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ORC POWER SYSTEMS

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Università degli Studi di Padova

National Technical University of Athens

Experimental performance evaluation of a multi-diaphragm pump of a micro-ORC system

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Milan, 15/09/2017

OBJECT

Multi-diaphragm positive displacement pump installed in a 4 kW ORC system for marine applications (small-scale marine ORC)

GOALS

- Experimental characterization of the multi-diaphragm pump
- Validation and improvement of the semi-empirical model of the pump developed by D’Amico et al. [1]

[1] F. D’Amico, P. Pallis, A.D. Leontaritis, S. Karellas, N.M. Kakalis, S. Rech, A. Lazzaretto. Semi-empirical model of a multi-diaphragm pump in an Organic Rankine Cycle test rig. 4th International Conference on Contemporary Problems of Thermal Engineering, September 14-16. Katowice, Poland, 2016.

OUTLINE

EXPERIMENTAL INVESTIGATION:

- The ORC test rig
- Pump performance (global and relative volumetric efficiency)
- Cavitation issues

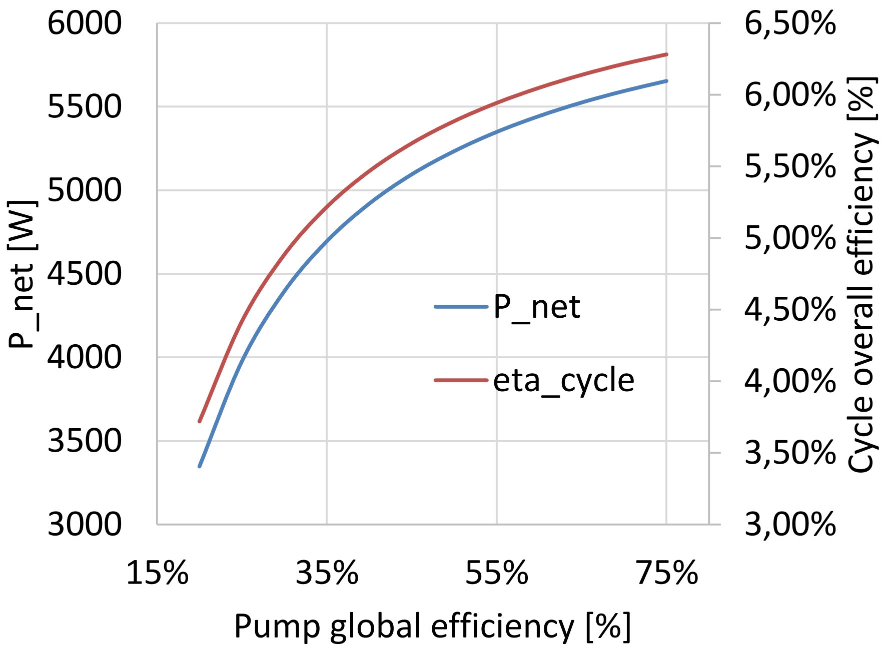
MODELLING:

- Recalibration of the pump model (addition of the electric motor efficiency)
- Prediction of the operation in different conditions from those used for the calibration procedure

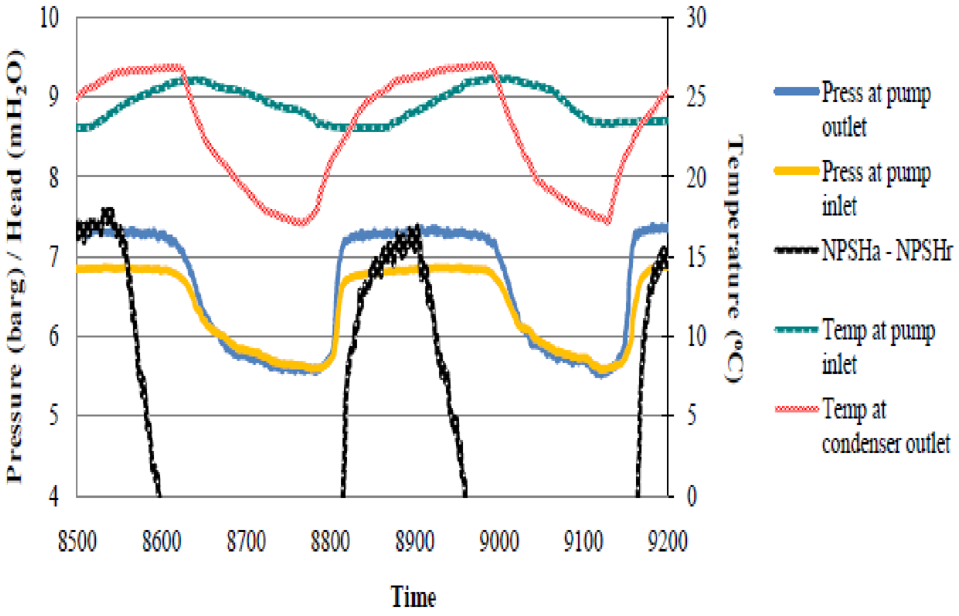
THE KEY IMPORTANCE OF THE PUMP IN SMALL-SCALE ORCs

Literature background: Few and low efficiency values available:

Suited pumps for small-scale ORCs	Typical efficiency values
Diaphragm (single/multi)	20 – 25%
Sliding vane	15 – 20%
Plunger	40 – 46 %



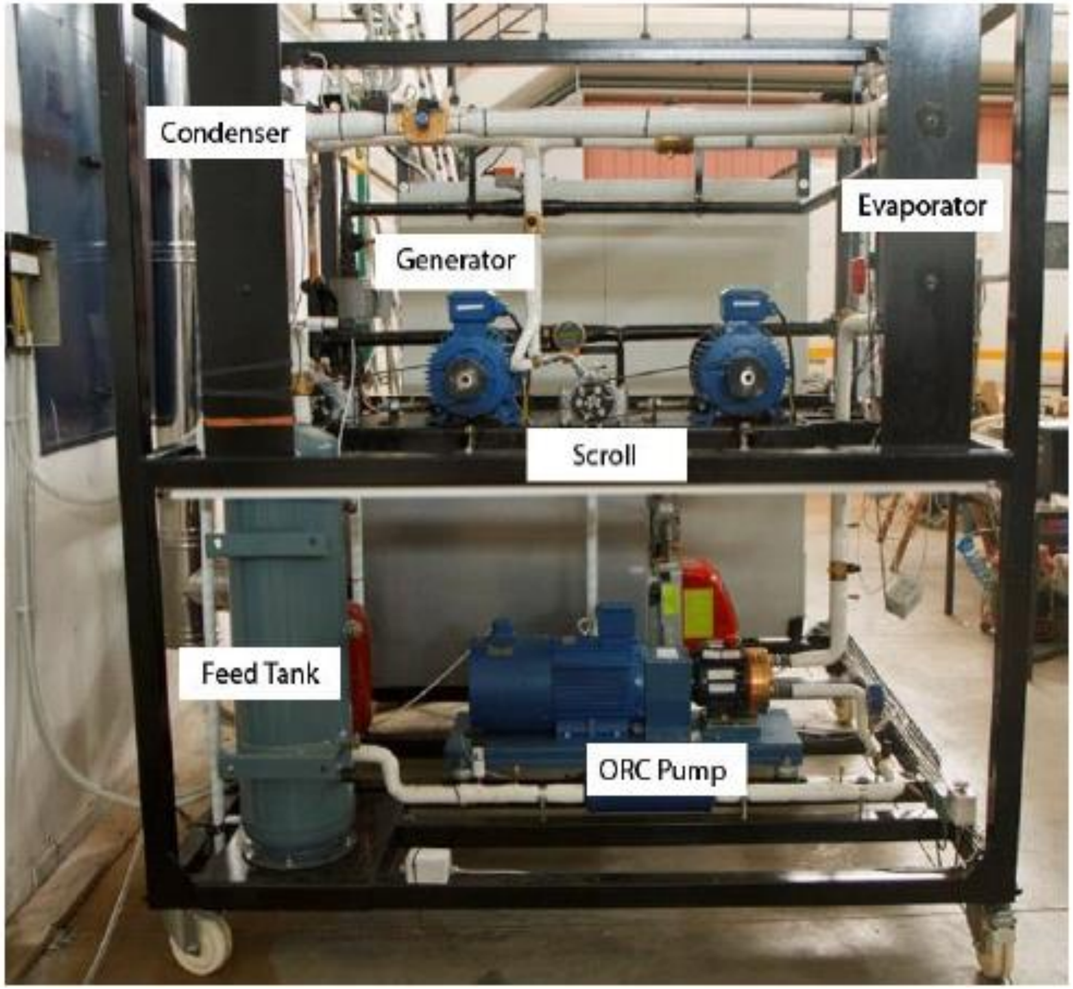
Cavitation occurrence (operational issues)



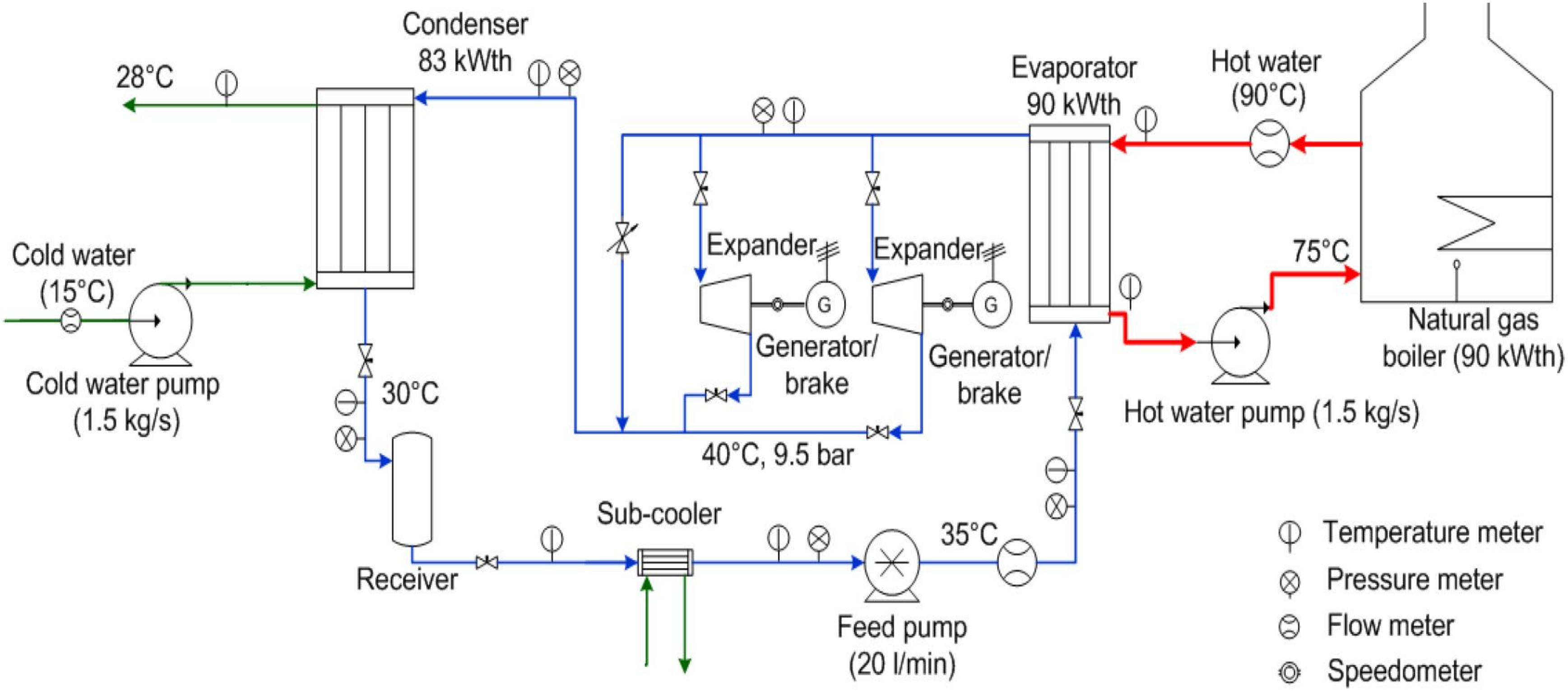
Impact on the operation of the total system:

- Mass flow rate and pressure drop
- System instability

THE ORC TEST RIG

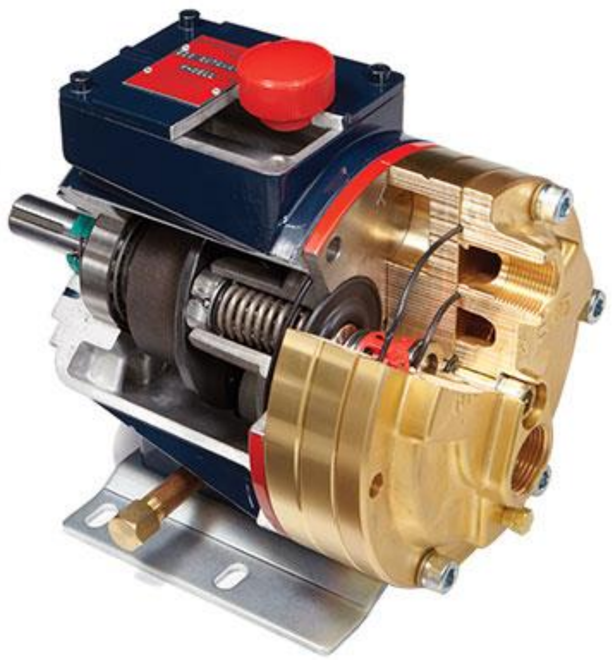


THE ORC TEST RIG



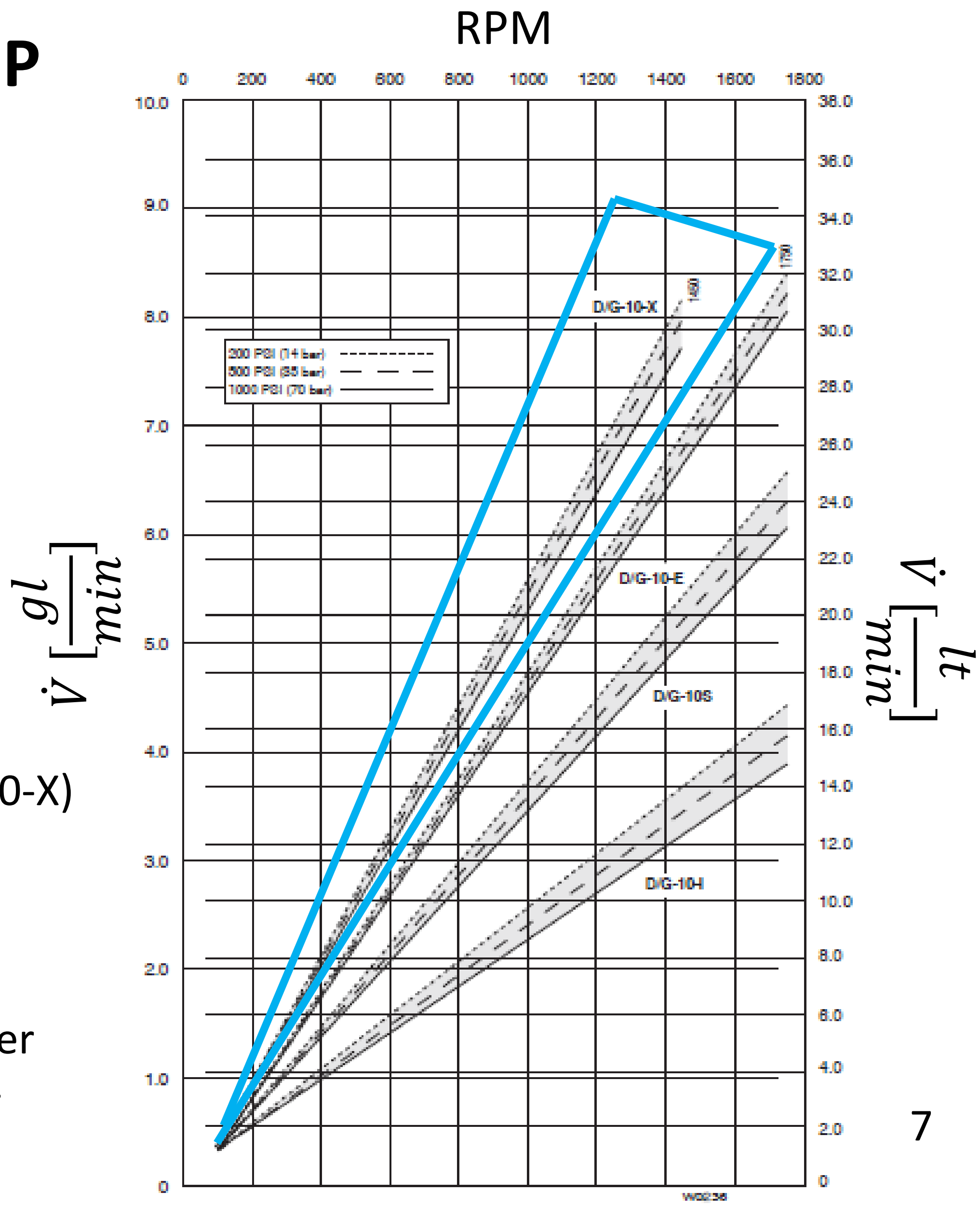
- ➢ Working fluid: HFC – R134a
- ➢ Small-scale power plant based on a standard ORC with 2 scroll expanders in parallel
- ➢ Presence of a sub-cooler in the pump suction line to avoid cavitation

THE PUMP



- Multi-diaphragm pump (Hydra Cell G10-X)
  Metallic head (brass);
  Diaphragm material: Buna-N
- TRANSMISSION CHAIN:

Electric motor – drive shaft – tapered roller bearing – fixed angle cam/wobble plate – hydraulic cells – diaphragms



EXPERIMENTAL CAMPAIGN:

1. PUMP PERFORMANCE

The experimental data were subdivided into **two categories** of operating conditions:

A. **Constant** pump rotational speed (**RPM**) and **variable** pressure difference (**Δp**):

- 1st **approach**: constant p_{max} and variable p_{inlet}
- 2nd **approach**: variable p_{max} and constant p_{inlet}

Tested speeds [RPM]	400, 470, 500, 530, 700, 900
Inlet pressure [bar]	$7{,}5 < p_{inlet} < 13$
Outlet pressure [bar]	$17 < p_{max} < 25$
Pressure difference [bar]	$7 < \Delta p < 17$

B. **Constant** pressure difference (**Δp**) and **variable** speed (**RPM**)

Δp [bar]	11, 12, 15
Speed variation range [RPM]	370 < RPM < 1000

EXPERIMENTAL CAMPAIGN:

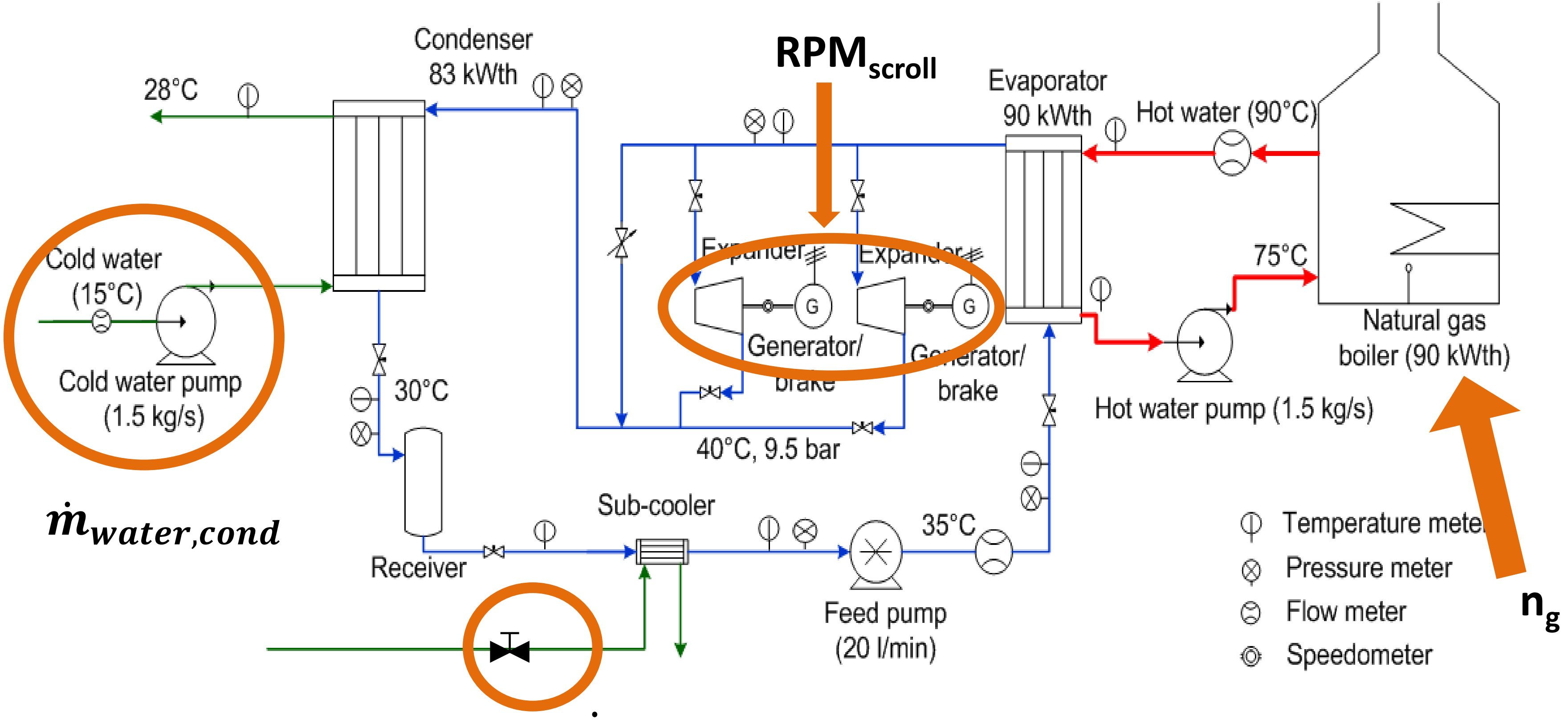
2. CAVITATION:

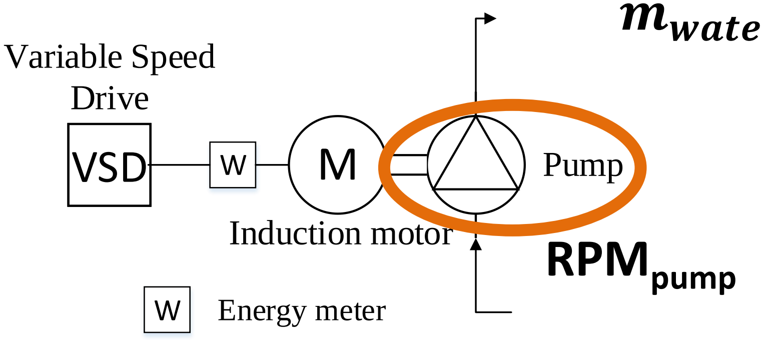
- evaluation of $NPSH_{available}$ for different working conditions

- 4 experimental tests: 2 at partial and 2 at full load (1 or 2 scroll expanders working)
- Enforcing cavitation + re-establishment of system stable operation

Tested speeds [RPM]	Inlet pressure [bar]	Outlet pressure [bar]
550	$7,3 < p_{inlet} < 8,3$	$9,7 < p_{max} < 25$
800	$8,5 < p_{inlet} < 9,5$	$11 < p_{max} < 22$
960	$7,8 < p_{inlet} < 8,8$	$11,9 < p_{max} < 23,8$

CONTROLLED QUANTITIES

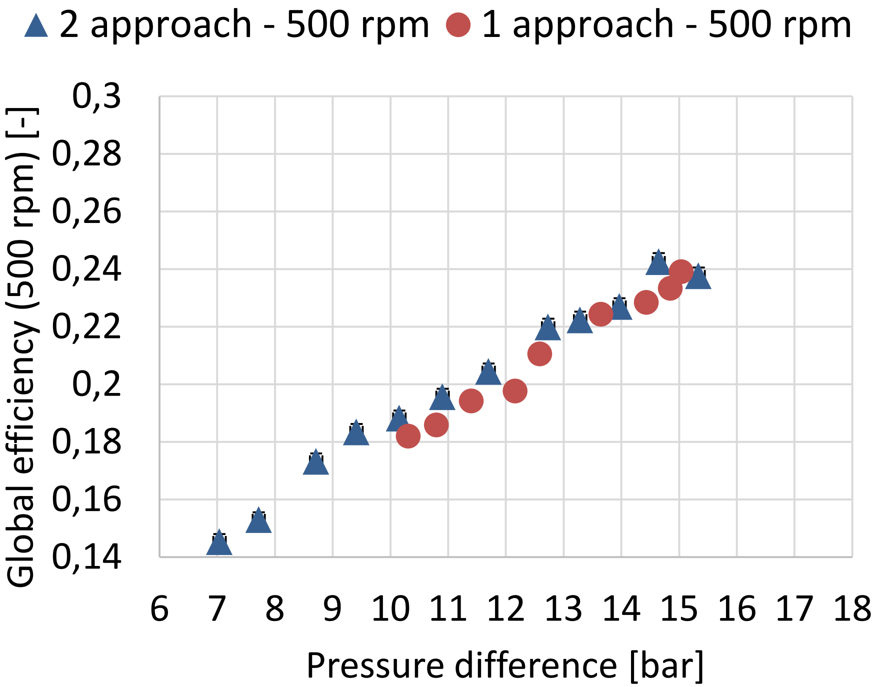


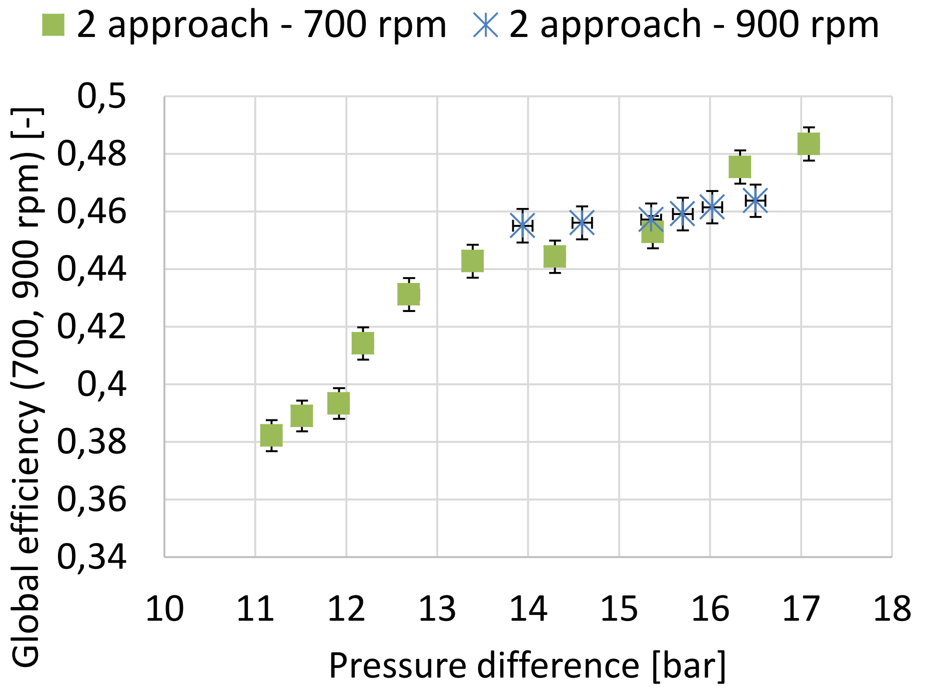


Scheme of the test rig used to measure the performance of the multi-diaphragm pump

CATEGORY A (variable *Δp* ; *constant RPM*)
GLOBAL EFFICIENCY

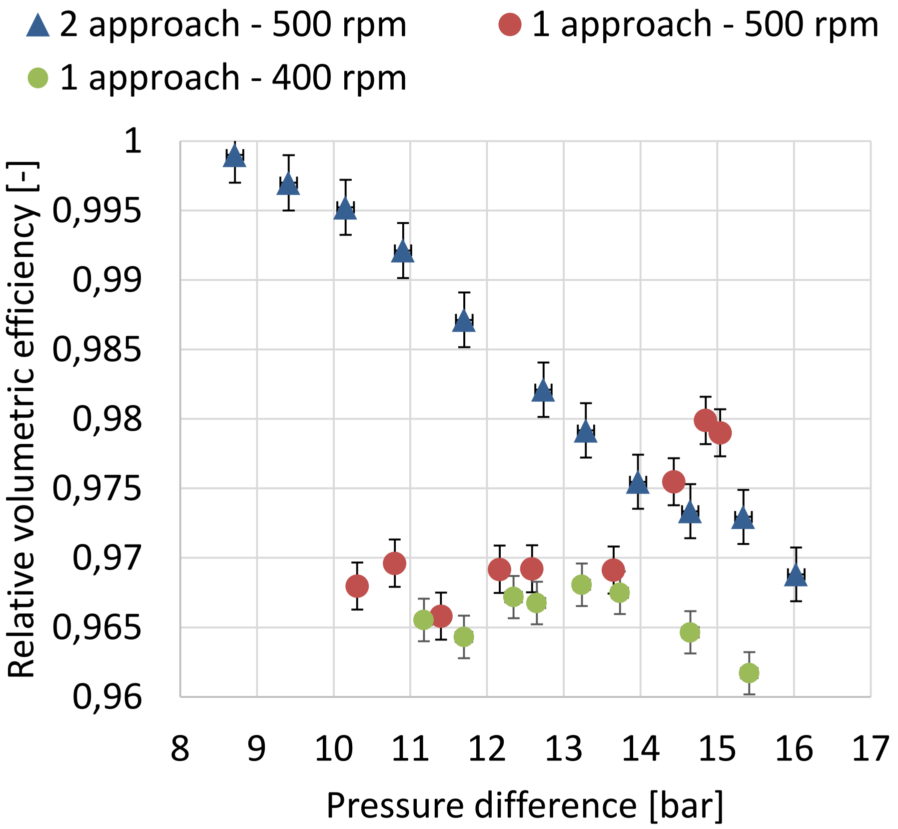
$$\longrightarrow \eta_{glob} = \frac{\dot{W}_{hyd}}{\dot{W}_{mot}} = \frac{\dot{V}\Delta p}{\dot{W}_{mot}}$$





- η_{glob}: increasing function of Δp in the range considered and does not depend on the applied approach
- Highest efficiencies: 45 ÷ 48% for n > 700 rpm
- η_{glob} increases with RPM reaching a maximum at intermediate speeds

CATEGORY A (variable Δp ; *constant RPM*)
RELATIVE VOLUMETRIC EFFICIENCY



$$\eta_{vol,rel} = \frac{\eta_{vol}}{\eta_{vol,manu}} = \frac{\dot{V}}{\dot{V}_{th}} \cdot \frac{\dot{V}_{th}}{\dot{V}_{manu}} = \frac{\dot{V}}{\dot{V}_{manu}}$$

$\dot{V}_{th}$= theoretical volume flow rate
$\dot{V}_{manu}$ = volume flow rate by manufacturer

- $\eta_{vol,rel}$ trend strongly depends on the applied approach :

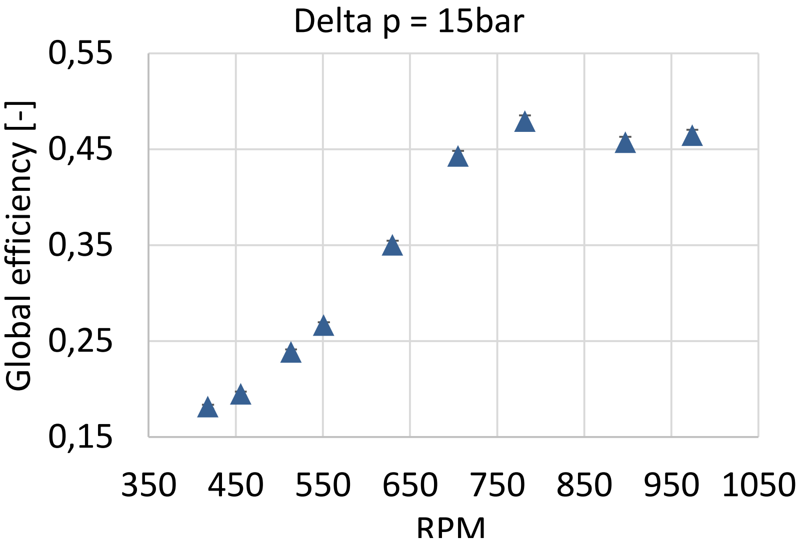
1st approach (●, ●): no clear trend identifiable;

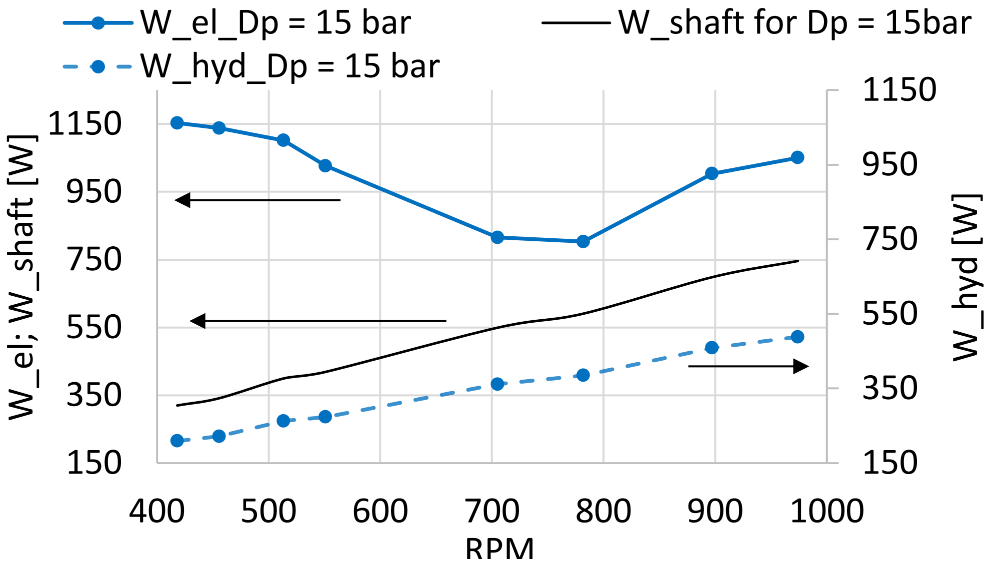
2nd approach (▲): expected trend

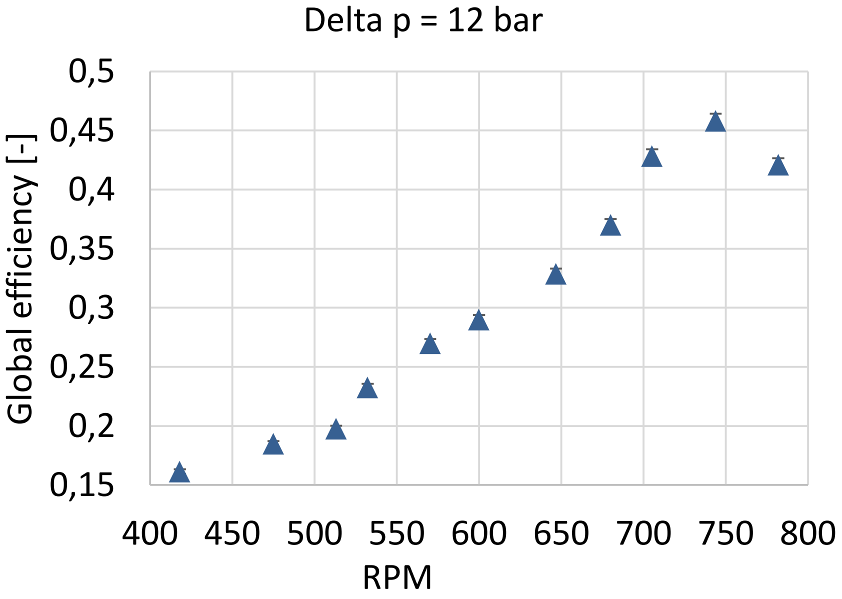
→ decrease of volumetric efficiency with the increase of Δp

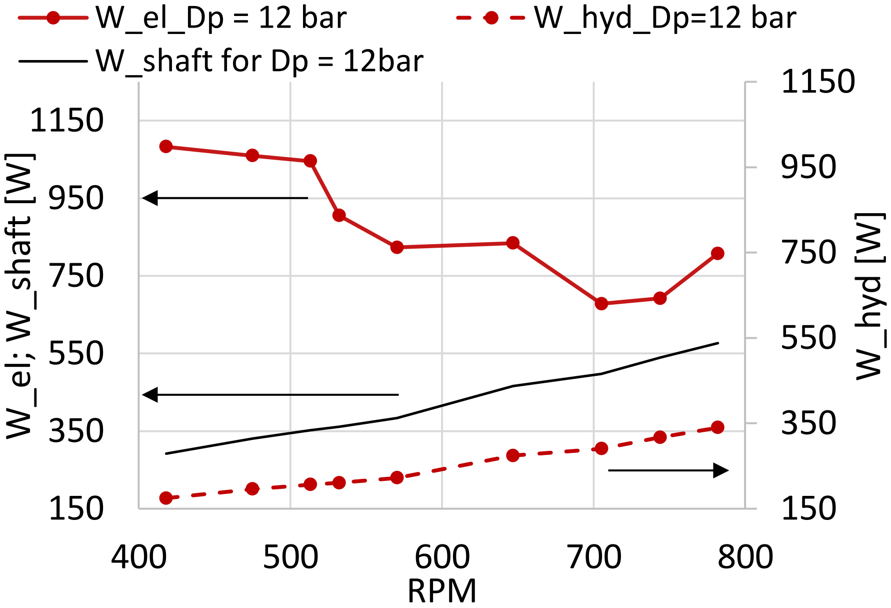
- The variation of p_{inlet} (1st approach) causes higher instabilities in the pump operation than the variation of p_{max} (2nd approach)

CATEGORY B (constant *Δp ; variable RPM*): GLOBAL EFFICIENCY VS POWER





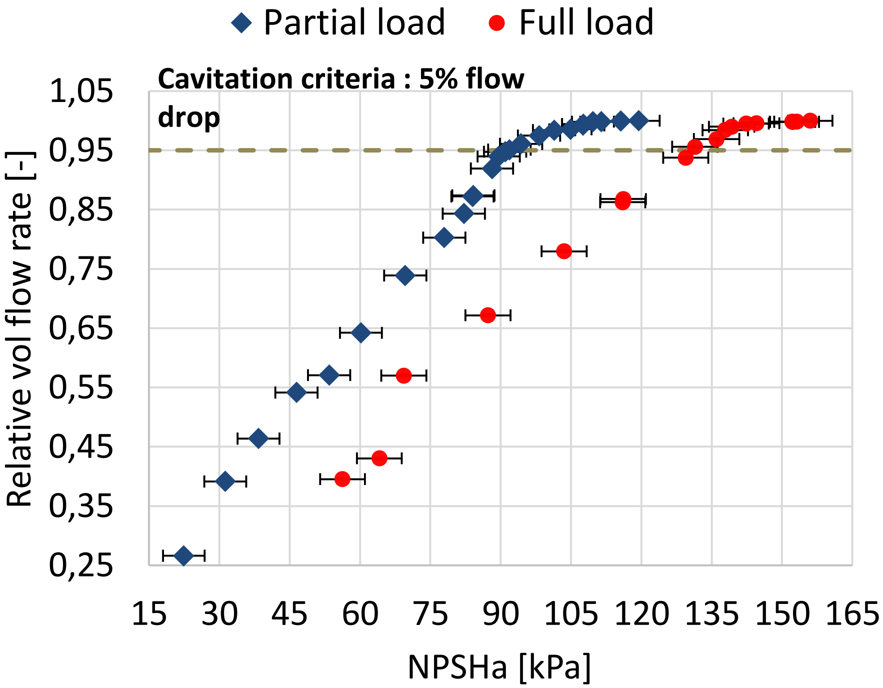


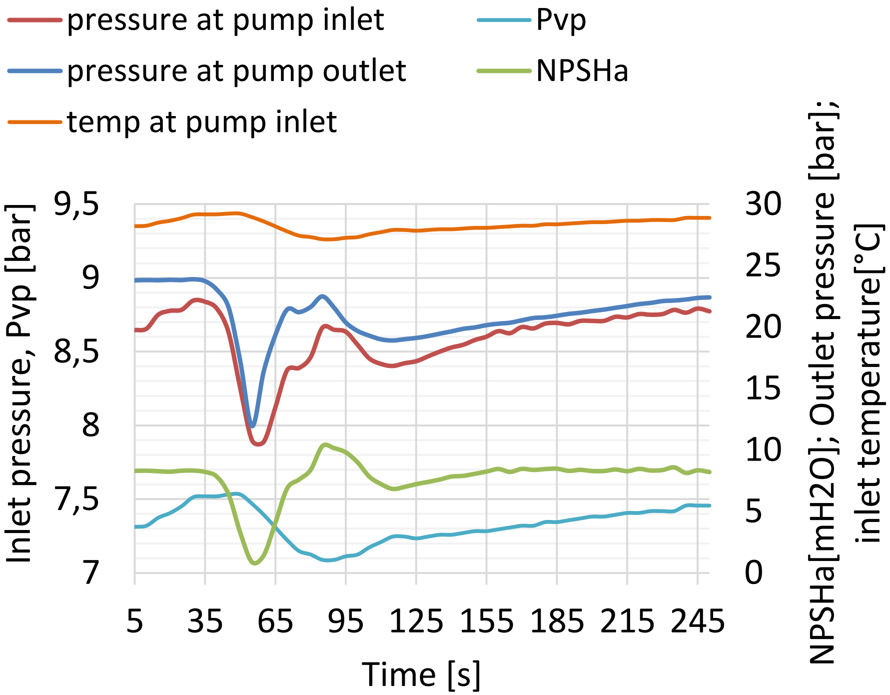


CAVITATION TESTS

$$Rel\ flow\ rate = \frac{\dot{V}_{real}}{\dot{V}_{nominal}}$$

$$NPSH_a = p_{in,pp} - H_a - Pvp$$





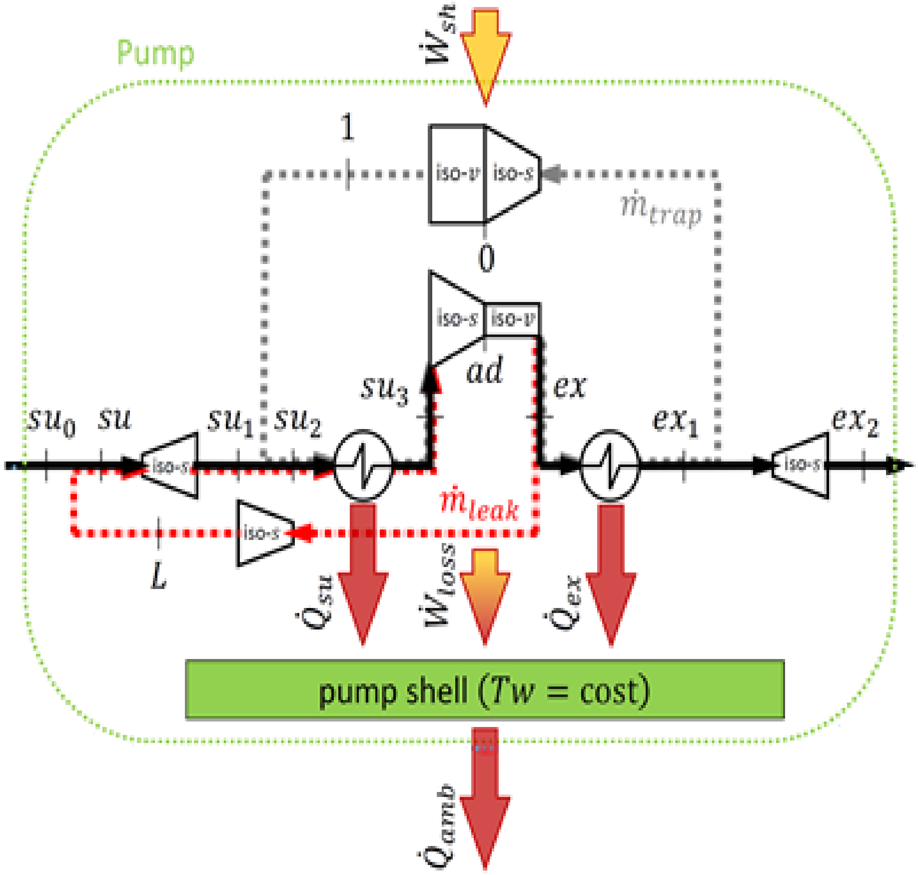
NPSH$_a$ decreased by:

- Decreasing the water flow rate (valve in the subcooling line)

Cavitation occurrence + system recovery

- ✓ Synchronized drop of NPSH and P
- ✓ Time lag in the decrease of T

SEMI-EMPIRICAL MODEL OF THE MULTI-DIAPHRAGM PUMP - D’AMICO ET AL. (2016)



- Model implemented in Engineering Equation Solver (EES)

- Pump operation represented by zero-dimensional thermodynamic processes:

$$\sum \dot{m} = 0; \quad \sum \dot{m}h + \dot{W} - \dot{Q} = 0;$$

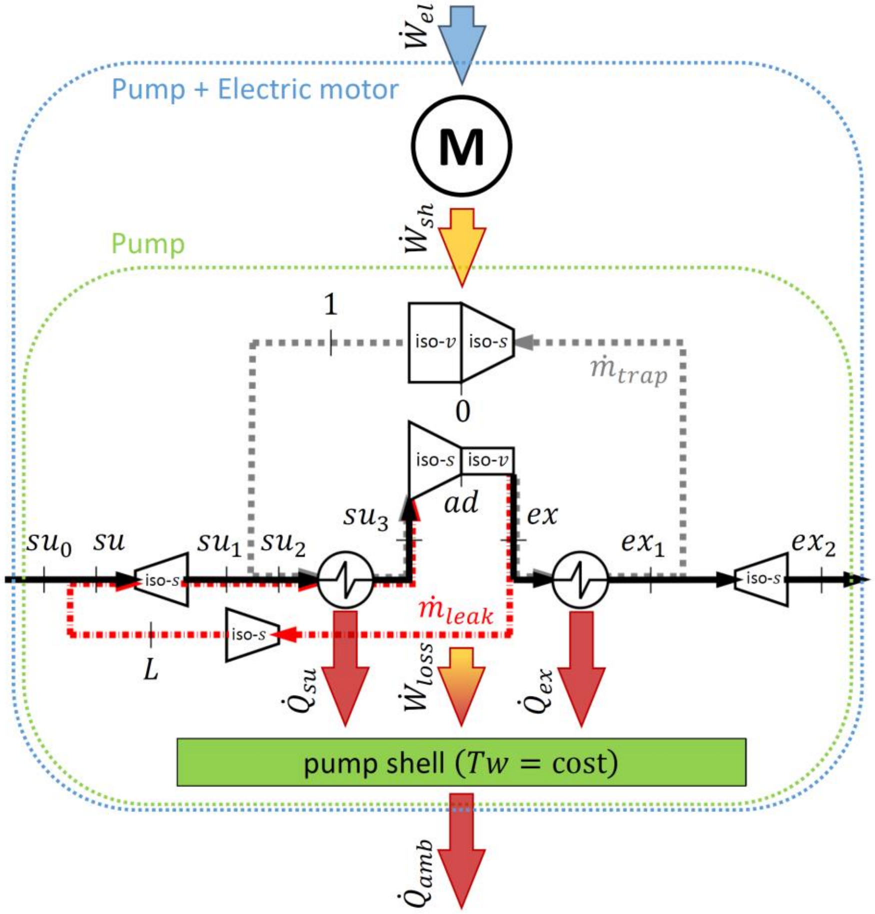
- Three mass fluxes considered:
  - $\dot{m}_{main}$ ←
  - $\dot{m}_{leak}$ ◂ – –
  - $\dot{m}_{trap}$ ◂ – –

- Lack of flow meter and energy meter

RECALIBRATION performed using experimental points in the whole operational range

21 Hz ≤ f ≤ 56 Hz

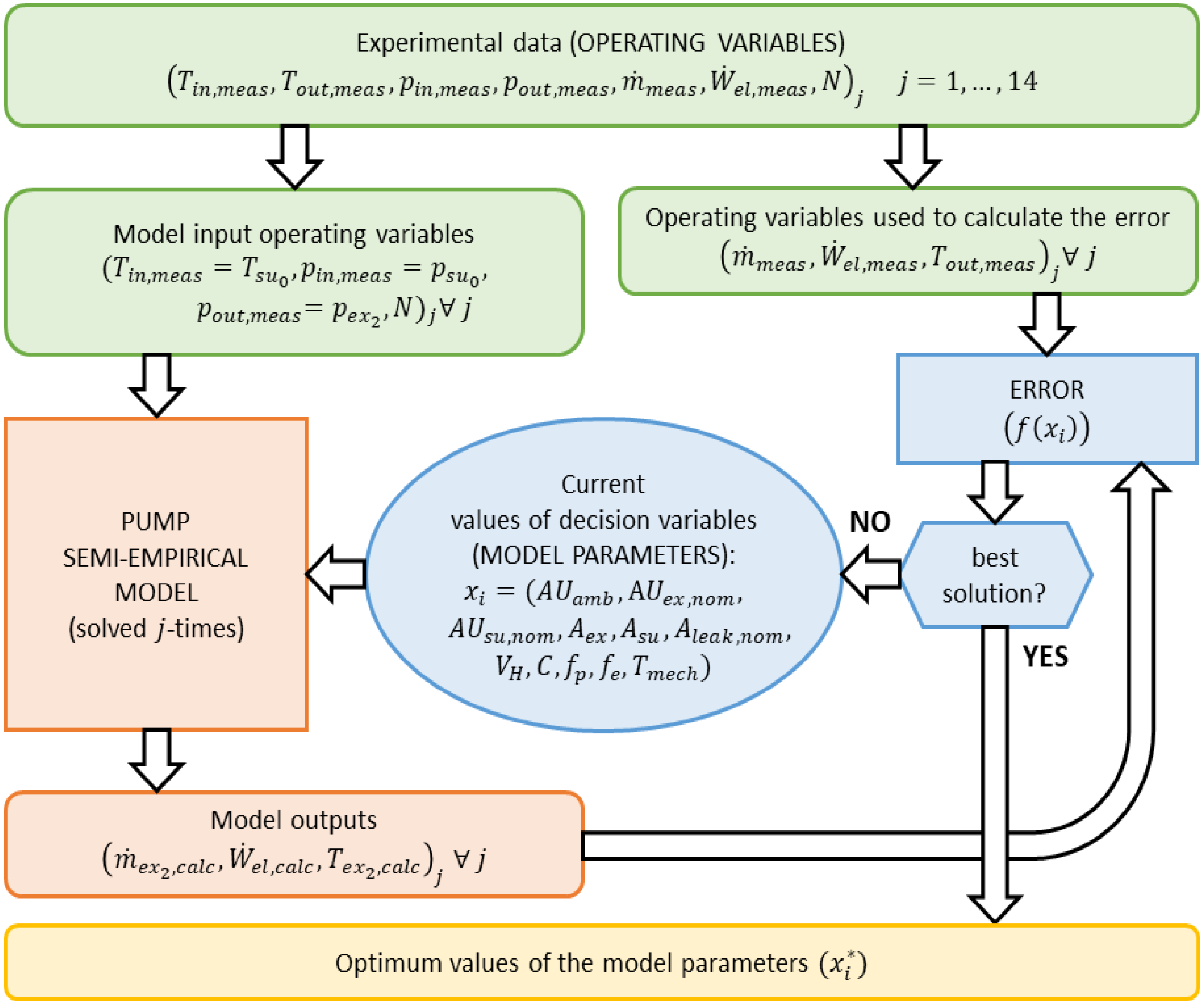
7 bar ≤ Δp ≤ 16 bar



- Model validated using points within the calibration range but not used for the calibration procedure
- Empirical equation to estimate the motor efficiency:

$$\eta_{mot} = \frac{W_{sh,pump}}{W_{el}} = f(N, Load)$$

CALIBRATION PROCEDURE



Experimental data (OPERATING VARIABLES)
$(T_{in,meas}, T_{out,meas}, p_{in,meas}, p_{out,meas}, \dot{m}_{meas}, \dot{W}_{el,meas}, N)_j \quad j = 1, \dots, 14$

Model input operating variables
$(T_{in,meas} = T_{su_0}, p_{in,meas} = p_{su_0},$
$p_{out,meas} = p_{ex_2}, N)_j \forall j$

Operating variables used to calculate the error
$(\dot{m}_{meas}, \dot{W}_{el,meas}, T_{out,meas})_j \forall j$

ERROR
$(f(x_i))$

PUMP
SEMI-EMPIRICAL
MODEL
(solved j-times)

Current
values of decision variables
(MODEL PARAMETERS):
$x_i = (AU_{amb}, AU_{ex,nom},$
$AU_{su,nom}, A_{ex}, A_{su}, A_{leak,nom},$
$V_H, C, f_p, f_e, T_{mech})$

NO

best
solution?

YES

Model outputs
$(\dot{m}_{ex_2,calc}, \dot{W}_{el,calc}, T_{ex_2,calc})_j \forall j$

Optimum values of the model parameters (x_i^*)

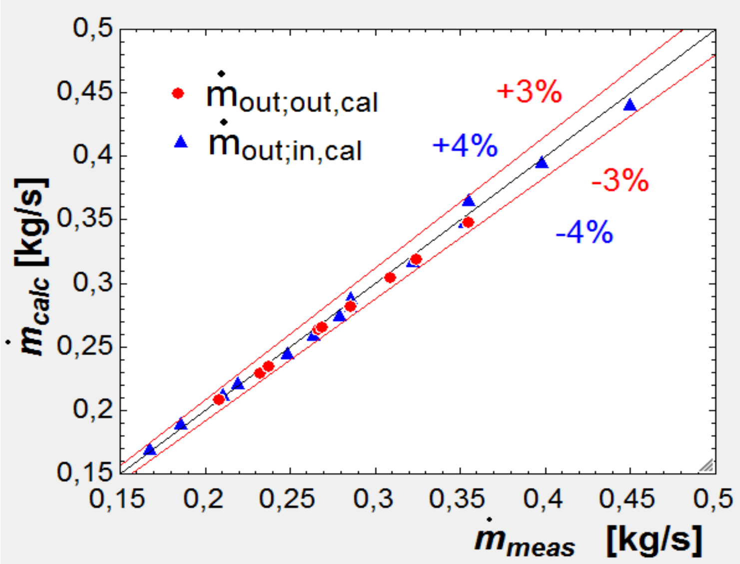
Calibration procedure → minimization of the error f(x_i) between calculated and measured values:

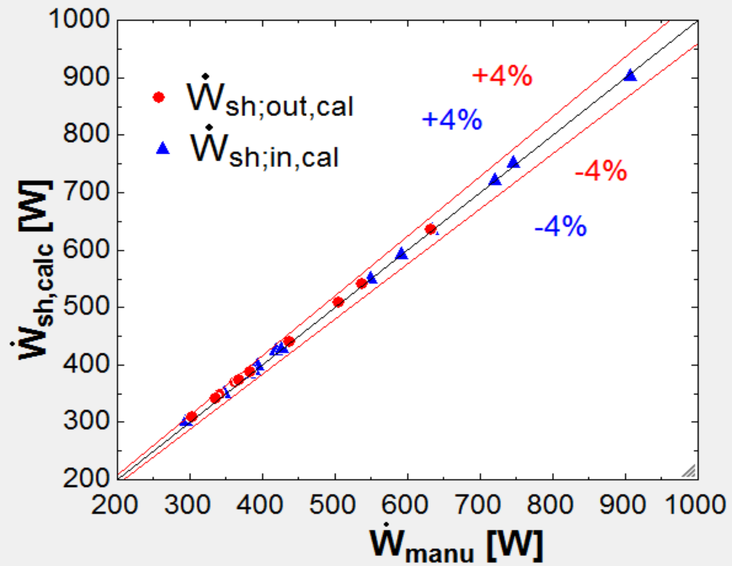
$$\dot{m}_{ex2,calc} \longleftrightarrow \dot{m}_{ex2,meas}$$

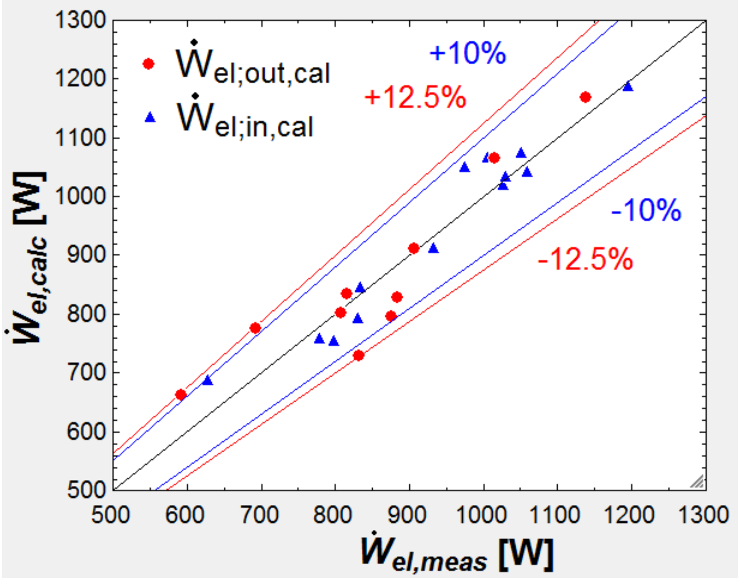
$$T_{ex2,calc} \longleftrightarrow T_{ex2,meas}$$

$$W_{el,calc} \longleftrightarrow W_{el,meas}$$

CALIBRATION AND SIMULATION RESULTS







- Good prediction of mass flow rate and shaft power

- Good accuracy also in the estimate of electric power points in spite of the simplified empirical model of the motor

$$\eta_{mot} = \frac{W_{sh,pump}}{W_{el}} = f(N, Load)$$

CONCLUSIONS

- **Experimental investigation** (multi - diaphragm pump):
  - Performance:
    - ✓ global efficiency → 46% at nominal operation (960 rpm – Δp = 16bar) (almost twice as much the values found in the literature)
    - ✓ relative volumetric efficiency → > 95% (decreasing with Δp)
  - Cavitation: NPSHa > 90 kPa → guidelines about the system response
- **Model**: good prediction of pump operation also considering operating points not used in the calibration procedure
  - ERRORS IN THE PREDICTION: $\dot{m} \leq 4\%$ $T_{out} \leq 0.67\%$ $\dot{W}_{manu} \leq 4\%$ $\dot{W}_{el} \leq 12.5\%$
- **Future work - importance of the motor in the pump system**
  - Further analysis of the motor operation
  - Need to install a torque meter to split pump and motor

Thank you for your attention!

Ευχαριστώ για την προσοχή σας!

Grazie per l’attenzione!

QUESTIONS

- **Shaft power equation** (by manufacturer):

$$\dot{W}_{sh,pump}[kW] = \frac{\dot{V}\Delta p}{511} + \frac{15 * rpm}{84428}$$

- **Acceleration head factor (H_a)** in the NPSH$_{available}$ expression (according to manufacturer):

$$H_a = \frac{C * V * L * N}{K * G}$$

C = Constant determined by type of pump (in this case 0.066 for Hydra Cell D/G10)

L = actual length of suction line ([m])

V = Velocity of liquid in suction line ([m/s])

N = RPM of crankshaft ([rpm])

G= Gravitational constant

K = Constant to compensate for fluid compressibility

- **Electric motor efficiency** – empirical equation:

$$\eta_{mot} = \frac{W_{sh,pump}}{W_{el}} = f(N, Load) =$$

$$-5.31247914 \cdot 10^{-2} \cdot N + 1.22264560 \cdot 10^{-4} \cdot N^2 - 1.18454157 \cdot 10^{-7} \cdot N^3 + 4.13550565 \cdot 10^{-11} \cdot N^4 +$$
$$+1.509897 \cdot 10^1 \cdot Load - 1.44009810 \cdot 10^2 \cdot Load^2 + 6.45006093 \cdot 10^2 \cdot Load^3 +$$
$$-1.04246533 \cdot 10^3 \cdot Load^4 + 7.9110469$$