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IHT controlled serial hydraulic hybrid passenger cars

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ABSTRACT

The 'hybrid' is a fairly new serial hydraulic hybrid drive line concept. The term 'hybrid' is used for hydraulic drive lines in which four constant displacement Floating Cup (FC) wheel drive units are pressure controlled by means of two Innas Hydraulic Transformers (IHT). It has been shown in earlier publications that a four wheel drive hybrid drive line with wheel hub motors realises reductions of fuel consumption and CO₂ emission of more than 50%.

In modern passenger cars, there is a large gap between - on one hand - the engine power and wheel torque necessary to fulfil the maximum performance requirements and - on the other hand - the quite modest power and torque demand in normal, every-day use. This gap is the main reason for the rather bad energy efficiency of modern passenger cars. Contrary to other hybrid concepts, the hybrid can bridge this gap and achieve large reductions without compromising performance.

This paper shows how the use of floating cup technology, the pressure amplification capability of the Innas hydraulic transformer (IHT) and a four wheel drive line topology, come together to realise this. After that a serial hybrid front wheel drive line is presented, which can realise similar reductions but uses only one IHT and two FC drive units, which attach to the front axle differential.

Finally, some ideas for other serial hybrid drive lines are given.

NOMENCLATURE

β	IHT control angle	-
m	empty curb weight	kg
A_{fr}	frontal area	m ²
C_w	drag coefficient	-
r_{dyn}	dynamic tyre radius	m
f_{roll}	rolling resistance coefficient	-
V_{max}	maximum vehicle speed	km/h

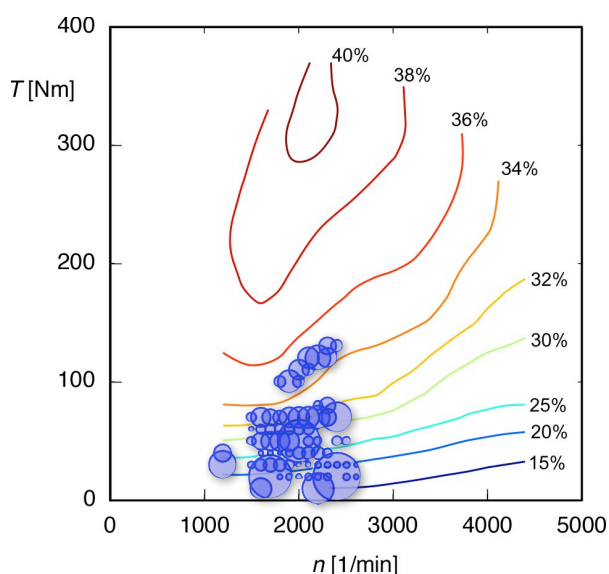
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1 INTRODUCTION

Driven by customer requirements, heavy competition and legislation, the average passenger car has evolved into the versatile machine it is today. Modern passenger cars are comfortable and safe and have low emissions. In order to realise the required acceleration, gradability and top speed, they have also become very powerful machines, with a high maximum wheel torque and with a large installed engine power. In average every-day driving, however, the required torque and power are much lower.

Because the conventional mechanical transmission, even if it is continuously variable, directly connects the internal combustion engine (ICE) to the wheels, the load power requirement at any point in time is equal to the ICE power requirement. Hence, the ICE is forced to run at operating points with low power, where its efficiency is much lower than its maximum efficiency.

Figure 1 illustrates this. It shows the operating points of the 100 kW diesel engine in an average family car, driving the New European Driving Cycle (NEDC), which is the European standard for fuel consumption and emission measurements. The specifications for this car are given in the same figure. The area of each bubble in the plot, is proportional to the amount of energy spent in that point of operation.



empty curb weight: $m = 1450 \text{ kg}$

frontal area: $A_{fr} = 2,26 \text{ m}^2$

drag coefficient: $c_w = 0,26$

dyn. wheel radius: $r_{dyn} = 0,315 \text{ m}$

rolling resist. coeff: $f_{roll} = 0,008$

maximum speed: $v_{max} = 190 \text{ km/h}$

gear ratios: 4.04/2.37/1.56/1.16/0.85/0.67

differential gear ratio: 3.5

Figure 1: Use of the ICE Engine in an average family car, driving the NEDC

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In the near future, very restrictive legislation for CO₂ emissions will be imposed to the passenger car industry. As CO₂ emissions are directly proportional to fuel consumption, this implies that ways have to be found to substantially reduce the fuel consumption of passenger cars with an internal combustion engine. *Figure 1* shows that there are basically two possible strategies to realise this.

- Replace the ICE by a less powerful or a 'down sized' ICE.

A less powerful engine will operate nearer to its efficiency 'sweet spot' and will thus consume less fuel and emit less CO₂. The obvious downside is a reduction in vehicle performance.

Down sizing, often presented as a new strategy, is in fact nothing else than improving the power density of the engine, thereby reducing the relative influence of the divers engine losses. This is an obvious improvement process, which has been pursued since the beginning of automotive industry.

- Break the hard link between the ICE and the wheels and introduce an energy storage. The ICE is then operated in on/off mode. When it is on, it delivers energy to the storage at operation points near the efficiency sweet spot. When the storage is full, the ICE is switched off.

The wheels are driven from the storage. Thus the installed ICE power can remain high in order to meet performance requirements but without the usual penalty in average efficiency. The storage also enables brake energy recuperation.

The latter strategy is preferable, as it realises low emissions without compromising the performance of the vehicle. The resulting vehicle type is generally called 'hybrid'. Depending on the type of storage system used, a further distinction can be made: 'electric hybrid' if batteries are used, 'hydraulic hybrid' on case of accumulators.

For any hybrid, if it has to realise a large reduction in CO₂ emissions while also complying to normal performance requirements, it is of the utmost importance that the components driving the wheels, also have high part load efficiencies. If they have not, the very same situation which is avoided at the primary side of the drive line - prolonged operation in low efficiency regimes - would occur at the secondary side. There also, the wheel drive units have to be laid out for the maximum performance but will operate mostly in part load conditions. A lower part-load efficiency of these

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units, as is the case with conventional hydraulic or electric drivelines, would largely spoil the gain achieved at the engine side.

Still fairly recently, Innas BV has introduced a new type of serial hydraulic hybrid drive line for passenger cars /Ach07/, based on the Floating Cup (FC) displacement principle and the Innas Hydraulic Transformer (IHT) it had developed earlier /Ach97/. This type of drive line was named 'hy^drid' in order to clearly distinguish it from the term 'hybrid', which the general public has come to associate with the electric hybrid. In a joint research project with the IFAS of the RWTH Aachen, it was shown that the hydrid drive line concept can reduce the fuel consumption and the CO₂ emission of an average family car by roughly 50%, without compromises regarding its performance /Ach08/.

The FC and IHT technologies are the key enabling factors for the success of the hydrid drive line concept, as with these two technologies, high part load efficiencies can be realised for the wheel drive.

This paper centres on IHT controlled serial hydrid drive lines for passenger cars. Starting with the hydrid four wheel drive that was presented in the earlier publications [Ach09], the second chapter explains how IHT control, drive line topology and the use of FC technology enable high part load efficiencies, while meeting all performance specifications of passenger cars with a conventional mechanical drive line. In the third chapter, a hydrid front wheel drive system is presented. Finally, in the fourth chapter some further alternative hydrid drive line topologies are discussed.

2 SERIAL HYDRID FOUR WHEEL DRIVE

The hydrid drive line on which the earlier publications on this subject are based, is presented in **Figure 2**. It is based on a hydraulic grid, the Common Pressure Rail (CPR), which consists of a high pressure line and a low pressure line. An accumulator is connected to each line.

A fixed displacement pump, driven by the ICE, upholds the pressure potential between the two lines of the CPR. When the pressure in the high pressure line drops between the lowest acceptable filling level of the accumulator, the ICE is switched on

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and oil is pumped from the low pressure line to the high pressure line. When the pressure crosses the highest acceptable filling level, the ICE is switched off.

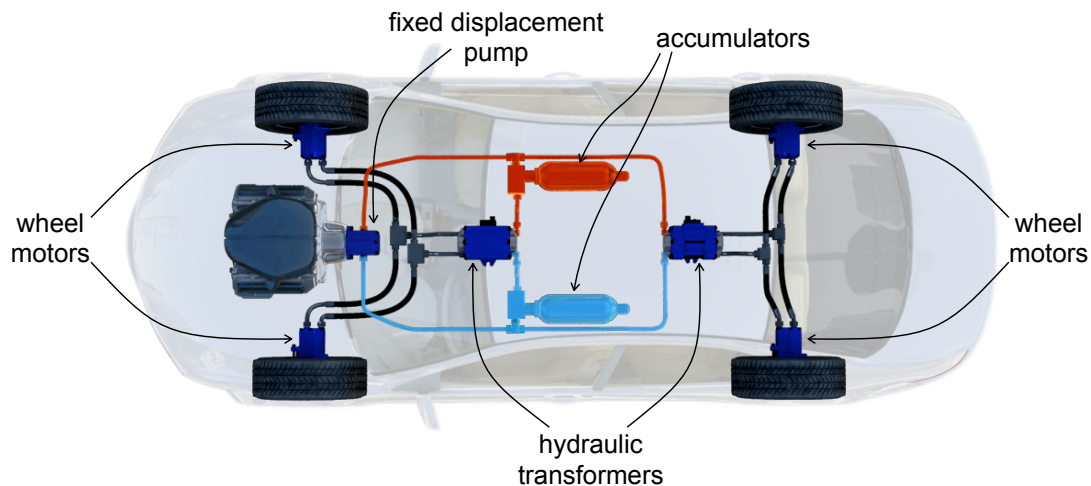


Figure 2: The hydrostatic drive line for the hybrid four wheel drive.

With the desired pressure range of the high pressure accumulator defined, the size of the fixed displacement FC pump can be chosen such, that this pressure range translates to an ICE torque range that spans its best efficiency area. For the ICE of *Figure 1* for instance, a torque range from 50% to 100% of the maximum ICE torque, is a good choice. At a 200 to 400 bar pressure range - which is a good choice for a bladder accumulator - the required size for the fixed displacement pump is 56 cm³.

In this way the strategy described in **Chapter 1** is realised: when it runs, the ICE is forced to operate at high efficiencies, low-efficiency part load operating points are avoided totally. As was explained before, if this strategy is to be successful, it is imperative that the associated efficiency gain at the energy source, is not lost in the transmission system. Also this transmission system, if no compromises are made with respect to vehicle performance, has to be laid out for high maximum load requirements but will operate relatively lightly loaded during most of the time.

At this secondary side of the drive line, the gap between the maximum load and the average load, is mostly due by the required maximum wheel torque. This maximum wheel torque is determined by requirements like the acceleration performance, gradability with maximum load and maximum trailer weight or maximum gradability at

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curb weight. For the average family car, for instance, 50% empty gradability is a quite common requirement. For the vehicle parameters presented above, this implies a maximum total wheel torque requirement of around 2050 Nm. **Figure 3** shows the gap between this maximum torque requirement and the torque demand on the NEDC and FTP75 drive cycles.

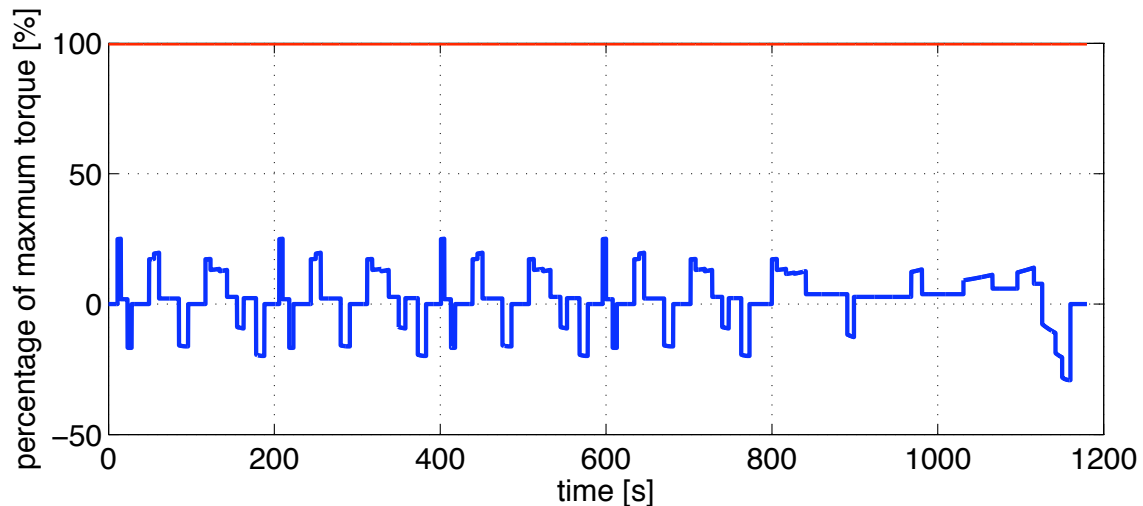


Figure 3: Maximum torque and torque on the NEDC and the FTP75.

In the hybrid drive line presented here, three factors enable the bridging of this gap. They will be discussed in the next three paragraphs:

Four wheel drive.

In this hybrid lay-out, each wheel gets its own fixed displacement wheel motor (which may be integrated in the hub or drive the wheel through a wheel shaft). The two wheel motors per axle are switched in differential mode. For each axle, there is one IHT, controlling the pressure differential over the two wheel motors of that axle.

When maximum torque is required, both the front and the rear axle drive systems are used. During the normal every day driving represented by the NEDC or the FTP75 cycles, only the front wheels are driven, the rear wheel drive system is inactive.

In this way, the gap between maximum and average requirements is diminished greatly.

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Pressure amplification.

The IHT can amplify the pressure in the CPR. It is laid out in such a way that at any pressure level within the 200 to 400 bar range chosen for this CPR, can be transformed to a maximum pressure of 500 bar at the wheel motors. With this pressure amplification capability, the equivalent wheel motor capacity for this average family car, necessary to realise the maximum total wheel torque of 2050 Nm, is approximately 260 cm³. If variable wheel motors would be used, the minimum rail pressure of 200 bar would be the determining case and the total equivalent wheel motor capacity would be 650 cm³. At normal every day use, these variable wheel motors would then operate in deep part load conditions.

High hydromechanical starting efficiency.

One of the distinctive differences between the FC principle and conventional axial piston displacement principles, is the total absence of pressure dependent side forces between the pistons and the walls of the cavities in which they move. As a result, the hydromechanical starting efficiency of FC units is much higher. The measurement results presented in **Figure 4** illustrate this.

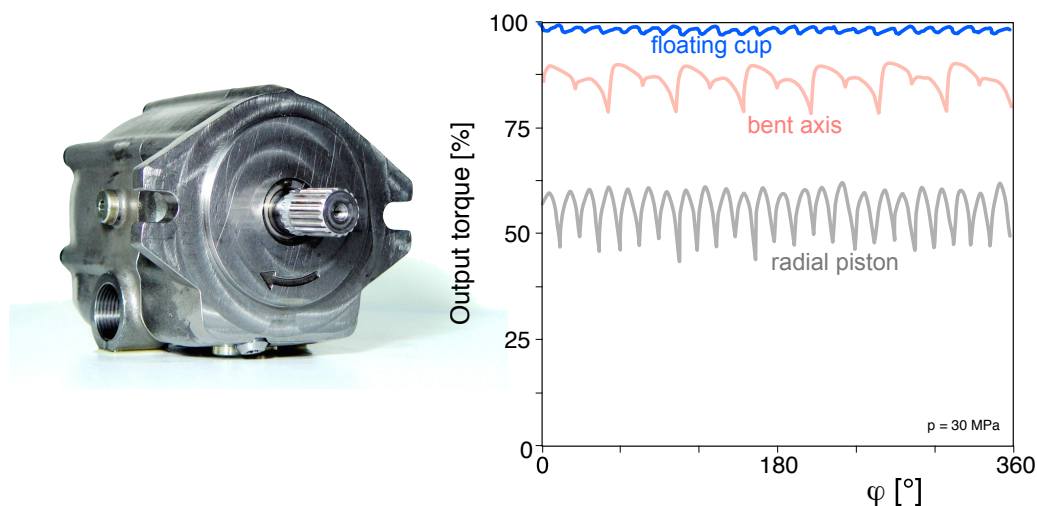


Figure 4: Low speed torque of axial piston units.

The figure also shows the effect of the larger number of pistons in the FC unit (typically 24 instead of the 7 pistons in the bent axis unit): the torque breakdowns which occur when a piston changes kidney, are much lower. Because of these two

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effects, smaller FC units can be chosen for the same start-up torque under load. Smaller units, again, mean that the system will operate less at part load conditions.

That the hybrid four wheel drive line of *Figure 2* indeed succeeds in avoiding deep part load operation, is illustrated in *Figure 5*, which shows at which efficiency points the FC IHT and the FC wheel motors operate during the NEDC. The area of the bubbles represents the amount of energy spent in that operating point.

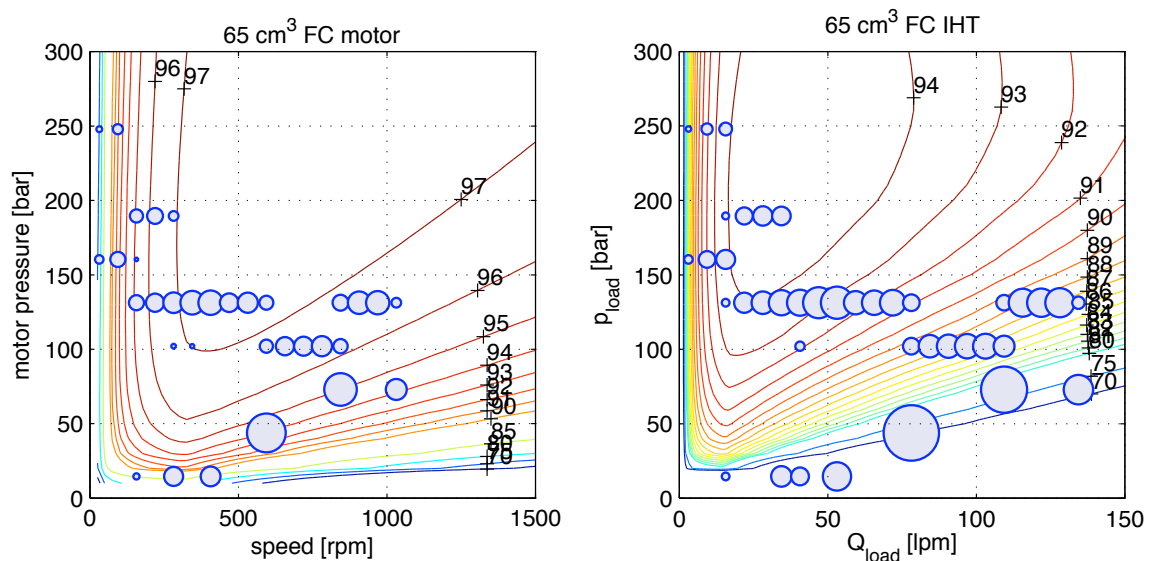


Figure 5: Operating points of the FC IHT and FC wheel motors on the NEDC

In this hybrid drive line, there are two FC IHTs, one for each axle. With this arrangement, the total wheel torque and the torque division between front and rear axle have both become continuously variable.

3 THE SERIAL HYBRID FRONT WHEEL DRIVE

If space and weight requirements are restricting - as may be the case with small passenger cars - the number of FC drive units and the number of FC IHTs can be reduced by driving the front wheels only, through the existing front axle differential gear. The resulting system topology is presented schematically in *Figure 6*.

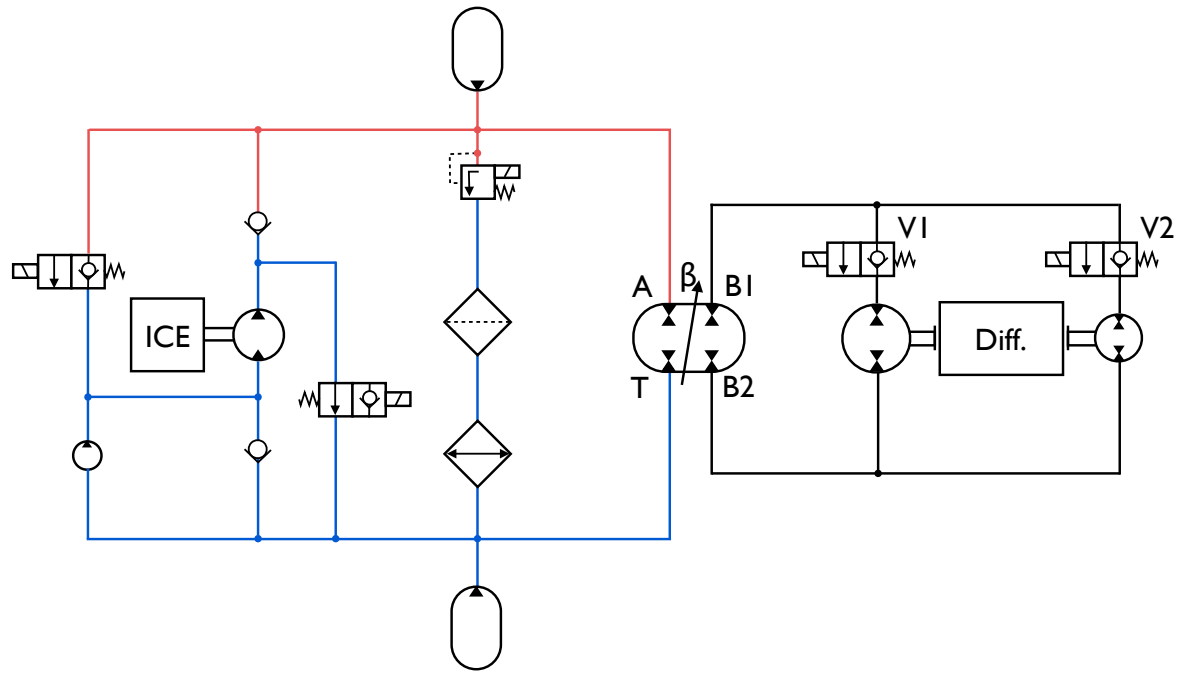


Figure 6: Topology of the Serial Hybrid Front Wheel Drive

In this topology, two FC units are attached to two pinions, each attaching to the ring gear of the front axle differential. Each unit can be connected to or disconnected from its pinion by means of a clutch. The clutches contain synchromesh rings and connect the drive unit to either the pinion, or to the foundation, thereby holding it in a fixed position.

The IHT design used in this alternative serial hybrid drive line, is the same as the one in the original hybrid drive line design. The design contains two integrated poppet valves, which together define in which quadrant the IHT can operate. These integrated valves are switched according to the drivers inputs (drive direction selector, gas pedal and brake pedal):

- If the driver selects forward driving and applies the gas pedal, the IHTs control angle (β) is turned in positive direction, resulting in a pressure at connection B1, which is a function of the control angle. The internal valves are switched in such a way that the pressurised oil can only flow out of connection B1 and oil at the low rail pressure level, can only flow back into the IHT through connection B2.
- If the driver has selected forward driving and applies the brake pedal, the IHTs control angle is turned in negative direction, B2 becomes the pressurised connection (pressure again a function of the control angle) and B1 the low

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pressure connection. The internal valves are switched to ensure that low pressure oil can only flow out of connection B1 and pressurised oil can only flow in to connection B2. This implies that the vehicle will brake to a standstill and remain that way.

- If the driver selects backward driving and applies the gas pedal, the IHT control angle is turned in negative direction, B2 is pressurised to a corresponding level and the valves ensure that pressurised oil can only flow out of B2 and low pressure oil can only flow into B1.
- When backwards driving is selected and the brake pedal is applied, the flow directions, determined by the internal valves, remain the same, but the pressurised side switches to B1 and the low pressure side to B2.

Valves V1 and V2 in the schematic of *Figure 6*, are integrated in the FC drive units.

When the driver selects forward driving, the positions of valves V1 and V2 determine which of the drive units will be subjected to the propelling pressure of output B1 of the IHT. Of course, in order to transfer this propelling pressure to a propelling torque, the clutch of that unit has to be engaged first, in order to connect it to the drive line.

By carefully selecting the drive units sizes, three 'gear steps' can be realised, depending on which drive units are connected to the IHTs output and coupled to the differential. When both units are connected, the maximum required total wheel torque can be realised. For lower torque regimes (like the every day driving represented by the standard NEDC or FTP75 drive cycles) only one unit will be connected. The gap between maximum torque requirement and the average torque demand, is thus reduced by splitting the different drive regimes over different units instead of splitting them over 4Wd or FWD, as it is done in the hybrid described in *Chapter 2*.

As soon as the swept volume, the port plate geometry, and the maximum speed of an IHT have been chosen, a relationship between its maximum output flow and its control angle can be established. The pressure transformation factor of the IHT is a function of its port plate geometry and its control angle.

With these relationships (for their derivation, see for instance /Ach97/), for given pressure levels in the two lines of the CPR, a relationship between output pressure

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and maximum flow can be constructed. Through the three possible gear steps, which represent three different total swept volumes coupled to the differential gear, this curve translates to three curves for the maximum total drive torque as a function of the maximum vehicle speed in that 'gear'.

For the vehicle presented in *Chapter 1*, these curves have been constructed for a lay-out with one 45 cm³ drive unit and one drive unit and one 30 cm³ drive unit, driven by a 65 cm³ IHT with a maximum speed of 3500 rpm. For the calculation, the pressures in the CPR lines are set to 400 and 10 bar, as this is the situation the accumulator state of charge controller will realise when the ICE has to deliver maximum power. The curves are shown in **Figure 7**.

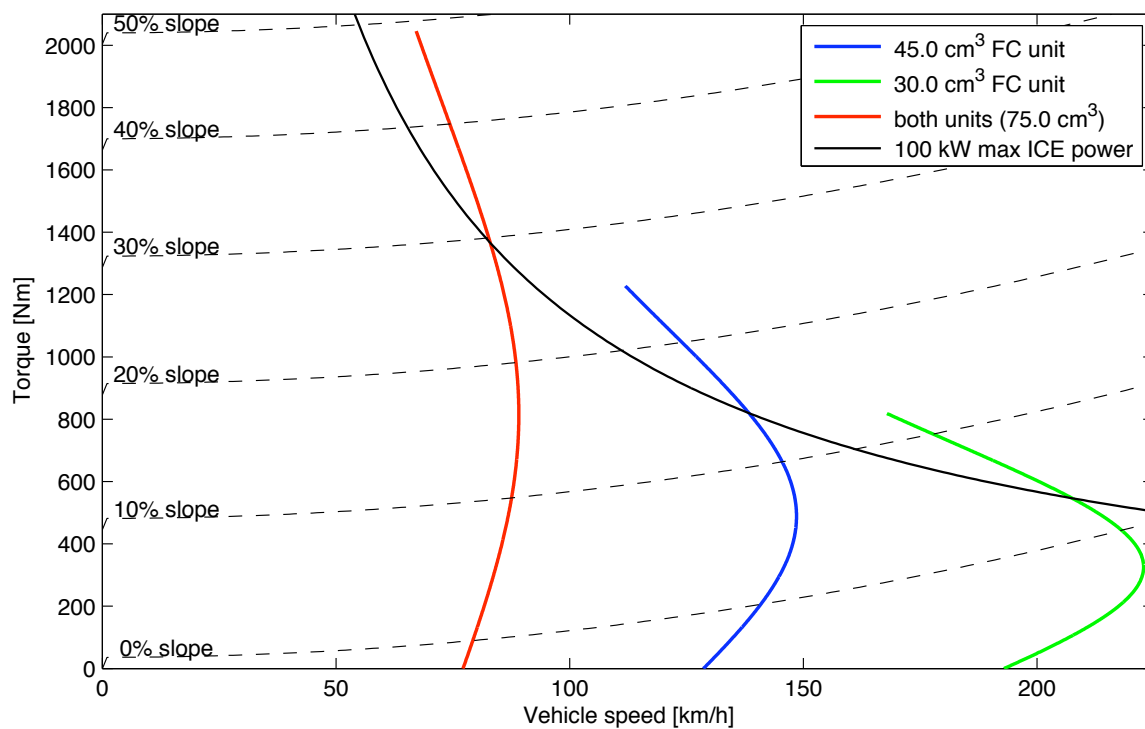


Figure 7: Topology of the Serial Hybrid Front Wheel Drive

As can be seen in the figure, with this arrangement with three possible 'gear-steps', the three curves closely follow the maximum power curve of the ICE

When, in forward propelling mode the drive units have to be connected or disconnected from the drive line, the IHTs control angle is first set to 0 degrees, thereby bringing the B1 and B2 pressure levels to the same low CPR pressure level. After that a synchronised operation of the appropriate valve and clutch, speeds up or

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decelerates the unit. When the unit has reached its end speed, the IHTs control angle is brought to the new required position. The whole process, in timing and feel, closely resembles a gear-shifting operation in an automatic gearbox.

In forward braking mode, a torque interruption, however briefly, is probably not acceptable. Hence, forward braking is always done with both drive units connected, thereby enabling smooth, continuous IHT control of the braking action.

For the energy efficiency of the hybrid vehicle, this is of very little consequence.

In order to keep the system lay-out simple, backward propelling and braking are both performed with both units coupled to the differential.

4 OTHER POSSIBILITIES FOR SERIAL HYBRID CONFIGURATIONS

Building on the principles described in the previous chapters, there are a lot more possibilities to realise serial hybrid drive lines.

For instance, in parallel to the hybrid circuit lay-out described in chapter 3, a mechanical gear box can be arranged. This gearbox can be a simple automated gear box, with one or two gears, which would only be used for driving at elevated speeds. This is interesting because above a certain speed, the constant speed fuel consumption of the the serial hybrid is higher than that of a vehicle with a conventional mechanical drive line. The reason for this is that at high constant speeds, the operating point of the ICE in the mechanical drive line also shifts into higher efficiency regions. So - for any serial hybrid drive line - there is a break-even constant speed, at which the lower serial drive-line efficiency is no longer off-set by the lower fuel consumption of the ICE. For hybrid passenger cars, this break-even speed typically lies at around 90 km/h. This a lot higher than for most electric hybrids but it may still be worthwhile to be able to switch to a mechanical drive above a certain speed. The extra gearbox may be a simple automated one, as its switching comfort can be augmented by using the parallel hydraulic system. Still, the added weight, costs and complexity have to be weighted against the gain.

Another interesting possibility may be to use a hybrid system similar to that of *Chapter 3*, connected to the transfer box of a typical 4WD Special Utility Vehicle. Such a 'SUV' typically has very demanding specifications. By leaving the 4WD and

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its torque division capabilities between front and rear axle mechanical, the number and size of the FC units can be minimised.

It is also possible to sacrifice the variable torque division between front and rear axle, that the drive line presented in *Chapter 2* offers, and use only one IHT. This IHT then drives both axles when high torque is required and only the front axle in normal daily drive conditions. This is a two regime set-up: a 4WD regime with fixed torque division and a FWD regime. A rear wheel drive (RWD) regime is theoretically also possible, but switching between FWD and RWD is not desirable for vehicle handling reasons.

CONCLUSION

The 'hybrid' serial hydraulic drive-line concept, when used in a passenger car, can realise large reductions in fuel consumption and CO₂ emissions, without sacrificing vehicle performance. It can realise this because of the use of floating cup technology, the pressure amplification capability of the Innas hydraulic transformer (IHT) and last but not least, by carefully choosing the drive line topology. The hybrid four-wheel drive topology, presented in earlier publications, is not the only possible one. Other topologies exist. Depending on the vehicle type, they may be preferable.

REFERENCES

- /Ach97/ **Achten, P.A.J., Fu, Z., Vael, G.E.M.,** *Transforming future hydraulics: a new design of a hydraulic transformer*, Proc. of the Fifth Scandinavian Int. Conf. on Fluid Power, SICFP '97, Linköping, Sweden, 1997
- /Ach07/ **Achten, Peter A.J.,** *The Hybrid Transmission*, paper 2007-01-4152, SAE International, 2007.
- /Ach08/ **Achten, P.; Vael, G., Kohmäscher, T., Sokar. M.,** *Energy Efficiency of the Hybrid*, 6th Int. Fluid Power Conf., Dresden, 2008.
- /Ach09/ **Achten, P., Vael, G., Murrenhof, H., Kohmäscher, T., Inderelst M.,** *Low-emission Hydraulic Hybrid for Passenger cars*, Automobiltechnische Zeitschrift, Volume 111, pp. 378-384, 05/2009.