sin\left(\frac{d\theta}{2}\right) = \rho r d\theta (-r\omega^2) \quad (\text{Eq. 11})$$

$$\sum F_\theta = (T + dT) \cos\left(\frac{d\theta}{2}\right) - T \cos\left(\frac{d\theta}{2}\right) - 2dF_f \cos(\gamma) = 0$$

$$\sum F_z = dN \cos(\alpha) - dF_f \sin(\gamma) \sin(\alpha) - dN \cos(\alpha) + dF_f \sin(\gamma) \sin(\alpha) = 0$$

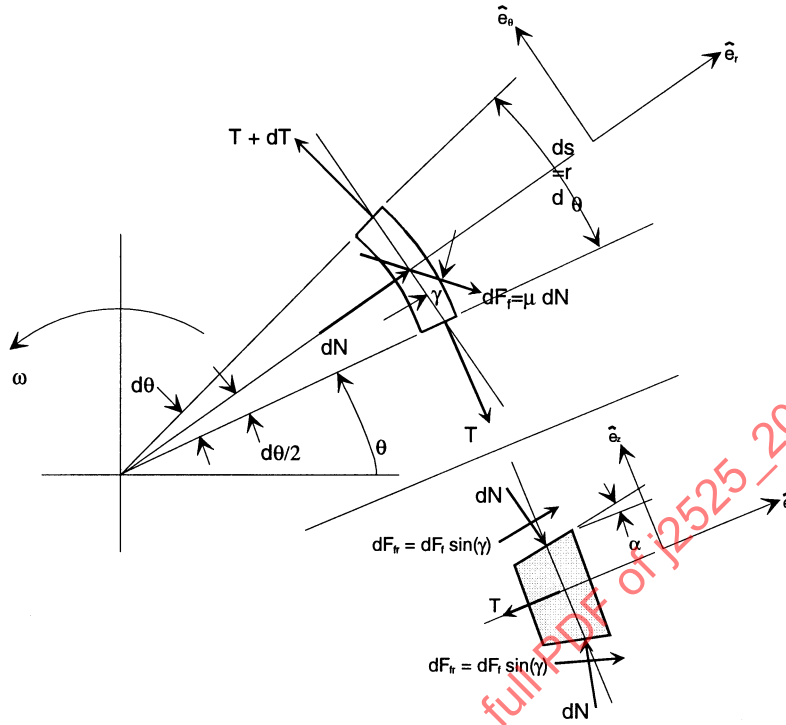


FIGURE 17—BELT VECTORS

Substituting $dF_f = \mu dN$, $\sin\left(\frac{d\theta}{2}\right) \approx \frac{d\theta}{2}$, $\cos\left(\frac{d\theta}{2}\right) \approx 1$ (since $d\theta$ is of infinitesimal size), and neglecting all order 2 and previous terms (see Equation 12):

$$\begin{aligned} 2dN[\sin(\alpha) + \mu \sin(\gamma) \cos(\alpha)] &= [T - \rho(r\omega)^2]d\theta \\ dT &= 2\mu dN \cos(\gamma) \end{aligned} \quad (\text{Eq. 12})$$

Eliminating dN and collecting like terms yields (see Equation 13):

$$\frac{dT}{[T - \rho(r\omega)^2]} = \frac{\mu \cos(\gamma)}{[\sin(\alpha) + \mu \sin(\gamma) \cos(\alpha)]} \quad (\text{Eq. 13})$$

Integrating this over the pulley wrap angle yields (see Equation 14):

$$\ln \frac{T_1 - \rho(r\omega)^2}{T_2 - \rho(r\omega)^2} = \frac{\mu \cos(\gamma)}{[\sin(\alpha) + \mu \sin(\gamma) \cos(\alpha)]} \quad (\text{Eq. 14})$$

7.2.4.1 *Torque Relation*—In order to solve Equation 14, an additional equation relating the tensions is required. For a purely tension belt, torque is transmitted through tension differences in the belt. From Figure 17:

$$\text{OutputTorque}^* = (T_1 - T_2) \frac{d_2}{2} \quad (\text{Eq. 15})$$

where:

$$T_1 > T_2$$

$$\text{OutputTorque}^* = \text{OutputTorque}_{\text{Actual}} \times \text{Safety - Factor}$$

Solving Equations 14 and 15 simultaneously, and assuming the acting friction angle is constant and equal to zero leads to Equation 16:

$$T_1 \approx \frac{\text{OutputTorque}}{r_2 \left(1 - e^{\frac{-\mu\beta}{\sin(\alpha)}} \right)} \quad (\text{Eq. 16})$$

$$T_2 \approx T_1 - \frac{\text{OutputTorque}^*}{r_2} \quad (\text{Eq. 17})$$

7.2.4.1.1 *Center Pull Force*—The center pull force induced upon the sheave support bearings is created by the tensile forces in the belt. Since the centrifugal component of the tensile force does not add to this, the center pull (CP) is shown in Equation 18:

$$\text{CP} = (T_1 + T_2) \cos(\delta) - 2\rho(r_2\omega)^2 \cos(\delta) \quad (\text{Eq. 18})$$

7.2.4.1.2 *Clamping Force Mechanisms*—The clamping force, which is applied to the movable sheave, can be accomplished by mechanical or hydraulic means. The mechanical means is typical for snowmobiles and other small recreational vehicles. It consists of a torque-sensitive cam and spring mechanism for the driven sheave and a speed-sensitive flywheel and spring mechanism for the driving sheave. This method of applying the clamping force will not be covered in this paper for it is not used in current automotive applications. For additional details, it is suggested that the reader investigate other sources such as SAE 720708 and SAE 730003.

7.2.4.1.3 *Hydraulic*—The basic principle of the hydraulic means to applying and varying the clamping force is to apply hydraulic pressure to the back of the movable sheave. The pressure required is thus shown in Equation 19:

$$\text{Pressure} = \frac{\text{Force}_{\text{Required}}}{\text{Piston Area}} \quad (\text{Eq. 19})$$

With the sheaves rotating, the pressure required is composed of the static supply pressure plus the rotational dynamic pressure. The rotational dynamic pressure can be determined from Equation 20:

$$\text{Pressure}_{\text{Dynamic}} = \frac{1}{4} \rho_{\text{Fluid}} \omega^2 (r_0^2 + r_1^2 - 2r_{\text{free}}^2) \quad (\text{Eq. 20})$$

where:

- r_0 = Piston outer radius
- r_1 = Piston inner radius
- ρ_{Fluid} = Density of fluid
- r_{free} = Free surface radius of rotating fluid

The combination of the static supply pressure plus the rotational dynamic pressure make up the total pressure required. Compensation for the dynamic pressure must be made in order to not over-stress the belt. Balance dams are often used in the sheave systems to eliminate large portions of the dynamic pressure.

7.3 Belt Alignment—The sheaves should be aligned so the belt can move as a solid column from one sheave to the other with a minimum of twisting of the belt. Alignment is defined as the lateral dimension from one fixed sheave to the other fixed sheave at the pitch point of the belt. Any difference between this dimension and the width of the belt at the pitch point is defined as misalignment. This will cause the belt to bend in order to enter the sheave. The amount of misalignment changes as a function of ratio due to the geometry of the fixed belt length and the sheaves. It is desirable to operate in areas with minimum misalignment.

Belt alignment for tension style belts is in general less critical than push style belts, since the torque transfer does not rely on a column to transmit power. CVT chain belts can accommodate reasonable misalignment.

Belt misalignment (as defined by Figure 18) is an invariable geometric consequence of the variator system. Though this statement is true, it is necessary to minimize the extremes. There are two ratio points in which the belt operates with zero misalignment. It is the designer's choice to determine at which two ratios this will occur. Once a ratio of zero misalignment is chosen, determining the misalignment values is as follows:

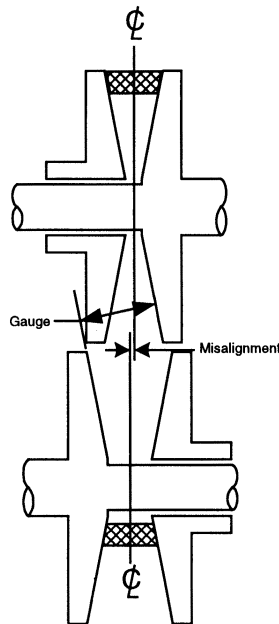


FIGURE 18—BELT ALIGNMENT

Using the ratio value of zero misalignment, solve Equation 2 and 4 simultaneously to obtain d'_1 ; then use Equation 1 to obtain d'_2 . Now for every other ratio, repeat the process to determine the other values of d_1 (i) and d_2 (i). The misalignment is thus shown in Equation 21:

$$\text{Misalignment}(i) = \left[\frac{d_1(i) - d'_1}{2} \right] \tan(\alpha) - \left[\frac{d'_2 - d_2(i)}{2} \right] \tan(\alpha) \quad (\text{Eq. 21})$$

Figure 19 shows typical values:

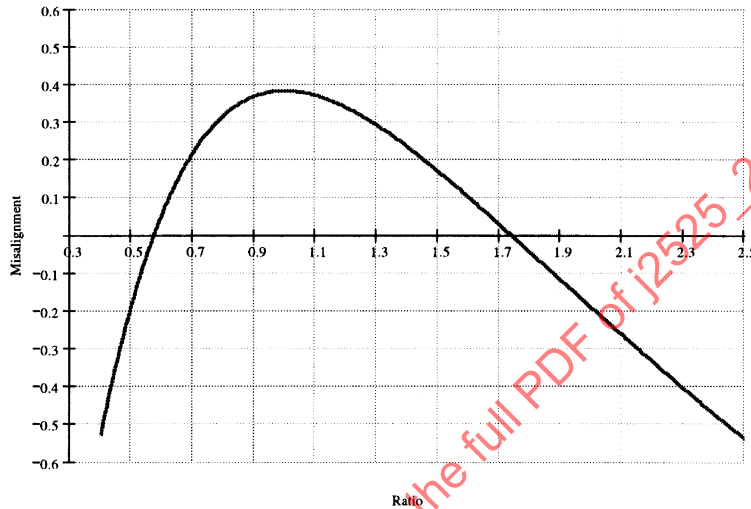


FIGURE 19—BELT ALIGNMENT GRAPH

Determining the gauge dimension which is used to set the alignment is as follows in Equation 22:

$$\text{Gauge} = \left[\left(c - \frac{d'_1}{2} - \frac{d'_2}{2} \right) \tan(\alpha) + w \right] \cos(\alpha) \quad (\text{Eq. 22})$$

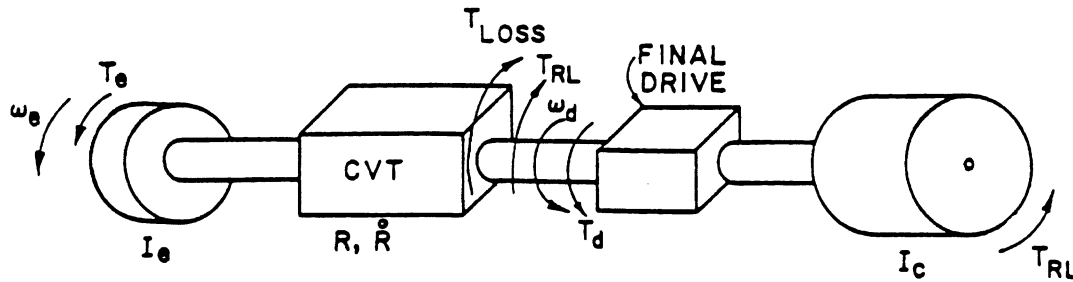
where

w = width of the belt at the pitch line.

Factors which influence misalignment are tolerances in the gauge, tolerances in the center distance stack, deflection of the sheaves under load, and thermal effects of the case supporting structure (center distance changes and axial clearance (end-float) changes).

7.4 Ratio Changing—The physical aspects of variator ratio changing must be understood in order to design control systems and ensure hardware life. The ratio and the rate of change of ratio both affect vehicle dynamics. Maintaining a constant or steady-state ratio requires a balance between the primary and secondary belt clamping forces. Changing ratio involves deviating from this equilibrium balance. There are limits to this deviation, beyond which the hardware is damaged.

The governing equation of a CVT-equipped vehicle is shown in Figure 20. From this equation, it can be seen that the vehicle acceleration depends upon both the ratio and the rate of change of ratio. It is important to note that the rate of change of ratio term is preceded by a negative sign. This sign can become important during a downshift when vehicle acceleration is desired. Too much change in ratio during the downshift can result in an initial deceleration instead of an acceleration.



Ref. SAE 840048

FIGURE 20—DRIVE-TRAIN SCHEMATIC AND DYNAMIC EQUATION

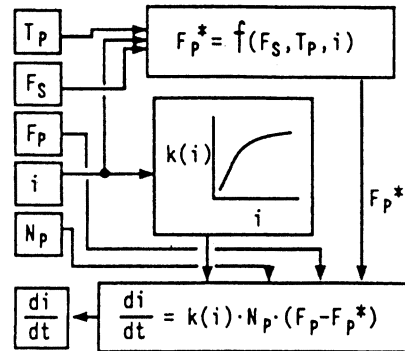
$$\dot{\omega}_d = \frac{-\dot{R}I_e\omega_e}{I_eR^2 + I_{cd}} + \frac{T_eR - T_{RL} - T_{Loss}}{I_eR^2 + I_{cd}} \quad (\text{Eq. 23})$$

where:

- I_{cd} = Vehicle inertia + CVT output inertia (reflected at the driveshaft)
- I_e = Engine Inertia + CVT input inertia (reflected at input shaft to CVT)
- R = CVT ratio (ω_e/ω_d)
- $\dot{R}$ = Ratio rate
- T_e = Torque generated in engine
- T_{Loss} = Torque loss in CVT evaluated at drive shaft
- T_{RL} = Road load torque evaluated at drive shaft
- $\dot{\omega}_d$ = Driveshaft rate of change of speed
- ω_e = Engine speed

In a conventional automatic transmission, gear ratios are fixed. In a belt type CVT, speed ratios can vary between the Underdrive ratio limit and the Overdrive ratio limit. To hold or maintain a speed ratio in between these limits requires a balance between the primary and secondary belt clamping forces. This equilibrium requirement has been characterized in terms of the primary to secondary force ratio.

Changing the variator ratio involves deviating from the equilibrium force balance. A simple and practical expression for shift speed (di/dt) is shown in Figure 21. This rate of change of ratio is basically proportional to the primary speed and the difference between the actual primary clamping force and the primary clamping force required to hold a constant ratio. As the difference of the actual primary clamping force from the equilibrium force increases, the ratio rate of change increases. Because of this relationship, a simplified view of the variator controls is that the primary sheave determines the speed ratio and the secondary sheave sets the torque capacity.



Ref. JSAE 9636330

FIGURE 21—PARAMETERS INFLUENCING RATIO RATE IN CREEP MODE

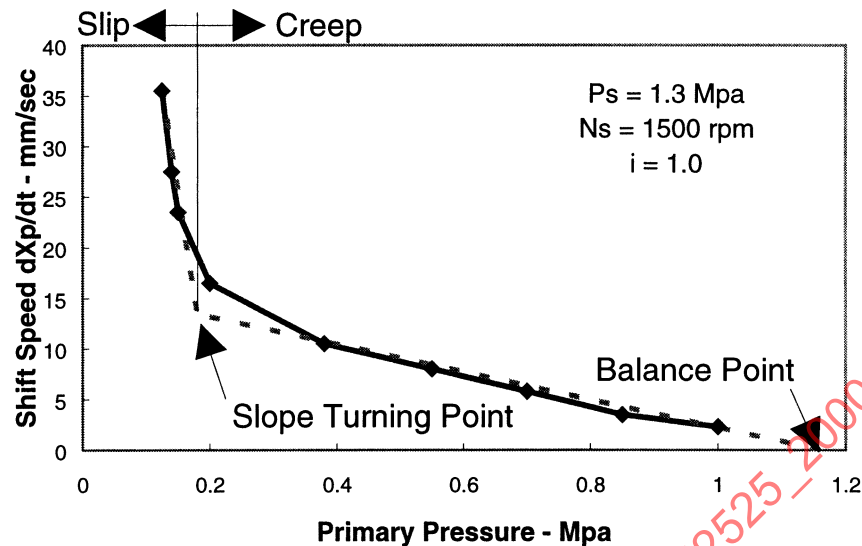
where:

- F_p = Actual Primary clamping Force
- F_p^* = Primary clamping force to hold a constant ratio
- F_s = Secondary clamping force
- i = Pulley ratio
- $k(i)$ = Proportionality constant with pulley ratio
- N_p = Primary rotating speed
- T_p = Primary torque

Unfortunately, there are limits to the rate of change of variator ratio. JSAE Paper 9636330 describes two different phases of variator shifting characteristics: Creep Mode and Slip Mode. These two phases are identified in Figure 22. In creep mode, the shift speed is proportional to the pulley rotating speed. In effect, the belt “walks” up or down the sheave. The empirical expression for shift speed, given previously, is for the creep mode. In the slip mode, the shift speed is independent of the rotating pulley speed. Because slip mode is the result of insufficient clamping forces, damage occurs to the belt and pulleys. For this reason, operation in the slip mode must be avoided. Figure 23 illustrates the belt movement on the primary pulley in both these modes.

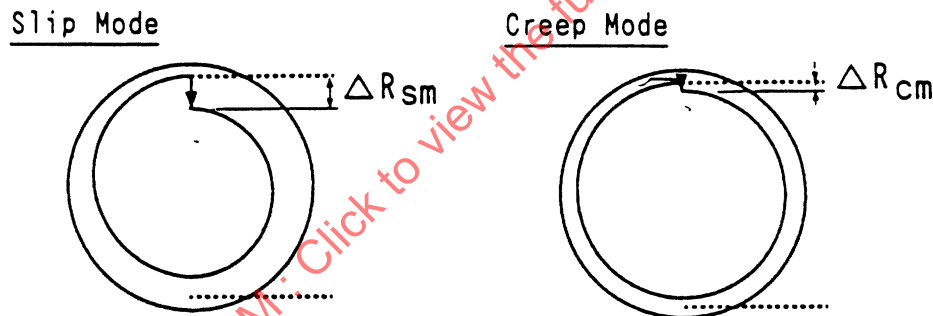
It should be noted that the sizing of the hydraulic pump is critical to preventing belt slip. When the flow demand to a sheave exceeds the ability of the pump to provide flow, the hydraulic pressures in that sheave will drop below the minimum level for clamping and the belt will slip. The pump output flow minus the system demand must always be greater than the flow rate required by the sheaves.

8. **Controls Design**—Before selecting the variator control approach, the functional requirements must first be understood. The type of controls hardware can be selected, depending on the desired calibration flexibility that is desired. The types of automotive controls implemented to date involve hydraulics and a combination of mechanical or electric actuators, where electronic controls provides more calibration flexibility. If electronic actuators are used, software strategy becomes a part of the control system. Further discussion on these topics is provided as follows.
- 8.1 **Control Functional Requirements**—A control system for a belt variator must clamp and schedule the movement of the belt. Belt clamping forces, on both the primary and secondary sheaves, must be sufficient to avoid the slip mode during the ratio change. Excessive clamping should also be avoided, so as to not lower both component life and power transfer efficiency. Belt ratio scheduling involves not only maintaining a ratio during steady-state operation, but also controlling the rate of change of ratio during transient conditions.



Ref. JSAE 9636330

FIGURE 22—CORRELATION BETWEEN PRIMARY PRESSURE AND SHIFT SPEED



Ref. JSAE 9636330

FIGURE 23—BELT ENTRY TRACK ON THE PRIMARY PULLEY

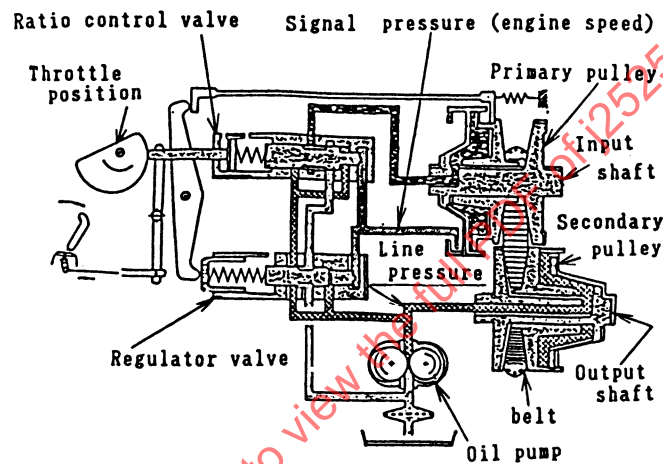
The control system for the variator must provide the following functions:

- Control belt clamping and ratio scheduling continuously.
- Clamp both the primary and secondary sheaves sufficiently to prevent belt slip without excessive clamping at any given input torque and belt ratio.
- Vary the rate of ratio change from zero to values that at least match the driveability of vehicles equipped with conventional stepped ratio automatic transmissions.
- Hold a constant ratio without oscillations.
- Change ratio rapidly without driver perceived overshoot or oscillations.
- Downshift to an acceptable launch ratio during a panic stop before the vehicle comes to rest.
- Provide a "limp home" capability in the event of an electrical malfunction.

For this last requirement, simply defaulting to underdrive is not acceptable because of potential engine damage at high vehicle speeds. Default to overdrive protects the engine, but results in a vehicle that is barely able to launch on level ground. One approach is to freeze the ratio at the time of the malfunction. Another approach is to provide for a backup limp home control system. A third concept is to default to a ratio around 1:1.

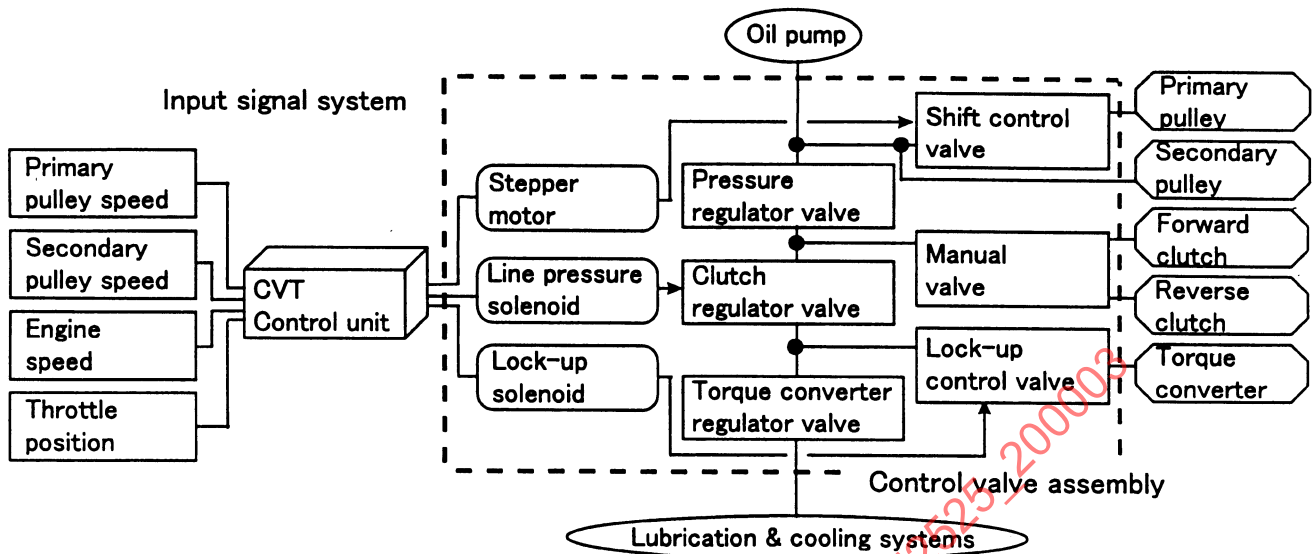
8.2 Types of Controls—A number of different variator control systems have been implemented. These systems have two common characteristics. The belt clamping function has, for most applications, been implemented in an open loop manner. The ratio scheduling has been implemented in a closed loop manner.

Belt clamping control consists of an open loop control system. Input torque to the variator primary sheave is inferred indirectly from engine sensors. Early production CVTs implemented clamping by mechanical and hydraulic devices, as shown in Figure 24. The clamping of the secondary sheave was a function of the engine speed and ratio. The engine speed signal was provided by a pitot tube in a radial, oil filled, cavity of the primary pulley. For these early transmissions, the ratio signal came from one of the moveable sheave positions through a lever arrangement. Later production CVTs implemented clamping by electro-hydraulic devices. One such scheme is shown in Figure 25. A microprocessor generates a torque proportional signal through a variable force solenoid to a hydraulic regulator valve. This signal is based on input signals from throttle position, engine speed, and primary speed electrical sensors.



Ref. SAE 880481

FIGURE 24—EARLY PRODUCTION CVT MECHANICAL—HYDRAULIC CONTROL SYSTEM



Ref. SAE 980823

FIGURE 25—CURRENT PRODUCTION CVT CONTROL SYSTEM

Ratio scheduling control consists of a closed loop control system, using the primary speed as a feedback signal. Referring back to Figure 24, early production designs used a throttle cam to set the desired primary speed and a hydraulic pitot tube, as described previously, for the feedback signal. The latest designs employ speed sensor feedback in conjunction with a microprocessor control module for added flexibility. Referring to Figure 25, Nissan uses a microprocessor-controlled stepper motor to set the desired ratio and a sheave position follower for feedback.

Of the variator control systems proposed to date, the hydraulic valve arrangements all fall into one of three generic types: Master-Slave, Partner Principle, and Independent Coupled. All three systems are illustrated in Figure 26. An integral part of implementing these control systems is the details of sheave apply area geometry.

In the Master-Slave control system, the ratio is set with the Primary sheave (Master) and the torque capacity is set with the Secondary sheave (Slave). Approximately a 1.8 force ratio is required to maintain extreme overdrive ratio. To satisfy the force ratio at extreme overdrive, the Master-Slave system employs a Primary apply chamber with approximately twice the area of the Secondary apply area. At extreme overdrive, the primary pressure is slightly less than the secondary pressure. As the steady-state belt ratio changes toward underdrive, the Primary pressure is a smaller percentage of the Secondary pressure. This system is documented in SAE 880481.

In the Partner Principle approach, both torque capacity and ratio scheduling are set with both sheave assemblies through a 4-way flow/pressure valve. The apply chambers have equal areas. In order to achieve the force ratio for overdrive, the primary apply pressure has to be approximately twice the secondary pressure. At extreme underdrive, the two sheave pressures are nearly equal. This approach is described in JSAE 9636286. A variation of this approach uses two separate apply chambers on both the Primary and Secondary assemblies. One chamber is used for clamping. The other chamber is used for ratio scheduling.

With the Independent Coupled control system, torque capacity is set on both sheave assemblies and ratio is scheduled with the primary sheave assembly. The system also uses the same approximate 2 to 1 area ratio of the Master-Slave system. The required sheave pressures in this system are the same as the Master-Slave system. The difference between this system and the Master-Slave system is that the primary sheave is controlled by pressure instead of flow. The independent coupled is system is describe in more detail in JSAE 9636321.

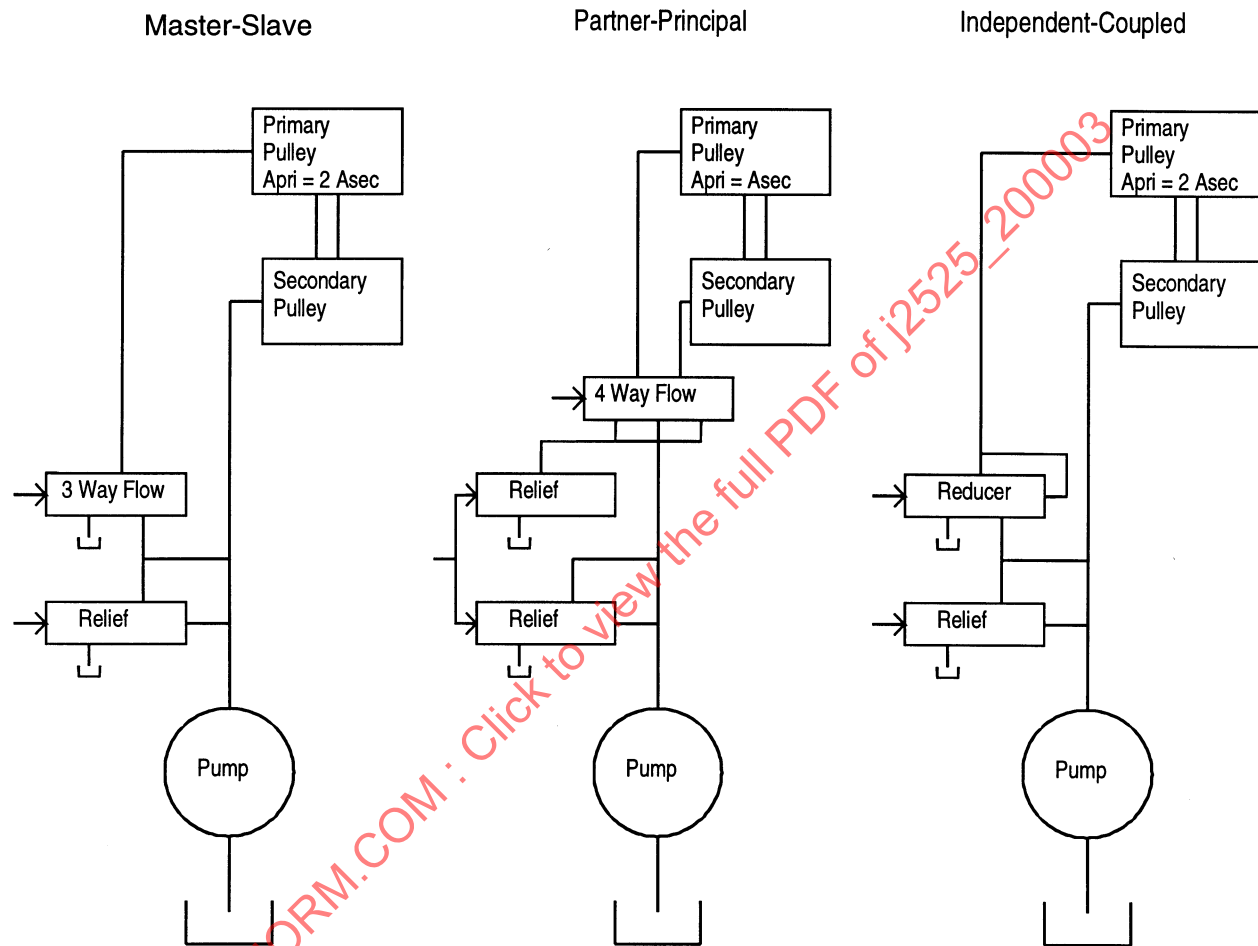


FIGURE 26—TYPES OF VARIATOR CONTROLS

8.3 Strategy Goals—The scheduling aspects of ratio changing affects not only vehicle fuel economy, but also performance and driveability. While this paper will not cover strategy concepts because these are OEM specific, there are a few CVT driving characteristics that must be avoided. SAE 925062 list four CVT driveability concerns that were prevalent with the early production mechanical - hydraulic control systems:

- Lack of acceleration feeling
- The constant engine speed and related sound during accelerations (sometimes referred to as the "Rubber Band Effect")
- Engine speed falling while entering a road curve and rising upon exiting the curve
- Lack of ratio adaptation at the end of the braking process

The goals of the control strategy should be to eliminate or minimize these driveability concerns.

9. **Fluid**—CVT lubrication requirements are in many respects similar to AT requirements, with the added need to lubricate and control friction within the belt and at the belt-sheave (belt-pulley) interface. Also, with the elimination of high energy engaging bands and clutches, clutch durability may be less of an issue in some designs. Since the requirements are similar, automatic transmission fluids (ATFs) have been used successfully in CVT applications. However, it is not yet clear whether a common CVT-ATF fluid is the most desirable approach. CVTs are a new technology and, therefore, there is little information available in the open literature on fluid tests and specifications. Accordingly, this section will draw heavily on methods developed for ATFs. A listing of oil tests and references for this section are contained in Appendix B.

9.1 Power Transmission

- 9.1.1 **BELT TO SHEAVE FRICTION**—High belt-sheave friction is necessary for transmission of high torque while using minimal clamping force. Dynamic (sliding) friction should be equal to or higher than static friction; that is the friction coefficient (μ) versus sliding velocity (V) curve should be positive or at least flat. A negative μ - V curve can cause stick-slip vibration, noise and, potentially, surface damage. Also, periods of excessive torque loading or a decrease in the friction level produced by the fluid could cause incipient slip to escalate to gross slip with catastrophic results. Belt-sheave friction and wear tests have been developed by Van Doorne Transmissie (VDT) and some OEMs. However, none of these tests have been made public at this time. Lubrizol and VDT have published information on an intra-belt noise test (see 2.1.8(1)). A number of friction bench tests have been proposed (see Table 1):

TABLE 1—METAL FRICTION BENCH TESTS

Test Type	Source
Belt element - Ring	Lubrizol, VDT (see 2.1.8 (1))
Belt element - Ring	Exxon (see 2.1.8 (2))
3 Belt Elements - Disk	Exxon (see 2.1.8 (2))
Block - Ring	Nissan (see 2.1.8 (3))

Friction and wear characteristics are not simply intrinsic properties of materials or fluids. They are performance attributes which vary with operating conditions, history, and system aging. Therefore, it is important to select a test method which duplicates the expected operating envelope (component speed, load, temperature, etc.) as closely as possible and to include sufficient system aging time.

- 9.1.2 **PLATE FRICTION AND FRICTION DURABILITY**—Clutch and band tests are designed to ensure adequate static and dynamic friction, a favorable friction-speed relationship and durability in the resulting system. The composition and properties of the friction material and fluid, as well as the design and operating envelope of the component, influence these properties in question. Tests are generally similar from one OEM to another, however details of the test procedures vary considerably, particularly with regard to test severity. This variation reflects the OEM's assessment of an appropriate duty cycle, differences in friction materials and fluids, and competitive factors such as, for example, the OEM's target for service intervals. Variation in clutch test procedures for North American and Japanese OEMs has been documented in an ILSAC report (see 2.1.8 (4)).

Shifting clutch performance is evaluated by measuring dynamic and static torque, differences between static and dynamic torque values, and operating parameters such as stop time. Some differences exist in exactly how static and dynamic torque values are measured. However, Figure 27 will give a general picture. Refer to the test specification for the SAE #2 clutch friction test for more details.

A = Low Speed Dynamic Torque
C = Low Speed Dynamic Torque
E = 0.25 Static Torque

B = Midpoint Torque
D = Max Static Torque

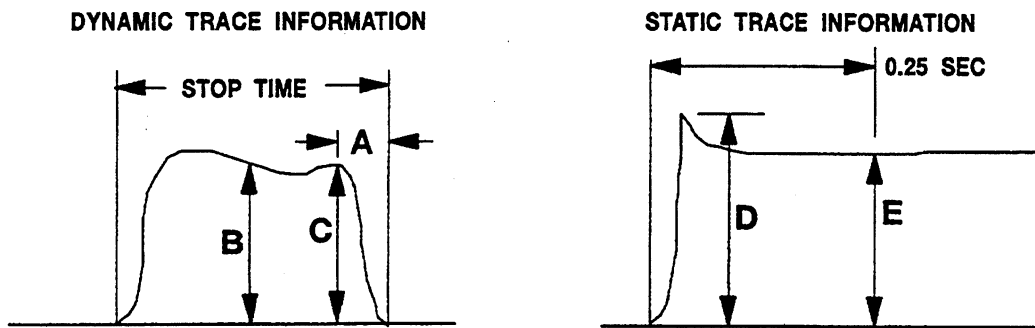


FIGURE 27—CLUTCH FRICTION TESTS

A rise in friction coefficient at the (low speed) end of engagement, or "rooster tail", leads to harsh engagements and is generally avoided in automotive applications. The engagement curve can also be drawn as a friction coefficient versus velocity (V) curve. A "rooster tail" is then seen to be equivalent to a negative μ -V curve.

Torque converter clutches are a relatively recent development. These are of two types: lock-up and modulated (slipping). A negative μ -V curve can lead to harsh engagements in a lock-up clutch. In a slipping clutch, this characteristic can lead to self-generated oscillations or "shudder." At this time, there are no published specification tests. General Motors recently introduced a vehicle dynamometer test that evaluates ATFs for shudder tendency in their modulated torque converter clutches. This test is now included in the DEXRON®-III specification (see 2.1.8 (5)).

9.2 Lubrication

- 9.2.1 **ANTIWEAR**—At Van Doorne Transmissie, oils are qualified using a variator and transmission durability test. However, this test has not been published as an industry test procedure. Since variator wear has so far been a relatively small problem (see 2.1.8 (1)) there is no standard variator wear test procedure. Proposed tests have focused on gear, pump, and general antiwear performance issues. The FZG rig, which employs spur gears and a vane pump are reasonable representations of actual components. The other bench tests are useful in characterizing the antiwear activity of fluids under various conditions, thus assuring good antiwear characteristics, but direct comparisons with CVT or AT hardware issues have not been made public at this time.
- 9.2.2 **METAL CORROSION**—Bench tests used to screen for corrosion protection are listed in Appendix B. Additional information is gathered from inspection of parts which have completed oxidation and cycling tests. The corrosion requirements for CVTs are the same as for ATF.

9.3 Hydraulic Properties

- 9.3.1 **VISCOSITY AND TEMPERATURE EFFECTS**—CVT viscosity requirements are very similar to requirements for an ATF. Typical ATF viscometric test methods and limits are given in Appendix B.
- 9.3.2 **AIR ENTRAINMENT AND FOAM**—The need to minimize air entrainment and foam in the CVT is similar to that in the AT. However, test methods in general use are aimed at measuring foam collapse and do not give an adequate measure of air entrainment, which results from high energy mixing of oil and air. Both processes are discussed in (see 2.1.8 (10)), which also presents a proposed test method for air entrainment.

9.4 Useful Fluid Life

- 9.4.1 OXIDATION STABILITY—Ensuring oxidative stability is important for establishing drain intervals. Oxidation degrades most fluid performance attributes: viscosity increases, dispersancy (the ability to keep degraded material in suspension) decreases, friction and wear characteristics deteriorate, metals corrode (and ultimately varnish) and sludge forms. As with ATF, the aim for CVT fluids should be to prevent any significant deterioration in a key performance attribute over the fluid service interval. No one degradation parameter can indicate whether a fluid is “good.” For example, viscosity change will not serve as a reliable indicator of friction or wear behavior. Therefore, it is important to not only track oxidation stability by chemical tests, but also perform functional tests. Functional tests include: transmission aging tests such as the GM cycling test, component tests which include aging such as the Ford friction durability tests, or by establishing correlation between a bench test and transmission failure modes, as in the Aluminum Beaker Oxidation (ABOT) test.
- 9.4.2 SHEAR STABILITY—CVT fluid shear stability requirements are currently similar to those for ATFs.
- 9.4.3 VOLATILITY—Evaporation leads to loss of fluid. Since low viscosity volatile fluid constituents are lost preferentially to those with higher viscosity, the overall viscosity increases. Evaporation has become more of an issue with the introduction of fluids with a $-40\text{ }^{\circ}\text{C}$ viscosity on the order of 10 000 to 20 000 mPa*s, so tests are beginning to be introduced to address this issue.

9.5 Safety

- 9.5.1 FLASH AND FIRE POINTS—The flash and fire points are related characteristics determined by passing an open flame over a cup containing fluid while the fluid temperature is gradually raised. The fluid temperature at first ignition is the flash point and the fluid temperature at which a flame is sustained is the fire point. It is not clear what limits are really necessary for safety and, accordingly, some variation is seen among specifications. Method: ISO 259/ASTM D 92

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APPENDIX A

CVT CROSS SECTIONS

A.1 CVT Cross Sections—See Figures A1 to A4.

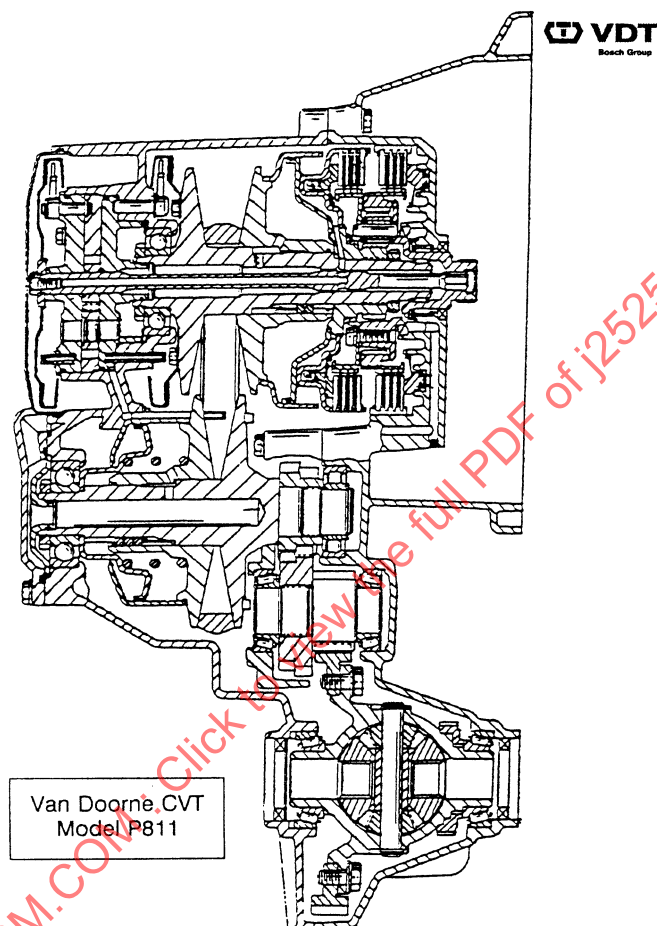


FIGURE A1—CVT CROSS SECTION—VAN DOORNE CVT MODEL P811