

Article

Model-Based Optimization of a Sliding Vane Rotary Pump for Micro-Organic Rankine Cycle

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Abstract: The residential sector is one of the main sectors responsible for the atmospheric emission of CO₂. Hence, a significant effort is required to develop technological solutions to enable decarbonization. The integration of Organic Rankine Cycle (ORC)-based units with renewable sources at a micro-scale of cogeneration units is commonly believed to be one of the most important technological alternatives. Indeed, an ORC-based unit allows the exploitation of low-temperature heat sources in the production of electricity. The low power scale of this application (1–5 kW) and the severe operating conditions call for the reliable and proper design of components. Particularly critical is the pump, as the experimental analyses available in the literature show its efficiency rarely exceeds values of 0.3. The most suitable technology is volumetric, and among those available, the sliding vane types are interesting candidates. However, low efficiency leads to a significant erosion of the power produced by the expander, limiting the achievement of high-efficiency values. What is more, in the literature, there is a lack of development of optimization strategies to improve the performance of this machine. To fill this knowledge gap, in this present paper an optimized sliding vane rotary pump was designed. Thanks to a comprehensive experimentally validated model, the pump performance was assessed for a wide range of operating conditions. Results confirmed that a disk-shaped configuration also ensures the best efficiency is achieved for small-scale pumps. Moreover, the model allowed for a detailed analysis of efficiency, evaluating the volumetric, fluid dynamic and mechanical behaviors. Results demonstrated that the weakest point was the mechanical efficiency, which was between 0.45 and 0.55. The best configuration was that involving four blades, the adoption of graphite and a clearance gap between the rotor face and casing of 10 μm. These design solutions improved efficiency by up to 25%, with a maximum value equal to 0.50, which is close to double with respect to the usual values. A final remark concerns the operating robustness of the machine, as the efficiency demonstrated weak variations even when wide operating conditions were considered.

Keywords: sliding vane rotary pump design; Organic Rankine Cycle (ORC)-based power unit; mechanical design; model-based design optimization; micro-cogenerative systems



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1. Introduction

The residential sector significantly impacts CO₂ emissions, producing about 11% of total CO₂ emissions in the E.U. [1]. To decarbonize this sector, systems and technologies capable of exploiting the potential of renewable energy sources must be implemented. Solar energy is surely the most promising option since it is widely available worldwide and can be easily converted into electrical and thermal energy [2,3]. Many solutions

have been proposed to address these challenges. Photovoltaic (PV) and solar thermal panels can directly and efficiently convert solar radiation into electricity or thermal energy, respectively [4]. However, the intermittent and non-homogeneous nature of the solar source necessitates energy storage systems and control strategies, which remain expensive and rely on components (batteries, inverters, etc.) that use rare and harmful materials [5]. In this context, power units based on the Organic Rankine Cycle (ORC) represent another promising technology. In fact, they can be used as power units or combined heat and power (CHP) systems, recovering waste heat from renewable energy sources such as solar and biomass [6]. Their appeal for small power applications (1–5 kW), like those required in the residential sector, stems from their adaptability to various heat sources, simple structure, high reliability, and proven technological maturity at medium-to-large scales [7,8]. Moreover, micro-ORC units mitigate the losses associated with the transportation and distribution of electric energy, improving grid reliability and sustainability [9]. Compared to simple ORC power units, CHP systems have higher conversion efficiency and provide greater flexibility in using solar energy based on energy demand [10]. Additionally, micro-ORC power systems can be used in combination with heat pumps to realize Carnot batteries, offering a reliable and low-cost solution for energy storage in buildings [11].

ORC-based power units have been widely investigated in the literature to optimize their performance. Indeed, a small temperature gap between the heat source and the environment requires a major effort to improve their efficiency. The main focus has been directed toward working on the fluid selection [12,13], as well as on the optimization of design parameters [14,15] and the implementation of innovative cycle configurations [16]. Furthermore, optimized designs of plant components [17,18], as well as suitable control strategies for the power unit [19,20], have been proposed to improve the ORC efficiency. With respect to mechanical components, expanders have been extensively studied, being crucial in converting fluid enthalpy into useful work [21]. Volumetric expanders have also been considered and when employed in ORC units and combined with heat pumps, realize Carnot batteries [22]. For such systems, the use of reversible machines as compressors or expanders depending on the specific operating mode (charging or discharging) has also been proposed. The main aim is to optimize the overall efficiency and reduce the number of components [23].

Nevertheless, the efficiency of ORC-based units, as stand-alone systems or in Carnot battery concepts, is also influenced by the working fluid pump, which has not been investigated deeply in the literature as have expanders. In fact, the impact of the pump efficiency on the overall efficiency is not negligible, as happens in Rankine Cycles. The energy consumption of the pump is significant due to the low critical temperature of organic fluids, which negatively affect the Back Work Ratio (BWR), i.e., the ratio between the pump consumption and expander output power [24]. The technology considered for the pump plays a pivotal role as well. In fact, the pump efficiency results from the combined effect of fluid dynamics and mechanical losses, which depend on the pump technology (centrifugal or volumetric pumps) and geometrical features (pump size, aspect ratio, etc.), as well as the operating point (revolution speed, pressure ratio) [25]. Furthermore, cavitation issues must be prevented since they cause noise and vibrations, reducing the efficiency and, eventually, blocking the pump [26]. Several studies have already been undertaken to optimize the use of pumps for ORC-based units. In [27], a diaphragm metering pump was investigated. Results showed that the mass flow rate was independent of the outlet pressure, but the BWR increased at lower evaporation temperatures. Moreover, the effects of four different organic fluids on the performance of the diaphragm pump were assessed in [28]. The isentropic efficiency was found to be very variable (between 57.2% and 93.5%) depending on the fluid and the operating conditions (volumetric flow rate and pressure

difference). A multi-diaphragm pump integrated into a small ORC unit (4 kW_{el}) was also characterized experimentally in [26]. The maximum global efficiency reached by the pump was in the range of 45 to 48%, with a maximum volumetric efficiency of 95%. Furthermore, in [29], a mathematical model capable of calculating even incipient cavitation conditions of a multi-diaphragm pump integrated into an ORC-based unit was proposed. Other types of pumps have also been explored. The performances of centrifugal and piston pumps operating in a 10 kW prototype were assessed in [30]. The centrifugal pump ensured higher mass flow rates and isentropic efficiencies, but the piston pump reached a better global efficiency, resulting in lower energy absorption. In [31], control strategies for the working fluid in a multistage centrifugal pump and the cooling water pump of an ORC condensing system were investigated under variable condensation conditions. The performance of the working fluid pump seriously impacted the BWR, which increased from 0.05 to 0.19 when functioning in winter. Rotational speed played an important role, with pump efficiency varying from 14% to 39% when the speed increased from 1600 to 2300 rpm. In [32], the characteristics in terms of mechanical efficiency, mass flow rate, and electricity consumption were evaluated experimentally for a multistage centrifugal pump. An increase in pump speed resulted in higher variation ranges in the flow rate and outlet pressures. Under these conditions, the maximum mechanical efficiency reached almost 62%.

Despite the efforts already made in the literature to optimize the design and use of pumps for ORC-based units, low pump efficiency still represents one of the main limitations to achieving high-efficiency values. This is particularly limiting for small-scale ORC-based units, for which the BWR increases when the power of the unit decreases. What is more, in the literature, there is a lack of development of optimization strategies for pumps conceived for these applications. To fill this knowledge gap, in this present research, a comprehensive analysis was carried out to outline an optimized design of a sliding vane rotary pump (SVRP) conceived to be installed in a 1 kW ORC-based unit of a micro-cogeneration system. Due to the complex thermo–fluid dynamic phenomena characterizing this type of pump and also the expected limits of the mechanical efficiency, which decreases when the size of the machine is reduced, a mathematical model adopting a zero- and mono-dimensional approach was adopted. The model was validated on experimental data collected on a larger SVRP analyzed by the authors in a previous work [33]. Once validated, the model was used to characterize the SVRP performance, finding the weakest point. Optimal design approaches were then found and inspected to provide a significant improvement with respect to the actual body of knowledge.

2. Materials and Methods

2.1. Theoretical Model

Volumetric machines are characterized by complex thermo–fluid dynamic phenomena. Hence, to support the design analysis, it is fundamental to develop a reliable and comprehensive mathematical model. What is more, in the literature, there are no codified design approaches to assist in the sizing phase. For this reason, experimentally validated software platforms are essential to provide an effective design optimization of the machine. This explains the wide interest observed in the literature about the modeling of volumetric pumps. The authors developed a comprehensive mathematical model of a sliding vane rotary pump employed in an Organic Rankine Cycle (ORC)-based power unit in a previous work [34]. The model was built following an integrated zero (0D) and mono (1D)-dimensional thermo–fluid dynamic analysis in the GT-Suite™ Environment. The model structure is reported in Figure 1.

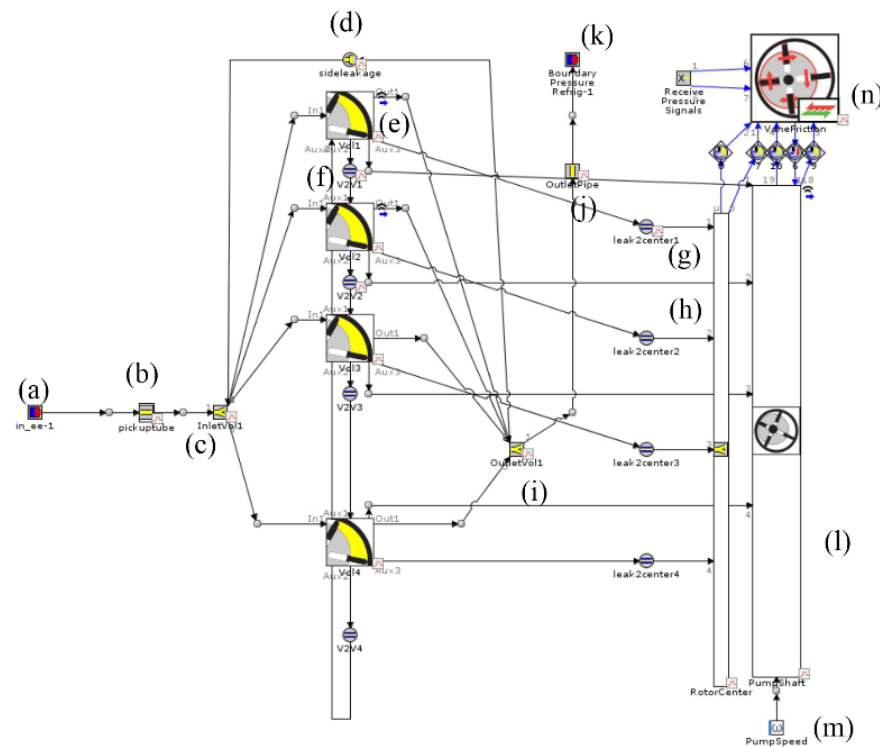


Figure 1. Scheme of the SVRP model.

The following main components of the model can be recognized in Figure 1:

- (a) Intake plenum, in which the inlet boundary conditions, such as inlet total pressure and temperature of the working fluid, can be set;
- (b) Inlet pipe discretized in multiple sub-volumes. For each, the mass, momentum and energy conservation equations are solved. In this way, the complex fluid-dynamic phenomena characterizing the intake phase can be modeled;
- (c) Intake manifold allows the connection of the inlet pipe with the chambers performing the intake phase;
- (d) Side leakage equivalent orifice, which allows for the representation of equivalent leakage losses from the exhaust up to the intake across the gap between the rotor face and the casing (element (1) in Figure 2a). According to the adopted procedure, the leakage area is represented by an equivalent orifice;
- (e) Represents the chamber of the pump, which is treated as a lumped capacity whose volume varies according to the rotation angle;
- (f) This component allows the representation of the leakage between the adjacent chambers across the gap between the blade tip and the stator inner surface (element (2) in Figure 2b). Such leakage is represented through the Poiseuille–Couette approach;
- (g) The same components can also be adopted to represent the leakage between the blade side and the rotor slot with the Poiseuille–Couette equation (element (3) in Figure 2b);
- (h) This element represents the volume between the blade bottom and the rotor side. It is used to evaluate the pressure (and consequently the force F_p) under the blade, which exerts a fundamental role in the definition of the normal action F_n , pushing the blade itself against the stator's inner surface (Figure 2c);
- (i) The exhaust manifold connection with the chamber performing the discharge process and the outlet pipe;
- (j) The outlet pipe, for which the same description of the inlet pipe can be considered;
- (k) Outlet plenum, which represents the holder of the outlet boundary conditions such as outlet static pressure;

- (l) The pump shaft drives the rotation, setting the actual rotating angle according to which the chambers assume a given volume and perform a certain phase (intake or exhaust). Such a component is the holder of the pump geometry and port configuration;
- (m) This component ensures the rotating speed of the shaft as a boundary condition is set;
- (n) Represents a compound to treat the mechanical losses due to dry and viscous friction effects. All sources of losses are considered (between the blade tip and the stator's inner surface, the blade side and rotor slot, the rotor face and machine casing, and the bearings). An amount greater than 95% of the total power lost is due to the dry contact of the blade tip and the stator's inner surface. This source of loss depends on the centrifugal force F_c and the pressure exerted under the blade, as discussed in point (h) and reported in Figure 2c. According to this approach, the dry force between the blade tip and the stator's inner surface can be written as follows (Equation (1)):

$$P_{mech,lost} = f N_V F_N R_a \omega = f N_V (F_c + F_p) R_a \omega \quad (1)$$

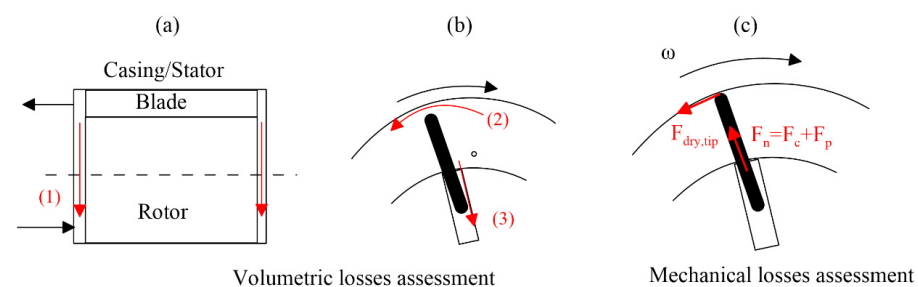


Figure 2. Scheme of the main volumetric (a,b) and mechanical (c) losses: (a) side leakage (1) between rotor face and casing; (b) leakages between adjacent chambers through blade tip (2) and blade side (3); (c) forces acting on the blade.

2.2. Model Features

The mathematical model briefly discussed in this paper was widely described in a previous work [34] by the authors, where more details can be found. Friction losses and leakages are certainly the most critical processes to focus on. The gap between the blade tip and the stator's inner surface is responsible for the leaks among the vanes, while the power absorbed by mechanical friction, which allows mechanical efficiency to be defined, is certainly the friction factor. In real conditions, both processes are also influenced by the oil, which is mixed with the working fluid in a very small quantity (5% in this case). Due to the complexity of a direct determination in the modeling, the two effects have been experimentally identified in such a way as to reduce the error between the experimental and theoretical data. In fact, the measurement of the indicated power and the shaft power allows the identification of the friction coefficient. The flow rate measurement compared with the theoretical value also allows the evaluation of the volumetric efficiency and an equivalent mean gap between the blade tip and the inner stator surface [34].

Indeed, an empirical relation (Equation (2)) was found thanks to the analysis of the experimental results, which showed a linear decrease in the friction factor (f) with rotation speed. The following equation can be then used to represent this parameter, which cannot be identified without an experimentally based analysis.

$$f = -k_1 \omega + k_2 \quad (2)$$

where k_1 is 0.0002 [1/RPM], ω is measured in RPM and k_2 is equal to 0.5153.

The reduction in the friction factor f with revolution speed is explained by the fact that when the pump speed increases, the capability of the oil film to avoid dry contact of the blade with stationary surfaces increases as well. Moreover, the increase in speed leads

to an enhancement in the centrifugal force, ensuring the blade assumes a radial direction and avoiding contact between the blade side and rotor slot (Figure 2c).

This improvement allows a more accurate prediction of power lost due to friction to be made and, consequently, the whole power required by the pump. Indeed, the pump power (Equation (3)) is equal to the sum of indicated power (Equation (4)) and power lost due to friction, basically coinciding with Equation (1) [34].

$$P_{\text{pump}} = P_{\text{ind}} + P_{\text{mech,lost}} \quad (3)$$

$$P_{\text{ind}} = \frac{\oint \sum_{i=1}^{N_v} p_i dV_i}{t_{\text{cycle}}} \quad (4)$$

The indicated power represents the area of the chamber pressure trace as a function of volume over a cycle. Consequently, this power is the one provided by the machine components to the working fluid. Adding to it, the lost mechanical power allows the assessment of the whole pump shaft power.

Therefore, the model can evaluate the pump efficiency (Equation (5)) given by the ratio between the hydraulic power and the global one [34].

$$\eta_{\text{pump}} = \frac{P_{\text{hydr}}}{P_{\text{mech}}} = \frac{Q \Delta p}{P_{\text{mech}}} \quad (5)$$

where Q is the effective flow rate delivered by the pump, which differs from the theoretical one due to the volumetric losses evaluated by the model (Equation (6)), which allows the volumetric efficiency to be defined (Equation (7)) [34].

$$Q_{th} - Q = V_{\text{disp}} \omega - Q \quad (6)$$

$$\eta_{\text{vol}} = \frac{Q}{Q_{th}} = \frac{Q}{V_{\text{disp}} \omega} \quad (7)$$

It is important to observe that Equation (5) can be decomposed in different contributions by multiplying and dividing for proper quantities [34]. Indeed:

$$\eta_{\text{pump}} = \frac{Q \Delta p}{Q_v \Delta p} \frac{Q_v \Delta p}{P_{\text{ind}}} \frac{P_{\text{ind}}}{P_{\text{mech}}} = \eta'_{\text{vol}} \eta_{\text{ind}} \eta_{\text{mech}} = \eta_{\text{fluid}} \eta_{\text{mech}} \quad (8)$$

Equation (8) represents the efficiency chain showing the different sources of losses:

- η'_{vol} represents a volumetric efficiency evaluated differently from Equation (5). Indeed, in Equation (8), Q_v represents the flow rate effectively circulating inside the chamber (but not necessarily completely sent due to the presence of leakages), while in Equation (5), the theoretical flow rate Q_{th} is used as a reference. Indeed, Q_v and Q_{th} could not be coincident, as some volumetric losses could be lost without entering all chambers (and consequently receiving work). Hence, as Equation (8) represents the impact of volumetric efficiency on the machine power losses for the adopted approach, Q_v is considered;
- η_{ind} represents the indicated efficiency, and is evaluated as the ratio between the indicated power and mechanical one. In the form in which it is written, such contribution allows the extra power received by the fluid due to an increase in pressure caused by the over- or under-pressure phenomena occurring inside the chamber to be expressed. These determine the growth of power provided by the pump components to the working fluid, keeping the useful effect of the machine constant. It is worth noting that, due to the difficulty in evaluating Q_v in a closed form, η'_{vol} and η_{ind} are

- multiplied, returning the fluid–dynamic efficiency η_{fluid} which considers the effects of the volumetric and over- and under-pressure phenomena on the whole pump power;
- η_{mech} represents the mechanical efficiency and describes the impact of mechanical losses on the whole pump power.

The model evaluates all these sub-contributions, allowing the weaknesses of the machine to be understood and, consequently, how to direct the design to improve the pump performance.

2.3. Model Validation

In order to assess the improvement in the model accuracy with respect to the previous version [34], the model presented in this paper was validated against the experimental data reported in [34]. The experimental analysis refers to an SVRP employed in an ORC-based power unit for a Waste Heat Recovery application in a Sliding Vane Rotary Air Compressor, and the heat source was represented by the lubricant oil. The reference pump geometry is reported in Table 1, whereas the considered working fluid was R236fa and the number of blades was 4.

Table 1. SVRP geometry.

Parameter	Value
Stator diameter [mm]	32.5
Rotor diameter [mm]	30
Pump length [mm]	30
Blade length [mm]	8.5
Blade thickness [mm]	3
Blade weight [g]	5.8

Table 2 shows the experimental data. It can be seen that the mass flow rate provided by the pump increased from 0.071 to 0.12 kg/s as the pump speed rose from 800 to 1300 RPM. The pump’s maximum power was 289 W, whereas the minimum was 168 W.

Table 2. Experimental data.

Parameter	Measurement Uncertainty	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
Mass flow rate [kg/s]	±9% [kg/s]	0.11	0.103	0.094	0.086	0.085	0.116	0.071	0.119
Intake temperature [°C]	±0.5 °C	24.2	23.6	22.3	20.4	20.2	25.7	18.9	29.5
Exhaust temperature [°C]	±0.5 °C	25	24.7	23.2	21.4	21.2	26.5	20	30.4
Intake pressure [bar]	±0.1 bar	3.4	3.3	3.3	3.2	3.3	3.7	3.2	3.7
Exhaust pressure [bar]	±0.1 bar	12.4	12.1	11.4	10.8	10.6	13.4	9.9	12.4
Pump speed [RPM]	±1 RPM	1200	1100	1000	900	900	1250	800	1300
Shaft power [W]	±3 W	289	265	230	198	179	275	168	272

In Figure 3, the comparison between the experimental and theoretical values was carried out for the mass flow rate and pump power. These quantities were selected to assess the model accuracy because a proper prediction of mass flow rate allows the fluid dynamic behavior of the machine and, consequently, the leakage losses to be predicted. This assessment was also fundamental for the analysis of the indicated power, which made a tighter estimation of mechanical losses and the calculation of the pump power lost, which was the other validation quantity. In Figure 3a, it can be seen that the theoretical predictions coincided with the experimental data, showing a fully satisfactory prediction.

The consequence of this was the low relative Root Mean Square Error (*RMSEr*) defined in Equation (9).

$$RMSEr = \sqrt{\frac{1}{NP} \sum_{i=1}^N \left(\frac{x_i - \hat{x}_i}{\hat{x}_i} \right)^2} \quad (9)$$

where *NP* is the number of populations, x_i is the *i*-predicted quantity and $\hat{x}_i$ is the corresponding experimental value. For mass flow rate, *RMSEr* was 0.4%. A good value for *RMSEr* was also achieved for the power prediction, being equal to 2.3%. Hence, the results suggest a significant improvement in the model accuracy with respect to the previous version, allowing a more detailed representation of the pump behavior to be provided and enabling a more effective optimization of the pump design.

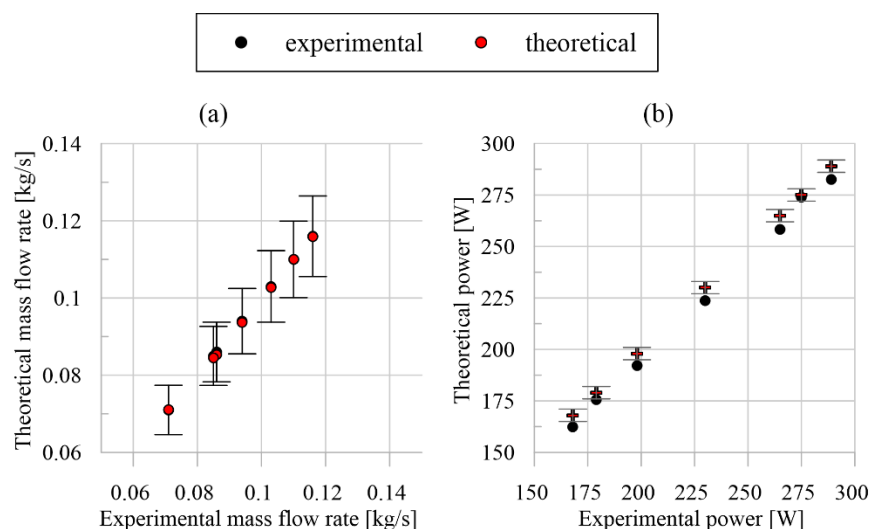


Figure 3. Comparison between the experimental and theoretical mass flow rate (a) and pump power (b).

3. Results

3.1. Pump Design and Characterization

The validated model represents a fundamental tool for refining the pump design, which deserves a preliminary theoretical task. Indeed, the volume capability of the pump should be defined according to the functional requirement of the application in which the pump is involved. It is important to observe the application of the pump for the case at hand in regard to a micro ORC-based unit with a flow rate significantly lower than that considered in the validation case was used. A further difference was the working fluid, as was used in the novel design R245fa was considered, that being the fluid adopted in the considered ORC application. With respect to the pressure level, the inlet pressure was significantly smaller, whereas the outlet pressure was comparable. This means that the designed pump must withstand a larger pressure difference. As a design point, the one with the maximum expected mass flow rate (corresponding to the maximum thermal power available at the hot source) was considered. Given the micro-ORC unit had a volumetric expander, the maximum pressure grew linearly with the mass flow rate [35]. Therefore, the maximum (pump exhaust) pressure was considered. The design point is reported in Table 3. As for design revolution speed, a value of 1000 RPM was considered a good compromise between the need to reduce the speed to contain the friction effect without reaching too low a value, which could compromise the lubricating oil film.

Table 3. Pump design point and requirements.

Parameter	Value
Inlet pressure [bar]	1.4
Outlet pressure [bar]	11.5
Inlet temperature [°C]	18
Blades number	4
Mass flow rate [kg/s]	0.06
Volumetric efficiency	0.6
Pump speed [RPM]	1000
Blade density	7200 kg/m ³
Number of blades	4

Given the quantities in Table 3 and considering Equation (7), a volumetric capability equal to 4.4 cm³ was considered. The SVRP assumed different shapes, ensuring the same volumetric capability. Indeed, if the pump presented a predominantly radial dimension, the SVRP assumed a disk shape, whereas if the axial length was dominant, a finger shape was instead adopted. For SVRPs to provide a larger mass flow rate (for heavy-duty engine cooling), the authors observed that the disk shape configuration allowed the best pump performance.

To demonstrate that the same conclusions also apply to the SVRP when an organic working fluid was used and a significantly lower mass flow rate, the validated software platform presented in Section 2 was used.

To set a direct optimization, all possible shapes assumed by the pump with a stator diameter varying between 30 and 34 mm and pump width in a range between 20 and 60 mm were considered. For all cases, the rotor diameter was kept equal to 30 mm. In Figure 4, all these combinations are reported. The only cases considered were those colored in red, which met the volume capability request. Indeed, only the couple (stator diameter–pump length) ensuring a volume capability equal to 4.4 cm³ was achieved were considered. A further refinement was carried out considering the design to allow minimum power to be achieved. It is interesting to observe that the best design value corresponded with the maximum diameter (33 mm) and minimum pump width (17 mm). Therefore, the SVRP presented a stator diameter of 33 mm, a rotor diameter of 30 mm and an axial length of 17 mm. This analysis confirms that for a small-sized pump conceived to work with organic fluid, the disk shape allows the pump power requirement to be minimized.

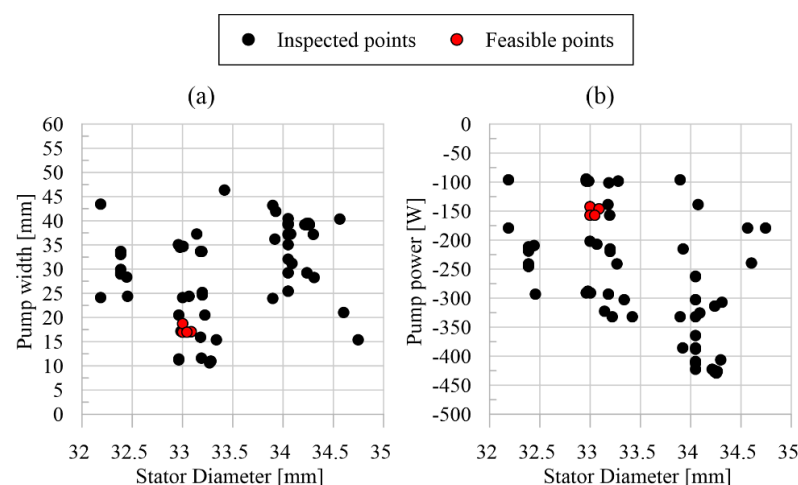


Figure 4. Inspected and feasible design points in terms of stator diameter and pump width (a) and power needed by each pump design (b). Negative values for pump power were considered due to the adopted convention related to considering negative power provided by the machine to the working fluid.

The configuration with four blades was considered for the first design, similar to the prototype used for the validation. In the following, two alternative solutions with five and six blades were considered and compared with the 4-blade SVRP. In Figure 5, the characteristic curves of the SVRP are reported for a delivered volumetric flow rate varying from 1 to 5 L/min and a pressure rise ranging from 5 to 14 bar. The revolution speed varied from 500 to 1500 RPM. The iso-efficiency curves are also reported. Figure 5a shows the typical trend of a volumetric pump characteristic curve with a slope decreasing with the enhancement of pressure. This decrease was mitigated by an increase in pump speed which decreased the impact of leakage losses. Characteristic curves showed that the pump was able to fulfill its functional requirement for the whole operating range. Concerning global efficiency, maximum values were achieved corresponding to flow rates between 4 and 5 L/min.

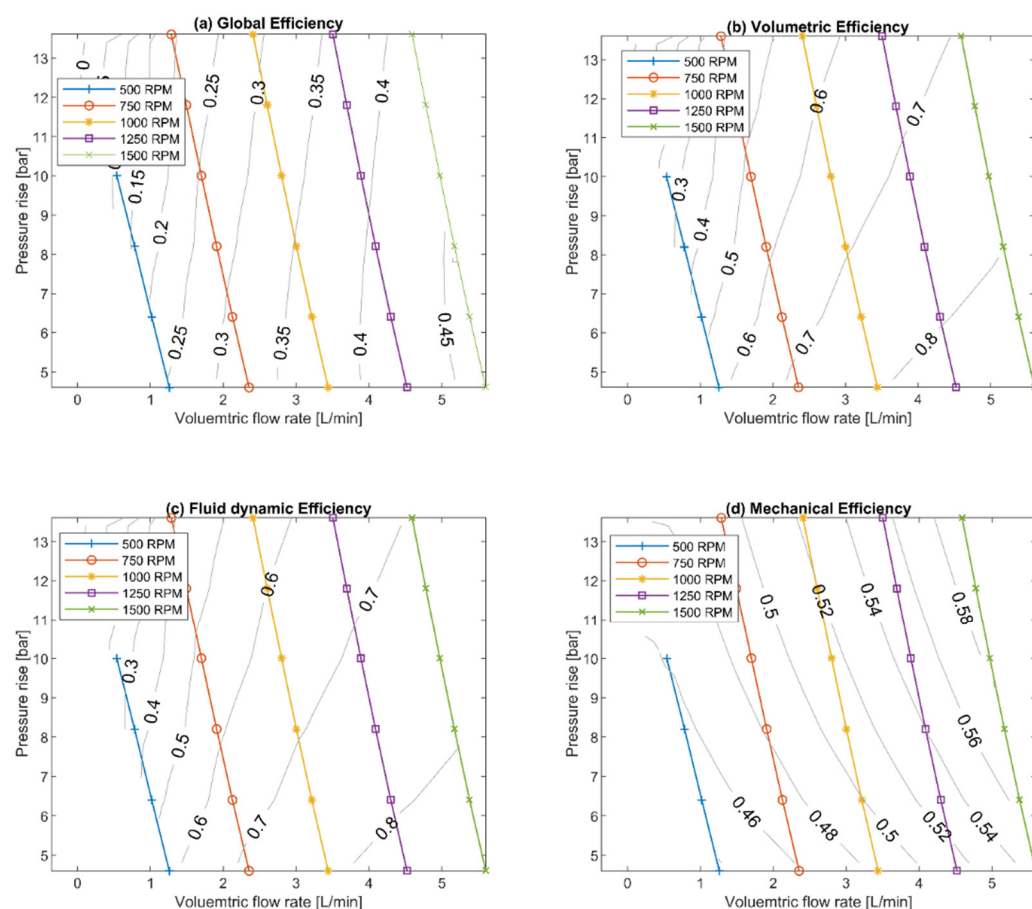


Figure 5. Four-blade SVRP characteristic curves in terms of pump pressure rise as a function of R245fa volumetric flow rate for different pump speeds (500–1500 RPM). Characteristic curves are plotted on iso-level curves of global (a), volumetric (b), fluid dynamic (c) and mechanical (d) efficiency.

In these conditions, the global efficiency was between 0.4 and 0.45. This value represents a significant improvement with respect to the values available in the literature for volumetric pumps of the same size. Indeed, pump efficiency rarely exceeds a value of 0.3, as also confirmed by the experimental analysis carried out by the authors in a previous work [34]. Indeed, this application's results are particularly critical for the achievement of higher efficiency. This is because the hydraulic power assumed low values (Equation (5)), in particular, when the elaborated flow rate was low. Hence, the impact of the friction power was significant (Equation (8)).

Indeed, by observing the volumetric (Figure 5b), fluid–dynamic (Figure 5c) and mechanical (Figure 5d) efficiency, the weakest behavior was mechanical, which had a maxi-

maximum value of the order of 0.54 and a minimum value (0.45) achieved for a low delivery flow rate. Volumetric and fluid dynamic efficiencies assumed satisfying values (higher than 0.7) for a wide operating area (flow rate higher than 3 L/min), but they showed a significant decrease with pressure rise. Indeed, this latter quantity is the main driver of the leakage losses [34], thus penalizing the machine's volumetric and fluid dynamic behaviors. An interesting aspect is that an increase in pressure produced an opposite effect on mechanical efficiency. Indeed, the growth in pressure was related to the enhancement of indicated power (the numerator of mechanical efficiency in Equation (8)). Hence, considering that the global efficiency is the product of fluid dynamic and mechanical efficiencies, the pump's performance depends on pressure rises, as can be seen in Figure 5a, where the efficiency slightly decreased when the pressure rose.

Hence, the main quantity affecting global efficiency is the delivered flow rate. A further interesting aspect revealed by the present analysis is that for low flow rate, the pump presented efficiencies ranging between 0.2 and 0.3, i.e., higher than those typical of micro-ORC-based power units. In fact, commercially available pumps that can be used to pressurize the working fluid of micro-ORC units generally have maximum efficiencies below 25% (Table 4). Hence, the SVRP analyzed in this paper had better performances compared with other commercial options.

Table 4. Commercially available pumps for micro-ORC units.

Manufacturer	Pump Name	Pump Technology	Maximum Speed [rpm]	Maximum Flow Rate [L/min]	Maximum Pressure [bar]	Nominal Power [W]	Efficiency [%]
Wanner Engineering, Inc. (Minneapolis, MN 55403, USA)	Hydra Cell P200 Series	Diaphragm	600	4.3	69	220	20
Fluid-o-Tech (Corsico 20094, Milano, Italy)	Fluid-o-Tech CO/MO 50-200	Rotary vane	1725	4	20	-	-
Alfa Laval Italy (Monza 20900, Italy)	Alfa Laval DuraCirc	Circumferential piston	1000	30	26	-	-
Casappa S.p.A. (Lemignano di Collecchio 43044, Parma, Italy)	Casappa POLARIS 10.1	Gear	4000	4	260	400	15–25
Giordan srl (Schio 36015, Vicenza, Italy)	Giordan 351L	Gear	300	11.7	10	750	20–25
KRAL (Milano 20156, Italy)	Kral K Series	Screw	4000	5.20	16	-	-

The future direction is indicated by the analysis of the different efficiency contributions. As Figure 5 shows, it is essential to reduce friction power losses, which assume values significantly lower with respect to the other contributions in the efficiency chain. In the following sections, strategies to improve the proposed design are discussed.

3.2. Optimization of the Proposed Design

The analysis of the efficiency chain outlined that the optimization strategy should be mainly oriented toward reducing friction power losses. Significant room for improvement is shown in the volumetric and fluid dynamic efficiencies despite the impact of the corresponding losses being lower than that exerted by the mechanical ones. An increase in blade numbers should theoretically provide a reduction in the angular extension of the chambers and, consequently, the angular interval of the pressure difference between adjacent vanes, which is the main driver of volumetric losses. On the contrary, a reduction in the chamber angle with the blade number provided a reduction in its volume and the available time for each chamber for the filling and emptying phases. What is more, the addition of blades provided a detrimental effect on mechanical efficiency due to the growth

in dry contact regions (Equation (1)). To show the effects of the adoption of a higher number of blades, two alternative design solutions with five and six blades were considered. The pump dimensions were slightly varied to keep the same volume capability (4.33 cm^3), thus addressing the fluid volume reduction caused by the addition of blades. In Figures 6 and 7, the global (a), volumetric (b), fluid dynamic (c) and mechanical (d) efficiencies are reported, respectively, for the 5- and 6-blade SVRP. It can be noticed that an increase in blades had a detrimental effect on global efficiency, which diminished with respect to the values obtained in the case with four blades (Figure 5). This is basically due to the strong reduction in mechanical efficiency, whose maximum value decreased to 0.52 and 0.46 for five and six blades, respectively.

Observing the trend reported in Figures 5a, 6a and 7a, it can be noticed that the global efficiency of the four-blade pump was higher over the entire operating conditions. Indeed, the pump achieved a maximum efficiency of 45%, whereas pumps with five and six blades had an efficiency close to 0.35. This result is related to the balance between volumetric, indicated and mechanical efficiencies. Comparing Figure 5b,c, Figure 6b,c and Figure 7b,c respectively, it can be seen that a reduction of 13–15% occurred in the case of five and six blades for volumetric and fluid dynamic efficiency. The decrease became even more severe (up to 20%) for the mechanical efficiency (Figures 5d, 6d and 7d). Moreover, mechanical efficiency increases when the number of blades decreases.

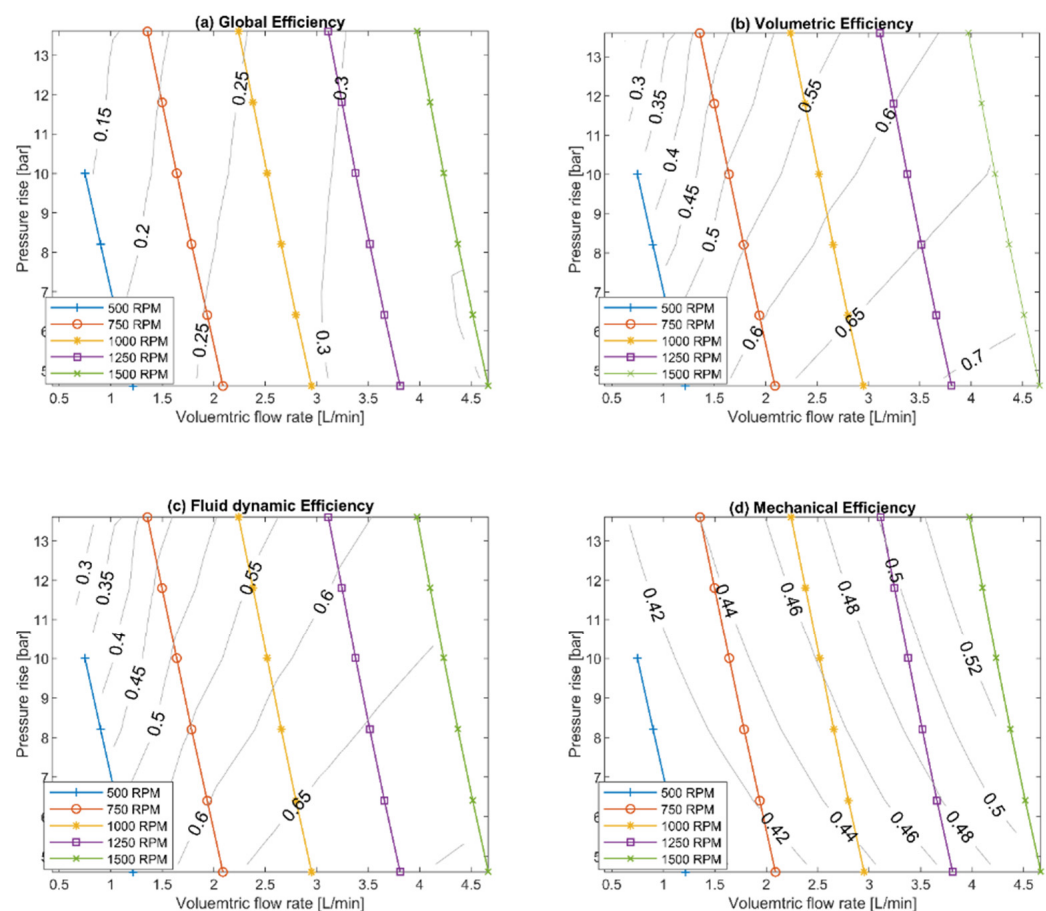


Figure 6. Five-blade SVRP characteristic curves in terms of pump pressure rise as a function of the R245fa volumetric flow rate for different pump speeds (500–1500 RPM). Characteristic curves are plotted on iso-level curves of global (a), volumetric (b), fluid dynamic (c) and mechanical (d) efficiency.

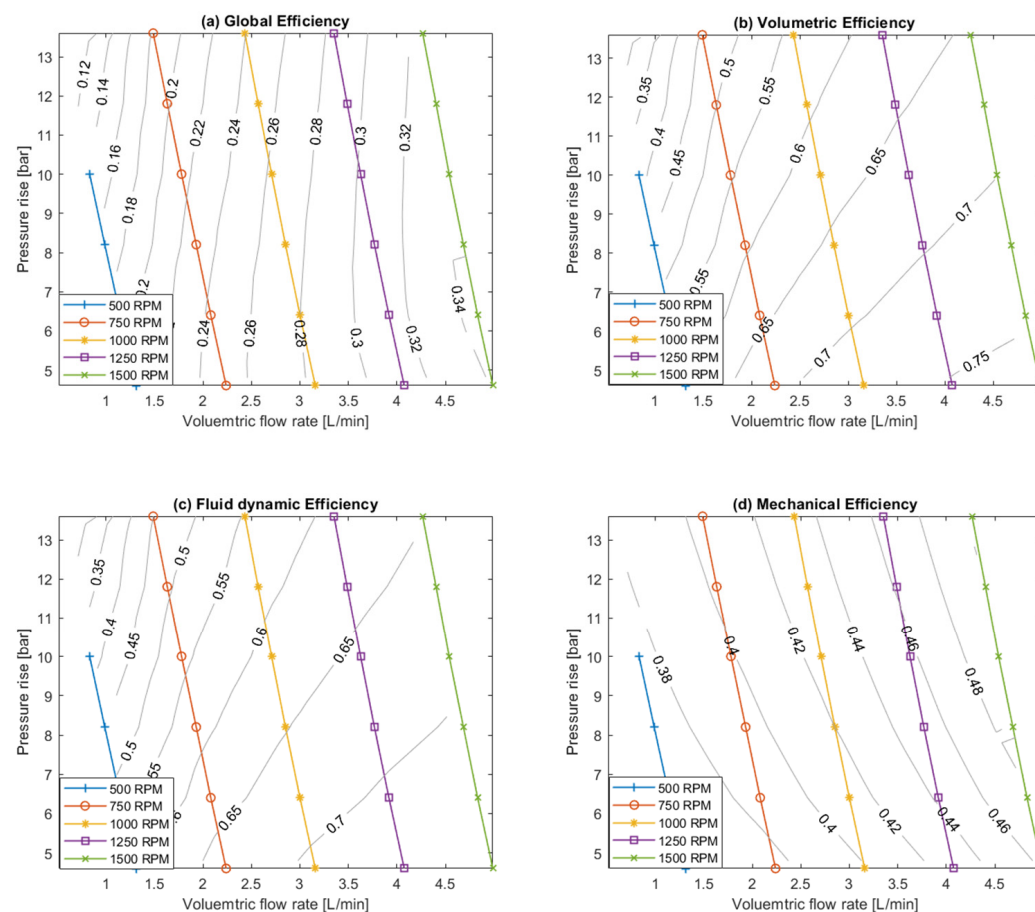


Figure 7. Six-blade SVRP characteristic curves in terms of pump pressure rise as a function of R245fa volumetric flow rate for different pump speeds (500–1500 RPM). Characteristic curves are plotted on iso-level curves of global (a), volumetric (b), fluid dynamic (c) and mechanical (d) efficiency.

This means that the benefits related to a reduction in the angular interval of the pressure difference among adjacent chambers were eroded by a reduction in the chamber volume.

The analysis of leakage losses between two adjacent chambers is reported in Figure 8a, 8b and 8c for four, five and six blades, respectively.

These figures show that an increase in the number of blades allowed the angular difference between the pressure profiles of adjacent chambers to be reduced. In fact, the pressure difference between adjacent chambers drives volumetric losses, worsening the filling of the considered chambers. The higher the angular duration of the pressure difference was, the larger the mass leaked to the following chambers. These figures show that an increase in the number of blades allowed the angular differences between the pressure profiles of adjacent chambers to reduce. Pressure differences between adjacent chambers, in fact, drove volumetric losses, worsening the filling of the considered chambers. The higher the angular duration of the pressure difference was, the larger the mass leaked to the following chambers. Hence, an increase in blades contributes to this direction. Indeed, with an increase in the number of blades from four to six, the angular difference between consecutive chambers decreased from 90° to 60° . Nevertheless, this angular reduction was not sufficient to provide a significant benefit to the volumetric efficiency. Indeed, despite the angular difference being lower, the average mass flow rate between two consecutive chambers was lower than the case of four blades. Indeed, for the design case reported in Table 3, the average flow between a chamber and the following in the sense of rotation was equal to 0.34 g/s, 0.36 g/s and 0.37 g/s for an SVRP with four, five and six blades,

respectively. This means that volumetric losses not only depend on pressure differences but also on chamber filling which improved with the adoption of a larger chamber achieved with a pump with four blades.

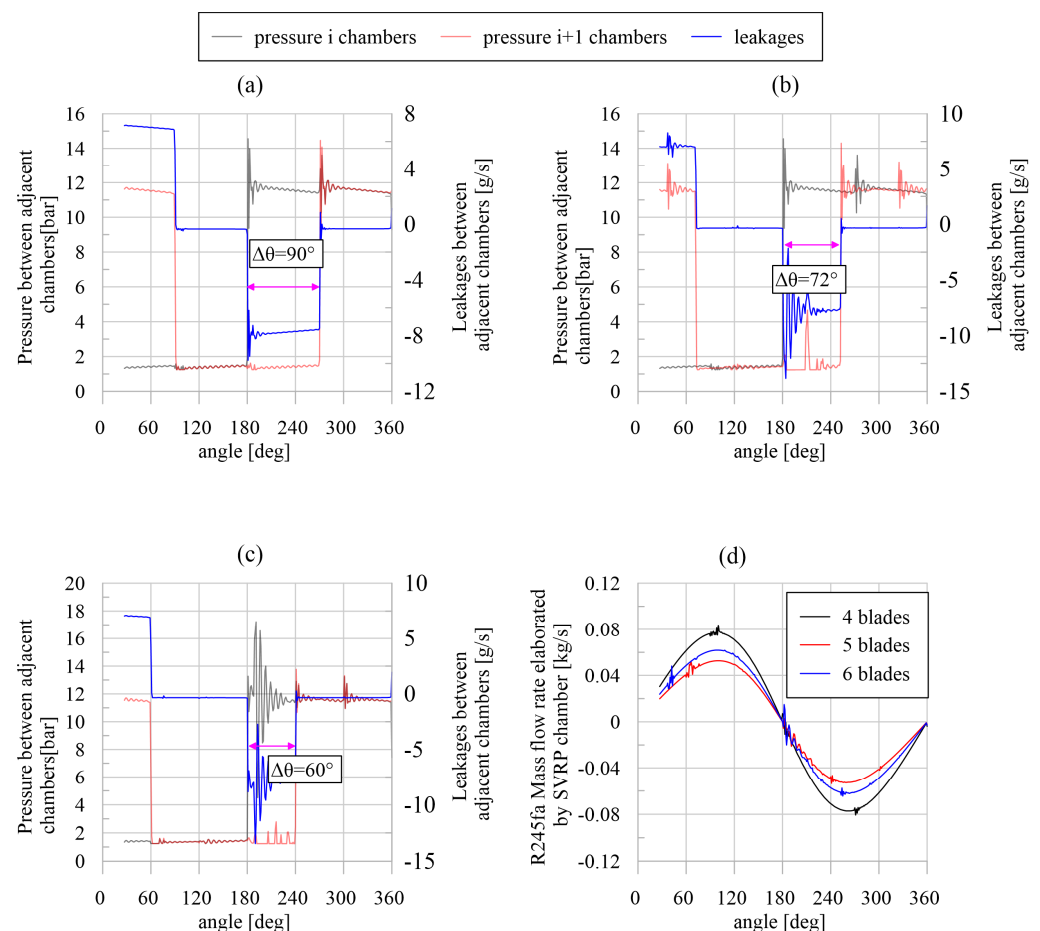


Figure 8. Pressure of adjacent chambers (red and black line) and the corresponding leakage as a function of the rotation angle for an SVRP with 4 (a), 5 (b) and 6 (c) blades. R245fa mass flow rate elaborated by each chamber as a function of blade number and rotation angle (d). Leakage flow rates were considered positive when the flow went from chamber i to chamber $i + 1$ (a–c). In (d), the mass flow rate was considered positive when it entered the chamber.

Moreover, a further benefit of a 4-blade SVRP was that the flow rate delivered (Figure 8d) by each chamber was higher. Hence, the impact of leakage was lower, amplifying the benefits of this solution. In addition to leakages between adjacent chambers, the ones between the casing and rotor face were analyzed for the different cases. In Figure 9, this leakage is reported for the different SVRPs. In the case of the SVRP with four blades, this leakage was higher with respect to the other SVRPs. Indeed, the leakage between the rotor face and machine casing decreased from 20 g/s to 12 g/s when the blades increased from four to six, respectively. This loss was proportional to the mass flow rate provided by the machine, which was higher in the case of four blades. Nevertheless, this value of leakages was evaluated according to a clearance gap equal to 50 μm , which is taken from the original prototype on which the experimental analysis was carried out. Hence, this value can be further reduced as this gap is related to geometrical tolerances. To observe the benefits related to this clearance reduction, the case in which such a gap was equal to 10 μm was considered. In Figure 9, by adopting this clearance gap, the corresponding leakages diminish to 7 g/s, which was more than half the baseline case. If this task is adopted, the

volumetric efficiency of the pump grows from 64% to 78%, justifying the higher cost needed to achieve a lower constructive tolerance.

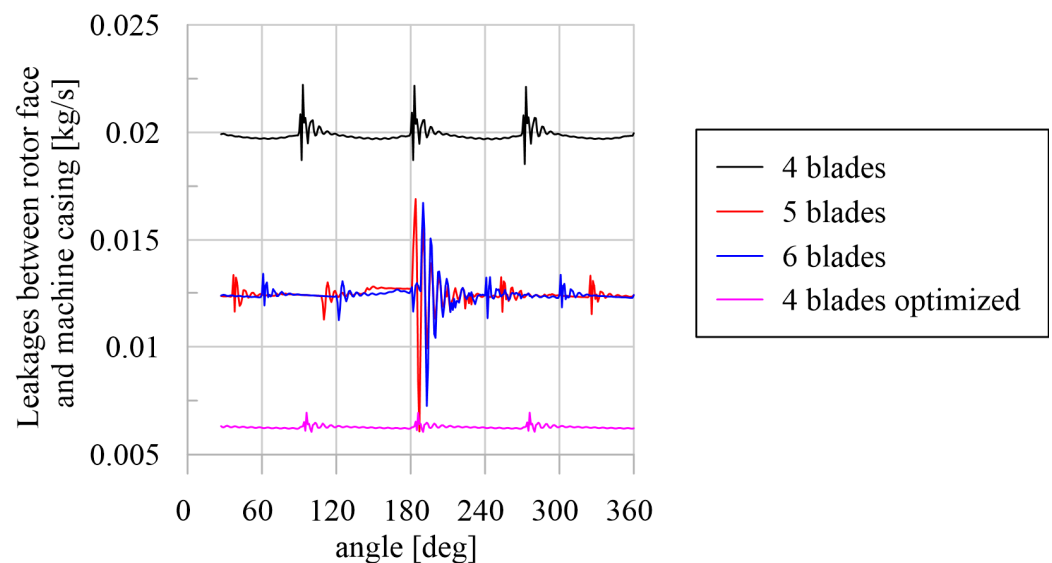


Figure 9. Leakages between the casing and rotor face for a 4-, 5-, 6-blade SVRP and in the case of an optimized 4-blade SVRP.

Hence, an optimized 4-blade SVRP was considered. In addition to the clearance gap reduction, lighter blades made of graphite were also considered to reduce the lost friction power. The applicability of this material was successfully verified by authors in [33], showing that no penalties are observed on volumetric efficiency. To assess all the benefits introduced by the optimized SVRP, its performances were characterized thanks to the theoretical model over the same operating region, considered in Figures 5–7.

It can be noticed that the global efficiency grew significantly in all operating regions (Figure 10a), reaching a maximum value of 0.5. Even in the worst operating conditions, the SVRP efficiency was never below a threshold of 0.3, which is the maximum value observed for the current pumps operating in micro-ORC-based power units. This improvement was made possible by an increase in volumetric and fluid dynamic efficiencies, as reported in Figure 10b and 10c, respectively. Such an efficiency gain is related to the reduction in the clearance gap between the rotor face and casing in the case of volumetric behavior. With respect to fluid dynamic efficiency, the improvement is due to the combined effect of a rise in volumetric efficiency and a reduction in friction losses. This latter effect is due to the adoption of lighter blades, as unequivocally shown in Figure 10d. A final remark should be made about the operating robustness of the machine. Indeed, despite the SVRP operating in a wider operating range, the efficiency varied in a narrow range (0.3–0.5).

To show the impact of pump efficiency improvement on the whole plant behavior, in Figure 11a,b, the power and efficiency of a reference micro-scale ORC-based power unit are reported where a baseline pump (diaphragm pump) was substituted with the optimized 4-blade SVRP. The reference micro-scale ORC-based unit was deeply characterized by the authors in [35]. From Figure 11a, if the ORC-based unit adopted the optimized SVRP, the power was higher for almost the entire analyzed operating range (30–54 g/s).

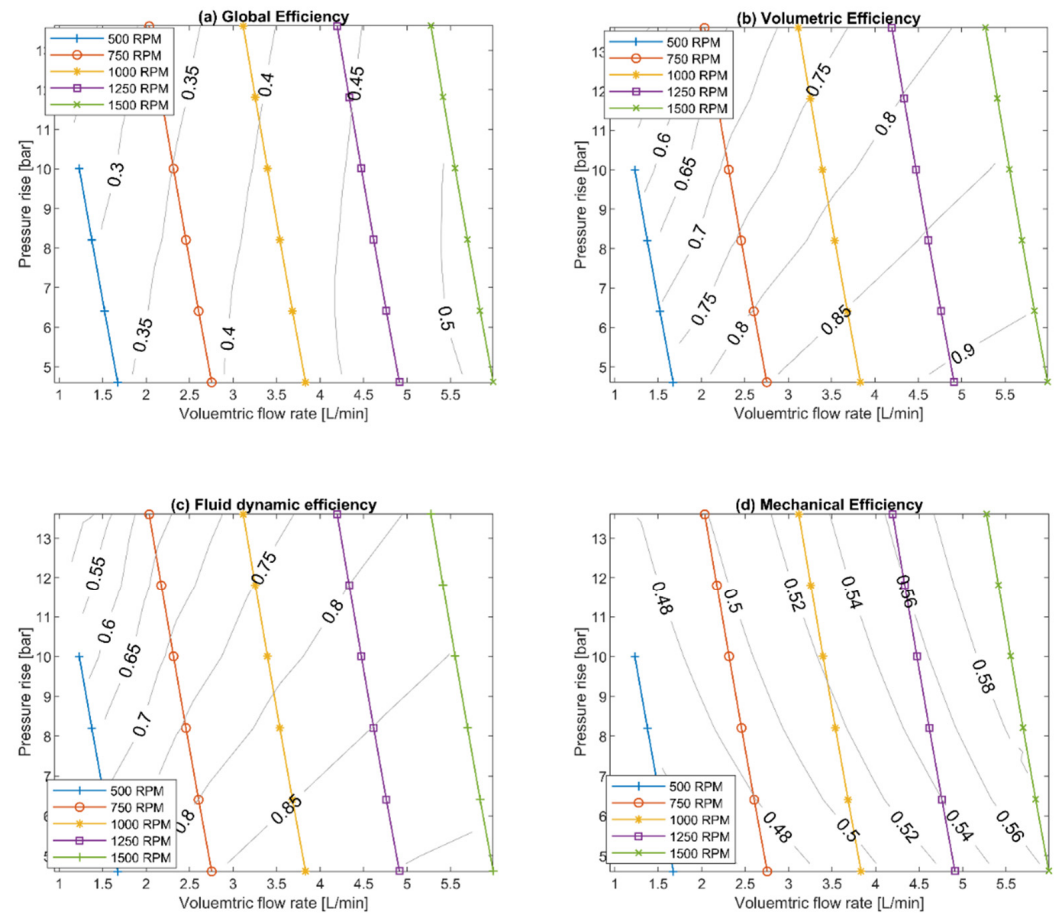


Figure 10. Global (a), volumetric (b), fluid dynamics (c) and mechanical (d) efficiency in the case of a 4-blade optimized pump.

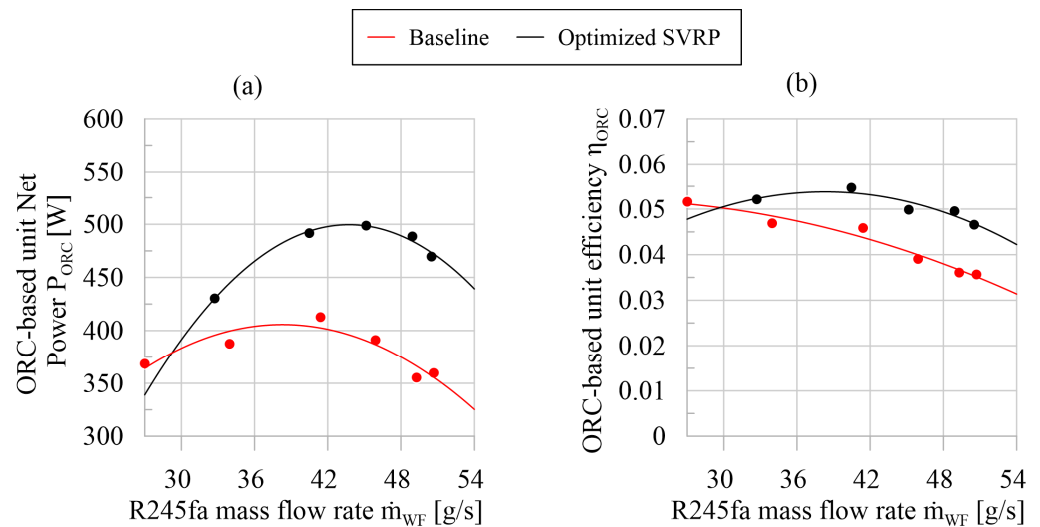


Figure 11. Comparison between the ORC-based unit net power P_{ORC} (a) and efficiency η_{ORC} (b) for a plant adopting the baseline pump and the optimized SVRP.

For a WF mass flow rate lower than 30 g/s, the optimized pump presented a lower efficiency than the baseline. These values were not considered because they were too low for significant power production. In fact, as shown in [35], the minimum flow rate for ORC-based unit activation was 30 g/s.

Above this flow rate, the power improvement varied between 6% and 37%. These results reflect on the ORC efficiency, evaluated as the ratio of ORC net power and the thermal power available at the evaporator (Equation (10)):

$$\eta_{ORC} = \frac{P_{exp} - P_{pump}}{\dot{m}_{HWCp,HW}(T_{HW,in} - T_{HW,out})} \quad (10)$$

It is worth noting that by adopting the optimized SVRP, the maximum power and efficiency shifted toward a higher mass flow rate. This means that when the mass flow rate increases to follow thermal power availability, the pump erodes a lower amount of expander power, thus ensuring greater producibility. This is possible thanks to the higher SVRP efficiency with respect to the baseline case. Indeed, optimized SVRP efficiency varied between 34% and 52% in the considered operating range, suggesting a significant improvement, considering that baseline pump efficiency ranges between 17% and 25% in the same operating interval.

A comparison was also carried out in terms of pump efficiency and power when varying the working fluid, as shown in Figure 12a and 12b, respectively. The figures show that the global efficiency was slightly larger for the R123 case, while the mechanical power was quite coincident in the three cases. This means that the larger values achieved with R123 in terms of efficiency were due to the fact the volumetric and mass flow rates are higher for this fluid than the other fluids (Figure 13a and 13b, respectively). Hence, the hydraulic power was larger, and given the mechanical power was the same, the efficiency grew. It is worth mentioning that the mechanical power was equal for the three cases, as the friction power largely depends on the dry contact between the blade and the inner stator surface. Moreover, given the pressure rise was the same and the volumetric losses were comparable, the indicated power was equal for the three situations. Despite these low differences, the proposed design has been demonstrated to be suitable for other working fluids. This conclusion is particularly interesting, considering the short operating life of organic working fluids [36].

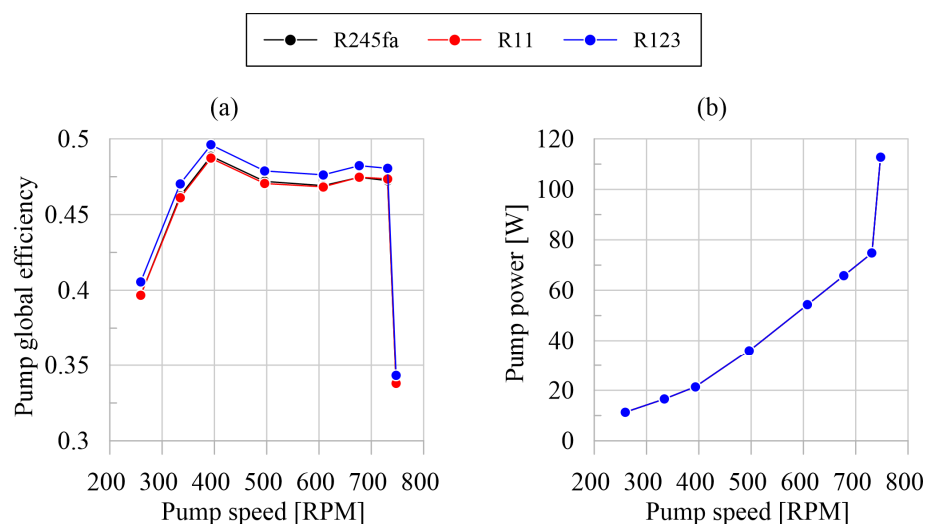


Figure 12. Pump global efficiency (a) and pump power (b) as function of pump speed for R245fa, R11 and R123.

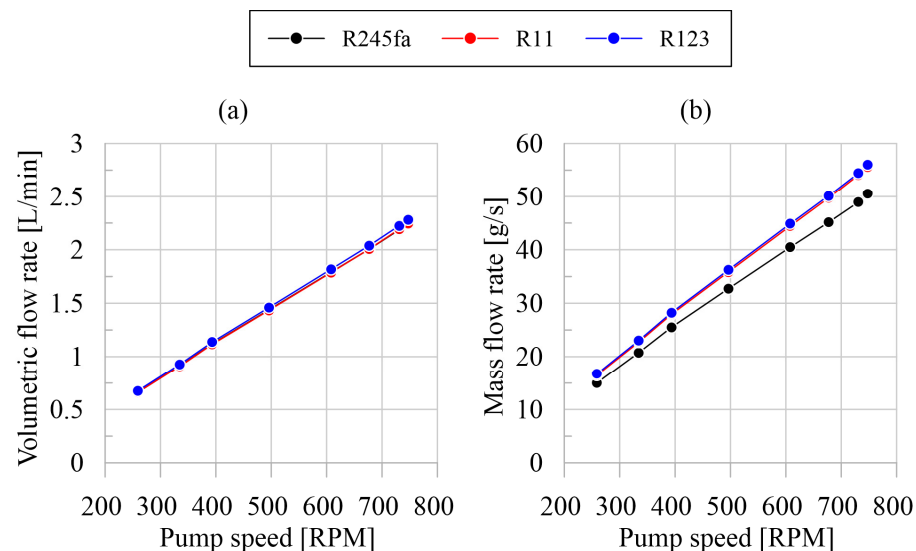


Figure 13. Volumetric flow rate (a) and mass flow rate (b) as a function of pump speed for R245fa, R11 and R123.

4. Conclusions

In small-scale ORC-based power units, low pump efficiency represents one of the main factors limiting higher efficiency. Indeed, due to the low power produced by the expander, the impact on the pump is significant, and for applications below 1 kW (typical of micro-cogeneration systems), the pump efficiency rarely exceeds values of 0.3. What is more, there is a lack of experimental and theoretical analyses aimed at improving the pump design to achieve higher efficiency in the literature. For these reasons, in this present paper, a model-based analysis was carried out to outline an optimized Sliding Vane Rotary Pump for a 1 kW micro-ORC-based power unit. To fulfill this goal, a novel integrated zero (0D) and mono (1D) thermo–fluid–dynamic model was developed and experimentally validated on experimental data collected on an SVRP conceived to work in a larger ORC unit. The validation phase confirmed the reliability of the model and the improvement with respect to a previous model developed by the authors thanks to a Root Mean Square Error equal to 0.4% and 2.3% for the mass flow rate and pump power prediction, respectively. Once the consistency and accuracy of the model were assessed, the model was used as a software platform to optimize the pump design. As a preliminary step, a novel design of the pump was developed, considering the functional requirements of the application at hand (mass flow rate equal to 60 g/s and pressure rise of 12 bar). Thanks to the model, all the possible pump configurations fulfilling the designed volumetric capability were considered, and the results showed that the disk shape configuration allowed the power required by the pump (150 W) to be minimized. This is in line with the results found by the authors for a larger SVRP in engine cooling applications. Once the design was set, the model-based analysis allowed the following novel results to be outlined:

- Considering the efficiency chain analysis, the weakest contribution was the mechanical efficiency, which assumed values between 0.45 and 0.6. In addition, the volumetric efficiency improved, presenting values between 0.6 and 0.8;
- The increase in blade number while keeping the volumetric capability constant resulted in a deterioration in the mechanical efficiency of 0.6 to 0.46, and, consequently, of the whole efficiency. Moreover, slighter penalties were observed in terms of volumetric and fluid dynamic efficiency due to the higher impact of leakage on the elaborated flow rate. Hence, the SVRP with four blades was the best design solution. This result is particularly interesting as, theoretically, an increase in blades allows the

angular pressure difference to be reduced. At the same time, they provided a reduction in chamber volume and filling time. As a whole effect, a reduction in volumetric efficiency was observed. A further experimental analysis will be carried out to explore this aspect;

- Another useful design point was the reduction in the clearance gap between the rotor face and casing of 50 μm to 10 μm , which allows the volumetric efficiency to be improved from 0.64 to 0.78, justifying the reduction in constructive tolerances;
- The integration of clearance gap reduction with the adoption of graphite blades allowed a significant increase in efficiency of up to 25% to be attained. As a result, the SVRP efficiency achieved a maximum efficiency of 0.5, whereas, in the worst operating region, values higher than 0.3 were always guaranteed;
- The adoption of the optimized SVRP in a reference ORC-based power unit allowed the power production to be boosted up to 37% with respect to the baseline case, in which a diaphragm pump was used. This aspect provided an equivalent benefit to plant efficiency, and it was possible thanks to the improvement in pump performance. Indeed, in the analyzed operating range, the optimized SVRP efficiency varied between 34% and 52%, whereas the baseline one was between 17% and 25%. This demonstrates how important it is to improve the pump behavior, whose impact on the whole plant power is significant, particularly for small-scale ORC-based units;
- It was also observed that the proposed design results are suitable for other working fluids (R11 and R123).

The analysis showed that the 4-blade sliding rotary vane pump represents a better solution for this kind of application. To confirm this theoretical conclusion, experimental activity will be performed considering a larger-scale pump. Moreover, different materials for the blade (using polymers) and pump stator will be considered in order to further reduce the friction losses, which are the main constraint to achieving larger efficiency values. This choice has been verified for larger machines and other sliding vane machines.

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Nomenclature

Symbols

c_p —specific heat at constant pressure (kJ/kgK)
 dV_i —chamber volume variation
 f —friction factor
 F —force (N)
 N_v —vane number
 $\dot{m}$ —mass flow rate (kg/s)
 P —power (W)

fluid—fluid dynamic efficiency
 global—global
 HW—hot water
 hyd—hydraulic
 i—index of chambers
 ind—indicated
 lost—power lost
 mech—mechanical

p—pressure (bar), (Pa)	N—Normal Force
p_i —chamber pressure (bar), (Pa)	ORC—Organic Rankine Cycle
Q—flow rate (L/min)	Out—outlet
R_a —actual radius (from rotor center up to blade tip)	p—pressure force (under the blade)
T—Temperature (°C), (K)	pump—pump
t_{cycle} —time of a complete cycle	th—theoretical
V—Volume (cm ³)	v—Flow rate effectively circulating inside chambers
Subscripts	Greek letters
c—centrifugal	Δp —pressure rise (bar)
disp—displaced	η —efficiency
Exp—expander	ω —pump speed (rpm)

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