TEST RIG FOR ORGANIC VAPOURS: DESIGN AND SIMULATION OF OPERATION

A. Spinelli, M. Pini, V. Dossena, P. Gaetani, C. Osnaghi
Laboratorio di Fluidodinamica delle Macchine
Dipartimento di Energia, Politecnico di Milano
Via Lambruschini 4, I-20156, Milano, Italy

ABSTRACT

A blow-down wind tunnel for real gas applications has been designed and the main processes involved in its operation has been simulated in order to verify its behaviour. The facility is aimed to characterize an organic vapour stream, representative of expansions taking place in Organic Rankine Cycles (ORC) turbines, by independent measurements of pressure, temperature and velocity. ORC turbine performances are expected to strongly benefit of flow description and design tools validation provided by experimental data.

This paper discusses how the facility has been designed and presents its final layout. The dynamic models of plant components are also described and the facility operation is discussed via the simulation results.

A straight axis planar convergent-divergent nozzle represents the test section for early tests, but the test rig can also accommodate linear blade cascades. High fluid densities and temperatures to be reached resulted in an unaffordable thermal power to be provided in case of continuous loop, therefore a blow down operating scheme has been adopted. A wide variety of working fluids can be tested with adjustable operating conditions up to maximum temperature and pressure of 400 °C and 50 bar. The test rig operational mode is unsteady, but the inlet nozzle pressure can be kept constant by a control valve.

In order to estimate the duration of both set-up and experiments and to evaluate the time trend of fluid properties during the main processes, the plant operation has been dynamically simulated using *Dymola®*, an object oriented simulation code, able to treat both fluid-dynamic and automation components; they have been either selected in a database or created through *Modelica* routines.

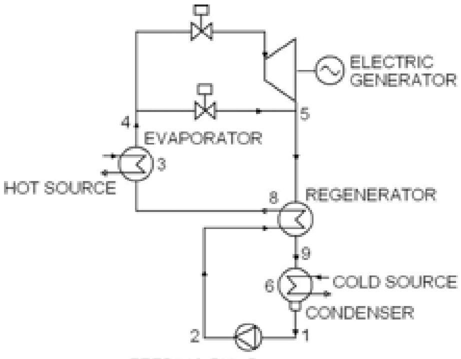
Design and simulation have been performed with a lumped parameter approach using *Siloxane* MDM and *Hydrofluorocarbon* R245fa as reference compounds and *FluidProp®* for properties calculation. Depending on the fluid and test pressure, experiments may last from 12 seconds to several minutes while the plant set-up requires a few hours.

The design calculation and dynamic simulation shown how the set-up and test duration make the facility suited to perform the desired experiments.

INTRODUCTION

Organic Rankine Cycles (ORC) represent a viable technology for the exploitation of energy from low/medium temperature sources, such as renewable or heat-recovery sources, with applications to low/medium electrical power generation and Combined Heat and Power (CHP) plants.

An Organic Rankine Cycle consists in a Rankine Cycle employing an organic compound as working fluid [1]. For the above cited applications, the ORC technology is usually preferred over steam cycle due to the simplicity of plant components, the high reliability and the low operational costs [2, 3], exhibiting at the same



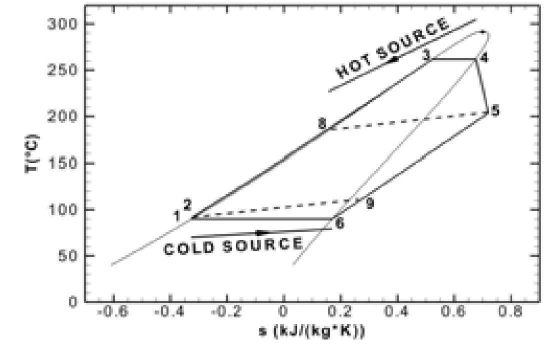


Figure 1: Left: Typical regenerative Organic Rankine Cycle power plant. Right: Corresponding Regenerative Organic Rankine Cycle

time a good thermodynamic cycle efficiency [4].

Further improvement of the overall efficiency of ORC plants can be achieved by better designs for the turbine blade passages, which currently give turbine efficiencies around 80-85 % due to the lack of validated design tools. Indeed, expansions through ORC turbines occur mainly in the so-called dense gas region, namely close to the vapour saturation curve and to the critical point resulting in highly complex passage flows.

First source of complexity is related to strong non-ideal behaviour of vapours, whose accurate prediction requires the use of complex real-gas thermodynamic models. These models are already embedded in specialized CFD codes aimed at calculating real gas flow fields, see for example [5-8], but no experimental data of dense gas flows are currently available for comparison. Moreover, the dense gas flows are further complicated by low values of speed of sound, see [9], thus resulting in strongly supersonic flows, despite the low velocities within ORC turbine blade passages. Therefore, shock waves are always present and significant reduction in machine performances can occur at off-design conditions.

To investigate the real-gas flows of organic compounds in typical ORC operating conditions, the *Test Rig for Organic Vapours* (TROVA) is currently under construction at the Politecnico di Milano, Italy, see [10]. In the facility, expansion flows of different dense gases will be investigated by independent measurements of pressures, temperatures and velocities.

The facility implements an Organic Rankine Cycle (either sub-critical or super-critical) where the expansion process takes place within a nozzle replacing the turbine. A straight axis convergent–divergent nozzle has been chosen as the simplest geometry representative of an ORC turbine blade passage. In order to reduce the required input thermal power, a batch operating facility has been selected.

The design of the TROVA facility and its dynamic simulation represent the work preliminary to the construction of the test rig and are reported in the present paper. The section *Test Rig conception and design* illustrates how the apparatus has been conceived and designed to operate with different working fluids and operating conditions; section *Final layout and operation* describes the setup arrangement and processes included in the thermodynamic cycle it performs; motivation to the choice of plants components are also given. The section *Simulation models* reports how plant models have been built and how the unsteady simulation has been carried out; description of the 1D steady calculation for the nozzle flow is also presented. Results are discussed in the *Simulation results* section, which bring to the *Conclusions*.

TEST RIG CONCEPTION AND DESIGN

The construction of a dedicated facility, namely the TROVA, has been decided as a consequence of strong limitations involved in performing measurements on industrial turbines; these machines are, in fact, not equipped with accesses required to apply optical techniques and the use of pressure probes would require real-gas calibration tunnel which is still not available. Moreover availability of plants for test is usually restricted and the control of operating condition is not based on experiment requirements. Among optical techniques, Laser Doppler Velocimetry (LDV) is required to verify the consistency of thermodynamic models of fluids, while Schlieren techniques allow for shock waves localization, also of great interest.

The facility design requires the definition of the test section type, the fluids to be tested, the flow main parameters and the plant arrangement to feed the test section.

As a test section a straight axis convergent–divergent nozzle has been selected, being the simplest geometry representative of an ORC turbine blade passage. It generates a quasi-1D expansion flow exhibiting a large isentropic core, thus making it possible to measure temperature and pressure fields without the use of calibrated probes, see [10], since the quasi-1D flow field can be obtained from total temperature and total pressure probes at the nozzle inlet and by static pressure taps at different section along the nozzle axis. The nozzle is planar and can easily accommodate a glass window.

Among fluids categories suited for ORC applications (e.g. *Hydrocarbons*, *Organofluorines*, *Haloalkanes* and *Siloxanes* [11]) particular interest has recently risen on *Siloxanes* and *Hydrofluorocarbons* for high (up to 400 °C) and low (up to 200 °C) temperature applications respectively. It is essentially due to their non-toxicity and their low Ozone Depletion Potential (ODP) and Global Warming Potential (GWP). Therefore *Siloxane* MDM and *Hydrofluorocarbons* R245fa has been chosen as reference compounds for plant design and for early tests.

The main flow parameters, namely the maximum values of the nozzle throat area, the upstream stagnation conditions and the expansion ratio, has been selected according to the possibility of pressure probes insertion at throat and of exploring thermodynamic regions up to supercritical fluid conditions. A nozzle throat area A_{th} = 314 mm^2 (namely an equivalent diameter D_{eq} = 20 mm) guarantee a negligible blockage effect in case of insertion of typical miniaturized pressure probe for transonic applications (with head diameter $D_{head} \approx 2$ mm). Total pressure and total temperature (p_{T6} , T_{T6} according to figure 2) and expansion ratio β are reported in table 1; notice that expansion ratio is such that all real-gas phenomena take place within the nozzle. For the two fluids, the above parameters allow for calculation of Mach number M_7 at nozzle exit, of exit-to-throat area ratio for design conditions and of the mass flow; the calculated values are also reported in table 1. Being ORC passage flows normally supersonic, the calculation has been performed for the nozzle chocked flow by solving the set of equations (1). Mass flow is easily obtained from equation (2).

The Mach number (M) is a given parameter, density (ρ) and entropy (s) are taken as independents, enthalpy (h) and speed of sound (a) are thermodynamic functions specified by the fluid model within *FluidProp®* libraries.

$$\begin{cases} s = s_6 \\ h = h_{T6} - \frac{1}{2}M^2 a^2 \\ h = h(\rho, s) \\ a = a(\rho, s) \end{cases} \quad (1) \;;\; m = \rho_{th} A_{th} a_{th} \quad (2)$$

	MDM	R245fa
p_{T6} (bar)	25	37
T_{T6} (°C)	310	159.2
β	25	18.5
D_{eq} (mm)	20	20
A_7/A_{th}	11.8	5.06
M_7	2.25	2.30
$M_{(t=0)}$ (kg/s)	6.25	5.4

Table 1: Flow parameters for MDM and R245fa

Concerning the plant arrangement, the implementation of either a gas cycle (depicted in figure 2, left) or a phase transition cycle (figure 2, right) have been considered; in both cases the expansion process occurs through the nozzle replacing the turbine. The gas cycle option has been discarded due to the complexity of compressor design and to its cost; the phase transition cycle (which mimic an actual Organic Rankine Cycle) has also been rejected in its continuous loop configuration, due to the very high thermal power to be supplied (around 2.5 MW). The final solution adopted has been the construction of a phase transition cycle operating using a batch scheme, allowing for the reduction of inlet thermal power to 30 kW. The final layout of the blow down facility and the thermodynamic cycle it implements are reported in figures 4 and 3 respectively.

The facility includes a high pressure vessel HPV, to store and evaporated the fluid under test; the test section, where vapour expansion takes place and measurements are performed; a low pressure vessel LPV, to collect and condense the exhausted vapour, and a pump to finally pump back the liquid to the HPV. Notice that the batch operation require the HPV pressure to be higher than the nozzle inlet pressure, which can be kept constant by the control valve MCV.

First step involved in the design is to define volumes, maximum operating pressure and temperatures required for the two tanks to perform tests complying the constraint of a minimum duration, which has been fixed in $t_{min} \approx 15$ s, to be compatible with selected measurement techniques. Volumes, pressure and temperatures to be calculated have been fixed as parameters and the transient flow through the nozzle has been calculated as a sequence of steady state by applying equations (1) and (2) for different time steps. The update of HPV and LPV thermodynamic conditions have been carried out by numerically solving the mass and energy balances for the unsteady flows of HPV and LPV, using a lumped parameter approach and *FluidProp®* routines. The main hypothesis assumed to perform the calculation have been the adiabatic emptying and filling processes for HPV and LPV, the constant total enthalpy throttling process throughout MCV and the isentropic expansion through the nozzle (for further details see [10]). At the end of calculation volumes, pressures and temperatures have been iteratively changed to match test duration and economic constraints. Table 2 reports HPV and LPV volumes and maximum pressures and temperatures allowing for experiments of a minimum duration of 12 s (considered as satisfactory), while table 3 illustrates vessels parameters selected to perform the reference tests of table 1 and the corresponding duration.

FINAL LAYOUT AND OPERATION

An overview of the final layout of the *Test Rig for Organic Vapor* and of its operation is given here referring to the plant sketch of figure 4 and to the related thermodynamic cycle of figure 3; note that different plant

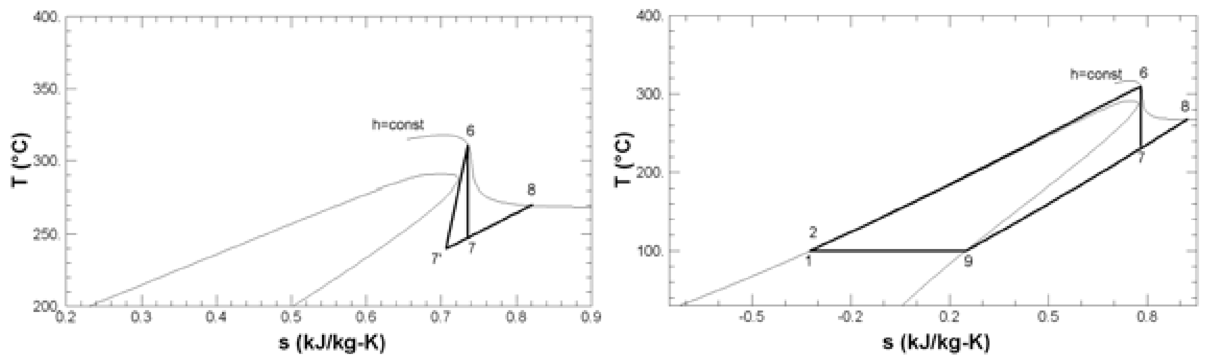


Figure 2: Options for Test Rig continuous cycle. Left: gas cycle. Right: phase transition cycle

sections are labeled in Figure 4 according to the thermodynamic states that are expected to occur. Further details can be found in [10].

A proper mass of the fluid under test is stored in the high pressure vessel HPV and heated up to super-heated/super-critical conditions (process 2→4). Temperature and pressure are higher than those corresponding to the nozzle stagnation conditions; the limiting values for the plant are p = 50 bar, T = 400 °C, as reported in table 2. The heating system consists of four electrical band heaters externally clung to the vessel to better control (by PID regulator) the maximum temperature on vessel external wall.

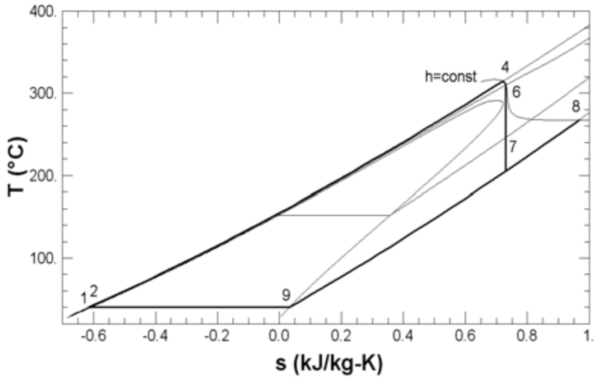


Figure 3: Test Rig thermodynamic cycle

Through an isenthalpic throttling process (4→6), performed by the main control valve MCV, the nozzle is fed at a constant stagnation pressure p_{T6} for a time t_{feed}, representing the maximum allowed experiment duration t_{ex}; this time is limited by the balancing of the HPV total pressure and the nozzle inlet pressure ($p_{T4} = p_{T6}$). Depending on the fluid and test conditions, t_{ex} ranges from 12 seconds (see table 3) to a few minutes. The MCV is equipped with a PID controller which operates using the total pressure at nozzle inlet as feedback signal. Within a rest plenum, located just ahead the test section, the flow is slowed to rest and stagnation conditions are measured by means of thermocouples and total pressure probes.

	HPV	LPV
V (m^3)	1	5.6
p_{max} (bar)	50	20
T_{max} (°C)	400	400

Table 2: Vessels design parameters

	MDM	R245fa
$p_{T4(t=0)}$ (bar)	50	50
$T_{T4(t=0)}$ (°C)	315	176.5
t_{ex} (s)	12	28.5

Table 3: Vessel conditions for reference tests of Table 1

Currently, the test section consists of the nozzle but a linear blade cascade can also be accommodated. A planar nozzle has been adopted and equipped with a glass window to allow for optical access. Laser Doppler Velocimetry (LDV) measurements are used to obtain direct measurements of velocity field. Optical access is also required to carry out shock wave localization by means of Schlieren technique. A sequence of pressure taps located on the wall opposite to the window, allow for static pressure measurements along the nozzle axis. Despite the constant inlet pressure, the nozzle flow is unsteady due to the change in total enthalpy related to the HPV emptying process; the change in total enthalpy during the nozzle characteristic time is extremely low, therefore the nozzle flow can be treated as a sequence of steady flows with transient operating conditions, which can be obtained at each time through the measurement of total pressure p_{T6} and total temperature T_{T6} .

The vapor exiting the nozzle is slowed to rest (7 → 8) and collected in the low pressure vessel LPV, where it is subsequently de-superheated (8 → 9) and condensed (9 → 1). These processes are carried out by two heat

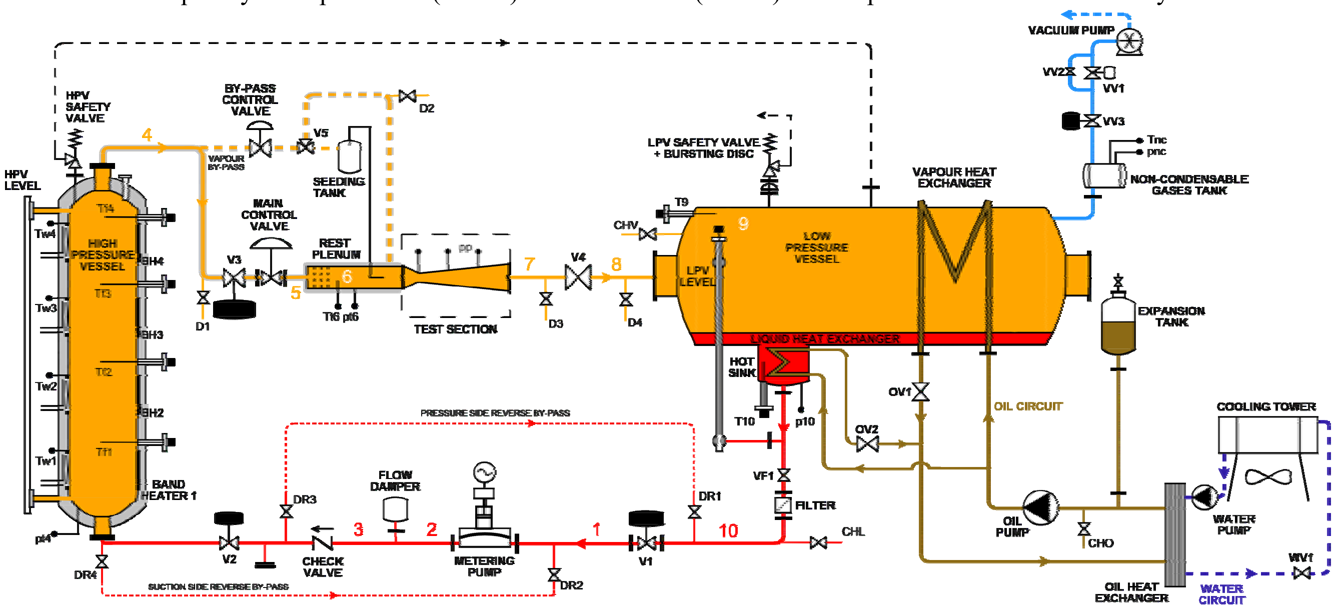


Figure 4: Final layout of the Test Rig for Organic Vapours

exchangers; a first one is placed on the LPV external wall to de-superheat and condense the exhausted vapor; a second one is located in the vessel hot sink, where the condensate is collected, and allows for liquid cooling. Both heat exchangers are fed by thermal oil as cooling media; the oil cooling is carried out, through a plate heat exchanger, by water coming from a cooling tower circuit.

The facility loop is finally closed by fluid compression to the HPV (1 → 2) through a membrane metering pump MP, suited to handle the low mass flow rate to be compressed; a series of valves allow for the opening/closure of the circuit and, if needed, for flow reversal.

A vacuum section, which includes a vacuum pump VP and a group of valves to regulate the mass flow, is also required for de-aerating purposes; the whole plant or part of it can in fact operate at pressures lower than atmospheric pressure.

SIMULATION MODELS

Due to the unsteady behaviour of the test rig, the main processes involved in the facility operation have been dynamically simulated in order to assess the time required for the circuit set-up and the duration of experiments; these characteristic times are strongly influenced by control loops action, which has been included in the calculation. To evaluate the plant set-up time, the simulated processes were the heating/evaporation of the fluid stored the HPV (process 2 → 4 in figure 3) and the de-superheating/condensation of the vapour exhausted to the LPV (8 →1), while the calculation of the test duration required to model and to simulate the throttling process (4 → 6) and the isentropic expansion through the nozzle (6 → 7). A batch simulation, reproducing the actual plant operation, has been carried out using *Dymola*® (Dynamic Modelling Laboratory) an object-oriented dynamic software where complex models (e.g. a power plant) can be built by assembling simpler models with a hierarchical approach. The software is based on the modelling language *Modelica* (first presented in 1997, see [12]), which a-causal approach permits the re-use of built models for different applications. The batch simulation presented here implies that results of one process calculation provide initial data for the following one.

The lack of many components included in the TROVA, such as the supersonic nozzle, imposed the construction of a self-made library. Hence, using *Modelica* code and a lumped parameter approach, several plant components have been built and included in the specific *TestRig* library, on which basis the three main plant sections (heating/evaporation, test, de-superheating/condensation) have been modeled.

The heating model includes the fluid, the heat transfer coefficients across the HPV wall, the thermal inertia of the HPV vessel and the thermal losses from the insulated and non-insulated surfaces. Concerning the fluid, the *Boiler* model has been built implementing the unsteady mass and energy balances for a closed volume coupled with *FluidProp*® routines, by which all thermodynamic properties have been calculated as functions of density ρ and specific internal energy per unit mass u. The transport properties, namely thermal conductivity k and dynamic viscosity μ have been taken from industrial experimental data. The *Boiler* model has been connected to the *Boiling* components to simulate the convective heat exchange process between the fluid and the HPV internal wall, namely to calculate the convective coefficients at each time step. Depending on the fluid state, the model perform this calculation by automatic selection of the proper correlation among those implemented (i.e. the Rohsenow correlation for the boiling liquid state [13] and the free convection Churchill-Chu correlation [13] for single phase states). In fact, during the heating process the coexistence of liquid and vapour phases results in strong differences on heat transfer mechanisms (and therefore in convective coefficients) at different location in the HPV, depending on the position of the two-phase interface. This fact in turn causes significant changes on wall temperatures along HPV axis, influencing the action of regulators, based on wall temperature control. To properly account for these temperature differences, within the *Boiler* model, the fluid volume has been divided into four zones, each coupled to the related band heater and connected to a single *Boiling* model. At each time step, temperature and pressure correspond to saturation conditions, while other properties depend on the fluid phase. Concerning the vessel segment where the two-phase

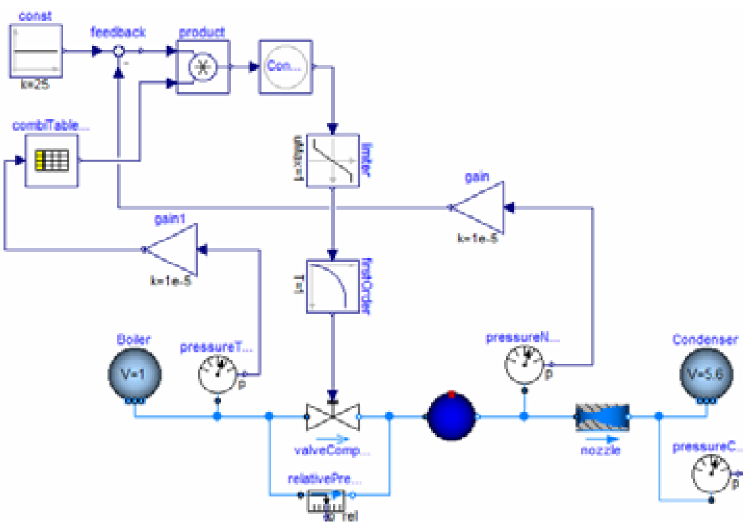


Figure 5 : Dymola model of the test including the control loop

interface is located, the convective coefficient has been calculated by properly averaging values obtained from the two correlations cited above. It has to be noticed that the heating model allows to reach supercritical states. Simulations reported here implement the proportional and the integral parts of a PID controller for each heater; the feed-back variables are the wall external temperatures provided by thermocouples.

The test model (depicted in figure 5) comprises the two vessels (HPV, LPV), the main control valve, the rest plenum and the supersonic nozzle. The control valve (MCV) model has been built by implementing the control loop of figure 5 over a valve component able to treat compressible throttling processes. A polynomial correction function, acting on the valve opening signal, has been included in order to better follow the non linear emptying process of HPV. In addition, a switching component, allowing for commutation between manual and automatic control, has been included; it guarantees stability of valve behaviour by forcing a fixed opening position for a time approximately corresponding to plenum filling time. Concerning the nozzle model, a lumped parameter approach has been adopted to write non-linear equations of motions (see set of equations 3) between the main sections (inlet, throat, normal shock section and outlet).

The condensation model has been built by a structure similar to the one adopted for the heating system, but with a simpler approach, due to the absence of a control system. The fluid model implements the natural convection correlation of Churchill-Chu, to treat heat exchange between super-heated vapour and vessel wall, the Nusselt correlation [13] for film condensation within horizontal cylinders and the conduction model through the wall. The heat exchange between the thermal oil and the LPV outer wall has been modelled implementing the Dittus-Boelter correlation [13] for forced convection through the pipe representing the heat exchanger located at the vessel external surface.

Besides the dynamic simulations, a quasi 1-D steady code has been written (using Fortran language) to calculate all thermodynamic properties and velocity along the axial coordinate of the nozzle. Being the unsteady flow through the nozzle well approximated by a sequence of steady flows, the solution represents the quasi 1D flow field at the time corresponding to the given boundary conditions. This code can also be used for initializing the unknown variables during the dynamic simulations of the test process. An example of non-linear system, implemented within both the 1D code and *Modelica* model of the nozzle, is given by the set of equations 3, which reduces to equation 4 and allow for calculation of choked conditions at the throat.

$$\begin{cases} M_{th} = 1 \\ s_{th} = s_0 \\ h_{t0} = h_{th} + \dfrac{v_{th}^{\;2}}{2} \\ v_{th} = a_{th} \\ h_{th} = h_{th}(p_{th}, s_0) \\ \rho_{th} = \rho_{th}(p_{th}, s_0) \\ T_{th} = T_{th}(p_{th}, s_0) \\ a_{th} = a_{th}(p_{th}, s_0) \end{cases} \quad (3)$$

$$F(p_{th}, s_0) = h_{t0} - h_{th}(p_{th}, s_0) - \frac{[M_{th} \cdot a_{th}(p_{th}, s_0)]^2}{2} \quad (4)$$

RESULTS OF SIMULATION

In this section the simulation results are presented for the three main processes (heating, test and condensation), further details can be found in [14]. Calculations have been carried out for the same two fluids employed as reference for design (MDM and R245fa). In case of MDM, two operating conditions have been considered; the first one, labelled MDM(1), corresponds to a supercritical state at nozzle inlet and matches conditions taken as reference for plant design; a second one, labelled MDM(2), corresponds to an expansion process typical of actual ORC turbines. Concerning R245fa, the only selected operating condition corresponds to the design reference test, with supercritical state at nozzle inlet.

Heating system

Thermodynamic conditions at the end of the heating process have been selected in order to obtain single-phase expansions within the real gas region of fluids. The heating duration and the final total thermodynamic conditions are listed in table 4.

Fluid	**MDM(1)**	**MDM(2)**	**R245fa**
p_{T4} (bar)	50	50	50
T_{T4} (°C)	315	311,4	176.5
ρ_{T4} (kg/m^3)	480,82	486,92	460,80
h_{T4} (kJ/kg)	372,95	364,03	498,17
heating time (s)	25000	25000	10000

Table 4: Final HPV thermodynamic conditions, and heating process duration

By means of the four PID regulators controlling the band heaters, significant temperature overshoots have been avoided. Simulations have been carried out implementing a proportional–integral (PI) controller. For all cases, the regulator has been tuned after several simulations by selecting a proportional gain value of K_P = 750 W/°C and an integral time Ti = 1000 s; these values allowed to match set point parameters (p_{T4}, T_{T4}) with a reasonably rapid increase in temperature and an almost zero over-elongations around the set point (see figure 6).

An important difference occurs in the isochoric heating process of MDM and R245fa. For both MDM(1) and MDM(2) cases the process starts from a two-phase state; as the temperature and pressure increase, the isochoric line cross the liquid saturation curve, namely the fluid state correspond to subcooled liquid and subsequently supercritical conditions are achieved. On the contrary the heating process for R245fa is such that the fluid crosses a superheated vapour states to finally reach supercritical conditions. The different behaviour depends on the fluid mass initially charged to the HPV vessel and on the initial temperature. The heat transfer coefficients vary according to the fluid thermodynamic states, assuming therefore higher or lower values depending on the heat exchange mechanisms involved, namely either the boiling of liquid or the natural convection of a single phase fluid (subcooled liquid, superheated vapour or supercritical fluid), as an example see figure 6. For all discussed cases the external wall temperature of the HPV insulation layer remains below 60 °C for the whole process, as it is required by safety constraints; the thermal losses caused by non-insulated components (e.g. the charging and discharging pipelines) exhibited maximum values below 10% of the total inlet thermal power.

Test simulation

During the test, a fraction of the total mass of fluid stored within HPV is discharged to the LPV; the discharged mass depends on the nozzle and HPV operating conditions and is reported in table 2 for the discussed cases. Table 2 also summarizes test durations and thermodynamic states of fluids contained in the two vessels at the experiment end.

A single test concludes when pressures across the control valve are balanced and the test section can no longer be fed; the HPV pressure is equal to the nozzle set-point pressure (p_{T4} = p_{T6}) as can be seen in table 5 for MDM(1) and R245fa experiments, exhibiting a duration of 12 and 28.6 seconds respectively. As a test conclusion threshold, the occurrence of a normal shock wave at nozzle exit can also be selected, if the study of normal shock waves entering the nozzle is not of interest. This can be the case of the MDM(2) test, where a normal shock wave forms at nozzle exit, as a consequence of the increasing pressure of the LPV due to the filling process; according to this criterion the test last for t_{ex} = 93 s (see table 5), while the criterion of pressure balance across the control valve imposes a test duration of about 270s. For all cases, test durations appeared adequate to obtain reliable measurements of nozzle flow field.

Fluid	**MDM(1)**	**MDM(2)**	**R245fa**
p_{T4} (bar)	25	17.7	37
T_{T4} (°C)	308	302.5	158.1
p_{T9} (bar)	2.27	4.44	8.33
T_{T9} (°C)	275	277	104.8
p_{T6} (bar)	25	10	37
Discharged mass (kg)	75	149	155.6
Test time (s)	12	93	28.6

Table 5: Final HPV and LPV thermodynamic conditions, discharged mass and test durations

A fast achievement of the set-point value for the nozzle inlet pressure p_{T6} has been obtained (with almost no oscillations in the valve opening process, see figure 7), by optimizing the time of manual operation mode for the

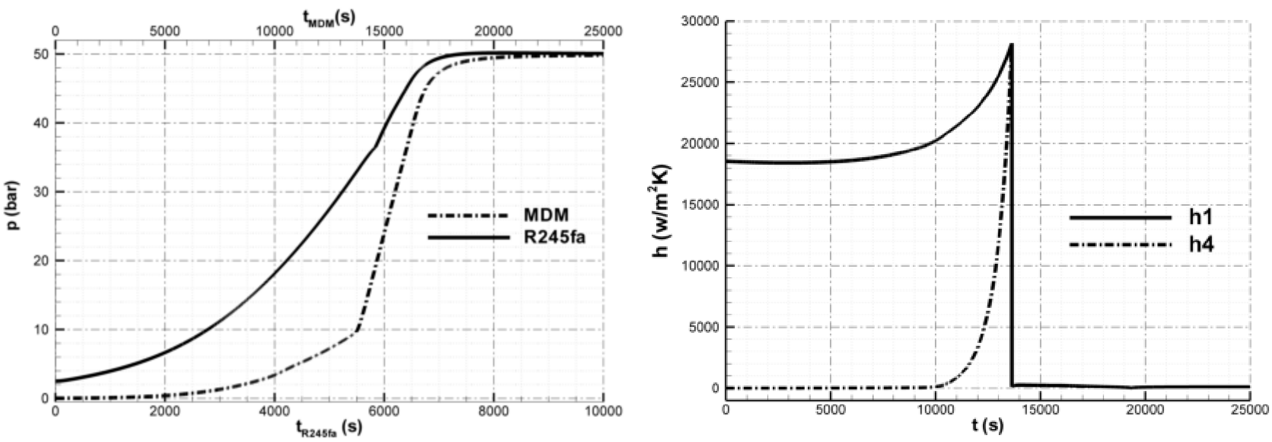


Figure 6: Left: pressure trends of MDM(1) and R245fa within HPV.
Right: heat transfer convective coefficients for MDM(1) in lower (h_1) and upper (h_4) HPV zones

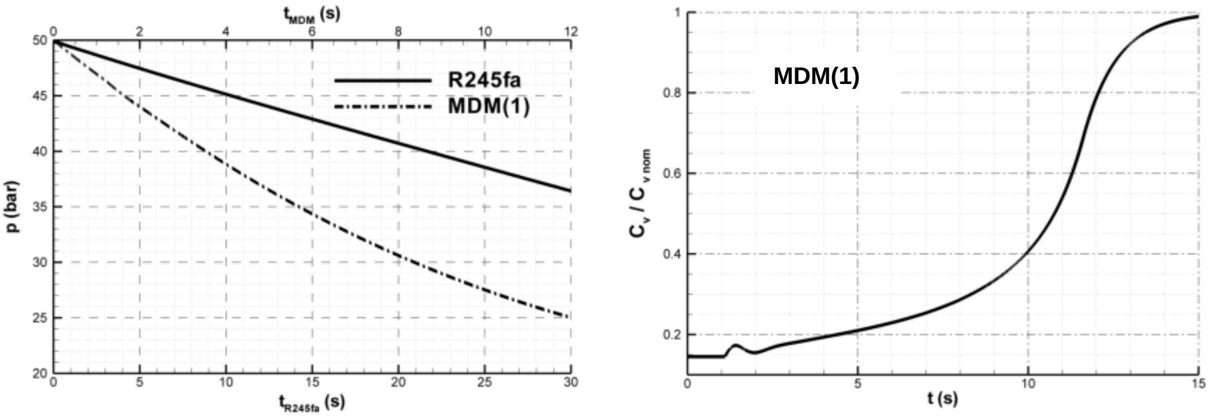


Figure 7: Left: HPV pressure trends for MDM(1) and R245fa tests.
Right: opening trend of main control valve for MDM(1) test.

control valve.

For all discussed cases the nozzle flow is choked at experiment start. The time trends of pressure threshold values at nozzle exit (see figure 8) namely supersonic-design pressure p_{sup}, subsonic-design pressure p_{sub} and exit-normal shock pressure p_{shock}, allow to define the actual nozzle flow regime at each time step, by simple comparison with the LPV pressure p_{LPV}; the under-expanded or over-expanded nature of the flow can be detected, as well as the occurrence of normal shocks in the divergent portion or the flow unchocking. Figure 8 shows how the nozzle is under-expanded until p_{LPV} line crosses p_{sup} line at t = 6.15 s for MDM(1), while for R245fa case the flow is over-expanded during the entire test. The flow remains choked for each test and the subsonic design pressure is slightly lower than inlet one.

Figure 9 depicts the pressures and Mach number trend along the nozzle axis for MDM(1) and R245fa cases. The quasi 1D flow field has been obtained by implementing the above cited Fortran code, for fixed values of p_{T6} and T_{T6} and for three different reference values of LPV pressures, corresponding to different nozzle regimes. It can be noticed that, for MDM(1) case only, the Mach number is not monotonically increasing along the nozzle axis as in the case of a polytropic ideal gas. This phenomenon is a real gas effect related to the value of the fundamental derivative of gasdynamics Γ, defined as $\Gamma = 1 + \rho / a(\partial a / \partial \rho)_s$. In fact in case of compounds exhibiting high molecular complexity (as it is the case of MDM), thermodynamic regions are present, close to the liquid-vapour saturation line and to the critical point, where Γ<1, namely the speed of sound increase as the density decrease along the expansion; as a consequence the Mach number can decrease if the increase in the speed of sound overcomes the increase in fluid velocity. R245fa does not exhibit this behaviour, having a relatively low molecular complexity. For MDM(2) experiment the Mach number trend is similar to R245fa one, namely the expansion does not cross regions where Γ<1.

Condensation system

The fluid condensation represents a non-critical process with a characteristic time of an order of magnitude lower than the heating process; no particular cares were required to perform the fluid cooling, therefore no control loops have been included The process duration is proportional to the discharged mass during the test

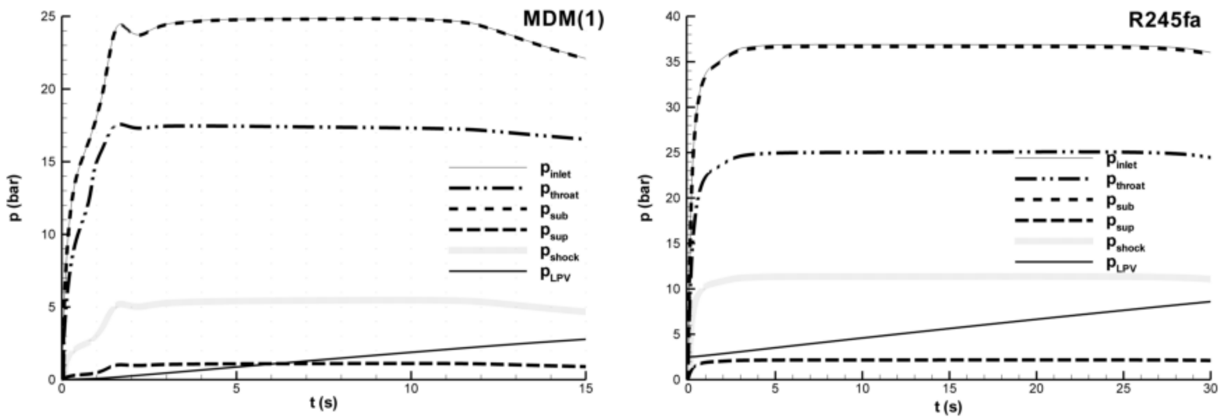


Figure 8: Pressure threshold values for MDM(1) (left) and R245fa (right) tests

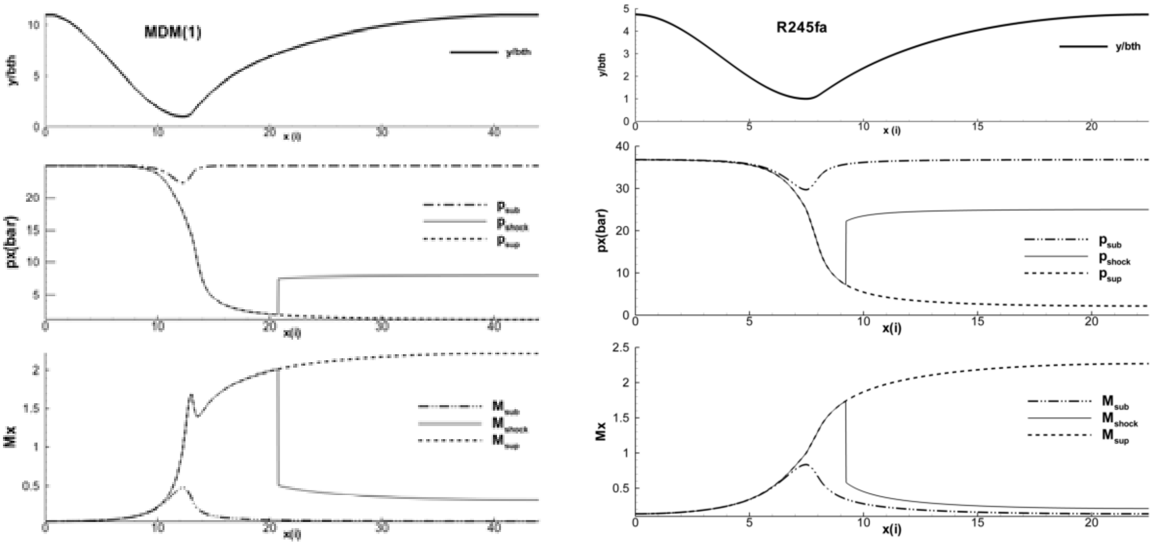


Figure 9: Pressures and Mach trends along the axial coordinate of the nozzle for three different LPV pressure values. Left MDM(1), Right R245fa.

phase and to the selected condensation temperature, which has been fixed at T_{cond} = 40 °C for all the discussed tests (see table 6). For MDM(1) and MDM(2) cases condensation time is, respectively, 1000 s and 1500 s, while R245fa case required around 2000 s for a complete condensation. Table 6 reports the obtained results.

Condensation system

The fluid condensation represents a non-critical process with a characteristic time of an order of magnitude lower than the heating process; no particular cares were required to perform the fluid cooling, therefore no control loops have been included. The process duration is proportional to the discharged mass during the test phase and to the selected condensation temperature, which has been fixed at T_{cond} = 40 °C for all the discussed tests (see table 6). For MDM(1) and MDM(2) cases condensation time is, respectively, 1000 s and 1500 s, while R245fa case required around 2000 s for a complete condensation. Table 6 reports the obtained results.

As in the heating process, the final condensate state depends on the discharged mass and final temperature. For desired final T_{T9} (see tab. 6) both fluids are in two-phase conditions. The slope variation of the curves depicted in figure 10, in the temperature – time plane, results as a consequence of the change in pressure and in convective heat transfer coefficient values during the cooling process. The discontinuity exhibited in the first derivative corresponds to the end of de-superheating an the beginning of condensation process

Fluid	MDM(1)	MDM(2)	R245fa
p_{T9} (bar)	0.0124	0.0124	2.51
T_{T9} (°C)	40	40	40
condensation time (s)	1000	1500	2000

Table 6: LPV thermodynamic conditions at condensation process end

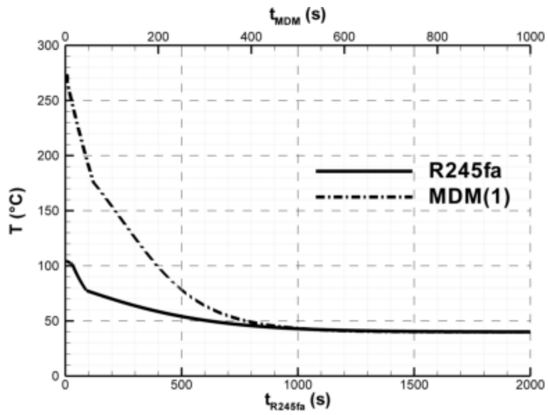


Fig. 10: Temperature trends of MDM(1) and R245fa during the condensation process

CONCLUSIONS

The design and the dynamic simulation of a test rig for investigations on organic vapours has been presented. The facility has been conceived to investigate expansions representative of ORC turbine blade passages, aiming to characterize these flows and to provide experimental data extremely useful to validate CFD codes together with fluid thermodynamic models.

The test rig implements a batch operating Organic Rankine Cycle to feed a planar convergent–divergent nozzle, replacing the turbine, where independent measurements of pressures, temperatures and velocities can be performed. The plant has been designed using MDM and R245fa as reference fluids under cost and test duration constraints.

In order to assess the time required for plant set-up and experiment durations and to properly operate the rig, the facility behaviour has been dynamically simulated by using the object-oriented software *Dymola®*. The three main processes involved in the test rig cycle (heating, test and condensation) have been modelled and simulated. Results of simulation showed how the test rig is suited to perform the desired investigations.

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