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Structural Design of a Harsh Environment - 4 Legged Jack-Up Boat

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INTRODUCTION

The *Irish Sea Pioneer* is a new generation liftboat, termed an Operating Support Vessel (OSV). It is self-propelled with a hull length of 180 feet (55 meters), a beam of 92 feet (28 meters), four lattice legs 242 feet (73 meters) long, and four 360° azimuthing thrusters. The elevating system is rack and pinion electric drive, using AC motors with infinitely variable speed control and jacking speeds up to 12 feet (3.7 meters) per minute. She was built by Bollinger Shipyards, Inc, in Louisiana for Halliburton Inc., with completion in the summer of 1995. She will operate on the Liverpool Bay Project under charter to BHP for the next 15 years, providing wireline and other services to three unmanned platforms and various other facilities in the field. The three unmanned platforms are minimal facilities, without cranes, helidecks, accommodations, and power or pumps for fire fighting. When elevated alongside an unmanned platform, the *Irish Sea Pioneer* provides all these facilities.

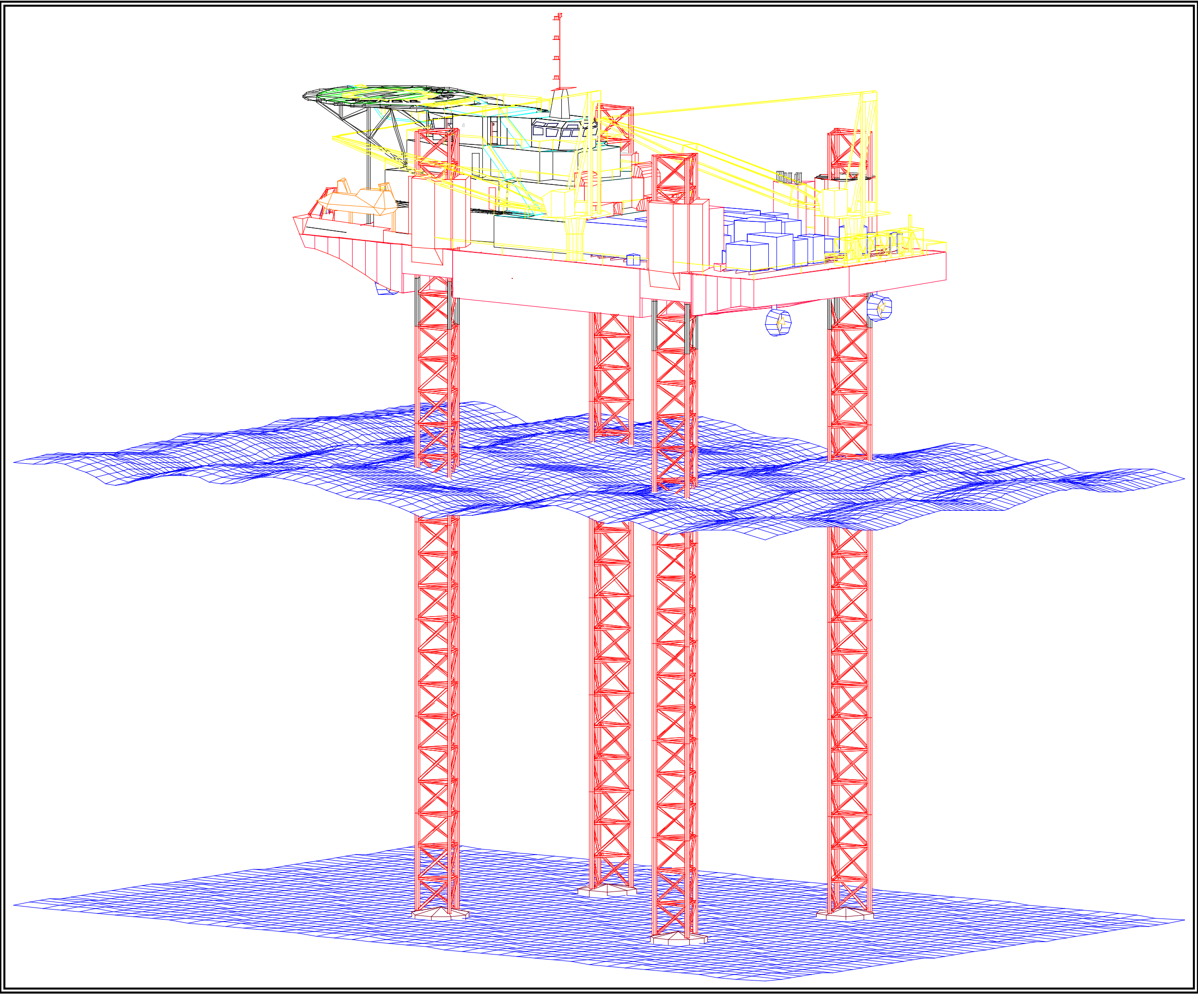

FIGURE 1 - *Irish Sea Pioneer* Elevated

VESSEL MISSION

The OSV, the *Irish Sea Pioneer*, is to service the Lennox, Hamilton, and Hamilton North unmanned platforms in Liverpool Bay, as well as the manned Douglas facility and the floating storage facility. Additionally, the OSV may be used for other in-field activities, including pipeline survey and diver support. Inter-platform distances range from 10 to 21 km. The shortest platform to shore distance is 5 km. The OSV transits under her own power from shore base (Liverpool) to site, and from site to site in the field. She has accommodations for 42 people and can carry sufficient fuel and fresh water for a 21 day mission.

The water depths at the platforms where the *Irish Sea Pioneer* must elevate vary from 24 feet (7.3 m) at LAT to 128 feet (39 m) at the deepest site, including HAT and a 50-year storm surge. The main deck is to be raised to be level with the weather deck of each platform, resulting in air gaps of up to 94 feet (28.6 m). She must manoeuvre into position alongside each platform using her own propulsion (four 360º azimuthing thrusters) and jack up with a relatively tight positional tolerance. When elevated, cranes on the OSV are used to lift coiled tubing and other equipment for well servicing from the OSV aft deck onto the platforms.

Frequent moves between platforms are required and must be accomplished within a few hours. As many as 50 field moves a year may take place. This frequent move requirement, and the requirement for self-propulsion, constitute the greatest difference between the design philosophies for liftboats and jack-ups.

ENVIRONMENTAL DESIGN CONDITIONS

Table 1 - Operating Environmental Conditions on Liverpool Bay

Environmental Condition	1-year	10-year	50-year
1-min wind speed @ 10m height	30.3 m/s (59 kt)	35.3 m/s (69 kt)	38.8 m/s (75 kt)
Predominant direction for storm winds	W-N-W	W-N-W	W-N-W
Significant wave height, Hs	5.5 m (18.0 ft)	6.9 m (22.6 ft)	7.8 m (25.6 ft)
Associated zero crossing period, Tz	8.0 seconds	8.9 seconds	9.5 seconds
Maximum (3-hour) wave height, Hmax	9.5 m (31.2 ft)	11.8 m (38.7 ft)	13.3 m (43.6 ft)
Associated period range, Tmax	8.4 - 11.2 seconds	9.3 - 12.5 seconds	10.0 - 13.3 sec
Current @ 1 meter water depth	120 cm/s (2.3 kt)		137 cm/s 2.7 kt)
Strongest currents from west			
Extreme minimum air temperature	-2.8 ° C		
Extreme maximum air temperature	25.2 ° C		
Extreme minimum sea surface temp.	2.6 ° C		
Extreme maximum sea surface temp.	21.5 ° C		
Max. water depth for operations	127.6 feet including 50-year storm surge and tide		

The sea bed in this area is generally sandy, with a hard clay beneath dense sand in some locations. Spud can penetrations in the range 1 m to 3 m are anticipated.

DESIGN REQUIREMENTS AND CONSTRAINTS

The principal design requirements are associated with the OSV's mission. The OSV is classed by ABS as ✠A1 Self Elevating Unit ACCU and has a UK Certificate of Fitness as an Offshore Installation and a Self Elevating Mobile Unit on the UK Continental Shelf. ABS acted as the Certification Authority on behalf of the UK Health and Safety Executive (HSE).

The performance requirements are listed below.

Platform Support Requirements

The OSV provides well workover services, personnel accommodation, platform control, platform utilities support, platform fire fighting assistance including water supply, and platform maintenance facilities. A helideck and accommodations for 42 people are part of the platform support requirements.

The OSV can provide the following support services (Reference 1 provides further information):

- Hydraulic Workover
- Wireline Operations
- Slickline Operations
- Coiled Tubing Operations
- Well Testing
- Well Stimulation
- General Well, Pipeline and Platform Maintenance, Including Pigging
- Diving Support Operations
- Remotely Operated Vehicle (ROV) Support Operations
- Construction Support Operations
- Accommodation Support

The large open workdeck area is used to carry specialized equipment for the above services. This is illustrated in Figure 2.

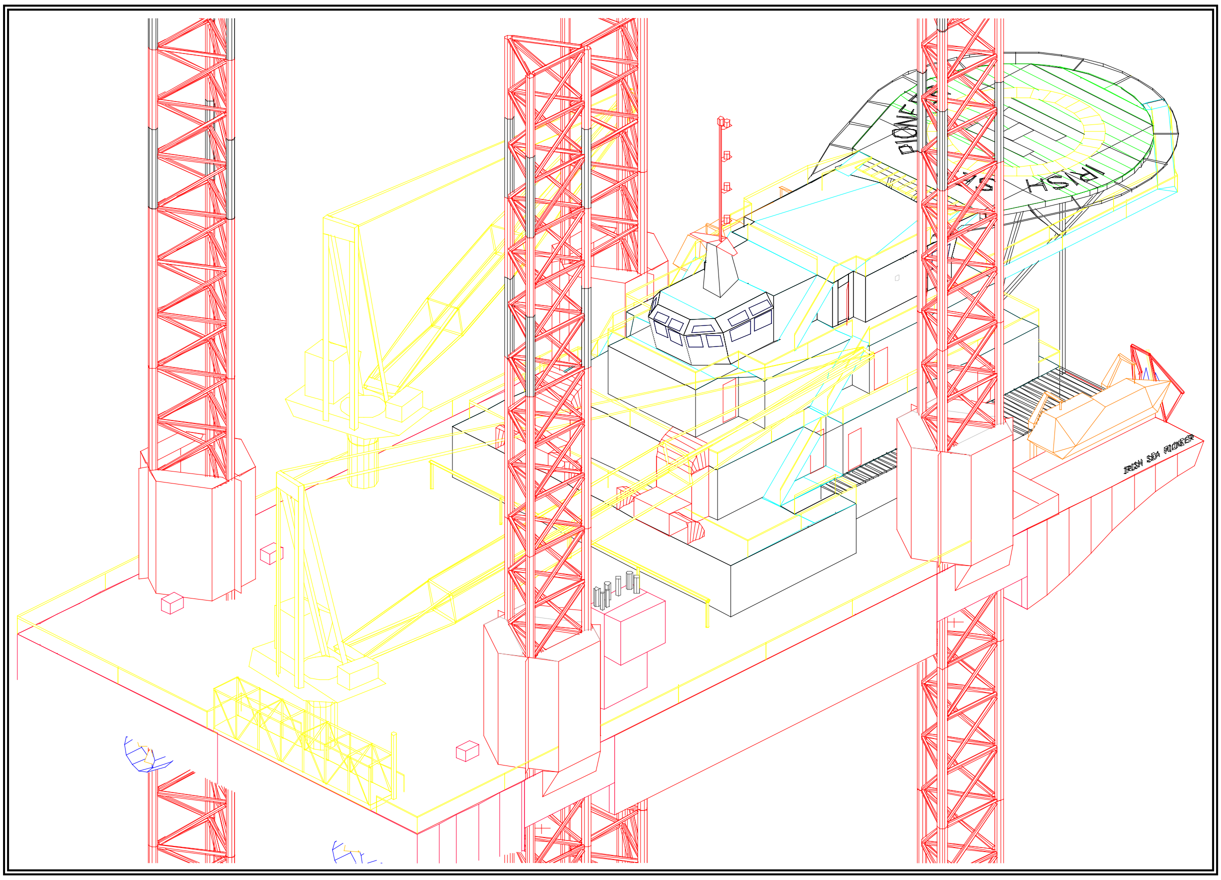

FIGURE 2 - Showing Large Workdeck Area and the Two Cranes

Variable Load Requirements

In order to perform the platform support work a variable deck load of 500 kips (2224 kN, or 227 tonnes) is required. The deck area required for this equipment is around 5000 square feet (465 square meters).

The requirement to house 42 people drives the size of the superstructure and the power for the hotel load. A helideck is also a requirement.

Elevating Requirements

Each of the four legs are powered by 12 LeTourneau Number 4 AC motors each capable of developing 30 kW. The jacking motors are controlled by a Siemens AC Variable Speed Drive System which is capable of achieving infinitely variable speeds ranging from 0-12 ft/min (3.66 m/min) when raising or lowering the legs and 0-6 ft/min (1.83 m/min) when raising or lowering the vessel.

The four legs can be controlled as a unit or independently. The system allows automatic vessel leveling, automatic load balancing between legs, and load monitoring on each leg. Braking is provided by a regenerative braking system dissipating the power generated into resistor banks. The motors are equipped with a fail-safe braking system using disc brakes which can only be released after power has been applied to the motors. This Siemens control system enables the electric motors to develop full torque at 0 rpm.

Transit Requirements

The OSV is required to make a transit speed of 6 knots (3.1 m/s) in 2 m (maximum wave height) seas against a 20 knot (10.3 m/s) wind.

PRIMARY VESSEL SYSTEMS

Electrical Power Generation

The OSV is equipped with six electrical generators, giving a total installed electric power of 4160 kW (another 2238 kW of direct drive diesel propulsive power is available). These comprise two ships service generators, each rated at 350 kW and 600 V, three auxiliary generators, each rated at 1070 kW and 600 V, and an emergency generator rated at 250 kW and 480 V. There are four switchboards, one ships service switchboard, two auxiliary switchboards, and an emergency switchboard. All of the generators and switchboards are situated in diverse locations to provide redundancy in the event of a fire in any compartment. Also various boards and generators can be connected in parallel in order to provide redundancy of supplies. Essential systems such as the Fire and Gas/Emergency Shutdown (F&G/ESD), Vessel Management System (VMS) and Navaids have dedicated uninterruptible power supply (UPS) battery systems to provide power in the event of a total blackout.

Two auxiliary generators and auxiliary switchboard Number 1 are located in Machinery Room Number 1. The third auxiliary generator and the two ships service generators are located in Machinery Room Number 2 along with auxiliary Number 2 switchboard and the ships service switchboard. The emergency generator and switchboard are located in a compartment on the second level of the deck house.

The vessel has a power management system which controls and monitors the electrical generating plant. This will monitor loads and shed non-essential loads and/or bring additional generators on line to maintain power supplies to essential items. The system can also be controlled remotely from the bridge where there will be facilities for monitoring, start/stop, synchronization and paralleling, load shedding, and control of generator circuit breakers.

All essential loads are supplied from the emergency switchboard which, in normal operation, is fed from the ships service board. The ship service board has both a 600 V and a 480 V bus. The 480 V bus is fed from the ship service 600 V bus via two separate 600/480 V transformer banks. The auxiliary boards Numbers 1 and 2 have only 600 V buses to feed the large power consuming equipment. Auxiliary board Number 2 can be paralleled with either auxiliary board Number 1 or the ship service board.

The large auxiliary generators are designed primarily to supply power for the forward thrusters, the jacking system, and the platform service fire pumps and associated submersible sea water lift pumps. A schematic view of the machinery arrangement is shown in Figure 3.

The emergency generator is designed to supply those loads which are essential for the safety of the vessel. The emergency generator will start automatically upon sensing loss of power to the emergency switchboard. It is designed to be able to start up and supply power within 45 seconds. Essential loads are maintained via battery powered UPS during this interval.

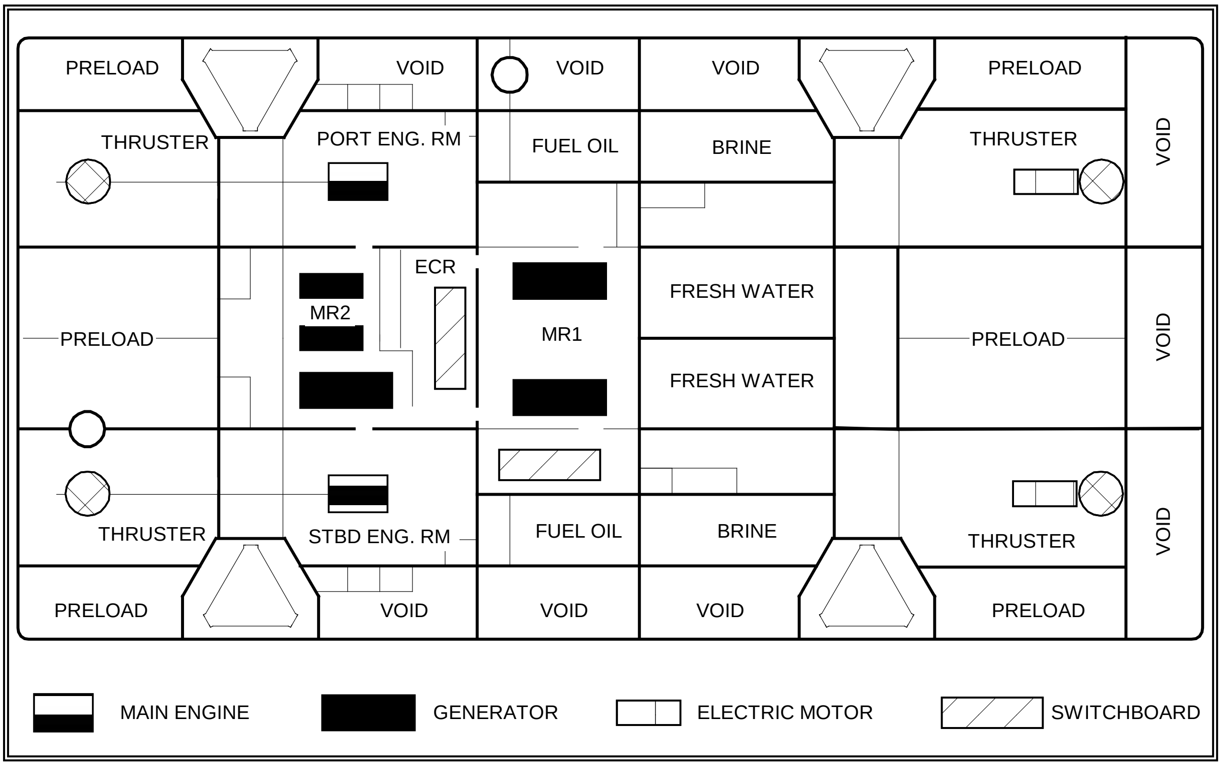

FIGURE 3 - Machinery Arrangement Schematic, Showing Watertight Bulkheads

Propulsion

The OSV develops its propulsion and manoeuvring power from four Aquamaster 1401 azimuthing thrusters. The aft units derive their power from two Caterpillar 3512 diesel engines, each rated for continuous operation of 1500 hp (1119 kW) at 1800 rpm. The forward units are each directly coupled to a 1200 hp (895 kW) Reliance electric motor which is controlled by a Siemens variable speed AC drive.

The net thrust (Bollard pull) for each unit is as follows:

Forward units each	37,770 lbf (168 kN)
Aft units each	42,940 lbf (191 kN)

The installation of multiple thrusters under the hull of a liftboat will subject the thruster to various effects which will cause a reduction in their effective thrust. In order to minimize this reduction the following effects were considered:

- Mutual thruster interference. A thruster exposed to the high speed outflow jet of another thruster is subject to a certain reduction in thrust. The thruster operates at higher inflow velocity which causes a reduction in thrust even if constant power can be maintained. This interference was avoided by restricting certain thruster azimuth positions in the thrust allocation logic of the control system.
- Hull interference. The deflection of the propeller jet by the hull causes a direct reduction of the available thrust. This was minimized by lowering the center of each propeller even with the vessel baseline and providing gradually sloping lines in both the bow and stern of the vessel.

- Appendage interference. At certain azimuth angles, the outflow jets of the thrusters are directed towards the legs and spud cans. The resulting drag force is directly opposed to the thrust and must be deducted from the net effective force. While in transit these forces were significantly reduced by recessing the spud cans into the hull so they are nearly flush with the baseline of the vessel.

Each thruster has an independent solid state electronic control system provided by the manufacturer. Triplicated direct thruster controls are provided, one each in the forward and aft station on the navigation bridge and one in the engine control room (ECR) below decks. These systems individually control thruster direction and power.

In addition there is a 'master' control system which allows control of all thrusters simultaneously via a single joystick control. This system is supplied by SIMRAD and is software based. The software calculates the vectors required by the thrusters based on the movement of the joystick by the operator and controls the thrusters to the appropriate orientations. The joystick system is self-diagnostic and can automatically recover from three out of four thruster failures utilizing whatever power is available to maneuver the vessel. An internal failure in the joystick system itself causes an alarm condition and transfers control to the individual thruster control units. The thruster control systems are powered via the emergency switchboard and are provided with back-up supplies via the vessel UPS.

The propulsion system was designed to allow the vessel to safely approach a live platform in a 1.1 m significant sea state (2 m maximum wave height) with a 1.9 knot current and 30 knot winds. This arrangement allows sufficient capability to meet all manoeuvrability requirements even after the failure of any one thruster.

Firewater

The firewater system provides protection for the OSV by means of a fire hose distribution network throughout the vessel, fire monitors, and a sprinkler system within the deckhouse. The OSV also provides firewater to the platforms it is designed to service. This is done by means of attaching hoses from the OSV to the platform firewater and deluge systems. The system is designed such that firewater protection is available at all times to the OSV whether afloat, jacking, or jacked up and servicing a platform.

The philosophy that a failure of a single component should not render the firewater system useless is used throughout the design by having redundancy of equipment, separation of equipment, and installation of isolation valves.

The distribution of water is divided into two rings. The first is the ship service fire main ring to provide water to the OSV. A separate ring is used for delivery of fire water to the platform the OSV is servicing. Both of these mains utilize jockey pumps to maintain pressure. The platform ring can, through pressure reducing valves, also act as a backup to the OSV ring.

To prevent damaged pipes from rendering the OSV fire ring inoperative, two isolation valves can be closed to effectively divide the system in two. Fire stations are supplied by the ring in such a manner that damage to half of the fire main ring would not prevent any area of the OSV from being without fire hose protection.

When jacked up, submersible pumps lift water to the OSV. Three submersible pumps are located on the top of each of the two forward spud cans. The ship service submersible pump is to provide the OSV with water to cool the diesel engines and provide water to the ship service fire pump suction. The rating of this pump is 800 US gallons per minute (gpm) at 97 psi (182 m^3/hr at 6.7 bar). This pump is run whenever the legs are submerged.

The two other pumps are used only in the event that the OSV has to provide fire water to the platform. The rating of these pumps is 2200 US gpm at 97 psi (500 m^3/hr at 6.7 bar).

The pumps in one leg are capable of providing all water necessary to supply the vessels ship service supply and fire main, as well as supply the platforms fire main. A completely redundant system is provided in the opposite leg.

The main ship service fire pump is an electric motor driven centrifugal pump dedicated to the vessel fire ring main. It is sized to meet the largest delivery required for the vessel, which is to provide deluge water by way of fire monitors to a 50 ft x 30 ft (15.24 m x 9.14m) deck area at a rate of 450 US gpm (102 m^3 /hr).

In order to provide firewater to a platform the vessel has boost pumps to deliver 3435 US gpm at 102 psi (780 m^3/hr at 7.0 bar). Two electric motor driven centrifugal pumps, piped in parallel, provide this. For redundancy two identical pumps in a separate compartment serve as backups. In addition, each of these four pumps can be used as a backup or an addition to the OSV ship service fire pump.

The submersible pumps supply water via a riser (pipe) which runs inside of the lattice structure of the leg and is sized at 250 mm, nominal diameter, to carry flow from all three submersible pumps. This pipe runs to the top of the leg and mates to a hose that runs down to a take-up reel on the OSV deck. This allows uninterrupted water flow during jack-up or jack-down operation. There is no separate raw water tower as found on many jack-ups.

The hose section is sized at 150 mm, nominal diameter, to carry flow from the engine cooling/OSV fire water supply submersible pump that is running during jacking and while the OSV is jacked-up.

Once the OSV is in the jacked-up position a 250 mm jumper hose is connected to the 250 mm pipe inside the leg via take-offs with shut-off valves and couplings that are located at various intervals along the leg. This connection supplies the water to the OSV's platform fire pump suctions.

The ship service fire pump can be activated manually from the bridge deck F&G panel, the engineer's operating station, the pump motor starter, or from manual call points located throughout the vessel, including locations near the helipad.

Automatic operation will occur if pressure loss sensors in the OSV fire main ring detect low pressure, for example, if a fire hose nozzle or a sprinkler head is opened.

Vessel Management System

The Vessel Management System (VMS) is the centralized machinery control system which is required to allow the vessel to achieve ACCU classification by ABS and operate with unattended machinery spaces. The system is currently configured with more than 600 operating/monitoring points and is designed to allow full control of the machinery from the bridge deck or Engine Control Room (ECR) without the need for operator intervention. The main control station is in the ECR but is capable of delegating control to the bridge.

The system is the G-DATA ship automation system provided by PRAXIS Automation Technology. It consists of self running process control stations and operator work stations linked by a highly reliable, redundant high speed LAN network running on coaxial cable. The redundancy is provide by installing two separate cable loops running along different routes where practicable.

There are four work stations from which the vessel can be operated, two in the bridge and two in the ECR. The work stations are completely independent and each acts as a hot standby for any other station. All of the vessel's primary systems are operated/monitored via this VMS, including main engines, generators, thruster units, jacking system, and ratchet chocks. Figure 3 shows the overall system as a block diagram.

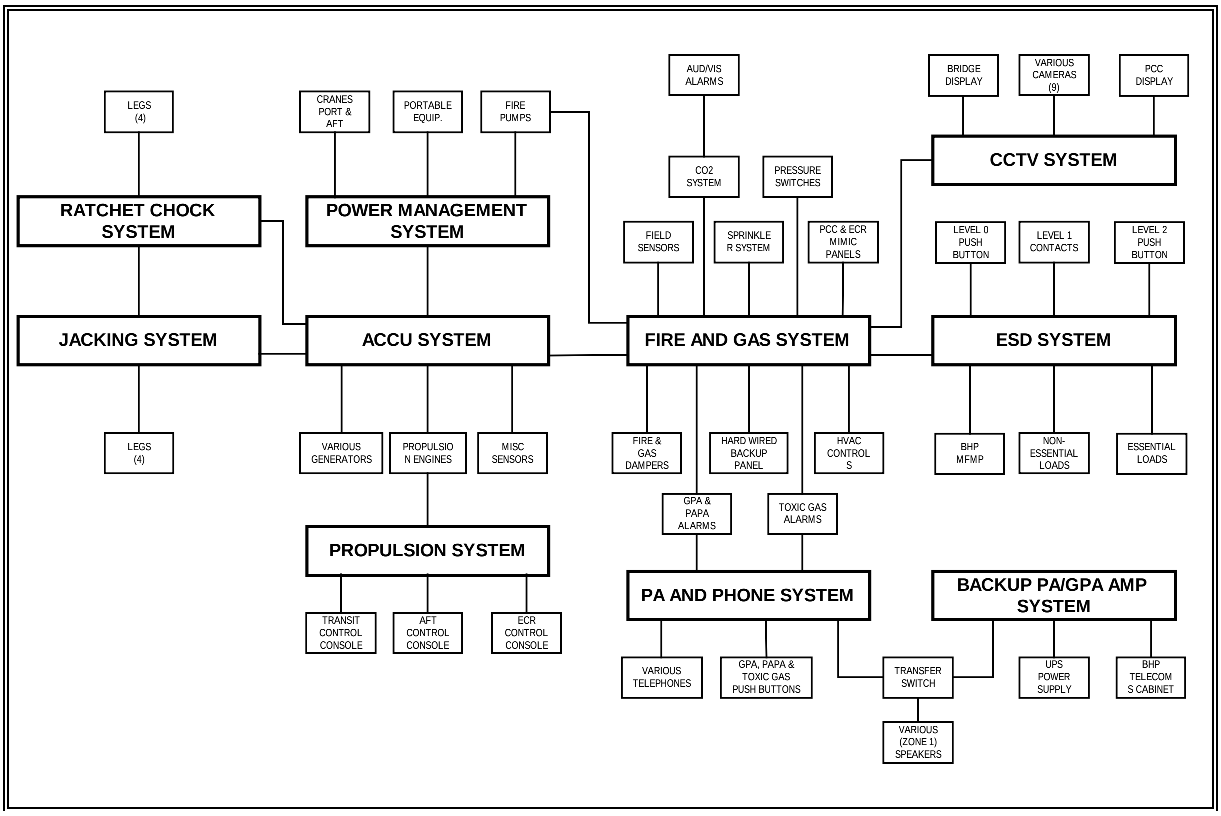

FIGURE 4 - Overview of Vessel Management System

Vessel Safety Systems

The *Irish Sea Pioneer* has a number of dedicated safety systems, include the following:

Fire and Gas Detection System

The OSV is equipped with a comprehensive F&G detection system covering all areas of the vessel and all types of hazard which could be encountered. Fire detection is achieved by means of smoke, heat, or flame detectors (or combinations of these) located throughout the vessel. Multiple detectors are used to allow a voting system in order to avoid spurious trips.

Gas detectors (both flammable and toxic) are located on the OSV external areas and in all HVAC intakes. The F&G system is also supplied with spare capacity to allow the connection of outputs from detectors located on deck mounted equipment. The F&G system is linked to the OSV emergency shutdown and alarm systems which provide audible alarms and executive actions based on detector responses.

Emergency Shutdown System

This system is designed to provide output signals which will shut down appropriate items of the OSV plant as well as provide electrical isolations. There will be an interface between the OSV ESD and the platforms ESD when the OSV is in combined operations.

General Alarm & Public Address System

The OSV has a general alarm system with voice over capability for public announcements. The dial telephone system is linked to the PA system to allow announcements to be made from any phone point. The system will be linked to the F&G system for automatic alarm under certain conditions and will also have manual call points.

Fixed Fire Fighting Systems

The OSV has three primary fixed fire fighting systems, the fire water system via hoses and monitors with foam injection capability, the accommodation sprinkler system, and a CO_2 system for machinery spaces. In addition there is a dedicated galley hood snuffing system.

Portable Fire Fighting Equipment

All areas of the OSV are equipped with portable extinguishers of a type appropriate for the nature of the fire hazards in that area. These will include CO_2, water, and foam extinguishers.

Cranes

The OSV is equipped with two Manitex cranes on the aft work deck, one located on the port side near amidships and one located on the starboard side near the stern. The port side crane has an 80 ft (24.4 m) boom and has a maximum rated load of 36 tonnes. The stern crane has a 110 ft (33.5 m) boom and is rated for 35 tonnes. Both cranes have man rated whip lines. The port crane is designed for supply boat operations. The stern crane is for lifting equipment to and from the platforms.

The cranes are elctro-hydraulically powered and fitted with joystick type controls. The cranes are each fitted with two 200 HP electric hydraulic pumps which are powered via the auxiliary switchboards Numbers 1 and 2. The two pumps on each crane are connected to different switchboards so that loss of a switchboard does not cause total loss of power to the cranes. In addition, if power was lost totally, the system is designed to lock hydraulically and the load can than be set down in a controlled manner.

<u>TRANSIT AND AFLOAT STABILITY</u>

The *Irish Sea Pioneer* is classed by ABS as a ✠A1 Self Elevating Unit and satisfies similar afloat stability criteria as jack-up drilling rigs. Figure 5, below, shows the righting and heeling arm curves for 100 knot wind conditions, with the vessel in its fully laden condition (displacement 4000 tonnes), with the cans flooded and the legs lowered 24 feet (7.32 m). The area ratio curve is also shown in Figure 5 and is seen to peak at just above 1.4 at a heel angle of around 20º. The vessel vertical center of gravity (VCG) is approximately 34 feet (10.4 m) above the baseline (keel) in this condition. Down flooding is seen to occur at around 21.3º and a positive range of righting arm is seen to extend past the 30º required by the HSE. The air intakes to machinery spaces are the first downflooding points. They were positioned by the requirement to be at least 4 m above the <u>damaged</u> waterplane with the OSV subject to a 50 knot wind.

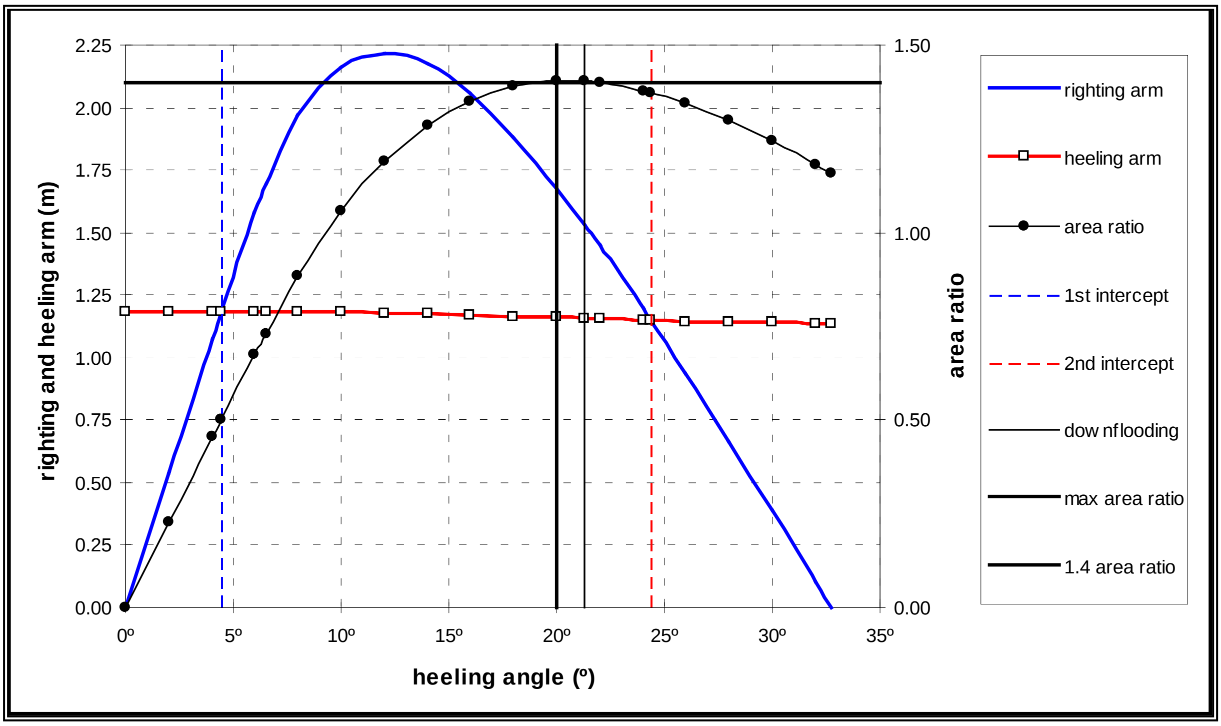

FIGURE 5 - Stability curves for fully laden vessel with legs lowered 24 ft (7.32 m) (100 Knot Wind).

Figure 6 shows the righting and heeling arm curves for 70 knot wind conditions, with the vessel in its fully laden condition, with the cans flooded and the legs fully raised.

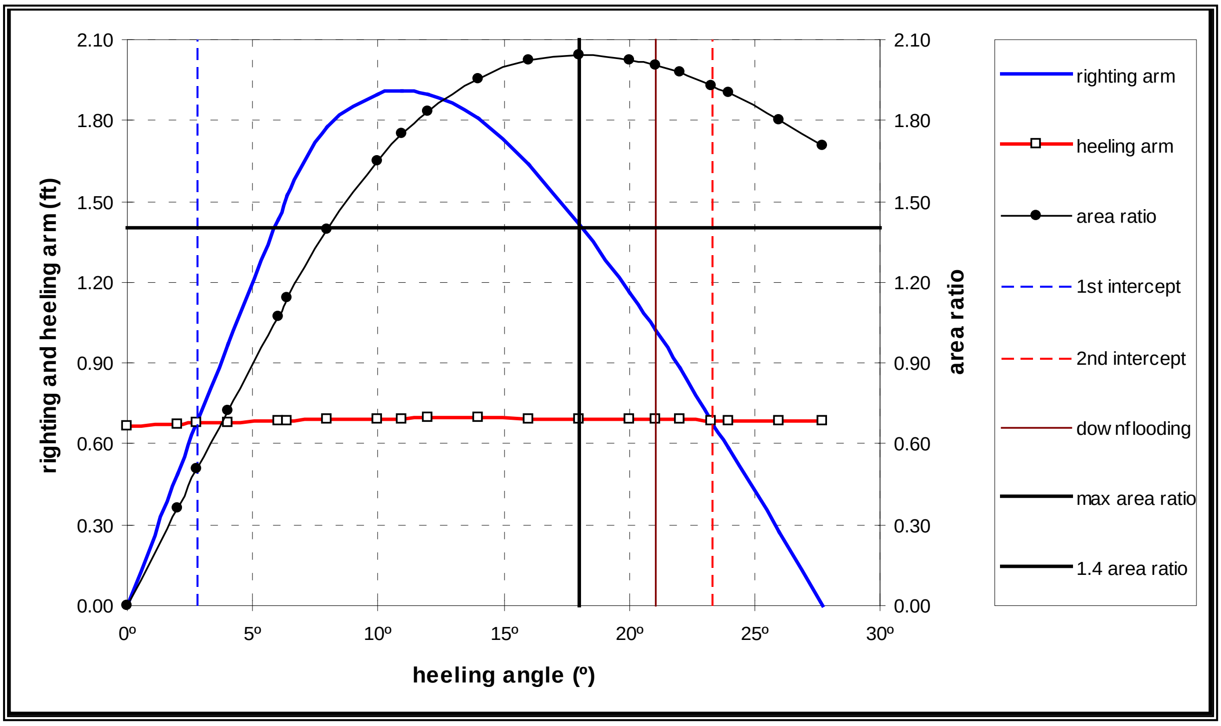

FIGURE 6 - Stability curves for fully laden vessel with legs fully raised (70 Knot Wind).

The area ratio curve is also shown in Figure 6 and is seen to peak at just over 2.0 (well above the 1.4 minimum requirement) at an angle of around 18º. The vessel vertical center of gravity (VCG) is approximately 38 feet (11.6 m) above the baseline (keel) in this condition. Down flooding is seen to occur at around 21.2º and a positive range of righting arm is seen to extend to nearly 30º (although this is not a requirement). Note that the area ratio peaks at around 18º and drops slightly (in this case) at the angle of first downflooding. In other cases investigated, a larger difference between the peak of the area ratio curve and the area ratio at either the second intercept, or downflooding, has been observed. The point here is that the maximum value of the area ratio in the range of stability is the governing value, not the area ratio at the limiting angle.

The equilibrium heel angle for 70 knot winds is seen in Figure 6 to be around 3º. The critical wind direction is on the beam. Deck edge immersion occurs at 6.4º for the full load condition (heeling with little or no trim). This compares with a maximum equilibrium heeling angle of 4.4º under a 50 knot wind in damaged condition. The damage conditions considered two compartments as being free flooding if the damage point was at a bulkhead between two compartments at the headlog, the transom, or the sideshell. For several damage conditions considered, this is more severe than the HSE requirements. The HSE requirements permit single compartment damage where watertight bulkheads are more than 3 m apart. Penetration into the hull of 1.5 m was considered to cause damage to an internal compartment, in accordance with the HSE requirements.

The maximum allowable VCG curves are shown in Figure 7. Note that the upper curve, corresponding to 70 knot wind conditions, is the governing curve for the vessel's normal operations. When the OSV moves from location to location on the Liverpool Bay project, or goes back to port, the limiting transit conditions are regarded as being equivalent to those for a jack-up field move. Note also that the curves in Figure 7 have been produced without consideration of the additional buoyancy forces that will result at large angles of inclination as parts of the superstructure are immersed. The upper curve is changed negligibly when the superstructure buoyancy contribution is included. However, an increase of around one foot to the maximum allowable VCG, at load line displacement, would occur for the 100 knot curve. Conservatively, the beneficial effect of superstructure displacement has not been allowed in formulating the maximum allowable VCG curves.

Note that the limiting conditions for going onto and coming off locations are governed by a combination of vessel motion responses (to first order waves) and wind/current forces. These are described later in this paper and correspond to environmental conditions significantly less severe than those characterized by a 70 knot wind. Consequently, it is anticipated that under normal operating conditions the OSV will not make transits in 70 knot winds and should certainly never be exposed to 100 knot winds in the afloat condition.

Under full load conditions (displacement 4000 tonnes) the OSV has a draft of 11 feet (3.35 m) and a freeboard of 5 feet (1.52 m) when on even keel. Some slight trim is anticipated reducing the aft freeboard to 4.5 feet (1.37 m).

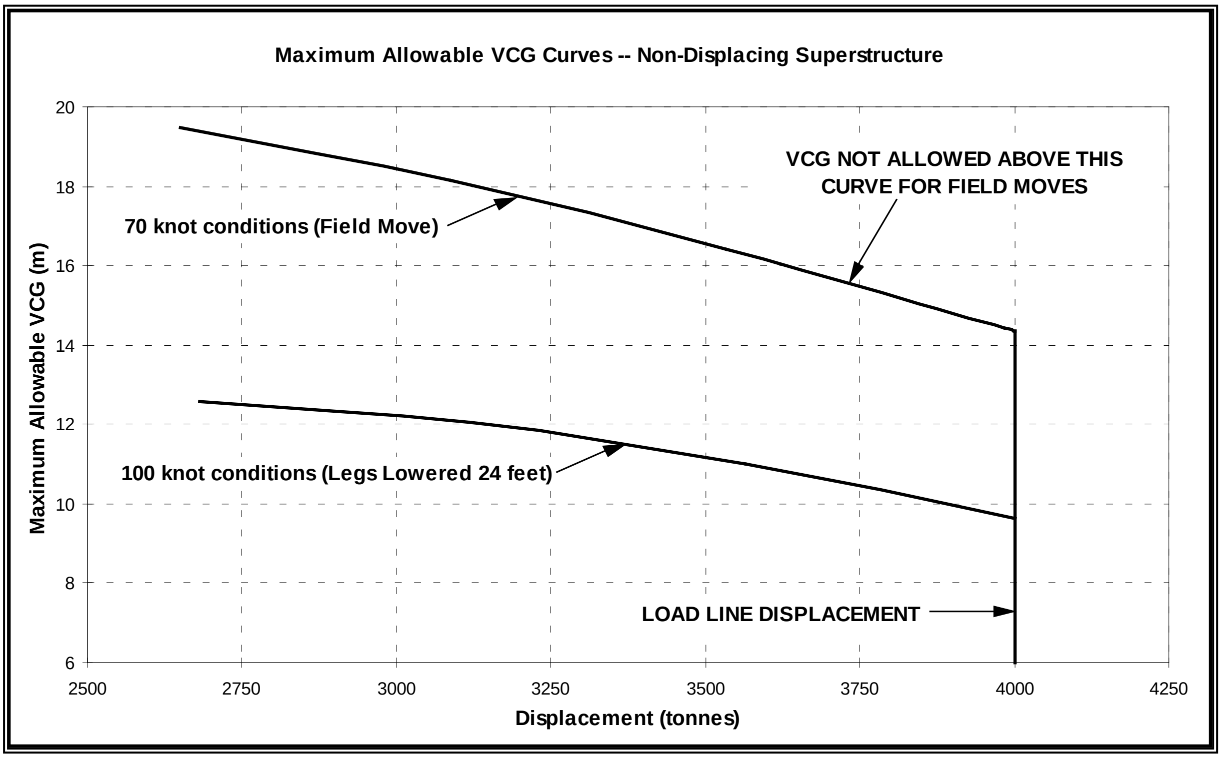

FIGURE 7 - Maximum allowable VCG curves for *Irish Sea Pioneer*.

<u>VESSEL MOTIONS IN WAVES</u>

The *Irish Sea Pioneer* will operate in relatively mild sea states when afloat in transit. The vessel has a natural roll period in the fully loaded condition (with the legs fully raised (VCG approximately 38 feet) of around 12 seconds and a pitch period of around 8 seconds. The roll and pitch response amplitude operators (RAOs) for beam and head seas are illustrated in Figure 8.

The roll motion is non-linear with wave height. Note that the RAO in Figure 8 corresponds to a 2 m wave height. Figure 9 shows the predicted non-linearity.

Figure 10 shows the anticipated roll and pitch motion responses to waves with 1 m, 2 m, and 3 m height and with periods ranging from 4 to 20 seconds. The pitch responses are calculated for <u>head</u> seas while the roll responses are calculated for <u>beam</u> seas. The OSV is in the full load condition with the legs fully raised.

The motion predictions are based partly upon the AQWA suite of programs from WS Atkins, in the UK, and partly upon a series of model tests undertaken in the UK in the late 1970s. The model tests included a barge with similar characteristics to that of the *Irish Sea Pioneer*.

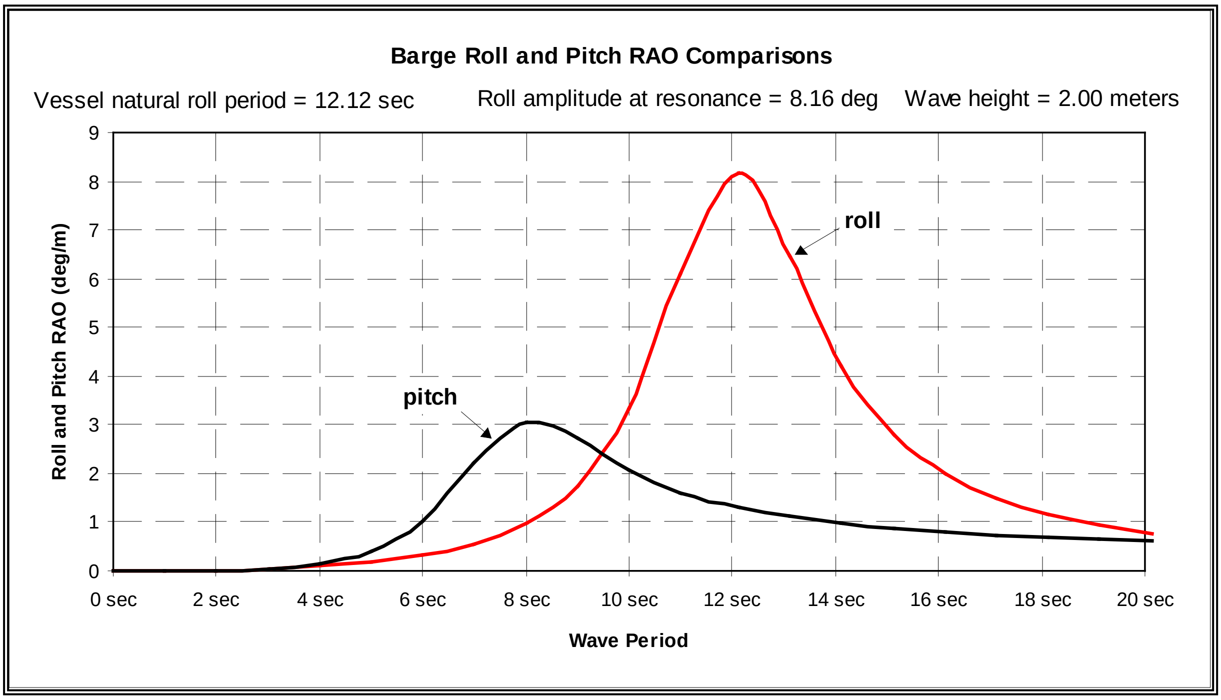

FIGURE 8 - Roll and Pitch RAOs in 2.0 Meter Wave Height

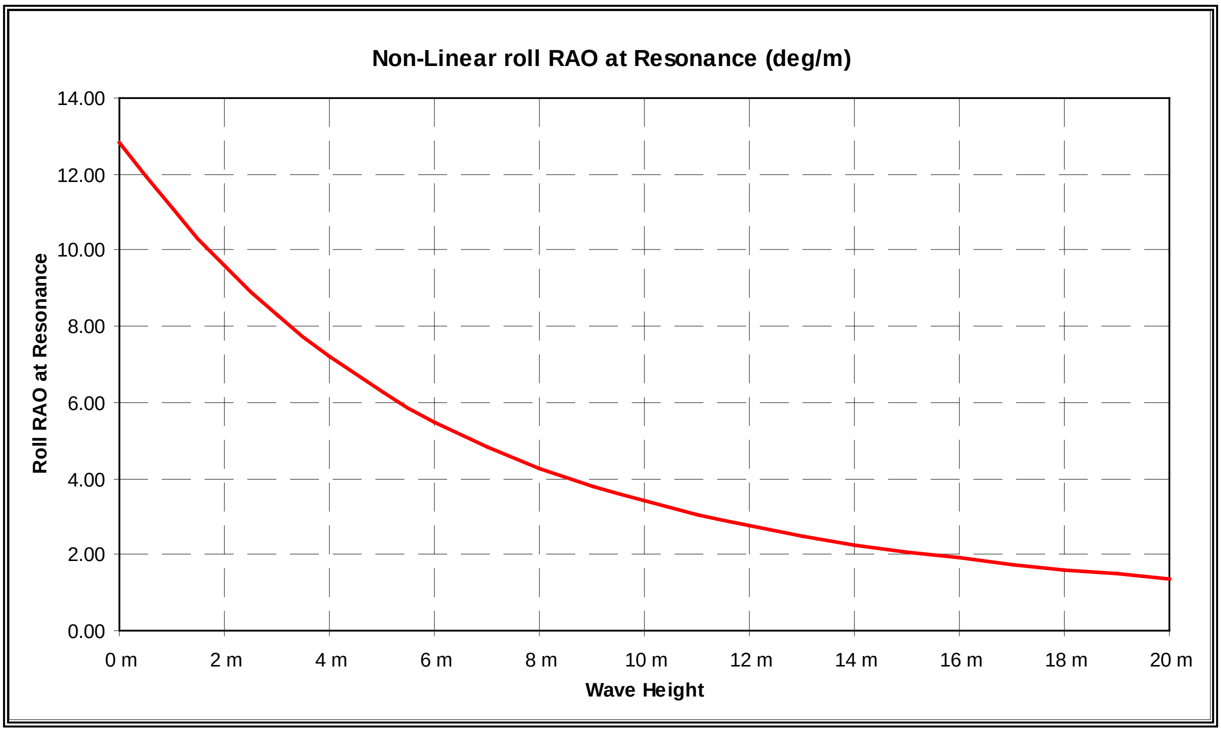

FIGURE 9 - Roll Non-Linearity With Wave Height

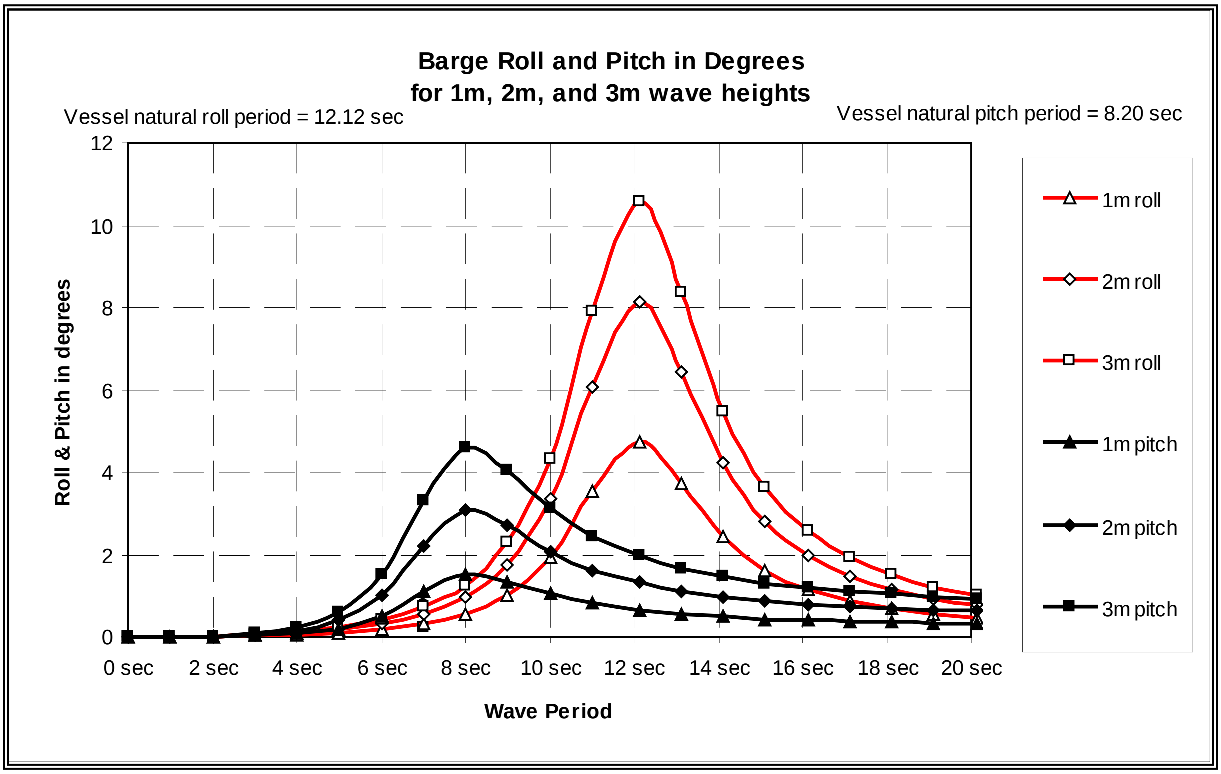

FIGURE 10 - Angular Motion Response Predictions in Relatively Mild Seas

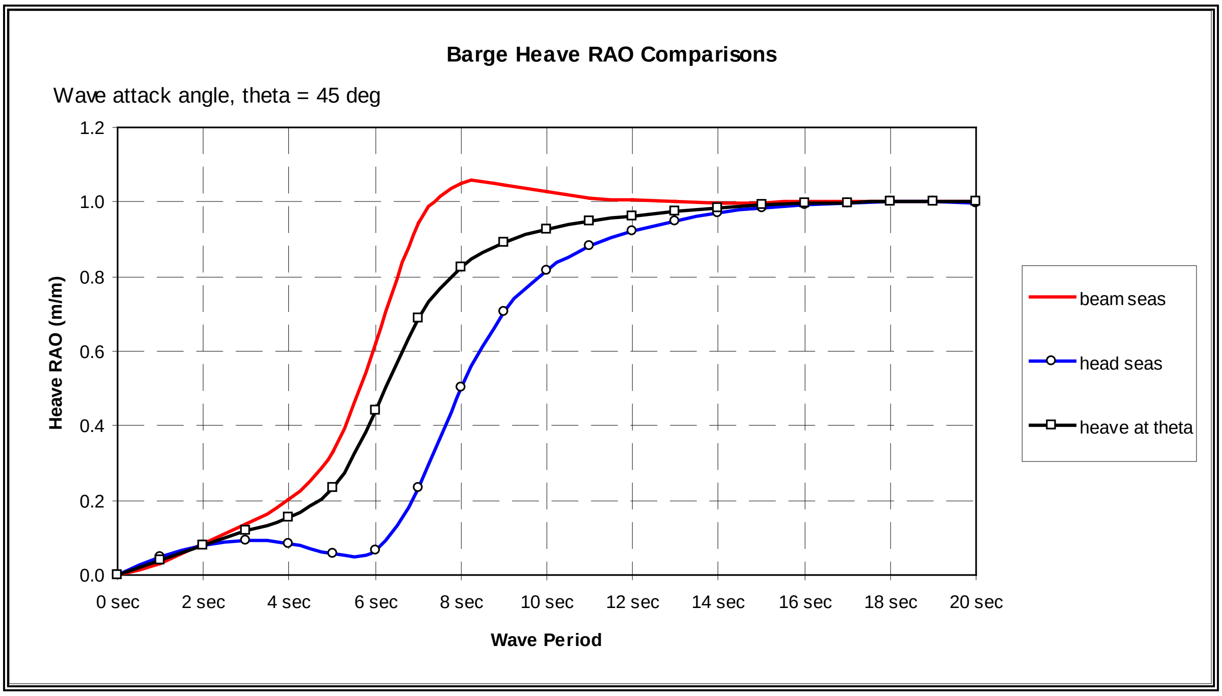

FIGURE 11 - Heave RAOs

Figure 11 shows the heave RAOs for head seas, quartering seas, and beam seas. The heave (and pitch) motions can be described by RAOs which are linear with wave height.

LEG IMPACT WITH SEA BED WHEN MOVING LOCATION

The force resulting from vessel motion when a can first contacts the seabed is a function of the impact velocity, the loaded condition of the vessel, the leg length extended, the hardness of the seabed, and the stiffness of the leg and hull structure, including the pinion stiffnesses. The direction of travel of the spud can during the impact will influence the magnitude of horizontal force, resulting in leg bending, and the magnitude of the vertical force, resulting in leg compression and pinion axial loading. Note that the pinions will also share part of the bending moment occurring as a consequence of horizontal forces generated at the cans.

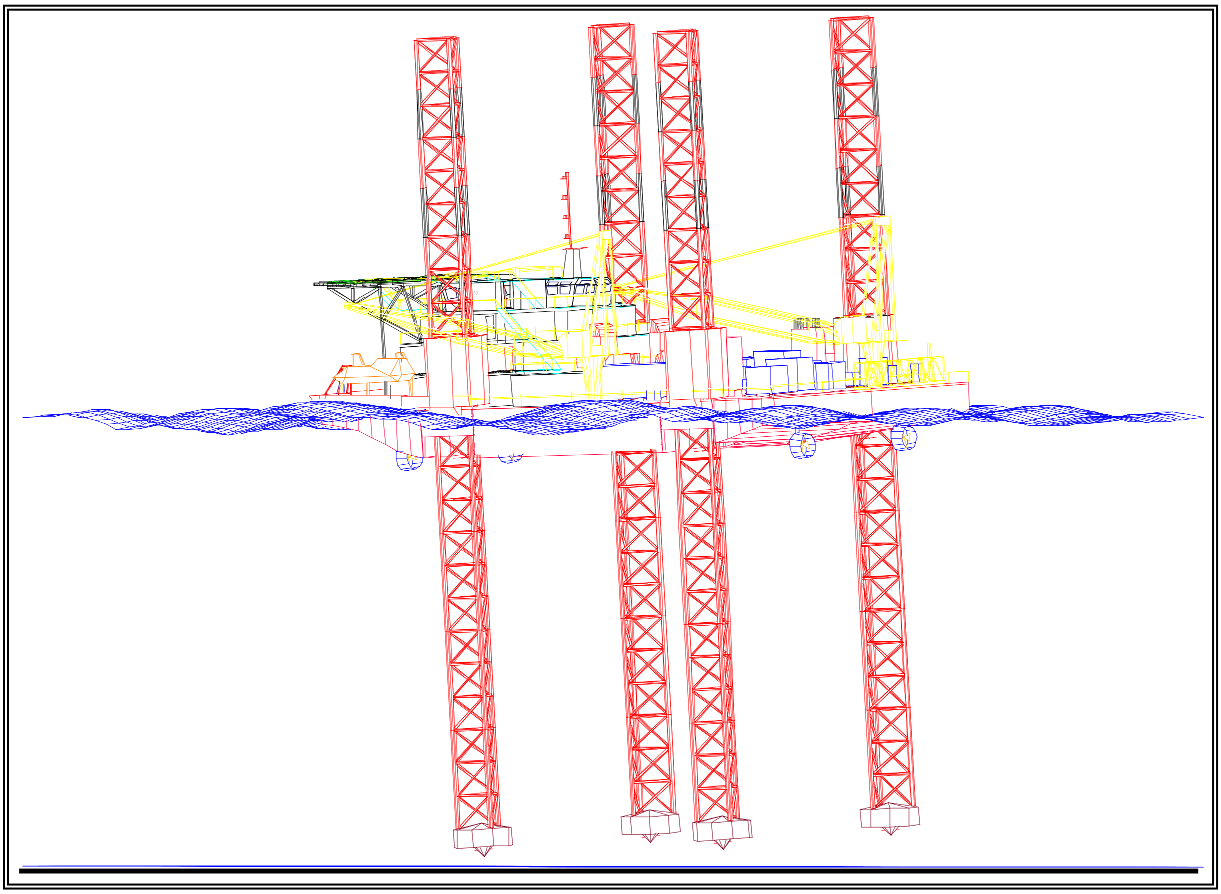

FIGURE 12 - Going Onto Location in Relatively Rough Seas

A target maximum allowable leg impact force is set to 3,800 kips (17 MN) vertical component, with varying amounts of horizontal component being allowed, depending upon the length of leg extended. The target allowable vertical force is 75% of the maximum design vertical load for the pinions on a single leg (5000 kips = 22 MN). The method for computing the leg impact force (described in Reference 2) results in the following expressions for the horizontal and vertical forces at the spud cans:

$$P_{Hsand} := \frac{2\cdot\pi}{T_{roll}}\cdot\theta\cdot\sqrt{\frac{I_m\cdot k_H}{1+\frac{k_{legVertSand}}{k_H}\cdot\left(\frac{d}{H_{lower}}\right)^2}}$$

$$P_{Vsand} := \frac{2 \cdot \pi}{T_{roll}} \cdot \theta \cdot \sqrt{\frac{I_m \cdot k_{legVertSand}}{1 + \frac{k_H}{k_{legVertSand}} \cdot \left[\frac{d}{(H_{lower})}\right]^2}}$$

Where:

P_{Hsand}	=	maximum horizontal force at can on impact
P_{Vsand}	=	maximum vertical force at can on impact
T_{roll}	=	period of angular motion
θ	=	amplitude of angular motion
I_m	=	structural and hydrodynamic added inertia of OSV
k_H	=	overall horizontal stiffness of leg/sea bed system.
$k_{legVertSand}$	=	overall vertical stiffness of leg/hull/jacking system/sea bed

The calculated impact force makes the assumptions that:

- Only one leg touches the bottom
- The lower end of the leg is stopped immediately when the leg touches the bottom
- The bottom stiffness is accounted for in the leg stiffness
- The rotational energy of the OSV is absorbed by the leg, hull, jacking system, and soil

The impact force is a direct function of the impact velocity in the above method. As the leg length extended below the hull becomes shorter, the allowable impact velocities tend to become larger, for a constant allowable impact vertical force. However, the horizontal forces tend also to increase as leg length is decreased, as a consequence of increased horizontal stiffness. In 20 feet (6.1 meter) water depth the allowable vertical impact force has been reduced to 50% of the pinion design maximum values. This is partly to protect the thrusters, partly to reduce horizontal forces, and partly because shallow water motion responses are less reliably predicted than those in deeper water. Nevertheless, the allowable shallow water motions that result from this reduction are slightly greater than those allowable in 50 feet (15.2 meter) water depth, where the 75% design pinion force is the limiting value. Figure 13, below, shows the allowable motions for going onto a new location.

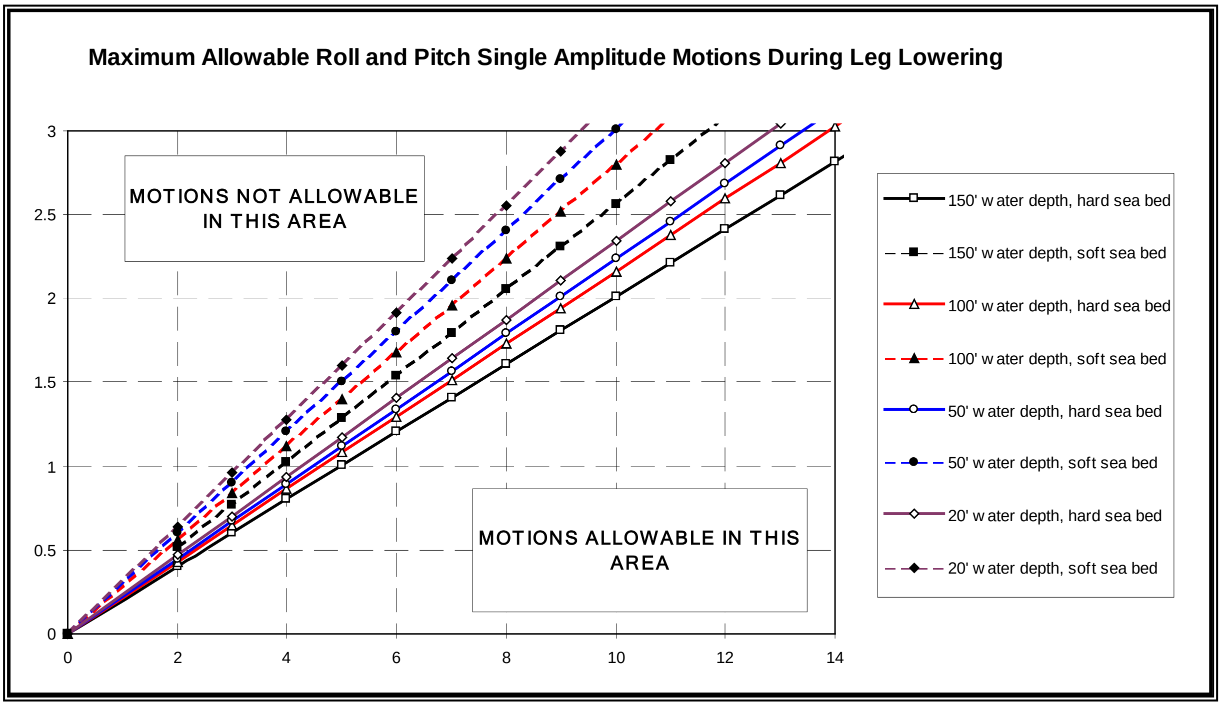

FIGURE 13 - Maximum Allowable Motions for Going Onto a New Location

Table 2 summarizes maximum resulting decelerations and velocities involved.

Table 2 - Maximum Can Decelerations and Velocities (corresponding to motions in Figure 13)

Allowable Deceleration Values			Allowable Velocity Values		
Water Depth	Hard sea bed	Soft sea bed	Water Depth	Hard sea bed	Soft sea bed
150 ft	0.089 g	0.089 g	150 ft	0.87 ft/sec	1.11 ft/sec
100 ft	0.110 g	0.110 g	100 ft	0.94 ft/sec	1.21 ft/sec
50 ft	0.118 g	0.118 g	50 ft	0.97 ft/sec	1.31 ft/sec
20 ft	0.078 g	0.066 g	20 ft	1.02 ft/sec	1.39 ft/sec

Figure 14 shows the predicted maximum can vertical velocities caused by combined roll and pitch motions in 1 meter high regular long-crested waves with varying angles of attack (90º corresponds to beam waves). Figure 15 shows the same data for 2 meter wave heights. When wave spreading is also considered the maximum motions are reduced for a given wave height. The effect of wave spreading is considered to increase the effective wave height that would cause the angular motion predictions in Figures 14 and 15, to around 1.5 meters and 2.75 meters, respectively. However, this probable increase has conservatively been neglected in calculating the maximum likely sea states for safe jacking operations.

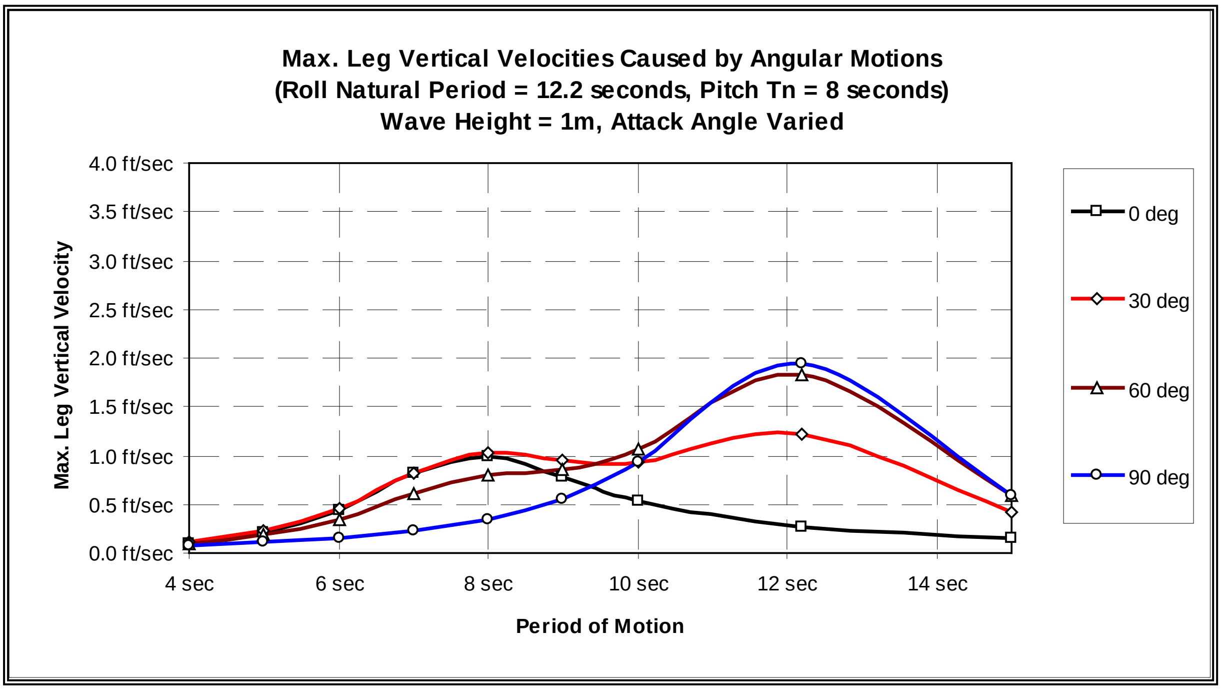

FIGURE 14 - Maximum Can Vertical Velocities in 1 meter Waves

The cut-off maximum allowable velocities are seen from Table 2 to be around 1.0 feet per second (0.3 m/s) for water depths in the range 20 to 100 feet (6 m to 30 m), and slightly less than this in water depths from 100 up to 150 feet (30 m to 46 m). From Figures 14 and 15, the OSV is predicted to have allowable angular motions for location moves in maximum waves of up to 2.0 meters in height at wave periods less than 6.0 seconds for any direction of wave attack. For beam seas, the OSV's motions, in the full load condition, are predicted to be acceptable for waves of up to 2.0 meters in height at wave periods of up to 9 seconds. For waves with longer periods than 10 seconds, the OSV must be oriented to head into the waves if maximum wave heights are up to 2.0 meters. In maximum wave heights of 1.0 meters or less, the OSV can safely go onto location with any wave attack direction in wave periods of up to 10 seconds. In longer period waves of maximum height 1.0 meters, she should be headed into the waves (or have the waves on the stern) in order to have allowable motions.

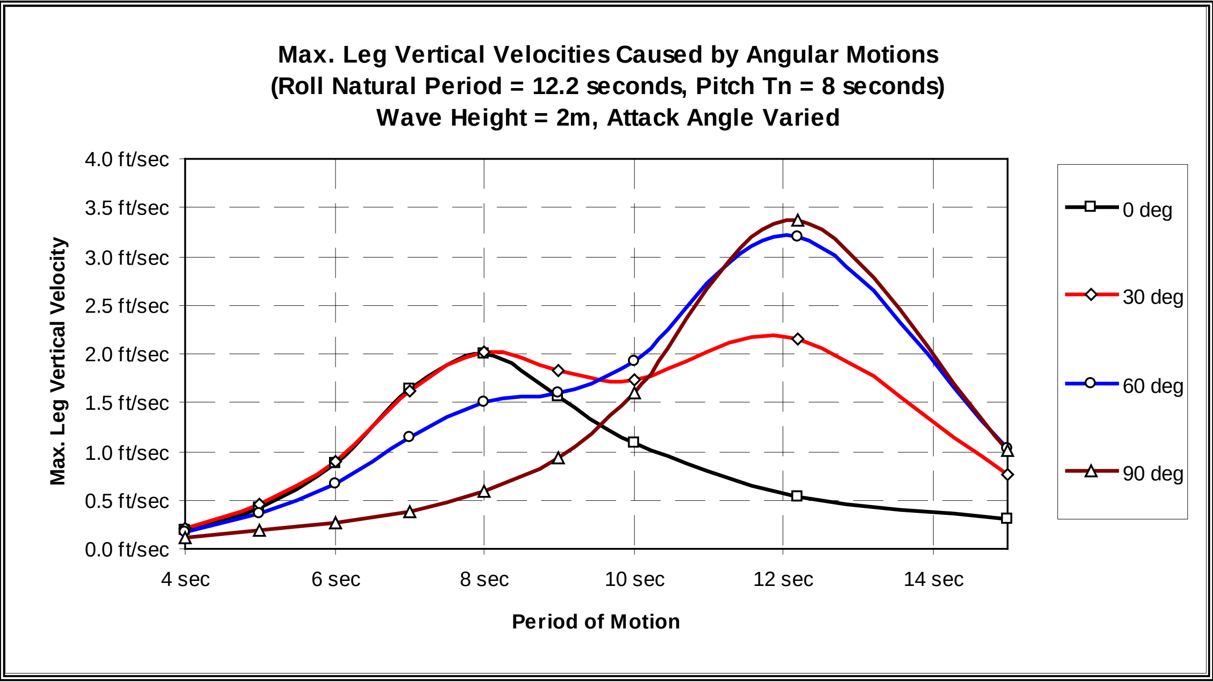

FIGURE 15 - Maximum Can Vertical Velocities in 2 meter Waves

Table 3 shows the Beaufort scale of wind and associated sea conditions. An estimate of the wave period is given, based upon wave steepness.

Table 3 - Beaufort Scale of Wind and Associated Sea Condition

Beaufort number	min. wind speed (knots)	max. wind speed (knots)	prob. ht. of waves (m)	prob. max. ht. of waves (m)	min prob period of prob max wave (steepness=1/14)	max prob period of max wave (steepness=1/24)	Ref. 3	Sea Conditions (from Reference 1)
0	0	1	0.0	0.0	0.00 sec	0.00 sec	calm	Flat calm, mirror smooth
1	1	3	0.1	0.1	0.95 sec	1.24 sec	light air	Small wavelets, no crests
2	4	6	0.2	0.3	1.64 sec	2.15 sec	light breeze	Small wavelets, crests glassy but do not break
3	7	10	0.6	1.0	3.00 sec	3.92 sec	gentle breeze	Small wavelets, crests begin to break
4	11	16	1.0	1.5	3.67 sec	4.80 sec	moderate breeze	Small waves, becoming longer, crests break frequently
5	17	21	2.0	2.5	4.74 sec	6.20 sec	fresh breeze	Moderate waves, longer, breaking crests
6	22	27	3.0	4.0	5.99 sec	7.84 sec	strong breeze	Large waves forming, crests break more frequently
7	28	33	4.0	5.5	7.03 sec	9.20 sec	near gale	Large waves, streaky foam
8	34	40	5.5	7.5	8.20 sec	10.74 sec	gale	High waves of increasing length, crests form spindrift
9	41	47	7.0	10.0	9.47 sec	12.40 sec	strong gale	High waves, dense streaks of foam, crests roll over
10	48	55	9.0	12.5	10.59 sec	13.87 sec	storm	Very high waves, long overhanging crests. Surface of sea white with foam
11	56	66	11.5	16.0	11.98 sec	15.69 sec	violent storm	Exceptionally high waves, sea completely covered with foam
12	67	80	14.0	20.0	13.40 sec	17.54 sec	hurricane	The air filled with spray and visibility seriously affected

Ref. 1 Oxford Companion to Ships and the Sea, Kemp, 1976
Ref. 2 Oceanography and Seamanship, Van Dorn, 1974
Ref. 3 Times Atlas and Encyclopedia of the Oceans, Couper, 1989
Ref. 4 Piloting, Seamanship and Small Boat Handling, Chapman, 1981

The limiting conditions for the OSV to move onto location are approximately characterized by Beaufort Force 5. From the environmental data for the Liverpool Bay area, it has been determined that there are around 20 days in the year where waves will limit location moves.

Figure 16 illustrates the shallow water conditions for going onto location at lowest astronomic tide (LAT) at the Lennox Platform.

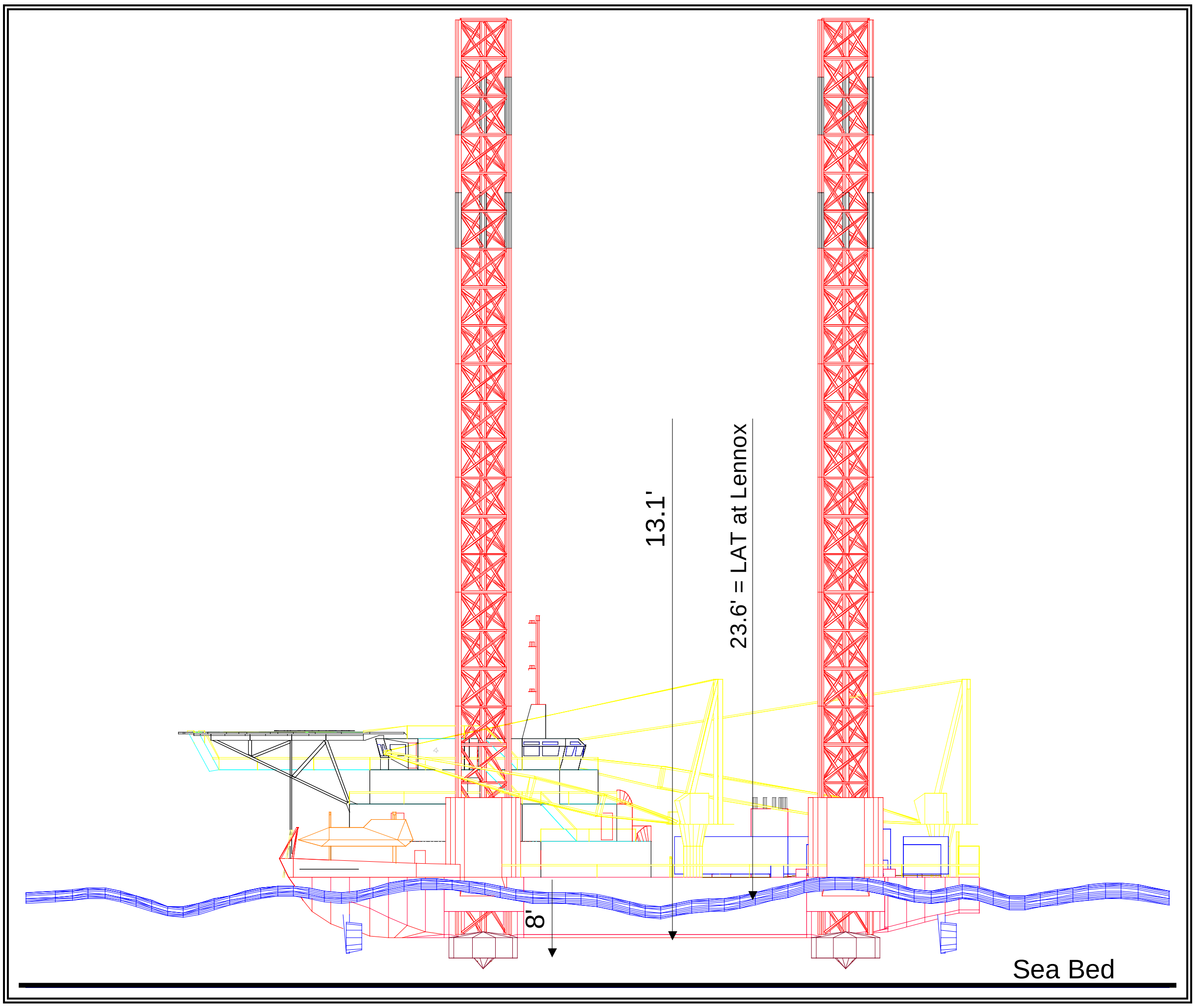

FIGURE 16 - OSV Going onto Location in Limiting Sea Conditions at LAT at Lennox

PRELOADING

Once the hull is level, and the loads on the legs have been equalized, the preloading operation may commence. One forward leg and the opposite leg, across the diagonal at the stern, will be preloaded first. In order to achieve this, the hull will be gradually lowered down the two legs which are not being preloaded. The hull sag will be about three inches on each corner as this occurs. One forward leg not being preloaded will carry some load if the forward legs are more heavily loaded than the stern legs. However, in this condition one stern leg will end up carrying no hull load at all. It is important that the jacking system is not used to try and raise the unloaded stern leg. At full preload conditions, the stern leg not being preloaded will have zero torque on all pinions.

RATCHET CHOCKS

Owing to increased loads associated with increased hull weight on the elevating pinions in extreme storm conditions, it was elected to design and build devices called "ratchet chocks" for *The Irish Sea Pioneer*. These chocks are shown disengaged in Figure 17. The intent is to set the full weight of the hull upon the ratchet chocks when the hull is elevated. This is done by engaging the ratchet chock arms with the rack teeth on the leg chords, and then slowly and carefully lowering the hull on each leg onto the ratchet chock devices, until there is zero torque left on the elevating pinions. The brakes on the motors are then engaged. Hull self-weight remains on the ratchet chocks while the hull is elevated and storm (vertical) loads are taken by the main elevating pinions.

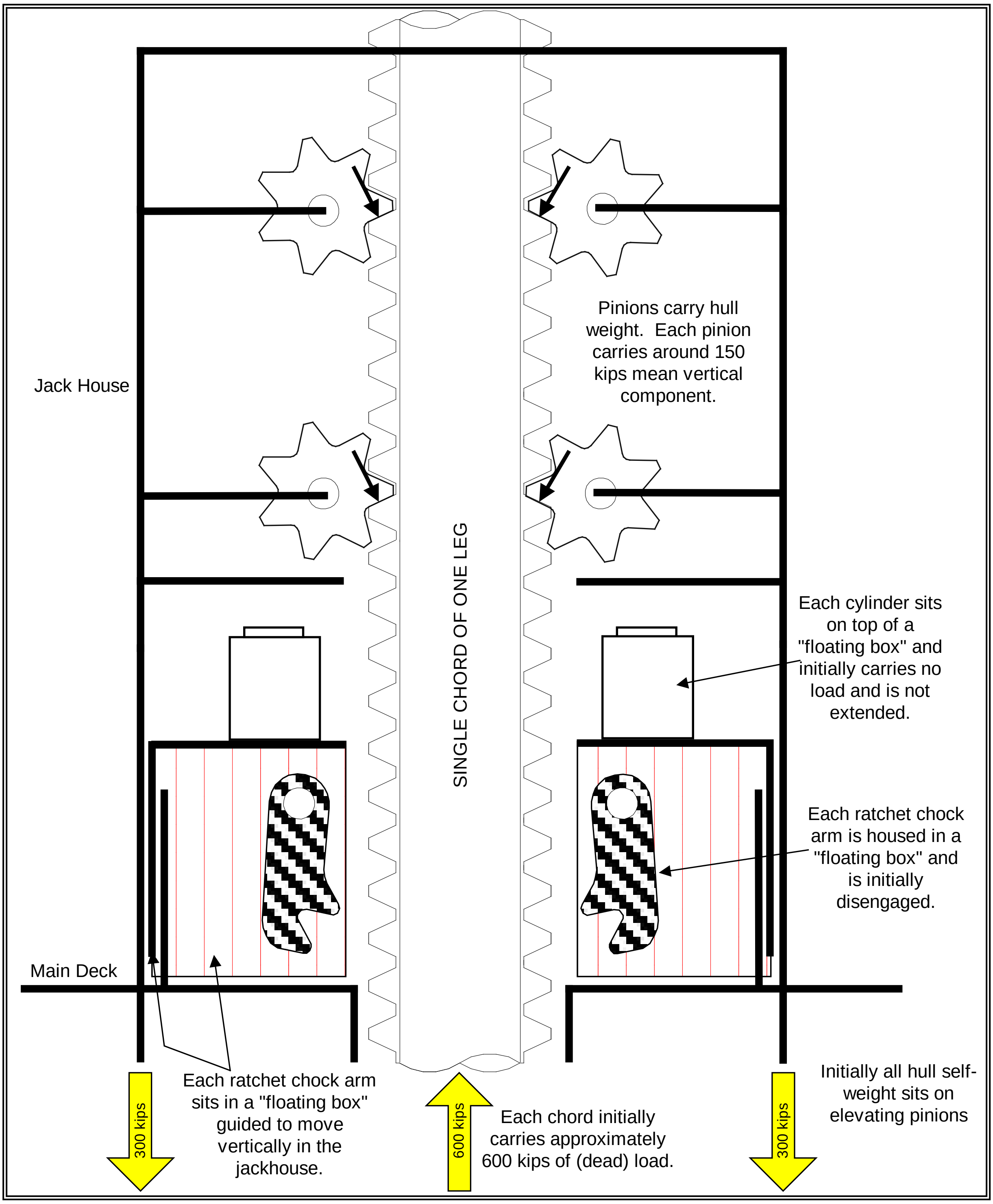

FIGURE 17 - Schematic View of Elevating System, Ratchet Chocks Disengaged

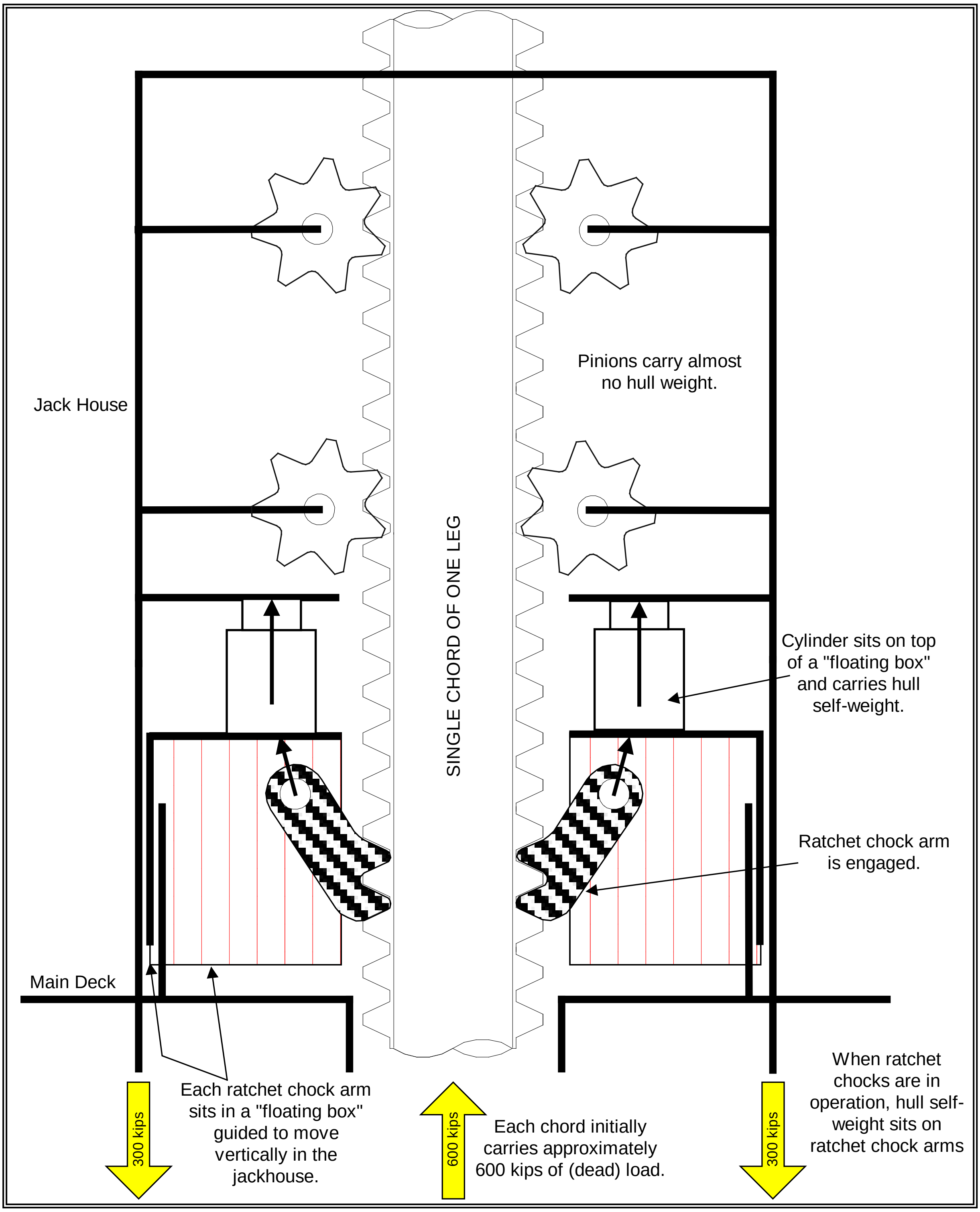

FIGURE 18 - Schematic View of Elevating System, Ratchet Chocks Engaged

Figure 18 shows a schematic view of the elevating system with the ratchet chocks engaged. The load path is seen to go through the chord rack, through the ratchet chock arms, via the floating boxes and into the hydraulic cylinders, finally reaching the jackhouse structural steel. As a consequence of the 25º pitch angle on the rack teeth, the reaction vector between the rack teeth and the chock teeth passes between the chock arm pins and the rack. Consequently, a moment is induced tending to engage the chock arms tighter as more vertical load is imposed. The chock arms, like the elevating pinions, are 6 in (150 mm) wide and are centralized on the 4 in (100 mm) wide rack. The rack, the pinions, and the ratchet chock arms are all made from steel with 100 ksi (690 MPa) yield strength.

Note that the selected geometry results in an inability of the ratchet chocks to apply an upwards load to the rack without their being a force tending to kick the arms out. Consequently, there is the possibility of raising the hull slightly in order to disengage the arms should they become tightly jammed.

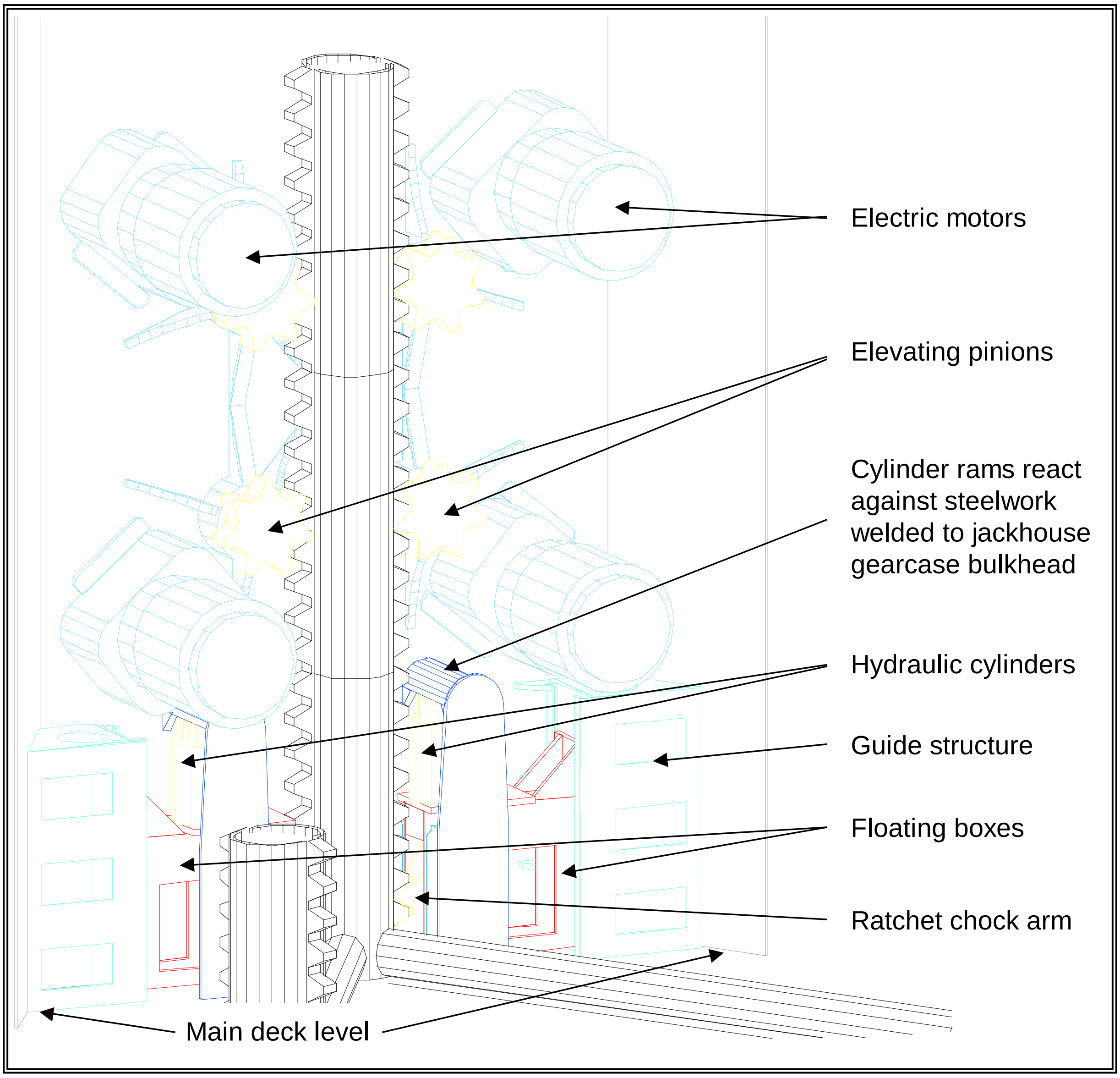

FIGURE 19 - Cut-Away View of Jackhouse & Leg, Ratchet Chocks Engaged

Figure 19 shows a view of the ratchet chocks inside the jackhouse. They are located just above main deck level. Details including the air cylinders, hydraulic lines, jack motor air cooling ducts, etc, have been omitted for clarity.

A rubber block concept was originally proposed instead of the hydraulic cylinders. However, the rubber stiffness properties required could not be achieved in the time available. It should be noted that various other means of reducing pinion loads during storms were investigated before ratchet chocks were selected. These included adding additional idle pinions with various means of braking, including full gearboxes and conventional brakes. A more promising approach was to use additional opposing pairs of final drive pinions with arms which could be engaged on splines on the drive shafts when needed. These arms would be rotated on the drive shafts, when engaged and would push together against large rubber blocks, A concept of limited brake slippage on the main elevating pinions was also pursued but this met with resistance from ABS.

Engagement of Ratchet Chocks

When the hull is in position for elevated operations, the ratchet chocks will be engaged. In order to do this, air cylinders are provided which push the lower part of the arms into the rack teeth. Before commencing the engagement process, a visual check must be made to ensure that no chock teeth are just a little too high. If this was the case, a partial engagement could result. The hull is lowered slowly on the elevating pinions, and the chock arms engage into the rack teeth with the aid of the air cylinders. Since each leg may be at a different rack phase angle, different amounts of hull lowering occur at each leg, before all ratchet chocks are engaged. Furthermore, because of construction tolerances, there are rack phase differences between the ratchet chock engagement positions on each leg.

From the point at which the hull is perfectly level, the ratchet chock engagement process proceeds to the point at which all chocks are engaged, while the hydraulic cylinders are fully retracted. The cylinders have a 7 in (175 mm) stroke. The gaps above the cylinder rams are reduced during the arm engagement process by a maximum of 6.3 in (160 mm) as the floating boxes are raised as each ratchet chock arm is engaged. The 6.3 in represents the pitch of the rack and represents the worst case scenario where, just before the engagement process commences, one chord is 6.3 in out of phase with another chord.

A large number of Monte Carlo simulations have been performed accounting for known construction tolerances and the completely random phasing possible for individual legs. These simulations have considered that it is necessary to make each of the forward pair of leg loads equal and the aft pair of leg loads equal. However, depending upon vessel trim when afloat and ballast requirements when elevated, the forward pair of legs typically carries a heavier load than the aft pair of legs. Consequently, the hull is leveled on the forward pair of legs independently of being leveled on the aft pair of legs. When this is accounted for, the minimum gap above the hydraulic cylinders (before they are actuated) is typically 3 in (75 mm).

After all chocks are engaged, the hydraulic cylinders are activated. The forward pair of legs have a separately powered hydraulic circuit from the aft pair of legs. Each leg has its own hydraulic controls. The load on each pair of legs is predetermined from knowledge of the pinion elevating loads. The hydraulic pressure is set such that all hull weight is carried by the hydraulic rams. Once the rams have been activated, the pinion loads are again checked using the Siemens control system and the elevating equipment. The pinion loads are all now close to zero, The brakes on the pinions are subsequently set, and the elevating system is powered down.

Ratchet Chock Behaviour During Storms

The bending moments in the legs induced by storms are partly reacted by the elevating pinions, partly reacted by the ratchet chocks, and partly reacted by the upper and lower guides. Depending upon the load condition, around 40% of the moment at any leg, in the absence of the ratchet chocks, is taken out by the pinions in a vertical couple. This results in higher pinion loads, for example, on an inboard chord, and lower loads on outboard chords. As a consequence of the load redistribution, there are deflections in the vertical direction of the pinions and the racks upon which they react. The hydraulic system on each individual leg is designed to equilibrate the force applied by each ram. This would effectively result in no moment coming from the ratchet chock system if the system were perfect. However, as a consequence of the dynamic nature of the loads and as a consequence of friction in the system, there is an inevitable small contribution to the pinion moments from the ratchet chocks.

As well as causing bending moments in the legs, storm loads cause variations in the mean axial loads carried by each leg. Mean axial leg load changes result in additional mean vertical forces being absorbed by both the ratchet chocks and the elevating pinions. The amount of load absorbed by each system is inversely proportional to the vertical stiffness of each system. The hydraulic system has a vertical stiffness which is dependent upon the hydraulic accumulator in the system. This stiffness has been set to nominally 400 kips per inch (70 kN/mm) per ram. This compares to a stiffness of around 2800 kips per inch (490 kN/mm) for the pair of corresponding pinions and gear trains. Consequently, around 90% of the increased (or decreased) axial leg load changes are absorbed by the elevating pinions. The net result is that the self-weight of the hull is carried by the ratchet chocks and storm loads are carried by the elevating pinions.

Note that if differential leg settlement occurs during storms, or at any other time, the elevated weight distribution shifts from a being equally distributed on each leg to being carried largely on a single pair of legs, across one diagonal. It was previously noted that around 3 in (75 mm) of hull sag would occur at diagonally opposed corners of the rig during preload. A differential settlement of 6 in (150 mm) at one leg, compared to the other three, would result in similar hull support conditions. In effect, one pair of legs across a diagonal would become unloaded, and the other pair would be supporting the hull weight. If such conditions occur in service, they will be detected by a leg load monitoring system, and corrective action will be taken. The versatility of the variable speed jacking system, coupled with the hydraulic system, permits relatively straightforward hull level adjustments to be made, even including the disengagement of the ratchet chocks and their re-engagement one or more rack pitch lengths away.

STRUCTURAL DESIGN AND ANALYSIS

Hull Design

The OSV has a 180 ft (54.9 m) long barge shaped hull with flange plate construction, satisfying ABS requirements for ocean going barges, as well as satisfying ABS Mobile Offshore Drilling Units (MODU) Rules. The main scantlings are all sized to satisfy ABS rules, at a minimum. In places, extra heavy horizontal girders have been added at the mid-height of bulkheads in order to reduce the possibility of elastic buckling during preload. Additionally, half frames have been added at the bow to minimize the possibility of local damage in heavy seas. Table 4 summarizes the main plate thicknesses (36 ksi, 241 MPa material).

Table 4 - Hull Plate Thicknesses

Item	Property
Side Shell Plate	1/2" Thick
Bottom Shell Plate Amidship	3/8" Thick
Bottom Plate at Ends	7/16" Thick
Deck Plate Amidship	5/16" Thick
Deck Plate at ends	5/16" Thick
Longitudinal Bulkhead Plating @ 14' off centerline(top Half)	5/16" Thick
Longitudinal Bulkhead Plating @ 14' off centerline(bottom Half)	3/8" Thick
Longitudinal Bulkhead Plating @ 35' off centerline(top Half)	5/16" Thick
Longitudinal Bulkhead Plating @ 35' off centerline(bottom Half)	3/8" Thick
Longitudinal Bulkhead Plating @ 35' off centerline(between Legs)	5/8" Thick
Transverse Bulkhead between legs (frame 9,11 ,28,30)	5/8" Thick

The jackhouses are constructed entirely from steel plate most of which has a yield strength of 70 ksi (469 MPa). There are three vertical bulkheads extending from the base of the hull to the top of the jackhouses which carry the gear cases and jacking equipment. Three other vertical bulkheads link the gear case bulkheads to form a hexagon in plan view. The leg guides are contained within the jackhouse, with their top and bottom edges being flush with the upper and lower steelwork.

A strong box of steel made up from (36 ksi yield) 3/4 in (19 mm) thick top plate, 3/4 in thick bottom plate, and 5/8 in (16 mm) thick side plates runs between each jackhouse as shown in Figure 20. The thickness of the plates making up this box and the associated stiffening is governed by structural requirements for the OSV when elevated. The hull width is limited to 92 ft (28 m) by a requirement to transit through limited width waterways. The transverse leg spacing is made as wide as possible within the hull. The triangular legs are oriented with two chords outboard and one chord inboard, thereby placing the center of the triangle as far outboard as possible. The jackhouses are cantilevered slightly out over the sideshell.

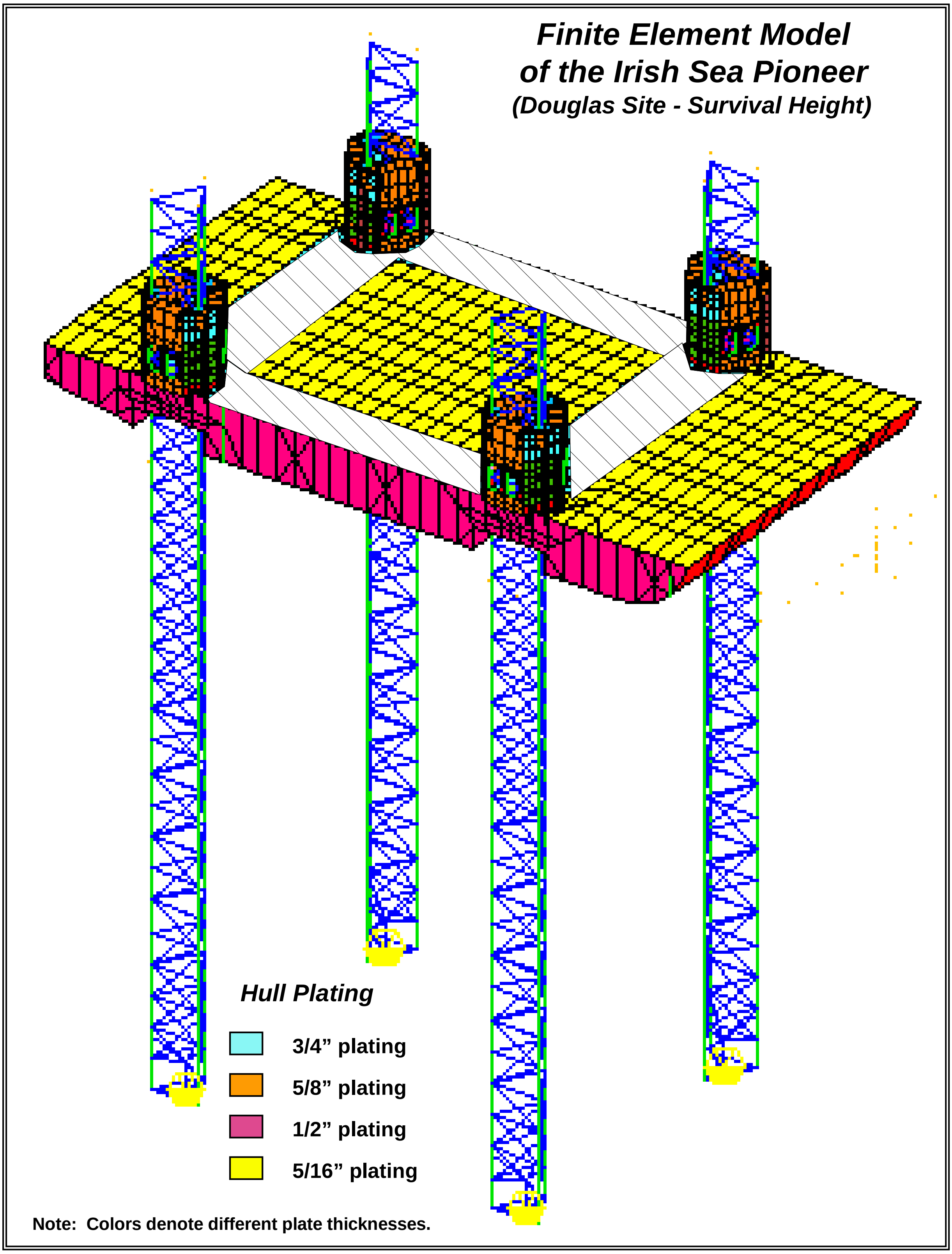

FIGURE 20 - Overview of Structural Model

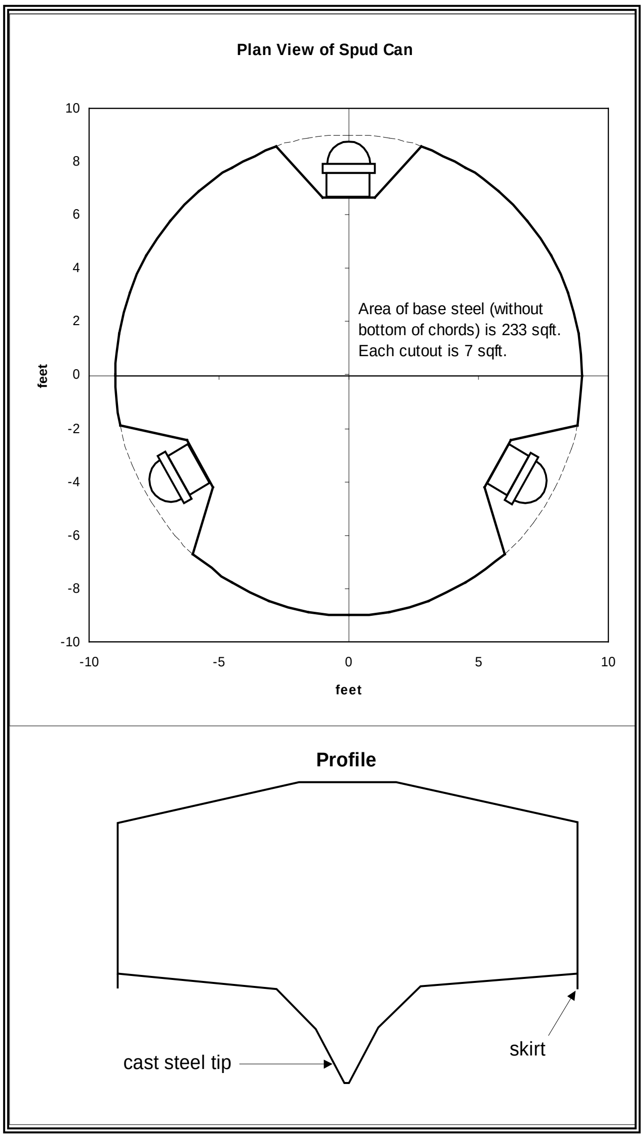

FIGURE 21 - Spudcan Schematic Model

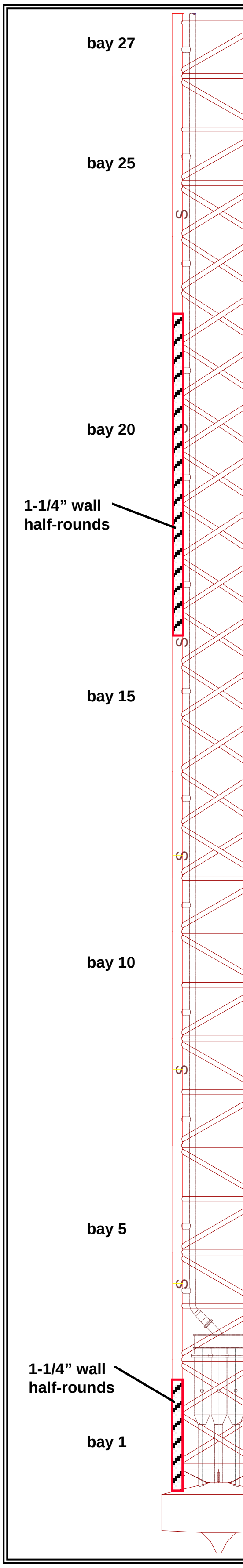

FIGURE 22 - Leg

Spudcans

The spudcans are designed specifically for the relatively hard sea bed conditions typical of the Liverpool Bay sites. Since the sea bed is sandy and the water depth is relatively shallow (varying from 7.3 m to 39 m) scour is expected beneath the cans, particularly during storms. In order to minimize the amount of scour, the spudcans are designed to penetrate a minimum of 1 m into the sea bed and to minimize flow disturbance. They also have skirts 150 mm long. The average bearing pressure under the cans at preload, assuming the full can base area is in contact with the soil, is relatively high at 16.3 kips/ft^2 (0.78 MPa). A maximum can penetration of 3 m is anticipated.

The main body of the can is cylindrical (18 ft = 5.5 m diameter) with radial bulkheads and a central vertical tube. The can steel is low strength (36 ksi = 241 MPa) and connection of the high strength chords is external as shown schematically in Figure 21. The can has a cast steel central tip provided by LeTourneau. The legs and spudcans were built by LeTourneau with the final spudcan detailed design also performed by LeTourneau based on the above design philosophy provided by Stewart Technology Associates. Can penetration predictions were made by MARSCO based upon geotechnical reports provided by Fugro-McClelland.

Each spud can weighs approximately 72 kips (33 tonnef) and is designed to be capable of taking full storm load on a 120º sector or on the tip. The can is also capable of carrying a large bending moment into the leg.

Leg Design and Guide System

The four legs of the vessel are 242 ft (73.8 m) long from can tip to leg top. Each leg weighs 420 kips (190 tonnef). The legs are triangular in plan view and have a combination of Z-braces and X-braces as shown in Figure 22. The less expensive Z-braces are used in areas where leg reaction forces with the hull are relatively low. The X-brace system is used in areas where the largest reaction forces with the hull occur. All braces are tubular, 9 inches (229 mm) outer diameter (OD) and have 3/8 in (9.5 mm) wall thickness. The brace steel has a yield stress of 85 ksi (587 MPa). The half round pipes on the chords are cut from 18 in (457 mm) OD pipe with 1 in (25 mm) wall thickness throughout most of the leg and 1-1/4ths in (31.7 mm) wall thickness in the region of bays 19 through 21. Each leg has a total of 27 bays with a short leg extension above bay 27, referred to as bay 28.

The leg chords are split tubes with a 4 in (100 mm) thick rack extending across the center of the chord. The rack is guided around the teeth with C-shaped guide shoes as shown in Figure 23. The upper guides are 4 feet (1.2 m) high and have a 1/4 in (6 mm) gap at each side of the rack teeth and at the tips of the rack teeth. Hence each chord has its rack guided on three faces on both sides of the chord thereby providing strong torsional restraint without inducing high contact forces at the rack teeth tips.

The lower guides are 4 feet (1.2 m) high and have a 1/4 in (6 mm) gap between the wear plates and the rack teeth tips. At the sides of the rack teeth, larger gaps of up to 1 in (25 mm) are provided. The widest gaps never close. One chord on each leg has a 5/8 in (16 mm) gap which closes under certain design conditions in extreme storms. This arrangement results in a restraint system which minimizes combined axial compressive stresses and bending stresses in the chords when the bottom of the lower guide is at the mid-height of a bay.

The leg design is governed by elevated hull storm survival conditions at the deepest water depth. The spud cans are treated as being pinned for this condition although some rotational restraint from the sea bed is anticipated. The legs are capable of surviving OSV afloat motions, in their fully raised positions, significantly in excess of the ABS minimum requirement of 15º in 10 seconds.

The butt welds between rack sections were subject to crack tip opening displacement (CTOD) tests. A CTOD value of 0.07 was achieved for the welds made by LeTourneau. Fracture mechanics indicated a fatigue life of well in excess of 20 years for these joints. Fatigue analysis of brace and chord joints, using a conventional S-N curve approach indicated fatigue lives well in excess of 20 years throughout the leg structure. Surprisingly, the joints with the lowest fatigue lives were in the upper bays of the leg where the upper guides are positioned at the Douglas and Hamilton sites.

Finite Element Modeling of the Guides and Legs

The NISA finite element system from Engineering Mechanics Research Corporation, Troy, Michigan, was used for the design and analysis of the *Irish Sea Pioneer*. The hull was modeled with thin plate elements, general purpose beam elements, pipe elements, point masses, and spar elements. The legs were modeled with general purpose beam elements and pipe elements. For dynamic analysis, point masses were added to simulate hydrodynamic added mass.

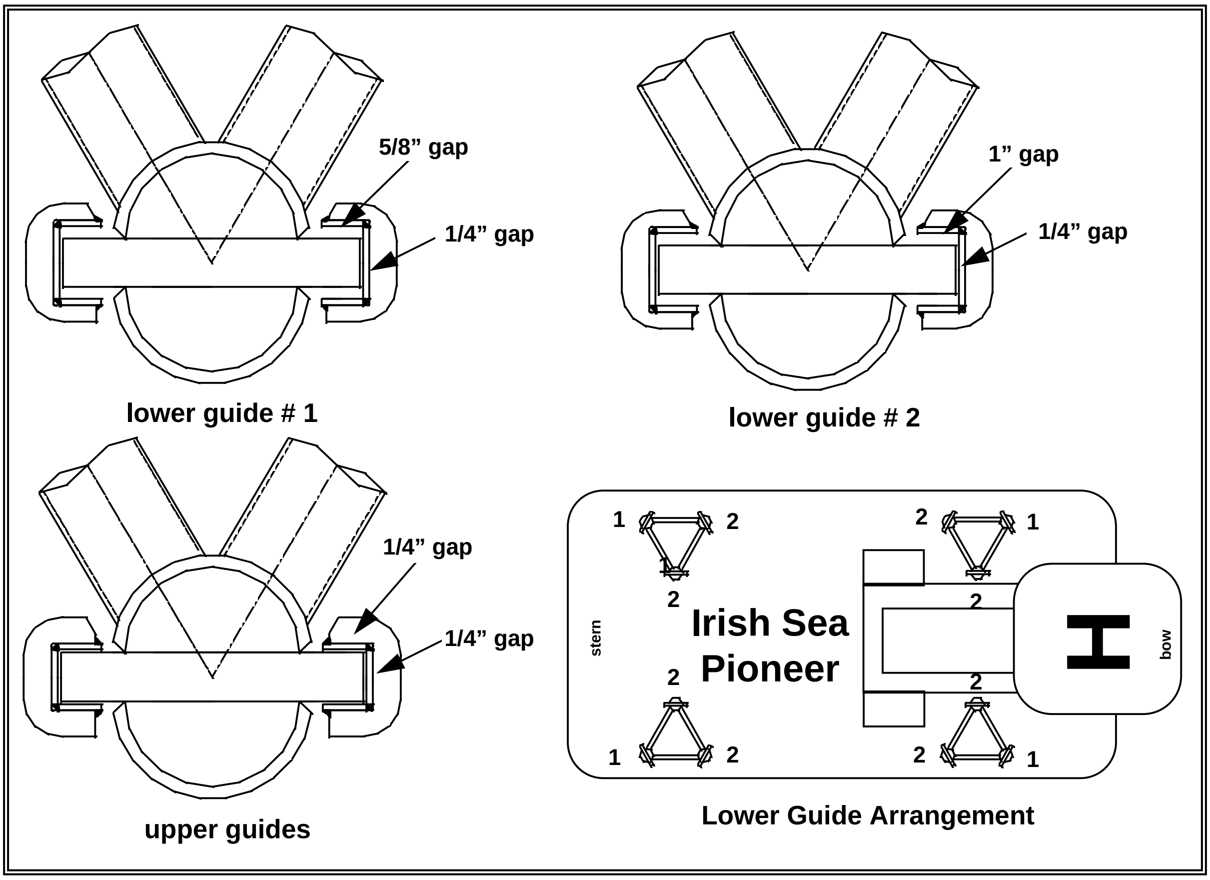

FIGURE 23 - Guide Arrangements

To model the leg/guide interaction, 3D friction/gap elements were used at each of the upper and lower guides. To model each guide structure a system of 18 gap elements was used. These gap elements were divided into three elevations (top of guide, middle of guide, and bottom of guide). At each elevation 6 gap elements and 4 rigid links tied the chord to the hull structure as shown in Figure 24. A total of 432 gap elements were used throughout the model to simulate the guide/chord interaction of the four legs.

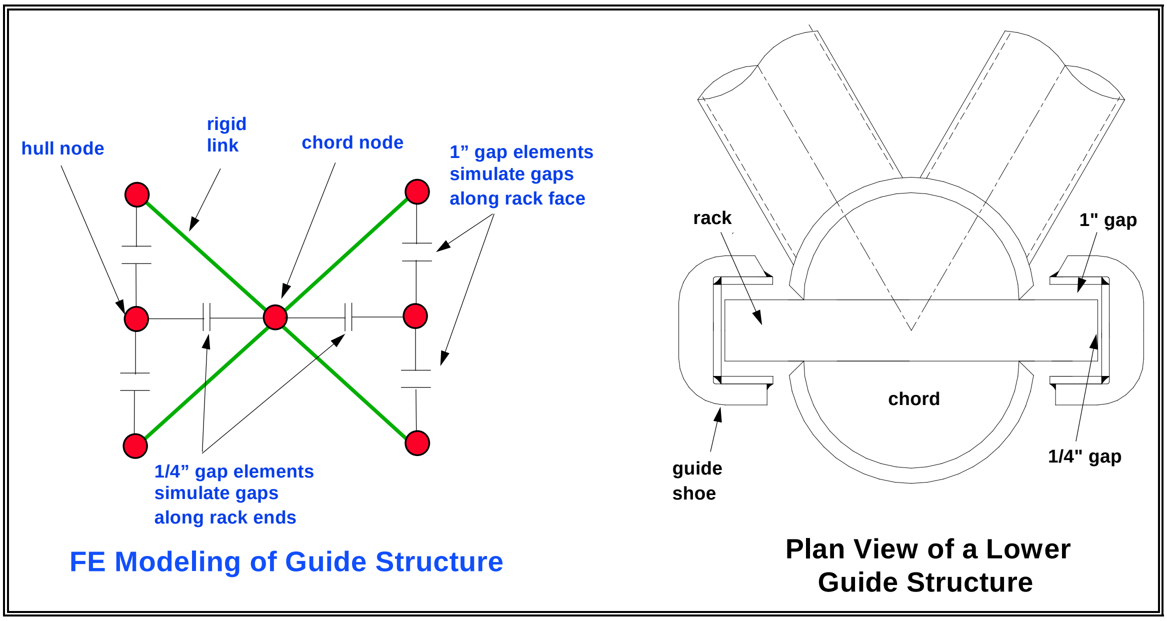

FIGURE 24 - Schematic View of Gap Elements in Guide Structure

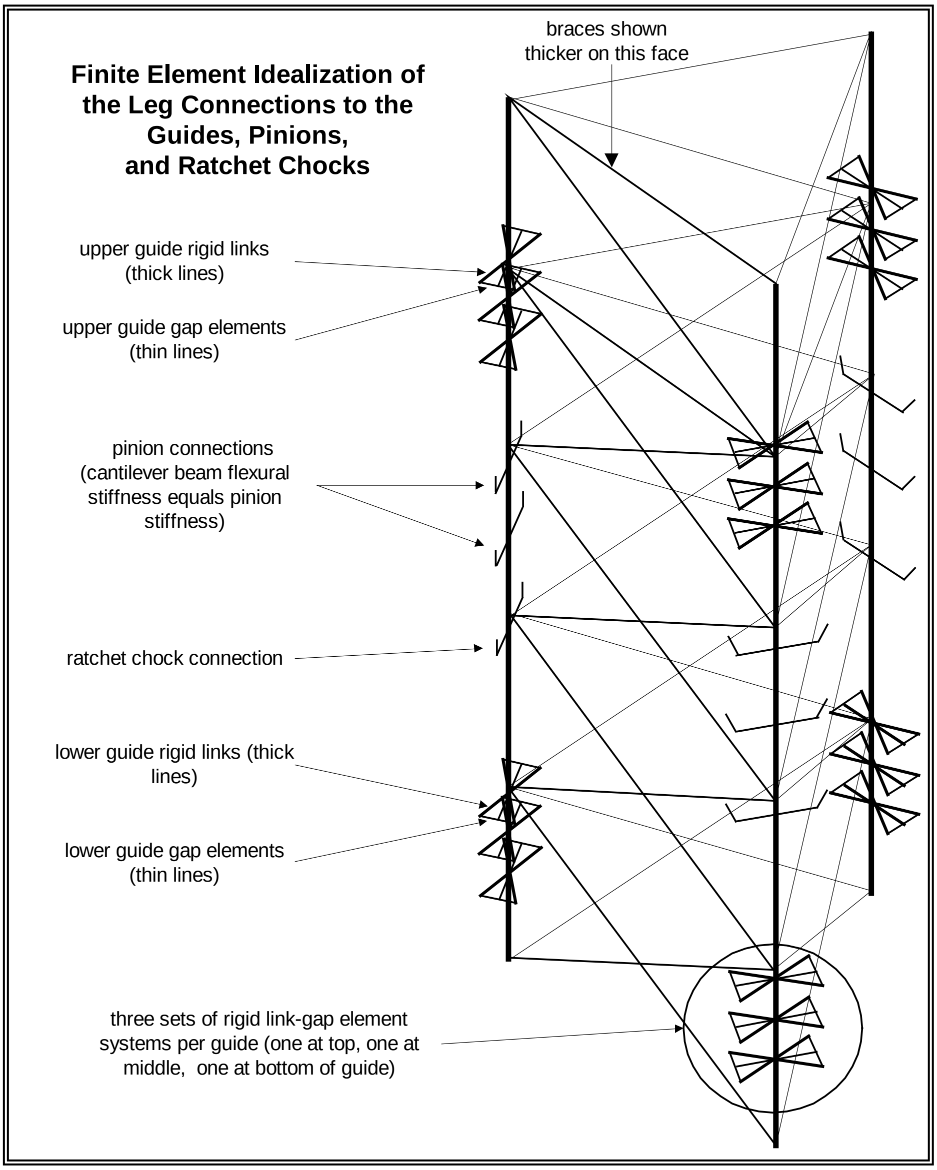

FIGURE 25 - Schematic View of Leg Connections to Hull at Jackhouse

Figure 25 shows a schematic view of the leg connections to the hull in the area of the jackhouse, with gap elements and rigid links. The vertical load transfer from the hull to the legs, via the pinions and ratchet chocks, was modeled using cantilever beams and spar elements. This permitted correct representation of the vertical stiffness of each system, as previously described. The local model also reflected the pitch angle of the rack, resulting in loads inclined from the vertical traveling back into the jackhouse steel plating.

Finite Element Modeling of the Hull

The hull is constructed of 6,360 plate elements and 1640 beam elements. All main compartments in the hull have been modeled. The majority of the beam elements are used to model the transverse frames of the vessel which are spaced every 5 feet. Stanchions and the main engine foundation girders comprise the rest of the beam elements in the hull model. The only beams present in the hull of the Irish Sea Pioneer and not present in the structural model are the longitudinal deck and bottom stiffeners. Figure 26 displays the structural model details within the hull below deck. A convention of coloring coding each plate element group to reflect the plate thickness was adopted.

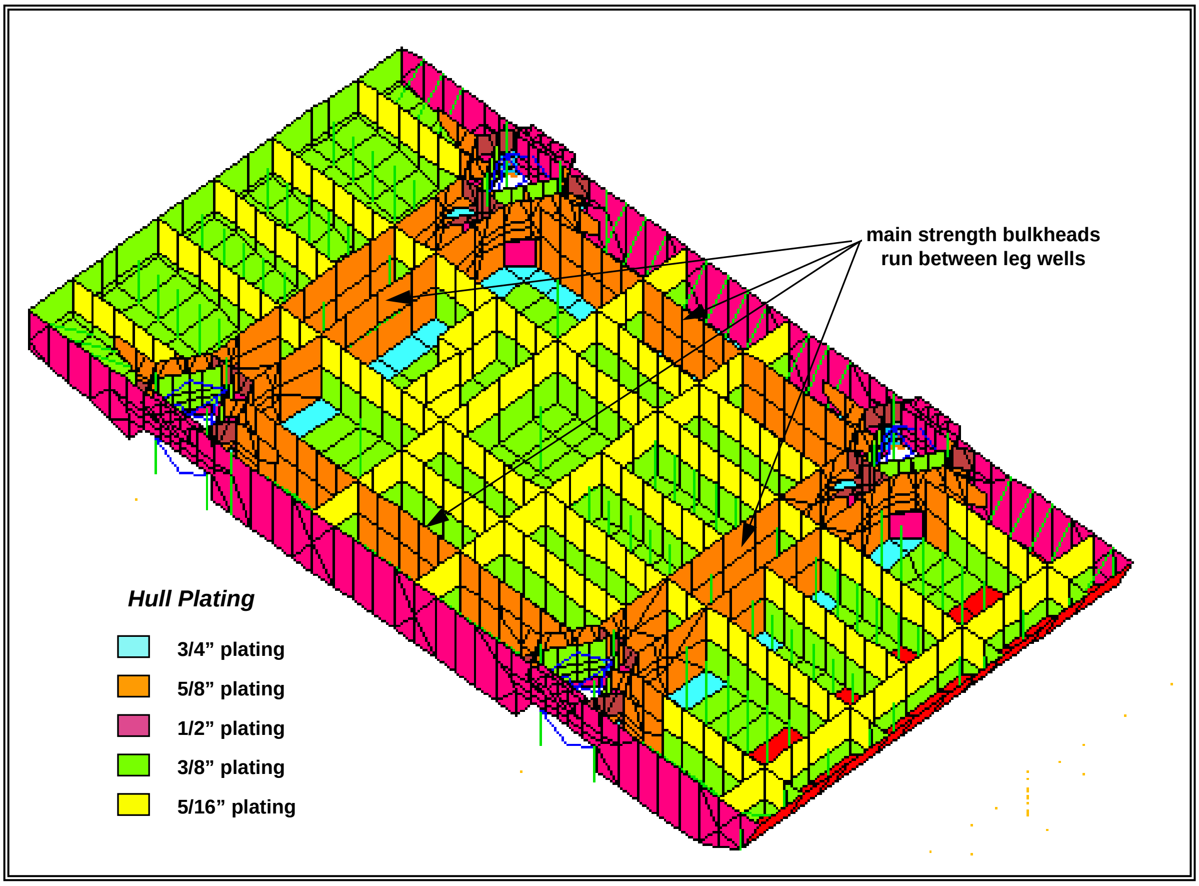

FIGURE 26 - FE Model of Hull with Deck Cut Away

The jackhouse, being integral with the hull, is the most detailed portion of the hull model. Some details of the FE model in the region of the jackhouse are shown in Figure 27. Like the hull, the main elements used in the jackhouse are thin plate elements and general purpose 3-D beam elements. In order to save weight, and to provide access to equipment within the jackhouse, rather large cutouts are provided within several of the vertical bulkheads. These cutouts are modeled in the FE model. The FE model was used as a true design tool for the jackhouse. 3-D AutoCad models of the jackhouse were developed directly from the FE model which was used to optimize plate thicknesses, cut-out maximum allowable sizes, stiffeners, etc.

Non-Linear Analysis

All primary load cases were applied using the gap elements, which are themselves non-linear, as well as using geometric non-linearity caused by large deflection behaviour, thereby accounting for the so-called P-delta effects directly in the FE analysis. Dynamic effects were accounted for separately.

Guide Gap Width Optimization

As the hull and legs were being constructed the overall weight of the vessel rose almost 20 percent. The increase in weight was due to changes in the hull structure, an additional level to the superstructure, a heavier helideck, and numerous equipment additions and weight changes. The additional elevated weight began to drive the chord stresses up near the allowable limits of the material for the 50 year storm conditions. In order to reduce the leg stresses without making structural changes to the legs differential gaps widths were analyzed and eventually incorporated into the design.

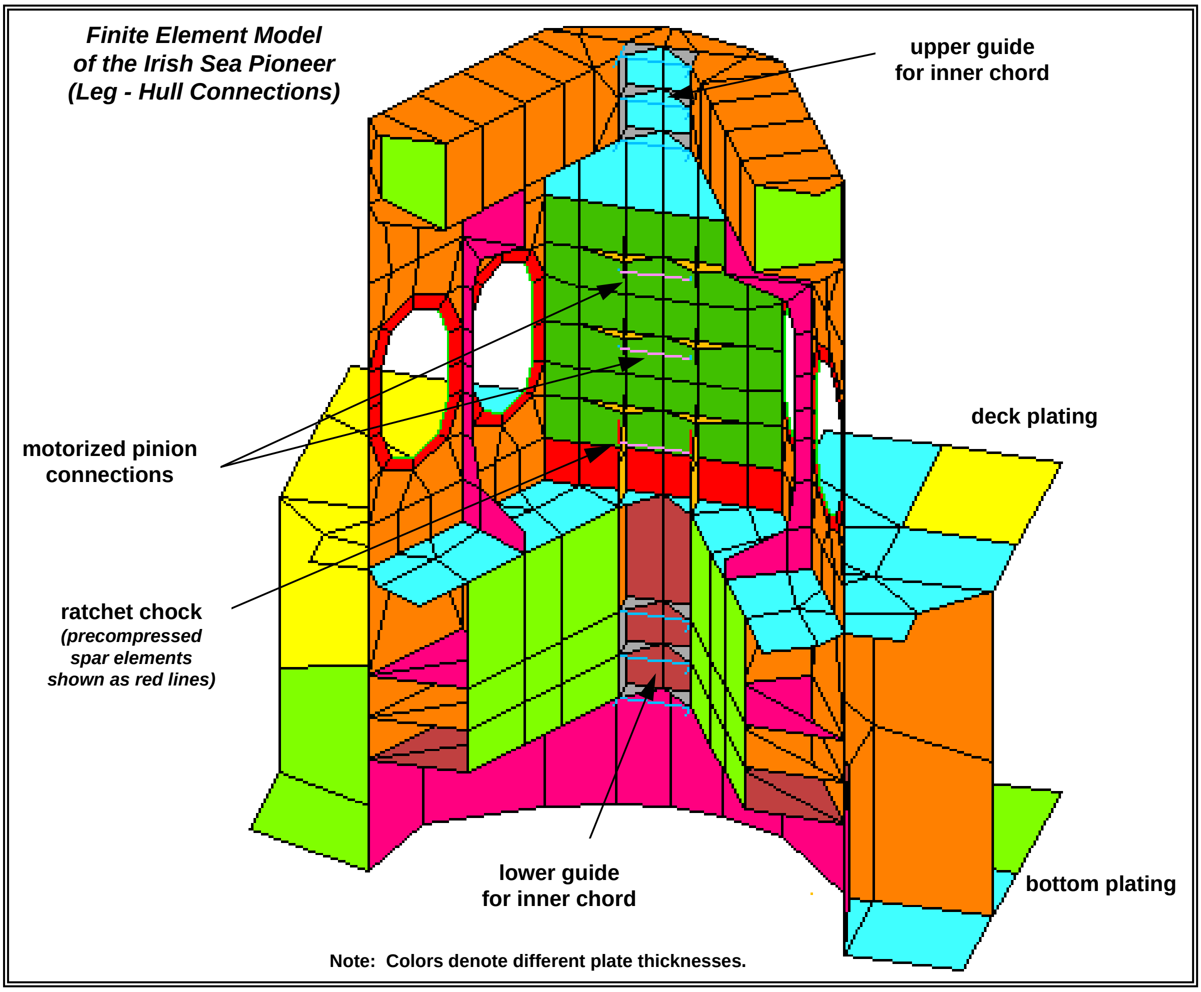

FIGURE 27 - FE Model of Jackhouse With One Side Cut Away

It is important to note that the gaps in the guides are not predicted by the detailed FE model to close over the full guide height. Where gaps close the guides are predicted to have a contact length of just a few inches with the sides of the rack, or with just two or three tooth tips. Hence a common practice of assuming a large percentage of the guide to be in contact with the chord is considered (for this structure) to be non-conservative when calculating bending stresses in the chords.

The logic behind differential gap widths within the guides is relatively straightforward. During storm conditions large bending moments caused by environmental loads must be taken out at the hull by the guides and pinions. Guide forces, especially if the bottom of the lower guide is at the mid-height of a bay, induce large bending moments in the chords. In some cases the chords with the highest axial loads can also experience the largest guide reactions and bending loads. There is little that can be done to reduce axial loads in the chords during storm conditions, but guide gap width adjustments can reduce, and even eliminate guide reaction forces on some chords, for certain critical load directions. Figure 28 shows pictorially how the differential gap widths in the guides work.

Notice in Figure 28 when there is a large gap width along the face of the inner chord the horizontal reaction is taken entirely by the two outer chords. The two outer chords, under this loading condition, have much lower axial compressive loads than the inner chord. The beam load case, shown above, is relatively easy to understand. Other load cases give rise to gap closure behaviour which is not as intuitive.

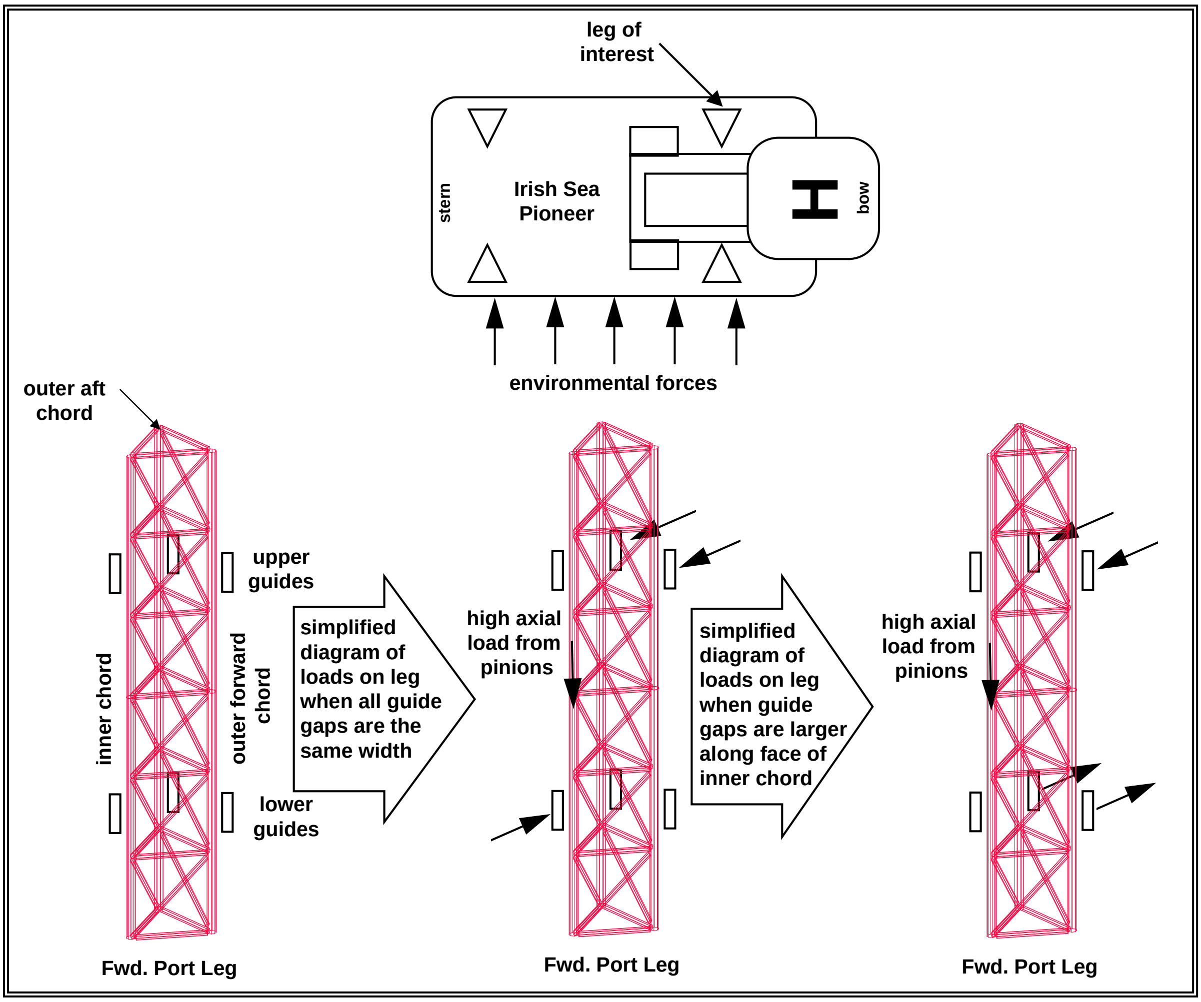

FIGURE 28 - Effect of Gap Widths at Lower Guides

For the *Irish Sea Pioneer*, the beam load case proved to be the most arduous condition for the chords.

To optimize the guide gap widths numerous finite element runs were conducted using a single leg model.

The single leg model was identical to the legs in the detailed finite element model. It had the same system of gap elements and rigid links used to model the guides. Each node that was attached to the hull in the global model was fixed in translation for the single leg model. Horizontal and vertical loads equal to the footing reactions found from detailed model runs were applied to the bottom of the leg. Wave and current forces were also applied to the single leg. Different lower guide gap widths were analyzed using this model for environmental loads between the directions of 0º (load from the stern toward the bow) and 90º (loads from the starboard to the port). From these finite element runs a gap width arrangement which gave the lower overall stresses in the chords and braces was found. Additional FE runs were conducted, accounting for construction tolerances of the leg and hull and guide plate wear, to assure the guide gap arrangement would work properly. The optimal guide gap widths found from this investigation were that incorporated into the global finite element model of the *Irish Sea Pioneer*.

SUMMARY

This paper provides a rather detailed description of the main systems and structure of the *Irish Sea Pioneer*, a new generation of offshore liftboat termed an Operating Support Vessel (OSV). This vessel has been built for a specific mission involving the servicing of unmanned platforms in Liverpool Bay, off the west coast of the UK. Some of the unique features of the vessel and its design include

- variable speed AC electric elevating system,
- ratchet chocks to support the hull self-weight when elevated, and
- differential gap widths in the leg guides and methodology of analysis.

Since the vessel may make fifty location moves in one year, emphasis is also given to afloat stability, vessel motions and limiting conditions when the legs tag bottom during location moves. A detailed description of both the hull and jackhouse design, as well as the leg and spudcan design, is provided.

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