PMM replacement project

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Laboratory Memorandum

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P. Thorburn

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PMM REPLACEMENT PROJECT

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SUMMARY

Planar motion mechanism (PMM) test techniques are used to determine hydrodynamic coefficients for mathematical models of ship manoeuvring [1]. An IOT PMM [2] was commissioned in 1996 and has been used successfully in a number of tank tests since then. The 1996 version of reference [2] was released when the PMM was commissioned. Since then, the apparatus has undergone many changes. The 2005 version describes the current PMM control system. The first part is a user manual and the second, a programmer's guide. The PMM control has six built-in standard tests: static tests (model yawed), sway only, sway and yaw coupled (pure yaw), coupled motion with static yaw, surge, and turning circles (constant yaw rate).

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PMM REPLACEMENT PROJECT

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1.0 INTRODUCTION

Planar motion mechanism (PMM) test techniques are used to determine hydrodynamic coefficients for mathematical models of ship manoeuvring [1]. An IOT PMM [2] was commissioned in 1996 and has been used successfully in a number of tank tests since then. The 1996 version of reference [2] was released when the PMM was commissioned. Since then, the apparatus has undergone many changes. The 2005 version describes the current PMM control system. The first part is a user manual and the second, a programmer's guide. The PMM control has six built-in standard tests: static tests (model yawed), sway only, sway and yaw coupled (pure yaw), coupled motion with static yaw, surge, and turning circles (constant yaw rate).

In September 2004, a small group began to investigate PMM use and capability at IOT, with the main objective of enhancing the institute's capability in this area. Group members are Paul Thorburn (Chair), Christopher Williams, David Molyneux, Michael Sullivan, Michael Lau, Tony Randell and Don Spencer (Oceanic). Initial objectives were:

- 1.1 determine requirements for a PMM
- 1.2 produce a concept design to meet the requirements
- 1.3 provide a cost estimate for recommendations. Recommendations may include improvements the present PMM and/or a proposal for a new PMM

A previous investigation in 2002 ended when there was no budget available to pursue improvements to the IOT device. A series of meetings began on September 28, 2004 and is continuing. Discussions, actions and results to date are outlined in this report. All information related to this project is available on an IOT computer network folder (pccommon\PMM).

2.0 DISCUSSION SUMMARY FROM MEETINGS, SEPTEMBER 2004 TO JUNE 2005

2.1 Research and commercial testing needs are the same. Open water tests using the PMM have been satisfactory; tests in ice with higher loads have had more difficulties. IOT is the only known user of a PMM in ice.

2.2 Other PMM installations

- 2.2.1 FORCE Technology, Danish Maritime Institute, Denmark; Shallow Water Manoeuvring Basin (25x8x0.8m) and a Deep Water Towing Tank (240x12x5.5m) which are equipped with a PMM (for captive ship manoeuvring tests in shallow and deep water) and a VPMM (a vertical PMM for measuring added-mass and damping effects for floating and submerged bodies), www.force.dk
- 2.2.2 HSVA, Germany; refer to Appendix A, HSVA PMM CPMC Information
- 2.2.3 MARIN (2), Holland
- 2.2.4 SSPA, Sweden has an X-Y carriage on a large (88m x 39 m) basin

2.2.5 NSTL, India has a device supplied by Cussons
2.2.6 Samsung and KORDI, Korea; refer to Appendix B, KRISO PMM Specifications

2.3 Components and considerations for a PMM device

2.3.1 mechanical frame; stiffness is an important factor for operation in ice and for potential yacht testing
2.3.2 attachment methods for Ice Tank and Tow Tank carriages
2.3.3 control system for test motions
2.3.4 instrumentation; modular so that measuring devices such as load cells and dynamometers can be readily changed to match expected loads
2.3.5 calibration methods and devices
2.3.6 software simulations for load prediction
2.3.7 data acquisition
2.3.8 data analysis to produce coefficients

2.4 Research Questionnaire; a questionnaire was prepared and distributed to IOT researchers by Christopher Williams in October 2004. This questionnaire is attached as Appendix C, PMM Questionnaire. No returns were received.

2.5 Yacht testing using a PMM; requirements for tacking simulation include freedom to roll (list), heeling at a substantial angle (> 35º), with high moments. Speed decreases as tack direction changes, then increases. Active rudder with tow force through the mast may be a possibility. Stiffness of the apparatus may be a problem. Present sway velocity is too low for yacht testing. Minimum required is 1.0 m/s and maximum, 1.35 m/s. Loads are all adequate for yachts.

2.6 Load prediction; Michael Lau is using Ice Tank test results on the Terry Fox to validate DECICE software simulations. If valid, then DECICE can be used to predict loads to determine if a proposed test is within the capability of the PMM, and the load cell range required.

2.7 Finite Element Analysis (FEA); at least three FEA studies have been done on the PMM. Refer to Appendix D, PMM Dynamometer and Frame Finite Element Analysis. Ahmed Derradji modeled the PMM to Ice Tank Carriage connection in 2004 and concluded that there are rigidity and vibration issues. There is limited transfer of load from the PMM to the carriage frame as it is not stiff enough for the application. Mark Dawe studied the PMM dynamometer in January 2005 [3]. Flex links in the dynamometer may also limit stiffness.

3.0 DESIGN AND SPECIFICATION

3.1 From January to April 2005, an engineering work term student in the Design and Fabrication Group was assigned the tasks of assembling performance requirements for a PMM, preparing a specification based on these requirements, and preparing a detailed plan for refurbishment of the present IOT PMM [3]. In addition, refer to Appendix E, PMM Performance Capability. The Design and Fabrication Group is now reviewing this design, the PMM to Ice Tank Carriage attachment method, and the dynamometer. A possible option is to have two dynamometers; one for high range loads and one for low range. As well, modular design will permit easier exchange of measuring components in the dynamometers as required.

3.2 The objectives of present design activities are to produce the technical and financial information required to prepare a proposal for an extensive refit of the present PMM. As most of this work would be done by an outside supplier, sufficient detail for a bidding process is needed. A second option is a completely new PMM. Information required is similar to that needed for the refit with detailed description, specifications and concept design to support a bidding process.

4.0 REVISED PROJECT OBJECTIVES

4.1 The PMM Replacement Project 42_2079_10 was revised in May 2005 to include interim measures that could be done to improve the reliability and performance of the present PMM. As the present device is likely to be in service until at least 2008, this work is worthwhile. Two tasks have been identified.

4.2 The first interim task is replacement of the PMM computer with a newer, more reliable model and an upgrade of its operating system from Windows 95 to Windows 98. This upgrade to a 32 bit operating system will permit use of new components of the Compumotor control software. The present front end program will be separated into two sections, one for user interface and motion definition, and one for motion control. The PMM operational computer will only be used to control the device and not for development or modification of software.

4.3 The second interim task is to study the PMM to Ice Tank Carriage connection method and design improvements that can be implemented in the short term.

4.4 There are also PMM tests with the Terry Fox icebreaker model planned for the Ice Tank in July 2005. These tests are a repeat of those done in Project PJ953 and will address issues of model heading control and non-repeatability of results from the original set.

4.5 Schedule; the interim tasks of improving the computer hardware, operating system and PMM software and procedures for managing software changes should be complete by September 2005. Study of the attachment method for the Ice Tank carriage should be complete by July 2005. Implementation of changes in the attachment method will depend on the outcome of the study. Proposals for refurbishment of the present PMM and for a complete new device should be complete by October 2005.

5.0 NEXT PHASE

Discussions with IOT staff involved in PMM installation and operation will be held with the objective of identifying other interim measures that could improve operations with the present PMM. Description, concept design and specifications will be submitted to potential suppliers to obtain cost estimates. A proposal for consideration by IOT management should be ready by early November 2005.

6.0 CONCLUSIONS

As the present PMM will remain in service for another three to four years, any improvements that can be done with reasonable effort and expense are worthwhile. The most pressing are stabilizing the control software and implementing version control, and making the Ice Tank carriage to PMM connection more rigid. There may be other desirable changes identified as the study continues, including documentation of operational procedures.

Suppliers with experience in PMM design and fabrication are few. It may be possible to contract with local mechanical companies to develop a new PMM. Detailed specifications for all aspects of capability, design and operational procedures will be required for this approach.

7.0 REFERENCES

[1] International Towing Tank Conference, 2002, Recommended Procedures, Testing and Extrapolation Methods, Manoeuvrability, Captive Model Test Procedure
[2] Spencer, Donald, 2005, Planar Motion Mechanism Controller User Guide, Oceanic Consulting Corporation, INT099-02
[3] Dawe, Mark, 2005, PMM Refurbishment Plan, National Research Council, SR-2005-05

Appendix A

HSVA PMM CPMC Information

HSVA PMM CPMC Information (from www.hsva.de), 11 March 2005, C. D. Williams

Figure 1: PMM in HSVA Towing Tank. CPMC is computerized planar motion carriage.

CPMC

maximum speed, longitudinal 3.0 m/s

maximum speed, transverse 1.9 m/s

maximum yaw rate 24^{o}/s

Manoeuvring Experiments

Manoeuvring tests are performed in HSVA’s large towing tank using the Computerized Planar Motion Carriage (CPMC), which provides the basis for superior predictions of the manoeuvring and course keeping qualities of surface ships. The CPMC has two fundamentally different operating modes, which enable a wide range of services and research activities. In both, the captive and the tracking mode each run is completely computer controlled from model stand-still to stand-still.

In the towing mode it guides a captive ship model, which is usually free to sink and trim, along a predetermined curvilinear path at predetermined speed while the resulting hydrodynamic forces are measured as functions of time. This mode is especially useful for research purposes and for a validation of CFD results, as well as for the direct determination of hydrodynamic coefficients of manoeuvring equations of motion, like added masses and damping functions.

In the tracking mode the CPMC automatically tracks a model manoeuvring freely under the action of a temporally and/or spatially predetermined sequence of rudder and propeller manoeuvres. This mode is especially useful for directly analysing the manoeuvrability of ships, as well as for an indirect determination of hydrodynamic coefficients. These coefficients allow an accurate simulation of several rudder manoeuvres as: turning circles, spiral and pull out tests.
Also special simulation algorithms are available at HSVA, which enable to calculate the temporal course of typical engine manoeuvres, like acceleration turns, coasting and stopping manoeuvres.

Appendix B

KRISO PMM Specifications

KRISO PMM Specifications

Built by Mitsui (Japanese Company that also built IOT carriages).

Strongback	
Length & Quantity	1.5 m 1 set
	4.0 m 1 set
Gimbal for model attachment	
Roll	Free(up to ± 30 deg.)
Heave	Free(up to ± 100 mm)
Heel	Locked desirable angle
Measurement System	Rotary encoder(100 pulses/rotation
Measurement accuracy	±0.01 deg. (360 deg./36000 pulses)
Measurement display	4 - digit digital meter
Recording output	Max. ±10 Vpp, 10 mA
Sway motor	
Type	Low inertia DC servo motor
Cooling system	Totally enclosed
Rating output	3080 W, 100 kg.cm, 3000 rpm
Sway control	
Amplitude setting	3- digit digital switch
Setting range	±0.0 ~ ±1.5 m
Control system	Thyristor Leonard control
Control accuracy	±1 mm
Measurement System	Rotary encoder (1000 pulses/ rotation)
Measurement accuracy	±0.1 mm (3000 mm/3000 pulses)
Measurement display	4 - digit digital meter
Recording output	Max. ±10 Vpp, 10 mA

Function generator

Frequency setting	Analog digital switch
Setting range	0.01 ～ 10 Hz
Function	$\sin\omega t$, $\cos\omega t$, $\sin(\omega t+\alpha)$
Accuracy	±1.4 %

Yaw motor

Type	Low inertia DC servo motor
Cooling system	Totally enclosed
Rating output	3080 W, 100 kg•cm, 3000 rpm

Yaw control

Amplitude setting	3- digit digital switch
Setting range	±0.0 ～ ±40.0 deg.
Control system	Thyristor Leonard control
Control accuracy	±0.1 deg.

Load cell

Capacity and quantity	20 kg 1 set (for X-force) 100 kg 1 set (for X-force) 200 kg 2 sets(for Y-force) 50 kg 2 sets(for Y-force)
Output	2 mV/V ±0.15 %
Linearity	0.05 % FS
Hysteresis	0.05 % FS
Zero drift	0.005 % FS/°C
Stability	0.005 % Reading/°C
Temperature compensation	-10 ～ +75 °c
Allowable overload	500 % (Capacity : 100 kg, 50 kg, 20 kg) 150 % (Capacity : 200 kg)

Drift angle setting

Setting	Mechanical setting (manual)
Setting range	0 ~ 360 deg.
Setting increment	0.1 deg.

Elevation setting

Setting	Mechanical setting (a motor drive)
Setting range	600 mm
Setting increment	1 mm

Appendix C

IOT PMM Questionnaire

IOT PMM Questionnaire 26 October 2005 C. D. Williams

In preparation for the meeting next Tue 26 Oct at 11:00, please send me your replies to the following questions that were raised at the last meeting. If everyone contributes then we can build up a list of requirements for a PMM. If you wish to provide us with information for several different types of devices, tests, etc. please just copy and paste the same set of questions each time. A sample response is provided below following the questions. If the questions don't make sense for your application, please provide some requirements that do match your application.

Thanks,
Chris

For which type of device are you providing the specifications below?
[ ] Surface ship, open water
[ ] Submerged vehicle, open water
[ ] Surface ship in ice
[ ] Oscillating foil
[ ] Other

Supplementary questions:
At what point within the model are these loads to be measured? ____________________
At what point within the model are these loads to be reported? ____________________
Typical number of data acquisition system channels required ______________
Any special analysis programming required? If 'yes', please specify
____

For a sinusoidal pure-sway motion, what is a typical
Amplitude of the sway component of the motion ______________
Period of the sway component of the motion ______________
Maximum sway rate ______________________
Maximum drift angle ______________
Carriage speed ________________
Range of loads to be measured (FX, FY, FZ, MX, MY, MZ) ______________________

For a sinusoidal pure-yaw motion, what is a typical
Amplitude of the sway component of the motion ______________
Period of the sway component of the motion ______________
Amplitude of the yaw component of the motion ______________
Period of the yaw component of the motion ______________
Maximum yaw rate ______________________
Maximum drift angle ______________
Carriage speed ________________
Range of loads to be measured (FX, FY, FZ, MX, MY, MZ) ______________________

For an arc of a circle, what is a typical

Radius of the arc _______________
Length of arc traversed _____________
Time to complete the arc _____________
Carriage speed _______________
Range of loads to be measured (FX, FY, FZ, MX, MY, MZ) ______________________
At what point within the model are these loads to be measured? ____________________
At what point within the model are these loads to be reported? ____________________
Typical number of data acquisition system channels required _____________
Any special analysis programming required? If 'yes', please specify

____

<u>For a straight line run with constant yaw angle, what is a typical</u>
Range of yaw angles _________________
Carriage speed __________________
Range of loads to be measured (FX, FY, FZ, MX, MY, MZ) ______________________
At what point within the model are these loads to be measured? ____________________
At what point within the model are these loads to be reported? ____________________
Typical number of data acquisition system channels required _____________
Any special analysis programming required? If 'yes', please specify

____

<u>For a controlled zig-zag motion, what is a typical</u>
Range of heading angles _______________________
Range of path widths _____________________
Range of path lengths _______________________
Range of yaw rates ______________________
Carriage speed ________________
Range of loads to be measured (FX, FY, FZ, MX, MY, MZ)

____

Any other considerations that were not mentioned above that will affect the requirements for a PMM?

____

-------------------------- sample --------------------------
For which type of device are you providing the specifications below?
[x] Submerged vehicle, open water

<u>For a sinusoidal pure-sway motion, what is a typical</u>
Amplitude of the sway component of the motion: 0.4 to 4 m
Period of the sway component of the motion: 5 to 50 sec
Maximum sway rate: 0.12 to 0.5 m/s
Maximum drift angle: always zero
Carriage speed: 2 m/s
Range of loads to be measured (FX, FY, FZ, MX, MY, MZ): FX about 52 N; FY 35 to 210 N; FZ approx zero; MX 0.4 to 2.6 N.m, MY approx. zero; MZ 25 to 100 N.m

For a sinusoidal pure-yaw motion, what is a typical
Amplitude of the sway component of the motion: 0.4 to 4 m
Period of the sway component of the motion: 5 to 50 sec
Amplitude of the yaw component of the motion: +/-14 deg
Period of the yaw component of the motion: 5 to 50 sec
Range of yaw rates: 1.8 to 18 deg/sec
Maximum drift angle: always zero
Carriage speed: 2 m/s
Range of loads to be measured (FX, FY, FZ, MX, MY, MZ) FX is about 52 N; FY is 10 to 157 N; FZ approx zero; MX 0.1 to 2 N.m, MY approx. zero; MZ 8 to 81 N.m

For an arc of a circle, what is a typical
Radius of the arc: 6 to 45 m
Length of arc traversed: 38 to 67 m
Time to complete the arc: 19 to 34 sec
Carriage speed: zero to 2 m/s
Turning rate: 2 to 18 deg/sec
Drift angle: 3 to 18 deg
Range of loads to be measured (FX, FY, FZ, MX, MY, MZ): FX about 52 N; FY 22 to 160 N; FZ approx. zero; MX 0.3 to 2 N.m; MY approx. zero; MZ 16 to 77 N.m

For a straight line run with constant yaw angle, what is a typical
Range of yaw angles: 0 to 90 deg
Carriage speed: 2 m/s
Range of loads to be measured (FX, FY, FZ, MX, MY, MZ): FX zero to 50 N; FY zero to 500 N at 30 deg; FZ approx. zero; MX zero to 6.2 N.m; MY approx. zero; MZ zero to 187 N.m at 45 deg.

For a controlled zig-zag motion, what is a typical
Range of heading angles: zero to 82 deg
Range of path widths: 3 to 8.8 m
Range of path lengths: 35 to 40 m
Range of yaw rates: 5.2 to 13 deg/sec
Carriage speed: 2 m/s
Range of loads to be measured (FX, FY, FZ, MX, MY, MZ): FX about 52 N; FY 42 to 120 N; FZ approx. zero; MX 0.5 to 1.5 N.m; MY approx. zero; MZ 30 to 64 N.m
------------------ end of sample ---------------------

Are there any other considerations that were not mentioned above that will affect the requirements for a PMM?

Carriage Speed [m/s]	2	2	2	2	2	2
	FX	**FY**	**FZ**	**MX**	**MY**	**MZ**
Manoeuvre	[N]	[N]	[N]	[N.m]	[N.m]	[N.m]
Turning circles	52	22 to 160		0.3 to 2		16 to 77
Zig-zags	52	42 to 120		0.5 to 1.5		30 to 64
Pure sway (max 0.5 m/s)	52	35 to 210		0.4 to 2.6		26 to 100
Pure yaw (max 18 deg/sec)	52	10 to 157		0.1 to 2		8 to 81
Yaw wiggle (max 20 deg)	52	23 to 325		0.3 to 4		17 to 128
Straight-ahead towing with the aft diveplanes deflected (max fin angle 30 deg; max fin lift force at 15 deg)	52		0 to 123	n/a	0 to 94	
Straight-line towing with hull at static yaw (drift) angle (max angle is 90 deg; max load occurs at 30 deg)	0 to 52	0 to 500		0 to 6.2		0 to 187
Overall range	0 to 52	0 to 500	0 to 123	0 to 6.2	0 to 94	0 to 187

Table 1. Estimated loads on C-SCOUT during various PMM manoeuvres

Appendix D

PMM Dynamometer and Frame Finite Element Analysis

PMM Dynamometer and Frame Finite Element Analysis John Bell 24 July 2002

In June of this year the PMM dynamometer was fitted with an aft vertical flexible link in place of the existing pin in a hole and a two axis frictionless table. This meant that the dynamometer was completely fitted out with flexible links for six degrees of freedom.

Load cells were not fitted for measuring roll. Thus only three load cells were present, one in drag and two in sway. The drag load cell was a 100 lbs capacity Interface SSB and the sway load cells were 500 lb capacity SSB's. All of the flexible links except the drag link were 7075 T6 aluminum links with necked down sections measuring .100/.105" in diameter with lengths to suit their location. The drag link was a .06" dia. 17-4 PH in the H900 condition and had a greater capacity than the loadcell.

To confirm the capacity of the dynamometer in this configuration for use in upcoming tests a finite element analysis was completed for the dynamometer.

A second analysis was completed for the PMM Frame with a simplified model of the dynamometer and an attached ship model. This second analysis was carried out to investigate possible weakness in the PMM mounting frame.

PMM Dynamometer Analysis

Model

See the attached figure.

The PMM dynamometer was largely made up of standard structural shapes welded together to form two frames.

The upper frame was mounted to a heavy steel tube supported by 4" diameter flanged ball bearing units.

The lower frame was attached to the upper frame through six flexible links and three load cells.

A third frame attaches to the lower frame of the dynamometer and then to the model through linear bearings. This frame allowed the model to heave, pitch and roll freely.

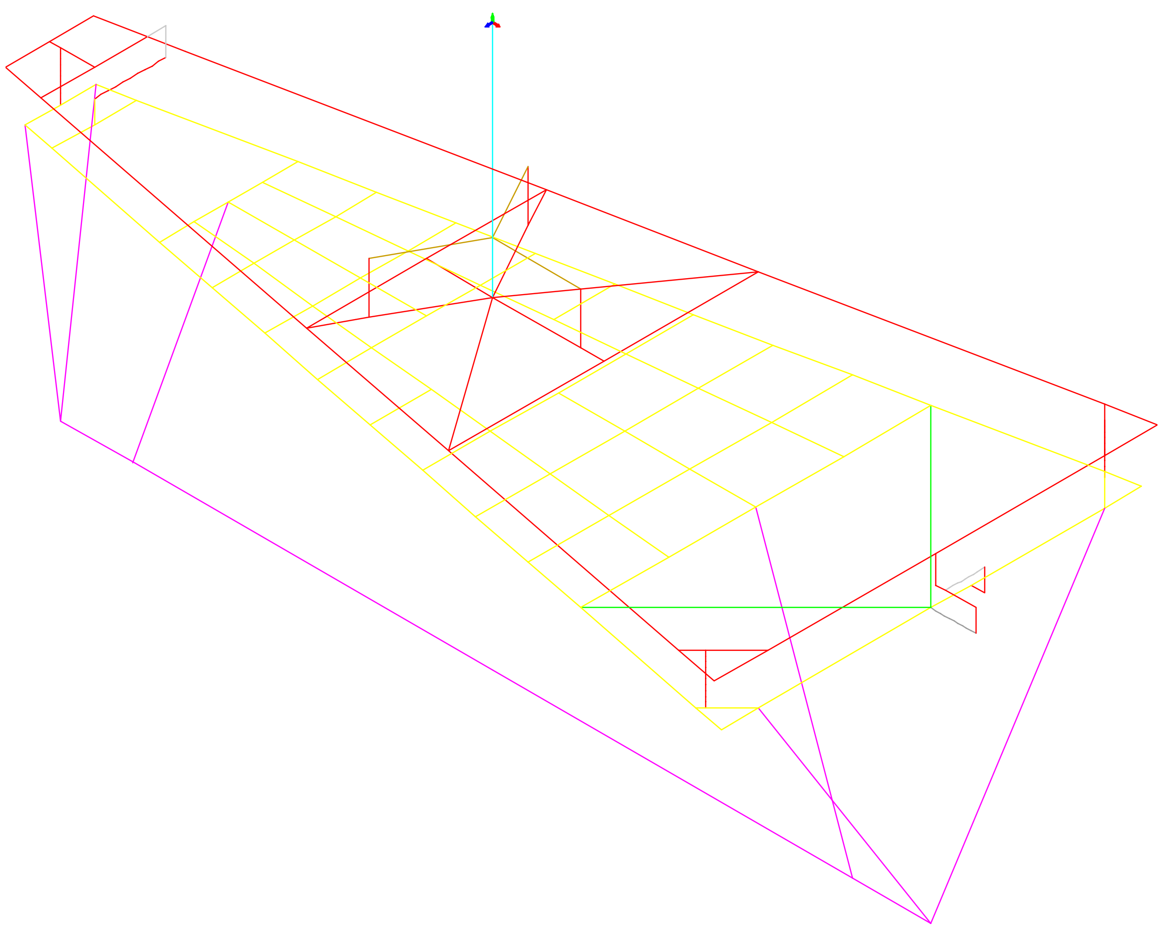
These three frames and the flexible link – load cell combinations were modeled in Algor using beam elements. The free to pitch and heave condition was ignored because no analysis was to be done with forces applied vertically.

A ship model of 400 kg displacement and length overall of 4 m was represented by a single beam element with the appropriate mass. The natural frequency modes were not affected by heave or pitch of the model. Therefore modeling of the displacement of the model was not required. An average position of 24 inches below the lower dynamometer frame was assumed for the model and its associated forces.

The dynamometer model was grounded where the 4” diameter bearings support the main vertical tube. In reality this was not a perfect ground and allowance was made in the results to take into account the reduction in natural frequency which would result.

The load cells were modeled as small cantilever beams having the same spring constant in bending as the real load cell. The load cell spring constants were evaluated using the stated capacity and deflection from the manufactures data sheets. A round steel rod having the same spring constant in bending was then calculated and the properties were assigned to the load cell beam element.

Element Model



The element model was created in Cadkey . The original wire frame part was used as a template to guide in the placement of center lines for all of the major structural elements. This grid of lines was then imported into Algor where it was separated out into group levels so that individual properties and material types could be assigned.

The pipe frame used to join the model to the lower dynamometer frame was representative only.

The ¼" aluminum plate on the lower frame was modeled by a grid of ¼ x 3 flat bars. Plate elements in this situation would not have provided correct results because the forces from the model would be predominantly normal to the plane of the elements and thus not supported in the analysis. Beam elements were modeled because they supported forces in three directions at each node and would therefore be supported during the analysis.

BEDIT Preprocessor

To allow for future use of this fea model the properties assigned to each group in the bedit preprocessor are listed here. The listed order is not meant to convey orientation.

Dynamometer Mounting Tube
Group 1 4" OD x 2.5" ID Steel Pipe

A – 7.66	S – 3.83	J – 21.3	I – 10.65	Z – 5.32

Upper Dynamometer Frame
Group 2 4" x 5.4 lbs/ft Steel Channel

A- 1.562	S - .74	J - .045	I – 3.82/.379	Z – 1.91/.354

Lower Dynamometer Aluminum Plate
Group 3 ¼" x 3" Aluminum Flat Bar

A - .75	S - .625	J - .015	I - .563/.004	Z - .375/.031

Lower Dynamometer Frame
Group 4 2 x 4 x ¼" Aluminum Channel

A – 1.875	S - .833	J - .059	I – 4.41/.695	Z – 2.21/.495

Ship Model to Lower Frame Connection Elements
Group 5 1.315 OD x 1.049" ID Aluminum Pipe

A – .489	S – .244	J – .172	I – .086	Z – .131

Ship Model
Group 8 3.5" dia. Aluminum Bar

A – 9.62	S – 8.66	J – 14.73	I – 7.37	Z – 4.21

Gear Webs
Group 9 3/4” x 3” Steel Flat Bar

A – 2.25	S – 1.875	J - .357	I – 1.69/.105	Z – 1.125/.281

Gear Bolts
Group 10 .5” dia. Steel Bar

A – .196	S – .176	J – .006	I – .003	Z – .012

Flexible Link Body
Group 11 3/4” dia. Aluminum Bar

A – .442	S – .4	J – .031	I – .016	Z – .041

Flexible Link Necked Section
Group 12 .1” dia. Aluminum Bar

A – .0079	S – .007	J – 9.82e-6	I – 4.91e-6	Z – 9.82e-5

Drag Load Cell Model
Group 13 .461” dia. Steel Bar

A – .167	S – .15	J – .0044	I – .0022	Z – .01

Aft Side Force Load Cell Model
Group 14 .65” dia. Steel Bar

A – .332	S – .3	J – .0175	I – .0087	Z – .027

Frwd Side Force Load Cell Support
Group 15 7/16 x 5/8 Aluminum Bar

A – .273	S – .228	J – .011	I – .0089/0044	Z – .0285/.02

Frwd Side Force Load Cell Model
Group 16 .548” dia. Steel Bar

A – .236	S – .212	J – .0089	I – .0044	Z – .016

<u>Analysis Results</u>

Two types of analysis were carried out on the dynamometer model. Linear elastic stress analysis and natural frequency mode shape analysis.

The linear elastic stress analysis assumed nominal forces. A drag force of 100 lbs, a sway force of 100 lbs and a sway moment of 1000 ft lbs respectively were applied.

Displacement Results

Yaw moment applied – maximum displacement of the ship model was .3" which corresponded to an yaw angle of .26 degrees.

Drag force applied - maximum displacement of the ship model was .06".

Yaw force applied - maximum displacement of the ship model was .188". This mode involves twisting of the narrow aft end of the model half of the dynamometer.

Stress Results

Yaw moment applied – maximum stress showed up in the side force flexible links, 25,000 psi. Higher than normal stresses were also observed in the bolted connection to the 4" diameter shaft, 10,000 psi.

Drag force applied – maximum stress showed up in the drag flexible link and mounting, 20,000 psi.

Sway force applied – maximum stress showed up in the roll flexible links, 13,500 psi.

Natural Frequency Results

The results for the natural frequency analysis had good agreement with hand calculations. The lowest frequency was a combination of yaw and sway with Algor giving a 9.2 Hrz result. This should be lowered by at least 10% because of the boundary condition assumptions, giving a prediction of about 8 Hrz.

Maximum Load Capability

Given the present load cells and flexible links and the load plane 24" below the dynamometer lower frame.

Yaw moment – The maximum stress is showing up in the forward side force flexible link. The stress shown is 25 kpsi. The flexible link material yields at 60 kpsi. Therefore the yield yaw moment is 60/25 x 1000 = 2400 ft lbs. The ½" diameter bolts and webs for the yaw gear are showing a stress of 11 kpsi. The gear material will yield at

35 kpsi therefore an increase in yaw moment capacity above 2400 ft-lbs. would have to be accompanied by a redesign of the dynamometer to 4” shaft connection.

Drag force – The maximum stress is showing up in two places – the drag flexible link and the forward vertical flexible link with both showing 20 kpsi. The present drag load cell has a name plate capacity of 100 lbs and this would be the present limit of the dynamometer. The flexible links would yield at 60/20 x 100 = 300 lbs of drag if a larger drag load cell were substituted.

Sway force – The maximum stress is showing up in the roll and forward side force flexible links simultaneously. All three show a stress of 14 kpsi. The side force that will cause yielding of the links is therefore 60/14 x 100 = 430 lbs. To reach the load cell capacities of 500 lbs each would require new flexible links.

Conclusions / Recommendations For the Dynamometer

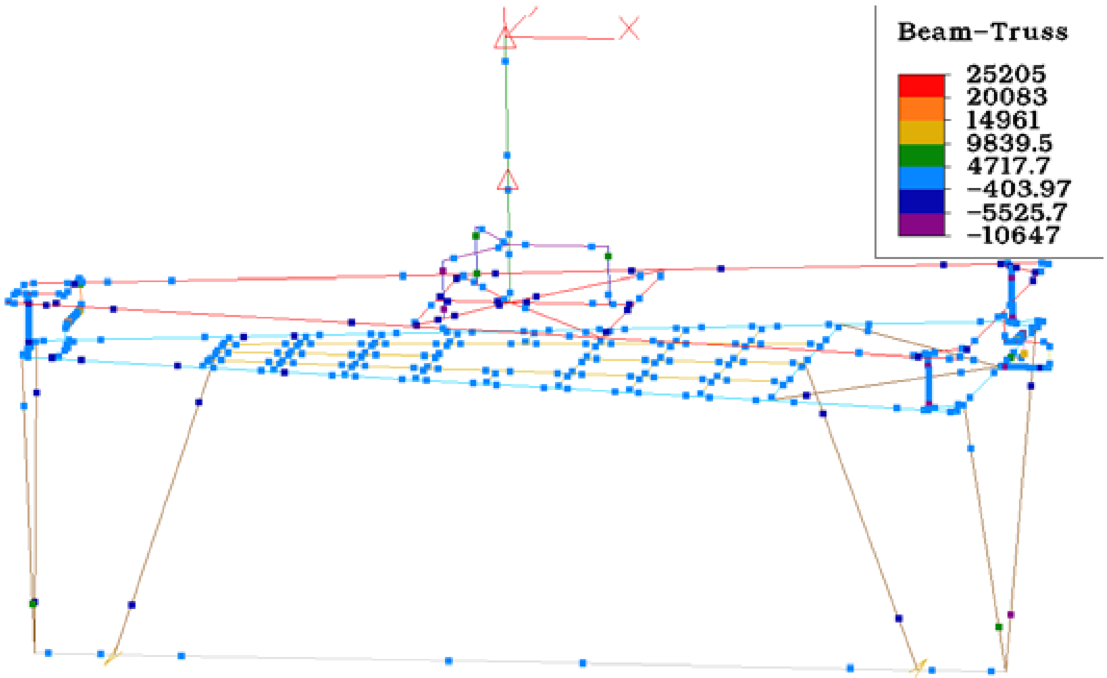
The basic concept/design of the PMM dynamometer appears sound.

However, the FEA and a visual inspection revealed weak points in the execution/fabrication.

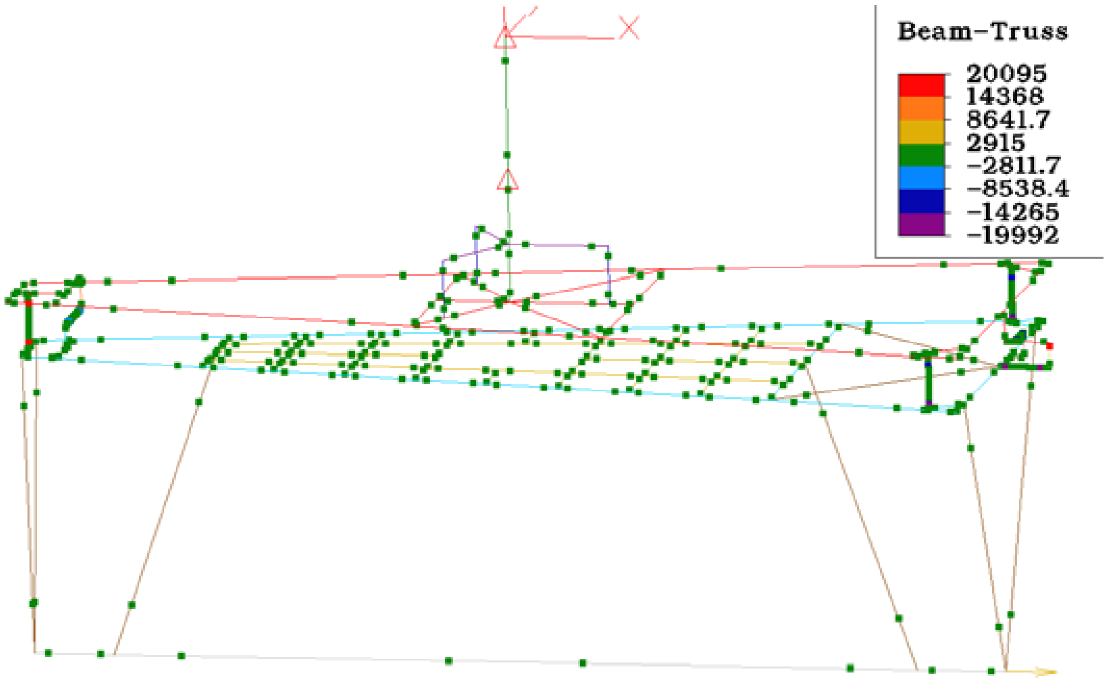
1. The aft side force flexible link mount is poorly welded and has a large slot which cuts the mount in half.
2. The drag load cell mounting is poorly aligned and overhangs the frame. The long overhang creates unwanted moments and thus noise in the drag results.
3. The aft (narrow) end of the lower aluminum frame is poorly designed to resist torsion. The long overhang down into the model creates moments which induce torsion and lead to excessive deflection out of the measuring plane for the dynamometer at this end.
4. The connection to the 4” shaft is weak in yaw. Again the long moment down into the model is inducing high stresses in the gear web section which it is not designed to resist.

From a practical point of view it would be considerably easier to repair/ maintain/calibrate this dynamometer if it could be removed from the PMM as a unit.

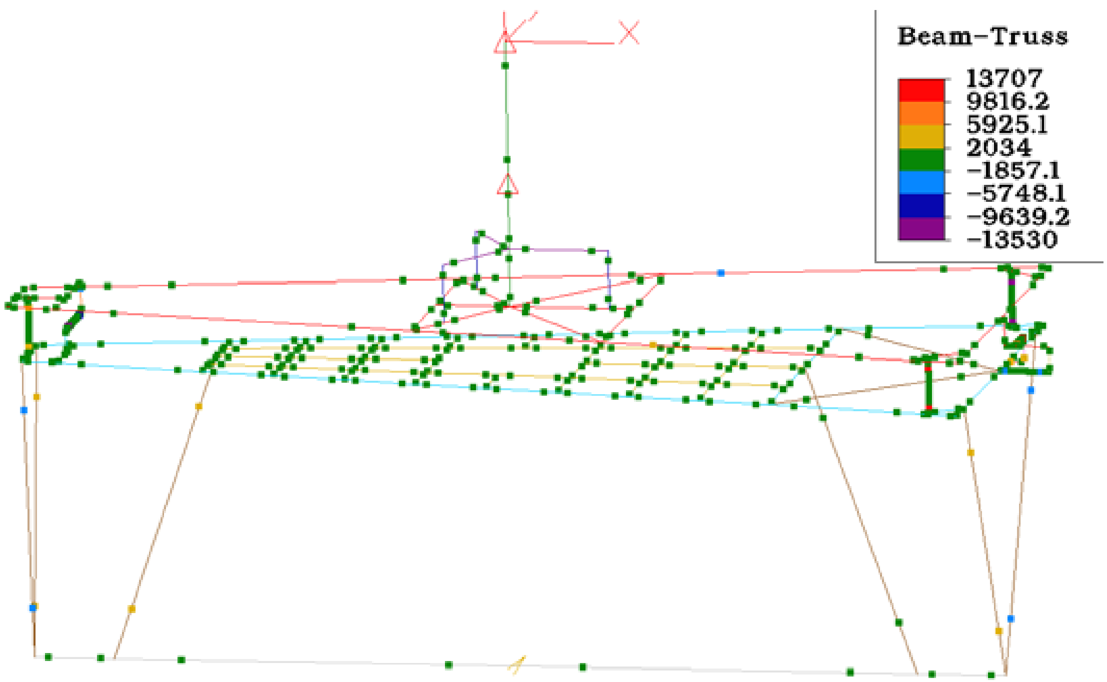
Yaw Moment Result



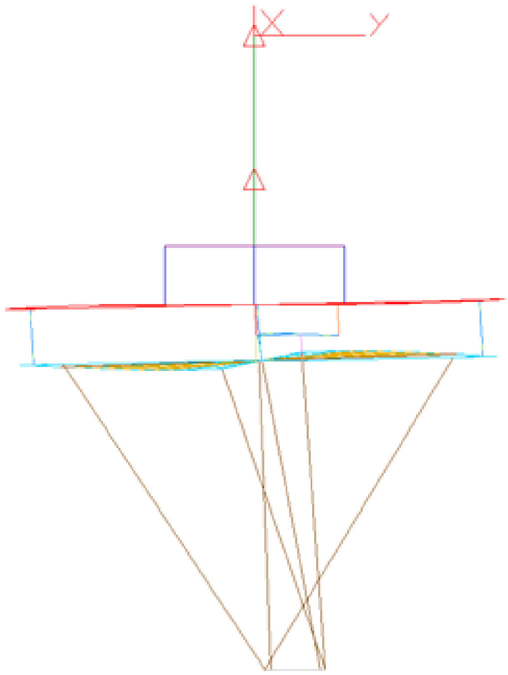
Drag Force Result



Sway Force Result



Natural Frequency Result



<u>PMM</u> <u>Frame</u> <u>Analysis</u>

The objective in performing this analysis was to see if the PMM was deflecting excessively because of the small size of the long horizontal members. To check this the frame was modeled with a simplified dynamometer and ship model. The weight of entrained water and the effect of buoyancy on pitch was not modeled. A linear natural frequency analysis was then carried out to see if deflections were excessive.

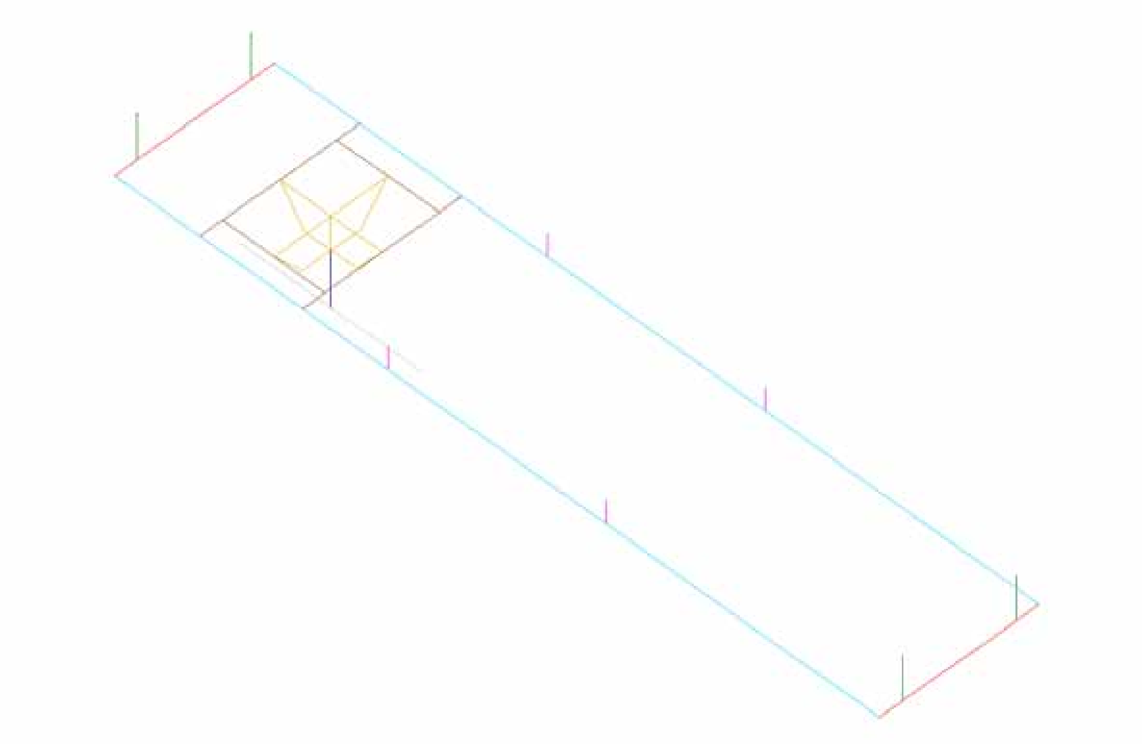
The ship model was chosen at 3 m long and 400 kgs.

The PMM frame is made up of two long horizontal 6” box tubes. These 6”tubes are supported by large channels which are attached to the carriage structure at each end. The 6” box tubes are also supported at roughly the 1/3 chord points by hangers which are attached to the measuring beams on the carriage. Each of these mounting points was represented by a grounded boundary condition.

The PMM has a carriage which supports the dynamometer and model. This carriage was placed in the center of the largest span between supports and modeled with beam elements.

The tube connecting the dynamometer to the PMM carriage was also modeled. This tube was carried down to the ship model.

Frame Model



BEDIT Preprocessor

Again to allow for future use the properties assigned in the preprocessor are listed and the I2,I3 / Z2,Z3 properties may not be in order and orientation of beams should be checked when properties are assigned.

End Channel Supports
Group 1 4" x 1.5 x 3/16" Steel Channel

A – 1.31	S – .63	J - .026	I – 3.21,0.286	Z – 1.61,0.276

End Channels
Group 2 12" x 3" x ¼" web Steel Channel

A – 5.65	S – 3,2.4	J - .351	I – 123.7,4.52	Z – 20.6,2.16

Model Dyno
Group 3 4” x ¼” Box Tube Steel

A – 3.75	S – 2	J – 13.2	I – 8.83	Z – 4.41

Horizontal Rails
Group 4 6” x ¼” Box Tube Steel

A – 5.75	S – 3	J – 47.5	I – 31.74	Z – 10.6

Dynamometer Carriage
Group 5 8” x 2.375 x ¼” wed Steel Channel

A – 3.594	S – 2,1.5	J - .155	I – 33.85,1.86	Z – 8.43,1.08

Mid Span Supports
Group 6 3” x ¼” wall Box Tube Steel

A – 2.75	S –1.5	J – 5.2	I – 3.5	Z – 2.33

Ship Model
Group 8 11” Dia. Aluminum Bar Stock

A –94.2	S –84.7	J – 1411	I – 706	Z – 129

Dynamometer Mounting Tube
Group 9 4” Sch 40 Steel Pipe

A –6.2	S –3.1	J – 61.9	I – 30.9	Z – 9.34

Analysis Results

The natural frequency analysis proved sensitive to the properties given to Group 9 the Dynamometer Mounting Tube. Analysis was carried out with 3 sets of properties for this beam. The properties of the actual 4” mechanical tube in the dynamometer, a 6” Sch 40 Pipe and a 12” OD x 11.5 ID steel tube. The results were as follows;

4” Mechanical Tube

1st Mode	6.7 Hz	Yaw
2nd Mode	7.2 Hz	Pitch

6” Sch 40 Pipe

1st Mode	10.4 Hz	Pitch
2nd Mode	11.2 Hz	Sway

12” Tube

1st Mode	12.4 Hz	Pitch
2nd Mode	12.8 Hz	Sway

Although the natural frequency analysis of the dynamometer in isolation showed acceptable values the combination of the dynamometer and frame showed that the 4” tube was a weak point in the model. Increasing the mounting tube size to the next standard size up in pipe (6” schedule 40) produced a dramatic increase in the natural frequencies.

The 12” Tube was tried to see where the ceiling was in perusing this line of inquiry. The 12 inch tube was unrealistically large and took the mounting tube out of the critical path of the analysis. The frequency results for this tube size indicate the natural frequency of the PMM frame with the carriage at center of the largest span.

Conclusions / Recommendations

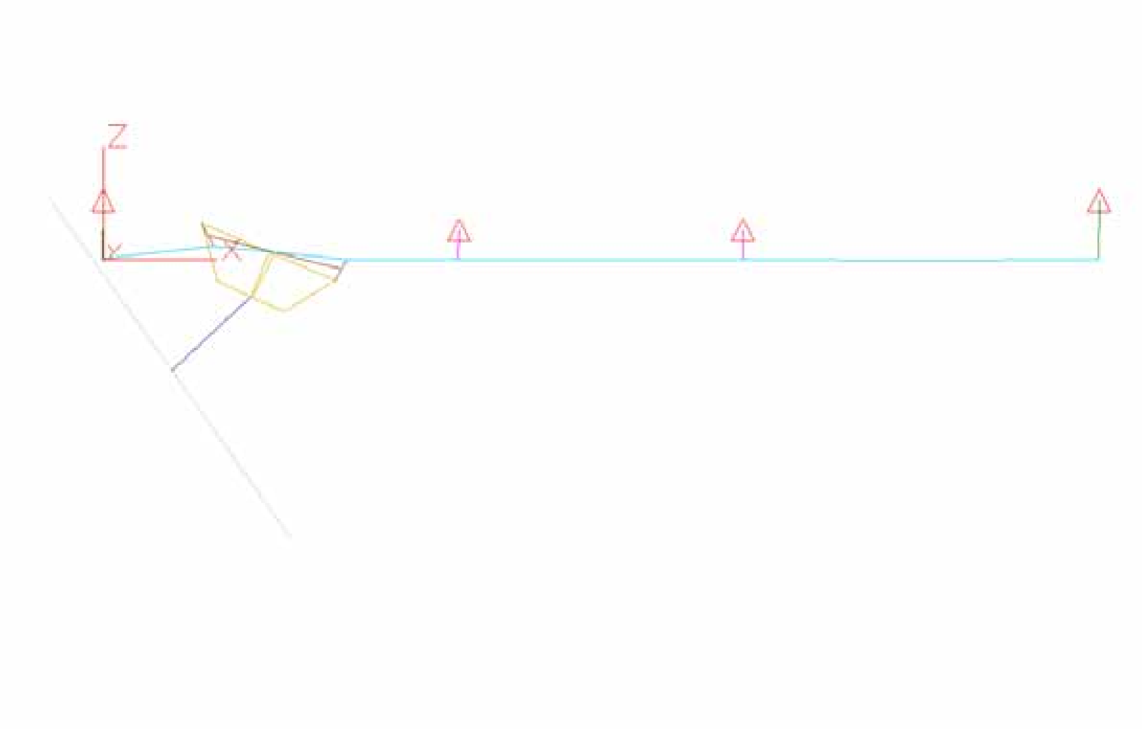
The PMM frame appears to be stiff enough to handle a 400 kg model when the frame is provided with support at the ends and at the center span hangers. The frame without the end supports would be unstable.

The frame model results indicated that a 10 % reduction in the natural frequency of the dynamometer when it was added to the carriage was overly optimistic. A reduction of 25% may be more realistic. Therefore the 1st mode for the combined dynamometer and frame is probably closer to 9.2 x .75 = 7 Hz.

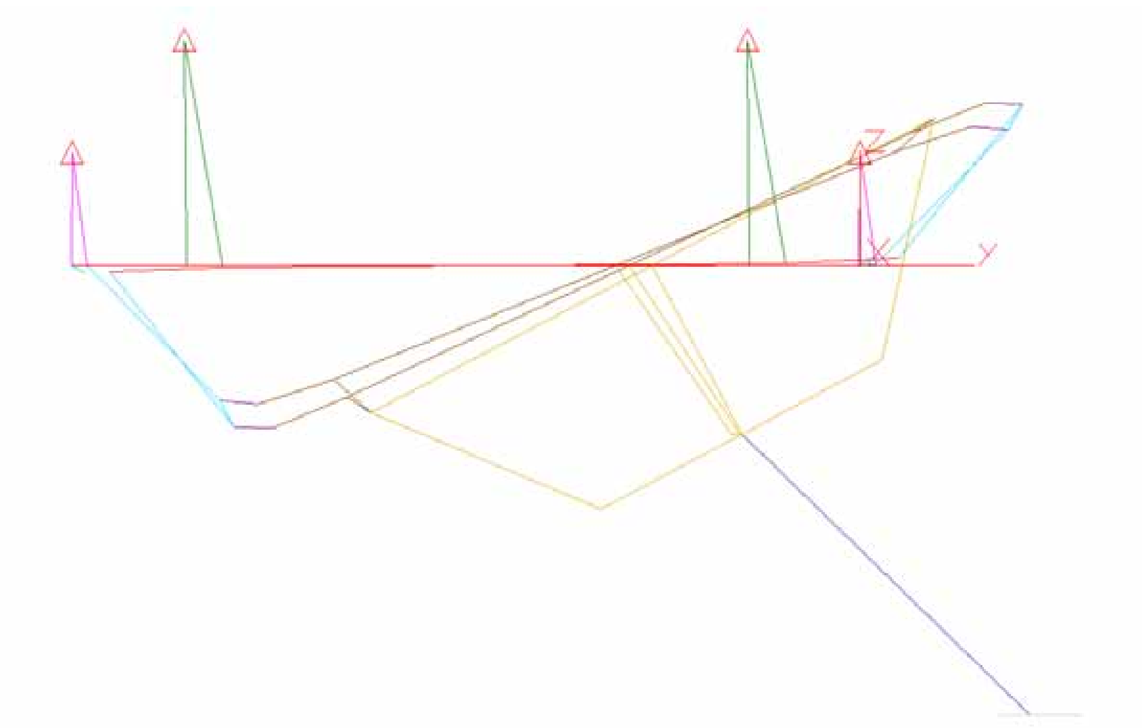
<u>File Location</u>

All files pertaining to this study are located on CadUser\ Projects \421017\ PMM

6" Tube 1st Mode Results
Pitch



6" Tube 2nd Mode Results
Yaw



Appendix E

PMM Performance Capability

PMM Performance Capability **Mark Dawe** **January 2005**

After examining several sources of data from projects that have been tested on the PMM, the following data has been obtained:

PMM Performance Capability

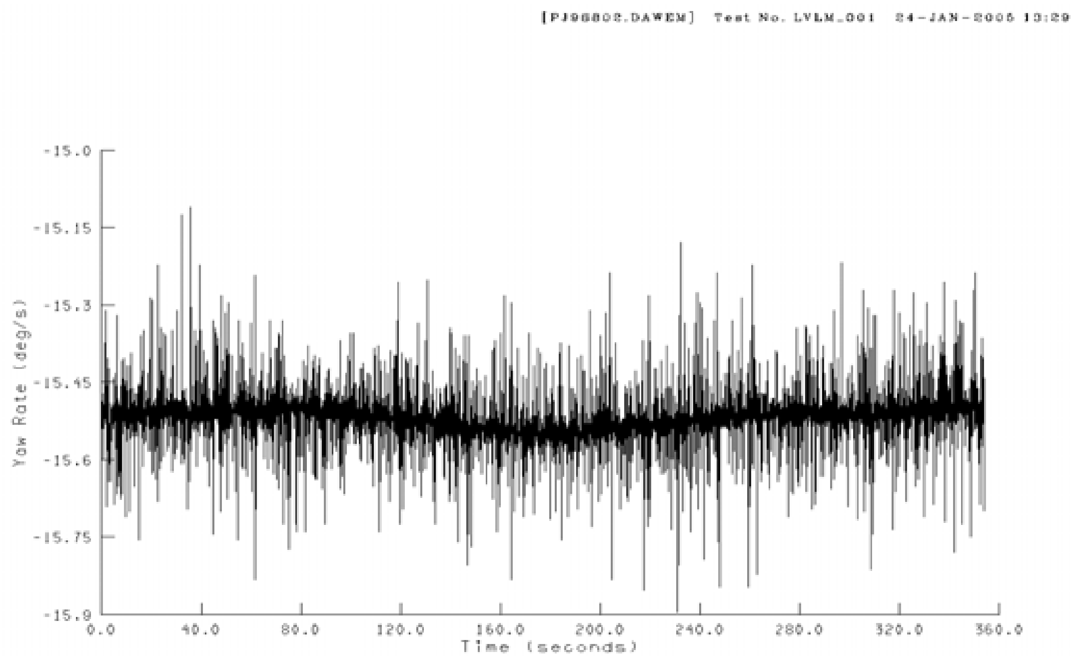
Max Carriage Speed	1.5 m/s
Max Sway Velocity	0.14 m/s
Max Yaw Angle	105 deg
Max Yaw Rate	16.1 deg/s
Max Sway Force (fwd/aft)	2233 / 1150 N
Max Surge Force	1301 N
Max Heave Force	123 N
Max Port/Stbd Roll Force	1195 / 1564 N

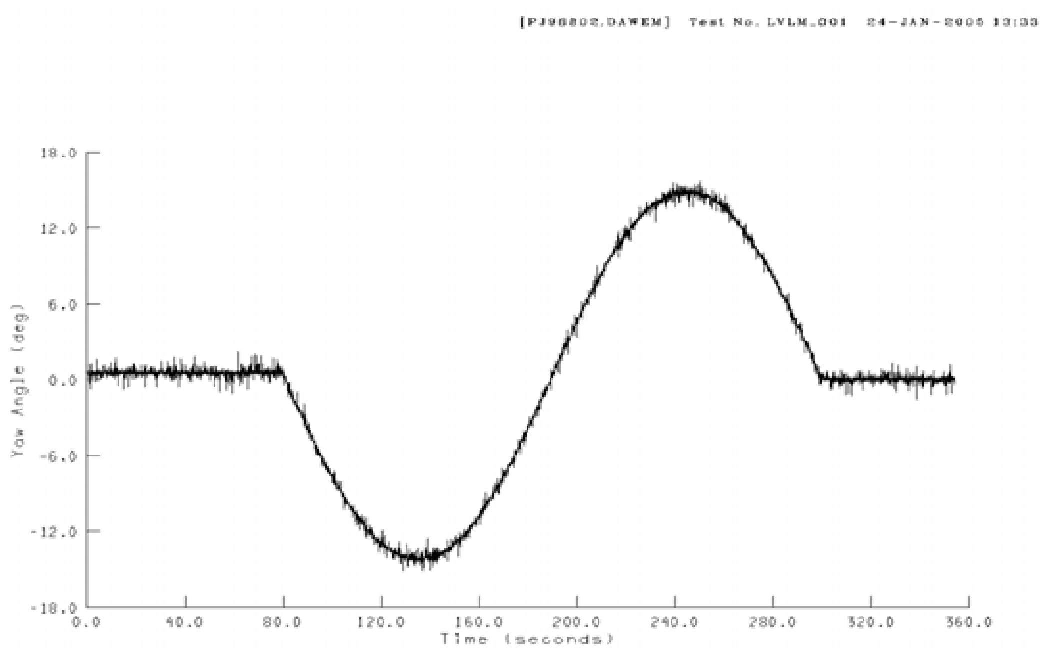
The above results represent the maximum values that could be obtained from several projects. The projects that were analyzed include the Terry Fox, Canadian Patrol Frigate, Escort Tug, C-SCOUT, and a Samsung Tanker. Most of the maximum results came from the analysis of the Samsung Tanker with the only exceptions being the Max Carriage Speed that was found in the Escort Tug tests and the Max Heave Force that was taken from a C-SCOUT estimate.

Results represent a broad variety of tests. The types of tests include:

1) Circular Motion Tests
2) Pure Yaw Tests
3) Yaw + Drift Tests

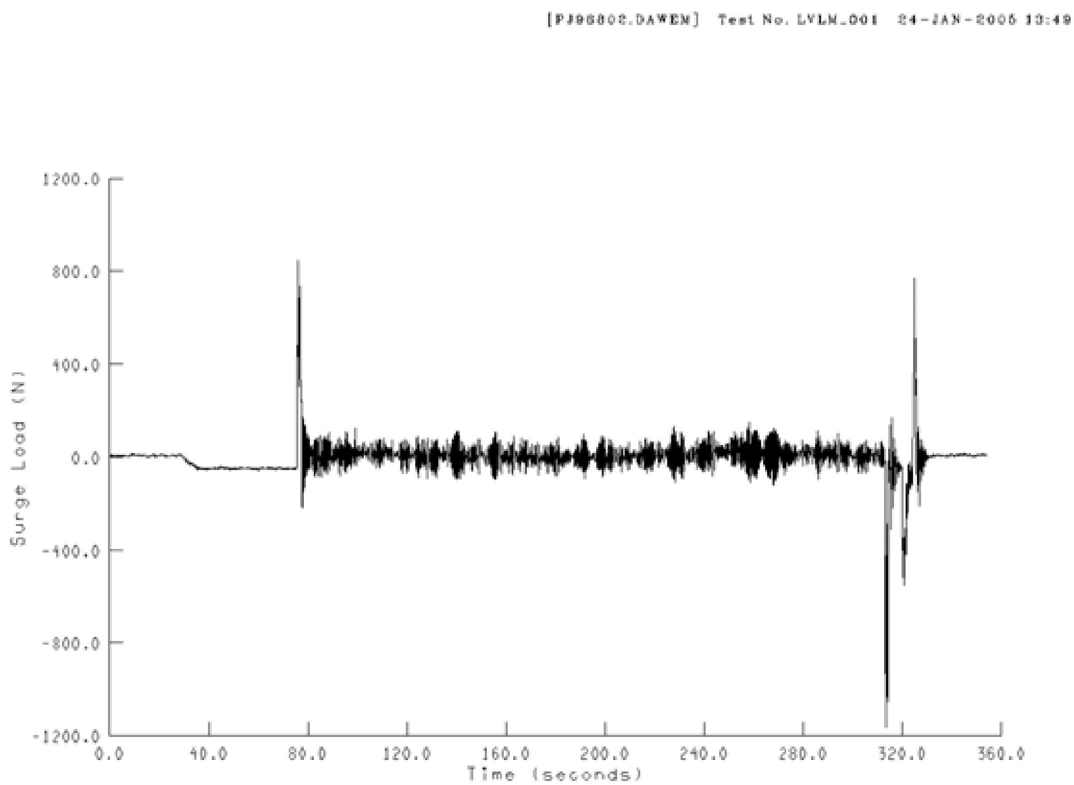
The Samsung Tanker was tested in 63mm thick ice that included level, pack, and rubble ice. The maximum carriage speed for the tests was 0.4 m/s. There were two items of note in the Samsung tests. First of all, the yaw rate for the test was fairly constant throughout each run in the tests which does not seem consistent with the yaw angle, as seen below:

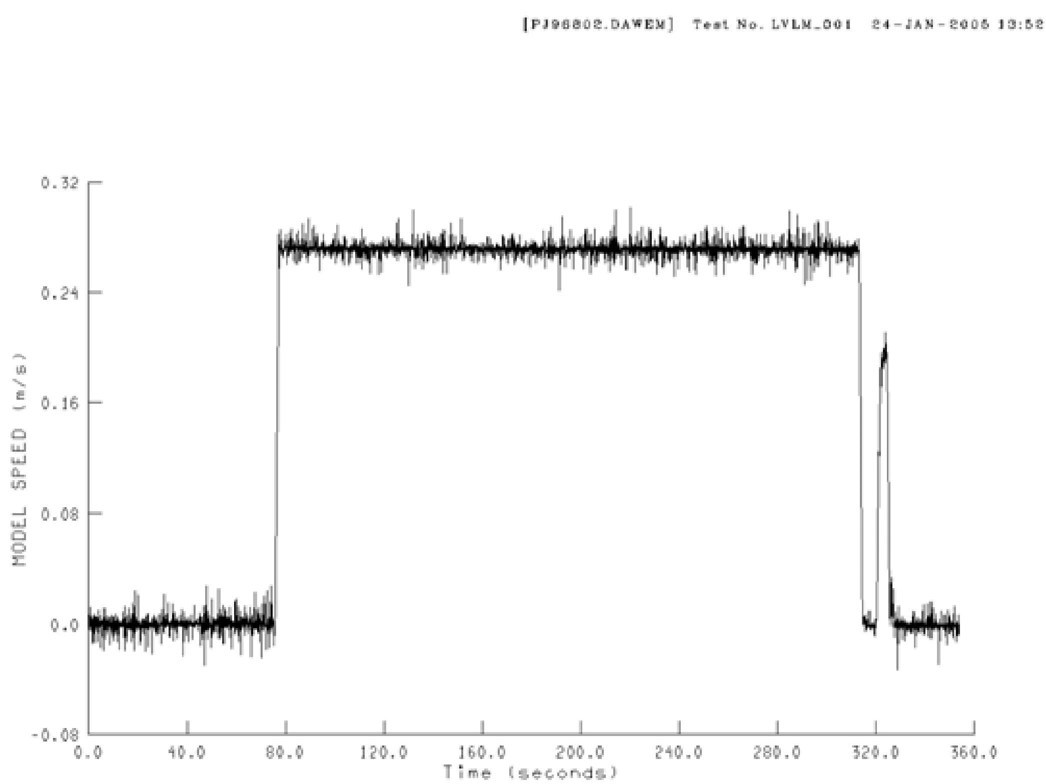




It may be possible to conclude from the above graphs that the yaw rate was not recorded or measured correctly and therefore the yaw rate may not need to be this high in actual testing. The only other yaw rate data that was obtained was from the Terry Fox and the highest rate in those tests was 5.5 deg/s.

The second item of note is the maximum surge loads seen in the summary are based on instantaneous loadings that occur during the initial acceleration and subsequent deceleration of the tow carriage. This can be seen from the graphs of the surge loads and carriage speed on the following page.


[PJ96802.DAWEM] Test No. LVLM_001 24-JAN-2005 13:49
1200.0
800.0
400.0
0.0
-400.0
-800.0
-1200.0
Surge Load (N)
0.0
40.0
80.0
120.0
160.0
200.0
240.0
280.0
320.0
360.0
Time (seconds)


[PJ96802.DAWEM] Test No. LVLM_001 24-JAN-2005 13:52
0.32
0.24
0.16
0.08
0.0
-0.08
MODEL SPEED (m/s)
0.0
40.0
80.0
120.0
160.0
200.0
240.0
280.0
320.0
360.0
Time (seconds)

PMM Design Loads

Max Sway Velocity	0.50 m/s
Max Yaw Angle	+/- 175 deg
Max Yaw Rate	20 deg/s
Max Sway Force (fwd/aft)	3000 / 2000 N
Max Surge Force	2000 N
Max Heave Force	1000 N
Max Port/Stbd Roll Force	2000 / 2000 N

Above are some rough loads that may be suitable to use for the design of a new PMM. The values are slightly higher than the loads seen in the test data that was analyzed since there may be other tests that would yield higher maximums. Maximum sway velocity and yaw rates would be for a large model like the Samsung Tanker.

Design Specifications:

1) As listed above in the PMM Design Loads, the various forces that are stated are structural design forces. A new or refurbished PMM should have sufficient stiffness such that these loads do not cause any undesirable deflection in the apparatus that would have negative impacts on test data. This includes both the rails and PMM carriage itself.
2) The yaw rate and sway velocity must be high enough to satisfy present and future testing requirements. Values given in the PMM Design Loads may not be required for larger models or for testing in general. However, to keep pace with other PMMs, it may be desirable to have elevated capabilities. Looking at the PMM that was used to test the Esso Osaka (document in PMM common folder),

its capabilities include: sway velocity of 2 m/s with an amplitude of 5 m, and maximum yaw rate of approximately 17.2 deg/s.

3) Should be able to restrain the motion of the model in various directions, i.e. roll, heave, and pitch as desired / needed.
4) Implement an emergency brake on the sway motion of the PMM carriage.
5) Reduce the overall height of the current PMM for ease of installation and removal in the Ice Tank.