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(Research Article)

A comparison between diesel-electric and mechanical propulsion plants for a small cruise ship

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Abstract

The paper reports the comparison between diesel-electric and mechanical propulsion plants, employed to a small cruise ship propulsion. The plants prime movers are the Rolls-Royce Bergen marine diesel engines. Both engine versions, characterized by similar rated power but different power lint curve and specific fuel contours trend in the power-speed plan, are tested in all the considered propulsion plant types by simulation. In addition, the diesel-electric propulsion plants are simulated considering the diesel generators working in both constant or variable speed. The comparison between all the considered ship propulsion plants are presented in tabular and graphical form and commented.

Keywords: Simulation; Diesel engines comparison; Diesel-electric propulsion; Marine propulsion plants comparison

1. Introduction

In the shipping the themes of fuel costs and polluting emissions reduction, required by the IMO [1-7], are held in high regard. To improve the marine engines efficiency and as consequence reduce the fuel consumption and carbon dioxide emissions, the use of electric propulsion is now the norm for cruise ships [8], and more and more often also in other ship types (ie: ferries and cruise ferries, yachts, naval ships) [9-18].

In some vessel types, as ferries and even more cruise ships, due the travel schedule and distance between the departure and arrival ports, the browsing speed can vary significantly. In these ships the propulsion plant engine(s) often runs in very different conditions respect their Normal Condition Rating (NCR), which is the condition for which the overall propulsion plant has its maximum efficiency, including propulsion engines.

The engine efficiency depending to its operating diagram power limit-speed curve and specific fuel contours values and trend. In the conventional mechanical ship propulsion plants the engine power limit curve, versus the engine speed, influences the propeller working conditions. In the engine operating diagram the propeller working curve must have a shape to make the engine work with specific fuel consumption (*sfc*) as low as possible, maintaining an adequate value of the engine margin. To achieve this, controllable pitch propellers (CPP) are often used in these propulsion plants type.

In the diesel electric propulsion plants, the required electric power is satisfied to the electric generator engines, those can operate in constant speed or, more recently, in variable one. In this last case, the diesel generator engine required power is delivered at the speed corresponding to the lower value of the *sfc* in the engine operating diagram. Another advantage of the diesel electric propulsion is that the propeller speed is independent to the engine one, therefore, for

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each ship speed, it is possible select the propeller speed which, in addition to providing the necessary thrust, allows it to operate with the better performance.

In addition to the advantages in terms of efficiency and emissions, compared to traditional mechanical propulsion plants, the diesel electric propulsion does not provide a mechanical connection between the propulsion engine(s) and the propeller, in them therefore the engine room(s) can be conceived and allocated in order to optimize the exploitation of the ship space.

In a previously authors paper [19] was compared the performance of a small cruise ship mechanical propulsion plant, to the variation of the main engine type (two Rolls-Royce Bergen diesel engines series are tested), characterized by similar power and speed but different power limit curve and *sfc* contours trend in the engine operating diagram [20]. Starting to the results obtained in [19], in this paper the current small cruise vessel propulsion plant, considered in [19], is substitute, by simulation, with a diesel electric propulsion one, adopting alternatively the same Bergen engines compared in [19]. Both engine types are tested in constant and variable diesel generator speed in the diesel-electric propulsion plants.

The diesel engines propulsion plants calculated results, for different ship speeds and season (winter and summer), are compared between them and with those of the mechanical propulsion plant, already determined in [19]. The comparisons results are reported and commented in the paper.

2. Case study

The considered vessel is the small cruise ship Spirit of Oceanus of Cruise West shipowning company. The currently ship and propulsion plant main data are reported in Tab. 1.

Fig. 1 shows the current vessel mechanical propulsion plant layout, composed by two independent shaft lines, each of them comprises a four stroke diesel engine (MAIN-DE in Fig. 1), a shaft electric generator (SEG), a reduction gear (G), a controllable pitch propeller (CPP).

Table 1 Main ship data

Overall length	80.7 m
Breadth	15.3 m
Draught	3.9 m
Displacement	3000 t
Maximum speed (design draught)	18.5 kn
Original propulsion diesel engines	2x2289 kW
Shaft electric generators	2x800 kWe
Diesel generators	2x1050 kWe
Summer hotel electric power	984.7 kW
Winter hotel electric power	727.8 kW
Passenger	120
Crew	72

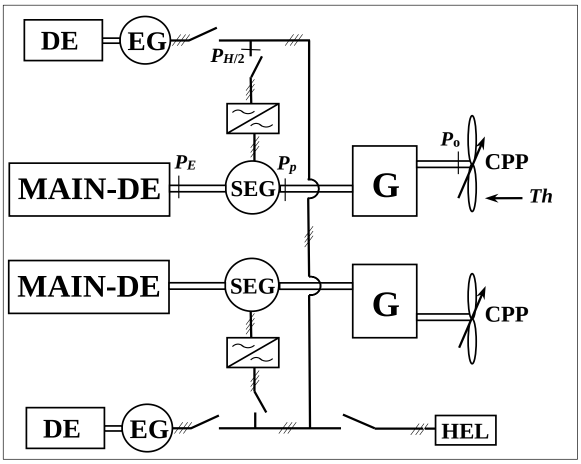

Figure 1 Current mechanical ship propulsion plant layout

The two shaft electric generator (SEG) satisfied the vessel hotel electric load (HEL in figure) during the browsing, while the two diesel electric generators (DE-EG blocks in Fig. 1) are used during stops in harbor.

The current vessel propulsion plant of Fig. 1 is modelled by the authors in [19], where each plant component (hull, CPP, gear, SEG, MAIN-DE) is simulated in a MATLAB-SIMULINK© modular code. Each module simulates the pertinent propulsion plant component performance by correlations and/or tabular data, as descried in [19]. In that article the simulator is employed to determine, for different ship speeds, the propeller thrust and power (*Th* and *P*o respectively in Fig. 1). In each shaft lines the power to propulsion before the SEG (P_p in figure) is calculated by:

$$P_p = \frac{P_o}{\eta_R \eta_G \eta_S} \quad \text{..............} (1)$$

where: η_R, η_G and η_S are the propeller rotary, the gear and the overall propeller shaft lines efficiency respectively. To these efficiencies are assumed the follows values: η_G = 0.98, η_S = 0.98, η_R = 0.99.

The single MAIN-DE brake power (P_E in Fig. 1) is determined with:

$$P_E = P_p + \frac{P_H / 2}{\eta_{AC/AC} \eta_{SEG}} \quad \text{..................} (2)$$

where: P_H is the ship hotel electric power (whose summer and winter values are reported in Tab. 1), $\eta_{AC/AC}$ the electric frequency converter efficiency and η_{SEG} the shaft electric generator efficiency. The values of these efficiencies are: $\eta_{AC/AC}$ = 0.97; η_{SEG} = 0.96 respectively.

In this paper the mechanical ship propulsion simulator is employed to calculate the propeller thrust (*Th*) for different vessel speeds. This data is essential, being in this article the original mechanical propulsion plant replaced by a diesel-electric propulsion one, in two different variants, as reported below.

3. Diesel Electric Propulsion Plants

In this article two diesel-electric plants are considered, with constant or variable diesel-generators speed, to replace the current mechanical propulsion system of the vessel in question. The two diesel-electric plant layouts are shown in Figs. 2a and 2b respectively.

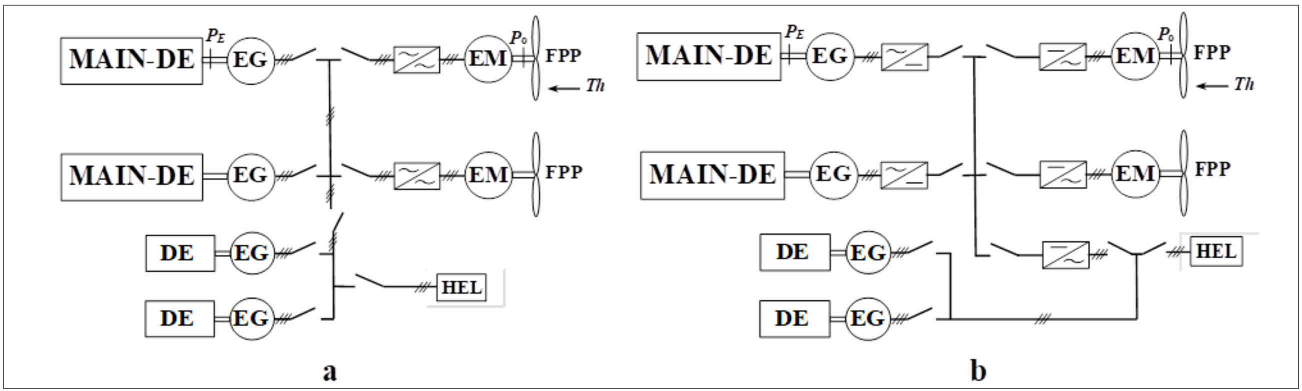

Figure 2 Diesel electric plant layouts, with constant speed electric generators (a) and variable one (b)

In both schemes the MAIN-DE engines satisfy both the propeller power and the hotel electric one (HEL in Fig 2). Likewise to the mechanical propulsion plant of Fig. 1, also in these one the two diesel electric generators (DE-EG blocks in Fig. 2) are used during stops in port.

Both diesel-electric plants use fixed pitch propellers (FPP in Fig. 2) instead of controllable pitch propellers employed in the mechanical one (see Fig. 1). The FPP solution, already employed in others analogues propulsion plants [11,14,17], is acceptable in a small ship with diesel-electric propulsion, thanks to the possibility of varying the propellers speed independently to the diesel-generators one, allowing (together with the bow-thruster, not shown in the plant layouts of Fig. 2) adequate ship control in the maneuvers of departure-arrival in port. Moreover, the FPP propellers have a better efficiency and a significantly lower cost referred the CPP ones.

Starting to the selected ship speed, from the mechanical propulsion plant simulator [19], the required propeller thrust (*Th* in Fig. 2) is determined, and from the FPP open water characteristics we obtain the shaft speed that minimize the propeller power in undisturbed water (*P*o in Fig. 2).

In the diesel-electric propulsion plant with constant speed diesel-generators (DG-cs) (Fig. 2a) the total MAIN-DE brake power (P_E) is calculated with:

$$P_E = \left(\frac{2\,Po}{\eta_R \eta_{EM} \eta_{AC/AC}} + P_H \right) \frac{1}{\eta_{EG}} \qquad (3)$$

where: η_{EM} and η_{EG} are the propellers electric-motors and electric-generator efficiency. An equal value of 0.97 is assumed for both efficiencies. In this plant the P_E can be produced by a single or both MAIN-DE.

In the diesel-electric propulsion plant with variable speed diesel-generators (DG-vs) (Fig. 2b), the overall MAIN-DE brake power (P_E) is determined also in this case starting by the propellers power on undisturbed water (*P*o) and to the hotel electric load (HEL) values, by equation:

$$P_E = \left(\frac{2\,Po}{\eta_R \eta_{EM} \eta_{DC/AC}} + \frac{P_H}{\eta_{DC/AC}} \right) \frac{1}{\eta_{AC/DC} \eta_{EG}} \qquad (4)$$

with: $\eta_{DC/AC}$ and $\eta_{AC/DC}$ the direct-alternate and vice versa electric energy converter efficiency. 0.97 value is assumed for both efficiencies. Also in this case only one or both the MAIN-DE can be used to generate the P_E.

4. Main diesel engines

Similarly to what done in [19], the propulsion engines currently used in the cruise ship propulsion plants, and in its mechanical propulsion plant simulator, are substitute with analogue power and speed diesel engine type, manufactured by Rolls-Royce Marine [20].

Tab. 2 reports the main data of this four-stroke marine diesel engine (Rolls-Royce Bergen C25:33L8P, declined in two series which differ mainly in the maximum rotational speed (900 or 1000 rpm of maximum speed (N_{MAX})), and in the normalize constant *sfc* contours trend in the power-speed operating diagram, reported in Fig. 3.

Table 2 Engines main data and Maximum Continuous Rating performance parameters

Engines parameters	**RR C25:33L8P 900 rpm N_{MAX}**	**RR C25:33L8P 1000 rpm N_{MAX}**
Cylinder numbers	8	8
Bore	250.0 mm	250.0 mm
Stroke	330.0 mm	330.0 mm
Brake power	2560 kW	2665 kW
B.m.e.p.	26.4 bar	24.7 bar
Maximum speed	900 rpm	1000 rpm
Norm. MCR *sfc*	1	1.005

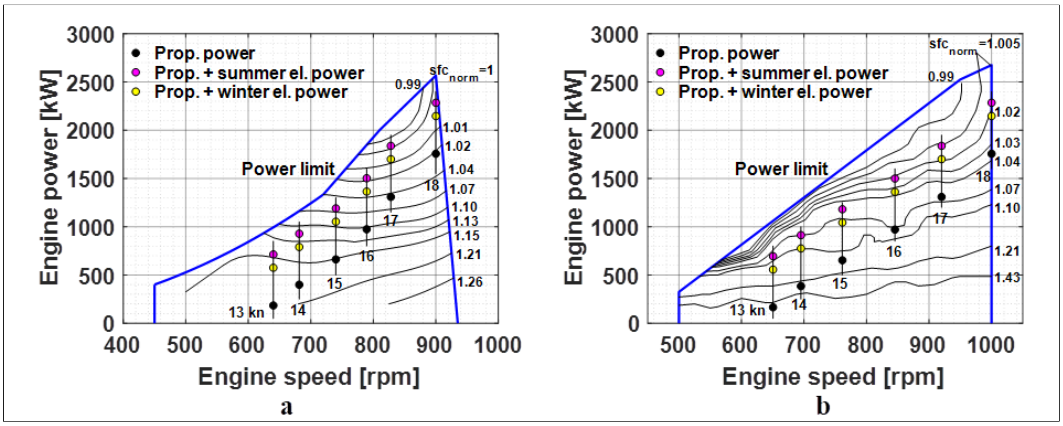

Figure 3 900 (a) and 1000 (b) rpm N_{max} Bergen engines normalized *sfc* contours and mechanical propulsion plant working conditions vs ship speed on engines power-speed plan

In Tab. 2: RR means Rolls-Royce, b.m.e.p. is the brake mean effective pressure, MCR is the engine Maximum Continuous Rating, the 'Norm. MCR *sfc*' is the engine MCR *sfc* normalized by dividing to the RR C25:33L8P 900 rpm N_{MAX} MCR load condition value one. The *sfc* contours reported in Fig. 3 are normalized in the same way used in Tab 2.

Fig. 3 put in evidence the substantial different power limit curves and *sfc* contours trend of the two engine versions. In the 900 rpm N_{MAX} engine (Fig. 3a) the *sfc* contours are characterized by a typical trend for these engines type [22-26]. The 1000 rpm N_{MAX} one *sfc* contours (Fig. 3b) have a trend that differs substantially from that typical of these engines type equipped with single turbocharger.

This marked difference in the power limit curves and *sfc* contours trend of the two engine versions led substantial differences in the mechanical propulsion plant performance of the considered ship [19]; a similar investigation is carried out in this article using both RR engine versions in diesel-electric propulsion plants of Fig. 2.

As reported in Tab. 1, two hotel electric power values are considered depending the navigation season (winter or summer), corresponding to 13 and 29 °Celsius of reference winter and summer ambient temperatures respectively. During this last season, a engines 0.8% *sfc* increase is considered, for the reasons given in [27]. In the same season an analogue engines *sfc* increase is computed also in [19].

5. Propulsion plants comparison

A first comparison between the different propulsion plants is carried out for several values of the ship speed, included in the range 13-18 knots, with one knot of step speed.

Fig. 3 [19] reports, in the engine power-speed plane, the vessel propulsion power (P_P, black marker in figure), pertinent the mechanical propulsion plant of Fig. 1, calculate by eq. (1). In Fig. 3 is reported the overall engine power (P_E, eq. (2) and plant scheme of Fig. 1), in case of winter (yellow marker) and summer (cyrano marker) navigation for the selected vessel speeds.

An engine margin (maximum value of difference between the power limit and the maximum power supplied by the engine, at the same speed) not less to ten percent of the power limit value is chosen. With a Pitch-propeller Diameter ratio (*P/D*) equal to 1.05 this engine margin condition is respected (cyrano markers in Fig. 3), in both engine versions, only with ship speeds of 18 and 17 knots. The 900 rpm N_{MAX} Bergen engine do not respect the engine margin limit starting from 16 knots of ship speed, due its power limit curve trend compared to that of 1000 rpm N_{MAX} Bergen engine, as shown in Figs. 3a and 3b. As reported in [19], to comply the chosen engine margin, the propeller *P/D* ratio had progressively reduced, as the ship speed decreased, as reported in Fig. 4a. The figure shows that for the 900 rpm N_{MAX} engine the *P/D* ratio reduction starts from 16 knots of vessel speed, while in the 1000 rpm N_{MAX} one this ratio reduction starts to 14 knots one.

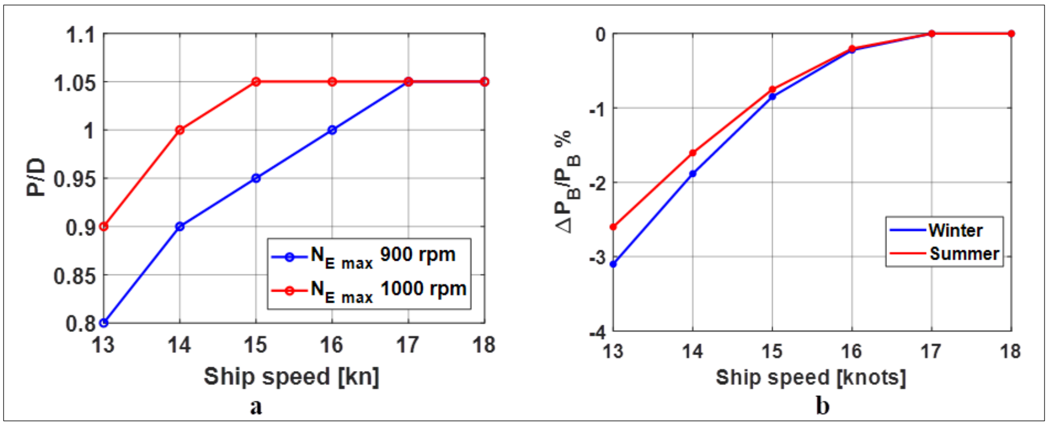

Figure 4 Mechanical propulsion plant: (a) CPP *P/D* laws vs ship speed, (b) 900 and 1000 rpm N_{MAX} engines brake power percentage differences vs ship speed in winter and summer conditions

With a lower *P/D* ratio, to parity of propeller thrust for obtain the desired ship speed is need increase the propeller speed, and in the mechanical propulsion plant also engine one, thus bringing the considered operating point to an engine speed whose power limit value is higher, as reported in Fig. 3. As reported in [19], the propeller *P/D* ratio reduction reduces the propeller efficiency, to parity of propeller thrust, and therefore increase the engine required power. Fig. 4b shows the 900 and 1000 rpm N_{MAX} engine brake power difference in winter and summer condition versus the ship speed. The data reported in Fig. 4b are calculated by the generic equation:

$$\Delta x/x\% = \frac{x_{1000\,\mathrm{rpm}\,N\,\max} - x_{900\,\mathrm{rpm}\,N\,\max}}{x_{900\,\mathrm{rpm}\,N\,\max}} 100\ [\%] \qquad \text{………………..(5)}$$

where: x is the generic variable, $x_{900\,\mathrm{rpm}\,N\,\max}$ and $x_{1000\,\mathrm{rpm}\,N\,\max}$ are the generic variable (x) referred to 900 rpm N_{MAX} and 1000 rpm N_{MAX} engines respectively.

Fig. 4b shows that, starting to 16 knot ship speed, the 900 rpm N_{MAX} Bergen engine need greater brake power, referred the 1000 rpm N_{MAX} one. This difference increase to the ship speed decreasing (more in winter navigation), this is due to the less propeller Pitch-propeller Diameter *P/D* ratio required to 900 rpm N_{MAX} engine, referred to the 1000 rpm N_{MAX} one, as visualized in Fig. 4a.

5.1. Diesel electric propulsion plants comparison

Fig. 5 reports the 900 rpm N_{MAX} (Fig. 5a) and the 1000 rpm N_{MAX} (Fig. 5b) engines working conditions of both diesel-electric plants with diesel-generators operating at constant and variable speeds (plants of Figs. 2a and 2b respectively), for all the considered ship speeds in summer navigation. In both figures the red numbers indicate the number of active diesel-generators.

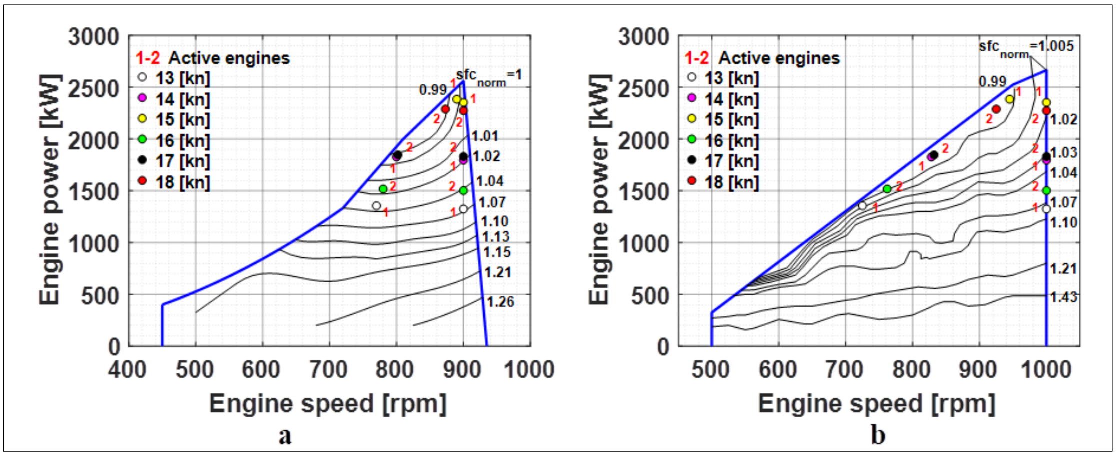

Figure 5 900 (a) and 1000 (b) rpm N_{MAX} Bergen engines normalized constant *sfc* contours and constant or variable diesel-generators speeds diesel-electric propulsion plants in summer working conditions vs ship speed on engines power-speed plan

The total MAIN-DE brake power required to the plant with diesel-generators operating at constant speed (Fig. 2a) is determined by eq. (3), while that of the plant having diesel-generators operating at variable one (Fig. 2b) is calculated with eq. (4). The number of MAIN-DE actives depend to the propulsion ad hotel required power.

In the propulsion plant with diesel-generators operating in variable speeds (MAIN-DE in Fig. 2b), once determined the total power demand (eq. (4)), and consequently the number of active diesel-generators and its brake power, the relative rotational speed selected corresponds to the condition of diesel engine minimum *sfc* to the brake power it must deliver.

The comparison between the summer navigation (Fig. 5) and the winter one (Fig. 6), put in evidence that, to ship speed parity, MAIN-DE delivered power is greater in the case of summer navigation, due to the greater electrical hotel power, as reported in Tab. 1.

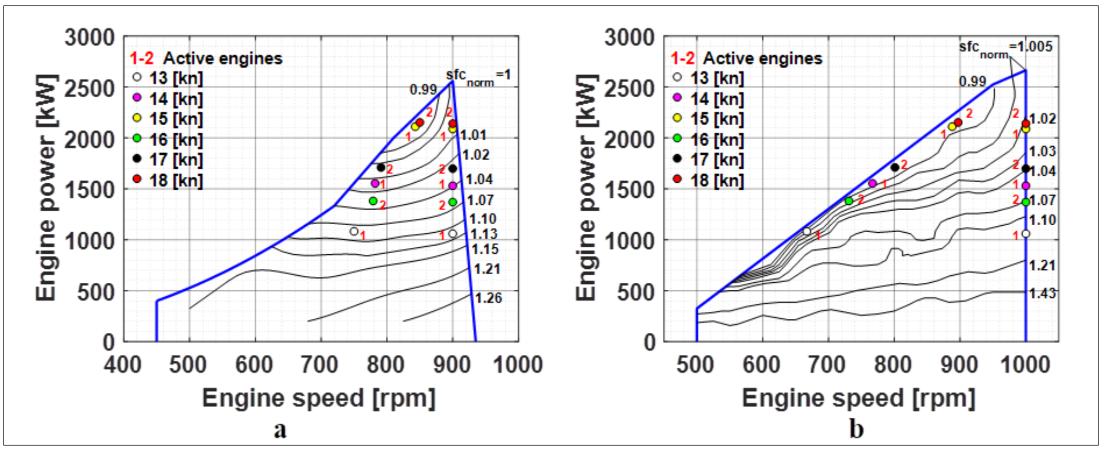

Figure 6 900 (a) and 1000 (b) rpm N_{MAX} Bergen engines normalized constant *sfc* contours and constant or variable diesel-generators speeds diesel-electric propulsion plants in winter working conditions vs ship speed on engines power-speed plan

Fig. 7 reports the comparison between the overall MAIN-DEs variable speed diesel generators (DG-vs) delivered power (P_E) versus the constant speed one (DG-cs), for several vessel speed and season (summer or winter).

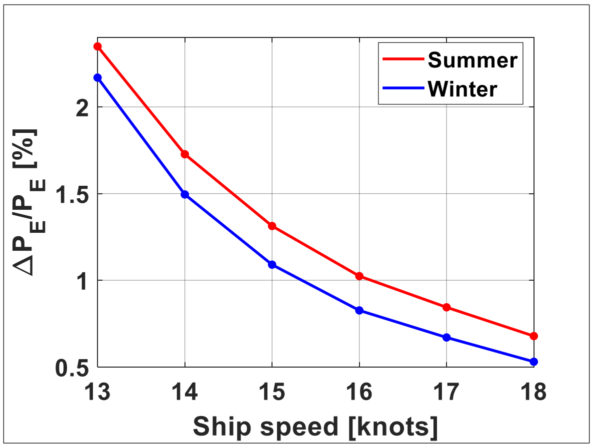

Figure 7 Variable speed diesel generators vs constant speed one overall diesel engines brake power difference for different ship speeds in summer and winter navigation

The data shown in Fig. 7 are determined by:

$$\Delta P_E / P_E \% = \frac{P_{E\,DG-vs} - P_{E\,DG-cs}}{P_{E\,DG-cs}} 100 \ [\%] \quad \text{.....................} \ (6)$$

with: $P_{E\,DG\text{-}vs}$ the overall MAIN-DEs variable speed diesel-generators delivered power and $P_{E\,DG\text{-}cs}$ the overall MAIN-DEs constant speed one.

Fig. 7 shows that the propulsion plant with DG-vs overall MAIN-DEs power is always higher, to same ship speed and season, compared to that of DG-cs one, this difference is greater in summer conditions. This is due to the greater number of voltage and frequency conversion systems present in the DG-vs plant, compared to the DG-cs one, as shown in Figs. 2b and 2a respectively. The difference increases as the ship's speed decreases, because this fact increases the power percentage of the ship's hotel load.

Before presenting and complying the following figures, it should be noted that in all considered diesel-electric propulsion plants types and engines, from the ship speed of 15 knots downwards there is the transition from two active MAIN-DEs to one, as shown also in Figs. 5 and 6. From 15 knots ship speed to descend the total power (P_E) required by the propellers and by the hotel load is less than the maximum power that can be supplied by a single MAIN-DE.

The comparison between the propulsion plant with DG-cs and DG-vs, referred the overall MAIN-DE efficiency (η_E) and nautical mile fuel consumption (m_f) is reported in Fig. 8. The data reported in this figure are calculated with:

$$\Delta x / x \% = \frac{x_{DG-vs} - x_{DG-cs}}{x_{DG-cs}} 100 \ [\%] \quad \text{..................} \ (7)$$

where: $x_{DG\text{-}vs}$ e $x_{DG\text{-}cs}$ are the generic variables referred to variable speed and the constant one diesel generators respectively.

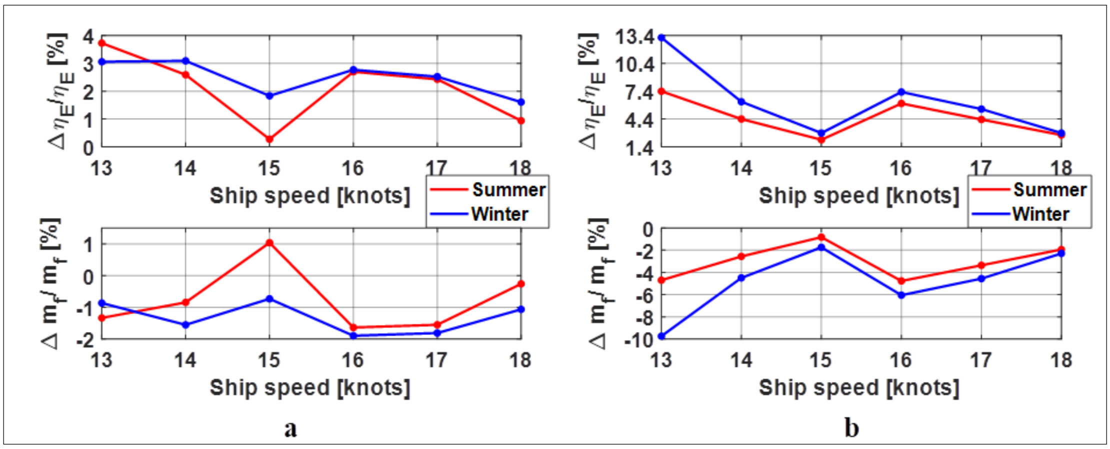

Figure 8 DG-vs plants vs DG-cs one 900 (a) e 1000 (b) rpm N_{MAX} engines efficiency and nautical mile fuel consumption percentage differences vs ship speeds in summer and winter seasons

As the regard of 900 rpm N_{MAX} engine, Fig. 8a shows that the DG-vs plant has a higher engines efficiency referring the DG-cs one, especially in winter navigation, as consequence the nautical mile fuel consumption differences versus ship speed is almost specular, with the difference that for the latter parameter the differences are approximately two percentage points lower than the engine's efficiency. This fact is due to the greater engine power, to the same ship speed, required by the DG-vs plant, as already reported in Fig. 7.

Fig. 8b put in evidence the greater advantage of the 1000 rpm N_{MAX} engine employed in the DG-vs plant versus the same engine running in the DG-cs one. This 1000 rpm N_{MAX} engine advantage, compared to the 900 rpm N_{MAX} one (see Fig. 8a), can be explained by observing its *sfc* contours trend on the engine power-speed plan in Figs. 5b and 6b, that permit to DG-vs plant to run the engine always in low *sfc* conditions, which is not possible in the case of 900 rpm N_{MAX} engine, as shown in Figs. 5a and 6a.

Fig. 9 reports the differences between the 900 and 1000 rpm N_{MAX} engines employed in the same propulsion plant type, with DG-cs (Fig. 9a) and DG-vs (Fig. 9b). Also in this case the comparison between the two engines type is carried out to overall MAIN-DE efficiency (η_E) and nautical mile fuel consumption (m_f), for different ship speeds in summer and winter seasons. The data shown in Fig. 9 are determined by eq. (5).

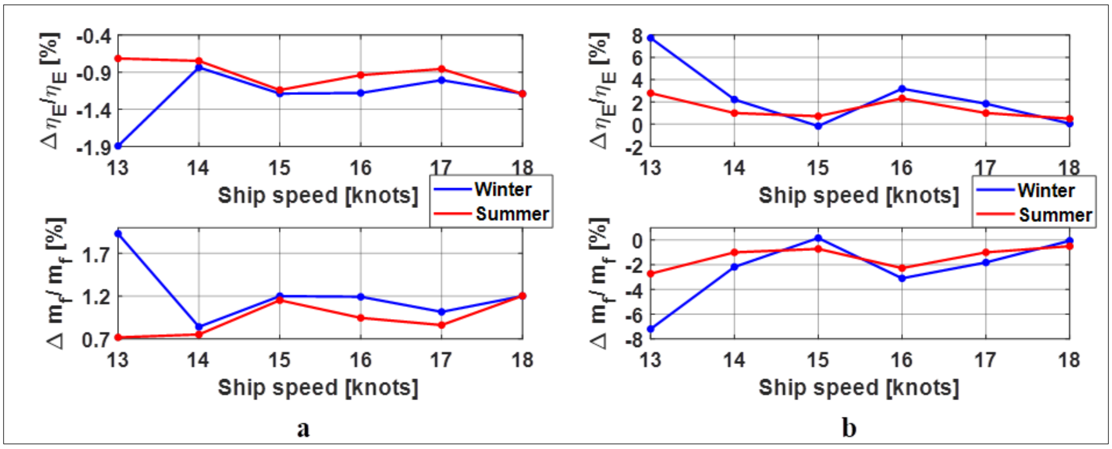

Figure 9 900 and 1000 rpm N_{MAX} engines working in the DG-cs (a) and DG-vs (b) plants comparison of efficiency and nautical mile fuel consumption percentage differences vs ship speeds in summer and winter seasons

Fig. 9a data, referred the DG-cs propulsion plant, show that the 900 rpm N_{MAX} engine works with an efficiency greater than about 1% compared to 1000 rpm N_{MAX} one, for almost all the considered ship speeds and seasons. As consequence the nautical mile fuel consumption percentage differences between the two engines are characterized by a specular trend with respect to those of the engines efficiency differences with similar values but with opposite sign.

On the contrary Fig. 9b shows that, in the case of DG-vs plant, in the summer the 1000 rpm N_{MAX} engine works with a percentage efficiency greater than about 2% referred the 900 rpm N_{MAX} one. In winter navigation the 1000 rpm N_{MAX} engine efficiency advantage becomes less constant but on average higher, at ship speeds of 14, 16 and 17 knots, and above all at 13 knots where the higher 1000 rpm N_{MAX} engine efficiency from 900 rpm N_{MAX} reaches 8%. To 15 knots of vessel speed (speed from which and for lower values only one MAIN-DE works) the efficiency of both engines type is equal (see Fig. 9b) in winter season, while in summer one the 1000 rpm N_{MAX} engine efficiency is 1% greater refer that of 900 rpm N_{MAX} one. The two engines nautical mile fuel consumption percentage differences trend is almost specular with percentage differences similar to those of the engine efficiency. In the DG-vs propulsion plant, the advantage of the 1000 rpm N_{MAX} engine, respect the 900 rpm N_{MAX} one, is due to the aforementioned particular trend of the 1000 rpm N_{MAX} engine constant *sfc* contours, reported in Figs. 5 and 6.

5.2. Diesel-electric and mechanical propulsion plants comparison

Before starting the comparison between diesel-electric and ship mechanical propulsion plants, is useful reports the main comparison parameters between the two Bergen engines versions, employed in the mechanical propulsion plant of Fig. 1. The data reported in Fig. 10, taken from the authors paper [19], show, in terms of engine efficiency and ship nautical mile fuel consumption in calm sea, a 900 rpm N_{MAX} engine less advantage, referring the 1000 rpm N_{MAX} one, in

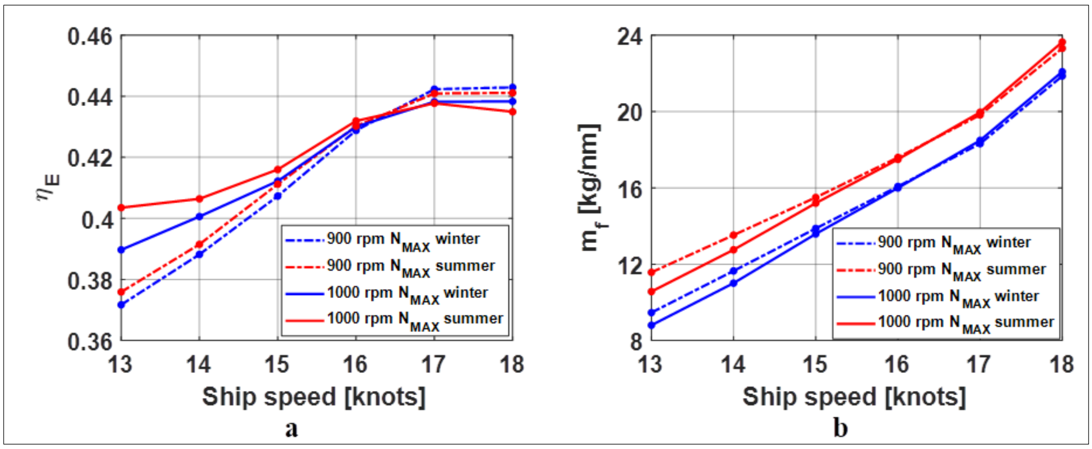

Figure 10 Mechanical propulsion plants 900 and 1000 rpm N_{MAX} MAIN-DEs overall efficiency (a) and nautical mile fuel consumption (b) vs ship speed in summer and winter navigation

the 17 and 18 knots vessel speed, a substantial parity at 16 knots, and a 1000 rpm N_{MAX} engine advantage starting to 15 knots (that grows more and more as ship speed decreases, mainly in summer season) referring the 900 rpm N_{MAX} one.

Fig. 11 shows, in percentage terms, the DG-cs and mechanical propulsion plants (a) and DG-vs and mechanical one (b) overall MAIN-DEs brake power difference, for different ship speeds and summer or winter seasons. The data presented in the figure are determined by:

$$\Delta P_E / P_E \% = \frac{P_{DE} - P_{MECH}}{P_{MECH}} 100 \; [\%] \quad \text{.................} (8)$$

with: P_{DE} e P_{MECH} are the MAIN-DEs brake power of diesel-electric and mechanical propulsion plants respectively.

The engines brake power percentage differences determined by eq. (8) are referred to the same Bengen engine type (900 rpm N_{MAX} engine or 1000 rpm N_{MAX} one).

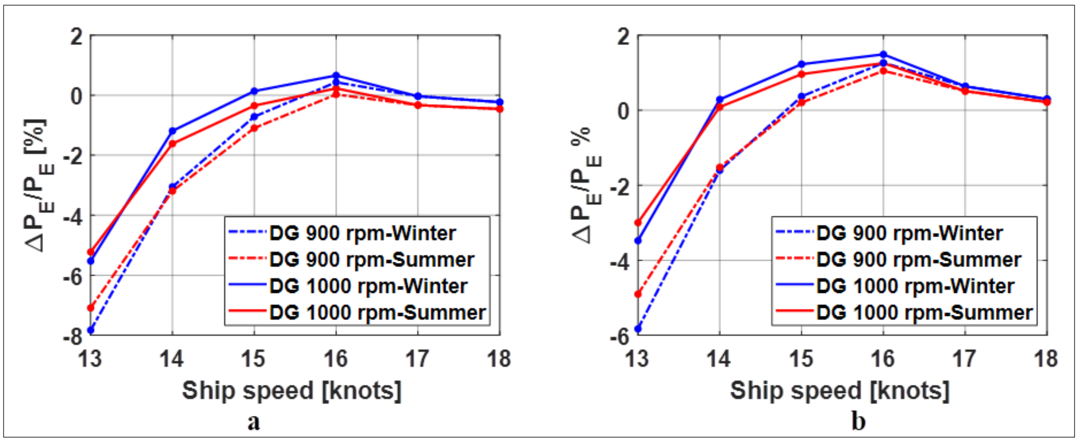

Figure 11 Overall MAIN-DEs brake power percentage difference between DG-cs and mechanical propulsion plants (a) and DG-vs and mechanical one (b) vs ship speeds in summer and winter seasons

As reported in Fig. 11a, to 17 and 18 knots of vessel speed the DG-cs plant the MAIN-DEs brake power is slightly less than that of mechanical plant, especially in the summer season. This is due mainly to FPP greater efficiency, used in diesel-electric plants, compared to those CPP of the mechanical one. Starting to 16 knots, to the ship speed reducing the mechanical MAIN-DEs brake power increases more and more compared to that of DG-cs, mainly in the case of 900 rpm N_{MAX} engine. This is due to the mechanical plant CPP *P/D* reduction to the vessel speed decrease (as visualized in Fig. 3), which results in an ever lower propeller efficiency.

In the DG-vs plant, already from 18 knots of ship speed the MAIN-DEs brake power is greater than that of the mechanical plant, as shows in Fig. 11b. This is due to the greater number of frequency and voltage conversion components present in the DG-vs plant referred the DG-cs one (see Figs. 2a and 2b), which entails an electricity transport line lesser efficiency from generators to propellers and HEL of the DG-vs plant respect the DG-cs one. Starting from a ship speed of 15 knots and with its decreasing, the mechanical plant advantage becomes less and less, until it cancels out and becomes negative at ship speeds slightly less than 15 knots for the plant with 900 rpm N_{MAX} engines, and about 14 knots in the plant employing the 1000 rpm N_{MAX} ones; this for the above mentioned reasons regarding the propeller *P/D* reduction as the ship speed decreases in the mechanical plant.

Fig. 12 reports the percentage difference between the DG-cs and mechanical propulsion plants pertinent the overall MAIN-DEs efficiency and nautical mile fuel consumption, referred the 900 rpm N_{MAX} engines (a) and 1000 [rpm] N_{MAX} one (b), for different ship speeds in summer and winter seasons. The data reported in Fig. 12 are determined by eq. (8).

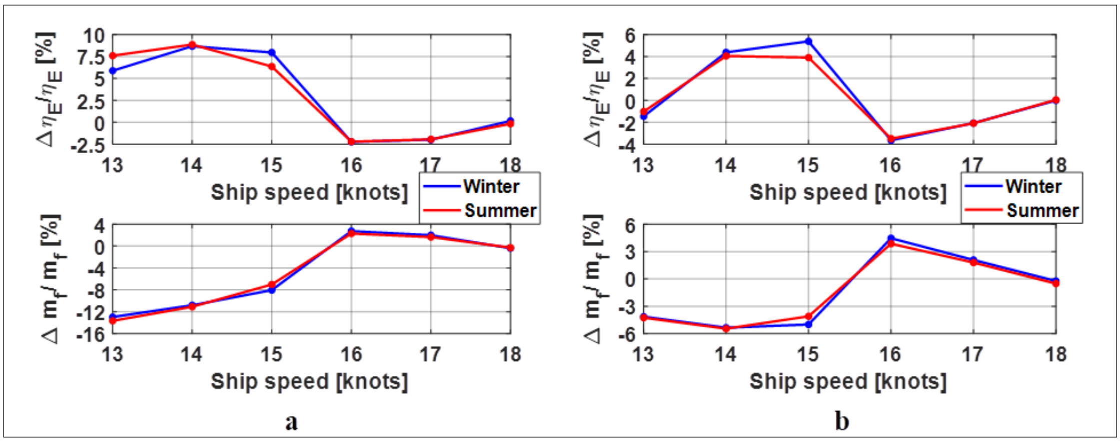

Figure 12 DG-cs versus mechanical propulsion plants overall MAIN-DEs efficiency and nautical mile fuel consumption percentage differences for 900 rpm N_{MAX} engines (a) and 1000 [rpm] N_{MAX} one (b) vs ship speeds in summer and winter seasons

Fig. 12a, referred to 900 rpm N_{MAX} engines, show that to 18 knots ship speed the mechanical plant MAIN-DEs efficiency is practically equal to those of the diesel-electric one, and a little better at 17 and 16 knots. For speeds equal or less than 15 knots the only active engine efficiency of the diesel-electric plant becomes significantly higher than that of the two engines of the mechanical one, because at low ship speeds the only MAIN-DE active of DE plant works under higher load conditions and therefore with less *sfc* (Figs. 5a and 6a). This occurs in both winter and summer navigation and obviously also reflects on nautical mile fuel consumption, as shown in Fig. 12a.

The data shown in Fig. 12b relates to the comparison between the same previously examined plants, with the only difference that both plants use the 1000 rpm N_{MAX} engines. The considerations that are drawn from the figure are generally similar to those relating the plants with 900 [rpm] N_{MAX} engines (Fig. 12a). The only substantial difference between the two engines is that, to ship speeds equal to or less than 15 knots, the differences between the mechanical propulsion plant and the diesel-electric one are smaller for the 1000 rpm N_{MAX} engines, compared to 900 rpm N_{MAX} ones (Figs. 12b and 12a respectively).

Fig. 13 reports the same data comparison presented in Fig. 12, in this case referred to mechanical and DG-vs propulsion plants. The data presented in the figure are calculated with eq. (8).

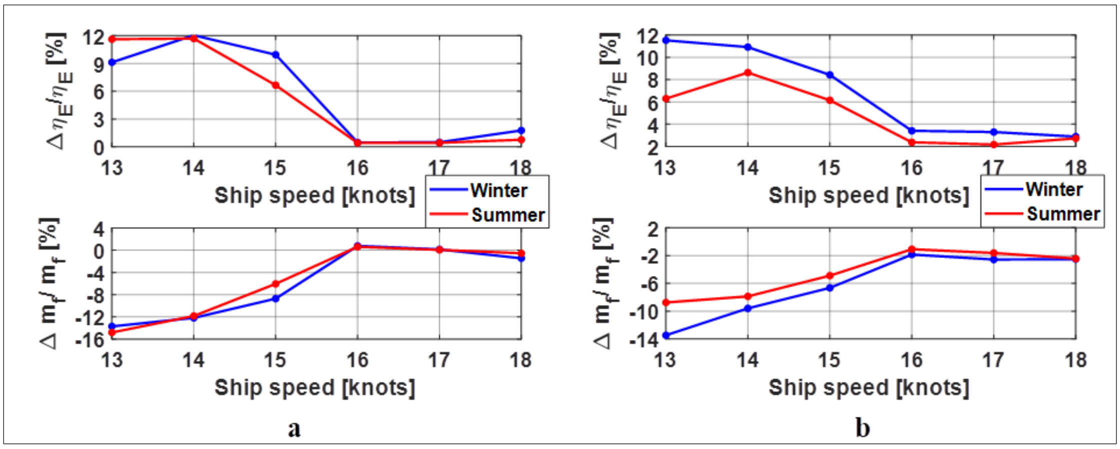

Figure 13 DG-vs versus mechanical propulsion plants overall MAIN-DEs efficiency and nautical mile fuel consumption percentage differences for 900 rpm N_{MAX} engines (a) and 1000 [rpm] N_{MAX} one (b) vs ship speeds in summer and winter seasons

In the DG-vs propulsion plants, the possibility of varying the MAIN-DEs rotational speed allows these engines to run in the minimum *sfc* condition at the power required by the plant (to ship propulsion and HEL), as shown in Figs. 5 and 6. The same engines installed in the mechanical plant works in the same conditions (visualized in Fig. 3), and therefore efficiency, of the case considered in Fig. 12. This entails a greater efficiency difference in favour of the MAIN-DEs operating in the DG-vs propulsion plants, compared to DG-cs one, as can be seen from the comparison between Figs. 12 (pertinent the DG-cs propulsion plants) and 13 (referred the DG-vs ones). This fact has a positive impact also in the nautical mile fuel consumption, as reported in Fig. 13 with respect to Fig. 12. Similarly to DG-cs plants, the DG-vs plants advantages, compared to the mechanical ones, occurs mainly at ship speeds equal or less than 15 knots (see Fig. 13), for the same reasons already reported in the comment of Fig. 12.

The greatest difference between summer and winter conditions in the plant with 1000 rpm N_{MAX} engines (Fig. 13b), compared to 900 rpm N_{MAX} one (Fig. 13a), especially at low ship speeds, is due mainly to the already remarked difference in the *sfc* contours between these two engines (Figs. 6a and 6b).

At reduced ship speeds, the lesser difference observed between the DG-vs and mechanical plants using the 1000 rpm N_{MAX} engine, compared to that employing the 900 rpm N_{MAX} one, mainly in winter navigation (see Fig. 13), is due to fact that at these ship speeds the mechanical plant with 1000 rpm N_{MAX} engines is characterized by an engines efficiency, and therefore nautical mile fuel consumption, significantly better referring the mechanical plant employing the 900 rpm N_{MAX} engines, as seen in Fig. 10.

6. Conclusion

The paper reports a comparison between mechanical and diesel-electric propulsion plants for a small cruise ship propulsion. Both versions of the prime movers Rolls-Royce Bergen marine diesel engine are employed in all plants. This engines type, characterized by a similar rated power, presents very different power lint curve and specific fuel contours trend in the operating diagram. To this reason, both diesel-electric plants with diesel-generators operating at constant and variable speed have been tested. The propulsion plants comparison is carried out with reference to summer and winter seasons (conditions that differentiate the ship's electrical consumption and the engines efficiency).

The main considerations drawn from the propulsion plants comparison can be summarized as follows:

- To lover ship speeds (13-15 knots) the mechanical propulsion plant employing the 1000 rpm NMAX engines is characterized by a less brake power Fig. 4b and higher efficiency Fig. 3, and consequently a lower nautical mile fuel consumption, compared to the same plant type with 900 rpm NMAX ones. These differences are also due to 1000 rpm NMAX engines more favorable propeller P/D law referring the 900 rpm NMAX ones (Fig. 4a). In the 16-18 knots ship speed range the two engine types performances are practically equivalents.
- In the diesel electric plants, despite that in the DG-vs one the power required to the MAIN-DEs progressively increases as the ship speed decreases, compared to that of the MAIN-DEs operating in DG-cs plants Fig. 7, the 1000 rpm NMAX engine sfc contours particular trend on power-speed plan allow an advantage, in terms of engine efficiency and fuel nautical mile consumption, compared to the same engines operating in the DG-cs plant, at 13-14 knots ship speeds and in lesser form at 16-17 knots one. In the plants with 900 rpm NMAX engines, the engine efficiency and nautical mile fuel consumption differences between DG-vs and DG-cs plants are much smaller than in the plants with 1000 rpm NMAX engines (Fig. 8).
- Always in the diesel electric plants, the efficiency and nautical mile fuel consumption differences between the two engines types (900 and 1000 NMAX) are not very consistent at all ship speeds in the DG-cs plants (Fig. 9a), in the DG-vs ones, the 1000 NMAX engine, due its more favourable specific fuel contours trend in the operating diagram, has better efficiency and nautical mile fuel consumption at lower ship speeds, referring the 900 NMAX one, especially at 13 knots ship speed (Fig. 9b).
- The comparison between the mechanical and the diesel electric propulsion plants shows generally little differences in the engines efficiency and nautical mile fuel consumption at high ship speeds (16-18 knots). The DG-cs diesel electric plants have better engine efficiency and nautical mile fuel consumption, vs the mechanical ones, to ship speed equal or less to 15 knots, especially with the 900 NMAX engine (Fig. 12). In the same ship speeds interval, the first mentioned diesel electric plants advantage vs the mechanical ones become more consistent in DG-vs plants, regardless to the used engine type (900 or 1000 NMAX one) (Fig. 13).
- The season in which the navigation takes place does not seem have a great influence on the plants data comparison. Some advantages are found more often in the winter season, especially in favour of the 1000 rpm NMAX engine (Figs. 8 and 13).

The greater efficiency of the diesel electric propulsion plants, compared to the mechanical ones, detected at lower ship speeds (13-15 knots), is due mainly to the fact that in this ship speeds interval only one MAIN-DE is active, which works under high load conditions, and therefore with reduced *sfc* values. (Figs. 5 and 6). In the same ship speeds interval, in the mechanical plants both engines are actives, therefore they operate at reduced power and therefore with high *sfc*. (Fig. 3) At high ship speeds (16-18 knots), in all the analyzed diesel-electric propulsion plants both MAIN-DEs are active, as it happens also in the mechanical plants. To this reason, at high ship speeds, the diesel-electric and mechanical propulsion plants efficiency, and thus the nautical mile fuel consumption, differ little from each other.

In conclusion, the examined diesel electric propulsion plants are convenient, respect the mechanical ones, in lower navigation speed, while are not convenient for travel with ship speeds near to the maximum one. These considerations, obtained from the comparison between two marine engines fuelled with traditional liquid fuels are also valid for marine engines fed with alternative fuels, such as natural gas, or likely to be used in the near future, such as: methanol, ammonia and hydrogen.

Compliance with ethical standards

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Disclosure of conflict of interest

The authors declare no conflict of interest.

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