RAY W. HERRICK
LABORATORIES

Feasibility Study of ICE Bottoming ORC with Water/EG Mixture as Working Fluid

***Davide Ziviani*[a]*,* *Donghun Kim*[a]*, Swami Nathan Subramanian*[b]*,*

James E. Braun[a] *and Eckhard A. Groll*[a]

[a] *Purdue University, School of Mechanical Engineering, Ray W. Herrick Laboratories, West Lafayette, IN, USA,*

[b] *Eaton Corporate Research & Technology, Southfield, MI, USA*

4th International Seminar on ORC POWER SYSTEMS

ORC 2017
13 - 15 September
MILANO, Italy

Outline

- Introduction
- Affordable Rankine Cycle
- Thermophysical Properties of Water/EG Mixture
- Thermodynamic Cycle Modeling
- Results
- Conclusions
- Future Work

Introduction (1/2)

- Heavy Duty Diesel Engines (HDDEs) reject a considerable amount of energy to the ambient

- In order to meet the U.S. Department of Energy (DOE) break thermal efficiency (BTE) goals, WHR by means of ORCs has been identified by U.S. engine manufacturers as viable solution

- Research on ORC systems applied to passenger and commercial vehicles has flourished in recent years
  - ☐ Subcritical and transcritical cycles (e.g., Amicabile et al. 2015)
  - ☐ Cascade cycles (e.g., Chen et al. 2017)

Introduction (2/2)

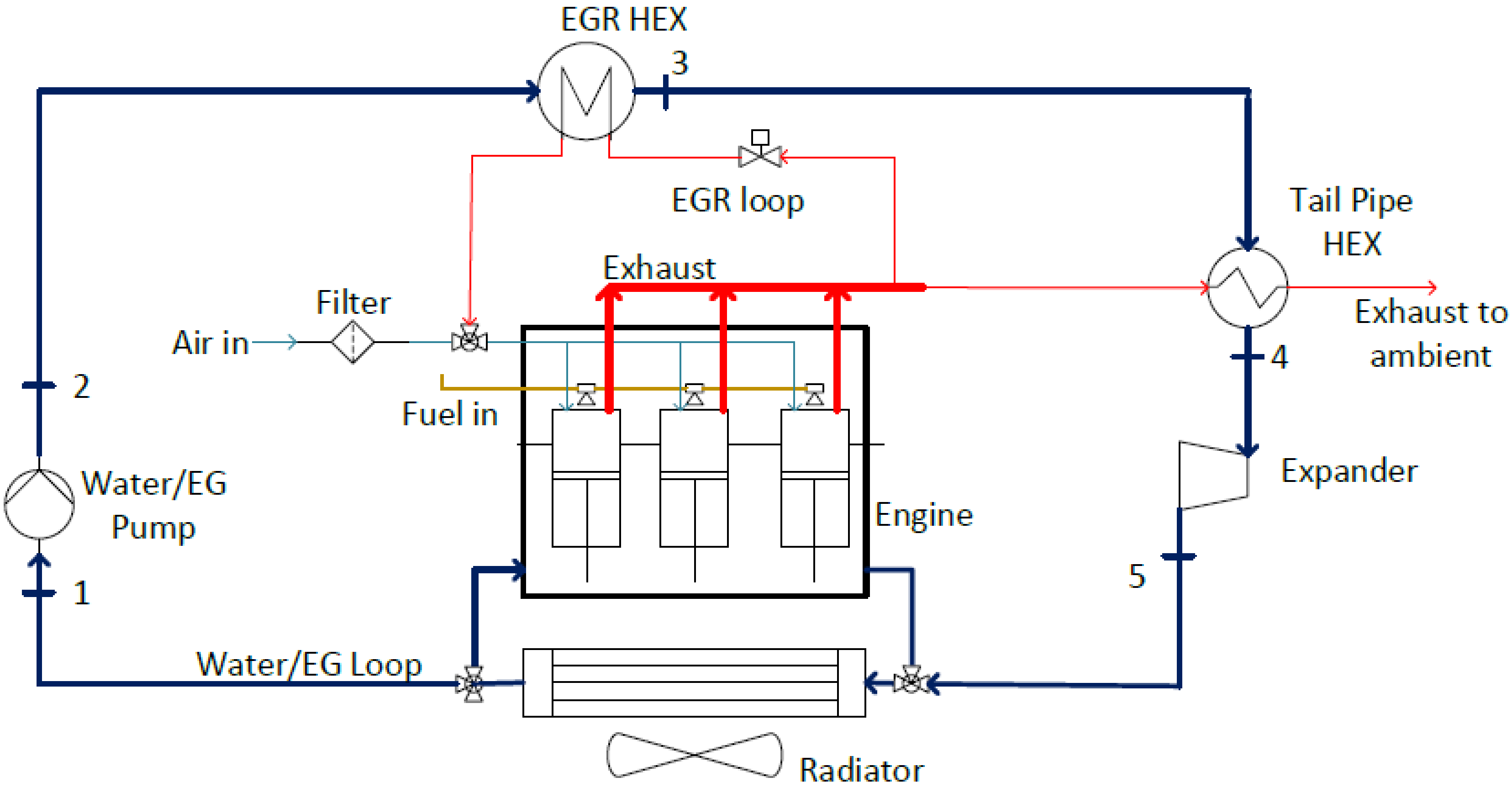
- Cost, complexity, environmental concerns and safety considerations are major issues that hold back OEMs from adopting ORCs in vehicles

- Return of investment period for the end customer is not highly attractive by using current technology (3 to 4 years payback period)

- An affordable Rankine cycle (ARC) system is proposed in order to obtain real benefits of WHR on the road and reduce the costs by 50% with a targeted payback period of 1.4 to 2 years

- A novel ORC architecture proposed within the ARC project is based on using the engine coolant as the working fluid.

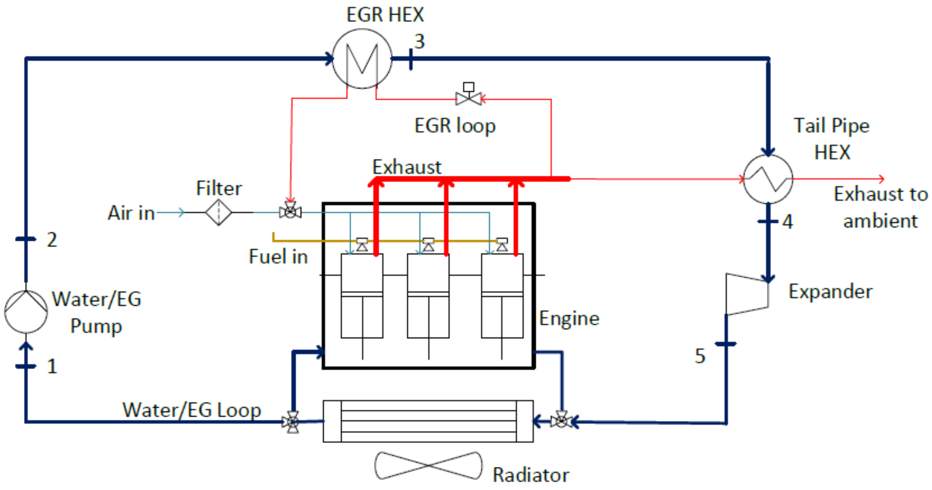
Affordable Rankine Cycle (1/2)

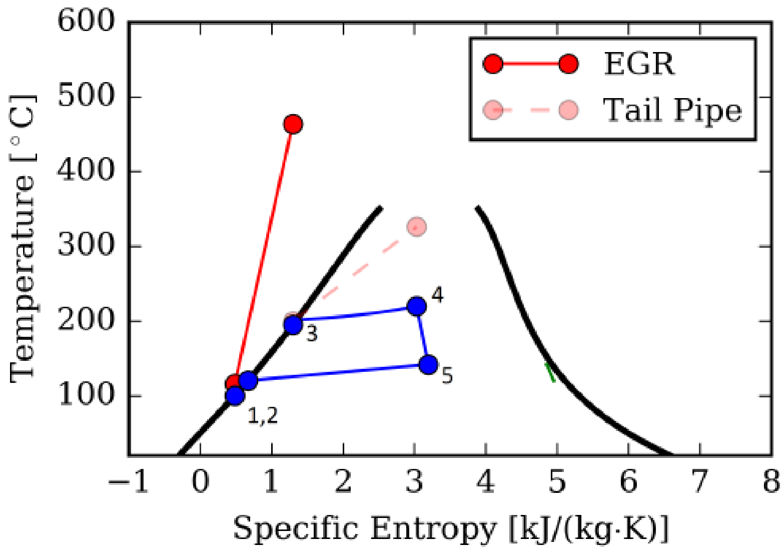


D. Ziviani (dziviani@purdue.edu)

Affordable Rankine Cycle (2/2)

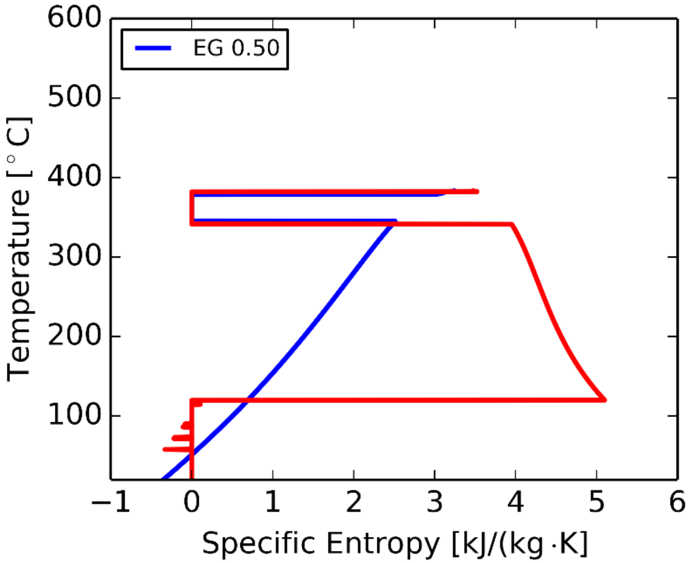
- ❑ Limitations arise concerning the maximum heat rejection rate.
- ❑ To ensure normal operation of the truck engine, the following constraints are taken into account:
  - Return temperature of engine recirculating gases
  - Maximum engine coolant temperature at expander inlet
  - Exhaust tail pipe boiler exit temperature





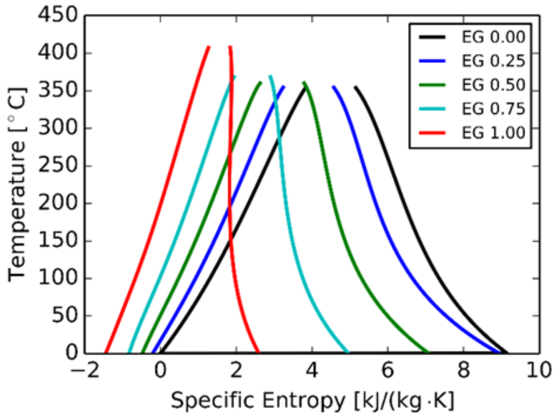
Thermophysical Properties of Water/EG Mixture (1/2)

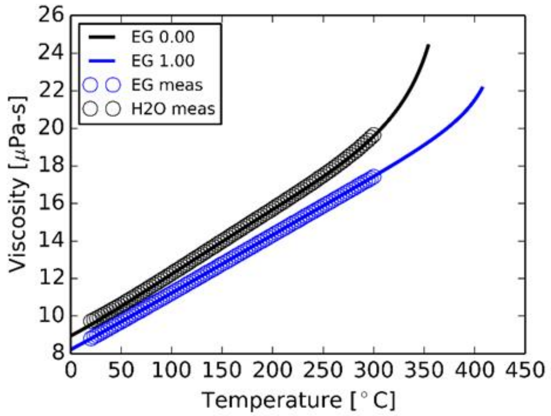
- The working fluid is a binary mixture of water and ethylene glycol.
- Few studies are available about thermodynamic and transport properties, e.g., Teja et al. 2003 and Dai et al. 2011.
- As the mixture phase-change is an important aspect, VLE conditions need to be obtained to understand the effect of concentration shifting.
- Original Water/EG mixture REFPROP file had issues:

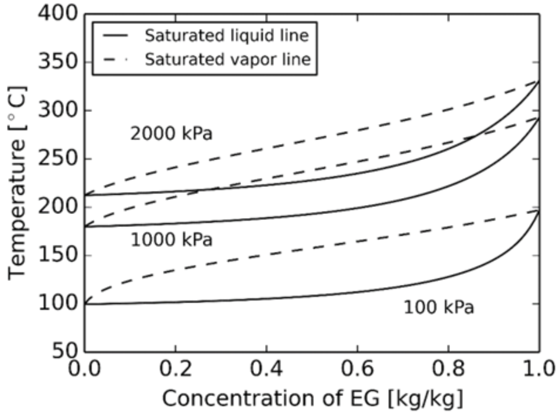


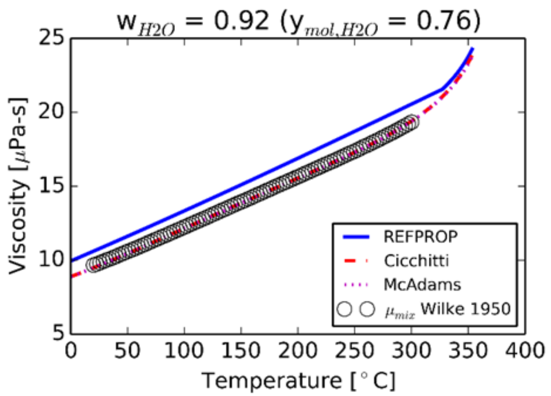
D. Ziviani (dziviani@purdue.edu)

Thermophysical Properties of Water/EG Mixture (2/2)









- ☐ Updated Water/EG mixture available within REFPROP 10.0 release

Thermodynamic Cycle Model (1/2)

- A steady-state cycle model has been developed in EES
- Heat inputs are determined from the engine operation
- The total heat rate available at the EGR and at the exhaust tail pipe:

$$\dot{Q} = \epsilon_{\mathrm{HX}} \dot{m} \Delta h$$

- The heat rejected by the radiator :

$$\dot{Q}_{\mathrm{cond}} = \dot{m}_{\mathrm{water/EG}} \Delta h_{\mathrm{radiator}}$$

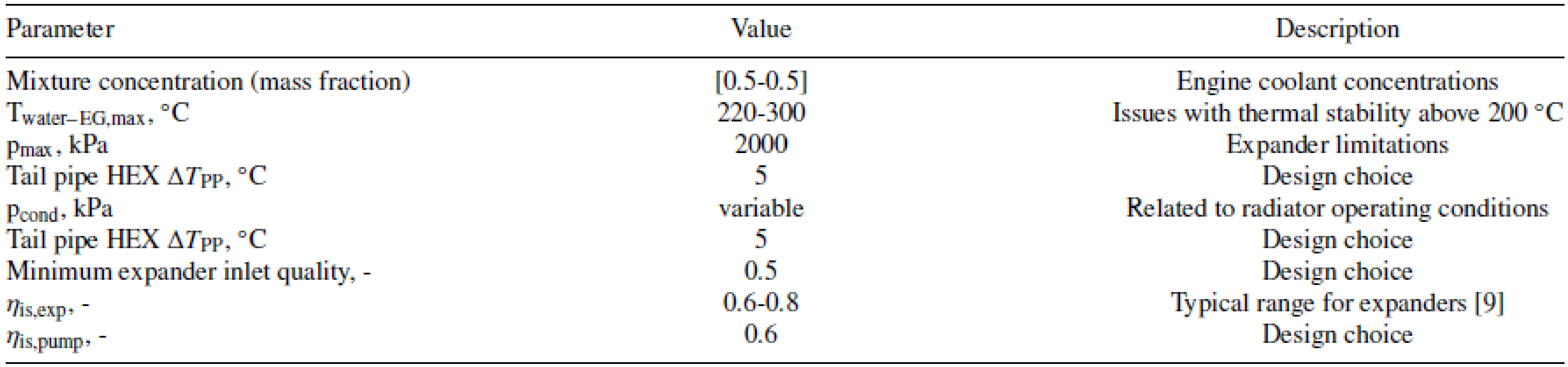
- Pump and expander are modeled by assuming the isentropic efficiencies
- The cycle performance and the benefits of the ARC system are quantified in terms of ORC thermal efficiency and Break Power (BP) improvement:

$$\eta_{\mathrm{ORC,net}} = \frac{\dot{W}_{\mathrm{exp}} - \dot{W}_{\mathrm{pump}}}{\dot{Q}_{\mathrm{EGR,in}} - \dot{Q}_{\mathrm{TP,in}}} \qquad BP = \frac{\dot{W}_{\mathrm{exp}} - \dot{W}_{\mathrm{pump}}}{\dot{W}_{\mathrm{engine}}}$$

D. Ziviani (dziviani@purdue.edu)

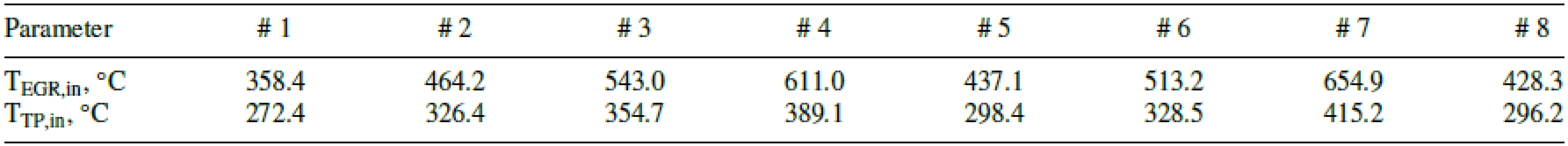
Thermodynamic Cycle Model (2/2)

- ❑ Cycle model assumptions:



Parameter	Value	Description
Mixture concentration (mass fraction)	[0.5-0.5]	Engine coolant concentrations
$T_{water-EG,max}$, °C	220-300	Issues with thermal stability above 200 °C
p_{max}, kPa	2000	Expander limitations
Tail pipe HEX ΔT_{PP}, °C	5	Design choice
p_{cond}, kPa	variable	Related to radiator operating conditions
Tail pipe HEX ΔT_{PP}, °C	5	Design choice
Minimum expander inlet quality, -	0.5	Design choice
$\eta_{is,exp}$, -	0.6-0.8	Typical range for expanders [9]
$\eta_{is,pump}$, -	0.6	Design choice

- ❑ Engine operating conditions:



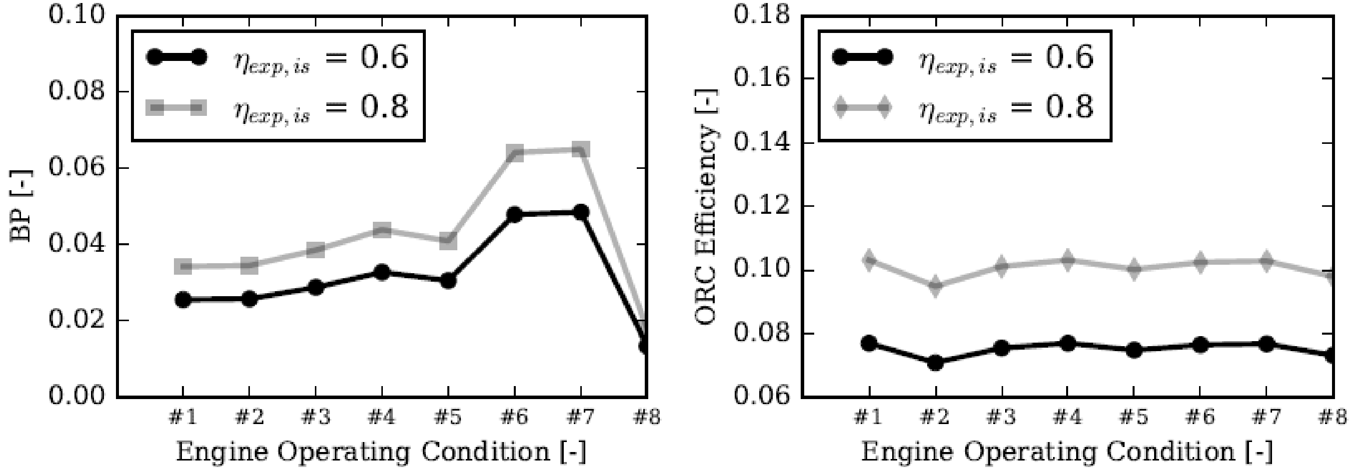
Parameter	# 1	# 2	# 3	# 4	# 5	# 6	# 7	# 8
$T_{EGR,in}$, °C	358.4	464.2	543.0	611.0	437.1	513.2	654.9	428.3
$T_{TP,in}$, °C	272.4	326.4	354.7	389.1	298.4	328.5	415.2	296.2

D. Ziviani (dziviani@purdue.edu)

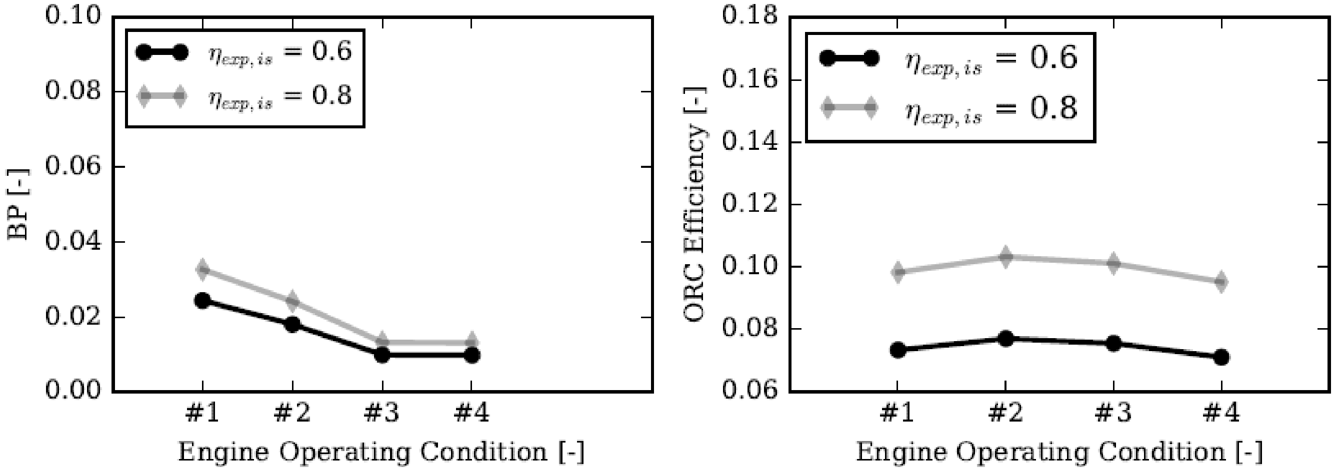
Effect of the Engine Operating Conditions (1/2)

$$T_{\text{exp,in}} = 220\text{°C}$$

Without heat rejection limitations

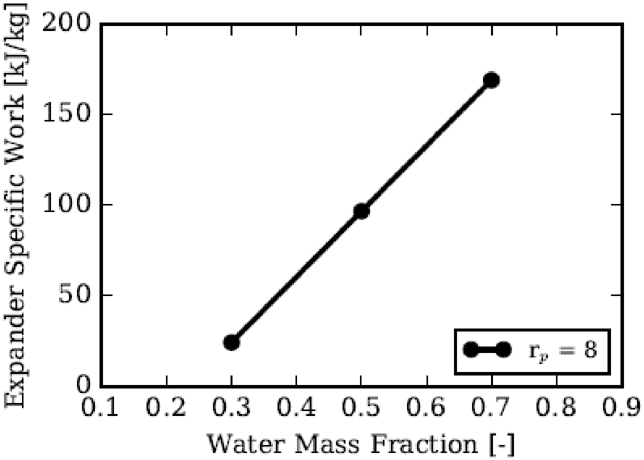


With heat rejection limitations

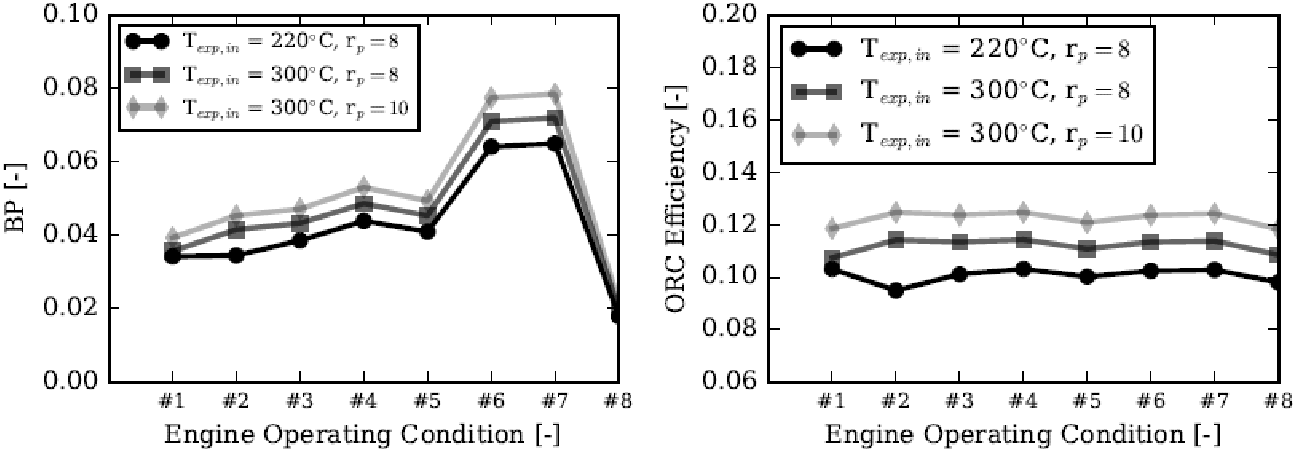


Effect of the Engine Operating Conditions (2/2)

- Variation of the water-EG mixture concentration



- Relaxed constraints on temperature and pressure



Effect of Expander Internal Volume Ratio (1/2)

- ❑ PD expanders are characterized by fixed internal volume ratio $r_{\mathrm{v,in}}$
- ❑ The theoretical internal specific work can be computed as:

$$w_{\mathrm{in,th}} = w_{\mathrm{is,exp}} + w_{\mathrm{V=const,exp}}$$

$$w_{\mathrm{is,exp}} = h_{\mathrm{su}}(T_{\mathrm{su}}, p_{\mathrm{su}}) - h_{\mathrm{in}}(v_{\mathrm{in}}, s_{\mathrm{su}})$$

$$w_{\mathrm{V=const,exp}} = v_{\mathrm{in}}(p_{\mathrm{in}} - p_{\mathrm{ex}})$$

$$v_{\mathrm{in}} = r_{\mathrm{v,in}} V_{\mathrm{s}}$$

- ❑ In the case of roots expanders:

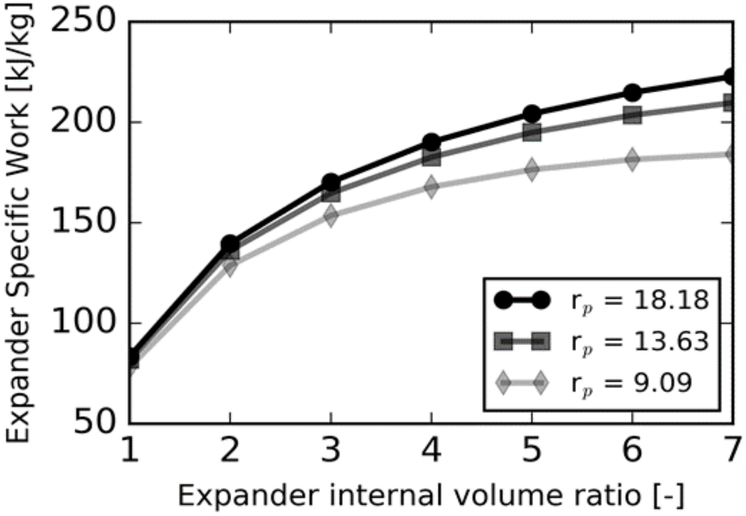
$$w_{\mathrm{in,th}} = w_{\mathrm{V=const,exp}} = V_{\mathrm{s}}(p_{\mathrm{in}} - p_{\mathrm{ex}})$$

- ❑ The actual specific work is affected by mechanical losses:

$$w_{\mathrm{in}} = w_{\mathrm{in,th}} \eta_{\mathrm{mech}}$$

Effect of Expander Internal Volume Ratio (2/2)

- ☐ To evaluate the influence of the expander volume ratio on the performance of the ARC running with a water/EG (0.5-0.5) mixture:
  - $p_{\mathrm{cond}} = 110\ kPa;\ p_{\mathrm{evap}} = 1000\ \mathrm{kPa}, 1500\ \mathrm{kPa}, 2000\ \mathrm{kPa}$
  - $T_{\mathrm{EGR,in}} = 430\ \mathrm{C};\ T_{\mathrm{TP}} = 294.4\ \mathrm{C}$



- ☐ $4 < r_{\mathrm{v,in}} < 6$ would be suitable to optimize the system

D. Ziviani (dziviani@purdue.edu)

Conclusions

- Water-EG mixture has been proposed as working fluid of an ORC for WHR within heavy-duty trucks
- A thermodynamic cycle model has been developed to investigate the potential improvements on the engine BTE.
- Simulation results showed that the employment of water-EG is heavily conditioned by engine operating conditions and high temperature limitations
- The maximum BP improvement obtained was 6.94% for engine operating point #7
- Although the initial parametric studies showed some potential for ARC architecture, additional work is needed to improve the performance especially under dynamic conditions
- A dynamic cycle model with different control strategies will be further developed

D. Ziviani (dziviani@purdue.edu)

References

Amicabile, S., Lee, J.I., Kum, D.. A comprehensive design methodology of organic Rankine cycles for the waste heat recovery of automotive heavy-duty diesel engines. Applied Thermal Engineering 2015;87:574–585.

Chen, T., Zhuge, W., Zhang, Y., Zhang, L.. A novel cascade organic Rankine cycle (ORC) system for waste heat recovery of truck diesel engines. Energy Conversion and Management 2017;138:210–223.

T., S., Teja, A.S.. Density, viscosity, and thermal conductivity of aqueous ethylene, diethylene, and triethylene glycol mixtures between 290 K and 450 K. J Chem eng Data 2003;48:198–202.

Dai, J., Wang, L., Sun, Y., Wang, L., Sun, H.. Prediction of thermodynamic, transport and vapor-liquid equilibrium properties of binary mixtures of ethylene glycol and water. Fluid Phase Equilibria 2011;301:137–144.

Future Work (1/2)

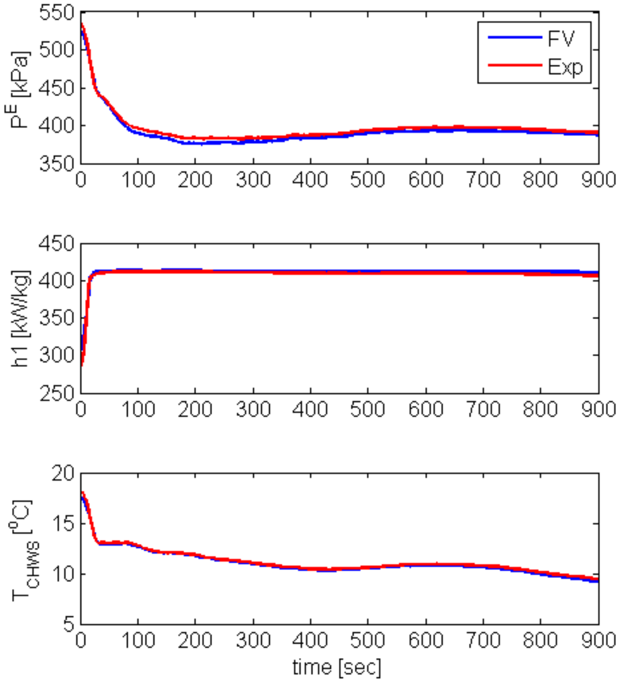
- WHR from a vehicle is a highly transient problem
- The optimization of the ORC during real operation requires a proper control strategy
- The development of a control strategy can be done by employing a dynamic model
- Since HXs influence the transient behavior of an ORC significantly, two dynamic models have developed
  - Moving Boundary Method (MB)
  - Finite Volume Method (FV)
- Challenges: binary-mixture, accuracy vs. computational speed, switching algorithm, numerical instabilities among others

D. Ziviani (dziviani@purdue.edu)

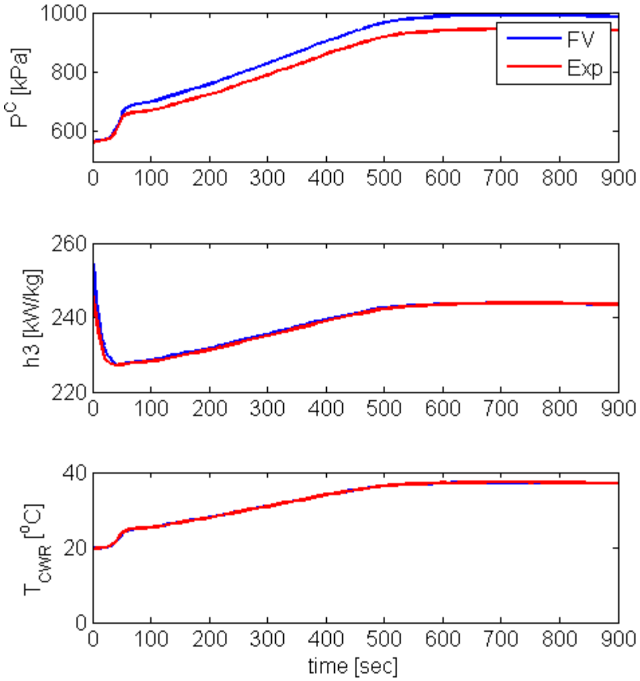
Future Work (2/2)

- Comparison between experimental and numerical results for FVM under dynamic conditions:

Evaporator

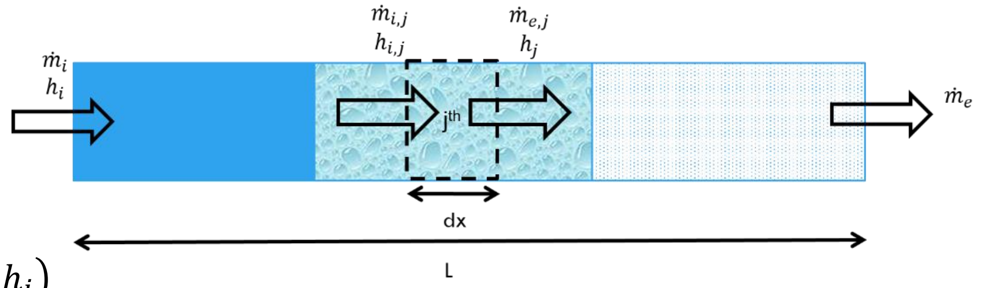


Condenser



Appendix (1/2)

- FV method is considered in this presentation

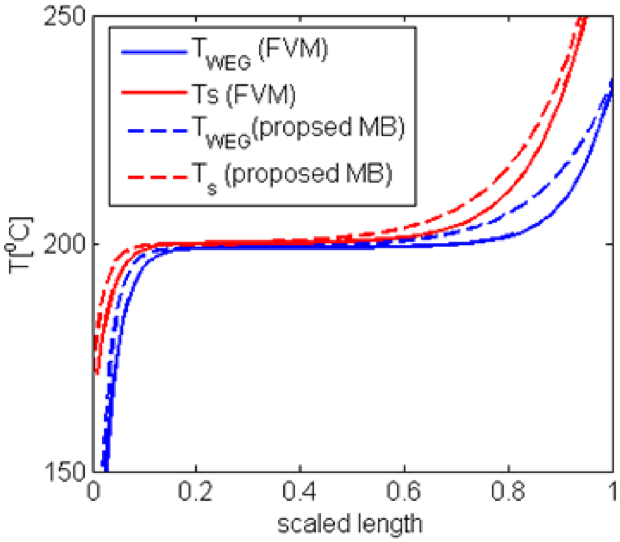


$$\frac{\partial \rho_j}{\partial p}\dot{p} + \frac{\partial \rho_j}{\partial h}\dot{h}_j = \frac{1}{Adx}\left(\dot{m}_{i,j} - \dot{m}_{e,j}\right)$$

$$\left(h_j\frac{\partial \rho_j}{\partial p} - 1\right)\dot{p} + \left(h_j\frac{\partial \rho_j}{\partial h} + \rho_j\right)\dot{h}_j = \frac{1}{Adx}\left(\dot{m}_{i,j}h_{i,j} - \dot{m}_{e,j}h_j\right)$$

$$\{h_i\,(t), \dot{m}_i(t), \dot{m}_e(t), \dot{m}_s(t), T_{s,i}(t)\} \longrightarrow \{p(t), h_e(t), T_{s,e}(t)\}$$

- Numerical comparison between MV and FV



Appendix (2/2)

- ❑ Profiles of model inputs (T_w: water temperatures, m_w: mass flow rates of water)
- ❑ Profiles of model outputs and experiments (m_r: refrigerant flow rate)