PRE-DESIGN OF WASTE HEAT RECOVERY ORC SYSTEMS FOR HEAVY-DUTY TRUCKS BY MEANS OF DYNAMIC SIMULATION

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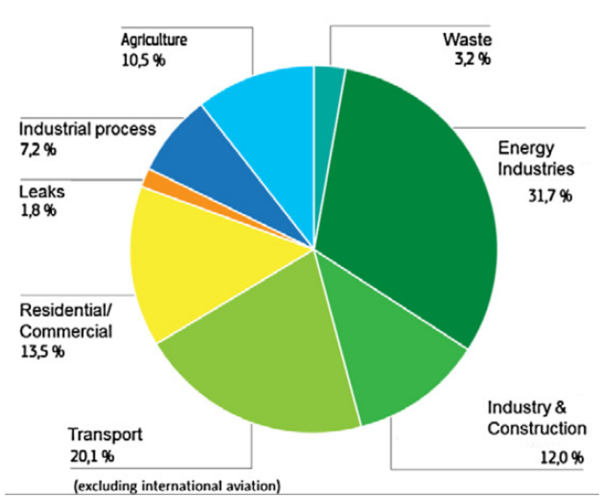
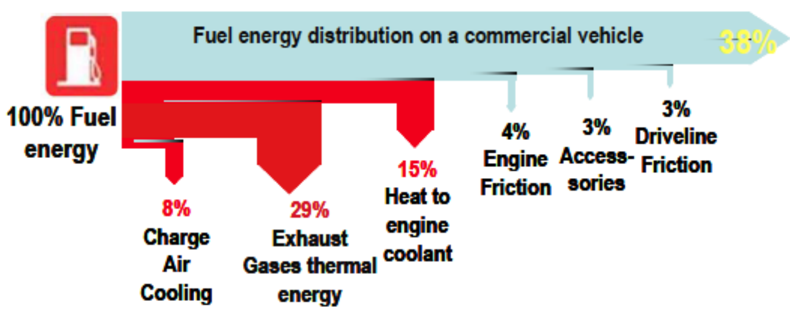
Introduction

Context

- Reduce fuel consumption is necessary
  - to reduce GHG emissions (HD represents ¼ of EU road transport emissions)
  - to increase competitiveness of transportation by trucks (fuel=28% of the total operating cost of the truck)
- How could we reduce fuel consumption?
  - Waste heat valorization is a promising solution
  - Even with a large engine efficiency, 50-60% of fuel energy is lost in waste heat

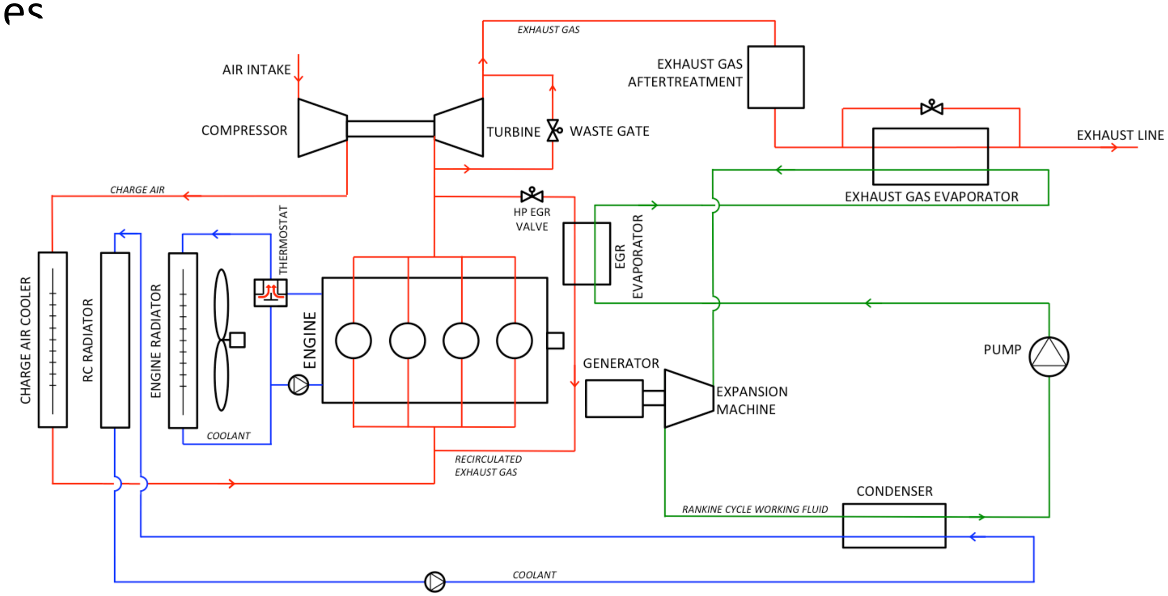




Typical energy distribution on a euro 5 engine

Introduction
ORC technology

- Among the WHR techniques, the Rankine cycle is one of the most promising ones

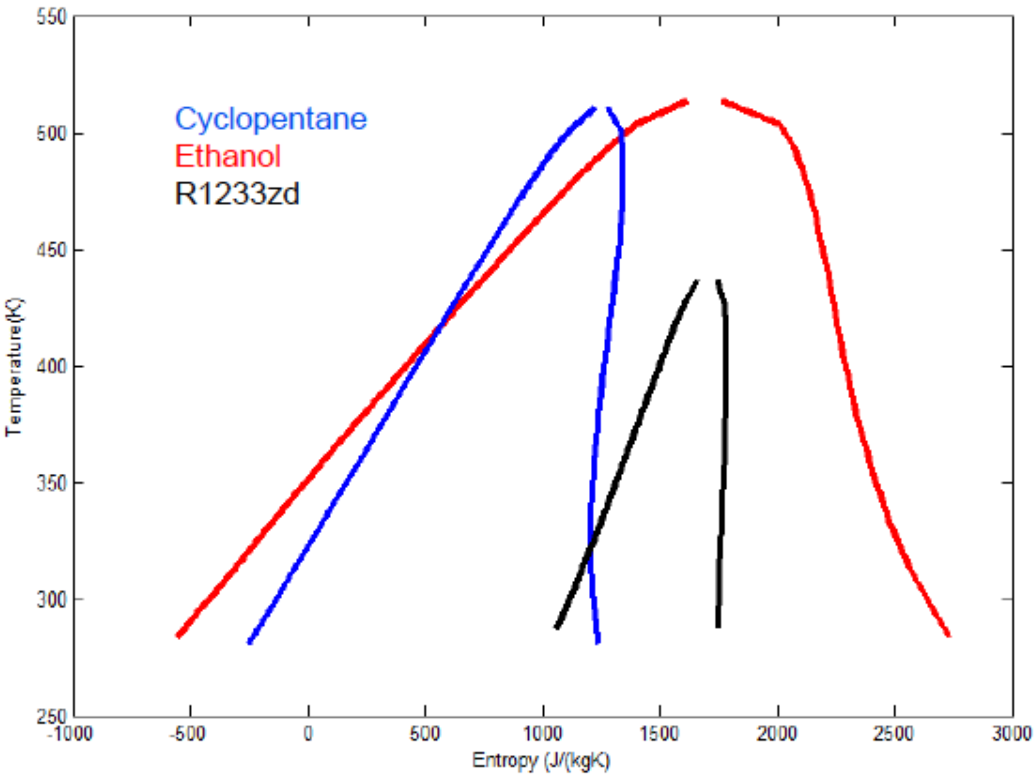


Many possible architectures for given boundary conditions

- However, R&D activities are still necessary to find the most appropriate architecture (working fluid, heat source/sink, expansion machine, etc.) in order to reach an acceptable economical profitability and to increase reliability

Introduction

ORC technology – working fluids



Criteria:

- ✓ Good thermodynamics performance (for low and high t° heat sources)
- ✓ Non-flammable, non toxic
- ✓ Low cost
- ✓ Compatibility with aluminum
- ✓ Dry fluids
- ✓ ….

Introduction

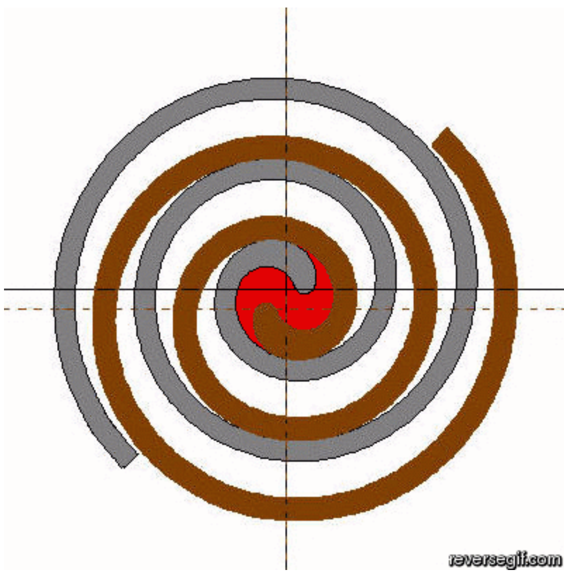
ORC technology – working fluids

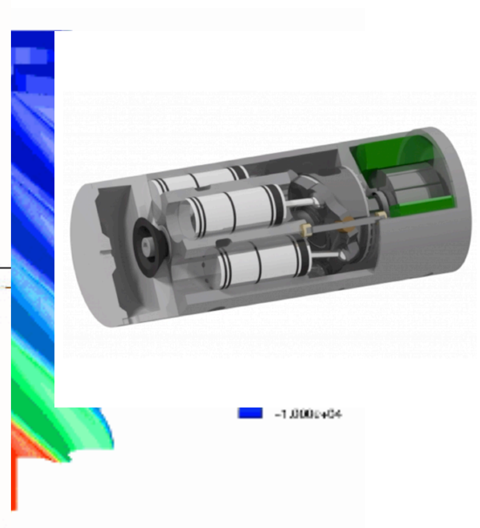
Volumetric expanders

Scroll

Screw

Piston

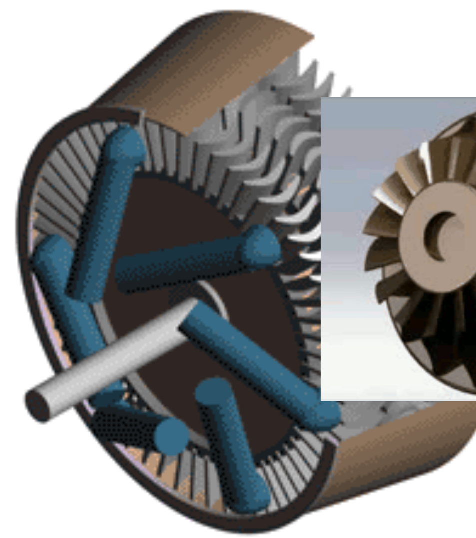




Turbomachines

Axial

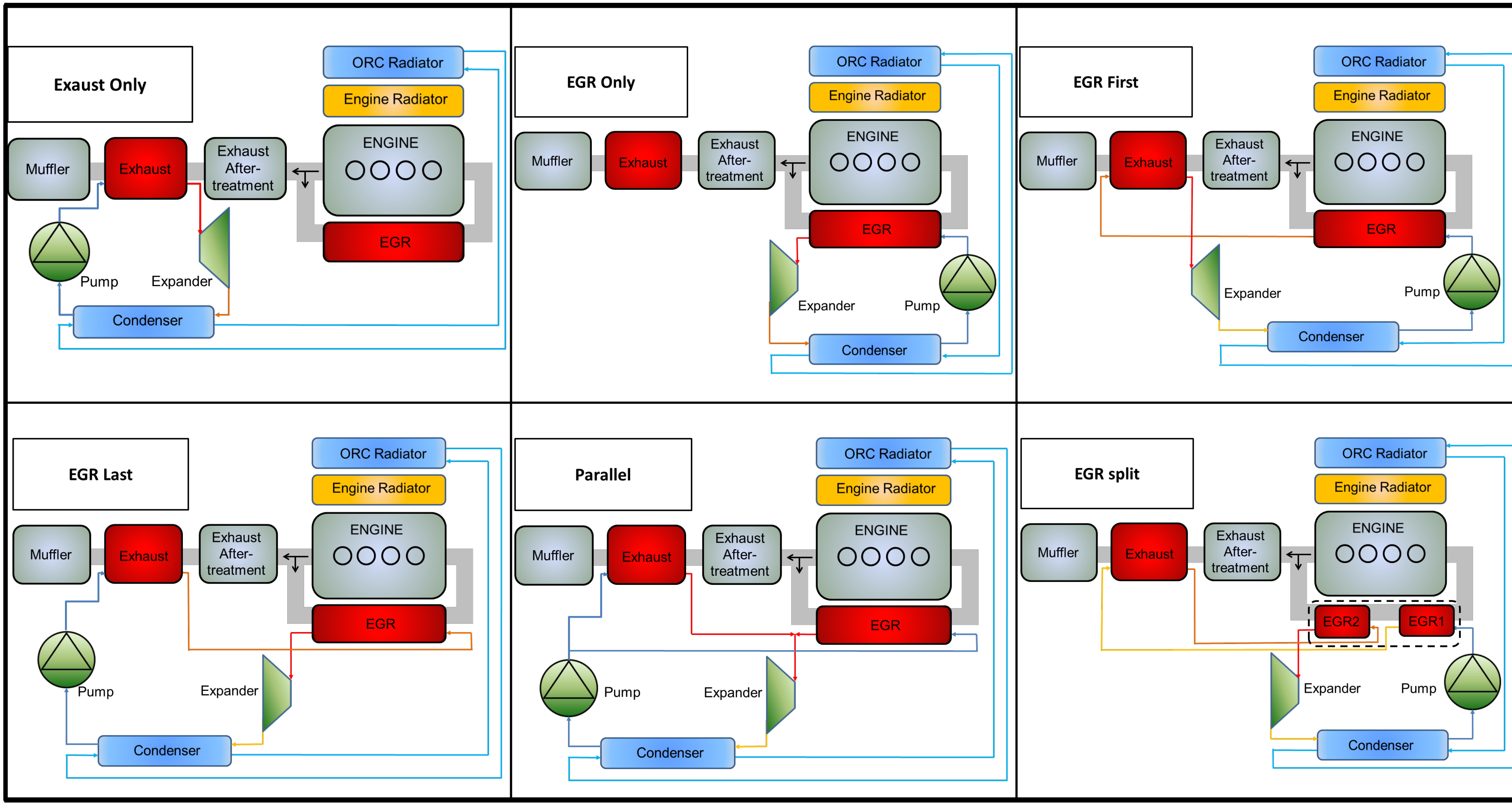
Radial



Introduction

ORC technology - heat sources and heat sinks

➔ 6 topologies



Introduction

ORC technology – previous work

- ✧ Previous work showed (VTMS London - 2017)
  - ✓ Ethanol + screw expander minimizes the Specific Investment Cost (EUR/kW)
  - ✓ Results are quite similar with scroll and piston
  - ✓ EGR last configuration (#4) does not ensure enough cooling of EGR

- ✧ Dynamic simulation on driving cycle should help refine the results

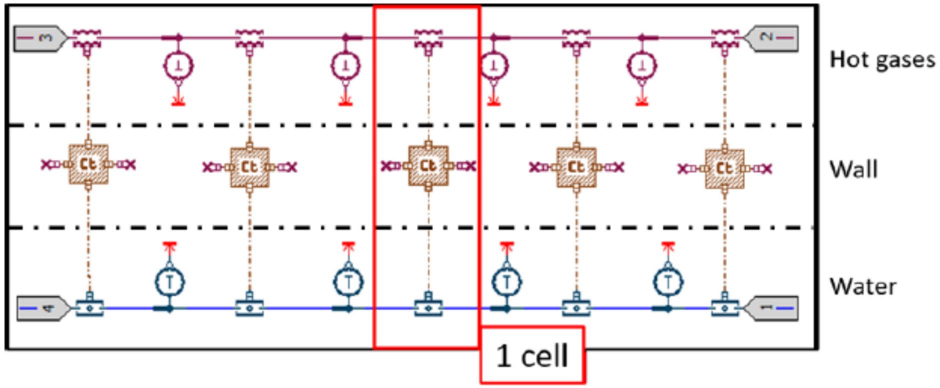
Content of the presentation

1. Introduction
2. **Dynamic simulation model**
3. Control strategy
4. Simulation results
5. Conclusions

Simulation model

Heat exchangers

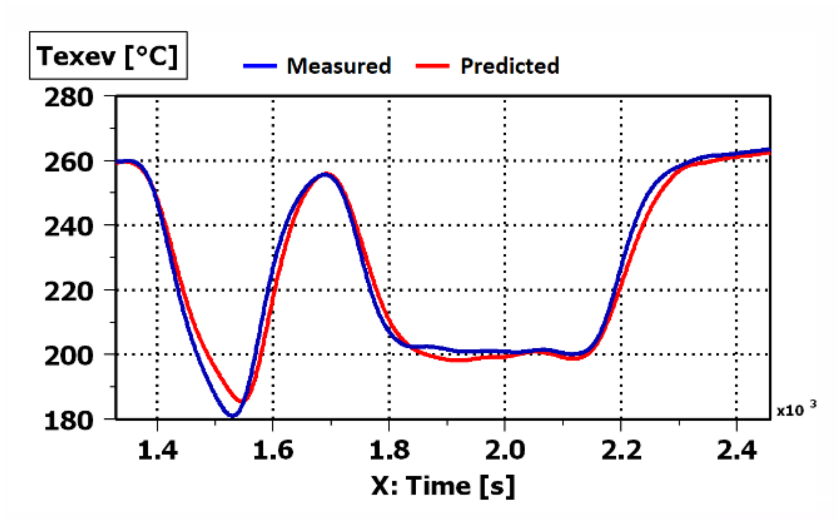
- Heat exchangers "concentrate" most of the dynamics of the ORC system
- Each side considered as a 1-dimensional tube in the flow direction
- Amesim (Siemens) modeling platform
- Finite volume approach: energy, mass and momentum balances are expressed and solved for each volume

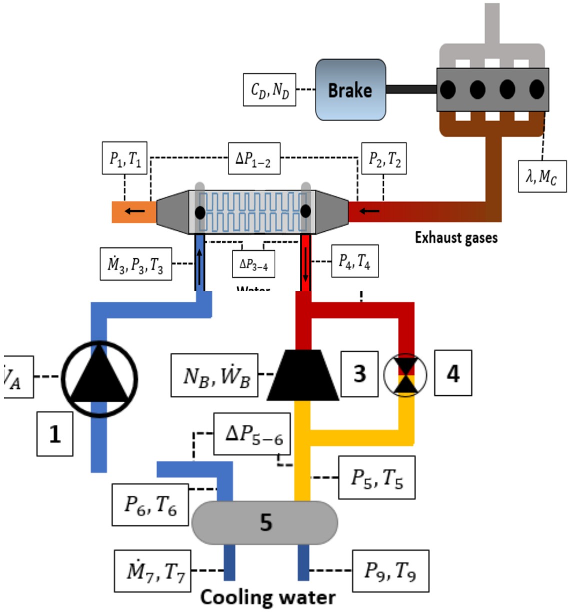


- Heat transfer coefficients are adjusted in order to reproduce results by steady-state heat exchanger models previously developed in Matlab.

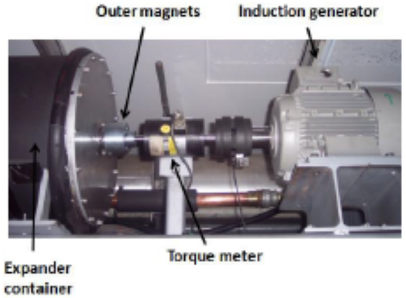
Simulation model
Heat exchangers

- Experimental validation of the model
  - Shell and plate heat exchanger
  - Connected to the tailpipe of a passenger car gasoline engine
  - Exhaust gas to water heat exchanger
  - Upwards and downwards steps on the pump flow rate



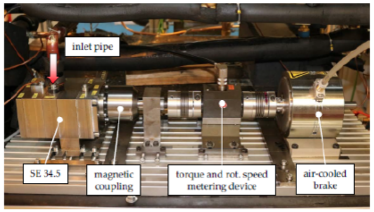


Simulation model
Pump and expansion machine



Scroll

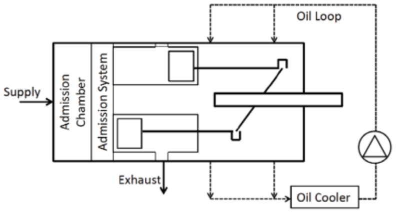
- 120 cm3 (compressor mode)
- Oil-free
- Tested with R245fa
- Built-in volume ratio close to 4.2
- Nominal power of 2 kWe



(Collaboration with Technical University Dortmund)

Screw

- 20 cm3
- Unsynchronized
- Tested with R245fa
- Built-in volume ratio close to 2.5
- Nominal power of 2 kWe

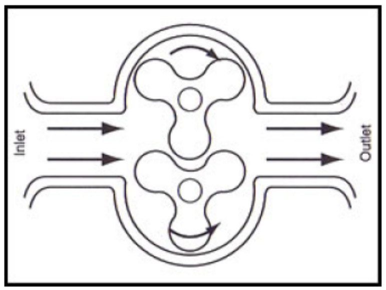


Axial-piston

- 195 cm3 (total cylinder)
- External oil lubrication loop
- Tested with R245fa
- Built-in volume ratio close to 5
- Nominal power of 4 kWe

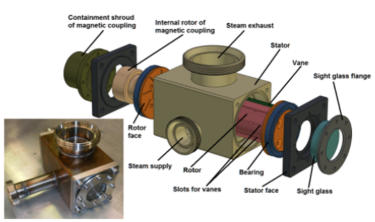
Simulation model

Pump and expansion machine



Roots

- 100 cm3
- Tested with R245fa
- Nominal power of 3.5 kWe
- Volume ratio close to 1



(Collaboration with
Czech technical university in Prague)

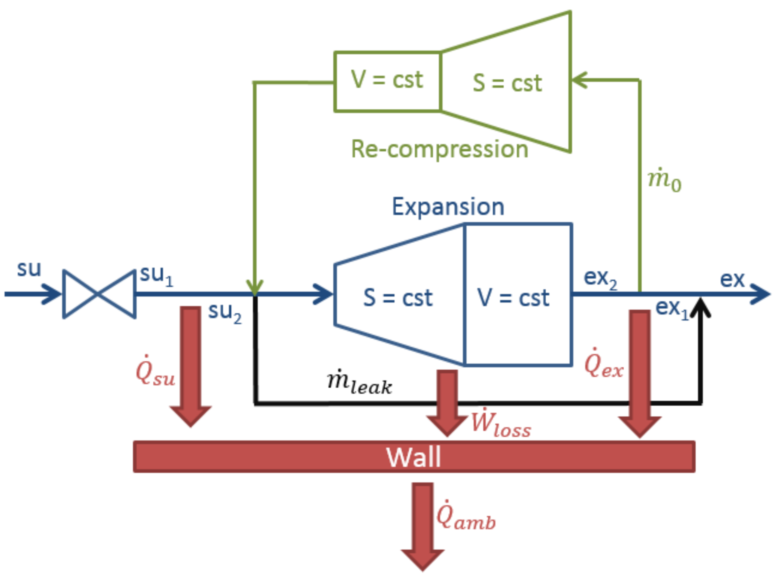
Vane

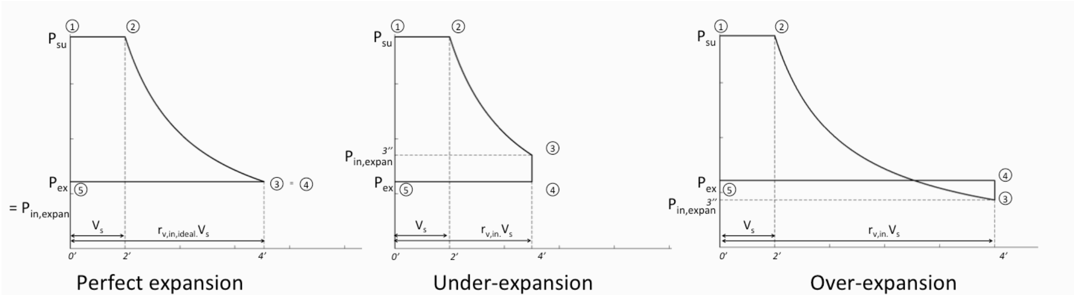
- 60 cm3
- Tested with siloxane
- Nominal power of 1 kWe

Simulation model

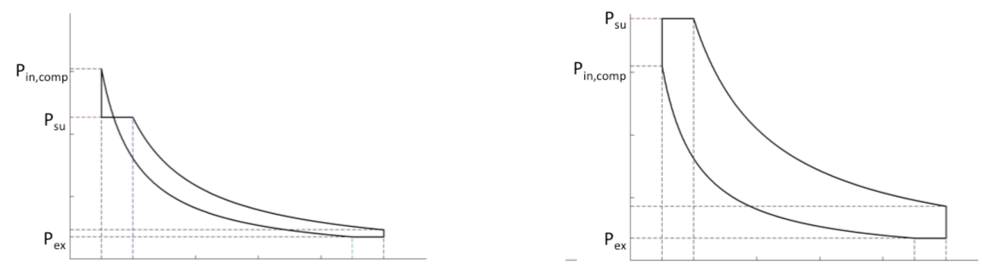
Pump and expansion machine

- Development of a generic grey-box model (lumped parameter model)
- Accounts for: inlet pressure droop, heat transfer between the fluid/machine/ambient, mechanical losses, leakages, under-/over- expansion/compression





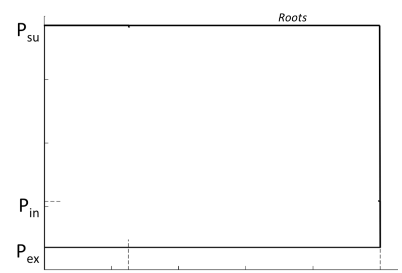
scroll-screw-vane



Over-compresion

piston

Under-compresion

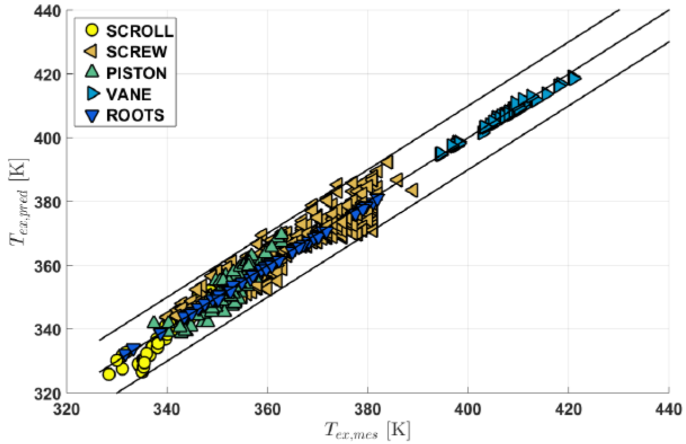


roots

Simulation model

Pump and expansion machine

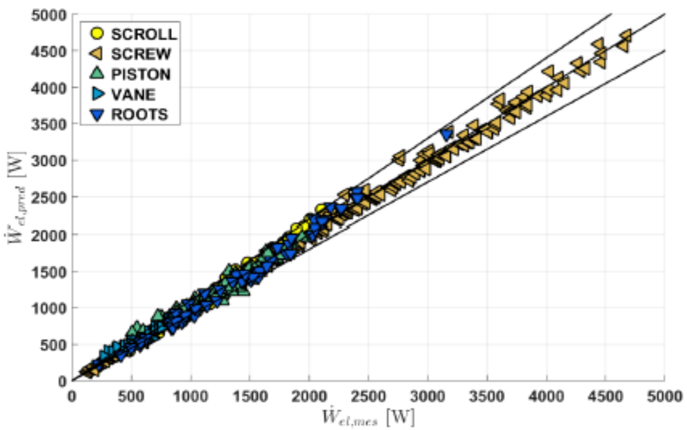
- Experimental validation of the model



Predicted outlet t°

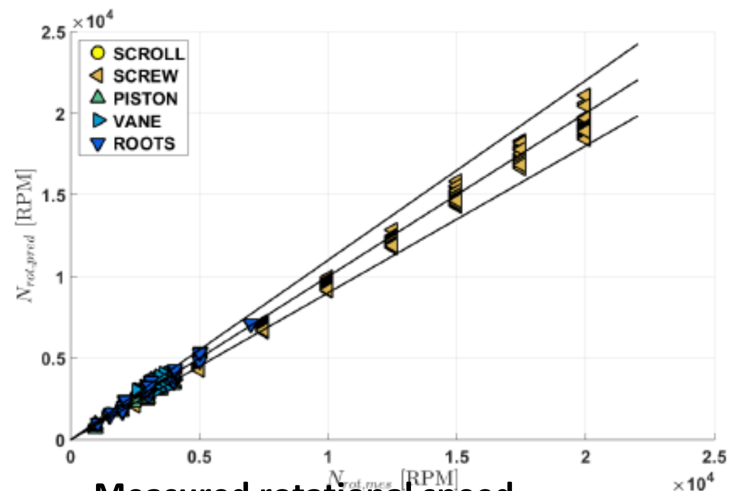
Measured outlet t°

R245fa test bench



Predicted shaft power

Measured shaft power



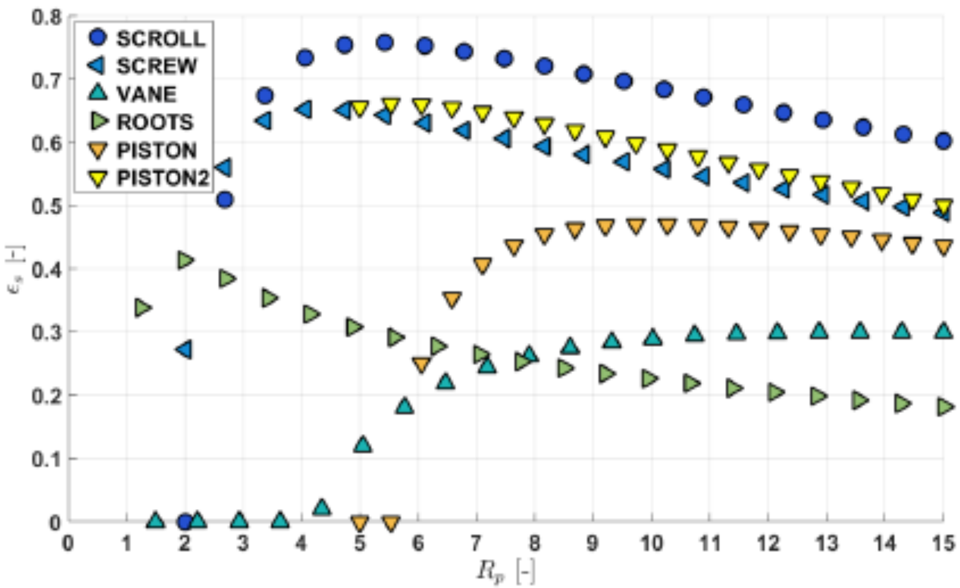
Predicted rotational speed

Measured rotational speed

Simulation model

Pump and expansion machine

- Prediction on the performance of the machines at their nominal rotational speed

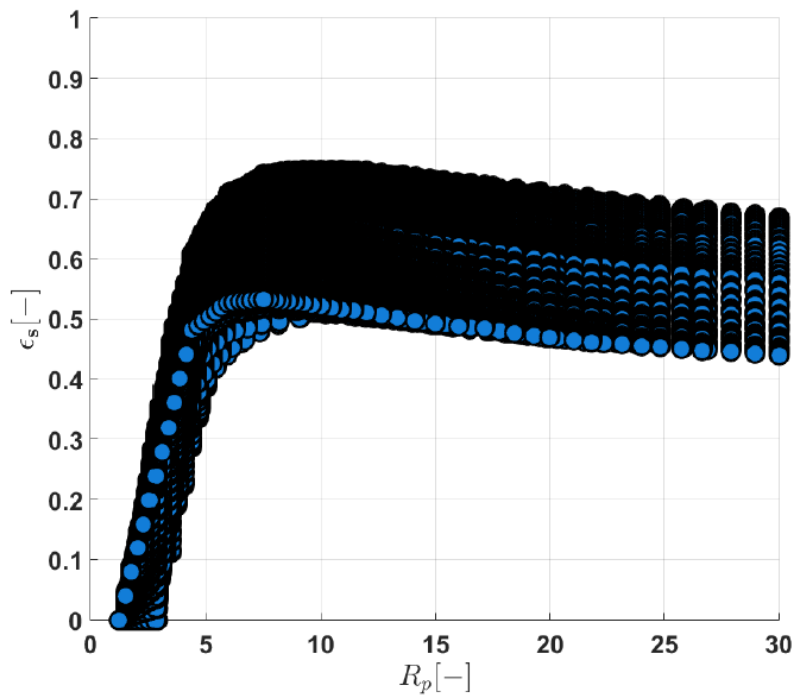


- ✧ Piston expander suffers from large clearance volume and large mechanical losses
- ✧ Leakages affect the performance of all machines
- ✧ Over-expansion losses is visible at low pressure ratio

Simulation model

Pump and expansion machine

- Dynamics of expanders (and pumps) is very limited compared with that of the heat exchangers (much smaller time constants)
- Grey-box model is used to derive operating maps of isentropic efficiency and filling factor as function of inlet/outlet pressures, inlet temperature, rotational speed



Example of map for the screw expander

- Filling factor:

$$\phi = \frac{\dot{M}}{\dot{M}_{th}} = \frac{1}{\epsilon_v}$$

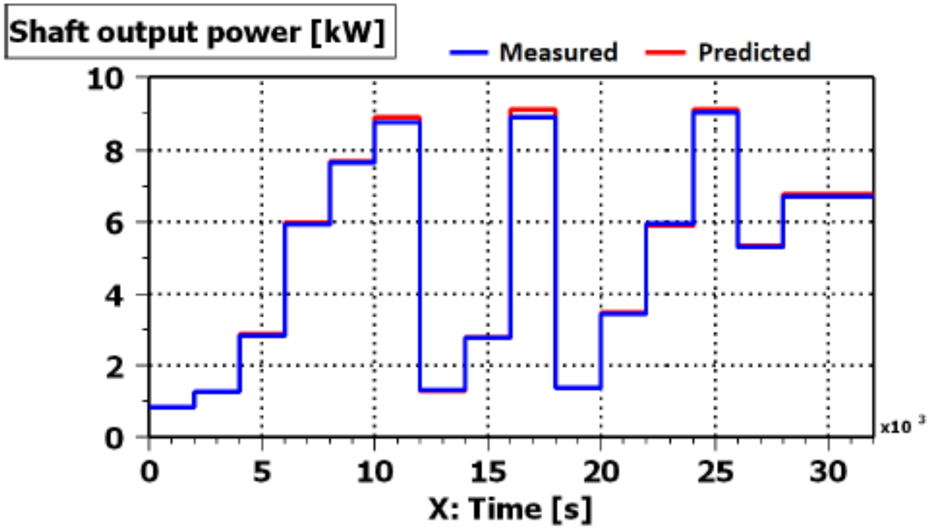
- Isentropic effectiveness:

$$\epsilon_s = \frac{\dot{W}}{\dot{M}\left(h_{su} - h_{ex,s}\right)}$$

Simulation model

Complete ORC system

- The complete ORC model is built by assembling components models
- Amesim model in steady-sate regime is compared to a previously developed steady-state model (built in Matlab) => good agreement



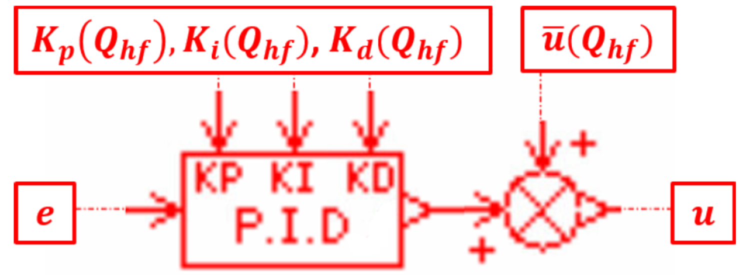
Control strategy

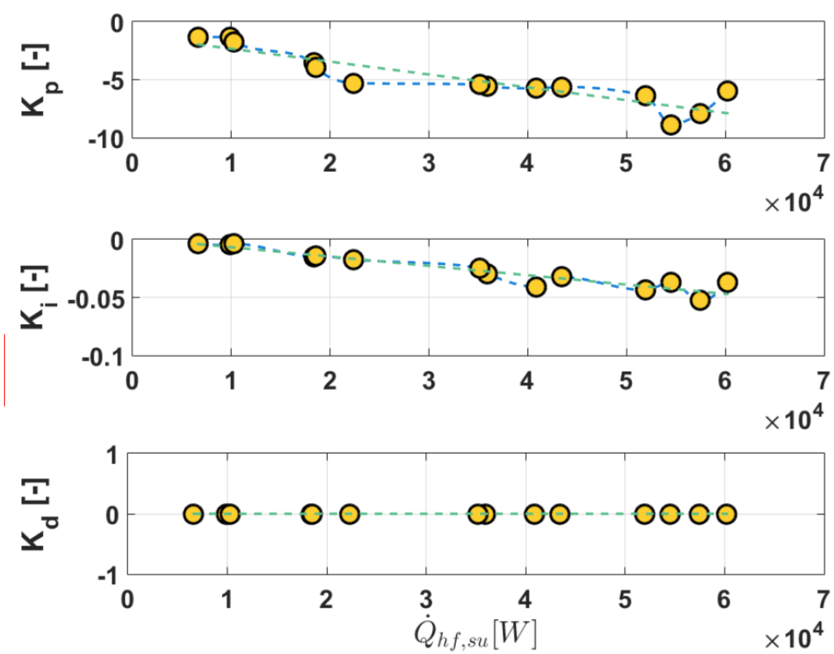
Gain-scheduled PID

- In order to conduct dynamic simulation, control (even simple) must be implemented
- Pump speed controls the superheat
- Expander speed controls the high pressure
- The system to control is non-linear
- Multi-linear model approach is considered = combination of linear models to approach the real system
- 17 operating points are considered

⇒ 17 FOPTD transfer functions

⇒ 17 sets of PID parameters interpolated as function of the heat source power ("gain-scheduling")



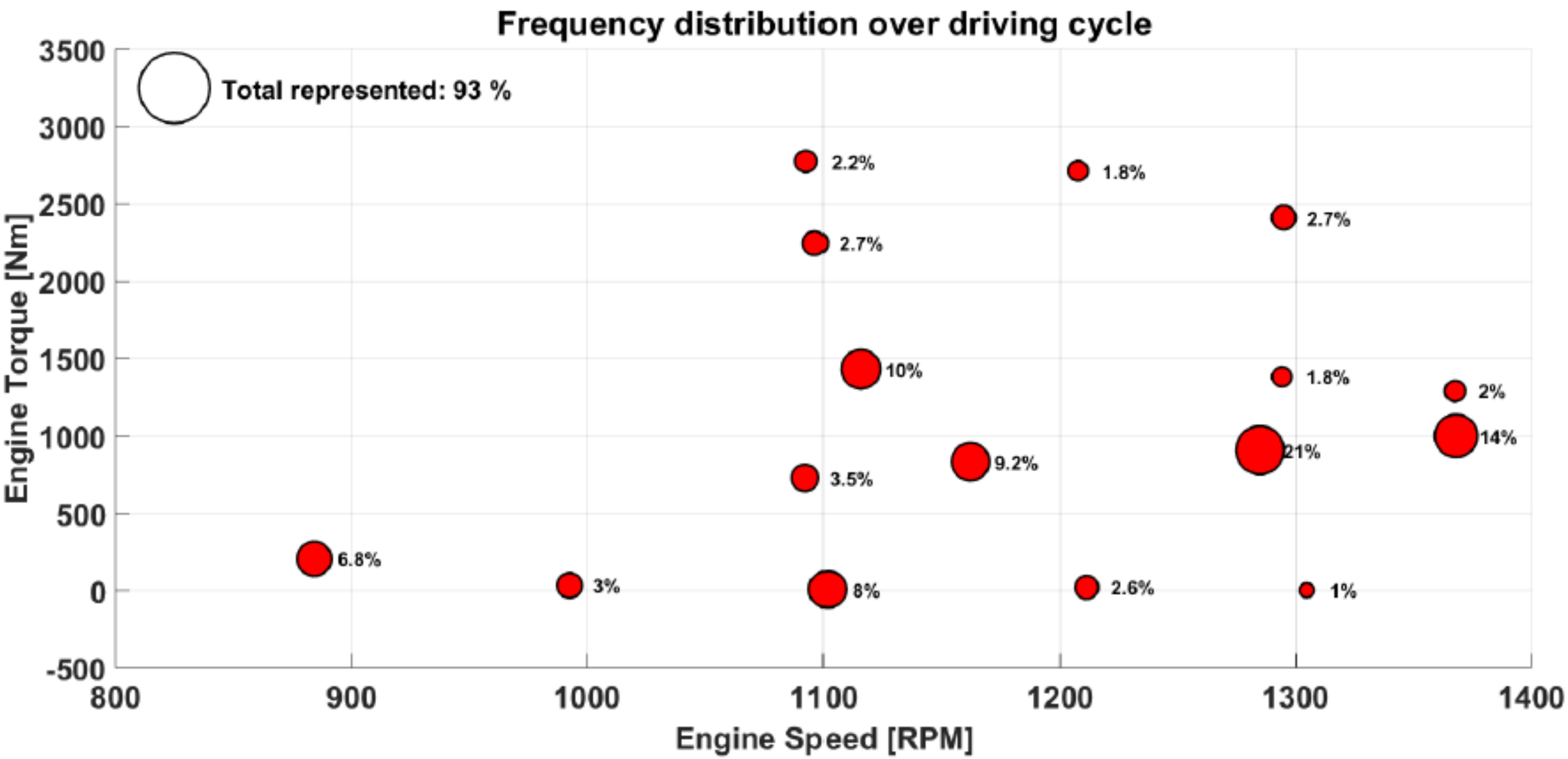


Example for the pump

Heat source power: $\dot{Q}_{hf,su} = \dot{M}_{EGR} C_{p,EGR,su}(T_{EGR,su} - T_{wf,sat}) + \dot{M}_{EG} C_{p,EG,su}(T_{EG,su} - T_{wf,sat})$

Control strategy

Operating points used for the gain-scheduled PID

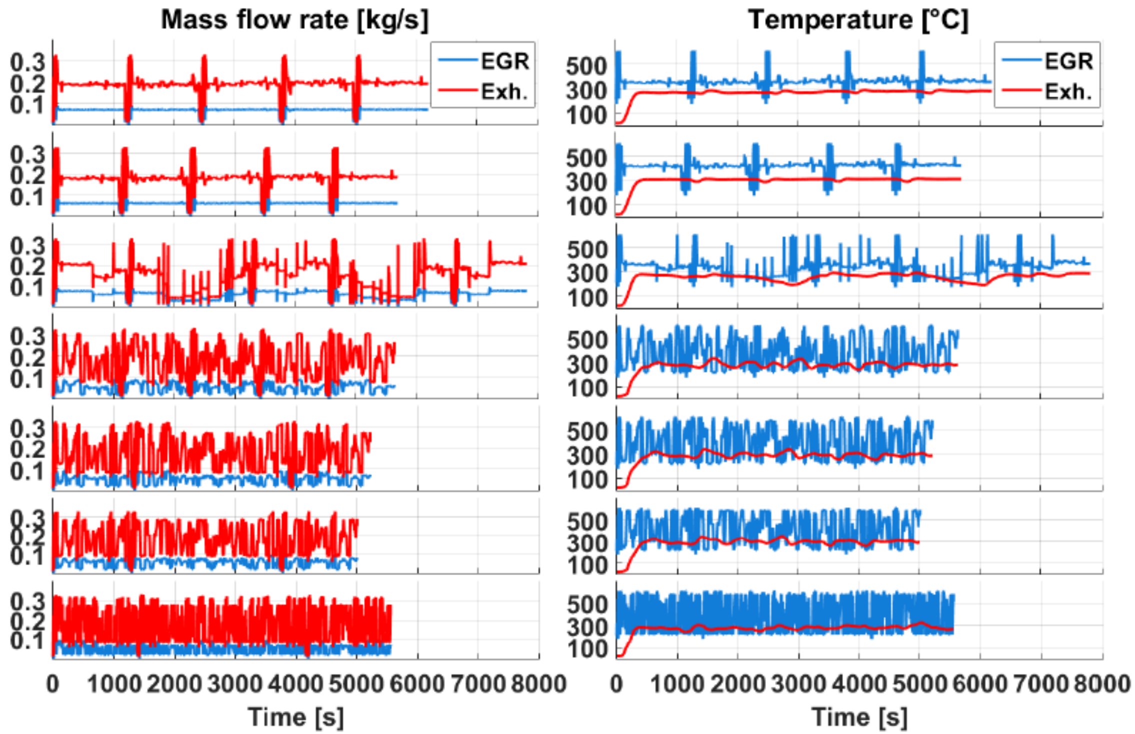


Simulation results

Driving cycle

- Driving cycle split into 7 phases (to represent all conditions met by a long-haul truck)
- Each phase is an independent driving cycle
- Each phase has a weight according to its contribution to real life of the truck

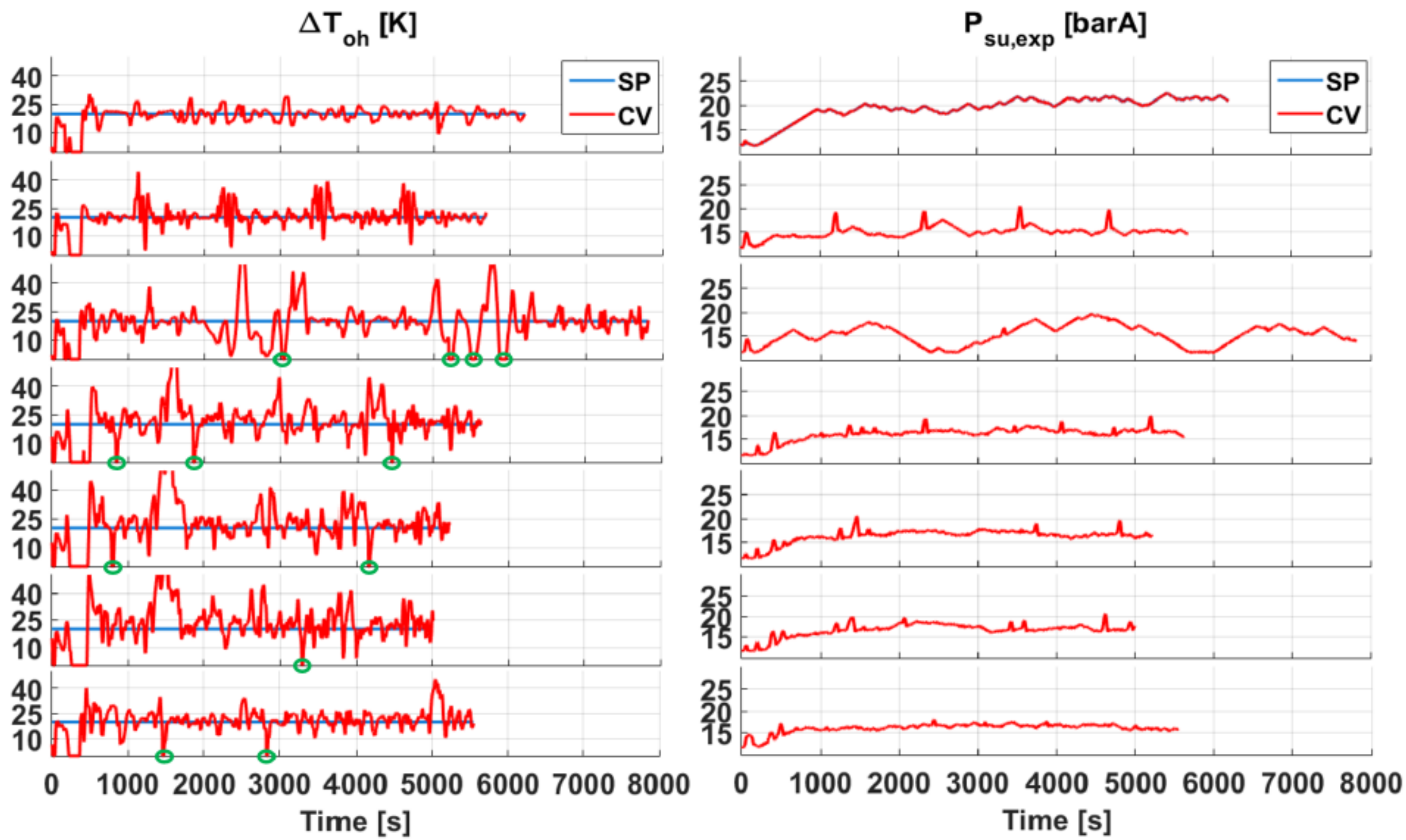
Driving cycle	1	2	3	4	5	6	7
Road type	Extra urban	Highway	Highway	Extra urban	Extra urban	Extra urban	Hilly
Vehicle speed	Mid	High	Mid	Low	Mid	High	High
Weight	0.1	0.1	0.5	0.075	0.1	0.075	0.05



Simulation results

Performances of the controllers

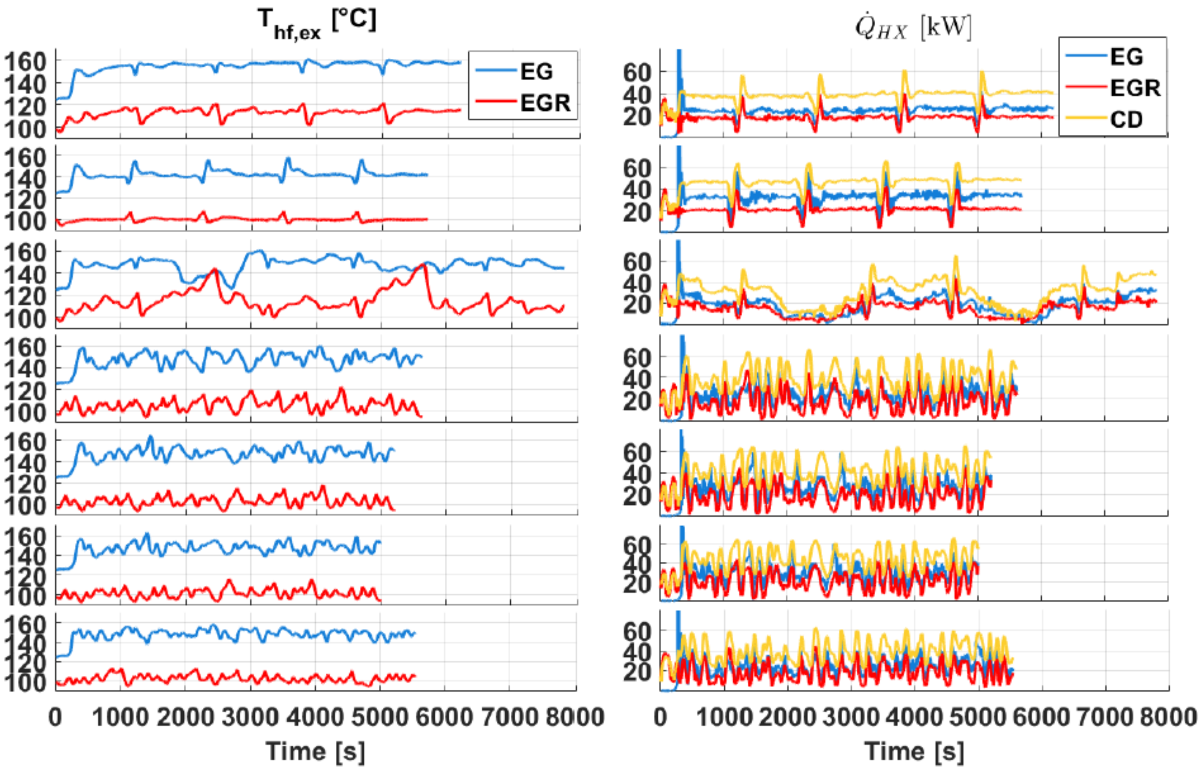
- Superheat set point: 20K (pump control)
- Optimal evaporating pressure set point identified based on a steady-state model and interpolated as function of heat source power (expander control)



Simulation results

Gas outlettemperartures and additional cooling load

- Additional cooling load approx. ranges between 20 and 40 kW (EGR does not yield additional cooling load)
- Recirculated exhaust gas outlet temperature must be between 100-120°C (NO_X emission reduction)
- Tailpipe gas outlet temperature >100°C (condensation)

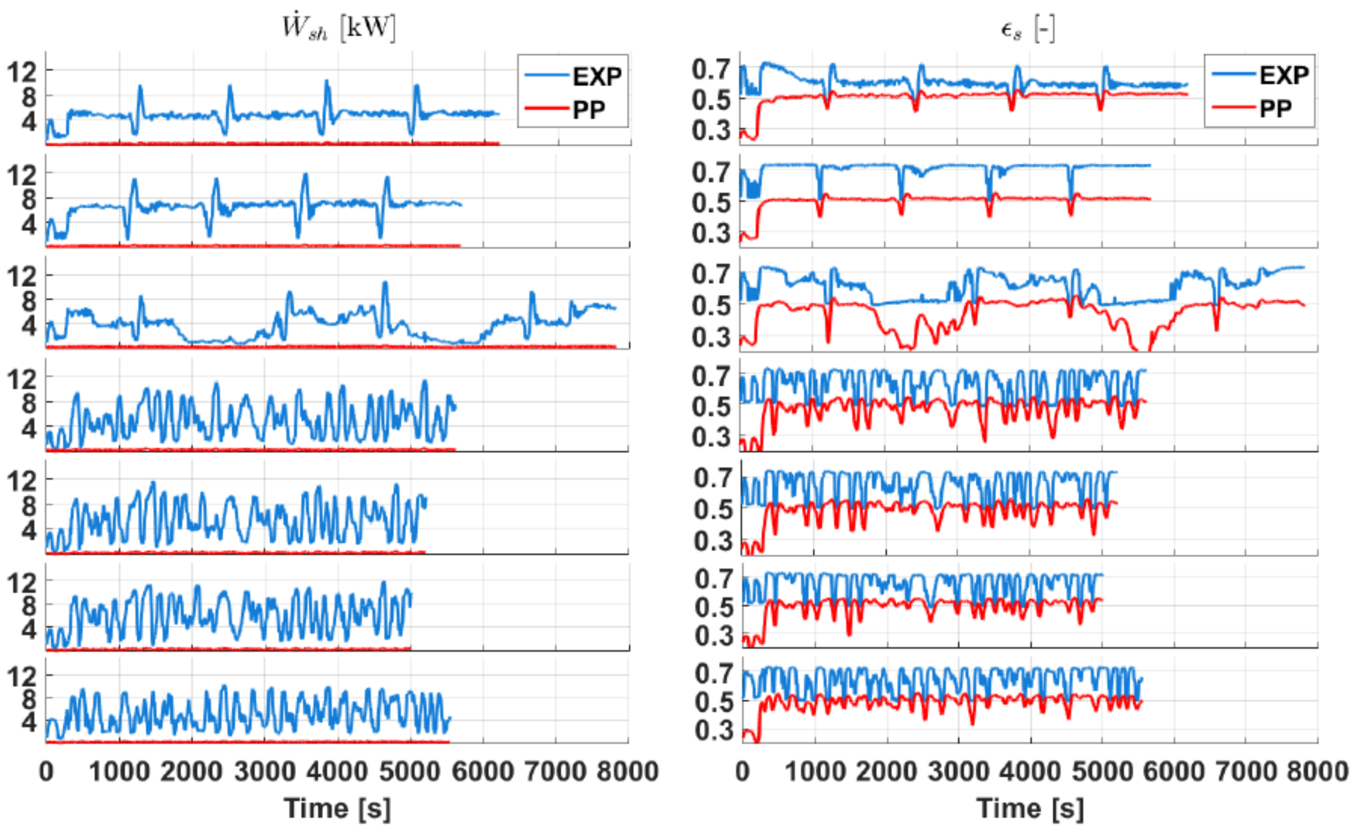


(screw expander + ethanol)

Simulation results

Gas outlettemperartures and additional cooling load

- Time evolution of the pump and expander shaft power and isentropic efficiencies



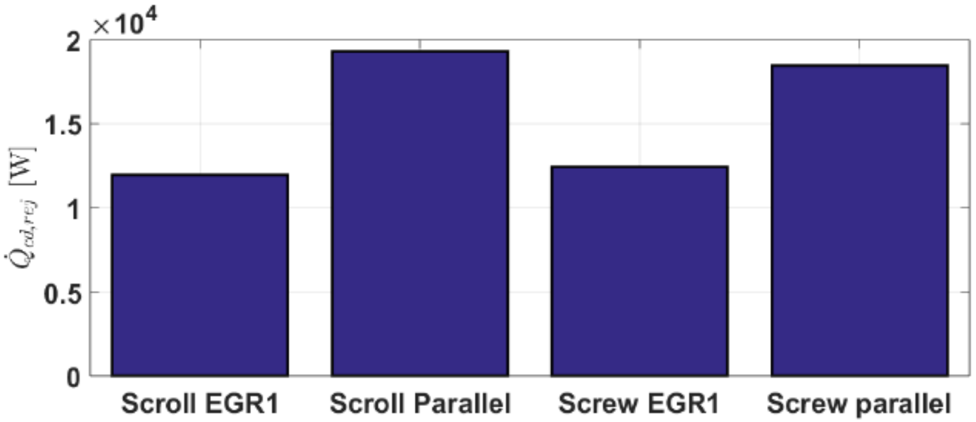
(screw expander + ethanol)

Simulation results

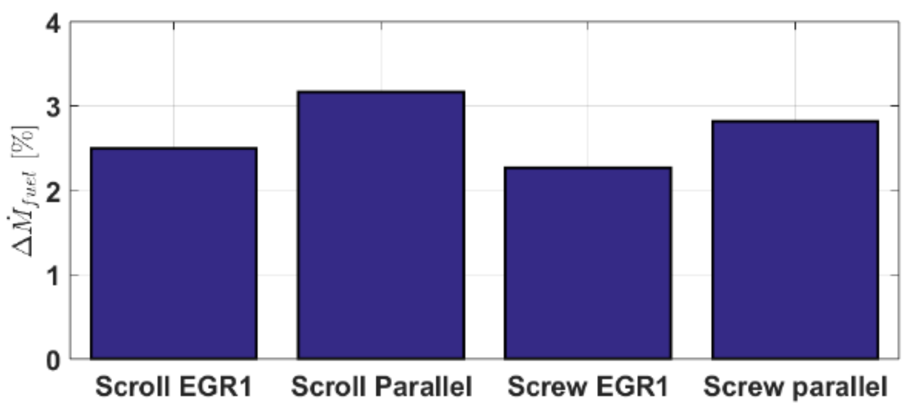
Average values of the composite driving cycle

<u>Energy performance indicators</u>

- Working fluid: ethanol
- Parallel architecture yields largest fuel savings, but also largest additional cooling load
- Fuel saving in the range of 2.3 to 3.2%



Additional cooling load



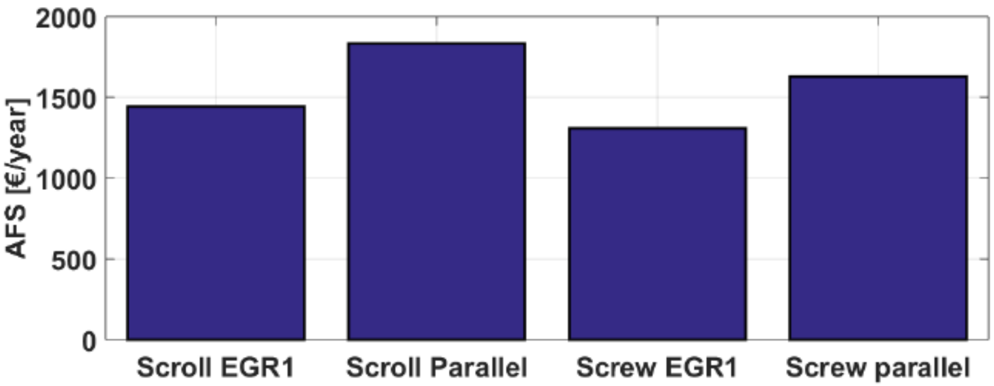
Fuel saving

Simulation results

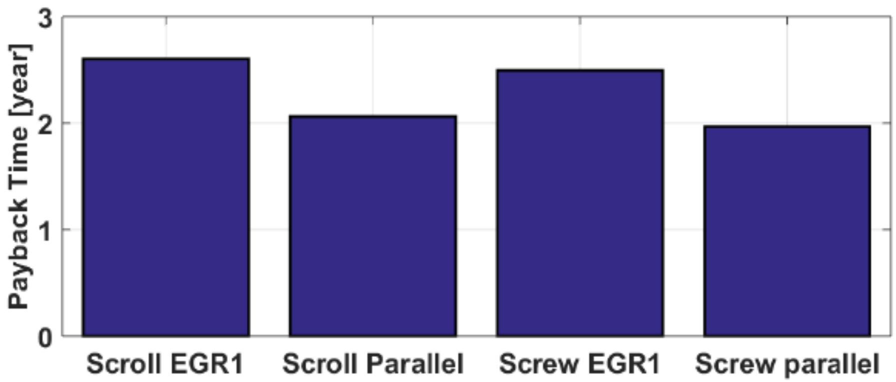
Average values of the composite driving cycle

Economic indicators

- Assumptions: 150,000 km/y; 35 l/100 km; 1.1 EUR/l; truck owner cost =1.5 TIC
- ORC weight: loss of trailer load (not taken into account)
- Payback time range from 2 years (parallel) to 2.5-2.6 years (EGR first)



Annual money savings



Fuel saving

Conclusions and future work

- Dynamic simulations of driving cycles give more accurate results regarding energy and economical performance
- An ORC dynamic model is built in Amesim platform (Siemens)
- Components and system models are validated by means of steady state and/or dynamic experimental data.
- Dynamic simulation requires controllers. 2 gain-scheduled PID controllers are implemented (control of the superheat and evaporating pressure)
- Performance is evaluated on a composite driving cycle
- Fuel consumption reduction from 2.3% (EGR first) to 3.2% (parallel)
- Payback time from 2 years (parallel) to 2.5-2.6 years (EGR first)
- Cooling load limitation and additional fan consumption must be taken into account
- Impact of ORC on engine warm-up phase could be taken into account

Thank you for your attention!

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