Adaptive PI Control of an Organic Rankine Cycle Power Plant

F. Padula * **R. Sandrini** ** **G. Cominardi** ***

* *Dipartimento di Ingegneria dell'Informazione, University of Brescia, Italy. e-mail: fabrizio.padula@ing.unibs.it*
** *Dipartimento di Ingegneria Meccanica e Industriale, University of Brescia, Italy. e-mail: roberto.sandrini@ing.unibs.it*
*** *Turboden s.r.l., Pratt & Whitney Power Systems. e-mail: giuseppe.cominardi@turboden.it*

Abstract: In this paper a PI based adaptive control system is developed to control an organic Rankine cycle power plant. In particular, the control of the turbine speed during the transient between the normal operation mode of the system (connected to the grid) and the island-mode (connected to a stand-alone load) is of main concern. A control algorithm suitable for the implementation on a programmable logic controller is developed. The designed controller is adaptive with respect to different plant thermodynamic boundary conditions and takes into account the limitations imposed to the control variable. An object oriented simulator of the system is developed and the controller is tuned via simulation.

Keywords: PI Control; Adaptive Control; ORC Power Plant.

1. INTRODUCTION

Organic Rankine Cycle (ORC) plants have nowadays reached a good level of development (Hung et al., 1997). In order to meet the requirements of the market and the new regulations concerning the production of electricity from renewable plants, ORC plants have to work properly under different conditions and in different operation modes. In particular, the ORC plants are required to be able to deliver power to the grid as well as to work in island-mode, condition under which the speed of the turbine is no longer imposed by the grid frequency and the required power is lower. In this context, the transitions from the nominal operating point (grid connected) to a different one (stand-alone) are often critical for the ORC plants and they are a challenging control problem too. A good control strategy has to be developed to allow ORC turbines to quickly adapt to new configurations without excessive speed overshoot and undershoot.
It is well known that, in order to develop an effective control strategy, information about the dynamic behaviour of the controlled plant are needed. The dynamic modeling of energy systems, in the last few years, has been widely employed for the analysis of energy conversion power plant (Colonna and van Putten, 2006, 2007; Quoilin et al., 2011). Because of the complexity and the nonlinearity of the ORC plants, it appears sensible to model the plant by means of a suitable software simulator. Using `Dymola`, based on `Modelica` language, (Casella and Leva, 2003) the dynamic model of a section of an ORC power plant is analysed under different thermodynamic boundary conditions.
Based on the dynamic behaviour of the turbine, a control strategy is developed to regulate the transient from the design condition to the island-mode, when the plant is disconnected from the grid and the electric power absorbed by the stand-alone load is significantly lower. In particular, while reducing the power generated by the plant up to the value required by the stand-alone load, the control system is required to avoid large fluctuations around the turbine speed set-point.
The control system operates on the plant through different actuators: the valve motors and the electric brake. The most challenging aspect of the control problem is that the actuators, for technological an economical reasons, have strong limitations, in particular in terms of capability of quickly change the control variable. Moreover, the control strategy has to be designed considering that different thermodynamic boundary conditions give rise to different dynamic behaviours. Finally, despite the complexity of the required control task, the algorithm has to be simple in order to implement it on a Programmable Logic Controller (PLC).
An adaptive Proportional-Integral (PI) plus Feed-Forward (FF) controller is developed to control the valves, whereas, for the brake, a FF strategy is designed, based on an integral criterion of energy dissipation. The parameters of the control system are automatically adapted depending on the different working conditions. The choice of using a PI controller lies on the fact that they are undoubtedly the most diffused controller in industry. Indeed, despite its simple structure, the PI controller allows to achieve satisfactory performance both in term of control response and robustness with a good cost/benefit ratio (Visioli, 2006; Åström and Hägglund, 2006) and it is also suitable for adaptive technique (Åström et al., 1993).
The effectiveness of the proposed solution is highlighted by means of several simulations, under different conditions.
The paper is organised as follows: in Section 2 the layout of the system is described while in Section 3 the limitations on the control variables are described and in Section 4

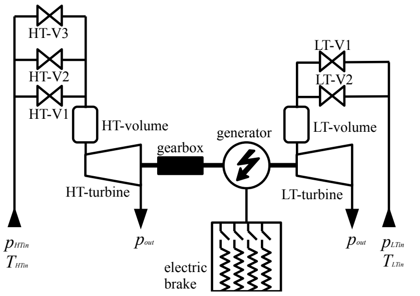

Fig. 1. Functional diagram of the ORC plant power section.

the tools used to simulate the plant are indicated. The proposed control strategy is explained in Section 5 and simulation results are shown in Section 6. Finally, in Section 7, conclusions are drawn.

2. ORC PLANT

The system to be controlled is a section of a geothermal ORC power plant, whose functional diagram is shown in Figure 1. The mechanical side of the ORC plant is mainly composed by the two turbines, HT (that stands for high temperature/pressure turbine) and LT (low temperature/pressure), and the relative piping and valves. The HT side of the plant works at higher inlet temperature and pressure, $T_{HT_{in}}$ and $p_{HT_{in}}$ respectively. The LT side of the plant works at lower inlet temperature and pressure, $T_{LT_{in}}$ and $p_{LT_{in}}$ respectively. Both the turbines work at the same discharge pressure p_{out}. The set of values $\{T_{HT_{in}}, p_{HT_{in}}, T_{LT_{in}}, p_{LT_{in}}, p_{out}\}$ represents the thermodynamic boundary conditions of the plant. The plant could work under different boundary conditions, in particular, the most critical conditions are the winter ones, with maximum produced power P_{max}. The turbine inlets are regulated by the valves HT-V1, HT-V2, LT-V1, LT-V2 and HT-V3. The first four valves are kept completely open when the plant delivers electric power to the grid, while, when working in island-mode, they are completely closed and the LT turbine is switched off. Viceversa, the *control* valve HT-V3 is used to control the plant when it works disconnected from the grid.

Between the valves and the turbines there are dead volumes, they represent the volumes of the pipes and of the turbines casings. Considering dead volumes is significant when modeling ORC plants. Indeed, using an organic fluid, the plant works at relatively low temperature and high volumetric mass flow, consequently the diameters of the pipes are big and the dynamics induced by the dead volume capacities cannot be neglected.

By means of a synchronous electric generator, the mechanical energy produced by the turbines is converted into electric energy. The LT turbine is directly connected to the generator, whereas the HT turbine is connected to the generator through a speed reducer.

The energy produced by the generator is delivered to the grid when the plant is connected. When the network slips, part of the produced energy is still absorbed by the stand-alone load (mainly, the plant services), while the rest is dissipated through the electric brake (during the transient) or is transformed into kinetic energy by accelerating the machineries' rotors.

Table 1. Normalised turbine characteristics.

Turbine	ω_{des}	P_f	K_m	K_a	β_c
HT	1	1	1	1	1
LT	0.5557	0.8	1	1.3872	1

To model the expander, the condition of choked flow is considered. The mass flow $\dot{m}$ is a function of the actual pressure ratio $\beta = \frac{p_{in}}{p_{out}}$ and of the critical one β_c according to the following equations:

$$\begin{aligned} \dot{m} &= K_m K_a p_{in} && \text{if } \beta \geq \beta_c \\ \dot{m} &= K_m K_a \beta_c p_{out} \frac{(\beta - 1)}{\beta_c - 1} && \text{if } \beta < \beta_c, \end{aligned} \tag{1}$$

where K_a and K_m are characteristic parameters of the turbine. The energy balance is considered in order to calculate the mechanical power of the turbine

$$\dot{m}(h_{out} - h_{in}) + P_f \frac{\omega^2}{\omega_{des}^2}, = \tau\omega, \tag{2}$$

where P_f is a power factor to describe the mechanical energy loss, ω is the actual speed, ω_{des} is the design speed, h_{in} and h_{out} are, respectively, the inlet and the outlet enthalpies and τ is the generated torque.

The enthalpy at the turbine outlet is calculated considering the isentropic efficiency performance curve $\eta(\omega, \beta_v)$, being the volumetric ratio $\beta_v = \frac{\rho_{in}}{\rho_{out}}$. Note that this is a common procedure, details can be found in (Dixon, 2005). The isentropic efficiency is a characteristic curve of the machine, provided by the constructor. The values of the characteristic parameters of the HT and LT turbines, normalised with respect to the ones of the HT turbine, are in Table 1.

The valves are used to regulate the mass flow feeding turbines. The following equations describe their behaviour

$$\begin{aligned} \dot{m} &= \frac{27.3}{3600} K_v \sqrt{\frac{\rho_{in}\Delta p}{1e5}} \left(1 - \frac{\Delta p}{p_{in} 3 x_t F_g}\right) \\ F_g &= \frac{C_{p_{in}}}{C_{v_{in}} 1.4}, \end{aligned} \tag{3}$$

where $C_{p_{in}}$ and $C_{v_{in}}$ are, respectively, the inlet constant pressure and constant volume heat capacity of the fluid, ρ_{in} is the inlet density, Δp is the pressure drop along the valve and K_v (flow coefficient) and x_t are parameters depending on the valves opening degree $x \in [0, 1]$, where 0 means valve completely closed and 1 completely open. Again, the relations between K_v, x_t and x are known in the form of table to be interpolated.

Dead volumes are very important when considering the dynamic behaviour of an ORC plant, because they add a strong dynamic related to the flow accumulation in the pipes and in the casings of the turbines. A dead volume introduces a dynamic of higher order than the one dues to a rotational inertia, thus it makes the system more difficult to be controlled. Energy and mass balances are considered to describe the behaviour of a volume

$$\begin{cases} \dot{M} = \dot{m}_{in} - \dot{m}_{out} \\ \dot{E} = \dot{m}_{in} h_{in} - \dot{m}_{out} h_{out} \\ M = \rho V \\ E = Mu, \end{cases} \tag{4}$$

where $\dot{m}_{in}$ and $\dot{m}_{out}$ are, respectively, the inlet and outlet mass flow, h_{in} and h_{out} are the relative enthalpies, M and E are the mass and the energy of the fluid inside the volume, u is the fluid specific energy and V is the dimension of the volume. The considered volumes, normalised with respect to the HT one, are of 1 for HT side and of 0.6494 for the LT one.

The electric generator is a synchronous voltage controlled generator. The generator has an embedded control system that rules the rotor excitement in order to keep the produced voltage constant. Accordingly, we work under the assumption that the output voltage is constant. Thus, the generator exerts a reaction torque proportional to the power of the load, according to the following equation

$$\tau_r = \frac{P}{\omega}, \tag{5}$$

being P the electric power of the load and ω the speed of the generator.

A system such as the ORC plant object of this work has typically big rotating masses, mainly depending on the generator and turbines rotor. For the purpose of dynamic modeling, a unique inertia $J = 12.8575$ (normalized with respect to the HT turbine one) can be considered. The dynamic equation of the rotating masses can be written as

$$J\dot{\omega} = \tau_t - \tau_r, \tag{6}$$

where τ_r is defined in (5), while τ_t is the torque acting on the generator crank. Note that, besides the dead volumes, inertia introduces the other dynamic equation.

The electric brake dissipates the excess of power produced during the transient of the system from the normal operation mode (grid connected) to the island one (stand-alone) in order to avoid excessive speed overshoot. It is a loadbank composed of four resistors whose electrical resistances are one the half of the other. Combining the resistor according to a binary code, it is possible to obtain sixteen different values of braking resistance. It also means that sixteen levels of power dissipation are available. The component employed on the considered ORC plant has a resolution of $R = 0.0537$ and a maximum power dissipation of 0.8048 (normalised with respect to the maximum power P_{max}).

The main characteristic of an ORC plant is that the working fluid is an organic fluid instead of water. The working fluid of the considered plant is **hfc234fa**, an hydrofluorocarbons organic fluid.

3. VALVES AND BRAKE ACTUATION

Despite its constructive simplicity, the brake has a very complex actuation system that introduces strong limitations in its usability. The resistors are connected to the generator through power switches. All the power switches have the same limitation: a finite and asymmetric commutation time (t_{on}, t_{off}) and an asymmetric latency time between a commutation and the next possible one (T_c, 0). In particular, the time required by a switch to open is $t_{on} = 0.05$ s, while the time required to close is $t_{off} = 0.03$ s. When a switch is open and it has been already closed before, it cannot start to close again since the commutation time $T_c = 4$ s has elapsed from the previous closure, whereas, when it is closed it can start to open immediately. The main non ideality in the valves actuation lies in the limited actuation speed (slew rate). The valves cannot move faster than a certain speed $\dot{x}_{max}(x)$, depending on the valve position x. The reason of this limitation lies in the need of reducing the actuator size and cost. The control valve HTV3 has constant maximum actuation speed of $\dot{x}_{max_3}(x) = 45$ %/s that strongly influences the control loop behaviour.

4. SIMULATION APPROACH

From the previous sections it clearly appears that simulating this system, by itself, is a complex task. In particular, since the fluid equations are not known in a closed form, obtaining an explicit model, even if strongly nonlinear, is very difficult. To overcome this problem, a `Modelica` based object-oriented simulator has been developed under `Dymola` environment. The properties of the fluid are computed using the `ExternalMedia` library (Casella and Richter, 2008) together with `FluidProp` (Colonna and Van der Stelt, 2004), while part of the components have been modeled by means of the `ThermoPower` thermodynamic library (Casella and Leva, 2009).

5. CONTROL STRATEGY

The control strategy can be broadly divided in two main parts: the the brake controller and the HT-V3 valve controller. The control system sampling time is 0.015 s (it is the time of the PLC cycle). The desired speed, or set-point value, will be denoted as $\bar{\omega} = \omega_{desLT}$ (see Table 1).

The disconnection from the grid is detected by using a threshold system: the control system starts to work at the time instant t^*, when the speed crosses a certain threshold. After several simulations a threshold value, normalised with respect to ω_{desLT}, of 0.0127 over the set-point speed has been chosen. This choice is the consequence of a trade-off between the necessity of a prompt action of the control system (tested by simulations) and the one of avoiding a grid slip hoax (according to *Turboden s.r.l.* experience).

5.1 Brake Control

The brake control is a non linear FF strategy. Let be P_t the mechanical power produced by the plant, P the power of the load (stand-alone) and P_b the braking power. Due to the limitation in the brake actuation, it is impossible to dissipate the excesses of power during the transient, since it would mean to precisely follow the $P_t(t) - P$ signal with the brake power $P_b(t)$; hence an integral strategy has been studied. Let us suppose to force the speed to be $\omega = \bar{\omega}$, and to close all the big valves (respecting their actuation constraints) keeping the opening degree of the control valve HT-V3 constant, such as at the end on the transient $P_t = P$. Repeating this operation several times, with different load values, and with the most critical thermodynamic boundary conditions (winter, see Section 6). For each simulation we integrate the signal $P_t(t) - P$ and, eventually, we calculate the mean value of all the integration results, obtaining a normalised value (with respect to the P_{max}) $I_p = 5.6336$ s. It represents the average amount of energy to be dissipated by the brake during the transient in winter conditions, provided that $\omega = \bar{\omega}$. Of course, in practical cases, this condition is translated into the hypothesis that the control is good enough to guarantee small speed displacements from the set-point value $\bar{\omega}$.

During the control system startup there is an unavoidable speed overshoot depending on the delay that always occurs between the instant when the grid is disconnected t_0 and the one t^* in which the velocity crosses the threshold. Thus, when the brake starts working at the time t^* it has to be $P_b(t^*) > P_t(t^*)$, in order to avoid additional over-speed. In island-mode P_t is no longer measurable, but considering that $P_t(t^*) \approx P_t(t_0)$ the previous condition becomes $P_b(t^*) > P_t(t_0)$ [1]. Moreover, the brake control signal $P_b(t)$, because of the latency time T_c, must avoid to request the closures of previously closed and reopened switches, since it would take 4 s, that, compared to the transient dynamic, is a large time interval. A simple and effective signal to satisfy all the conditions is a two step signal, whose integral is I_p, whose first step highness P_1 is the smallest one, compatible with the brake resolution, such as $P_1 > P_t(t_0)$ and whose second step highness P_2 is the half of the first one, rounded by excess according to the brake resolution. Using this solution, it is guaranteed that each switch is required to close only once. Nevertheless, a degree of freedom on the duration t_1 of the first step with respect to the one t_2 of the second step remains. Via simulation it has been shown that, using a single step with $P_1 > P_t(t^*)$, a strong speed undershoot is obtained, in particular in winter conditions, when the produced power is big and, consequently, P_1 is big too. Keeping in mind that the role of the second step is to reduce the speed undershoot, its duration can be parameterised with respect to a coefficient α, increasing or reducing it depending on the produced power. The coefficient α can be computed as follows

$$\alpha = \max\left\{\alpha_{min}\frac{P_{max}}{P_t},\ 1\right\}, \tag{7}$$

where it has been experimentally found that a good value is $\alpha_{min} = 0.7$. The corresponding braking times are

$$\begin{aligned} t_1 &= \alpha\left(\frac{P_t(t_0)}{P_{max}}\right)^2 \frac{I_p}{P_1} + t_{on} - t_{off} \\ t_2 &= (1-\alpha)\left(\frac{P_t(t_0)}{P_{max}}\right)^2 \frac{I_p}{P_2}, \end{aligned} \tag{8}$$

where the therm $\left(\frac{P_t(t_0)}{P_{max}}\right)^2$ is an adaptive term that compensates the thermodynamic boundary conditions: when the plant produces less power (non-winter working conditions) the braking times are reduced. Also note that the therm $t_{on} - t_{off}$ is needed in order to compensate the asymmetrical delay of the brake switches. For the same reason, when passing from the first to the second step, the switches that open, have to be delayed of $t_{on} - t_{off}$.

5.2 Valve Control

The valve control is an adaptive PI plus FF strategy with saturation of the control variable. Since the winter conditions are the most critical and the risk of instability is greater, the FF action has been calibrated in these conditions. Experimentally, the mechanical power generated by the plant when all the valves, except HT-V3, are closed, has been computed for different degrees of opening of the HT-V3 valve, to generate the FF curve h_{FF}, as Figure 2 shows. The FF action is always active, since the stand-alone load P is known, through the FF curve the control valve is always (both in island and normal operation modes) kept open according to the following equation:

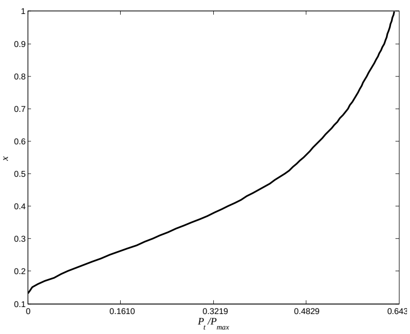

Fig. 2. The FF curve normalised with respect to the winter power P_{max}.

$$\bar{x}_{FF} = \frac{p_{HT_{in_w}} - p_{out_w}}{p_{HT_{in}} - p_{out}}\sqrt{\frac{p_{out}}{p_{out_w}}}h_{FF}(P), \tag{9}$$

where the term that multiplies $h_{FF}(P)$ is an adaptive term that compensates the FF action when the system works in conditions different from the winter ones and the subscript w stands for winter. Of course, the adaptive term could be calculated analytically, but it would require the fluid database and complex calculation, whereas the proposed equation is very simple and can be easily implemented on a PLC, moreover simulations will show that it gives effective results. Note that, keeping HT-V3 valve opened at the $\bar{x}_{FF}$ value does not influence the power produced for the grid because HT-V3 is in parallel with the big valves completely opened, thus its hydraulic impedance can be neglected (this valve is smaller than the other ones). Conversely, when the plant disconnects from the grid, the control valve is already opened at the right position, thus it partially avoids the problems induced by the slew rate limitation and consequently it also avoids undershoot caused by the great valve excursion required together with the valve sluggish behaviour.

Besides the FF action, the valve is controlled by an event driven PI controller that starts to control if one of the following conditions occurs:

(1) $\bar{\omega} - \omega < 0$ *for the first time*, it avoids excessive speed undershoot;
(2) the brake switch off and $\bar{\omega} - \omega > 0$, it avoids excessive speed overshoot;
(3) a certain time t_{PI} is elapsed, it avoids to trust too much in the FF action that is blind with respect to the actual process output.

Indeed, it would be meaningless to try to compensate the overshoot at the beginning of the transient using HT-V3, when all the big valves are still open. Moreover it would cause the accumulation of the integral action, with consequent oscillations. Further, the discontinuous proportional action, because of the limited valve slew rate, would introduce a phenomenon similar to the integrator wind-up, reducing the stability margin and inducing further oscillation.

The PI, as well as the FF action, is adaptive with respect to the thermodynamic boundary conditions, indeed its equation is:

[1] $Pt(t_0)$ is measurable because, when in normal operation mode, all the produced energy goes into the grid, thus it is easily measured.

$$\bar{X}_{PI}(s) = \frac{p_{HT_{in_w}} - p_{out_w}}{p_{HT_{in}} - p_{out}} \sqrt{\frac{p_{out}}{p_{out_w}}} K_p \frac{1+T_i s}{T_i s} (\bar{\omega} - \Omega(s)), \tag{10}$$

where T_i is the integral time constant, K_p is the proportional one and the upper-case letter means Laplace transform. It has been shown, via simulations, that the system in winter configuration shows an underdamped behaviour and bigger gain. On the contrary, with higher discharge pressure and/or lower inlet pressures, the system is overdamped and has a smaller gain. Thus, since the PI parameter has been tuned in winter conditions, in order to avoid sluggish behaviour in different conditions the open loop gain is increased.

Eventually the HT-V3 valve controller output will be $\bar{x}(t) = \bar{x}_{PI}(t) + \bar{x}_{FF}(t)$.

Finally, an anti-windup strategy has been implemented since the valve naturally saturates at 0 and 1. A back-calculation solution has been employed, a detailed explanation about this technique can be found in (Visioli, 2003). It has only to be stressed that the saturation is considered each time the whole controller output signal $\bar{x}(t)$ crosses the saturation limits, not only considering the PI output. Via repeated simulations it has been found an effective set of tuning parameter: $K_p = 0.05$, $T_i = 5$ s, $t_{PI} = 1.5$ s, anti-windup tracking time constant $\frac{T_i}{0.9}$ s.

6. SIMULATION RESULTS

The control system has been tested under different thermodynamic boundary conditions. The stand-alone load power normalised with respect to the winter produced power P_{max} is $P = 0.2495$, while the inlet temperatures, normalised with respect to the HT inlet one, expressed in Celsius degrees, are the same in all the tests: $T_{HT_{in}} = 1$ and $T_{LT_{in}} = 0.6210$. The values of the different inlet pressures, normalised with respect to the HT inlet one in winter conditions $p_{HT_{in_w}}$, are in Table 2. The first simulation is done making the control system to work in the worst thermodynamic boundary conditions, the winter ones. Indeed, in winter the outlet pressure is lower than in the other conditions, consequently, the produced power is higher as well as the risk of a big speed overshoot. Moreover, the plant dynamic is underdamped and the plant dc-gain is higher.

Table 2. Normalised boundary conditions and simulation results.

Test	$p_{HT_{in}}$	$p_{LT_{in}}$	p_{out}	overshoot [%]	undershoot [%]
1 winter	1	0.3734	0.0586	3.69	1.96
2 summer	1	0.3734	0.2033	2.03	0.53
3 hybrid	0.7106	0.3734	0.0586	3.11	0.47

When the grid slips, the speed increases very quickly, as Figure 3 shows, because of the great excess of produced power. The maximum speed is attained when the controller starts to work, because the first braking step is higher than the excess of produced power. It means that the maximum overshoot does not depend on the controller and, in order to reduce its value, the only feasible way is to speed-up the disconnection detection. On the other hand it would mean to increase the risk of a grid slip hoax.

In the meantime the HT-V1, HT-V2, LT-V1 and LT-V2 valves are closing according to their finished slew rate characteristic. The control valve HT-V3 is kept open at the FF value (9), see Figure 3.

The rest of the transient strongly depends on the controller. Because of the integer energy balance criterion used when controlling the brake, an undershoot occurs, but thanks to the two steps braking signal, the undershoot value is always lower than the overshoot one. It is interesting to note that the produced power remains high during the first part of the first braking step. It depends on the high order dynamic introduced by the dead volumes, on the limited slew rate of the valves HT-V1, HT-V2, LT-V1 and LT-V2, that are closing, and on the strongly nonlinear valve hydraulic behaviour (3).

When the speed crosses for the first time the set point value, the PI controller starts to work and the speed quickly reaches the set point value as well as the produced power reaches the stand-alone load one.

It is worth stressing that, despite the strong actuator limitations, the proposed control algorithm allows a speed displacement lower than 4 % from the FF value in view of a load reduction grater than 75 %, see Table 2.

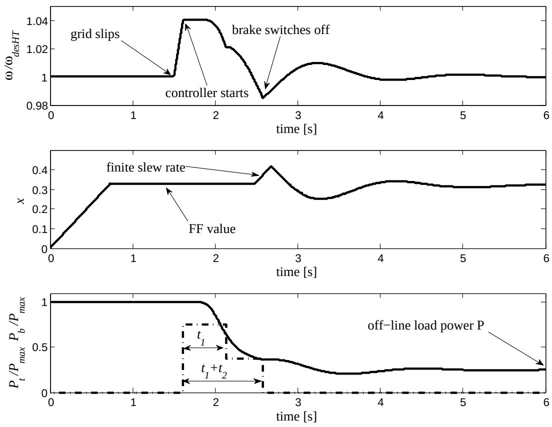

Fig. 3. Normalised speed ω, HT-V3 position x, normalised generated power P_t and normalised braking power P_b (dash-dot line) with winter boundary conditions.

Besides the winter conditions, the summer ones are the other nominal situation under which the control system is required to work. In these conditions the turbine outlet pressures are strongly increased (plus 247 % with respect to the winter case) and, consequently, the produced power is lower. Moreover the plant dynamic is sluggish and a new parameter tuning would be necessary without an adaptive controller in order to achieve satisfactorily performance. On the contrary, using the proposed solution, the controller is automatically adapted to the new plant dynamics greatly simplifying the tuning procedure. It is worth noting that, using the adapted tuning parameters in winter configuration, would lead to instability because of the faster dynamic and the higher dc-gain.

Also the valve FF action (9) is automatically increased using the adaptive correction therm, indeed, in winter it is slightly over 0.3, while in summer it is slightly under 0.8, see Figure 4. Note that, without this correction, the PI would have been to control a long run valve excursion causing disastrous results because of the valve limited slew rate.

Finally the brake signal is a single step one, indeed, due to the low produced power, the coefficient α (7) saturates at 1. In this case the PI starts to work when the brake

switches off because no undershoots occur during the braking action.
As expected, the control of the plant in summer conditions is easier and the obtained undershoot and overshoot are lower, see Table 2, even if the speed response appears to be slightly more sluggish. On the contrary, the produced power quickly reaches stand-alone load one, see Figure 4.

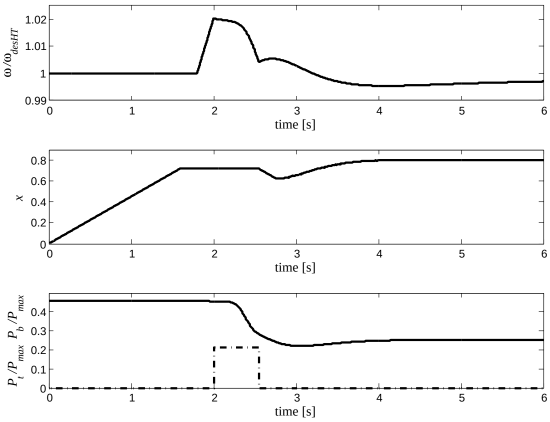

Fig. 4. Normalised speed ω, HT-V3 position x, normalised generated power P_t and normalised braking power P_b (dash-dot line) with summer boundary conditions.

The third simulation is done under hybrid working conditions, quite far from a reasonably expected plant working conditions, thus very useful to test the control system robustness and adaptive capability. The obtained results are quite satisfactorily, see Table 2, and Figure 5 shows how the adaptive strategy works properly when regulating the PI parameters (10), the FF value (9) and the energy dissipation through the braking times t_1 and t_2 (8).

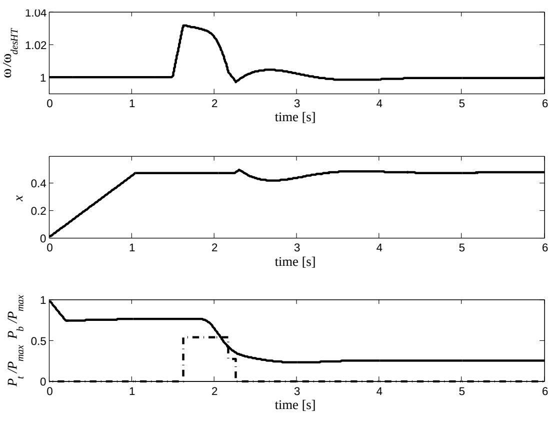

Fig. 5. Normalised speed ω, HT-V3 position x, normalised generated power P_t and normalised braking power P_b (dash-dot line) with hybrid boundary conditions.

7. CONCLUSION

An effective PI adaptive control strategy for a section of an ORC power plant has been developed.
The strong limitations due to the control devices are taken into account when designing the control system. The resulting control algorithm meets the industrial requirements of simplicity, thus it can be implemented on a PLC. Moreover, it is clear that most of the limits of the modeled control system depends on the actuators. Thus, exasperating the search of performance by increasing the complexity of the control strategy, while the limitations lie in the physical limits of the actuator, is not sensible.
The plant has been modeled in an object oriented environment and, by simulating, a set of suitable control parameter has been found.
The proposed solution is adaptive with respect to the thermodynamic boundary conditions and work properly under strongly different turbine inlet and outlet pressures. The effectiveness of the proposed solution has been proven via simulation.

8. ACKNOWLEDGEMENTS

The authors gratefully thank professor Antonio Visioli and professor Costante M. Invernizzi from University of Brescia for their valuable suggestions.

REFERENCES

Åström, K.J. and Hägglund, T. (2006). *Advanced PID Control*. ISA Press, Research Triangle Park, USA.

Åström, K.J., Hägglund, T., Hang, C.C., and Ho, W.K. (1993). Automatic tuning and adaptation for PID controllers - a survey. *Control Engineering Practice*, 1, 699–714.

Casella, F. and Leva, A. (2003). Modelica open library for power plant simulation: design and experimental validation. In *Proceedings of the 3rd International Modelica Conference*, 41–50.

Casella, F. and Leva, A. (2009). ThermoPower library. https://modelica.org/libraries/ThermoPower.

Casella, F. and Richter, C. (2008). ExternalMedia library. http://www.modelica.org/libraries/ExternalMedia.

Colonna, P. and Van der Stelt, T. (2004). FluidProp: a program for the estimation of thermophysical properties of fluids. http://www.fluidprop.com/.

Colonna, P. and van Putten, H. (2006). Dynamic modeling of steam power cycles. part i - modeling paradigm and validation. *Journal of Applyed Thermal Engineering.*

Colonna, P. and van Putten, H. (2007). Dynamic modeling of steam power cycles. part ii - simulation of a small simple Rankine cycle system. *Journal of Applyed Thermal Engineering.*

Dixon, S. (2005). *Fluid Mechanics and Thermodynamics of Turbomachinery.* Butterworth-Heinemann, 5th edition.

Hung, T., Shai, T., and Wang, S. (1997). A review of organic rankine cycles (ORCs) for the recovery of low-grade waste heat. *Energy*, 2, 661–667.

Quoilin, S., Aumann, R., Grill, A., Schuster, A., Lemort, V., and Spliethoff, H. (2011). Dynamic modeling and optimal control strategy of waste heat recovery Organic Rankine Cycle. *Journal of Applyed Energy.*

Visioli, A. (2003). Modified anti-windup scheme for PID controllers. *IEE Proceedings – Control Theory and Applications*, 150(1), 49–54.

Visioli, A. (2006). *Practical PID Control.* Springer, London, UK.