

UDC 629.341-232:629.341.032.22:532.1

DESIGN PROPOSAL FOR A HYDROSTATIC CITY BUS TRANSMISSION

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Abstract: A city bus is probably one of the harshest operating environments for a vehicle transmission. The average operating speed is low, while frequent stops require swift acceleration from standstill. The options for the placement of transmission and suspension components are highly restricted as the passenger space must be as large as possible, with a completely flat vehicle floor for ease of passenger movement, and ground clearance is kept as low as possible to facilitate the entry and exit of passengers. Therefore, it is common to place the engine and gearbox at the rear end, and power is transmitted to the wheels via a propeller shaft and a rigid rear axle, effectively ruling out any completely level floor design. An improvement is offered by the hydrostatic transmission, using high pressure fluid for the power transmissison. The fluid flows from a pump to a motor via relatively small diameter pipes, enabling the design of a completely level floor. The hydrostatic transmission offers continuous variation of transmission ratios and the possibility that hydraulic motors can be placed very close to the wheels, thus enabling all- wheel drive and increased passenger space. Finally, hydraulic accumulators can be added to a hydrostatic transmission in order to recover and reuse kinetic energy which would otherwise be lost by braking.

Keywords:

- hydrostatic power transmission
- vehicle kinetic energy recovery system
- city bus

1. INTRODUCTION

A city bus transmission presents a serious challenge in powertrain design. The vehicle is operated at low speed in stop and go mode, with frequent full accelerations, and even overload conditions with excessive passengers during the rush hour. The designer also faces placement and size constraints when designing the transmission, as the vehicle floor must be completely flat, and the vehicle itself must have a low ground clearance to enable the passengers to easily enter and exit. Additionally, the usable passenger space in the interior of the vehicle must be as large as possible. It has, therefore, become a standard to place the engine and gearbox longitudinally at the rear end of the vehicle, and the power train is completed by a propeller shaft and a rigid rear axle, effectively blocking the design of a completely flat floor (Figure 1).

The engine is almost always a turbocharged diesel engine, as it needs a high power to weight ratio combined with high torque. The gearbox is usually an automatic with six to seven gears, which

provides the best compromise between gearbox cost and reduced fuel consumption.

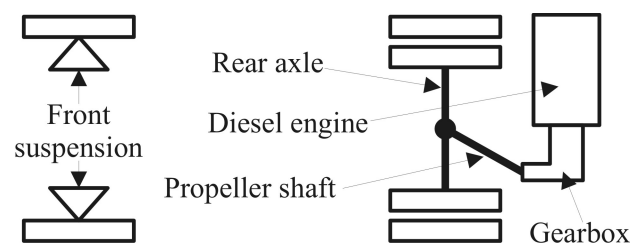


Figure 1. Schematic of a conventional city bus transmission layout

A solution can be provided with continuously variable transmission ratios, which is easily achieved using a hydrostatic transmission. Hydrostatic transmissions transmit power using high pressure fluid flowing from a pump to a motor using small diameter pipes, thus enabling the design of a completely flat vehicle floor. The motors can be placed near the wheels, increasing the available passenger space, and all wheel drive may be applied

with little weight increase and no reduction in passenger space.

2. VEHICLE PERFORMANCE

2.1. Basic Requirements

A two-axle bus [1] with the following characteristics was considered for the purposes of this article: rated engine power $P = 287$ kW, engine torque $T_{\text{mot}} = 1900$ Nm at $1000 \dots 1400$ min^{-1} , and torque at maximum rated power $T_{\text{Pmax}} = 1600$ Nm at 1700 min^{-1} . The maximum gross weight is $m = 19000$ kg, length $l = 10$ m, height $h = 2,8$ m, and width $w = 2,5$ m. The tyres have a dimension of 275/70 R 22,5, resulting in an outer thread radius $r = 0,764$ m. The vehicle uses an automatic gearbox with the following ratios: $i_1 = 3,43$, $i_2 = 2,01$, $i_3 = 1,42$, $i_4 = 1,00$, $i_5 = 0,89$, $i_6 = 0,59$. The rear axle ratio is $i_d = 9,82$.

The performance was estimated based on the following requests: a fully loaded vehicle must be able to climb a 28% slope, move from standstill in the same conditions with an acceleration of at least 1 ms^{-2} , and finally be able to reach a maximum speed of at least 80 km/h on level terrain. The weight distribution ratio between the front and rear axles is assumed to be 1:2. The coefficient of static friction can be assumed to be $\mu = 0,8$ for a concrete or tar surfaced road, while the rolling resistance factor equals $\mu_k = 0,02$.

The wheels are subject to the highest torque T_{kmax} during startup from standstill:

$$T_{\text{kmax}} = T_{\text{mot}} \cdot i_1 \cdot i_d \cdot \eta_{\text{meh}} = 57600 \text{ Nm} \quad (1)$$

It should be noted that according to [2], the overall efficiency of the mechanical transmission may be assumed to be $\eta_{\text{meh}} = 0,9$ using a black box approach.

The highest tractive force when moving from standstill F_{max} must be smaller than the friction force acting between the rear wheel and road surfaces F_{tr} in order to avoid wheelslip. For the purposes of this work, it has been assumed that 1/3 of the vehicle weight acts on the road surface over the front axle, while the remaining 2/3 act over the rear axle:

$$F_{\text{max}} = \frac{T_{\text{kmax}}}{r} = 75390 \text{ N} \quad (2)$$

$$F_{\text{tr}} = \frac{2}{3} \mu \cdot m \cdot g = 99410 \text{ N} \geq F_{\text{max}} \quad (3)$$

The mechanical transmission [3] also determines the maximum vehicle speed v_{max} :

$$v_{\text{max}} = \frac{1700 \cdot 2 \cdot r \cdot \pi}{60 \cdot i_6 \cdot i_d} = 23,5 \frac{\text{m}}{\text{s}} = 85 \frac{\text{km}}{\text{h}} \quad (4)$$

The outside forces acting on the vehicle (rolling resistance force F_k , drag force F_z and vehicle weight component force $G \sin \alpha$) must be balanced with the tractive force F_v in all cases, however, only two forces are important in order to determine the vehicle performance.

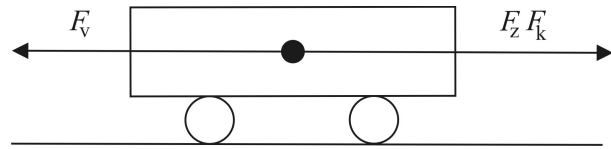


Figure 2. Forces acting on the vehicle on level terrain

In the first case, the vehicle is moving at maximum speed on flat terrain (Figure 2), and maximum rated engine power is used. The vehicle weight component equals zero, so the tractive force must be balanced against the rolling resistance force and the drag force [4]:

$$F_v = F_k + F_z + G \sin \alpha \quad (5)$$

$$F_k = \mu_k \cdot m \cdot g = 3730 \text{ N} \quad (6)$$

$$F_v = \frac{T_{\text{Pmax}} \cdot i_6 \cdot i_d \cdot \eta}{r} = 10920 \text{ N} \quad (7)$$

$$F_z = \frac{1}{2} \cdot \rho \cdot C \cdot A \cdot v^2 \quad (8)$$

In order to calculate the drag force, the density of air is assumed to be $\rho = 1,13$ kg/m^3 , while the front cross section of the vehicle equals $A = 7$ m^2 . As the front side is in fact a vertical flat surface with little to no aerodynamic refinements, it is safe to assume that the drag factor equals $C = 1$. By combining (5,6,7,8) into (9) it is possible to calculate the theoretical maximum vehicle speed as follows:

$$v_{\max t} = \sqrt{2 \frac{F_v - F_k}{\rho \cdot C \cdot A}} = 41,6 \frac{\text{m}}{\text{s}} = 150 \frac{\text{km}}{\text{h}} \geq v_{\max} \quad (9)$$

The equation (9) makes it clear that the theoretical maximum speed regarding drag is significantly greater than the maximum speed determined by the transmission.

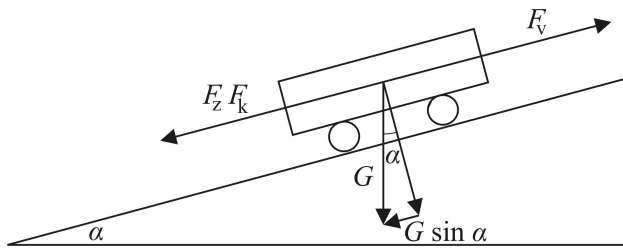


Figure 3. Forces acting on the vehicle on a climb

In the second case (Figure 3), the vehicle is accelerating from standstill on a slope. In this case, F_z equals zero and may be neglected, but the component force $G \sin \alpha$ must be taken into account. The acceleration from standstill a may be calculated using the following expressions:

$$m \cdot a = F_{\max} - F_k - m \cdot g \cdot \sin \alpha \quad (10)$$

$$a = \frac{F_{\max} - F_k - m \cdot g \cdot \sin(\arctan(0,28))}{m} = 1,13 \frac{\text{m}}{\text{s}^2} \geq 1 \frac{\text{m}}{\text{s}^2} \quad (11)$$

2.2. Mechanical Transmission Performance

A diesel engine is a constant power prime mover, meaning that the product of traction effort and vehicle speed remains constant for a given engine power output.

According to the engine manufacturer data, engine power grows in a linear manner with constant torque in the $1000 - 1400 \text{ min}^{-1}$ area, while in the $1400 - 1700 \text{ min}^{-1}$ area the torque begins to drop following the increase of engine speed. By using these figures, we can calculate traction effort of the vehicle in all gears for any combination of engine speed, and i.e., vehicle speed (Figure 4).

It is obvious that the graph in Figure 4 is not a continuous hyperbola, but a series of straight lines connected to hyperbolic segments, which respectively correspond to the constant torque and decreasing torque engine ranges. Therefore, there

are significant gaps in the curve, which can be remedied by using a stepless transmission.

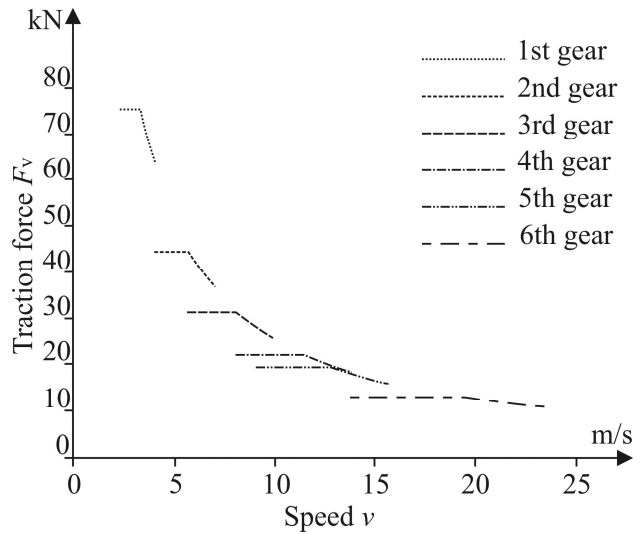


Figure 4. Traction force graph for the vehicle with mechanical transmission

2.3. Stepless Transmission Types

A stepless transmission can provide an indefinitely variable number of transmission ratios between its designed top and bottom ratios. Looking from a design perspective, such transmissions may be achieved using a friction drive with variable diameter wheels, a belt drive with variable diameter pulleys or with an electrical or hydrostatic transmission.

So far, pure friction drive has been applied to variable speed machine tools, while the belt drive system with variable diameter pulleys has been successfully used on cars and motorcycles. Besides, there have been no applications to low speed, high torque prime movers yet.

Since this work deals with a vehicle subject to a stop and go usage pattern, it has also to be concerned with an electric or hydrostatic transmission system which should have an option to store the brake energy of the vehicle and reuse it for moving from standstill.

An electric transmission consists of a generator connected to the prime mover and an electric motor connected to the wheels of the vehicle via a mechanical transmission. Transmission ratios are varied by control circuitry which modifies the characteristics of the motor and generator. This circuit enables the use of accumulators or supercapacitors for energy storage and reuse.

However, such systems are very heavy and bulky when compared to conventional mechanical transmissions in the low power size range, so they are confined to large applications, such as locomotives, ships and mining equipment.

The basic hydrostatic transmission consists of a hydraulic pump and a hydromotor in a closed loop system. The transmission ratio may vary by changing the system pressure, the pump flow rate, or the hydromotor displacement. Energy may be stored by compressing hydraulic fluid in relatively small hydro – pneumatic accumulators. Their mass

and volume can be reduced by using composite materials and technologies already applied in the production of natural gas storage tanks.

Furthermore, the hydrostatic transmission does not experience any significant mechanical wear by design, and unlike batteries, hydraulic accumulators may be used through an unlimited number of loading and unloading cycles. Also, the hydrostatic transmission offers an additional advantage in the placement of motors next to the wheels without a propeller shaft or a rigid axle, enabling the design of true low floor vehicles.

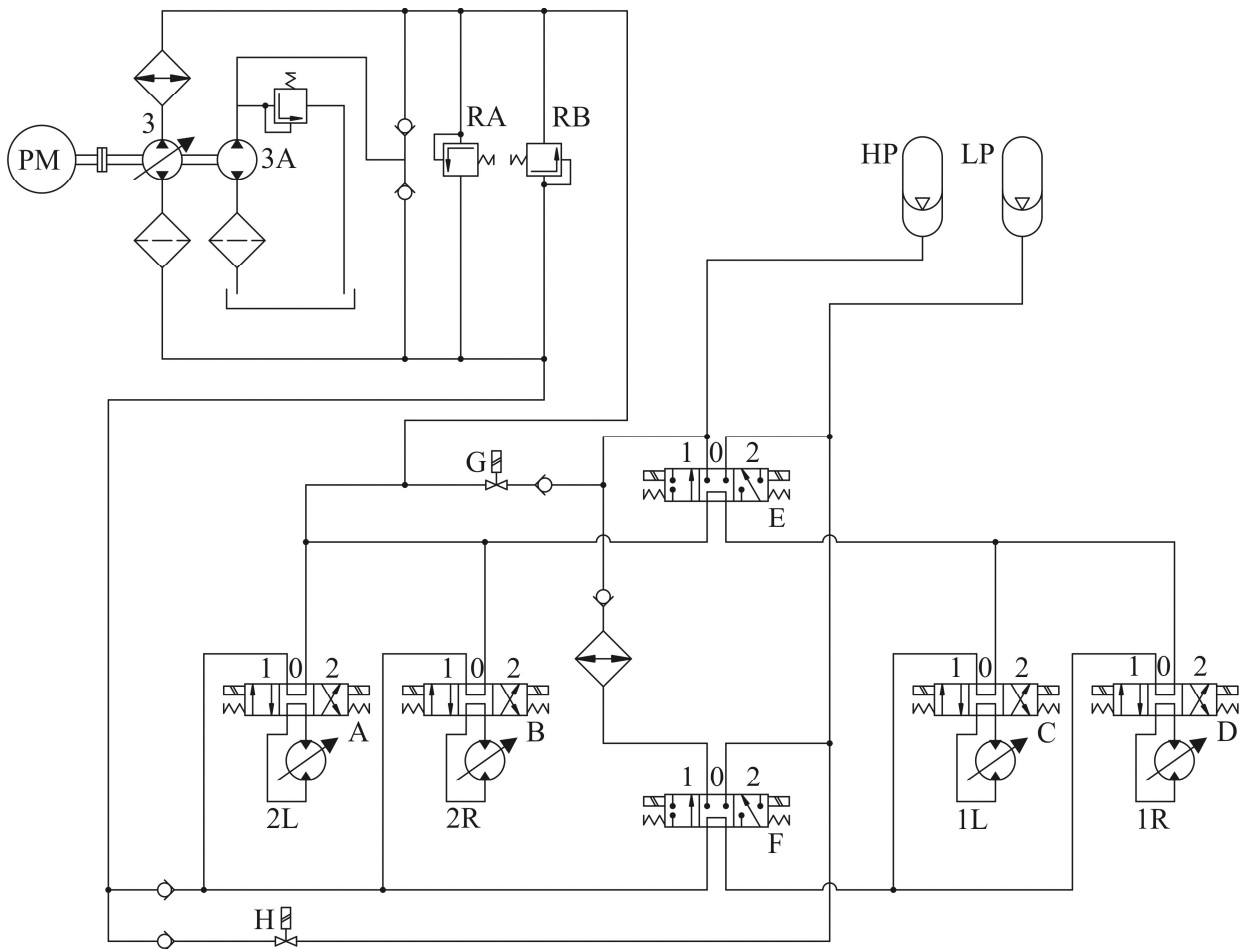


Figure 5. Scheme of hydrostatic transmission

3. HYDROSTATIC TRANSMISSION

3.1. System Description

The transmission (Figure 5) consists of four variable displacement bent axis piston hydromotors, 1L and 1R on the front wheels, and 2L and 2R on the rear wheels, a variable displacement pump 3, an auxiliary pump 3A, 4 port 3 position

electromagnetically controlled directional valves A, B, C, D, E, F, electromagnetically controlled gate valves G, H, a high pressure bladder type accumulator HP and finally a low pressure bladder type accumulator LP. Two relief valves, RA and RB, protect the system from overpressure.

During transmission, the prime mover PM (diesel engine) powers the main pump 3 which delivers the oil from the low pressure side to the high pressure

side of the system. High pressure oil is cooled in an oil cooler. The auxilliary pump 3A is combined as a single unit with the main pump 3 and compensates any oil lost.

The forward movement is achieved by moving the directional valves A, B, C and D into position 1, while directional valves E and F remain in position 0. The appropriate transmission ratio is achieved by varying the flow rate of the pump and motors.

The reverse movement is achieved by moving the directional valves A, B, C and D into position 2, while the directional valves E and F remain in position 0 as in the case of forward movement. It is interesting that a vehicle with hydrostatic transmission has exactly the same performance regardless of the direction in which the vehicle is moving, which would be of particular interest in a bi-directional application for guided busways or confined spaces.

The moving vehicle can coast with reduced mechanical resistances by moving the directional valves A, B, C and D into position 0, and electromagnetic clutches are disengaged at the same time to disconnect the motor output shafts from the wheels.

In engine brake mode, the motors work as pumps, and deliver high pressure oil to the main pump 3,

which acts as a motor and adds energy to the diesel engine.

In kinetic energy recovery mode, the directional valve E is shifted to position 1, directional valve F to position 2, after which the motors 1L and 1R deliver oil from the low pressure accumulator LP into the high pressure accumulator HP. An oil cooler is also provided in this circuit to cool the oil during kinetic energy recovery mode. The motors 2L and 2R remain connected to the pump 3 and continue to work in engine brake mode.

When moving from standstill, the stored kinetic energy can be used by shifting the directional valve E to position 2, and directional valve F to position 1. The motors 1L and 1R are powered by high pressure oil stored in accumulator HP, and oil is returned to the low pressure accumulator LP. The accumulator HP may be recharged at any time while the engine is running by activating the gate valves G and H while the directional valves E and F remain in position 0.

Figure 6 shows the placement of the various transmission components on the bus. The accumulators will be roof mounted in a way similar to compressed natural gas tanks on existing vehicles.

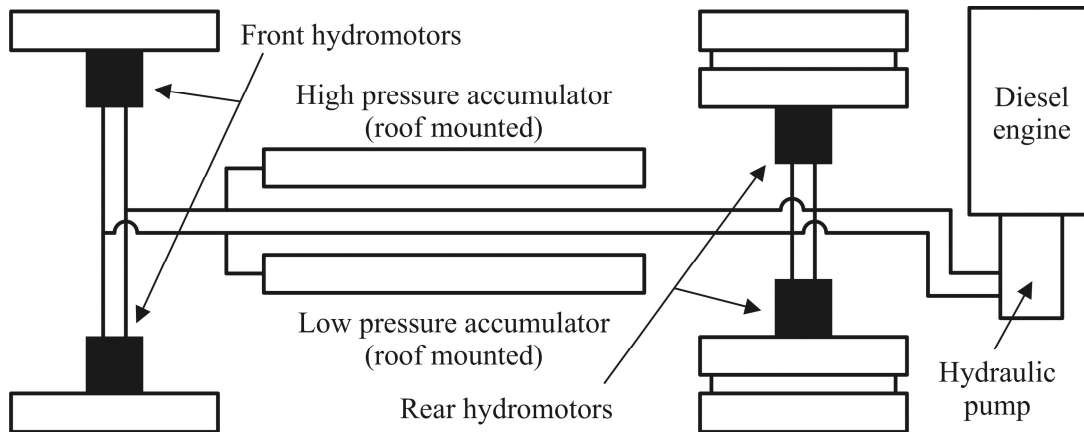


Figure 6. Arrangement of the hydrostatic transmission components

Table 1. Overview of hydromotor characteristics

Motor	Displacement V_g [cm ³]	Maximum flow rate q [dm ³ /min]	Maximum speed at full displacement n_{Vmax} [min ⁻¹]	Maximum permissible speed n_{max} [min ⁻¹]	Maximum torque T [Nm]
AA6VM 80	0...80	312	3900	7350	509
AA6VM 200	0...200	580	2900	5100	1273

3.2. System Characteristics

The hydrostatic transmission is designed regarding two extreme operating conditions: first, maximum torque when moving off from standstill, second, maximum flow rate when moving at maximum speed.

The transmission is assumed to operate with a constant pressure difference $\Delta p = 400$ bar. The pump outlet pressure is $p_2 = 430$ bar, and pump inlet pressure is $p_1 = 30$ bar, so as to avoid pump cavitation.

The total traction force developed by the hydrostatic transmission must match the force F_{\max} developed by the mechanical transmission, and has to be distributed to the front (F_1) and rear (F_2) hydraulic motors with regard to the vehicle weight distribution:

$$F_1 = \frac{1}{3} \cdot \frac{1}{2} F_{\max} = 12565 \text{ N} \quad (12)$$

$$F_2 = \frac{2}{3} \cdot \frac{1}{2} F_{\max} = 25130 \text{ N} \quad (13)$$

Assuming the same tyre size as in the case of mechanical transmission, the respective motor torques are $T_1 = 9600$ Nm for the front motors, and $T_2 = 19200$ Nm for the rear motors.

Such torques may be achieved by using directly coupled radial piston motors or axial piston motors coupled via planetary reduction gears. Axial piston motors will be used, because they can operate as pumps during engine braking and kinetic energy recovery. The motors (Table 1) AA6VM 80 for the front wheels, and AA6VM 200 for the rear wheels have been selected from [5].

After selecting the motors, the gear ratios of the respective planetary reduction gears $i_{r1} = 18,92$ and $i_{r2} = 15,13$ may be chosen. Those ratios determine the maximum allowed number of wheel revolutions $n_1 = 388,5 \text{ min}^{-1}$ and $n_2 = 337,1 \text{ min}^{-1}$. From here it is possible to adjust $n = 320 \text{ min}^{-1}$ as the common

value for both front and rear motors, which results in maximum vehicle speed of $v_{\max} = 92 \text{ km/h}$.

The flow rate q [dm^3/min] and the motor torque T [Nm] can be calculated using the expressions (14,15) [6], in which n [min^{-1}] is the current motor speed, Δp [bar] the pressure difference between the motor inlet and outlet, V_g [cm^3] the current motor displacement, η_V the volumetric efficiency, and η_{MH} is the hydromechanical efficiency.

$$q = \frac{V_g \cdot n}{1000 \cdot \eta_V} \left[\frac{\text{dm}^3}{\text{min}} \right] \quad (14)$$

$$T = \frac{V_g \cdot \Delta p \cdot \eta_{MH}}{20 \cdot \pi} [\text{Nm}] \quad (15)$$

Now it is possible to create the hydrostatic transmission traction effort diagram (Figure 7). The curve actually denotes the condition of maximum engine power, while the conditions of lower engine power may be found in the area enclosed by the coordinate axes and the traction force curve.

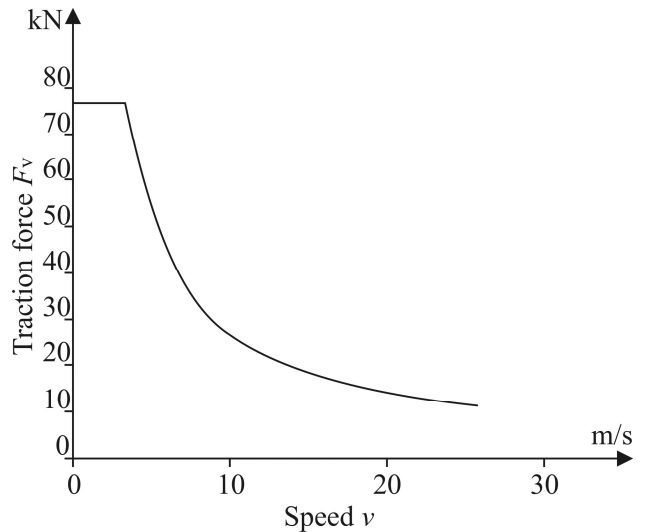


Figure 7. Traction force graph for hydrostatic transmission vehicle

Table 2. Overview of hydraulic pump characteristics

Pump	Displacement V_g [cm^3]	Maximum flow rate q [dm^3/min]	Maximum input power P_{\max} [kW]	Maximum permissible speed n_{\max} [min^{-1}]	Maximum input torque T [Nm]
AA4CSG 355	0...355	639	373	2000	1976

Table 3. Hydraulic accumulator specifications

	Volume [dm ³]	Gas volume [dm ³]	Bladder precharge pressure [bar]	Initial oil pressure [bar]	Final oil pressure [bar]	Initial oil volume [dm ³]	Final oil volume [dm ³]
High pressure accumulator	50	48,7	180	200	400	7	22
Low pressure accumulator	50	48,7	28	30	54	5	20

3.3. Prime Mover and Pump Selection

The prime mover and the hydraulic pump are selected on basis of the previously determined system characteristics. Assuming that there are no efficiency losses, both are slightly oversized to compensate for the calculation of (14,15). A diesel engine, MAN D2066 LF 32 has been selected from [7] as the prime mover. It has the following characteristics: rated engine power $P = 294$ kW, engine torque $T_{\text{mot}} = 1900$ Nm at $1000 \dots 1400$ min⁻¹, and torque at maximum rated power $T_{\text{pmax}} = 1600$ Nm at 1700 min⁻¹. The pump (Table 2) AA4CSG 355 has been selected from [8] to match the engine.

3.4. Engine Brake System

All road vehicles are capable of engine braking in order to reduce the load on the main friction brake system during extended use. During engine braking, the kinetic energy of the vehicle is transferred to the engine, where it is transformed into heat due to the internal resistances of the engine. This is also possible with hydrostatic transmission by operating the motors as pumps, driving the main pump acting as a motor, and finally transferring the kinetic energy of the vehicle to the diesel engine, where it is dissipated through the engine cooling system.

According to the manufacturer's data, the selected diesel engine offers a braking power of $P_k = 240$ kW. A constant pressure difference $\Delta p = 400$ bar is maintained during braking, while the desired braking power is determined by regulation of hydraulic motor displacement.

The system automatically shuts down at 10 km/h, after which the main friction brake system is used to completely slow down the vehicle.

3.5. Kinetic Energy Recovery System

The kinetic energy recovered by this system can be used to assist in moving from standstill, to drive with the engine turned off, or to assist the engine during acceleration. As the bus frequently stops and drives at low speeds, the system is the best used in moving the vehicle from standstill.

Hydraulic fluid is supposed to be incompressible, so the recovered energy must be stored by compressing gas in a hydraulic accumulator.

It is assumed that the system fully charges during a slowdown from $v_1 = 60$ km/h to $v_2 = 10$ km/h, so the recovered energy is:

$$E_k = \frac{1}{2} m (v_2^2 - v_1^2) = 2565586 \text{ J} \quad (16)$$

Bladder accumulators type HAB 50 (Table 3) have been selected from [9], as piston type accumulators are not feasible for vehicle use.

The high pressure accumulator works between the $p_{h1} = 200$ bar and $p_{h2} = 400$ bar, while the low pressure accumulator works between the $p_{n1} = 54$ bar and $p_{n2} = 30$ bar. It is accurate enough to assume that the energy is stored or released from the accumulator by an adiabatic process of the gas ($\kappa = 1,4$). The changes in gas volume, pressure, and the energy stored in the accumulators may be calculated using the equations from [6]:

$$p_2 = p_1 \left(\frac{v_1}{v_2} \right)^\kappa \quad (17)$$

$$E = \frac{(p_1 \cdot v_1 - p_2 \cdot v_2)}{(\kappa - 1)} \quad (18)$$

It is obvious from those equations that the recovered energy will not be completely spent on recharging the accumulators, because a part of it will be used to

overcome the pressure difference between the low pressure and high pressure accumulators.

The energy that can be stored by one high pressure accumulator equals $E_{VT} = 592751$ J, and this can be used to conclude that the system needs five high pressure and five low pressure accumulators.

The remaining energy stored in the low pressure accumulator is $E_{NT} = 60941$ J, so it is possible to calculate the total charging energy E_1 , as well as the total discharging energy E_2 :

$$E_1 = 5 \cdot (E_{VT} + E_{NT}) = 3266211 \text{ J} \quad (19)$$

$$E_2 = 5 \cdot (E_{VT} - E_{NT}) = 2661301 \text{ J} \quad (20)$$

An efficiency calculation based on (19, 20) shows that the total efficiency of the system is 81%.

Unlike piston accumulators which provide constant pressure fluid, bladder accumulators vary in pressure due to the change in the gas volume. This pressure variation also results in a torque variation of the motors supplied by the accumulator, so the control unit cuts them out as soon as they are not able to power the vehicle (e.g. startup on a slope under heavy load). Figure 8 shows this variation of pressure and motor torque as a function of fluid volume. The stored energy will suffice to cover a distance of about 120 m under full load on flat ground.

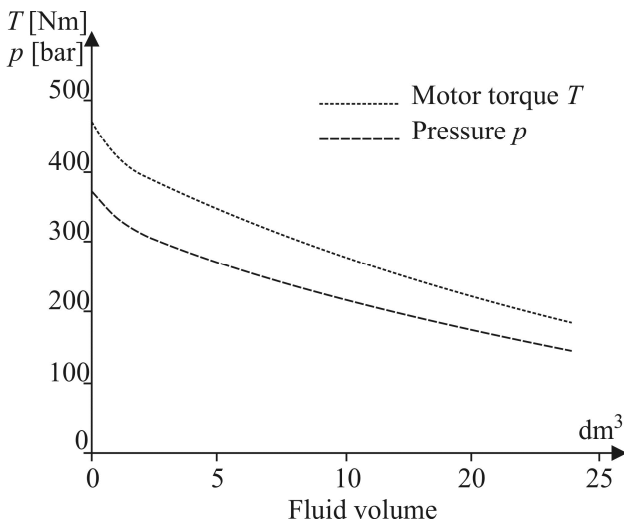


Figure 8. Diagram showing the variation of pressure and motor torque as a function of fluid volume.

4. CONCLUSION

The hydrostatic transmission enables power transmission with a continuously variable transmission ratio, meaning that the prime mover can be always operated in the optimal part of its power range. Power is transmitted using high pressure fluid flowing through small diameter pipes, so, there is no need for a large propeller shaft and rear axle. This enables the designer to lower the floor of the vehicle, which significantly improves the ease of access for passengers.

The hydrostatic transmission is interesting from an economic point of view as it brings a simple way to recover kinetic energy by using accumulators. Thanks to the technology developed for the compressed natural gas industry, the accumulators may be manufactured from lightweight composites, which significantly reduces their mass. Furthermore, to reduce environmental oil pollution, vehicle transmissions may create an important area for the application of water based fluids (with adequate protection from freezing), .

It can be expected that the application of hydrostatic transmission will expand from the current areas of machinery and construction vehicles to other areas in which the recovery of kinetic energy could bring significant energy savings, such as waste management, public transport and delivery services.

5. LIST OF SYMBOLS

adiabatic constant	κ ,	-
braking power	P_k ,	kW
coefficient of static friction	μ ,	-
cross section area	A ,	m ²
density	ρ ,	$\frac{\text{kg}}{\text{m}^3}$
displacement	V_g ,	cm ³
drag factor	C ,	-
drag force	F_z ,	N
engine torque	T_{mot} ,	Nm
force	F ,	N
friction force	F_{tr} ,	kN
front motor torque	T_1 ,	Nm
height	h ,	m
high pressure accumulator energy	E_{VT} ,	J
highest tractive force	F_{max} ,	kN
highest wheel torque	T_{kmax} ,	Nm
hydromechanical efficiency factor	η_{MH} ,	-
kinetic energy	E_k ,	J
length	l ,	m

low pressure accumulator energy	E_{NT} , J	rolling resistance factor	μ_k , -
mass	m , kg	rolling resistance force	F_k , N
maximum flow rate	q , $\frac{\text{dm}^3}{\text{min}}$	slope angle	α , rad
maximum input power	P_{\max} , kW	theoretical maximum vehicle	
maximum gross weight	m , kg	speed with regard to drag	v_{\max} , $\frac{\text{m}}{\text{s}}$
maximum permissible speed	n_{\max} , min^{-1}	torque at maximum rated power	$T_{P_{\max}}$, Nm
maximum vehicle speed	v_{\max} , $\frac{\text{m}}{\text{s}}$	total charging energy	E_1 , J
maximum speed at full displacement	$n_{V_{\max}}$, min^{-1}	total discharging energy	E_2 , J
mechanical transmission efficiency	η_{meh} , -	tractive force	F_V , N
number of revolutions	n , min^{-1}	tractive force on front wheel	F_1 , N
outer thread diameter	r , m	tractive force on rear wheel	F_2 , m^4
power	P , kW	transmission ratio	i , -
pressure	p , bar	vehicle weight	G , N
pressure difference	Δp , bar	volumetric efficiency factor	η_V , -
pump inlet pressure	p_1 , bar	width	w , m
pump outlet pressure	p_2 , bar		
rear motor torque	T_2 , Nm		

REFERENCES

- [1] MAN Bus Chassis [online] Available: www.manbusandtruck.com
- [2] Reimpell, J., Stoll, H., Betzler, J. W.: *The Automotive Chassis: Engineering Principles*, Butterworth – Heinemann, Oxford 2001
- [3] Stokes, A.: *Manual Gearbox Design*, Butterworth – Heinemann, Oxford 1992
- [4] Avallone, E. A., Baumeister, T.: *Marks' Standard Handbook for Mechanical Engineers*, McGraw - Hill, New York, 1996
- [5] Axial Piston Variable Motor AA6VM [online] Available: www.boschrexroth.com
- [6] Chapple, P. J.: *Principles of Hydraulic System Design*, Coxworth Publishing Company, Oxford 2003
- [7] D2066 LOH and LUH [online] Available: www.man-engines.com
- [8] Axial piston-compact unit AA4CSG [online] Available: www.boschrexroth.com
- [9] Bladder type accumulators model HAB [online] Available: www.boschrexroth.com

Received:
06.07.2011

Accepted:
25.10.2011

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