VOL.V No 1 APRIL 2008

JMR

JOURNAL OF MARITIME RESEARCH

Spanish Society of Maritime Research

JMR JOURNAL OF MARITIME RESEARCH

R. Hanzu-Pazara, E. Barsan, P. Arsenie, L. Chiotoroiu and G. Raicu

REDUCING OF MARITIME ACCIDENTS CAUSED BY HUMAN FACTORS USING SIMULATORS IN TRAINING PROCESS

J. A. Orosa, A. Baaliña and G. Iradi

WORK RISK MEASURES IN SEVER ENVIRONMENTS OF A SHIP

E. Melón, S. Iglesias, A. Bermejo and H. Sánchez

SEABED MAPPING

E. Eguía, A. Trueba and M. M. Milad

DESIGN AND SIMULATION OF A VIRTUAL TUBULAR HEAT EXCHANGE UNIT FOR EDUCATIONAL APPLICATIONS

T. Perez and T. I. Fossen

A DERIVATION OF HIGH-FREQUENCY ASYMPTOTIC VALUES OF 3D ADDED MASS AND DAMPING BASED ON PROPERTIES OF THE CUMMINS' EQUATION

C. S. Chas and R. Ferreiro

INTRODUCTION TO SHIP DYNAMIC POSITIONING SYSTEMS

VOL.V N.1
APRIL 2008

CONTENTS

Reducing of Maritime Accidents Caused by Human Factors Using Simulators in Training Process
R. Hanzu-Pazara, E. Barsan, P. Arsenie, L. Chiotoroiu and G. Raicu — 3

Work Risk Measures in Sever Environments of a Ship
J. A. Orosa, A. Baaliña and G. Iradi — 19

Seabed Mapping
E. Melón, S. Iglesias, A. Bermejo and H. Sánchez — 35

Design and Simulation of a Virtual Tubular Heat Exchange Unit for Educational Applications
E. Eguía, A. Trueba and M. M. Milad — 47

A Derivation of High-frequency Asymptotic Values of 3D Added Mass and Damping Based on Properties of The Cummins' Equation
T. Perez and T. I. Fossen — 65

Introduction to Ship Dynamic Positioning Systems
C. S. Chas and R. Ferreiro — 79

Journal of Maritime Research, Vol. V. No. 1, pp. 3-18, 2008

ISSN: 1697-4840

REDUCING OF MARITIME ACCIDENTS CAUSED BY HUMAN FACTORS USING SIMULATORS IN TRAINING PROCESS

R. Hanzu-Pazara[1], E. Barsan[2], P. Arsenie[3], L. Chiotoroiu[4] and G. Raicu[5]

Received 30 April 2007; received in revised form 10 May 2007; accepted 14 January 2008

Abstract

Given the increasing prevalence of automated systems on board ships, it is important that the human element is considered throughout their design, implementation and operational use. Automation can be beneficial to operators of complex systems in terms of a reduction in workload or the release of resources to perform other onboard duties. However, it can also potentially be detrimental to system control through increasing the risk of inadvertent human error leading to accidents and incidents at sea.

A team of researchers from our University had participated at a study together with students, study which wants to release the dangerous situation on sea based on human factors. In this scope has been used a web base simulator, bridge and liquid cargo handling simulators, developing applications in navigation and ship handling area, with different grades of difficulty and risk. These applications brings the future deck officers in usually situations on board, forced to use the present navigation technology and study their options and reactions in these cases, focus on situations with risk of errors, human errors, appearance.

Keywords: Bridge technology, human factors, simulation, Bridge management.

[1]Lecturer Nautical Sciences, Constantza Maritime University (raduhanzu@yahoo.com), Constantza, Romania. [2]Vice Dean Navigation Faculty, Constantza Maritime University (ebirsan@internav.com), Constantza, Romania. [3]Head of Navigation Department, Constantza Maritime University (arseniep@yahoo.com), Constatnza, Romania. [4]Professor Naval Arhitecture, Constatnza Maritime University (laurentiu_chiotoroiu@yahoo.com), Constantza, Romania. [5]Lecturer of Nautical Sciences, Constantza Maritime University (graicu@imc.ro), Constantza Romania.

INTRODUCTION

Over the last 40 years or so, the shipping industry has focused on improving ship structure and the reliability of ship systems in order to reduce casualties and increase efficiency and productivity. We've seen improvements in hull design, stability systems, propulsion systems and navigational equipment. Today's ship systems are technologically advanced and highly reliable.

With all of these the maritime incidents rate is still high, we have not significantly reduced the risk of accidents. There is because ship structure and system reliability are a relatively small part of the safety equation. The maritime system is a people system, and human errors figure prominently in these situations. About 75-96% of marine accidents are caused, at least in part, by some form of human error. Studies have shown that human error contributes to:

- 89-96% of collisions
- 75% of fires and explosions
- 79% of towing vessel groundings
- 84-88% of tankers accidents
- 75% of allisions

Therefore, if we want to make greater strides toward reducing marine accidents, we must begin to focus on the types of human errors that cause these and their relationship with the new technology on board.

In recent years a problem in international maritime training became obvious: the lack of experiential learning of entry-level officers or "lost apprenticeship" (Chiotoroiu L., et all 2006). Like in many other technical work systems the work processes on board have been an object of extensive automation. Human beings on board modern ships are needed predominantly for planning, control and supervision. However, in critical and unusual situations they have to step in actively. Such situations require flexible problem solving, improvisation and intuition. Furthermore, decreasing number of (experienced) crewmembers on board and the pressure of fast promotion into responsible positions have increased the "experiential learning gap' of junior-officers. This problem is most obvious in tanker shipping, due to the demands on junior officers which are higher in this branch of our industry as compared for example with the field of container shipping.

HUMAN FACTORS

Most people would agree with the old adage "to err is human". Most too would agree that human beings are frequent violators of the "rules" whatever they might be. But violations are not all that bad, through constant pushing at accepted boundaries they got us out of the caves!

As it is inevitable that errors will be made, the focus of error management is placed on reducing the chance of these errors occurring and on minimizing the impact of any errors that do occur. In large scale disaster, the oft-cited cause of "human error" is usually taken to be synonymous with "operator error" but a measure of responsibility often lies with system designers.

To find a human error is necessary to identify active and latent failure in order to understand why mishaps occur and how it might be prevented from happening again in the future.

As described by Reason, active failures are the actions or inactions of operators that are believed to cause the accident (Wagenaar W.A. and Groeneweg J., 1987).

In contrast, latent failures are errors committed by individuals elsewhere in the supervisory chain of command that effect the tragic sequence of events characteristic of an accident.

The question for mishap investigators and analysts alike, is how to identify and mitigate these active and latent failures. One approach is the "Domino Theory", which promoted the idea that like domino's stacked in sequence, mishaps are the end result of a series of errors made throughout the chain of command.

Violations and errors

Assuming that the rules, meaning safe operating procedures, are well founded, any deviation will bring the violator into an area of increased risk and danger. The violation itself may not be damaging but the act of violating takes the violator into regions in which subsequent errors are much likely to have bad outcomes.

This relationship can be summarized quite simply by the equation:

Violations + errors = injury, death and damage

The resultant situation can sometimes be made much worse because persistent rule violators often assume, somewhat misguidedly, that nobody else will violate the rules, at least not at the same time! Violating safe working procedures is not just a question of recklessness or carelessness by those at the sharp end.

Factors leading to deliberate non-compliance extend well beyond the psychology of the individual in direct contact with working hazards and include such organizational issues as:

— the nature of the workplace
— the quality of tools and equipment (IMO, 2000)
— whether or not supervisors or managers turn a "blind eye" in order to get the job done
— the quality of the rules, regulations and procedures
— organization's overall safety culture, or indeed its absence.

Unsafe acts and preconditions

A brief description of the major components and causal categories follows beginning with the level "nearest" the accident – unsafe acts.

The unsafe acts committed by operators generally take on two forms, errors and violations. The unsafe acts operators commit can be classified among three basic errors types and two forms of violations.

The basic error forms are:

- Decision Errors – this is one of the more common error forms, represent the actions or in-actions of individuals whose heart is in the right place, but they either did not have the appropriate knowledge available or just simply chose poorly.
- Skill-based Errors – is best described as those basic operating skills that occur with little or so significant conscious thought, are particularly vulnerable to failures of attention and/or memory.
- Perceptual Errors – when your perception of the world is different then reality, errors can, and often do, occur, like visual illusions or spatial disorientation.

Violations in general are defined as the willful departure from authority and are two distinct types of violations, as:

- Routine/Infractions – tend to be routine/habitual by nature constituting a part of the individual's behavioural repertoire and can be further broken down in: routine violations, optimizing violations and situational violations.
- Exceptional – appear as isolated departures from authority, not necessarily indicative of an individual's typical behavior pattern, nor is it condoned by management.

Preconditions for unsafe acts are described by two major subdivisions: substandard conditions of operators and substandard practices of operators.

The substandard conditions of operators are categorized as:

- Adverse Mental States – those mental conditions that affect performance;
- Adverse Physiological States – those medical or physiological conditions that preclude safe operations;
- Physical/Mental Limitations – those instances when the task requirements exceed the capabilities of the individual at the controls.

The substandard practices of operators are categorized as:

- Crew Resource Mismanagement – often the substandard practices of the team will lead to the conditions and unsafe acts;
- Personal Readiness.

Till now we presented the errors as individual error. But, in most cases the work is done in a team organization that means many persons.

Team error is one form of human error. The difference is that team error considers how a group of people made human errors when they work in a team or a group. Then we can define team error as human error made in group processes. Reason categorized human errors into three types: mistakes, lapses and slips. Mistakes and lapses arise in the planning and thinking process, whereas action slips emerge primarily out of these execution processes. Mistakes and lapses are more likely to be associated with group processes. Slips are errors in the action process of a single individual and are likely to be divorced from the activities of the team as whole.

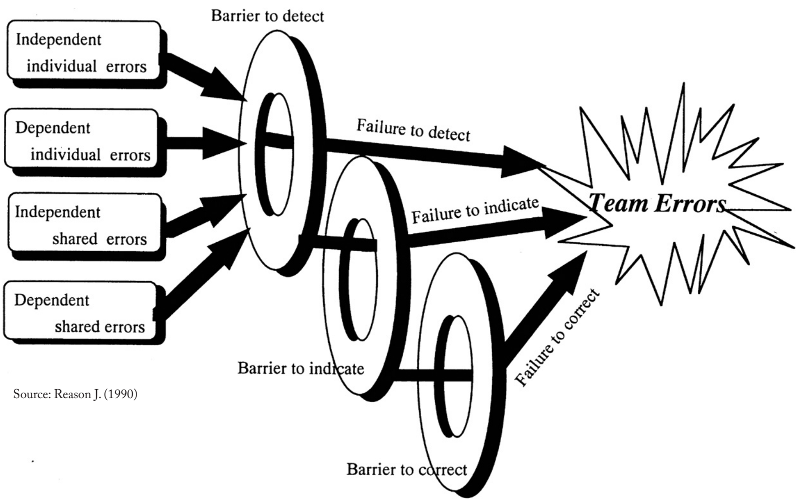

Figure 1. Team error process.

The error making process

Individual errors – are errors which are made by individuals. That is, an individual alone makes an error without the participation of any other team member. Individual errors may be further sub-divided into independent errors and dependent errors. Independent errors occur when all information available to the perpetrator is essentially correct. In dependent errors, however, some part of this information is inappropriate, absent or incorrect so that the person makes an error unsuitable for a certain situation.

Shared errors – are errors which are shared by some or all of the team members, regardless of whether or not they were in direct communication. Like individual errors, shared errors may also be sub-divided into two categories: independent and dependent.

The error recovery process

The error recovery process may fall into any one of three stages: detection, indication and correction.

1. Failure to detect – the first step in recovering errors is to detect their occurrence. If the remainders of the team do not notice errors, they will have no chance to correct them. Actions based on those errors will be executed.
2. Failure to indicate – once detected, the recovery of an error will depend upon whether team members bring it to the attention of the remainder. This is the second barrier to team error making. An error that is detected but not indicated will not necessarily be recovered and the actions based on those errors are likely to be executed.
3. Failure to correct – the last barrier is the actual correction of errors. Even if the remainder of the team notices and indicates the errors, the people who made the errors may not change their minds. If they do not correct the errors, the actions based on those errors will go unchecked.

ORGANISATIONAL FACTORS

The training of operators can only ever be part of the solution to reducing accidents. Organisational factors also play a significant part in accident causation.

The analysis of human factors in accident causation is still relatively immature in the maritime world (U.K. P & I Club, 1992). Although databases held by insurers and classification societies do include human error taxonomies, little analysis is undertaken to identify trends or patterns. Even less analysis has been attempted in assessing the significance or frequency of organisational factors such as the incidence of commercial pressure or the effects of organisational culture on accident causation.

The differences in organizational culture between shipping companies are a well known phenomenon, but there has been little work on understanding the effects of organizational culture on safe and efficient performance. In much the same way as we are striving to identify a set of behavioral markers to assess the competence of individuals, so there is a need to establish a set of organizational metrics to determine the competence of shipping companies to perform safely.

Not enough is known about the parameters governing functioning and performance of management systems. There is a little research evidence to indicate what makes a management system work or indeed what prevents it from working. Equally, not enough is known about the metrics that enable the status of a management system to be determined. Ideally, what is required is a set of "leading" indicators that will predict future performance so that interventions can be made before accidents occur.

The research conundrum is, first, to agree what constitutes organizational behaviour; second, in deciding which "behaviours" are leading indicators of profiency and third, in designing methods that can measure these indicators accurately.

HUMAN FACTORS ISSUES IN MARITIME INDUSTRY

What are some of the most important human factors challenges facing the maritime industry today? A study by U.S. Coast Guard found many areas where the industry can improve safety and performance through the application of human factors principles. The three largest problems were fatigue, inadequate communication and coordination on navigational bridge, and inadequate technical knowledge. Below are summaries of these and other human factors areas that need to be improved in order to prevent accidents (Huey, D. et all, 1993),.

Fatigue, has identified to be an important cross-modal issue, being just as pertinent and in need of improvement in the maritime industry as it is in the aviation, rail and automotive industries. Fatigue has been cited as the "number one" concern of mariners in different studies. It was also the most frequently mentioned problem in a recent Insurance and Classification Society survey. A new study has objectively substantiated these anecdotal fears: in a study of critical vessel casualties and personnel injuries, it was found that fatigue contributed to 16% of the vessel casualties and 33% of the injuries (IMO, 1999).

Inadequate Communications. Another area for improvement is communications between shipmates, between masters and pilots, ship-to-ship, and ship-to-VTS. Is stated that 70% of major marine collisions and allisions occurred while a pilot was directing one or both vessels. Better procedures and training can be designed to promote better communications and coordination on and between vessels. Bridge Resource Management is a first step towards improvement.

Inadequate General Technical Knowledge. This problem is responsible for 35% of accidents. The main contributor to this category is a lack of knowledge of the proper use of technology, such as radar and electronic charts. Mariners often do not understand how the automation works or under what set of operating conditions it was designed to work effectively. The unfortunate result is that mariners sometimes make errors in using the equipment or depend on a piece of equipment when they should be getting information from alternate sources.

Inadequate Knowledge of Own Ship System. A frequent contributing factor to marine casualties is inadequate knowledge of own ship operations and equipment. Several studies and accidents report have warned of the difficulties encountered by crews who are constantly working on ships of different sizes, with different equipment, and carrying different cargoes. The lack of ship-specific knowledge was cited as a problem by 78% of the mariners surveyed (McCallum M.C., et all, 1996). A combination of better training, standardized equipment design and an overhaul of the present method of assigning crew to ships can help solve this problem.

Poor Design of Automation. One challenge is to improve the design of shipboard automation. Poor design pervades almost all shipboard automation, leading to collision from misinterpretation of radar display, oil spills from poorly designed overfill devices, and allisions due to poor design of bow thrusters. Poor equipment design is cited as a causal factor in one-third of major marine accidents (Perrow C., 1984). The solution is relative simple: equipment designers need to consider how a given piece of equipment will support the mariner's task and how that piece of equipment will fit into the entire equipment "suite" used by the mariner. Human factors engineering methods and principles are in routine use in other industries to ensure human-centered equipment design and evaluation.

Decisions Based on Inadequate Information. Mariners are charged with making navigation decisions based on all available information. Too often, we have a tendency to rely on either a favored piece of equipment or our memory. Many accidents result from the failure to consult available information, such as than from a radar or an echo-sounder. In other cases, critical information may be lacking or incorrect, leading to navigation errors, for example, bridge supports often are not marked or buoys may be off-station.

Faulty Standards, Policies or Practices. This is an oft-cited category and covers a variety of problems. Included in this category is the lack of available, precise, written and comprehensible operational procedures aboard ship, for example, if something goes wrong, and if a well-written manual is not immediately available, a correct and timely response is much less likely. Other problems in this category include management policies which encourage risk-taking, like pressure to meet schedules at all costs and the lack of consistent traffic rules from port to port.

Poor Maintenance. Poor maintenance can result in a dangerous work environment, lack of working backup systems and crew fatigue from the need to make emergency repairs. Poor maintenance is also a leading cause of fires and explosions.

Hazardous Natural Environment. The marine environment is not a forgiving one. Currents, winds and fog make for treacherous working conditions. When we fail to incorporate these factors into the design of our ships and equipment and when we fail to adjust our operations based on hazardous environment conditions, we are at greater risk for accidents.

SIMULATORS AND THE PROCESS OF TRAINING

The use of simulation in providing solutions to the problems of risk and crisis management and the optimal use of crew resources has a long established pedigree in maritime training (Barnett, M.L. et all 2002).

The early simulators consisted of real radars, located in a set of cubicles, and fed with simulated signals. Individuals or teams could learn the skills of radar plotting under the guidance of an instructor working at a separate master console. Other navigational aids in the simulator were fairly basic and certainly did not include a visual scene.

Bridge simulators with a nocturnal visual scene made their appearance later and allowed teams to conduct simulated passages in a realistic environment but with only a few lights available to indicate other vessels and shore lights.

Simulator-based training courses were introduced primarily to train the skills of passage planning and the importance of the Master/Pilot relationship (Hensen H., 1999). This training initiative developed into the Bridge Team Management courses that are conducted today on many simulators world-wide and contain many of the elements to be found in Crew Resource Management courses developed in other industries, such as aviation. These courses were developed to focus on the non-technical skills of flight operations and include group dynamics, leadership, interpersonal communications and decision making.

Bridge Resource Management courses are a more recent initiative, adapted directly from the aviation model for training the non-technical skills of resource management, and are not always based on the use of simulators.

The 1980s saw the introduction of Engine Room simulators and towards the end of that decade, cargo operations simulators also became available. These types of simulator have primarily been used to train officers in the handling of operations, including fault finding and problem diagnosis, and increasingly to train teams in the skills of systems, resource and risk management.

Many types of simulator: bridge, engine and cargo control room, have tended to emphasise a physically realistic environment in which these exercises occur, although the of PC-based simulators for training some tasks is increasingly widespread.

In some parts of the world, simulators have been developed which have very high levels of physical fidelity, for example, multi-storey engine room mock-up and bridge simulators including features such as 360 degrees day/night views, pitch and roll, and full vibration and noise effects.

The only mandatory requirements in the maritime domain for the development of the non-technical skills of crisis management are those of the International Maritime Organization's (IMO) Seafarer's Training, Certification and Watchkeeping Code (International Maritime Organization, 1995). Table A-V/2 of this code specifies the minimum standard of competence in crisis management and human behaviour skills for those senior officers who have responsibility in emergency situations.

The competence assessment criteria detailed within the Code are not based on specific overt behaviours, but rather on generalized statements of performance outputs, and as such are highly subjective and open to interpretation.

Although these standards of competence indicate that IMO recognizes the need for non-technical management skills, both the standards and their assessment criteria are immature in comparison with the understanding of non-technical skills, and their assessments, within an industry such as civil aviation.

Simulation and modeling

Sea trials are a poor method of investigating human performance issues. Firstly, crisis situations cannot be replicated safely in live systems. Secondly, only a limited number of people can participate in a sea trial making the observed results difficult to extrapolate to entire marine community. Thirdly, it is difficult to control for variables in sea trials making it very difficult for investigators to identify cause-and-effect relationship. As a result of these three shortcomings, new technology and processes are often introduced with little knowledge of its impact on human performance.

The work being done to improve simulation technology has resulted in the growth of open systems that are modular and recyclable. The efforts to define High Level Architecture and the development of readily available Run Time Infrastructure are examples of development in simulation technology. We believes that such developments will ultimately reduce the costs associated with simulation projects in the maritime sector making simulation and modeling more accessible for commercial applications.

The lesson to be learned from the other sectors, like aviation and military, is that simulation has a much wider application than training in the reduction of human error. Simulation is aggressively being pursued as a tool to address latent human factors issues that are present in the general failure types. The maritime community has not picked up on simulation to the same extent as the other sectors.

Although simulation is now a mandatory component in mariner training , the use of simulation to address the human factors in maritime operations is sporadic and haphazard. As far as human performance is concerned, the maritime industry continues to rely upon "trial-and-error" when implementing new technology or work processes.

The following examples illustrate the potential benefits of insertion simulation and modeling into the maritime innovation cycle.

An example of how simulation and modeling could reduce the latent error in maritime systems would be mission rehearsal. The mission rehearsal process could be used to evaluate the impact of new technology or doctrine on human performance, and result in improved design; effective operational and contingency plans; efficient publicly funded infrastructure; and, improved regulatory controls. To use mission rehearsal, the operating conditions would need to be modeled, and then professionals would participate in a series of simulations.

Simulation and modeling, in this sense, represents a tool to generate "artificial experience" that would significantly improve professional judgment in the consultation process, especially with respect to human performance.

Mitigating the impact of psychological precursor to human error involves using modeling and simulation to observe and quantify human performance of mariners, particularly their cognitive performance in operational context.

It would be important, for example, to conduct baseline studies to determine the stress absorbing capacity of mariners under a variety of operational conditions. The baseline studies would serve to highlight operations and situations where over-simulation is likely to occur.

Another important area of research would to be look at the fatigue induced by highly integrated displays. Most automated display systems have the potential to be integrated with other displays. The question becomes one of the degree to which these displays should be integrated in order to maintain optimal human performance.

Error detection and correction, as well as crisis management, are the last defences in the human error chain. At the present time, maritime simulators are used to train mariners how to detect and correct errors, and to a limited extent on how to manage a crisis.

The scope and amount of simulator based training in crisis management needs to increase. Areas such as initial actions to an spill oil and a distress incident need to be incorporated into simulation training programs. Simulation based training in the corporate emergency procedures also needs to be conducted to ensure that mariners are not reaching for the emergency manuals to find out what to do once an incident happens.

An improved understanding of how crisis management is conducted in the maritime environment is also required to the development of effective decision support systems.

Simulation and modeling capabilities needed for innovation

The use of simulation and modeling in the innovation cycle demand a higher degree of flexibility in simulation technology than required for the training function. Simulators need to be able to accept input from a variety of model data, and need to be able to interact with other simulators in unusual and unique situations.

Open systems with modular and recyclable components are required in order to mobilize the boarder academic, scientific, engineering and corporate communities to integrate simulation and modeling into the innovation process.

A recent requirement to introduce simulation into an offshore oil and gas emergency command and control training exercise illustrates the benefits of modular simulation. For the purpose of this exercise, an input from the process control system is required for the participants. The Faculty of Navigation and Maritime Transport at our parent university, Constanta Maritime University, use a process control simulator for research and training purposes. Through an internet connection, we propose to utilize the simulator to present a process control display to the participants of the training exercise (Barsan E., 2006).

The above example illustrates another important concept. Simulation and modeling components need not be concentrated within a single organizational unit to be successful. Open systems that permit various facilities to collaborate on a project specific basis can be equally, if not more, effective.

With the example of the process control simulator, as the Faculty of Navigation and Maritime Transport adds capabilities to their simulator through research activities, the emergency command and control exercises utilizing the simulator benefits as well.

Such an organizational concept is not without problems, and requires careful planning and management to be successful.

Our University is in the process of defining its system architecture requirements to permit a broad scope of collaborations amongst its simulation and modeling units.

The Virtual Center of Marine Simulation is in the process of defining the organizational structure required to mobilize the University simulation and modeling community to deal with maritime issues.

VIRTUAL SIMULATOR TRAINING PROCESS

Inside the project develop by our university about web based simulation training; four tankers related IMO Model Courses was implemented with the new instructional design strategy (Familiarization, Oil tanker, Chemical Tanker, Gas Tanker). All courses have combined three different but interdependent content levels:

Theory Modules (with online-assessment)
Simulator Exercises (with online assessment)
Practical Training Tasks (planned in form of a tanker qualification record book)

The instructional design considered the general aim of the courses. In the Familiarization course the simulation-modules and/or practical tasks were implemented into the theory. Since in many cases the course was made prior to sailing on a tanker, a lot of tasks could be fulfilled either by simulation or on-board practice (Brown A., 2000).

It should be clear that all courses covered more than the requirements of the IMO-model courses. Every course starts with a "pre-test" to ensure that all participants are starting with at least a comparable level of knowledge. The main menu will follow the IMO-model course

The structure of the theory modules themselves is different. A linear structure of explanations seems to be appropriate for those modules describing physical phenomena such as pump- characteristics. A non-linear structure is able to cover wider fields of knowledge such as cargo-hazards. "Story telling" with imbedded explanations and simulation modules represents another useful ways for competence based modules such as “cargo operations”. The complete solution will be offered on the worldwide market with a learning management system running on a server in Norway. In this way, the solution is available not only to the above mentioned partners but to all maritime training institutes worldwide.

1. *Theory module* (in .pdf or Macromedia format). The theory is presented in a comprehensive way and can be found and access on the LMS. Moreover, an exercise and evaluation form are shown from the beginning to the trainees:

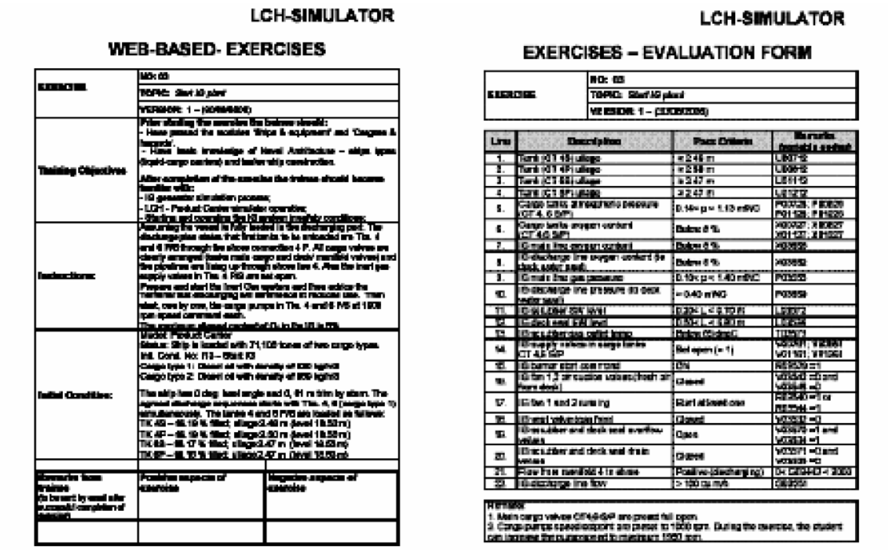

Figure 2 Standard exercise and evaluation form

2. *Step by step demonstrator* (simulator interactive demonstration, ViewletBuilder format). This represent all the steps (with pop-up messages) the students or the trainee need to follow when a specific process is running on the simulator.
3. *Simulator exercises* (with e-Coach and evaluation editors): When creating simulators exercises, a certain procedure shall be followed. It has many steps, starting with the need analysis for the exercise to be carried out and finishing with implementing in the LMS after successful testing and improvement the package. Each exercise has an evaluation form consisting of various evaluation criteria. All these evaluation figures became evaluation actions inside the simulator. After the student/ trainee runs a specific e-learning package and therefore a simulator exercise, the LMS monitors and track all his exercise and finally, by accessing a .txt file, the instructor have the students exercise results.

```
EVALUATION REPORT - Mon Apr 23 20:06:10 2007

SCENARIO    : Start IG
REPORT MODE: manually print
SIM. TIME   : 00:03:30

 ID : Description          (trigger:    ID) LOW-LIM   HIGH-LIM WEIGHT    SUM
---------------------------------------------------------------------------------
+E01: IG discharge line o (trigger: +X01) 0.00      5.00     2.00      0.00
+E02: IG main line oxy co (trigger: +X01) 0.00      8.00     2.00      0.00
+E03: Atm. pressure CT4S  (trigger: +X01) 0.14      1.13     1.00      0.17
+E04: Atm. pressure CT4P  (trigger: +X01) 0.14      1.13     1.00      0.17
+E05: Atm. pressure CT6S  (trigger: +X01) 0.14      1.13     1.00      0.17
+E06: Atm. pressure CT6P  (trigger: +X01) 0.14      1.13     1.00      0.18
+E07: IG main line gas pr (trigger: +X01) 0.10      1.40     1.00      0.00
+E08: IG discharge line p (trigger: +X01) 0.40      1.00     1.00      0.36
+E09: IG scrubber SW leve (trigger: +X01) 0.20      0.70     1.00      0.19
+E10: IG deck seal SW lev (trigger: +X01) 0.50      0.80     1.00      0.00
+E11: IG scrubber gas out (trigger: +X01) 20.00     65.00    1.00      0.00
+E12: IG fan1 air suction (trigger: +X01) 0.00      0.00     2.00      0.00
+E13: IG fan2 air suction (trigger: +X01) 0.00      0.00     2.00      0.00
+E14: IG vent valve       (trigger: +X01) 0.00      0.00     2.00      0.00
+E15: IG supply valve CT4 (trigger: +X01) 1.00      1.00     2.00      0.00
+E16: IG supply valve CT4 (trigger: +X01) 1.00      1.00     2.00      0.00
+E17: IG supply valve CT6 (trigger: +X01) 1.00      1.00     2.00      0.00
+E18: IG supply valve CT6 (trigger: +X01) 1.00      1.00     2.00      2.00
+E19: Tank ullage CT4S    (trigger: +X01) 2.48      25.00    1.00      0.04
+E20: Tank ullage CT4P    (trigger: +X01) 2.50      25.00    1.00      0.04
+E21: Tank ullage CT6S    (trigger: +X01) 2.47      25.00    1.00      0.03
+E22: Tank ullage CT6P    (trigger: +X01) 2.47      25.00    1.00      0.03
+E23: Flow from manifold  (trigger: +X01) -1000.00 2000.00   1.00      0.00
+E24: IG scrubber SW drai (trigger: +X01) 0.00      0.00     1.00      0.00
+E25: IG deck seal SW dra (trigger: +X01) 0.00      0.00     1.00      0.00
+E26: IG discharge line f (trigger: +X01) 0.00      8000.00  2.00      0.00
+E27: IG discharge line o (trigger: +X00) 0.00      5.00     2.00      2.67
+E28: IG main line oxy co (trigger: +X00) 0.00      8.00     2.00      0.00
+E29: Atm. pressure CT4S  (trigger: +X00) 0.14      1.13     1.00      0.09
+E30: Atm. pressure CT4P  (trigger: +X00) 0.14      1.13     1.00      0.09
+E31: Atm. pressure CT6S  (trigger: +X00) 0.14      1.13     1.00      0.09
+E32: Atm. pressure CT6P  (trigger: +X00) 0.14      1.13     1.00      0.09
+E33: IG main line gas pr (trigger: +X00) 0.10      1.40     1.00      0.00
+E34: IG discharge line p (trigger: +X00) 0.40      1.00     1.00      0.50
+E35: IG scrubber SW leve (trigger: +X00) 0.20      0.70     1.00      0.12
+E36: IG deck seal SW lev (trigger: +X00) 0.50      0.80     1.00      0.00
+E37: IG scrubber gas out (trigger: +X00) 20.00     65.00    1.00      0.00
+E38: IG fan1 air suction (trigger: +X00) 0.00      0.00     2.00      0.73
+E39: IG fan2 air suction (trigger: +X00) 0.00      0.00     2.00      0.17
+E40: IG vent valve       (trigger: +X00) 0.00      0.00     2.00      0.23
+E41: IG supply valve CT4 (trigger: +X00) 1.00      1.00     2.00      0.00
+E42: IG supply valve CT4 (trigger: +X00) 1.00      1.00     2.00      0.00
+E43: IG supply valve CT6 (trigger: +X00) 1.00      1.00     2.00      0.00
```

Figure 3. Exercise results.

The encouraging results obtained in final tests by our students give us the right to consider that the use of multimedia tools, computer program and web enable simulation modules must be constantly improved and extended. Also, the interactive methods prove to be efficient and have to be developed widely in the future. Distant learning combined with simulators will make a new and flexible training approach possible. Therefore, we can finally consider that e- Learning has a great and positive impact on the maritime education field and moreover learning combined with training will be by far the most effective way to increase skills and competence

CONCLUSIONS

Without addressing the chronic issue of human error, the maritime transportation system all over the world, already feeling the effects of spiraling costs associated with accidents, will have difficulty in absorbing the sweeping changes currently underway. Without mitigating the impact of human error, innovation in the maritime sector may introduce more cost than benefit and not be sustainable in the long run.

In order to reduce the number of accidents caused by human error, effort will have to focus on reducing latent error, mitigating the impact of psychological precursors, and improving the crisis management capability within the maritime community. Such an effort will not only reduce the accident rate, but will help to stimulate the innovation process by making new initiatives more likely to succeed.

Simulation, modeling and web base simulation training represents an important capability to ensure that innovation delivers on its promise of improved activity. To achieve this goal a concentrated effort is required to incorporate maritime simulation, modeling and web base training process into the innovation cycle.

REFERENCES

Anthony Peterson M.M. (2002), Simulation and Modelling in Innovation, Proceedings of The 29th Annual General Meeting of the International Marine Simulator Forum, Keelung, Taiwan.

Barnett, M.L., Gatfield, D., Habberley, J (2002), Shipboard crisis management: A Case Study, *Procceding International Conference Human Factors in Ship design and Operation*, Royal Institution of Naval Architects. pp 131-145.

Barnett M.L. (2004), The development of simulator-based scenarios from the analysis of recent maritime accidents, Proceedings of the Advances in International Maritime Research Conference, Tasmania, Australia.

Barsan E. (2006), Bridge and engine room teams cooperation, in loss of remote control scenarios, International Navigation Simulator Lecturers Conference, ISBN 8-8901-2481-X, Pub.Algraphy S.N.C., Genova, Italy.

Brown A., (2000), Optimum Risk Tanker (ORT) - A Systematic Approach to a TAPS Tanker Design, *Ship Structure Committee (SSC) Symposium*, Arlington, VA, US.

Chiotoroiu L., Dinu D., Hanzu-Pazara R., Pana I. (2006), Simulation Models in Maritime Distant Learning: Tankers Topping-Off, *Conference "TECHNONAV '2006"*, Constanta, Romania.

Hensen H. (1999), *Ship Bridge Simulators: A Project Handbook*, Nautical Institute, ISBN 1870077504, London, UK.

Huey, Beverly Messick and Christopher D. Wickens (1993), *Workload transition – Implications for Individual and Team Performance*. Washington D.C: National Academy Press.

International Maritime Organization (1999), MSC/Circ.565 *Fatigue as a contributory factor in Marine Accidents*, International Maritime Organization., London, UK.

International Maritime Organization (2000), MSC/Circ.982 *Guidelines on Ergonomic Criteria for Bridge Equipment and Layout*, International Maritime Organization, London, UK.

McCallum M.C., Raby M., and Rothblum A.M. (1996), *Procedures for Investigating and Reporting Human Factors and Fatigue Contributions to Marine Casualties*. Washington, D.C.: U.S. Dept. of Transportation, U.S. Coast Guard Report No. CG-D-09-97. AD-A323392.

Perrow C. (1984), *Normal Accidents: Living with High-Risk Technologies*. Basic Books, pp. 215-218..

Reason J. et al (1991), *Errors in a team context,* Mohawe Belgirate Worksho.

Rothblum, A.M., *Human error and marine safety*, Volume 4 in U.S. Coast Guard Risk-Based Decision-Making Guidelines, U.S. Coast Guard Research and Development Center, 2006.

The International Maritime Human Element Bulletin: www.he-alert.org

U.K. P&I Club, (1992), The United Kingdom Mutual Steam Ship Assurance Association (Bermuda) Limited. Analysis of Major Claims. www.ukpandi.com

Wagenaar W.A. and Groeneweg J. (1987), *Accidents at sea: Multiple causes and impossible consequences*. Int. J. Man-Machine Studies, 27, 587-598.

Journal of Maritime Research, Vol. V. No. 1, pp. 19-34, 2008

Printed in Santander (Spain).
ISSN: 1697-4840

WORK RISK MEASURES IN SEVER ENVIRONMENTS OF A SHIP

J. A. Orosa[1], A. Baaliña[2] and G. Iradi[3]

Received 18 April 2007; received in revised form 28 April 2007; accepted 20 January 2008

ABSTRACT

Ships are a clear example of thermal indoor environment with characteristics changing in short intervals of time. In the engine room the conditions used to be extreme as well.

In this study, we have carried out a monitoring of air temperature and relative humidity in several locations of a merchant ship that covers the sea lane Las Palmas-Barcelona.

Subsequently, it has been determined from those indoor temperature and relative humidity data, the corresponding parameters of thermal comfort (predicted mean vote, PMV; predicted percentage of dissatisfied, PPD and acceptability, Acc) and heat stress of sever exposures in the engine room.

Key words: Work-risk measures, heat strain, thermal comfort, PMV, PPD.

INTRODUCTION

The thermal parameters of indoor environments need a suitable study, knowledge and processing in relation with the industrial safety, because high or low air temperatures and not controlled heat sources, may induce lower productivity rates, higher accidents rates and health hazards.

A clear example of thermal indoor environment with extreme characteristics changing in short intervals of time is a ship. In this study, we have carried out the

[1] Profesor Universidade da Coruña, E.T.S. Náutica y Máquinas (jaorosa@udc.es), A Coruña, Spain. [2] Professor Universidade da Coruña (alvarobi@udc.es), A Coruña, Spain. [3] Marine engineer, Universidade da Coruña (jaorosa@udc.es), A Coruña, Spain.

sampling of air temperature and relative humidity in a merchant ship that covers the sea lane Las Palmas-Barcelona-Las Palmas.

In the indoor environment of a ship, it is the engine room that is important having in mind its special characteristics related with safety at work. Temperature and relative humidity data from the engine room and other locations have been analysed to obtain comfort indexes by means of new models applied to this environment.

From this analysis of real conditions, it has been possible to define limit time that a person can work without heat stress in accordance with spanish standards NTP.

METHODS

By means of Gemini® data loggers, a monitoring of air temperature and relative humidity has been carried out in a merchant vessel during the voyage Las Palmas-Barcelona and return in the winter season of 2001. The engine room together with the control room, the dinning room for officers and the bridge has been the sampling locations. At the same time, outdoor data have been also obtained for comparing purposes. More than 11,000 measurements have been collected.

As previously stated, we have paid special attention to the engine room environment. Thus, we have collected data both from the control engine room and the proper engine room.

To obtain measurements of work environments, data-loggers were located to a height near the centre of gravity of workers when they remain in the usual position of work. In those places where this position was not possible to fix, the sampling points were moved away from heat sources such as walls or air conditioning equipments at least 0.6 m, for avoiding interferences. This is the case of the engine room where several work places, those were anticipated hotter and colder and in the centre of place, have been assessed according to recommendations of INNOVA (1997).

ANALYSIS OF MONITORED VARIABLES

Because ASHRAE standards (2001) only discuss about the different comfort zones, the European standard ISO 7730 (1994) has been analysed in relation to the required conditions of indoor environments. Table 1 lists these conditions for different seasons and all practical applications.

As opposed to the real conditions in other locations, it has been obtained an average temperature in the control engine room of 19.76 ºC. This temperature is too low compared to that in the engine room, so this fact can cause both a thermal shock for workers and high-energy consumption for air conditioning.

Average relative humidity in the bridge and dinning room has reached 50%, very close to the maximum value recommended in standards (see Table 1). Whereas in the control engine room average relative humidity was 40%, in the engine room the value was 25%, in clear opposition to the rest of locations and under the minimum value of 30% recommended.

Table 1. Reference values for indoor conditions.

Applications	Summer		winter	
	Temperature (ºC)	Relative humidity (%)	Temperature (ºC)	Relative humidity (%)
All	>23	30-65	18-22	30-65

Given the influence of outdoor conditions on indoor environment, we have collected outdoor data during the monitoring period from December 2001 to February 2002. The results obtained were an average temperature of 23 ºC, with maximum values of 27 ºC and an average relative humidity of 60 %. The weather was predominantly sunny and not much cloudy.

The assessment of indoor environments is based in the study of comfort indexes, however the variables involved in the definition of such indexes must remain in the range provided by standards. For this reason, we have carried out a statistical analysis of temperature, relative humidity and enthalpy data. Results in each location are shown in Tables 2, 3 and 4.

The average temperature of the air both in the bridge and dinning room has remained about 22 ºC (see Table 3). In the engine room the average was 32.5 ºC, with peaks of 38.5 ºC. These results are out of the allowed values for hot environments and can produce different health disorders (see Figure 1).

As fast as indoor temperature increases the first psychical disorders appear such as loss or difficulties of concentration. Finally, physiological disorders as heart and circulatory system overload could yield.

According to the indications of NTP 18 (Heat stress evaluation of sever exposures) (NTP, 18) and NTP 350 (Heat stress evaluation required sweating index) (NTP, 350) standards, we got the corresponding graphics in Figures 2 and 3. They are based in human body thermal balance and they showed the maximum time that a worker could remain in sever exposures as engine rooms. The minimum time that the same worker must be at the control room to lose the accumulated heat was also calculated.

This study is refered to a standard worker with 70 kg in weight and light clothes. The Figures show that the exposition must stop when the internal temperature increases 1 ºC, because the maximum evaporation is lower than the required for the thermal balance.

Figures express the relationship between time and globe temperature because the ship ambient changes during the voyage.

Using the obtained graphics for this engine room we can affirm that the worker must be in the engine room for 17 minutes and must have a rest at the control room for at least 10 minutes in order to get the suitable heat release.

Table 2. Statistical analysis of temperature data collected in different sampling locations of the ship.

Temperature T (ºC)	**Dinning room**	**Bridge**	**Engine room**	**Control engine room**
Average	22.05	21.41	32.50	19.76
Standard deviation	1.76	2.14	2.83	1.33
Maximum	25.40	26.90	38.50	27.30
Minimum	17.10	16.20	25.40	17.40

Table 3. Statistical analysis of relative humidity data collected in different monitoring locations of the ship.

Relative humidity, RH (%)	**Dinning room**	**Bridge**	**Engine room**	**Control engine room**
Average	50.66	49.66	24.90	41.17
Standard deviation	8.19	8.67	4.15	6.58
Maximum	69.90	73.80	33.90	70.50
Minimum	28.10	21.90	16.20	30.30

Table 4. Statistical analysis of enthalpy data collected in different sampling locations of the ship.

Enthalpy, h (kJ/kg)	**Dinning room**	**Bridge**	**Engine room**	**Control engine room**
Average	43.03	41.58	52.20	34.58
Standard deviation	5.24	6.56	6.47	2.79
Maximum	54.76	52.07	66.39	66.39
Minimum	31.11	1.00	35.37	35.37

Figure 1: Influence of temperature on heat illness.

Temperature	Biologic response	Heat disorders
20 ºC	Comfortable	Full capacity
↓	Discomfort Irritability Concentration difficulties Decrease in intellectual capacity	Psychical disorders
	Increase in work mistakes Decrease in handiness More accidents	Psychical and physiological disorders
	Decrease in efficiency of heavy works Disturbance of metabolism Cardiac-circulatory system overload Heavy fatigue	Physiological disorders
35-40ºC	Maximum temperature bearable	

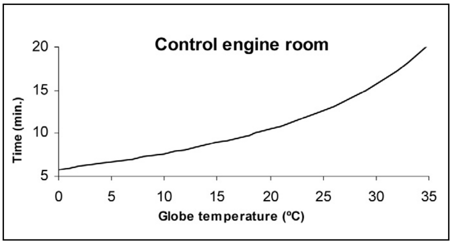

Figure 2. Minimum time that mus be elapsed in the control engine room.

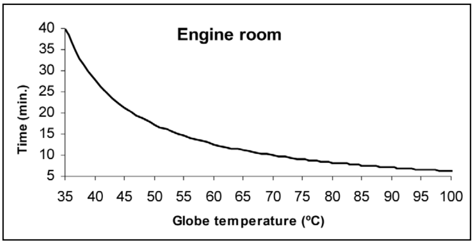

Figure 3. Máximum time that must be elapsed in the engine room.

STUDY OF COMFORT CONDITIONS

To keep thermal comfort and avoid disorders two conditions must be fulfilled. The first is that the combination of skin and deep body core temperatures leads to a neutral feeling of comfort. The second involves the energy balance between the body and the environment. In this sense, the total metabolic heat produced by the body should be equal to the heat loss from the body.

The comfort equation developed by P. O. Fanger [3] relates physical parameters that can be measured with the neutral thermal feeling experimented by a "typical" person:

$$M - W = H + E_c + C_{res} + E_{res} \tag{1}$$

M = metabolic rate, W/m^2. It is the rate of chemical energy transformation from aerobic and anaerobic activities, in heat and work.
W = rate of mechanical work performed, W/m^2.
H = heat exchange from the skin by convection, conduction and radiation, W/m^2.
E = evaporative heat exchange, W/m^2.
E_C = evaporative heat exchange through the skin, in conditions of neutral thermal feeling, W/m^2.
C_{res} = respiratory heat loss by convection, W/m^2.
E_{res} = respiratory heat loss by evaporation, W/m^2.

Through measurement of physical parameters, the comfort equation provides an operative tool whereupon it can be assessed under what conditions the thermal comfort in an occupied space is achievable. The thermal comfort can be quantified through indexes defined in standard ISO 7730 (Fanger, 1970). The Predicted Mean Vote (PMV) is derived from the heat balance before mentioned and provides an indication of the thermal sensation by means of a scale of 7 points, from -3 (cold sensation) to +3 (hot sensation), where 0 means a neutral thermal sensation. Another comfort index is the Predicted Percentage of Dissatisfied (PPD) that provides information on thermal sensation by predicting the percentage of people likely to feel too hot or too cold in a given environment. PMV values of -3, -2, +2 and +3, means thermal discomfort in the PPD index. Both indexes are influenced by physical activity and clothing. The physical activity is quantified through the metabolic rate. The human body maintains a minimum rate of heat production at about 60 W during sleeping. The metabolic rate is often expressed in Met, which means a heat production of 58 W/m^2 of body surface. On the other hand, clothing acts as an insulation reducing the heat loss from the body. A magnitude called Clo is normally used to quantify the insulation of the different clothes. In terms of thermal resistance 1 Clo is equivalent to 0.155 m^2 ^{o}C/W.

In this study, we have used the PMV models developed by the Institute for Environmental Research Kansas State University (ASHRAE, 1985) for the assessment of the thermal comfort conditions in indoor environments. The expression of the model is the following:

$$PMV = at + bP_v - c \tag{2}$$

Where t is the temperature and P_v is the vapour partial pressure. Right constants a, b and c must be used to take into account sex and time of exposure to the indoor environment. Such constants have been adapted to the existing conditions in the studied environments by means of a thermal comfort data-logger 1221 from Innova. Values of 1.2 Met and 1 Clo were assumed for calculations.

PPD has been also studied for the same environment. The index has been defined by means the following equation, taking into account the PMV values previously obtained:

$$PPD = 100 - 95 \cdot e^{-\left(0{,}03353\,PMV^4 + 0{,}2179\,PMV^2\right)} \tag{3}$$

Temperature and relative humidity values collected in the ship have been introduced in the models just detailed and corresponding PMV and PPD values has been calculated for each indoor location. Tables 5 and 6 show the statistical analysis of results.

In order to make comparisons, the thermal acceptability (Acc) has been also calculated. This new index, introduced by Fanger (1970), has been used by Simonson et al. (2001) to assess indoor environments as a result of their adaptation to any kind of thermal conditions but with loss of accuracy. The index is related with enthalpy through the following equation:

$$A_{cc} = a \cdot h + b \tag{4}$$

Where a and b are empirical coefficients whose values for clean air are -0.033 and 1.662 respectively. Table 7 shows the average acceptability in the engine room.

Once averages and standard deviations have been assessed, it has been quantified what values have been broken the standards according with ASHRAE. Such specifications set a PMV range that goes from -0.5 to +0.5 as adequate. This interval is equivalent to a PPD lower than 10 %.

To make the interpretation of results easier, Figures 4 to 11 show the relative frequency graphs of PMV. The cumulative PPD curve has been also plotted.

Table 5. Statistical analysis of calculated PMV values in different locations of the ship.

PMV	Dinning room	Bridge	Engine room	Control engine room
Aerage	0.21	0.19	2.15	0.52
Standard deviation	0.51	0.48	1.01	0.37
Maximum	1.74	2.13	3.75	1.77
Minimum	-0.96	-0.91	1.54	-0.90

Table 6. Statistical analysis of calculated PPD values in different locations of the ship.

PPD	Dinning room	Bridge	Engine room	Control engine room
Average	11.31	8.80	76.61	13.54
Standard deviation	10.51	12.8	34.88	5.20
Maximum	64.13	82.60	99.99	65.92
Minimum	5.00	0.00	53.06	5.00

Table 7. Average acceptability in the engine room.

Acc	Dinning room	Bridge	Engine room	Control engine room
Average	0.23	0.30	-0.05	0.52

Bridge and dinning room

Figures 10 and 11 show a cumulative PPD in the bridge and dinning room close to the neutral thermal conditions. The last one is the location more comfortable, followed by the bridge; since 90% and 78% of collected data are included in the 10 % of PPD, according with the ASHRAE standard. These results mean that it is not possible the energy optimisation of indoor environment, because outdoor and indoor air temperatures maintain similar values. Nevertheless, in these environments relative humidity usually exceeds 55% established by ASHRAE and ISO standards. For this reason, it would be advisable the use of dehumidifiers for protecting both occupiers and electrical devices of the bridge.

Engine room

The most extreme conditions have been found in the engine room (see Figures 4 and 8). PMV and PPD values of 2.15 and 76.61 %, has been obtained. These extreme values together with the fact that nearly all calculated PPD data exceed 10 % show that in this location the thermal sensation is very hot.

From Simonson's studies [8] can be deduced that acceptability is better with a lower enthalpy. He has established the value of 50 kJ/kg as the upper limit for enthalpy. Above this value the air perception is unbearable, independently of the indoor air quality. In our case, the enthalpy value in the engine room exceeds 52 kJ/kg and thus the calculated acceptability has been -0.05.

Values of relative humidity are low because high outdoor temperature. These conditions may cause hyperthermia, vasodilatation, sweat glands activation, increase of peripheral circulation and electrolytic changes of sweat by loss of salt content. As a possible solution to this problem, an increase in ventilation rates may achieve some decrease in temperature.

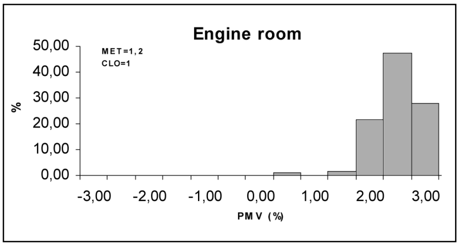

Figure 4. Relative frequency of PMV in the engine room.

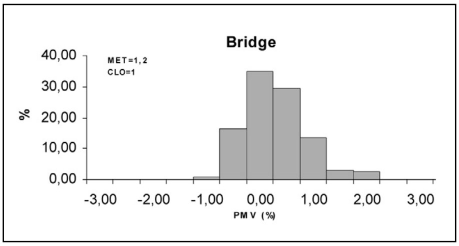

Figure 5. Relative frequency of PMV in the bridge.

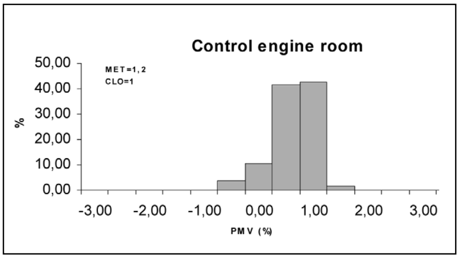

Figure 6. Relative frequency of PMV in the control engine room.

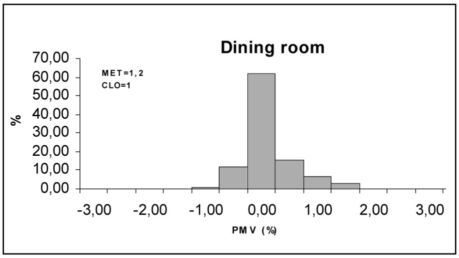

Figure 7. Relative frequency of PMV in the dinning room.

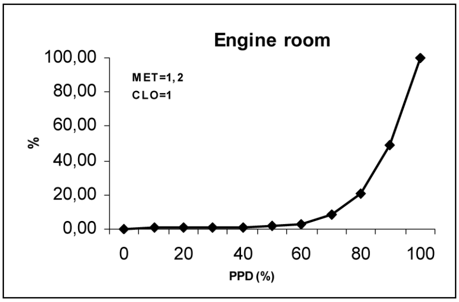

Figure 8. Cumulative frequency of PPD in the engine room.

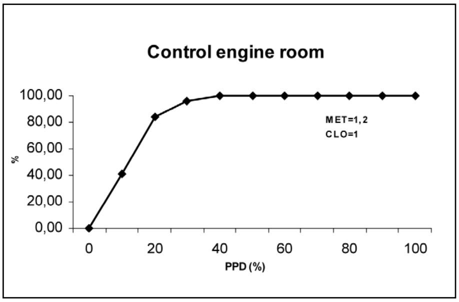

Figure 9. Cumulative frequency of PPD in the control engine room.

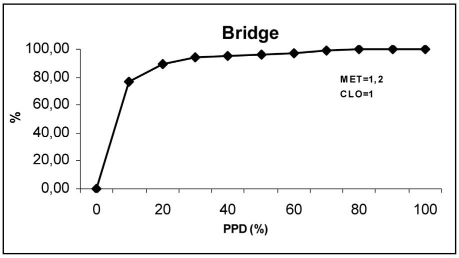

Figure 10. Cumulative frequency of PPD in the bridge.

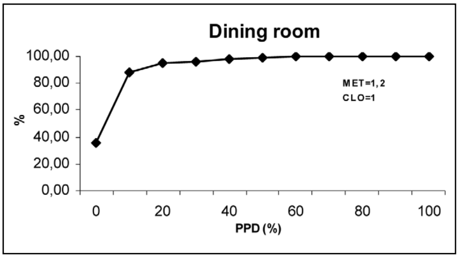

Figure 11. Cumulative frequency of PPD in the dinning room.

Control engine room

In this location the PMV is close to 0.5, the optimum condition for energy saving, the average PPD is 13 % and 40% of calculated PPD data are within 10 % fixed in standards (see Figures 6 and 9). These results lead to a limit condition in the control engine room. Besides, the average temperature of 19.76 ºC and the average relative humidity of 24.9% (lower than the minimum reference value of 30 % from ISO 7730 [3]) are too low which means a high-energy consumption in air conditioning.

To get an energy saving, the temperature should be higher than 20 ºC as in accordance to the air acceptability criteria, to maintain the same PPD, the enthalpy must be the same as well. For this reason, a temperature increase of 1ºC leads a relative humidity decrease of 5% and the lower limit of 30% already mentioned would not be fulfilled. All these facts lead to set the existing temperature conditions as optimum for the existing relative humidity.

A possible improvement may be to increase the air renovations with outdoors, to cause an increase of both relative humidity and temperature towards values more suitable.

Another solution to correct the low relative humidity may be to avoid the possible dehumidification of conditioning air in the chiller by means air conditioning unit to be replaced by another one with higher surface area of heat exchange that causes a higher external surface temperature for the same heat rate.

CONCLUSIONS

- Ships show very different thermal environments that must be studied with greater depth.
- Options of energy saving or thermal comfort improving in the bridge and the dinning room are very limited.
- The engine room shows air conditions out of any recommendations from standards, so it is suggested an increase in ventilation for taking preventive measures against work risks.
- The control engine room shows limit conditions of thermal comfort with temperature values too low that lead to an excessive energy consumption. As the outdoor air conditions are suitable, an increase of renovations with outdoor air can be proposed.
- Taking into account our results it would be necessary to take the following work risk-preventing measures:
  - — Drinking water. Sources of drinking water must be available close to work locations and workers must be informed about the necessity of drinking frequently.
  - — Acclimatization. Workers starting new or going back to work require an exposure time for achieving acclimatization.
  - — Metabolic heat. Adjusting length and frequency of breaks and work periods, and work rates may be reduced the metabolic heat release. If it is possible works must be scheduled in time of less heat. Work periods into engine room must not be higher than twenty seven minutes and, after it, worker must be about ten minutes in the control room.
  - — Workers must be kept under constant watch by a trained colleague for detecting any symptom of heat strain.

REFERENCES

ASHRAE (1985). HVAC Fundamentals. ASHRAE. Atlanta.

ASHRAE STANDARD (2001). Proposed Revision to an American National Standard Thermal Environmental Conditions for Human Occupancy. FIRST PUBLIC REVIEW DRAFT.

Fanger, P. O. (1970) Thermal comfort analysis and applications in environmental engineering. USA: McGraw-Hill.

INNOVA Air Tech Instruments A/S (1997). Thermal Comfort [online]. Denmark . Available from: http://www.innova.dk/books/ thermal/
[Accesed 1 April 2007].

ISO 7730 (1994). Moderate thermal environments. Determination of the PMV and PPD indexes and specification of the conditions for thermal comfort.

NTP 18 (Heat stress evaluation of sever exposures).

NTP 350 (Heat stress evaluation required sweating index).

Simonson et al. (2001). Improving indoor climate and comfort with wooden structures. Technical research centre of Finland. Espoo.

PREVENCIÓN DE RIESGOS EN AMBIENTES EXTREMOS DE UN BUQUE

Resumen

Los buques son un claro ejemplo de ambiente interior cuyas características varían en breves intervalos de tiempo. En este trabajo se han muestreado las variables de temperatura y humedad relativa en diversas zonas de un buque que realiza periódicamente la misma ruta.

Una vez analizadas las condiciones de temperatura y humedad relativa, se han definido los índices de confort térmico por medio de modelos definidos por la Ashrae, prestando especial atención a la sala de máquinas, dadas sus características especiales en torno a la seguridad laboral.

También se han calculado los períodos de trabajo en la sala de máquinas y descanso en la sala de control para prevenir la aparición de los síntomas de estrés térmico.

MÉTODOS

Las condiciones de temperatura y humedad relativa han sido muestreadas por medio de Gemini data loggers durante una ruta típica. Las zonas analizadas han sido; la sala de máquinas, sala de control, puente, comedor de oficiales y ambiente exterior. En total, se han recogido más de 11000 mediciones.

Para que las condiciones muestreadas sean lo más representativas posible, dichos data loggers se han ubicado cerca del centro de gravedad de los trabajadores pero alejados de fuentes térmicas como paredes, y equipos de acondicionamiento de aire.

Los resultados obtenidos se han comparado con las indicaciones de la normativa ISO 7730 en función de sus indicaciones, las cuales se han resumido en la Tabla 1.

Tabla 1. Valores de referencia para las condiciones interiores.

Verano		Invierno	
Temperatura (ºC)	Humedad relativa (%)	Temperatura (ºC)	Humedad relativa (%)
>23	30-65	18-22	30-65

RESULTADOS

Estudio de las variables termodinámicas

Los resultados se muestran en las tablas 2 y 3.

La temperatura promedio del aire en el puente y en el comedor de oficiales ha permanecido en torno a 22ºC. En la sala de máquinas el valor promedio ha sido de

Temperatura T (ºC)	Comedor	Puente	Sala de máquinas	Sala de control
Promedio	22,05	21,41	32,50	19,76
Desviación típica	1,76	2,14	2,83	1,33
Máximo	25,40	26,90	38,50	27,30
Mínimo	17,10	16,20	25,40	17,40

Tabla 2. Análisis estadístico de la temperatura muestreada en las diferentes zonas del buque.

Humedad relativa (%)	Comedor	Puente	Sala de máquinas	Sala de control
Promedio	50,66	49,66	24,90	41,17
Desviación típica	8,19	8,67	4,15	6,58
Máximo	69,90	73,80	33,90	70,50
Mínimo	28,10	21,90	16,20	30,30

Tabla 3. Análisis estadístico de la humedad relativa muestreada en las diferentes zonas del buque.

32,5 ºC con picos de hasta 38,5 ºC. Estos valores superan los valores permitidos para ambientes térmicos de forma que se puede originar problemas de salud. A medida que aumenta la temperatura interior aparecerán los primero signos de desordenes térmicos como la pérdida o dificultad de concentración, hasta llegar a originar la sobrecarga del sistema circulatorio.

A pesar de que en la mayoría de las zonas del buque se respetan las indicaciones de la normativa, en la sala de máquinas la humedad relativa promedio es de 25 ºC. Dicho valor es claramente inferior al valor mínimo recomendado.

Estudio de los índices del confort

La Ecuación de Confort nos proporciona una herramienta operativa con la cual, midiendo unos parámetros físicos, podemos evaluar bajo que condiciones podemos ofertar comodidad térmica en un espacio habitado. Dicha comodidad puede ser definida mediante el índice PMV de Voto Medio Previsto (Predicted Mean Vote). El índice PMV predice el valor medio de la sensación subjetiva de un grupo de personas en un ambiente determinado mediante un rango de sensación térmica de 7 puntos, desde -3 (frío) a +3 (caliente), donde el 0 representa una sensación térmica neutra.

Para predecir cuánta gente está insatisfecha en un ambiente térmico determinado, se ha introducido el índice de Porcentaje de Personas Insatisfechas PPD (Predicted Percentage of Dissatisfied). En el índice PPD la gente que vota - 3, - 2, +2, +3 en la escala PMV se considera térmicamente insatisfecha.

Para el estudio de las condiciones de confort térmico existente en dichos ambientes interiores, se han empleado los modelos de PMV desarrollados por el Instituto para la Investigación Ambiental de la Universidad del Estado de Kansas. Su estructura es la siguiente:

$$PMV = at + bP_v - c \tag{2}$$

También se han definido el porcentaje de insatisfechos ante ese mismo ambiente de estudio. Este índice se ha definido por medio de la ecuación 2, determinada a partir de los valores de PMV anteriormente obtenidos.

$$PPD = 100 - 95 \cdot e^{-\left(0{,}03353\, PMV^4 + 0{,}2179\, PMV^2\right)} \tag{3}$$

Las condiciones de temperatura y humedad relativa obtenidas han sido introducidas en los modelos creados, de forma que se han determinado los valores de PMV correspondientes.

Los resultados han mostrado para la sala de máquinas unos valores de 2,15 de PMV y la mayoría de los valores superan el 10 % de insatisfechos, por lo que la sensación térmica es muy calurosa.

En la sala de control el 40% de las mediciones están dentro del 10% de PPD y el PMV promedio ha sido de 0,5, por lo que se puede decir que dicho ambiente se encuentran optimizados térmicamente. A pesar de tener esa temperatura, los valores de humedad relativa están por debajo del límite inferior fijado por la normativa.

En función de las normativas NTP 18 y 350 se han determinado los gráficos que definen el tiempo máximo de estancia en la sala de máquinas y mínimo necesario en la sala de control para poder perder el calor acumulado y evitar posibles riesgos laborales. En concreto, se recomiendan períodos de trabajo de 17 minutos y descansos, en la sala de control, de 10 minutos.

CONCLUSIONES

Los buques presentan ambientes térmicos muy variados que deben ser estudiados con mayor profundidad.

La sala de máquinas presenta unas condiciones ambientales fuera de cualquier normativa, por lo que se sugiere un aumento de la ventilación para la prevención de posibles riesgos laborales.

La sala de control presenta una situación límite de confort térmico, pero a una temperatura demasiado baja, por lo que se origina un consumo energético excesivamente elevado. Se ha propuesto un aumento del número de renovaciones con el ambiente exterior, dado que éste presenta unas condiciones más adecuadas.

Es necesario acatar una serie de medidas para la prevención de riesgos laborales ante ambientes tan extremos:

Agua potable: debe existir una fuente adecuada de agua potable cerca del lugar de trabajo, y los trabajadores deben estar informados de la necesidad de ingerir agua con frecuencia.

Aclimatación: aquellos trabajadores nuevos o aquellos recién incorporados (por baja o vacaciones) o aquellos que estén asignados a trabajos más ligeros, deben tener un período de aclimatación previo antes de incorporarse definitivamente a pleno trabajo. Este es el caso del embarque de personal procedente de largos periodos de descanso, lo cual es un suceso muy habitual.

Calor metabólico: puede reducirse el calor interno generado mediante ajustes en la duración del período de trabajo, la frecuencia y duración de los intervalos de descanso, el ritmo del trabajo y la mecanización del trabajo. El estudio ha mostrado períodos de trabajo en la sala de máquinas de 17 minutos y periodos de descanso en la sala de control de 10 minutos.

Vigilancia por un compañero: los trabajadores deben ser observados por un supervisor entrenado que pueda detectar a tiempo cualquier síntoma de sobrecarga térmica.

Journal of Maritime Research, Vol. V. No. 1, pp. 35-46, 2008

ISSN: 1697-4840

SEABED MAPPING

E. Melón[1], S. Iglesias[2], A. Bermejo[3] and H. Sánchez[4]

Received 11 June 2007; received in revised form 18 June 2007; accepted 3 February 2008

ABSTRACT

Navigational charts are the marine equivalent to topographic maps. Both use measures of height/depth points and outlines to depict an image.

The arrival of multibeam hydrographic echo sounders in the early eighties marked a true revolution in seabed mapping, achieving results for marine measurements almost equivalent to those obtained on land.

There came a change from validating specific data to having a continuous registration of the seabed. These technologies have even indirectly made it possible to obtain a highly precise verification of the composition of the marine seabed with errors smaller than 10 centimetres.

The equipments for bathymetric studies are installed in ships arranged for mapping the seabed. These are usually oceanographic ships for mapping deep waters and small-size boats for shallow waters. In the first case, the detailed studies of deep zones serve as an important research component, whereas in the case of bathymetries for coastal or shallow waters, the foundation is more technical, as in the case of navigable channels, which must be periodically dredged, of harbour works, regeneration of beaches, etc.

Key words: Hydrography, Bathymetry, Echo sounders, Multibeams.

INTRODUCTION

Hydrography, in general, is similar in many aspects to research on land and many of the techniques used are the same, or extensions of them. Navigational charts are

[1] Profesor de Universidad, Universidad de la Laguna (emelon@ull.es), Santa Cruz de Tenerife, Espana. [2] Profesor Titular, Universidad de la Coruña (emelon@ull.es), La Coruña, Espana. [3] Profesor Titular, Universidad de La Laguna (emelon@ull.es), Santa Cruz de Tenerife, Espana. [4] Alumno Segundo Ciclo, Universidad de La Laguna (emelon@ull.es), Santa Cruz de Tenerife. Espana.

the marine equivalent to topographic maps. Both use measures of height/depth points and outlines to depict an image, but where the person using the map can visually verify the details shown, the user of the chart cannot do it. The person using a chart, therefore, blindly depends on the precision and reliability of the hydrographer´s work.

Marine measurements have always been less precise than their equivalents on land, although in the last few years with the technological development of transducers and positioning systems (differential GPS, dynamic positioning, inertial systems, etc.) an important evolution has been achieved, obtaining results close to those obtained on land. Due to this evolution, seabed mapping has lately experienced an enormous improvement in its accuracy. From the archaic but effective sounding lead, which prevailed up until the 20th century, there has been a change to the current multibeam echo sounders that map the seabed with great exactitude.

The need for knowing the depth of specific shallow zones has always existed, understanding that the safety of the boat depended on this data in the case of calling at a port or bay. Knowing the precise draft in conditions of low tide, permitted entrance of particular ships with a maximum draft equal to the one measured plus the established margin of safety. In addition to draft data, it is equally important to identify the composition of the seabed. Evidently, it is not the same for a ship to run aground on rock or coral reef, as on a muddy bed, for in the latter the possibility of recovering the ship in case of stranding is much more feasible. In the same way, depending on the type of seabed existing, some anchors will be more adequate than others.

The equipments for bathymetric studies are installed in ships arranged for seabed mapping. These are usually oceanographic ships in the case of deep waters and small-sized ones in shallow waters. In the first case, the detailed study of the deep zones has an important research component, whereas in the case of the bathymetries of coastal or shallow waters, the foundation is more technical, as in the case of navigable channels that must be periodically dredged, of harbour works, regenerations of beaches, etc.

DESCRIPTION OF AN ECHO SOUNDER

The echo sounder is a telemetry system based on echoes adapted to the marine environment. This apparatus emits sound waves with a specific frequency, which after reflecting off the seafloor are received once again. Subsequently, the apparatus, using a simple calculation, finds the distance between the echo sounder and the seabed. Given that data for the speed of sound is known as well as the elapsed time between emission and reception, the distance fulfilled by the wave in half of this time, that is, the distance from emission of the wave until it touches ground is the depth between the boat and the seabed.

There are two types of echo sounders, multibeam and single-beam, which moderately differ from each other in regards to range, frequency, amplitude of beams, etc.

Source: http://www.hydroacoustics.com/

Figure 1. Image of a multibeam echo sounder scanning (Hydroacoustics).

As its own name indicates, a multibeam echo sounder has several beams, against only one of a single beam. The latter emits a beam of waves directed downwards with an angle of 1º.5. This means that with the single beam we will only know the depth existing right below the ship. On the contrary, the multibeam emits a multitude of beams, between 101-80 depending on the model, attaining coverage of 6 to 7 times the depth. That is, if we are in an area of 50 metres depth we will know the exact draft of a strip of 350 metres in width. The great advantage of the multibeam technology is the elevated coverage of the floor, which makes it possible to scan a large floor surface with a single pass ensuring that, contrary to the single beam echo sounder, no dead zones have been left between non-adjacent lines. Remember the sinking of the "Urquiola" at the entrance of A Coruña, because of "needle" rocks that had not been detected when the navigational chart was elaborated.

In order to minimize the errors we have to take into account a series of variables, which are:

Speed of sound in water

The waves emitted by an echo sounder are sound waves with a frequency of between 50 and 200 KHz. in the case of the multibeams, and between 2 and 600 KHz. in the single beams. The travelling speed of these waves depends on the density of the medium through which they move, which in our case is water. At a greater water density, there is a greater travelling speed. The problem is that water does not always have the same density in the same zone, varying seasonally, and is very different from some geographic areas to others, not to mention the differences between the density of water in ocean and that of a river or a lake. For this reason, it is necessary to introduce the density data into the working software of the echo sounder in order to correctly calibrate its travelling speed. This data must be obtained and corrected prior to each data collection. In order to do this, a density profile is fulfilled by launching a C.T.D. instrument (Conductivity, Temperature, Depth) to the maximum depth in which we are going to measure and introducing the density values of each depth at the time of processing the data in the echo sounder computer.

Placement of the transducer

The echo sounders have an emitting head, denominated transducer, which is usually placed against the hull of the ship. Its own coordinates must be well calculated with respect to a central point of the ship, which will serve as point 0, since the greater or lesser precision of the measurement will depend on it. The system is placed with a differential GPS, which in turn is based in reference to the localization point of the echo sounder. The draft variations produced in the ship must be corrected in the multibeam software since if, for example, the ship drafts 1 more metre because it is loaded with fuel, the measurement obtained will be 1 metre less than the real one.

Reference system

The depth measurements taken by the echo sounder must be spatially referenced, that is, a latitude and longitude must be assigned to each point of measurement of the depth. The ship is equipped for this with a positioning system, logically differential, which gives each depth sounding its latitude and longitude. This system is composed of a differential GPS (DGPS) and a movement sensor. There are currently several systems for obtaining a differential positioning; the most common ones are the ones previously mentioned, the DGPS, RTK (Real Time Kinematic) and the Omnistar system.

The RTK is fixed on a base station situated on a land point with known coordinates that receives a GPS signal with it´s standard error. This station, by knowing its real position with an exactitude down to the millimetre - without error -, sends this error in latitude and longitude made by the radio satellites, to the GPS station of the ship, which can consequently correct it directly and apply it to its position. The accuracy obtained with this system is of less than 5 centimetres in longitude and latitude. The inconvenience is that the radio signal has a maximum range of 10 kilometres and that the station must be placed on land and have known coordinate points - normally geodesic points - near our working zone.

The Omnistar system is simpler, as it is a based on the ship receiving the already corrected and very accurate signal from a satellite. This satellite does not belong to the GPS system, but it belongs instead to a company that rents its signal through payment of an annual fee for its reception. This company

Source:
http://www.novatel.com/products/index.htm

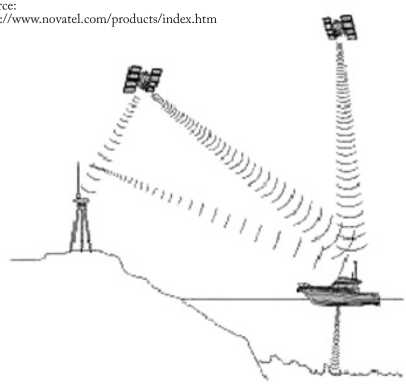

Figure 2. Scheme of the omnistar system (novatel).

sends the correction through powerful radios to its satellites and these, in turn, send it to "its clients". The accuracy is somewhat worse than the RTK system, having an error range comprised between 10 and 15 centimetres in longitude and latitude.

The advantage of this system with respect to the RTK is that it does not need a land station and that it is only necessary to have a small size receiver antenna in the ship. The inconvenience is the yearly payment to rent the satellite.

Movement sensor

In order to keep the typical movements of a ship in the water from altering the measurement of an echo sounder, a movement sensor is installed in a central area of the ship, that is, over the centreline of the ship and as close as possible to the water line.

While navigating the ship experiences movements through the hydrodynamic action of the water such as the roll, pitch and yaw. Furthermore, there are movements that occur due to the action of an unbroken wave that vertically elevates the ship without producing a pitch.

Source:
http://www.tss-international.com/products2.html

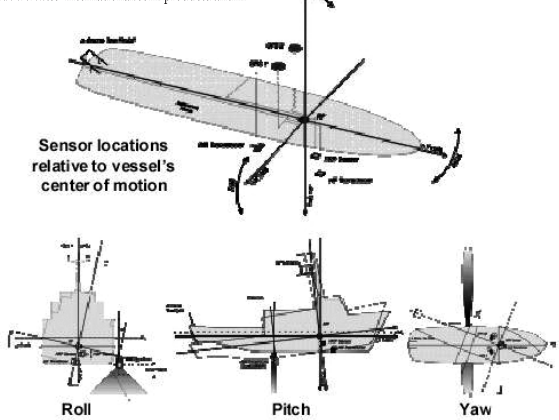

Figure 3 – Scheme of the movements of a ship (mahrs).

The movement sensor corrects these deviations from the "rest" position of the ship instantly, for which at the time of calculating the sounding in each point, this offset is already corrected.

Apart from these elements, it is of great importance to take into account the effect of the tide, given that it significantly alters the bathymetric measurement. That is, if we are measuring depths at one same point, it will vary according to the amplitude of the existing tide. This makes it necessary to have a tidal amplitude record of the working zone, which can be introduced, subsequently, in the data processing program of the echo sounder in order to correct its effect. There are different methods for this purpose, of which we will discuss the two main ones: the direct measurement of the elevation over the geoid given by the differential GPS, and the data taken from a tide recorder installed in the zone.

In the case of the differential GPS the method is simple: the geoid of the zone being measured is conveniently selected and the data of its elevation over the ship are gathered. This will give us a graph showing an elevation caused by the tide and some small elevations caused by sporadic waves. The series of data can be directly introduced into the echo sounder or can be done later after fulfilling a statistical cleaning of the data. The inconveniences of this procedure are that the biggest error of the GPS is found within the vertical data and, furthermore, that the ellipsoid must be well selected.

In regard to the tide gauge method, it is based on obtaining a temporary series of elevation data, but in this case, issued from a tide gauge installed close to the working area. These equipments are usually found in commercial ports and are generally well calibrated. They are property of the port Authorities, these being the ones responsible for their maintenance. Their data can be downloaded from the Internet almost in real time, thereby obtaining very accurate data. In the case that the closest port does not have a tide gauge, one can be installed with the drawback that a height with which to reference the tide measure is needed. Normally, the ports have known heights referred to the Port Zero in question.

The reference level, on which the bathymetric measurement obtained with the echo sounder is based on, is essential, since depending on that it uses a same measurement it will have different values. Normally the data will be referred to the Lowest Equinoctial Spring Tide or Port Zero, which is an even safer value for navigation as it includes the effect of the atmospheric pressure.

PROCEDURE IN A BATHYMETRIC MEASUREMENT

Demarcation of the Working Area

When planning a campaign for bathymetric measurement the demarcation of the working zone is fundamental. For this purpose, the sounder has navigation software, as if it were a chartplotter, which guides the skipper of the vessel and displays the areas which are being measured. A navigational chart is introduced in this software and if possible with the greatest detail of the coastal shapes.

It is necessary to have an order of magnitude of the surface, which can be measured in a day´s work in order to adjust the daily plots. These parcels will overlap with each other.

It is important to have a strict method in regard to filling the information, since each zone is associated to the tide taken on that same date.

The calibration of the speed of the sounders and of the position of the sounder´s transducer must be fulfilled prior to commencing the bathymetric surveys, as well as calibrating the speed of sound in water and the position and draft of the transducer (of the sounder). Theoretically, it should not move from its correct position but, for example, a slight variation in the list of the boat would cause errors in the measurement.

The calibration is fulfilled by placing an iron plate at a well-known distance from the water line, which will provide its accurate draft. This operation must be fulfilled in a sheltered area where the ship will experience the least movement making the calibration easier. In turn, the roll, pitch and yaw of the ship must be calibrated.

Methodology of the bathymetric measurement

The transducer installed has the capacity to read data in an angle that oscillates between 90º and 160º. Due to the way that a multibeam sounder works, the coverage will depend on the depth, finding a gradual coverage increase with the increase of the draft. The multibeams make it possible to measure 100% of the seabed, this scanning remaining reflected in the ship's navigator which is observed both by the skipper of the vessel and by the operator of the echo sounder; the first one to fill the area to be measured with the successive passes, and the other one to control the parameters of the echo sounder which will optimise its functioning.

To aid in this process of "filling in", that is, of covering with passes the entire seabed, it is convenient for the lines to be as straight as possible to ensure the quality of the data being collected. For this, the operator traces parallel straight lines so that the ship can follow them as if they were courses, and in this way the navigator follows them attaining a regular cover of the seabed.

The data at the extremes of the scan are the ones that can incur the most errors, since it has a lower angle of incidence, leading to usually eliminating the data obtained during processing. This processing is based on a computer program that statistically eliminates erroneous data. For this reason, the data between two consecutive passes must overlap to a certain degree.

http://www.novatel.com/products/index.htm

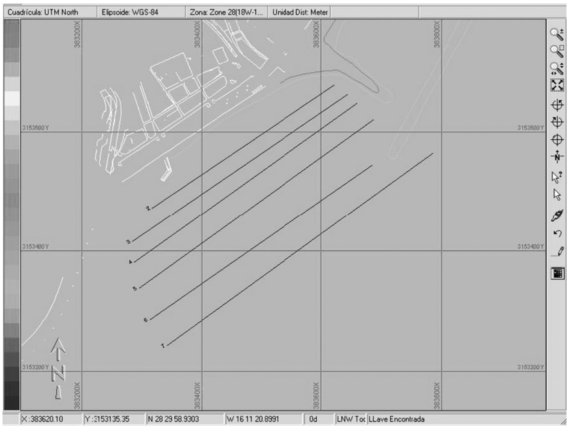

Figure 4. Image of the guide system of the ship with the lines of planned measurement.

Due to the irregularities that appear in the seabed, such as sudden changes in levels, rocks, etc, the lines that are being performed might not be as uniform as desired, for which the work must be revised to avoid leaving areas of the seabed unmeasured.

In the instance that the accuracy of the survey so requires it, like for example in the case of measurements in the interior of docks, navigable channels, dredging material volumes or dumping of harbour work material, the following procedures can be fulfilled to increase accuracy:

Two passes along each line of planned measurement.

The speed of each one of these passes must be the minimum possible to maintain the correct governing of the boat. In this way a larger number of points in the measure is obtained.

Following the conclusion of the measurements, it is then necessary to proceed to recording the data obtained by the sounder on a CD for its post processing. In this way we ensure having at least two copies of the work.

Methodology of the post process of the data obtained

During the data processing of the information gathered during the field bathymetry campaign, and in addition to the bathymetry and positioning data, we integrate the tide correction corresponding to the work period extracted from the tide recorder or the DGPS.

There are several software models with which to process the data although their functioning is basically similar. The data are statistically cleaned to eliminate wrong data produced by strange rebounds or acoustic noise and are transformed into bathymetric curves exportable to different formats, such as for example, the CAD or in 3D models.

All of these processes are fulfilled graphically and remain reflected in the data tables.

Source: http://www.sidmar.es/sidmar.htm

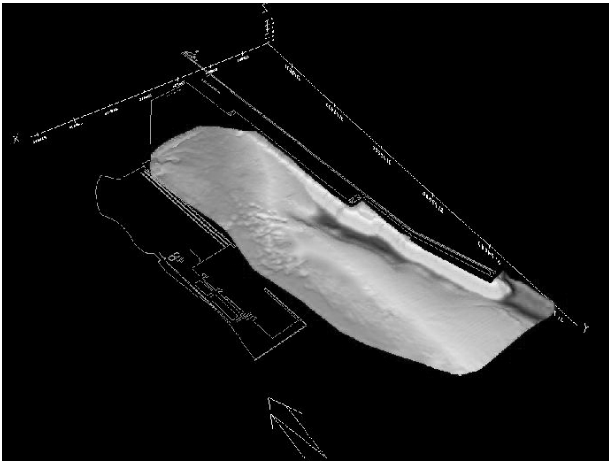

Figure 5. Image of the result of a bathymetry in the interior of a port.

CONCLUSIONS

The arrival of multibeam hydrographic echo sounders at the start of the eighties implied a true revolution in seabed mapping. There was a change from validating sporadic data to having a continuous record of the seabed. These technologies even make it possible indirectly to obtain a highly precise verification of the composition of the seafloor with errors smaller than 10 centimetres.

These methods are becoming popular at a very fast pace in the drafting of maritime work projects, given that a better project will arise from good data. Some Port

Authorities already have precision bathymetries in their ports, with perfectly demarcated depths under each mooring bollard.

It is to be expected that in the next few years its use will become even more generalized, with the subsequent increase in navigation safety in areas of shallow waters.

REFERENCES

Capasso I. and Fede S. (1981) Navigazione. Milano: Ulrico Hoepli.

Granata, T., Duarte, C. and Garcia, E. (1999) Modification of the bottom boundary layer by the seagrasses. Madrid: Estuarine, coastal and shelf science.

Ingham, A. (1992) Hydrography for the Surveyor and Engineer. Plymouth: Blackwell Science.

Lowrie, W. (1997) Fundamentals of Geophysics. Cambridge: University press, 354.

Pérez, F. (2006) Los sondadores monohaz. Madrid: Revista General de Marina, Agosto-Septiembre, 273-289.

Pepkin, B., et al (1997) Oceanography. San Francisco: W. H. Freeman and Company.

Pickard, G., et al (1990) Descriptive Physical Oceanography An introduction. Oxford: Pergamon.

Sheriff, R. (1989) Geophysical Methods. New York: Prentice Hall.

Summerhayes, C. and Thorpe, S. (1996) Oceanography, an illustrated guide. Southampton: Ed. Manson.

Tetley, L. and Calcutt, D. (1991) Electronic Aids to Navigation. London: Edward Arnold.

US Corps of Engineers (2003) Shore Protection Manual. Washington: US Corps of Engineers.

Hydro Acoustics: http://www.hydroacoustics.com/frameset.htm

Hypack: http://www.hypack.com/

Novatel: http://www.novatel.com/products/engines.htm

Sidmar. Bernhard Pack S.L.: http://www.sidmar.es/sidmar.htm

T.S.S. S.G. Brown: http://www.tss-international.com/products2.html

CARTOGRAFIADO DE FONDOS MARINOS

La hidrografía, en general, es similar en muchos aspectos a la investigación en tierra y muchas de las técnicas empleadas son las mismas, o una extensión de ellas. Las cartas náuticas son el equivalente marino a los mapas topográficos. Ambos utilizan medidas de puntos de altura/profundidad y contornos para mostrar una imagen, pero mientras que la persona que utiliza el mapa puede verificar los detalles mostrados con una inspección visual, el usuario de la carta no lo puede hacer.

Las mediciones marinas siempre han sido menos precisas que sus equivalentes en tierra, aunque en los últimos años con el desarrollo tecnológico de transductores y sistemas de posicionamientos (GPS diferencial, posicionamiento dinámico, sistemas inerciales, etc.) se ha logrado una gran evolución, obteniendo unos resultados casi equivalentes a los obtenidos en tierra. Debido a esta evolución el cartografiado del fondo marino ha mejorado enormemente su precisión en los últimos tiempos. Desde el arcaico pero efectivo escandallo, que prevaleció hasta el siglo XX, se ha pasado a las actuales ecosondas multihaces que cartografían el fondo marino con una gran exactitud.

Desde siempre, se ha hecho necesario conocer la profundidad que existe en determinadas zonas someras, pues de ese dato dependía la seguridad del barco en caso de una recalada a un puerto o bahía, dado que, si se conocía el calado con exactitud en condiciones de bajamar, se permitía la entrada de determinados barcos con un calado que, como máximo, fuese igual al medido más el margen de seguridad establecido. Además del dato del calado, es importante conocer la composición del fondo. Evidentemente, no es lo mismo que un barco toque fondo sobre roca o arrecife coralino, que sobre un lecho fangoso, pues en este último caso es bastante factible recuperar el barco en caso de varada. Del mismo modo, dependiendo del tipo de fondo existente, serán más adecuadas unas anclas que otras.

Los equipos de estudios batimétricos se instalan en barcos preparados para el cartografiado de fondos. Suelen ser barcos oceanográficos en el caso de cartografías de aguas profundas y barcos de pequeño tamaño en las de aguas someras. En el primer caso, la función del estudio detallado de las zonas profundas tiene un importante componente investigador, mientras que en el caso de las batimetrías de aguas costeras o someras, el fundamento es más técnico, como en el caso de los canales navegables que se deben dragar periódicamente, las obras portuarias, regeneraciones de playas, etc.

La ecosonda es un sistema de telemetría basado en ecos adaptado al medio marino. Este aparato emite las ondas de sonido a una frecuencia determinada que, tras rebotar en el fondo marino, las recibe de nuevo. Tras esto, el aparato mediante un sencillo cálculo halla la distancia entre la ecosonda y el fondo. Dado que se conocen los datos de velocidad de desplazamiento de las ondas y el tiempo entre emisión y recepción, la distancia recorrida por la onda en la mitad de este tiempo, es decir, la distancia desde que la onda se emite hasta que choca con el fondo, es la profundidad entre el barco y el fondo.

Hay dos tipos de ecosonda, la multihaz y la monohaz, que difieren bastante una de la otra en cuanto a alcance, frecuencia, amplitud de los haces, etc.

Como su propio nombre indica, una ecosonda multihaz tiene varios haces, por uno sólo de una monohaz. Esta última emite un haz de ondas dirigidas hacia abajo con un ángulo de 1º,5. Esto implica que con la monohaz conoceremos solamente la profundidad existente justo debajo del barco. En cambio, la multihaz emite multitud de haces, entre 101-80 según el modelo, consiguiendo una cobertura de 6 a 7 veces la profundidad.

La gran ventaja de la tecnología multihaz es la elevada cobertura del fondo que permite barrer una gran superficie de fondo con una sola pasada por encima, lo que permite asegurar que, a diferencia de la ecosonda monohaz, entre líneas no adyacentes no nos hemos dejado ninguna zona muerta.

Para minimizar los errores hay que tener en cuenta una serie de variables que son:

— Velocidad del sonido en agua
— Posición del transductor
— Sistema de referencia
— Sensor de movimiento

Aparte de estos elementos, es de suma importancia tener en cuenta el efecto de la marea ya que altera de forma importante la medición batimétrica.

PROCEDIMIENTO EN UNA MEDICIÓN BATIMÉTRICA

Delimitación del Área de trabajo

En la planificación de una campaña de medición batimétrica es fundamental la delimitación de la zona de trabajo. Para esto, la sonda tiene un software de navegación, como si fuera un chartplotter, que guía al patrón de la embarcación y le muestra las zonas que va midiendo. En este software se introduce una carta náutica, a ser posible con el mayor detalle de las formas de la costa.

La calibración de la velocidad de las sondas y de la posición del transductor de la sonda se deben realizar antes de empezar los trabajos batimétricos, así como también se llevarán a cabo los calibrados de la velocidad del sonido en el agua y de la posición y calado del transductor de la sonda.

La calibración se realiza poniendo una plancha de hierro a una distancia de la línea de flotación perfectamente conocida. De este modo sabremos exactamente el calado del mismo. Esta operación de debe realizar en una zona resguardada para que el barco se mueva lo menos posible y la calibración sea más fácil. A su vez se debe calibrar el cabeceo (roll), balance (pitch) y alineamiento (yaw) del barco.

CONCLUSIONES

La aparición de las ecosondas hidrográficas multihaces a principios de los años ochenta, supuso una auténtica revolución en el cartografiado de los fondos marinos. Se pasó de validar datos puntuales a tener un registro continuo del fondo. Estas tecnologías incluso permiten indirectamente verificar la composición del fondo marino, siendo su precisión muy alta, con errores menores de 10 centímetros.

Estos métodos se están popularizando a pasos agigantados en la redacción de proyectos de obras marítimas, pues de unos buenos datos, saldrá un mejor proyecto. Algunas Autoridades Portuarias ya disponen de batimetrías de precisión de sus muelles, con profundidades perfectamente delimitadas bajo cada noray.

Es de esperar que en los próximos años su uso se generalice aún más, con el consiguiente aumento de la seguridad de la navegación en zona de aguas someras.

Journal of Maritime Research, Vol. V. No. 1, pp. 47-64, 2008

ISSN: 1697-4840

DESIGN AND SIMULATION OF A VIRTUAL TUBULAR HEAT EXCHANGE UNIT FOR EDUCATIONAL APPLICATIONS

E. Eguía[1], A. Trueba[2] and M. M. Milad[3]

Received 24 August 2007; received in revised form 9 September 2007; accepted 22 February 2008

ABSTRACT

One of the tasks of an industrial plant designer is the choice of the most suitable elements that will make up the said plant, so that all the devices complement and are synchronised with each other. If the industrial plant includes several heat exchanger units, each one of these should match its particular working conditions, which will be determined by the work process each exchanger performs. When designing and choosing the most appropriate heat exchanger for a set work process, certain international standards must be observed and likewise the recommendations of the classifying societies and the methods currently used in industry all have to be taken into account. In view of this, and taking into account the most advanced methods in heat exchange unit design, a computer program has been created that enables us to design a virtual heat exchanger that fits each of our needs and on which we can simulate different thermal processes and analyse its behaviour in all kinds of situations.

Keywords: Heat exchanger, design, simulation, software.

INTRODUCTION

In any heat exchange unit calculation, a series of economic factors that are going to have a direct effect on the final performance of the installation must be assessed. It

[1] University Professor, Department of Sciences & Techniques of Navigation and Ship Construction (eguiae@unican.es), Cantabria University, Santander, Spain. [2] Assistant Professor, Department of Sciences & Techniques of Navigation and Ship Construction (truebaal@unican.es), Cantabria University, Santander, Spain. [3] Ph.D. in Marine Sciences (milad@ono.com), Cantabria University, Santander, Spain.

is essential to carry out a study of the variables that have an influence both on heat transfer and on losses of head. Resistance to heat transfer is minimised when working with a high Reynolds Number. This enables a heat exchanger to be projected with a low heat exchange surface area thus reducing the cost of the product. However, a high Reynolds Number leads to an increase in loss of head, thus making it necessary to use an impulse pump with a greater range, hence making the product more expensive (Kreith and Bohn, 2001).

A factor that must be taken very much into account during the design of a heat exchanger unit is the internal fouling of the tubes. Biological deposits adhered to the inside surface of the tubes represent a huge problem both for land and for sea based industry, as they reduce heat transfer and therefore the cycle's thermal performance (Eguía, 1998). Thus, when the time comes to make calculations, the fall in the transmission coefficient over the period when the exchange unit is active will have to be taken into account. Likewise, a differentiation will have to be made in the working conditions depending on whether it is new or if it has already been in service and for how long.

Furthermore, for certain inlet conditions and certain operating fluid velocities, it is necessary to consider the concept of marginal transmission. Heat flow is directly proportional to the mean temperature difference of the operating fluids, and it decreases as the heat exchange surface area rises. Therefore, in the heat exchange unit financial calculation we must take into account that an increase in the surface area corresponding to the marginal transmission implies an additional cost. This cost will have to be assessed versus the increase in calories recovered. Normally we consider economic approximation to the lower difference in temperatures taken at each of the ends (it is necessary to introduce an efficiency factor specific for each kind of exchanger unit into the calculation (Rohsenow et al., 1998)).

It should be emphasized that the optimisation calculation for a heat exchanger is a complex process, as it forms part of a whole installation and cannot be dealt with in isolation. All of this complex calculation process is simplified when simulators are created that enable exact and rapid data to be obtained by taking all the technical and economic imperatives of today's industry into account.

This has therefore led to the creation of a thermal-hydraulic design program for shell and tube heat exchangers like the one shown in Figure 1. It starts from initial data supplied by the client such as the operating fluid inlet and outlet flows and temperatures, and its main features are:

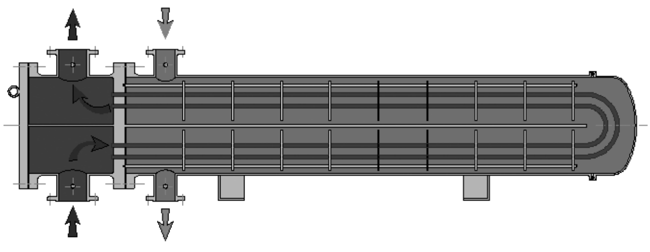

Figure 1. Shell and tube type heat exchanger.

— Improvements in the heat exchanger design and evolution process.
— Ease of calculation of the thermal-hydraulic process and of heat exchange.
— Calculation of the fluid stream convection coefficients.
— Calculation of the overall heat transmission coefficient.
— Calculation of dimensionless values such as the Reynolds Number, Nusselt Number, etc.
— Heat transmission by conduction in multiplayer pipelines and walls.
— Automatic attainment of the properties of certain fluids such as fresh water, salt water and air.
— Calculation of the absolute viscosity of 99 fluids, based on their temperature and density.
— Automatic unit conversion, making them easier to be handled and enabling the user to introduce data in any available unit.

The calculated data may be converted or transformed into the unit favoured by the user.
— Calculation of the thermodynamic properties of water vapour and Mollier diagrams up to 1000ºC.
— A Moody diagram for friction in conduits and pipelines.
— The graph presentations have a real zoom function and are designed for easy reading.
— Working by use of independent windows and calculation modules.
— Data is saved on a standard database, thus enabling it to be transported and shared with other programs.
— It employs modern programming technology and an "intelligent" data introduction module.

The design process is carried out using the method recommended by TEMA (TEMA, 1999), given that the said method may be applied in computer assisted design techniques (Leong et al., 1998).

Once the heat exchanger has been designed, the target is that it fits as good as possible the heat exchange process that it is going to perform in normal real conditions, simulating different operating situations:
— Normal or design conditions in which the heat exchanger works at full capacity and without going into head losses.
— Abnormal conditions with unpredictable situations that have not been taken into account during the calculation and design phase.
— Critical conditions in extreme operating situations before entering into head losses.

For example, in the case of conventional land or sea based thermal plants using seawater as their coolant fluid, the temperature of the said coolant would vary depending on the geographical area and the season of the year. It is therefore necessary to use a simulation program to find out how the heat exchanger will behave before such variations in temperature or to what temperature it will work at an acceptable performance level.

Hence, a simulation program has been created in order to study and predict heat exchanger behaviour in a variety of working conditions. It allows us to (Kotake and Hijikata 1993):

— Simulate functioning in normal conditions while varying environmental parameters within the set margins.
— Study the effect caused by a variation in each of the exchanger parameters and features on the other parameters and environmental conditions.
— Calculate its working range limits.
— Study heat exchanger behaviour in extreme situations and identify potential failures under these conditions.
— Study and analyse behaviour in abnormal conditions that may arise for a short period.
— This simulation module is an interesting tool for studying and learning how a heat exchanger works, obtaining a more reliable and balanced design.

THE HEAT EXCHANGER DESIGN PROCESS

The design of a heat exchanger for a specific process involves handling of a series of data and parameters that are going to determine its features. It is the designer's job to obtain the said data and parameters for the work process for which it is going to be designed. Likewise, it is the designer who should choose the design method to be followed (Minkowycz et al., 1988).

Definition of the Process

The client determines the heat transmission process that is required in his plant. An example process scenario might be a 1–2 shell and tube type heat exchanger to cool 19 kg/s of an organic liquid from 71ºC to 49ºC, using water as the coolant fluid with an inlet temperature of 21ºC. The outlet temperature must not exceed 49ºC. The following aspects would then be set:

— Material of which the tubes are made: steel tubes (14 SWG)
— Outer diameter of the tubes: 19 mm
— Length of the tubes: 243.8 mm
— Arrangement: Triangular and 25.4mm separation between tubes
— The coolant fluid fouling factor: 0.00018 m^2 K/W

The physical properties of the fluid to be cooled also have to be taken into account for the given range of working temperatures:

— Dynamic Viscosity $\mu =$ 0.45 centiPoise
— Density $\rho =$ 882 kg/m^3
— Specific Heat $c_p =$ 0.93 kJ/kg K
— Prandtl Number $N_{pr} =$ 4.356

The program automatically calculates the physical properties of a water coolant; therefore it is not necessary to use tables when working with either fresh or salt water.

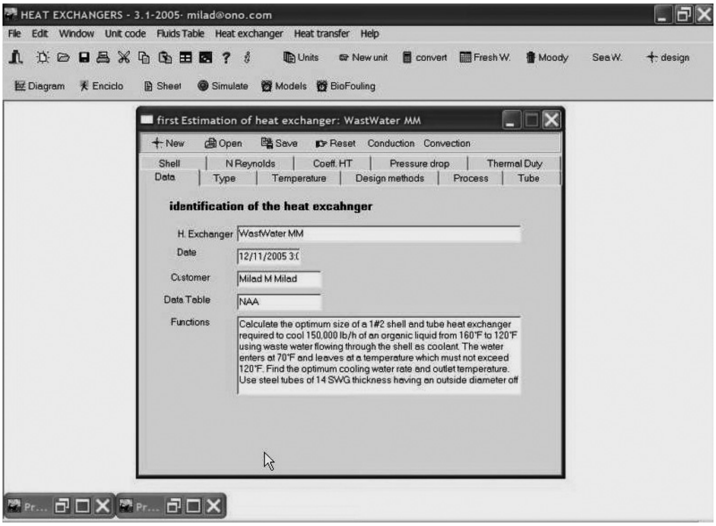

Figure 2. Main window of the design program.

To Create a New Design

The design module is started from the main program window (Figure 2). Access to the heat exchanger primary estimation window is attained by introducing the following variables into their corresponding windows (Kim et al., 2002):

— *Heat exchanger identification data:* Name, date, author and process description. This data is saved on a database. The said database may store different designs, which may be modified whenever required.

— *Choice of the type of heat exchanger identification*: The choice of the heat exchanger and the flow type will depend on the fluids that take part in the process, e.g. liquid-liquid, vapour-liquid, etc.

— *Process temperatures*: Here we introduce the inlet and outlet temperatures and mass flow rates along both the tube side and the shell side.

— *Mean caloric temperatures*: These are necessary to automatically calculate the physical properties of the fluids, the convection coefficients both on the shell side and on the tube side, and also the estimated fouling coefficients must be introduced. Likewise, the properties of the operating fluids, both those inside the tubes and through the shell, are also introduced. If an organic liquid is employed, the user must put this data in manually; however if the coolant fluid is water, its properties are already included in the program. If other fluids are used, their dynamic viscosity can be calculated from their density. The program has a list of 99 fluids available.

— *Calculation of the energy involved in the process:* In the *Process* window we can calculate the heat exchanger effectiveness and the thermal capacity relationship of the fluids. This data is necessary to calculate the effective temperature correction factor. We then go on to calculate the mean logarithmic temperature and the effective temperature. Likewise, the correction factor is calculated by clicking on the "Correction Factor" button and selecting the type of heat exchanger, following the TEMA classification. It has to be borne in mind that the correction factor must be over 0.75. If this is not the case it is advisable to change the configuration. We then go on to calculate the heat flows, the overall heat transmission coefficient and the heat transfer surface area required by the process.

— *Tube dimensions:* If the client supplies this data, it is introduced into the corresponding data cells. However, if no data is supplied, there is a dropdown menu showing tube diameter, pitch and arrangement (Serna and Jimenez, 2004).

— *Shell Dimensions:* First we select the number of shell pitches from the dropdown list (following the TEMA classification), and then we calculate the number of tubes to thus obtain the diameter of the shell. We should also introduce the tolerance between the shell and the tube bundle; this value should lie between 10 and 20 mm. As shell dimensions are standardised, we will have to choose the shell that just exceeds the calculation results obtained. In this section, we also calculate the shell nozzle that will condition the head losses due to the fluid velocity, cavitation and erosion of the tubes. Mass fluid state, density and rate enable us to derive the minimum nozzle diameter, the maximum velocity and the distance from the nozzle to the tube bundle.

— *Calculation of the Reynolds Number:* Firstly, we calculate the pitch section of the tube bundle, i.e. the sum of the sections of all the tubes that form the tube bundle and this is treated as if it were a single tube with a section that is the total we have calculated. Then we calculate the wear in the tubes and the Reynolds Number. These values will subsequently be used to obtain the film by convection coefficient in the tubes.

— *The overall heat transmission coefficient:* Due to the temperature difference between the wall and the core of the main stream of the two fluids, we must add the so-called *viscosity factor.* It is therefore necessary to calculate the tube wall temperature and obtain the viscosity of both fluids at this temperature. The program carries out these operations automatically. Furthermore, the option exists for introducing the *fouling factor* into the calculation.

— *Tube wall temperature:* Due to the temperature difference between the wall and the core of the main stream of the two fluids, we must add the so-called viscosity factor. It is therefore necessary to calculate the tube wall temperature and obtain the viscosity of both fluids at this temperature. The program carries out these operations automatically. The tube fouling coefficient is similarly applied.

— *The convection coefficient in the tubes:* This module includes the calculation, amongst others, of the Nusselt Number, the convection coefficient and the pressure drop in the tube bundle. It is also possible to carry out flow and heat transmission analyses in tubes that are independent of the heat exchanger as a whole. We must begin by selecting the type of configuration to be analysed from the three options available: The first option, "pipelines and conduits" is suitable for analysing and calculating the flow in an isolated pipeline in which fluid flows and there is a heat energy exchange with the outside. In this case, there is a drop down list for the user to select the option he requires from the various calculation formulae, depending on the pipeline and flow characteristics and his particular preferences. The second option, "tube bundle", refers to the calculation for the heat exchanger as a whole: To calculate the convection coefficient we must go through the following steps:

- The properties of the fluid inside the tubes are derived; both at mean and wall temperatures.
- The calculations can be made *step by step* or in *automatic* mode.
- The fluid velocity inside the tubes is calculated.
- The mean temperature inside the tube is calculated.
- The Nusselt Number is calculated.
- The convection coefficient is calculated.

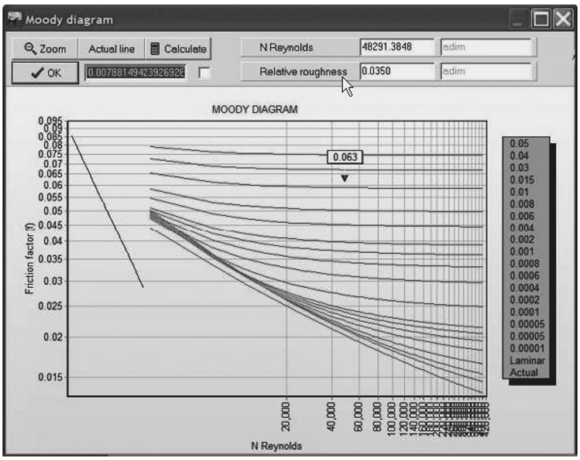

Figure 3. Moody diagram showing the current work point, the relative roughness lines and the friction factor.

The Moody diagram enables us to obtain the pressure drop in the tube bundle. Setting the relative roughness value, the friction factor is calculated and when we close the "Moody" window, the pressure drop in the tubes is calculated automatically. The Moody Diagram module (Figure 3) shows the point we are currently at and the relative roughness lines together with the results of the friction factor calculation.

On concluding the calculation and closing the convection in the tubes module window, we go back to the design window that now shows the results in their corresponding data cells.

— *Calculation of the flow parameters and the shell deflectors:* The profile of the shell fluid stream between the deflectors is set as a combination of cross, counter-current and parallel flows. The calculation of the deflectors is a process that directly affects heat exchanger performance; therefore an independent module has been created to calculate and design all the shell flow properties. We have to go through the following steps:

- The calculations can be made step by step or in automatic mode.
- If the system of units is incompatible, the program automatically undertakes the conversion.
- The "*basic data*" window shows the values calculated in other sections that are necessary here, and it is here that we choose the type of arrangement of the tube bundle (triangular, square, at 30º or at 45º).
- In the "*J Factors*" window the following values are calculated: The transverse area or the equivalent diameter which corresponds to the section the fluid passes through in the shell; the minimum tolerance required between the shell and the flow deflectors obtained from the tables; the maximum space between the deflectors; the tolerance between the tubes and the deflectors, the value of which may range from 5 to 10 mm; and the shell film coefficient correction factors, "J_c, J_L, J_B, J_r, J_s", which are involved in the calculation of the convection coefficient of the outside of the tube bundle. These are calculated by the program.

The "*Shell film coefficient*" window shows the parameters that are automatically calculated by the program and we can obtain the physical properties of the shell fluid by clicking on "*Shell fluid properties*". Calculations are also made for the shell wear, the Reynolds Number, the total J factor expressed numerically or in graph form versus the Reynolds Number and the shell film coefficient.

— The "*Pressure drop*" window serves to calculate the pressure drop correction factors, "R_L, R_b, R_s", and the pressure drop in a bank of ideal tubes to which we will later add the most suitable correction factor for the heat exchanger to be designed. We can also obtain a graph representation of the friction coefficient versus the Reynolds Number. The pressure drops in the shell are then calculated.

— *Thermal balance and design verification:* The aim of this module is to check that the caloric energy exchanged complies with the safety margin on the real process. The safety coefficient is a function of the fouling factor admitted by the heat exchanger before losses imply working beyond the design point. If the safety margin is small, the downtime programmed for its cleaning will be more frequent and the running periods will be shorter, whereas if the safety margin is too large we will have an oversized heat exchanger for the process to be undertaken. Therefore, the ideal situation is that the safety coefficient be close to the expected fouling factor.

Once the heat exchanger design has been concluded, we will only need to print off the data obtained and send this to the manufacturing firm.

AUXILIARY MODULE

In the event of operating with water vapour, an auxiliary module is available that shows the work line depending on the vapour properties in the Temperature-Entropy (T-s) diagram, the Enthalpy-Entropy (h-s) diagram and the Pressure-Enthalpy (p-h) diagram. It is also possible for the user to create his own vapour tables.

THE SIMULATION PROGRAM

The design program allows the user to simulate and analyse the calculated exchanger or alternatively an already existing exchanger whose operation the user wants to analyse.

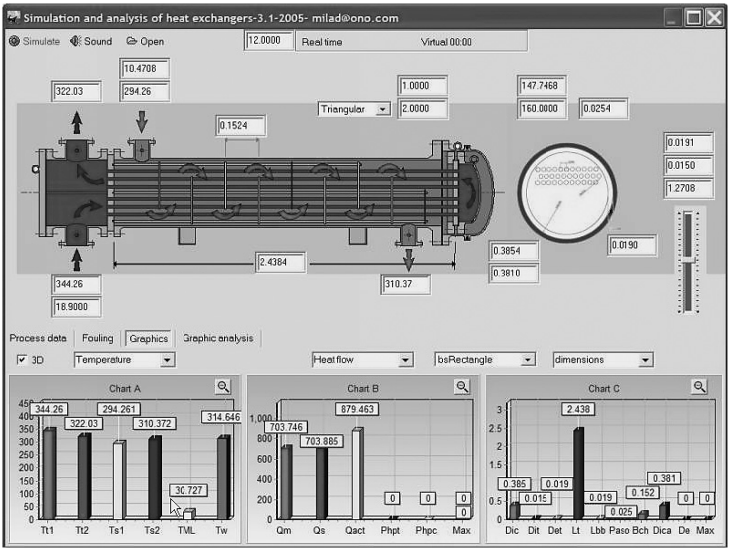

Figure 4. Main window of the simulation and analysis module.

A Description of the Program

In the top part of the main heat exchanger simulator and analysis module shown in Figure 4, there is a shell-tube type heat exchanger unit with its corresponding inlet and outlet nozzles. There are several boxes superposed on the drawing showing data regarding some of the features of the thermal process and the heat exchanger dimensions. The inlets of the cooled and coolant fluids show their temperature and rate of mass flow and the outlets indicate the temperature of the fluids. It likewise shows the dimensions of the exchanger: Tube length, the tube's internal and external diameter, pitch between the tubes and the tube diameter/pitch relationship, the total number of tubes, the shell diameter, arrangement of the tubes on the tube plate, the space between deflectors and the number of tube and shell pitches.

To access additional information for each parameter, we only need to click on the data cell to call up a more complete information box.

The bottom part shows a box of several superposed pages that give us information about:

— *Heat exchange process data.* In this section we can obtain the following numerical data: the physical properties of the operating fluids (their viscosity, density, specific heat, Prandtl Number and thermal conductivity); flow parameters (mass flow rates, velocities, wear and Reynolds Number); and general information concerning the thermal process (logarithmic mean temperature, correction factor, thermal efficiency, current and required area for heat transmission and a list of thermal capacities).

— *Fouling.* We are shown the general outcomes of the simulation: film coefficients, thermal energy transmitted by each fluid, current overall coefficient and the overall coefficient required by the thermal process under consideration, fluid velocities, cutting forces, resistance to fouling, the friction factor, pressure drop and pump force necessary for each fluid to circulate. As the program automatically determines the friction factors, we must access the Moody diagram module to find these results.

— *Graphs.* This section shows the outcome of the simulation and the exchanger dimensions and parameters in graph form, showing us in real time the variation of any parameter of the simulated heat exchange process. The graphs are ordered by variables (temperature, pressure, etc.) and can be selected using a dropdown menu.

— *Graph Analysis.* The graph analysis window offers us a graphic representation of the evolution of any parameter in relation to another parameter. We are able to simulate an infinite number of heat exchangers for a given thermal process to thus determine which is the most appropriate for that particular process.

RESULTS

Design Module

Once the client has defined the process and we have followed the abovementioned design module steps, the data obtained is tabulated so that the manufacturing firm has all the necessary parameters in design order to build the heat exchanger. An example of this is shown in Table 1 for a shell with an external diameter of 43.2 cm and an outlet temperature of 49°C.

Tube Inlet Temperature	21	°C

PARAMETER	RESULT	UNIT
Tube Outlet Temperature	49	ºC
Tube Mass Flow Rate	6	kg/s
Shell Inlet Temperature	71	ºC
Shell Outlet Temperature	49	ºC
Shell Mass Flow Rate	19	kg/s
DTML Logarithmic Mean Temperature	24.896	K
Effective Temperature Correction Factor	0.94969016	dimensionless
Effective Temperature	23.64	ºK
Tube Heat Flow	703.46	KW
Shell Heat Flow	703.46	KW
Estimated Overall Heat Transmission Coefficient	1,059	W/m^2K
Heat Transfer Area	28.091	m^2
Capacity Relation	0.799	dimensionless
Effectiveness	0.555	dimensionless
External Tube Diameter	19	mm
Internal Tube Diameter	15	mm
Tube Pitch	25.4	mm
Tube Length	243.8	cm
Number of Tube Pitches	2	dimensionless
Space between Reflectors	15.2	cm
Number of Tubes	192.5	dimensionless
Number of Shell Pitch	1	dimensionless
Internal Shell Diameter	419	mm
Mean Tube Temperature	35	ºC
Mean Shell Temperature	60	ºC
Pitch Section (Bundle)	181	cm^2
Mass Velocity (Tube)	1,041	kg/s m^2
Reynolds Number (Tube)	34.706	dimensionless
Pitch Section (Shell)	17.897	mm^2
Mass Velocity (Shell)	339.120	kg/s m^2
Equivalent Diameter	0	mm
Reynolds Number (Shell)	8.830	dimensionless

PARAMETER	RESULT	UNIT
Wall Temperature (Clean Wall)	45	ºC
Bundle Viscosity (Clean Wall)	0.45	centiPoise
Bundle Film Coefficient	11.617	W/m^2 K
Shell Viscosity (Clean Wall)	1	centiPoise
Shell Film Coefficient	1.763	W/m^2 K
Clean Overall Coefficient	1.478	W/m^2 K
Estimated Bundle Film Coefficient	2.840	W/m^2 K
Estimated Shell Film Coefficient	4.261	W/m^2 K
Bundle-Shell Tolerance	19	mm
Friction (Bundle)	0.0029579	dimensionless
Pressure Increase (Bundle)	1.2	kg/cm^2
Friction (Shell)	0.121986	dimensionless
Pressure Increase (Shell)	0.023	kg/cm^2
Required Heat	703.4648	kW
Required Area	20.124	m^2
Required Transmission Coefficient	991.743	W/m^2K
Tube Conductivity	30	W/mK
Fouling Factor	0.00018	W/m^2K
Shell Nozzle Diameter	6	cm
Density (Shell)	882	kg/m^3
Specific Heat (Shell)	1.675	kJ/kg K
Dynamic Viscosity (Shell)	0.45	centiPoise
Prandtl Number (Shell)	4.356	dimensionless
Conductivity (Shell)	0.1731	W/m K
Density (Tube)	995.264	kg/m^3
Specific Heat (Tube)	4.1725	kJ/Kg K
Dynamic Viscosity (Tube)	0.8321	centiPoise
Prandtl Number (Tube)	5.614	dimensionless
Conductivity (Tube)	0.616	W/m K
Viscosity (Tube)	0.8138	cst
Maximum Shell Nozzle Velocity	2.126	m/s
Minimum Nozzle Diameter	60.4	mm
Nozzle-Bundle Distance	20	mm

Table 1. List of the data obtained for a shell with an external diameter of 43.2 cm and an outlet temperature of 49 ºC.

SIMULATION PROCESS

Once we are aware of the heat exchanger design features and of the parameters of the thermal process that take place therein, we can use the *SIMULATION module* to view the evolution of the different variables in relation to any other variable and thus analyse the behaviour of the heat exchanger under all possible working conditions.

— *Variation of the tube outlet temperature in relation to the flow volume in the shell.* If we introduce a variation interval in the rate of mass flow, after a few seconds of simulation we obtain the graph shown in Figure 5. During the simulation process we can observe how the bar graphs show the evolution of the heat exchanger parameters. We can also vary simulation speed by clicking on the *Speed Bar*, in order to better perceive how the parameters vary. Likewise, these results may be simulated under the effects of fouling. We thus aim to analyse how biofouling affects the operation of the exchanger and its localised effect on a specific parameter. Figure 5 shows the comparative effect of fouling on tube outlet temperature versus the shell mass flow rate in a clean exchanger (green line) and in an exchanger with fouling (purple line). We can observe that when there is biofouling for a resistance of 0.00018 m^2 K/W, there is a 60 % increase in the rate of mass flow, which represents a considerable economic loss. This explains the importance of study and research into fouling and the methods for its elimination, especially in condensers and/or coolers in conventional, nuclear, land or sea-based thermal power stations. It also shows the variation of shell fluid inlet temperature and mass flow rate versus the tube fluid outlet temperature. Figure 5 shows that a variation in the shell mass flow rate leads to an inverse variation in the tube outlet temperature. The red line represents the scenario of a variation in shell fluid inlet temperature. This simulation allows the user to predict how the heat exchanger will work and the point to which it can fulfil its mission without entering into losses.

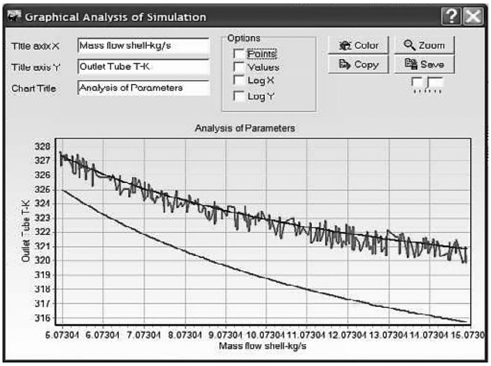

Figure 5. Variation of the tube outlet temperature in function of the mass flow rate in the shell with and without simulation of the effect of fouling.

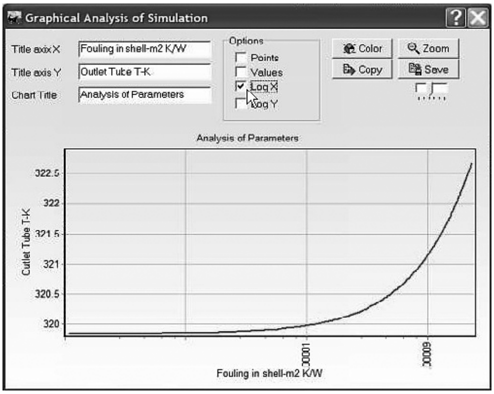

Figure 6. Variation of the tube outlet temperature in function of the resistance of the deposited biofouling.

— *Variation of the tube outlet temperature in relation to the biofouling deposited*. Figure 6 shows how the tube outlet temperature increases with the resistance to heat transfer exercised by the biofouling layer. Likewise, we can represent the effect of biofouling on the overall heat transmission coefficient or on the shell outlet temperature.

- *Variation of the mass flow rate and temperature of the fluid to be cooled.* When this occurs the heat exchanger suffers great thermal tension. Thus to avoid the heat exchanger entering into the area of low thermal performance, it is advisable to take these variations into account during its design or when deciding on the most suitable model. If the maximum mass flow rate of the coolant fluid is constrained by the features of the circulation pump and by the maximum velocity allowed to avoid tube erosion, we need to determine the margins for normal operation and calculate the variations it can bear to consider that it is operating at a satisfactory level. The first step is to calculate the maximum rate of mass flow allowed in the shell. To do so, we simulate a variation in the shell fluid velocity with the mass flow rate, thus attaining the maximum mass flow rate for the maximum velocity allowed that assures no damage will be done to the tubes. The variation in tube temperature versus flow volume or inlet temperature is simulated in the same way. The behaviour of the heat exchanger proves highly interesting when mass flow rate and temperature are varied at the same time, as shown in Figure 7. In this simulation we can view the heat exchanger work field when the outlet temperature is kept constant and the mass flow rate and the inlet temperature are varied. We can see that if the target is to maintain fluid outlet temperature constant (316 K/43ºC) the mass flow rate and inlet temperature must not exceed 348 K/75ºC and 20 kg/s or it would go into losses. It shows the heat exchanger limits for a fluid inlet temperature of 70º to 80ºC and a variation in mass flow rate from 15 to 25 kg/s with a maximum coolant flow volume of 21 kg/s.

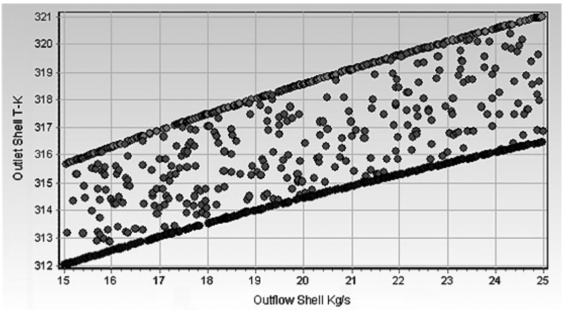

Figure 7. Heat exchanger behaviour when varying the flow volume and the temperature of the fluid to be cooled. Work field.

— *The effect of the tube length on the heat transmitted.* As the exchanger surface area increases more calorific energy is transmitted between the two fluids. In this simulation the tube length varies between 2 and 3 m and the variation on heat transmitted, the tube outlet temperature and the thermal efficiency are all analysed.

The results shown in Figure 8 indicate that the pressure drop in the tube bundle increases as the tubes increases in length. A variation can similarly be simulated in the outlet temperature or in the transmission of calorific energy when there is an increase in the length of the tubes.

— *The effect of the tube arrangement.* Figure 9 shows the variation in shell pressure drop with the variation of the mass flow rate according to how the tubes are arranged on the tube plate (triangle, square or inverse triangle). We can observe that the pressure drop varies according to the tube arrangement; therefore we must bear this factor in mind when working with critical pressure drops. We may likewise do the same with any other variable on the heat exchanger.

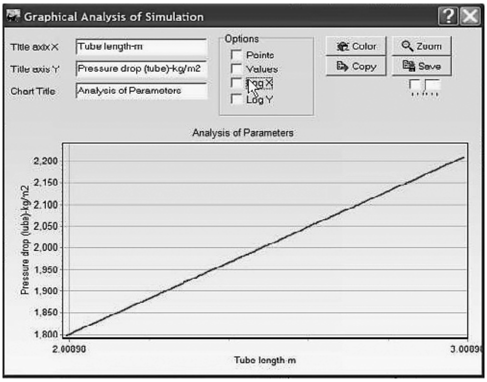

Figure 8. Pressure drop variation in relation to the tube length.

CONCLUSIONS

We have attained a profitable, reliable and didactic tool to assist engineers in industrial plant design and one that fosters an understanding of the processes and mechanisms that take place in heat exchangers. This tool facilitates engineers' work, because it allows them to easily design a heat exchanger, to assess and improve a heat exchanger that has already been designed, to simulate heat exchanger operation in different working conditions, to simulate and analyse biofouling growth and its effects on the designed heat exchanger's performance, as well as to analyse the design method recommended by TEMA.

The design results obtained are comparable to those obtained in real practice in industry, attaining an optimal design that matches the requirements of the heat exchanger's real functions.

A computer simulation of a heat exchanger is a complementary and necessary tool for obtaining an efficient and financially profitable design. It enables us to study the effect caused by a variation in any parameter on the other parameters, on the heat exchanger behaviour and on its performance. Furthermore, the simulations that can be carried out are unlimited and depend only on the user's requirements and imagination.

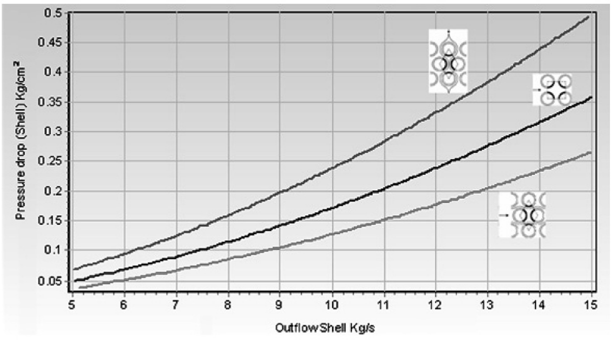

Figure 9. Variation of the pressure increase in the shell depending on the arrangement of the tubes.

REFERENCES

Eguía, E. (1998). The Problem of Biofouling in Heat-Condensing Interchangers Cooled by Water of Sea. Book Opening of the Academic Course 1998/1999. Cantabria University. Spain.

Kim, J.K., Lee, G.C., Zhu, F. and Smith, R. (2002). Cooling System Design, Heat Transfer Engineering,23(6):49–61.

Kotake, S. and Hijikata, K. (1993). Numerical Simulations of Heat Transfer and Fluid Flow on a Personal Computer. Transport Processes in Engineering, 3, Elsevier.

Kreith, F. and Bohn, M.S. (2001). Principles of heat transfer, Sixth edition, Brooks/Cole, Thomson.

Leong, K.C., Toh, K.C. and Leong, Y.C. (1998). Shell and Tube Heat Exchanger Design Software for Educational Applications, Int. J. Engng Ed. Vol. 14, Nº 3, pp. 217-224.

Minkowycz, W.J., Sparrow, E.M., Schneider, G.E. and Pletcher, R.H. (1988). Handbook of Numerical Heat Transfer, ISBN: 0-471-83093-3.

Rohsenow, W.M., Hartnett, J.P. and Cho, Y.I. (1998). Handbook of Heat Transfer, Third Edition, McGraw Hill.

Serna, M. and Jimenez, A. (2004). An Efficient Method for the Design of Shell and Tube Heat Exchangers, Heat Transfer Engineering, 25(2): 5–16.

TEMA (1999). Standards of TEMA, 8th Edition, Tubular Exchanger Manufacturers Association, New York.

DISEÑO Y SIMULACIÓN DE UN INTERCAMBIADOR DE CALOR TUBULAR VIRTUAL PARA APLICACIONES EDUCATIVAS

RESUMEN

Una de las tareas del diseñador de una planta industrial es la elección más adecuada de los aparatos que componen dicha planta, para que todos los dispositivos se complementen y estén sincronizados. Si en la planta industrial se disponen varios intercambiadores de calor, cada uno de ellos debe adecuarse a sus condiciones de funcionamiento particulares, las cuales serán función del proceso de trabajo que cada intercambiador realice. En el diseño y elección del intercambiador de calor más adecuado para un proceso de trabajo determinado, deberán respetarse unas normas internacionales y tenerse en cuenta las recomendaciones de las sociedades clasificadoras y los métodos utilizados en la industria actual. Con esto, teniendo en cuenta los métodos más avanzados en el diseño de intercambiadores de calor se crea un programa informático que nos permite diseñar un intercambiador de calor virtual adecuado a cada una de nuestras necesidades en el que podemos simular diferentes procesos térmicos y analizar su comportamiento en todo tipo de situaciones.

PROCESO DE DISEÑO DEL INTERCAMBIADOR DE CALOR

En el diseño de un intercambiador de calor para un proceso determinado se manejan una serie de datos y parámetros que lo van a caracterizar, siendo tarea del diseñador obtenerlos en función del proceso de trabajo para el que va a ser diseñado. Así mismo, el diseñador deberá elegir el método de diseño a seguir.

Definición del proceso

El cliente establece el proceso de transmisión de calor que necesita en su instalación, por ejemplo, un intercambiador de calor del tipo tubo-carcasa 1-2 en el que refrigerar 19 kg/s de un liquido orgánico de 71 ºC a 49 ºC utilizando como fluido refrigerante agua con una temperatura de entrada de 21 ºC, no debiendo exceder la temperatura de salida de 49 ºC, fijándose a continuación los siguientes aspectos:

— Material de construcción de los tubos: tubos de acero 14 SWG
— Diámetro exterior de los tubos: 19 mm
— Longitud de los tubos: 243.8 mm
— Disposición: triangular y 25.4 mm de paso entre tubos
— El factor de fouling por parte del fluido refrigerante: 0.00018 m^2 K/W

También se tendrán en cuenta las propiedades físicas del fluido a refrigerar en el intervalo de temperaturas de trabajo dadas:

— Viscosidad dinámica μ = 0.45 centiPoise
— Densidad ρ = 882 kg/m^3
— Calor especifico cp = 0.93 KJ/kg K
— Número Prandtl N_{pr} = 4.356

Las propiedades físicas del agua de refrigeración las calcula automáticamente el programa, por lo que no es necesario utilizar tablas ni con agua dulce ni con agua salada.

Crear un diseño nuevo

Se inicia el *módulo de diseño* en la ventana principal del programa (Figura 2) y se accede a la ventana de estimación primaria del intercambiador de calor introduciendo las siguientes variables en las ventanas correspondientes:

— Datos de identificación del intercambiador de calor
— Elección del tipo de intercambiador de calor
— Temperaturas del proceso
— Temperaturas medias calóricas
— Cálculo de la energía del proceso
— Dimensiones de los tubos
— Dimensiones de la carcasa
— Cálculo del Nº de Reynolds
— Coeficiente global de transmisión de calor
— Temperatura de la pared del tubo
— Coeficiente de convección en los tubos: en este módulo se calcula entre otros, el número de Nusselt, el coeficiente de convección y la caída de presión en el haz tubular.
— Cálculo de los parámetros del flujo y los deflectores en la carcasa
— Balance térmico y comprobación del diseño

PROGRAMA DE SIMULACIÓN

Descripción del programa

En la parte superior de la ventana principal del módulo de simulación y análisis del intercambiador de calor que se muestra en la Figura 4 se dispone un intercambiador de calor del tipo tubos-carcasa con las toberas de entrada y salida. Sobre el dibujo del intercambiador de calor hay sobrepuestas varias celdas que muestran datos sobre algunas de las características del proceso térmico y las dimensiones del intercambiador de calor. En las entradas del fluido refrigerado y del refrigerante se muestran la temperatura y el caudal y en las salidas se pueden ver la temperatura de los fluidos. Así mismo se presentan las dimensiones del intercambiador: longitud de los tubos, diámetro exterior e interior del tubo, paso entre los tubos y relación diámetro/paso tubular, número total de tubos, diámetro de la carcasa, disposición de los tubos en la placa tubular, espacio entre deflectores y número de pasos de los tubos y carcasa.

Para acceder a la información adicional de cada parámetro, basta con hacer doble clic sobre la celda para que aparezca un cuadro informativo más extenso.

En la parte inferior se muestra un cuadro de múltiples páginas sobrepuestas que nos ofrecen información sobre:

— Datos del proceso de intercambio de calor.
— Fouling.
— Gráficos.
— Análisis gráfico.

CONCLUSIONES

Se ha conseguido una herramienta rentable, fiable y didáctica para ayudar el ingeniero en el diseño de las plantas industriales y que permite comprender los procesos y mecanismos que tienen lugar en los intercambiadores de calor. Esta herramienta facilita las tareas del ingeniero permitiéndole diseñar fácilmente un intercambiador de calor, analizar y mejorar el intercambiador de calor diseñado, simular el funcionamiento del intercambiador de calor en diferentes condiciones de trabajo, simular y analizar el crecimiento del biofouling y sus efectos sobre el rendimiento del intercambiador de calor diseñado, así como analizar el Método de diseño recomendado por TEMA.

Los resultados de diseño obtenidos son comparables a los obtenidos en la práctica real en la industria, consiguiéndose un diseño óptimo que cumple con las funciones reales del intercambiador de calor.

La simulación de un intercambiador mediante ordenador es una herramienta complementaria y necesaria para obtener un diseño eficiente y económicamente rentable. Se puede estudiar el efecto de la variación de cualquier parámetro sobre los demás, sobre el comportamiento del intercambiador de calor y sobre su rendimiento. Además, las simulaciones que se pueden realizar son ilimitadas y dependen de las necesidades e imaginación del usuario.

Journal of Maritime Research, Vol. V. No. 1, pp. 65-78, 2008

ISSN: 1697-4840

A DERIVATION OF HIGH-FREQUENCY ASYMPTOTIC VALUES OF 3D ADDED MASS AND DAMPING BASED ON PROPERTIES OF THE CUMMINS' EQUATION

T. Perez[1,3] and T. I. Fossen[2]

Received 5 September 2007; received in revised form 15 September 2007; accepted 15 March 2008

Abstract

This brief paper provides a novel derivation of the known asymptotic values of three-dimensional (3D) added mass and damping of marine structures in waves. The derivation is based on the properties of the convolution terms in the Cummins's Equation as derived by Ogilvie. The new derivation is simple, and no approximations or series expansions are made. The results follow directly from the relative degree and low-frequency asymptotic properties of the rational representation of the convolution terms in the frequency domain. As an application, the extrapolation of damping values at high frequencies for the computation of retardation functions is also discussed.

INTRODUCTION

The ability to predict ship responses and loads in waves is an important tool in the design of marine structures and motion control systems. One method of constructing time-domain models consists of using the data generated by the hydrodynamic codes to compute the different elements of the so called Cummins's equation of ship motion (Cummins, 1962). This equation contains some convolution terms which describe fluid memory effects. The kernel of the convolution terms (impulse responses) can be computed from the frequency-dependent potential damping. For

[1] Centre for Complex Dynamic Systems and Control—CDSC (tristanp@ieee.org), The University of Newcastle, Callaghan, NSW 2308, Australia. [2] Department of Engineering Cybernetics, Norwegian University of Science and Technology—NTNU, Norway. [3] Corresponding author (tristan.perez@ntnu.no).

these computations to be accurate, it is often necessary to compute the hydrodynamic damping for very high frequencies-even when at these high frequencies the response of the vessel is negligible.

When frequency-domain hydrodynamic codes are used to compute added mass and damping; there is a limitation on the highest frequency that can be reached, and this may limit the accuracy of the retardation function. Indeed, in order to have accurate results of added mass and damping, it is recommended to use a panelling size such that the panel characteristic length is less than 1/8th of the wave length (Faltinsen, 1993). This implies that in order to reach high frequencies, the size of the panels need to be reduced–with the consequence of a very large number of computations and time. One way to alleviate this problem consists of extrapolating the potential damping based on its high-frequency asymptotic trend.

The asymptotic values of damping and added mass have been discussed in the literature in terms of expansions and the Kramer-Kronig relations (Greenhow, 1986). The contribution of this paper is to provide a much simpler and more elegant derivation of such asymptotic values based on the properties of the convolution terms in the Cummins's Equation. The results obtained show that in the limit as the frequency tends to infinity, both damping and added mass tail towards their asymptotic values at a rate of ω^{-2}, which, as discussed by Damaren (2000), seems at odds with the traditional results of the hydrodynamic literature, but coincide as the limit is taken to high frequencies.

LINEAR TIME-DOMAIN EQUATIONS OF MOTION

For a marine structure at zero forward speed a linear time-domain model takes the following form (Cummins, 1962)[a]:

$$[\mathbf{M}+\mathbf{A}]\,\ddot{\boldsymbol{\xi}}+\int_0^t \mathbf{K}(t-\tau')\,\dot{\boldsymbol{\xi}}(\tau')\,d\tau'+\mathbf{C}\,\boldsymbol{\xi}=\boldsymbol{\tau} \tag{1}$$

Where the motion and force variables are

$$\boldsymbol{\xi}=\begin{bmatrix}\xi_1\\ \vdots\\ \xi_6\end{bmatrix},\quad \boldsymbol{\tau}=\begin{bmatrix}\tau_1\\ \vdots\\ \tau_6\end{bmatrix}$$

Note that **bold symbols** represent vector and matrix quantities.

[a] The lower limit in the integral term id (1) as derived by Cummins (1962) is -∞; however, under the assumption of causality, K(t)=0 for $t<0$, the a lower limit of 0 yields the same value for the convolution term in (1).

These variables represent the motion and excitation forces of the structure due to the waves with respect to a coordinate system fixed to the mean free surface. The matrix **A** is a constant added-mass matrix. The matrix **K**(t) is a retardation function; i.e., its entries are impulse responses from the velocities to the part of the radiation forces that represent the fluid-memory effects.

Ogilvie, (1964), took the Fourier transform of (1) obtained the following relations with the frequency-dependant added mass and damping:

$$\mathbf{A}(\omega) = \mathbf{A} - \frac{1}{\omega}\int_0^{\infty} \mathbf{K}(t)\sin(\omega t)\, dt \tag{2}$$

$$\mathbf{B}(\omega) = \int_0^{\infty} \mathbf{K}(t)\cos(\omega t)\, dt \tag{3}$$

From a theoretical perspective, the retardation functions can be evaluated from the frequency-dependent damping:

$$\mathbf{K}(t) = \frac{2}{\pi}\int_0^{\infty} \mathbf{B}(\omega)\cos(\omega\, t)\, d\omega \tag{4}$$

When a hydrodynamic code based on panel methods is used to compute the potential damping for different frequencies, there are limitations on the highest frequency for which the damping can be computed. In order to have accurate results of added mass and damping, it is recommended to use a panelling size such that the panel characteristic length is less than 1/8$^{\text{th}}$ of the wave length (Faltinsen, 1993). This implies that in order to reach high frequencies, the size of the panels need to be reduced–with the consequence of a very large number of computations and time.

Because of the finite-frequency **Ω**, there is an error in the computed retardation function:

$$\begin{aligned} \hat{\mathbf{K}}(t) &= \frac{2}{\pi}\int_0^{\Omega} \mathbf{B}(\omega)\cos(\omega t)d\omega \\ \mathbf{Error}(t) &= \frac{2}{\pi}\int_{\Omega}^{\infty} \mathbf{B}(\omega)\cos(\omega t)d\omega \end{aligned} \tag{5}$$

To minimize the error, it is necessary to increase the frequency **Ω** as much as possible. This can be achieved by considering the high-frequency trend of the damping.

PROPERTIES OF THE CONVOLUTION TERMS

It also follows from Ogilvie (1964) that[b]

$$\mathbf{K}(j\omega) = \mathbf{B}(\omega) + j\omega\,[\mathbf{A}(\omega) - \mathbf{A}(\infty)] \tag{6}$$

and also

$$\lim_{\omega \to 0} \mathbf{K}(j\omega) = \mathbf{0} \tag{7}$$

$$\lim_{\omega \to \infty} \mathbf{K}(j\omega) = \mathbf{0} \tag{8}$$

$$\mathbf{K}(t = 0^{+}) \neq \mathbf{0} \tag{9}$$

$$\lim_{t \to \infty} \mathbf{K}(t) = \mathbf{0} \tag{10}$$

Equations (7) and (8) establish the asymptotic value at low and high frequencies respectively. Equation (9) establishes that the initial time value of the retardation function is different from zero. Equation (10) establishes the bounded-input bounded-output stability of the convolution terms. For further details about these properties, see the APPENDIX.

Due to the linearity of the convolution terms, these can be represented in the frequency domain by a rational function:

$$K_{mn}(j\omega) = \frac{P_{mn}(j\omega)}{Q_{mn}(j\omega)} = \frac{b_p (j\omega)^p + b_{p-1}(j\omega)^{p-1} + \cdots + b_1 (j\omega)}{(j\omega)^q + a_{q-1}(j\omega)^{q-1} + \cdots + a_1 (j\omega) + a_0} \tag{11}$$

Note that in (11), we have omitted the constant b_0 term in the numerator polynomial so as to account for the fact that the convolution terms are zero at $\omega = 0$ –*c.f.* (7).

Equation (8) indicates that (11) must be strictly proper; i.e., the relative degree must be greater than zero (the relative degree is the difference between the degree of the denominator minus the degree of the numerator). Equation (9) has further implications on the relative degree of (11). Indeed, condition (9) follows from (2):

[b] The reader is reminded that we are considering the case of zero forward speed; and therefore $B(\infty)=0$. We will comment on the extensions to forward speed in a later section.

$$\lim_{t\to 0^+} \mathbf{K}(t) = \lim_{t\to 0^+} \frac{2}{\pi}\int_0^\infty \mathbf{B}(\omega)\cos(\omega\, t)\, d\omega = \frac{2}{\pi}\int_0^\infty \mathbf{B}(\omega)\, d\omega \neq \mathbf{0} \tag{12}$$

Note that regularity conditions are satisfied for the exchange of limit and integration; and the last equality follows from energy considerations, which establish that the diagonal terms of $\mathbf{B}(\omega)$; namely $B_{ii}(\omega) > 0$–see Faltinsen (1990).

From the initial value theorem of the Laplace Transform, it follows that

$$K_{mn}(t=0^+) = \lim_{s\to\infty} s\, K_{mn}(s) = \lim_{s\to\infty} \frac{b_p s^{p+1}}{s^q} = b_p \quad (if\;\, q = p+1) \tag{13}$$

which will be different from zero if and only if $p+1=q$. Therefore, the relative degree of (11) must be equal to 1 whenever $K_{mm}(t=0^+) \neq 0$ (the integral of the off-diagonal terms–those which are not uniformly zero due to symmetry of the structure–could be zero; but it is not so in most practical cases.).

HIGH-FREQUENCY ASYMPTOTIC VALUES OF DAMPING AND ADDED MASS

We now show the main contribution of the paper. Because of the rational representation (11)) of the frequency response of the convolution terms (6)), the following relations hold for the damping and added mass:

$$B_{mn}(\omega) = \mathrm{Re}\{K_{mn}(j\omega)\} = \mathrm{Re}\left\{\frac{P_{mn}(j\omega)}{Q_{mn}(j\omega)}\right\} = \frac{\mathrm{Re}\{P_{mn}(j\omega)Q_{mn}(-j\omega)\}}{Q_{mn}(j\omega)Q_{mn}(-j\omega)} \tag{14}$$

$$\omega[A_{mn}(\omega) - A_{mn}(\infty)] = \mathrm{Im}\{K_{mn}(j\omega)\} = \mathrm{Im}\left\{\frac{P_{mn}(j\omega)}{Q_{mn}(j\omega)}\right\} = \frac{\mathrm{Im}\{P_{mn}(j\omega)Q_{mn}(-j\omega)\}}{Q_{mn}(j\omega)Q_{mn}(-j\omega)} \tag{15}$$

From (11), it follows that

$$Q_{mn}(-j\omega) = (-1)^q (j\omega)^q + a_{q-1}(-1)^{q-1}(j\omega)^{q-1} + \cdots + (-1)q_1(j\omega) + a_0,$$

and then

$$\begin{aligned} Q_{mn}(j\omega)\, Q_{mn}(-j\omega) = {} & (-1)^q (j\omega)^{2q} + a_{q-1}(-1)^{q-1}(j\omega)^{2q-1} + \cdots + (j\omega)^q a_0 \\ & + a_{q-1}(-1)^q (j\omega)^{3q-1} + \ldots + a_{q-1}(j\omega)^{q-1} a_0 + \ldots + a_0^{\,2}. \end{aligned} \tag{16}$$

Similarly,

$$P_{mn}(j\omega)\,Q_{mn}(-j\omega) = b_p(-1)^q(j\omega)^{q+p} + b_p a_{q-1}(-1)^{q-1}(j\omega)^{q+p-1} + \cdots + b_p(j\omega)^p a_0 + \ldots + b_1 a_0(j\omega).$$

From the relative degree constraint ($p = q - 1$), the latter becomes

$$P_{mn}(j\omega)\,Q_{mn}(-j\omega) = b_p(-1)^q(j\omega)^{2q-1} + b_p a_{q-1}(-1)^{q-1}(j\omega)^{2q-2} + \cdots + b_{q-1}(j\omega)^{q-1}a_0 + \ldots + b_1 a_0(j\omega). \tag{17}$$

Using the properties of the power of the imaginary unit:

$$j^m = \begin{cases} (-1)^{\frac{m}{2}} & m - even \\ (-1)^{\frac{m-1}{2}}\, j & m - odd \end{cases} \tag{18}$$

on (16)) and (17)), it follows that in the limit as $\omega \to \infty$,

$$B_{mn}(\omega) \quad \to \quad \frac{b_p a_{q-1}(-1)^{q-1}(j\omega)^{2q-2}}{(-1)^q(j\omega)^{2q}} = \frac{b_p a_{q-1}}{\omega^2} := \frac{\beta_{mn}}{\omega^2} \tag{19}$$

$$j[A_{mn}(\omega) - A_{mn}(\infty)] \to \frac{b_p(-1)^q(j\omega)^{2q-1}}{\omega(-1)^q(j\omega)^{2q}} = j\frac{-b_p}{\omega^2} := j\frac{\alpha_{mn}}{\omega^2} \tag{20}$$

Since we are taking the limit as $\omega \to \infty$, the results above are obtained by considering the term of highest order in (16) that is real and the corresponding real and imaginary highest order term in (17).

The results (19) and (20) establish that, at high frequencies, the damping and the added mass tend to their asymptotic values at a rate of ω^{-2} provided that the integral of the damping with respect to the frequency is different from zero (which gives the relative degree 1); and this applies to the diagonal as well as to the off-diagonal terms.

These results are in agreement with those previously published in the literature of hydrodynamics–for example Greenhow (1986) has shown via series expansions that

$$B_{mn}(\omega) \quad \rightarrow \frac{\beta'_{mn}}{\omega^2} + \frac{\beta''_{mn}}{\omega^4} \quad as \quad \omega \rightarrow \infty\,,$$

$$[A_{mn}(\omega) - A_{mn}(\infty)] \rightarrow \frac{\alpha'_{mn}}{\omega^2} + \frac{\alpha''_{mn}}{\omega^4} \quad as \quad \omega \rightarrow \infty\,.$$

But these are dominated by the terms proportional to ω^{-2} as the frequency increases–and therefore revert to the results of (19) and (20).

It is also interesting to note from (20), that the sign of $A_{mn}(\omega) - A_{mn}(\infty)$ is opposite that that of

$$b_p = \frac{2}{\pi}\int_0^\infty B_{mn}(\omega)\, dt\,.$$

Therefore, for example, for the diagonal terms which are known to be positive ($B_{nn}(\omega) > 0$), $A_{nn}(\omega) \rightarrow A_{nn}(\infty)$ always from below as $\omega \rightarrow \infty$.

The derivation of the above results is simple and elegant. It is based solely on the interpretation of the properties of the convolution terms discussed in the APPENDIX. In particular the property of relative degree being equal to 1 plays a key role—which holds whenever the integral of the damping with respect to the frequency is different from zero. Furthermore, the derivation makes no use of series expansions as the results previously presented in the literature—see Greenhow (1986) and references therein.

ILLUSTRATION EXAMPLE: MODERN CONTAINERSHIP

As an example of application, we can consider the vertical plane motion of a modern container ship. The hydrodynamic coefficients were computed with WAMIT, and the panel sized were dimensioned so as to compute up to a frequency of 2.5 rad/s—at which there is no appreciable response.

Figure 1 shows the potential damping computed by the hydrodynamic code (in solid line), and the extrapolation at a rate of ω^{-2} (dashed line). As we can appreciate, for couplings 33 and 55 the ω^{-2} tail is a very good approximation. However, for the couplings 35 and 53 the approximation is not as good. For these couplings, it would be necessary to increase the frequency of the computations more, or, alternatively, to use the higher order expansion by Greenhow (1986) ($\beta' \omega^{-2} + \beta'' \omega^{-4}$), which is also shown in Figure 1. Figure 2 show the plots in logarithmic scale (both axis) so as better appreciate the trends.

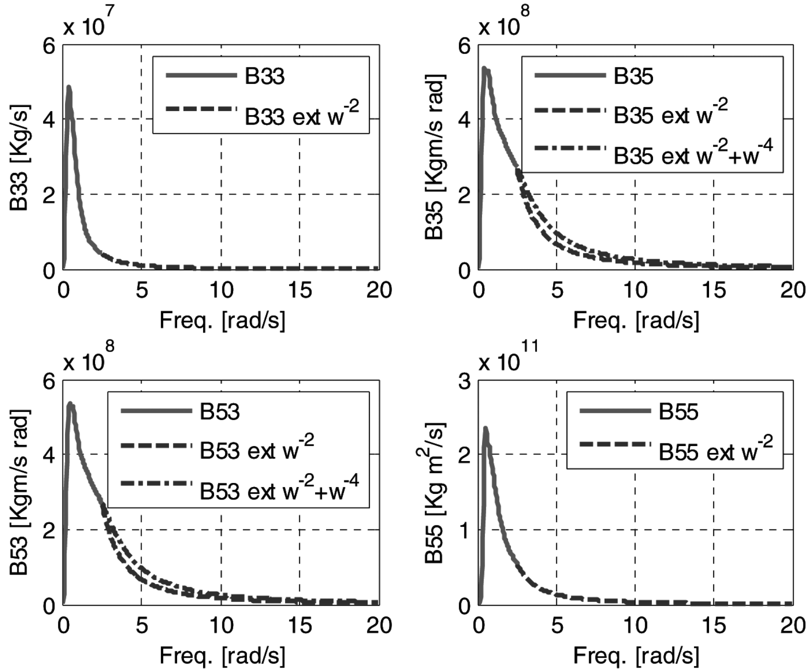

Figure 1 Potential damping linear scale. Solid–computed with a hydrodynamic code. Dashed–extrapolation at a rate of ω^{-2}.

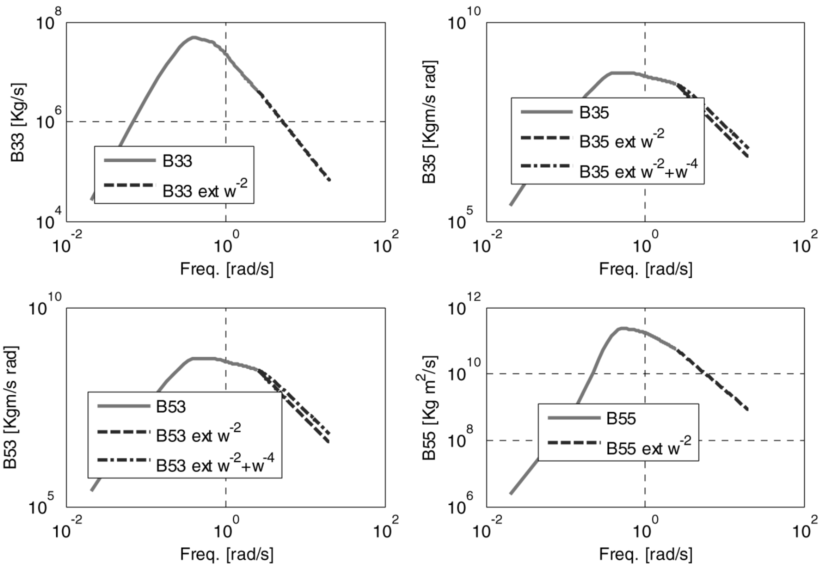

Figure 2 Potential damping log-log scale. Solid–computed with a hydrodynamic code. Dashed–extrapolation at a rate of ω^{-2}.

Figures Figure 3, Figure 4, and Figure 5 show the retardation functions computed using the damping with the extrapolations in (4). The solid lines correspond to the damping computed up to 2.5rad/s only; whereas the dashed lines correspond to the damping extrapolated with a tail proportional to ω^{-2} and for the 35 and 53 with $\beta' \omega^{-2} + \beta'' \omega^{-4}$.

As we can see, in all cases there is a smearing of the error when considering the damping over the higher frequencies. Note also, the error in the initial value of the retardation; which comes from neglecting the area under the tails.

The use of the retardation functions computed from a damping up to a low frequency can give rise to unnecessary dynamics if the retardation functions in used for time-domain identification for convolution replacement–for example, using the method recently proposed by Kristiansen, et al. (2005). Therefore, the results discussed here have application to this field.

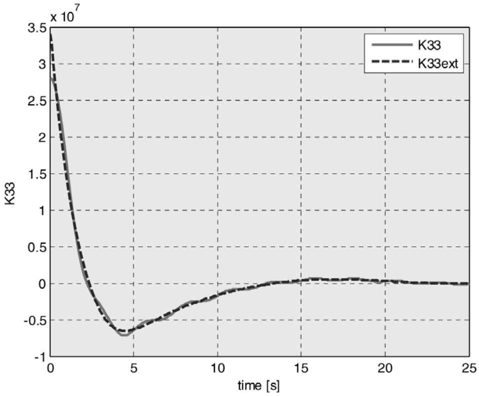

Figure 3. Retardation function 33. Solid–omputed from damping without extrapolation. Dashed–computed from damping with extrapolation ω^{-2}.

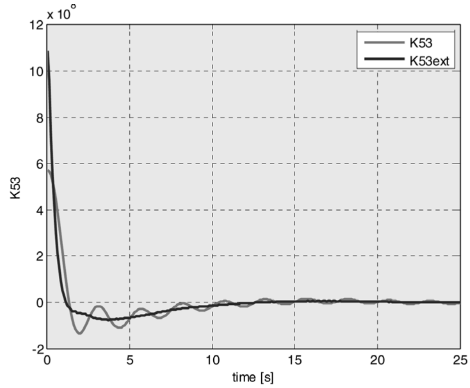

Figure 4. Retardation function 35. Solid–computed from damping without extrapolation. Dashed–computed from damping with extrapolaton $\beta'\omega^{-2}+\beta''\omega^{-4}$).

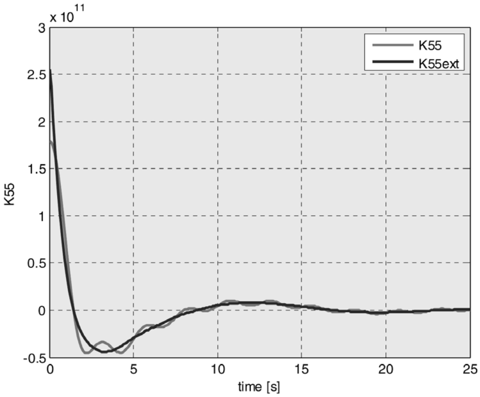

Figure 5. Retardation function 55. Solid–computed from damping without extrapolation. Dashed–computed from damping with extrapolatio ω^{-2}.

FORWARD SPEED CASE

For the case of a constant forward speed case, the Cummins equation (1) becomes

$$[\mathbf{M}+\mathbf{A}(\infty)]\,\ddot{\boldsymbol{\xi}}+\mathbf{B}(\infty)\,\dot{\boldsymbol{\xi}}+\int_0^t \mathbf{K}(t-\tau')\,\dot{\boldsymbol{\xi}}(\tau')\,d\tau'+\mathbf{C}\,\boldsymbol{\xi}=\boldsymbol{\tau} \tag{21}$$

where the constant damping and the retardation function depend on the forward speed U.

The time- and frequency-domain relation revert to

$$\mathbf{A}(\omega)=\mathbf{A}(\infty)-\frac{1}{\omega}\int_0^\infty \mathbf{K}(t)\sin(\omega t)\,dt \tag{22}$$

$$\mathbf{B}(\omega)=\mathbf{B}(\infty)+\int_0^\infty \mathbf{K}(t)\cos(\omega t)\,dt \tag{23}$$

$$\mathbf{K}(t)=\int_0^\infty \big[\mathbf{B}(\omega)-\mathbf{B}(\infty)\big]\cos(\omega t)\,dt \tag{24}$$

$$\mathbf{K}(j\omega)=\mathbf{B}(\omega)-\mathbf{B}(\infty)+j\omega\,[\mathbf{A}(\omega)-\mathbf{A}(\infty)] \tag{25}$$

Therefore, it follows that the results derived in Section 6—as well as the results in the APPENDIX—are valid for the case of constant forward speed modulo substitution $\mathbf{B}(\omega)$ by $\mathbf{B}(\omega)-\mathbf{B}(\infty)$.

CONCLUSION

In this paper, we have considered the consequences the properties of convolution terms in the Cummins' Equation on the high frequency asymptotic values of the added mass and potential damping. We have presented a new derivation of known asymptotic results without using expansions. We have shown that whenever the integral of the potential damping over the frequencies is not zero–as it happens in most practical cases—the hydrodynamic damping and added mass tend to their asymptotic values at a rate of ω^{-2} in the limit as the frequency is very high. The results agree with those previously presented in the literature, but the derivation, however, is believed to be novel, simpler and more elegant. We have also discussed the application to the computation of retardation functions, which can then be used for time-domain system identification of a convolution replacement for simulation of marine structures in waves.

ACKNOWLEDGEMENTS

The authors would like to thank Mr Reza Taghipour, from the Centre for Ship and Ocean Structures (CeSOS) at NTNU–Norway for computing the hydrodynamic data used in the application example with WAMIT.

REFERENCES

Cummins, W., (1962). The impulse response function and ship motions. Schiffstechnik 9 (1661), 101–109.

Damaren, C.J. (2000). Time-domain floating body dynamics by rational approximations of the radiation impedance and diffraction mapping. Ocean Engineering 27, 687–705.

Faltinsen, O.M. (1990) Sea Loads on Ships and Ocean Structures. Cambridge University Press.

Greenhow, M., 1986, High- and low-frequency asymptotic consequences of the Kramers-Kronig relations, J. Eng. Math., 20, 293-306.

Ogilvie, T., 1964. Recent progress towards the understanding and prediction of ship motions. In: 6th Symposium on Naval Hydrodynamics.

Kristiansen, E., A. Hjuslstad and O. Egeland (2005). State-space representation of radiation forces in time-domain vessel models. Ocean Engineering 32, 2195–2216.

Parzen, E. (1954) "Some Conditions for Uniform Convergence of Integrals" Proceedings of the American Mathematical Society, Vol. 5, No. 1. (Feb., 1954), pp. 55-58.

Perez, T. and T. I. Fossen (2006) "Time-domain Models of Marine Surface Vessels Based on Seakeeping Computations." 7th IFAC Conference on Manoeuvring and Control of Marine Vessels MCMC, Portugal, Sept.

Unneland, K., and T. Perez (2007) "MIMO and SISO Identification of Radiation Force Terms for Models of Marine Structures in Waves." IFAC Conference on Control Applications in Marine Systems (CAMS). Bol, Croatia, Sept.

APPENDIX:

Properties of the Convolution Terms in the Cummins Equation.

Low-frequency asymptotic value: $\lim_{\omega \to 0} \mathbf{K}(j\omega) = \mathbf{0}$

The proof of this statement follows from (6):

$$\mathbf{K}(j\omega) = \mathbf{B}(\omega) + j\omega\,[\mathbf{A}(\omega) - \mathbf{A}(\infty)] \ .$$

In the limit as $\omega \to 0$, the potential damping B(ω) $\to 0$ since there cannot be generated waves (Faltinsen, 1990). Therefore, the real part of K($j\omega$) tends to zero. The imaginary part also tends to zero since the difference A(0) − A(∞) is finite, which follows from (2):

$$\mathbf{A}(0) - \mathbf{A}(\infty) = \lim_{\omega \to 0} \frac{-1}{\omega} \int_0^\infty \mathbf{K}(t) \sin(\omega\, t)\; dt = -\int_0^\infty \mathbf{K}(t) \lim_{\omega \to 0} \frac{\sin(\omega\, t)}{\omega}\; dt = -\int_0^\infty \mathbf{K}(t)\; dt \ ,$$

from which the result follows. Note that regularity conditions are satisfied for the exchange of limit and integration (Parzen, 1954): $f_n(t) = K(t) \sin(2\pi t/n) / (2\pi/n)$ converges uniformly to $f(t) = K(t)$ as $n \to \infty$.

High-frequency asymptotic value: $\lim_{\omega \to \infty} \mathbf{K}(j\omega) = \mathbf{0}$

The proof of this statement also follows from (6):

$$\mathbf{K}(j\omega) = \mathbf{B}(\omega) + j\omega\,[\mathbf{A}(\omega) - \mathbf{A}(\infty)] \ .$$

In the limit as $\omega \to \infty$, the potential damping B (ω) $\to 0$ since there cannot be generated waves (Faltinsen, 1990). The imaginary part also tends to zero and this follows from (2) and the application of the Riemann-Lebesgue lemma (Ogilvie, 1964):

$$\lim_{\omega \to \infty} \omega[\mathbf{A}(0) - \mathbf{A}(\infty)] = \lim_{\omega \to \infty} \int_0^\infty -\mathbf{K}(t) \sin(\omega\, t)\; dt = \mathbf{0} \ \cdot$$

Initial time value of the retardation function: $\mathbf{K}(t = 0^+) \neq \mathbf{0}$

This follows from (4),

$$\lim_{t \to 0^+} \mathbf{K}(t) = \lim_{t \to 0^+} \frac{2}{\pi} \int_0^\infty \mathbf{B}(\omega) \cos(\omega\, t)\; d\omega = \frac{2}{\pi} \int_0^\infty \mathbf{B}(\omega)\; d\omega \neq \mathbf{0} \, .$$

Note that regularity conditions are satisfied for the exchange of limit and integration (Parzen, 1954); and the last equality follows from energy considerations, which establish that the diagonal terms of B(ω); namely $B_{ii}(\omega) > 0$–see Faltinsen (1990).

Final time value of the retardation function: $\lim_{t\to\infty} \mathbf{K}(t) = \mathbf{0}$

This follows from (4), by application of the Riemann-Lebesgue Lemma:

$$\lim_{t\to\infty} \mathbf{K}(t) = \lim_{t\to\infty} \frac{2}{\pi} \int_0^\infty \mathbf{B}(\omega) \cos(\omega\, t)\, d\omega = \mathbf{0}\,.$$

This property establishes the necessary and sufficient conditions for bounded-input-bounded-output stability of the convolution terms.

Passivity of K($j\omega$): The damping matrix for a structure with zero forward speed and without current is symmetric and positive-semi definite, $B(\omega) = B^T(\omega) \geq 0$ (Faltinsen, 1990), from which it follows the positive realness of K($j\omega$); and thus, the fact that K($j\omega$) is passive. A derivation of passivity in terms of energy functions can be found in Damaren, (2000) and Kristiansen et al (2005). From this it follows that the diagonal elements ($B_{mn}(\omega)$ for terms $m=n$) are positive-semi definite and their integral with respect of ω is different from zero; the integral of the off-diagonal terms (those which are not uniformly zero due to symmetry of the structure), however, could be zero. As commented by Unneland and Perez (2007), the diagonal terms $K_{ii}(j\omega)$are passive, but the off-diagonal terms need only be stable–see previous property.

Journal of Maritime Research, Vol. V. No. 1, pp. 79-96, 2008

ISSN: 1697-4840

INTRODUCTION TO SHIP DYNAMIC POSITIONING SYSTEMS

C. S. Chas[1] and R. Ferreiro[2]

Received 24 September 2007; received in revised form 19 October 2007; accepted 30 March 2008

ABSTRACT

The aim of this work deals with the general review of conventional dynamic positioning systems. A summary of the main elements of a DP vessel and a brief explanation of the most relevant tasks is presented, followed by an IMO and NMD classification of DP vessels, depending on the redundancy of the ship equipment.

Key Words: Dynamic positioning, position reference systems, IMA classification, NMD classification

INTRODUCTION

Ship's Dynamic Positioning (DP) is a procedure that automatically maintains a vessel's position and heading (station keeping) by using her own propellers and thrusters (Balchen(a) et all., 1976), (Balchen(b) et all., 1980), Balchen(c) et all., 1980) (Grimble(a) et all., 1980), (Sorensen et all., 1996). This allows operations at sea where mooring or anchoring is not feasible due to deep water, congestion on the sea bottom (pipelines, templates) or simply the place where DP operations are needed.

DP is defined by the (IMCA, 2003), (International Marine Contractors Association) as: *"A system which automatically controls a vessel's position and heading exclusively by means of active thrust"*.

This definition includes remaining at a fixed location, precision maneuvering,

[1] Professor ETS. Náutica y Maquinas, Dept. Ingeniería Industrial (ferreiro@udc.es), A Coruña, Spain.
[2] Doctoral Student, ETS Náutica y maquinas, Dept. Ingeniería Industrial, (crisolchas@hotmail.com), A Coruña, Spain.

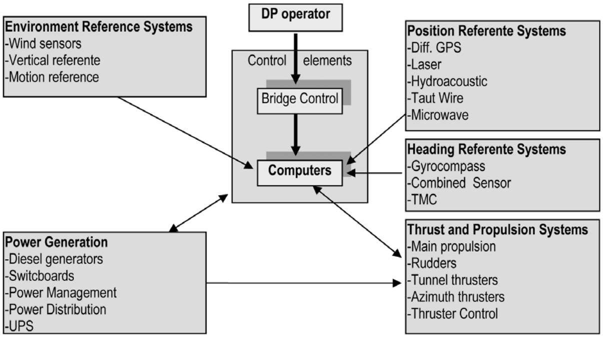

Fig. 1. Schematic diagram of a general DP system, showing the tasks and resources.

tracking and other specialist positioning abilities. To infer the mentioned positioning capabilities, a set of well defined tasks must be carried out. Furthermore, to carry out such tasks a set of resources or DP equipment is essential. Such tasks and resources are depicted in figure 1. In figure 2, the general arrangement of a DP system is shown. The necessary equipment to implement a DP system is also shown. With regard to this figure, a set of sensors acquires the necessary information, which is processed before entering the control algorithm (Grimble(b) et all.,1980), (Fossen et all., 1994), (Fossen et all., 1999), Strand et all., 1995), (Strand et all., 1999). The control algorithm computes the thruster setpoints as function of position, speed and environment conditions to operate the thrusters according power demand.

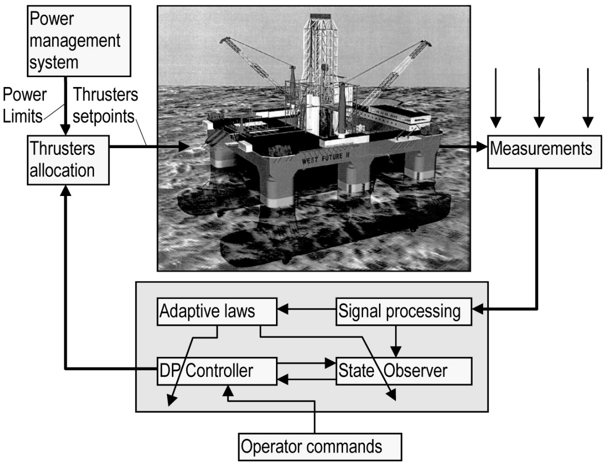

Fig. 2. The general arrangement of a DP system.

Any vessel can move in six degrees of freedom, three rotations (yaw, pitch and roll) and three translations (surge, sway and heave). Dynamic positioning conventionally is concerned with the automatic control of surge, sway and yaw. Surge and sway are related to the position of the vessel, while yaw is defined by the vessel heading.

Every vessel is subjected to forces from wind, waves and tidal movements (currents) as well as forces generated from the propulsion system and other external elements as fire monitors. The movement of the vessel with changes of position and heading is the result to these forces. Position is measured by position reference systems, while heading information is provided from gyrocompasses.

The vessel must be able to control position and heading within acceptable limits facing the external forces. If these forces are measured directly, the control computers can apply immediate compensation.

The DP control system calculates the offsets between the measured values of position and heading and the required values, and after that it calculates the forces that the thrusters must generate in order to reduce the errors to zero.

DP vessels duties would be:

Coring, exploration drilling (core sampling), production drilling, diver support, pipelay (rigid and flexible pipe), cable lay and repair, multi-role, accommodation or "flotel" services, hydrographic survey, pre- or post-operational survey, wreck survey, salvage and removal, dredging, rockdumping (pipeline protection), subsea installation, lifting (topsides and subsea), well stimulation and workover, platform supply, shuttle tanker offtake, floating production (with or without storage), heavy lift cargo transport, passenger cruises, mine countermeasures, oceanographical research, seabed mining, rocket launch platform positioning, repair/maintenance support to military vessels, ship-to-ship transfer and maneuvering conventional vessels.

Some advantages of dynamic positioning system are:

Vessel is fully self-propelled, setting-up on location is quick and easy, vessel is very maneuverable, rapid response to weather changes is possible, rapid response to changes in the requirements of the operation, versatility within system, ability to work in any water depth, can complete short tasks more quickly, thus more economically, avoidance of risk of damaging seabed hardware from mooring lines and anchors, avoidance of cross-mooring with other vessels or fixed platforms and can move to new location rapidly.

Te main disadvantages are:

High capex and opex, can fail to keep position due to equipment failure, higher day rates than comparable moored systems, higher fuel consumption, thrusters are hazards for divers, can lose position in extreme weather or in shallow waters and strong tides, position control is active and relies on human operator (as well as equipment) and requires more personnel to operate and maintain equipment.

The first DP control systems did not adapt to the actual sea conditions and vessel and thruster errors, that's the reason why it was necessary control improvements in station keeping accuracy (Loria et all., 2000) or optimal filtering based on the Kalman filter (Grimbre(b) et all., 1980), which provides an algorithmic procedure of recent implementation to improve maneuvering performance, and fast digital data transmission.

The elements of a DP system are computers, the control console, position reference systems, the heading reference, the environment reference, power systems and propulsion systems as shown in figures 1 and 2. (Bray, David 1997), (Kongsberg Maritime), (UKOOA).

Depending on the level of redundancy, DP computers and/or sensors and actuators are installed in single, dual or triple configurations.

The bridge console is the facility for the operator (DPO) to send and receive data located close to position reference system control panels, thruster panels and communications. Since some DP operations require better than 3m relative accuracy, a DP control system requires data at a rate of once per second to achieve high accuracy.

In next section a brief description of the position reference system is described. In section 3,4,5,6 and 7, coordinate system, environment disturbances, power systems, propulsion system and class requirements are respectively briefly described.

POSITION REFERENCE SYSTEMS

All DP vessels have position reference systems (PRS), independent of the vessel's normal navigation suite. There are five types of PRS in common use in DP vessels:

– Hydroacoustic Position Reference (HPR).
– Taut Wire.
– Differential Global Positioning (DGPS).
– Laser-based systems (Fanbeam and CyScan).
– Artemis.

For any operations requiring DP redundancy it is necessary to utilize three position references. Two PRSs are not adequate, because if one has failed it would have contradictory reference data, whereas three systems provide two-out-of-three data.

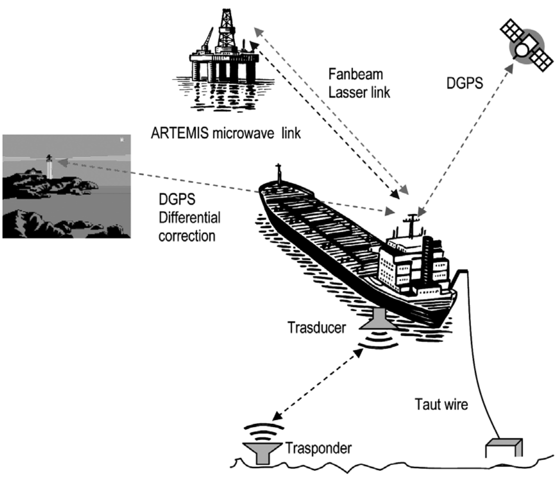

Fig. 3 Position reference systems.

If three PRSs are required, the DPO should choose systems that are different; because this reduces the probability of common-mode failure.

Hydroacoustic Position Reference (HPR).

This system consists of one or more transponders placed on the seabed and a transducer placed in the ship's hull. The transducer sends an acoustic signal (by means of piezoelectric elements) to the transponder, which is triggered to reply. As the velocity of sound through water is known, the distance is known. Because there are many elements on the transducer, the direction of the signal from the transponder can be determined. Now the position of the ship relative to the transponder can be calculated. Disadvantages are the vulnerability to noise by thrusters or other acoustic systems. Furthermore, the use is limited in shallow waters because of ray bending that occurs when sound travels through water horizontally.

There are three types of acoustic position reference systems in common use: ultra or super-short baseline systems (USBL or SSBL), short baseline systems (SBL) and long baseline systems (LBL).

Ultra- or Super-Short Baseline Acoustic System

The principle of position measurement involves communication at hydroacoustic frequencies between a hull-mounted transducer and one or more seabed-located transponders. The ultra or super-short baseline (SSBL) principle means that the measurement of the solid angle at the transducer is over a very short baseline (the transducer head).

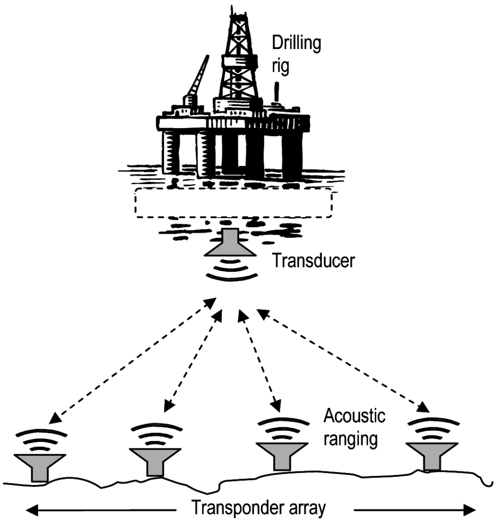

Fig. 4. LongBaseLine acoustic system.

An interrogating pulse is transmitted from the transducer, which is received by the transponder on the seabed. The transponder replies and the transmitted reply is received at the transducer. The transmit/receive time delay is proportional to the slant and range. The angles and range define the position of the ship relative to that of the transponder. But the measured angles must be compensated for values of roll and pitch.

The problem of this system is that the performance of an acoustic system is often limited by acoustic conditions in the water. And because of the nature of angle measurement, the accuracy deteriorates with increasing water depth.

Long Baseline System

It is the typical one in case of deepwater locations. It uses an array of three or more transponders laid on the seabed in the vicinity of the worksite. Usually the array forms a pentagon (5 transponders) on the seabed, with the drill ship at the centre above.

Once calibrated for position, individual interrogation of three or more of this array from a vessel's transducer will giver a series of ranges to the transponders, hence vessel position. The position should theoretically be located at the intersection of imaginary spheres, one around each transponder, with a radius equal to the time between transmission and reception multiplied by the speed of sound through water.

The angle measurements are not required at the transducer, thus a major source of error is eliminated and errors in range measurements caused by ray bending are less significant.

Short Baseline System.

It is like a long baseline system with an exception, there is an array of transducers (hydrophones), spread along the underside of the DP vessel and the baseline(s) are the distances between them. As the array is located on the ship, it needs to be corrected for roll and pitch.

They are typically installed in drilling rigs and semisubmersible barges.

Taut Wire Position Reference.

Taut wire is a useful short range position reference system, particularly where the vessel may spend long periods in a static location and where the water depth is limited. Also may be used on mobile equipment, where the vessel needs to maintain a location relative to a moving vehicle.

A normal configuration of a taut wire consists of a crane assembly on deck (usually at the side) and a depressor weight on a wire rope is handled by a constant tension winch. At the end of the boom of the crane are located angle sensors, which detect the angle of the wire. When the depressor weight is lowered to the seabed, the system switch to mooring mode (constant tension). The winch operates to maintain a constant tension on the wire and hence to detect the movements of the vessel.

The length of the wire deployed and the angle of the wire defines the position of the sensor head. This information is displayed on the DP system. Angles data are corrected from roll and pitch movements, and also the offset between the location of the sensor head and that the vessels centre of rotation.

The accuracy of the system will depend upon the depth of the water, the mooring tension, the wire angle to the vertical and the strength of tide.

Taut wire systems have limitations on wire angle because of the increasing risk of dragging the weight as angles increase. A typical maximum angle is 30 degrees (DP system will initiate a warning). At 35 degrees an alarm will occur and the DP system will reject this position reference system.

Advantages:

— High accuracy, especially in moderate depths of water.

— Good reliability (regular maintenance carried out).

— Quick and easy to set up.
— No need for assistance from external sources to set up or operate.
— System is mechanical, so on-board repair is possible.

Disadvantages:
— The system is of a short range only, especially in shallow water.
— Susceptible to strong tides (resulting in inaccuracies).
— Accuracy deteriorates in deep water.
— Maximum depth limitation.
— Adversely affected by debris or ice conditions.
— Possibility of weight dragging (positional errors).
— Wire may provide obstruction to underwater operations.
— System is not geographically referenced (relative positioning only).
— Wire may provide obstruction to underwater operations.
— Depressor weight may cause damage to seabed hardware when landing.
— If the vessel needs to move, taut wire must be continuously re-plumbed.
— Susceptible to mechanical damage.
— Limitations in range due bilge keel or other vessel structures.

The DGPS Position Reference System

The Global Positioning System (GPS) has typical accuracies available from the GPS Standard Positioning Service of 20m, but GPS accuracy is not adequate for DP purposes.

In order to improve GPS accuracy, differential corrections are applied to GPS data. This is done by establishing reference stations at known points on the WGS 84 spheroid. The pseudo ranges derived by the receiver are compared with those computed from the known locations of the satellites and reference stations, obtaining a correction.

Most DGPS services accept multiple differential inputs obtained from an array of reference stations widely separated. Network DGPS systems provide greater stability and accuracy, and remove more of the ionospheric error than obtainable from a single reference station.

The accuracy obtainable from DGPS systems is in the area of 1-3m dependent upon the distances to the reference stations, ionospheric conditions, and the constellation of satellites available.

Some DP operations require the positioning of a vessel relative to a moving structure. For example, a shuttle tanker loading via a bow loading hose from the stern of an FPSO (Floating Production Storage and Offloading). Since the FPSO can weathervane, the stern of the FPSO describes the arc of a circle, as well as it appears surge, sway and yaw motions, providing a complex positioning problem for the shuttle tanker.

An Artemis and a DARPS system (Differential, Absolute and Relative Positioning System) are configured to handle this problem. For the measurement of relative position by GPS, differential corrections are not needed, as the errors induced are the same for the shuttle tanker as they are for the FPSO.

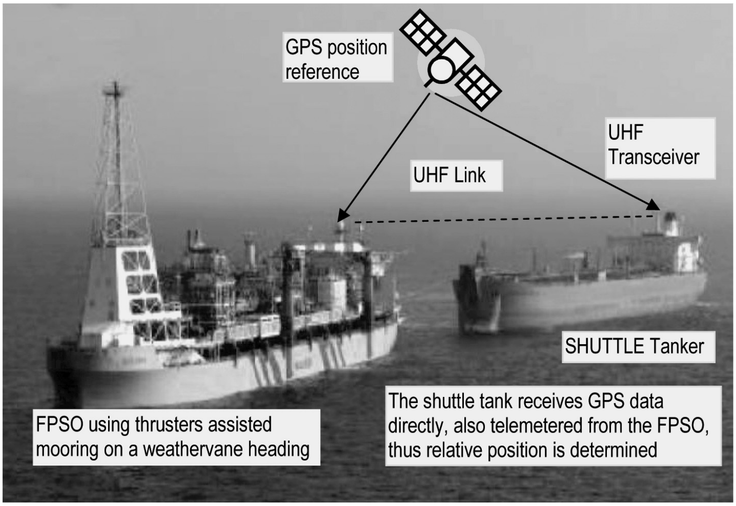

Fig. 5. Relative GPS.

There is another satellite system; the GLONASS (the Global Navigation Satellite System). It is similar in design and operation to the American GPS, but this is from Russia. The problem is that nowadays the number of available satellites is not good for positioning. So it is used as a position reference for DP and there are combined GPS/GLONASS receivers.

But the higher orbital inclination of GLONASS satellites (65°), compared to the GPS constellation (55°), results in better satellite availability in higher latitudes.

Laser-Based Position Reference

There are two laser DP position references in use: Fanbeam and CyScan.

Both systems lock onto a single target and/or a number of targets on the structure, from which position must be maintained. Light pulses are sent and received so that range and bearing can be measured.

The Fanbeam system is manufactured by MDL of Aberdeen and it consists of two units, the laser scanner and the Universal Display Unit (UDU). The laser consists of an array of gallium arsenide semiconductor laser diodes. These produce a laser with a 20° vertical fan with a horizontal divergence of less than 4 miliradians, pulsed at 5.000 Hz. The pulses are emitted through a transmitter lens. Reflected received light is directed onto an array of photo sensitive diodes to produce an electrical signal. The range is determined from echo ranging; the accuracy is improved by averaging a number of returns from a target.

It is a very straightforward system, as only a small prism needs to be installed on a nearby structure. Some risks are that the Fanbeam could be locked on other reflecting objects and block the signal.

The scanner unit in a DP vessel should be placed in a location affording an unobstructed view of the horizon.

Ranges vary according to weather conditions, when the systems will be affected by reduced optical visibility. It is typically more than 500 meters.

The advantages of the Fanbeam systems are:
— Low cost in comparison with the other reference systems.
— Ease of installation.
— Passive target, no power supply required.
— Target does not require any support services.
— High accuracy.

But it has some disadvantages as:
— It will not operate with the sun shining directly into the lenses.
— The lenses can be affected by condensation, rain and salt spray.
— The system operation is impaired by fog, snow and heavy rain.
— The system may be confused by bright lights close to the target at night.
— The system may suffer interference from reflective items.

Artemis

Artemis is a trade name for a system produced by the Christian Huygenslaboritorium BV in Netherlands. The basis of position by Artemis is that of obtaining a range and bearing of a mobile station from some known fixed location. A low power microwave link is established between the two stations, using directional tracking antennae.

Specifications of the system:

Frequency:	9.2 – 9.3 GHz.
Range:	10m – 30,000m
Range accuracy:	0.1 – 1.0m
Bearing accuracy:	0.02 degrees.
Horizontal beamwidth:	2 degrees to the “half power “ points
Vertical beamwith:	22 degrees to the “half power” points
Antenna tracking:	maximum 15 degrees / second
Power supply:	24V DC, consumption approximately 40w

The signal delays are directly proportional to the distance between the fixed and the mobile stations. Normally, the fixed station is located on board a platform or other fixed location, while the mobile is located on a vessel. The mobile station acts as the Master; control and operator input is from the mobile. The mobile unit is interfaced to the DP system.

Corrections must be applied within DP algorithms to allow for the differences in location between the antenna and the vessel centre of rotation.

The system consists in two identical Artemis Basic Units, which are configured as a fixed and as a mobile unit. The operator interface is provided by means of an Extended Operator Panel (EOP) at the mobile station, and by a Basic Operator Panel (BOP) at the fixed Station. The EOP is the user interface aboard the vessel and is conveniently mounted adjacent the DP operator console, or integrated within it.

The antenna consists of a tracking slotted waveguide aerial, fitted in two halves. The tracking antenna has a horizontal beam width of 2º and a vertical beam width of about 22º. The radiated energy is about 100 mW, vertically polarized. It is arranged that the antennae at the fixed and mobile stations maintain a radio link by tracking the antennae in azimuth. Without considering the vessel movement, the two antennae track so as to face each other, with the antennae normal to the direction of the signal. Left and right halves of the antenna applies signals separately to four ports. The signal are applied equally but opposite in phase. This means that the output will be zero only if the antenna is perpendicular to the incoming signal direction. If this is not the case, drive motor will act to rotate the antenna to reduce the perpendicular error to zero. By this means the two antennas continually tracks each other.

To operate the system it is necessary to establish a microwave link between the mobile and fixed stations. The platform is contacted and a request is made to use the Artemis system. The platform personnel will switch the system on and connect their BOP. After selecting the required frequency channel and running a number of checks, they will aim the fixed antenna at the vessel. The vessel will do the same for the mobile. Then both units automatically "lock" and antennae will track. Range and bearing are displayed at the mobile end.

It is necessary to calibrate the fixed station in order to be properly referenced for azimuth. This is done by fitting a small telescope on the top of the antenna, to the same direction that the antenna is addressed. The objective is to determine the true grid bearing of a visual reference object (another platform for example) before use the system.

Advantages of the Artemis system:

— Long range.
— High accuracy.
— Possible to geographically reference the position data.
— Very convenient when inside the 500m zone.

Disadvantages:
— Requires a fixed station established on a nearby installation.
— Fixed units need to be correctly calibrated and configured.
— Specially designed units needed for hazardous areas.
— May require assistance from platform personnel to set up.
— May suffer interference from platform personnel.
— May suffer interference from heat or precipitation.
— May suffer line-of-sight interruption.
— Vulnerable to power supply problem at fixed end.
— Interference from 3cm radar.

CO-ORDINATE SYSTEMS

Position information from position-reference systems may be received by the DP system in several formats. Typically for DP applications the co-ordinate system used may be Cartesian (relative) or geodetic.

For the DP system to handle earth-referenced type of data it is necessary to configure the DP system to accept geodetic data, or global references, such as GPS.

A DGPS system provides co-ordinates in terms of latitude and longitude referenced to the WGS84 datum. Most offshore operations are conducted using UTM (Universal Transverse Mercator). This is a flat-surface, square-grid projection defined by a UTM zone number, and a Northing and Easting distance from the zero point of the zone.

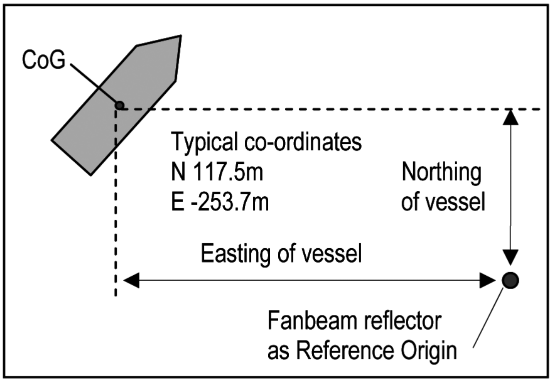

Fig. 6. Local or Cartesian reference co-ordinates.

UTM is a cylindrical projection with the axis of the cylinder coincident with the plane of the Equator; thus the line of contact between the cylinder and the sphere is a meridian. It cannot be used to chart the whole terrestrial surface. The useful scope of the projection consists of a zone 6°of longitude in width, with the centre upon the contact or “Central” meridian. Within this zone distortions are minimal.

ENVIRONMENT REFERENCE

There are three main environmental forces which cause the vessel to move away from her setpoint position and/or heading. They are the forces created by wind, waves and current.

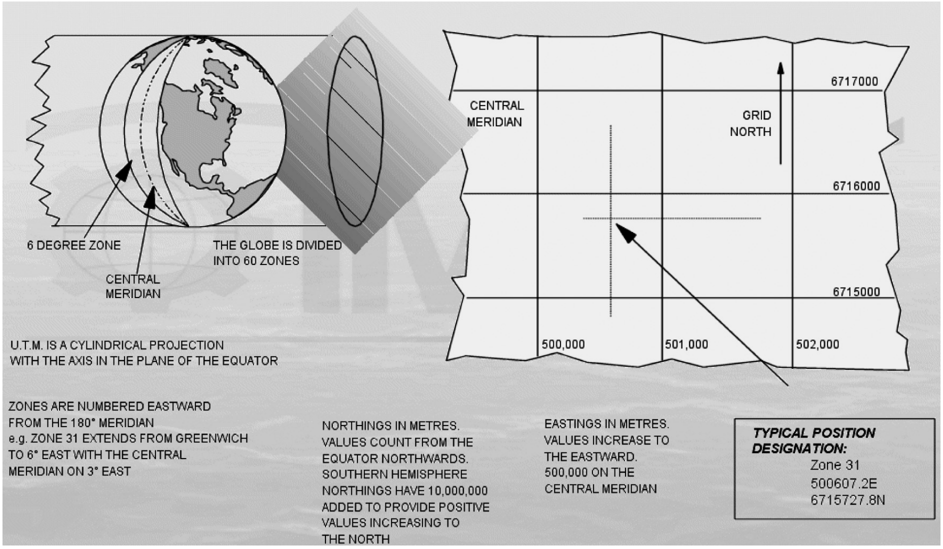

Fig. 7. Universal Transverse Mercator Co-Ordinate System.

In the case of current meters, they are not used in general, because they are expensive and generally the current forces change slowly.

For waves the DP control system does not provide direct compensation. In practice, the frequency of the waves is such that it is not feasible to provide compensation for individual waves. Wave drift forces build slowly and appear in the DP control system as current or sea force.

All DP systems have wind sensors (usually a rotating-cup anemometer). This data is used to calculate wind-induced forces acting upon the vessel's hull and structure, allowing these forces to be compensated before they cause a position or heading change.

The roll, pitch and heave motions of the vessel are not compensated for by the DP control system, but it is necessary for the DP control system to be provided with accurate values of roll and pitch. This is to allow compensation to be applied to all the various position reference sensor inputs for their offset from the centre of gravity of the vessel. Instrumentation to measure these values is provided in the form of a vertical reference sensor (VRS), vertical reference unit (VRU) or a motion reference unit (MRU).

A recent development is the provision of a system which utilizes two or more DGPS receivers with antennae mounted some distance apart. The GPS fixes and motion-sensors provide data on vessel position, heading, roll, pitch and heave values.

POWER SYSTEMS

Power needs to be supplied to the thrusters and all auxiliary systems, as well as to the DP control elements and reference systems.

The thrusters on a DP vessel are often the highest power consumers on board.

The power generation system must be flexible in order provide power rapidly on demand while avoiding unnecessary fuel consumption. Many DP vessels are fitted with a diesel-electric power plant with all thrusters and consumers electrically powered from diesel engines driving alternators.

In the event of an interruption to the ship's main AC supply, batteries will supply power to computers, control consoles, displays, alarms and reference systems for a minimum of 30 minutes.

PROPULSION SYSTEMS

The DP capability of the vessel is provided by her thrusters. In general, three main types of thruster are fitted in DP vessels; main propellers, tunnel thrusters and azimuth thrusters.

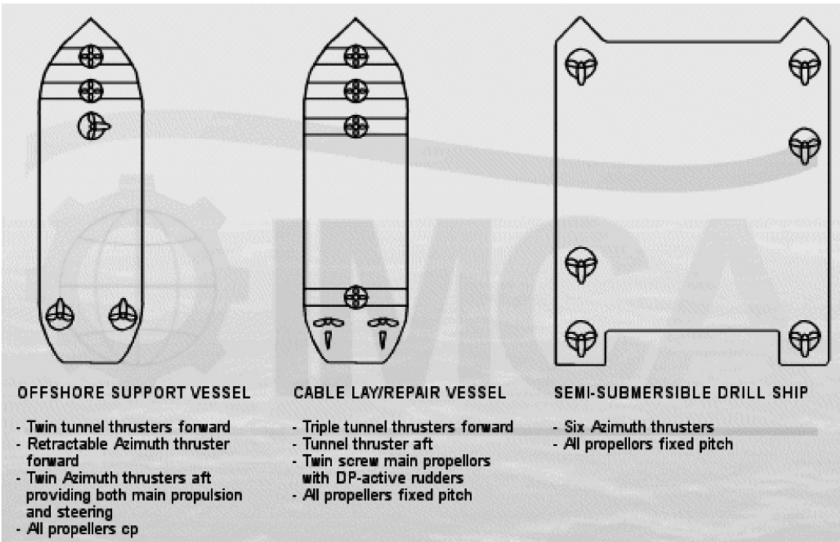

Fig. 8. Typical Propulsion System Layouts.

In DP vessels propellers may be controllable pitch (cp) running at constant rpm or variable speed.

DC motors or frequency-converter systems enable variable speed to be used with fixed-pitch propellers. Main propellers are usually accompanied by conventional rudders and steering gear.

In addition to main propellers, a DP must have well-positioned thrusters to control position. Typically, a conventional monohull-type DP vessel will have six thrusters; three at the bow and three aft. Forward thrusters tend to be tunnel thrusters, operating athwartships. Stern tunnel thrusters are common, operating together but controlled individually, as are azimuth or compass thrusters aft. Azimuth thrusters project beneath the bottom of the vessel and can be rotated to provide thrust in any direction.

CLASS REQUIREMENTS

Based on IMO (International Maritime Organization) publication 645 the Classification Societies have issued rules for Dynamic Positioned Ships described as Class 1, Class 2 and Class 3.

Equipment Class (EC) 1 has no redundancy. Loss of position may occur in the event of a single fault.

Equipment Class 2 has redundancy so that no single fault in an active system will cause the system to fail. Loss of position should not occur from a single fault of an active component or system such as generators, thruster, switchboards, remote controlled valves etc. But may occur after failure of a static component such as cables, pipes, manual valves etc.

Equipment Class 3, which also has to withstand fire or flood in any one compartment without the system failing. Loss of position should not occur from any single failure including a completely burnt fire sub division or flooded watertight compartment.

Classification Societies have their own Class notations:

Table 1. Class Classification

Description	IMO E.Class	LR E.Class	DnV E.Class	ABS E.Class
Manual position control and automatic heading control under specified maximum environmental conditions	-	DP(CM)	DNV-T	DPS-0
Automatic and manual position and heading control under specified maximum environmental conditions	Class 1	DP(AM)	DNV-AUT DNV-AUTS	DPS-1
Automatic and manual position and heading control under specified maximum environmental conditions, during and following any single fault excluding loss of a compartment. (Two independent computer systems).	Class 2	DP(AA)	DNV-AUTR	DPS-2
Automatic and manual position and heading control under specified maximum environmental conditions, during and following any single fault including loss of a compartment due to fire or flood. (At least two independent computer systems with a separate backup system separated by A60 class division).	Class 3	DP(AAA)	DNV-AUTRO	DPS-

Where IMO leaves the decision of which Class applies to what kind of operation to the operator of the DP ship and its client, the Norwegian Maritime Directorate (NMD) has specified what Class should be used in regard to the risk of an operation. In the NMD Guidelines and Notes No. 28, enclosure A four classes are defined:

— Class 0 Operations; where loss of position keeping capability is not considered endangering human lives, or causing damage.
— Class 1 Operations; where loss of position keeping capability may cause damage or pollution of small consequence.
— Class 2 Operations; where loss of position keeping capability may cause personnel injury, pollution, or damage with large economic consequences.
— Class 3 Operations; where loss of position keeping capability may cause fatal accidents, or severe pollution or damage with major economic consequences.

Based on this, the type of ship is specified for each operation according the following:

— Class 1: DP units with equipment class 1 should be used during operations where loss of position is not considered to endanger human lives, cause significant damage or cause more than minimal pollution.
— Class 2: DP units with equipment class 2 should be used during operations where loss of position could cause personnel injury, pollution or damage with great economic consequences.
— Class 3: DP units with equipment class 3 should be used during operations where loss of position could cause fatal accidents, severe pollution or damage with major economic consequences.
— Class 2 or 3 systems should include a "Consequence analysis" function that continuously verifies that the vessel will remain in position even if the worst single failure occurs.

The IMO Guidelines also specify the relationship between equipment class and type of operation. DP drilling operations and production of hydrocarbons, for instance, require equipment class 3, according to the IMO.

CONCLUSIONS

A general review of conventional dynamic positioning systems, including a summary of the main elements and tasks to be carried out on DP operations, has been presented. In the same way IMO and NMD classification of DP vessels, depending on the redundancy of the ship equipment has been defined. From this general review it is concluded the existence of a variety of DP missions which can be carried out under safety requirements that demand the application of the state of the art in marine engineering technology.

REFERENCES

Balchen(a), J. G., Jenssen, N. A., and Sælid, S., (1976) Dynamic positioning using Kalman filtering and optimal control theory, *IFAC/IFIP Symposium on Automation in Offshore Oil Field Operation*, Amsterdam, The Netherlands, 183–186.

Balchen(b), J. G., Jenssen, N. A., and Sælid, S., (1980) Dynamic positioning of floating vessels based on Kalman filtering and optimal control, *Proc. of the 19th IEEE Conf. on Decision and Control*, New York, NY, 852–864.

Balchen(c), J. G., Jenssen, N. A., Mathisen, E., and Sælid, S., (1980) Dynamic positioning system based on Kalman filtering and optimal control. *Modeling, Identification and Control MIC*-1(3): 135–163.

Bray, David, *Dynamic Positioning, Vol. 9,* England, Oilfield Publications Limited, (1997).

Fossen, T. I., (1994) *Guidance and Control of Ocean Vehicles,* John Wiley & Sons, Ltd., Chichester, England.

Fossen, T. I. and Strand, J. P., (1999) Passive nonlinear observer design for ships using Lyapunov methods: experimental results with a supply vessel, (Regular Paper) *Automatica* AUT-35(1): 3–16, January.

Grimble(a), M. J., Patton, R. J., and Wise, D. A., (1980) The design of dynamic positioning control systems using stochastic optimal control theory, *Optimal Control Applications and Methods* 1, 167–202.

Grimble(b), M. J., Patton, R. J., and Wise, D. A., (1980) Use of Kalman filtering techniques in dynamic ship positioning systems, *IEE Proceedings Vol. 127*, Pt. D, No. 3, 93–102.

IMCA (2.003): *Guidelines for the Design & Operation of Dynamically Positioned Vessels*, http://www.imca-int.com.

KONGSBERG MARITIME A S, P.O. Box 483, N-3601 Kongsberg, Norway, www.kongsberg.com

Loria, A., Fossen, T. I., and Panteley, E., (2000) A cascaded approach to a separation principle for dynamic ship positioning, *IEEE Transactions of Control Systems Technology*, to appear in 2000.

Strand, J. P. and Fossen, T. I., (1998, 89–95) Nonlinear output feedback and locally optimal control of dynamically positioned ships: experimental results, *Proc. of the IFAC Conference on Control Applications in Marine Systems (CAMS '98)*, Fukuoka, Japan, October 27–30.

Strand, J. P. and Fossen, T. I., (1999) Nonlinear Passive Observer for Ships with Adaptive Wave Filtering, in *New Directions in Nonlinear Observer Design*, H. Nijmeijer and T. I. Fossen, Eds., Springer-Verlag London Ltd., 113–134.

Sørensen, A. J., Sagatun, S. I., and Fossen, T. I., (1996) Design of a dynamic positioning system using model based control, *Journal of Control Engineering Practice* CEP-4, 3, 359–368.

UKOOA – OIL AND GAS FOR BRITAIN.FPSO Committee. Tandem Loading Guidelines, Volume 1. http://www.ukooa.co.uk/issues/fpso/docs/tandemvolume1.pdf

INSTRUCTIONS FOR AUTHORS

http://www.jmr.unican.es

Description

Journal of Maritime Research (JMR) publishes original research papers in English analysing the maritime sector international, national, regional or local. JMR is published quarterly and the issues—whether ordinary or monographic—include both theoretical and empirical approaches to topics of current interest in line with the editorial aims and scope of the journal.

The objective of JMR is to contribute to the progress, development and diffusion of research on the maritime sector. The aim is to provide a multidisciplinary forum for studies of the sector from the perspective of all four of the following broad areas of scientific knowledge: experimental sciences, nautical sciences, engineering and social sciences.

The manuscripts presented must be unpublished works which have not been approved for publication in other journals. They must also fulfil all the requirements indicated in the 'Technical specifications' section. The quality of the papers is guaranteed by means of a process of blind review by two scientific referees and one linguistic reviewer. Although the journal is published in print format, the processes of submission and reception of manuscripts, as well as revision and correction of proofs are all done electronically.

The publication process

Submission and acceptance of manuscripts

Papers should be sent via the journal's web site at http://www.jmr.unican.es, following the guidelines indicated for the submission of manuscripts by authors. Submitting the manuscript involves sending electronically via the web site three different types of information: general information about the article and the authors, an abstract and the complete paper.

The general information about the article and the authors, as well as the acknowledgements, is to be introduced by means of an electronic form, in which the following data are requested:

- ☐ Title of the paper. Subtitle.
- ☐ Field of the manuscript presented (Experimental Sciences, Engineering, Nautical Sciences or Social Sciences) and sub-field.
- ☐ E-mail of the author to whom correspondence should be addressed.
- ☐ Personal data of each author including name, professional category or position, institution or organisation, postal address and e-mail.
- ☐ Acknowledgements, if included, should be no longer than 150 words without spaces between paragraphs.

The manuscripts submitted which are in line with the aims and scope of JMR will be submitted to a blind review process. The objective is to assess the scientific quality (two referees) and ensure that the text is linguistically acceptable (one reviewer). Only those works which are considered adequate from a scientific point of view will be reviewed linguistically. Manuscripts which do not pass the linguistic filter will be admitted with the condition that the author make linguistic and stylistic modifications as indicated.

Authors will be informed of the acceptance, rejection and/or modifications required no later than four months after the receipt of the manuscript. Once the paper has been accepted, authors can check on the status of their article via the web site. Any doubts or questions should be addressed to the Editor at jmr@unican.es

Copyright

After manuscripts have been definitively accepted, the authors must complete the copyright agreement assigning the copyright to JMR so that it can be published.

Proofs

Once the copyright transfer agreement has been completed, the Editors will assign the paper to an upcoming issue of JMR and proofs will be sent as a PDF file to the author who has corresponded with JMR throughout the reviewing process. The corrected proofs must be returned by e-mail to the Editor of JMR within 96 hours of receipt.

Reprints

Once the paper has been published, the authors will be sent a total of 12 free reprints, which will be sent the postal address of the corresponding author. Authors may also request, via the JMR web site http://www.jmr.unican.es, a maximum of 10 complete journals at 50% of the cover price.

Style and technical specifications

Manuscripts should be sent to JMR's web site in a format that is recognisable to Microsoft Word (.doc) in any of its versions for Windows. The maximum length of manuscripts is 23 double-spaced pages (approximately 7000 words) including the abstract, references and appendices. Manuscripts will be submitted using a specific application of the electronic form used to send personal data. The page layout should follow these guidelines:

- ☐ Size: DIN A4 (29 cm by 21 cm)
- ☐ Margins, 3 cm: top, bottom, left, and right.
- ☐ Font: Times New Roman, normal style, 12-point type.
- ☐ Double spacing should be used for all the paper except for the references which are to be single-spaced.
- ☐ 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.

The abstract is to be presented on one page and should include the following information:

- ☐ Title and subtitle of the paper
- ☐ Field and sub-field of the work presented.
- ☐ Abstract, which is to be no longer than 200 words, and should have no spaces between paragraphs.