Dynamic Modeling of a Hybrid Propulsion System for Tourist Boat

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Received: 28 August 2018; Accepted: 26 September 2018; Published: 28 September 2018

Abstract: Interest in designing more efficient and versatile ships comes from increasingly stringent regulations on emissions. In this context, a possible solution to overcome these limits may be the replacement of marine propulsion systems based on diesel engines with hybrid architectures. This paper provides a dynamic analysis of a hybrid marine propulsion system (HPS) consisting of an internal combustion engine and an electric engine coupled with a battery pack. A dynamic simulation of a daily working cycle was carried out based on a real load demand. The instantaneous behavior of each component was evaluated. A brief summary of the HPS performance, varying the battery pack capacity, was provided together with an estimation of its impact on the system efficiency. Referring to this last point, the adoption of a hybrid system has permitted a decrease in the specific consumption, on a given route, of about 2% with respect to the case where the propulsion is entrusted only to the diesel engine.

Keywords: hybrid propulsion; energy storage; dynamic model; marine system; ship propulsion

1. Introduction

Maritime transport has been a fundamental element of human civilization throughout its history, enabling rapid movement for people, growth in trade and the exploration of new territories. Over the years, boats have evolved to accommodate a growing need for people and goods transportation, with the aim of ensuring fast, economical and safe travel. This impulse for innovation in the marine sector is reflected in the evolution of propulsion systems, passed from sailing and steam propulsion to the introduction of diesel engines at the beginning of the twentieth century.

Diesel propulsion has been the traditional architecture adopted by most ships in the last century. However, for about 30 years, growing concerns about the environmental impact of climate-changing and polluting gases [1,2] together with the regulations imposed by the IMO (International Maritime Organization) have caused a substantial change in the design approach of propulsion [3,4].

In the recreational sector, environmental protection has, instead, been achieved through Directive 2013/53/EU [5], which has replaced the previous 94/25/EC. The legislation imposes new requirements for environmental defense, lowering the previous limits on emissions and noise from internal combustion engines. A fundamental challenge that occurred is also the control of the exhaust emissions produced in bays and ports [6]. Many research works deal with this issue in order to estimate emissions due to in-port ship activity based on real and empirical data [7–13].
Many shipbuilders are concentrating their efforts on implementing exhaust gas treatment systems in order to comply with MARPOL (MARitime POLlution) emissions regulations [14]. Moreover, some of them are actively exploring alternatives for energy efficiency and/or pollutant emissions reduction, ranging from the use of cleaner fuels such as LNG (liquefied natural gas) to the hybridization of propulsion systems through a greater electrification. By entrusting the propulsion to an electric motor, also driven by diesel engines, there is the possibility to increase the propulsion system efficiency reducing emissions.

The interest in designing more efficient and versatile ships has amplified the variety of hybrid propulsion systems (HPSs) and power supply architectures, including also hybrid storage systems (ESS). In [15] the authors summarize and analyze the benefits and drawbacks due to the adoption of hybrid propulsion together with the strategies that can improve HPS performance of future smart ships.

Thanks to the increasing power density of lithium-ion battery technologies [16], HPS coupled with ESS has become a realistic option for many maritime applications. For example, real HPS applications can be found in naval vessels [17,18], offshore vessels [19,20], research vessels [21], yachts [22], ferries [23,24], towing vessels [25].

In these applications, the whole electrical demand fluctuates significantly over time and in some cases involves steep power increases and decreases. Consequently, ESS adoption (i.e., batteries, super capacitors, etc.) can provide peak shaving, load smoothing, frequency control, improved quality of power supply and, above all, can enable to switch off all internal combustion engines for a limited period [18,26] reducing noise and pollutant emission. Moreover, batteries can also provide back-up power during failures of diesel generators [24]. Considering the recharging issue, two options can be considered: charging from the grid, when the ship is moored reducing local emissions [15], or charging during the cruise at sea, exploiting the internal combustion engine (ICE) in its operation at higher efficiency.

It is important to emphasize that the benefits of adopting HPS are not only technological or environmental, but also economic. When the only electric users are the auxiliary loads, and only a small fraction of the required propulsive power is converted into electricity, the losses associated with this conversion can raise fuel consumption [27]. Extra electrical equipment can lead to increased weight, size and cost. But vessels that frequently operate at low speed, in protected areas or in ports, can find advantages in the adoption of HPS [28] directly exploiting the electric propulsion.

In technical literature, dynamic modelling and analysis along a real working route are not investigated for HPSs and its components (battery pack and ICE). For these reasons, the present work aims to fill this gap providing a dynamic analysis of the power flow interaction among the HPS components for marine applications emphasizing, at the same time, the importance of proper battery sizing.

In this research work, the authors have analysed the behavior of an HPS constituted as follows:

- a direct mechanical drive that provides propulsion at cruise and high speeds with high efficiency;
- an electric motor, coupled to the same shaft through a gearbox or directly to the propeller shaft, which provides propulsion at low speeds, avoiding pollutants in ports and protected areas, as well as inefficient diesel engine operation in these operating conditions;
- a battery pack which supports the electric motor operation and is charged by the ICE at cruising speed [15].

In this paper, we propose a simulation dynamic model that allows us to estimate the behavior and the performance of the whole HPS in a real case study. Specifically, after the identification of the ship workload, dynamic models were developed both for the diesel engine and the electric propulsion in the Matlab-Simulink environment.
Thanks to the integration of these models, it was possible to characterize the performance of the hybrid system, both in terms of efficiency and effectiveness, going on to check if the energy stored on board is enough to cover the daily route.

2. Overview of Hybrid Marine Propulsion System (HPS) Architectures

As anticipated, hybrid architectures can be considered among the most promising solutions to reduce pollution and greenhouse gas emissions [29]. Despite the reduction in fuel consumption usually related to the use of these technologies, their introduction on small vessels represents a technical challenge, due to the need to maximize the space for people and goods, consequently minimizing the space available for the energy storage and the propulsion system. Moreover, the typical work profiles for this type of boat are characterized by frequent speed transients combined with short charging times.

From the technical point of view, hybrid architectures can be classified in 3 main categories:

- Series hybrid architecture: the propulsion is entrusted entirely to an electric motor, which is powered by one or more electric generators driven by ICEs [30]. The energy storage system is connected to the bus and it can be used separately from or together with the generators [31]. This configuration is characterized by the decoupling of the propeller from the ICE, allowing it to operate at peak efficiency; the efficiency increases obviously with the number of ICEs, as a consequence of a better partition of the required load. This solution can be problematic in small ships due to restrictions on volumes and weights [29].

- Parallel hybrid architecture with both ICE and electric motor mechanically connected to the crankshaft. The electric motor is powered by the energy storage system or by some other source of energy (i.e., other ICEs, fuel cells, etc.). The motors can be used both simultaneously and separately; in most cases the electric component is used at low speeds whereas the ICE is used at high speeds, while the coupled use is available in cases of additional requested power [30]. This architecture reduces the number of components compared to the series hybrid one and allows the optimization of the size of every energy source [29].

- Series-parallel hybrid architecture (also known as power-split), a combination of the previous two inheriting their advantages. ICE can either move the shaft mechanically, as in the parallel configuration, or be disconnected, as in the series configuration. This architecture has greater efficiency because is capable of switching from a series configuration at low speeds to a parallel configuration at high speeds. The main disadvantage comes from a higher cost of production and added difficulty of management [29].

The hybrid architecture analyzed in this research work is a parallel one, with the possibility of completely detaching the ICE from the transmission shaft in case of full electric propulsion. This architecture allows switching between the two engines, but not their simultaneous exploitation.

3. Load Demand

To analyze the behavior of the hybrid propulsion system, a typical load demand of a 10 m tourist vessel used for excursions to a sea cave was considered. This working cycle, supplied by Nautica Salpa, concerns the profile of the power absorbed by the boat hull as obtained through CFD (Computational Fluid Dynamics) analysis known the boat speed at different regimen conditions.

The characterization of the load request was fundamental for the right sizing of the propulsion system, both for the endothermic and electric sections.

Figure 1 shows the trend of the required power to the hull of a single working cycle lasting about 1 h and a half (4380 s). Four phases can be distinguished, characterized by different load demand and boat operation (electric or endothermic mode): maximum speed (16 knot), cruise (12 knot), low speed (5 knot) and maneuvers.
For clarity, after the maneuver phase and the exiting from the port, guaranteed by electric propulsion, the acceleration, for around 700 s, up to the maximum speed in the open sea is entrusted to the ICE. Subsequently, the boat decelerates to reach cruising speed. In this phase the propulsion is carried out through the ICE which also contextually recharges the battery pack. In the surroundings of the cave (1860 s), the boat slows down and the propulsion passes to full electric for the duration of the visit (maneuvers included). At the exit, the internal combustion engine is re-ignited and the second charging phase takes place during the return to the port. Once it has arrived, the entrance and all the maneuver phases are performed in the electric mode.

This last condition is the same for the three sections of Figure 1 characterized by spikes, or rather, the maneuvers to exit the port; close to the cave and entering the port are characterized by the same profile of required load depicted in Figure 2.

Moreover, Figure 2 shows the detail of the power required during the maneuver phase together with the engine rotational speed. Table 1 provides a brief description of each navigation phase. It is emphasized that the battery pack recharging will be realized during the cruise phase.
Table 1. Working cycle description.

<table>
<thead>
<tr>
<th>Navigation Mode</th>
<th>Power (kW)</th>
<th>Speed (knot)</th>
<th>Crankshaft Speed (rpm)</th>
<th>Propulsion Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maneuvers</td>
<td>4.78 ÷ 75</td>
<td>5.14 ÷ 12.4</td>
<td>1300 ÷ 3100</td>
<td>Electric</td>
</tr>
<tr>
<td>Low speed</td>
<td>4.78</td>
<td>5</td>
<td>1300</td>
<td>Electric</td>
</tr>
<tr>
<td>High speed</td>
<td>140</td>
<td>16</td>
<td>4100</td>
<td>Diesel</td>
</tr>
<tr>
<td>Cruise speed</td>
<td>72.39</td>
<td>12</td>
<td>3100</td>
<td>Diesel</td>
</tr>
</tbody>
</table>

4. HPS (Hybrid Propulsion System) Sizing

4.1. Internal Combustion Engine (ICE)

The ICE selected for this marine application is the FNM Marine Diesel Engine (Atella, Potenza, Italy) 30 HPE. This 4-stroke supercharged engine is characterized by the following data [32]:

- Max shaft power: 184 kW (250 hp) at 4100 rpm.
- Max torque: 525 Nm at 2300 rpm.
- Total displacement: 2998 cc.
- Bore/stroke ratio: 95.8/104 mm.
- Cylinders: 4 in line.
- Fuel: diesel.
- Weight: 335 kg.

Figure 3 shows the maximum power and torque curves provided by Costruzioni Motori Diesel spa (Naples, Italy) [32]. It can be highlighted that the maximum power is reached at 4100 rpm, while the torque maximum value, equal to 525 Nm, is found at 2270 rpm.

Figure 3. Maximum engine power vs shaft rotation speed.

In order to determine the power profile required by the propeller, a conventional method used for the boat propulsion sizing was used. This procedure permits us to determine a mathematical
relationship (i.e., cubic function) between the propeller load demand and the engine rotation speed (rpm).

Specifically, the cubic starting point is the maximum engine power that, in this way, overlaps the maximum power required by the propeller (point 1 in Figure 4). The choice of the cubic exponent, equal to 2.5, was carried out in order to guarantee for every operating regime a propeller power greater than the load to the hull determined through CFD analysis.

Moreover, the difference between the maximum engine power and the propeller power ($\Delta W$ in Figure 4) represents a power surplus usable in case of the boat has to face heavy sea conditions ensuring performance in line with the nominal ones (i.e., speed of navigation).

4.2. Battery Pack

For the battery pack sizing, in order to evaluate the energy to be delivered, only the electric load profile required by the propeller was considered.

This profile was calculated starting from the propeller curve previously determined known the shaft rotational regime required in the maneuver phase (see Figure 2) and during boat operation at low speed (see Table 1).

Moreover, to determine the power required to the battery pack (Figure 5), the efficiencies of the electric engine (0.93) and transmission (0.98) were applied to the propeller load. The profile obtained is depicted in Figure 5, while Figure 6 shows a zoom relative to the maneuver phase.

In this discussion, in order to simplify the analytic approach, the mechanical coupling issues among the ICE, the electric motor and the transmission were not deeply analysed. Moreover, a generic electric motor able to manage the power exchanged with the battery maintaining an ideal efficiency of 93% was chosen. This value can be considered typical of a three-phase asynchronous motor. Moreover, a transmission efficiency of 98% was set.

The battery pack was sized to meet the operating cycle load demand both in terms of power and stored energy (about 8 kWh for cycle).
Figure 4. Propeller curve construction.

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Figure 5. Electric load required by the propeller.

Figure 6. Propeller load in maneuvers.

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The battery pack was sized to meet the operating cycle load demand both in terms of power and stored energy (about 8 kWh for cycle).

Table 2 shows the characteristics of two lithium iron phosphate (LFP) battery packs sized compatibly with this application.

Table 2. Batteries datasheet.

<table>
<thead>
<tr>
<th>Battery Model</th>
<th>WB-LYP40AHA Thunder Battery</th>
<th>WB-LYP60AHA Thunder Battery</th>
</tr>
</thead>
<tbody>
<tr>
<td>Module capacity</td>
<td>40 Ah</td>
<td>60 Ah</td>
</tr>
<tr>
<td>Operative voltage</td>
<td>3.00 V</td>
<td>3.0 V</td>
</tr>
<tr>
<td>Nominal current</td>
<td>40 A</td>
<td>60 A</td>
</tr>
<tr>
<td>Number of modules in series</td>
<td>90</td>
<td>100</td>
</tr>
<tr>
<td>Number of parallel strings</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Bus voltage</td>
<td>270 V</td>
<td>277 V</td>
</tr>
<tr>
<td>Maximum charging rate</td>
<td>1 C</td>
<td>1 C</td>
</tr>
<tr>
<td>Continuous C-rate</td>
<td>3 C</td>
<td>3 C</td>
</tr>
<tr>
<td>Pulse C-rate</td>
<td>10 C</td>
<td>10 C</td>
</tr>
<tr>
<td>Charge current</td>
<td>40 A</td>
<td>60 A</td>
</tr>
<tr>
<td>Continuous discharge current</td>
<td>120 A</td>
<td>180 A</td>
</tr>
<tr>
<td>Charge power (1C)</td>
<td>11 kW</td>
<td>18 kW</td>
</tr>
<tr>
<td>Continuous discharge power (3C)</td>
<td>32 kW</td>
<td>54 kW</td>
</tr>
<tr>
<td>Peak power</td>
<td>108 kW</td>
<td>180 kW</td>
</tr>
<tr>
<td>Battery pack capacity</td>
<td>10.8 kWh</td>
<td>18 kWh</td>
</tr>
<tr>
<td>Battery pack weight</td>
<td>144 kg</td>
<td>230 kg</td>
</tr>
</tbody>
</table>
These configurations are able to provide discharge powers to cover the propeller demands both in case of continuous (11.47 kW) and peak requests as in the case of the maneuver phases (up to 101.88 kW), maintaining at the same time a bus voltage close to 270 V.

5. HPS Dynamic Modelling

As mentioned in the introduction, the Simulink dynamic model allowed us to simulate and, subsequently, analyse the HPS operation on a true working cycle. At the same time, the model permitted us to evaluate the profiles of instantaneous operating conditions of the hybrid propulsion system, in terms of battery state of charge (SOC), engine consumption, efficiency, crankshaft angular velocity, etc. The model was essentially divided into two main blocks: the “internal combustion engine” and the “battery pack” sections.

5.1. Internal Combustion Engine Block

The ICE model was developed in the Simulink environment to characterize its dynamic behavior varying the required power. The engine was modelled using a cycle mean value model [33,34] together with differential equations for the calculation of the engine crankshaft speed and the turbocharger shaft speed.

In Figure 7 a global overview of the subsystems that constitute the model is reported.

The main input of the model is the assigned working cycle (Figure 8). It is imposed in terms of the rotation speed of the engine crankshaft that is, in this type of application, directly linked to the propeller required power (see the propeller curve in Figure 4). This load demand includes also a mechanical transmission efficiency of 98%.

Moreover, the model needs, in the first simulation step, the initial crankshaft rotational speed and that of the turbocharger shaft beyond the initial value of the speed regulator. After every iteration, the model calculates the instantaneous supplied power and torque.

An overall description of how the model works is provided below. In the Supplementary Material Section, the governing equations, block by block, are described and discussed. In Figure 9, the layout of the “Engine” block is shown.

At the first iteration, the model compares the initial value of the engine crankshaft speed with the set one in order to determine, through the PI (Proportional Integral) subsystem (Figure 9), the speed regulator angle.

Considering such a parameter together with the engine crankshaft speed and the related consumption at full load (available from the test bench characterization and implemented in the code through a look-up table), the fuel flow rate injected in the cylinders is then obtained in the

![Figure 7. Global overview.](image-url)
“fuel pump” subsystem (Figure 9). This value is used in the “Cylinders” subsystem together with information about pressure and temperature in the intake manifold (calculated in “Compressor subsystem”), to determine the power and the torque.

The parameters calculated at the end of the combustion process are exploited in the “Turbine” block to determine the torque produced by the turbine in the exhaust manifold. Applying the conservation of the angular momentum, between the torque absorbed by the compressor and the initial value of the turbocharger rotational speed, a new rotation regime is obtained.

Contemporaneously, the torque produced by the engine is compared with the one required by the propeller and so, through an integration function, the new crankshaft speed for the next simulation step is calculated.

![Figure 8. Rpm and power required from the internal combustion engine (ICE).](image)

![Figure 9. ICE block detail.](image)
Model Validation

The validity of the results of the dynamic simulations was verified with respect to the data recorded at the test bench, characteristic of the operation in stationary conditions. Consequently, step variations in terms of the crankshaft rotational speed required to meet the working points tested at the test bench were imposed as the model input. In this simulation, the engine works at full load.

Figure 10 shows the comparison in terms of crankshaft speed between the calculated values (dotted) and the imposed steps (black). It can be seen how, after a short initial transient, for each step the model achieves the imposed rotation regime. The little initial deviation is due to the braking torque, which hinders the speed variations, and to the non-instantaneous response of the PI controller. Moreover, Figure 11 shows the comparison between simulated and experimental values of power and torque. Even here, after a very short transient, the calculated values coincide with the experimental ones.

![Figure 10. Comparison between imposed (input) and calculated (output) crankshaft speed.](image1)

![Figure 11. Comparison between simulated and experimental values of torque and power.](image2)
From what above, it is possible to state that the model represents with a good approximation the ICE real operation at full load. This result allows the model application to the case study with a good confidence in the results accuracy.

5.2. Battery Pack Model

A dynamic model of the battery pack was developed to analyse its transient behaviour in the HPS. Among several parameters, this model evaluates the profile of the power to/from the battery and, consequently, the energy amounts stored and delivered to the electric motor during one or more working cycles.

The authors have tuned the model in previous research works [35,36]; specifically, a \( Q \) (Ah) capacity was set to determine the battery \( SOC \) trend according to the battery datasheet characteristics. In detail, the open circuit voltage \( (V_{ocv}) \), the change of internal resistance during charging \( (R_{ch}) \) or discharge \( (R_{dis}) \) were used to characterize the battery behaviour.

In this model, battery current \( (I_{bat}) \) and voltage \( (V_{bat}) \) are characterized as:

\[
I_{bat} = V_{ocv} - \sqrt{\frac{V_{ocv}^2 - 4R_{bat}^{int}P}{2R_{bat}^{int}}} \tag{1}
\]

\[
V_{bat} = V_{ocv} - \frac{I_{bat}}{R_{bat}^{int}} \tag{2}
\]

where \( P \) is the power required/delivered to the battery and:

\[
V_{ocv} = \begin{cases} V_{ocv} = f_1(SOC) & \text{charge} \\ V_{ocvd} = f_2(SOC) & \text{discharge} \end{cases} \tag{3}
\]

\[
R_{bat}^{int} = \begin{cases} R_{ch} = f_3(SOC) & \text{charge} \\ R_{dis} = f_4(SOC) & \text{discharge} \end{cases} \tag{4}
\]

Equations (3) and (4) were implemented by using look-up tables based on experimental data. In [35], \( V_{ocv} \) and \( R_{bat}^{int} \) trends of the LFP battery are reported.

The determination of \( SOC \) was carried out as follows:

\[
SOC = SOC_{ini} - \int \frac{\eta I_{bat}}{Q} \tag{5}
\]

and

\[
\eta = \begin{cases} \eta_{ch} = \frac{V_{ocv} - I_{bat}R_{ch}}{V_{ocv}} & \text{charge} \\ \eta_{dis} = \frac{V_{ocv} - I_{bat}R_{dis}}{V_{ocv}} & \text{discharge} \end{cases} \tag{6}
\]

where \( SOC_{ini} \) is the initial value of \( SOC \) and \( Q \) (Ah) represents the battery capacity. In the present study battery capacity and the nominal storage power were set as indicated in Table 2.

The Simulink battery model is shown in Figure 12. Specifically, the battery characteristics (open circuit voltage and internal resistance) were implemented in the “Battery Imare” sub-block.

![Figure 12. Battery model.](image-url)
6. Simulations and Results

In this chapter the simulation results are presented and discussed together with the most relevant aspects due to the adoption of an HPS. The overall dynamic model of the HPS is applied to the considered working cycle to characterize the operating points of the HPS and its performance with respect to the case of full ICE propulsion.

6.1. HPS Simulation Results: ICE Propulsion Phase

Below, the results of the simulations performed on the diesel engine in the case of hybrid architecture are presented. Figure 13 shows the output of the model in terms of crankshaft rotational speed and power supplied in the event, as a preliminary simulation, that the battery pack is not charged.

![Figure 13. Working cycle without battery charge.](image)

As can be seen, the calculated speed trends follow the imposed one with no significant deviations in the whole operating range. The only section in which the two trends diverge is corresponding to the maximum speed. In Figure 13, for completeness, a detail of this negligible deviation is provided.

In order to evaluate how the two different battery packs sized at paragraph 4.2 affect the engine performance, the cycle conditions with charging power of 11 kW and 18 kW (only during the cruising phase) were simulated and compared.

Figure 14 depicts the torque (a) and power (b) profiles obtained. The bottom of this figure shows the fuel flow rate (c) required during the endothermic phase of the working cycle. It is evident how, in the cruising phase, this value increases proportionally to the power supplied by the engine. It moves from 5.69 g/s for the case without battery charging to 6.35 and 6.8 g/s in the case of 11 and 18 kW charging power respectively. Similar considerations can be made considering cumulative consumption per cycle. Specifically, the total consumption increases, with respect to the case of no charging (17.35 kg), of 5.5% (18.3 kg) and 9% (18.9 kg) for 11 kW and 18 kW respectively.

Instead, it can be highlighted that the fuel specific consumption slightly lowers during the cruising phase, increasing the battery charging power (219.4 g/kWh with no charging vs. 219 g/kWh and 218.8 g/kWh at 11 kW and 18 kW respectively). This behavior is due to the higher efficiency at increasing load closer to full load conditions.
Figure 14. Torque (a), power (b), fuel flow rates (c) profiles with different battery charging power (0, 11, 18 kW).

6.2. HPS Simulation Results: Electric Propulsion Phase

In these simulations, the model input is the electric load profile shown in Figure 5. To analyze the battery pack behavior, charging phases during cruising were added to the load curve.
By convention, the charge and discharge power values have negative and positive sign respectively. Figure 15 shows the overall working cycle for what concerns the battery pack operation.

In Figure 16, a higher depth of discharge for the 10.8 kWh battery pack can be noted. Specifically, it allows only 6 consecutive trips to be supported, while the 18 kWh pack is able, from the second cycle, to be fully recharged allowing it to be disengaged from the need to recharge during docking.

6.3. Further Considerations on Simulation Results

The results obtained can be also exploited to evaluate the advantages of adopting a hybrid architecture in terms of environmental benefits and overall system efficiency. Hybrid propulsion allows navigation and maneuvers within ports and protected areas in full electrical mode, without
acoustic and pollutant emissions. Furthermore, this is achieved with further advantages in term of fuel consumption reduction.

To quantify these benefits provided by the HPS, due to a higher mean efficiency, the reference case is represented by the diesel engine utilization to cover the overall working cycle considered. To this end, a simulation was carried out to determine the overall and the average specific fuel consumption in one cycle.

On the other hand, to evaluate the specific consumption in hybrid configurations, the energy required to bring the battery back to full charge was also considered (equal to the mean of difference values between initial and final SOC of each cycle of Figure 16). This value, related to the electricity consumption at the recharging column, was converted into an equivalent fuel consumption, or better still additional fuel, considering the specific consumption of the engine at 3100 rpm (cruising speed).

This consideration is cautionary as the average generation efficiency of the thermal power plants that provide electricity to the recharging column is certainly higher than that of the small ICE considered here for the boat’s propulsion.

All the data obtained are shown in Table 3.

<table>
<thead>
<tr>
<th>Architecture</th>
<th>Fuel Consumption (kg)</th>
<th>SOC&lt;sub&gt;ini&lt;/sub&gt;-SOC&lt;sub&gt;final&lt;/sub&gt; (%)</th>
<th>Additional Fuel (kg)</th>
<th>Propeller Energy Demand in One Cycle (kWh)</th>
<th>Average Specific Consumption (g/kWh)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Only Diesel Engine</td>
<td>19.03</td>
<td>-</td>
<td>-</td>
<td>82.7</td>
<td>230.1</td>
</tr>
<tr>
<td>HPS with 10.8 kWh battery pack (40 Ah module)</td>
<td>18.305</td>
<td>14.6</td>
<td>0.36</td>
<td>82.7</td>
<td>225.7</td>
</tr>
<tr>
<td>HPS with 18 kWh battery pack (60 Ah module)</td>
<td>18.91</td>
<td>0</td>
<td>0</td>
<td>82.7</td>
<td>228.6</td>
</tr>
</tbody>
</table>

From this table it can be deduced that a hybrid architecture is definitely advantageous in terms of consumption. In fact it allows the supply of power, produced at cruise operation and so under ICE high efficiency, at operating regimes far from the optimal. Moreover, ICE efficiency at cruise speed slightly increases, due to the higher operating load for battery charging, as demonstrated by the specific consumption data previously discussed.

What is stated above makes the HPS sized in this work more efficient than endothermic propulsion, despite efficiencies lower than 1, although high, being considered for the battery pack and the electric machine that are added to the efficiency chain.

7. Conclusions

This paper provides a dynamic analysis of a hybrid propulsion system which integrates an internal combustion engine and an electric motor supported by a battery pack.

Thanks to the development of an HPS dynamic model, the instantaneous behaviour of each component is analysed on a real working cycle of a tourist craft intended for tourist visits at a particular sea cave.

Specifically, the ICE model allowed us to verify if the chosen model is appropriate to sustain the cycle loads during the cruising and high-speed phases, respecting the assigned speed curve.

The battery model was, instead, used to check the charge status of the ESS during the daily working cycle and for sufficient autonomy from external recharge.

From the simulation results, it is possible to affirm that the hybrid system is able to offer low-speed navigation at zero emissions during cave visits and maneuvers in port and protected areas.

As regards the two types of batteries tested, it can be concluded that the 10.8 kWh battery pack with 40 Ah modules is the optimal choice in terms of weight and costs and, moreover, it ensures a range of 6 consecutive mission cycles without external recharge.
Instead, the 18 kWh battery pack (60 Ah modules) is able to provide higher starting powers and the possibility to avoid the recharge needed during docking times but, on the other hand, this solution in more expensive and weighty.

From an energy point of view, the adoption of a hybrid architecture has led to an improvement in the overall efficiency of the propulsion system on the considered route. With reference to the specific consumption, the HPS adoption permits us to decrease it of about 2% in the case of 10.8 kWh battery pack with respect to the case in which only the diesel engine is intended to cover all of the route.

**Author Contributions:** Conceptualization, L.B. and P.A.O.; Data curation, L.B., F.I., N.P. and P.A.O.; Formal analysis, L.B.; Methodology, P.A.O.; Software, N.P., P.A.O. and L.T.; Writing—original draft, L.B. and P.A.O.; Writing—review and editing, G.B. and F.G.

**Acknowledgments:** The research activity has been carried out within the project “iMARE: ibrido MAriino per imbaRcazioni ad elevata efficienza Energetica”, funded by the Italian MISE. Project partners are acknowledged.

**Conflicts of Interest:** The authors declare no conflict of interest.

**References**

1. Rehmatulla, N.; Calleya, J.; Smith, T. The implementation of technical energy efficiency and CO$_2$ emission reduction measures in shipping. *Ocean Eng.* 2017. [CrossRef]


7. Saxe, H.; Larsen, T. Air pollution from ships in three Danish ports. *Atmos. Environ.* 2004. [CrossRef]


23. Ovrum, E.; Bergh, T.F. Modelling lithium-ion battery hybrid ship crane operation. *Appl. Energy* 2015. [CrossRef]

24. Zahedi, B.; Norum, L.E.; Ludvigsen, K.B. Optimized efficiency of all-electric ships by dc hybrid power systems. *J. Power Sources* 2014. [CrossRef]


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