REDUCING FUEL USAGE AND CO_2 EMISSIONS FROM TUG BOAT FLEETS: SEA TRIALS AND THEORETICAL MODELLING

(DOI No: 10.3940/rina.ijme.2012.a1.220)

A J Murphy, **S J Weston**, Newcastle University, UK and **R J Young**, Svitzer, UK

SUMMARY

Sea trials on a harbour tug have been conducted and are explained. The experimental results for fuel consumption per unit transport effort, under free-running (transiting) conditions, are presented and engine speed-propulsor pitch combinations for improved fuel economy are identified. A simplified analytical approach to predict fuel consumption, including the coupled engine-propulsor-hull system, is described. This rationale is combined with experimental observations and, consequently, performance maps present the complete operating envelopes of the harbour tug under both free-running and towing conditions. This combined approach proved to be effective and can be applied to the study of other tug vessels. As a consequence of this research, the engine control system on the harbour tug was modified to permit it to operate fully within the region of best fuel economy during free-running. The results from the bollard-pull predictions provide insight for the design and operation of harbour tugs in the future.

NOMENCLATURE

A_e	Effective propulsor area (m^2)
f_L	Load factor (s^{-1})
k, k_1 k_2, k_3	Coefficients (Units vary as relevant in narrative)
n	Rotational speed ($rev.s^{-1}$)
P_i	Ideal power (W)
P_D	Delivered power (W)
P_O	Baseline power loss (W)
P_P	Pitch-related power loss (W)
T	Thrust (N)
V_a	Advance velocity (ms^{-1})
V_e	Effective velocity (ms^{-1})
V_s	Vessel velocity (ms^{-1})
θ_e	Effective hydrodynamic pitch ($m.rev^{-1}$)
θ_n	Nominal pitch setting
ρ	Density (kgm^{-3})
ΔV	Difference in velocity (ms^{-1})

1. INTRODUCTION

With the current imperative for shipping to reduce its environmental impact through all stages of the ship's life cycle, fuel quality, fuel usage and exhaust gas treatment, leading to the eventual constituency of gas emitted from ships is the focus of much attention. The principal emphasis, when using diesel and fuel oils, is on reducing the emission of oxides of nitrogen, sulphur and carbon (NO_X, SO_X & CO_X) together with reducing the release of particulate matter (PM).

The prevailing legislation is specified in the "Regulations for the Prevention of Air Pollution from Ships" contained in Annex VI of the MARPOL Convention [1]. For NO_X emissions, the limits have been set in terms of g/kWh as a function of nominal engine rotational speed, although this takes no account of the actual in-service condition or operation of the engines. To mitigate SO_X emissions, the sulphur content of marine fuels is being progressively restricted in terms of percentage sulphur by mass, with provision for the use of post-combustion sulphur treatment as an alternative. Thus, these measures are focussed on fuel quality and marine engine technology, although, do not impose an absolute limit on these emissions as a function of, say, cargo tonne-miles, which will also be reliant on the design and operation of the complete ship-propulsor-engine arrangement.

The control of CO_2 emissions is also being addressed with recommendations from the IMOs Marine Environment Protection Committee (MEPC). These recommendations recognise that the mass of carbon dioxide produced is a direct function of fuel carbon content and efficiency of fuel use. These recommendations include guidelines for an Energy Efficiency Design Index (EEDI) to influence the CO_2 emissions from future ships [2] and recommendations for improving the efficient operation of cargo carrying ships through the development of a Ship Energy Efficiency Management Plan (SEEMP) [3].

The particular focus of this paper is on addressing the fuel efficiency and exhaust emissions from the harbour tug boat sector of the shipping industry. The principal challenges in this work extend beyond those already faced by the cargo shipping industry because of the highly variable nature of the duty cycles on tug boats, the fact that they operate their engines in part-load and transient-load conditions for significant amounts of time and that the metrics for environmental impact, as recommended by MEPC, are not so obvious as CO_2 per tonne-mile of cargo.

Furthermore, while the IMO legislation on ship emissions recognises the geographical significance of ship emissions by yet further limiting exhaust emissions when operating within designated Emission Control Areas (ECA), for harbour tugs, which operate exclusively within port regions of high population density, yet further improvements to local air quality may be possible through improved understanding, monitoring and operational management.

This paper presents results from sea trials on a harbour tug and, coupled with sparse engine and vessel performance information available for in-service tugs, resulted in a modification to the engine operational controls to reduce the fuel consumption and hence exhaust gas emissions for free-running operations. In addition, a simplified analytical rationale is used to predict the fuel consumption characteristics of the harbour tug in both free-running and towing modes of operation over the entire operational envelope of the engines and propulsors, which can also be applied to other vessels.

2. HARBOUR TUG BOAT OPERATIONS AND FUEL USAGE

Harbour tugs have a varied operating profile in terms of vessel speed and engine load including: pulling/pushing operations at, or close to, zero vessel speed with the propulsor operating in bollard-pull condition; and transits between locations, referred to as the free-running condition, at a variety of vessel speeds. Harbour tugs are designed with zero-speed performance in mind, relating only to their static bollard-pull; this being the principal factor in determining the size of vessel the tug is capable of assisting. Tug design over recent years has seen a steady growth in bollard-pull capability and related installed power and, although hulls have become wider and deeper to accommodate the larger engines and thrusters required to achieve these higher bollard-pulls, principal dimensions have remained fundamentally unchanged. This has resulted in even the most modern of tug designs being unsuited to efficient free-running.

By definition, a harbour tug will be required to burn fuel to achieve the required assistance to the ship it is towing. Therefore, generally, fuel consumption, when providing towing assistance, is beyond the control of the tug operator and will depend on a wide range of factors including: the size of the towed ship, the relative berth locations, environmental conditions, the size and number of other tugs assisting and the actions of the tug master or pilot. However, operational experience reveals that approximately 50% of all fuel consumed by a harbour tug is while operating in the free-running mode, in transit from the tug berth to, and between, job locations and back. Information regarding the free-running efficiency of harbour tugs is not readily available to operators, mainly due to the previously low cost of fuel and to the concentration on bollard-pull performance over fuel efficiency.

One of the most common, and effective, propulsion systems used in harbour towage is the Voith Schneider unit – due to its ability to provide close to maximum thrust in any direction almost instantaneously without the need for rudders. This is achieved through the propulsor pitch being applied, and vectored, independently to the engine rotational speed, allowing the master to leave the engines at constant speed and simply applying vectored pitch to achieve the desired effect. Whilst in some cases the engine speed can be arbitrarily varied, it is most common to find three or four preset engine speeds controlled by push buttons. In either case, this has the result that there are multiple combinations of settings of the engine and propulsor for achieving the same free-running vessel speed and, in current practice, is entirely at the discretion of the tug master. And operational experience indicates that, during free-running, the preferred combinations of engine speed and propulsor pitch setting is highly varied between tug masters; often motivated by varying perceptions of comfort or the *feel* of the vessel. Indeed, some tug masters pilot the vessels stern-first during free-running because they perceive this reduces the overall vibration levels and/or this approach *apparently* produces a smoother ride of the vessel through the water.

Clearly, fuel usage during this transiting period could be influenced by the tug operator, for example, through allowing sufficient time for the tug to attend the assisted vessel at an economic speed. Therefore, understanding the relative efficiency with respect to free-running speed for each mode of operation is vital for tug operators and tug masters to allow improved emissions and reduced fuel consumption. Furthermore, understanding the fuel-usage during bollard-pull, or assisting duties, may yet lead to further reductions in fuel consumption and resulting exhaust gas emissions.

3. RESEARCH STRATEGY

For a given tug boat there is typically only sparse information available in terms of the hull, propulsor and engine systems specifications and, what is available, usually dates back to the build date. Certainly, the relevant performance characteristics in terms of efficiencies and interactions that are required to make a complete and accurate analysis for fuel usage under a variety of conditions are not available. Therefore, the strategy adopted in this study combined empirical observations, over a series of sea-trials using an existing tug boat, with a simplified theoretical model. This allowed the limited available vessel component specifications to complement the experimental observations, resulting in a solvable approach to characterising the complete hull-propulsor-engine system that can be applied more generally.

4. SEA TRIALS

The sea trials were carried out in conjunction with a company operating a fleet of harbour tugs on the River Tees in the UK (amongst others). The tug boat used is a typical example of a harbour tug servicing a significant cargo-shipping port and is 30.6m LOA, capable of a nominal bollard-pull of 40 tonnes. The tug is propelled with twin Voith Schneider 28G11/185 propulsors, each separately driven by a Ruston 6RK270M diesel engine with a Maximum Continuous Rating (MCR) of 1450kW each. This vessel was built in 1992.

The propulsion system is controlled from the wheelhouse. Push-buttons allow the engines to be set at one of four preset rotational speeds of approximately 380, 570, 620 and 740 rpm, nominally referred to as "13%", "50%", "70%" and "100%" in the onboard literature – this "%" designation will also be used for the purposes of this paper. The pitch setting of the Voith Schneider propulsors is analogue, controlled with levers, graduated from 0 to 10, ahead and astern, corresponding to zero and maximum pitch. For free-running propulsion, both propulsor pitch levers are set to the same value and steering is achieved using a wheel which vectors the net thrust of the two propulsors. Sophisticated manoeuvres, required during towing operations, can be performed by splitting the two propulsors; applying different pitch values to each independently. The *hydrodynamic* pitch corresponding to the numerical settings of the propulsor levers is not known and therefore analysis of available data and sea-trials measurements, as described in Section 4.1 and subsequently used in Section 5.2(c), were required to account for this in the vessel performance analysis.

Each propulsion engine on this vessel is fitted with fuel consumption meters and the instantaneous fuel consumption rate is calculated by differentiating between the rate at which fuel is delivered to, and spilt back from, the engine using a pair of gear flow meters.

The vessel trials were carried out on the River Tees, following typical free-running vessel tracks for the vessel when in operation. Preparatory trials were conducted to establish the feasibility of the experiments, data collection and analysis procedures, data quality, etc. and the data reported in this paper is from a subsequent set of trials carried out during the course of a single day when the weather conditions were fair with light wind conditions.

It proved possible to establish a matrix of test conditions, in terms of engine rotational speed and pitch setting combinations, covering the full range of possible combinations except that on the lowest engine speed setting the load (torque) limit of the engines is reached at approximately pitch setting 6 of the propulsors. In keeping with accepted practice [4], trials were conducted in paired groupings with approximately the same vessel track being negotiated down-river and up-river so that wind and tidal effects are averaged out to give an accurate correlation between vessel speed through the water, propulsion settings and fuel consumption. Each individual trial was conducted in the absence of other vessel traffic and lasted for up to five minutes. Data was only recorded once the vessel had been travelling on a straight track at a steady speed for a period of time and engine operating conditions had stabilised – established through continual monitoring of salient engine temperatures. The trials results, presented in terms of litres of fuel consumed per nautical mile, are presented in Section 6.2/Figure 3.

To perform the subsequent analysis of the complete hull-propeller-engine system, as presented in Section 5, it was also possible to estimate the engine delivered power by recording the fuel rack position (graduated 0-10) at the given engine rotational speed and referencing this in proportion to the same calculation for the engine at maximum power; achieved with the fuel rack in position 10 at maximum rotational speed. Furthermore, while a limited set of Specific Fuel Consumption (SFC) data is available for the engines from their final inspection reports prior to installation on the vessel in 1992, these engine delivered power data, coupled with fuel consumption rate data, was used to update this information to account for the prevailing in-service condition and hence SFC (in terms of kg/kWh at the shaft) of the engines.

4.1 VESSEL SPEED AND BOLLARD-PULL PERFORMANCE

To assist in characterising the performance of the vessel, particularly in terms of establishing a relationship between nominal propulsor pitch setting and effective hydrodynamic pitch, and to perform the analysis of the observed fuel consumption data and subsequent modelling, the behaviour of the vessel in terms of speed through the water was analysed. In addition, as is typical for a harbour tug, a wheelhouse data sheet is available that provides guidance on the bollard-pull characteristics of the vessel for a variety of pitch settings across the range of engine rotational speeds - these curves are transcribed and presented as part of Figure 5. These have not been measured as part of this study, consequently their pedigree is not assured, e.g. they may simply be estimates and/or guidelines from the propulsor manufacturer; and if originally generated from measurements, the vessel age and condition may have changed. Nevertheless, for a complete investigation, they provide an added datum from which to reference current observations and have the added advantage, in a more general sense, that this sort of data is generally available for all harbour tugs.

While, as included in Figure 5, for the benefit of the tug crew, it is normal to present vessel speed and bollard-pull data on an axis of engine speed with contours for propulsor pitch, alternative approaches were investigated to discover if the propulsor performance could be characterised, in terms of hydrodynamic pitch, as discrete from the coupled propulsor-hull-engine system. Furthermore, utilising the data for the effects of propulsor performance, from two independent sources, under distinctly different operating conditions, namely, with vessel forward speed and under bollard-pull conditions, permitted checks to be made on the self-consistency and fidelity of the two data sets. The approach that provided greatest insight, as presented in Figure 1, was to view the vessel speed and bollard-pull data against an axis dubbed the, *load-factor*, f_L. That is, since thrust is a function of both engine rotational speed and propulsor pitch setting, the load-factor is defined as the product of the nominal pitch setting (0-10) and the engine rotational speed in rps, i.e.

$$f_L = n\theta_n \quad (1)$$

In Figure 1, data obtained for each preset engine rotational speed setting are presented as separate sets, as indicated in the legend, whereas the trend lines have been calculated for the complete set of data in each case.

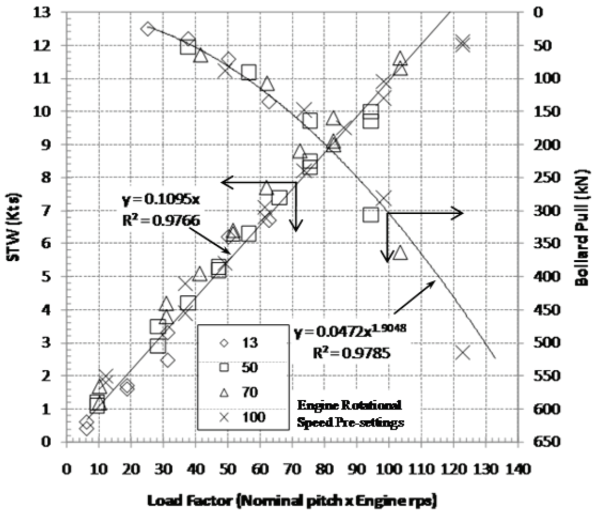

Figure 1: Vessel Speed Through the Water (STW) in knots and bollard-pull in kN as a function of load factor, f_L. Measured data at different engine rotational speed settings as indicated in the legend.

From the data presented in Figure 1, it is apparent that, for both independent data sets, there is consistency of resulting vessel speed and bollard-pull for a given load factor, f_L, with no discernable trends between the different rotational engine speed data sets. This indicates that varying the nominal pitch settings approximately corresponds to a linear variation in the effective hydrodynamic pitch of the propulsors. That is, for a given vessel speed of advance, the propulsor advance, or effective pitch in units of metres per engine revolution, is linearly proportional to the nominal pitch setting. Similarly, for a given level of thrust in bollard-pull condition, the effective pitch in units of Newton's of thrust per engine revolution is linearly proportional to nominal pitch setting.

Given these observations, the form of the respective curves in Figure 1 reveals a consistency of behaviour confirming the fidelity of the data which have come from two independent sources, namely, measurements made in this research at forward-speed and, separately, an analysis of pre-existing bollard-pull data supplied with the vessel. Considering the bollard-pull data: propulsor thrust is expected to exhibit proportionality to the square of the product of effective pitch and rotational speed (i.e. see Equations (2) & (3) for which $V_s = 0$, in bollard-pull conditions). Since nominal pitch, as just observed, is proportional to effective hydrodynamic pitch, then propulsor thrust should also exhibit proportionality approximately to the square of load-factor, as apparent directly from the bollard-pull data in Figure 1. Furthermore, this result is self-consistent with the measured observations at forward-speed. That is, propulsor thrust and ship resistance are proportional to each other, and the latter has a dependency on the square of vessel advance speed, and therefore, given the bollard-pull data indicates thrust is proportional to the square of nominal pitch, the approximate linear dependency of vessel advance speed on load-factor, apparent in Figure 1, is consistent. The modest deviations from these simplified relationships account for higher order effects in vessel resistance and losses from the hull-propulsor-engine system; these are considered in Section 5.

5. ANALYTICAL MODELLING

5.1 RATIONALE AND PHYSICAL BASIS FOR MODELLING

Following normal practice [5], to accurately predict the fuel consumption of a vessel, for example, during vessel design, it would be necessary to have complete details of all resistance components of the vessel (hull, appendages, air resistance, etc.). This would be coupled with complete performance curves for the propulsors and a full characterisation of the propulsor-hull interactions. Finally, a full characterisation of the engine performance envelopes with associated shafting and mechanical losses would be required. There are various methods available, of varying levels of sophistication and cost, to estimate all the afore-mentioned parameters. The methods selected for use would reflect the relative value of the vessel and the relative importance of the particular parameter to that vessel.

In reality, such detailed information is rarely available for any vessel, particularly not harbour tugs, and even if theoretically obtainable, the numerous un-predictable factors, such as hull roughness & fouling, wind and sea conditions, the exact state of the engine due to age and servicing, etc. means that to fully predict the in-service performance of a vessel is virtually impossible without recourse to empirical observations. Furthermore, given these uncertainties, in reality, only an approximate performance map is practicably applicable.

Noting these limitations, it was nevertheless desirable to provide an analytical rationale to aid the interpretation of sea trial results. In addition, recognising that it is not practical to undertake extensive sea trials on every tug boat in a fleet, the analytical rationale adopted provides a means by which a limited number of discrete trials data points can be used to extend the understanding of the vessel performance over a continuous performance envelope.

The desired outcome from combining the trials data and an analytical interpretation of these was the performance maps for both free-running condition and towing operations (bollard-pull), continuous across all combinations of pitch and engine speed, as presented in Figures 4 & 5 - not achievable using trials data alone. The approach adopted proved to be pragmatic and economical to apply. This makes it effectively applicable to other cases studies and is particularly useful to assess the performance of a large number of, in this case, harbour tugs, across an entire fleet.

To combine experimental measurements with the analytical approach first requires an estimate of the SFC of the engines, as a function of delivered power and the delivered power for either a given ship speed or bollard-pull force. As noted in Section 4, it is an economical task to update the engine SFC data from pre-installation trials data, as a function of MCR. Because this data is associated with delivered shaft power, it already accounts for losses within the engine. It is then required to interpret the free-running trials data and the bollard-pull chart to determine the power delivered to the propulsion system under varying combinations of pitch and engine rotational speed.

The approach adopted is to model ideal propulsor thrust generation and propulsor absorbed power according to basic momentum theory [5] and then use the trials observations to account for losses from the system.

For ideal thrust generation,

$$T = \rho A_e\left(V_a + {}^1/_2\,\Delta V\right)\Delta V \tag{2}$$

in which:
ρ, is the known water density,
A_e, is the effective area of the propulsor normal to the direction of travel,
V_a, is the advance velocity of the propulsor relative to the surrounding undisturbed fluid, idealistically modelled as the forward speed of the vessel, V_s, herein, and
ΔV, is the change in velocity of the water as it passes through the propulsor.

Idealistically, the change in velocity of the water,

$$\Delta V = (n\theta_e - V_s), \tag{3}$$

in which, n, is the rotational speed of the propulsor or, as used in this case, engine rotational speed, as long as the gear-ratio constant of proportionality is either in- or ex-cluded from any subsequent calculations as appropriate and, θ_e, is the effective hydrodynamic pitch of the propulsor in units of metres per unit revolution of propulsor or engine as appropriate.

Similarly the *ideal* propulsive power can be predicted from momentum theory, see the first term in Equation (5), which accounts for the change in kinetic energy of the water flowing through the plane of the propulsor. In this approach, power losses were assumed to consist of two parts. Firstly a base-line power loss, P_O, which is required to rotate the shaft and propulsor at a given speed, n, when set to zero pitch, accounting for mechanical friction losses in the shafting and propulsor mechanism as well as the hydrodynamic friction losses from the propulsor in a zero-pitch condition. The other element of power loss is assumed to be a function of increasing propulsor pitch, P_P. I.e.

$$P_D = P_i + P_O + P_P. \tag{4}$$

The reason for this approach is that it provided a rational physical basis, and hence a function, with sufficiently few unknown coefficients that they can be reliably populated using the limited available data. Yet, while remaining solvable, this approach provided a relatively accurate description of the observed trends in the measured data, as presented in Figure 2. Populating Equation (4) with the relevant parameters and unknown coefficients provides the complete equation used to predict total delivered power as presented in Equation (5). Details of the respective terms and analysis method for this delivered power equation are provided in Section 5.2.

$$P_D = {}^1/_4\,\rho A_e(n\theta_e + V_s)\left(n^2\theta_e{}^2 - V_s^2\right) + k_1 f(n) + k_2 n^3\theta_e^2 \tag{5}$$

5.2 ANALYSIS PROCEDURE

To solve Equation (5) the coefficients, k_1, k_2 and the effective hydrodynamic pitch, θ_e, which varies according to propulsor pitch setting, are required and these are considered next.

5.2(a) Zero Pitch Losses

The variable, k_1, to account for zero-pitch losses, is found from trials observations. In this case this was achieved through observing the fuel consumption rate at the four different set engine rotational speeds with the pitch levels set to zero and back-calculating a delivered power through the known SFC for the engines. Data was directly available for the SFC down to 10% of MCR from the factory trials on the engines prior to installation on board, and therefore was directly used in the cases of 100, 70 and 50% engine speed settings where the zero pitch losses were greater than 10% MCR. For the cases of 13% engine speed, where the zero pitch losses were less than 10%, the SFC curves were extrapolated backwards using a curve of the form given for the characteristic behaviour of engine fuel consumption on low load, as published in [6]. An alternative approach, using the fuel rack position multiplied by engine speed could also be used in practice.

5.2(b) Non-Zero Pitch Losses

The rationale for the last term in Equation (5), accounting for losses due to increasing propulsor pitch is, following standard wing-theory for lifting surfaces [7], and hence propulsors of Voith Schneider type (or, indeed, screw propellers), the drag coefficient of a lifting surface will generally be a square-law to angle of attack, or for propulsors, pitch and speed of advance. Therefore the total drag force will be in a form,

$$\mathrm{Drag} = \mathrm{kV_e^2\theta_e^2} \tag{6}$$

where V_e is some effective velocity and the constant, k, accounts for fluid density, a reference area and drag coefficient.

This drag force acting at speed will constitute a power loss, i.e. in a form,

$$\mathrm{Power\ Loss} = \mathrm{kV_e^3\theta_e^2} \tag{7}$$

For Voith propulsors, with a vertical axis of rotation and a relatively high rate of rotation, a reasonable reference speed can be related to that of the rotational speed of the propulsor. Thus, the resulting form of power loss due to pitch and rotational speed was modelled according to,

$$P_P = k_2 n^3 \theta_e^2 \tag{8}$$

Finding variable k_2 requires that pitch-related power losses and the effective hydrodynamic pitch are actually known for at least one condition. The approach adopted is explained in Section 5.2 (c).

5.2(c) Effective Hydrodynamic Pitch

While the nominal pitch, θ_n, at each vessel speed is known, the effective hydrodynamic pitch is not. However, having noted from Figure 1, the fact that the nominal pitch when at forward-speed is linear with respect to the effective hydrodynamic pitch, it only remains to find this constant of linear proportionality, k_3, for,

$$\theta_e = k_3 \theta_n \tag{9}$$

For this to be accomplished, at least one instance where the effective pitch is known is required. In fact, finding both pitch loss variable, k_2, and the constant of proportionality for effective hydrodynamic pitch, k_3, is actually a set of two simultaneous, albeit non-linear, equations for these two unknowns, i.e.

$$P_{D1} - P_{O1} = {}^1\!/_4\, \rho A_e (n_1 k_3 \theta_{n1} + V_{s1}) \left(n_1^2 k_3^2 \theta_{n1}{}^2 - \mathrm{V_{s1}^2}\right) + k_2 n_1^3 k_3^2 \theta_{n1}^2 \tag{10}$$

$$P_{D2} - P_{O2} = {}^1\!/_4\, \rho A_e (n_2 k_3 \theta_{n2} + V_{s2}) \left(n_2^2 k_3^2 \theta_{n2}{}^2 - \mathrm{V_{s2}^2}\right) + k_2 n_2^3 k_3^2 \theta_{n2}^2 \tag{11}$$

For the purposes of this study there were nine instances for which the total delivered power, under known conditions was estimated and thus an iterative scheme to best fit coefficients, k_2 and k_3 was used.

6. PRESENTATION OF SHIP TRIAL RESULTS

6.1 VALIDATION OF THE ANALYSIS PROCEDURE

Figure 2 presents the corresponding relationship between estimated power and resulting calculated delivered power from the engines. The modelling rationale proved successful, since the assumption that a single variable to account for pitch related losses and a single variable to account for effective hydrodynamic pitch, while apparently simple, does fit the measured data well.

This has the further implication that an approximate resistance curve for the vessel could also be back-calculated using Equations (2) & (3). While this equation assumes the thrust is generated from an ideal perspective, the fact that the effective hydrodynamic pitch for the, apparently idealised, delivered power is computed from fitting the curves to empirical measurements, it has intrinsically accounted for the physics of propulsor-hull interaction, wake losses, etc.

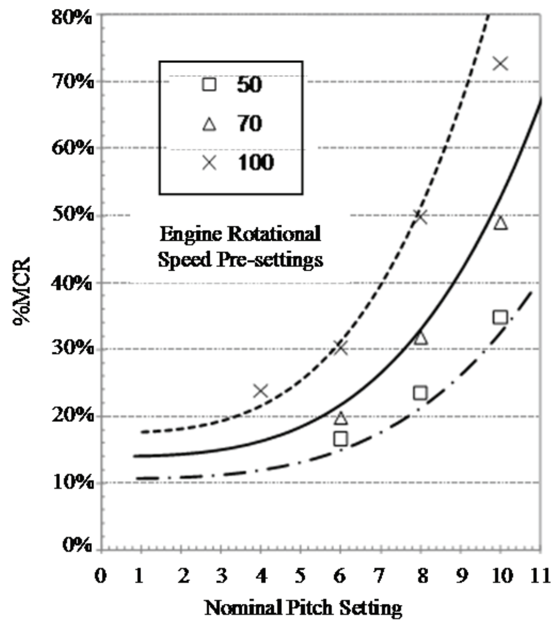

Figure 2: Predicted and estimated % MCR as a function of nominal pitch. Contours for predictions and data points for measurements at different engine rotational speeds.

6.2 FUEL CONSUMPTION DATA

While fuel consumption, as measured on board, is in units of litres per hour, this does not fairly represent a useful unit to judge the efficiency of the system. Following the rationale of the IMO's EEOI [8], a more meaningful unit is to present fuel consumption as a ratio to useful effort. For a harbour tug, the useful work is providing assistance to other vessels and it is somewhat difficult to put a denominator on this, since a denominator of "per job" does not take into account the size of assisted vessel (and hence the useful "cargo" tonnage assisted). However, during free-running operation, i.e. in transit from berth to job location or between job locations, the useful work carried out from the perspective of that particular tug can be simply viewed as transit progress, or transport of the asset itself. Therefore, the fuel consumption data in Figure 3 is presented in terms of litres per nautical mile through the water at different vessel forward speeds. The data sets are presented for each engine rotational speed setting and lines connecting the same pitch setting data from each set are added for clarity. The continuous contours are the equivalent predicted fuel consumption using the methods detailed in Section 5.

The experimental data generally fall into clearly separate groups, excepting some amount of scatter in the data with apparently some overlap between the 100 and 70% sets at a vessel forward-speed of about 10 knots. For each engine speed setting, the experimental data reveals an optimum vessel forward-speed and hence optimum pitch setting, in terms of fuel usage. This optimum transit speed tends to reduce as the engine speed is reduced and although there is necessarily sparse data for the lowest engine rotational speed setting due to engine over-load issues, at the more practical free-running engine rotational speeds (nominally 100, 70 & 50%) this transit speed tends to correspond to a consistent propulsor pitch setting of around about 6.

Additionally, there is a trend for the fuel consumption to reduce as the engine speed is reduced at a given vessel forward-speed, generally indicating that a crude strategy to adopt is to use the lowest engine speed setting available, capable of achieving the required speed within the pitch limit of the propulsors.

Aside from the minor overlap between the experimental data in the 10 knot region, there is generally good agreement between the analytical approach adopted and the experimental observations and this therefore provided a reliable means to produce a continuous performance map for the coupled engine-propulsor-hull system over the full operating range of the tug boat.

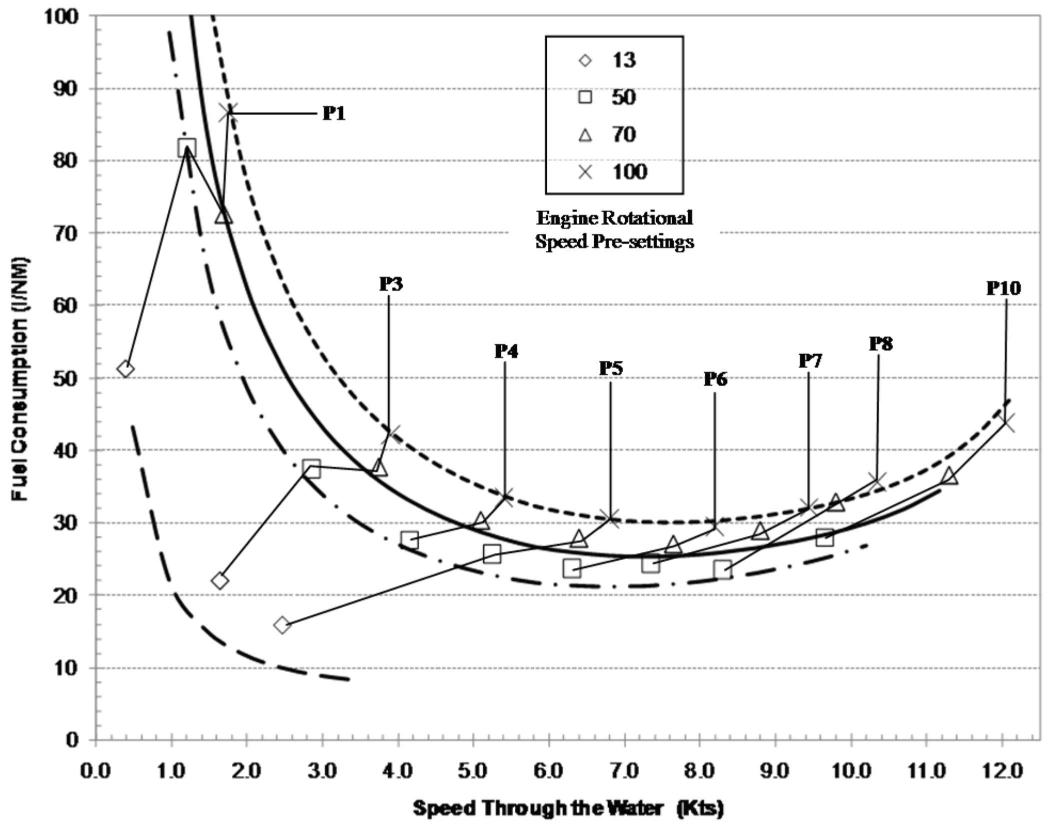

Figure 3: Fuel consumption rate in litres per nautical mile as a function of vessel speed. Contours and data sets for predictions and measurments for different engine rotational speeds.

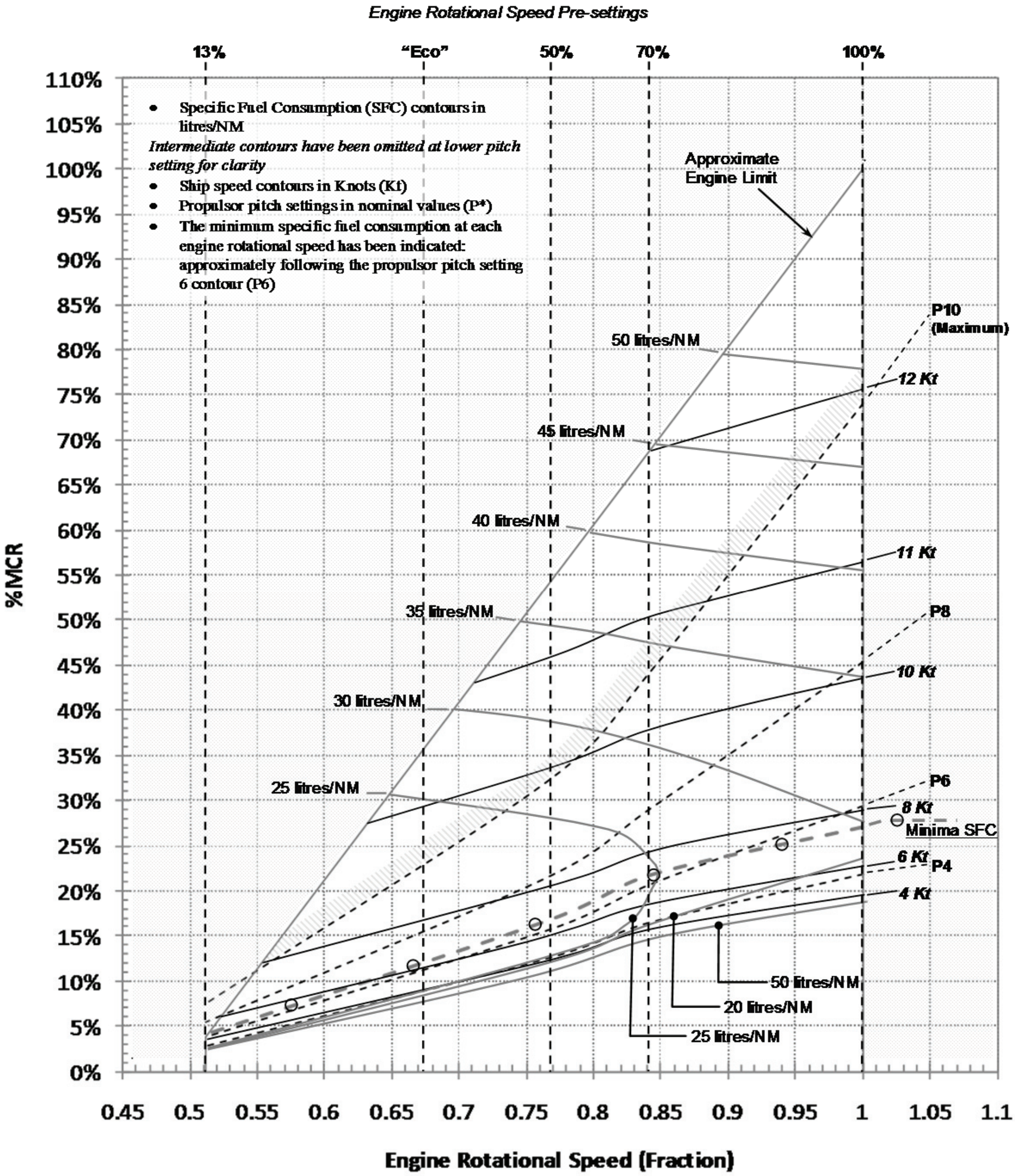

Figure 4: Vessel free-running perfromance map.

7. FORWARD SPEED AND BOLLARD-PULL PERFORMANCE MAPS

Having established that the analytical approach to estimating the fuel consumption of the vessel gave good agreement with directly measured results it was used to calculate two performance maps spanning the complete operational envelope of the engine-propulsor-hull system. As illustrated in Figure 4, a free-running performance envelope combines engine rotational speed (with the preset settings highlighted), engine power, vessel forward-speed and propulsor pitch setting. This configuration of presentation is used in other applications to illustrate, so-called, combinator control schedules for controllable pitch propellers (CPP) as can be found in the publically available literature from most leading CPP manufacturers, but unusually in this case, the performance diagram has also been complimented with a contour map of specific fuel consumption in terms of litres per nautical mile, which has accounted for the total engine-propulsor-hull performance, rather than simply the SFC of the engines in terms of fuel mass per unit shaft energy output (e.g. g/kWh). The form of the performance map in Figure 4 is a highly informative tool for operators and tug crew alike.

For free-running, the physical operating limits for the vessel are bounded by the maximum pitch (P10) line or the engine limit line at low engine rotational speed, which combined, run approximately diagonally in a direction bottom left to top right, and the minimum and maximum engine rotational speed lines running vertically on the left and right hand sides. A line connecting points of minimum fuel consumption at each engine rotational speed has also been added and this approximately follows the contour for pitch setting 4 (P4) as noted in Section 6.2.

This presentation clearly and conveniently gives information on the fuel consumption implications for any pitch and engine speed setting for a given vessel forward-speed. Notably, there is a relatively flat-bottomed "valley" of low fuel consumption toward the lower left corner of the approximately triangular region of the operational envelope, sided at the lower edge by a steep "cliff" of rapidly increasing fuel consumption rate as propulsor pitch is reduced below pitch setting 6 (P6).

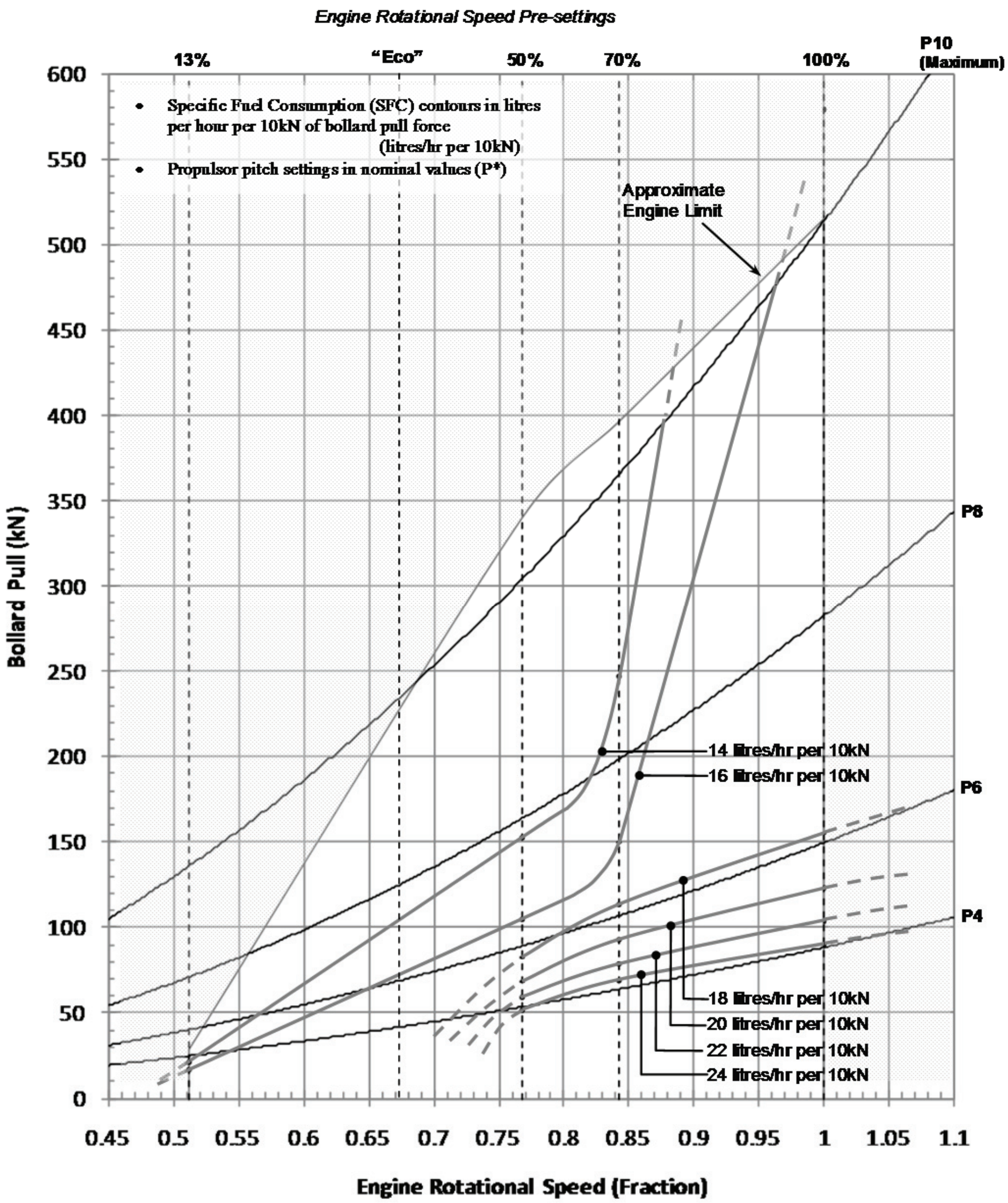

Figure 5: Bollard-pull performance map.

In addition and unusually, using the analytical approach, it has also been possible to follow a similar rationale to produce a corresponding fuel consumption map for the tug in bollard-pull condition. This is illustrated in Figure 5 in which bollard-pull force, on the vertical axis, can be determined for engine rotational speed and propulsor pitch combinations, as usually presented in the wheelhouse of the tug. In this case, fuel consumption has been expressed in a ratio to useful output, in units of litres per hour per unit bollard-pull force (in 10kN for convenience). This reveals a general decrease in specific fuel consumption as engine speed is reduced while pitch is increased.

8. DISCUSSION

While considering the tug at forward speed, operating from purely a fuel economy point of view, it would be desirable to select engine speed and pitch combinations in the valley of low specific fuel consumption while remaining at pitch settings above 4 to avoid straying into the “cliff” region where fuel economy deteriorates rapidly. Pragmatically, when free-running, the transit duration becomes an additional factor to consider and particularly low free-running speeds means that the vessel performance becomes susceptible to the influence of external factions such as wind and tide which can be detrimental to vessel handling, steerage and traffic avoidance can become problematic. Furthermore, economic factors such as crew costs and crew working hours begin to dominate the operational considerations As a consequence of this research and the practical consideration the engine control system was modified with the inclusion of an additional engine rotational speed at ~67% of maximum (dubbed “Eco” setting) and this allows the tug crew to operate well within the valley of low fuel consumption, yet still being able to achieve practically reasonable forward-speeds (typically 6-8kts), with the resulting consequence of net fuel savings.

The pitch and engine speed settings during bollard-pull operations, as noted in the introduction, are not, as yet, always within the direct control of the tug operators. For the sake of being able to rapidly respond in terms of manoeuvrability and bollard-pull levels in complex and safety-critical towing and pushing operations, the current practice is to set the engines at high rotational speeds and use the highly responsive propulsion pitch settings to be able to rapidly go through the full range of thrust conditions. This strategy also reduces the complexity of the control of the vessel from the perspective of the tug master because the only variable is the pitch settings rather than having to manually coordinate both engine speed and pitch during these rapidly changing situations.

For the sake of providing engineering insight, the *index* used in this research to compare fuel (or ultimately emissions) to *useful work* during bollard-pull operations is litres (of fuel) per hour per unit bollard-pull force and this illustrates a new way of scrutinising these operations, highlighting the opportunities for fuel usage (and emission) reduction at modest levels of bollard-pull force if some of the operation could take place at lower engine rotational speeds. And, while the tugs under current scrutiny do not have this facility, automating the pitch and engine speed in a combinator mode might eventually facilitate reduced fuel usage during towing operations while remaining safely controllable by the tug master.

9. CONCLUSIONS

Using a limited set of sea trials, this research has presented the fuel consumption, in terms of litres per nautical mile, for a typical harbour tug boat under free-running conditions and this has been used to characterise the performance under free-running conditions for a variety of engine rotational speeds and propulsor pitch combinations.

A simplified theoretical approach has been shown to give good agreement with the measured results and applying this rationale, performance maps covering the full range of operating conditions have been produced. This has allowed the identification of a region within the operational envelope at which it is desirable from an improved fuel consumption perspective, while also recognising practical considerations for tug boat operation. This approach offers an efficient method for characterising other harbour tug vessels in the same terms, allowing further studies to reduce the fuel consumption and the resulting emissions for harbour tug fleets. Ultimately, these studies have led to modifications on the particular harbour tug under consideration to permit it to operate well within the economic region of operation.

The analytical approach has, unusually, also permitted a characterisation of harbour tug performance in terms of fuel consumption under bollard-pull, or towing, conditions. This informs the understanding of how existing tug operation might be modified to further reduce fuel consumption and emission production. This also informs the rationale for future tug design and highlights the need to consider what useful numerators and denominators could be used to fairly develop exhaust emissions indices and efficiency measures for vessels such as tugs, undertaking complex and varied operations.

10. ACKNOWLEDGEMENTS

The authors would like to thank the tug crews and Svitzer for their support and assistance for this research.

11. REFERENCES

1. INTERNATIONAL MARITIME ORGANISATION, *MARPOL 73/78, International Convention for the Prevention of Pollution from Ships, Annex VI, Prevention of Air Pollution from Ships*, 2005.

2. INTERNATIONAL MARITIME ORGANISATION, Interim Guidelines on the Method of Calculation of the Energy Efficiency Design Index for New Ships, *Marine Environment Protection Committee (MEPC)*, 2009.

3. INTERNATIONAL MARITIME ORGANISATION, Guidance for the Development of a Ship Energy Efficiency Management Plan (SEEMP), *Marine Environment Protection Committee (MEPC)*, 2009.

4. INTERNATIONAL TOWING TANK CONFERENCE (ITTC), Full Scale Measurements Speed and Power Trials, Preparation and Conduct of Speed/Power Trials, *Recommended Procedures and Guidelines, Revision 3, 10pp,* 2005.

5. LEWIS, E. V., Ed., Principles of Naval Architecture, Vol. 2: Resistance, Propulsion and Vibration, Pub. *The Society of Naval Architects and Marine Engineers, 327 pp,* 1988.

6. WOUD, H. K. and STAPERSMA, D., Design of Propulsion and Electrical Power Generation Systems, *Pub. IMarEST, 494pp, ISBN 1-902536-47-9, 2002.*

7. KATZ, J. and PLOTKIN, A., Low-Speed Aerodynamics, *2nd Edition, Pub. Cambridge University Press, 613 pp*, 2000.

8. INTERNATIONAL MARITIME ORGANISATION, Guidelines for Voluntary Use of the Ship Energy Efficiency Operational Indicator (EEOI), *Marine Environment Protection Committee (MEPC)*, 2009.