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High Performance Winches for High Performance Tugs – Winch and HMPE Rope Limitations

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SYNOPSIS

In the past two years, Markey Machinery Company has been expanding its line of High Performance Render/Recover™ Hawser Winches to meet the demands for escort tugs that are being called upon to operate under increasingly strenuous offshore conditions.

The new flagship model is four 760hp DESDF-48WF-760hp double drum hawser winches developed for Moran Towing and Groupo Boluda for use on tugs to be based at the new Energia Costa Azul LNG terminal located near Ensenada, Mexico on the Baja California peninsula. These tugs are required to bring tankers to dock in 3m 10-second period sea conditions and were preceded by 100hp, 200hp and 250hp versions developed for other customers.

Markey is employing cutting-edge technologies such as 3D-modelling, finite element analysis and numerical modelling in conjunction with model-testing to increase its capability to produce designs that meet the increasingly demanding requirements for ship assist and escort. As a contributing partner in the Marin Safe Tug project, the company is working to improve the overall safety and efficiency of the companies that operate at exposed offshore and near-shore terminals.

INTRODUCTION

The Markey ARR winches and PSR Plasma HMPE ropes fitted to tugs servicing Sempra Energy's Energia Costa Azul LNG terminal near Ensenada, Mexico, are among the first tug components to benefit from extensive modelling of ship-assist operations in open ocean service, including piloting and docking manoeuvres, tug motion analysis, and the numerical modelling of winch and rope performance. We have learned much in the run-up to these projects and have at hand the ability, once these tugs go into service, to close the loop on the design assumptions and simulations fundamental to the specifications for this terminal. By this time next year we will have real world data that will undoubtedly help to refine our assumptions and equations. We hope to have the opportunity to report on these matters next year. Nevertheless, this real world data will provide only one additional point on the industry's operational experience curve, and hardly enough, even when combined with anticipated improvements in simulation and modelling, to allow spot-on predictions beyond these conditions, such as higher seas, shorter periods, steeper lead angles, and higher bollard pulls. With our current knowledge, can we at least predict how far can we push the limit of winch and rope technology?

WHAT ARE THE MECHANICAL AND MATERIAL LIMITS?

Examination of the average trend in high performance assist and escort winches and ropes during the past 20 years reveals increases in winch power from 55kW to more than 550kW, generally without commensurate increases in tug direct bollard pull. Current winch and rope technology is intended for operation in 3m seas, and upward towing angles limited to 10 degrees or less. How far can we go in applying recently developed tug and machinery technology to dynamic sea conditions, using traditional and advanced winches, such as Markey's ARR Escort Winch and HMPE ropes?

The primary forces, geometry and motions of importance to the winch and rope system are:

- Bollard pull (BP) required by the pilot to be applied to the assisted vessel chock;
- Hawser lead angle from the tug staple or bullnose to the assisted vessel chock;
- Accelerations and inertia of the tug staple or bullnose due to hull motion;
- Geometric and motion limits of traditional tugs.

Winch inertial limits

The primary function of a high-performance winch in dynamic seas is to maintain a predictable average line tension, while avoiding slack line conditions. Therefore, the winch must accelerate quickly enough to keep up with the changing directions of inhaul and payout, as well as maintain the maximum speed in either direction. As applications move towards larger

waves, are there inherent limitations in machine capability to meet these requirements?

The characteristics that dictate the winch power must first be evaluated. If the working line tension is held constant, the required winch power becomes directly related to the maximum line speed. Based on the Pierson-Moskowitz sea spectrum, *Figure 1* shows the vertical tug velocity as sea state increases. It is assumed that the tug follows a sinusoidal wave form, and that this movement is proportional to the relative line length between the tug and tanker.

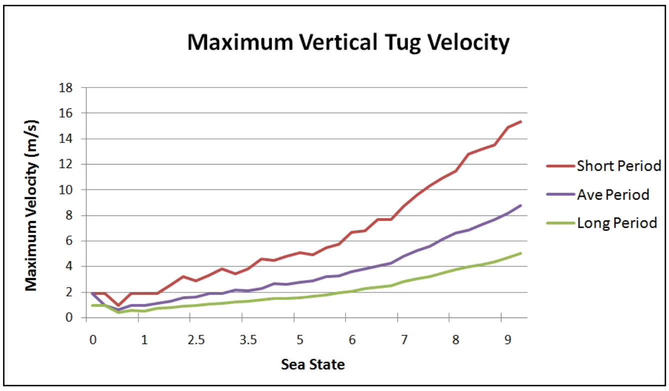

Figure 1: The vertical tug velocity as sea state increases.

The same assumptions are used to calculate the maximum tug accelerations shown in *Figure 2*. Although a wide range of values is predicted, the sea state has little effect on the acceleration. Therefore, higher sea states require higher maximum speed and power, but do not necessarily require faster acceleration.

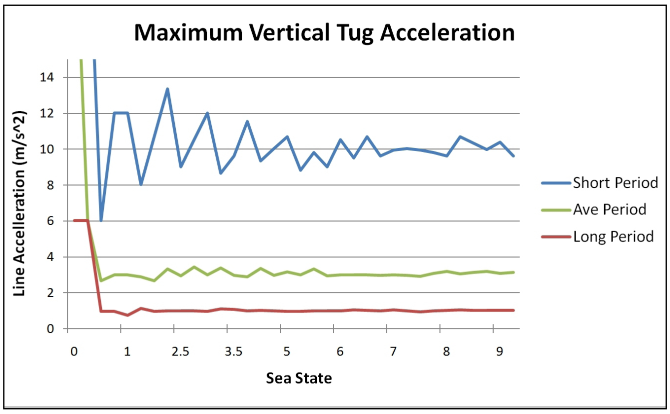

Figure 2: Maximum tug accelerations.

Nevertheless, as winch power increases, there are a number of factors that affect acceleration. Larger components make the winch drivetrain heavier, requiring more power to maintain a given acceleration. In addition, the gear ratio also tends to decrease in order to maintain a constant line pull. This significantly increases the total inertia seen by the motor because it is considered as the square of the gear ratio.

The following equation represents the inertia of a shaft in a power transmission, as reflected on to another shaft. $I_{shaft1} = I_{shaft2} (1/R^2)$

If shaft2 is the drum shaft, and shaft1 is the motor shaft, I_{shaft2} is the moment of inertia of the drum shaft assembly and I_{shaft1} is the inertial value of that drum shaft as seen by the motor. R is the gear ratio between the two shafts. As the gear ratio is reduced, the time required by the motor to reach maximum speed is increased.

However, these effects are counteracted when considering the acceleration of the line, rather than the motor. Although the motor requires more time to accelerate, the decreased gear ratio results in higher line speeds. Therefore, the line acceleration does not suffer the same reduction as the motor acceleration. This is seen in the following study of two theoretical winches.

The first winch is modelled with two electric motors, providing 567kW to a single-speed, single-drum gear-train with approximately 85 per cent mechanical efficiency. The first gear reduction at the input shaft is 2:1. The second winch is based on four motors, delivering 1134kW directly to the second shaft, while eliminating the original input shaft from the previous model. The remaining gears and shafts see the same torque loads, but rotate at twice the speed. This maintains the final line pull while doubling the power, and dividing the gear reduction by two. The results are shown in *Table 1*.

Winch Power (KW)	567	1134
Line Pull (tonnes)	23	23
Line Speed (m/s)	2.14	4.28
Total Inertia at Motors (kg*m^2)	109.0	255.9
Line Acceleration (m/s^2)	2.71	4.61

Table 1: Winch acceleration factors.

Despite the increase in inertia, the line acceleration increases due to higher line speeds that result from a given motor speed. There are several additional factors not considered here, such as drivetrain layout optimisation to reduce inertia, and operational features such as multiple speeds and drums, which tend to increase inertia.

These results suggest that increased acceleration is not required as applications move towards heavier seas. In addition, winch acceleration is not compromised as power increases to provide higher maximum line speeds. However, more studies are needed to investigate more complex and irregular wave forms and their effects on tug motions.

Hawser thermal limits

The second critical component is the hawser line connecting the tug to the tanker. Typically, a synthetic hawser is spooled on the winch drum, leading forward to the tug staple and up to the tanker at some lead angle. This creates a point of fatigue at the staple, exaggerated by the constant back-and-forth tug motions. This has been, and continues to be, studied extensively for various applications, and by various

parties. It is found to be highly complex, with several non-linearities and apparent inconsistencies. The mechanics of rope strength and fatigue is beyond the scope of this paper, and requires additional research to be fully understood. However, an investigation of known failure modes, considered in the context of dynamic offshore conditions, presents some insights as to the likely questions and concerns of this portion of the escort tug system.

A simplified diagram of the rope-staple interface is shown in *Figure 3*. Fl is the line tension, Theta is the lead angle, and Fr is the resultant force of the staple acting on the line. The line friction is neglected for the purpose of calculating Fr. Increasing Theta causes a roughly linear increase in the resultant force Fr, as shown in *Figure 4*.

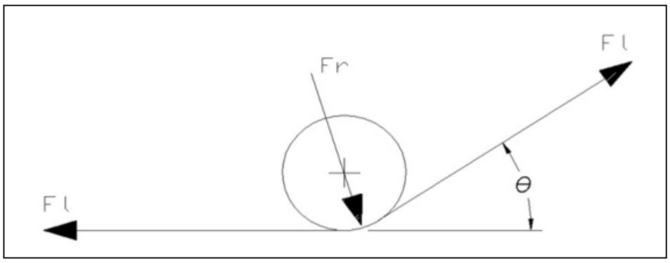

Figure 3: Rope-staple interface.

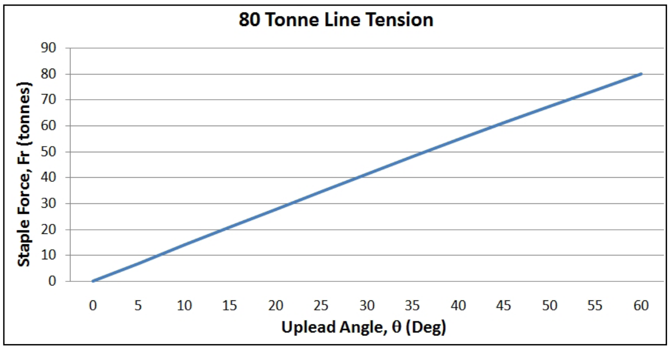

Figure 4: Line force on staple due to uplead angle.

As a result of these forces, friction and abrasion occur both at the staple surface, as well as within the rope braid itself. This causes heating and mechanical damage that reduces fibre strength.

On the rope surface, direct friction with the staple can be calculated based on somewhat predictable friction coefficients. If we assume a vertical line lead, then the line velocity is equal to the vertical tug velocity found earlier. As the tug moves away from the tanker, horizontal movements become more significant, but are still related to the general sinusoidal wave form. For simplicity and general trend observations, the line speed is assumed equal to the vertical tug velocity and a constant line lead of 10 degrees is considered. *Figure 5* shows the heat generated from friction with an 80-tonne line tension.

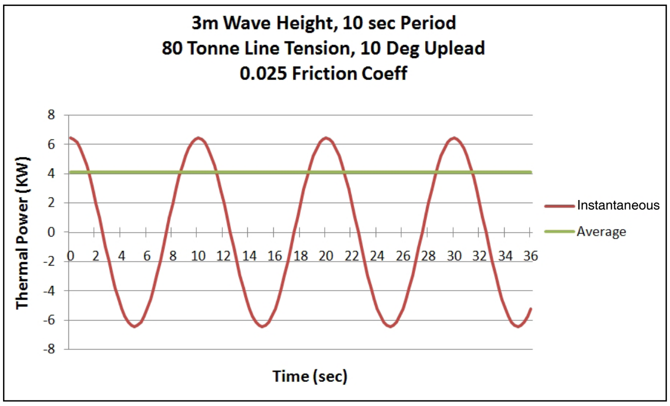

Figure 5: Thermal power from friction.

In this example, a maximum of 6.4kW of heat is generated at the staple-rope interface. An average value of 4.1kW is found by multiplying 2/pi. Therefore, the rope and staple must dissipate this heat without reaching excessive temperatures in the rope fibres. As the sea state increases, the heat generated by staple friction increases in proportion to the velocity, as shown in *Figure 6*.

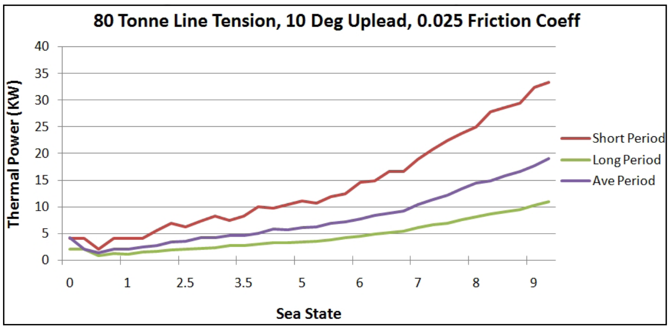

Figure 6: Average thermal power from staple friction.

The dissipation of surface heat is perhaps the most easily addressed, as several methods are available. The staple itself provides a large thermal mass with good conduction properties. In addition, water spray cooling could be easily added.

Internal rope friction and abrasion is much more complex to understand and address. Experiments have observed melted fibres at the rope core, as well as significant sliding movement between individual braids, suggesting the significance of these factors on overall rope strength and fatigue resistance.

Internal friction is caused by at least three factors: firstly, the simple tensioning of a straight section of rope causes the fibres to stretch relative to each other, resulting in fatigue damage; secondly, the bending of the rope over a surface, or sheave, causes the rope cross-section to deform and internal friction to occur; and thirdly, as a friction force is applied to the rope surface, a shear force is generated within the braid, again causing internal friction. This is seen on powered sheaves as well as tug staples.

The magnitude of the effects of these factors is based on the fibre strength at elevated temperatures, braid

configuration and friction coefficient between the rope fibres. Because the individual fibres deform relative to each other, the contact pressure between them, as well as the geometry and friction properties, vary depending on the braid and load. To further complicate matters, the ropes are often coated for different surface properties, introducing yet another variable for internal friction. Once heat is generated by the friction, more issues arise in predicting the heat dissipation. The changing internal geometry affects the insulation value of the rope, as well as the effectiveness of external water sprays.

However, despite these monumental complexities, it can be generally assumed that larger ropes will experience greater internal heating effects. This is due to both the increased friction from larger internal deformations, as well as increased insulation from the environment. To quantify these effects, additional research is needed on large diameter ropes at full operating loads. In addition, real-world data is needed to qualify the experiments and establish reasonable, simplifying assumptions for new applications.

Ultimately, the most significant limiting factors of the winch-hawser system appear related to line pull. This requires a heavier winch, disproportionally increasing the power required to maintain acceleration. In addition, the internal damage in larger ropes is likely to be more significant. Conversely, it is shown that higher winch speeds can likely be accommodated with proportional increases in power. Although the line does require more heat dissipation at higher speeds, a smaller line size allows more effective cooling. With continued research, combined with real-world data collection, we are advancing our understanding of these effects. This will allow more accurate predictions for future applications, as well as optimising the energy management potential of the many winch and rope system components.