ANCHOR WINDLASSES: A DESIGN PROPOSAL TO STANDARDISE REGULATIONS

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SUMMARY

Shipbuilding is an increasingly competitive field. As the key players in this activity, shipyard managers have to make their counterparts in related industries aware of two conflicting needs. On the one hand, they must produce high quality components that are technologically advanced. On the other, they need equipment that lets them build vessels at a reasonable price. With this second aim in mind, it would be useful to standardise both design and manufacturing processes. At the same time, ship owners always impose regulations. The manufacturers of deck equipment must therefore adapt their product designs to comply with these regulations, some of which have aspects in common. However, guidelines also differ on many points. This makes it difficult to reach desirable levels of standardisation in equipment design. The situation can be summed up in a phrase: for each vessel, a specific design. This article will first provide a comparative overview of current legislation. A proposal for anchor windlass design is then presented in an attempt to make the various regulations more cohesive. The objective here is to reach an acceptable degree of standardisation and, consequently, lower costs by applying economies of scale.

NOTATION

a	band of the brake
Q	minimum breaking load for the chain
C	traction coefficient
d_c	chain diameter (mm)
D_t	drum diameter (mm)
F_r	cable-lifter-retaining force (Kg)
h	anchoring depth (m)
i	reduction ratio
K_1	coefficient for calculating chain breaking tension
K_2	coefficient for calculating chain breaking tension
L	chain length (m)
m	geometric coefficient for band on brake
M_f	braking torque
N_m	r.p.m. drive motor
P	pressure on brake material
p_a	anchor weight (kg)
p_c	chain weight (kg)
$P_{max.}$	maximum admissible pressure on brake lining (N/mm^2)
F	pump flow $(l/min.)$
V_g	cylinder volume of motor (cm^3)
Vs	hoisting speed (m/min)
η_m	windlass efficiency
η_e	hawse pipe efficiency
η_t	transmission efficiency
η_v	volumetric–efficiency (0.8 – 0.95)
θ	angle contained in band brake radials (rad)
μ	friction coefficient for brake lining

1. INTRODUCTION

1.1 CURRENT REGULATIONS

Each ship will be equipped with components that help it remain in one position, from which it can be anchored. This equipment makes up the anchoring gear. Some of it will be fixed onto the deck: the windlass, stopper, hawse pipe and chain locker. Other elements comprise the moving devices needed for anchoring: the chain cable, which include anchors, chains and accessories. Nevertheless, all of the gear in the anchoring system will be conditioned by the shape and size of the vessel being developed, as well as the regulations that correspond to that vessel. This is the case with every aspect of the project.

The people carrying out a design process must always adapt themselves to the multitude of regulations that are directly applicable, as well as to the ones specified by the classification society chosen by the ship owner. These regulations have been developed to preserve the ship's integrity, which is a common objective. It may be concluded, therefore, that everyone is thinking along the same lines in terms of which aspects need to be dealt with in the regulations and the best way to do so.

Nevertheless, there are great differences in how classification societies handle the operation and design of windlasses. An overview of these differences is provided here. Among the societies included in this table are: American Bureau of Shipping- (ABS), Bureau Veritas- (BV), Det Norske Veritas-(DNV), Germanicher Lloyd- (GL), Rina (RN) and Lloyd´s Register of Shipping- (LR). The International Association of Classification Societies (IACS), standardise, in part, their requirements related to anchoring, mooring and towing in its document "Requirements concerning mooring, anchoring and towing" (2007). However, this document only focuses on equipment number and anchor testing. There is no IACS document to standardise design regulations for the piece of equipment used in manoeuvring the mooring lines: the anchor windlass.

The International Standard Organisation (ISO) is made up of national entities from over 156 countries, including most of the European ones. Technical Committee 8

(TC8) deals with ship and marine technology. This committee has developed standards related to the equipment being examined here.

Carral et al [1] looks at a situation that is similar to the one discussed here. In this case, the study is about the design of a towing winch and examines the third largest bibliographic source for naval architecture, the one provided by the International Maritime Organisation (IMO). In contrast, the IMO has no regulations on anchoring.

There is a wide range of regulations and they have marked differences. Consequently, the project manager of a team is conditioned by the vessel's classification society at the moment of carrying out the design process. This circumstance is a far cry from design and manufacture processes, in which recommended economies of scale are sought by standardising components.

ISO Standards become the EN-ISO when they are accepted by the European Committee for Standardisation. By studying and comparing these standards, it is possible to establish a common core from which a common regulation can be developed for the future. Thus, there would be significant savings in equipment design and manufacturing costs. These reductions will affect three different areas: design, supplies and production. The first of these- design- will be affected because there is a greater number of common components and therefore fewer conceptual design drawings. Secondly, fewer suppliers are needed, especially when it comes to cast iron pieces. Economies of scale are involved with the third area of savings. A greater number of pieces is produced for each model. .As for assembling the equipment, the learning curve concept comes into play. Thanks to this, there are reductions of around 30% when four units of the same model are built. When all of these factors are taken into account, equipment costs may be decreased by over 25%.

2. COMPARING A SELECTION OF RELEVANT REGULATIONS

As mentioned earlier, this selection will include both classification society regulations and ISO standards. Simply looking at the former will be of great interest. In the case of tugs and their anchoring gear, Allan [2] has carefully examined the scope of each regulation. By doing so, it is possible to reach the conclusion that the regulations mainly cover operational aspects. Scant reference is made to the parameters of windlass design and manufacture [1]. The next section looks at how various regulations deal with calculating the operative parameters for the equipment related to traction and braking force.

For anchoring windlass design, both classification society and ISO propose using a concept of minimum breaking load for the chain (Q) as a variable on which the most relevant features of the equipment depend. Among these are traction and braking. This value will be obtained from the three tables included in the regulations for the three chain qualities normally considered (Q_1, Q_2, Q_3), However, Lloyd´s Register (Eq 1) and Bureau Veritas (Eq 2) propose an alternative as they take into account the following formulae for this calculation:

$$Q\ (N) = K_1 \cdot d_c^2\ (44 - 0.08\, d_c) \tag{1}$$

$$Q\ (KN) = 9.807 K_2 \cdot d_c^2\ (44 - 0.08\, d_c)10^{-3} \tag{2}$$

The coefficient values are:

Table 1 Coefficient K for calculating breaking force for the chain.

Grade	K_1 (Lloyd´s Register of Shipping)		K_2 (Bureau Veritas)
	Chain stopper	No chain stopper	
Q_1	4.41	7.85	1
Q_2	6.18	11.0	1.4
Q_3	8.83	15.7	2

For calculating windlass traction, the traction coefficient C is employed. This is the value on which the classification societies and ISO coincide. It reflects the grade of chain to be used. To define the nominal traction for the windlass, coefficient C is multiplied by the square number for the chain diameter. The following table provides the coefficients given by the main classification societies and ISO standard 4568 [3].

$$T = C \cdot d_c^2 \tag{3}$$

Table 2 Coefficient C to calculate the nominal traction for the windlass

Chain grade	DNV	Lloyd´s Register	Bureau Veritas	RINA	Germanicher Loyd´s	ISO 4568
Q_1	37.5	37.5	49.8	37.5	49.8 (*)	37.5
Q_2	42.5	42.5	56.5	42.5	56.5 (*)	42.5
Q_3	47.5	47.5	66.5	47.5	66.5 (*)	47.5

* Germanicher Lloyd´s uses this as a formula for anchorage depth greater than 100 m.

$$T = d_c^2 \cdot [C + 0.218 \cdot (h - 100)] \tag{4}$$

The braking force acts on the cable- lifter so that the gear does not slip. In the case of anchoring windlasses, the braking force specified in the regulations are defined as a percentage related to the chain's breaking load (Q), as shown in Table 3.

Table 3 - Braking force as a percentage of breaking load for the chain

Force	DNV / G.L / ISO/RINA	L.R.S.	Bureau Veritas	ISO 4568
With chain stopper	45 %	45 % (*)	45% (**)	45 %
W/o chain stopper	80%	80%	80%	80%

* Calculating breaking load with Equation (1)
** Calculating breaking load with Equation (2)

Until now, this study has focused on aspects related to how the windlass is operated. However, reference should also be made to other factors for design and manufacture. Here, the standard that comes into play is ISO 4568–2006 -"SHIPBUILDING SEA – GOING VESSELS-WINDLASSES AND ANCHOR CAPSTANS" [3]. It is the one that places the greatest attention on windlass design parameters. Table 4 summarises classification society regulations and then compares these with content from ISO 4568.

3. PROPOSAL FOR STANDARDISATION REGULATIONS AND WORKING HYPOTHESES

This proposal for achieving standardisation consists of adopting broad criteria based on the content in ISO 4568. The ISO standard has features that are in line with the classification societies related to traction and braking force. At the same time, the ISO has suitable design and manufacture specifications in terms of the cable- lifter, speed control, warping end and control devices, while the classification societies stand out for being silent on these matters (Table 4).

Table 4 – Comparison between classification society regulations and ISO standards on anchoring windlass design

	Classification soc. reg.	*ISO 4568–2006*
DESIGN AND MANUFACTURE		
Cable- lifter	Indicates they must be declutchable/ no mention of geometry	Minimum of 5 points, in accordance with ISO 1704. Declutchable
Warping end	Not mentioned	Optional, to comply with ISO 6482
Strength requirements	With chain stopper: 0.45 x the chain's breaking load; no stopper: 0.80 x the breaking load	With chain stopper: 0.45 x the chain's breaking load; no stopper: refer to classification soc. standards
Control braking system	Some call for brake by electric windlass	With a brake capacity equivalent to 1.5 of nominal traction when electrically operated; 1.3 x when operated by other means
Emergency stop	Some require this	Within reach of equipment
Protection	Some call for operational control (electric and hydraulic)	Must have torque limit switch
Control devices	Not mentioned	Manual, returning to neutral position
Speed control	Not mentioned	Adjustable from zero to nominal speed
OPERATION		
Nominal traction	Related to chain diameter and grade, but there are different procedures for calculating chain grade	Take into account: chain diameter value, chain grade and anchorage depth (under or over three shots)
Maximum traction	1.5 x nominal traction	1.5 x nominal traction
Nominal speed	No less than 0.15 m/s.	No less than 0.15 m/s.

In the European Union (EU), more uniform standards help eliminate technical obstacles within the home market. They make it possible to evaluate product conformity through a system of accreditation, certification, testing and calibrations. Product development is also influenced by economic and technical reasons given that innovation must go hand and hand with this normalisation. For this reason, the European Committee for normalisation is increasingly merging ISO and EN –ISO standards in the field of ship construction, pleasure craft, equipment and machinery.

Fulfilling standardised regulations is one step. Another involves accepting hypotheses based on rational assumptions. With these two steps it is possible to establish initial design conditions suitable for the chosen calculation process. In the absence of other conditions specified by the ship owner, initial ones will guide windlass design.

3.1 PROPOSAL FOR MORE COHESIVE REGULATIONS

The proposal for making windlass design more cohesive will include a set of minimum conditions the gear must fulfil so that the vast majority of existing standards are embraced. In this way, these standards could make up a proposal for standardising regulations:

- When the windlass is idle, the cable- lifter is declutched and the brake is activated. The mechanical components of the windlass, including the foundations, must be capable of withstanding the pull applied on the pitch circle diameter of the cable- lifter equal to 45 % of the chain's breaking load [3].
- If the windlass is to be employed without the chain stopper, the static load must be equal to 80 % of the chain's breaking load [3].
- When the windlass is moving, and with the aim of considering the dynamic effects resulting from the anchoring manoeuvre, its mechanical components must be capable of withstanding a load perpendicular to the windlass's axle. The application point of this load is to correspond with that of the pitch circle diameter of the cable- lifter cable-lifter. Its value is to be equal to the one corresponding to the following expression [3]:

$$71.25 \cdot {d_c}^2 \text{ (N)} \tag{5}$$

- The braking force will be such that, if the windlass is going to work in conjunction with the stopper chain, with the cable- lifter declutchable from the motor, the brake must be capable of withstanding- without slipping- a static load of corresponding to 45 % of the chain's breaking load, this being applied along the pitch circle diameter of the cable- lifter [3].
- The brake holding load for a windlass without the chain stopper will be 80 % of the chain's breaking load [3].
- Should the brakes be activated manually, the levers must have a maximum scope of movement of 600 mm. If the brakes work by means of a hand wheel, this wheel must move in a clockwise motion [5].
- With power brakes, the system must be designed in such a way that, if the equipment's power supply fails, the brake will immediately and automatically start working [3].
- The cable- lifter will have a coupling device that allows it to engage and disengage from the prime mover when it is not in use. Its shape and dimensions must respect ISO 1704 [6].
- If there is a reduction gearbox, it must be lubricated on a regular basis and to a sufficient level if the equipment tilts up to 15 degrees [5]. Furthermore, it should also have a display panel for the oil level. Along those points in which grease lubrication is needed, there must be lubricating devices, such as grease nipples that are adjusted to ISO 7824 specifications. The gears must not be lubricated with grease.
- If there are hydraulic windlasses, the system's working pressure must not exceed 70% of the maximum permissible continuous pressures specified by the component's manufacturer. When compared with all the other components of the system, it must have the lowest nominal pressure [4].
- If the gearbox is reversible or it is a direct drive windlass, its system must ensure that the brakes on the motor are activated whenever the motor stops, intentionally or not. With the particular case of electrically operated windlasses, it is essential to use an electrical negative brake whose braking force is 1.5 times nominal traction [3]. For hydraulically operated windlasses, brake valves are used. They are to have a braking power that is 1.3 times normal traction.
- Hoisting speed must be 9 m/min, which will be tested by raising the anchor and chain from 85 m. to 57.5 m. in depth [3]
- If there is a warping end, ISO 6482 [7] will apply.

3.2 WORKING HYPOTHESES

In order to design the windlass, it is essential to establish a range of initial conditions. These have to be carefully thought through and then incorporated into the set of standardised guidelines presented in point 3.1.

- When the windlass is in motion, its components must be capable of withstanding a force that is, in minimum terms, over 25 % of the nominal force during its expected service life, as indicated by the manufacturer.[4]
- If it is a dual anchor windlass, in addition to the forces mentioned earlier, it must be capable of withstanding 50 % of the total of each one of the forces applied on the cable- lifters [4]
- The windlass motor must be able to operate for at least 30 minutes at the power that corresponds with the following value, calculated for the value p_c corresponding to four shots of the chain:

$$P(w) = \frac{8.7 \cdot ((p_a + p_c) \cdot \mathrm{Vs}}{60 \cdot \eta_m \cdot \eta_e} \tag{6}$$

- If there are no data about how much the chain and anchor weigh, the windlass motor has to be capable of operating for 30 minutes at the following power [4]. Please note that the expression does not comply with GL and BV requirements, in which case, the coefficient will be 66.5.

$$P(w) = \frac{47.5 \cdot {d_c}^2 \cdot \mathrm{Vs}}{60 \cdot \eta_t} \qquad (7)$$

- The windlass motor must be able to operate for at least two minutes, at a power that exceeds 50 % of the one calculated in the section above [3].
- The windlass motor and its operating and control systems must be prepared to withstand being started up six times in 30 minutes with a resting period of 1 hour [4].
- The average hoisting speed will not fall below 9 m/min., nor go above 12 m/min. while there is nominal rotation and the motor is working at full load [3,4].
- Calculations for ball bearings will comply with ISO 281, at a 90% reliability level and with an effective service life of 10 years. Lubricating options include grease, an oil bath or splash lubrication. If grease chambers are chosen, these will then have, through easy access, grease nipples that comply with ISO 7824.
- Bearings are calculated at a reliability level of 90 %. Their effective service life must be five years under nominal loads and speeds [8].
- The axle, clutch, cable- lifter and gearbox calculations are to have a 99% reliability level. Their fatigue during their service life will be along the same lines as the one for the windlass, and under a nominal workload [20].
- To ensure that the lines and ropes last longer, if the windlass has warp ends, their diameter will be six times that of the rope and 16 times that of the line [10].
- If it is electrically operated, it is advisable to have a squirrel cage, three-phase asynchronous motor. It must have, as a minimum, F class insulation and B class heating. The degree of protection should be (UNE 20324 – CEI 144) [11] or [12] IP-560 above deck and IP-540 in other cases [4].

4. DESIGN PROCESS FOR WINDLASS BASED ON MORE COHESIVE REGULATIONS

The first step here entails calculating the equipment number using the values tabulated by the International Association of Classification Societies (IACS) [13]. These values are related to diameter, total length of the chain cable and anchor mass. They help determine the anchoring gear that the windlass must manoeuvre. By considering the standardised regulations and working hypotheses, it is possible to determine the main characteristics of the windlass in accordance with the data provided by the client. On the whole, the minimum data needed are:

- Windlass type and operation.
- Number of anchoring lines.
- Chain diameter and grade.

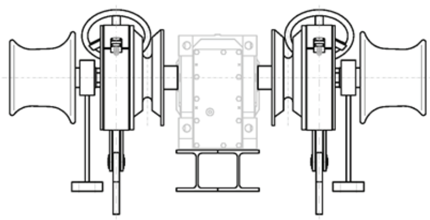
Additional data are needed to complete basic information: hoisting speed, whether there are auxiliary warping ends and how the windlass is operated.

To define an anchoring windlass, these parameters must be considered [4]:

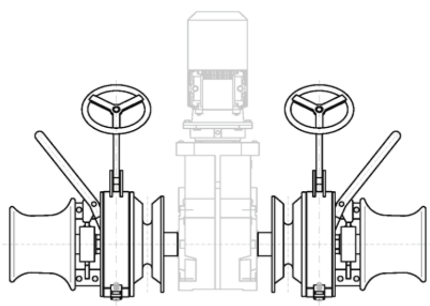
- Windlass type (single or symmetrical double cable- lifter, combined or not mooring device).
- Operational and technical features of the gearbox in terms of reduction ratio.
- Average and instantaneous power of the windlass motor.
- Geometry and dimensions of cable- lifter.
- Warping end dimensions.
- Brake type and dimensions.

4.1 WINDLASS TYPES

Windlasses can be classified into two groups: dual anchor devices with two anchoring lines and single anchor ones, with only one anchoring line. At the same time, the latter group can be sub-classified as either horizontal or vertical according to their position in relation to the transmission axle of the cable- lifter. In general dual-anchor windlasses (Figure 1) are used with small chains, whose diameters are between 22 and 30 mm. With very small chains, whose diameters are below 20.5 mm, and those whose diameters exceed 30 mm, single-anchor windlasses are used [9]. (Figures 2 and 3).

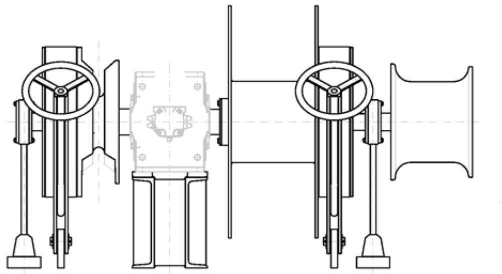


Rear view

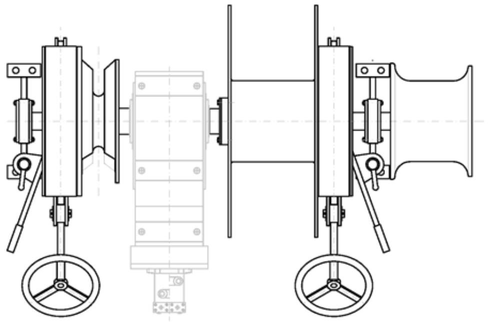


Frontal view

Figure 1 – Double anchor windlass for a 26 mm chain

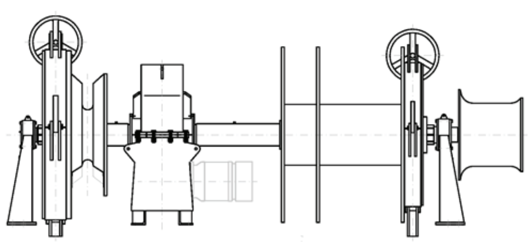


Frontal view

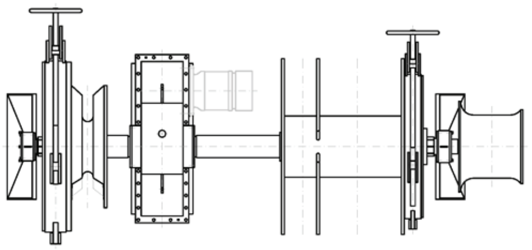


Rear view

Figure 2 – Single anchor windlass for a 40 mm chain and mooring reel



Frontal view



Rear view

Figure 3 – Single anchor windlass for 58 mm chain and split mooring reel

4.2 OPERATING SYSTEM; TRANSMISSION RATIO

In terms of the way they operate, the most common systems are electric and hydraulic. Among electric motors, the asynchronous ones with alternate current stand out. The hydraulically-operated ones mostly use rapid axial pistons motors connected to variable flow pumps. Radial piston motors are suitable for higher powered equipment [14], as the reducer gear is smaller. However, they have the drawback of being expensive and difficult to maintain. Both systems have one point in their favour: they make it possible to vary the hoisting speed. Moreover, they offer a constant torque, which is independent of the speed.

On the other hand, small windlasses, with chains up to 30 mm, use asynchronous motors that have softstarters. For chains between 30 and 70 mm, there is a preference for hydraulic systems with axial pistons or electric systems with frequency converters. When chains are over 70 mm, hydraulic systems predominate; these have radial pistons.

In general, windlasses have a mechanical reducer gear and it is necessary to determine their transmission ratio. When the transmission ratio for a reducer is the quotient between the number of times the input shaft, near the operating motor, rotates and the number of times the output shaft, near the cable- lifter, turns, the following expression works for the calculation [15]:

$$i = \frac{N_m}{N_b} = \frac{4.10^{-2} N_m \cdot d_c}{V_s} \quad (8)$$

With electrically operated devices, the motor speed corresponds with the one that is under full load, 1500 rpm. In the case of windlasses with auxiliary warping ends for mooring manoeuvres, if the required speed is around 30 m/min [10], frequency converters are particularly interesting.

If a hydraulic system is chosen, the motor speed will depend on cylinder volume, as well as the flow supplied by the pump that feeds it, that is [16]:

$$N_m = \frac{F \cdot 1000}{V_g \cdot \eta_v} \quad (9)$$

4.3 AVERAGE AND INSTANTANEOUS POWER OF THE WINDLASS

The windlass motor's average power will depend on its number of cable- lifters, the size and grade of its chain, its hoisting speed, the mechanical efficiency of the motor and the geometry of the Hawser pipe. If the data known about the anchoring line include its cable diameter, the following formula can be applied to obtain the average power [4]:

$$P_m(w) = \frac{63.7 \cdot K \cdot V_s \cdot d_c^2}{60 \cdot \eta_m} \quad (10)$$

The hoisting speed being V_s, values between 9 and 11 m/min. will be taken. Therefore, the value for K will be 0.91 for single cable- lifter windlasses and 1 for double cable- lifter models, while the windlass efficiency will range between 0.5 – 0.7 [4]. If data are available for chain grade and length, anchor weight, the performance of the hawse pipe (0.5 – 0.7) as well the reducer gear, it will be better to use the following formula [4]:

$$P(w) = \frac{85.3 \cdot (P_a + 0.02 \cdot L \cdot d_c^2) \cdot v_s}{60 \cdot \eta_m \cdot \eta_e} \tag{11}$$

Windlass efficiency is generally not known, but it is possible to obtain approximate this value by multiplying the unitary performance for each movement of its gears and for the cable- lifter. As a guide, these typical performance rates may be cited t [4]:

- A pair of cylindrical gears with a transmission ratio i $\leq$ 8: η = 0.98 – 0.95
- A pair of conical gears with a transmission ratio i $\leq$ 10: η = 0.95 – 0.90
- An arrangement of reversible worm gears with endless screws η = 0.5 – 0.7
- An arrangement of irreversible worm gears with endless screws η = 0.4 – 0.5
- Warp end performance may be given as 0.95

To hoist the anchor from the sea bed, the motor has to surpass the anchor's holding power. Thus, for two minutes the motor should be run at instantaneous power, which is calculated below [4]:

$$P(w) = \frac{(20.6 \cdot P_a + 0.2 \cdot d_c^2 \cdot L) \cdot V_s}{6.12 \cdot \eta_m \cdot \eta_e} \tag{12}$$

4.4 GEOMETRY AND DIMENSIONS FOR THE CABLE- LIFTER ON THE WARPING END

Cable- lifters are defined by the diameter of the chain to which they are geared. Another factor is the number of points in their geometry. Their shape must allow them to be in contact with at least two snugs along the chain. This position is determined by the angle at which the cable- lifter hugs the chain. Its angle value will be, at the same time, determined by the relative position between the Hawse pipe on deck or the chain stopper and the spurling pipe through which the chain passes to the case. For the chain to wind correctly in the cable- lifter, it is necessary to test it once the gear is placed on its pole on board.

ISO 21-1985 [16] specifies the shape taken by a cable-lifter with a diameter of 44 mm, with the opportunity of having five, six or seven snugs. Figure 4 shows its shape and defining parameters. For chains over the diameter mentioned above, ASTM F- 765 – 93 is applied [18]. It is important to calculate the pitch circle diameter of the cable-lifter, which is achieved with this formula [4]:

$$D_b = 1.27 \cdot 10^{-2} \cdot d_c \tag{13}$$

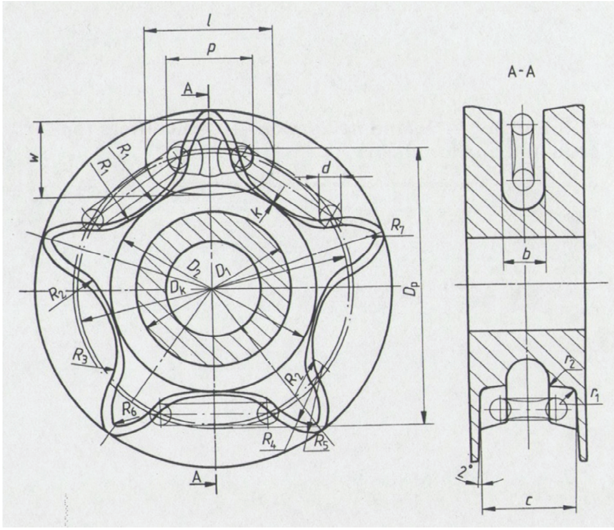


Figure 4 – Five-snug cable- lifter with its main parameters – ISO 21

Furthermore, the cable- lifter is freely mounted on the windlass axle. It receives its drive torque through the dog clutch. Given its high radial load and low speed, it is normally mounted over bronze bearings. Ball bearings are only used when the equipment has a chain that is less than 24 mm

Warping ends in anchoring windlasses are used to turn the mooring lines near where the forecastle deck is located. They share an axle with the windlass. To obtain a suitable hoisting speed, they should rotate twice as many times as the cable-lifter. The simplest way to achieve this is to vary the motor's drive speed. Another option, for winches with chains over 70 mm, is to run them at an intermediate reduction. Their Z shape lets them fit neatly on deck.

The warping end diameter must be of a size that stops the line or rope from deteriorating as it is handled by the equipment. Thus, the minimum diameter of the warping end must be over six times that of the rope [10, 15].

4.5 BRAKE TYPE AND DIMENSIONS

Every cable- lifter must have a brake that allows it to reduce its speed or stop whenever the chain is wound out. Moreover, the brake will hold back the equipment during anchoring. In most cases, a band brake is involved. It is differential because it is self-energising and can even be automatic. (Figure 5)

To obtain the necessary force from the band on top of the brake drum, windlasses include systems that multiply the force made on the lever or hand wheel. (Figure 5)

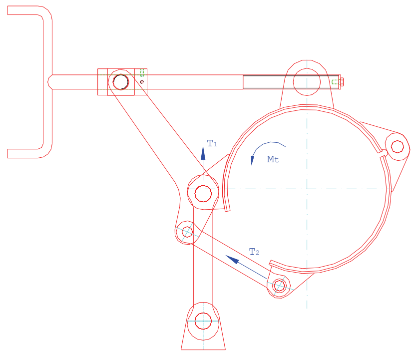


Figure 5 Band brake at a fixed point and in differential mode

Brake calculations must ensure that they can withstand the maximum static load, as specified by the ISO. It is also important to check that the maximum pressure over the brake linings falls within admissible levels.

Braking torque goes hand in hand with the braking force acting on the mooring drum attached to the cable- lifter, as demonstrated below [19]:

$$M_f = F_r \cdot \frac{D_t \cdot i}{2\,\eta_t} \qquad (14)$$

When the drum brake diameter is linked to the diameter of the cable- lifter on which it rests, coefficient "i" will be the transmission ratio for the differential brake. This coefficient will correspond to the multiplication of the force applied to the lever or hand wheel, whose mechanical efficiency will be η_t. The windlass is normally alongside the chain stopper. In this case, the cable- lifter's retaining force F_r can be found in the value for 45% of the chain's breaking load [3]. This is in agreement with the expression in Table 3.

$$F_r = 0.45 \cdot Q \qquad (15)$$

The width for the band of the brake with the same name is obtained with the following expression [19]:

$$a = F_r \frac{m}{m-1} \cdot \frac{2}{D_t \cdot P_{max}} \qquad (16)$$

Where m is the geometric coefficient of the band interacting with the angle found within the band brake (θ), the friction coefficient for the brake lining is (μ). The value P_{max}, maximum point pressure admissible for the brake lining's material, will depend on the material used. Moreover, for the brake geometry, the average pressure over the brake must be lower than the average admissible pressure on the material of choice.

5. CONCLUSIONS

When opting for an anchoring windlass design, classification society specifications and ISO standards must be taken into account. The IMO, in contrast, has no regulations on this subject.

There is a great deal of common ground between what has been produced by the classification societies and ISO 4568: the sum total of working specifications related to traction, speed and braking force. However, the classifications societies are noticeably reticent over aspects of design and manufacture, such as the geometry and dimensions of the warping ends, cable- lifter, operating systems and control devices.

Using ISO standards as a base, it is possible to produce regulations that are cohesive, and, at the same time, guarantee that the main bulk of classification society standards are met. In this way, it is also possible to establish design and manufacture specifications that make it easier to standardise components. Along with established codes, there are solid working hypotheses. With all of these, a design can be proposed that determines the main characteristics of the windlass in accordance with the data provided by the client.

With this solid base, procedures have been put forward for determining windlass type (single or double ł anchor), as well as operating modes. One can also define the gear box through the reduction ratio, the average and instantaneous power for the windlass motor, cable- lifter geometry and dimensions, warping end dimensions and, finally, brake type and dimensions.

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APÉNDICE 1

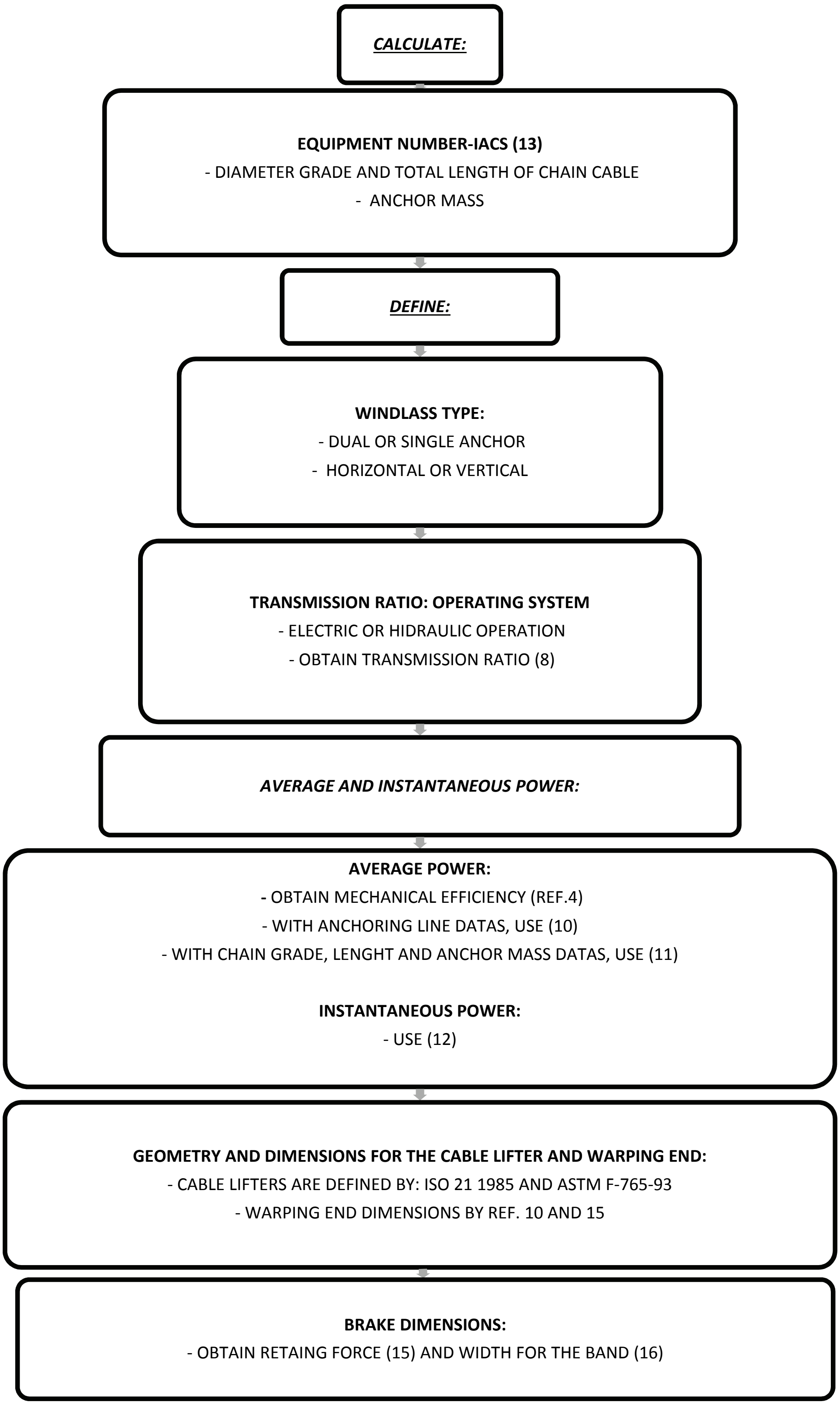

Figure 6 Windlass flow design diagram based on standardised regulations

APPLICATION OF FUZZY ANALYTIC HIERARCHY PROSES FOR ERROR DETECTION OF AUXILARY SYSTEMS OF SHIP MAIN DIESEL ENGINES

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SUMMARY

Ship engine room has a structure which has to meet a number of needs with regard to administrative conditions. Therefore, when the complicated structure of engine room are considered, even a simple mechanical failure, if no measures taken abruptly, grows into irreversible condition, causing losses that cannot be compensated. A well-qualified ship engine conductor along with an effective error detection system is needed to detect failure and act immediately against any engine impairments possible. This study aims to manage troubleshooting in main engine auxiliary systems which cover cooling, lubricating and cooling oil and fuel systems. The study is also thought to be a good reference for maintenance processes for marine engineering operators. Breakdown of main engine equipment are examined and troubles hooting program is developed for using Fuzzy Analytic Hierarchy Process (F-AHP) determine solution methods and causes of such breakdowns. In this paper, a fuzzy Multi Criteria Decision Making (MCDM) methodology was proposed to determine the most effected system of the ship main diesel engine. The results showed that fuel system was the most effected alternative, as being followed subsequently by cooling system, governor system, air supply system and oiling system. The results were based upon the opinions of three experts groups who ranked the ship main diesel engine systems alternatives according to twenty-nine criteria expert selected.

1. INTRODUCTION

Engine room is composed of main engine auxiliaries designed to supply the power necessary for operating the ship and fuel, oil, exhaust, cooling, air supply, and control systems that meet all the operational needs. Engine room is under the category of systems with high complexity; also it is made up of sub-systems that meet a number of needs. Diesel engines account for 98% of the resources producing power to operate ships. Marine diesel engines are more likely to encounter sudden and unexpected breakdowns than those stemming from long term use and wear in time. Overlooking, ignoring, or being unable to notice small impairments are among the reasons for large scale breakdowns [1].

Any simple engine failure may lead to another one unless it is noticed at a short time and measures are taken. These failures, occurring subsequently, grow into such an extent that they can lead to losses which cannot be reversed along with causalities. The important thing is to curb and take action against those failures before they become impossible to overcome. Any possible main engine failure can easily be detected by means of effective main engine failure detection. In addition to observed symptoms and detected failures, Frequency of the failures and their relations with auxiliary systems must also be taken into consideration to take the account of the possible causes of failures, which lengthens its productivity.

In addition, checking the pressure and heat of exhaust, combustion air, oil, and cooling water, as well as checking the turbocharger along with marine diesel engine would be instrumental in detecting failures [2].

Sharma *et al.* [3] in their work presented Failure Mode and Effect Analysis (FMEA) listing all possible failure modes and their causes of industrial system. So as to avoid failures in ship engine auxiliaries enjoying PROLOG programming language, Cebi *et al.* [4] set up an expert failure detection system. Through an application which they had developed for ship cooling system, they formed action tables displaying what to do in case of emergency taking the types of failures encountered before and changes in indicative value limits. As a result of their study, it is emphasized that detecting failures in time and shortening the intervening time in the trade ships in the critical seas while maneuvering will raise the operational efficiency. Unlugencoglu [5] developed a troubleshooting programme by using C# programming language to determine solution methods and causes of such breakdowns in main engine auxiliary systems which cover cooling, lubricating and cooling oil and fuel systems. Ozsoysal [6] studied the possible reason or reasons of failure exhaust and its effects on the damage size at high speed marine diesels in Turkish ambulance boats. Gourgoulis [7] studied turbo engine driven electro generators used in maritime engineering for the auxiliary electrical power supply system of the ship. He made failure analysis and besides to provide solutions for real operating problems.

In this study, six failure types of high importance, first seen in marine diesel engine, have been determined. Possible causes for these failures were categorized into subtitles. Based on expert group decision, the article has been demonstrate which type of the failure was the most critical and which system was the most influenced by the failure.

2. FUZZY AHP METHODOLOGY

A set of methods resting on fuzzy sets theory have been set forth to acquire the final assessment thanks to expert's remarks. Cholewa [8] suggested various axioms for fuzzy weighted opinions. Analysis by Dubois and Koning [9] resulted in various fuzzy set aggregation connectives to evaluate the suitability as for social choice functions. Kacprzyk *et al.* [10] focused on fuzzy preference relation. Authors concluded vitality of fuzzy relations by all experts, by means of which authors attained a resulting preference relation from the point of view of individual fuzzy preference relations with the aim of choosing the best option. Mohammad *et al.* [11] put a new approach to handle problem of parametric form of fuzzy numbers and applied it to a case study of diversion of water. Lee [12] established an iterative approximation procedure to collect individual opinions into an optimal consensus. Jiang and Fan [13] worked on the possibility degree for triangular fuzzy number and introduced a new method based on judgment matrix. Xu and Da [14] stressed the possibility degree of interval number and several properties proved to be true. Yeh and Chang [15] offered a hierarchical weighting method to analyze weights, and additionally submitted an algorithm for grouping MDCM to involve criteria weights out of decision makers' subjective judgments. Ma et al. [16] formulated a decision support system relevant with a model to promote the satisfaction throughout the whole process in the multi-criteria group decision making. Fan and Liu [17] gave rise to a method for group decision-making dependent on the multi-granularity uncertain linguistic information.

Linguistic variable: A Linguistic variable refers to a variable whose values are not numbers but words or sentences in a natural or artificial language. In this paper, such statements are used for making comparison of auxiliary system selection evaluation criteria through five basic linguistic terms which are "absolutely important", "very strongly important", "essentially important" "weakly important" and "equally important" with regard to a fuzzy five level scale [18]. In this present study, the computational technique based on the fuzzy numbers is explained below in Table 1.

Table 1. Membership function of linguistic scale, [19]

Fuzzy number	Linguistic scales	Scale of fuzzy number	Scale of Reciprocal Fuzzy Number
$\tilde{1}$	Equally important (EQ)	(1,1,3)	1/3,1,1
$\tilde{3}$	Weakly important (WK)	(1,3,5)	1/5,1/3,1
$\tilde{5}$	Essentially important (ES)	(3,5,7)	1/7,1/5,1/3
$\tilde{7}$	Very strongly important (VS)	(5,7,9)	1/9,1/7,1/5
$\tilde{9}$	Absolutely important (AB)	(7,9,9)	1/9,1/9,1/7

The linguistic variables presented in Table 1 are used to demonstrate the superiority or weakness status of AHP method by the five designated groups in the criteria-criteria comparison.

Alternatives measurement: It is referred to the use of the measurement of linguistic variables to indicate the criteria performance (effect-values) by means of statements such as "very good", "good", "medium good", "fair", "medium poor", "poor", "very poor". The evaluators are requested to conduct their subjective judgments and each linguistic variable can be demonstrated by a Triangular Fuzzy Number (TFN) within the scale range of 0–10, as presented in Table 2.

Table2. Fuzzy evaluation scores for the alternatives [20]

Linguistic terms	Fuzzy score
Very poor (VP)	(0, 0, 1)
Poor (P)	(0, 1, 3)
Medium poor (MP)	(1, 3, 5)
Fair (F)	(3, 5, 7)
Medium good (MG)	(5, 7, 9)
Good (G)	(7, 9, 10)
Very good (VG)	(9, 10, 10)

The linguistic variables presented are used to demonstrate the superiority or weakness status of F-AHP method by the five designated groups in the alternative-criteria comparison in Table 2.

Moreover, the evaluators can subjectively give their personal range of linguistic variable that can display the membership functions of each evaluator's expression values. Take E_{ij}^k to denote the fuzzy performance value of evaluator k towards alternative i under criterion j, and all of the evaluation criteria will be displayed by $E_{ij}^k = (LE_{ij}^k, ME_{ij}^k, UE_{ij}^k)$. Since each evaluator's perception differs from one another in their experience and knowledge, and the descriptions of the linguistic variables show an alteration as well, this study uses the idea of average value to integrate the judgment values of m evaluators in fuzzy type, that is,

$$\hat{E}_{ij}^k = 1/m(LE_{ij}^k, ME_{ij}^k, UE_{ij}^k) \tag{1}$$

demonstrates the average fuzzy number of the decision-makers' judgment which can be represented by a triangular fuzzy number as $LE_{ij}^k, ME_{ij}^k and UE_{ij}^k$. The end-point values LE_{ij}, ME_{ij} *and* UE_{ij} can be figured out by the method proposed by Buckley [26], that is,

$$LE_{ij}^k = \frac{\sum_{k=1}^{m} LE_{ij}^k}{m}; ME_{ij}^k = \frac{\sum_{k=1}^{m} ME_{ij}^k}{m}; UE_{ij}^k = \frac{\sum_{k=1}^{m} UE_{ij}^k}{m} \tag{2}$$

Fuzzy synthetic decision: Besides the fuzzy performance values, the evaluation of the weights of each criterion of auxiliary systems selection must be incorporated by the computation of fuzzy numbers located at the fuzzy performance value (effect-value) of the integral assessment. According to the each criterion weight obtained by FUZZY-AHP, the criteria weight vector $\widehat{W} = (\widehat{W}_1 \ldots\ldots \ldots \widehat{W}_j \ldots\ldots \widehat{W}_n)^t$ j can be acquired, while the fuzzy performance matrix $\widehat{E}$ of each of the alternatives can also be derived from the fuzzy performance value of each alternative under n criteria, that is, $\widehat{E} = \widehat{E}_{ij}$ From the criteria weight vector $\widehat{W}$ and fuzzy performance matrix $\widehat{E}$, the ultimate fuzzy synthetic decision can be carried out, and the obtained result will be the fuzzy synthetic decision matrix $\widehat{E}$, that is,

$$\widehat{R} = \widehat{E} o \widehat{w} \quad (1)$$

The symbol "o" denotes the computation of the fuzzy numbers including fuzzy addition and fuzzy multiplication. Due to the complexity of the calculation of fuzzy multiplication, it is usually indicated by the approximate multiplied outcome of the fuzzy multiplication, and the approximate fuzzy number $\widehat{R}$ i, of the fuzzy synthetic decision of each alternative can be represented as $\widehat{R} = (L\widehat{R}_i, M\widehat{R}_i, U\widehat{R}_i)$, where, $L\widehat{R}_i, M\widehat{R}_i \; and \; U\widehat{R}_i$ are the lower, middle and upper synthetic performance values of the alternative i, that is:

$$LR_i = \sum_{j=1}^{n} LE_{ij} x Lw_j ; MR_i = \sum_{j=1}^{n} ME_{ij} x Mw_j ; UR_i = \sum_{j=1}^{n} UE_{ij} x Uw_j ; \quad (4)$$

Ranking the fuzzy number: The outcome of the fuzzy synthetic decision obtained by each alternative is a fuzzy number. Consequently, it is necessary to employ a non-fuzzy ranking method for fuzzy numbers in order to make comparisons of each alternative. That is to say, the defuzzification procedure is needed to locate the Best Nonfuzzy Performance value (BNP). Methods of defuzzified fuzzy ranking such as Mean of Maximal (MOM), Center of Area (COA), and a-cut are generally included. It is an easy and applicable method for utilizing the COA method to find out the BNP, and it is not necessary to appeal to the preferences of any evaluators. Therefore, the COA method is used in this study. The BNP value of the fuzzy number $\widehat{R}_i$ can be reached by the equation below:

$$BNP_i = [(UR_i - LR_i) + (MR_i - MR_i)]/3 + LR_i \quad \forall i \quad (2)$$

According to the value of the acquired BNP for each of the alternatives, the ranking of the auxiliary systems can be proceeded.

F-AHP Methodology steps of application is summarized as follows in Figure 1.

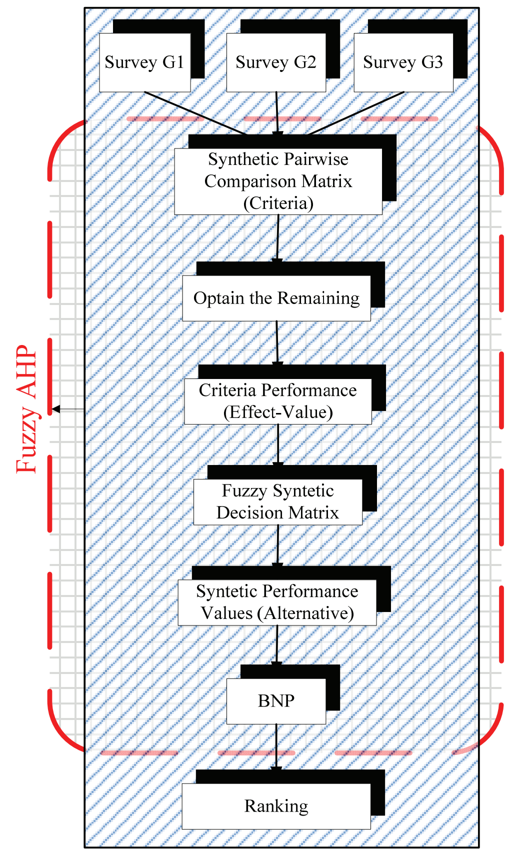

Figure 1. F-AHP method

Step 1: Construct pairwise comparison matrices among all the criteria in the dimensions of the hierarchy system.

Step 2: Calculation the elements of synthetic pairwise comparison matrix by using the geometric mean method suggested by Buckley [21] :

$$\hat{a}_{ij} = (\hat{a}_{ij}^1 \otimes \hat{a}_{ij}^2 \otimes \ldots \otimes \hat{a}_{ij}^n)^{\frac{1}{n}} \quad (6)$$

Step 3: In the same way, we can obtain the remaining $\widehat{r}_i$:

$$\widehat{r}_i = (\hat{a}_{i1}^1 \otimes \hat{a}_{i2}^2 \otimes \ldots \otimes \hat{a}_{in}^3)^{\frac{1}{n}} \quad (7)$$

Step 4: For the weight of each dimension, it can be performed as follows:

$$\widehat{w}_i = \widehat{r}_i \otimes (\widehat{r}_1 \oplus \widehat{r}_2 \oplus \ldots \oplus \widehat{r}_n)^{-1} \quad (8)$$

Step 5: Alternatives measurement: Using the measurement of linguistic variables to demonstrate the criteria performance (effect-values) by expressions.

$$\hat{E}_{ij}^k = 1/m(LE_{ij}^k, ME_{ij}^k, UE_{ij}^k) \quad (9)$$

Step 6: The end-point values LE_{ij}, ME_{ij} and UE_{ij} can be solved by the method put forward by Buckley, (1985), that is,

$$LE_{ij}^{k} = \frac{\sum_{k=1}^{m} LE_{ij}^{k}}{m}; ME_{ij}^{k} = \frac{\sum_{k=1}^{m} ME_{ij}^{k}}{m}; UE_{ij}^{k} = \frac{\sum_{k=1}^{m} UE_{ij}^{k}}{m} \quad (10)$$

Step 7: Fuzzy synthetic decision matrix $\widehat{R}$, that is,

$$\widehat{R} = \widehat{E} o \widehat{w} \quad (11)$$

Step 8: Synthetic performance values of the alternative i, that is:

$$LR_i = \sum_{j=1}^{n} LE_{ij} x Lw_j; MR_i = \sum_{j=1}^{n} ME_{ij} x Mw_j; UR_i = \sum_{j=1}^{n} UE_{ij} x Uw_j; \quad (12)$$

Step 9: Ranking the fuzzy number: The BNP value of the fuzzy number $\widehat{R}_i$ can be found by the following equation:

$$BNP_i = [(UR_i - LR_i) + (MR_i - MR_i)]/3 + LR_i \qquad \forall i \quad (13)$$

Step 10: Evaluation is done according to the results.

3. A REAL CASE APPLICATION FOR SHIP DIESEL ENGINE TROUBLE SHOOTING

When the causes and signs of faults encountered in marine diesel engines are investigated, it is seen that they are mostly the indicators of another malfunction. There is a reason in each case of failure and that reason may occur in the course of operation. The hierarchical structure applied in this study to overcome the operational problems of the machine assessment for ships is demonstrated in Figure. 2.

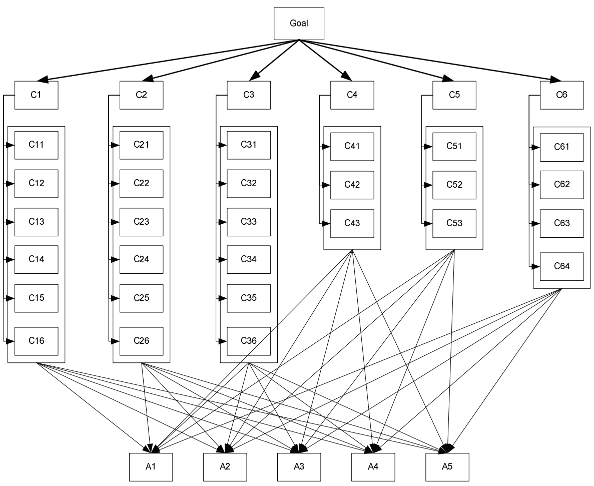

Figure 2. The hierarchical structure for ship engine operation system alternatives assessment.

The main dimensions of the criteria for evaluating and selecting the systems of engine operation for the alternative ship were obtained with an extensive research and consultation with three groups in which one professor from the department of Naval Architecture and Marine Engineering was involved. The groups were requested to grade the criteria dimensions in terms of their accuracy, sufficiency and significance in order to validate the content of these criteria for engine failure evaluation. Reasons of failures in the main engine systems were derived from former reports, maintenance logbooks, and the acquired data were combined with the personnel's experiences. When the examination of these failures are taken into account, it appears that there are six types of failures of high priority that come forth as shown in Table 3. Failures are coded as where i is the number of related failure.

C1. High heat level in all exhaust cylinders of the engine
- C11. Fuel oil quality
- C12. Fuel injector problems
- C13. Fuel oil pump failures
- C14. Fuel oil leakage in cylinders
- C15. Air fun not working fully
- C16. Wrong adjustment of governor

C2. Fluctuation in engine rotations
- C21. Dirty fuel oil filter
- C22. Fuel oil pump pressure
- C23. Fuel oil temperature
- C24. Insufficient intake air
- C25. Mechanical failure in the turbocharger
- C26. Wrong adjustment of governor

C3. Sudden shut down of the engine while it is working usual
- C31. Low level fuel oil tank
- C32. Insufficient intake air
- C33. Oil pressure
- C34. Oil leakage,
- C35. Insufficient cooling water
- C36. Fuel oil pump failures

C4. Rise in the oil level in crankcase while the engine is working
- C41. Cooling water leakage
- C42. Shut off valve on oil tank open
- C43. Fuel oil leakage

C5. Fire in the Scavenging area
- C51. Dirty scavenging manifold inlet
- C52. Abrasive oil ring and piston
- C53. Air cooler problem

C6. Surge in the turbocharger
- C61. Exhaust valve burns
- C62. Insufficient turbocharger oil
- C63. Low level oil in the governor
- C64. Insufficient intake air
- C65. Scavenging pressure high

Table 3: Weights of dimensions and criteria for decision-maker groups

Dimension and Criteria	Local weights			Overall Weights			BNP
C1	0,048	0,102	0,263				0,138
C1.1	0,167	0,327	0,647	0,012	0,047	0,182	0,380
C1.2	0,102	0,213	0,470	0,007	0,030	0,132	0,262
C1.3	0,054	0,113	0,259	0,004	0,016	0,073	0,142
C1.4	0,033	0,072	0,137	0,002	0,010	0,038	0,081
C1.5	0,122	0,230	0,391	0,009	0,033	0,110	0,247
C1.6	0,022	0,045	0,101	0,002	0,006	0,028	0,056
C2	0,025	0,050	0,172				0,083
C2.1	0,100	0,244	0,619	0,007	0,035	0,174	0,321
C2.2	0,064	0,167	0,476	0,005	0,024	0,134	0,236
C2.3	0,051	0,125	0,315	0,004	0,018	0,088	0,163
C2.4	0,068	0,178	0,454	0,005	0,025	0,128	0,234
C2.5	0,035	0,095	0,239	0,002	0,014	0,067	0,123
C2.6	0,075	0,191	0,444	0,005	0,027	0,125	0,236
C3	0,131	0,323	0,848				0,434
C3.1	0,066	0,129	0,256	0,005	0,018	0,072	0,150
C3.2	0,156	0,286	0,496	0,011	0,041	0,140	0,313
C3.3	0,224	0,364	0,555	0,016	0,052	0,156	0,381
C3.4	0,037	0,069	0,156	0,003	0,010	0,044	0,087
C3.5	0,032	0,061	0,132	0,002	0,009	0,037	0,075
C3.6	0,046	0,090	0,187	0,003	0,013	0,053	0,108
C4	0,131	0,351	0,747				0,410
C4.1	0,201	0,319	0,473	0,014	0,046	0,133	0,331
C4.2	0,183	0,270	0,407	0,013	0,039	0,115	0,287
C4.3	0,264	0,411	0,664	0,019	0,059	0,187	0,446
C5	0,046	0,124	0,312				0,161
C5.1	0,280	0,441	0,818	0,020	0,063	0,230	0,513
C5.2	0,232	0,417	0,659	0,016	0,060	0,185	0,436
C5.3	0,072	0,143	0,235	0,005	0,020	0,066	0,150
C6	0,021	0,050	0,144				0,072
C6.1	0,152	0,339	0,711	0,011	0,048	0,200	0,401
C6.2	0,068	0,114	0,286	0,005	0,016	0,080	0,156
C6.3	0,185	0,396	0,790	0,013	0,057	0,222	0,457
C6.4	0,066	0,151	0,339	0,005	0,022	0,095	0,185

When engine failures separated from each other based on basic characteristics above with the intention of categorizing are technically examined, it appears that each has a relationship with a different system. Failures have been established in accordance with the opinion of specified groups. Factors for failures are concerned, auxiliary systems connected with the failures can be categorized as follows;

A1. Fuel System
A2. Cooling System
A3. Oiling System
A4. Governor System
A5. Air supply System

Heat operating value, critical for operating marine diesel engines, are the values of cooling water and oil which act as the main factor that cools the engine and keeps the heat stemming from fuel out of running engine away. In addition to these values, heat value of exhaust gases is the factor that gives important information about combustion process, combustion productivity, and power obtained from the engine.

At ship operations, an extensive intervention is required to control heat of oil and cooling water, depending on irregular alterations in marine diesel engine load. More effective energy gain and safer marine diesel engine operation are ensured keeping values of cooling water and oil heat at an optimum level. The heat of the cylinder wall cooling water can affect the formation of oil film at the cylinder wall. Operation algorithm of central cooling system which is frequently encountered marine diesel engines is shown in the Figure 3.

In diesel engines, fuel and governor systems are required to work perfectly to gain desired power and rotation.

Rotation intervals for the engines to work safely are determined by the engine manufacturers. Operating the engine out of this range and for a longer period causes the exhaust heat to increase. As the engine rotation increases, emission of the exhaust gases flow rises, and this end up in increase in turbine rotation. The control of the amount of fuel sent to the injector from fuel pump is ensured by the governor to operate at a stable speed.

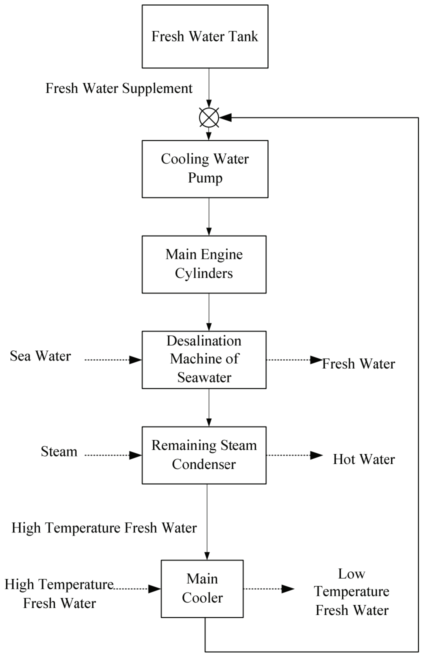

Figure 3. Structure of Main Engine HTFW System, [22]

Proper functioning of diesel engine and turbo charger is prevented by high level exhaust gases heat, blocked filters, unwanted substances stuck in the compressor or the turbine. Excessive dirt and blockage in air supply filters cause fire in suction manifold. Difficulty in pushing the gases in the area exhaust thorough the chimney with the force of counter pressure and decrease in the inlet pressure cause the engine to fail to bear the load.

The weights for the criteria for decision making groups can be found as shown in Table 3. And we listed the final BNP value of groups in Table 4. From the FAHP results, for the decision maker groups, we find the first two most important aspects are Sudden shut down of the engine while it is working (**C3:0,434**) and Rise in the oil level in crankcase while the engine is working (**C4: 0,410**); whereas the least important is Surge in the turbocharger **(C6: 0.07).** The important first two sub-criterias in Sudden shut down of the engine while it is working are Oil pressure **(C33:0.381)** and Insufficient intake air **(C32: 0.313)** according to the decision maker groups, the least is Insufficient cooling water **(C35: 0.075).** In addition, the important sub-criteria's in Rise in the oil level in crankcase while the engine is working are displayed in order of arrival Fuel oil leakage **(C43: 0.446)**, Cooling water leakage **(C41: 0.331)** and Shut off valve on oil tank open **(C42: 0.287)** for the experts groups. However, the first two important dimensions in least important criteria are Low level oil in the governor **(C63: 0.457)** and Exhaust valve burns **(C61: 0.401)**, and Insufficient turbocharger oil is the least **(0.156)**.

These results indicate that the decision making groups are worried about the safety of managing Sudden shut down of the engine while it is working, in addition, the decision making groups also cares about the Rise in the oil level in crankcase while the engine is working which will be considering the convenience of freighter operating. The decision making groups focus on the related professional issues for Sudden shut down of the engine while it is working, but they deem that the Oil pressure and insufficient intake are certain to be safe under calculations, so they ranked it with the most importance.

As for the criteria hierarchy, all decision maker groups deem dirty scavenging manifold inlet (C51) to be the most important (0,513). This may reflect the operating performance and combustion process efficiency of engine. Dirty scavenging manifold inlet was followed in importance by Low level oil in the governor (C63:0.457), Fuel oil leakage (C43: 0.446), and Abrasive oil ring and piston (C52: 0.436) for decision maker groups. On the other hand, all decision maker groups rely Wrong adjustment of governor (C16) to be the least important by (0.056). This may not lead to a serious fault but it can cause more fuel consumption. Wrong adjustment of governor was follow up Insufficient cooling water (C35: 0.075), Fuel oil leakage in cylinders (C14: 0.081), Oil leakage (C34: 0.087) and Fuel oil pump failures (C36: 0,108).

We can obtain the BNP values of other alternatives for comparison purposes; finally, details of the results are presented in Table 4.

Table 4: Ranking by criteria weightings

ALTERNATIVES	BNP	RANKING
A1: Fuel System	8,773	1
A2: Cooling System	5,894	2
A3: Oiling System	4,811	5
A4: Governor System	5,795	3
A5: Air Supply System	5,196	4

As can be seen from the alternative evaluation results in Table 4, the Fuel System is the most affected alternative (BNP value: 8,773) by errors considering the weights of all decision maker groups. The results in Table 3 reflect the common consensus that changes in criteria weights may affect the evaluation outcome to a certain degree. Besides, the Oiling System has the least affected alternative (BNP value: 4,811) by errors relative to other alternatives, which is the most common perception among the groups.

4. CONCLUSION

In the engine room, all engines work in an integrated manner and due to this reason, any fault happening in any

system can quickly affect the whole system. A small failure may grow to a failure of the whole system to turn the situation into a life-threating danger. This shows that in any case of engine breakdown or failure, the engine operators need to address cause as quick as possible. That cause must be easily found and corrected by expert applications.
In this study, the hierarchical structure adapted to the troubleshooting of main diesel engine auxiliary systems which cover cooling, lubricating oil, governor, air supply and fuel systems.

The major causes of system errors have been determined by evaluation of experts using F-AHP method. The way in which systems are affected from possible defects revealed. Besides this, operator indicated any fault which will primarily intervene. Summing all together the alternative Fuel System is the most affected system when failures of this kind occurred.

The study is also thought to be a good reference for maintenance processes for ship engine officers. Future research in this direction is really needed, in order to provide policy-makers a wider perspective on the ship diesel engine troubleshooting systems control.

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