Design of a mooring system for model testing of a downward breaking moored cone in ice

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SUMMARY

This document describes the preliminary design of a model mooring system for Phase 2 of the Ice Forces on Conical Structures project. The purpose and objectives of the project are outlined, as well as background information on the full-scale Kulluk conical drilling barge that is being modelled in these experiments. The methods used to obtain mooring system parameters are described, and the results presented.

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DESIGN OF A MOORING SYSTEM FOR MODEL TESTING OF A DOWNWARD BREAKING MOORED CONE IN ICE

SR-2005-01

Jeff Stanley
Michael Lau

April 2005

ABSTRACT

This document describes the preliminary design of a model mooring system for Phase 2 of the Ice Forces on Conical Structures project. The purpose and objectives of the project are outlined, as well as background information on the full-scale Kulluk conical drilling barge that is being modelled in these experiments. The methods used to obtain mooring system parameters are described, and the results presented.

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1.0 INTRODUCTION

In 2003, a project was commissioned by the Institute for Ocean Technology to examine the dynamic response of conical structures in sea ice and to consolidate the current information on this topic into a database, which would be expanded in the future with knowledge gained from additional experiments. This information on floating conical structures has applications for the oil and gas industry, as these vessels are a proven economical alternative to fixed structures in deeper water and marginal oil fields where sea ice poses hazards to operations. The dynamic response of these platforms refers to the forces that are imposed on their hulls and mooring systems, and how they move in response to ice loads. Gaining a better understanding of how these platforms respond to ice loads would allow for improved operational safety and efficiency in terms of responding to severe ice conditions and preventing off-site emergency rig moves or production disruptions in response to false alarms. Also this information is important in the design of mooring systems and drilling risers, which are heavily influenced and can be easily damaged by excessive vessel motions.

Tests of a 1:40 scale model of the conical drilling barge “Kulluk” (See Appendix B) were conducted and divided into two phases: Phase 1 was completed in early 2004, with the Kulluk model being tested in a variety of level ice conditions and velocities while rigidly mounted to the tow post. The purpose of these tests was to establish a base line for comparison with a moored structure, which will be tested in Phase 2. Phase 2 aims to repeat the Phase 1 tests with a moored structure, and also to vary the range of mooring system stiffnesses used to determine what effect that has on loads and vessel motions. In preparation for the Phase 2 tests, research was conducted into past experiments and literature concerning the Kulluk to determine the most appropriate way to model the full-scale mooring system, the findings of which are presented in this report. The objectives of this document are thus as follows:

i) Outline the design of a mooring system to be used in the Phase 2 model testing using equipment and facilities available at IOT.
ii) Determine the relevant parameters that will govern the behavior of the simulated mooring system.
iii) Verify that the design accurately reflects the performance of the full-scale vessel.

2.0 BACKGROUND

2.1 The Kulluk

The Kulluk [Fig. 1] was a prototype moored drilling platform with a downward-breaking conical hull shape, designed to operate in ice-infested waters. Construction began in 1982 and the finished vessel was delivered in 1983 to Gulf Canada Resources Ltd. It was intended for exploratory drilling in the Beaufort Sea in water depths of 20-60m, where ice conditions limited the effectiveness of conventional drillships. The Kulluk displaces approximately 28 000 tonnes, has a main deck (maximum) diameter of 82m, a waterline diameter of 70m and an operating draft of 11.5m. Its conical, downward-sloping hull allowed the vessel to break ice in flexure regardless of the direction from which the ice was approaching. The design also includes a deflector ring at its bottom to prevent ice rubble from traveling under the hull and fouling the mooring lines or drilling risers. The Kulluk was designed to be able continue drilling operations without icebreaker support in solid sea ice 1.2m thick. The vessel was kept on station by a radially symmetrical system of twelve 3 ½" (90mm) wire rope mooring lines, which were routed from on-deck winches through the hull and out the bottom.

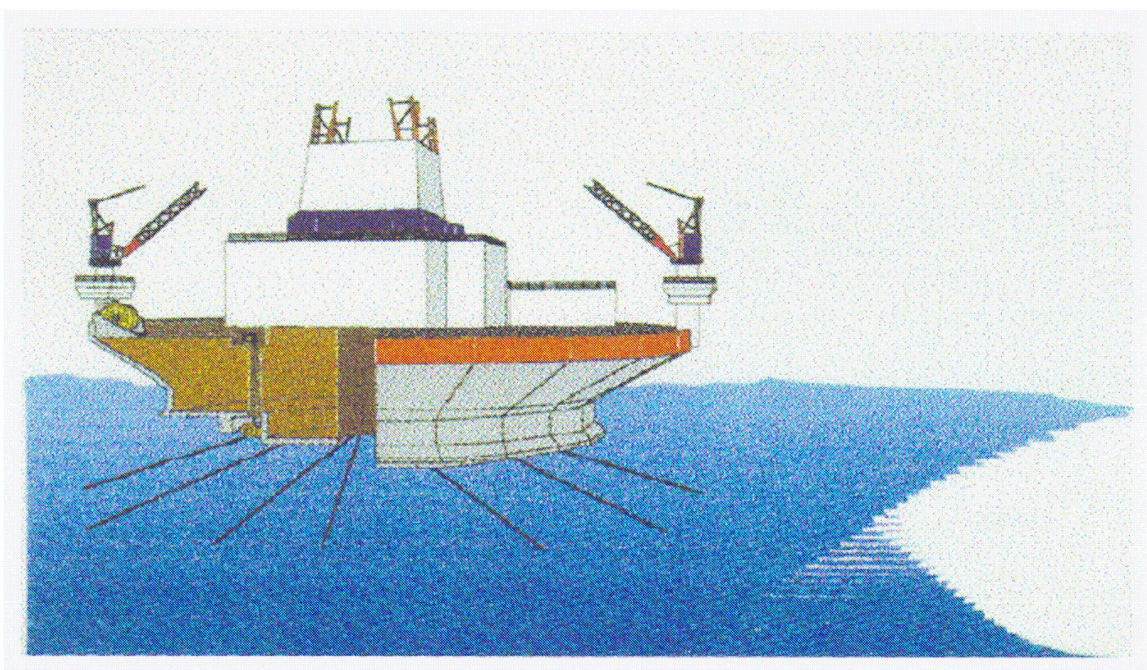

Figure 1: The Kulluk (Source: Wright et al (1999))

2.2 Full-Scale Behavior

As a general rule, the maximum allowable excursion of the Kulluk over its target while drilling was 5% of the water depth (Wright et al, 2000). This restriction prevents excessive loads from developing in its risers. In a "survival mode" where drilling risers have been disconnected, the Kulluk is permitted to move off station by 10% of the water depth. Considering its design depth, for routine operation this gives a full-scale excursion of 1-3m. It should be noted that environmental conditions in the Beaufort Sea are varied and somewhat different than what the Kulluk was designed to withstand. Maximum ice thickness can reach 1.8m, and that due to environmental forces it is often broken up into pack ice (large numbers of relatively small pieces) and large ice flows rather than monolithic, unbroken sheets. A number of different phenomena occur with broken ice, including the formation of pressure ridges – places where the ice sheet has buckled and the rubble accumulations have frozen together – which are usually several times thicker than the surrounding ice. Ice ridges and older (multi-year) ice, which are much thicker and stronger than the norm, were scattered amongst the normal sea ice and would have created excessive loads on the Kulluk if they struck the vessel. This made it necessary to constantly employ icebreaker support to fragment or push the ice before it impacted with the rig.

Under normal operation, the Kulluk's twelve-line mooring system was able to withstand global loads of 750 tonnes and individual line loads of 260 tonnes. In a survival mode, the maximum load increased to 1000 tonnes and peak line loads of 390 tonnes (Wright et al, 1999). The individual mooring lines had the following properties (Provided by Gulf Canada Resources Ltd. in the HSVA study (Evers et al, 1983)):

Diameter:	90mm
Breaking Load:	5.1 MN (520 tonnes)
Unit Weight (Dry):	33.78 kg/m
Unit Weight (Submerged):	28.00 kg/m
Length:	Varied with line configuration and water depth
Preload/Pretension:	Varied with line configuration and water depth

Using the difference in dry and submerged unit weight and assuming the density of sea water to be 1026 kg/m^3, the cross-sectional area of the wire rope can be calculated. If the cable segment is assumed to be 1m long, then the difference between its dry and submerged unit weights will be the mass of the volume of seawater that the cable segment displaces; which is equal to 5.78 kg. The volume of the displaced seawater is equal to the volume of the cable segment, and is found using:

$$V = {}^{m}\!/_{\rho} \tag{1}$$

where m is the mass in kg, ρ is the density of seawater in kg/ m^3 and V is the volume in m^3. Equating this to the volume of the cable segment:

$$V = A \bullet \ell \tag{2}$$

where V is volume in m^3, A is the cross-sectional area in m^2, and ℓ is the segment length in m, the cross sectional area can be calculated and is equal to $0.00563m^2$. Note that this figure represents the total area of the steel components in the wire rope, and excludes the spaces between the strands that would be filled by water.

2.3 Past Experiments

The Kulluk was the prototype for a drilling barge concept that had never before been attempted. The hull was extensively tested both before and after it began drilling operations and has been the subject of numerous studies. Part of the reason for so much interest in the Kulluk is that as part of its day-to-day operations, extensive records were made of ice loads, ice conditions, and other pertinent data that could be used by researchers for comparison with experimental results. Detailed, accurate, and complete full-scale performance data of this nature are very rare and as such, it continues to be used in studies.

Comfort et al (1999) summarized the setup and findings of the most pertinent Kulluk studies to date. These are presented in Table 1:

Table 1: Existing Test Data

Institution	Year	Model Scale	Full-Scale Line Stiffness Equivalent	Full-Scale Global Stiffness Equivalent
ARCTEC Canada Ltd (ACL)	1982	1:30	-	-
Hamburgische Schiffbau-Versuchsanstalt GmbH (HSVA - Hamburg Ship Model Basin)	1983	1:45	625kN/m	3.75MN/m (Calculated)
Iowa Institute for Hydraulic Research (IIHR)	1985	1:45	567kN/m (Calculated)	3.4MN/m
Iowa Institute for Hydraulic Research (IIHR)	1988	1:45	167kN/m (Calculated)	1.0MN/m

For more information see Appendix B, which contains a bibliography of the numerous papers which were reviewed and a more detailed summary of the experimental data listed above.

2.4 Scaling Law

In order to accurately test a model of a vessel to determine its full-scale characteristics, one must not only create an accurate scale model, but also determine how to scale the environmental factors and how to compensate for that which cannot be scaled. These are things ranging from reaction forces, velocities, and distances to mechanical properties such as elasticity and flexural strength. A table of scaling factors (in terms of model scale λ) is given by Michael (1978) along with information on scaling laws. For the purposes of designing the mooring system we are only concerned with the scaling factors listed in Table 2.

Table 2: Scaling Factors

Variable	Scaling Factor
Length/Distance/Thickness (m)	λ
Area (m^2)	λ^2
Volume (m^3)	λ^3
Mass (kg)	λ^3
Density (kg/ m^3)	1
Time (s)	$\lambda^{1/2}$
Velocity (m/s)	$\lambda^{1/2}$
Acceleration (m/s^2)	1
Force (N)	λ^3
Spring Stiffness k [1]	λ^2
Flexural Strength	λ
Crushing Strength	λ
Shearing Strength	λ
Elastic Modulus	λ
Rigidity Modulus	λ
Poisson's Ratio	1
Friction Coefficient	1

These factors allow for most of the mooring system parameters to be brought to model scale. However, despite this there are some elements that should be scaled, but cannot be. In order to accurately reflect the full-scale elastic characteristics, the model mooring line would have to be made of a material that possessed the scaled-down properties (such as the elastic modulus) of the full-scale mooring line as well as dimensions. This is not feasible to achieve, and most often model mooring lines are made of steel cable, which has the same material properties of the steel in the full-scale cable.

Also, owing to the physical limits of the facilities it is usually not feasible to use model moorings of scaled-down length (when scaling down mooring cables that

[1] Derived from $F = kx$ relationship.

are hundreds of meters long, the model-scale equivalents are too long to fit into the test tanks). Without any compensation, this would result in a model mooring system that is vastly stiffer by comparison to full-scale, and thus a poor reflection of its behavior. To remedy this, researchers determine the overall (global) restoring characteristics of a full-scale mooring system in terms of how the vessel will respond in six degrees of freedom (the x, y or z directions and rotations about these axis) when a certain overall load is applied by waves, wind or ice. For the purposes of the response of a moored system due to ice loads, the important directions are surge (horizontal motion along the x-axis, which runs the length of the vessel), pitch (rotation about the y-axis), and heave (vertical movement in the z-direction). The single most important direction is surge, and as such the greatest emphasis is on properly modeling surge response. For most mooring systems surge offset is not a linear function of load, however in the Kulluk's case the response is linear, as shown by an operator provided load-displacement chart (Evers et al, 1983). This means that the stiffness (in the surge direction) of the entire mooring system can be modeled using a single constant, k_g, which can be very easily scaled down for use in model experiments. In model testing, spring arrangements are directly inserted into the mooring lines with stiffness constants selected to reflect the global restoring characteristics of the full-scale moorings. The proper selection of spring constants is therefore critical to any experiment. Section 2 details some of the calculations that were performed to determine the spring constants needed for this experiment and those used in others.

2.5 Design Specifications

For the purposes of this project, using a twelve point mooring system was not a preferred option. This is owing to the fact that one of the objectives of the design of the model mooring system was to make use of pre-existing equipment at IOT to reduce project expenditures. A brief inventory was conducted for mooring line equipment with the following results:

- Mooring Hardware: IOT has sufficient hardware to support 16 mooring lines, using two sets of 8 mooring posts and pulleys. At the time this inventory was conducted, 5 sets of posts and pulleys from the first set of mooring posts had been accounted for and were available. The set of mooring posts is of smaller diameter and proportion, and it is currently not clear if they would be adequate for this project. A sufficient number of these posts exist, but it is uncertain if they use the same or have their own pulley systems; which in the latter case would have to be located.
- Springs: IOT will not be able to provide springs in the stiffness ranges this project will require. All springs will likely have to be purchased.
- Load Cells: The number of load cells appropriately rated for the relatively high loads expected for the Phase 2 testing is limited. There are sufficient waterproof load cells in the 240lb range to accommodate a four-line system.

It was decided that a four-point mooring system would be the simplest mooring arrangement that would not compromise the movement characteristics of the full-scale Kulluk. A four-line system is also the most convenient in terms of materials, as there is sufficient equipment to model four lines without a need to purchase any additional pieces; save springs. Given that several different stiffnesses will be tested, utilizing a four-line system offers significant cost savings in terms of the springs that must be purchased. An above-water configuration was chosen owing to technical and time constraints, as an underwater mooring would likely be fouled by ice that passes under the model during high-speed tests, and it would require the use of divers and a great deal of special fabrication. It is also doubtful that the underwater carriage in IOT's ice tank could support the load of the model moving through ice. A number of different options for horizontal moorings were considered, but ultimately the technical constrains (outlined in more detail in Section 3) made the horizontal four-line mooring system the preferred choice.

3.0 DETERMINATION OF SPRING CONSTANTS

3.1 Full-Scale Characteristics

As stated in section 1.4, the most relevant parameter relating to the behavior of a mooring system is its global surge stiffness, that is, the constant that will reflect the distance the vessel will displace in response to a load on its entire mooring system from things such as surface ice pressing against the vessel. It is also convenient to be able to relate the individual line stiffness to the resulting global surge stiffness, which can be derived from trigonometry. The Kulluk's mooring lines in a typical arrangement (See Appendix A) were approximately 30° apart [Fig. 2]. If we assume these mooring lines are exactly 30° apart and of equal length (and therefore stiffness) the calculations are greatly simplified.

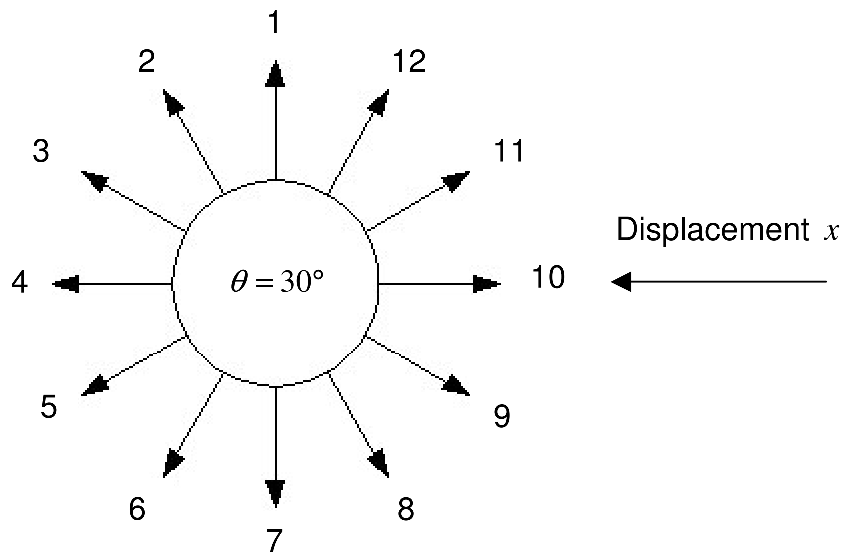

Figure 2: Full-Scale Mooring Line Arrangement

Assuming all pretensions are equal, a disturbance x will create a restoring force F which is equal to the global surge stiffness, k_g, multiplied by x. For convenience, positive x is taken to be horizontal displacement to the left. The contributions of lines normal to the direction of motion (Lines 1 and 7) are neglected, as the change in length is very small. The restoring force can be found by taking the difference in line tensions on both sides of the vessel. Since the arrangement is symmetrical, the tensions in the following pairs are equal: 3 & 5, 2 & 6, 9 & 11, and 8 & 12. Summing the individual line tensions, the following equation results:

$$F = (T + kx) + 2(T + kx_1)\cos 30^\circ + 2(T + kx_2)\cos 60^\circ - (T - kx) - 2(T - kx_1)\cos 30^\circ - 2(T - kx_2)\cos 60^\circ \quad (3)$$

where F is the restoring force, T is the individual line tension; x_1 is the component of motion along an angle of 30 degrees; and x_2 is the component of motion along an angle of 60 degrees. Substituting in $x_1 = x\cos 30^\circ$ and $x_2 = x\cos 60^\circ$ this simplifies to:

$$F = 2kx + 4kx\cos^2 30^\circ + 4kx\cos^2 60^\circ = 6kx \quad (4)$$

Therefore, global surge stiffness k_g is equal to 6 k for a twelve-line system.

3.2 Model Stiffness

Since the model mooring has only four lines [Fig. 3], another geometric relationship must be established between the individual lines and their global surge stiffness in a similar manner as the full-scale arrangment. The same displacement x in model scale would allow us to sum the difference in line tensions as before:

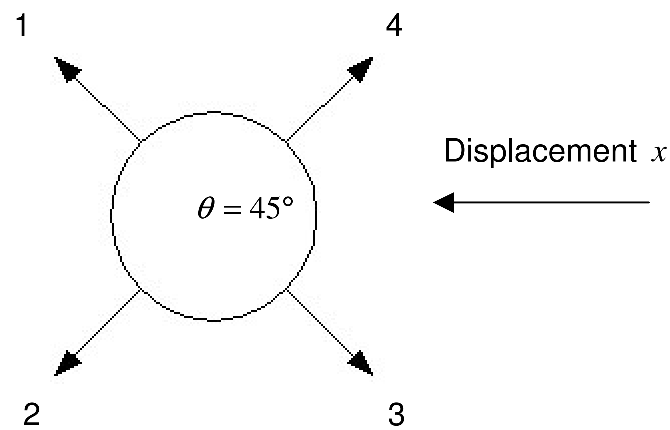

Figure 3: Model Mooring Line Arrangement

$$F = 2(T + kx_1)\cos 45^\circ - 2(T - kx_1)\cos 45^\circ \quad (5)$$

where F is the restoring force; T is the line tension; and x_1 is the component of motion along an angle of 45 degrees, which is $x\cos 45^\circ$. The above expression simplifies to:

$$F = 4kx\cos^2 45^\circ = 2kx \quad (6)$$

Therefore, k_g is 2 k for a four-line system.

3.3 Selection of Spring Characteristics

The springs used in the model mooring will govern the restoring forces exerted by the model mooring system, and as such they must be selected in a way that best approximates the characteristics of the full-scale vessel. Since one of the objectives of the Phase 2 testing is to make comparisons with past experimental results, it is also important to note the stiffness values used in those tests that are listed in Table 1. They will be included in the range of stiffness values for the Phase 2 testing.

A problem that needed to be overcome was the fact that information on the Kulluk's full-scale global stiffness and line properties was not easily available. Originally these values were estimated making use of the trigonometric relationships derived in Sections 2.1 and 2.2, existing mooring configuration data, and a formula for calculating the EA value of wire rope used in a mooring analysis program called MOORING_SYSTEM (Lau and Stanley, 2005). The EA value is the product of the effective elastic modulus and the cross-sectional area, and is the parameter commonly used to describe full-scale line stiffness. The formula from MOORING_SYSTEM provides an approximation for EA of a wire rope of a given diameter when that value is unknown, and was verified as being reasonably accurate for older wire rope mooring lines. This formula is:

$$EA = \frac{\pi}{4}(D/1000)^2 \bullet 87x10^9 \bullet 1.25 \tag{7}$$

where D is the diameter of the cable in millimeters.

In this case, 3 ½" becomes 88.9 mm and $EA = 675.0$ MN.

When the HSVA test documentation (Evers et al, 1983) became available, a more accurate EA value was calculated using the load-displacement curve for a water depth of 35m provided in the report by Gulf Canada Resources Ltd. For the HSVA study, a range of motion of –2m to 2m was chosen and the graph was assumed to be linear in this region, resulting in a line stiffness of 625 kN/m. Using the formula for effective stiffness constant:

$$k = EA/\ell \tag{8}$$

where ℓ is the full-scale line length, an EA can be calculated. A line length of 670m was given for a water depth of 35m. The resulting EA value was:

$EA = 418.75$ MN

The first method gives results that are approximately 60% higher. The second method should be used as it is based off operator-supplied information. Note that the line stiffness varies with the length of the mooring lines (which changes with water depth), and is another reason why it is important to test a range of stiffnesses. Actual Kulluk mooring configuration data such as that listed in Appendix A, obtained from vessel records (Wright et al, 1999), shows that mooring line length exceeded 1000m in some cases, which would significantly change the line stiffness. Table 3 contains a list of suggested k values for the experiment in which line length is varied from 500m to 1400m. Note that the given stiffness values from the HSVA and 1985 IIHR tests are in bold.

Table 3: Stiffness Matrix

Full-Scale Global Stiffness (kN/m)	Full-Scale Line Stiffness (kN/m)	1:40 Scale Global Stiffness (kN/m)	1:40 Scale Line Stiffness (kN/m)	Description
5025.00	837.50	3.141	1.570	Line Length 500m
4187.50	697.92	2.617	1.309	Line Length 600m
3750.00	**625.00**	2.344	1.172	Line Length 670m (**HSVA**)
3400.00	523.44	2.125	1.063	**1985 IIHR**
3140.63	523.44	1.963	0.981	Line Length 800m
2512.50	418.75	1.570	0.785	Line Length 1000m
2284.09	380.68	1.428	0.714	Line Length 1100m
2093.75	348.96	1.309	0.654	Line Length 1200m
1932.69	322.12	1.208	0.604	Line Length 1300m
1794.64	299.11	1.122	0.561	Line Length 1400m

The stiffnesses generated in this table must be verified as being consistent with independent sources such as other full scale mooring configurations and the values used in past experiments. Unfortunately third party information was not available at the time of the writing of this report therefore no further comparisons can be made. A recommendation has been made to investigate further.

3.4 Maximum Excursion

As outlined in Section 2.2, the full-scale Kulluk was restricted by its mooring system to move no more than 5% off station during normal operation. In a worst-case scenario, the model would move parallel to one of its mooring lines, creating a maximum spring extension or compression for that line equal to the full displacement. In a survival mode, the Kulluk would be able to move off-station by a distance equal to 10% of the current water depth. It is possible that this limit will be tested, so as an added margin of safety the model mooring system will allow for quadruple the normal operating displacement and double the survival mode option, which translates to 20% of water depth. This added margin of safety is important, as the spring must never extend beyond or collapse below the limits of its linear-elastic region. If this were to happen the characteristics of the mooring system would change, affecting measured loads, and there would be a risk that the springs may be permanently damaged.

3.5 Model Spring Pretension & Spring Length

The maximum excursion and spring stiffnesses are needed to determine the pretension that must be placed on the model mooring lines to ensure that the springs are stretched enough so that they can both expand and contract by 20% of water depth without leaving the spring's linear-elastic region. Extension beyond this range would lead to either permanent deformation, or the spring becoming totally slack. The spring pretension needed is the maximum excursion (which is the maximum possible change in line length, and thus spring length), multiplied by the spring constant k. The spring length needed to ensure that the range of motion falls within the elastic region is dependant on coil diameter, wire size, and the maximum traverse (the sum of the maximum extension and compression, in this case twice the maximum excursion). At present the calculation of spring length will be conducted at a later date by the Design & Fabrication department, as there is no general rule of thumb that can conveniently relate required spring length, using the parameters that have already been calculated.

3.6 Mooring System Parameters

At present line stiffness is a function of line length, and the maximum permissible excursions are functions of water depth. In order to calculate actual values for the permissible excursions and maximum forces that can be exerted on the system, line length must be related to water depth. Maximum force is calculated by multiplying the global model spring constant k_g by a distance equal to 10% of the given water depth. Water depth is extrapolated based on the assumption that the pretension on the Kulluk's mooring lines makes them completely taught, and that the angles of the mooring lines are kept constant.

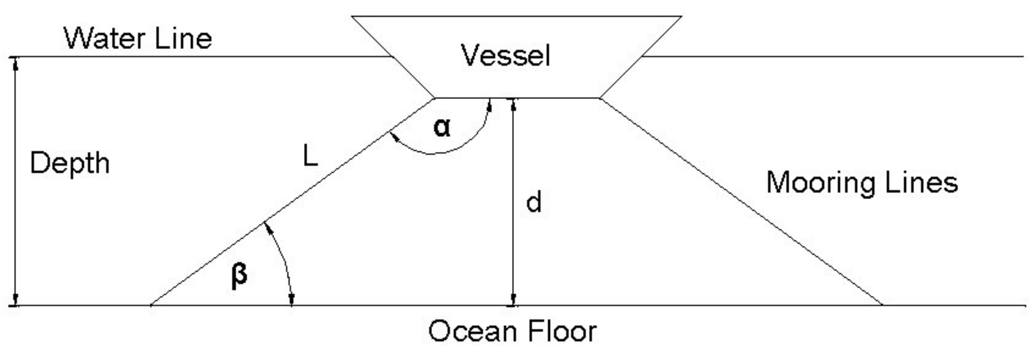

Figure 4: Depth Calculation

From an elevation drawing of the Kulluk (Pilkington et al, 1986), it can be determined that the distance from the vessel's bottom to the fairlead points (the points where the mooring lines are attached to the hull) is approximately 0.5m. Given that its operating draft is 11.5m, the fairleads are located 11.0m below the waterline. This means that the vertical distance d from the fairleads to the ocean floor is equal to the depth minus 11m. Using any known depth and line length L, the angles α and β can be calculated using trigonometry. It is most convenient to use β, so:

$$\beta = \sin^{-1}(d/L) \tag{9}$$

At present there are three data points that can be used in this equation (see Appendix A), from which an average value of β can be calculated [Table 4]. Using the same formula, but solving for d with the known angle β and line length, the water depth can be determined. This analysis is most valid for shallow water depths, where the sag in the line owing to its weight is at a minimum. The correlation between the values extrapolated from the average water depth and actual data has been plotted in Figure 5 and show reasonable agreement.

Table 4: Calculation of Average β

Source	**Line Length (Average) (m)**	**Depth (m)**	**Beta (Deg)**
HSVA	670	35	2.05
2J-44 Mooring Configuration (Wright et al (1999)	652	32	1.90
Kulluk Barge Report (Wright et al (2000)	917	52	2.56
Average	746.3	39.7	2.17

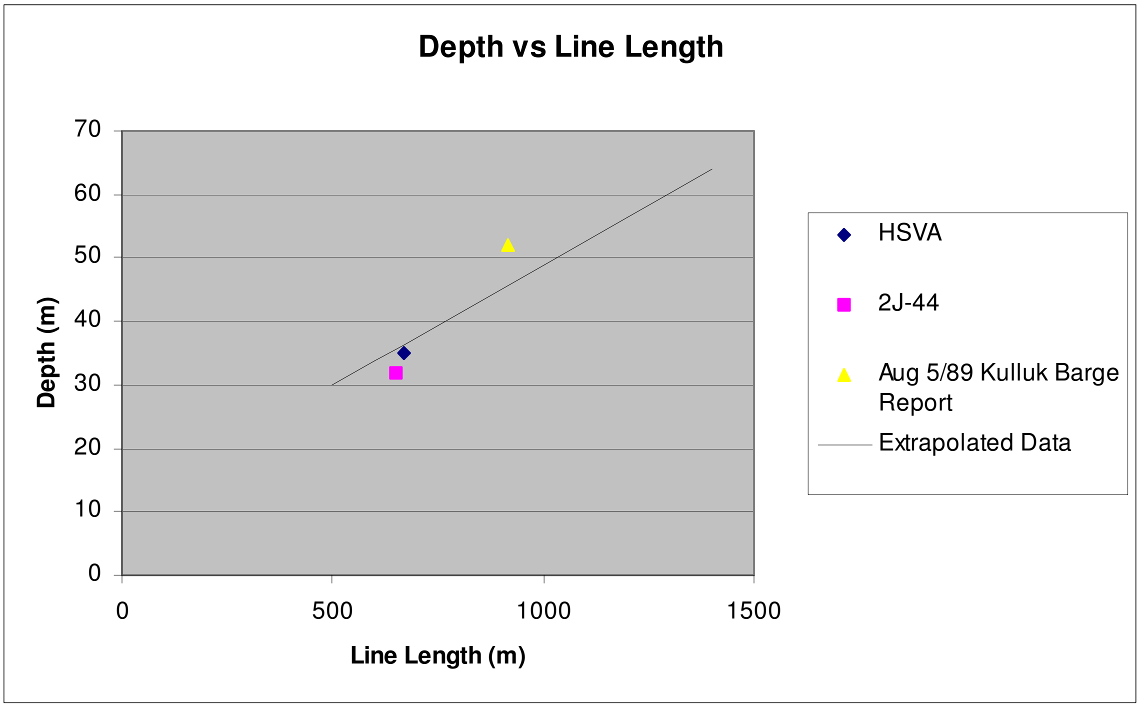

Figure 5: Depth versus Line Length

Table 5: Model Mooring System Parameters

Full-Scale Global Stiffness (kN/m)	Full-Scale Line Stiffness (kN/m)	1:40 Scale Global Stiffness (kN/m)	1:40 Scale Line Stiffness (kN/m)	Line Length (m)	Estimated Water Depth (m)	Pretension Needed (N)	Counterweight Needed for Pretension (kg)	Maximum Model Force for 10% Model Depth Excursion (N)	Maximum Spring Traverse (cm)
5025.00	837.50	3.141	1.570	500	29.9	675.23	68.86	235.02	15.0
4187.50	697.92	2.617	1.309	600	33.7	562.70	57.38	220.62	16.9
3750.00	**625.00**	2.344	1.172	670**(HSVA)**	36.4	503.91	51.39	213.10	18.2
3400.00	566.67	2.125	1.063	**1985 IIHR**	-	456.88	46.59	-	-
3140.63	523.44	1.963	0.981	800	41.3	422.02	43.04	202.63	20.6
2512.50	418.75	1.570	0.785	1000	48.9	337.62	34.43	191.83	24.4
2284.09	380.68	1.428	0.714	1100	52.7	306.92	31.30	187.91	26.3
2093.75	348.96	1.309	0.654	1200	56.4	281.35	28.69	184.63	28.2
1932.69	322.12	1.208	0.604	1300	60.2	259.71	26.48	181.87	30.1
1794.64	299.11	1.122	0.561	1400	64.0	241.16	24.59	179.49	32.0

4.0 TEST SETUP

The Phase 2 tests will be conducted in IOT's ice tank. The ice tank is 96m long, 12m wide and 3m deep with a useable length of over 70m. The ice tank is equipped with two carriages: a primary carriage for tow testing and a service carriage for ice control and measurement. The main carriage weighs 80 000kg and can travel at velocities ranging from 0.001 m /s to 4.0m/s with an accuracy of 0.1%. The model and mooring system must be fixed to the main test carriage in order to be towed through the ice sheet. The location of the posts which will act as anchors for the model mooring system, and the location of the points where the model moorings are attached to the model itself have a significant impact on experimental results as outlined in the following two sections.

4.1 Mooring Post Location

As was stated earlier, it is preferable if equipment that is readily available is used for the ice tank tests to come. After completing some measurements of both the Kulluk model and the ice tank's test frame, it has been determined that four mooring post and pulleys available at IOT would be adequate for these experiments. The post, spring and pulley arrangement will be pretensioned with counterweights and then clamped in place to prevent the counterweight from moving. Figure 6 shows the proposed test setup, while Figure 7 is a more detailed illustration of the setup of each of the four mooring posts.

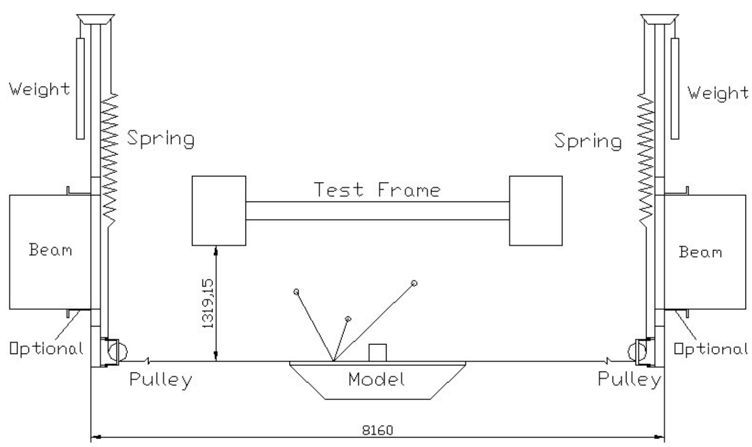

Figure 6: Mooring System

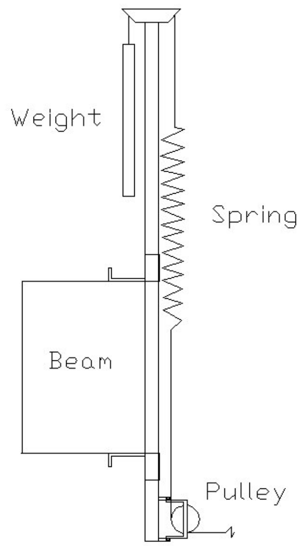

Figure 7: Mooring Post Configurations

Using a four-line configuration in an X-pattern, as shown in Figure 8, is the preferred option with each line at a 45° angle to the model's heading and the anchor points equally spaced from each other (in this case the spacing would be equal to the ice tank carriage frame's inner diameter of 8.16m). Several options for mounting mooring posts on the ice tank carriage were explored, including several positions on the test frame and carriage frame.

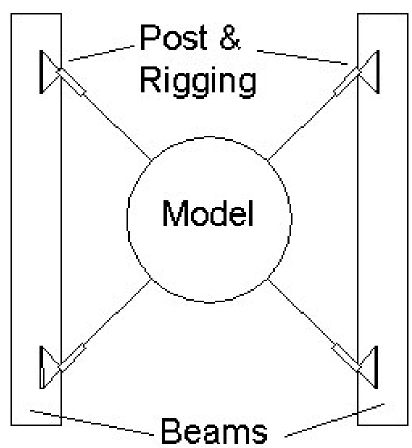

Figure 8: Mooring Lines – Top View

The maximum extension on the ice tank's test frame is 3m, but because the Kulluk model is slightly over 2m wide it does not leave sufficient clearance for the mooring hardware and allow for model movement. While there would be sufficient clearance to prevent the model from striking the mooring hardware, it would not be enough to prevent the model's displacement from significantly changing the angles of the mooring lines. The acceptable limit for this change in angle is no more than 5° (as the change in the cosine of the angle and thus force components would be more than 10%). The longer the model mooring lines are, the less they will be affected by the displacement of the model owing to loading. The two remaining options that would allow the use of longer mooring lines are to use the carriage frame, or a pair of fixed structural members that are part of the support for the test frame. Also, if loads on the model are similar to those in Phase 1, then the model will experience much greater movement within the ice tank. In this event it is preferable to have as much of a safety factor as possible, in which case the option to mount the posts on the carriage frame is preferable.

Further limits are imposed by the model's instrumentation. In order to facilitate the use of the Qualisys displacement tracking system the test frame must be raised to maximum or near-maximum position (approximately 1.3m above the water level) to allow sufficient clearance for the tree and markers used to track movement and to ensure that the infrared cameras have the model in view at all times. While this did not, in itself, create an issue, it had one notable side effect: The Kulluk model is wide and has a tendency to accumulate rubble, which results in relatively high ice loads. Since the test frame and connected beams would be raised high above water, the post would have a long unsupported end that may flex when loaded, influencing line tension readings. Any flexure of the mooring post would adversely affect the behavior of the mooring system and the experimental results. This may be remedied by using other means such as wires and cables to secure the free end of the post, but it was less of a problem with the carriage frame as it is closer to the water.

Using the carriage frame provides ample space for equipment and model motion, however the use of much longer cables presents another issue. A longer cable would stretch more under load and have greater influence on the system stiffness value. However, after consultation with the Design & Fabrication department, the influence of these variables was deemed negligible as the steel cables used in model moorings have a much higher stiffness than the springs inserted into the system

In order to accommodate the model motion with a safety factor to accommodate unexpectedly high ice loads, the carriage frame was chosen as the best option. The mooring posts would be affixed with clamps and other temporary means to avoid damaging the carriage frame; the exact configurations of which will be decided by the ice tank technical staff at the time of testing. The ice tank carriage beams have an inner spacing of approximately 8.16m wide x 9.45m long. Since the Kulluk model is symmetrical, it would be ideal to have all four posts arranged in a square pattern with individual posts being 8.16m apart, or approximately 8.09m center to center using the 7cm diameter mooring posts [Fig. 9]. Figure 10 shows the proposed layout of the mooring posts on the carriage frame.

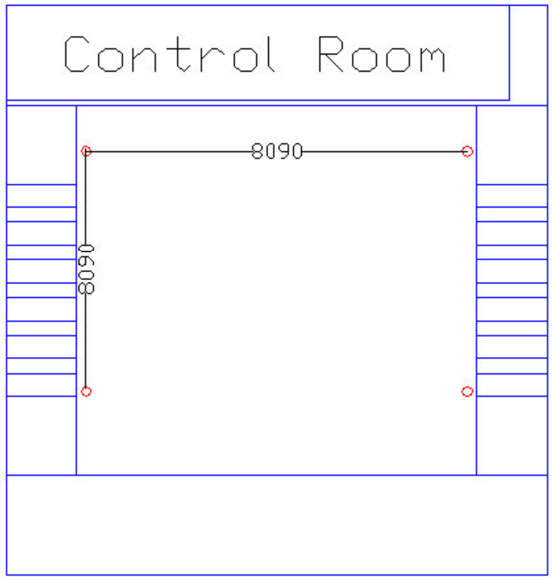

Figure 9: Ice Tank Carriage – Top View of Post Locations

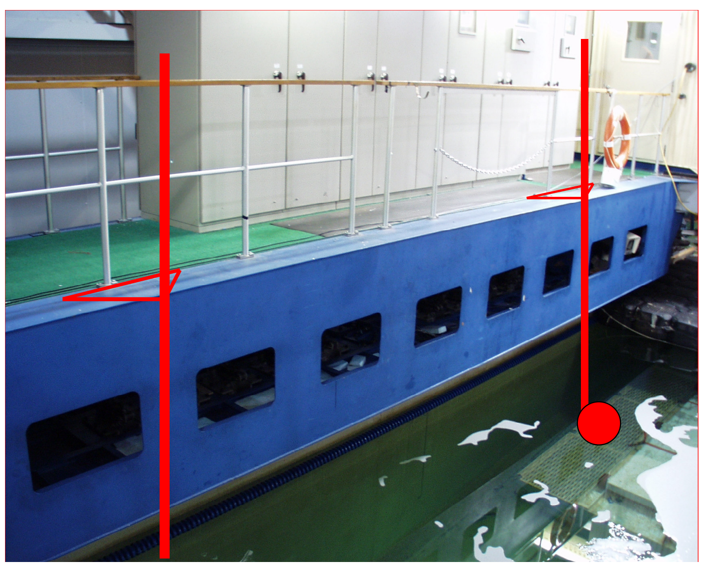

Figure 10: Mooring Post Setup on Carriage Frame

4.2 Fairlead Position

Ideally the model should be towed through its center of buoyancy, however in this case that is not feasible owing to the nature of the model and the test equipment. At present all mooring lines will need to attach directly to the “main deck” of the model, either at its edges or closer to its center. Attachment points near the edges of the model are more favorable at this point, as mooring lines located closer to the vessel’s center would possibly interfere with instrumentation. While it was not believed this would have a significant effect on horizontal motions, this arrangement would have an effect on the pitch response of the vessel. To gauge this, a comparison was made utilizing the MOORING_SYSTEM software (mentioned earlier) to simulate two mooring systems that are equivalent in every way save the locations of the fairleads on the vessel in a water depth of 80m. In one case the fairleads were attached on the edges of the main at a radial distance of 40.5m away from the center of gravity of the vessel, and in the second case they were attached below the waterline at a radial distance of approximately 18m from the vessel center (the location of the full-scale fairleads). The first case was taken as the base case for comparison as it reflects the full-scale configuration. As no information on the location of the Kulluk’s center of gravity was available, it was assumed to be at the geometric center of its near-circular hull, at the same elevation as its full-displacement water line (11.5m from keel).

Figure 11 shows a plot of surge offset versus surge force. It confirms the assumption that there would be negligible influence on vessel surge response owing to fairlead placement, as the plots of surge versus surge force for both cases lie on the same line.

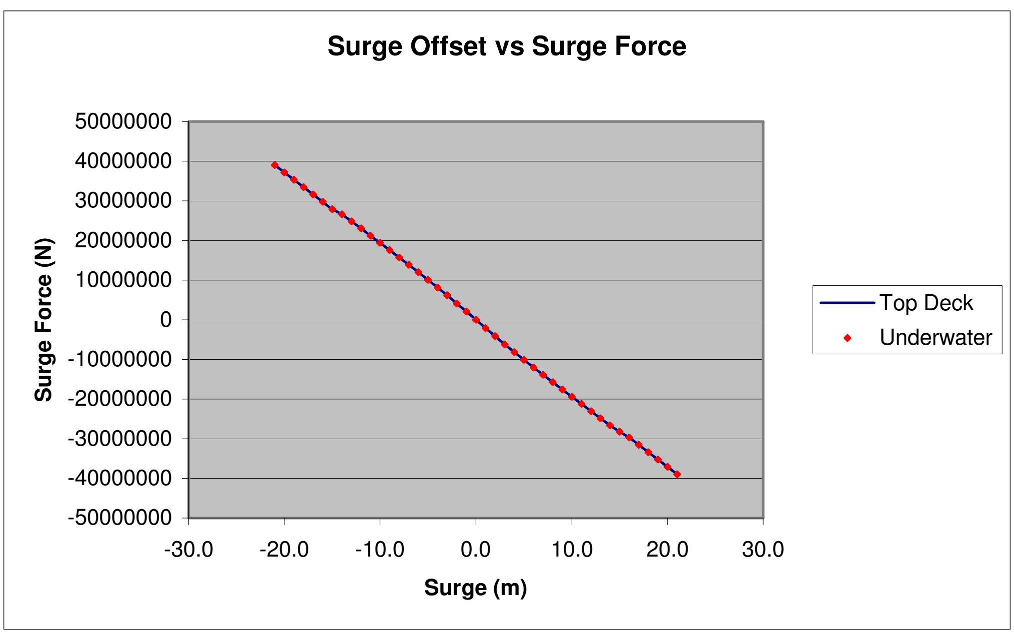

Figure 11: Surge versus Surge Force Curve showing effect of Fairlead Placement

The influence of fairlead placement on pitch response was of greater concern. Figure 12 shows a plot of pitch versus surge force for both cases. From this chart we are able to determine that the actual surge load resulting from a set pitch offset is approximately 13% lower for the on-deck fairlead case. The discrepancy was caused by the fact that the above-water fairleads have a smaller moment arm, being closer to the vessel's center of gravity. If measures were taken to ensure this distance was equal, then the magnitudes of pitch offsets and restoring pitch moments in this plot and the ones to follow would have a higher degree of agreement.

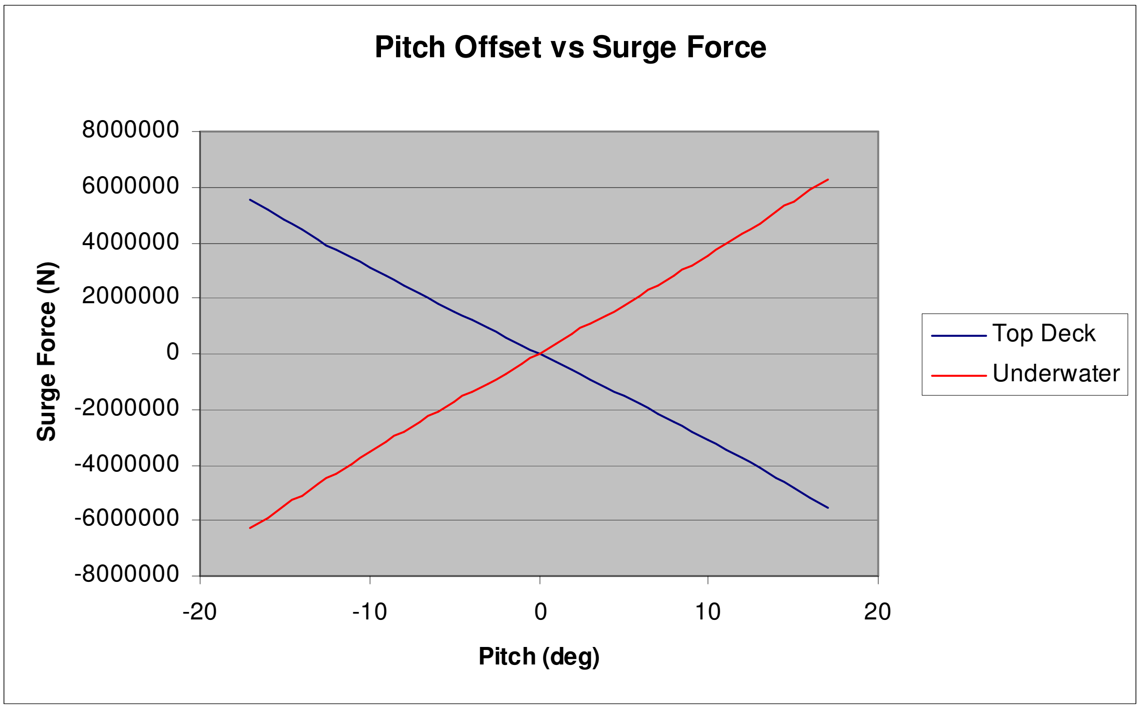

Figure 12: Pitch versus Surge Force Curve showing effect of Fairlead Placement

Figure 13 shows a plot of surge versus the restoring pitch moments. As expected, the moments are in opposite directions as the fairlead points are above and below the center of gravity, respectively. The error in the magnitude of the moment between the two configurations grows with greater displacement. For the 5% depth offset envelope, the on-deck fairlead case produces loads approximately 15% lower, while for the 10% depth offset envelope the loads are approximately 17% lower.

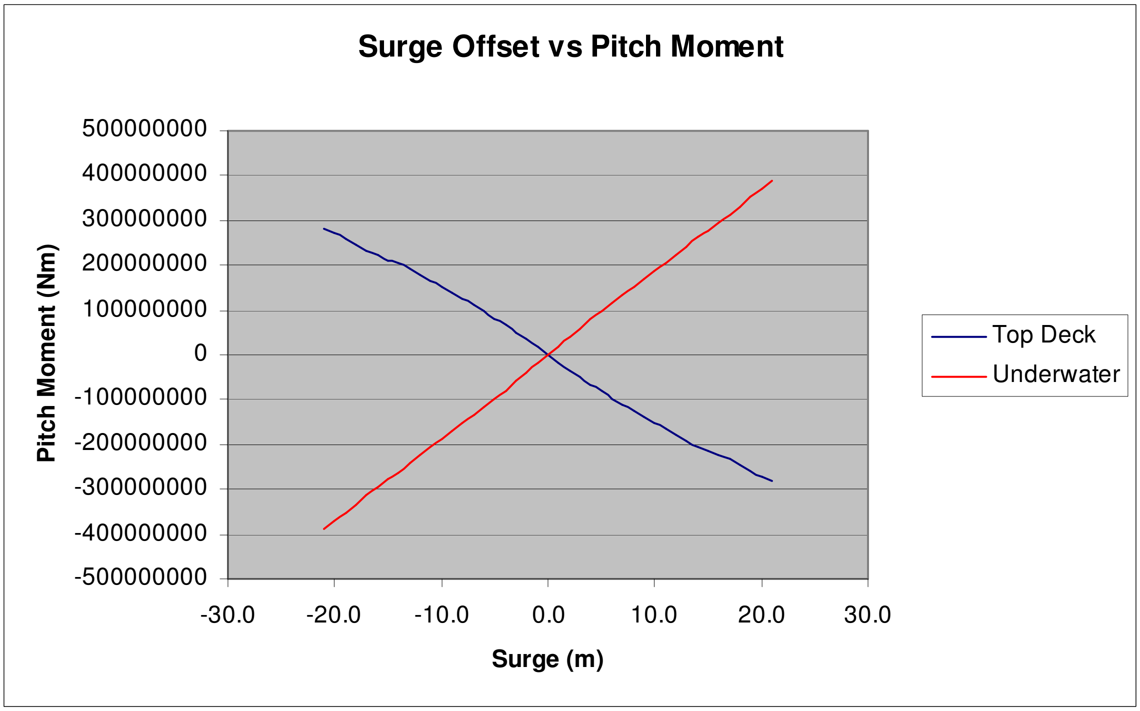

Figure 13: Surge versus Pitch Moment showing effect of Fairlead Placement

Figure 14 shows the relationship between pitch and restoring pitch moment for the two cases. At maximum, the on-deck fairleads case has a pitch moment approximately 20% lower than that for the underwater fairlead case.

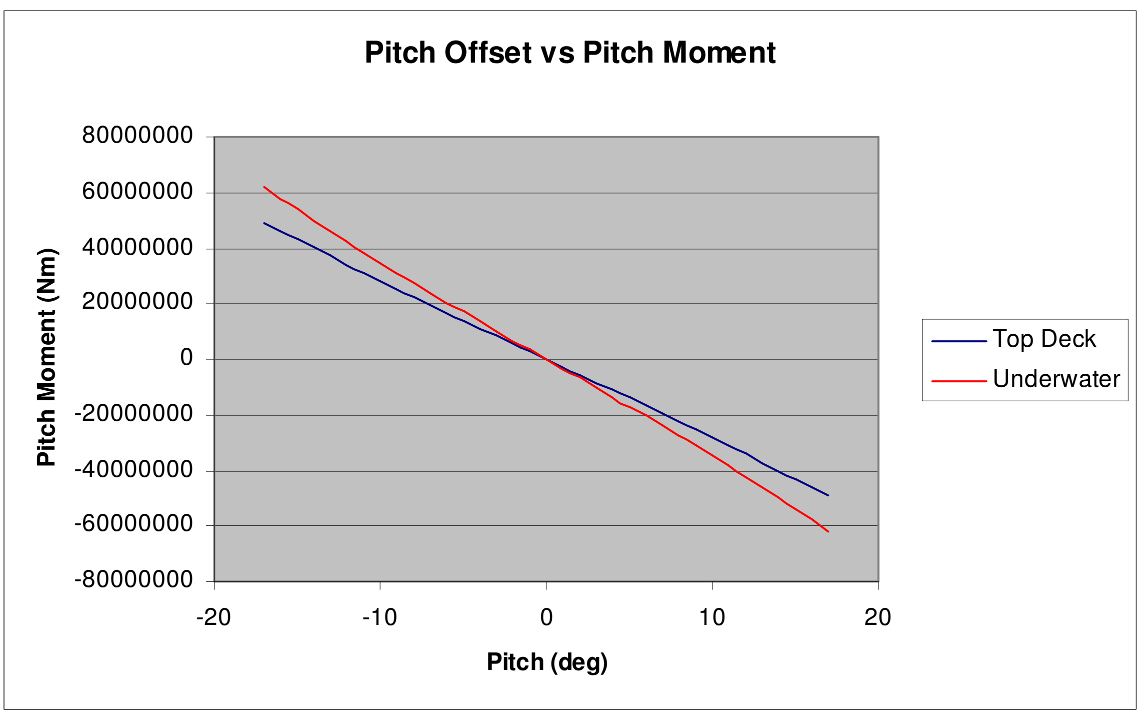

Figure 14: Pitch versus Pitch Moment Curve showing effect of Fairlead Placement

5.0 CONCLUSIONS & RECOMMANDATIONS

The work of preliminary design for the mooring system for the Phase 2 testing for the Ice Forces on Conical Structures project has been completed. The original design specifications have been met and the methods used to determine mooring system parameters have been outlined and verified as best as they could be with the available information. Through the analysis and research that was conducted, a number of ways to improve upon this design have been suggested:

- Effort should be made to acquire additional information on the experimental set-up of other experiments, and the stiffnesses used added to the stiffness matrix for the Phase 2 experiment.

- The stiffness matrix for the Phase 2 model experiments generated by the calculations presented in this report must be further verified as properly reflecting full-scale behavior. Should additional third-party information become available, stiffness characteristics relating to line length and water depth should be compared with those presented.

- The calculation for estimating water depth from line length could also be improved if information for additional mooring configurations is used. Further effort should be invested in acquiring more configuration data.

- Should further information about the precise location of the Kulluk's center of gravity become available, the mooring system simulation conducted with MOORING_SYSTEM should be repeated.

- Further investigation into changing the fairlead position on the model to lessen the discrepancy between simulated pitch response in the above-water and underwater cases for fairlead location should be made.

6.0 REFERENCES

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APPENDIX A

Kulluk Mooring Configuration

List of Tables

List of Figures

KULLUK Barge Report Belcher

Date: 05-Aug-89 Period: 0000-2359 MDT Lat/Long: 70 16.5 N / 141 30.8 W

ALERT @0400/06 Blue REASON: Fog & Ice HT:15 ST: 8 KT:-1 OT: 7

ENVIRONMENT as of 0400 06/08/89	MAX PAST 24 HRS	ACTUAL 0400	MAX NXT 24 HRS
Ice Conc (tenth):	8/10	8/10	9/10
Ice Thickness (m):	2.0	2.0	2.0
Ice Drift (knots):	0.4	0.4	0.4
Drift Dir. To (degT)	100	100	100
Ice Press.Potential:	No	No	No
Wind Speed (knots):	26	02	18
Wind Direction(degT)	290	140	220
Temp Min (degC):	6	8	10
Visib Min (n.miles):	0.75	0.75	0.5
Global Load (tonne):	101	71	150
Riser Angle (deg):	.56	.19	0.5
Offset (m):	.69	.23	0.75
Roll/Pitch (deg):	.73/.11	.71/.06	0.5
Wave Ht/Per (m/s):	n/a	n/a	n/a
Heave (m):	0	0	0

	miles	degT
Hazard/Ice:	4.0	280
Mvmt past24	5.19	087
Mvmt next24	7.5	090

Remarks/Outlook

Hazardous ice updrift is expected to be managed after moderate heavy ice breaking. Ice beyond 5 miles is easily managable.

(No gyro failures today, T6 senson dead again).

STABILITY DATA

Dspl:28403t	KM:45.8	KB:6.8	KG:14.0	GM:31.8	Dk/Ld:4369	Drft:11.4m	W/D: 52m

ANCHOR SYSTEMS

#	Wire Lgth	Wire Out	Pen	RAR #	Code	Pen	#1 Anc	MMS	F/A	Pen	Brng	#2 Anc	MMS	F/A	Pen	By
1	1130	895	20	171	IJOP	70	20SS	37	32		195					
2	950	859	20	172	IJLP	70	12S	12	32		225					
3	1130	942	20	160	IKNO	70	15M	29	-	70	256	15B	16	30	70	
4	1143	961	20	146	IKLN	70	15B	22	30	70	289	12B	21	30		
5	1000	905	20	143	IJNO	70	15S	9	32		315					
6	1130	1030	20	156	MNOP	70	15HS	40	32		348					
7	1130	928	20	169	IJMP	70	20SS	36	32		015					8
8	1130	909	20	174	IKMN	70	12S	10	32		045					3
9	985	891	20	153	IKLO	70	12HB	20	30	70	075	15L	32		70	2
10	1130	907	20	151	IJKM	70	15B	17	30	70	103	12B	14	30	70	7
11	1130	927	20	162	IJMN	70	15B	23	30	70	134	15B	19	30	70	
12	1130	858	20	154	IKNP	70	12S	13	32		165					

2nd piggybacks: #3 - 6 1/2 t trident MMS 35, #10 - 15t LWT MMS 25
#11 - 15t LWT MMS 26 #9 - 15t LWT MMS 24

Anchor Number:	1	2	3	4	5	6	7	8	9	10	11	12
Current Tension:	145	135	145	138	138	140	145	165	120	180	152	170
Peak Tension (24 hr):	150	150	169	195	150	140	170	185	195	200	198	197
Pre-tension:	205	195	190	185	195	205	210	200	195	200	190	200

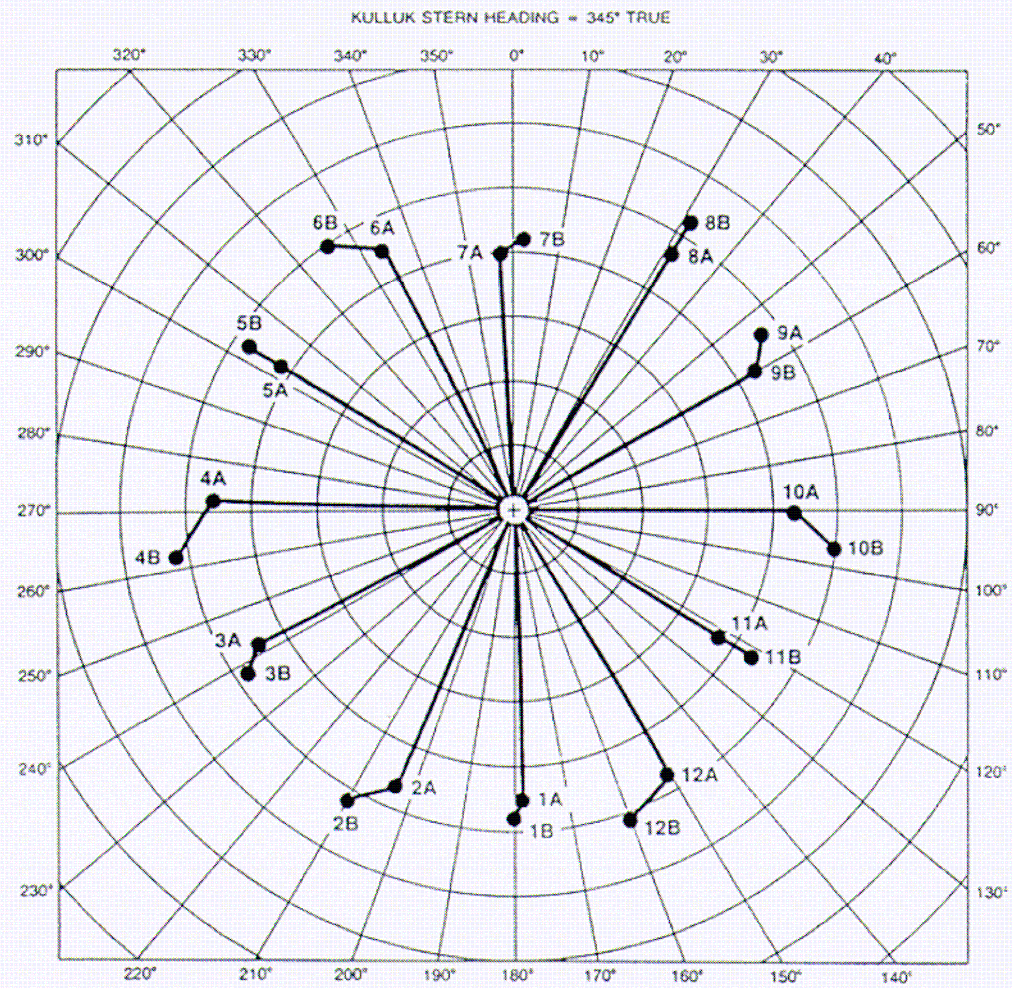
Table A1: Example of Kulluk Mooring Configuration (Source: Wright et al, 2000)

Kulluk Mooring System Characteristics

Location:	***Amauligak 2J - 44***
water depth	32m
number of lines	9
line diameter	*3 1/2"*
line type	wire rope
breaking strength	520 tonnes

Line Number	*Length (m)*	*Orientation (degrees true)*	*Pretension (tonnes)*	*Anchor(s)*
line 1	-	-	-	
line 2	639	164	195	9T Bruce, 9T Bruce
line 3	772	196	200	15T Bruce
line 4	-	-	-	
line 5	732	255	210	9T Bruce, 6.5T Bruce
line 6	530	285	60	15T Bruce
line 7	666	314	220	15T Bruce
line 8	-	-	-	
line 9	621	14	210	15T Bruce
line 10	601	45	180	15T Bruce
line 11	723	76	150	15T Bruce
line 12	584	106	70	15T Bruce, 9T Bruce

Table A2: Example of Kulluk Mooring Configuration (Source: Wright et al, 1999)



NOTE: STEV = STEVPRIS ANCHOR
BRU = BRUCE ANCHOR

DISTANCE (m)
0 200 400 600 800 1000 1200

#	TYPE	DISTANCE (m)	BEARING (°T)	#	TYPE	DISTANCE (m)	BEARING (°T)
1A	15T STEV	905	179	7A	15T STEV	806	358
1B	9T BRU	945	180	7B	9T BRU	848	001
2A	15T BRU	945	204	8A	15T BRU	940	031
2B	12T BRU	1029	210	8B	12T BRU	1046	031
3A	15T STEV	886	241	9A	15T STEV	854	060
3B	9T BRU	957	239	9B	9T BRU	973	054
4A	15T STEV	924	272	10A	15T BRU	887	090
4B	9T BRU	1010	262	10B	12T BRU	999	097
5A	15T STEV	821	303	11A	15T STEV	751	121
5B	12T BRU	957	303	11B	9T BRU	991	122
6A	15T STEV	913	332	12A	12T BRU	925	150
6B	12T BRU	995	326	12B	12T BRU	1002	155

KULLUK'S MOORING PATTERN AT AKPAK P-35
(09 JUNE TO 17 JUNE)

Figure A1: Example of Kulluk Mooring Configuration
(Source: Wright et al, 1999)

APPENDIX B

Kulluk and Experimental Information

List of Figures

List of Tables

BIBLIOGRAPHY

Summaries of Kulluk Full-Scale and Model Test Data

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Kulluk Model Data

Table B1: Kulluk Model Dimensions

	Platform model	Kulluk
Diameter at deck level (m)	2.025	81
Diameter at load waterline (m)	1.688	67.5
Diameter at hull bottom (m)	1.552	62
Depth (m)	0.46	18.4
Draft (m)	0.365	14.6
Displacement (m^3)	0.438	28000
Cone angle (°)	31.4	31.4

Figure B1: Kulluk Model

SUMMARY OF TEST DATA

Table B2: Summary of Model Tests with the Kulluk in Ambient Ice (Source: Comfort et al (1999))

Organiz. & Reference	Model Scale	Model Ice Material	Test Approach		
			Ice Pushed or Model Towed ?	Model Compliant or Fixed ?	Air Bubbler Included ?
ACL – Comfort et al, 1982	1:30	MOD-ICE	ice pushed	moored	no
	1:30	MOD-ICE	ice pushed	moored	yes
	1:30	MOD-ICE	ice pushed	Fixed	no
HSVA	1:45	Saline ice	model towed	moored	no
IIHR(1985)- Matsuishi et al, 1985 a,b	1:45	Urea ice	ice pushed	fixed	no
	1:45	Urea ice	ice pushed	Compliant- leaf spring used	no
IIHR(1988)- Nixon et al, 1988 a,b	1:45	Urea ice	ice pushed	leaf spring - 2 stiffnesses tested	no
	1:45	Urea ice	ice pushed	fixed	no

Table B3: Summary of Model Tests with the Kulluk in Managed Ice (Source: Comfort et al (1999))

Organiz. & Reference	Model Scale	Model Ice Material	Test Approach		
			Ice Pushed or Model Towed ?	Model Compliant or Fixed ?	Air Bubbler Included ?
ACL – Comfort et al, 1982	1:30	MOD-ICE	ice pushed	moored	no
	1:30	MOD-ICE	ice pushed	moored	yes
	1:30	MOD-ICE	ice pushed	Fixed	no
HSVA	1:45	Saline ice	model towed	moored	no
IIHR(1985)- Matsuishi et al, 1985 a,b	1:45	Urea ice	ice pushed	fixed	no
	1:45	Urea ice	ice pushed	Compliant- leaf spring used	no
IIHR(1988)- Nixon et al, 1988 a,b	1:45	Urea ice	model towed	leaf spring- 2 stiffnesses tested	no
	1:45	Urea ice	ice pushed	leaf spring- 2 stiffnesses tested	no

Legend:
ACL ARCTEC Canada Limited
AI ARCTEC Incorporated
HSVA Hamburg Ship Model Basin (German acronym)
IIHR Iowa Institute of Hydraulic Research
(associated with the University of Iowa)

Table B4: Summary of Model Test Ice Data
(Source: Comfort et al (2001))

Parameter Flex. Strength (kPa)	ACL tests 310-840	HSVA tests 560-720	1985 IIHR tests 1000-1100	1988 IIHR tests 800-1800
Density (kg/m^3)	911	910	no data[1]	no data[1]
Ice-model friction factors	dry: 0.15;0.11[2] wet: 0.14;0.12[2]	0.1	no data[1]	no dat a[1]
Ice-ice friction factors	0.79; 0.51[2]	surf-surf: 0.06 surf-bottom: 0.28	no data[1]	no data[1]

Notes:

1. Because the HSVA and the IIHR tests were both done using refrigerated ice, it is expected that the values for the IIHR tests would be reasonably similar t o those for the HSVA tests.
2. The two values are the static and the dynamic friction factors, respectively.

Table B5: Summary of Base Cases Used for Analysis in Kulluk Model Tests in Ambient Ice (Source: Comfort et al (1999))

Organiz-see Table 3.1 for references	Model Scale	Model Ice Material	Test Approach		
			Ice Pushed or Model Towed ?	Model Compliant or Fixed ?	Air Bubbler Included ?
ACL	1:30	MOD-ICE	ice pushed	moored	no
HSVA	1:45	Saline ice	model towed	moored	no
IIHR(1985)	1:45	Urea ice	ice pushed	compliant – 2.8 MN/m stiffness	no
IIHR(1988)	1:45	Urea ice	ice pushed	compliant – 2.8 MN/m stiffness	no

Table B6: Summary of Base Cases Used for Analysis in Kulluk Model Tests in Managed Ice (Source: Comfort et al (1999))

Organiz-see Table 3.2 for references	Model Scale	Model Ice Material	Test Approach		
			Ice Pushed or Model Towed ?	Model Compliant or Fixed ?	Air Bubbler Included ?
ACL	1:30	MOD-ICE	ice pushed	moored	no
HSVA	1:45	Saline ice	model towed	moored	no
IIHR(1985)	1:45	Urea ice	ice pushed	compliant – 3.4 MN/m stiffness	no
IIHR(1988)	1:45	Urea ice	model towed	compliant -1.0 MN/m stiffness	no
	1:45	Urea ice	ice pushed	compliant -1.0 MN/m stiffness	no

Table B7: Summary of Experimental Information From Other Sources

Source	Author(s)	Mooring Info	Ice Info	Speed Info
The Dynamic Behavior of a Floating Cable-Moored Platform Continuously Impacted by Ice Floes (1985)	M. Matsuishi and R. Ettema	Horizontal Leaf Spring System Equivalent Cable Stiffness: 1.7 kN/m Stiffness for Foundation Reaction: 17.3 kN/m Pitch stiffness = 35.1 Nm/degree Note: Used Leaf Spring, no mooring lines	Flexural Strength: 16.6-24.4kPa Thickness: 29-32mm Elastic Modulus: 8.2 - 14.8 More detail pg. 101	Ice Pushed Fixed: 0.04, 0.06, 0.08, 0.12 m/s Moored: 0.01, 0.02, 0.04, 0.08, 0.12 m/s
Source	**Author(s)**	**Mooring Info**	**Ice Info**	**Speed Info**
Model Study of a Cable-Moored Platform in Sheet Ice (1989)	Nixon, Ettema, Matsuishi, Johnson	1988 IIHR Stiffness = 0.5 kN/m (Leaf Spring) Stiffness for Foundation Reaction: 17.3 kN/m Pitch stiffness = 35.1 Nm/degree		