

Towards model-based control of a steam Rankine process for engine waste heat recovery

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Context



- **30% of the energy produced by internal combustion engine (ICE) is released as heat through exhaust gases**
 - partial recovery is possible via systems implementing the thermodynamic Rankine cycle
- **Intensive research dedicated to design issues**
 - comparatively little research on control issues
- **Control of “mobile” Rankine systems is difficult, due to**
 - multivariable and coupled nature of the process
 - fast transients
 - safe operating range of components

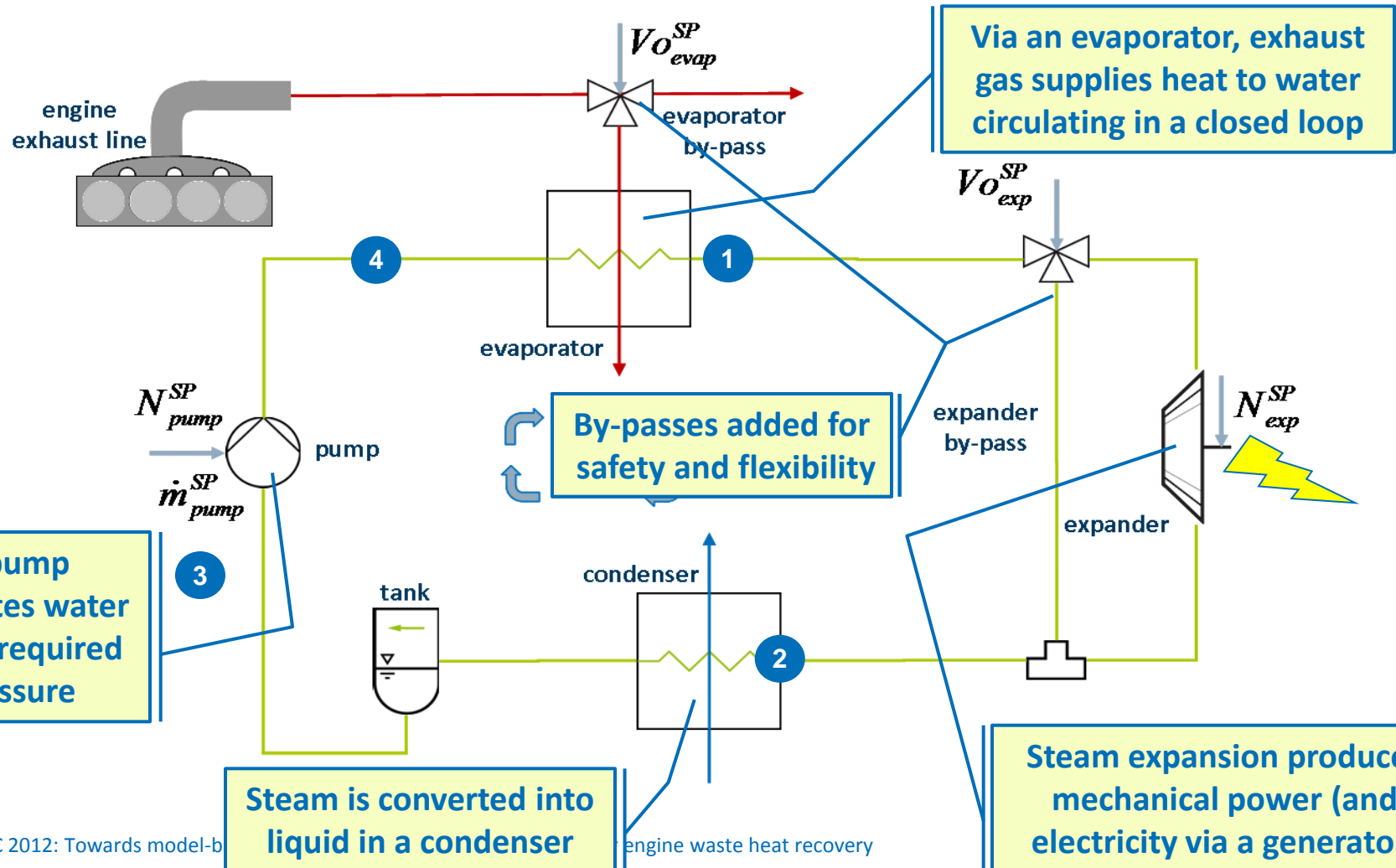
Objective and workplan

- **Developing a control system to be implemented on a test bench hosting a pilot Rankine system**

- **Steps**
 - **Developing a simulator for control prototyping**
 - **Control design**
 - **Supervisory layer to manage start-up and shut-down modes**
 - **Preliminary control strategy for power-production mode**
 - **Control validation on data coming from a motorway driving cycle**
 - **...**

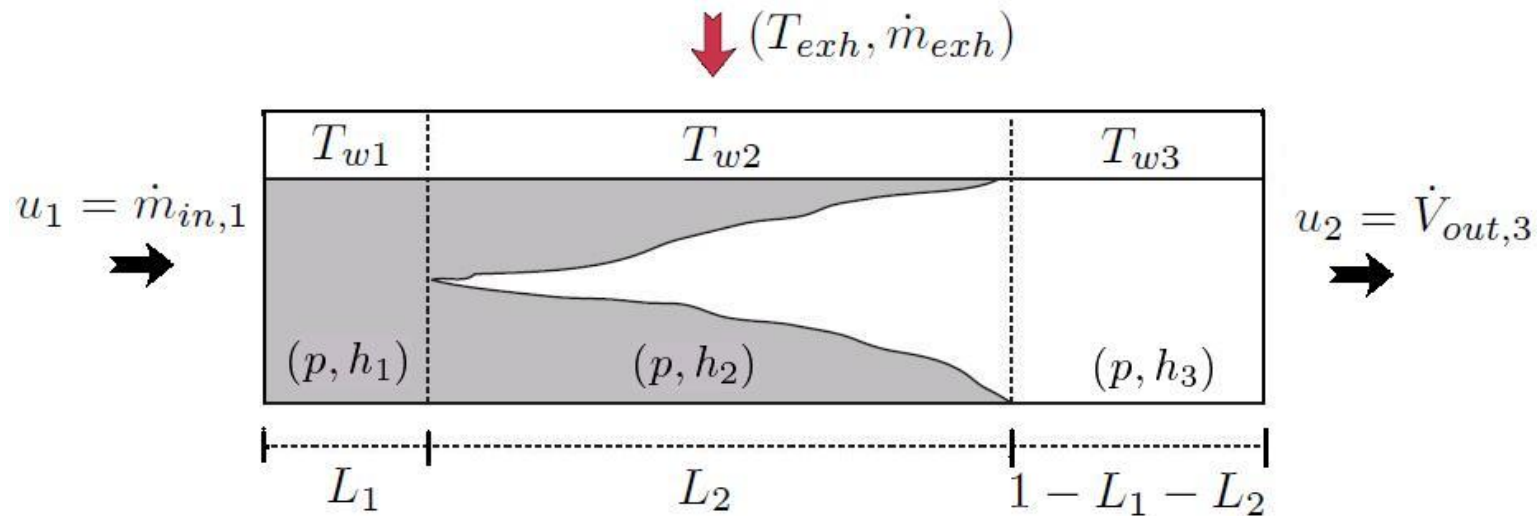
System layout

- Pilot Rankine steam process for recovering waste heat from the exhaust gas of a spark-ignition (SI) engine



Modeling of heat-exchangers: moving-boundary principle

- 3 volumes: liquid – two-phase – vapor



- Assumptions:

- uniform pressure
- negligible thermal conduction along the wall

- Equations derived from mass and energy balances

Modeling : evaporator (1)

■ Working fluid mass balance:

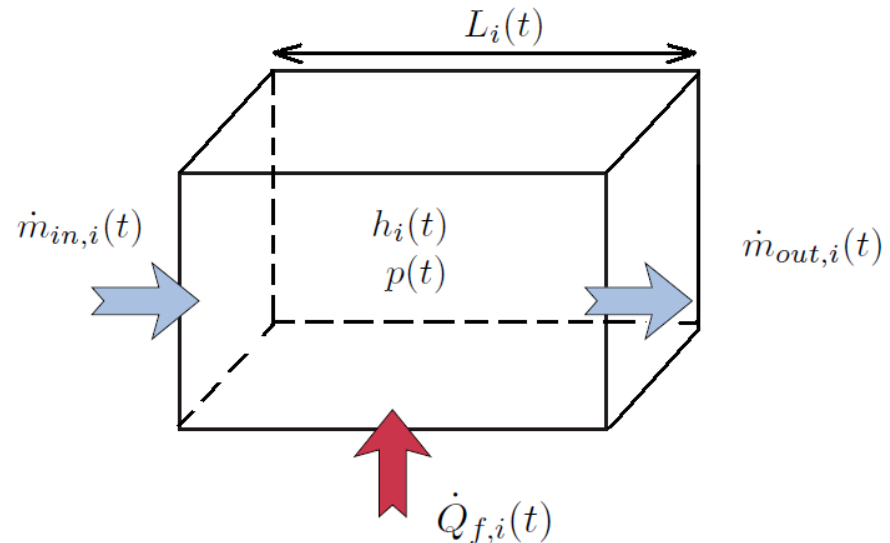
$$\frac{dm_i(t)}{dt} = \dot{m}_{in,i}(t) - \dot{m}_{out,i}(t)$$

Mass expression:

$$m_i(t) = \rho_i(p(t), h_i(t)) V_f L_i(t)$$

yields to:

$$\left(\frac{\partial \rho}{\partial p} \bigg|_{h_i} \frac{dp}{dt} + \frac{\partial \rho}{\partial h_i} \bigg|_p \frac{dh_i}{dt} \right) V_f L_i(t) + \rho_i(p(t), h_i(t)) V_f \frac{dL_i}{dt} = \dot{m}_{in,i}(t) - \dot{m}_{out,i}(t)$$



Modeling : evaporator (2)

■ Working fluid energy balance:

$$\rho_i(p(t), h_i(t)) V_f L_i(t) \frac{dh_i}{dt} = \dot{m}_{in,i}(t) \left(h_{in,i}(t) - h_i(t) \right) - \dot{m}_{out,i}(t) \left(h_{out,i}(t) - h_i(t) \right) + \dot{Q}_{f,i}(t) + \frac{dp}{dt} V_f L_i(t)$$

where

$$\dot{Q}_{f,i}(t) = V_f \alpha_f \left(T_{w,i}(t) - T_{f,i}(p(t), h_i(t)) \right) L_i(t)$$

■ Wall energy balance:

$$m_w c_{p_w} L_i(t) \frac{dT_{w,i}}{dt} = \dot{Q}_{exh,i}(t) - \dot{Q}_{f,i}(t)$$

$$\dot{Q}_{exh,i}(t) = \dot{m}_{exh}(t) c_{p_{exh}} \left(1 - \exp\left(-\frac{\alpha_{exh} S_{exh}}{\dot{m}_{exh}(t) c_{p_{exh}}}\right) \right) (T_{exh}(t) - T_w(t)) L_i(t)$$

Modeling: evaporator – expander DAE

■ Differential algebraic equations (DAE) system:

- evaporator balances with interface equations
- expander static model

$$\dot{m}_{exp} = N_{exp} \rho \eta_{exp}$$

- result in a DAE with 7 dynamic states

■ Fast / slow modes analysis:

- slow (wall inertia) T_{w1}, T_{w2}, T_{w3}
- middle (liquid inertia) L_1
- fast (pressure, vapor and two-phase) p, L_2, h_{out}

Model reduction

■ Fast / slow modes separation:

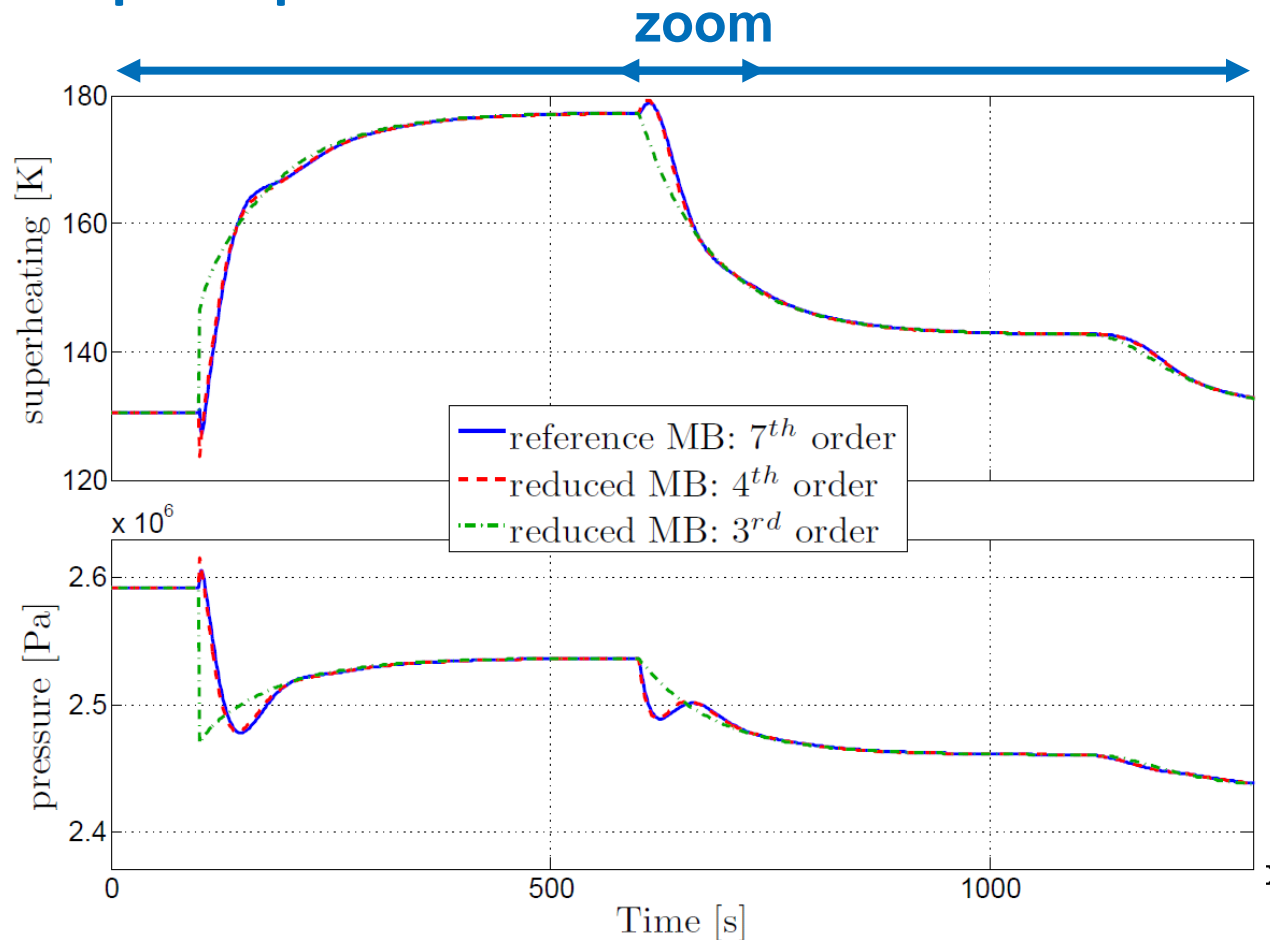
- fast dynamic states approximated by static variables
- result in 3rd or 4th order model

■ 3rd order model equations:

$$\left\{ \begin{array}{l} m_w c_w \frac{dT_{w,i}}{dt} = \dot{Q}_{exh,i} - \dot{Q}_{f,i} , \quad i = 1, 2, 3 \\ 0 = -0.5 V_f \rho_1 L_1 \dot{h}_{in} + \dot{m}_{in,1} (h_{in,1} - h_l) + \dot{Q}_{f,1} L_1 \\ 0 = \dot{m}_{in,1} (h_l - h_v) + \dot{Q}_{f,2} L_2 \\ 0 = \dot{m}_{in,1} (h_v - h_{out,3}) + \dot{Q}_{f,3} (1 - L_1 - L_2) \end{array} \right.$$

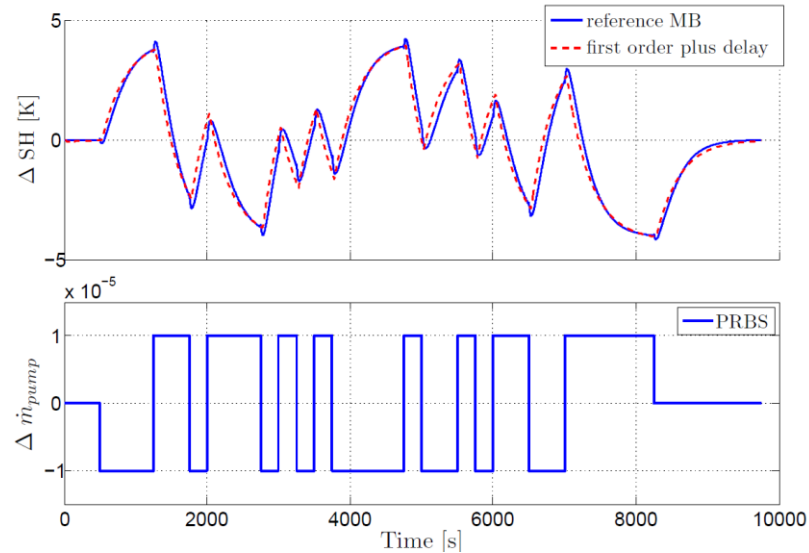
Model reduction: validation

■ Open-loop comparison:



Control: decentralized PI

■ Identification



■ Limits of decentralized PI control

“Hot” operating point			“Cold” operating point		
y^0	SH	200 K	y^0	SH	200 K
	p	2.5 MPa		p	2.5 MPa
u_2^0	\dot{m}_{pump}	0.005855 kg/s	u_1^0	\dot{m}_{pump}	0.0010156 kg/s
	N_{exp}	712.5 rpm		N_{exp}	134.3 rpm
d_2^0	T_{exh}	600°C	d_1^0	T_{exh}	400°C
	\dot{m}_{exh}	0.05 kg/s		\dot{m}_{exh}	0.02 kg/s
	T_{in}	70°C		T_{in}	30°C

TABLE I
DEFINITION OF “HOT” AND “COLD” OPERATING POINTS

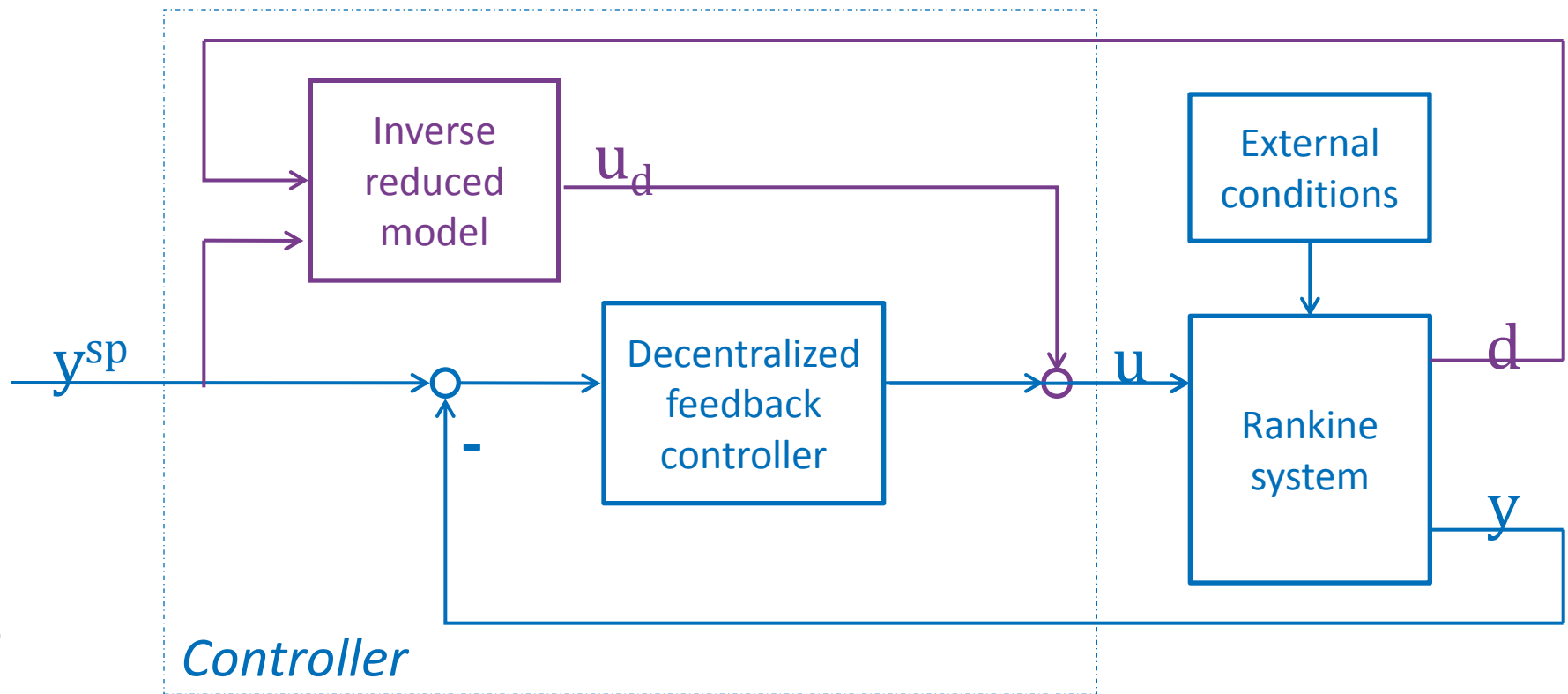
	K_p	τ_p	T_p
“Hot” operating point	$-1.36e^5$ K/kg.s	60.1 s	8.4 s
“Cold” operating point	$-4.28e^5$ K/kg.s	345 s	30 s

TABLE II
IDENTIFIED MODEL PARAMETERS AROUND THE TWO OPERATING POINTS

Control: nonlinear inversion-based

■ Decentralized feedback controller

- with inverse of reduced model in the feedforward path:



Control: nonlinear inversion equations

■ Dynamic (wall):

$$\frac{dT_{w,i}}{dt} = \frac{\dot{Q}_{exh,i} - \dot{Q}_{f,i}}{m_w c_w}, \quad i = 1, 2, 3$$

■ Pump mass flow:

$$u_{1,d} = \frac{(T_3 - T_{w3}) S_R \alpha_3}{h_v - h_{out} + \frac{(T_3 - T_{w3})(h_l - h_{in})\alpha_3}{\alpha_1(T_{w1} - T_1)} + \frac{(T_3 - T_{w3})(h_v - h_l)\alpha_3}{\alpha_2(T_{w2} - T_2)}}$$

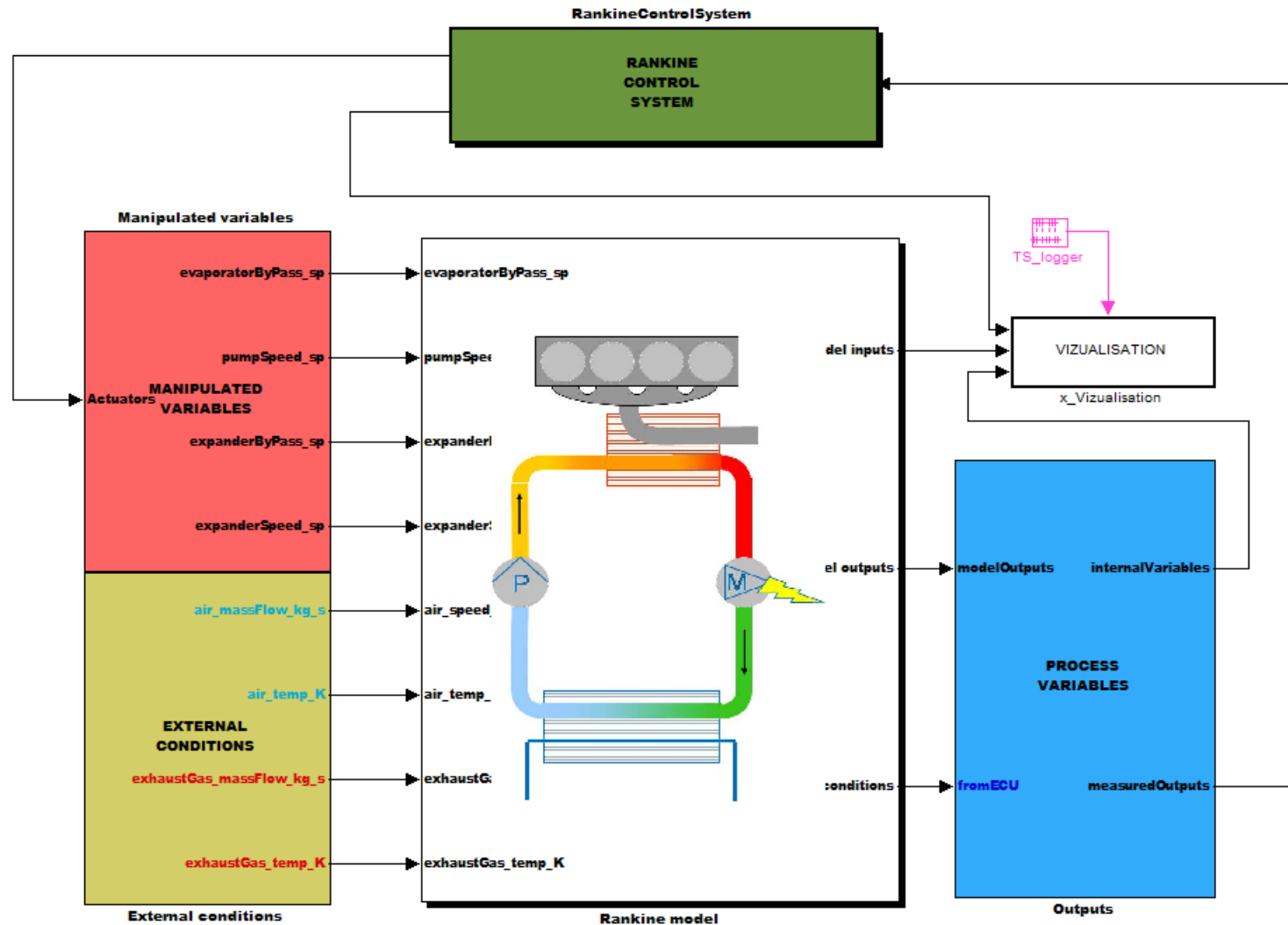
■ Expander speed:

$$u_{2,d} = \frac{u_{1,d}}{\rho_{out,3} \eta_{exp} V_{exp}}$$

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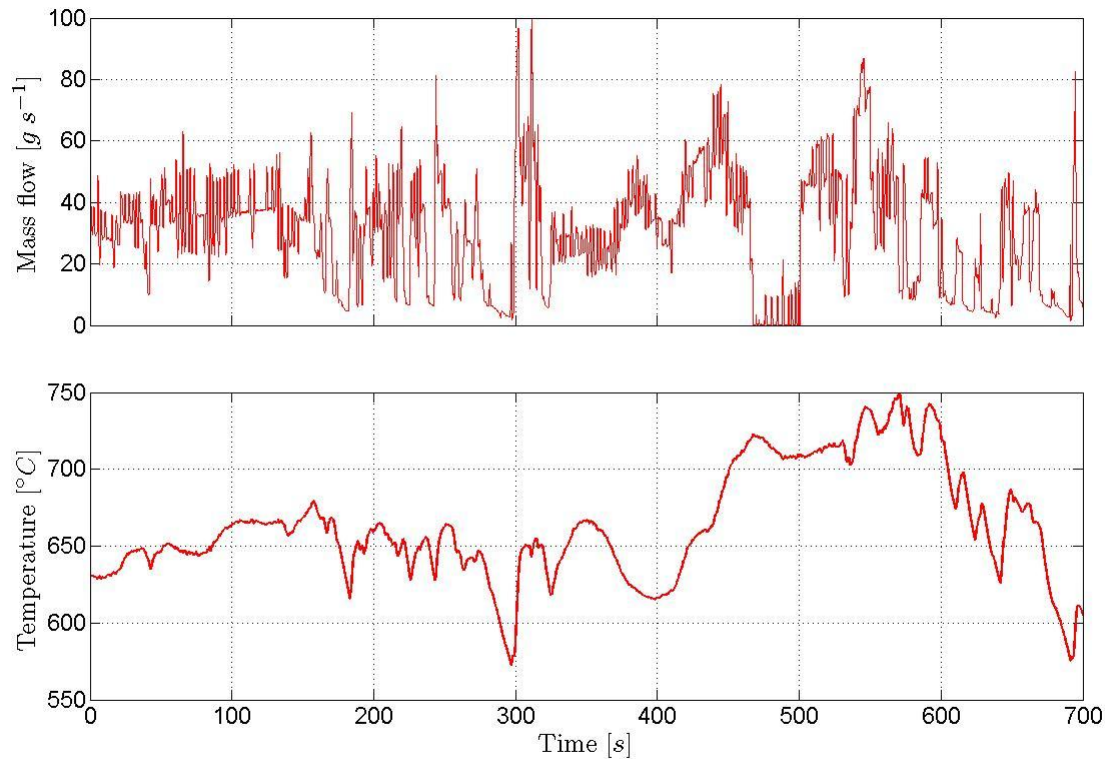


Control: co-simulation



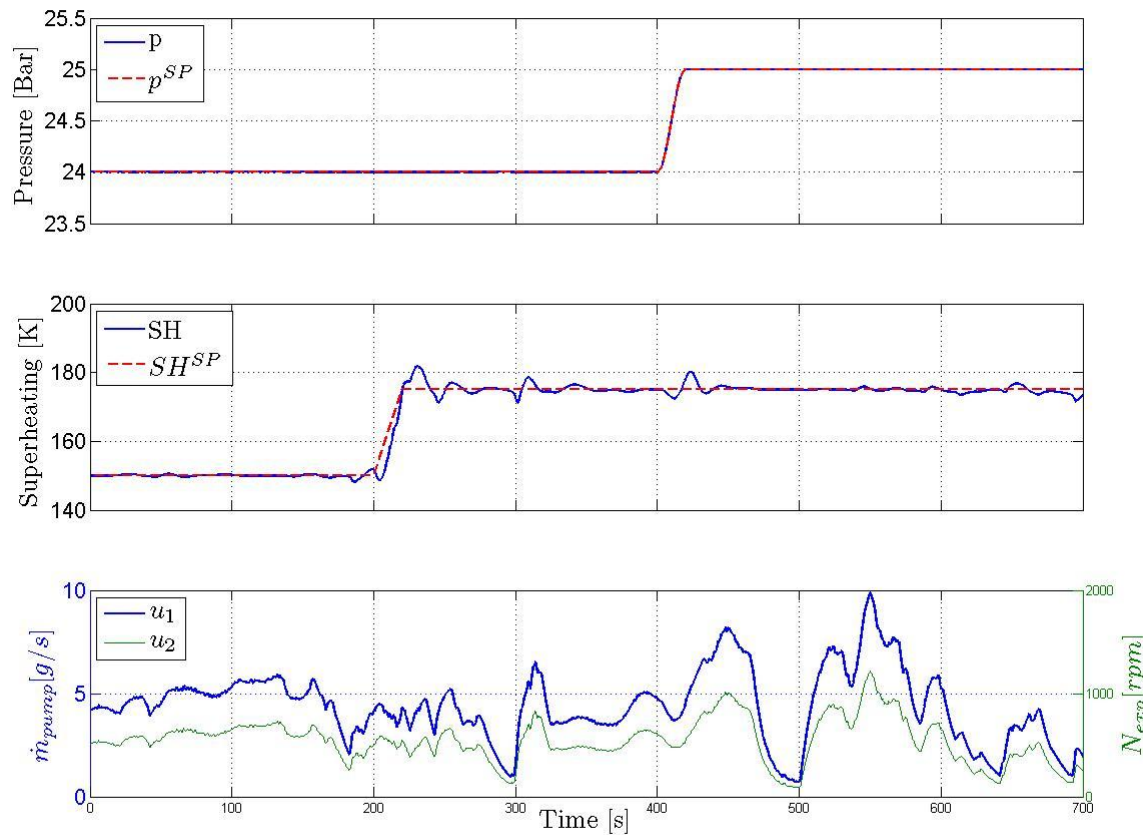
Simulation results with real data from a Renault Scenic with a 2L-F4RT engine

■ Exhaust gas conditions:



Simulation results with real data from a Renault Scenic with a 2L-F4RT engine

■ Pressure and temperature control:



Perspectives

- **Pursuing validation**
 - in simulation on broader range of driving profiles
 - on the experimental test bench

- **Improving the control strategy**
 - expander speed saturations
 - optimal set points