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CONCEPT AND OPERATION MODE OF THE ADVANCED ELECTRONIC CONTROL SYSTEM OF THE AZIMUTH PROPELLERS IN TUGS

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ABSTRACT:

"The remote control of azimuth propellers in tugs from the bridge (speed, thrust direction and clutch) is electronic and each manufacturer has his characteristic model, although the control system and the arrangement of the propellers to achieve the expected thrust are similar.

In this article, we analyse the remote electronic control devices in the bridge from the tug azimuth propellers in general, especially its performance and operation mode, without making any distinction between those located forward (tractor Z) or aft (ASD), since their foundation is similar and the only difference is the horizontal propeller position and consequently the direction of the remote controls from the bridge to achieve the expected thrust direction and speed".

Keywords: Tugs, Omnidirectional Propulsion, Azimuth propellers, thrust direction, *rpm* and clutch remote control system.

INTRODUCTION

Although the azimuth thruster concept is an innovation dating back more than fifty years ago, nowadays it still remains a novelty to many ship owners and operators. The basic idea behind an azimuth thruster is that the propeller can rotate 360 degrees round a vertical axis providing omni-directional controlled thrust. This

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means superior manoeuvrability for vessels equipped with azimuth thrusters. It also eliminates the need for a rudder and a reverse gear (i.e. the azimuth thruster itself also functions as a reduction gear[1]). Unlike the cicloidal propellers where Voith Turbo Marine is the only manufacturer -from that the generic name of *Voith propellers*-, the azimuth thrusters are manufactured by different firms and have been referred to under different names e.g. *Z-drives*, *rudder propellers*, *rotable propulsion units*, *omnidirectional thrusters,* although the brand name of a manufacturer like *Shottel*, *Ulstein Rolls-Royce*, *Steerprop*, *Niigata, Duckpeller* etc. is often used as a generic label for all azimuth thrusters.

The system consists of several different devices: azimuth thrusters, steering and control unit levers and shaft lines. As a whole, they might well be more accurately referred to as a "*propulsion and steering system*".

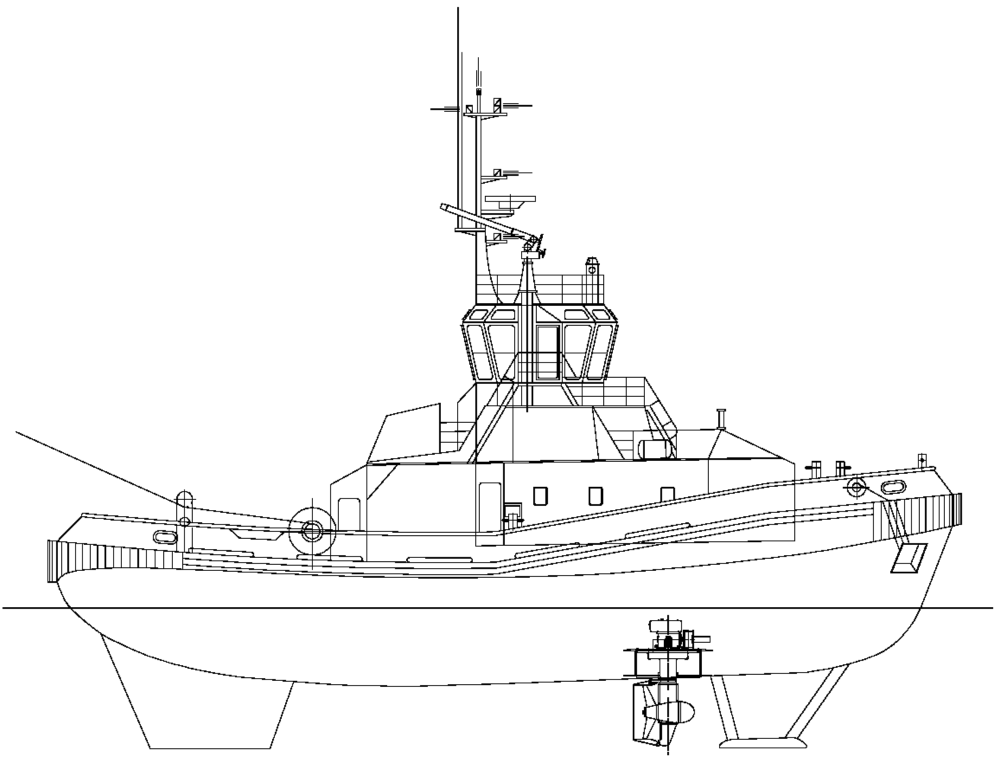

Figure 1. Tractor Z Tug. Drawing: author

[1] Thanks to the fact that the reduction gear system is located in the propeller unit, every part in the high-torque power transmission has also been moved into the unit itself.

In Spain, the tugs with this type of propellers are the tractor *Z*[2] (its propellers are mounted in the forepart of the tug´s hull, the same as the *tractor Voith*) and the *Azimuth Stern Drive (ASD)* whose propellers are located aft, the same as conventional tugs[3].

The remote control of these propellers from the bridge (speed and thrust direction) is electronic and each manufacturer has his own characteristic model, although the control system and the arrangement of the propellers to achieve the expected thrust are very similar.

In this article, the remote electronic control devices in the bridge from the tug azimuth propellers are analysed in general, especially their performance and operation mode, without making any distinction between those placed forward (*tractor Z*) or aft (*ASD*), since their basis is similar and the only difference is the horizontal propeller position and consequently the position of the remote controls from the bridge in order to achieve the expected thrust direction.

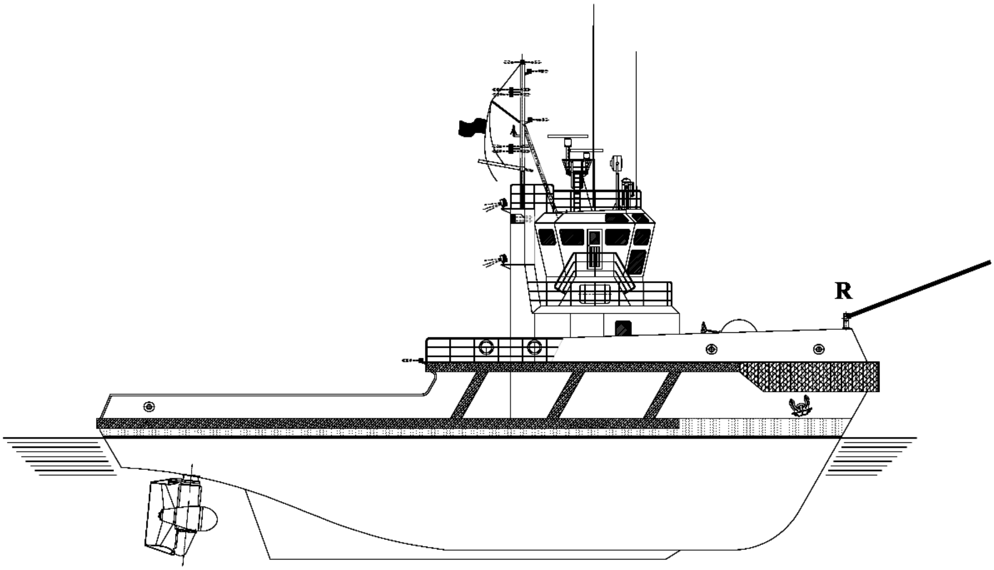

Figure 2. ASD Tug. Drawing: author

[2] So called because of the configuration of the drive shaft.

[3] There are two new states of the art tugs that also have this type of propellers, they are the *Ship Docking Module (SDM)* now rechristened in Europe –for contractual reasons– as *Asymmetric Tractor Tug (ATT)* which are "*Salvador Dalí*" and "*Ramón Casas*" recently delivered to the Spanish operator *Reyser, S.L.* for its own operations in the harbour of Barcelona and the *Rotor Tug* , a revolutionary concept characterised as an enhanced tractor tug with two propellers forward and where the large skeg aft is replaced by an azimuthing propulsion unit; but for the time being, this type of tugs are not working in Spanish harbours.

AN INDEPENDENT ELECTRONIC CONTROL SYSTEM FOR EACH AZIMUTH PROPELLER

General description

Generally, a control of this type for the azimuth propeller[4] of a tug consists of a follow-up remote control system from the bridge, which is independent for the control of each propeller.

The basic operation unit of the system is the control head, which has got different denominations according to the manufacturer. It basically consists of a single lever which can rotate 90º vertically and 360º horizontally for the thruster functions (steering, *rpm* and clutch). Thus, each thruster has its own independent control system, allowing all the azimuth propeller controls in only one hand:

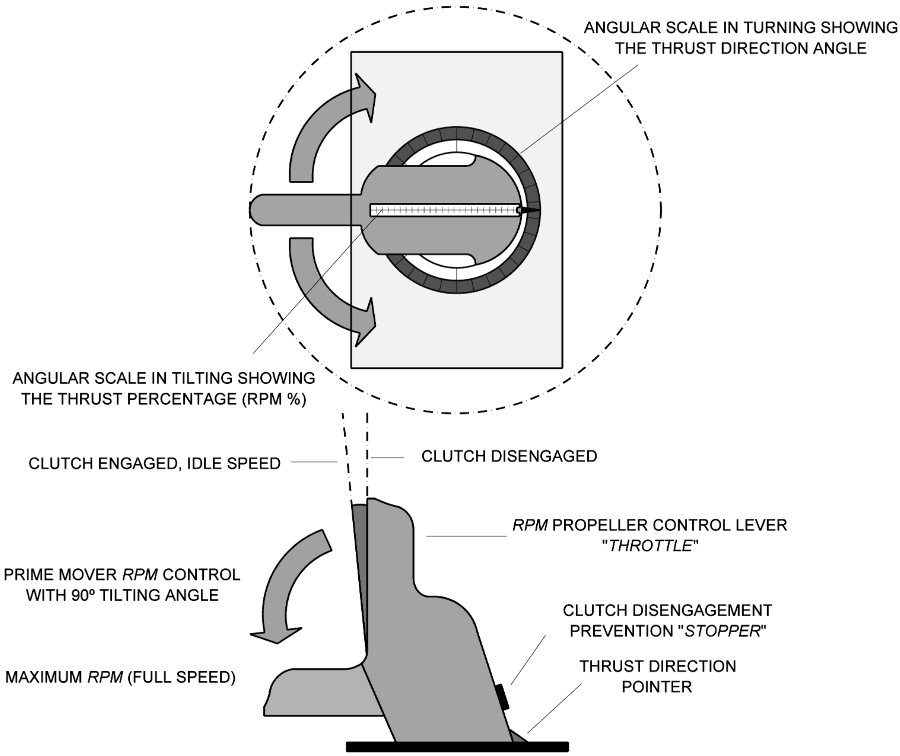

Figure 3. Drawing in ACAD of the control head with its main functions to a bird view (up) and to a side view (down) of an azimuth propeller from the manufacturer Aquamaster. Source: Aquamaster, drawing: author.

[4] Although their functionality is very similar, as well as its design, its denomination varies depending on the manufacturer; thus, *Schottel* denominates it *Copilot*, Aquamaster denominates it *Aquamaster Control Head* and Ulstein denominates it *Combilever*.

— Continuous horizontal control of the thrust direction (steering) throughout the 360º.

— Clutch control: the clutch remains disengaged when the lever is in vertical position with a tilting angle of between 0º and 5º, in which case, the propeller does not rotate (the highest number of degrees from the vertical position depends on each manufacturer). From this point, the power increases by tilting the lever further. When the lever is in 90º position, full power (*rpm*) is in use.

— Main engine speed control (*rpm* control): the prime mover´speed is controlled by tilting the control lever "*throttle*" in the vertical direction. The prime mover *rpm* of the two engines does not necessarily have to be adjusted. It is possible to stop and hold the ship in its place by thrust orientation only.

— Some manufacturers, as Aquamaster, incorporate clutch disengagement prevention by means of the "*stopper*". A "*stopper*" button blocks the return of the lever to the upright position to keep the clutch engaged while the engine idles.

The azimuth propeller follows the movements of the control head in the bridge with a certain delay "*follow-up control*", in such a way that the operator can concentrate on the real operation of the ship, as this control acts in a logical way.

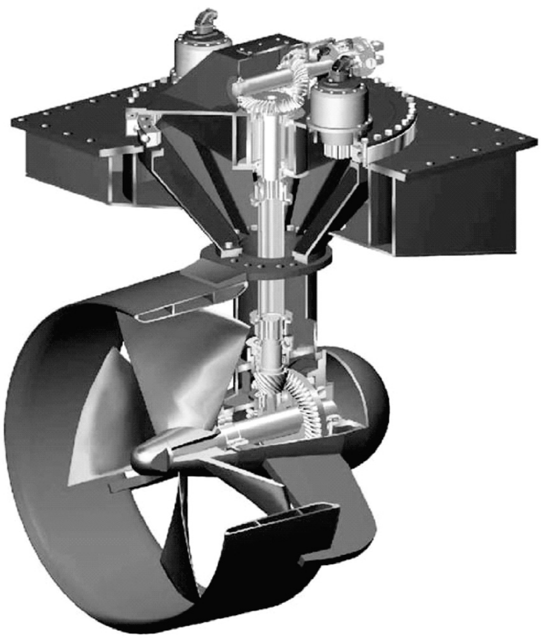

Figure 4. Diagram depicting the "Z" drive configuration of a typical azimuthing propulsion unit (in this case from the manufacturer Thrustmaster), cut away to show the drive shafts and gears. The steering gear, which turns the entire centre portion and propeller through 360º, is omitted for clarity reasons. Source: Thrustmaster.

The control statio

It usually consists of a small console provided with at least the following units:

—The control head, a single lever for the operation of the thruster functions –steering, *rpm* and cluth.

—A thrust direction indicator.

—A control panel of the propeller with different indicators.

Some manufacturers as *Duckpeller* and *Niigata* do not have the three functions in a single lever, but rather they have:

— a single control lever for both propellers named *unilever*, a kind of *joystick* which is a single control lever on the wheelhouse console that is used to control the direction of the tug as well as the speed for a given *throttle* setting.

— two levers named "*throttles*" for the *rpm* control of the main engines of each propeller.

The angle that the propellers make with the forward and aft line is controlled by the *unilever*.

The general principle is that the tug will go in the direction where the *unilever* is placed, with a combination of rotational and translational movements, made possible by the control system vectoring the propeller thrusts in various ways. With the *unilever* forward or aft in the centreline, the propellers drive the tug directly ahead or astern.

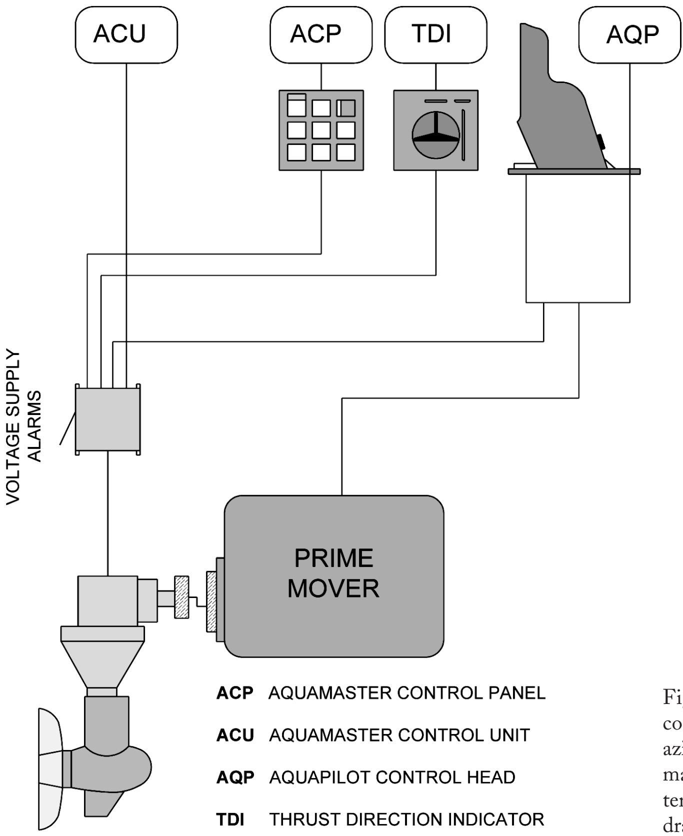

Figure 5. Diagram of a control system of an azimuth propeller by the manufacturer Aquamaster. Source: Aquamaster, drawing: author.

With the *unilever* placed off the centreline, a control system rotates the propeller units, so that the tug travels in the direction where the lever is placed.

Main engines are controlled by two *throttle* levers, while directional or steering control is carried out by means of only one *unilever*, in such a way that the electronic control system adjusts the thrust direction of the two *Z-pellers* in order to move the tug in the direction that the *unilever* is moved. With the *unilever* in the vertical or neutral position, the thrust of the two *Z-pellers* is at right angles to the vessel´s centre line on each side, thus holding the vessel stationary. Speed control, and thus propeller thrust, is also independent for each main engine.

There are combined clutch and engine speed control levers named *throttles,* located on the wheelhouse console adjacent to the *unilever*.

The tug's manoeuvrability may be further improved by the tug master by varying the speed of each engine in combination with the various *unilever* settings.

The secondary steering system: backup control system

The control system of the propellers includes a secondary *non follow-up* dependent back-up system usually known in the tug slang as secondary steering, although considering its function (thought more to be used mainly in case of the "*follow up control*" failure), we could denominate it more properly as emergency steering.

This back-up system is made up of some push-buttons which have a direct and independent electric connection with the propeller unit by electronic circuits, i.e. they are directly connected to each propeller.

Figure 6. Control Head of one of the two Aquamaster propellers (named Aquamaster Control Head) of the ASD tug "Sertosa Treinta y dos" working-based at La Coruña harbour. To the right of this control, the two push-buttons that constitute the secondary steering system can be appreciated. Photograph: author.

The thrust direction control

The control head of the azimuth propeller controls the turning of the propulsion unit. All the controls of the different manufacturers have some specific characteristics, but in general, their functions are very similar. Taking as an example that of the manufacturer Aquamaster, when the control lever is rotated horizontally an angle bigger than 0,7º, the propeller turning controller (*Aquamaster Turning Controller - ATC-)* senses the deviation angle and turns the propeller to the desired direction by controlling the hydraulic system and adjusting the turning speed, according to some preset values[5]. The horizontal turning speed of the propeller is partially proportional to the deviation angle between the actual direction of the propeller and the value settled down by the control head. When the deviation is more than 23º, the turning speed is higher; when it is smaller than this angle, the turning speed is proportional to the angle difference. The steering lever includes a circular scale where the thrust direction is shown in degrees.

The thrust direction indicator is electrically independent from the control system. The propeller speed indication is usually integrated to the thrust direction indicator connected to the propulsion unit.

The speed control

The main engine speed is controlled by tilting the propeller control lever vertically. Generally, the idling *rpm* range is 0º- 5º; from this point, the power increases by tilting the lever further towards full power *rpm* at 90º. The main engine speed does not necessarily have to be adjusted. It is possible to stop and keep position by only thrust orientation.

The clutch control

The propeller built-in clutch is controlled by tilting the propeller control lever through five degrees from the upright position. A microswitch, driven by the propeller, controls the thruster clutch. A *stopper* button usually blocks the return of the lever to the upright position inside the propeller control lever to keep the clutch engaged while the engine idles. For security reasons, starting the main engine is only possible when the clutch is disengaged. A clutch engaged bright and a clutch engaged prevention indicators with a warning light are all located on the propeller control panel.

Electronics in azimuth propellers

The hydraulic steering function system of the propeller is electronically controlled. The turning control is an intelligent, microprocessor-based unit with an

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5 This guarantees smooth steering without pressure shocks in the hydraulic system.

Figure 7. Left: Control Head of one of the two Schottel propellers (entitled Copilot) of the tug ASD "Bravyy" recently built by a Spanish shipyard to a Russian owner. Right: the thrust direction indicator of the propeller. Photographs: author.

automatic adjustment system and self-monitoring. All its settings are electrical and have a self-diagnostic display system in the propeller control unit which ensures the propeller´s smooth turning and shock-free steering at low hydraulic steering pressures, as well as a constant turning speed.

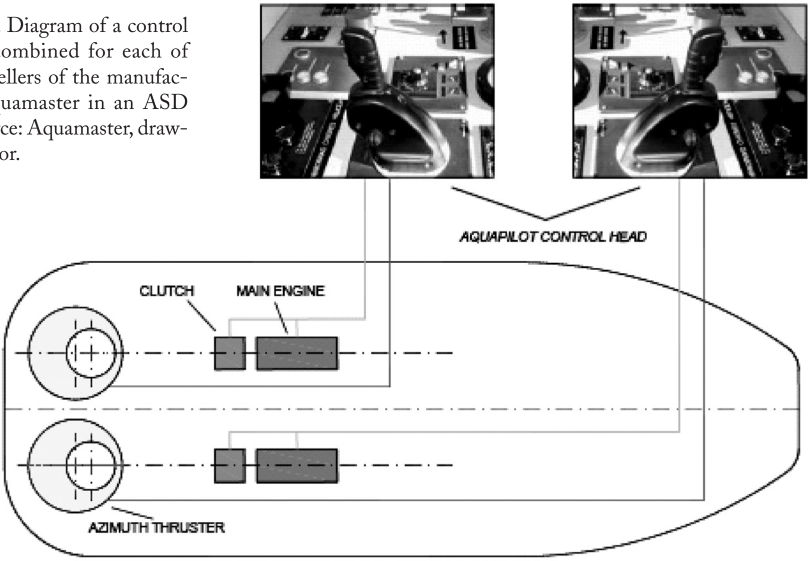
In figure 7 a control head of a *Schottel* azimuth propeller can be appreciated. This control consists of a horizontal wheel which controls the propeller thrust direction (the wheel has a pointer at console level that points the propeller direction exactly), and a small lever "*throttle*" from which the thrust of the propeller is controlled, modifying the *rpm* of the motor that drive it from the vertical to the horizontal position where full power (*rpm*) is in use.

The propeller lever synchronizing when there is more than one control station on the bridge

The state of the art wheelhouse´s designs of *tractor* and *ASD* tugs has good outside visibility close to 360º from the operating position[6], and consequently in general, the optimum solution is to have only one central steering position. So, at present, it is strange to see a bridge equipped with more than one control station. However, in case there is more than one control station in the bridge, each propeller lever control per propeller unit is synchronized with the same ones in another control station. Then, all the control levers will move together and any station can take control of the propeller by means of a push button control; in that case, the selection

6 The *ASD* always with their bow towards the assisted ship "*bow first*" and the *tractor Z* with their stern towards the assisted ship "*stern* or *skeg first*."

Figure 8. Diagram of a control system combined for each of the propellers of the manufacturer Aquamaster in an ASD tug. Source: Aquamaster, drawing: author.



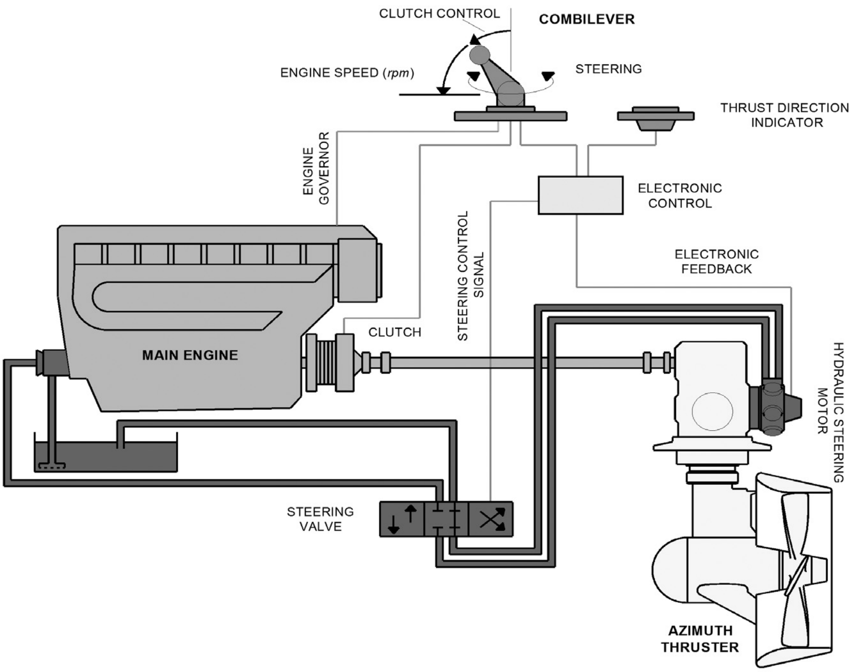


Figure 9. Detail of the control system of one of the two azimuth propellers of the manufacturer Ulstein in an ASD tug. In the diagram, a direct shaft line can be appreciated. Actually, at present and due to the new designs of the ASD tugs, the propellers are higher than their motors and the shaft line does not adopt such a simple disposition. Source Ulstein, drawing: author.

is indicated with a light, whereas the other control stations are now passive and will follow the one in command (now they are in slave mode).

In this case, the propeller control lever has the lever syncronizing unit which lets an alarm sound if the lever in the slave mode does not follow the lever in command. In the same way, the alarm is triggered (with indication and contact), when the operator turns or tilts the lever in the slave mode.

THE UNIQUE COMBINED CONTROL SYSTEM FOR THE TWO PROPELLERS

The manufacturers of the different azimuth propellers usually offer the option of installing a combined control (a kind of a *joystick*[7]), designed to control the movements of a ship equipped with several propellers (normally with up to four) by means of a single three-axis *joystick* which simultaneously controls all propulsion units.

The control programme is individually prepared for each vessel according to their individual hydrodynamic characteristics and steering performance.

There are already quite a number of different *joysticks* on the market[8] for the control of azimuth propeller tugs and, in short, they consist of a micro-computer controlled device for vessels equipped with more than one azimuth propeller. Through an input gear (a combination of a lever and horizontal wheel), the desired side-stepping direction and rotary motion of the vessel as well as the force of the movement are continuously fed at a set value for each *joystick* position. The direction of motion is indicated by the direction of the lever and the force of motion by the deflection of the lever[9].

Although it is not very common, there are tugs with this option; but it is necessary to point out two things: on the one hand, this is not suitable for a harbour tug because although the manoeuvre is simpler, its controls options are restricted to predeterminated parameters by the manufacturer as mentioned, following the client´s indications. On the other hand, the Clasification Societies demand that the tug has the possibility of individually controlling each propeller although a unique combined control system is installed. This is made not only as a security measure in case of a possible failure of the combined system, but also so that the tug Captain has the option to use all the functionality of an omnidirectional propulsion system which, otherwise, would be reduced if it only had the possibility of using the combined control system.

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[7] The word "*joystick*" is a term used in aviation and has its origin in the simple tiller and whipstaff for steering a boat or ship.t

[8] For example, Aquamaster denominates it "*Micropilot Control System*", while Schottel denominates it "*Master Pilot Control System*."

[9] For example, while the Aquamaster design constitutes a true *joystick* (this is a small lever that in the case of the *ASD tugs* controls the two azimuth propellers, but in the case of other types of ships it can control up to four azimuth propellers); in the case of Schottel, the design is similar to the independent control of each propeller (*copilot*).

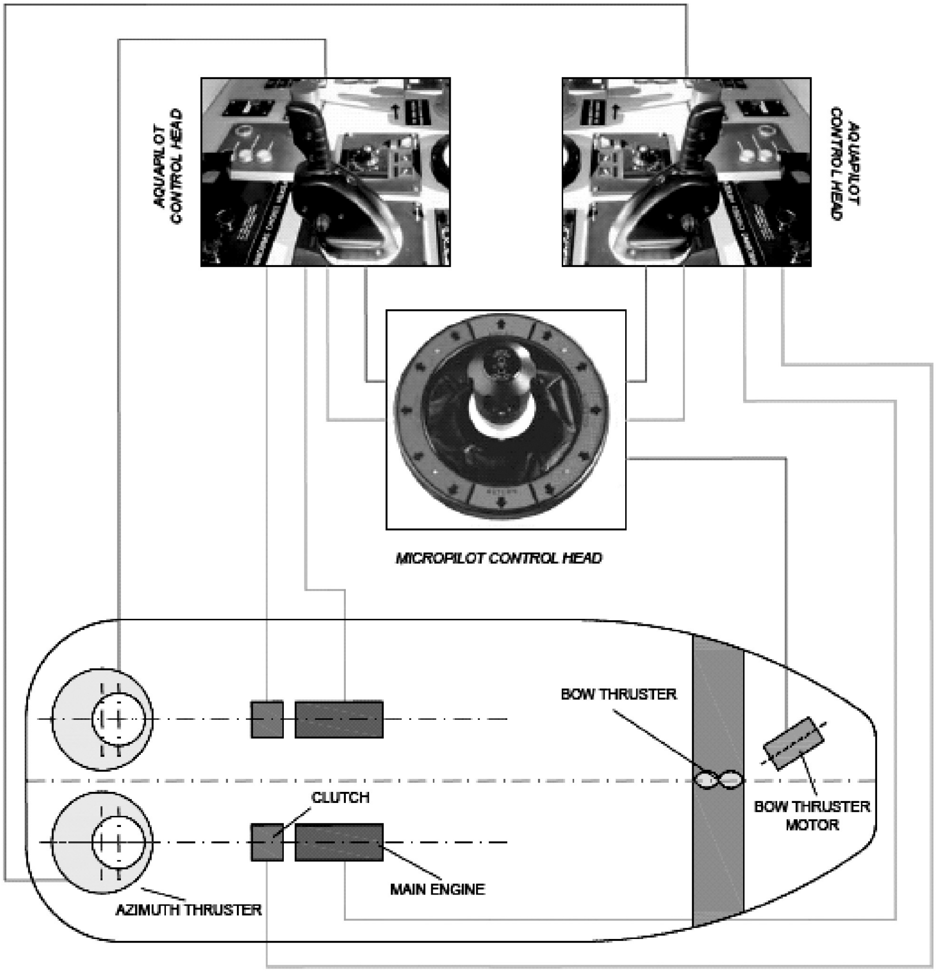

Figure 10. A typical Integrated Bridge System in an ASD Tug with an Aquamaster Micropilot Control System for two azimuth propellers and a bow thruster which controls every propulsion unit simultaneously. Source: Aquamaster, drawing: author

In conclusion, the Captain of an *ASD* or a *tractor Z* harbour tug with the necessary skill and with a combined control system should not use it to carry out the harbour assisting manoeuvres. On the contrary, he should use the independent electronic control system for each propeller. As a consequence, he gets a higher versatility which will improve the performance, especially the most difficult ones.

In figure 10 there is a diagram showing a combined control system by the manufacturer Aquamaster which controls all the propulsion units simultaneously[10]. As it can be appreciated, the lines that go to the propellers, which control the orientation of the thrust, and those that go to the respective motors that drive each one of them, control the *rpm*. In the case of the figure, an *ASD* tug with its two typical azimuth propellers aft and a tunnel bow thruster is represented, being able to control them all by means of the named *Micropilot Control System*.

CONCLUSIONS

The tugs with Azimuth propellers have got a remote electronic control from the bridge which controls both their orientation and *rpm* in their engines and the clutch.

In order to carry out the assistance manoeuvres with the best possible security, it is thought that the option of a combined control of the two Azimuth propellers in only one control, which some manufacturers offer, does not seem a recommendable solution in the case of the tugs as it restricts its manoeuvre capability. It limits the endless combinations of orientation and azimuth thrust to the parametres previously defined by the manufacturer in order to get the expected movement.

This electronic control system of the Azimuth propellers has been shown less hard and reliable compared with the mechanic control of the cicloidal propellers of the Tractor Voith tugs, although lately the azimuth propeller manufacturers are doing great efforts to improve its reliability. However, there are two disadvantages which are very difficult to overcome compared with the Voith mechanic control: on the one hand, higher keeping manteinance costs and on the other, in case of damage or breakdown, it may need more out-of-order time, as the manufacturer technical assistance is practically essential.

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10 In the state of the art tug denominated Rotor Tug which is endowed with two azimuthal propellers forward (the same as a *tractor Z*) and a third azimuth propeller aft in substitution of the skeg of the genuine tractor tugs, the manufacturer *Schottel* that has been the only one that has mounted the azimuth propellers in this type of tugs at the moment, has designed a system of combined control that controls the three propellers, denominated *Master Pilot Control System*. As it is the most natural thing, the tug is also equipped with an independent control for each one -denominated *Copilot*- which is not only advisable to give bigger versatility to the tug, but also obligatory, as the Classification Societies demand it.

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CONCEPTO Y MODO DE OPERACIÓN DEL SISTEMA DE CONTROL ELECTRÓNICO DE LOS REMOLCADORES DOTADOS DE HÉLICES ACIMUTALES

INTRODUCCIÓN

Aunque el concepto de hélice acimutal es una innovación que data de hace más de cincuenta años, aún hoy en día constituye una novedad para muchos armadores y operadores de buque. La idea básica que subyace detrás de una hélice acimutal es que puede girar 360º en torno a un eje vertical generando un empuje controlado omnidireccional en sentido horizontal, lo que significa una superior maniobrabilidad para los buques equipados con este sistema de propulsión suprimiendo al mismo tiempo la necesidad de un timón, de un sistema de inversión del sentido de giro de la hélice o de una reductora (la propia unidad de la hélice acimutal incorpora el sistema reductor de las rpm a las que gira el motor principal que le suministra potencia). A diferencia de las hélices cicloidales cuyo único fabricante es Voith Turbo Marine, de ahí el nombre genérico de hélices Voith con el que se les conoce; las hélices acimutales se fabrican por diferentes firmas motivo por el que se les conoce bajo diversos nombres tales como Z-drives, rudderpropellers, rotatable propulsion units, omnidirectional thrusters aunque en ocasiones el nombre de sus fabricantes como Shottel, Ulstein Rolls-Royce, Steerprop, Niigata, Duckpeller etc. se emplean como un término genérico aplicable para describir a todas las hélices acimutales.

Este sistema de propulsión se compone de diferentes dispositivos: las propias hélices acimutales, las unidades de gobierno y control de las mismas y las líneas de ejes. El sistema quizá podría denominarse en su conjunto con más propiedad como un “*sistema de propulsión y gobierno*”.

En España, los remolcadores que están equipados con este tipo de hélices son el tractor Z (sus hélices van en la parte de proa de casco al igual que los remolcadores *tractor Voith*) y los *Azimuth Stern Drive (ASD)* cuyas hélices van ubicadas a popa al igual que los remolcadores convencionales.

El control remoto de estas hélices desde el puente (velocidad y dirección del empuje) es electrónico y cada fabricante tiene su propio diseño, aunque el sistema de control y la disposición combinada de las hélices para conseguir el empuje en una determinada dirección es muy similar.

En este artículo se analizan en general los dispositivos de control electrónico remoto desde el puente de las hélices acimutales, especialmente su funcionamiento y modo de operación, sin hacer distinción entre aquéllas que van ubicadas a proa (*tractor Z*) o a popa (*ASD*), ya que su fundamento es similar y la diferencia radica en la orientación de la hélice y consecuentemente en la posición de los controles remotos desde el puente para conseguir el empuje que se pretende.

UN SISTEMA DE CONTROL ELECTRÓNICO INDEPENDIENTE PARA CADA HÉLICE ACIMUTAL

Se aborda inicialmente el estudio del típico sistema de control independendiente para cada hélice acimutal que generalmente consiste en un sistema seguimiento "*follow-up remote control system*" mediante control remoto desde el puente y que es independiente para el control de cada una de las dos hélices que montan estos remolcadores. La unidad básica de operación es la unidad de control principal, que recibe diferentes denominaciones según el fabricante de las hélices y que en general consiste básicamente en una simple palanca que se puede girar 90º en sentido vertical y 360 en sentido horizontal con lo que se consigue el control de la orientación, las *rpm* y el embrague de la hélice.

El sistema de control de las hélices incluye adicionalmente un sistema secundario sin seguimiento "*non follow-up*" generalmente conocido en el argot de los remolcadores como gobierno secundario, aunque en vista de su función (pensado para emplearse principalmente en caso de un fallo del sistema de seguimiento "*follow-up*" visto anteriormente), podríamos denominarlo más propiamente como un gobierno de emergencia en caso de fallo del sistema principal.

Este sistema alternativo de gobierno consiste en dos pulsadores que tienen una conexión eléctrica directa e independiente con cada una de las hélices en forma de circuitos electrónicos, esto es, están conectados directamente a cada hélice.

Aunque en los remolcadores de última generación no es normal, en el caso de que exista más de una consola de control en el puente, cada palanca de control de cada hélice se sincroniza con la palanca correspondiente que exista en cualquier otra estación de control, actuando de *esclava* la que no tiene el control de mando transferido.

EL SISTEMA DE CONTROL COMBINADO DE TODOS LOS SISTEMAS DE PROPULSIÓN DEL REMOLCADOR

Las fabricantes de las diferentes hélices acimutales, generalmente ofertan la opción de instalar adicionalmente un control combinado y simultáneo de todas las unidades propulsoras (una especie de *joystick*), diseñado para controlar los movimientos de un buque equipado con varias hélices (normalmente hasta cuatro) por medio de un único *joystick* que controla simultáneamente todas las unidades propulsoras.

El programa de control se prepara individualmente para cada buque, de acuerdo con sus características hidrodinámicas y capacidad de gobierno.

En el caso de los remolcadores existen diferentes *joysticks* en el mercado que comercializan los fabricantes de sus hélices y que en resumen consisten en un dispositivo compuesto de una microcomputadora controlada que está pensada para buques equipados con más de una hélice acimutal. La dirección del movimiento del buque se genera mediante la dirección de la palanca del *joystick* y la fuerza de ese movimiento mediante la desviación de dicha palanca con relación a la vertical.

CONCLUSIONES

Los remolcadores dotados de hélices acimutales van dotados de un sistema de control electrónico remoto desde el puente que controla tanto la orientación de las mismas, como las *rpm* de sus motores y el embrague.

Con el fin llevar a cabo las maniobras de asistencia con la mayor seguridad posible, se estima que la opción de un control combinado de las dos hélices acimutales en un solo mando que ofertan algunos fabricantes y que suelen instalarse en otro tipo de buques, no parece una solución muy recomendable en el caso de los remolcadores por cuanto restringe su capacidad de maniobra al limitar las infinitas combinaciones de orientación y empuje de sus hélices a los parámetros definidos previamente por el fabricante para conseguir el movimiento que se pretende.

Este sistema de control electrónico de las hélices acimutales se ha revelado desde su implantación algo menos robusto y fiable si lo comparamos con el control mecánico de las hélices cicloidales de los remolcadores *tractor Voith*, si bien es verdad que en los últimos años, los distintos fabricantes de hélices acimutales están haciendo grandes esfuerzos para mejorar su fiabilidad, aunque probablemente se mantengan dos desventajas difíciles de superar si se compara con el control mecánico de *Voith* y que son de una parte, unos mayores costes de mantenimiento y de otra, que en caso de avería, puede ocasionar unos períodos mayores fuera de servicio, al resultar prácticamente imprescindible la asistencia técnica del fabricante.

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COMPARISON OF TWO PI METHODS APPLIED TO FDI ON SHIPS DYNAMICS

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Abstract

Most of non-linear type one and type two control systems suffers from lack of detectability when model based techniques are applied on fault detection and isolation (FDI) tasks. This research is centred on frequency techniques applied to identify ship's model parameters (PI) including non-structured or partially known structured models using backpropagation neural networks as functional approximators. The results of the comparison of two strategies based in frequency techniques are presented. Such frequency techniques are:

— Mapping the frequency response associated to system parameters when a closed loop controlled ship is excited by the well-known harmonic balance test (HBT).

— Mapping the frequency response associated to system parameters when closed loop controlled ship is excited by a group of sinusoidal inputs added to the manipulated variable (CLFRT).

With achieved frequency response mappings, system parameters are associated by means of functional approximation techniques. In this case, Feedforward neural networks trained with backpropagation conjugate gradient algorithm are massively used. Finally, PI results are used in FDI tasks, where nominal plant parameters are matched against on-line estimated parameters on a parity space approach.

Keywords: Backpropagation, Conjugate gradient, Parameter identification, Fault detection, Frequency response, Harmonic balance, Neural Networks.

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INTRODUCTION

Safety in process industry can be strongly related to the detection and isolation of the features indicative of changes in the sensors actuators or process performance. In using model-based approaches, when the models describing the process are accurate, the problem of fault detection may be solved by observer-type filters. These filters generate the so-called residuals computed from the inputs and outputs of the process. The generation of these residual signals is the first stage in the problem of fault detection and isolation (FDI). To be useful in the FDI task, the residuals must be insensitive to modelling errors and highly sensitive to the faults under consideration. In that regard, the residuals are designed so that the effects of possible faults are enhanced, which in turn increases their detectability. The residuals must also respond quickly. The residuals are tested in order to detect the presence of faults. Various FDI methods have been previously reported, such as the papers of Willsky, A. S. (1976), Isermann, R. (1984), Frank, P. M. (1987a), Gertler, J. J. (1988), Patton, R. J. and Chen, J. (1991). Among the classic books on the subject are those of Himmelblau, D. M. (1978), Pau, L. F. (1981), Basseville M. (1986) .

Model based fault detection methods

Fault detection methods based on process and signal models include actuators, processes and sensors for which inputs and output variables must be precisely measured. Such methods deal mainly with parameter estimation, state observers and parity equation methods. If measuring system fails, fault detection methods based on the use of input/output measurements yields ambiguous and/or erroneous results.

A lot of research on model based fault detection methods has been carried out during the last three decades. In this section a brief list on process model based fault detection methods is given:

1.Fault detection with parameter estimation Gertler, J. J. (1988) Isermann, R. (1992), Isermann, R. (1993), Mosler, O., Heller and R. Isermann (2001), Newmann, D. (1991).
   - Equation error methods
   - Output error methods
   - Frequency techniques

2. Fault detection with state-estimation.
   (a) Dedicated observers for multi-output processes.
   - State Observe, excited by one output Clark, R. N. (1978a):
   - Kalman filter, excited by all outputs] Mehra, R. K. and Peschon, J. (1971):, Willsky, A. S. (1976),.
   - Bank of state observers, excited by all outputs Willsky, A. S. (1976),
   - Bank of state observers, excited by single outputs Frank, P. M. (1987a)

– Bank of state observers, excited by all outputs except one Frank, P. M. (1987a).
(b) Fault detection filters for multi-output processes Beard, R. V. (1971).

3. Fault detection with parity equations Isermann, R. (1984), Gertler, J. J. (1991):, Patton, R. J. and Chen, J. (1994):.
   (a) Output error methods.
   (b) Polynomial error methods.

4. Fault detection using analytical redundancy Ragot J., Maquin D., Kratz, F., 2000.
   (a) Static analytical redundancy.
   (b) Dynamic analytical redundancy.

No general method exists for solving all FDI cases. Successful FDI applications are based on a combination of several methods. Practical FDI systems apply analytical redundancy using the so-called first-principles like action-reaction balances such as mass flow rate balance, energy flow rate balance, force/torque/power balances and commonly, the mathematical balance of any cause-effect equilibrium condition.

As stated before, diagnosing techniques previously mentioned, when applied to non-linear type one and type two processes, suffers from lack of detectability. With regard to residuals, they are the outcomes of consistency checks between the plant observations and a mathematical model. The three main ways to generate residuals are parameter estimation, observers and parity relations. For parameter estimation, the residuals are the difference between the nominal model parameters and the estimated model parameters. Derivations in the model parameters serve as the basis for detecting and isolating faults.

In most practical cases the process parameters are partially not known or not known at all. Such parameters can be determined with parameter estimation methods by measuring input and output signals if the basic model structure is known. There are two conventional approaches commonly used which are based on the minimization of equation error and output error. The first one is linear in the parameters and allows therefore direct estimation of the parameters (least squares) in non-recursive or recursive form. The second one needs numerical optimisation methods and therefore iterative procedures, but may be more precise under the influence of process disturbances. The symptoms are deviation of the process parameters. As the process parameters depend on physically defined process coefficients, determination of changes usually allows deeper insight and makes fault diagnosis easier Isermann, R. (1984. These conventional methods of parameter estimation usually need a process input excitation and are especially suitable for the detection of multiplicative faults. Parameter estimation requires an input/output correct measuring system.

Some drawbacks of such methods are:

— the possibility of faulty measuring signals,

— an unknown model structure or

Goals to be achieved

Afore-mentioned diagnosing techniques, when applied to non-linear type one and type two processes, suffer from lack of detectability. For this reason the following work will be oriented to the problem of fault detection, fault isolation, and fault estimation by a novel parameter estimation method using residual generation on the basis of parity space approach. The proposed parameter estimation method is based on functional approximation techniques implemented with backpropagation neural network (BPNN) even under a faulty measuring system.

The main tasks to be carried out are:

— Implementation of a fault tolerant data acquisition method by means of frequency based techniques to achieve a consistent database

— Implementation of two PI methods based on the association of frequency responses with functional approximation

— Comparison of both methods on a ship model steering process

Subsequent sections are devoted to the description of such technique, resulting in an interesting complement or substitute to some conventional mentioned techniques.

FREQUENCY BASED PARAMETER ESTIMATION TECHNIQUES

Introduction

This research work is focused on the problem of fault detection, fault isolation on the basis of parameter estimation by functional approximation implemented with backpropagation neural networks associated to frequency techniques on non-linear type one and type two systems, for which serious problems with detectability exist.

Conventional parameter estimation techniques are affective if measuring system operates free of faults. That means, measurement equipment operates without drift errors. Consequently, when the possibilities of sensor drift errors exist, the methods described below, in this section, are proposed.

Process Characteristics Based on HB Tests

The output of a nonlinear system under the effect of a disturbance or a step input may go into a steady-state oscillation about an equilibrium point and still be considered stable. Such an oscillation is called a *limit cycle* and is a periodic though not sinusoidal oscillation whose amplitude and frequency are dependent only upon the magnitude of the input and the characteristics of the system for non linear systems and are dependent only upon the characteristics of the system for linear systems.

The ultimate amplitude and frequency are particular characteristics of all transfer functions. HB is a procedure formerly used in process controllers auto-tuning which is an attractive technique for determining the real time ultimate frequency and gain of a process. When stationary processes are under consideration, the technique does provide a useful tool for the process parameter changes detection as shown in this work. If a change in the values of ultimate frequency and ultimate amplitude is observed, this means that some parameters of transfer function have changed. The method requires a relay feedback or a closed loop controlled by a relay around the setpoint as shown in figure 1.

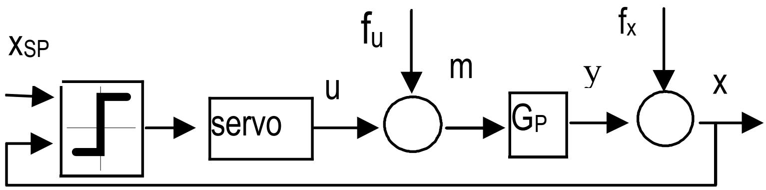

Fig. 1. Structure of the HBT

With regard to figure 2, a relay of height *h* is inserted as a feedback controller. The manipulated variable *m* is increased by *h* above the steady-state value. When the controlled variable *x* crosses the setpoint, the relay reduces *m* to a value *h* below the steady-state value. The system will respond to this "bang-bang" control by producing a limit cycle, provided the system phase angle drops below -180°, which is true for all real processes.

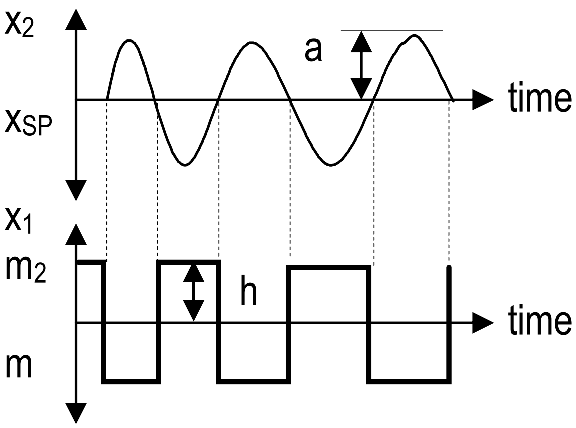

Fig. 2. Input output signals of HBT

The period of the limit cycle is the ultimate period (P_u) for the transfer function relating the controlled variable *x* and the manipulated variable *m*. So the ultimate frequency is given as

$$\omega_u = \frac{2\pi}{P_u} \qquad (1)$$

A series Fourier expansion of the relay output shows that the amplitude of first harmonic component is $\frac{4h}{\pi}$, and the error signal has the amplitude

$$a = \frac{4h}{\pi}\left|G(i\omega_u)\right| \tag{2}$$

The condition for sustained oscillation (limit cycle) is that

$$\arg G(i\omega_u) = -\pi \quad \text{and} \quad K_u = \frac{1}{\left|G(i\omega_u)\right|} \tag{3}$$

Consequently the ultimate amplitude of the transfer function is given by

$$a = \frac{4h}{\pi K_u} \tag{4}$$

where
h = height of the relay,
a = amplitude of the primary harmonic of the output x.

It follows that if any change in system parameters takes place, then the ultimate frequency, ultimate amplitude, or both, will change also. Such concept can be defined as a function of ultimate frequency and ultimate amplitude of primary harmonic of the output. This property is expressed as:

$$(\omega_u, a) = f(P_i); \quad i = 1, \cdots M \tag{5}$$

with P_i the system parameter set. So that, the condition to asseverate system parameter invariance, which means to confirm that no parameter has changed is

$$(\omega_u, a) = f(P_{iN}) = \omega_{uN}, a_{uN} \tag{6}$$

where ω_{uN} and a_{uN} are the nominal ultimate frequency and amplitude respectively corresponding to the nominal parameters set P_{iN}.

It should be noted that Eq. (5) and (6) give us approximate values for ω_u and a_u because the relay feedback introduces an additional nonlinearity into the system. However, for most systems, the approximation is close enough for engineering purposes.

Nevertheless, when systems transfer functions are influenced by any auxiliary or external variable, (variables different of the input/output of the transfer function), they should be taken into account. As consequence of the existence of such variables (6) can be rearranged as follows:

$$(\omega_u, a) = f(P_i, V_i); \tag{7}$$

If the relay of height *h* inserted as a feedback controller is externally forced to change its eight to a new value, which means to change the manipulated variable, then a different pair of ultimate period and amplitude is achieved. Such idea is expressed as

$$\begin{aligned} &m_1 \Rightarrow h_1 \Rightarrow (\omega_{u1}, a_1) \\ &m_2 \Rightarrow h_2 \Rightarrow (\omega_{u2}, a_2) \\ &\vdots \\ &m_i \Rightarrow h_i \Rightarrow (\omega_{ui}, a_i) \end{aligned} \tag{8}$$

Consequently, the application of (5) yields

$$\begin{vmatrix} (\omega_{u1}, a_1) \\ (\omega_{u2}, a_2) \\ \vdots \\ (\omega_{ui}, a_i) \end{vmatrix} = f(P_i, V_i) \tag{9}$$

Expression (8) states any pair of ultimate period and amplitude of the group described by (9) is function of the complete set of plant parameters and related external variables (coupling variables).

Process Characteristics Based on CLFRT

Frequency response is understood as the gain and phase response of a plant or other unit under test at all frequencies of interest. Although the formal definition of frequency response includes both the gain and phase, in common usage, the frequency response often only implies the magnitude (gain). In this study phase response must be considered.

The frequency response *H(f)* is defined as the inverse Fourier Transform of the Impulse Response *h(τ)* of a system.

$$H(f) = \int_{-\infty}^{\infty} h(\tau) e^{-j2\pi f \tau} d\tau \tag{10}$$

Frequency response measurements require the excitation of the system with energy at all relevant frequencies. The fastest way to perform the measurement is to use a broadband excitation signal that excites all frequencies of interest simultaneous, and use FFT techniques to measure at all of these frequencies at the same time. Using random noise excitation best minimizes noise and non-linearity, but short impulses or rapid sweeps (chirps) may also be used. The selected excitation signal for this study is of the type given as

$$f(\omega t) = \sum_{i=1}^{n} A_i Sin(\omega_i t) \tag{11}$$

with two or three relevant frequencies and same amplitude yielding for the case of three relevant frequencies

$$f(\omega t) = A_1 Sin(\omega_1 t) + A_2 Sin(\omega_2 t) + A_3 Sin(\omega_3 t) \tag{12}$$

Excitation function can be applied simultaneously or sequentially. Obviously when simultaneously, the CLFRT is faster that sequentially but under noisy systems accuracy is poorer.

When the desired resolution bandwidth of interest is less than about 100 kHz, the fastest way to measure the frequency response functions is to use FFT based techniques as it is done in this work.

For proper measurement, it is also important to take into account the nature of the type of signals that we are dealing with.

As a rule of thumb, if there is a given percent distortion or noise in the system, the error will be of the same order of magnitude. The output must be statistically correlated to the input. This assumption is normally true in high fidelity analog systems. However, in mechanical systems, as well as systems with complex transmission mechanism and/or with digital encoding, echo cancelling, and other adaptive techniques, this assumption may not be fulfilled. To account for all of the above, it can be used digital signal processing techniques, including FFT and cross-spectral methods.

The output of a stable nonlinear system under the effect of a sinusoidal continuous disturbance consists in a steady-state oscillation about an equilibrium point and still be considered stable. Such an oscillation similar to a *limit cycle* is a periodic though not sinusoidal oscillation whose amplitude $|G|_\omega$ and phase ϕ_ω is dependent only upon the magnitude of the input and the characteristics of the system.

When stationary processes are under consideration, the technique does provide a useful tool for the process parameter changes detection as shown in this work.

If a change in the characteristic values of the frequency response (amplitude and phase) is observed, this means that some parameters of transfer function has changed. The method requires a sine generator added to a closed loop controller as shown in figure 3.

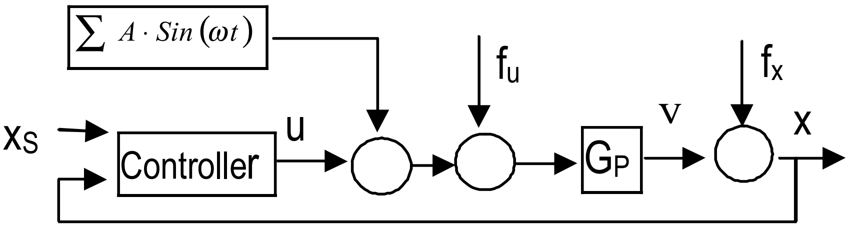

Fig. 3. Structure of the CLFRT

It follows that if any change in system parameters takes place, then, the magnitude and phase will change also. This property is expressed as:

$$(|G|_{\omega}, \phi_{\omega}) = f(P_i); \quad i = 1, \cdots M \tag{13}$$

with P_i the system parameter set. So that, the condition to asseverate system parameter invariance, which means to confirm that no parameter has changed, is

$$(|G|_{\omega}, \phi_{\omega}) = f(P_i) = |G|_{\omega N}, \phi_{\omega N} \tag{14}$$

where $|G|_{\omega N}$ and $\phi_{\omega N}$ are the nominal amplitude and phase respectively corresponding to the nominal parameters set P_{iN}.

It should be noted that Eq. (13) and (14) give us approximate values for $|G|_{\omega N}$ and $\phi_{\omega N}$ because the measuring system introduces an additional error into the system which must not be relevant. However, for most systems, the approximation is close enough for engineering purposes.

Nevertheless, when a system transfer function is influenced by any auxiliary or external variable, (variables different of the input/output of the transfer function), they should be taken into account. As consequence of the existence of such variables, (13) can be rearranged as follows:

$$(|G|_{\omega}, \phi_{\omega}) = f(P_i, V_i) \tag{15}$$

If a sinusoidal function of amplitude A inserted in parallel with a feedback controller is forced to change its frequency to a new value, then a different pair of amplitude and phase as frequency response is achieved. Such idea is expressed as

$$\begin{aligned} &A_1, \omega_1 \Rightarrow (|G|_{\omega 1}, \phi_{\omega 1}) \\ &A_2, \omega_2 \Rightarrow (|G|_{\omega 2}, \phi_{\omega 2}) \\ &\vdots \\ &A_n, \omega_n \Rightarrow (|G|_{\omega n}, \phi_{\omega n}) \end{aligned} \tag{16}$$

for identification purposes the amplitude of the excitation signal can be selected such that $A_1 = A_2 = \ldots A_n$ yielding

$$\begin{aligned} &A, \omega_1 \Rightarrow (|G|_{\omega 1}, \phi_{\omega 1}) \\ &A, \omega_2 \Rightarrow (|G|_{\omega 2}, \phi_{\omega 2}) \\ &\vdots \\ &A, \omega_n \Rightarrow (|G|_{\omega n}, \phi_{\omega n}) \end{aligned}$$

Consequently, the application of (15) yields

$$\left| \begin{matrix} (|G|_{\omega 1}, \phi_{\omega 1}) \\ |G|_{\omega 2}, \phi_{\omega 2}) \\ \vdots \\ (|G|_{\omega n}, \phi_{\omega n}) \end{matrix} \right| = f(P_i, V_i) \tag{17}$$

Expression (17) states that any pair of amplitude and phase is function of the complete set of plant parameters and related external variables (coupling variables).

Advantages of the methods

These methods has several distinct advantages over conventional parameter estimation methods:

a) It doesn't depend on the output measuring errors (drift of system output sensors)

b) No a priori knowledge of the system parameters is needed. The method automatically results in a sustained oscillation at the excitation frequency of the process. The only parameter that has to be specified is the frequency and amplitude of excitation signal or the relay eight for HBT.
c) They are closed loop tests, so the process will not drift away from the set-point. This is precisely why the methods works well on highly nonlinear processes. The process is never pushed very far away from the steady-state conditions

GENERAL PROCEDURE

To fulfil the requirements for training a feedforward backpropagation neural network, a database for every PI method is needed. For the case of HBT, a database relating ship parameters P_{ij} and the associated pairs of (a_{Uj}, P_{Uj}) must be achieved. Figure 4(a) shows the implementation of HBT to achieve the actual demanded data. For the case of CLFRT, a database relating ship parameters with the magnitude and phase responses are needed. Such pairs of ultimate values corresponding to actual plant parameters are nominal ultimate values if, and only if, plant parameters are nominal. In this situation it is assumed a fault free plant operation mode. Figure 4(b) shows the implementation of CLFRT to achieve the actual demanded data

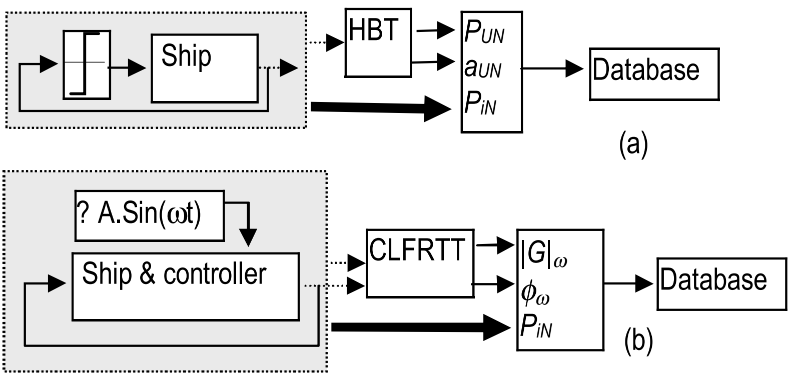

Fig .4. Tasks to achieve the necessary data to be used in NN training. (a), HBT. (b) CLFRT.

The simplest idea to identify only one parameter consists in applying a functional approximation technique based in the use of backpropagation neural networks properly trained as shown in figure 5. Figure 5 illustrates the case of a plant with known model structure and three (but could be any other quantity) accessible (known by any means) parameters P_1, P_2 and P_3. It shows a ship in which an HBT is executed. As consequence of applied HBT, a database is filled with achieved data. With the recent data contained into the database, a training session is performed and a NN based model for the patter parameter is achieved. The same sequence can also be performed under CLFRT in the same order that for HBT.

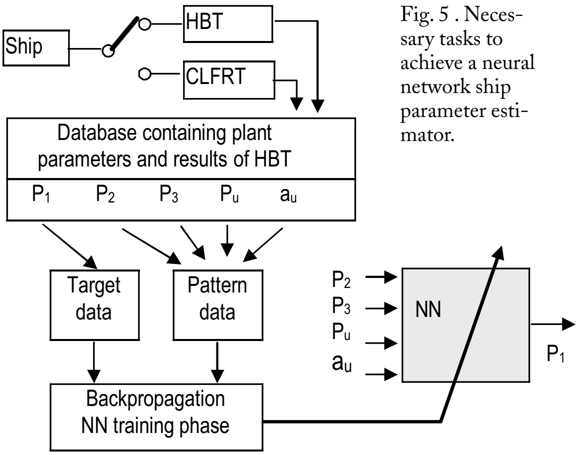

Fig. 5 . Necessary tasks to achieve a neural network ship parameter estimator.

Figure 6 shows the scheme adopted for comparison purposes on the basis of parameter estimation, which consists in a group of trained neural networks, ready to identify one or more plant parameters, w hen real time or actual data is applied to the inputs. So that, the tasks necessary to identify at least a plant parameter, requires again the on line HBT or CLFRT tasks to obtain the actual pair of ultimate gain and period or alternatively, the frequency response data. By introducing such actual values, including the rest of known parameters to the neural network inputs, it yields at the output, the actual value of the plant unknown parameter. The accuracy in the value of the estimated parameter is crucial because it will be straightaway applied on the last phase of the FDI task.

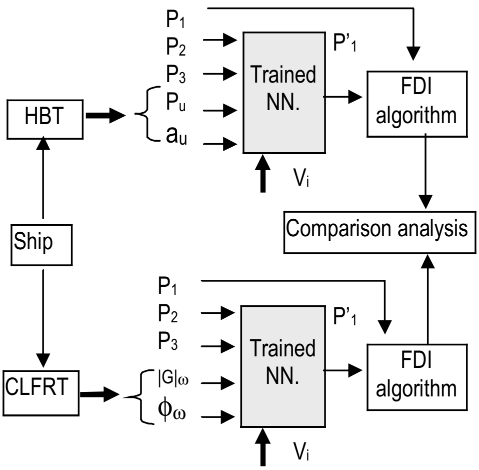

Fig. 6. Scheme of paramostic tasks and comparison of PI methods.

APPLICATION TO PI USING A NOMOTO MODEL

In order to validate the PI methods by comparison of model parameters accuracy, one of the simplest models of a ship is selected Ferreiro García R., Haro Casado M.(2006),. The transfer function from rudder angle δ to heading ψ is described for our purposes under a Nomoto model as

$$G(s) = \frac{\psi(s)}{\delta(s)} = \frac{b}{s(s+a)} \tag{18}$$

where according with Ästrom and Wittenmark (1989), model parameters can be approached as

$$a = a_0 \frac{u}{l} \quad \text{and,} \quad b \approx 0.5\left(\frac{u}{l}\right)^2 \frac{Al}{D} \tag{19}$$

with *l* the ships length in m, *u* the ship velocity in m/s, *A* the rudder area in m^2, and D, the ship's displacement in m^3. According described Nomoto model, the parameters *a* and *b* depends on the ship velocity *u* and ship displacement *D*. The rudder servo operates with a speed of 4 degrees/s limited to ± 30 degrees. The ruder area is assumed as 20 m^2. Consequently, the simulation model necessary to validate both the parameter identification procedures is of the type

$$G(s) = \frac{0.5\left(\frac{u}{l}\right)^2 \frac{Al}{D}}{s(s + a_0 \frac{u}{l})} \tag{20}$$

where the servo model is shown in figure 7.

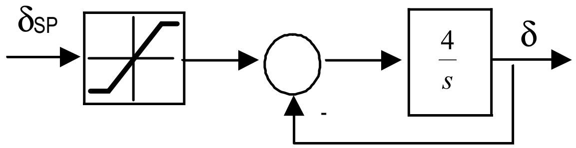

Fig. 7. Simplified rudder-servo model

The closed loop controller is an adaptive PID adjusted by gain scheduling to adapt its parameters as function of ship displacement under the criterion of minimum overshoot and response time. According such requirements, the PID parameters are shown in able I.

$D/10^3$ PID	4	6	8	10
Kp	4	3.4	2.9	2.6
Ti	120	135	160	190
Td				

Table I. PID parameters as function of displacement

Training procedure

Both PI methods are then applied by simulation on the closed loop model, and consequently, two databases shown in tables II and III were achieved. Input data to achieve the necessary database uses ship velocities from 4 to 10 m/s while ship displacement varies from 4000 to 10000 tons on the container ship, where rudder area is 20 m^2.

u	$D/10^3$	$\lvert G\rvert_{\omega 1}$	$\phi_{\omega 1}$	$\lvert G\rvert_{\omega 2}$	$\phi_{\omega 2}$
4	4	0,0801045	-68,0906	0,0416869	-89,5841
4	6	0,0815297	-73,8223	0,0415882	-95,779
4	8	0,0820465	-78,9383	0,0407358	-103,086
4	10	0,0813522	-83,8892	0,0394238	-109,301
6	4	0,07857	-63,2443	0,0413649	-81,0538
6	6	0,0788952	-65,1473	0,041328	-84,9855
6	8	0,0793861	-67,5302	0,0413907	-87,3135
6	10	0,0797526	-70,0747	0,041473	-90,4957
8	4	0,0779755	-61,4826	0,041303	-79,5276
8	6	0,0781531	-63,0128	0,0413122	-80,7345
8	8	0,0784074	-63,4854	0,0412631	-81,2487
8	10	0,0786551	-64,0515	0,0411353	-84,4931
10	4	0,0778708	-59,4287	0,0414477	-76,4577
10	6	0,0777866	-61,1995	0,0412255	-79,3707
10	8	0,0778946	-62,6008	0,0412056	-80,3609
10	10	0,0779463	-63,2211	0,0411875	-80,8526

Table II. Database for CLFRT method

The data of such consistent databases achieved on the basis of Nomoto model is used in backpropagation neural network training phase. The training algorithm selected is the conjugate gradient Fletcher-Reeves implemented on the Neural Network toolbox of Matlab under off-line training sessions. Consequently, the selected neural network architecture is defined by means of the Matlab expression:

net = newff(minmax (p), [10, 10, 1], {'tansig','tansig','purelin'},'traincgf')

which consists of a feedforward Backpropagation NN with two hidden layers and ten neurons per layer, trained by means of ,*'traincgf'* algorithm of Matlab.

After several training sessions with different data structures from the database, some of the training results are shown in table IV.

Mean Square Error (MSE) is an acceptable performance index to evaluate the estimates accuracy of training results as shown in table I. By comparing the values of rows corresponding to the index MSE, some differences are observed, which indicates the degree of accuracy that could be expected when on-line parameter estimation method is applied. For CLFRT, MSE = 0.0989 and for HBT MSE = 0.12088. Consequently, CLFRT procedure is expected to be more effective than HBT.

u	D/10^3	Pu	a
4	4	88.2353	0.181148
4	6	93.75	0.163727
4	8	93.75	0.109534
4	10	88.2353	0.0852799
6	4	68.1818	0.338723
6	6	75	0.270966
6	8	75	0.18365
6	10	75	0.157505
8	4	62.5	0.399597
8	6	62.5	0.282363
8	8	62.5	0.233929
8	10	62.5	0.174988
10	4	55.5556	0.623681
10	6	55.5556	0.351874
10	8	55.5556	0.290567
10	10	55.5556	0.245539

Table III. Database for HBT method

Some results

Figure 8 shows both PI methods ready to be used on parameter identification. The results of a PI session shows that the accuracy of parameters achieved by using CLFRT is better than using the HBT, according with the MSE value of table I. Consequently, with acceptable parameter identification values, the residuals of the parity space

Training algorithm	Epochs	MSE	Gradient	Test
Traincgf-srchcha	100/100	0.0989	5.426e-6	CLFRT
Traincgf-srchcha	100/100	0.1209	2.979e-6	HBT

Table IV. Training characteristics

approach are achieved in the FDI tasks. FDI task is then carried out using simple rules in a rule based task. Residuals necessary to implement the last phase of FDI task, are achieved by comparing the actual ship velocity with the estimated one. As consequence of such comparison, display 2 and display 3 of figure 8 shows respectively –0.1147 and 0.09682. Under the assumption of a fault free process, then CLFRT method is more accurate than HBT.

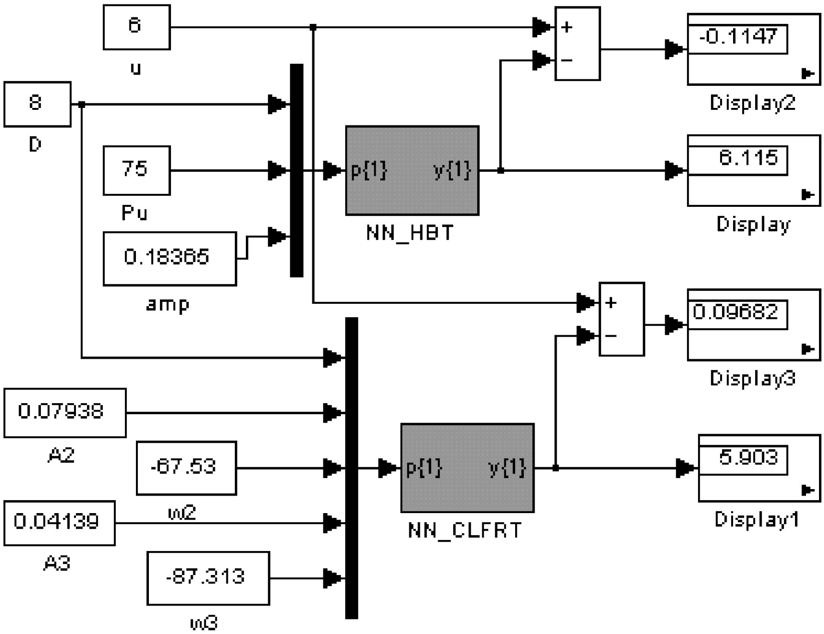

Fig. 8. Implementation of both PI methods for comparison purposes.

CONCLUSIONS

The aim of this research was to compare both PI strategies which consists in develop and implement a method to detect and isolate faults on the basis of parameter variation detection on a ship even when under faulty measuring systems (output sensor drift). Furthermore, the method is focused towards plants of type 1 and type 2 where conventional PI methods are not quite effective.

The most relevant advantages of these strategies are:

— PI method when applying HBT and CLFRT doesn't depend on the quality of measuring system in cases of sensor drift.

— The on-line test can be applied with the ship operating under nominal setpoints, without disturbing or interrupting the ship course for instance.

— The strategy is useful under partially known model structures.

As a result of comparison of both methods, CLFRT is more accurate than HBT, and hence, FDI task is more reliable.

The most relevant disadvantages of both frequency based methods are due to:

— The time necessary to estimate the parameters increase with both, the accuracy and number of parameters.

— The computational effort increase with the required accuracy and the number of parameters to be estimated.

ACKNOWLEDGMENTS

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COMPARACIÓN DE DOS MÉTODOS DE IDENTIFICACIÓN DE PARÁMETROS APLICADOS A LA DETECCIÓN Y AISLAMIENTO DE FALLOS EN LA DINÁMICA DE BUQUES

Resumen

La mayor parte de los sistemas no lineales de los tipos uno y dos sufren de escasez de detectabilidad al aplicar métodos de detección y aislamiento de fallos basados en modelos. Este trabajo está centrado en técnicas frecuenciales aplicadas para identificar parámetros del modelo de un buque incluyendo modelos no estructurados o parcialmente estructurados, utilizando aproximadoRes funcionales basados en redes neuronales por propagación hacia atrás. Se presentan los resultados de la comparación entre dos estrategias de identificación basadas en técnicas frecuenciales. Tales técnicas frecuenciales consisten en:

—Mapear la respuesta frecuencial asociada con los parámetros del buque cuando al aplicar el método del balance armónico en lazo cerrado (HBT).

—Mapear la respuesta frecuencial asociada con los parámetros del buque cuando se añade una señal de excitación a la salida del controlador por realimentación (CLFRT).

Las citadas respuestas frecuenciales acumuladas en una base de datos asociadas a los parámetros del buque, proporcionan los aproximadores funcionales para la identificación de parámetros. Se utilizan masivamente redes neuronales artificiales entrenadas por propagación hacia atrás mediante el algoritmo del gradiente conjugado. Finalmente, los resultados de las tareas de identificación de fallos son utilizados en la detección y aislamiento de fallos en línea y los resultados de ambos métodos de identificación son comparados entre sí para establecer baremo de calidad.

INTRODUCCIÓN

Este trabajo comienza describiendo los antecedentes de los métodos de detección y aislamiento de fallos en base a la estimación de parámetros. A continuación se describen dos métodos frecuenciales (HBT y CLFRT) de estimación de parámetros del buque en base a la utilización de redes neuronales como aproximadotes funcionales. Seguidamente, se realiza una aplicación de la identificación de parámetros utilizando los métodos frecuenciales propuestos con el modelo de Nomoto y posteriormente se comparan los resultados de ambos métodos para establecer un baremo de calidad.

RESULTADOS

La figura 8 muestra ambos métodos de identificación de parámetros listos para identificación. Los resultados de una sesión de identificación muestran que la precisión en la determinación de los parámetros conseguida mediante el método CLFRT es mejor que la conseguida mediante HBT, de acuerdo con el criterio MSE (mean square error) de la tabla I. En consecuencia, se aplican las técnicas de detección y aislamiento de fallos o anomalías con valores aceptables de identificación de parámetros que proporcionan los correspondientes residuos dentro del espacio de paridad. La tarea de detección y aislamiento es llevada a cabo por medio de razonamiento basado en reglas. Los residuos necesarios para implementar la última fase de aislamiento de anomalías se llevan a cabo comparando la velocidad real del buque con la velocidad estimada por el modelo dependiente de los parámetros estimados. Como consecuencia de tal comparación, se han obtenido resultados de los residuos de -0.1147 para HBT y 0.09682 para CLFRT, constatando que es de mayor calidad el método de CLFRT frente a HBT, bajo la suposición de un ensayo en condiciones nominales o libre de fallos.

CONCLUSIONES

El objetivo de este trabajo consistía en comparar ambos métodos de identificación de parámetros posteriormente utilizados para la detección y aislamiento de anomalías en base a modelos, aún ante fallos de medida de los sensores de salida debidos a desvíos de la misma (drift).

Además, el método está enfocado hacia plantas de los tipos uno y dos donde exhiben inherentemente falta de detectabilidad. Tal, es el caso de los buques.

Las ventajas más relevantes de tales estrategias son:

— Los métodos de identificación de parámetros basados en técnicas frecuenciales como HBT y CHFRT no dependen de la calidad de la medida cuando ésta está afectada de desvío constante.

— Las pruebas en línea pueden ser aplicadas con el buque operando en condiciones normales y dentro del punto de consigna nominal sin perturbar sus tareas de operación.

— Estas estrategias pueden ser utilizadas con estructuras de modelos parcialmente conocidas

Como resultado de las comparaciones de calidad de ambos métodos, se tiene que CLFRT es más preciso que HBT y por consiguiente, las tareas de detección y aislamiento resultan mas eficaces con el método CLFRT

Las desventajas mas relevantes de tales técnicas de identificación se enumeran a continuación como:

— El tiempo necesario para estimar los parámetros del modelo aumenta con la precisión requerida y el número de parámetros a determinar

— El esfuerzo computacional aumenta con la precisión requerida y el número de parámetros a ser estimados

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ASPECTS OF REMNANT LIFE ASSESSMENT IN OLD STEAM TURBINES

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ABSTRACT

In order to get a high operation availability and to stablish a maintenance and spares management strategy capable of develop the life extension of steam turbines, is essential suitable assessment of aging damage and remnant life for most relevant parts.

According with machine integity inspection results, life consumption and estimated remnant life results, future maintenance plans, equipment and even big components renewal, will be carried out in basis of efficience assessments data to get optimal safety, availability, reliability and efficiency operating conditions.

Therefore, in order to get these objectives, is essential to know the main aging deterioration mechanisms and realize how are affecting to different turbine components.

Only a few of all failure mechanisms that can happen in steam turbines affect directly to aging deterioration of components, and are produced by the effect of high temperatures kept for long time periods, as well as the sudden variations of these temperatures.

Keywords: life extensión, creep, termal fatigue, consumed life, remnant life.

INTRODUCTION

Most of steam turbines installed on electric power plants last years, were designed and manufactured to operate during 20 or 25 years, that is equal to 160.000 or 200.000 operating hours.

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But because of costs involving new units erection relate to repairing and refusbishment, most of plant owners prefer to continue existing machines operation. Generally, this life extension offers the chance of carrying out operational and enviromental changes and considering benefits of efficiency and reliabylity improvement using advantages offered by turbine new design components adoption.

All these considerations becomes manufacturers and owners to research and develop assessment technology and plant renewal, that contribute adequately to turbine extension life when applied.

Assessment consumed life and maintenance and renewal plans using these technology, are consolidated either as a basis to establish extensión life programs and as economically reliable maintenance methods.

Although plants were operated under good conditions and high efficiency without critical problems, long term operation periods can produce latent damage on turbine inner components. These damage can suddenly become and cause important mechanical problems.

Meanwhile are ligth and affecting only simple components, problems can be easily solved by means working maintenance habitual procedures. But if problems are importants or involves critical components, it can produce long time forced outage, causing important and unexpected economical waste.

OBJECTIVE

Damage produced by mechanisms of deterioration affecting turbine integrity are directly related with time, and are produced after long year turbine operation.

These mechanism produce a progressive damage on materials that turbine is manufactured with, reducing its mechanical properties. It also produce deviation and distortion on inner turbine parts, becoming to efficiency deterioration.

There are two of these mechanisms that demonstrates to be more significant to assess the life consumption of affected component, not only for its relation with time factor, but for the responsability of turbine components that they affect, such as rotors and casings, that are large and expensive and may produce disastrous damage.

The essential objective of this article is to show, according with damages produced by these two mechanisms, the aspects involving when assessing and analyze turbine remnant life in order to get life extension and recovery or improvement of efficiency.

METHODOLOGY

The methodology used has been based on experience obtained during several turbine overhauls in a power plant, wuth participation of own power plant staff, turbines manufacturer and technologists specialized on inspections and non destructive tests, using the top technology in inspection equipment.

Scopes on this kind of overhaul are often standardized for each type of turbine, although sometimes other machine requirement are considered. Scopes to be realized during overhaul period are also standardized, and for these type of failure mechanism are often the following:

— Complete inspection by means magnetic particles of rotors, specially in section change areas and sharpen radius.
— Ultrasonic testing in rotor center bore.
— Ultrasonic testing in rotor surface critical areas.
— Metallurgical inspection in critical areas on rotors, rows and casings.
— Magnetic particle inspection in steam chests.
— Magnetic particle inspection in steam inlet pipe sleeve and stationary row blades.

Obtained data from these inspections are compared with those obtained in previous inspections, thus it is possible to assess the evolution of existing or previously repaired defects.

Manufacturer experience and machine knowledge made neccesary its overhaul mediation in order to deal with technical improvement sheets and detected defects assessment, but it is also important the mediation of inspection specialized people, that can discover defects that are not easily detected due to its features.

TURBINE LIFE MANAGEMENT

Turbine life management consists of a continuous assessment of plant condition, monitoring operation conditions and improvement of operation and maintenance procedures. Also, a plan based on assessment of component condition and the need of restoration or replacement of critical components is required.

The process is concentrating on components whose failures have greater importance on safety, availability or maintenance

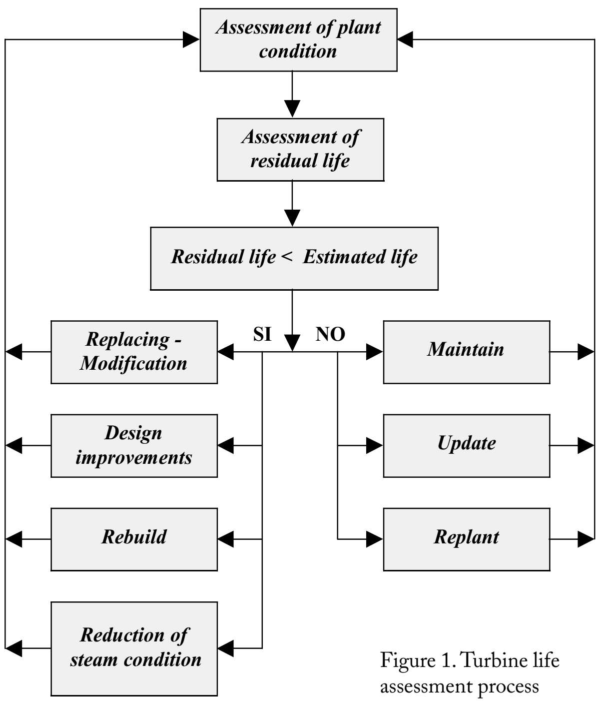

Figure 1. Turbine life assessment process

costs and on mechanisms affecting those components. The process is showed in schematic way on figure 1.

Creep

The phenomenon of creep consists of dislocations on grain boundaries of steel caused by high temperature working under the action of high centrifugal forces, so that a permanent deformation is produced on the afected component, as is showed on figure 2.

It can be deduced that more sensitive turbine parts to suffer from creep are those involved in rotatory movement, as rotors, pressure first HP and IP blade rows, center bore, serrations and blade roots, transition ratio between different rotor stages, balance holes, etc...

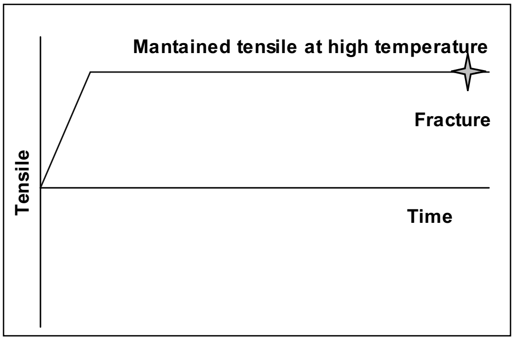

Figure 2 .- Creep mechanism

Low Cycle Fatigue

Low cycle fatigue is produced because of changes ocurring in stresses and temperatures during start, shutdown and load changes when unit is working according with load demand (see figure 3).

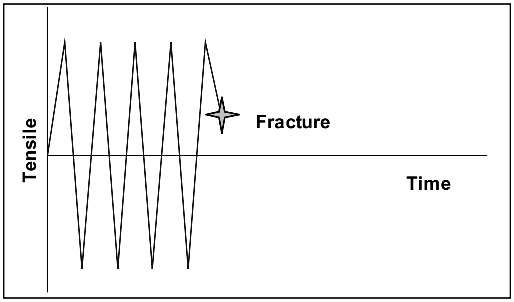

Figure 3. Fatigue mechanism

Turbine components with thick section construction, such as cylinders or high pressure casings, working under high pressure and temperature action, are affected by creep-low cycle fatigue, produced by combination of thermal fatigue caused by load changes and creep caused during high temperature and pressure operation. Schematic representation of mechanisms is showed on figure 4.

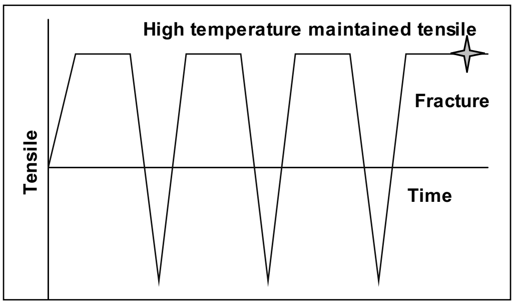

Figure 4. Creep-fatigue mechanism

REMANENT LIFE ASSESSMENT

Steam turbine components work under severe conditions, such as high temperature and pressure.

Will be then needed to know the amount of life consumed for the component until a failure was produced, in order to assess and calculate the remaining time until deffect produces failure. So that, a high machine availability level will be reached, because is possible to know, with certain grade of accuracy, the moment in which the component failure is going tobe produced.

The análisis is carried out for each failure mechanism, based in progress data records of produced deffects.

Creep Life Assessment

Creep life is assessed in basis of operation temperature and pressure stresses calculations at the critical areas using creep rupture strength tables for each material.

Creep under steady operation conditions is calculated taking into account both the centrifugal stresses set up by the rotation of the rotor an the thermal stresses set up by the temperature gradient in the rotor body. Additional loads imposed by the blades are also taken into account.

The method involves complex calculations, and an example ofresults can be saw in table 1. Calculations were made using following data, as well as a polynomial expression for temperature variation with respect to radius [1]

Bore diameter = 101 mm
Rim mean diameter = 919.5 mm
Blade centrifugal force at 3000 r.p.m. = 47.0 kN/Blade
Young modulus, E = 172900 MPa
Coef. Of thermal expansion = 0.1625×10^{-4} / °C
Cylinder outside diameter = 711 mm
Disc otside diameter = 876 mm
Number of blades = 110
Density = 7833 kg/m^3

Calculations are based on Von Mises criterion or Von Mises-Hencky theory, that stablish that failure on ductile materials occurs when the energy of distortion by volume unit reaches the same level or exceeds the distortion energy level of the same material when traction test yield strength is reached. This theory considers the energy associated at changes of material shape and is appropiate for ductile materials, because it shows very well the triaxial stress status at the component.

Von Mises theory is expressed as:

$$\sigma_v = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}} \qquad [6]$$

where $\sigma_1, \sigma_2, \sigma_3$, are the principal stresses acting on component.

Centrifugal force stress in cylinder				
Radius (mm)	Stress(MPa) Axial	Tangential	Radial	Radial displacement (mm)
50.5 355.5	10.25 –10.25	128.5 38.21	0.00 21.68	0.0366 0.0715
Centrifugal force stress in disc				
Radius (mm)	Stress(MPa) Axial	Tangential	Radial	Radial displacement (mm)
355.5 438		48.55 42.79	45.89 32.37	0.0715 0.0838

Termal stress in cylinder					
Radius (mm)	Stress (MPa) Axial	Tangential	Radial	Radial displacement (mm)	Temperature °C
50.5 355.5	22.32 –22.32	25.4 –20.98	0.00 1.51	0.005 0.031	526.7 537.9
Termal stress in discs					
Radius (mm)	Stress (MPa) Axial	Tangential	Radial	Radial displacement (mm)	Temperature °C
355.5 438.0		–13.71 –15.81	3.19 0.00	0.031 0.050	537.9 540.0

Table 1. Example of rotor creep results calculation

The assessment is subjected to uncertainties resulting from several factors, as:

– Accuracy of the predicted stress and temperature
– Material properties, such as creep toughness.
– Deviation of the operating temperature from design.
– Degradation of material properties due to service exposure.
– Deviation from design either at the manufacturing stage or by subsequent modifications.

Therefore, the creef life of rotors is assessed in stages, each successive stage being more detailed as the remnant life margin is reduced.

The first stage is the initial assessment based on global service data from the same family design rotors. The stresses at critical regions are calculated and the life at each region is assessed on the basis of the generic creep data for the rotor steel.

The assessed life of the rotor is taken to be the minimum obtained for the regions considered, and will be termed as "Assessed rotor creep life"(ACRL). (See table 2)

The rotor is allowed to run to 50% of the ACRL without further actino. If the perceived operating life of the rotor is grater than 50% ACRL, it is necessary to proceed to later stages.

Second assessment stage considers individual rotors, that requires the determination of creep average operating temperature for the rotor and examination of the records of the inspections carried out during manufacture of the rotor to establish the size of defects. Besides, the records of steel composition and heat treatment during manufacture are used to estimate the creep toughness and ductility of the rotor (see table 2).

Creep effect	1º stage blade fixing			
	Stress (MPa)	Temperature (ºC)	Life to rupture (h)	50 % ARCL (horas)
Rotor type 110	94	542	2.6×10^5	130000
Rotor type 94	89	542	1.8×10^5	90000

Tabla 2. Example of assessed rotor creep life calculation (ARCL)

The rotor life is then assessed using stresses derived from design calculations as in stage 1, but using the creep average operating temperature derived from the operating records and the creep data obtained from the manufacturing records.

Creep average temperature is increased for operation at off-design condition. This is particulary important on the high pressure turbine, where operating at different load to design rated load can change the steam entalphy and, as a result, temperature after governing valves. Temperature has an important effect on creep, because, as an example, a 10 ºC rise in the nominal steam temperature of 540 ºC will increase creep strain and creep rupture by a factor of about 2.

Third stage requires removal of the rotor from the turbine in order to take material samples and to carry out non-destructive testing (NDT). The procedure is similar to that of stage 2, but the creep strength and ductility are determined directrly by acelerated tests at higher than operating temperatures on samples of steel removed from the cooler part of the rotor, which has not been subjected to high temperature.

Magnetic particle inspection (MPI) is applied at the bore surface and at regions on the outer surface where creep or termal fatigue damage may be expected., and ultrasonic examination is carried out where this is posible. Replication of the surface in critical areas is used to check for creep cavitations or micro cracking which may be due to creep.

Once creep remnant lif is assessed for these stage, tue rotor is allowed to operate to 95% of ARCL.

Low Cycle Fatigue Life Assessment (Thermal Fatigue)

Thermal fatigue phenomenom is common on high temperature turbines, in this case, low cycle fatigue, that occurs mainly due to severe temperature reached during start up and initial stages of load generation. It is also afected by a combination of factors, sucha as deviations from the stipulated start-up procedures, poor designs, poor machining and poor material properties.

Excessive temperature increase during start up induces large stresses that could excede the yield limit of the material.

A typical stress-strain cycle during one cycle of start-up and shutdown is showed at figure 5 [2].

Considre a component subjected to an alternating tensile/comprensive load cycle in which the total stress variation is "$\Delta\sigma$". This load introduces within the component stresses ranging from the maximum comprensive effect "$-\Delta\sigma$" to the maximum tensile effect "$+\Delta\sigma$", that produce the subsequents strains "$-\Delta\varepsilon$" and "$+\Delta\varepsilon$". Normally, these loads are of equal magnitude in the tensile and comprensive direction.

The center point "O" represents the first put in service of the component, where it has not still experienced cycling load. When an initial tensile load is applied, a tensile stress is induced that produces a linear elastic strain "$\Delta\varepsilon_{ie}$" up to condition A.

After condition A, the extensión with increasing load continúes but no longer follows a linear relationship, being a plastic strain "$\Delta\varepsilon_{ip}$" and, therefore irrecoverable. Therefore, the total strain is:

$$+\Delta\varepsilon = \Delta\varepsilon_{ie} + \Delta\varepsilon_{ip}$$

At condition B, the load has reached its maximum tensile value. Now, as the load is reversed, the tensile stress reduces to zero and strain

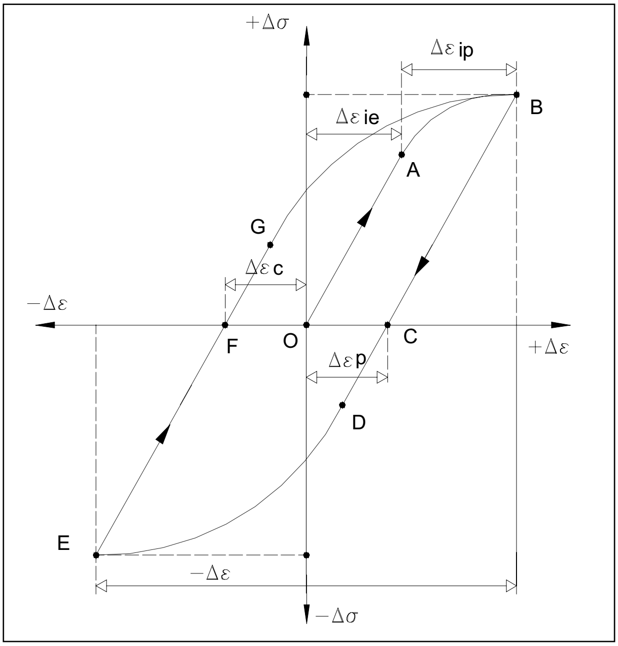

Figure 5. Typical stress-strain cycle for a turbine component material

reduces, but does not reach zero (C condition). A plastic strain "$\Delta\varepsilon_p$" remains at the component. At this condition, the direction of the load is reversed, the load becomes comprensive, initially following a linear (elastic) relationship to condition D, where the strain becomes plastic and follows a curved relationship from D to E, untill comprensive stress reach its maximum value at E, "$-\Delta\sigma$" . As a result of this, a total elastic and plastic strain "$-\Delta\varepsilon$" is produced.

Next, the compresive load is reduced from a maximum value, and so the compresive stress is reduced reaching zero at condition F, where there is still a residual strain remaining "$-\Delta\varepsilon_c$". The tensile load is again applied with the initial strain being elastic to G and the final strain being plastic to B.

This series of stress and strain are repeated causing the material of the component undergoes a cyclic loading and the accumulation of plastic strain continues to deform the component.

If comprensive and tensile stresses are of the same magnitude, the curve will be symmetrical about the point of origin O.

The total strain range of the component is the sum of "$+\Delta\varepsilon$" and "$-\Delta\varepsilon$", shown at the diagram as the strain range from B to E, that consists of two elements, the elastic "$\Delta\varepsilon_{ie}$" and the plastic "$\Delta\varepsilon_{ip}$" portions.

The elastic portion is represented by the ranges OA, BD and EG, but since the range OA is present only during the initial strain cycle, it can be neglected. The plastic strain ranges are from AB, DE and GB, and again, the range AB can be neglected.

Then, there is a relationship between the total and elastic strains for any material, that is taken by the expression:

$$+\Delta\varepsilon = B \cdot \Delta\varepsilon \cdot \Delta\varepsilon_p \cdot g$$

where

$\Delta\varepsilon$ is the total strain range

$\Delta\varepsilon_p$ is the plastic strain range

B and g are constants determinated for the material

This relationship is only approximate and applicable primarily to cycles of intermediate strain ranges.

The accumulated damage will lead to crack initiation as the number of start up increase. Thereafter, crack will grow with each additional start.

The remnant life is assessed as the time for the crack to grow to a depth at which the steady state stress could cause rupture in a relatively short time. It should be noted that the procedure for estimating the thermal fatigue carck growth rate includes the effect of the steady state stresses (for example, at the creep state).

Termal fatigue life is not assessed in a easy procedure, as is creep life. A similar first stage assessment can be considered using typical starting and shutdown procedures for a family of turbines. Calculations are then made to predict the stresses set up at the critical regions of cold, warm and hot start conditions due to the rate of rise of steam temperature. The calculated stresses are then used to predict the number of cycles required for crack initiation for each type of start.

An example of start up data and the induced termal stresses at the critical locations of a HP turbine rotor is given in figure 6.

The life expired is the sum of the number for each type of start accumulated to date divided by the number of cycles calculated for cold, warm and hot starts. That is to say if the calculated number of starts are N_f, N_t and N_c, and the number of accumulated starts to date are n_f, n_t y n_c

$$RL = 1 - \left(\frac{n_f}{N_f} + \frac{n_t}{N_t} + \frac{n_c}{N_c} \right)$$

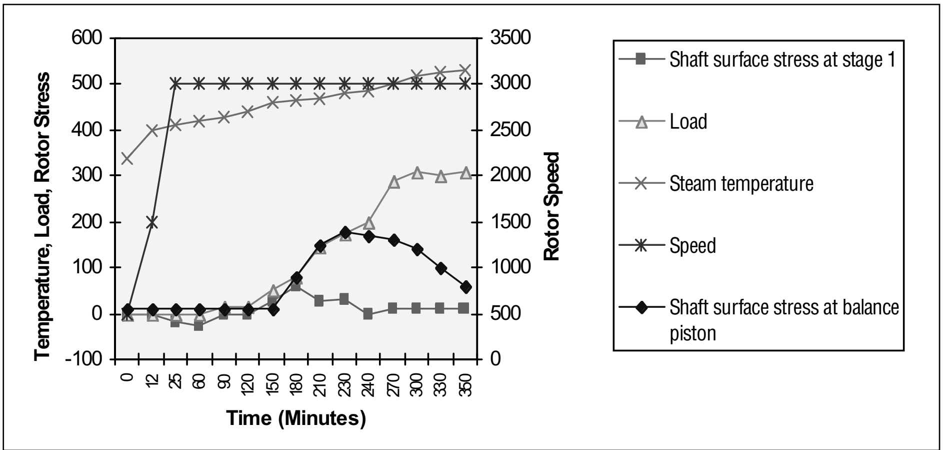

Figure 6. Termal stresses set up on rotor during start-up

Generally, cold starts are defined when the metal temperature falls below 200 °C. A start following a weekend shutdown can be classified as warm, and after an a trip is considered as hot.

Experience shows that normally cold starts induce about three times more damage than warm or hot starts. Therefore, an equivalent number of starts may be used to calculate the allowable safe number of starts before an inspection becomes necesary.

$$Equivalent\ starts = cold\ starts + \frac{1}{3}\left(warm\ starts + cold\ starts\right) \qquad [3]$$

High pressure and intermediate pressure rotors, valve chests or steam chests and cilindres must be inspected before they reach the number of equivalent starts.

Normally, the magnitude of the start up termal stress decreases with depth and crack growth may stop. However, propagation to a critical size where it could grow due to thermal cyclic stresses needs to be considered. Therefore, is advisable to adopt a policy against crack initiation when manufacturing, and the procedure of determination of crack growth rate by means monitoring it, at future inspections.

The calculated termal fatigue life indicates the risk of damage at rotor and regions of the rotor where damage is most likely to occur. An example of this is given in table 3 [4].

<table>
<tr><th rowspan="2">Component</th><th rowspan="2">Location</th><th rowspan="2">Stress (MPa)</th><th colspan="3">Nº of starts to cracking</th><th rowspan="2">Inspect at</th></tr>
<tr><th>Cold</th><th>Warm</th><th>Hot</th></tr>
<tr><td rowspan="2">HP Rotor</td><td>Balance piston</td><td>2.5</td><td colspan="3">Little risk of cracking below 10000 starts</td><td>Major overhau</td></tr>
<tr><td>Reaction stage</td><td>5</td><td colspan="3">Little risk of cracking below 10000 starts</td><td>Major overhaull</td></tr>
<tr><td rowspan="2">HP Cylinder</td><td>Inlet piping</td><td>1.5</td><td colspan="3">Little risk of cracking below 7000 starts</td><td>Major overhaul</td></tr>
<tr><td>Diaphragm groove</td><td>3</td><td>1700</td><td></td><td></td><td>Inspect before 2000 equiv. starts</td></tr>
<tr><td>IP Rotor</td><td>First Reaction stage</td><td>4</td><td>900</td><td>900</td><td>3300</td><td>Inspect before 1000 equiv. starts</td></tr>
<tr><td rowspan="2">HP Cylinder</td><td>Inlet piping</td><td>1.5</td><td colspan="3">Little risk of cracking below 10000 starts</td><td>Major overhaul</td></tr>
<tr><td>First stage stationary blade fixing</td><td>3</td><td>3000</td><td></td><td>2500</td><td>Major overhaul</td></tr>
</table>

Table 3. Estimated termal fatigue life and recommended inspections [5]

Figures 7 and 8 show the distribution of temperatures and thermal stresses at first HP blade grooves during a hot start.

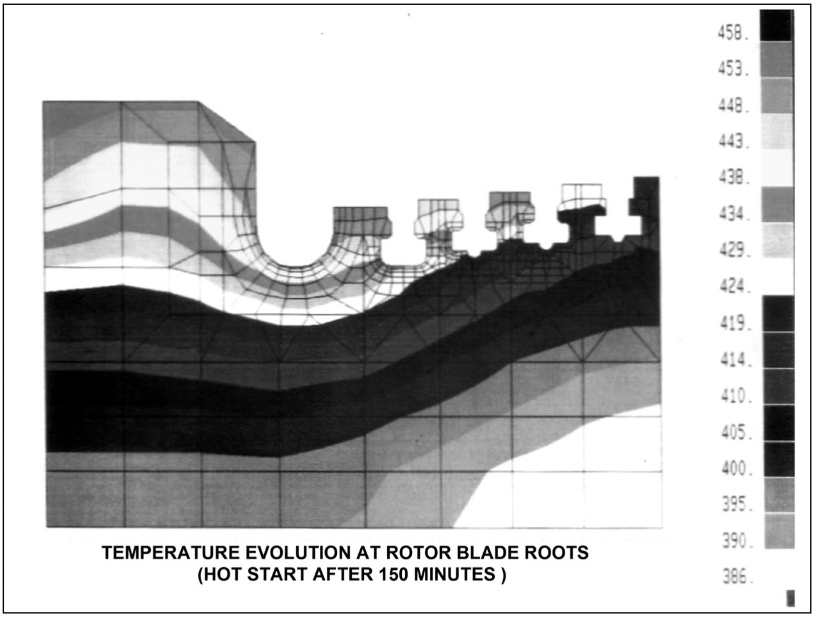

Figure 7. Rotor temperature distribution during a hot start

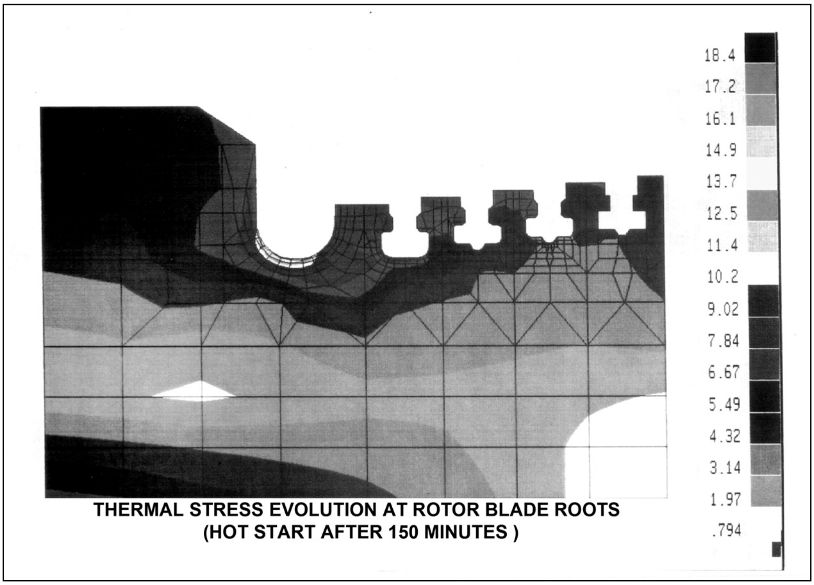

Figure 8. Rotor thermal stress distribution during a hot star

CONCLUSIONS

In order to get a suitable remnant life assessment of steam turbines and to guarantee its reliable operation, is essential to establish an inspection program affecting at least the mos critical parts of the machine, that are those sensitive to produce stress concentration, such as seal fin grooves, blade root serrations, balance holes regions, changing section regions, etc...

The specification and scope of neccesary inspections must be agree on manufacturer and inspections specialized people, and, as a minimum must include:

— Assessment of initial condition of component and operation and maintenance data records n order to identify critical components.
— Magnetic particle inspection of the complete rotor with particular attention to changes of section.
— Ultrasonic examination from the bore rotor. This technique, known also as Boresonic, achieves to inspect bore surface and blade fixing areas in order to look for cracking.
— Ultrasonic inspection from critical rotor surface regions, in order to complement magnetic particle or liquid penetrant tests and to get a suitable measurement from deffects found by means these techniques.
— Metallurgical examination by means replications from critical regions, in order to study the material metallurgical structure and to know its age deterioration grade.
— Ultrasonic inspection and magnetic particle inspection from steam chests. If possible, is advidable to measure the depth and length of known deffects and monitor it. Deffects found at earlier inspections, must be also measured to determine if its progress is as expected. Evaluation of chest integrity at these inspections will provide a guide to the frequency of further inspections and planning for eventual repair or replacement.
— Magnetic particle examination of the cylinder at steam inlet belt region and stationary blade rings.
— Set up a database of turbine operating statistics including hours of operation , number of starts, records of previous inspections.
— Monitoring, when posible, the termal stress and remnant termal fatigue life of high pressure and intermediate pressure, that are the ones affected by worst working conditions.

Knowing phenomenom affecting turbine integrity, evaluation of time progress of deffects caused by them, data obtained from operating records from own or another similar machines, review from critical components inspection results, are an essential tool to asses remnant life of turbine.

Assessment of this remnant life, will allow the establishment of economically capable preventive maintenance plans or improbé or renewal plans in order to guarantee availability and reliability of machines, and economically capable life extensión programs to allow to extend steam turbines operation.

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ASPECTOS DE LA EVOLUCIÓN DE LA VIDA RESIDUAL EN TURBINAS DE VAPOR VIEJAS

RESUMEN

Con en fin de conseguir una alta disponibilidad de operación e implantar un plan de mantenimiento y de gestión de repuestos económicamente viable para llevar a cabo el alargamiento de vida de las turbinas de vapor, es imprescindible una adecuada valoración del deterioro por envejecimiento de las mismas, así como de la vida residual de los componentes más relevantes.

En base a los resultados de inspecciones de integridad de la máquina, consumo de vida actual y vida remanente estimada se estructuran, de acuerdo con los resultados de la evaluación del rendimiento del ciclo, los planes de mantenimiento futuros, la renovación de equipos e incluso de grandes partes de la instalación para que la planta siga operando en unas condiciones de seguridad, fiabilidad y rendimiento óptimas.

Es fundamental, por tanto, para conseguir estos objetivos, conocer los principales mecanismos de deterioro por envejecimiento y saber de qué forma afectan a los diferentes componentes de la turbina.

De todos los mecanismos de fallo que pueden producirse en las turbinas de vapor, solamente algunos de ellos intervienen directamente en el envejecimiento de los componentes de las mismas, y son los ocasionados por el efecto de las altas temperaturas mantenidas durantes largos períodos de tiempo, así como las variaciones bruscas de estas temperaturas.

METODOLOGÍA

La metodología utilizada ha estado basada en las experiencias obtenidas durante sucesivas revisiones de turbina en una central termoeléctrica en las que han intervenido, además del personal de la empresa propietaria, los fabricantes de las turbinas y otros tecnólogos especialistas en inspecciones y ensayos no destructivos utilizando las últimas tecnologías en equipos y técnicas de inspección.

Los alcances de este tipo de revisiones suelen estar estandarizados para el tipo de turbina de que se trate, aunque a veces se tienen en cuenta otras necesidades de la máquina. También están estandarizados los alcances de las inspecciones a realizar durante la revisión, que suelen ser los siguientes para este tipo de mecanismos de fallo, termofluencia y fatiga de bajo ciclo:

— Inspección completa mediante partículas magnéticas de los rotores, prestando especial atención a los cambios de sección y radios de acuerdo.
— Inspección mediante ultrasonidos de los orificios centrales de los rotores.
— Inspección mediante ultrasonidos de las zonas críticas de las superficies de los rotores.

— Inspección metalográfica de las zonas más críticas en rotores, coronas y carcasas.
— Inspección mediante partículas magnéticas de las cajas de vapor.
— Inspección mediante partículas magnéticas de los manguitos de entrada de vapor y álabes de coronas fijas.

Los datos resultados obtenidos mediante estas inspecciones se comparan con los obtenidos en inspecciones anteriores, y de esta forma se puede evaluar la evolución de defectos existentes o que habían sido reparados.

La experiencia y conocimiento de la máquina por parte del fabricante de la misma, hace muy necesaria su intervención en la revisión, de cara a la expedición de recomendaciones y valoración de defectos encontrados, pero también es muy importante la intervención de especialistas en inspecciones que puedan sacar a la luz defectos que por su naturaleza no siempre son fáciles de descubrir.

CONCLUSIONES

Para una adecuada valoración de la vida remanente de una turbina de vapor y para asegurar la operación fiable de la misma, es fundamental la implantación de un programa de inspecciones que afecte al menos a las partes más críticas de la máquina, que son aquéllas susceptibles de producir concentraciones de esfuerzos, como los encastres de las pletinas de cierre, encastres o serratiles de álabes, las inmediaciones de los orificios de equilibrado, las zonas de transición bruscas debidas a cambios de diámetro, etc...

La especificación y alcance de las inspecciones necesarias han de ser consensuadas con el fabricante de la máquina y con los especialistas en materia de inspección, y como mínimo han de incluir las siguientes:

— Valoración inicial del estado de los componentes y estudio de los históricos de operación y mantenimiento para identificar los componentes críticos
— Inspección completa mediante partículas magnéticas del rotor, prestando especial atención a las zonas donde se producen cambios de sección.
— Inspección mediante ultrasonidos del orificio central del rotor. Con esta técnica, conocida también como Boresonic, se consigue inspeccionar, además de la superficie del orificio, las zonas de los encastres de los álabes en busca de posibles fisuras.
— Inspección mediante ultrasonidos en zonas críticas de la superficie del rotor, con objeto de complementar las inspecciones mediante partículas magnéticas o líquidos penetrantes y, en su caso, dimensionar los defectos encontrados con esas técnicas.
— Inspección metalográfica mediante réplicas en las zonas críticas, con objeto de estudiar la estructura metalográfica del material y determinar su grado de envejecimiento.

— Inspección mediante ultrasonidos y partículas magnéticas de las cajas de vapor. Si es posible, conviene dimensionar en profundidad o longitud los defectos encontrados y realizar un seguimiento de los mismos. En cuanto a los defectos ya conocidos de inspecciones anteriores, han de ser también dimensionados para determinar si su comportamiento es el esperado. La evaluación del estado de las cajas de vapor mediante estas inspecciones proporcionará una guía de la frecuencia con que se han de realizar las inspecciones y para planificar posibles reparaciones o sustituciones.
— Inspección mediante partículas magnéticas de la carcasa o cilindro interior de la turbina en la zona de los manguitos de entrada de vapor y en las coronas de álabes fijos.
— Establecimiento de una base de datos de las estadísticas de operación de la turbina, en la que se reflejen, además de las horas de operación, número de arranques, resultados de anteriores inspecciones, etc...
— Monitorización en lo posible de las tensiones térmicas y la vida remanente por fatiga térmica de los componentes de la turbina de alta presión y presión intermedia, que son las sometidas a las peores condiciones.

El conocimiento de los fenómenos que afectan a la integridad de la turbina, así como la evaluación del progreso con el tiempo de los defectos ocasionados por ellos, los datos obtenidos de la acumulación de históricos de funcionamiento de la máquina o de otras similares, la interpretación de los resultados de inspecciones de los componentes críticos, son una herramienta fundamental a la hora de valorar la vida remanente de la máquina.

La evaluación de esta vida remanente permitirá la implantación, de manera económicamente viable, de planes de mantenimiento preventivo o planes de mejora o renovación con objeto de asegurar la disponibilidad y la fiabilidad de las máquinas, y programas de extensión de vida económicamente viables que permitan prolongar la explotación de las turbinas de vapor.

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HIGH SPEED CRAFT VIABILITY ANALYSIS

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ABSTRACT

This paper presents a brief analysis on the applicability of high speed crafts in short sea trades, from different marine stakeholders point of view. The TRANSMAR research group of the Nautical Engineering and Sciences department from the Technical University of Catalonia, has continued the finished study (INECEU: Intermodality between Spain and Europe), proposing fast ships for serving some of the selected sea links in West Europe in it. The recent communication regarding the mid-term review of the European Commission's 2001 White paper on Transport, confirms that the evolution of modal split in freight transport from the year 2000 up to the year 2010, will show a steady share of 39% for the waterborne transport and a slight increase in the road transport. One of the four pillars of the European transport policy is the innovation in transport technologies and systems, improving its efficiency and logistics throughout the supply chain.

Key words: Short sea shipping, high speed crafts, logistical performances

INTRODUCTION

The European Commission and Member States drafted in the year 2001 a transport policy in order to get some goals beyond 2010. The reason of these programmes was the need to balance the share of the different transport modes, as the forecasted growth would be absorbed mostly by road transport. Still around 45% [1] of European Union foreign trade is carried by road and is consequently conditioned by traffic congestion or high fuel consumption, implying disadvantages related to

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pollution and safety. Meanwhile up to 39% of the before mentioned volume is carried by short sea shipping. Short Sea Shipping in European waters has been considered by national and European governments as one of the most feasible ways to alleviate the congestion that gets worse every day on the roads and highways across Europe. The European Union has presented the mid-term review of the European Commission's 2001 White Paper on Transport, confirming the waterborne transport share, but increasing the road participation. The Marco Polo II action, with an overall budget of 740 millions € over 7 years (from 2007 to 2013), should include funding for motorways of the sea and also for modal shift programmes together with the traditional Marco Polo programme actions.

The shipping scenario.

The short sea shipping (SSS) fleet can be classified per cargo type, mainly container, Ro/Ro and passenger transport, but keeping apart passenger traffics from the perspective of the Community objective, container and Ro/Ro transport are the segments in SSS traffics in which modal-shift could be carried out. Keeping in mind that Ro/Ro (Roll on/Roll off) activities concern the loading and discharging of a road vehicle, a wagon or an inter-modal transport unit on or off a ship on its own wheels or wheels attached to it for that purpose (United Nations (Economic Commission for Europe), European Conference of Ministers of Transport (ECMT) and the European commission (EC), 2001).

Figure 1. Image of a high speed craft serving the domestic traffics in Canary Islands

Source: http://www.fredolsen.es/lineas/Benchijigua_especificacions.asp.

High speed maritime transport could be seen as a possible solution to current challenges in seaborne transport. In a first view it could be understood that small boats operating at high speeds, would be less efficient than bigger ships sailing at conventional speeds. This is quite logical due to different sea keeping performances, the fuel prices and additionally we face the higher building costs in the fast side sometimes around two and a half to ten times de price of a conventional Lo/Lo ship per freight slot [2].

However from a global point of view and considering all the logistical chain aspects, the fast maritime transport concept reduces the transit time, increasing the overall transport process speed. Then high speed crafts would be an efficient logistics solution in specific SSS services, reducing the associated freight costs, because speed minimises the workshop costs, affecting the logistical chain global costs (Bendall et al, 2001). Mainly when talking about “just in time” goods, where speed is seen as a quality service aspect. This superior speed is reached through not only adding speed at sea or increasing frequencies, but through quicker and more efficient operations on port and in general on shore [3,4].

EUROPEAN TRANSPORT POLICY

Today transport policy stimulates and recognises the need of an efficient multimodal transport chain as an alternative to road transport. The European transport corridors are intended as an overall transport network, where the maritime legs will be the future accesses to the motorways of the sea and the ports will become quality links between the sea transport and their terrestrial continuation. These thinking were firstly anticipated in the European White Paper on Transport Policy, also alerting of an imbalanced share of the modes of transport within the intra European exchanges. In order to compensate them, the Short Sea Shipping was considered a reasonable choice for reaching the objectives of sustainability and possibly absorption of the estimated 50% of increment in heavy traffic, contributing also to the modal balance and alleviating the terrestrial bottlenecks [5].

The European Commission has been promoting research on maritime transport mostly during the fourth and fifth framework programmes and in several calls in the sixth programme. Some of the projects related to the fast maritime transport were S@S, FASS, EMMA, SPIN-HSV or TOHPIC, and other ones related to the short sea transport, analysis and promotion as RECORDIT, INSPIRE or REALISE, inter alia. SPIN-HSV identified the policy to achieve a safer and more competitive high-speed maritime transport in the logistics chain and TOHPIC project analysed in deep the way to optimise the fast ships handling at port by means of a software fitted to each port particulars. Fast ships need dedicated port infrastructures in order to increase the efficiency of those ships operability, mainly [6,7]:

1. The identification of the operational features of a HSC in the port interface, affording quick loading and discharging procedures together with cargo tracking and tracing.
2. Applicable ruling in HSC procedures as a set of recommendations that combined with wake wash effects study, affording the Masters to know the real effect of his ship.
3. The study of the port control management systems and the provision of recommendations for improved traffic, safety and efficiency.

From the perspective of the Community objective, aiming for modal shift, transport policy needs to promote SSS more intensively by creating the right conditions to let shippers and logistic service providers use it more frequently in their supply chains. So that, we primarily focus on the SSS container and Ro/Ro market, because these two markets constitute almost the entire SSS market, and they are the segments in SSS transport in which modal shift could be carried out. But the promotion of HSC is second step after the previous one would be accomplished, id est first of all it is needed attracting good flows that do not yet use SSS. Then if a sufficient market demand requires faster service, then operators will invest in HSC. Today HSC are only deployed in specific niche markets (Becker et altri, 2004). If the right socio-economic conditions are settled, the HSC transport services could be offered by themselves, and market demand for fast SSS services would grow in the future. That would be the time when the stakeholders will seek for HSC solutions.

HIGH SPEED CRAFTS ANALYSIS.

Different advantages could justify the HSC serving in short sea trades, keeping in mind the difference between what it can be considered as fast conventional services (24 to 30 knots) and the pure fast ships sailing at speeds 50% to 100% greater than a conventional vessel's speed [8]. A superior speed implies the service reduces the transit time, aspect to be considered for specific goods, so that it is possible to serve more tight schedules and reduce further the cargo waiting time in port. Thus appears the capacity to compete against road transport because for example a 30 knots speed means 56 km/h, a figure much higher than mean freight train speed in Europe or comparable to mean long haul road transport considering the congestions and driving time legal limits. Also at sea there are almost no congestion, leaving room to cargo increasing volumes and no tolls.

However delays in ports are too frequent, sometimes due to a poor management or lack of adequate devices, those delays affect not only the ship schedules but the overall logistical cost, reaching sometimes the 35% of the total cost (Lagoudis et al. 2002). Another disadvantage is the fuel price, because the elevate rate of consumption of those ships due to the high power engine plan needed for developing the required speed. The last is related to the pollutant emissions of high intensity together with the CO_2 and NOx emissions. Also sea keeping performances should be kept in mind, as fast small ships have not so good sailing ability in bad weather than conventional ones, also a larger vessel has a greater navigability at worse sea conditions and therefore is less sensitive to delays at open sea. But the largest vessels are only deployed for intercontinental transport and the maximum average SSS vessel speed is in the region of 20 knots for conventional container transport and 23 knots for conventional Ro/Ro transport (TNO et al, 2004). Moreover bigger fast ships can operate in a wider range of sea conditions but they are only viable when enough demand exists.

The geographic analysis in Europe

Some shipping companies have opted for the fast ships in some short sea services in routes competing directly with the shore transport. This situation varies form one country to another and most of the companies are placed around the Mediterranean sea in France; Greece, Italy and Spain and in the Atlantic basin mainly in Norway and United Kingdom. The Mediterranean and the North Sea show the largest share of Short Sea Shipping, with 30% and 27% respectively. Only in the Ro/Ro market, the Mediterranean countries carry out almost all their Ro/Ro transport through domestic transport. The 7.8 million trailers totally shipped in Europe include 2.3 million trailers in Greece and 1.6 million trailers in Italy (Ship-Pax, 2002) as the countries with largest domestic transport volume.

Splitted by countries, in France operate on a regular basis companies like SNCM and Corsica Ferries, in Greece there are a lot of small companies operating HSC but the main ones are Hellas Flying Dolphins, Superfast Ferries and Minoan Lines. In Italy Grimaldi operates 9 ships but only 4 of them sail at speeds higher than 22 knots, the Ro/Pax Eurostar Roma and Eurostar Barcelona for example cover in 18 hours the trip between Barcelona and Civitavecchia at 27 knots, SNAV, Caremar, Siremar, Tirrenia or Ustica Lines and in Spain Acciona-Trasmediterranea, Balearia, Buquebus, Fred-Olsen and Garajonay Express.

The short sea transport in Greece uses extensively fast ships for linking the high number of islands with the mainland supported by a strong demand, but the strict control of the Merchant Marine Ministry, the seasonal demand and meteorological conditions even during the summer with strong winds and short waves, are the negative aspects in the other hand (Karayanis, 2000).

French and Spanish scenarios, are examples of link between the insular provinces with the main land using fast vessels using also Ro/Pax services. In France the link between Nice and Corsica island, is carried out by means of a fast steel made mono hull belonging to SNCM Ferryterranée company (other previous twin ships Alisó and Ascó are sailing under Greek flag today). In Spain fast ships serving the route between Barcelona and Balearic Islands during summer season, are placed in wintertime in the Gibraltar strait route. Additionally there are specific and efficient services linking different ports within the Canary archipelago all year around. The special conditions surrounding Canary Islands, exposed to all type of weather and swell, have given nowadays to the big multi hull units the exclusivity in the coming future [9].

In northern Europe fast ships operate successfully as regular passenger services in Norway along its coast connecting the most northern cities with the south, mainly by two big companies (Fylkesbatane I ognogFjordane and HSDSjo AS) with around 30 ships.

In United Kingdom there are also an extensive fast short sea freight network served by companies as Stena Line covering 18 routes with 34 ships, but only 4 of them are HSC. This company built in 1996 the first commercial fast ferry in the world, the *Stena Sea Lynx* the precursor of HSS series, being under operational status today this called 900 series carrying passengers, cars and buses and the bigger ones called HSS 1500 with a bigger capacity. Superfast ferries a Greek company that successfully adopted the high speed in routes between Italy and Greece, winning the international recognition when was selected to operate between Rosyth and the European continent. Further the services were expanded between the ports of Hanko (Finland) and Rostock. Isle of Man Steam Packet operates the services between Isle of Man and Ireland, together with Liverpool and He sham, offering high quality services carrying passengers and vehicles between Douglas and four other ports, using fast ships between April to October and conventional ones in winter. The ships are the Seacat and the Superseacat, the first one a catamaran made of aluminium can reach the 35 knots and the second one made in Fincantieri (Italy) is a 100 meters of length mono hull with a fully laden speed of 38 knots. Other companies are *Speedferries* operating with a third generation wave piercing catamaran, and the dissolved companies *Hoverspeed* and *Irish Sea Express*, that finished their activity some years ago.

A good example of successful motorway of the sea can be found in Japan, where around 12 companies hold a wide net of fast freight services, using ships capable of developing more than 30 knots, assuming the share of the 25% of trucks involved in trips longer than 100 kilometres.

In brief and apart from the military services the need for the speed at sea is not perceived among the shippers and receivers and while no need for it would appear, the high speed crafts will not be used in mature short sea traffics.

The vessel's classification

A shipper has the opportunity of selecting different types of vessels for operating a short sea shipping service, depending on the required qualities of the transport in itself. The choice is based mainly on the kind of cargo and the market demand. The maximum mean speed among ships dedicated to short sea is around 20 knots for conventional container carriers and 23 knots for conventional Ro/Ro (Becker et al., 2004).

High speed crafts are used mainly for the passenger transport as the 92% of the 1600 European fast ships are used only as passenger ships and the other 8% combining freight and passengers mainly catamarans and mono hulls. In fact the last ones have spaces for trucks and cars that could be used for freight. In order to assume a cargo increase in the maritime transport, ships could enlarge their cargo capacity, and this is the tendency instead of increase the speed, but this implies more

Table 1: Classification of speed ranges depending on ship types and trip.

Type of transport	Type of vessel	Speed
Conventional transport	Conventional (multipurpose, small tankers inter alia)	12-15 knots
Container transport	Conventional container carrier	12-20 knots
Ro/Ro transport	Conventional Ro-Ro Fast ship (Ro-pax) High speed (Ro-Pax)	15-23 knots 23-30 knots 30-40 knots

Source: own based on Becker et al, 2004.

time at port and worse service to the customer [10], only to be managed by means of faster port handling. The size is a question of economies of scale, reducing the cost per mile and also capable of sail at higher speeds recovering delays and being less dependent of the weather conditions.

When talking about the HSC short sea, its development differs between conventional and fast ships, due to considerations of business, commercial, port services and geographical port lay out. From a business point of view, faster ships permit to reduce the number of units but maintaining the service frequency, but there is a move from cost of capitals to operational or variable costs. So that the economic viability depends not only from investment costs but also of technical developments the rate consumption per carried metric tonne. From a commercial point of view, the natural market of fast short sea traffics is reduced to passengers and high value and time sensitive freight. Conventional ships are more viable in less value and indistinct dependency of transit time freight. Fast ships then can offer benefits in some Ro-Pax services and even Ro-Ro services with short turnaround time in port. Some examples of natural scenarios for fast short sea services are:

1. Ro-Pax transport from mainland to islands as Great Britain, Ireland or Balearic islands.
2. Ro-Ro transport among islands or in fixed freight volumes as European mainland to Great Britain and Ireland.
3. Ro-Pax transport between points where the shore transport is longer or more difficult as Scandinavian countries, Italy to Europe or Spain and Italy to Greece and with the African coast.
4. Ro-Pax transport between points where competitive alternatives of similar distance carry low cost containerised freight.
5. Ro-Pax transport between islands in archipelagos where the capacity of passenger and freight competes against the plane as in Canary Islands.

From a port service perspective, fast ships require quicker port operations and efficient hinterland connections. It is concluded that a reduction in port time has a double effect on the total transport time compared with the same reduction in sailing time (Laine & Vepsalainen 1994) so that the fast vessel operative would be more benefited using Ro/Ro (Ro/Pax) ships than traditional Lo/Lo ships, because the need for timeless port operations.

From a geographical port layout point of view, the fast ship port facilities never should be placed in the middle of the port, but in the outer part of the port as they do not need for marshalling and consolidation areas, needed for a quick link to the hinterland. This location benefits also passengers as they has not cross over industrial areas within the port.

THE CHOICE BETWEEN CAPACITY AND SPEED

For carrying out an in deep analysis of the high speed maritime transport, previously we should describe the different alternatives of the existing transport modes, so that a brief description backs on the further comparative. Each existing ship design is made depending on the initial requirements and focused on certain economic needs. Today there is a wide range of fast mono and multi hull designs demonstrating a lack of standardisation that could be required by the operators. However the advantage is the opportunity for compare the alternative designs by the operator side, and then look for the best price in a competitive market.

Among the ship builders there are the ones defending the mono hull design because they consider them safer in critical situations and in the opposite side the builders defending the catamaran design as they need 30% less power for the same deadweight and their capacity of maintaining a high speed in calm seas. Building materials are also matter of discussion as for example Incat and Austal use aluminium, Fincantieri (builder of mono hulls) uses steel and other builders a combination of both. This last affects the needed power as it depends of weight and required speed, being the most common the diesel engines and gas turbines, existing some combination of them (CODAG). Shipbuilders are waiting for demand of vessels with speeds in excess of 33 knots seeing in the future speeds of more than 50 knots in the near future (Baird, 2004).

The kept in mind variable to decide the final design are the freight volume, the cost of the carried good (the purchase cost and the possible delay costs), the trip distance, the frequency, the transit time and the type of products as for example manufactured goods have an added value compared with commodities.

Based on secondary studies and the above mentioned characteristics, high-speed vessels compared with conventional, container and Ro/Ro vessels can offer the same degree of flexibility as container and Ro/Ro ships, they are suitable for high-cost products, they can operate in a very wide range of distances, their level of service is very high, the transit times achieved are short and the type of products they can transport are both commodities and manufactured goods. A comparative between fast vessels has been made between the container ships and the Ro/Ro and Ro/Pax ships developing speeds in excess of 25 knots. In the first case, their size and speed has been increased up to mean values of 21 knots and maximum values of 27.5 knots, so it can not be confirmed the existence of high speed container ships. However projects of such kind of vessels as the Norasia or Fast-ship examples were cancelled before being carried out due to: doubts on her sea-keeping performances, weak perspectives on a solid demand and the need for investors assuming such a risk.

Despite the above, today the conclusion is that the improvement of performance should be sought from increasing the round trip frequency of ships, (Blauwens et al, 2003 and Becker J.F. et al, 2004) regarding the transport service between seaport and an inland terminal. A higher frequency can be achieved not only by ship's propulsion together with hull's design but cargo-handling system. The investment on cargo-handling may provide higher return than investment in ship propulsion and this is the reason because the SSS operators expand their fleet on conventional ships rather than fast ships. As the largest ships are deployed on intercontinental routes because the economies of scale of a great ship decrease on the cost per mile basis and the need for additional speed to overtake delays in her departure that is an added value for the intermodal operator, the maximum SSS vessel speed is in the range of 20 knots for conventional container vessels and 23 knots for conventional Ro/Ro transport. Regarding Ro/Pax ships, there is a wide variety of them dedicated to short sea trades with speeds superior to 25 knots and designs mainly of monohulls but also catamarans and SES. Most of the fast vessels have tonnages bigger than 500 GT and some of them exceed the 2000 GT, so this means that fast ships are not intended to carry a lot of cargo and are more addressed to carry passenger and cars together with small volumes of cargo. At this point we have classified three categories of ships depending on their maximum speed.

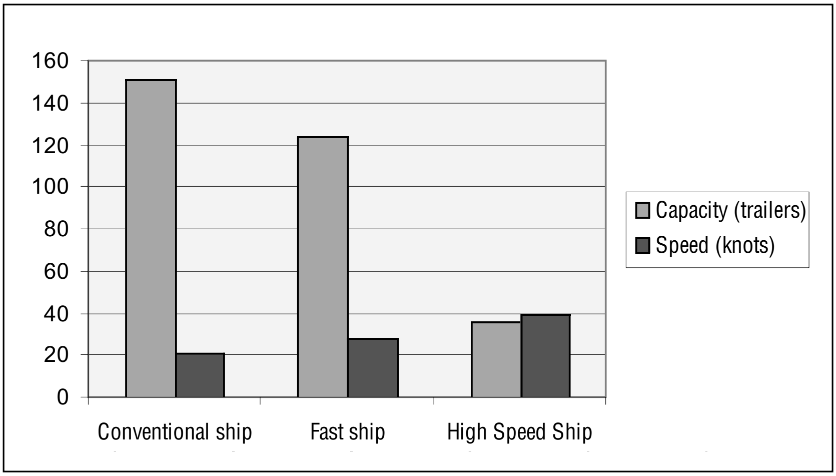
COMPARATIVE ANALYSIS.

From the precedent points and based also in previous studies on the high speed vessels particulars (Marchant, 2002), we are going to consider only the Ro/Ro only freight ships in this analysis, because the loading and discharge operations are quicker than in Lo/Lo ships and then cheaper, they need only a ramp but no cranes, and the port space needs are minor, together with some other technical particulars as less draught in general. However their freight capacity compared with same size Lo/Lo ships could be half, because goods are placed on wheels and the stiffeners in general are wider and need space for lifters, ramps, accesses together with dead spaces between trailers. We can say that in general Ro/Ro ships are double expensive per TEU slot than Lo/Lo ships, but in short distance traffics with the minor port costs make them a viable and efficient alternative.

Additionally it is thought that fast ships need to operate in short routes in order to ensure the frequency of one trip per day. In the other side fast conventional ships (superior deadweight, steel hull and speeds up to 30 knots) are used in longer routes with a limit of 500 miles that is a distance to be covered within 24 hours, a daily frequency can be ensured with two ships, that is the strategy followed by companies like *Superfast ferries*, *Sinihonkai ferries* or *Blue Highway*.

— Considering the classification of ships depending on their cruise speed:
— Conventional ship (speed less than 23 knots).
— Fast ship (Speed between 23 and 30 knots)
— High speed ship (speed in excess of 40 knots)

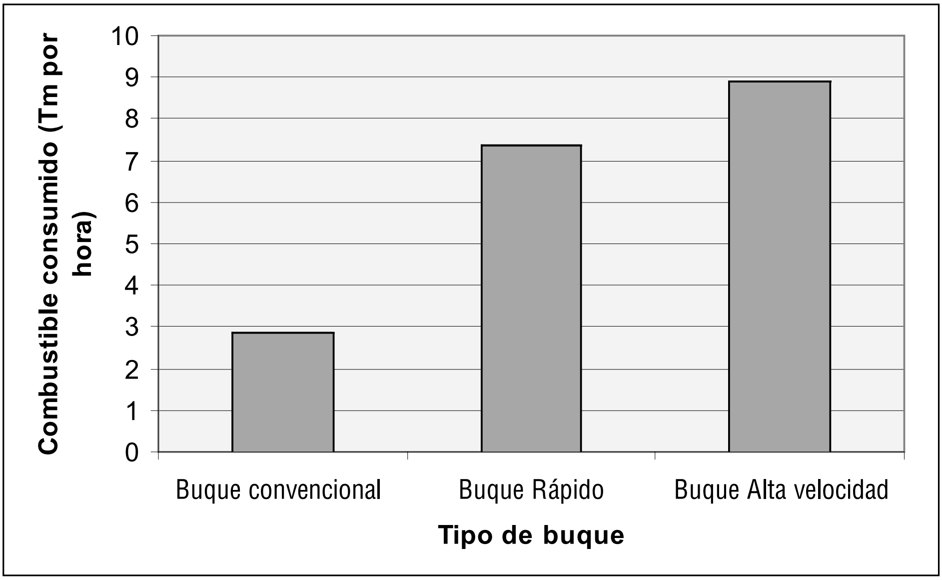
Figure 2. Freight capacity and speed rate by group of ships.



Source: Own based on Marchant, C. 2002

The main differences on deadweight capacity and speed, can be seen in the next diagram where it is seen that there is an opposite relationship between the speed and the deadweight capacity, although the freight levels are very similar between the conventional and fast ships, where technical considerations are not an obstacle for combine load capacity up to speed below 30 knots. A different situation exists when we talk about high speed crafts, where the freight possibilities are drastically reduced. Conventional ships offer more capacity but not only less speed because a minor fitted power but reduced fuel consumption. Fast conventional ships offer a mean term between conventional ships and high speed ships but still maintaining a high freight capacity. However the power needs use to be more than twice the previous group level, meaning an increase in fuel consumption and then increasing the operational costs.

Figure 3. Fuel consumption per hour



Source: Own based on Marchant, C. 2002

High speed crafts try to recover the high fixed costs, by means of only a crew and maximising the number of trips per day. They fit very well in a very high demand markets and shorter routes. Carrying out an analysis per trailer or freight slot, taking information from different published data and shipbuilders, we can confirm that the building cost per slot and the fuel consumption per slot are other variables to be kept in mind.

Comparing one example per group with their main particulars and performances, we would find the following figures:

— Conventional ship, this first case could be represented by the Stena Runner series with a 370 TEU's capacity or 2,700 lane meters (180 trailers x 15 m.), speed of 22 knots and a power plant based on four 5,760 kW (23.040 kW) engines coupled to twin variable pitch screws and a estimated consumption rate of 4,6 Tm /hour at 90%. Every freight slot cost around 100,000 €. An example in the Ro/Pax group would be the Fantastic owned by *Grandi Navi Velocci*, is a 7,150 deadweight tonnes ship with 1,850 lane meters (123 trailers x 15 m.), speed of 18 knots and a power plant of 4 engines developing a total output of 25,916 kW, coupled to twin variable pitch screws. The passenger and crew capacity in this case is 2,300 persons.

— Fast conventional ships, are a group where a superior speed supposes larger frictional forces and higher generated waves. There are ways to reduce these issues as reducing the wet surface by means of multihull designs or through a dynamic reduction of displacement. In this case we are going to use an example of a mono-hull design with a cargo capacity of 1,460 lane meters or 100 trailers or 4,000 Tm of deadweight and a maximum speed of 28 knots or 25 knots at 90% of power. A model of ship would be the Blohm & Voss Trailer ferry, used also in the EMMA EU project, propelled by twin 16,800 kW engines coupled to a single screw. The rate of consumption was around 6,0 Tm per hour and the building cost per slot reached 245,000 € or 2.5 times the previous example.

The Ro/Pax example in this case, is the *Eurostar Roma*, owned by Grimaldi Napoli with 5,717 tonnes of deadweight with 1,700 lane meters and a speed of up to 27 knots due to the four engines developing a maximum output of 31,680 kW coupled to twin variable pitch screws and consuming around 6 tonnes of fuel per hour.

— High speed catamaran ship with a cargo capacity of 100 TEU's and speed around 40 knots. There are quite examples as the HSS 1500 series of Stena with 1,500 tonnes of deadweight and a capacity of 1,500 passengers and 800 lane metres or 50 trailers or 375 cars. Her speed reaches the 40 knots and is propelled by 4 gas turbines developing 73,529 kW. Or the more recent *Benchijigua express* with similar dimensions and 1,350 passengers together with 727 lane metres for trailers or 123 cars. She develops 32.800 kW and reaches 40 knots at only 500 tonnes of cargo.

Figure 4. Conventional fast ship, Eurostar Roma.

Source: http://www.shortsea-es.org/casosexito/noticiasnewdesplegada.asp

CONCLUSIONS

The European Union is promoting the short sea shipping as a more sustainable transport mode. One of the ways to improve the competitiveness in front of the road transport and the air passenger transport is the extensive use of high speed vessels. However high speed crafts dedicated in some traffics can be more pollutant than other modes, mainly when speeds exceed from 25 knots or when reaching critical levels at 35 knots (depending on the routes, cargo conditions or road congestion). For example from the previous analysis a conventional ferry needs 0,125 MW per trailer that is a third of 0,350 MW needed by a fast conventional ship developing for example 28 knots (only 25% faster). A HSC sailing at 40 knots needs a mean of 1,120 MW per trailer that is ten times the conventional ferry power needs. The capital costs per trailer slot also differs depending on the type of ship, as the fast conventional ships costs around 70% more per trailer slot and the HSC up to four times the conventional ship cost.

It is evident that HSC reduces the sailing transit time however for maintaining the time earnings, the port operations should be reduced in order to diminish the total transit time. Some recommendations for facilitate the operability of such ships would be:

1. Reduction of waiting times derived from administrative and custom procedures and the need for one stop shop open 24 hours per day.
2. Investments in port infrastructures for reducing the port phase and then transit time.
3. The need for space in ports to accommodate the freight ready to be loaded, with almost no congestion in the facility accesses. This point is opposite to the needs of passengers who prefer to be near the city.
4. Today an accepted compromise is the use of passenger and freight ships, as HSC offer a limited cargo space and the costs could be covered with the passenger.

The showed ships as example represent different options in the short distance traffics. Each one can offer different performances, to be evaluated depending on the traffic or freight to be carried. In a general sense it can be confirmed that:

— The high speed craft needs to double the fast conventional power requirements.
— The high speed craft is almost a 50% faster than the fast conventional ship.
— The fast ship has around 89% more freight lane metres than the high speed craft.

However more speed means more power and then higher operational costs and more pollutant emissions. The additional speed can be profitable in specific routes and in specific conditions as a high demand for covering the high frequency or good weather all the year in order to exploit the ship all year around.

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ANÁLISIS DE LA VIABILIDAD DE LAS EMBARCACIONES DE ALTA VELOCIDAD

Resumen

Este artículo presenta un sintético análisis sobre la viabilidad de los buques de alta velocidad en los tráficos de corta distancia, obtenido a partir de la opinión de diferentes actores en el negocio marítimo. El grupo de investigación TRANSMAR del Departamento de Ciencia e Ingeniería Náuticas de la Universidad Politécnica de Cataluña, ha continuado profundizando en los resultados del estudio finalizado (INECEU: Intermodalidad entre España y Europa), proponiendo buques de alta velocidad para cubrir algunas de las líneas marítimas que se consideraron más viables en la Europa Occidental. La reciente comunicación relativa a la revisión a medio plazo del Libro Blanco de la Comisión Europea del 2001, confirma que la evolución de la distribución entre modos de transporte de carga entre los años 2000 y hasta el año 2010, mostrará un mantenimiento del 39% para el transporte marítimo y un ligero incremento para el transporte por carretera. Uno de los pilares de la política de transporte Europea es la innovación en las tecnologías y sistemas del transporte, mejorando su eficiencia y logística a través de la cadena de suministro.

Como conclusión se evidencia la voluntad de la Comisión Europea para promocionar el transporte marítimo de corta distancia, como un modo más sostenible. Uno de los posibles caminos para mejorar su competitividad frente al transporte por carretera y el aéreo, puede pasar por el uso extensivo de buques de alta velocidad. Sin embargo los buques de alta velocidad utilizados en algunos tráficos pueden ser más contaminantes que otros modos de transporte, principalmente cuando se superan velocidades de 25 nudos o la cifra crítica de lo 35 nudos (lógicamente dependiendo de la ruta, condiciones de carga y la congestión en la carretera). Por ejemplo del análisis realizado anteriormente un ferry necesita 0,125 MW de potencia por remolque, lo que supone un tercio de los 0,350 MW que necesita un ferry rápido que desarrolle 28 nudos (sólo un 25% más rápido). Un buque de alta velocidad navegando a 40 nudos, necesita una media de 1,120 MW por remolque transportado, lo que es diez veces más que la potencia necesaria para un ferry convencional. Los costes de capital por espacio de remolque transportable, también difieren dependiendo del tipo de buque, ya que los buques rápidos cuestan un 70% más por unidad de remolque, mientras que los buques de alta velocidad cuestan hasta cuatro veces más por unidad, que un buque convencional. Es evidente que los buques de alta velocidad reducen el tiempo de viaje, pero para mantener la ganancia en tiempo en la mar, las operaciones en puerto también deben de reducirse para poder mantener la ventaja en el tiempo de viaje. De modo que algunas recomendaciones en este sentido son:

1. La reducción de los tiempos de espera derivados de los procedimientos administrativos y aduaneros, además de la necesidad de poder realizarlos las 24 horas
2. Inversiones en infraestructuras portuarias que agilicen la operativa del buque.
3. La necesidad de espacios en Puerto que almacenen la carga lista para ser embarcada y una ausencia casi total de congestión en los accesos a la terminal. Este apartado se opone a la preferencia de los pasajeros al querer estar lo más cerca posible del centro de la ciudad.
4. El compromiso usado actualmente es el de utilizar buques mixtos, ya que los buques de alta velocidad tienen una capacidad limitada para la carga, peo pueden cubrir sus costes transportando pasaje.

Los buques usados como ejemplo representan opciones diferentes a utilizar en tráficos de corta distancia. Cada uno de ellos puede proporcionar prestaciones diferentes, cuya validez dependerá de las condiciones de la línea, tráfico o carga a transportar. En general podemos afirmar que:

— Los buques de alta velocidad necesitan el doble de potencia que los convencionales.
— Los buques de alta velocidad son en general casi un 50% más rápidos que los buques convencionales rápidos.
— Los buques convencionales rápidos disponen de un 89% más de capacidad de carga que los de alta velocidad.

Sin embargo más velocidad implica mayor consumo y mayores costes operacionales y sobretodo más emisiones contaminantes. Ese plus añadido en velocidad puede ser en algunos casos aprovechado en rutas y condiciones específicas tales como, una alta demanda para cubrir una alta frecuencia de rotación o unas condiciones de buque tiempo durante todo el año que permitan explotar el buque sin restricciones de tiempo.

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IGNITION QUALITY OF RESIDUAL FUEL OILS

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ABSTRACT

The relevance of residual fuel oil aromaticity for its ignition performance in diesel engines has been demonstrated previously and led to the concept of calculating aromaticity from known specification properties. Thus the Calculated Carbon Aromaticity Index (CCAI) can be calculated from density and viscosity, and provides a useful tool to rank the ignition quality of different residual fuel oils: the lower the number, the better the ignition characteristics.

Potential improvements to the CCAI concept have been investigated. The CCAI represents the aromaticity of the entire fuel. However, at low load engine operations ignition occurs at relatively low temperatures, when only part of the injected fuel may have evaporated. Under these conditions the high molecular weight, highly aromatic (asphaltene) components in all probability are not all vaporised. Thus the aromaticity of the (lighter) part of the fuel might be different from the bulk and possibly more relevant to ignition quality. A programme to investigate the relationship between ignition delay and the aromaticity of the fuel vapour under certain engine conditions has been carried out.

The relationship between ignition delay and CCAI was demonstrated to be valid at all engine conditions employed and comparable to the one found previously. No improvement could be realised by taking into account the micro carbon residue (MCR) content as a measure for the heavy fraction of the fuel, nor any other of the available fuel parameters. The results of Pyrolysis Combustion Mass Spectrometric Element (PCME) analysis of the fuels, providing detailed compositional information of the vapour at different temperatures, indeed confirm that the aromaticity of the lighter fuel fractions does not dominate the aro-

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maticity of the fuel vapour and therewith does not dominate the ignition performance of the fuel. However, in view of the still rather limited predictive power of the CCAI, other, not yet identified fuel parameters must play a role.

Key word: Carbon Aromaticity index, ignition, residual fuel oil

INTRODUCTION

In the absence of an indicator for ignition quality of residual fuels, like the Cetane Index for distillate fuels, extensive research by Zeelenberg et al.1,2 resulted in the concept of the Calculated Carbon Aromaticity Index (CCAI). Since its introduction in the early eighties an increase in the use of CCAI has been observed in the industry. Experience with the concept learns that it is a useful tool for ranking fuels roughly on ignition quality, but also that the CCAI is not a very accurate measure. Apparently fuel parameters other than carbon aromaticity play a role. Possible improvements of the CCAI have been sought in the quality of the light end components of fuels, assumed to ignite first.

THE SHELL CCAI CONCEPT

Ignition difficulties when using distillate fuels are almost unheard of. For many years the ignition quality of these fuels, such as gas oil, has been characterised primarily by a parameter known as Cetane Number, although to a lesser extent other methods such as Cetane Index or Diesel Index have been used also. Current international specifications for marine distillate fuels, such as the ISO 8217: 1996 and BS MA100: 1996, include a minimum limit for Cetane Number. Regrettably there is no similar widely recognised procedure for characterising the ignition quality of residual fuel oil. For a number of reasons the methods used for determining ignition quality of distillate fuels cannot be applied to residual fuel oils. Therefore in the early eighties Shell Research embarked upon a programme with the objective of gaining an understanding of the factors controlling the ignition performance of residual fuel oils, and to identify means of quantifying ignition quality.

Both the physical and chemical properties of residual fuel oil were found to have an influence on ignition performance. Physical properties are viscosity and temperature. Atomisation quality is greatly affected by fuel viscosity. Too high a viscosity at injection increases fuel droplet size, which hinders fuel/air mixing in the cylinder and extends ignition delay and combustion. Many engine designs now incorporate fuel management systems capable of operating at temperatures which allow a wide range of residual fuels to be burned without difficulty.

The relevance of the chemical composition of residual fuel oil on ignition was also demonstrated. This led to the recognition that ignition performance relates to fuel aromaticity. Since aromaticity is a difficult parameter to measure in the absence of specialist laboratory equipment, Shell developed the concept of calculating residual fuel aromaticity. The resulting Calculated Carbon Aromaticity Index (CCAI) can be calculated on the basis of specification properties viscosity and density. It is this parameter which has gained favour as the most practical and meaningful method for characterising ignition quality of residual fuel oils.

CCAI can be calculated from the following formula:

$$CCAI = D - 81 - 141Log\left[Log\left(Vk + 0.85\right)\right] - 483Log\left[\frac{T + 273}{323}\right]$$

Where: D = density at 15°C, kg/m³
Vk = kinematic viscosity (mm²/s) at temperature T°C

It must be stressed that CCAI is a unit-less number allowing ranking the ignition qualities of different residual fuel oils: the lower the number, the better the ignition characteristics. CCAI does not give an absolute measure of ignition performance since this is much more dependent upon engine design and operating conditions. For this reason no attempt has been made to include limiting values in international standards, since a value which may be problematical to one engine operated under adverse conditions may perform quite satisfactory in many other instances. Modern medium speed engines will tolerate CCAI values up to 870 to 875, and even values up to 890 and beyond are acceptable to some engine types. Medium speed diesel engines are sensitive to fuels having poor ignition characteristics, while low speed cross head engines may be more tolerant of higher CCAI values. The limits for viscosity and density in international marine fuel specifications in themselves provide a control of ignition quality for the main residual fuel oil grades. For example, a 380 mm²/s (@ 50°C) fuel oil at maximum specification density of 991 kg/m³ has a CCAI of 852, whilst a 180 mm²/s (@ 50°C) fuel oil with the same density has a CCAI of 861. Ignition characteristics improve with increasing viscosity and decreasing densi-

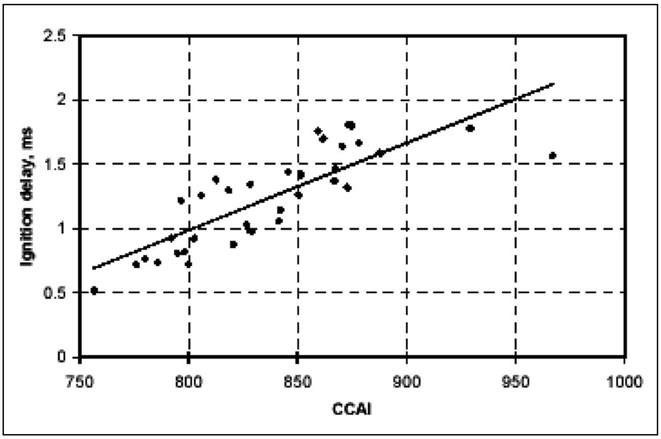

Figure 1: Correlation Ignition delay - CCAI

ty. Ignition difficulties can become more acute at lower fuel viscosity (e.g. < 100 mm^2/s @ 50°C) if there is not a significant corresponding reduction in density. This is one of the reasons for the lower density limits applying to the low viscosity grades in the international specifications.

The correlation between ignition delay and CCAI is not ideal (see Figure 1 for typical results obtained in a research engine). The scatter of data points around the regression line is rather large. At lower engine outputs this scatter is even larger. This is not necessarily due to experimental errors since cases have been reported where from two fuels with similar analyses one gave ignition problems at low engine output, but the second ran as normal. It is recognised that fuels may appear on the market whose poor ignition performance is not predicted by their CCAI value, but also that fuels which perform satisfactorily should not be rejected on basis of their too high CCAI value.

Thus the CCAI is based on bulk fuel properties only and does not require detailed chemical information of the fuel. The use of CCAI has been evaluated3,4 and it has been concluded that this parameter gives a rough estimate of the ignition performance of heavy fuels at best. In contrast, CCAI has been found to correlate very well with the ignition performance of distillate fuels. The latter could suggest that the ignition quality of residual fuels is more closely related to the quality of the distillate part of the fuel rather than to the bulk properties. This has lead to further research attempting to improve the CCAI correlation by including vapour composition.

INFLUENCE OF VAPOUR AROMATICITY

A critical element of the CCAI concept is the assumption that at the moment of ignition of the vapour, almost all of the injected fuel has vaporised, i.e. that the aromaticity of the vapour is identical to that of the bulk . Particularly at low temperatures, at engine start-up conditions and low load operation, this is not necessarily the case. The high molecular weight fraction of the fuel will not vaporise at all or only with great difficulty. Therefore the aromaticity of the bulk as indicated by the CCAI is not representative for the aromaticity of the igniting vapour4,5. Theoretically the following situations could exist:

- At low ignition temperatures, the vapour composition may be primarily that of the distillate diluents and contains hardly any heavy residue of the fuel. If the diluent has a parafinnic character the ignition performance will be better than predicted by the CCAI and worse for an highly aromatic diluen.
- At somewhat higher temperatures, the residue will also begin to vaporise and take part in the ignition process. Because the aromaticity is mainly contained in the high molecular weight part of the residue, the contribution to the nature of vapour is mainly parafinnic. The ignition performance will therefore be more or less the same for a fuel with a parafinnic diluent or might even improve in case of an aromatic diluent.

• At high temperatures almost all of the residue evaporates and contributes to the nature of the vapour phase. In this case the aromaticity of the vapour will be similar to that of the bulk and the ignition performance may be as predicted by the CCAI.

The Calculated Vapour Aromaticity Index (CVAI)

Work to better predict ignition quality by taking the aromaticity of the vapour phase into account was initiated by J.C. van der Werff, J.S.E.A.M. Naber and F.M. Wortel (unpublished work). A crucial element in the estimate of the vapour aromaticity, is the selection of a parameter for the amount of non-vaporisable aromatic carbon of the fuel. For pragmatic reasons the micro carbon residue (MCR) content has been chosen for this purpose. MCR is determined at 500°C close to the experimental conditions in the engine. Also MCR (or Conradson carbon residue, CCR) is an existing fuel specification. To obtain a measure for vapour aromaticity, the CCAI was corrected for MCR for which the following equation was derived (Appendix I):

$$CVAI = \frac{CCAI - 10.5c(MCR)}{1 - c\left(\frac{MCR}{100}\right)}$$

Correlation with ignition delay in test rig

The validity of this CVAI has been evaluated on a series of test fuels blended from short residues or thermally cracked residues with either kerosene or light catalytically cracked cycle oil (LCCCO). The choice of these components was to create so-called gap-fuels with a distinct difference between the aromaticity of the distillate and residual fraction. The ignition tests were performed in a fuel ignition test rig, consisting of an electrically heated, cylindrical combustion chamber of approximately 4 litre in which compressed air at a pressure of maximum 50 bar is heated to a temperature of maximum 600°C.

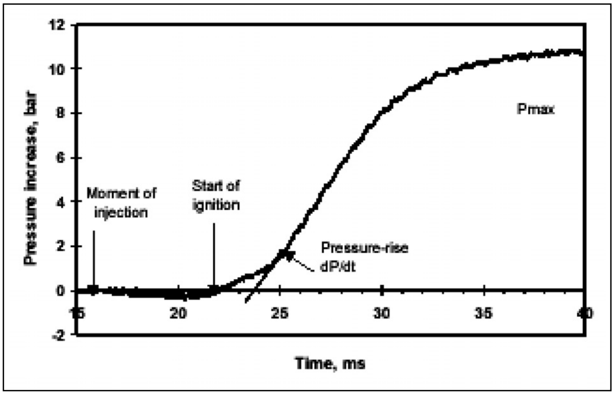

Figure 2: Pressure increase in the combustion chamber after injection of fuel.

At the test conditions one single amount of fuel is injected with a volume of maximum 0.15 ml. The moments of injection and ignition are derived from the fuel pressure or needle lift and cylinder pressure or light emission signal, respectively. A typical curve of the pressure development in the combustion chamber is shown in Figure 2.

The ignition delay of the test fuels was measured at three different temperature and air pressure conditions: 450°C and 45 bar, 490°C and 50 bar, and 525°C and 50 bar. Each fuel was measured 10 times at each condition.

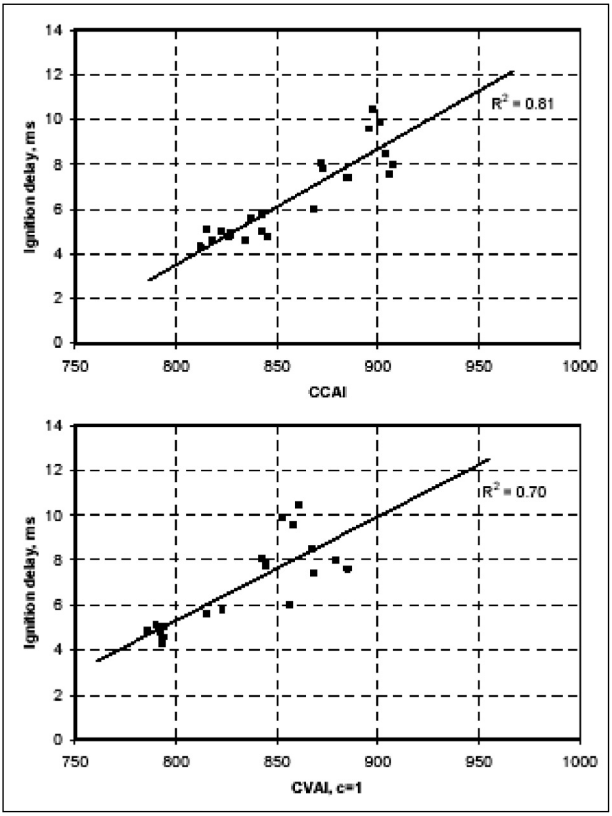

Figure 3: Relation between test-rig ignition delay and CCAI and CVAIc = 1. Temperature 490°C / Pressure 50 Bar

The results show that the high CCAI (>850) fuels with LCCCO as cutter, give considerable variation in results. On the other hand, the low CCAI fuels with kerosene as cutter give much less variation. This was particularly valid at 450 °C. This may be interpreted to be the result of incomplete residue vaporisation at this relatively low temperature, although it is not at all clear why this is not observed similarly with the high CCAI fuels.

		525 °C	490 °C	450 °C
CCAI	c = 0	0.92	0.81	0.85
CCAI	c = 0.5	0.93	0.79	0.90
CCAI	c = 1.0	0.88	0.70	0.93
CCAI	c = 1.2	0.84	0.64	0.92

Table 1: R2-values for linear regression of test-rig ignition delays versus CCAI and CVAI (Fuel 1-21).

Bore, mm	137
Stroke, mm	168
Volume at TDC, ltr	0.19
Compression ratio	13.8
Speed, min^{-1}	1000
Load, Nm	100 - 300
Power output, kW	10.5 - 31.5
Charge air pressure, bar	0.9 - 2.2
Charge air temperature, °C	30 - 60
Static injection timing, °CA BTDC	31

Table 2: Technical details and operating conditions of the AVL-Caterpillar 1Y540 engine used for ignition delay measurements.

The CVAI does not give a better correlation with ignition delay than CCAI. The regression coefficients decrease with increasing c-value (Table 1), i.e. increasing effect of correction for MCR, for the tests at 525°C and 490°C. Only for the tests at 450°C the regression coefficient tends to improve a little.

Correlation with ignition delay in test engine

To investigate the correlation of CVAI under realistic conditions an extended series of fuels was tested in an AVL-Caterpillar 1Y540 single cylinder 4-stroke high speed diesel engine.

The engine is fully instrumented for ignition parameter measurements, i.e. with pressure transducers in both combustion chamber and high-pressure fuel line approximately 10cm in front of the injector housing, injector needle lift sensor and shaft encoder for main axis angular position (degrees crank angle, °CA). The engine was run at well controlled conditions with respect to speed, load, air inlet temperature and pressure, cooling water and lubricating oil temperatures and pressures.

The fuel was supplied from a heated 60 litre container placed on a balance, allowing continuous monitoring of fuel consumption at the desired fuel temperature for a 12-15 mm^2/s injection viscosity. For each ignition measurement, combustion pressure, fuel line pressure and needle lift data were recorded by an AVL Indiskop 647 instrument, transferred to an IBM compatible PC and analysed with dedicated software for ignition delay and combustion hardness.

	At moment of injection		At Top Dead Centre	
Test Mode	°C	Bar	°C	Bar
I	411	21.0	487	31.2
II	423	27.8	499	41.6
III	456	39.3	537	58.9
IV	479	50.9	562	76.2

Table 3: Typical adiabatic compressed air temperatures and pressures calculated for the AVL Caterpillar 1Y540 engine.

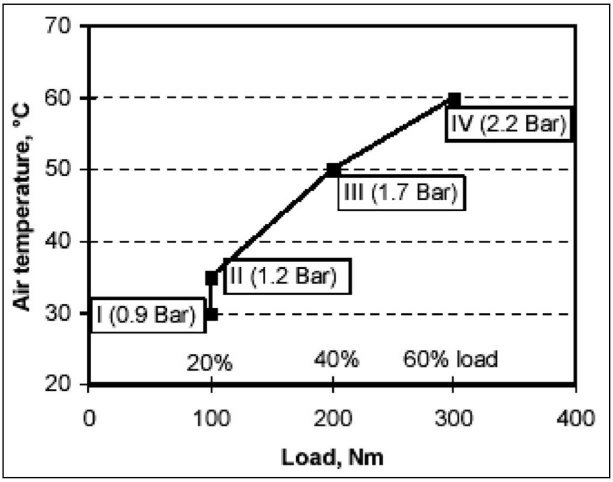

Figura 4: AVL-Cat engine test conditions. Variation of load and air temperature/pressure

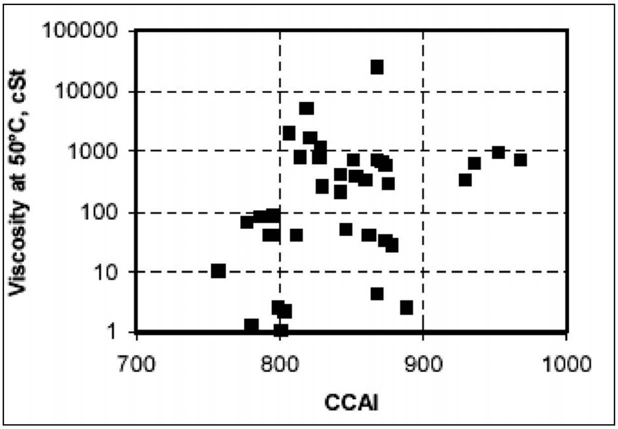

Figure 5: Viscosity/CCAI of ignition test fuels.

Test modes

The test sequence was designed to have in a single engine test both low and high ignition temperatures, thus allowing demonstration of both the CVAI and CCAI ignition prediction concepts. This was attempted with a sequence of 4 modes (I,II,III and IV) by stepwise raising engine load (100 to 300 Nm) and inlet air temperature (30 to 60°C) and pressure (0.9 to 2.2 bar), while maintaining engine speed (1000 rpm) and cooling water and lubricating oil temperatures (80°C) and pressures constant.

The cold engine was started on gasoil and then operated in approximately 5 minutes towards 20% power output with 1000 rpm speed (1400 rpm nominal) and 100 Nm load (360 Nm nominal) using naturally aspirated (0.9 bar) inlet air at a temperature of 30°C. Then the gasoil was quickly switched for the test fuel to start Mode I. For Mode II the air compressor and in line heater were activated giving inlet air with temperature of 35°C and pressure of 1.2 bar. For Mode III these parameters were raised to 50°C and 1.7 bar, respectively, allowing the power output to be raised to 40% with a load of 200 Nm. For Mode IV the inlet air temperature and pressure were raised further to 60°C and 2.2 bar, respectively, at engine load of 300Nm representing 60% power output. After Mode IV the engine was turned off and allowed to cool. Mode IV closely resembles the conditions on the MaK engine used previously by Zeelenberg1

Ignition parameters

Start of injection was set to occur at 31°CA BTDC (static). Injector opening was derived from both the needle lift signal and the fuel line pressure reaching 260 bar, as for every test the spring tension of the cleaned injector was adjusted to open at this pressure. This injector opening generally occurred at approximately 19°CA BTDC (dynamic). Start of ignition/combustion was derived from the combustion pressure trace, obtained from the cylinder pressure by mathematical correction for the compression pressure. The °CA scale was converted into a time scale assuming the speed of rotation to be constant. This was independently verified to be the case within the accuracy of the measurement. Ignition delay is defined as the time period in milliseconds (ms) between start of injection and onset of ignition and combustion.

The combustion hardness, or the rate of pressure rise at incipient combustion (dP/dt in kbar/s), is much less often quoted as ignition parameter. This is because it is not as well defined and easily determined as the ignition delay, and, maybe more importantly, previously a satisfactory correlation was established with CCAI. Thus, high combustion hardness was associated with high ignition delays and CCAI (R2 = 0.4 - 0.6). In this investigation the combustion hardness was computed from the smoothed combustion pressure/time trace in two ways (a and b), viz. from the slope of the straight lines created by the 0 and 5 Bar (dP/dt-a) and the 5 and 10 Bar (dP/dt-b) combustion pressure points, respectively.

Test variability

Ignition parameters were recorded after 5 min (test-1) and after 25 min (test-2) into every test mode, and each measurement was repeated 10 times.

One of the commercial fuels (RFO-Tank 11) was tested throughout the investigation as a reference fuel to assess test variability. The resulting 13 separate tests showed satisfactory overall repeatability of ignition delay (SD ≈ 8%), although single results (max., min) can deviate quite substantially. Much greater variation exists in the combustion hardness data, however, particularly when defined over 5-10 kbar/s combustion pressure rise (SD ≈ 19%).

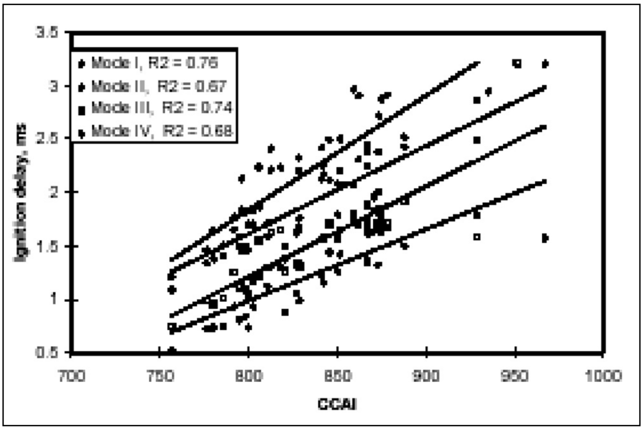

Figure 6: Relation between ignition delay and CCAI

Test Mode	Parameter	R^2
I	CCAI	0.76
	CCAI, MCR	0.76
	CCAI, S	0.78
	CCAI, MCR, S	0.79
	CCAI, MCR, FP	0.76
	CCAI, MCR, S, FP	0.80
II	CCAI	0.67
	CCAI, MCR	0.70
	CCAI, MCR, S	0.71
	CCAI, MCR, FP	0.75
	CCAI, MCR, S, FP	0.75
III	CCAI	0.74
	CCAI, MCR	0.74
	CCAI, MCR, S	0.74
	CCAI, MCR, FP	0.75
	CCAI, MCR, S, FP	0.75
IV	CCAI	0.68
	CCAI, MCR	0.72
	CCAI, S	0.76
	CCAI, MCR, S	0.76
	CCAI, MCR, FP	0.72
	CCAI, MCR, S, FP	0.76

Table 4: Summary of linear regression analysis attempts between ignition delay and several fuel parameters.

Test Mode	dP/dt-a	dP/dt-b
I	0.65	0.02
II	0.66	0.29
III	0.21	0.10
IV	0.04	0.38

Table 5: Linear Regression analysis between combustion hardness and CCAI. (R2-value).

The results from the first (test-1) and second series (test-2) of measurements in every test mode were always closely identical. This clearly suggests that following a change in engine operating conditions (thermal) equilibrium is quickly re-established.

Test fuels

In the test program a total of 39 different fuels were used. They are 22 heavy fuels specially blended from a selection of residue and diluent components, four commercial RFO's and 2 special high density fuels, and 8 light and 3 heavy fuel blending components. In Figure 5 the total variation in CCAI and viscosity (Vk50) values is illustrated, highlighting the wide CCAI range of 757 - 959 at viscosities between 1.1 and 26000 mm^2/s.

Ignition delay results

From the plots between ignition delay and CCAI for all test modes in Figure 6 it is clear that the relationships and also their slopes are very similar. Only the ignition delays become smaller upon going from Mode I to IV.

Correlation coefficients (R2-value) for linear regression are between 0.68 (Mode IV) and 0.76 (Mode I) as is shown in Table 4. These values are closely identical to the 0.77 (at inlet air temperature 60°C) and 0.71 (at inlet air temperature 45°C) found previously by Zeelenberg in the MaK engine (1).

With the present results no improvement of the correlation between ignition delay and CCAI could be realised by taking into account the MCR content nor other bulk fuel parameters sulphur (S) and flash point (FP) as is summarised in Table 4.

Clearly the ignition delays correlate best with CCAI alone and because this correlation is also hardly affected by the engine output, it must be concluded that the CCAI best represents the ignition performance of the fuel vapour. This may be either because all of the injected fuel is vaporised and contributes to the vapour aromaticity (i.e. CCAI = CVAI), or, alternatively, if not all of the fuel is vaporised the vapour aromaticity is proportional to the aromaticity of the bulk (i.e. CCAI = CVAI). This appears irrespective of the engine conditions employed.

Combustion hardness

The rate of pressure rise or combustion hardness data for the first (dP/dt-a) and second parts (dP/dt-b) of the combustion pressure show poor correlations with CCAI (Table 5).

In fact, only for dP/dt-a in Modes I and II and for dP/dt-b in Mode IV linear correlation coefficients are observed that may be comparable to the 0.61 (at inlet air temperature 60°C) and 0.45 (at inlet air temperature 45°C) found by Zeelenberg. Strangely enough, in these modes dP/dt-a decreases with CCAI, while dP/dt-b increases with CCAI. The latter result may well be in agreement with the positive correlation found by Zeelenberg.

Vapour composition analysis

To obtain further insight in the volatility of heavy fuels and their vapour composition, the fuels were analysed with Pyrolysis Combustion Mass Spectrometric Elemental analysis (PCME), which gives the volatility of the fuel by means of a True Boiling Point (TBP-PCME) temperature and also the elemental composition of the emitted vapour (Appendix II). From these data both the vapour carbon aromaticity index (VCAI, not to be confused with the CVAI which is calculated from CCAI and

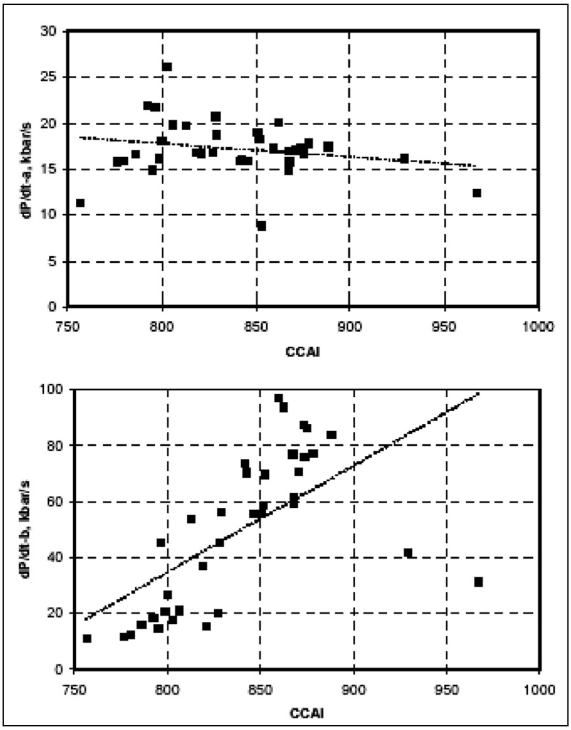

Figure 7: Relation between combustion hardness and CCAI as observed in Mode IV

MCR) and the aliphatic and aromatic carbon yields of the vapour were calculated for various arbitrary TBP-PCME temperature limits.

From the results it is concluded that strong correlations exist between CCAI values and the calculated vapour parameters VCAI, aliphatic carbon yield and aromatic carbon yield for TBP-PCME temperatures above 500°C. Not surprisingly, therefore, that also in plots of ignition delay against these parameters for the different TBP-PCME temperature limits the best linear correlations exist with data from temperatures above this arbitrary limit of 500°C. However, correlation coefficients are found less good (Figure 8) than for ignition delay/CCAI relationships. This may have been caused by errors in the determination of VCAI.

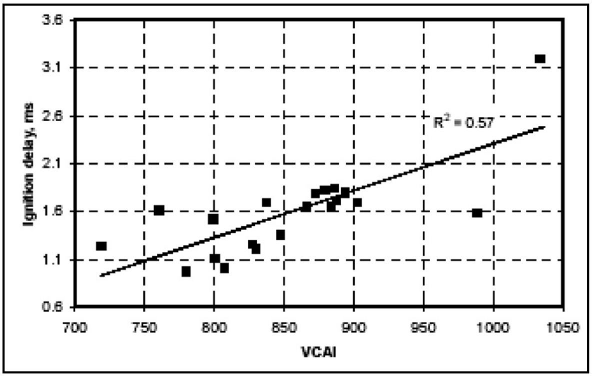

Figure 8: Relation between ignition delay and VCAI for Mode III (TBP-PCME up to 650°C)

CONCLUSIONS

Previous work on the ignition performance of heavy fuels in diesel engines has shown that at medium load (MaK) engine operation the CCAI best describes ignition performance. The CCAI is based upon fuel density and viscosity, and accurately represents the aromaticity of the bulk of the fuel. It can be used to rank fuels on ignition performance but does not offer a direct measure of ignition quality.

In the assumption that the aromaticity of the vapour from a fuel is not the same as that of the bulk of the fuel, the CVAI was derived by correction of the CCAI with the MCR content, assumed to represent the aromatic carbon of the bulk which cannot be evaporated.

Results presented in this paper show that ignition delay still correlates best with CCAI at all engine conditions employed, and these correlations were comparable to that found previously by Zeelenberg (R2-value ≈ 0.7). No improvement could be realised by either CVAI, taking into account the MCR content or the other available fuel parameters: sulphur content and flash point.

This conclusion is also supported by PCME studies. Thus, both ignition delay and CCAI correlate best with the aromaticity and also with the aliphatic carbon yield of the cumulated vapour formed up to (and above) a temperature of 500°C in this analysis.

Although bulk aromaticity may be the paramount fuel parameter determining its ignition performance, also other fuel parameters must play a role to explain the non-ideal correlation between CCAI and ignition delay. At this point it is worth mentioning that results from further tests indicate that one of these additional parameters may be contained in the processing nature of the blend components in the fuel.

The introduction of CCAI in the early 1980's made available a tool by which engine manufacturers could specify an acceptable range of ignition values for fuels to be used in their engines. The current work confirms CCAI as still the best available parameter to indicate ignition quality. But it cannot be used as an absolute measure of quality because of its limited accuracy.

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APPENDIX

Calculation of the aromatic carbon content of fuel vapour (CVAI)

In order to correct the CCAI for the aromatic carbon in the residue which cannot be vaporised the following procedure was defined:

- Calculation of the aromatic carbon content from theCCAI of the bulk of the fuel from the empirically established equation:

$$\%C_{ar} = -255 + 0.339CCAI$$

Calculation of the vaporisable aromatic carbon content by correction for the amount of non-vaporisable aromatic carbon with MCR:

A gram of residue contains on average 0.85 g carbon. Consequently, the mass of aromatic carbon per gram residue is given by:

$$mC_{ar} = \left[\frac{0.85x\%C_{ar}}{100}\right]$$

Per gram of residue, the mass of vaporisable aromatic carbon is given by:

$$mC_{ar}(vap) = mC_{ar} - MCR'$$

Where MCR' is the MCR content in g/g residue (%MCR/100), where it is assumed that Micro Carbon Residue consists of pure aromatic carbon.

Per gram of residue, the total mass of carbon in the vapour can be expressed as:

$$mC_1(vap) = 0.85 - MCR'$$

The mass percentage of aromatic carbon in the vapour is then given by:

$$\%C_{ar}(vap) = \frac{mC_{ar}(vap)}{mC_1(vap)} \cdot 100\text{,or}$$

$$\%C_{ar}(vap) = \frac{\frac{0.85x\%C_{ar}}{100} - MCR'}{0.85 - MCR'} \cdot 100\text{,or}$$

$$\%C_{ar}(vap) = \frac{\%C_{ar} - 1.18MCR}{100 - 1.18MCR} \cdot 100$$

However, depending on the test temperature, the full MCR correction may be over or under estimating the crackability of the larger aromatic structures into smaller components which will end up in the vapour phase. Therefore a correction factor *c* instead of 1.18 is used which can be optimised to obtain the best correlation, which results the equation.

$$\%C_{ar}(vap) = \frac{\%C_{ar} - c(MCR)}{1 - \frac{c(MCR)}{100}}$$

For *c=0* all the MCR is evaporated and CVAI and CCAI are identical.

• Calculation of CVAI from the inverse equation in the first step:

$$CVAI = 752 + 2.95x\%C_{ar}(vap)$$

This sequence of calculations can be reformulated into the following overall formula:

$$CVAI = \frac{CCAI - 10.5c(MCR)}{1 - c\left(\frac{MCR}{100}\right)}$$

CALIDAD DE IGNICIÓN DE LOS ACEITES COMBUSTIBLES RESIDUALES

RESUMEN

Con anterioridad ya se ha demostrado la importancia de la aromaticidad del aceite combustible residual con respecto a su actuación y ha dado lugar al concepto de cálculo de aromaticidad a partir de propiedades conocidas de las especificaciones. De este modo, el Índice de Aromaticidad Carbónica Calculada (Calculated Carbon Aromaticity Index, CCAI) se puede calcular a partir de la densidad y viscosidad, y proporciona una herramienta útil para clasificar la calidad de ignición de diferentes aceites combustibles residuales: cuanto más bajo sea el número, mejor serán las características de ignición.

Se han investigado las mejoras potenciales al concepto de CCAI. El CCAI representa la aromaticidad de todo el combustible. Sin embargo, en operaciones de baja carga del motor se da la ignición con temperaturas relativamente bajas, cuando sólo parte del combustible inyectado se puede haber evaporado. Bajo estas condiciones el alto peso molecular, los componentes altamente aromáticos (asfaltenos) con toda probabilidad no se han evaporado del todo. Así, la aromaticidad de la parte del combustible (más ligera) podría ser diferente del grueso y posiblemente más relevante para la calidad de la ignición. Se ha llevado a cabo un programa para investigar la relación entre el retraso en la ignición y la aromaticidad del vapor del combustible bajo ciertas condiciones.

Se ha demostrado la validez de la relación entre el retraso de la ignición y el CCAI en todas las condiciones de motor empleadas y comparables a la ya descubierta. Ninguna mejora se podría realizar teniendo en cuenta el contenido del residuo micro-carbónico (micro carbon residue, MCR) como medida para la fracción pesada del combustible, ni para cualquier otro de los parámetros de combustible disponibles. Los resultados del análisis de los combustibles mediante Elementos Espectrométricos de Combustión de Masa por Pirólisis (Pyrolysis Combustion Mass Spectrometric Element, PCME), proporcionando información detallada de la composición del vapor a diferentes temperaturas, confirma de hecho que la aromaticidad de las fracciones de combustible más ligeros no domina la aromaticidad del vapor del combustible y por consiguiente no domina el resultado de la ignición del combustible. Sin embargo, a la vista del potencial predictivo todavía limitado del CCAI, otros parámetros de combustible, aún no identificados podrían desempeñar un papel.

Palabras Clave: Índice de Aromaticidad Carbónica, ignición, aceite combustible residual

CONCLUSIONES:

Los trabajos precedentes sobre el resultado de la ignición de combustibles pesados en motores diesel han mostrado que el CCAI describe mejor el resultado de la ignición con el motor funcionando a media carga (MaK). El CCAI se basa en la densidad y viscosidad del combustible, y representa adecuadamente la aromaticidad del grueso del combustible. Se puede utilizar para clasificar los combustibles según su ignición pero no ofrece una medida de la calidad de ignición.

Asumiendo que la aromaticidad del vapor de un combustible no es la misma que la del grueso del combustible, el CVAI se derivó por corrección del CCAI con el contenido del MCR, que se supone representa el carbono aromático del grueso que no se puede evaporar.

Los resultados que se presentan en este artículo muestran que el retraso de la ignición se corresponde todavía mejor con CCAI en todas las condiciones del motor empleadas, y estas correlaciones fueron comparables a las descubiertas anteriormente por Zeelenberg (R^2-valor ≈ 0 0.7). No se pudo apreciar ninguna mejora CVAI, teniendo en cuenta el contenido de MCR ni con los demás parámetros de combustibles disponibles: contenido de azufre y punto de combustión.

Esta conclusión también está apoyada por los estudios de PCME. Así, tanto el retraso en la ignición como la CCAI se corresponden mejor con la aromaticidad y también con el residuo del carbono alifático del vapor acumulado que se forma a temperaturas de (y por encima de) 500°C en este análisis.

Si bien el grueso de la aromaticidad puede ser el principal parámetro que determine el resultado de la ignición, también otros parámetros del combustible pueden desempeñar un papel para explicar la correlación no-ideal entre el CCAI y el retraso de ignición. En este punto vale la pena mencionar que los resultados de pruebas ulteriores indican que uno de estos parámetros adicionales puede estar contenido en la naturaleza del tratamiento de la mezcla de componentes del combustible.

La introducción del CCAI a principios de la década de 1980 hizo posible una herramienta con la que los fabricantes de motores podían especificar una tasa de valores de ignición aceptables para los combustibles a utilizar en sus motores. El presente trabajo confirma CCAI como todavía el mejor parámetro disponible para indicar la calidad de ignición. Pero no se puede utilizar como una medida de calidad absoluta debido a su limitada exactitud.

Calidad de encendido

Los parámetros experimentales relacionado con la calidad de encendido de CCAI de CII desarrollados, proporcionan un método experimental para la clasificación de combustibles residuales

La calidad del combustible suministrado a la embarcación puede afectar a la navegabilidad, así como el subsiguiente tratamiento del combustible a bordo. En un mundo dónde es común la inconstante calidad del combustible, hay una necesidad por establecer medidas más adecuadas que regulen la efectividad del tratamiento del combustible a bordo.

Metodología:

La metodología utilizada en el presente artículo ha consistido en la consulta de diversas bibliografía, para poder entender correctamente la evolución y estado actual del tema tratado, se ha utilizado además de la bibliografía ordinaria la información disponible en la red, en portales de organismo internacionales solvente. La bibliografía ha sido tanto nacional como internacional.

Finalmente complementamos el articulo con la con la información más reciente en cuanto a la legislación correspondiente a dicho tema por parte del organismo correspondiente, la OMI.

Después de esta elección metodológica, se efectúo la revisión de los textos, figuras y tablas. Para entender esta tarea, utilizando como herramienta fundamental los textos ya seleccionados y a nuestra disposición.

La calidad de los productos comienza a forjarse en la etapa de diseño, siguiendo los máximos estándares de calidad a nivel internacional y teniendo en cuenta las mayores exigencias a las que serán sometidos al momento de ser utilizados.

Posteriorrmente, se elaboran a través de tecnología de última generación presente en cada una de las refinerías que posee la Compañía. Por último, en la etapa de distribución, almacenaje y comercialización, se lleva a cabo un estricto seguimiento que comprende monitoreos continuos de calidad y certificación.

Este sistema de gestión integral de la calidad asegura al cliente la utilización del productos con calidad de origen, permitiendo su trazabilidad desde el mercado a la refinería a través del número de Certificado de Calidad.

Fuel Testing

Residual Fuel, Fuel Oil Testing Analysis	**Fuel Oil Test Methods**
API Gravity / Density / Relative Density (Elevated temp)	D287 / D1298 / D4052
Ash Content Test	D482
Flash Point Test (PMCC)	D 93B
Pour Point Test	D 97
Sediment Content	D473
Sulfur Content Analysis	D4294
Viscosity - Kinematic at 40 & 100C (100 & 210F if necessary).	D445
Water Content	D95
S & W (Centrifuge)	D1796

Number 2 Fuel Oil Basic Testing Analysis	
API Gravity / Density / Relative Density	D287 / D1298 / D 4052
Appearance or Haze	D4176 / Colonial
Cetane Index (Calculation Only)	D976
Cetane Index (Calculation Only)	D4737A or B
Cloud Point	D2500/D5771/D5773
Color	D1500
Copper Corrosion	D130

Distillation	D86
Flash Point	D93A
Pour Point	D97/D5950/D5949
Sulfur Content	D4294
Sulfur Content (WDXRF)	D2622
Sulfur Content by Oxidative Microcoulometry	D3120
Sulfur Content by UV Fluorescence	D5453
S & W (Centrifuge) if required .	D1796 / D2709
Metals Analysis- Al, Si Ashing & Fusion	D5184 / IP377, AAS/AES
Metals Analysis- Al, Si, V, Ni, Fe, Na Ashing & Fusion	IP470, AAS/AES
Metals Analysis- Ni, V, Na Solvent Dilution	D5863B, AAS/AES
Metals Analysis - Ni, V, Fe Digestion & Ashing	D5863A, AAS/AES
Metals, (Graphite furnace for three elements) AAS/AES	—
Metals Testing- Ni, V, Fe Solvent Dilution ICP	D5708A
Metals Testing- Ni, V, Fe Digestion & Ashing ICP	D5708B
Metals Testing, Each additional outside scope of procedure	Various
Microscopic Characterization of Particulates (wear debris)	—
Molecular Weight (Includes necessary tests)	D2502
NACE Corrosion Test	TM0172
Nitrogen Content Kjeldahl	D3228
Nitrogen Content, Nitrogen Speciation	D4629, D5762 Chemiluminescence
Nitrogen - Bases	UOP269 / UOP313
Oxidation Stability (Accelerated)	D2274, D6468 (Octel F21)
Oxidation Stability (Oxygen Overpressure)	D5304.
Particulate Contamination	D6217/IP 415
pH Value	D1293
Polypropylene in Fuels, Qualitative.	BPMarine A.
Polypropylene in Fuels, Quantitative by FTIR	BPMarine B
Pour Point (Amsterdam Procedure)	Shell Method
Refractive Index	D1218
Salt Content	D3230 / D6470 / IP77
Sediment in Lube Oils	D2273
Sediment by Hot Filtration	D4870 / IP375
Sediment - Potential (Accelerated)	IP390 + IP375
Simulated Distillation	D2887 Sim Dis
Simulated Distillation	High Temperature Sim Dis
Storage Stability of Diesel	D4625
Stability Testing, Fuel Oil Peptization Value	P-value and P0/Frmax
Sulfur / Sulphur (Bomb Method)	D129
Sulfur Dioxide Emissions (Calculation Only)	—
Sulfur / Sulphur Speciation	—
Toluene or Xylene Equivalence	—
Ultimate Analysis (C.H.N. with O by difference.)	D5291
Unsulfonated Residue in Diesel	D483
Viscosity (Saybolt) in Diesel	D445 & D2161
Viscosity - Kinematic at 40 & 100C (100 & 210F if necessary)	D445
Viscosity - Kinematic at other temps	D445
Viscosity Index (Including tests)	D445 & D2270
Water Content (Volumetric Karl Fischer)	E203
Water Content (Coulometric Karl Fischer)	D6304
Wax Appearance in Distillates	D3117
Wax Content	UOP46
Water Separability (Demulsification)	D1401 / FTM3201

E TA PAS DE CONTROL DE CALIDAD

Elaboración

Cada refinería cuenta con un laboratorio altamente equipado donde se evalúa la calidad de todos los productos intermedios que salen de cada planta de proceso y fundamentalmente se efectúa la evaluación de la calidad de los productos finales.

Los laboratorios fijos y móviles de Servicio Técnico de Productos controlan la calidad de los combustibles y asisten técnicamente al cliente.

Distribución, almacenaje y comercialización

En esta segunda etapa, el combustible es transportado a través del sistema de distribución de la Compañía y almacenado en las distintas Terminales de Despacho. En esta instancia, el Departamento de Servicio Técnico de Productos efectúa una nueva evaluación de calidad dando lugar a la certificación del combustible y habilitando su despacho.

Inspecciones periódicas

La última etapa la constituyen las inspecciones periódicas que realiza Servicio Técnico en las distintas Terminales y demás puntos de distribución. Las mismas comprende la evaluación de la calidad del combustible y la verificación de las condiciones de almacenaje, de modo de asegurar en el tiempo la ausencia de ciertos elementos que podrían alterar las propiedades del combustible (agua, partículas, etc.)

La estructura de Servicio Técnico consta una serie de laboratorios distribuidos geográficamente a lo largo y ancho del país con cobertura sobre todo el territorio nacional. Están equipados con tecnología de última generación y operados por personal altamente calificado, que además de realizar los controles de calidad mencionados anteriormente cumplen la función de asistir técnicamente al Cliente.

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COLLISION OF FISHING VESSELS: LORENZ CURVES AND GINI INDICES

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Abstract

Fishing vessel collisions are accidents which, in most cases, give rise to other accidents such as explosions, water on board and even sinking. These are accidents which often have fatal consequences for the crews and in many cases signify the loss of the vessel. Such losses can be avoided or reduced if the fishing vessels are equipped with the appropriate technological means and their crews proceed correctly and promptly. Thus, national governments should include in their regulation of the fishing sector control mechanisms that guarantee that the vessels are endowed with the adequate resources and with crews well-trained in safety. These resources and their distribution must be assigned in accordance with the needs of the fleet, which requires an in-depth knowledge of the degree of concentration of accidentality by collision of the fishing sector of a country.

In this context, the aim of this work is twofold: first, to formalise a methodology for the fishing sector of a country which allows the inequalities in the concentration of collisions to be analysed; and second, to apply this methodology to the Spanish fishing fleet for the period 1994-2002. Thus, indices are constructed by region and by fishing types for two variables: fishermen and fishing vessels. In these areas, results are then obtained both for the concentration of collisions and for their associated Lorenz curves. An increase in the inequalities in the spatial and functional distribution of these accidents is observed.

Key words: Fishing vessel collisions, Concentration of fishing vessel collisions, fishing vessel collision accidents.

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1. INTRODUCTION

Fishing vessel collisions are accidents which can be avoided or reduced if the fishing vessels are equipped with the appropriate technological means and their crews proceed correctly and promptly. The planning and regulation of a fishing fleet must incorporate control mechanisms that guarantee that the vessels are endowed with the adequate resources and with crews well-trained in safety. If they do not fulfil these conditions, the consequences may be fatal. Thus, the improvement in safety at sea has, for several decades, been one of the main concerns of various supranational institutions, national governments and non-government organisations. However, the statistics indicate that fishing, in comparison with other sectors, has been and continues to be one of the most dangerous of human activities (ILO 1999).

The present work focuses on fishing vessel collisions. This is a type of accident which usually gives rise to other accidents such as explosions, water on board and even sinking, with fatal consequences for the crews and the service life of the vessel. There is little specific research on fishing vessel collisions and usually addresses such questions as: aspects of the international regulation on collisions such as those concerning reach (Carver 1992a) sailing in narrow waters (Carver 1992c), lights (Du, Yao 1992) and speed (Carver 1992b); causes of fishing vessel collisions (Deng 2000, Hou 2004); development and implementation of anti-collision measures such as integral control systems on the bridge (Cox, Puckett & Gowen 1977), systems base don hearing and sight (Lodge 1987, Noble 1980), vessel identification systems (Kao, Lee & Ko 2003) and those of crew operation controls (Ren, Yao 2004); structural measures of protection of fishing vessels such as the use of reinforcement in the structure (Evans, Boufounos 1977) and of more protective light elements (Pike 1978); and reports on actions of fishing vessels in collisions (Puthran 1978, Siler 1978); (National Transp. Safety Board 1985).

This work addresses the phenomenon of fishing vessel accidentality by collision from a perspective which is somewhat different from those developed in the above references. The aim is to analyse fishing vessel collisions from the point of view of their spatial and functional distribution. To this end, a twofold objective is set: first, to formalise a methodology for the fishing sector of a country which allows the inequalities in the concentration of collisions to be analysed; and second, to apply this methodology to the Spanish fishing fleet for the period 1994-2002.

The sections below present the formalisation of this methodology, the data used, the results obtained and the main conclusions drawn from the research.

2. METHODOLOGY

Inequality measurements are intended to underline the greater or lesser degree of proximity in the total distribution of the values of a variable in a population. They are thus indicators of the degree of distribution/concentration of the variable. In the field of health science, the research on accidentality makes widespread use of inequality analysis (concentration).

There are several studies on the methodological aspects of measuring inequalities in health. Among the most classical studies are those which analyse and examine the different indices of inequality in health and some others which propose a classification of the indices according to their level of complexity, possible use and the appropriacy of different measurements for the study of inequalities in health (Kunst, Mackenbach 1994, Pamuk, Lenzner & Brackbill 1993, Brown 1994/5, Wagstaff, Paci & van Doorslaer 1991). In more recent works, new indices have been incorporated for measuring health inequalities based on the notion of entropy (Bacallao et al. 2002).

Similarly and in keeping with the spatial analysis of this work, the differences in the health conditions of different geographical zones of one same country have been described (Cook 1990, Csaszi 1990, Kirchgässler 1990, Lahelma, Valkonen 1990).

From all of the inequality measures used concerning health, the Gini coefficient is selected for this work. This coefficient will be used to formalise the general equation which will allow us to determine the collision concentration indices for the fishing sector, for different population variables (fishermen, fishing vessels etc.) and different groups of these (regions, fishing types, etc.).

As an application of the Gini coefficient to the present analysis, equation (1) is the starting point

$$CII = \left(\sum_{i=1}^{k-1} (p_i - q_i) \right) \Big/ \left(\sum_{i=1}^{k-1} p_i \right) \qquad (1)$$

where:

FIC = Gini Coefficient of inequality (concentration) of collisions.

p_i is the accumulated percentage of the at-risk population variable analysed, (X), which has k individuals, and is determined by means of equation (2)

$$p_i = \sum_{i=1}^{i} \left(X_i \Big/ \sum_{i=1}^{k} X_i \right) = \sum_{i=1}^{i} X_i \Big/ \sum_{i=1}^{k} X_i \qquad 2)$$

q_i es el porcentaje acumulado de la variable poblacional accidentada en la colisión (CX) para un grupo que dispone de k individuos, que se determina mediante la ecuación (3).

$$q_i = \sum_{i=1}^{i}\left(CX_i \Big/ \sum_{i=1}^{k} CX_i \right) = \sum_{i=1}^{i} CX_i \Big/ \sum_{i=1}^{k} CX_i \qquad (3)$$

Substituting in (1), we get the general equation (4) by jeans of which the inequality in the concentration of collisions in the fishing sector can be determined for a specific group and period t.

$$CII / X(t) = \left(\sum_{i=1}^{k-1} \left[\left(\sum_{i=1}^{i} X_i \Big/ \sum_{i=1}^{k} X_i \right) - \left(\sum_{i=1}^{i} CX_i \Big/ \sum_{i=1}^{k} CX_i \right) \right] \right) \Bigg/ \left(\sum_{i=1}^{k-1} \left(\sum_{i=1}^{i} X_i \Big/ \sum_{i=1}^{k} X_i \right) \right) \qquad (4)$$

The variables to be used in the construction of the collision concentration indices are shown in Table 1. These are the at-risk population and the injured population variables.

G^a	At-risk Population Variable (X)		Injured Population Variable (CX)	
	Crew Members	Fishing Vessels	Crew Members	Fishing Vessels
1	$CM_1(t)$	$FV_1(t)$	$CCM_1(t)$	$CFV_1(t)$
...	...	...	...	...
I	$CM_i(t)$	$FV_i(t)$	$CCM_i(t)$	$CFV_i(t)$
...	...	...	...	...
K	$CM_k(t)$	$FV_k(t)$	$CCM_k(t)$	$CFV_k(t)$
Total	$\sum_{i=1}^{k} CM_i(t)$	$\sum_{i=1}^{k} FV_i(t)$	$\sum_{i=1}^{k} CCM_i(t)$	$\sum_{i=1}^{k} CFV_i(t)$

Table 1. Variables used in the construction of collision concentration indices

Two different groupings of variables are made in the present work, one corresponding to fishing regions and the other to typ es of fishing. The values of At-risk Population Variable and Injured Population Variable will change depending on the variable selected (Crew Members and Fishing Vessel) the variable groupings and the period (t).

The Gini coefficient is based on the Lorenz curve, which is a function of the accumulated frequency which compares the empirical distribution of a variable with a uniform (of equality) distribution. This uniform distribution is represented by the diagonal y=x. The greater the distance, or more precisely, the area between the Lorenz curve and this diagonal, the greater the inequality.

In its application to the analysis of accidents through collision in the fishing sector, the "p" axis represents the accumulated value of the at-risk population variable (censuses of fishermen and vessels) and the "q" axis the accumulated value of the injured population variable (fishermen injured and vessels collided.

3. DATA

The methodology described above will be applied to the Spanish fishing sector for the years 1994, 2002 and 2004 in order to determine the concentration of sinking accidentality and its evolution, per region and types of fishing. In Spain, the fishing fleet is distributed by base ports through 9 coastal Autonomous Communities (The Basque Country, Cantabria, Asturias, Galicia, Andalusia, Murcia, Valencia, Catalonia, Balearic Islands, Canary Islands) and two Autonomous Cities (Ceuta and Melilla).

In Spain, the sea accident statistics are drawn up by the Merchant Navy Council of The Ministry of Industrial Development. The data available is structured according to types of vessel (merchant, fishing, recreational), accident (collision, list, structural failure, technical failure, sinking , fire-explosion, grounding, flooding and others), cause of accident (human failure, material failure, bad weather and unknown), denotation of damage (total loss, man over board, hull damage, machine damage and others), and aspects of personal and material damage. As for the technical characteristics of the damaged vessel, the statistics only incorporate the general characteristics and a vessel identification code. Thus, any indicator of accidentality in the Spanish fishing sector is drawn up from these sources.

Since we intend to develop our analysis in the regional context and for types of fishing, the above data is insufficient as it does not incorporate these references. Thus, we have turned to the fishing censuses drawn up by the Secretary General of Fishing of the Ministry of Agriculture, Fishing and Food. As well as technical data on the fishing boats, these censuses incorporate the type of fishing (demersal trawl, gillnet, purse seine, long line and complementary activities) and the base port. From these two sources, a data base has been created with Microsoft Access with two tables linked by the common register of the vessel identification code

4. RESULTS

The concentration indices for collision accidents for the Spanish fishing sector for the period (1994-2002) are shown in Table 2. These have been structured by region (Autonomous Communities) and type of fishing, for the at-risk variables analysed: fishermen (crews) and fishing vessels.

The indices refer to crew members injured in collisions and collisioned Vessels.

4.1. Fishermen Accidentality Concentration

By Autonomous Community for the period analysed (1994-2002), the concentration of injured crews of Spanish fishing vessels increased by 74.57%, as can be

Year	Indices by Regions		Indices by Fishing Types	
	Fishermen	Fishing Vessels	Fishermen	Fishing Vessels
1994	0,346	0,352	0,587	0,681
1998	0,389	0,460	0,449	0,326
2002	0,604	0,597	0,616	0,493

Table 2. GINI collisions concentration indices for Spanish fishing sector for the period (1994-2002) by region and type of fishing.

observed from the variations in the fishermen affected by regions indices (Table 2). The associated Lorenz curves in Figure 1 show the distribution of accidentality by regions. These have been ordered from higher to lower accidentality. For the last year analysed, the regions of Murcia, Valencia, Cantabria and Galicia with a number of crews that make up 48.82% of the total census of Spanish fishermen, suffered 94.78% of the total number of accidents by collision of Spanish fishing vessels.

By fishing types, the degree of concentration of crew accidents varied for the years analysed, but for the whole period (1994-2002) it increased by 4.94%, as shown by the changes in the indices of fishermen affected by fishing types indices (see Table 2). The Lorenz curves in Figure 2 show the distribution of accidentality by fishing type for the Spanish fleet. As for the regions, the fishing types have been ordered from higher to lower accidentality. In the last year, fishermen working in the dermesal trawl and purse seine fishing types, which make up 42.63% of the country's fisherman population, suffered 84.35% of the total crew accidents by collision of the Spanish fleet.

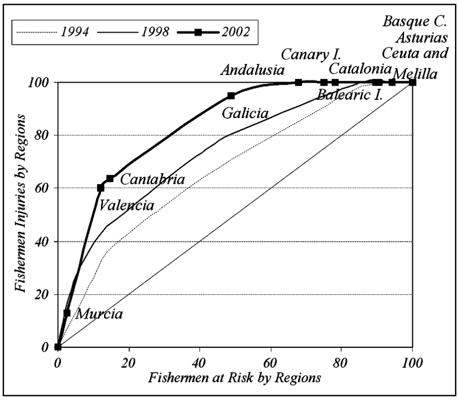

Fig. 1. Lorenz curves for Spanish fishermen injured in collisions, by region. The values in the two axes represent the accumulated % of the variables.

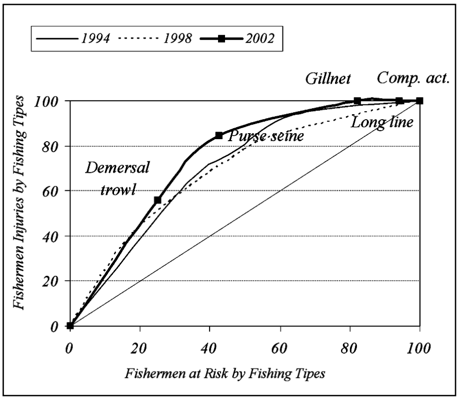

Fig. 2. Lorenz curves for Spanish fishermen injured in collisions, by fishing type. The values in the two axes represent the accumulated % of the variables.

4.2. Concentration of fishing vessel accidentality

By Autonomous Communities, the number of Spanish fishing vessels suffering collisions in the period analysed (1994-2002) increased its concentration by 69.60%, as shown by the variations in the indices for collisioned vessels by Regions (see Table 2). The distribution of accidentality by collisions shown by the Lorenz curve for the final period analysed (see Fig.3), indicates that 27.43% of the Spanish fishing fleet accounted for 75% of the collisions. The regions bearing the brunt of this accidentality were Valencia, Cantabria, Murcia and Andalusia.

By fishing types, the degree of concentration of collisions in the fleet varied greatly for the period analysed (1994-2002), although overall there was a reduction of 27.6%, as shown by the variations in the indices for vessels collisioned by Fishing Types (see Table 2). The Lorenz curve for the last period shows the distribution of fishing vessel collisions by fishing types for the Spanish fleet (Fig. 4). The fishing vessels working in dermesal trawl and purse seine fishing types, which make up 20.34% of the overall census for the country, accounted for 66.67% of the collisioned vessels.

This is due to the fact that the number of vessels dedicated to Gillnet and Purse Seine fishing make up 77.49% of the total Spanish fishing fleet. The use of a measurement of fleet size or capacity would probably eliminate this disparity since these vessels are smaller than the average of the fleet.

CONCLUSIONS

The indices of accidentality concentration by collision and their associated Lorenz curves prove to be a useful instrument in the analysis of the accidentality concentration of a country's fishing sector. Moreover, these tools could be applied to

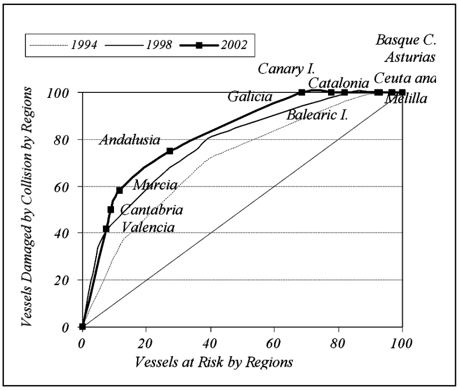

Fig. 4. Lorenz curves for collisioned Spanish fishing vessels, by regions.

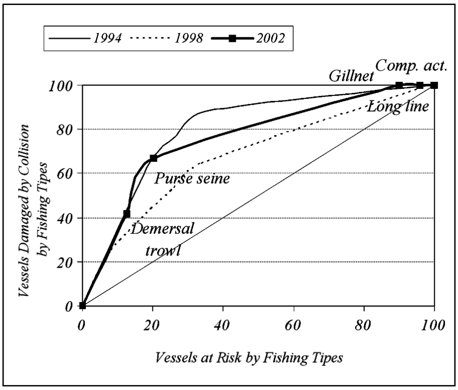

Fig. 5. Lorenz curves for collisioned Spanish fishing vessels by fishing types.

any other similar analysis in the spatial (local, national and supranational) and functional domain (vessel type, transport type, etc.) or other type of accident.

In Spain, in the period 1994-2002, a notable increase can be appreciated in the disparities in the regional distribution of collisions, both for fishermen and for fishing vessels, the Communities of Valencia, Murcia and Cantabria being those that suffered or concentrated the highest percentage of collisions.

By fishing types, for the same period 1994-2002, the inequality increased slightly with respect to fishermen injured in collisions and decreased significantly for fishing vessels. The highest rates of accidentality by collision both for fishermen and for the Spanish fishing fleet, took place in the fishing modalities of dermesal trawl and purse seine.

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COLISIONES DE BUQUES PESQUEROS: CURVAS DE LORENZ E ÍNDICES DE GINI

Resumen

Las colisiones de pesqueros son accidentes que en la mayoría de los casos desencadenan posteriormente otros accidentes como la explosión, la vía de agua e incluso el hundimiento. Se trata de accidentes que suelen tener fatales consecuencias para las tripulaciones y en numerosos casos suponen la pérdida del buque. Tales siniestros pueden evitarse o reducirse si los pesqueros cuentan con medios tecnológicos suficientes y las tripulaciones actúan correctamente y de forma inmediata. En tal sentido, los países deben incorporar en la ordenación de su sector pesquero los mecanismos de control adecuados que garanticen unos buques correctamente pertrechados de recursos y unas tripulaciones bien formadas en seguridad. Dichos recursos y su distribución deben asignarse de acuerdo con las necesidades de la flota, lo que requiere un conocimiento pormenorizado del grado de concentración de la siniestralidad por colisiones del sector pesquero del país.

En dicho contexto, el objetivo del trabajo que se presenta es doble. Por un lado, formalizar una metodología para el sector pesquero de un país, que permita analizar la desigualdad en la concentración de colisiones. Por otro, aplicar dicha metodología a la flota pesquera española en el período 1994-2002. Así se construyen índices por regiones y tipos de pesca, para dos variables: pescadores y buques. En dichos ámbitos, se obtienen resultados tanto de la concentración de colisiones como de las curvas de Lorenz asociadas. Se verifica un incremento de la desigualdad en la distribución espacial y funcional de dichos siniestros.

INTRODUCCIÓN

La ordenación de una flota pesquera debe incorporar mecanismos de control adecuados que garanticen unos buques correctamente pertrechados con recursos suficientes y unas tripulaciones bien formadas en seguridad. Si no se cumplen tales condiciones las consecuencias pueden ser fatales. En tal sentido, la mejora de la seguridad en el mar ha constituido durante varios decenios una de las principales preocupaciones de diversas instituciones supranacionales, administraciones nacionales y organizaciones no gubernamentales. Sin embargo, las estadísticas de accidentes indican que la pesca comparada con otros sectores ha sido y sigue siendo una de las actividades humanas más peligrosas.

Los trabajos de investigación sobre seguridad y siniestralidad en buques pesqueros comerciales son escasos. En los mismos son tratados aspectos relacionados con daños personales y/o materiales originados por la actividad pesquera, tales como:

la mortalidad de pescadores en los accidentes (Reily, 1985); equipos, medios de salvamento y percepción sobre normas de seguridad de buques pesqueros (Lagares, 1990; Poggie et al., 1995); las enfermedades y accidentes de pescadores (Goethe & Vuksanovic, 1995); la cognición del peligro y las actitudes de los pescadores en la seguridad al gestionar pesquerías (Poggie et al., 1996; Kaplan, & Kite-Powell, 2000); los daños a pescadores en el trabajo (Jansen, 1996); los riesgos y aspectos determinantes de las pérdidas personales y materiales en buques pesqueros (Dyer, 2000; Jin et al., 2001); y la formulación de modelos de probabilidad de accidentes en pesqueros (Jin et al., 2002).

En el presente trabajo centramos la atención en las colisiones de pesqueros. Se trata de un tipo de siniestro que suele desencadenar otros accidentes posteriores como la explosión, vía de agua y el hundimiento, con consecuencias fatales para las tripulaciones y la pérdida del buque. Las investigaciones específicas sobre colisiones de pesqueros son muy escasas y suelen analizar diversas cuestiones, como:

Aspectos del reglamento internacional de abordajes como los concernientes al alcance, la navegación en canales angostos, las luces y la velocidad.

Causas que originan las colisiones de pesqueros.

Desarrollo e implantación de medidas anticolisión como los sistemas de control integral en el puente, los sistemas basados en la audición y la visión, los sistemas de identificación de buques y los de control de operaciones de las tripulaciones.

Medidas estructurales de protección en la construcción de los pesqueros como la utilización de refuerzos en la estructura y de elementos ligeros de mayor protección.

Informes sobre actuaciones de los pesqueros en colisiones.

En este trabajo se aborda el fenómeno de la siniestralidad por colisiones de pesqueros desde una óptica diferente a la desarrollada en las referencias indicadas. Se pretende analizar las colisiones de los buques pesqueros desde el punto de vista de su distribución espacial y funcional. Con tal finalidad nos planteamos un doble objetivo. Por un lado, formalizar una metodología para el sector pesquero de un país, que permita analizar la concentración de colisiones, por regiones pesqueras y tipos de pesca. Por otro, aplicar dicha metodología a la flota pesquera española en el período 1994-2002.

En los epígrafes siguientes se presentan la formalización de la metodología, los datos utilizados, los resultados obtenidos y las conclusiones generales de la investigación desarrollada.

METODOLOGÍA

Las medidas de desigualdad tratan de poner de relieve el mayor o menor grado de proximidad en la distribución total de los valores de una variable en una población. Por tanto, son indicadores del grado de concentración de la variable. En el campo de las ciencias de la salud, las investigaciones sobre siniestralidad utilizan de forma habitual el análisis de desigualdad (concentración).

Existen numerosos estudios sobre aspectos metodológicos para la medición de las desigualdades en salud. Entre los trabajos clásicos más conocidos están los que analizan y examinan los diferentes índices de desigualdad en salud y aquellos otros que proponen una clasificación de los índices según su nivel de complejidad, posibles usos e idoneidad de las diferentes medidas para el estudio de las desigualdades en salud. En trabajos más recientes se han incorporado nuevos índices para medir las desigualdades de salud basados en la noción de entropía.

Asimismo y en línea con el ámbito espacial de análisis en que se centra el presente trabajo, se han descrito diferencias en las condiciones de salud entre diferentes zonas geográficas de un mismo país. Entre todas las medidas de desigualdad utilizadas en salud, en el presente trabajo nos centramos en el coeficiente de Gini. A partir del mismo se formaliza la expresión general que nos permita determinar índices de concentración de colisiones en el sector pesquero, para distintas variables poblacionales (pescadores, buques, etc.) y diferentes agrupaciones de las mismas (regiones, tipos de pesca, etc.).

CONCLUSIONES

Los índices de concentración de siniestralidad por colisiones y las curvas de Lorenz asociadas a los mismos se muestran como un instrumento útil en el análisis de la concentración de siniestralidad del sector pesquero de un País. Además dichas herramientas podrían aplicarse a cualquier otro análisis similar de ámbito espacial (local, nacional y supranacional) y funcional (tipo de buque, tipo de transporte, etc.) u otro tipo de siniestro.

En España, en el período 1994-2002, se aprecia un incremento importante de las disparidades en la distribución regional de las colisiones, tanto para los pescadores siniestrados como para los buques pesqueros, siendo las Comunidades de Valencia, Murcia y Cantabria, las que aglutinaron o concentraron un mayor porcentaje de las colisiones.

Por tipos de pesca, en el mismo período 1994-2002, la desigualdad se incrementa ligeramente respecto a los pescadores accidentados en las colisiones y se reduce sensiblemente respecto a los buques. Las mayores tasas de siniestralidad por colisiones se, tanto de los pescadores como de la flota pesquera española, se originaron en las modalidades de arrastre y cerco.

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- ☐ Notes, when necessary, are to be placed at the end of the paper and numbered in their order of appearance in the text. Mathematical derivations should not be included in these endnotes.

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- Methods
- Development (application and results)
- Conclusions
- Endnotes
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The way in which *bibliographic citations* are included in the text will depend on the context and the composition of the paragraph and will have one of the following forms:

- One author: Farthing (1987); (Farthing, 1987); (Farthing, 1987 pp. 182-5)
- Several authors: Goodwin and Kemp (1979); Ihere, Gorton y Sandevar (1984); Ihere et al.(1984); (Ihere et al., 1984)

The *bibliographic references* are to be arranged in alphabetical order (and chronologically in the case of several works by the same author), as is indicated in the following examples:

Books

Farthing, B. (1987) *International Shipping.* London: Lloyd´s of London Press Ltd.

Chapters of books

Bantz, C.R. (1995): Social dimensions of software development. In: Anderson, J.A. ed. *Annual review of software management and development*. Newbury Park, CA: Sage, 502-510.

Journal articles

Srivastava, S. K. and Ganapathy, C. (1997) Experimental investigations on loop-manoeuvre of underwater towed cable-array system. *Ocean Engineering* 25 (1), 85-102.

Conference papers and communications

Kroneberg, A. (1999) Preparing for the future by the use of scenarios: innovation shortsea shipping, *Proceedings of the 1st International Congress on Maritime Technological Innovations and Research,* 21-23 April, Barcelona, Spain, pp. 745-754.

Technical Reports

American Trucking Association (2000) *Motor Carrier Annual Report*. Alexandria, VA.

Doctoral theses

Aguter, A. (1995) *The linguistic significance of current British slang*. Thesis (PhD).Edinburgh University.

Patents

Philip Morris Inc., (1981). *Optical perforating apparatus and system*. European patent application 0021165 A1. 1981-01-07.

Web pages and electronic books

Holland, M. (2003). *Guide to citing Internet sources* [online]. Poole,
Bournemouth University. Available from:
http://www.bournemouth.ac.uk/library/using/guide_to_citing_internet_sourc.html
[Accessed 1 November 2003]

Electronic journals

Storchmann, K.H. (2001) The impact of fuel taxes on public transport — an empirical assessment for Germany. *Transport Policy* [online], 8 (1), pp. 19-28 . Available from:
http://www.sciencedirect.com/science/journal/0967070X
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