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A New Pump for CVT Applications

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ABSTRACT

For years, gear and gerotor type pumps have been common in Continuously Variable Transmissions (CVT). In their efforts to improve power density and efficiency of the CVT, engineers are showing interest in other pump types that can better cope with the increasing demands. The raising of CVT pressure levels, to enable higher torque and increased variator efficiency, makes that issues like the hydraulic system effectiveness and efficiency have become increasingly important.

Dual stroke vane pumps provide an answer. They basically form two independent, not necessarily identical, pumps that can work at different pressure levels. They enable the design of cost-effective solutions like switch-able or separated high/low pressure circuits within a single pump unit. This offers a better answer to the hydraulic power requirements of the CVT. Furthermore, the vane design is compact and provides high volumetric efficiency at low speeds even without measures like axial pressure compensation.

Understanding CVT requirements, Van Doorne's Transmissie has focussed on the development of the roller vane (RV) pump to avoid the unfavorable start-up performance, the typical wear behavior and the relatively high production cost accompanying traditional vane pump designs. This has led to several built designs for prototype CVT transmissions showing that the RV pump is ready for the market.

CVT PUMP REQUIREMENTS

Current CVT designs contain several functions besides the variator that have to be supplied with oil such as controls, clutches, torque converter, lubrication and cooling (**figure 1**). Usually the shifting variator imposes the critical demands in terms of pressure and flow. Lately also the idle/launch condition has gained significance by the influence of reduced engine idle speeds that affect the flow from the engine driven pump. The trend towards higher maximum pressure that allows reduction of the displacement volume of the pump enhances this effect.

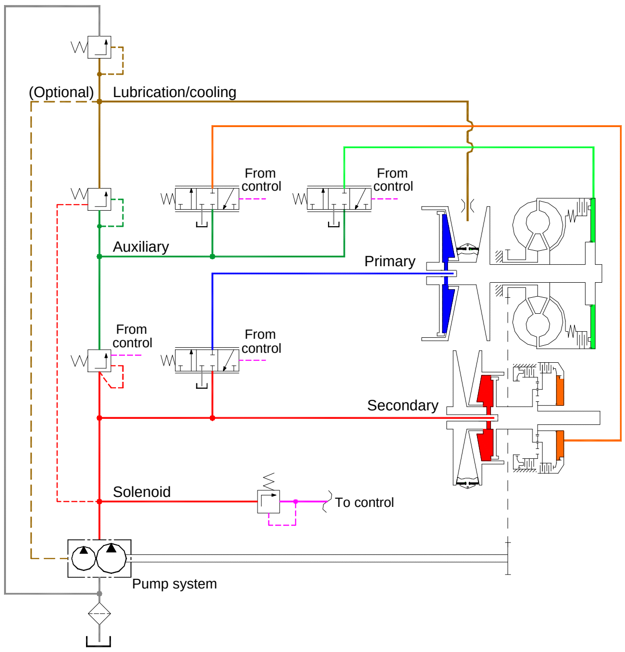


Fig. 1: Hydraulic functions in a CVT, cascade system

For most applications, the severe requirements are an emergency stop from overdrive towards underdrive at low engine speed and a kick-down starting from the same work-point shown in **figure 2**. Programmed step-mode features in CVTs can result in additional requirements. Coordinated powertrain control concepts based on drive torque control in some cases lead to more gradual engine speed and ratio transients, occasionally bringing some relief to these requirements.

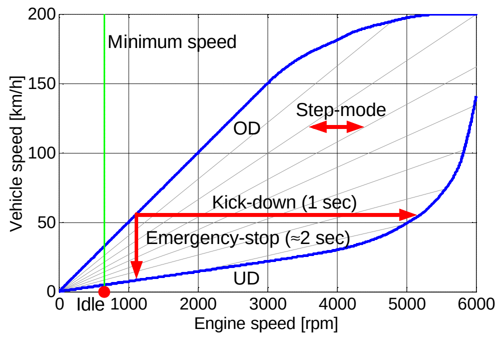


Fig. 2: CVT Variogram with shifts requiring large flow

The actuation system of a CVT influences transmission efficiency in two ways:

1. By its own power requirement that is based on variator and transmission component requirements.
2. By the difference between optimal and realizable belt clamping force, which affects variator efficiency.

Average hydraulic power during calm drive cycles can be as low as 300 Watt of which a considerable portion is consumed by the controls, lubrication and efficiency losses. The actuation system however must be designed to meet transients during which the pulley sheaves have to be displaced with 20 mm/sec at 50 kN clamping force. Even in case of a power exchange between the pulley cylinders, the maximum hydraulic power requirement can reach values close to one kilowatt caused by the large force ratio between the pulleys during fast shifts. This requirement leads to solutions incorporating engine driven pumps as the engine at this moment is the most convenient source to supply this kind of power.

Engine speeds lie in the range of 500 to 7500 rpm. An engine driven constant delivery pump, sized to meet the most critical situation, creates flow overshoot and related power loss in all other situations. Flow control valves do not offer a solution because the pump flow still has to be pressurized before the surplus is blown off. An optimization of the pump system, the hydraulic circuit and the use of continuously variable, switch-able or multiple pumps working at different pressures can realize a reduction in power consumption.

The displacement volume of the pump directly affects the power requirement of the hydraulic system. Maximum engine torque and maximum chosen system pressure influence this volume. For a system that basically resembles figure 1, equation 1 can be derived.

$$V_{th} \approx \frac{c_1 \cdot Tf \cdot T_{eng.\max}}{p_{s.\max} \cdot (1 - c_2 \cdot p_{s.\max})} + \frac{c_3 \cdot p_{s.\max}}{(1 - c_2 \cdot p_{s.\max})} + \frac{c_4}{(1 - c_2 \cdot p_{s.\max})} \qquad (1)$$

Where:

V_{th} = displacement volume of the pump [m^3]
$T_{eng.max}$ = maximum engine torque [Nm]
Tf = torque amplification factor [-]
$p_{s.max}$ = maximum secondary (system) pressure [N/m^2]
c_1 = constant dependent on variator parameters such as secondary cylinder surface, maximum shift speed, pulley angle, belt – pulley friction, clamping force safety, etc.. [-]
c_2 = constant defining pump leakage [m^2/N]
c_3 = constant defining hydraulic leakage [m^5/N]
c_4 = constant dependent on auxiliary flow [m^3]

The critical situation that in this approximation determines the displacement volume of the pump is assumed to be equal for most CVTs. The constants c_1 to c_4 therefore are applicable for multiple transmissions.
The torque amplification factor is a significant parameter. Most CVTs are equipped with a torque convertor that can amplify the maximum engine torque to the primary shaft.

In order to handle the increased torque, the cylinder surface or pressure level has to be enlarged. A surface increase leads to an enlarged displacement volume. In case a clutch is chosen as a launch device, there is no torque amplification and consequently the pump can be chosen smaller.
Figure 3 shows calculation results together with actual values from literature [1]/[2]/[3]/[4]. Obviously, leakage disturbs the linear relationship between torque, pressure and displacement volume, especially for smaller pumps.

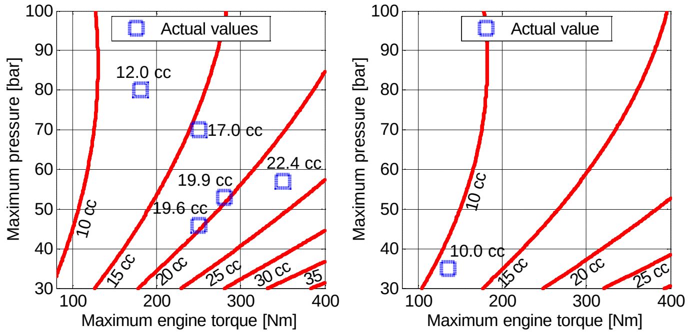


Fig. 3: Displacement volume CVT pump ([cc/rev]). Estimate for CVTs with torque convertor (left) or clutch (right) as a function of maximum engine torque and maximum pressure.

To reduce variator losses, the actuation system must be able to control low belt clamping forces during frequently occurring low engine torque [3]. A low minimum system pressure and small pulley cylinder surfaces enable this. Small cylinder surfaces however lead to a high maximum pressure level in case of maximum torque. The increasing maximum torque levels for CVT transmissions in general [1]/[4]/[5] are adding to this.
Higher pressures lead to extra leakage in the hydraulic circuit. These losses are partly compensated by the mechanical pump efficiency that initially rises with increasing pressure. The reduction of the minimum pressure level also adds to the compensation of the loss. The decrease of leakage initiates an optimization between hydraulic losses and variator losses that leads to an optimal maximum system pressure level.

The optimal maximum system pressure for the hydraulic system can be estimated by calculating the pump torque with the equations 1 and 2.

$$T_{pump} = \frac{p_s \cdot V_{th}}{2 \cdot p \cdot h_{hm}} \qquad (2)$$

Where:

T_{pump} = pump torque [Nm]
p_s = momentary secondary pressure [N/m^2]
h_{hm} = hydro-mechanical pump efficiency [-]

Figure 4 shows a pump torque estimate in relationship to momentary and maximum secondary pressure for a 250 Nm engine running at 1500 rpm. The straight lines represent constant variator clamping forces applied by the static pressure. The secondary cylinder surface and the pump displacement volume vary along these lines. The figures respectively show the situation for normal and negligible leakage of pump and hydraulic.

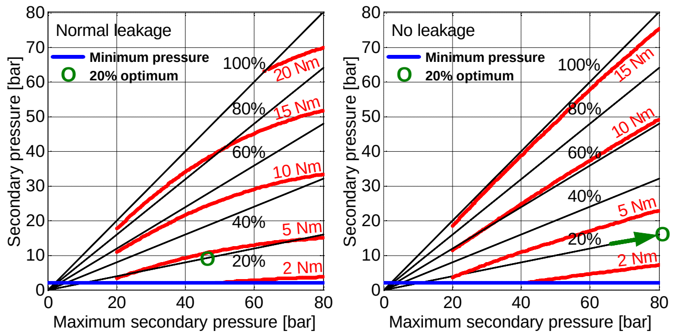


Fig. 4: Pump torque estimate dependent on secondary and maximum secondary pressure for a 250 Nm engine running at 1500 rpm. Normal leakage (left), no leakage (right)

At 20% clamping force, the figures show optimal maximum secondary pressures close to 45 bar and beyond 80 bar respectively. Both figures show a drop in optimal pressure for higher force percentages. Most of the time however the variator handles limited engine torque and therefore requires limited clamping force.
The reduction of leakage allows an increased maximum pressure level and leads to a reduction in power consumption. In case variator efficiency influenced by an optimal clamping force strategy is included in this calculation, this pressure is expected to rise further.

Pump systems must be able to cope with the increasing maximum pressures that in practice reach 60 to 80 bar. These values are considerably higher than the more generally known values of 35 to 45 bar.
Presently most CVT pumps are located in-axis behind the launch device. Relatively large pump rotors enable the primary shaft to pass through but result in large pump housings, susceptible to pressure raised deformation and internal leakage.

High maximum system pressures require smaller pump designs to reduce leakage. Small housings can be designed stiffer and offer weight advantages. To prevent drive losses, small pumps can be positioned at the end of the primary shaft. Optionally they are positioned behind the launch device and are remote driven by gear or chain-drive, which in some cases helps to reduce transmission length. It also prevents the need for a pump drive shaft going through the primary shaft.

Table 1 sums up some typical CVT pump requirements.

CVT requirement	Value/criterium
Pressures	0 to 80 bar
Speeds	500 to 7500 rpm
Pump size	7.5 to 23 cc/rev (CVT dependent)
Cavitation speed	>7500 rpm or measures to prevent
Temperatures (cold start)	-30 to 120° C (-30°C / 150 rpm)
Power saving capability	Multiple pumps/Switch-able/Variable
Pump location	In-axis/off-axis/end of axis

Tab. 1: CVT pump requirements

PUMP SYSTEM LAY-OUT

PUMP CHOICE

Currently, most CVT pumps are gear or gerotor type pumps based on designs from stepped automatic transmissions. Recently vane type pumps have been introduced into CVT from the field of power steering technology [6].
Typically leak-tightness of gear and gerotor type pumps is based on the tolerances of parts that define the critical radial clearance inside the pump [7]. Closer tolerances mean better efficiency but also more expensive parts for which the risk of degrading efficiency due to wear increases. In order to apply these pump types for higher pressure applications like CVT, cost increasing measures like radial pressure compensation, additional seals or complex rotor profiles [8] are required. Another solution is the increase of pump displacement volume, accepting the accompanying power losses.

In vane pump designs the discussed leakage is avoided. The contact between vane and cam seals off the radial clearance as shown in **figure 5**. Hereby a significant reason for low volumetric efficiency is made negligible.

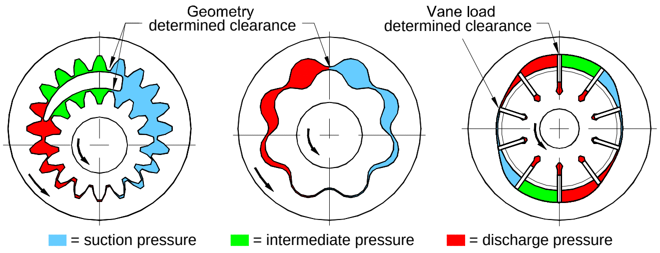


Fig. 5: Radial clearances in gear, gerotor and vane pumps.

At this moment the blade vane pump is pushed forward as a logical choice for CVT. On a technical level this choice can be disputed. The tilting of vanes, the large contact area and small play between rotor and vanes together with the presence of oil can lead to the problem of sticking vanes [9]. This disturbs the delicate radial force balance of the vane on which the pump is relying to prevent a short circuit between discharge and suction.

Typically the blade vane requires some level of discharge pressure that pushes it against the cam and overcomes viscous shear and friction between vanes and carrier. This has several disadvantages such as a reduced mechanical efficiency caused by friction between vane and cam, a minimum pressure requirement below which the pump can not operate and an unfavorable start-up behavior during which the absence of discharge pressure prevents the pump from becoming self-priming.
The roller vane pump, shown in **figure 6**, is already known from fuel pump technology and offers a solution to these shortcomings. Its working principle is based on a more positively composed radial force balance on the roller where centrifugal force can have a larger influence.

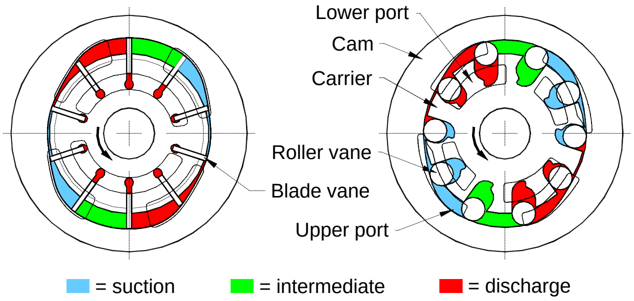


Fig. 6: Blade vane and roller vane, equal displacement.

Rollers do not stick due to the relatively large play and limited contact area between roller and carrier. It enables a good start-up behavior and allows suction and discharge pressure to become equal.
Further advantages follow from the roller geometry. The roller can rotate freely. The constantly changing contacts with the cam and carrier hardly experience wear. A degrading mechanical efficiency as found for blade vane pumps [10], caused by wear in the vane-cam contact, is not observed. The absence of a bending torque introduced by tangential forces, mechanically stressing and tilting the vane works advantageously.
In terms of cost the roller by its shape and tolerance requirements offers a 15% pump cost benefit compared to blade vane designs.

Table 2 shows a comparison between some of the pump types that knowingly are used for CVT.

Pumps / Criteria	*Ext. Gear*	*Int. gear*	*Gerotor (type)*	*Radial piston*	*Blade vane*	*Roller vane*
CVT pressures	0	0	-	+	0	+
Noise	0	+	0	-	+	+
Efficiency	0	0	-	0	+	+
Start-up	+	+	0	0	-	0
Variable displacem.	-	-	-	0	+	+
Wear compensation	-	0	-	+	0	+
Cost	0	-	0	-	0	+

Tab. 2: A comparison between CVT pumps

PUMP SYSTEM

From a system point of view dual vane type pumps offer advantages. They basically consist out of two independent pumps gathered around a single shaft. This offers the following functionality at the cost of adding one or two valves. **Figure 7** shows some examples:

- Switch-able between two or three flow delivery modes (single 1, single 2, dual)
- High speed cavitation prevention by boosting one pump stage with the other (parallel-series)
- Energy saving by flow reduction or pressure drop reduction (suction control).
- Separated pressure circuits (high-low pressure)
- Combinations

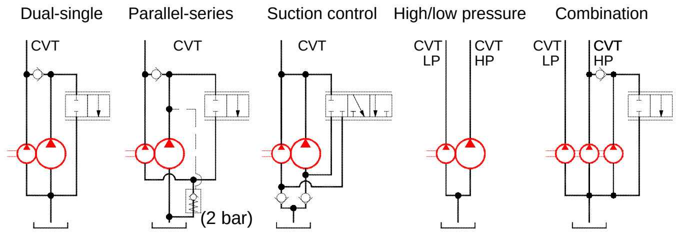


Fig. 7: CVT pump systems with dual/triple roller vane pump.

Lubrication pressure can be used to passively control the switch valves (figure 1). This saves a control valve. At approaching flow shortages, lubrication pressure drops and the pump switches to maximum delivery. During the switch the valve acts as a flow control valve (**figure 8**).

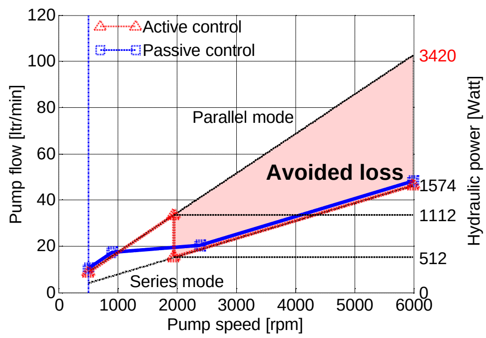


Fig. 8: Reduced power consumption for a parallel-series pump system with two flow delivery modes working at 20 bar.

The achievable power consumption reduction of the hydraulic system depends on the drive cycle and the pump system layout. For a NEDC cycle, power consumption can be reduced with about 60% compared to conventional systems. This reduces fuel consumption with about 1.4%. For increased vehicle speeds, secondary pressure drops as a result of reduced secondary torque and increasing centrifugal pressures in the pulleys. At high speed, power consumption reductions of 80% can be reached which translate to fuel consumption savings of up to 3.0%, despite the high engine power. During fast vehicle accelerations the power reduction of several kilowatts has a significant effect on vehicle performance.
Systems with a continuously variable pump can offer further efficiency advantages [11]. These however have the following drawbacks.

- A reduced volumetric efficiency caused by additional clearances between for instance cam and housing.
- The volumetric efficiency reduces with decreasing displacement volume of the pump.
- Pump flow control may require a minimum pressure level above minimum CVT pressure. An electro-mechanical pump actuation system may be required.
- Flow control increases variator control complexity.
- Relatively high cost.

The limited improvement compared with switch-able pump systems may not outweigh the extra cost. **Table 3** shows a comparison between some of the options.

System Criteria	*Constant delivery*	*Dual-single*	*Parallel -series*	*High low pressure*	*Variable displ.*
Efficiency	0	+	+	+	++
Noise	0	+	+	0	0
Simplicity	0	-	-	0	--
Cost	0	-	-	0	--

Tab. 3: Comparison of CVT pump systems (constant delivery pump is the reference)

ACTUATION SYSTEM

As mentioned, the fast shifting variator imposes the strictest demands on pump size. Flow requirements are influenced by the design of the actuation system. The surfaces of the pulley cylinders and their configuration in the hydraulic circuit play a role.
For the circuit layout, also the variator clamping force ratio KpKs is of importance. This parameter, shown in **figure 9**, determines the ratio between primary and secondary clamping force, required to maintain variator ratio under stationary conditions.

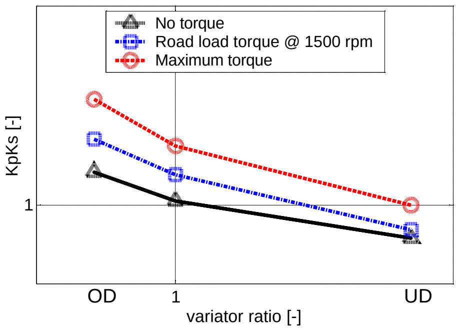


Fig. 9: Variator clamping force ratio KpKs.

Concerning the primary and secondary cylinder pressure, three circuits are known at this moment.

a. Circuit with larger secondary than primary pressure.
b. Circuit with free to choose cylinder pressures.
c. Circuit with two pressure levels per cylinder.

Figure 10 shows some examples. The first circuit typically contains a pressure control valve to control secondary pressure that provides the clamping force.

Primary pressure is derived from secondary pressure through a 3/3 directional valve that controls transmission ratio. This simple and low cost circuit has the following features.

- The cylinder surface ratio A_{pri}/A_{sec} is set to the maximum KpKs that is required to keep the variator in overdrive plus an additional safety for dynamic situations. Fast shifting towards underdrive, during which the secondary cylinder is filled, is most critical. The secondary cylinder surface therefore is an important parameter in determining pump size.
- Shift flow is not or only partially transferred between the cylinders. The pump has to raise most shift flow.
- The pressure control valve is located in the main stream. Secondary pressure can not be reduced below auxiliary pressures (also see figure 1).

In the second circuit the maximum cylinder pressure switches between the primary and secondary cylinder. The circuit contains one 3/3 valve per cylinder fed by line pressure that is controlled by a pressure control valve [1]/[2]. This circuit has the following features.

- The cylinder surface ratio A_{pri}/A_{sec} can be chosen to meet the most frequently occurring KpKs ratio. In this way primary and secondary pressure lie close to each other most of the time. Also for this circuit the secondary cylinder surface is an important parameter to determine pump size. For cost reasons it can also be decided to use equal cylinder surfaces.
- The pump has to raise all shift flow.
- Both cylinder pressures can be reduced to ambient pressure.
- The line pressure usually is controlled slightly above the maximum of secondary and primary pressure. As primary pressure can rise above secondary pressure, pump pressure will be high more often. Both effects influence the efficiency of the hydraulic system negatively.

The third circuit achieves a pump size reduction by applying cylinders with double surfaces that separate the functions clamping and ratio control. During shifting, the clamping volumes exchange flow. This circuit has the following features.

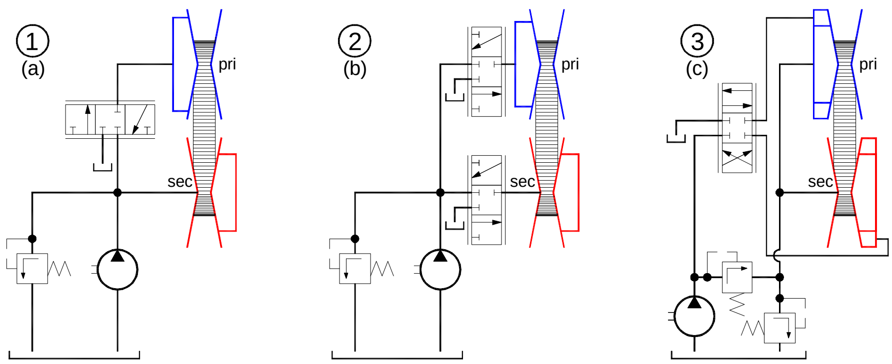


Fig. 10: Several hydraulic circuits for CVT actuation.

- Optimization of the cylinder surfaces with respect to hydraulic system efficiency points in the direction of larger clamping surfaces. This reduces clamping pressure without increasing flow. The difference between clamping and ratio control pressures can be optimized. Large surfaces do contradict with the wish to control low clamping forces, especially for systems with higher minimum system pressure.
- The pump must provide the flow to the ratio volumes plus some additional flow caused by the not one on one exchange between the clamping surfaces. The ratio surface of the secondary cylinder is an important parameter in determining pump size.
- The clamping and ratio control pressures are reduced from maximum circuit pressure. This is controlled by a line pressure valve, which can be overruled by the clamping pressure control valve.
- The clamping pressure can not be reduced to very low values.

In the discussed circuits the pump flow surplus is blown off to auxiliary functions or to sump in the absence of shifting. This influences efficiency negatively.
Up till now the pump system was seen as a separated unit, located at the start of the circuit and used to pressurize fluid from ambient to the required system pressure. The following concepts prove that when this principle is left, interesting alternatives can be found.

The fourth circuit in figure 10 contains two separate servo-pumps that directly control pressure [12]. Although good results are reported on power consumption the system requires two separate e-motor driven pump units.

The last circuit showed in figure 10 aims for a low-cost solution with standard components. It uses a 4/3 valve, a pressure control valve and a pressure differential valve. The circuit contains two pumps that can be integrated to form one dual vane type pump. Circuit features are:

- The cylinder surfaces are chosen such that the difference between secondary and primary pressure, basically being the pressure drop over the intermediate pump, is minimal most of the time.
- The feed pump provides the minimum variator pressure and compensates for leakage and compression flows during pressure increases.
- The minimum cylinder pressure can not be reduced to ambient pressure.
- The pressure difference valve minimizes the pressure drop over the pressure control valve.
- One of the pumps can optionally be driven electrical.

The reduced energy requirements of this circuit are based on the free to choose cylinder surfaces that realize reduced pressure differences over the pump.

Table 4 shows a comparison of the discussed actuation systems.

Actuation system / Criteria	1	2	3	4	5
Efficiency hydraulics	0	-	+	+	+
Efficiency variator	0	+	0	+	0
CVT controllability	0	0	0	-	0
Construction simplicity	0	0	-	-	0
Space envelope	0	+	-	-	+
Cost (cylinders included)	0	0	-	-	0

Tab. 4: Comparison of CVT actuation systems (system 1 is the reference)

ROLLER VANE PUMP DESIGN

The design of a roller vane pump contains some critical features that will be discussed here shortly:

- Roller kinematics and dynamics.
- Carrier design.
- Noise reducing measures, cavitation speed.

ROLLER KINEMATICS AND DYNAMICS

Cam profile design

The cam profile design of a vane pump is an important feature. The continuity of the cam profile has an impact on pump noise and flow pulsation through:

- The force balance working on the rollers.
- The upper and lower chamber volumes.
- The pressures inside the pump chambers.

Discontinuities in either of these characteristics must be prevented. Continuity is preferably maintained up to higher derivatives.
The cam profile contains parts with a constant radius and connecting parts. A pump stage has a minimum and maximum cam radius $R_{c.min}$ and $R_{c.max}$ (**figure 11**).

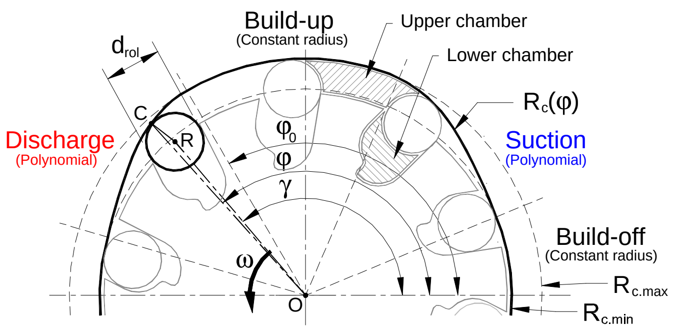


Fig. 11: Roller kinematics.

The displacement volume of a single stage can be determined with equation (3) where b_{cam} is the cam width. Contrary to most blade vane pumps, the lower pump chamber also contributes to the pump flow.

$$V_{th.stage} = p \cdot \left(R_{c.max}^2 - R_{c.min}^2\right) \cdot b_{cam} \qquad (3)$$

A continuous cam profile with a high degree of continuity can be obtained by applying polynomials for the parts that connect the constant radii. For a constant angular speed "ω", the local cam radius and its derivatives can be described as follows.

$$R_c(t)=\sum_{0\to n} c_n\cdot j^{\,n},\;\; j=w\cdot t,\;\;\to R_c(t)=R_c\big(j(t)\big)$$
$$\dot{R}_c(t)=R_c'(j)\cdot \dot{j}=R_c'(j)\cdot w \qquad (4)$$
$$\ddot{R}_c(t)=R_c''(j)\cdot w^2+R_c'(j)\cdot \dot{w}=R_c''(j)\cdot w^2,\;\;(\dot{w}=0)$$

At the transitions between the radius and the polynomial, discontinuities are prevented by assuring the polynomial derivatives are zero at these points, minimally up to the second derivative. In this way, six polynomial constants are known (radius, velocity and acceleration at start and end of the curve). To enable optimization of the polynomial, three additional constants are used resulting in an 8-order polynomial. These constants are chosen based on the minimization of the function f_{opt}:

$$f_{opt}=w_1\cdot\left|R_c'(j)\right|_{\max}+w_2\cdot R_c''(j)_{\max}-w_3\cdot R_c''(j)_{\min} \qquad (5)$$

The weight factors w_1, w_2 and w_3 in this function determine the importance of the following values:

- w_1. Maximum radial speed. Determines the momentary flow inside the pump caused by roller displacement.
- w_2. Maximum radial outward acceleration. Influences loss of roller-cam contact.
- w_3. Maximum radial inward acceleration. Influences force between roller and cam.

The radial outward acceleration of the cam curve negatively influences the contact force between roller and cam and is the main optimization parameter. In the early design stages the loss of contact is judged by comparing the centrifugal force, multiplied by a safety factor "S", with the (imaginative) cam acceleration force.

$$S\cdot m_{roller}\cdot\left(R_c(j)-\frac{d_{rol}}{2}\right)\cdot w^2>m_{roller}\cdot R_c''(j)_{\max}\cdot w^2\;\to$$
$$S\cdot\left(R_c(j)-\frac{d_{rol}}{2}\right)>R_c''(j)_{\max} \qquad (6)$$

The cam acceleration force in equation (6) basically is the force that is required to follow the maximum cam acceleration. The equation assumes that this is not dependent on the roller mass "m_{roller}" and angular speed "ω". In reality, effects like the pressure difference over the roller, viscous shear and the difference between the kinematics of a point on the cam profile and the roller mass point have some influence. These are taken into account by a comprehensive calculation that will be shortly addressed further on in this paper. In case required, the cam profile is adjusted to compensate for these influences.

An example of the optimization of the radius, radial velocity and radial acceleration is shown in the **figures 12 to 14** for a discharge polynomial from $R_{c.max}$ to $R_{c.min}$.

Roller diameter

The roller diameter is an important parameter. Pressure differences over its projected surface (diameter×length) create pressure forces working on the roller. This surface also affects momentary flows inside the pump when the roller is moving in a radial direction. To avoid large forces and high flows the roller diameter preferably should be as small as possible.

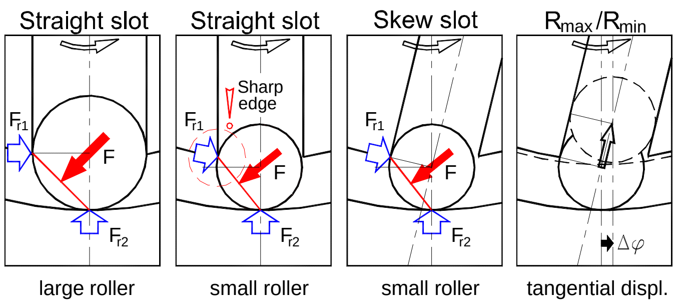


Fig. 15: Roller size and skew roller slot.

The displacement volume (equation 3) is primarily determined by the lift height of the roller $R_{c.max}$-$R_{c.min}$. For a straight slot, with its centerline running through the carrier center, the lift height is limited to approximately the roller radius. Limiting factor is the sharp contact between carrier and roller at the maximum cam radius as shown in **figure 15.**

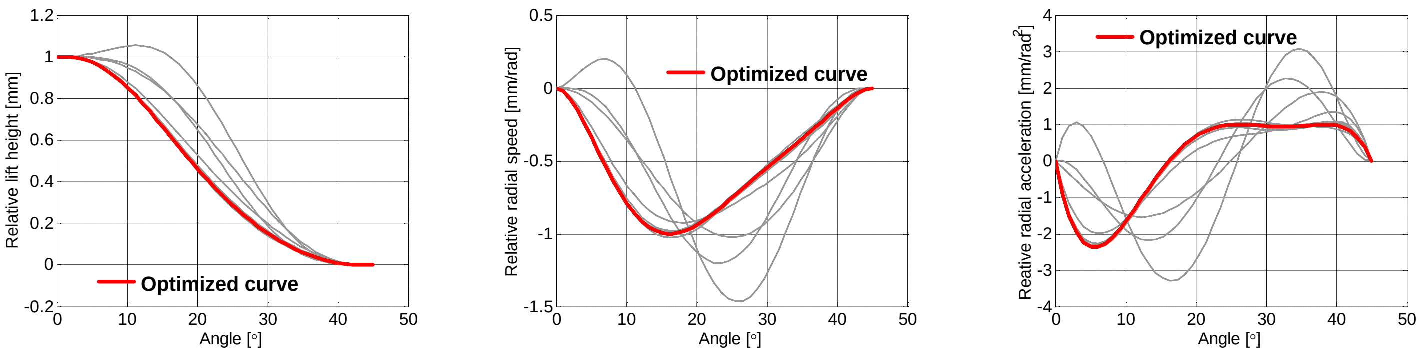


Fig. 12, 13 and 14: Optimized discharge polynomial.

For a skew slot an identical lift height can be obtained with a smaller roller. For skewness angles up to 15° the contact forces between roller and cam or carrier vary little. The smaller pressure load working on the smaller roller compensates for the increasing contact forces caused by the wedge effect that the roller is experiencing between cam and carrier. In figure 15 carrier and cam loads are practically identical for both diameters.

The skewness of the roller slot imposes a tangential acceleration on the roller when travelling between $R_{c.max}$ and $R_{c.min}$. This acceleration together with the coriolis acceleration is included in the force balance on the roller.

Internal pump pressures

The pressures working on the roller have an influence on its dynamical behavior. The roller passes through the phases shown in **figure 11** and **figure 16**.

1. Suction, roller surrounded by suction pressure.
2. Pressure build-up, roller forms a seal between suction and discharge pressure.
3. Discharge, roller surrounded by discharge pressure.
4. Pressure build-off, roller forms a seal between discharge and suction pressure.

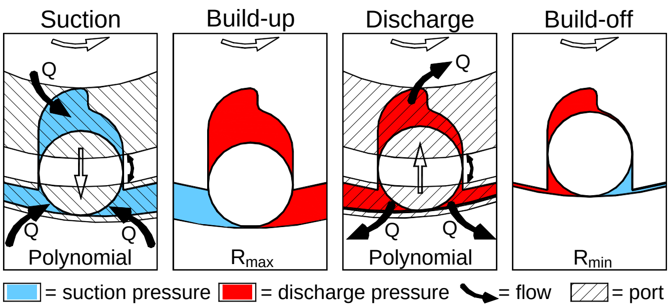


Fig. 16: Pressure phases.

During suction and discharge, when the chamber volumes are changing, pressures inside the chambers differ from the pressures in the suction and discharge channel. In these situations fluid runs through the ports connecting the pump chambers with these channels.
The ports form resistors that increase the chamber pressures during discharge and decrease them during suction. By varying the local port size, pressures inside the pump chambers can be affected helping to positively influence the contact force between roller and cam. This is only possible when the roller is moving in a radial direction during suction and discharge.

The timing of ports plays an important role in the dynamic behavior of the roller. It aims for a positive contact force between roller and cam. During pressurization this means that the lower pump chamber is pressurized first. During de-pressurization the upper chamber is de-pressurized first.

Friction forces

The friction forces that work on the roller are:

- Friction between roller and cam.
- Friction between roller and carrier.
- Viscous shear between roller faces and housing.

The main friction results from the contact forces between rollers and cam. Each roller constantly contributes to these friction losses dependent on the local contact force and the local coefficient of friction. Cam acceleration forces (imaginative) on the roller occur while it is on a polynomial. At the radii $R_{c.min}$ and $R_{c.max}$, discharge pressure related forces dominate the force balance. Geometry, load, material properties and oil properties influence the coefficients of friction in the roller-cam contact and the roller-carrier contact. This complex system determines whether the roller rolls or slides over the cam or carrier. The tribological behavior falls outside the scope of this paper.

Friction between roller and carrier occurs when the roller is travelling between $R_{c.min}$ and $R_{c.max}$ during suction and discharge. In these circumstances, the contact force between roller and carrier is small and friction is limited.

Viscous shear occurs between the faces of the roller and the surrounding housing. These relatively small losses are dependent on pump speed, oil temperature and axial play between rollers and housing.

Results kinematics and dynamics

A comprehensive computer program supports the design. It calculates important data based on a given geometry of cam, carrier, roller and ports for a given speed, pressure and temperature work point of the pump. The following calculation steps are performed:

1. Calculation of mass points, contact points and angles
2. Calculation of roller velocities and accelerations
3. Calculation of chamber volumes and gradients
4. Calculation of free port surfaces
5. Calculation of flow velocities through the ports and between the upper and lower chamber
6. Calculation of the related chamber pressures (by iteration)
7. Calculation of cavitation speed, based on flow velocities
8. Calculation of all loads and contact forces (normal, friction) working on the rollers (tribological model, partly by iteration).
9. Calculation of mechanical and volumetric pump loss and efficiency

Figure 17 shows a graphical representation of the forces working on the roller. This figure is an example. Values of forces are not on scale.

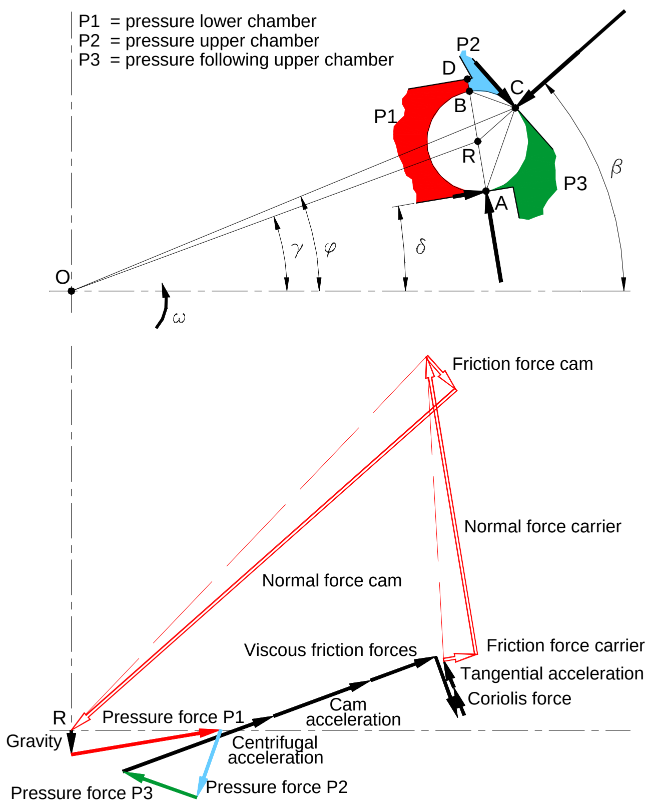


Fig. 17: Calculation of roller loads and contact forces.

Figure 18 shows some calculation results from the RV pump program. Clearly the contact forces between roller and cam show that during the constant radii the pressure difference over the roller leads to the maximum force.

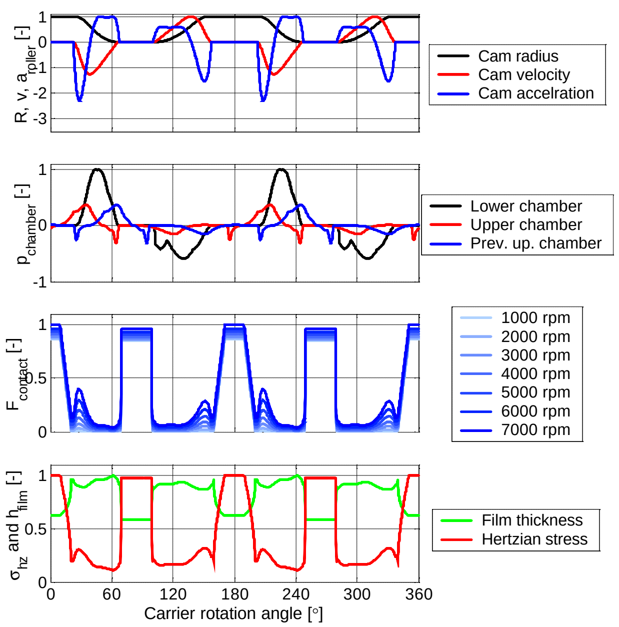


Fig. 18: RV pump program: some calculation results for a pump speed of 3500 rpm and a discharge pressure of 50 bar:
- roller kinematics, radius, velocity and acceleration
- chamber pressures from dynamic roller movements
- contact force roller-cam for several speeds
- oil film thickness / Hertzian stress in the roller-cam contact.

CARRIER DESIGN

The carrier geometry influences the dynamical behavior of the roller. The roller slot shape and the clearance between roller and slot are important. Most of the time the roller is in contact with the drive side of the slot (**figure 19**). During pressure build-up the pressure difference over the roller pushes the roller against the drive side. During pressure build-off, the pressure difference pushes the roller towards the opposite side. This motion is damped by the presence of oil that has to be squeezed from the clearance between roller and slot, defined by the radius "R_1". This clearance temporarily forms a seal between discharge and suction pressure during the roller motion. When the suction ports are reached and the pressure difference over the roller reduces, the roller travels back.

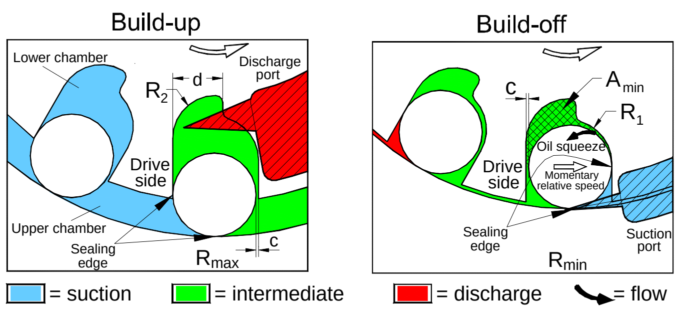


Fig. 19: Carrier design, roller slot.

The surface A_{min} influences the lower port area and is optimized within the following constraints.

- Allow flow to pass the port with a defined resistance.
- Reduce the dead chamber volume.
- Maintain the structural strength of the carrier (R_2).

Pressure build-up and build-off requirements determine the clearance "c" between roller and slot. During pressure build-up, the lower chamber is pressurized first. The upper chamber is pressurized by the lower chamber through the clearance to maintain a positive pressure force on the roller. This requirement defines the lower clearance limit. The allowed tangential motion of the roller defines the upper limit. A typical range of values for this clearance is 0.03-0.09 mm.

NOISE REDUCTION MEASURES

Noise can be caused by the following phenomena:

- Large pressure gradients.
- Large flow velocities, introducing cavitation.
- Large force gradients on pump parts.
- Transmission properties such as foaming, excessive leakage, inlet line and filter layout.

Large pressure gradients occur at the pressure transition areas between suction and discharge. Measures to reduce these gradients are based on gradual pre-compression or pre-decompression of chamber volumes by either pressure grooves, chamber volume changes, separate accumulator volumes or combinations.

These methods provide an optimization for one speed and pressure work-point. For the RV pump, grooves are used. In this way pre-compression pressure, fed by discharge pressure, exceeds that discharge pressure less easily and consequently does not cause loss of contact between roller and cam. Pre-compression by chamber volume decrease forced on by the cam profile provides a constant pressure increase independent on discharge pressure. This pressure increase starts in the upper chamber and can temporarily rise above lower chamber pressure dependent on the clearance between roller and slot. It can also rise above discharge pressure as shown in **figure 20** (2a, p_1) possibly resulting in a loss of contact between roller and cam.

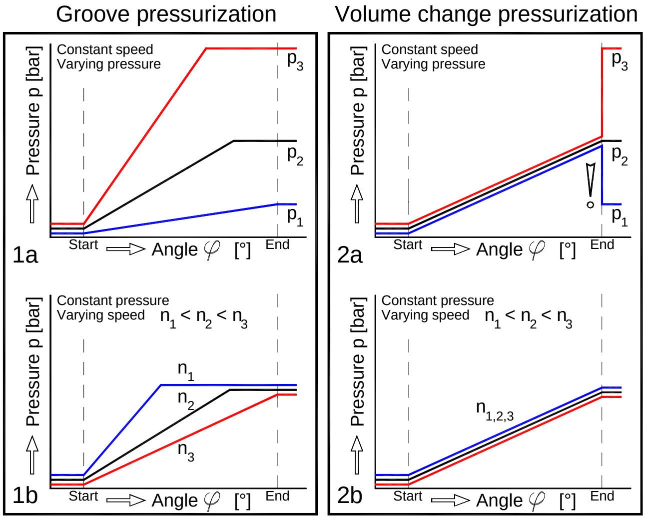


Fig. 20: Pressure build-up by groove or volume decrease.

The pressure grooves are typically chosen somewhat too large to optimize noise in all work points of the pump.

High flow velocities inside pumps can cause noise problems. They usually result from the high peripheral velocity of parts rotating at high speed. Oil must basically be able to follow that velocity which can be a problem at locations of low static pressure.
At the inlet, oil must be able to follow the maximum combination of tangential roller velocity ($R_c \cdot \omega$) and axial oil velocity through the suction port ($v_{port} \cdot \omega$). The maximum available pressure difference to generate this velocity is the ambient pressure (sump), minus oil vapor pressure, minus pressure loss in the inlet line. The cavitation speed of the pump can now estimated.

$$v_{port} = \frac{dV}{dj \cdot A_{port}(j)} \qquad (7)$$

$$\frac{1}{2} \cdot r_{oil} \cdot \left((R_c \cdot w_{cav})^2 + (v_{port} \cdot w_{cav})^2 \right) = p_{amb} - p_{vap} - \Delta p_{line} \rightarrow$$

$$\Delta p_{line} = c_f \cdot w_{cav}^2$$

$$n_{cav} = \frac{30}{p} \cdot \sqrt{\frac{2 \cdot (p_{ambient} - p_{vap})}{r_{oil} \cdot (R_c^2 + v_{port}^2)_{max} + 2 \cdot c_f}} \qquad (8)$$

Where:

v_{port} = axial oil velocity through port [m/rad]
dV/dj = momentary chamber volume change [m^3/rad]
$A_{port}(j)$ = momentary port surface [m^2]
r_{oil} = oil density [kg/m^3]
R_c = momentary cam radius [m]
p_{amb}-p_{vap} = ambient minus vapor pressure [N/m^2]
Dp_{line} = pressure loss sump to pump inlet port [N/m^2]
c_f = inlet line/pump constant [Nsec^2/m^2]
n / w_{cav} = cavitation speed pump [rpm] / [rad/sec]

Pumps with larger cam radius R_c like in-axis pumps appear to cavitate sooner. The flow velocity v_{port} however is small. For pumps with identical displacement volume but smaller cam radius, the flow velocity increases due to the smaller ports. Cavitation speed therefore remains almost constant for pumps with identical displacement.
Figure 21 shows cavitation speeds of several roller vane pumps with varying displacement volume. The cavitation speed for a hypothetical 20 cc pump remains almost constant according to the described model.

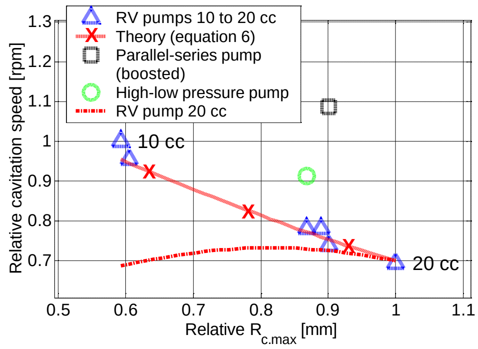


Fig. 21: Measured cavitation speeds for several values $R_{c.max}$.

The parallel-series layout (see figure 7) forms a solution to cavitation. In series mode an intermediate pressure of 2 bar is controlled. This rise of effectively $p_{ambient}$ prevents cavitation in the second stage. The pressure of 2 bar is too low to cause cavitation damage or noise in the first stage.
High-speed cavitation inside a RV pump usually starts in the upper suction port that is located at the larger radius near the cam. The port areas and pressure grooves are optimized to let cavitation start at the highest speed possible. In some cases cavitation can be completely avoided within the working range of the pump. The start of cavitation in the upper port avoids the loss of contact between roller and cam at the suction as the under-pressure pulls the roller against the cam. The connecting clearance between the pump chambers assures that pressure peaks are virtually equal in both chambers and do not lift the roller from the cam.

As discussed earlier, large force gradients are prevented as much as possible by measures such as cam profile, port timing and pressure groove optimization.

A lot can be done in the transmission to prevent noise. Already mentioned is the pressurization of the inlet by the parallel-series set-up.

Other methods use the oil surplus returned to the pump by the CVT hydraulic to improve pump inlet conditions [6]. With respect to this, also issues like choice of oil, anti-foaming additives, de-aeration, leakage prevention, filter location and filter optimizations are important.

MEASUREMENT RESULTS

In this section some of the results from RV pump measurements are discussed. In recent years Van Doorne's Transmissie has built a large number of prototype CVT transmissions for automotive customers in support of their development. **Figure 22** shows two results for end-axis/off-axis and in-axis mounting where new production techniques like lost foam aluminum housing manufacturing have been used. The P930 design comprises a dual-single pump with integrated switch hydraulic.

Fig. 22: P962 constant delivery (10 cc/rev) prototype (left) and P930 dual-single delivery (20 or 10 cc/rev) prototype.

Other results concern the P962 constant delivery concept, the P960 parallel-series concept and the P980 concept of which the following results will be shown.

- Volumetric and mechanical efficiency.
- Start-up behavior.
- Noise measurement

VOLUMETRIC AND MECHANICAL EFFICIENCY

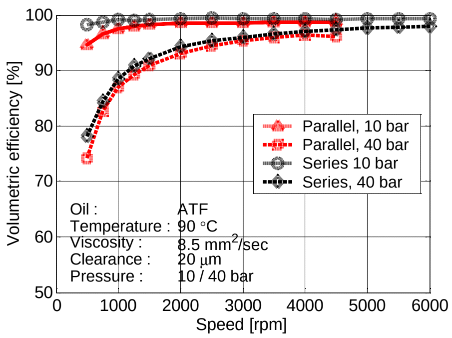
Figure 23 shows the volumetric efficiency of the parallel-series roller vane pump as a function of speed for the two separated stages (also see figure 5 and 6 and literature [13]).

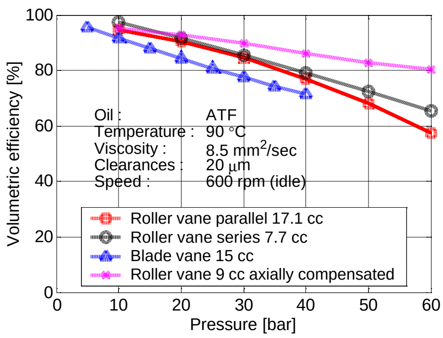
In the CVT, the pump typically runs in parallel mode up to 1500 – 2500 rpm. Then the pump is switched to series mode. The first pump stage is larger than the second pump stage (9.4 and 7.7 cc/rev).
The pump does not have axial pressure compensation. The efficiency in parallel mode is lower than in series mode due to some extra losses in the switch hydraulic.

Figure 24 shows the volumetric efficiency at a speed of 600 rpm where volumetric efficiency typically is low. This condition is critical during launch when the clutch has to be filled and pressure can rise steeply dependent on throttle position. The results are compared with data from a commercially available vane pump with axial pressure compensation. Still, roller vane efficiency remains above the blade vane design for both pump modes.
The fourth line in the graph shows the result of the high-pressure stage of a P980 roller vane pump with axial pressure compensation. This pump has a 9cc high-pressure stage and a 6cc low-pressure stage. The additional pressure compensation results in a considerable improvement in volumetric efficiency at this speed. The improvement reduces at higher speeds. As pressure compensation is a cost increasing feature its use should be dependent on the critical pump speed level in terms of flow requirement.

Figure 25 shows the mechanical efficiency of the same roller vane pump at 80°C and 1000 rpm. In series mode the pump shaft is not balanced anymore. The lower mechanical efficiency during this mode is caused by losses in bearings and switch hydraulic. The efficiency of the blade vane pump lies slightly above the roller vane efficiency. The planetary rotor results, taken from literature [8], lie below the RV pump results.

The result shows that the roller vane pump achieves good efficiency at high pressures. The efficiency improvement that it offers compared to other pump types directly translates in a CVT efficiency increase. For a NEDC cycle, a typical 10% improvement in volumetric efficiency that helps to reduce the displacement volume of the pump leads to a 0.4% reduction in fuel consumption. Besides this, transmission efficiency also benefits from the variable flow options that the roller vane pump offers.





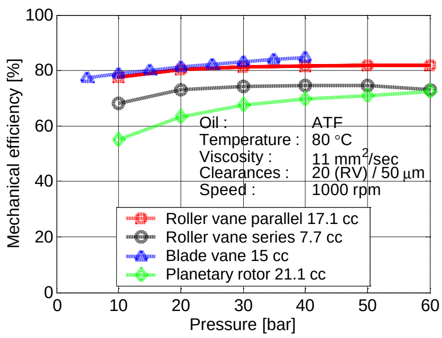


Fig. 23: Volumetric efficiency RV pump as a function of speed.
Fig. 24: Volumetric efficiency roller versus blade vane pump.

Fig. 25: Mechanical efficiency roller vane versus blade vane and planetary rotor pump.

START-UP MEASUREMENT

The priming speed of the roller vane pump is dependent on the moment the rollers get in contact with the cam. At that moment the roller functions as a sealing element. During start-up, the centrifugal force on the roller has to overcome the gravitational force. Discharge pressure is not available yet. Equation (9) shows how priming speed is estimated (see figure 17 for definition of angles).

$$m_{roller} \cdot w^2 \cdot \left(R_{c.\max} - \frac{d_{rol}}{2}\right) \cdot \cos(g - d) = m_{roller} \cdot g \cdot \sin(d) \rightarrow \qquad (9)$$

$$n_{start} = \frac{30}{p} \cdot \sqrt{\frac{2 \cdot g \cdot \sin(d)}{(2 \cdot R_{c.\max} - d_{rol}) \cdot \cos(g - d)}}, \quad (0 < d < p)$$

For angles (0>δ>π) gravity supports roller-cam contact.

Figure 26 shows the priming speed at 20°C and 90°C oil temperature for the 9cc high-pressure stage of the P980 RV pump. The test rig suction filter with 60 micron mesh is identical to the transmission filter. The suction channel of the test rig shown in figure 26 is 150 mm longer than the transmission suction channel. The inlet porting of the pump is positioned in a way that gravity positively influences the sealing function of the roller (equation 9).

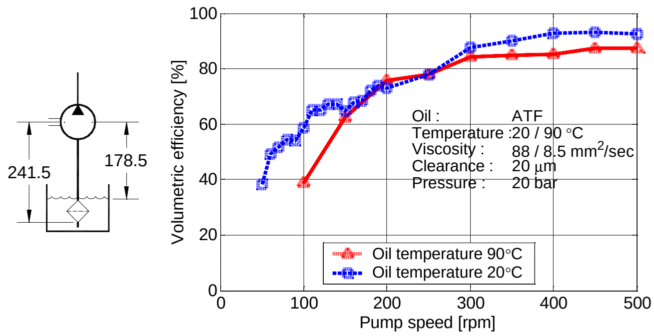


Fig. 26: Volumetric efficiency RV pump at low speed.

Pump speed started at 0 rpm and was increased. The measurement started at 50 rpm. For the temperature of 20°C flow was already available at 50 rpm. For the temperature of 90°C flow became available at 100 rpm. Due to the higher viscosity at lower temperatures, the volumetric efficiency was higher at 20°C. The lower temperature also contributed to the earlier priming of the pump.
The roller vane principle allows flow to become available at very low speeds. Contrary to this, literature shows that for blade vane pumps a minimum speed must be taken into account that lies around 600 rpm [14].

NOISE

For a specific transmission project, the noise behavior of a P962 roller vane pump was compared with the noise behavior of a conventional external gear pump with equal displacement (10 [cc/rev]). A blade vane pump with identical displacement volume was not available for this test. **Figure 27** shows an example of the test results.

Up to 4700 rpm the external gear pump generated significantly more noise than the roller vane pump. By applying special design features like cam continuity, carrier styling and pressure groove optimization a low noise design has been developed that lends itself very well for automotive applications like CVT.

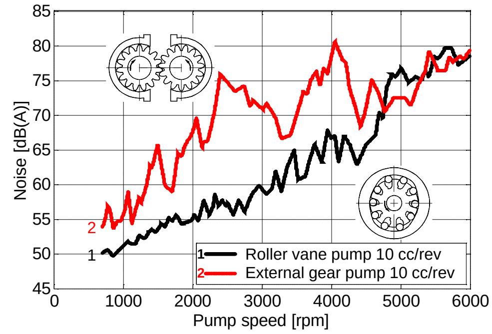


Fig. 27: Noise behavior. Comparison between an external gear and a P962 RV pump at 20 bar.

CONCLUSION

After extensive research, development and prototyping activities, roller vane technology has been made available for CVT.

The roller vane principle offers a dedicated pump concept that answers to the typical requirements imposed by the CVT transmission. Its high efficiency compared to conventional automotive gear and gerotor designs is expected to result in a fuel consumption improvement of about 0.4% over a NEDC cycle. Its multiplicity further adds up to 1.4% to this improvement, either by applying the pump as a multiple delivery mode system or by using it to supply multiple pressure levels in separate circuits. The most efficient configuration depends on the specific CVT layout and load cycle.
The roller vane pump can very well cope with the increasing pressure levels up to 80 bar that are foreseen for CVT in the future. Also for these pressure levels the design remains to enjoy quiet running characteristics.

Compared to the blade vane design the roller vane design offers specific advantages in terms of wear behavior, start-up and low speed running. The specific configuration of its internals that is characterized by relatively low tolerance requirements and cheap parts leads to a pump cost benefit of about 15% compared to conventional vane designs.
The design offers flexibility in positioning it in any transmission layout either in- or off-axis.

With this paper Van Doorne's Transmissie aims to contribute to the goal of improved power density, improved efficiency and reduced cost of the Continuously Variable Transmission and extend its acceptance level and application range.

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