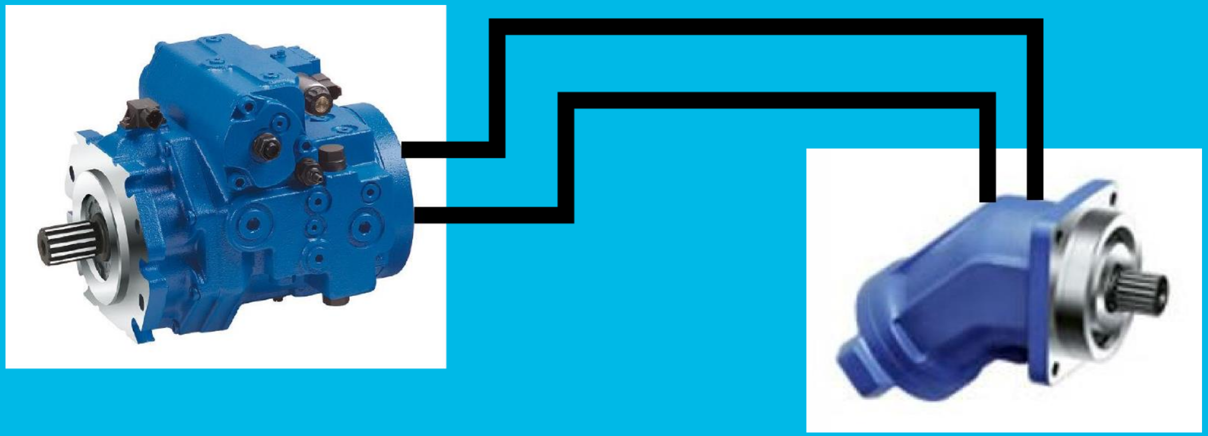
Hydro-Mechanical Transmissions

Liselott Ericson

Fluid power systems



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Agenda

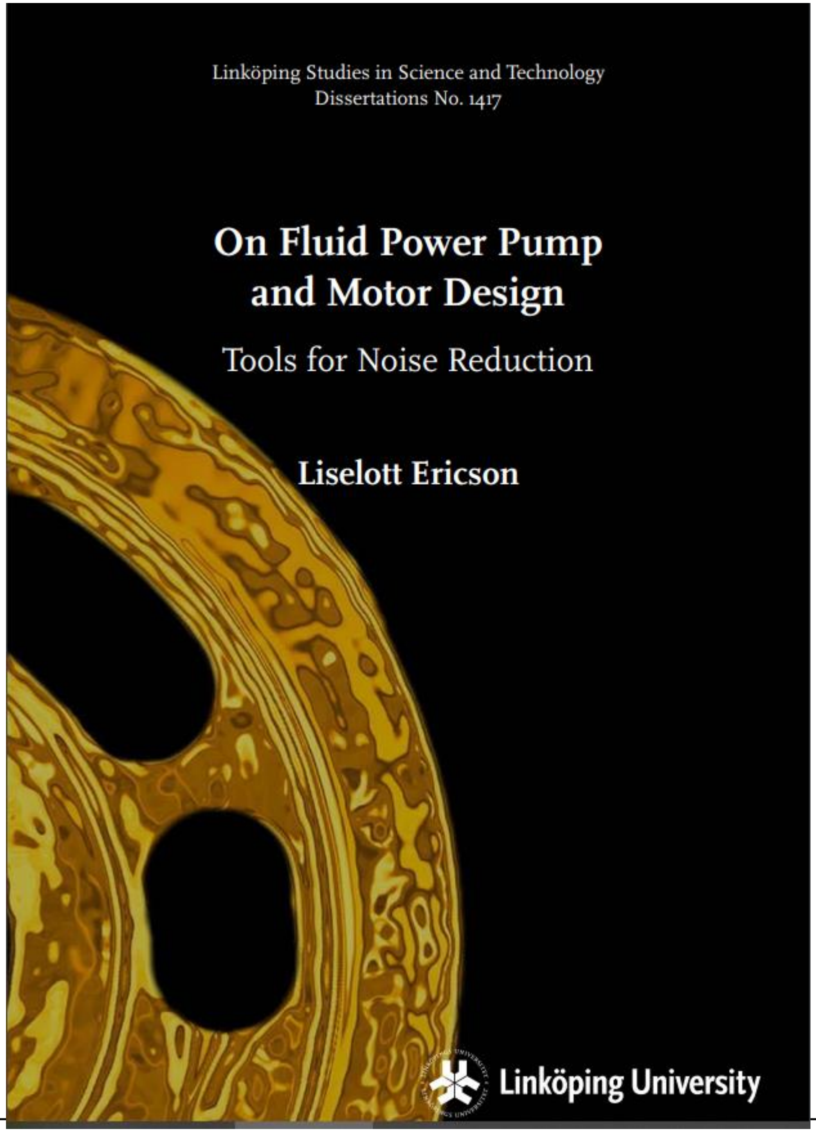
- But first a little bit about me
- Short course: Hydrostatic- and hydro-mechanical vehicle transmission. Static sizing considerations and design.

Research

- Master of Mechanical Engineering, Linköping University, 2006
- PhD in fluid and mechatronic systems, Linköping University, 2012
- R&D engineer at Parker Hannifin, 2011-2012
- Researcher at LiU since 2013
- Since 2019: Senior Associate Professor at Fluid and Mechatronic Systems

PhD thesis

- Investigated method to analyze noise in pumps and motors
- Investigated ununiformed placement of pistons
- Investigated differences for pump and motor flow pulsations

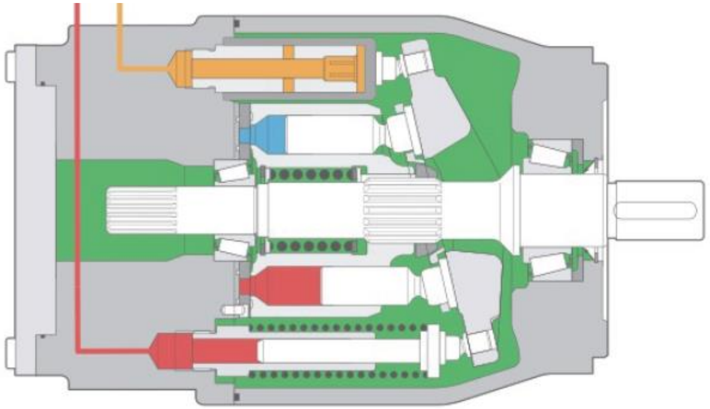


Current projects

- Energy efficient compact electrohydraulic component and system solutions for mobile machinery, E-hydraulics
- Efficient carbon free forest machines
- Improvements of proportional valve for electric autonomous mobile machine

Summary: Mobile machinery in different configurations

Own focus area – Hydraulic pumps/motors



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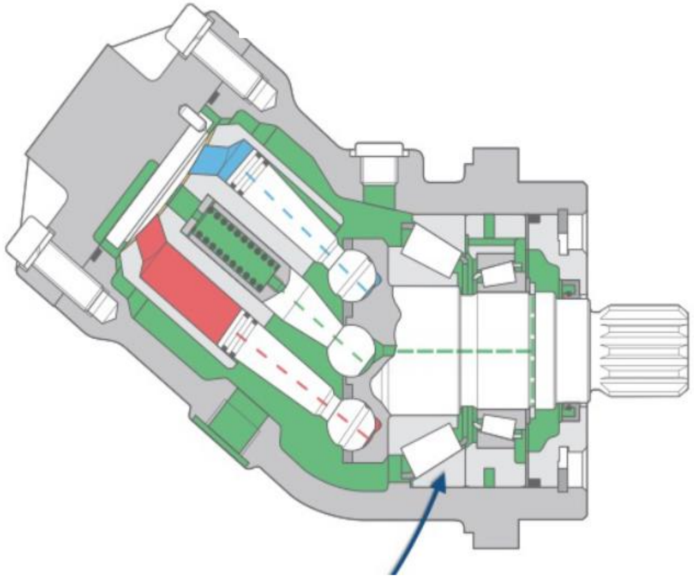

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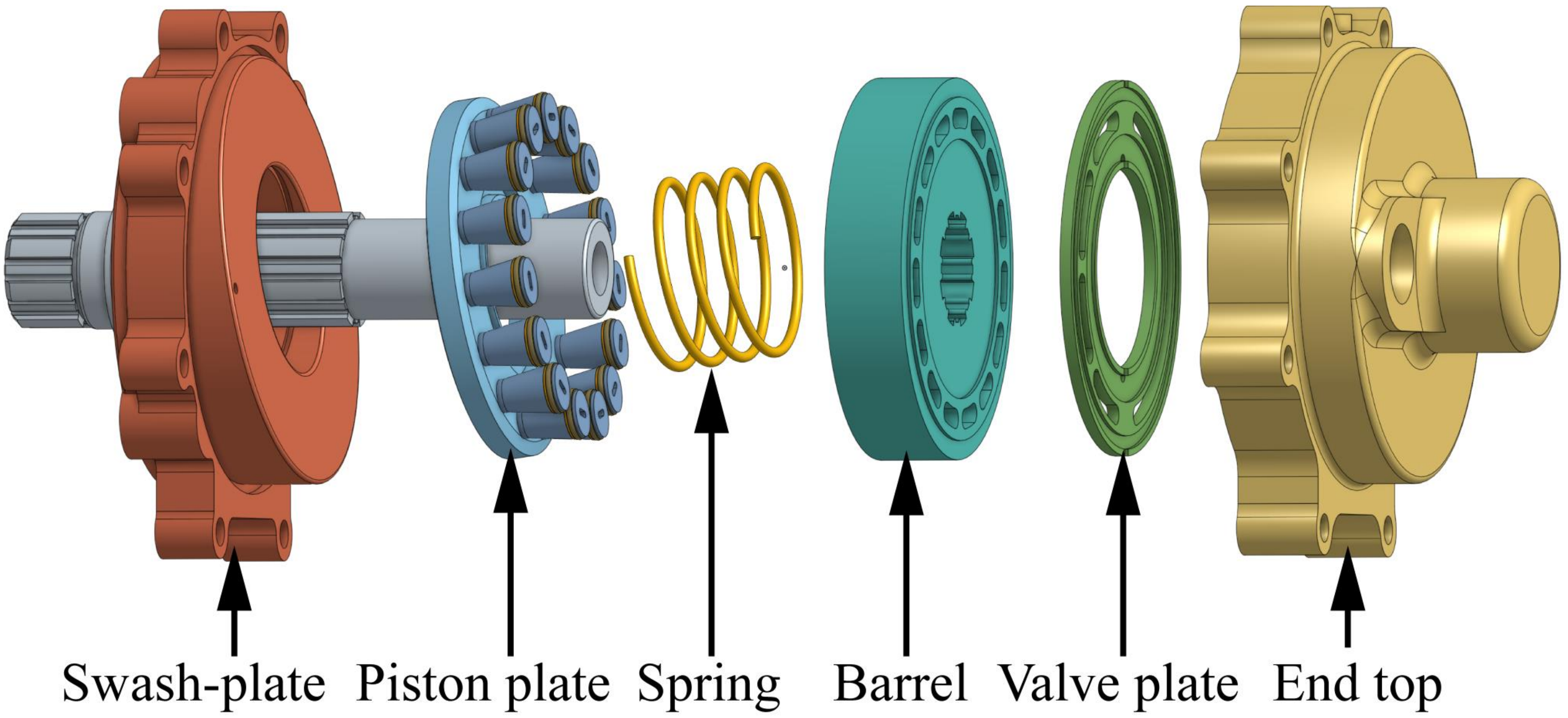
Swash-plate piston machine

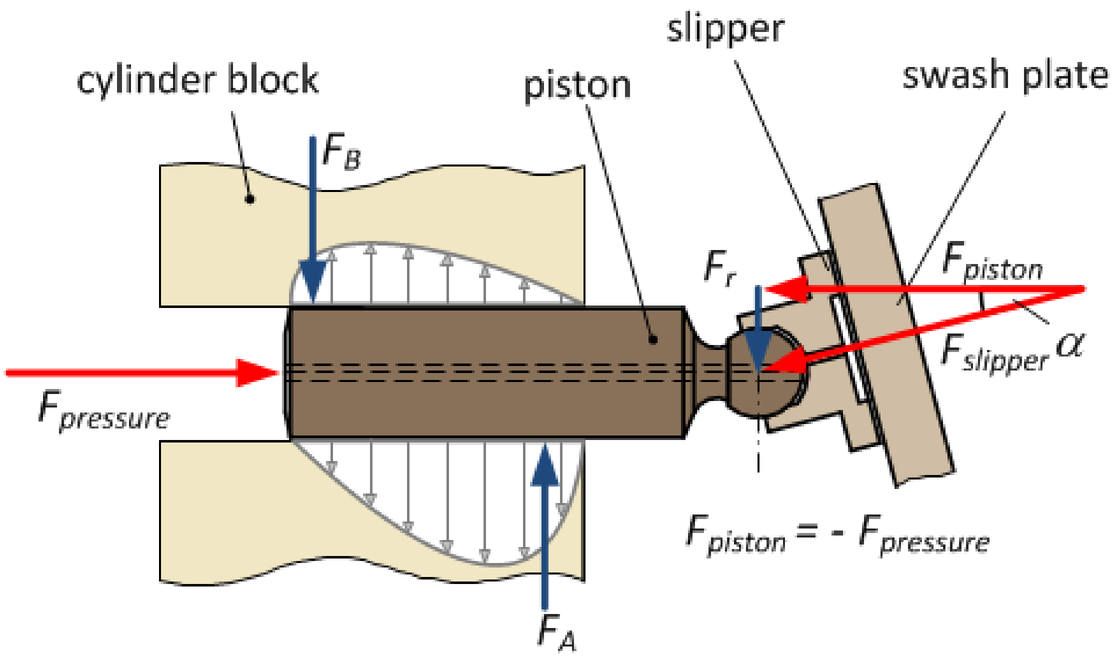
Bent-axis piston machine

Floating piston machine

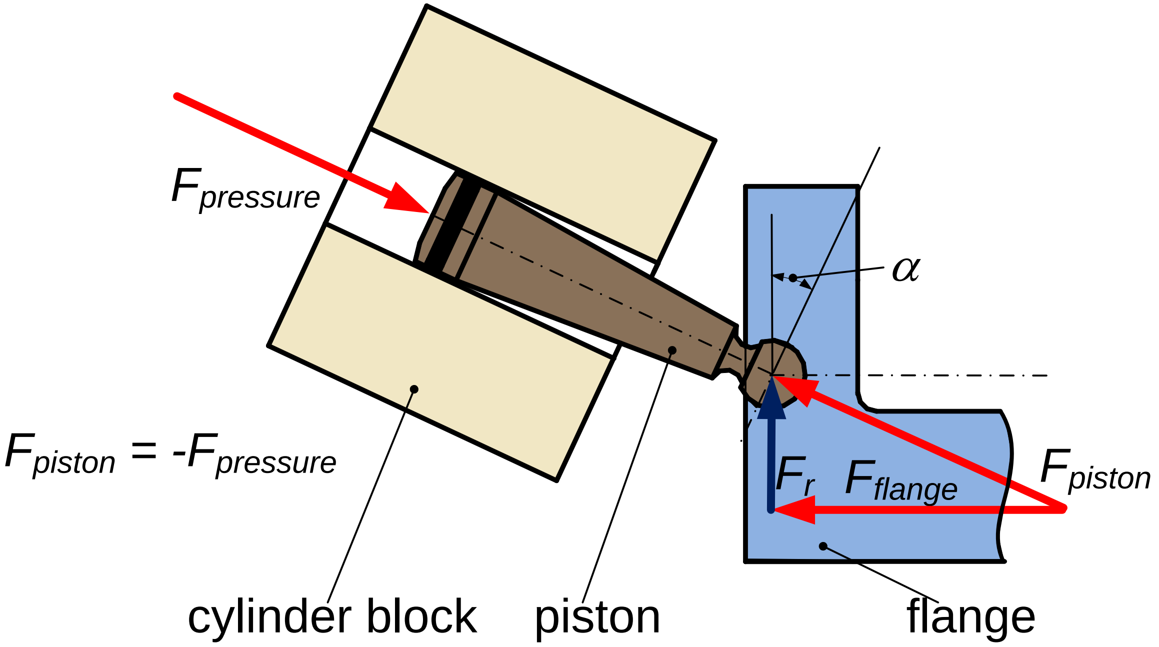
Main components of the floating piston



How is the torque created in traditional machines?



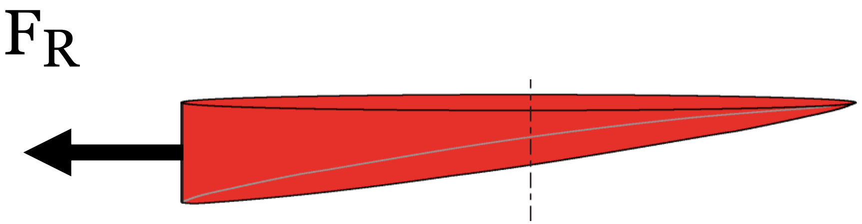
Swash-plate piston machine

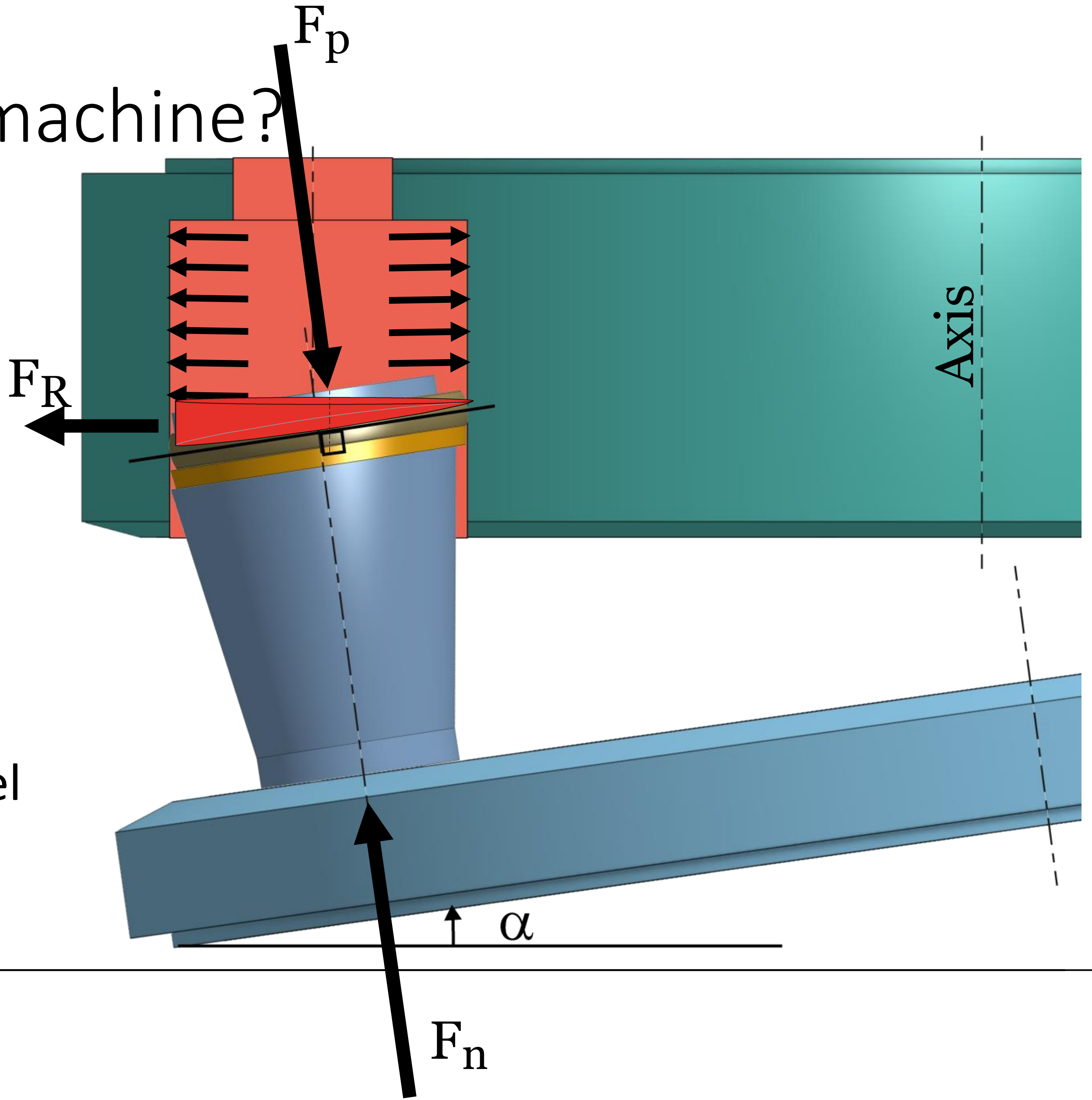


Bent-axis piston machine

Hubertus Murrenhoff, Presentation at Hydraulikdagarna 2015

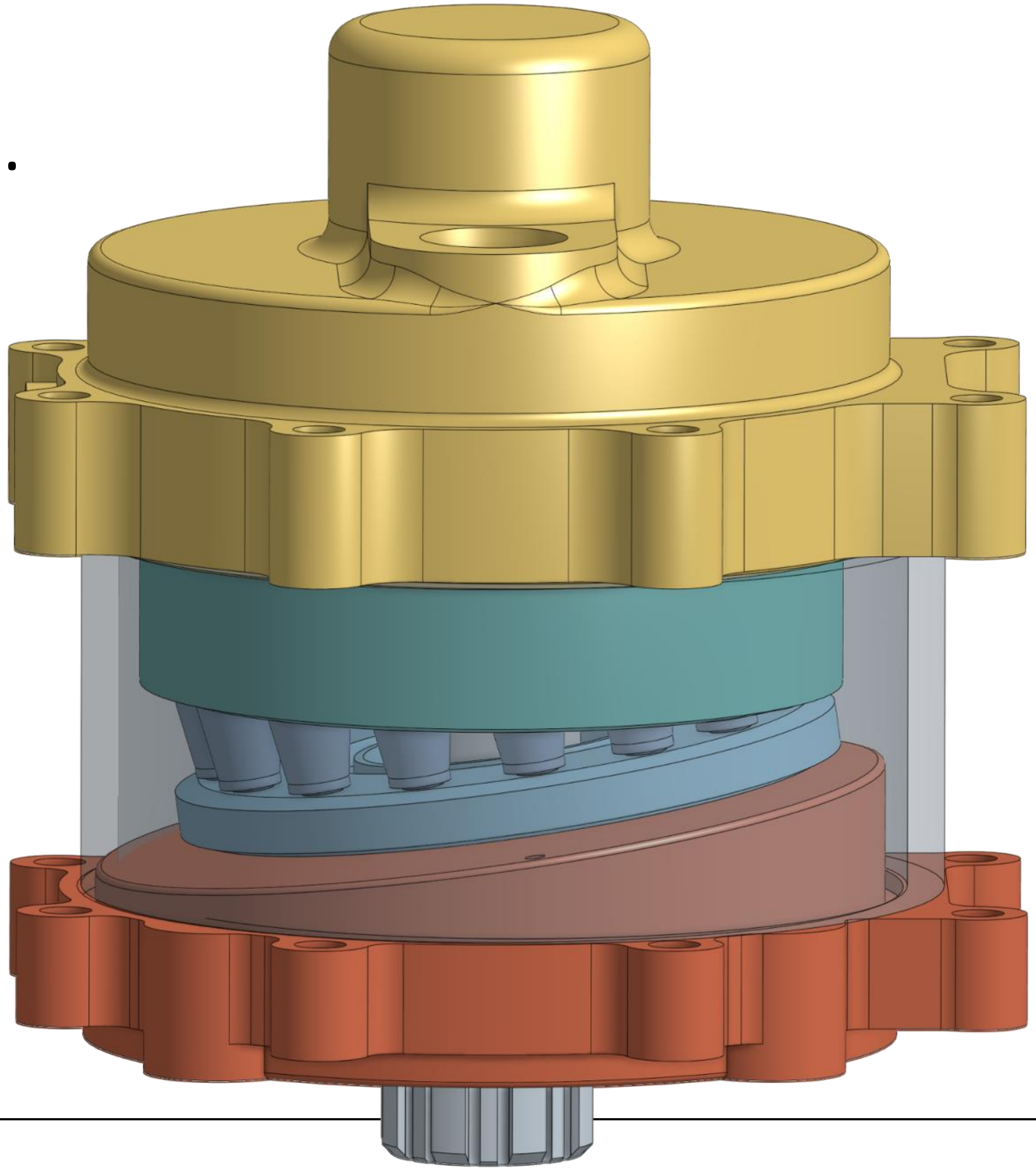
And in floating piston machine?





- The piston is always in angle α
- No radial loads on pistons
- The force acting directly on barrel
- No forces on sliding interfaces

Not a final product YET…..



Teaching – fluid power systems

- Fluid power systems (first course in hydraulic systems)
- Hydraulic servo systems
- Fluid power systems – advanced course
- Advanced project course – mechatronics
- Electro-hydraulic systems

Hydro-Mechanical Transmissions

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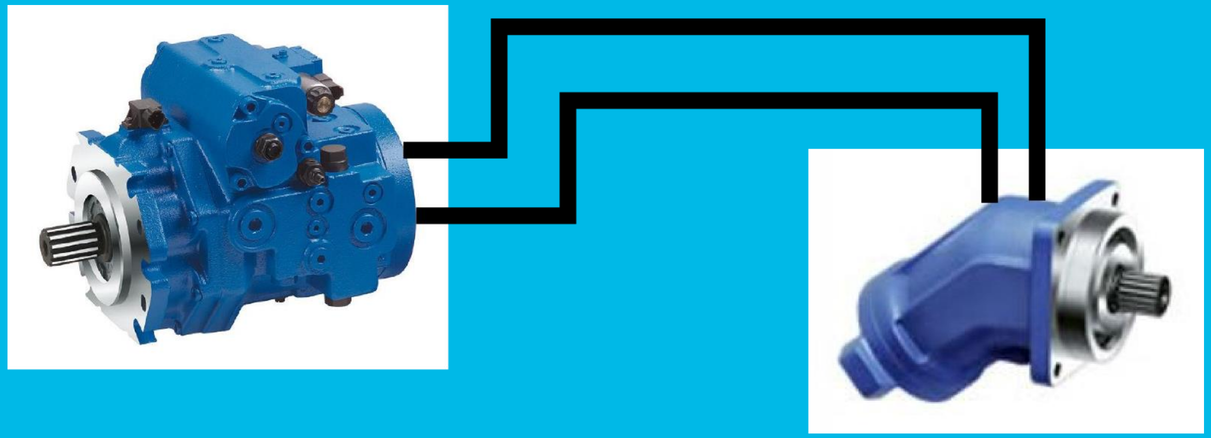

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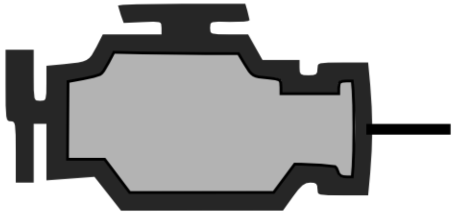
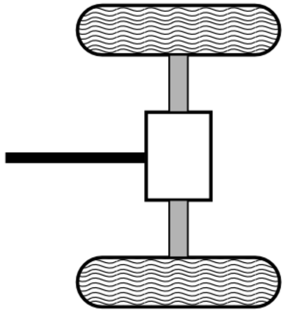
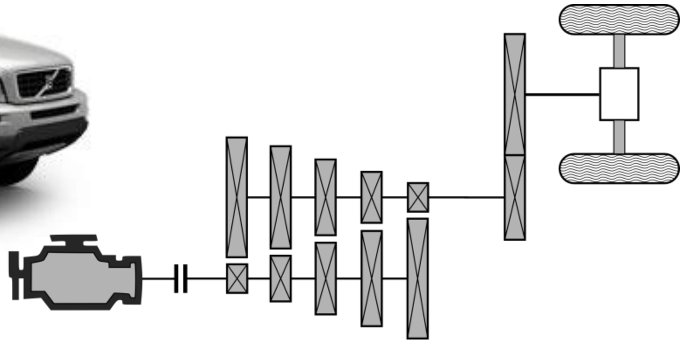


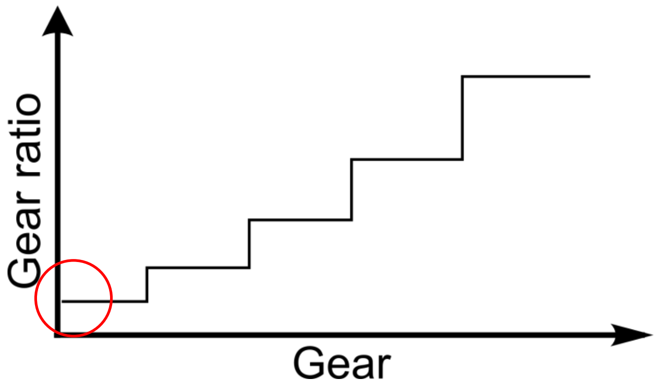
Transmission

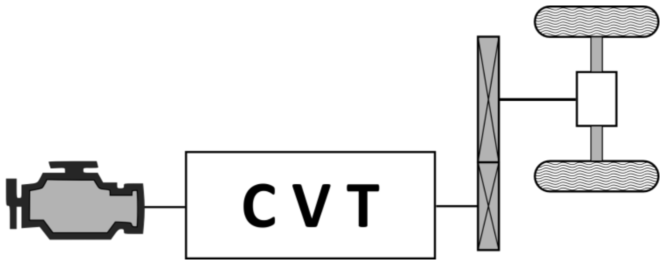


Transform rotational mechanical power from the engine to the wheels

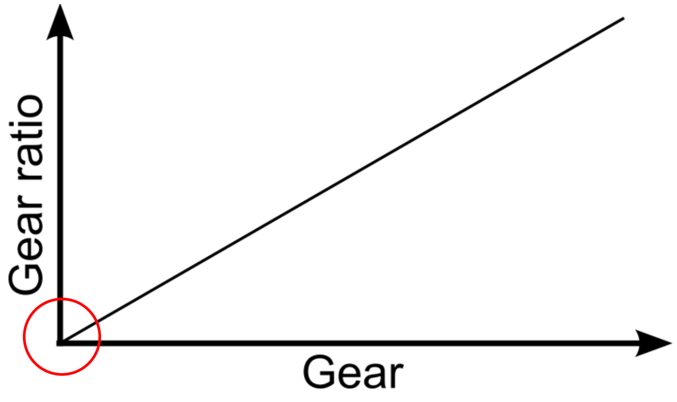
Car vs Wheel Loader

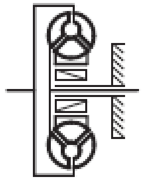
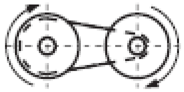
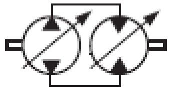
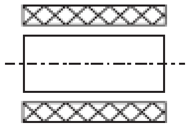






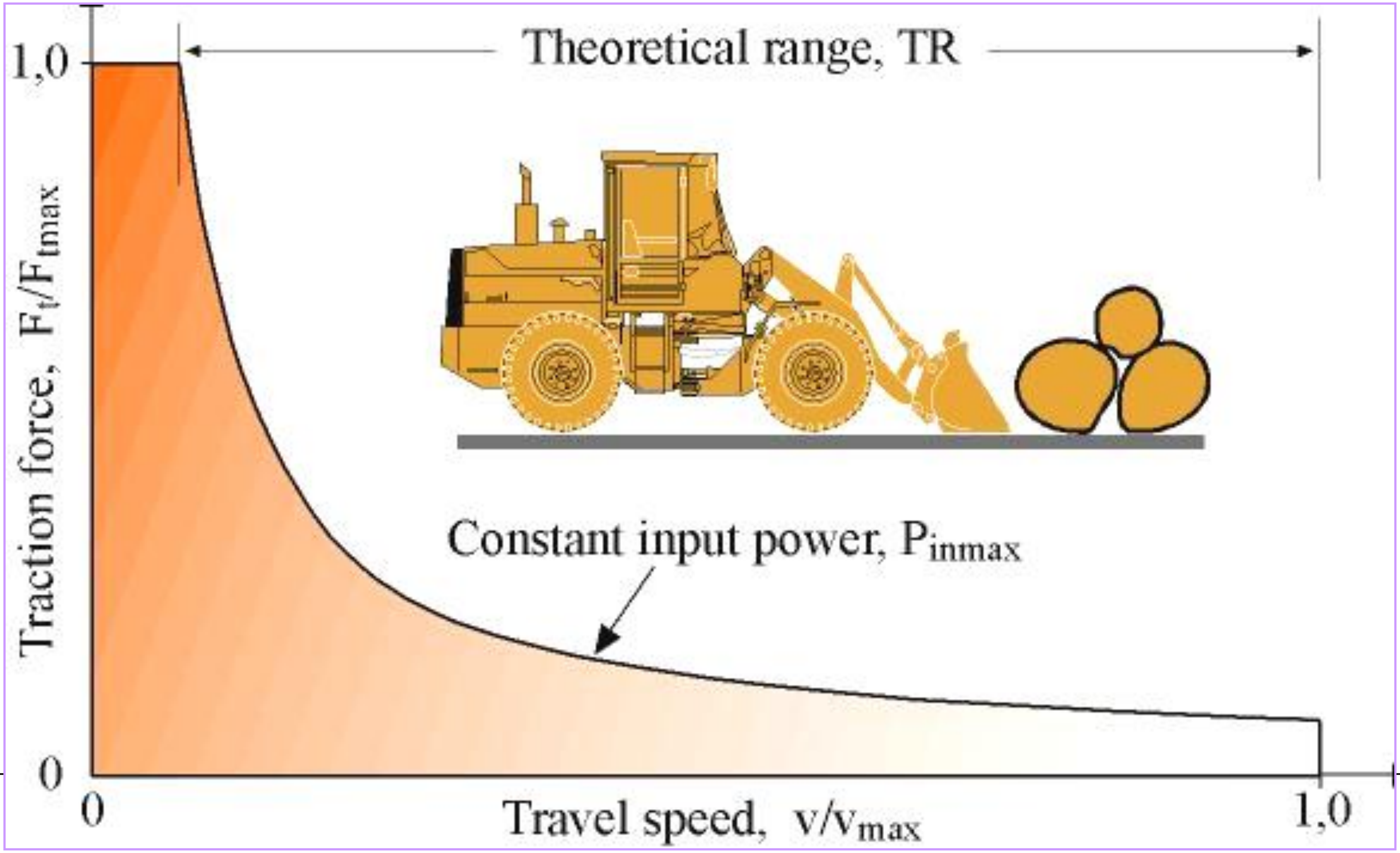
Continuously Variable Transmissions



Type of CVT			Principle of energy transmission	Ratio control	Application	Efficiency
1	Hydrodynamic		Mass forces at pump and turbine	Usually automatically by load	Important for pass. cars and construction machinery	poor
2	Mechanical		Traction forces within friction contacts	Radius of traction force	Important for passenger cars	excellent
3	Hydrostatic		Hydrostatic forces at pump(s) and motor(s)	Displacement of the units	Important for mobile machinery	moderate
4	Electrical		Electro-magnetic forces at generator(s) and motor(s)	Frequency of current or electric flux or load	Upcoming	moderate

Ref: K. Renius and R. Resch, Continuously Variable Tractor Transmissions, ASAE Distinguished Lecture Series, Tractor Design No. 29

Typical traction/speed characteristics for a wheel loader



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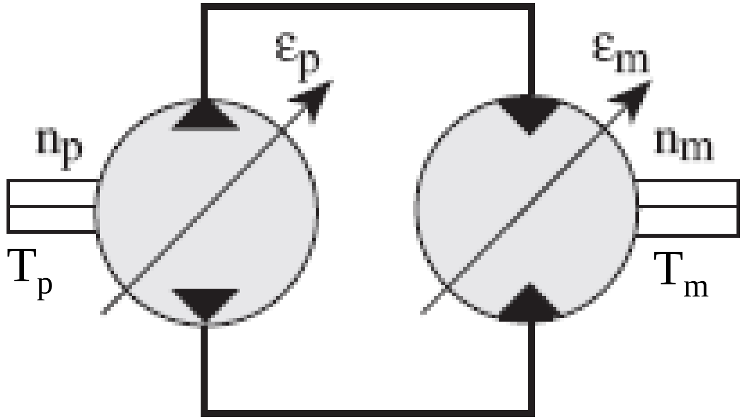
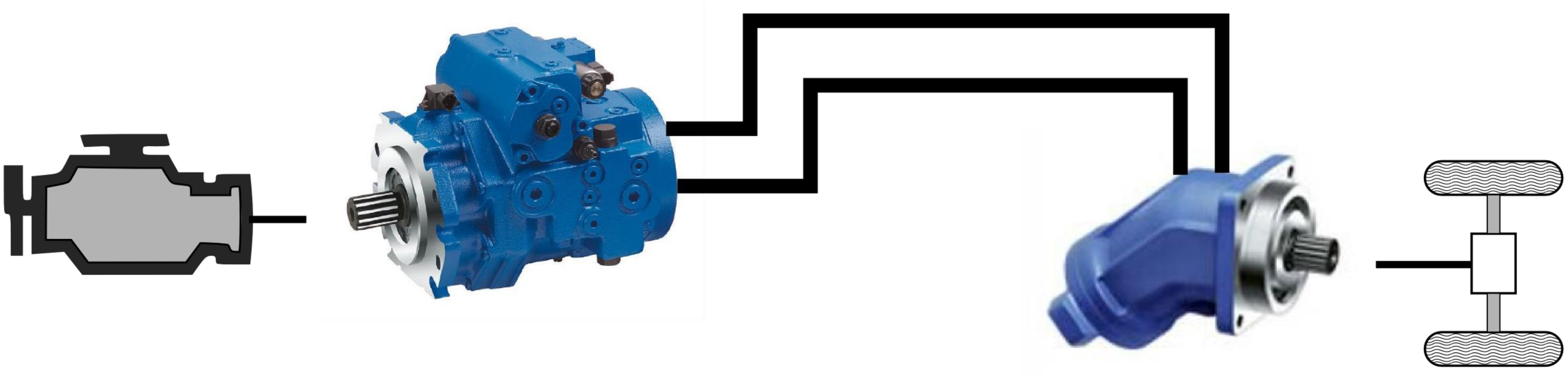
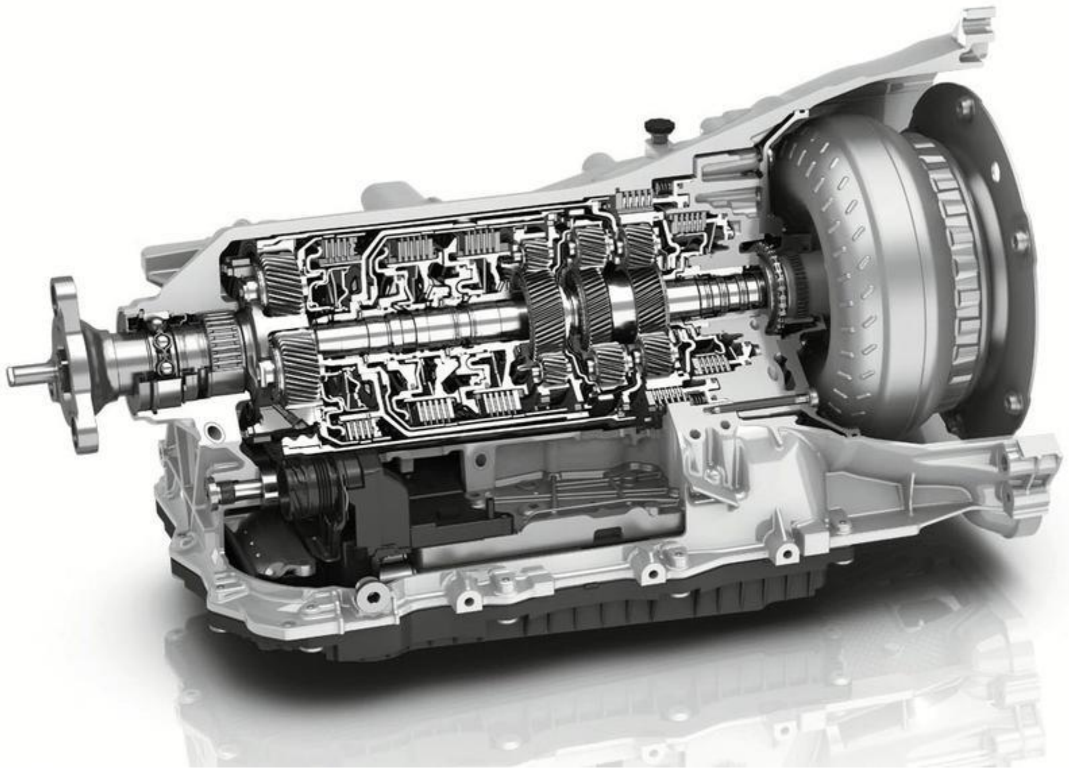
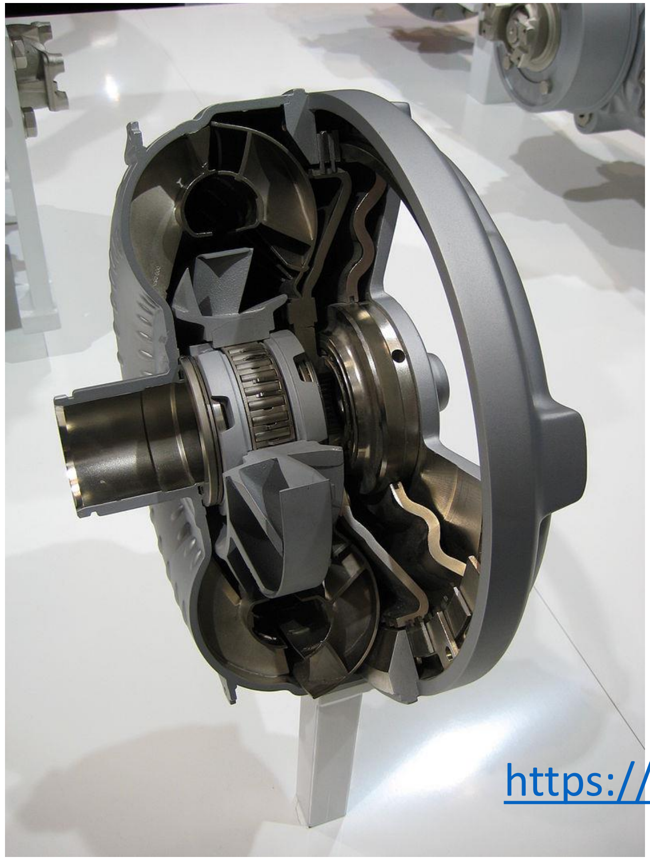
Hydrostatic transmission





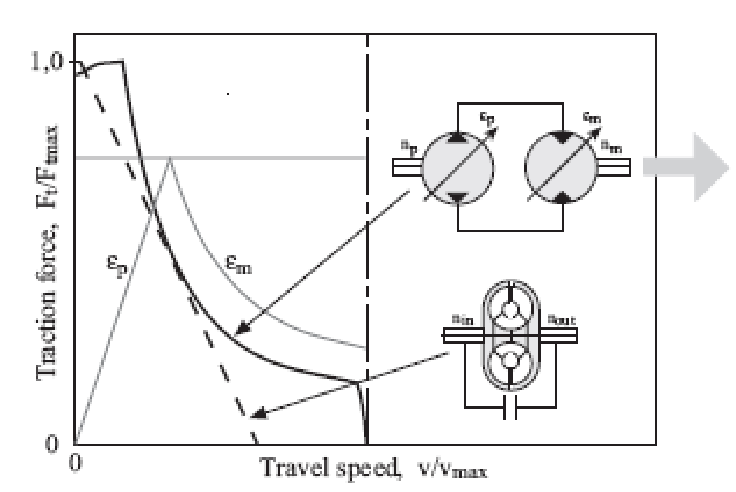
Torque converter – Hydro-dynamic clutch



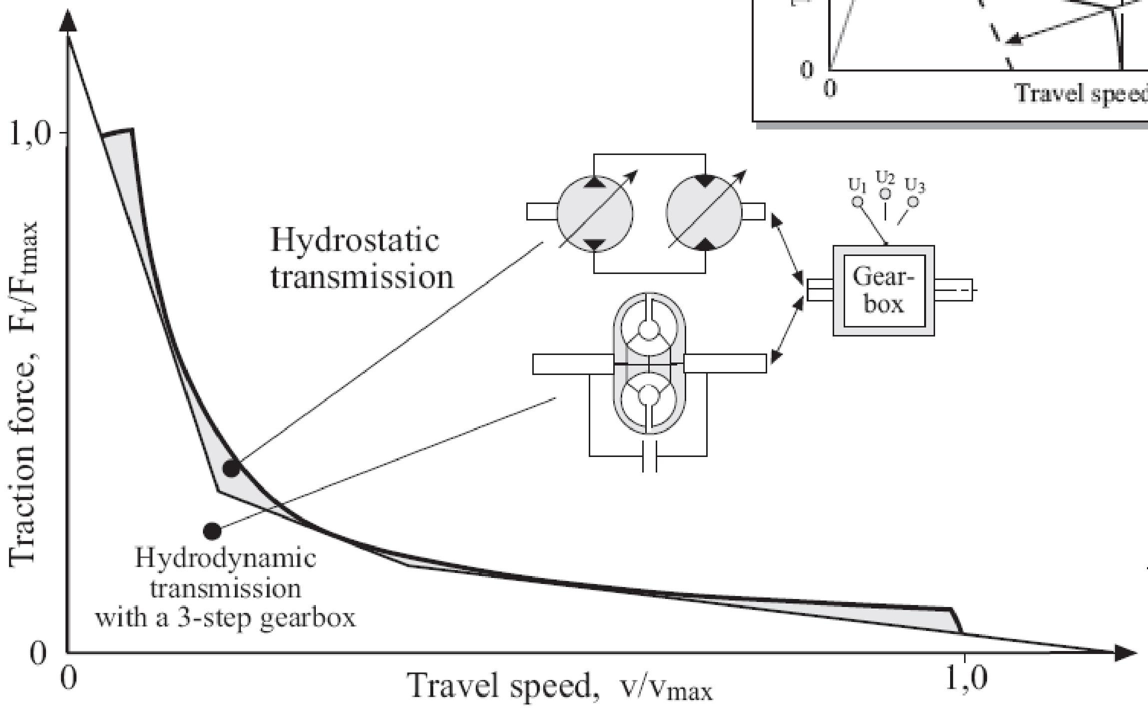


https://youtu.be/z5G2zQ_3xTc

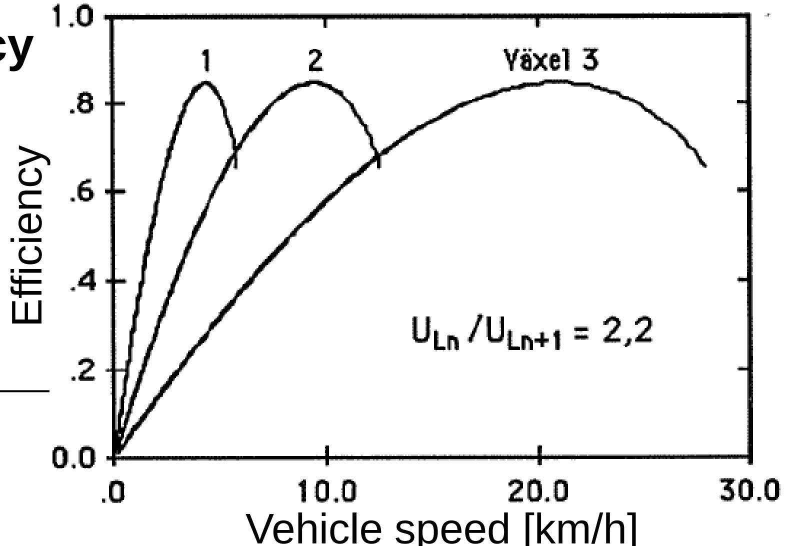
Hydrostatic and Hydro-dynamic Drivetrain



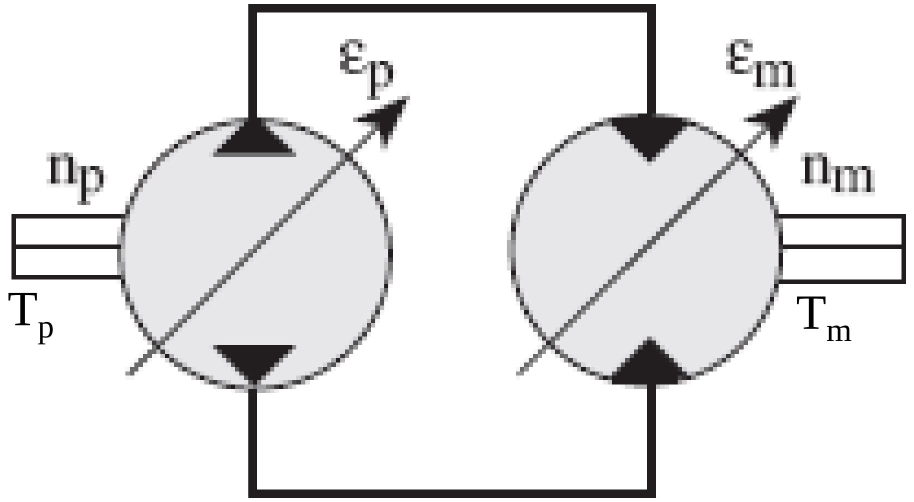
- Large convertion range
- High maneuverability
- Hydrostatic breaking
- High traction force at low engine speed
- High efficiency
- Simple reversing by pump control
- Open design, single-wheel drive
- Low heat development in blocked point

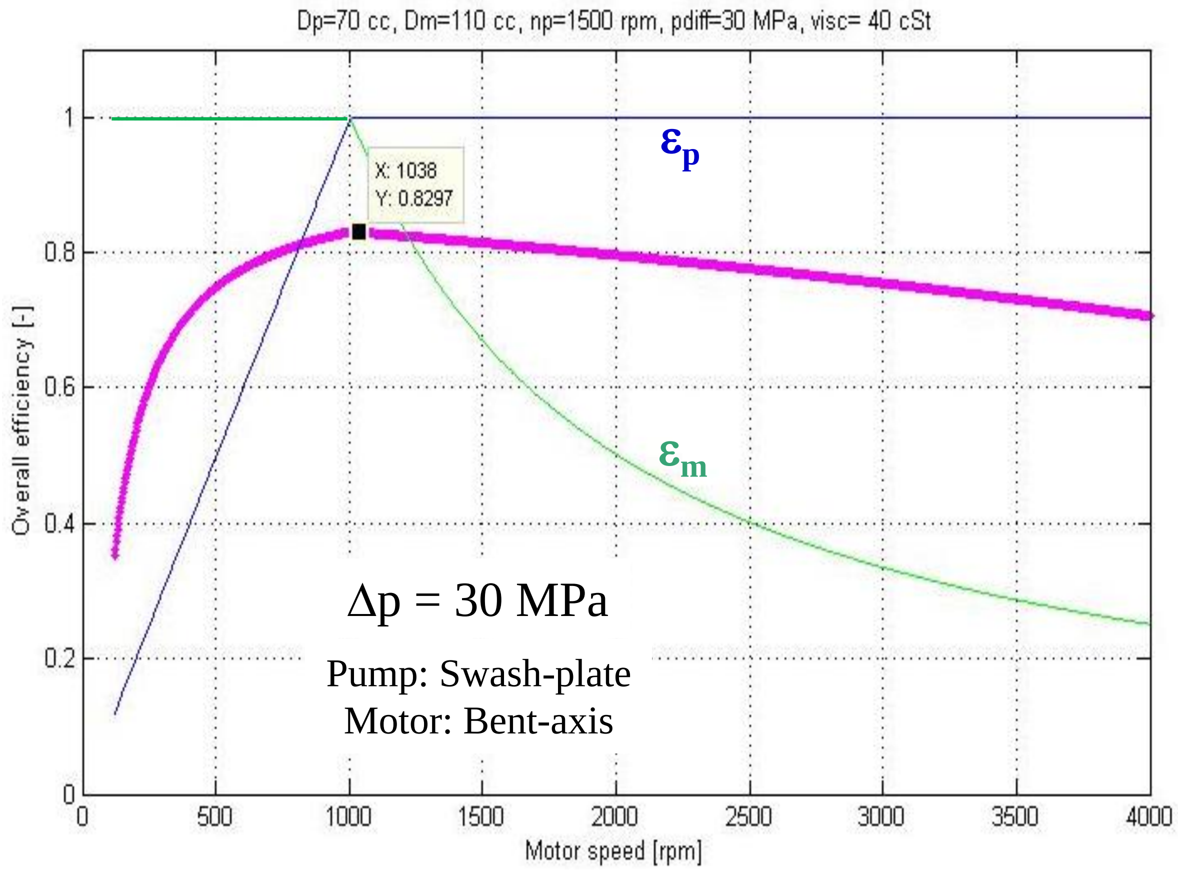


Torque converter efficiency

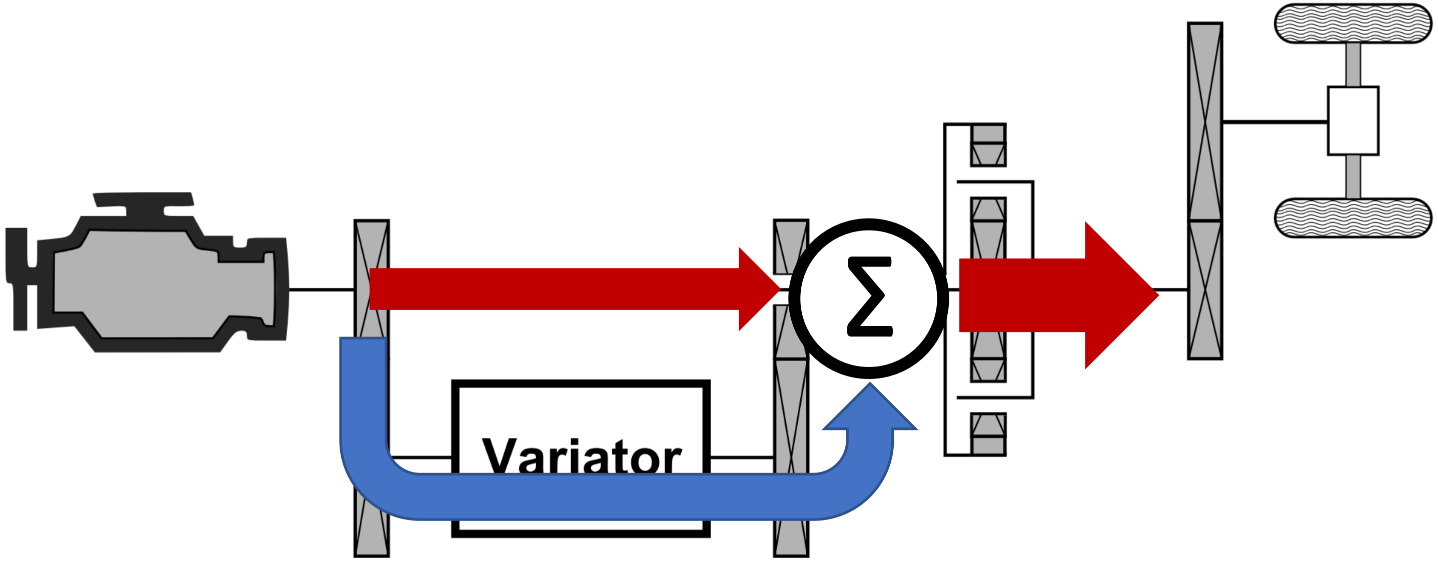


Hydrostatic Transmission





Power-Split



Hydraulic transmission applications

Power split

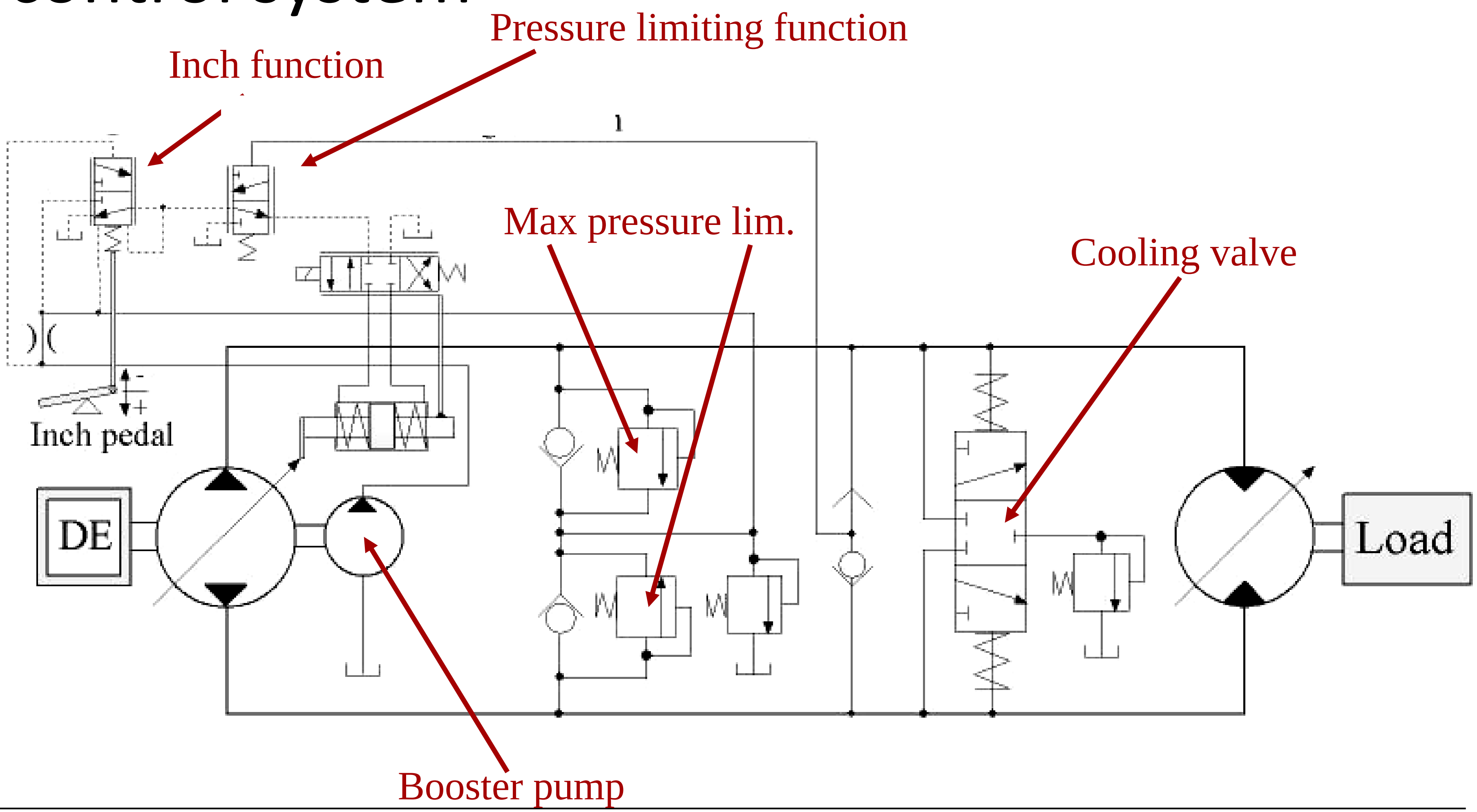
Hydrostatic transmission

Hydrostatic transmission

Torque converter

Hydraulic transmission applications

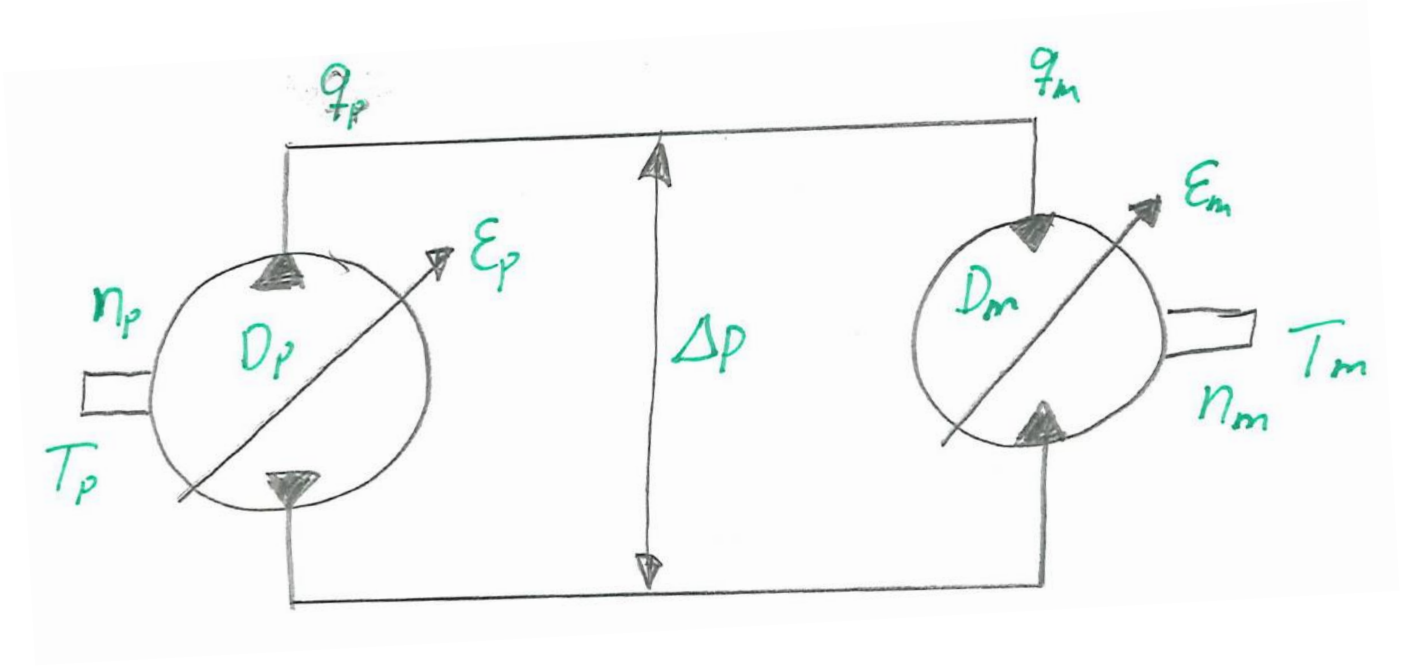
Transmission control system



Some calculations for sizing the hydrostatic transmission

- Speed ratio
- Max power
- Comparing different designs

Sizing the hydrostatic transmission



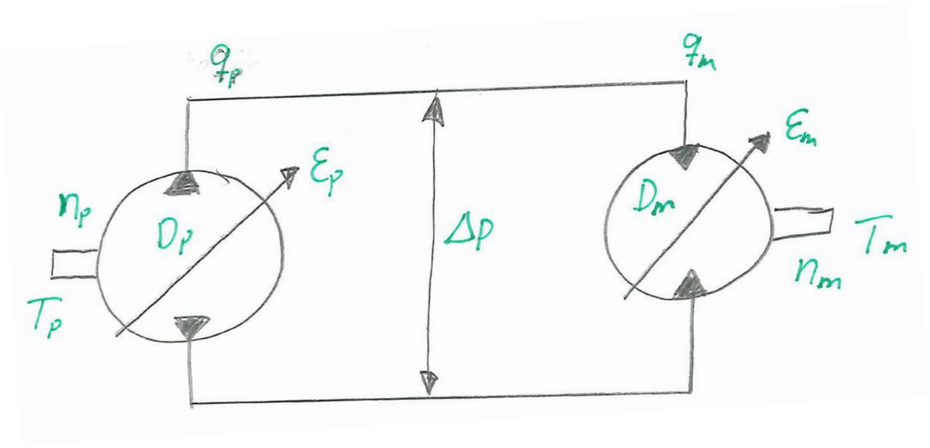
Assumptions

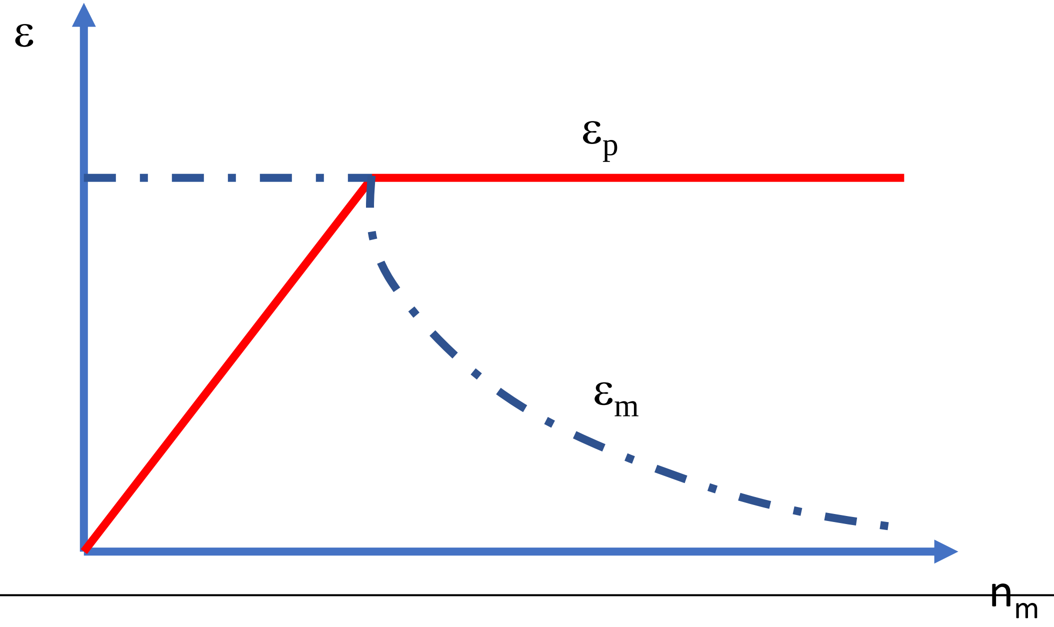
- Constant pump speed
- TR values and dimensioning is made when maximum power used

Speed ratio

$$\left.\begin{aligned} q_p &= \varepsilon_p D_p n_p \\ q_m &= \varepsilon_m D_m n_m \end{aligned}\right\} \quad q_p = q_m \Rightarrow$$

$$\frac{n_m}{n_p} = \frac{\varepsilon_p D_p}{\varepsilon_m D_m}$$





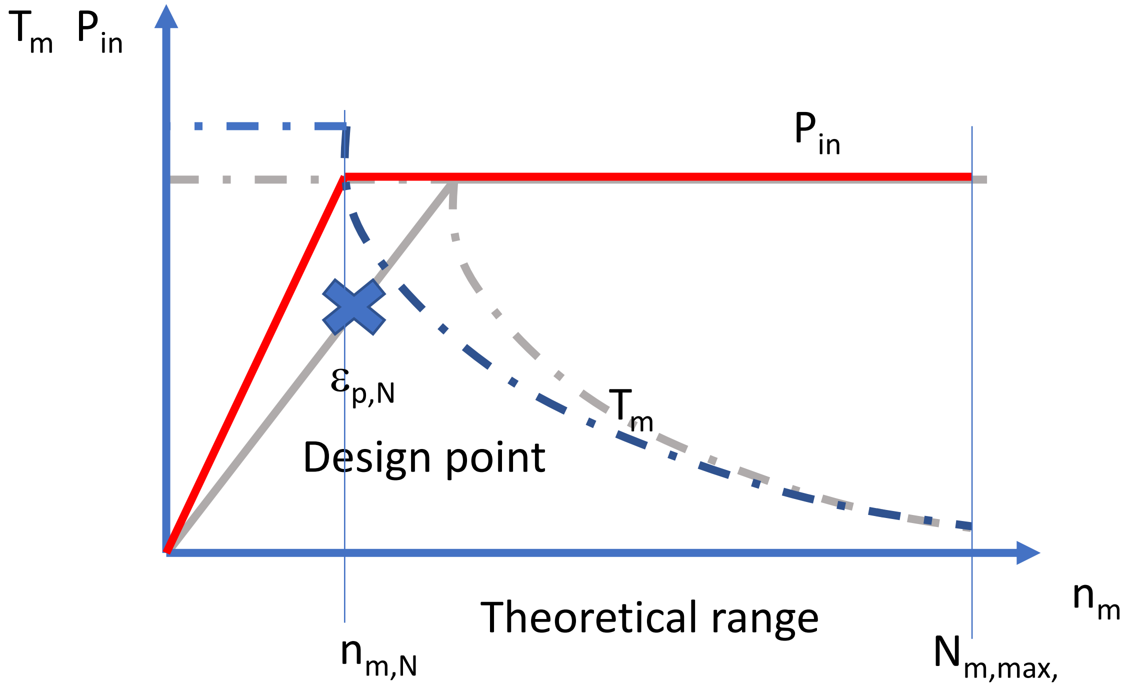
Transmitted power and Traction torque

- Transmitted power

$$P_{in} = T_p n_p 2\pi = \frac{\varepsilon_p D_p n_p \Delta p}{\eta_{hmp}}$$

- Traction torque

$$T_m = \frac{\varepsilon_m D_m \Delta p}{2\pi} \eta_{hmm}$$



Theoretical range

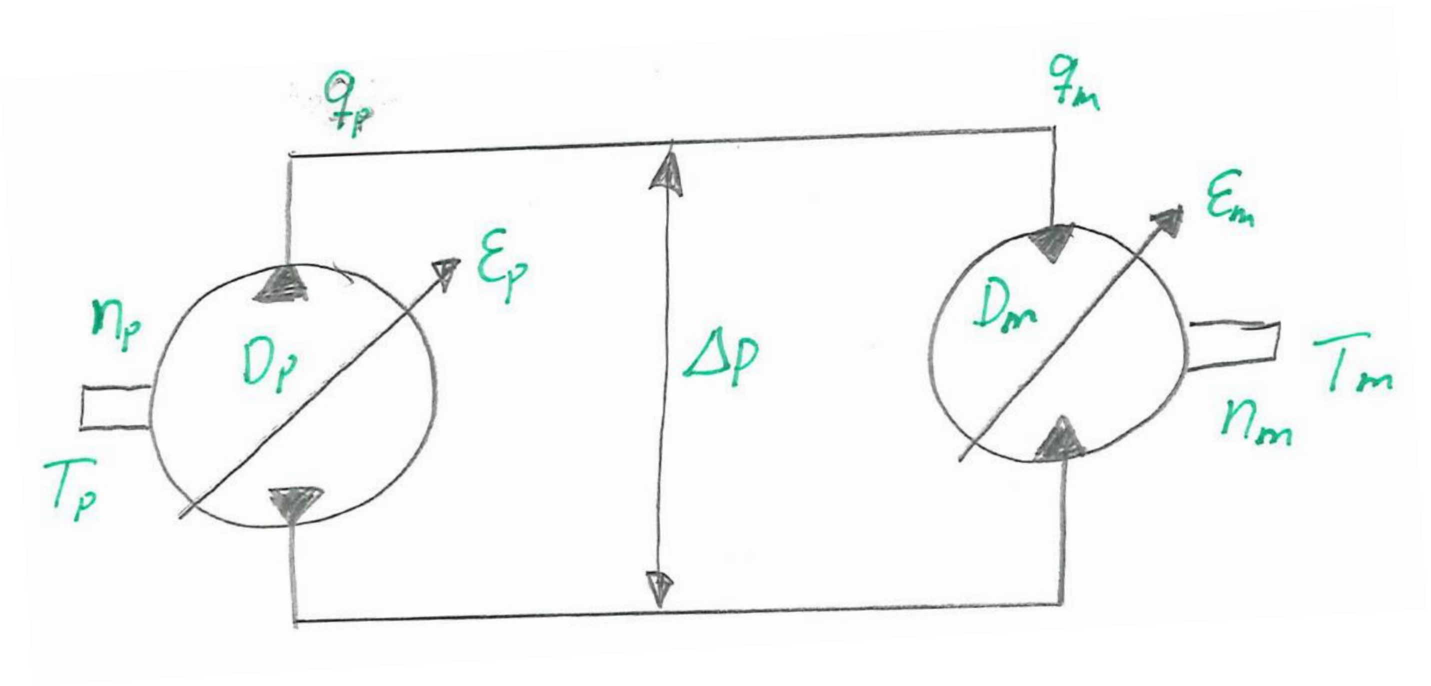
$$TR = \frac{n_{m,max}}{n_N}$$ or $$TR = \frac{1}{\varepsilon_{m,min}\varepsilon_{p,N}}$$

- TR – range where maximum power can be transmitted throught the transmission
- TR =5-25
- For higher values, the motor size gets too big

Sizing the transmission pump/motor unit

- To compare different designs of hydro-mechinacal transmissions, we can look on the sizing of the components

Simplest example – 1 pump driving 1 motor



- Motor displacement is calculated at max speed:

$$P_{out,max} = P_{in,max} = \varepsilon_{m,min} D_m n_{m,max} \Delta p$$

where $\Delta p = \varepsilon_{pN} \Delta p_{max}$

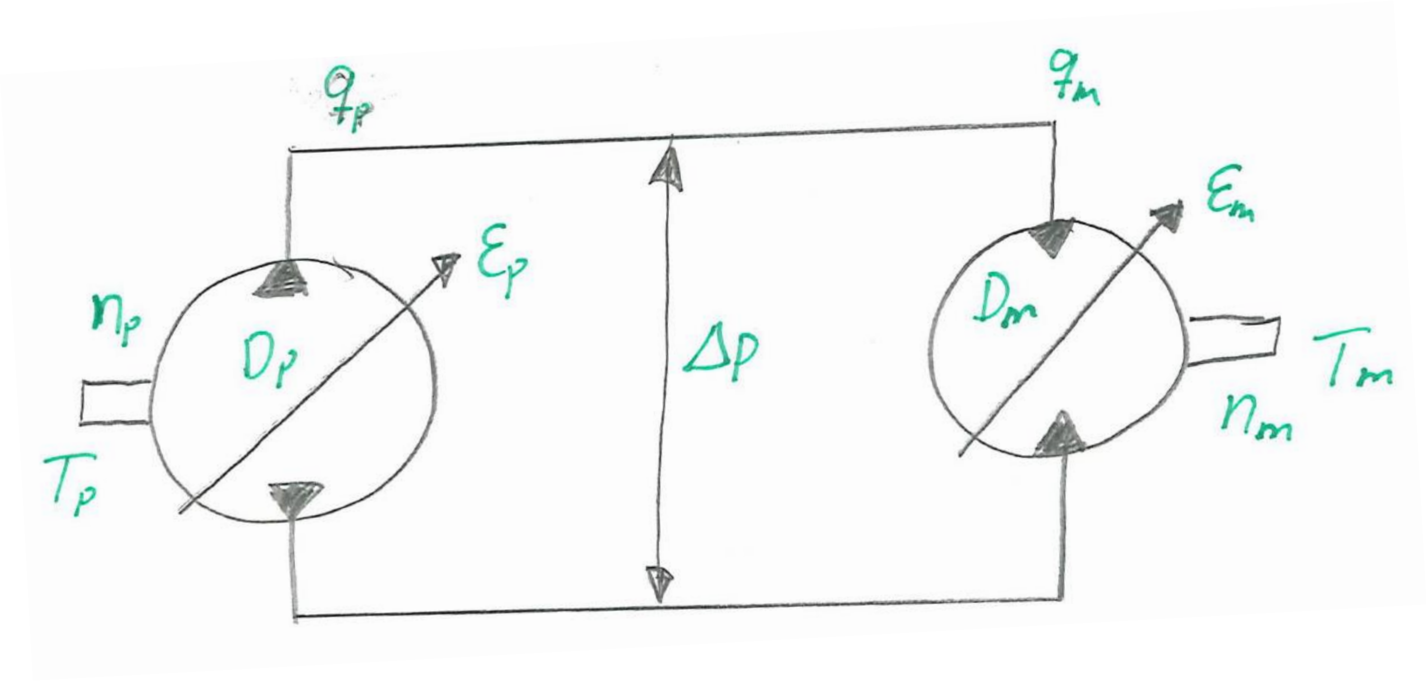
- Motor displacement

$$D_m = \frac{P_{in,max}}{\varepsilon_{m,min} n_{m,max} \varepsilon_{pN} \Delta p_{max}}$$

- With TR equation

$$D_m = \frac{P_{in,max} TR}{n_{m,max} \Delta p_{max}}$$

Simplest example – 1 pump driving 1 motor



- Pump displacement at the design point:

$$P_{in,max} = \varepsilon_{pN} D_p n_p \Delta p_{max}$$

- With TR equation

$$D_p = \frac{P_{in,max} \, TR \, \varepsilon_{m,min}}{n_p \, \Delta p_{max}}$$

- And motor displacement equation

$$D_p = \frac{\varepsilon_{m,min} \, n_{m,max} \, D_m}{n_p}$$

Hydrostatic transmission with mechanical gearbox

Speed ratio:

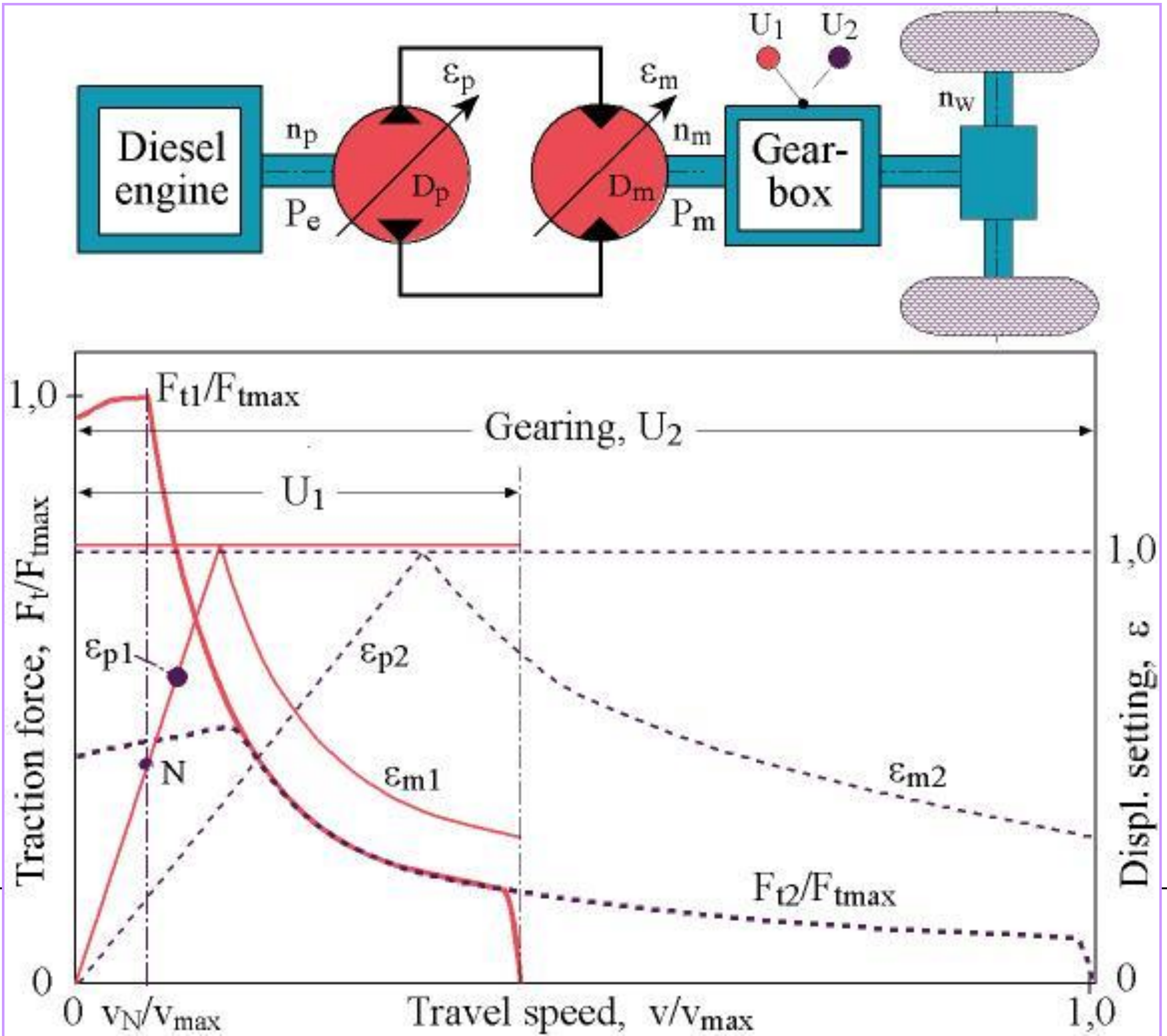
$$\frac{n_w}{n_p} = \frac{\varepsilon_p D_p}{\varepsilon_m D_m} \cdot \frac{U_1}{U_2}$$

TR-value:

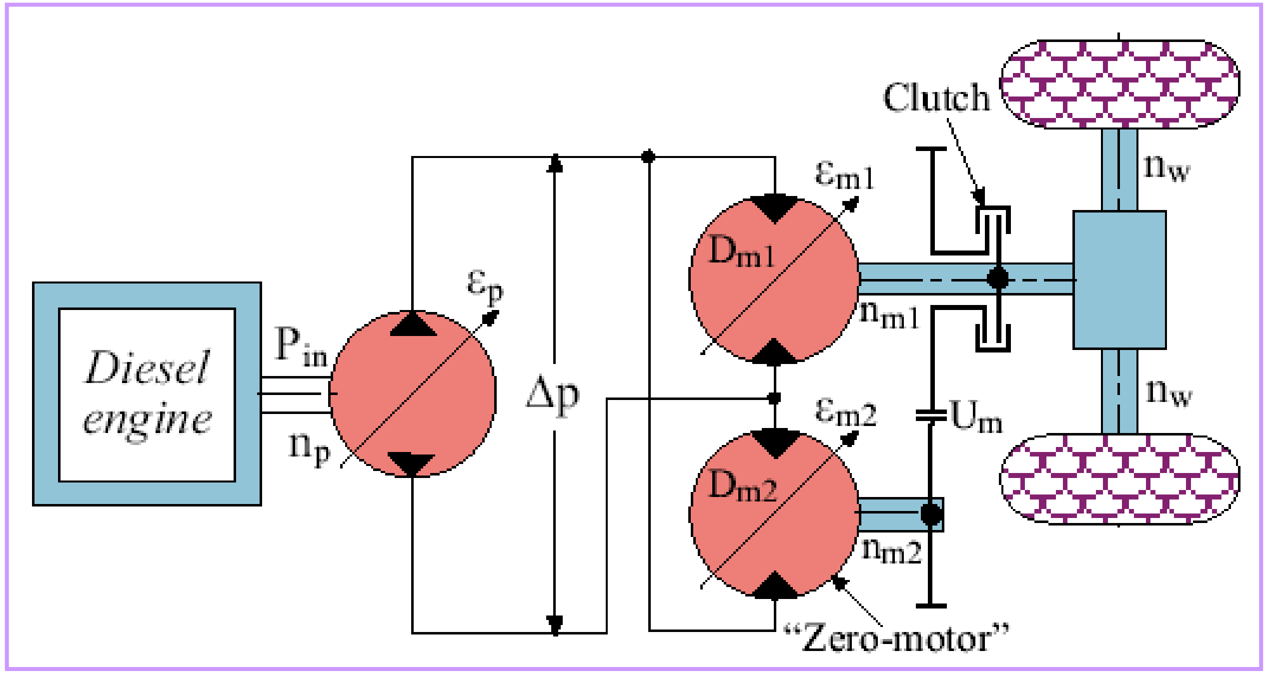
$$TR = \frac{n_{mmax}}{n_{mN}} \cdot \frac{U_{max}}{U_{min}}$$

Motor displacement:

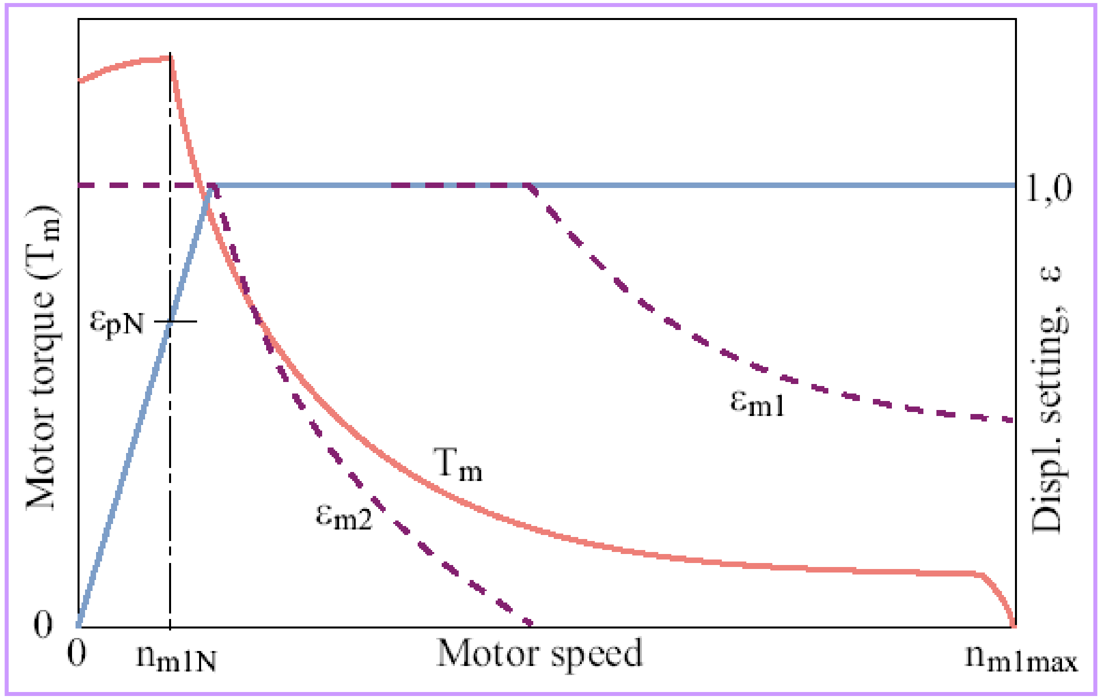
$$D_m = \frac{P_{in,maxTR}}{n_{m,max}\Delta p} \cdot \frac{U_{min}}{U_{max}}$$



Hydrostatic two-motor transmission with a dis-connectable zero-motor

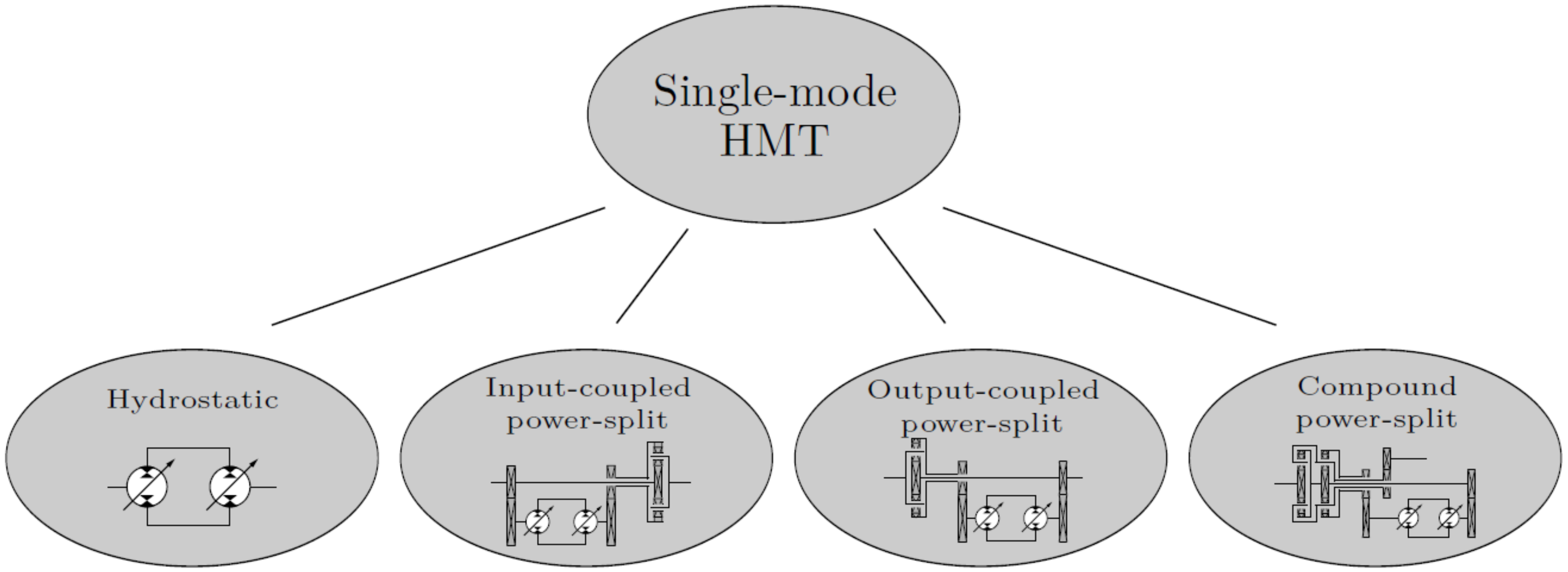


$$\frac{n_{m1}}{n_p} = \frac{\varepsilon_p D_p}{\varepsilon_{m1} D_{m1} + \varepsilon_{m2} D_{m2} U_m}$$

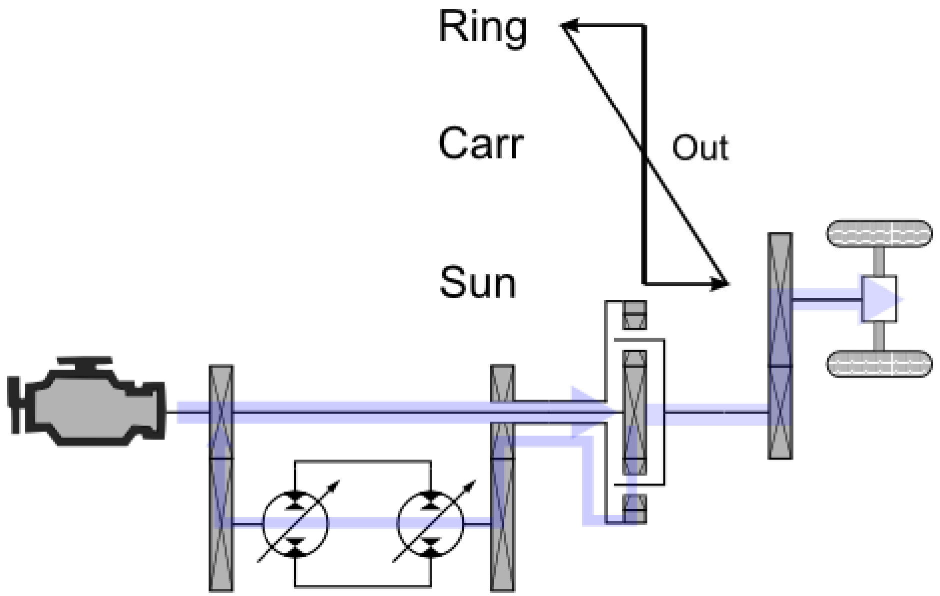


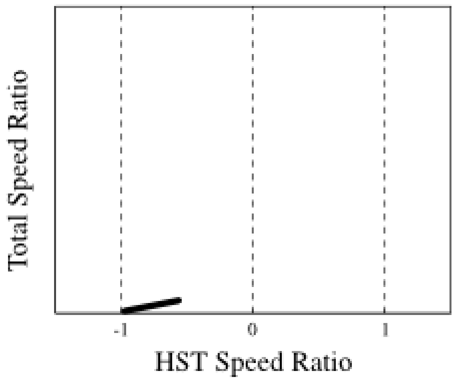
$$TR = \frac{n_{m1\max}}{n_{m1N}\dfrac{1+\dfrac{D_{m2}}{D_{m1}}U_m}{\varepsilon_{pN}\varepsilon_{m1\min}}}$$

Power split transmissions



Input-coupled





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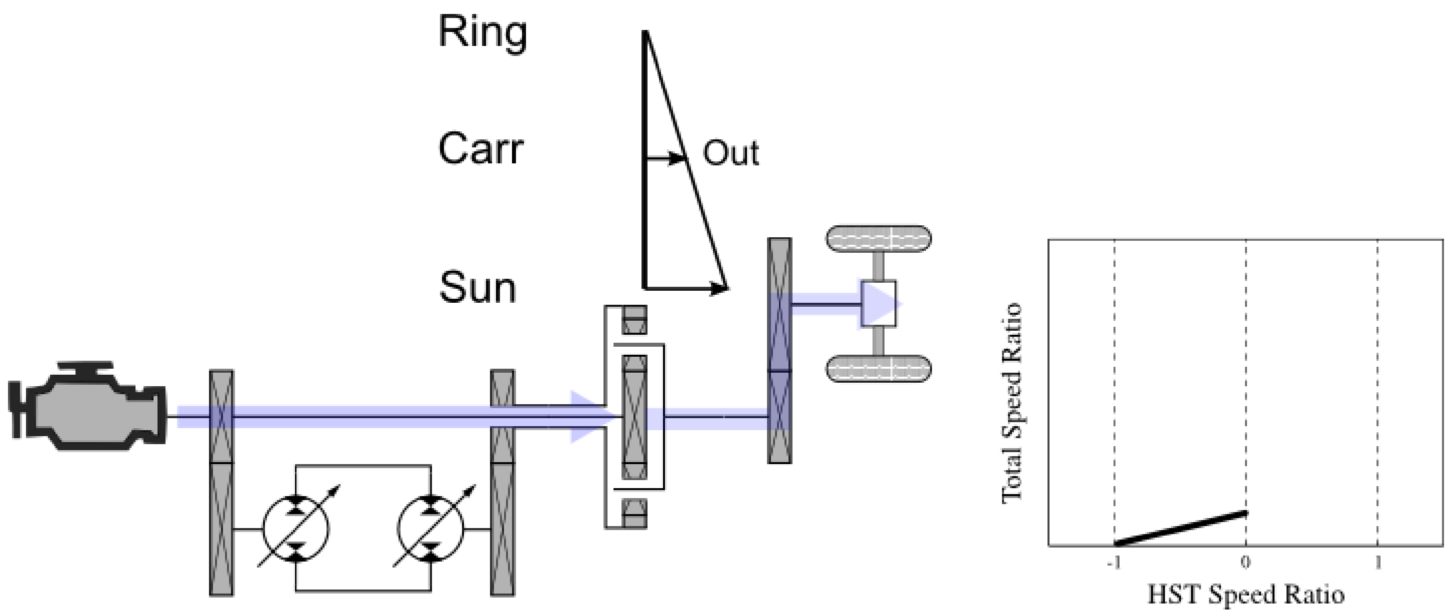

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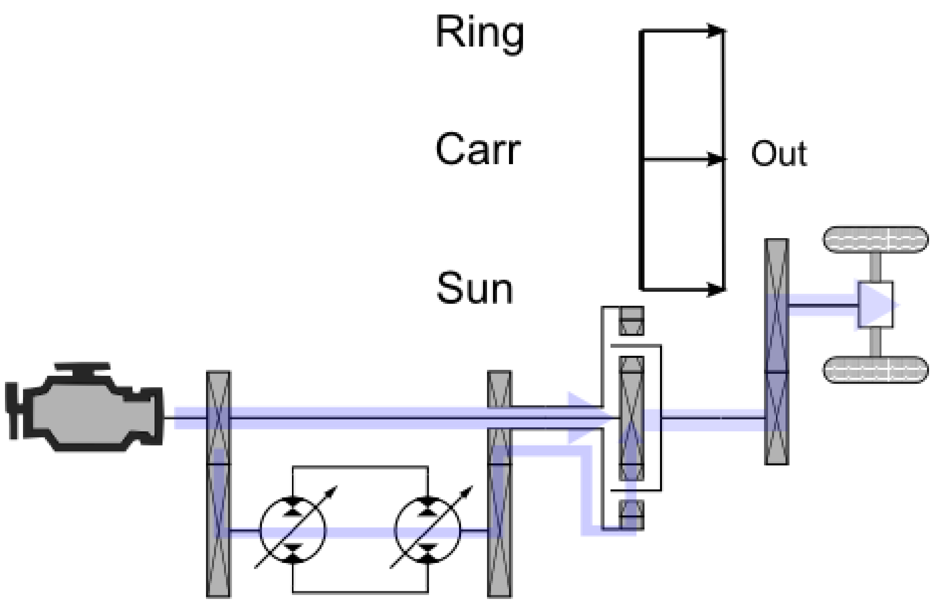
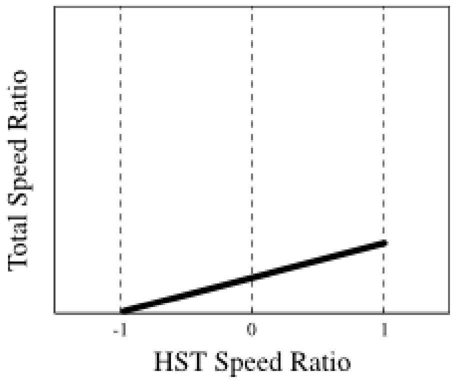
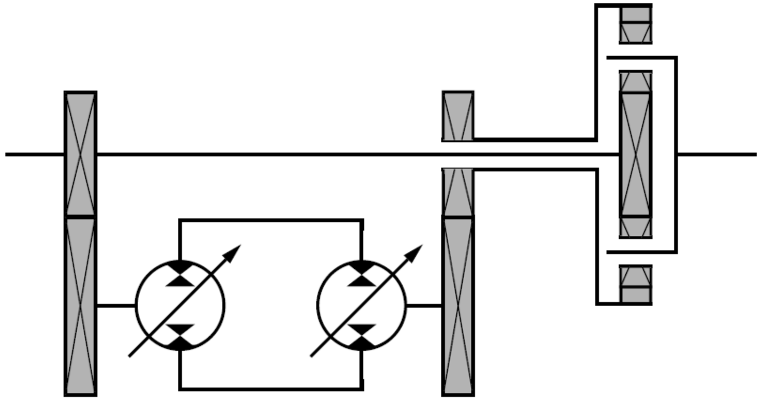
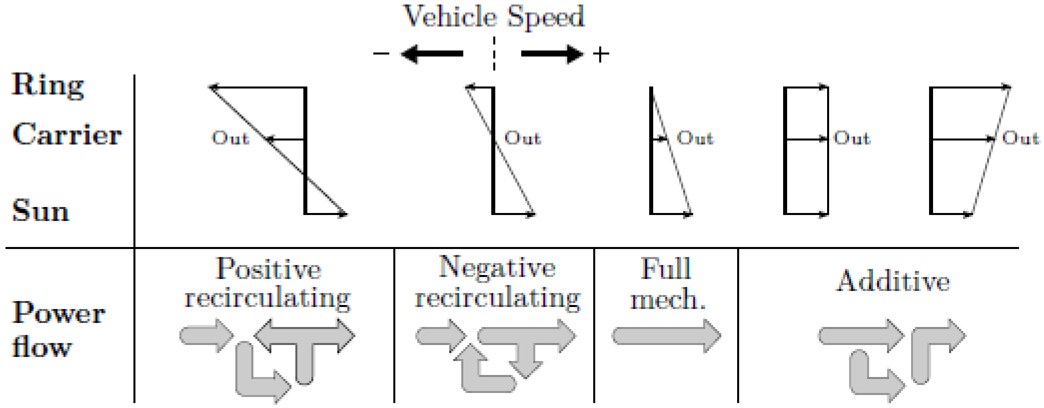
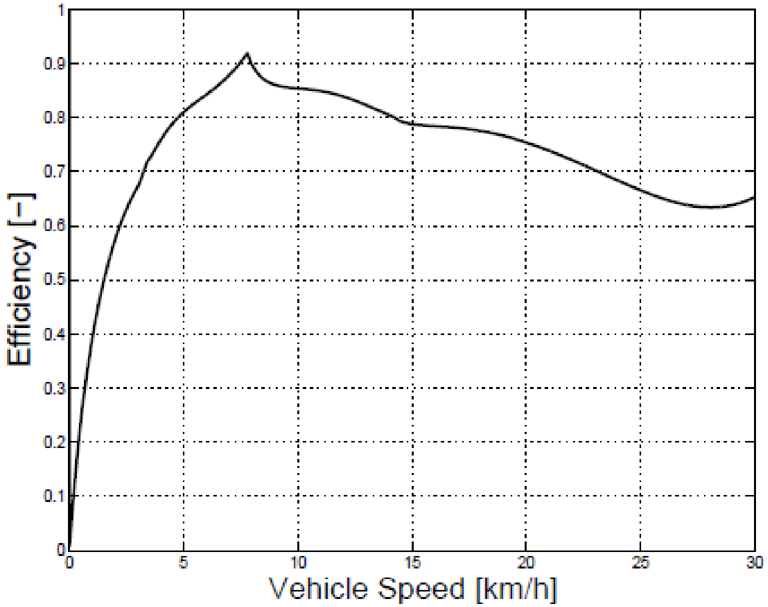


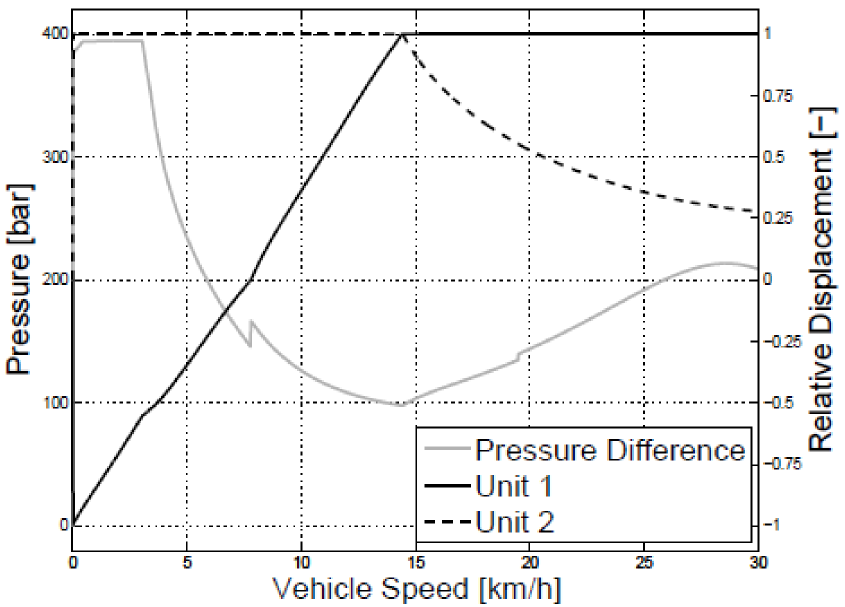


Input-coupled power split

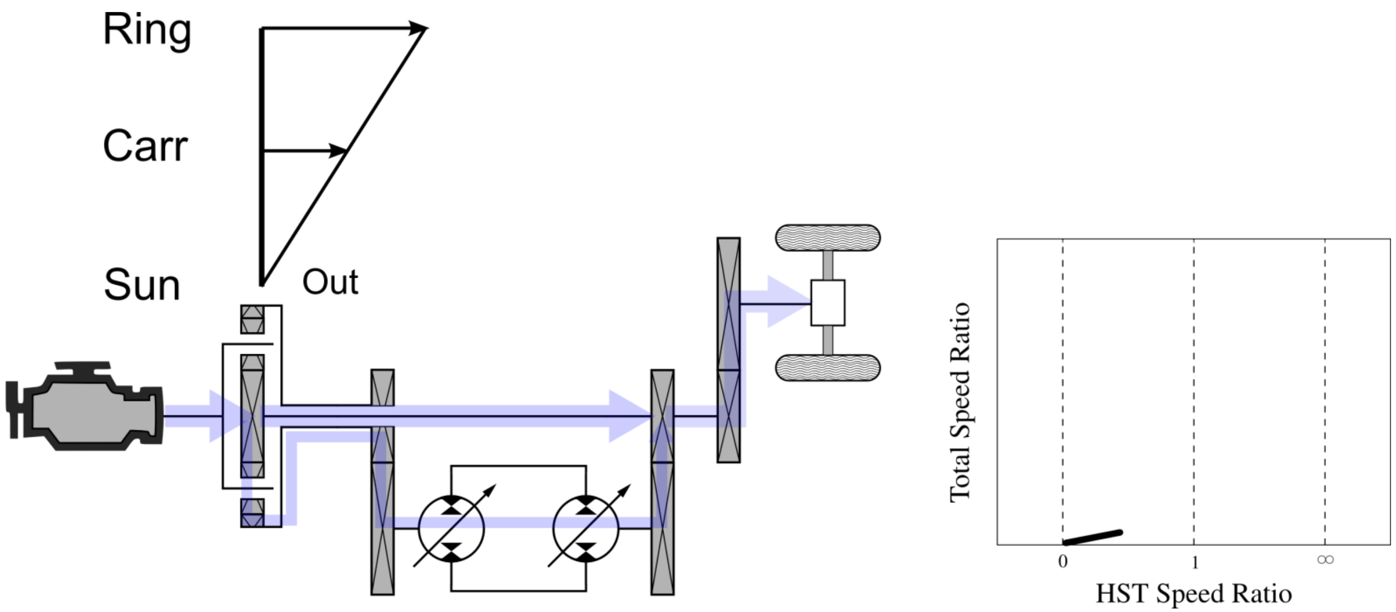




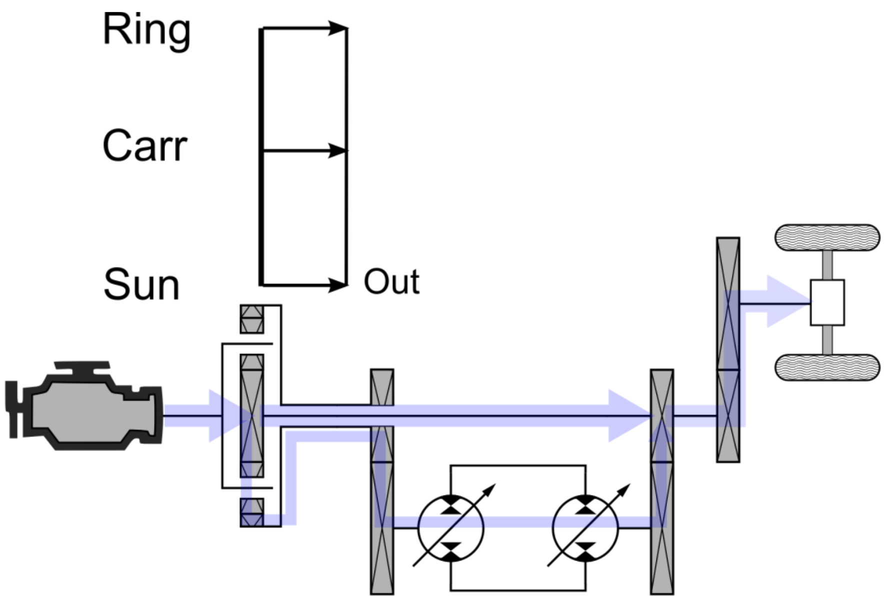


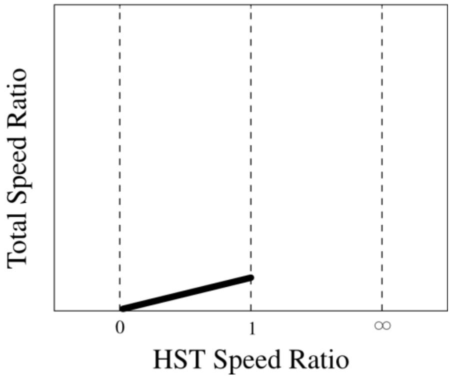


Output-coupled

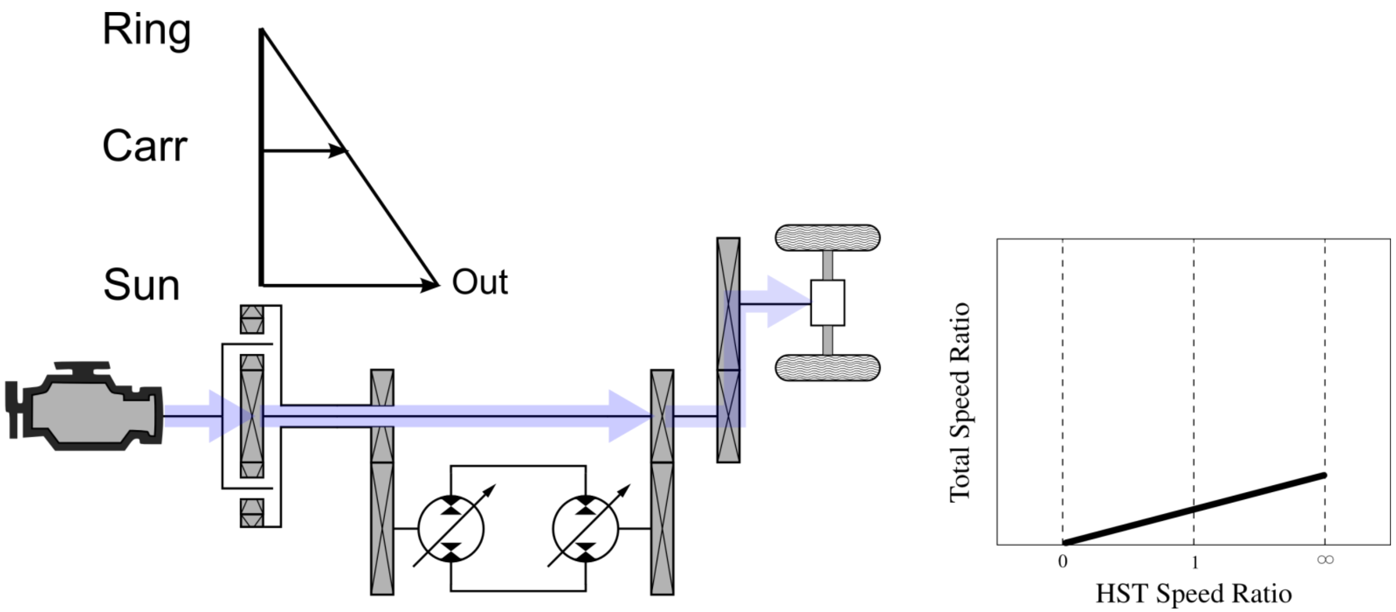


Output-coupled

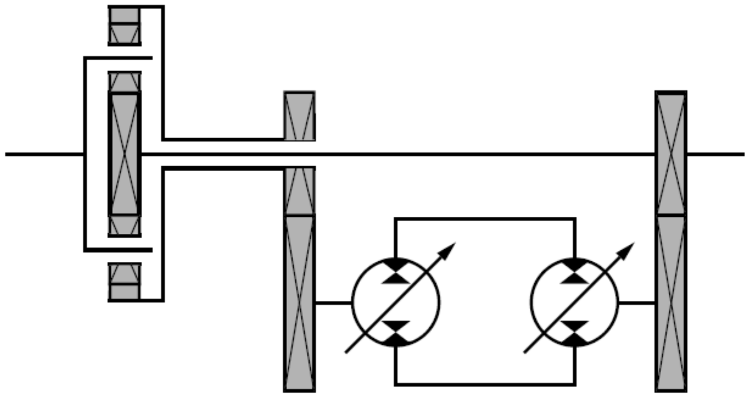


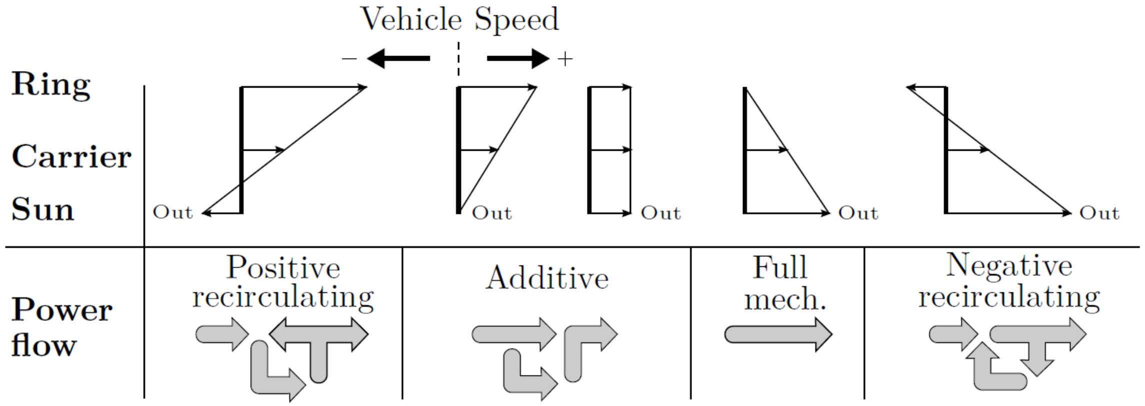


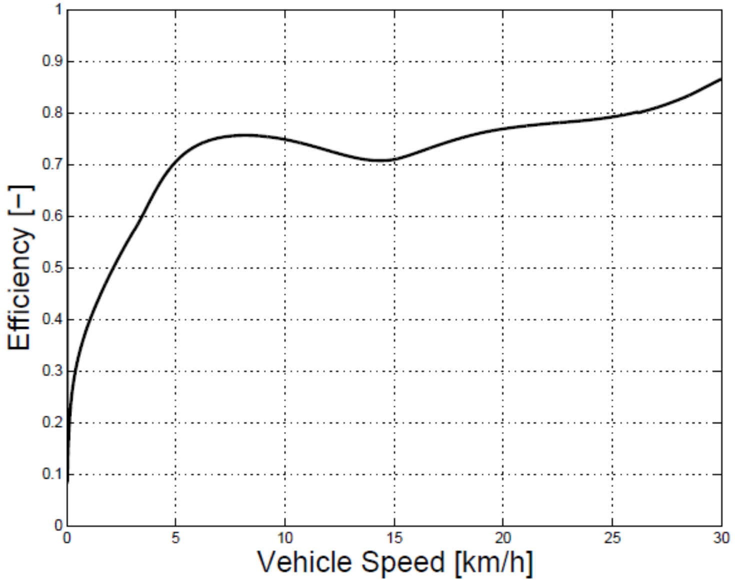
Output-coupled

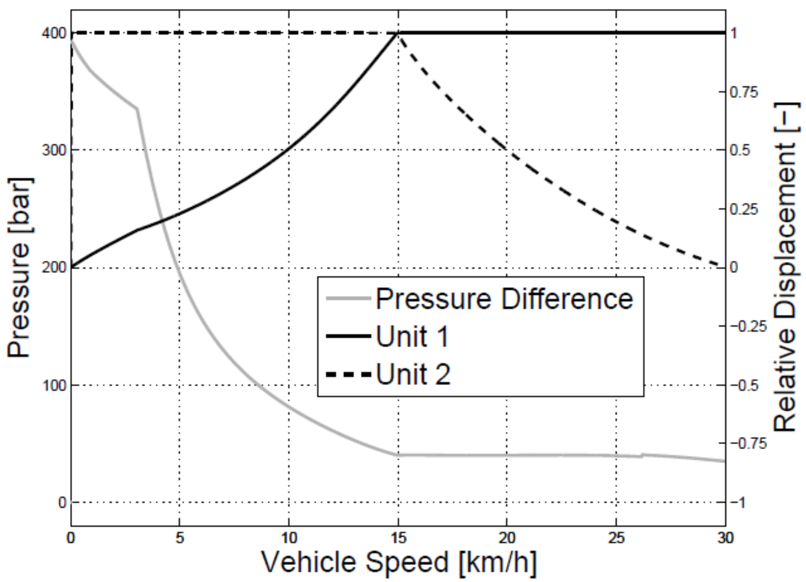


Output-coupled power split

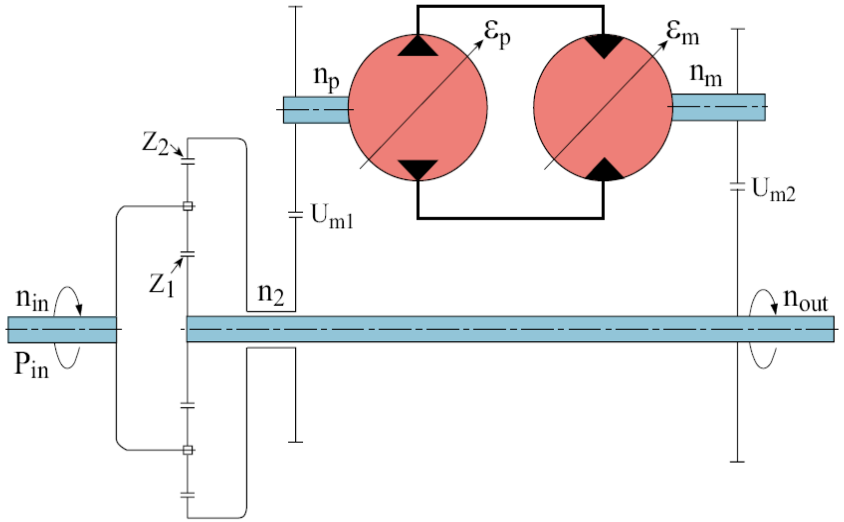


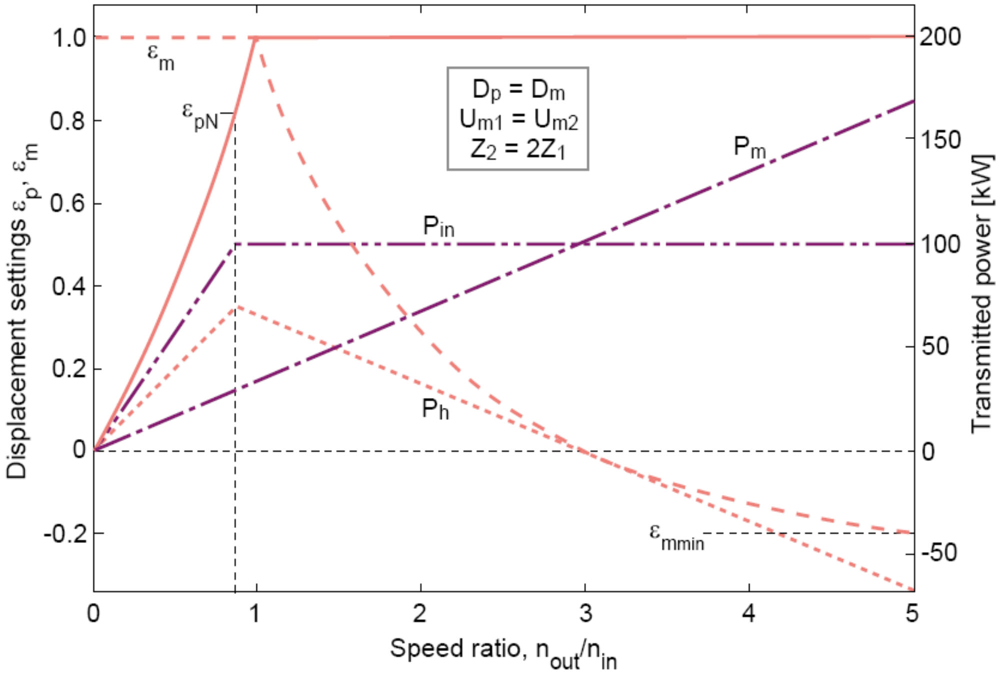






Output-coupled power split – ex.

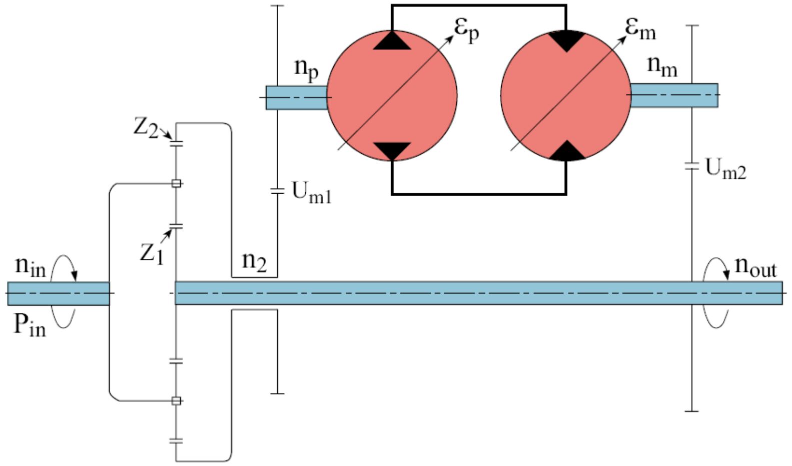




Speed ratio:

$$\frac{n_{out}}{n_{in}} = \frac{Z_1 + Z_2}{Z_1 + \frac{\varepsilon_m D_m}{\varepsilon_p D_p} \frac{U_{m2}}{U_{m1}} Z_2}$$

Output-coupled power split – ex.



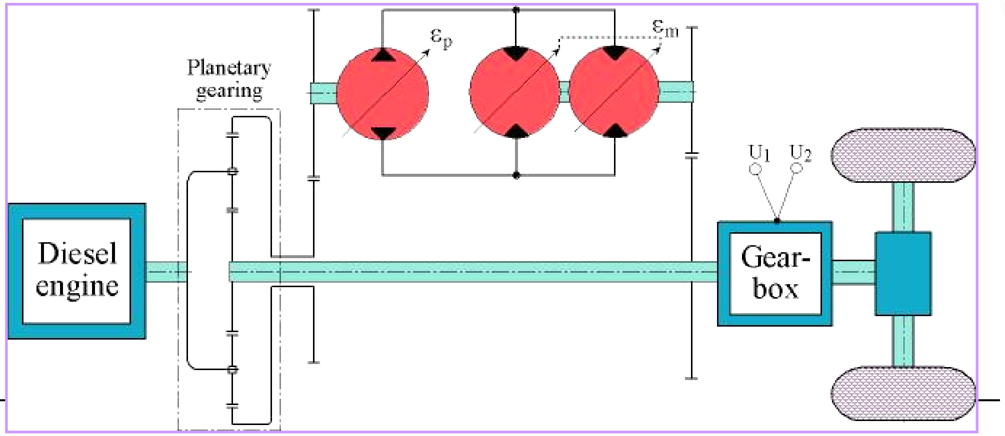
$$TR = \frac{1 + \frac{D_m}{\varepsilon_{pN} D_p} \cdot \frac{U_{m2} Z_2}{U_{m1} Z_1}}{1 + \frac{\varepsilon m_{mmin}}{D_p \frac{U_{m2} Z_2}{U_{m1} Z_1}}}$$

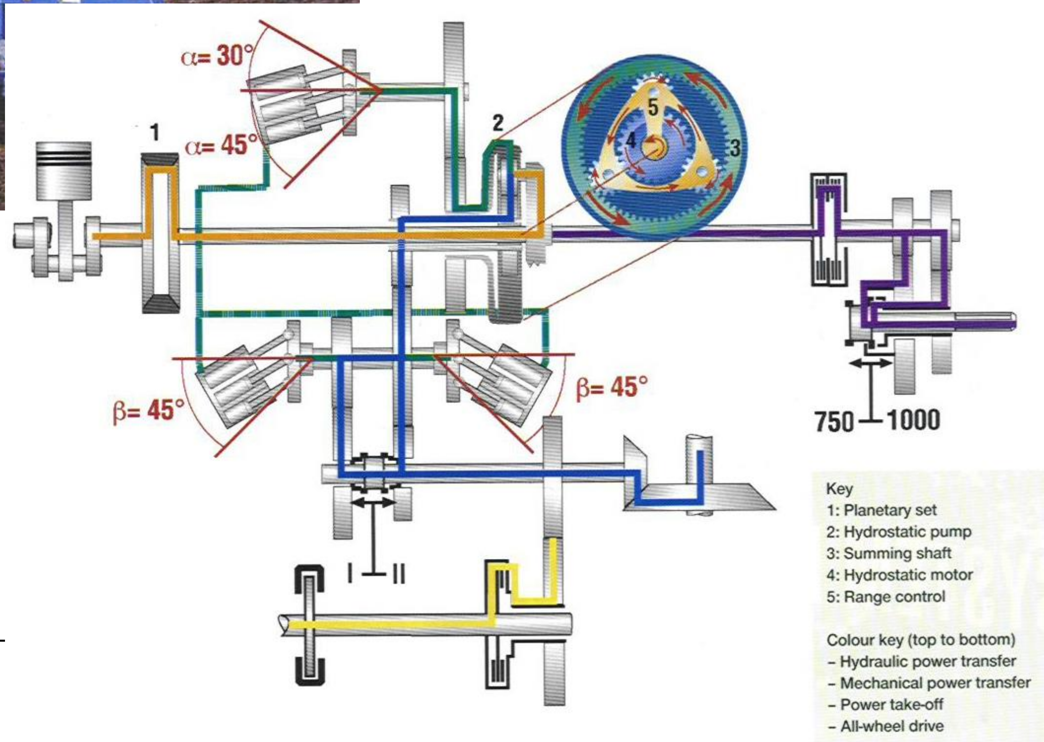
$Z_2 = 2Z_1$, $D_p = D_m$ och $U_{m1} = U_{m2}$

gives $\frac{n_{out}}{n_{in}} = \frac{3}{1 + 2 \cdot \varepsilon_m / \varepsilon_p}$ *and* $TR = \frac{1 + 2/\varepsilon_{pN}}{1 + 2\varepsilon_{mmin}}$

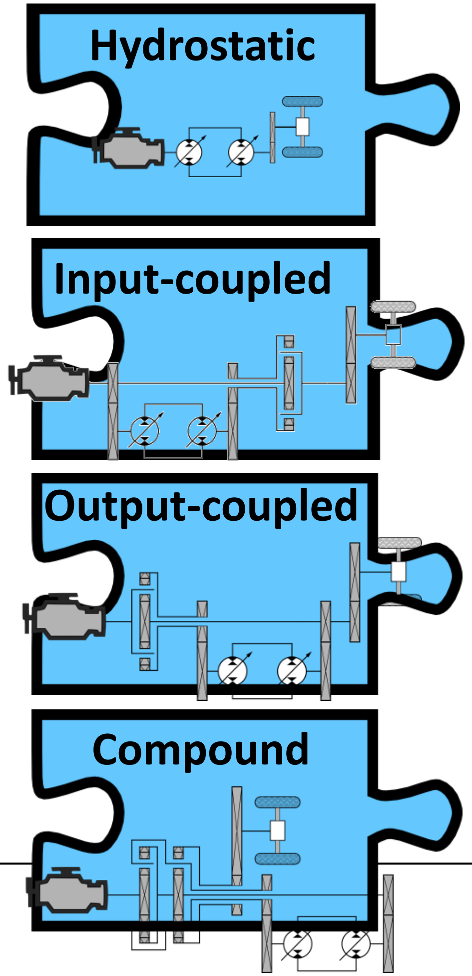
Common Hydro-Mechanical Transmission

Fendt Vario



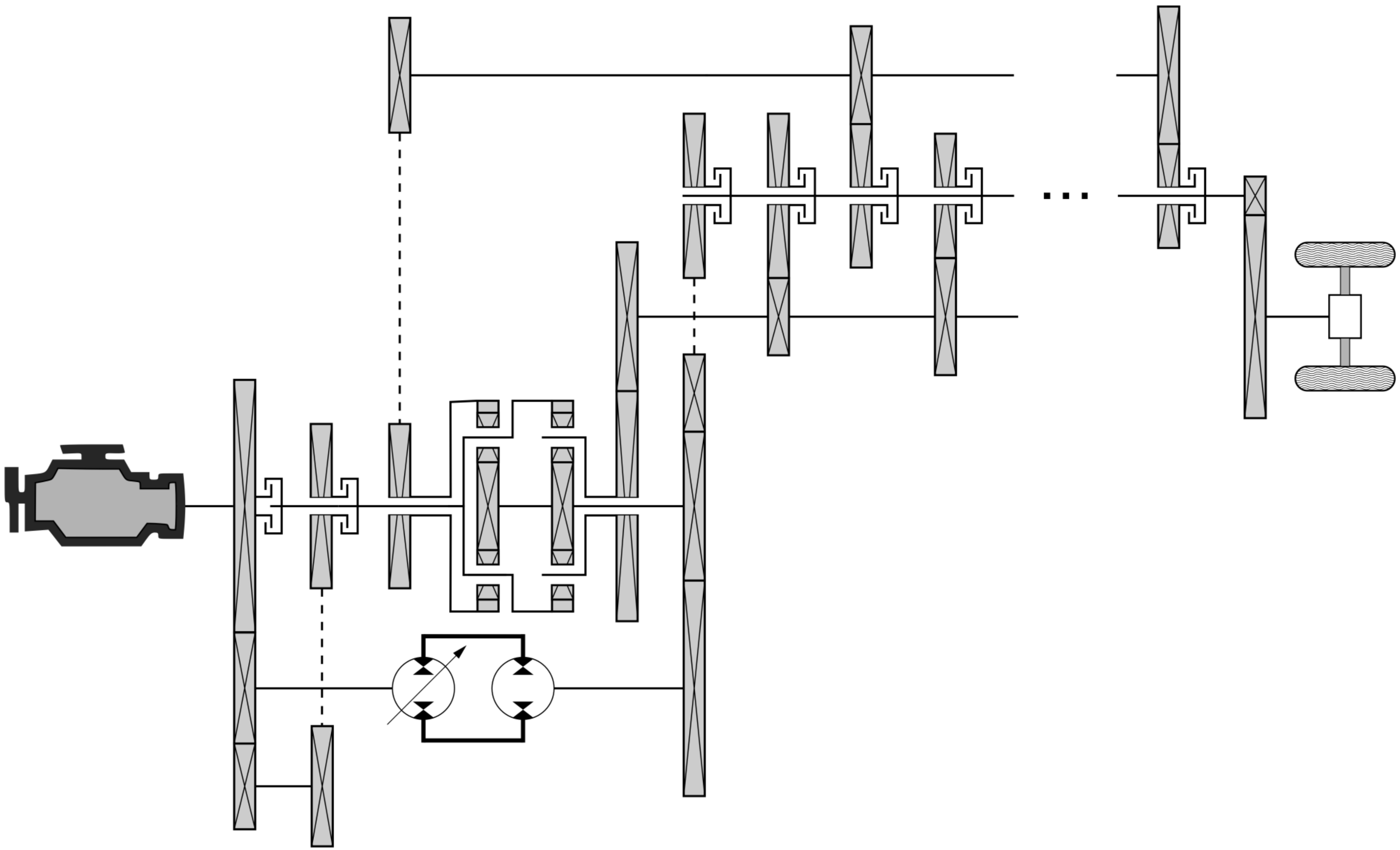


Using multiple modes



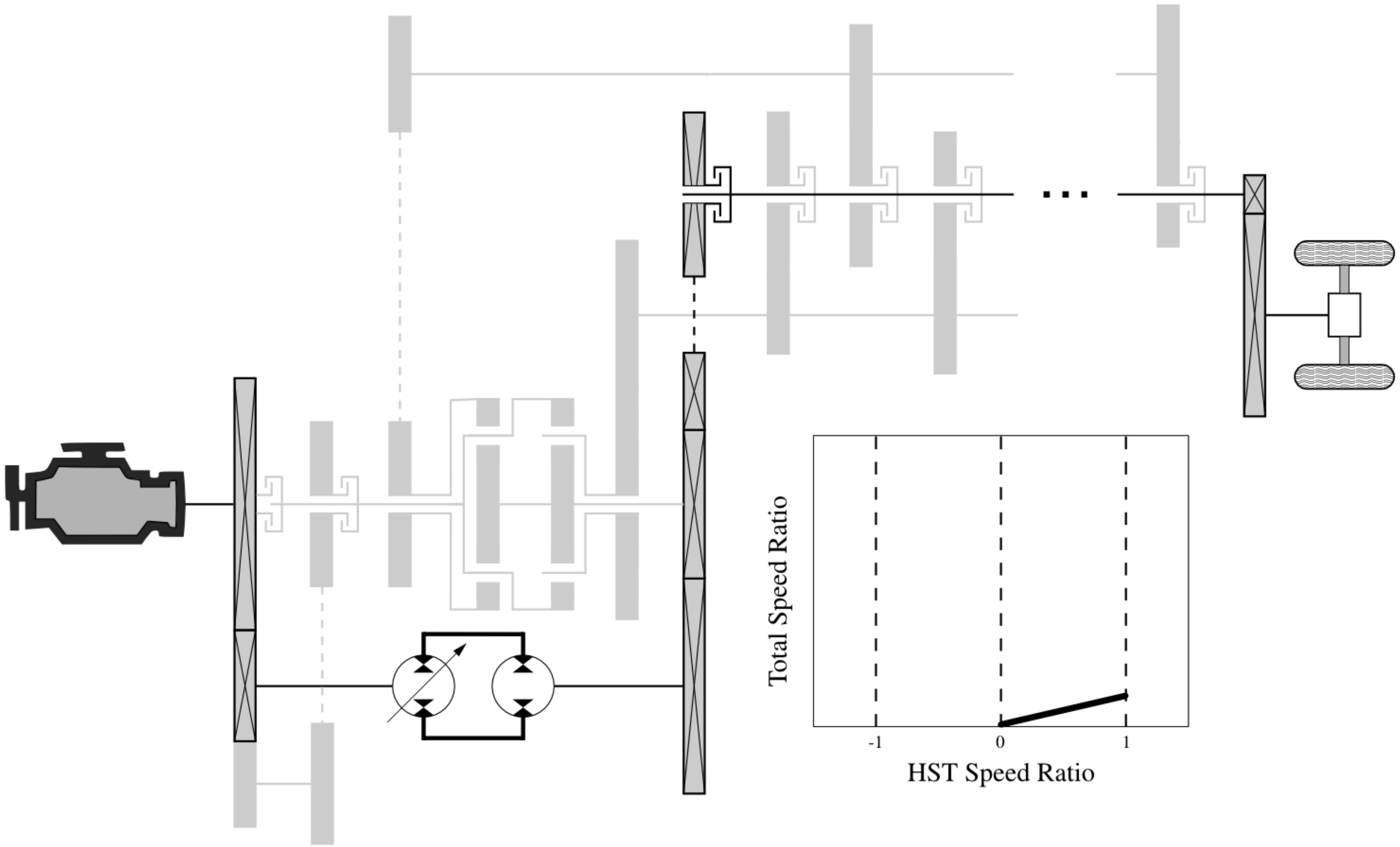
Complete Transmission

Example: The ”Jarchow-concept”

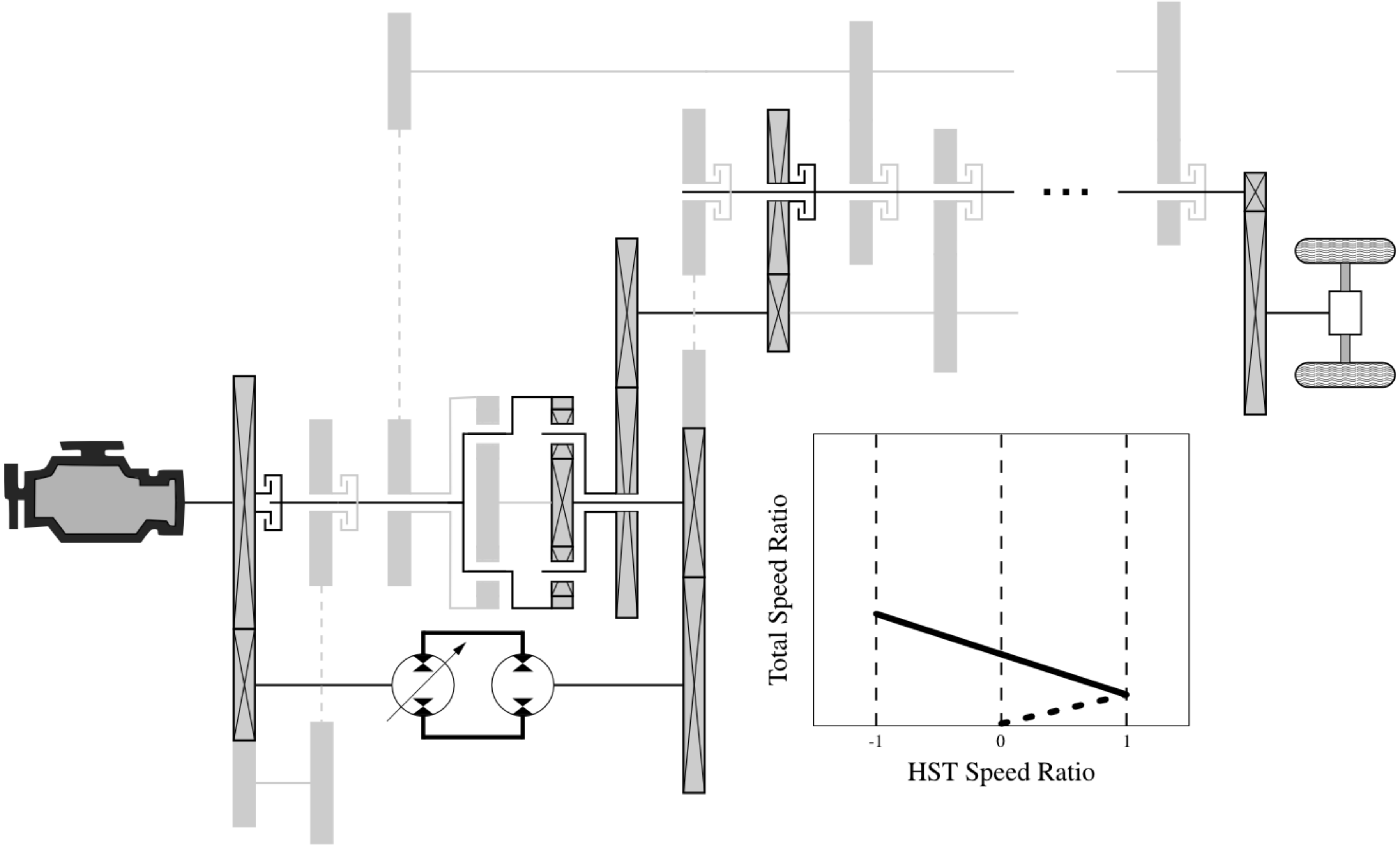


Hydrostatic mode + 1,2,3,...,m input-coupled power-split modes

Mode 1



Mode 2



Mode 3