DYNAMIC MODELS FOR A HEAT-LED ORGANIC RANKINE CYCLE

- turbine and alternator models

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ASME ORC 2013 NL Conference -
Rotterdam, October 7-8, 2013

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Content I

Introduction

Motivation for the work

- 150 biomass based power plants all over Europe, 140 are heat-led
- many facilities are suffering from economic difficulties
  (fuel prices $\uparrow$, feed-in tariffs $\downarrow$, maintenance $\uparrow$)
- heat-led systems have limited degrees of freedom
- previous analyses have shown problems of control system regarding varying loads
- therefore, necessity for control optimisation

System overview I

Table : Design data of case study plant

-	Value	Unit
Thermal input	6356	kW
Temperature of source	300/240	$^\circ$C
Thermal output	5300	kW
Temperature of sink	80/60	$^\circ$C
Electric output	950	kVA
Mass flow	20	kg/s
Gross design efficiency	16.38	%

System overview II

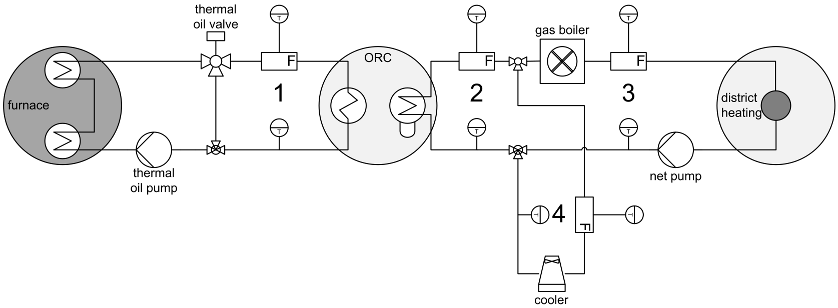

Figure : furnace, transfer system, ORC, cooling unit, district heating

- Fluids: thermal oil (T66), Octamethyltrisiloxane (MDM), water
- Heating curve is based on amb. temperature /ORC control is heat-led

Cycle layout

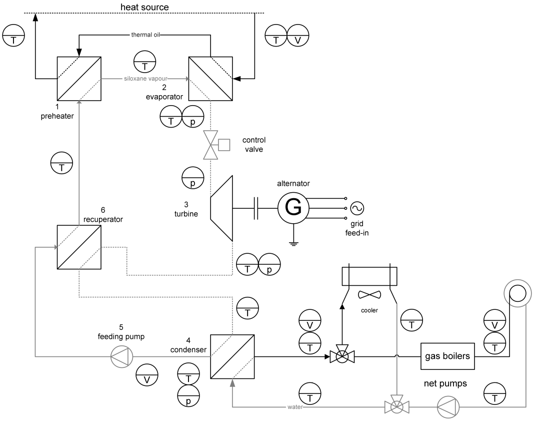

Figure : cycle including all relevant sensors

S7 / M-Bus → OPC gateway → OPC server/client → data base

Simulation scope I

Overall scope

- find compensation strategies for the controller
- find a suitable winter, summer and intermediate operation strategy

Partial scope

- dynamic generator model: improve validation, simulate start-stop procedures, alternator oscillation

$\rightarrow$ a very accurate and robust turbine, alternator and drive shaft model with a wide load range is required.

Simulation method I

Method

- physical turbine model with empirical correlation for stage efficiency
- empirical alternator model based on design data of manufacturer
- physical drive train model
- validation through: measured electric output, $\sum \dot{m}$, frequency
- validation data for three different load ranges: high, medium, low

Tools

- steady state calculations, correlation fitting $\rightarrow$ MATLAB
- dynamic simulation $\rightarrow$ Dymola 2013 / modelica
- $ExternalMedia$ Lib, $FluidPropMedium$ Package, REFPROP [3]
- $ThermoPower$ Lib

Turbine I

Turbine

- type: impulse, axial, single stage, super-sonic
- nozzle: 24 x De Laval, $\alpha = 19°$, $\eta_{noz} = 92\%$
- turbine speed 3000 RPM $\rightarrow \bar{u} \sim 160$ m/s
- isentropic outflow velocity: 300 m/s
- lubrication system: separate
- frequency control: alternator/grid
- flow control: full admission, control (by-pass) valve for start-up
- mean rotor diameter 1 m

Turbine II

Figure : turbine, diffuser and aux. compounds

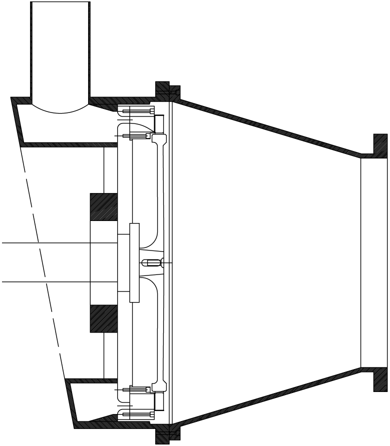

Figure : cross section of turbine and diffuser

Turbine III

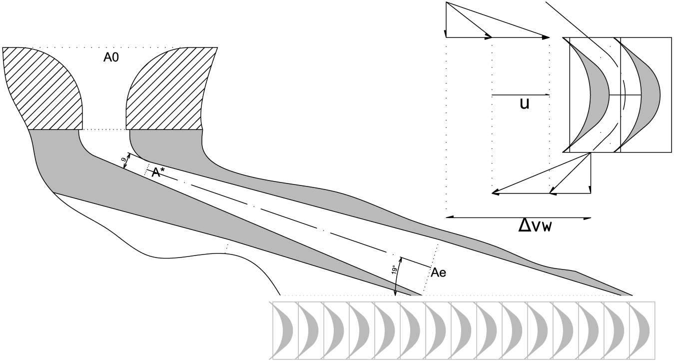

Figure : wrapped section of turbine nozzle and rotor

Turbine IV

Based on Stodola's Law of Cones [4] [5], Cooke [2] [1] proposed a model for multi-stage steam turbines:

$$\dot{m}_{turb} = k_T \times \sqrt{\rho_{in} \times p_{in}} \times \sqrt{1 - \left(\frac{p_{out}}{p_{in}}\right)^{\frac{\kappa+1}{\kappa}}} \quad (1)$$

The exponents of the pressure ratio r_s is usually set to 2 for steam turbines. The question is: what happens if one uses the κ of the organic fluid?

$\rightarrow$ two model variants are tested: $\kappa = constant$, $\kappa = f\left(p, T\right)$

Turbine V

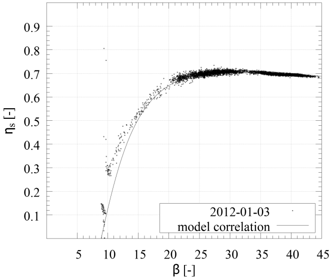

Figure : measured isentropic efficiency of turbine vs. correlation

$$\eta_s = a \times atan\left(b \times \beta^2 + \frac{c}{\beta}\right) + d \times \beta + f \quad (2)$$

Alternator I

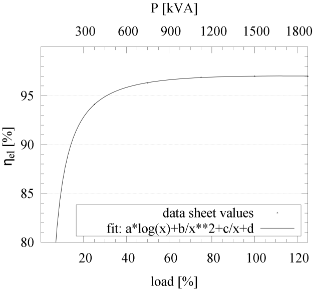

Figure : electric efficiency of alternator

- synchronous engine (50$\pm$0.5 Hz), rated power 1500 kVA
- water cooled with separate cooling unit

Drive train I

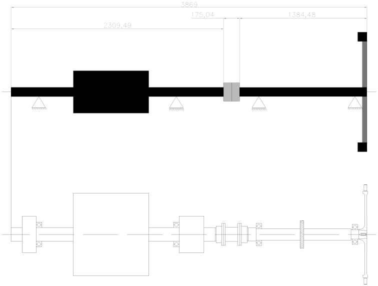

Figure : entire drive train, including bearings and couplings

- the friction has constant, linear and quadratic compounds
- simplification: bearing friction is a quadratic correlation $f(\omega)$
- tensor is calculated via measured shut down

Drive train II

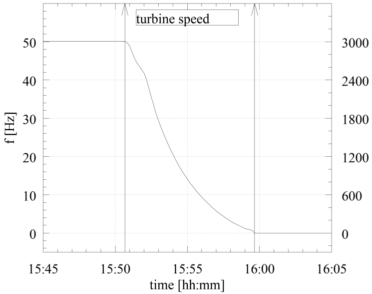

Figure : turbine shut-down vs. time

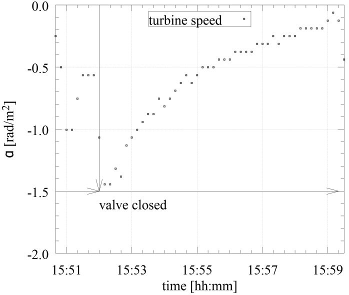

Figure : angular acceleration of rotor

- friction torque max. 63 Nm
- tensor 97.9 kg m^2

Results I

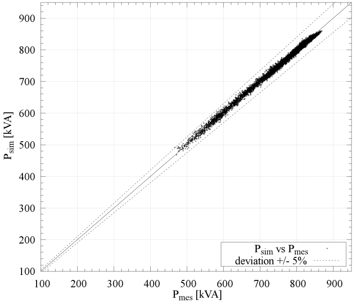

Figure : simulated vs. measured (12-01-01)

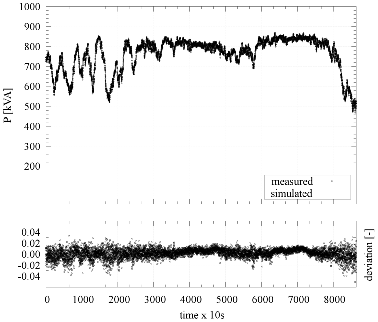

Figure : simulated and deviation vs. time (12-01-01)

Results II

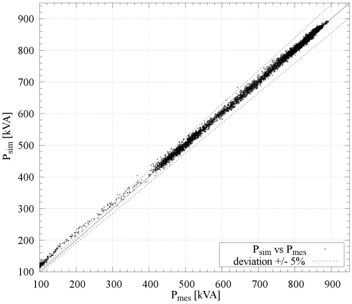

Figure : simulated vs. measured (12-01-02)

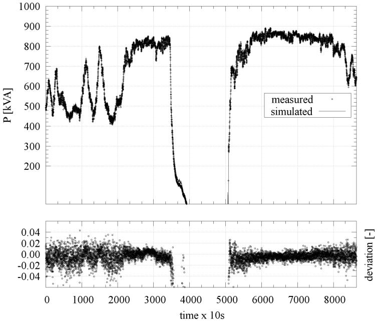

Figure : simulated and deviation vs. time (12-01-02)

Results III

model	date	measured $\bar{P_{el}}$	W	simulated $\bar{P_{el}}$	W	deviation
-	[YYMMDD]	[kW]	[kWh]	[kW]	[kWh]	[-]
	12-01-01	760.28	18247	756.62	18159	-0.4821%
$\kappa = \bar{\kappa}$	12-01-02	592.47	14219	589.21	14141	-0.5491%
	12-01-03	338.38	8121	337.09	8090	-0.3823%
	12-01-01	760.28	18247	756.62	18159	-0.4824%
$\kappa(p, T)$	12-01-02	592.47	14219	589.17	18158	-0.5567%
	12-01-03	338.38	8121	337.06	8094	-0.3900%

Table : comparison of model and measured values for three data sets (1 day / 10 second steps)

Results IV

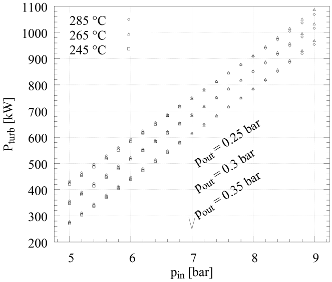

Figure : parameter study of turbine model

→ turbine provides maximum with low super-heating

Conclusions I

Conclusions

- measured values of isentropic efficiency give a good prediction for higher pressure ratios ($\beta \geq 15$)
- both model variants have a good prediction quality, the $\kappa = const$-model needs less CPU
- k_t, $\tau_{fric}(\omega)$, polytropic exp, alternator: 4 parameters, η_s 5 parameters
- low super-heating is favourable, possible optimisation of the evaporator level
- for load changes under 10% both models have less than $\pm 2\%$ dispersion, maximum of $\pm 5\%$
- for 400 to 900 $\mathrm{kW_{el}}$ models have less than $\pm 2\%$ dispersion
- the cumulated work error for one day of operation is less than -0.5%

Conclusions II

Next steps

- testing the models with oscillating rotation frequency with data of higher precision

Acknowledgements

Thank you for your attention - Bedankt voor uw aandacht!

This work has been founded by the European Union within the framework of EraSME and INNOnet.

Thanks to the modelica people

[1] D.H. Cooke. Modeling of off-design multistage turbine pressures by stodola's ellipse. *Proceeding PEPSE User Group Meeting*, pages 205–234, 1983.

[2] D.H. Cooke. On prediction of off-design multistage turbine pressures by stodola?s ellipse. *Trans. ASME, J. Eng. Gas Turbines Power*, 107:596–606, 1985.

[3] E.W. Lemmon, M.L. Huber, and M.O. McLinden. Nist standard reference database 23: Reference fluid thermodynamic and transport properties refprop. *NIST - Standard Reference Data Program*, Version 9.0, 2010.

[4] A. Stodola. *Steam Turbines - With an Appendix on Gas Turbines and the Future of Heat Engines*. Van Nostrand, Princeton, New Jersey, 1906.

[5] Walter Traupel. *Thermische Turbomaschinen - Geänderte Betriebsbedingungen, Regelung, Mechanische Probleme, Temperaturprobleme*. Springer DE, Berlin, 2001.