Design Summary Of The

USNA 380 FOOT HIGH PERFORMANCE TOWING TANK

Prepared For The

16th AMERICAN TOWING TANK CONFERENCE

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By

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I. INTRODUCTION

The new engineering laboratory complex under construction at the U. S. Naval Academy will include a high performance ship model towing tank of advanced design. This towing tank will be the primary facility of the hydro-mechanics laboratory where student education and faculty research will be carried out in Hydro-Acoustics, Ocean Engineering, Hydrodynamics and Naval Architecture.

A conceptual facility study was conducted by members of the USNA Naval Systems Engineering Department faculty and the results were presented at the 15th ATTC in Ottawa. Subsequent to this study, an indepth design and developmental effort was accomplished by the AAI Corporation under contract to the Chesapeake Division of the Naval Facilities Engineering Command. This study centered on the high and low speed towing carriages and their propulsion, velocity control and data transmission systems, the rail system and the acoustic provisions. These development studies resulted in the preparation of preliminary performance specifications for the aforementioned systems. The detail design drawings and specifications for the towing tank equipment and systems have been completed and the U.S. Navy will procure this equipment in FY 1972.

This paper presents a resume of the facility's performance capabilities and descriptions of the various major subsystems which have envolved through the various study and design activities.

II. PHYSICAL CHARACTERISTICS

The physical dimensions for the towing tank were established from a combination of criteria based on the nature of the testing envisioned, architectural and other physical constraints and economic limitations. The final cross-sectional configuration is shown in Figure 1. The total tank length is 380 feet, the usuable track length is 359 feet, the minimum distance at steady-state speed is 150 feet, and the total rail length is 414 feet. It should be noted that the rail location is such that model visibility is unrestricted along the entire usable length on both sides of the tank.

III. PERFORMANCE CHARACTERISTICS

The preliminary performance requirements for the facility were established through the efforts reported on in Reference 1. A further refinement was made through the accomplishment of the development studies, the results of which are given in Reference 2. During the design phase it became necessary to modify or change certain of these requirements. The final specifications for the towing tank are shown in Figure 2.

IV. DESCRIPTION OF MAJOR SYSTEM ELEMENTS

The facility layout shown in Figure 3 indicates the pertinent features and their locations within the architectural envelope provided by the building designers. Of notable interest are the control room, drydock area and the observation and service platforms. The isometric view shows the arrangement of walkways and steps throughout the towing tank area, with the pertinent elevations indicated. (Elevation + 5.00 feet is the normal maximum water level in the basin) Also shown at the drydock area in the plan view is the

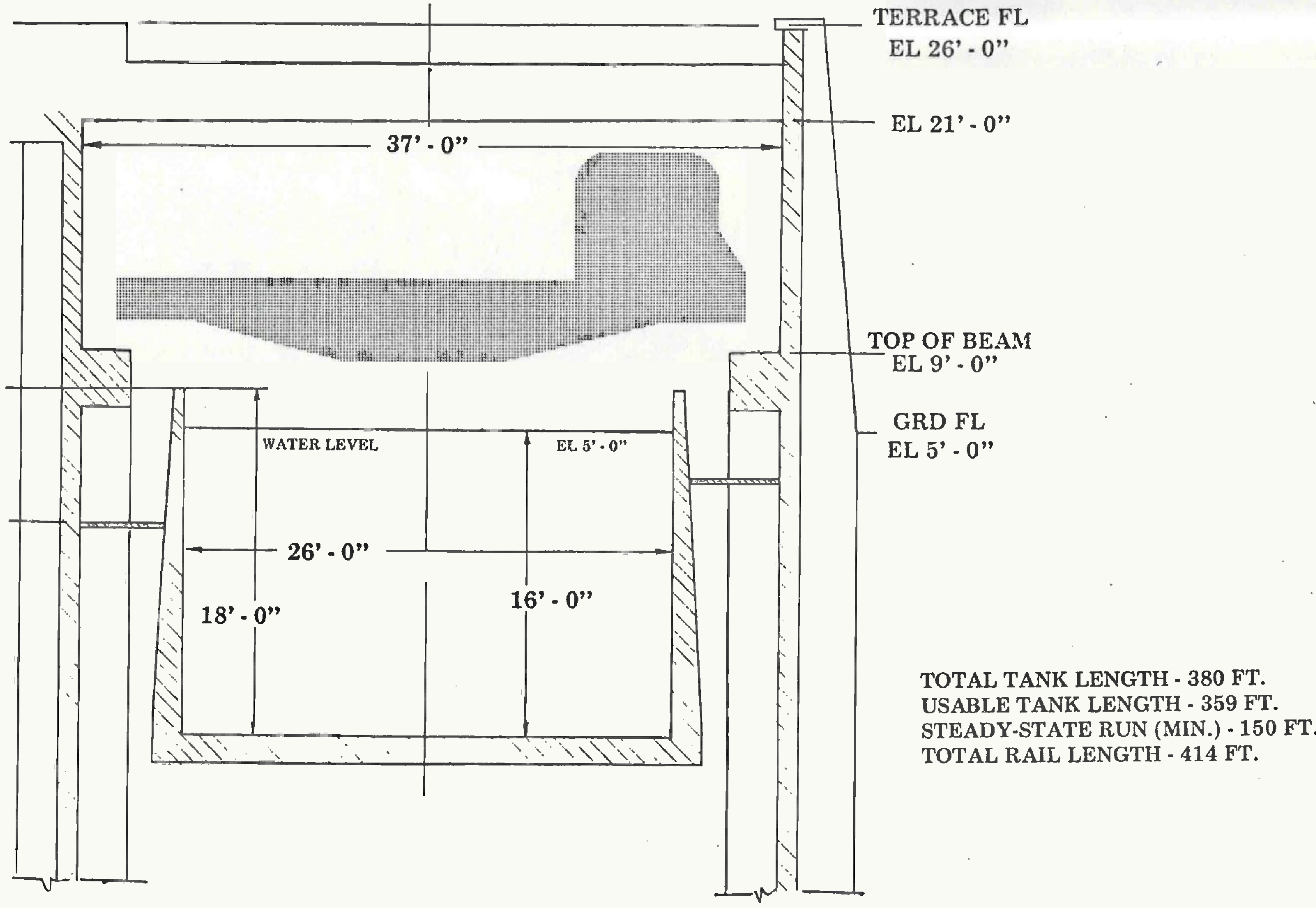


FIGURE 1 TOWING TANK CROSS SECTION WITH LOW SPEED CARRIAGE

		HIGH SPEED CARRIAGE	LOW SPEED CARRIAGE
	Primary Usage	For towing submerged models at high speed and for acoustic tests. Will be an unmanned carriage.	For towing surface models (seakeeping and resistance tests). Capable of accommodating 22 persons for observation and data monitoring
	Max Steady-State Velocity (fps)	50	25
	Minimum Steady-State Run (Ft.)	150	270
	Minimum Speed Range	15:1	100:1
Steady-State Speed Accuracy	Avg. Drift Over Succeeding 50 Ft. Sections in a Given Run	up to 5 fps - ± .001 fps 5 fps to 25 fps - ± .02% 25 fps to 50 fps - ± .1%	up to 5 fps - ± .001 fps 5 fps to 25 fps - ± .02%
	Repeatibility Over 150 Ft. Run-To-Run	up to 5 fps - ± .001 fps 5 fps to 25 fps - ± .02% 25 fps to 50 fps - ± .1%	up to 5 fps - ± .001 fps 5 fps to 25 fps - ± .02%
	Transients, Avg. Over 0.2 Sec.	up to 10 fps - ± .020 fps 10 to 50 fps - ± .2%	up to 10 fps - ± .020 fps 10 fps to 25 fps - ± .2%

FIGURE 2a SUMMARY OF PERFORMANCE CAPABILITIES

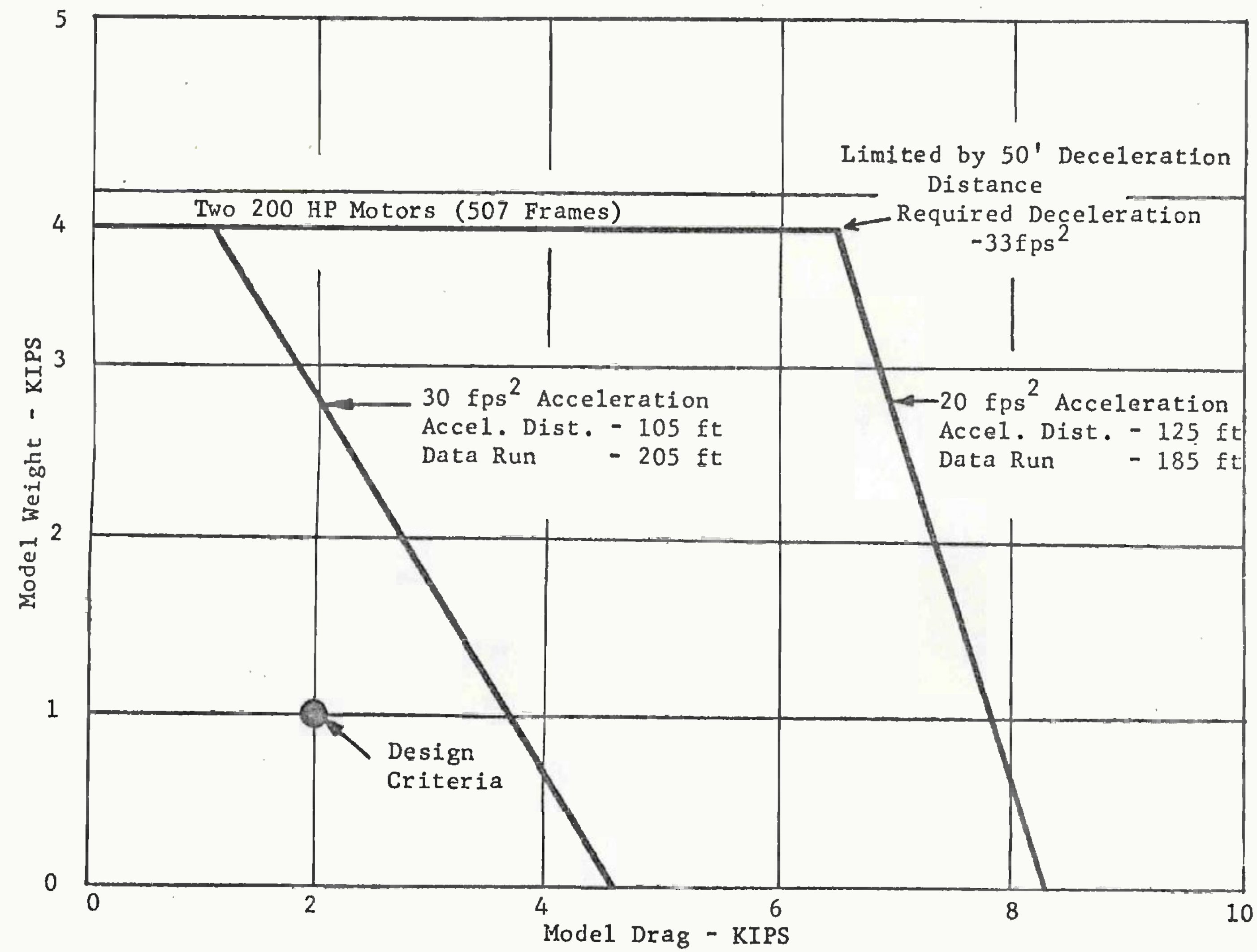


FIGURE 2b MODEL WEIGHT AND DRAG CAPABILITY - HSC

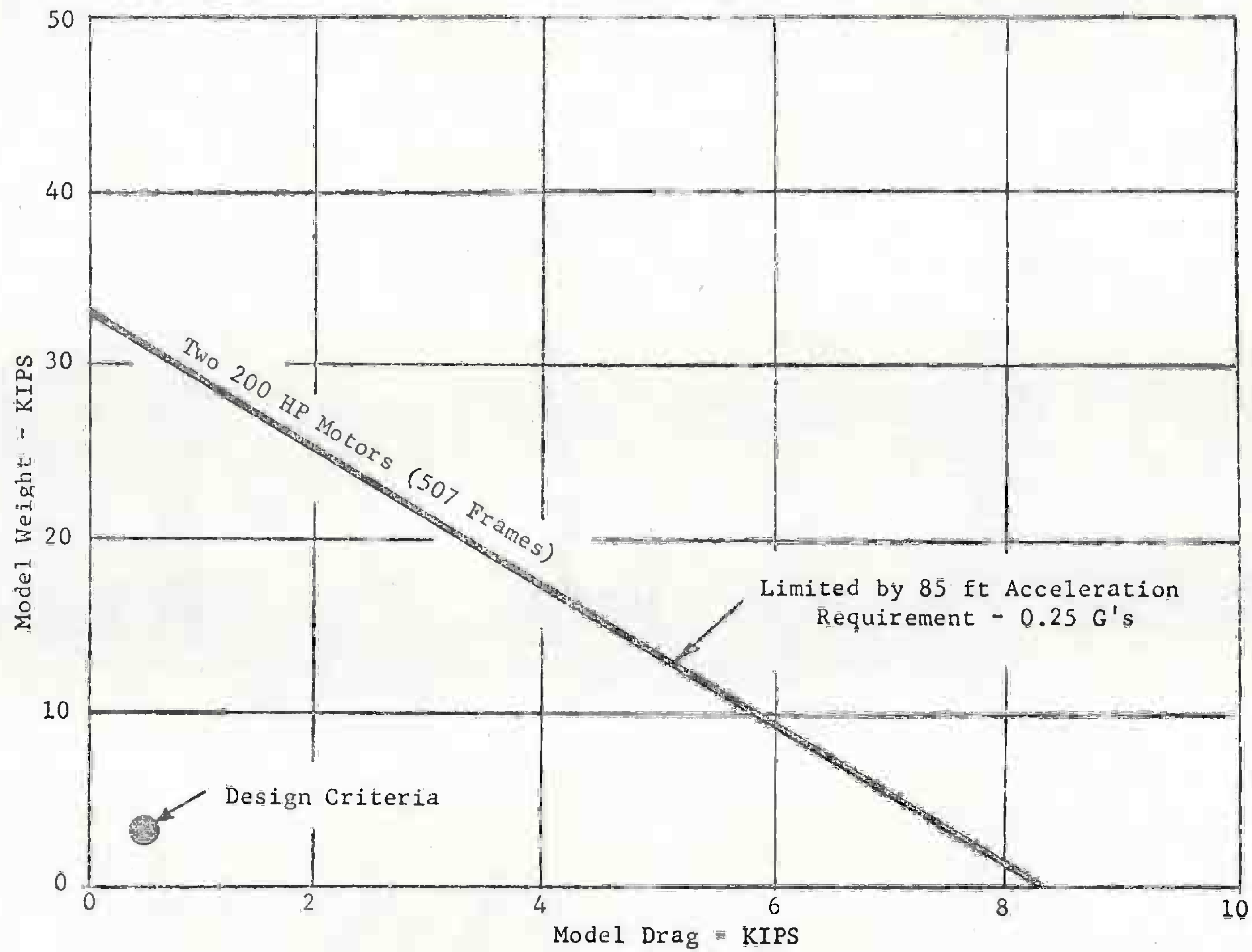
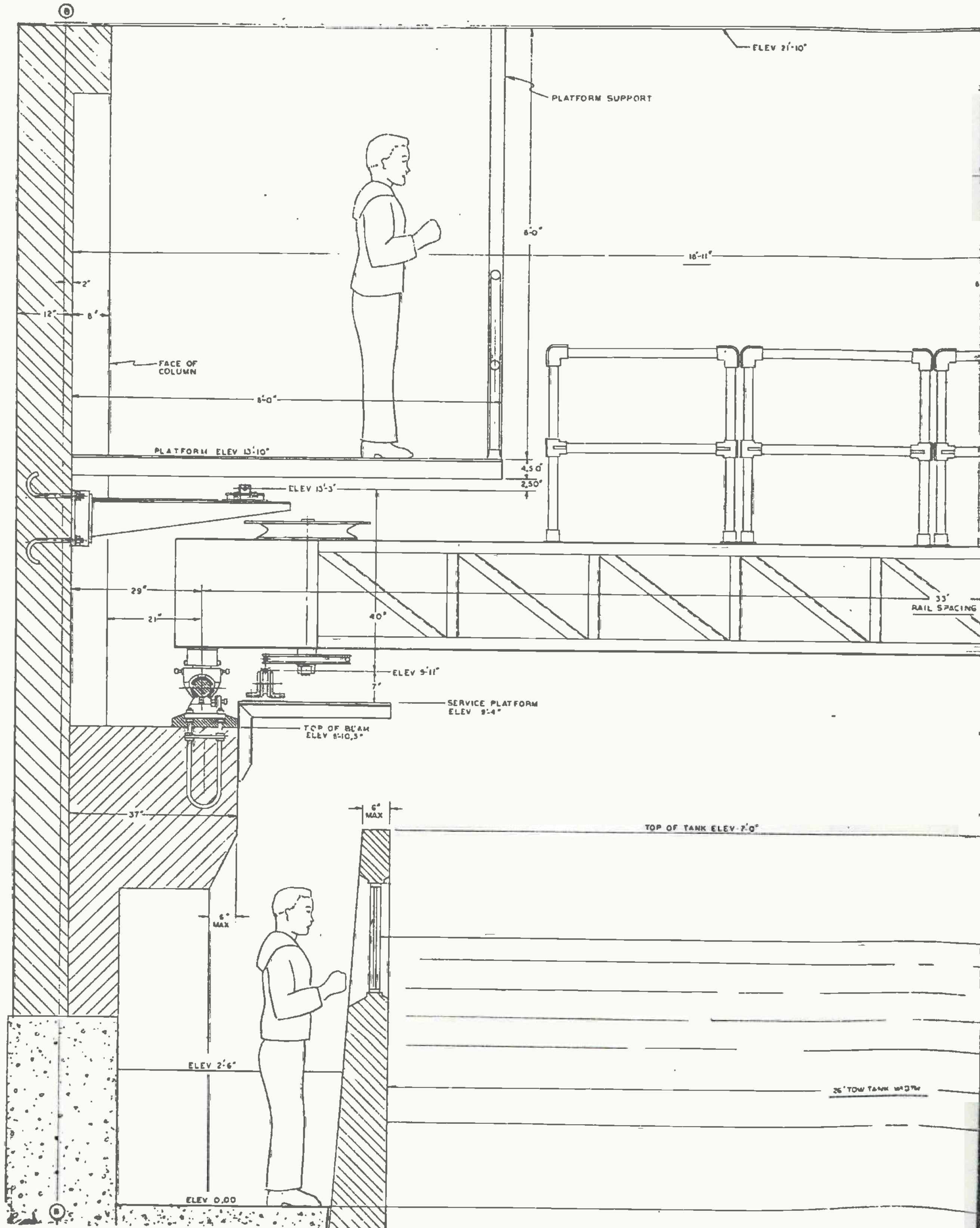


FIGURE 2c MODEL WEIGHT AND DRAG CAPABILITY - LSC



ELEV 21'-10"
PLATFORM SUPPORT
8'-0"
16'-11"
2"
12"
8"
FACE OF COLUMN
8'-0"
PLATFORM ELEV 13'-10"
4.50"
2.50"
ELEV 13'-3"
29"
21"
40"
33"
RAIL SPACING
ELEV 9'-11"
7"
SERVICE PLATFORM ELEV 9'-4"
TOP OF BEAM ELEV 8'-10.5"
6" MAX
TOP OF TANK ELEV 7'-0"
37"
6" MAX
ELEV 2'-6"
ELEV 0.00

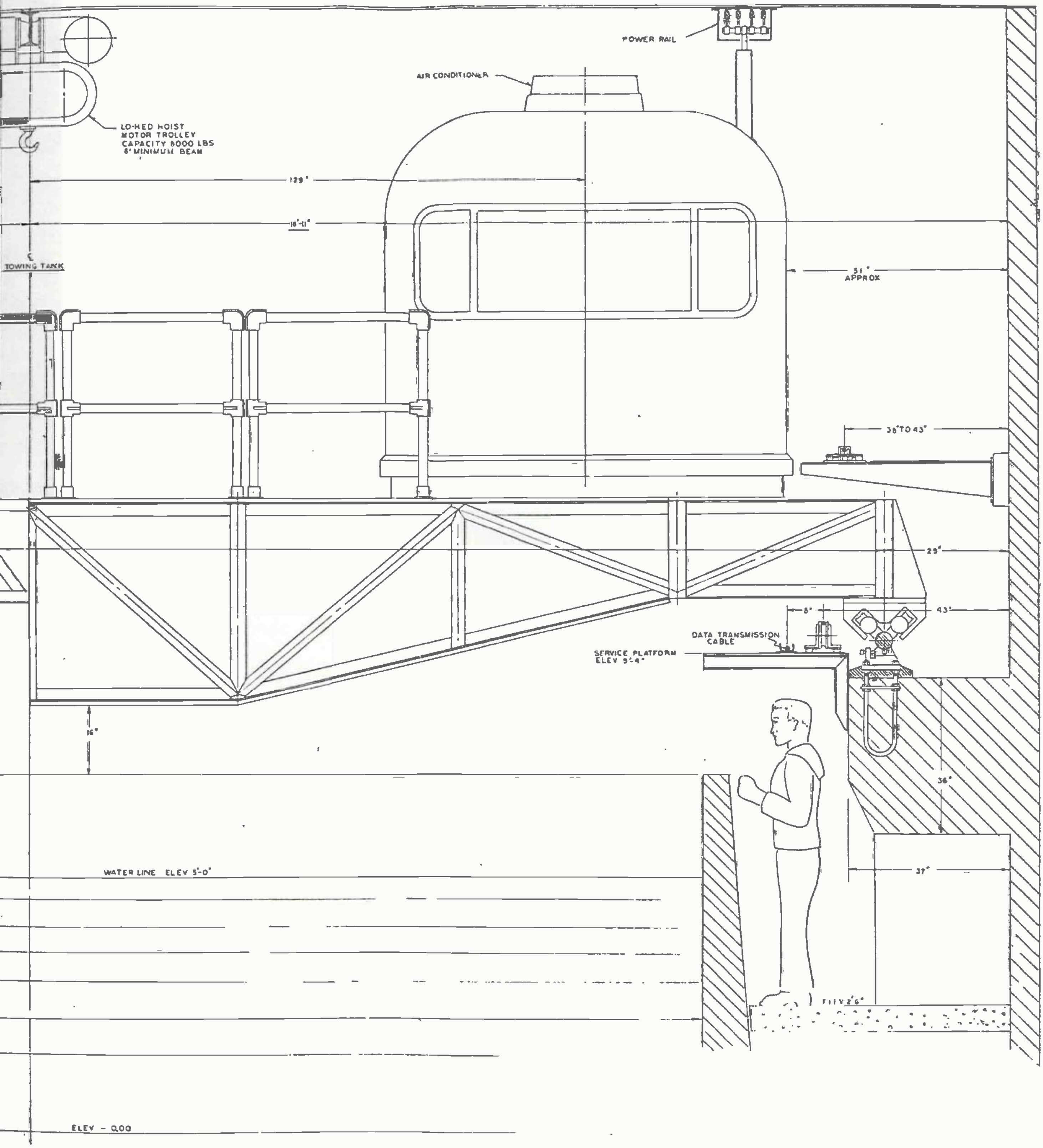
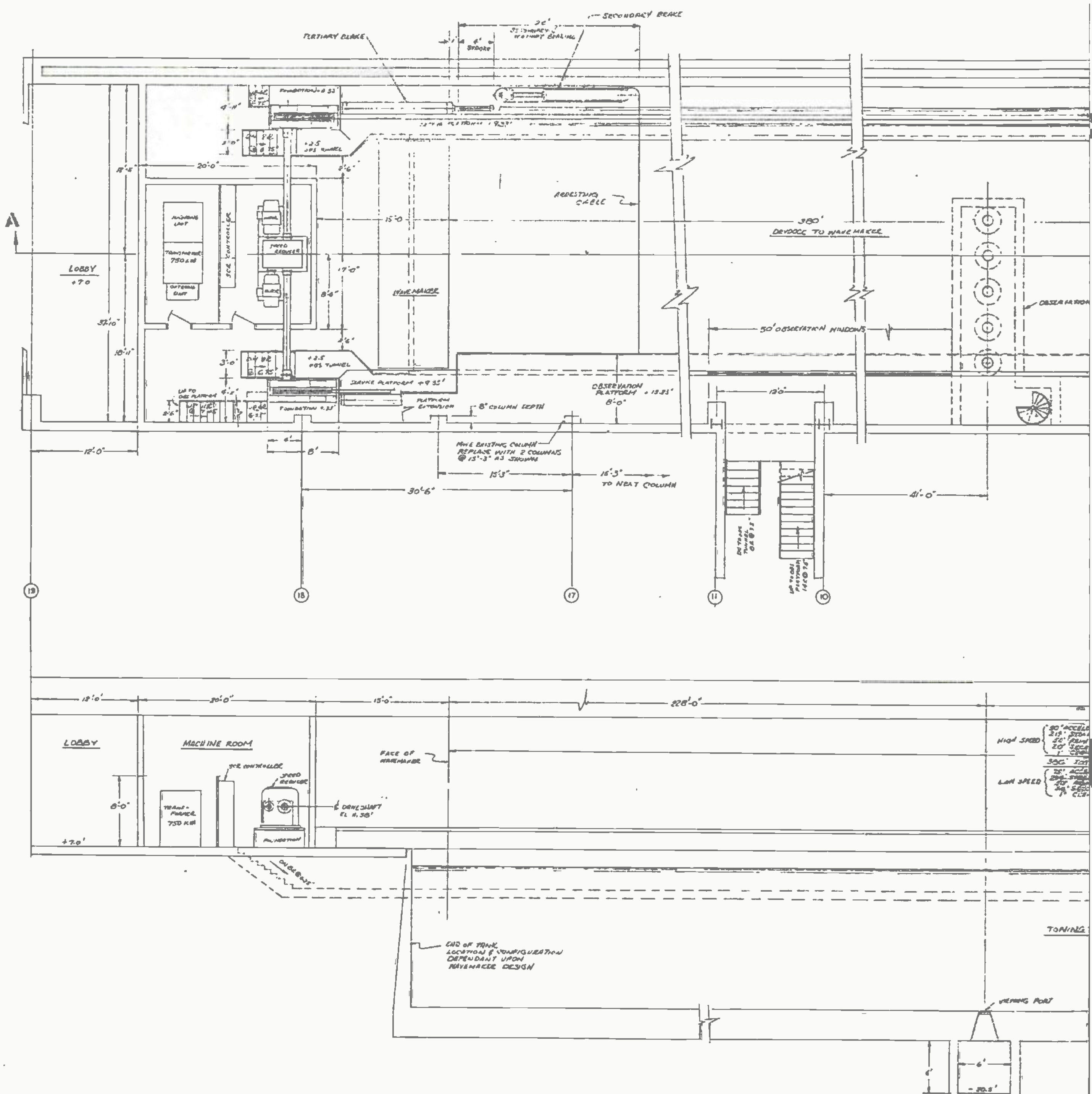


FIGURE 3a FACILITY LAYOUT



SECONDARY BRAKE
TERTIARY BRAKE
A
LOBBY
+7.0
SCR CONTROLLER
TRANSFORMER 750 kW
SPEED REDUCER
WAVEMAKER
ARRESTING CABLE
380'
DRYDOCK TO WAVEMAKER
50' OBSERVATION WINDOWS
OBSERVATION PLATFORM +13.25'
8'-0"
SERVICE PLATFORM
8" COLUMN DEPTH
MOVE EXISTING COLUMN REPLACE WITH 2 COLUMNS @ 15'-3" AS SHOWN
15'-3"
15'-3" TO NEXT COLUMN
30'-6"
41'-0"
12'-0"
20'-0"
15'-0"
228'-0"
LOBBY
MACHINE ROOM
SCR CONTROLLER
SPEED REDUCER
TRANSFORMER 750 kW
FACE OF WAVEMAKER
HIGH SPEED
LOW SPEED
8'-0"
+7.0'
END OF TANK LOCATION & CONFIGURATION DEPENDANT UPON WAVEMAKER DESIGN
VIEWING PORT
TOWING
6'

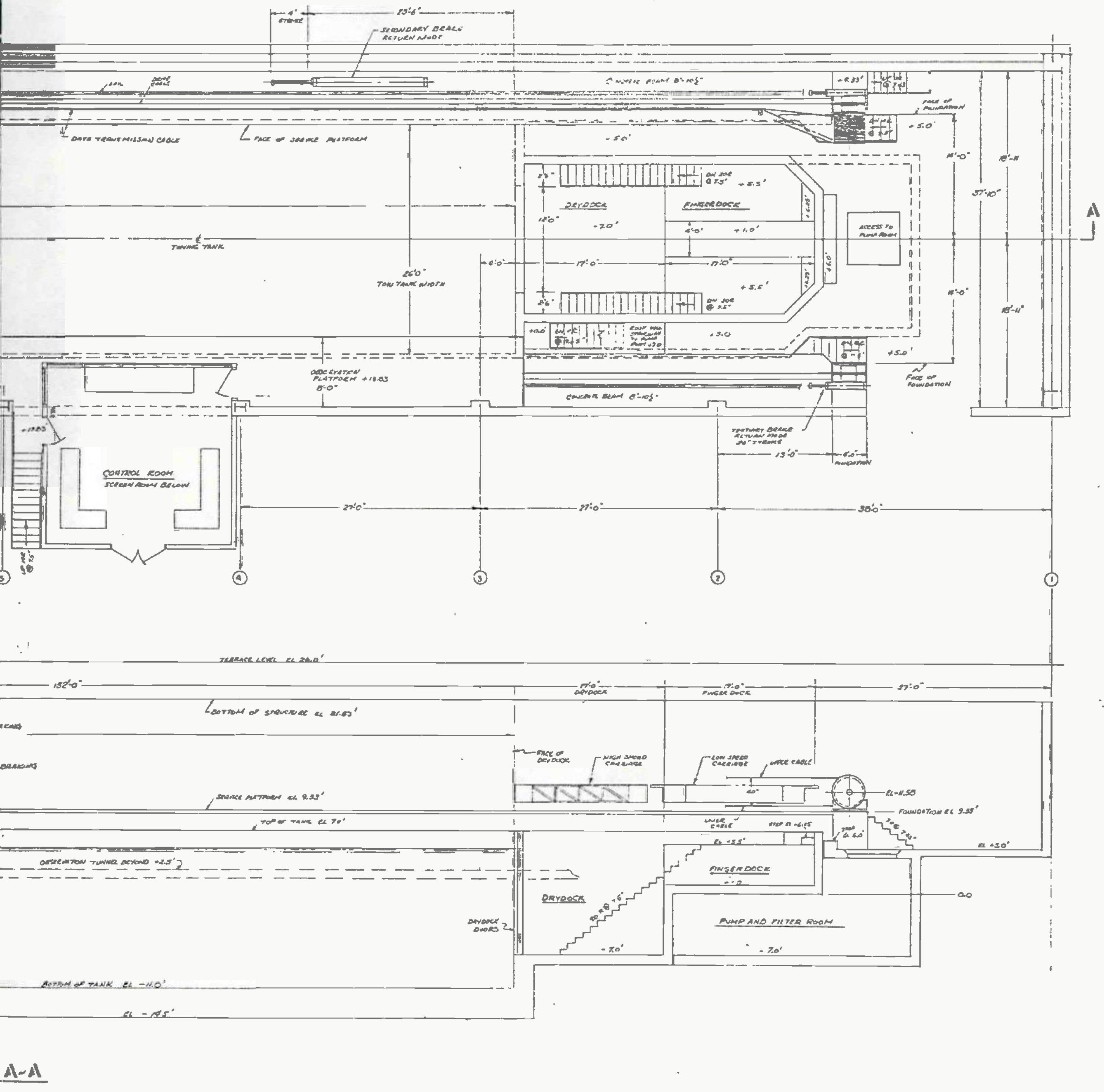


FIGURE 3b FACILITY LAYOUT

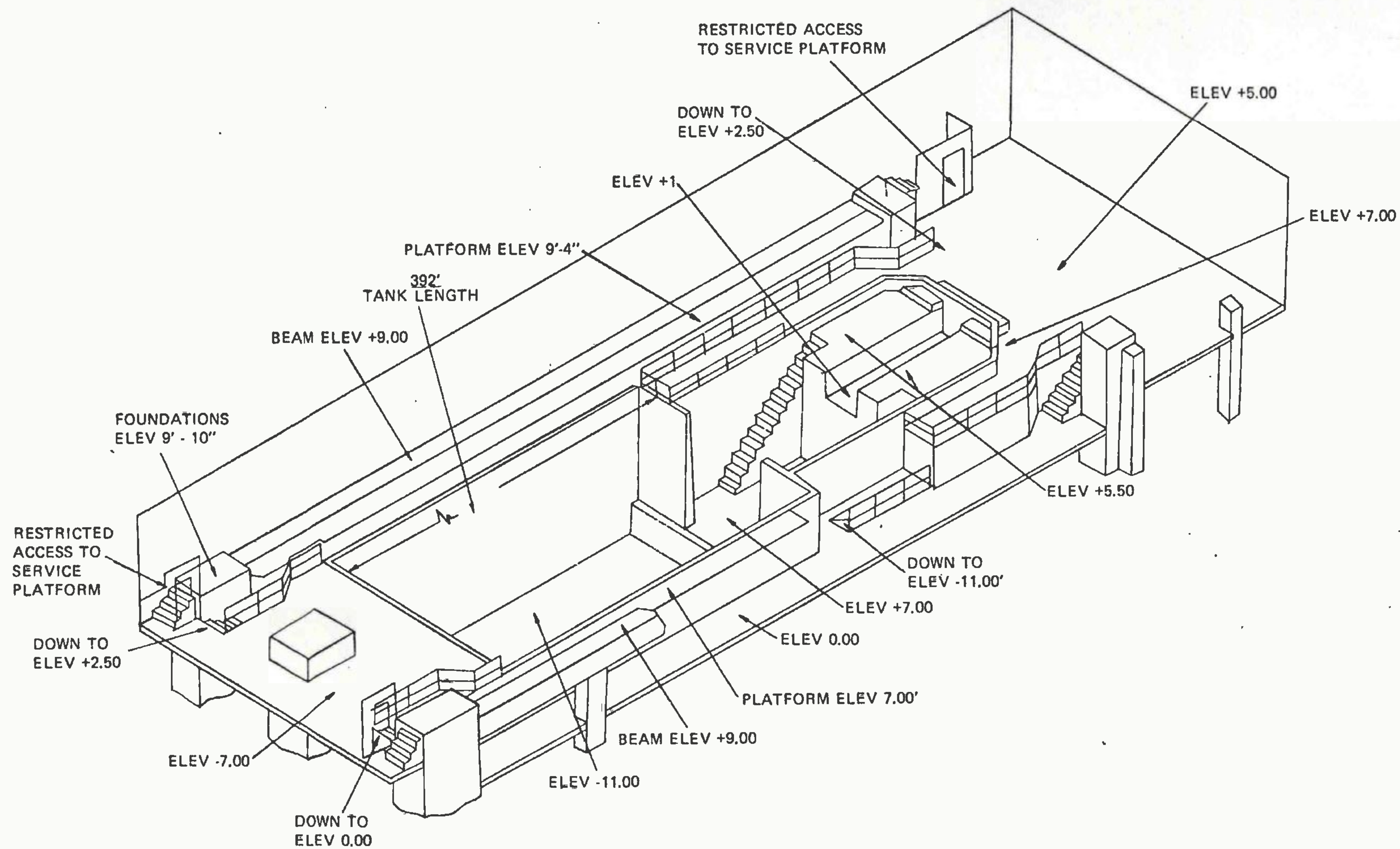


FIGURE 3c FACILITY LAYOUT

maximum width available, with the building configuration, for the movement of equipment and models into the preparation area from the outside. Unauthorized entry onto the service platforms is prohibited by adequate screened enclosures which are located as shown at each end of the tank. These are lockable and are provided primarily for safety reasons, since, during a test, the carriages pass directly over the service platforms with insufficient clearance to allow anyone to occupy the platforms during a test. The following are descriptions of the major system elements.

A. Propulsion System

This section discusses the factors leading to the selection of the propulsion system for the high and low speed carriages. The studies described herein were also conducted to establish the feasible parameters and the functional requirements for propelling the carriages.

1. Evaluation of Candidate Systems

Investigations into the current methods of propelling model towing carriages and a detailed review of the operational and performance requirements for the USNA towing tank system, indicated that only three basic types of propulsion systems should be considered. These were the Linear Induction Motor (LIM), on-board electric motors powering drive wheels, and off-board electric motors driving a tape or cable attached to the carriages. Each of these propulsion systems is considered in the following discussions.

a. Linear Induction Motor

The Linear Induction Motor (LIM) is an AC machine similar to a Squirrel Cage Induction Motor, except that the rotor and stator have been in effect, unrolled into flat elements. Development programs have demonstrated that the rotor or secondary member can be configured as a flat metal sheet, and that the stator or primary winding can be energized to produce a thrust between the primary and secondary members.

Two configurations are possible and both were considered. The first contains a short primary with a very long secondary where the primary is located on-board the moving vehicle. This system is being developed for high speed trains, where the primary power is on board or can be picked up from trackside using a trolley or third rail system. In the second configuration, the primary is in the rail and the secondary is on the vehicle. This scheme is more attractive for application to the towing tank.

With the primary and secondary used in the configuration of the "unrolled" induction motor, large attractive forces are developed between these two components, which must be restrained by the rail systems. To overcome this problem the LIM is most often used in a double sided configuration, i.e., the two primaries with the secondary member located between them. This configuration gives double the thrust of the single sided version and because the attractive forces are balanced, effectively controls the airgap flux and shortens the magnetic path.

The advantages of using the LIM for propulsion in the towing tank application are:

- It is the only on-board electrical drive system capable of generating the thrust necessary to accelerate the High Speed Carriage to maximum velocity in the distance available.

- This thrust is generated without contact between the fixed and moving portions allowing greater freedom in the design of the guide rails and the carriage suspension.

- Noise and vibration from the motor are minimal.

- As a direct drive system, it eliminates components of low resonant frequency from the speed control system servo loop. This permits higher gain at higher frequency and consequently shorter time to damp out speed transients, resulting in longer data runs.

The disadvantages of the LIM are:

- It is a state-of-the-art component. Currently there are no off-the-shelf designs available for the motor, the power conversion system, or the control system. This results in high development costs to adapt the existing knowledge in the field to this particular application.

- There are high risks in interpolating between the existing LIM systems particularly in regard to the requirements for speed control accuracy and speed variation. The existing LIM's range from the high speed train propulsion system to a system for opening and closing curtains with essentially no other sizes developed between them.

- It appears from the investigations that a 10-1 speed range is the best that can be expected from current generation LIM's.

The speed range problem can be illustrated in Figure 4. It can be seen that there are two factors which affect speed: power frequency and pole pitch. Frequencies below 10 Hz are not desirable and small pole pitches are not possible because finite copper windings are required. Elaborate

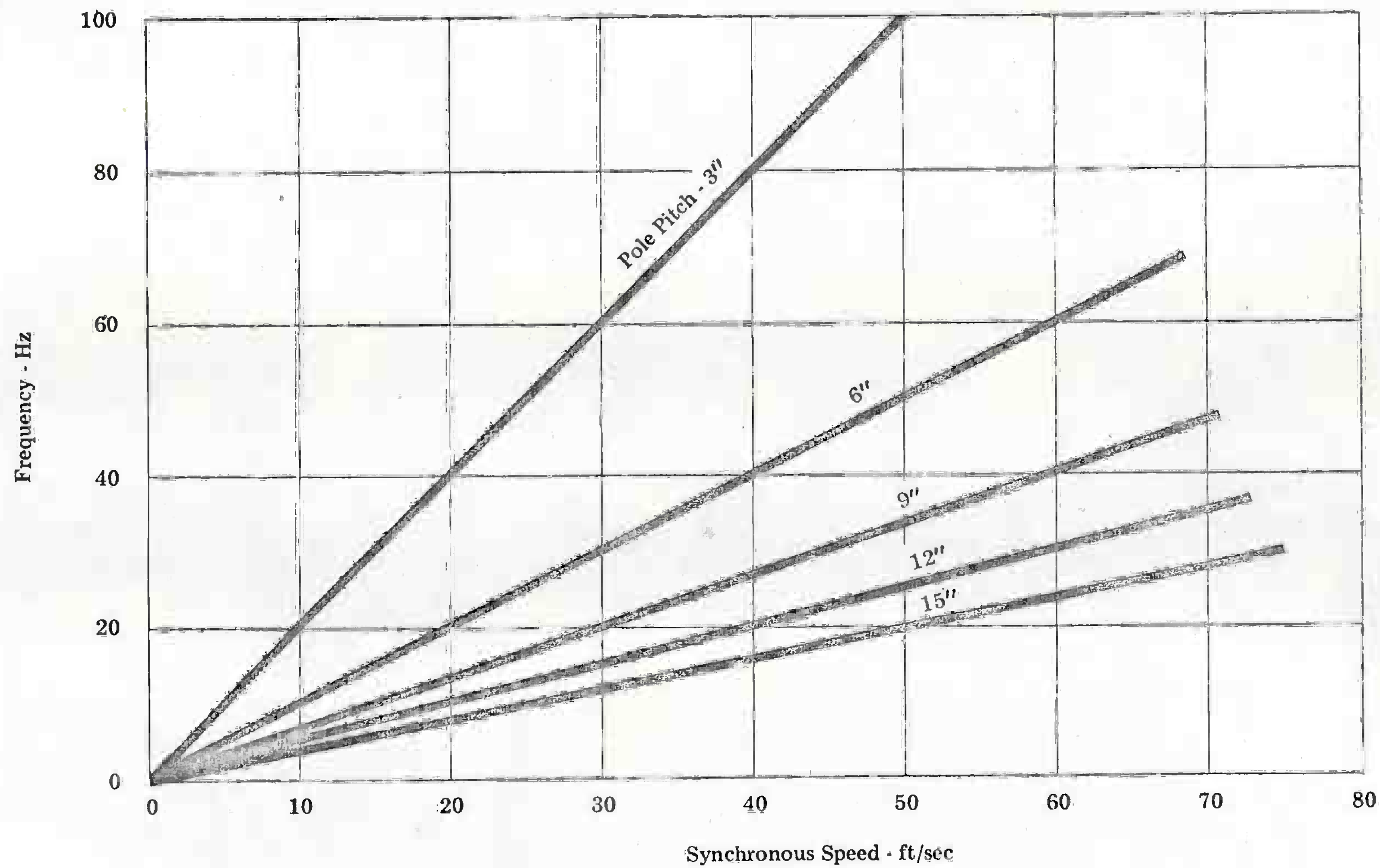


FIGURE 4 SPEED RANGE CAPABILITIES OF LIM

combinations where winding configurations are switched for various speeds can conceivably provide speed ranges of 30 to 1.

b. On-Board Motor

(1) High Speed Carriage

The primary advantage of on-board propulsion is improved speed control made possible by a stiff, high frequency drive train. However, the noise and vibration associated with this system make it completely unsuitable for flow noise testing. Furthermore, it can be demonstrated that on-board motors cannot even accelerate their own weight without a substantial reduction in the data run time. In order to accelerate or decelerate the carriage to or from 50 fps within 40 feet of travel, a peak acceleration of 2 g's is required. If a carriage weight of 5000 pounds is assumed, the total horsepower rating of the motor can be determined as follows:

$$HP = \frac{\text{Force x Speed}}{550} = \frac{\text{Acceleration x mass x speed}}{550}$$

$$HP = \frac{5000 + \text{motor weight}}{5.5} \quad \text{(for the assumed parameter values)}$$

Let N = motor weight per horsepower

$$HP = \frac{5000 + N \times HP}{5.5}$$

$$HP = \frac{5000}{5.5 - N}$$

It can be seen that the required horsepower approaches infinity as N approaches 5.5 pounds per horsepower. Conventional, watertight, DC motors are available with a weight of approximately 10 pounds per peak, short term horsepower, which makes them unsuitable as an on-board drive. Watertight AC induction motors are available at a minimum weight of about 8 pounds per peak horsepower and consequently could not be used. The added weight, noise and complexity of a multi-range gear shift such as an automotive transmission precluded its use to extend the motor capability.

(2) Low Speed Carriage

Since the low speed carriage can accelerate to 25 fps within 50 feet of available track length at .25 g's, an on-board drive appears more feasible. Assuming a low speed carriage weight of 20,000 pounds the following expression for required horsepower can be derived.

$$HP = \frac{20,000}{88-N}$$

If N equals 10, an on-board propulsion system of 260 horsepower is required. Although the motors would only weigh 2600 pounds, an additional 8000 pounds of controllers, gear boxes, drive wheels, guide wheels and preload wheels would be required. Since the maximum tractive coefficient which can be developed between a steel drive wheel and a wet rail is .085, 40,000 to 50,000 pounds of drive wheel preload is required. If the wheels are not preloaded, 160 feet of track would be required for acceleration and 140 feet for deceleration, leaving very little track for the data run.

c. Off-Board Motor

(1) DC Motors

The off-board drive has the advantage of accelerating only its own rotor in addition to the load and a towing cable or tape with drive sheaves. The net mass to be accelerated is estimated at 250 slugs or 8000 pounds, for the high speed carriage. The required horsepower is 1450 short term, or 363 continuous rating. A 400 HP, 1750 RPM, DC shunt motor with limited current to prevent burnout is designed to operate at 400% of continuous rated current with a 20% duty cycle, provided the overload does not last for longer than one minute. Thermal protection is also provided to protect the motor should the overload duty cycle be inadvertently exceeded.

The DC motor drive can be controlled open loop over a 10-to-1 speed range to an accuracy of better than 10%, and if a tachometer feedback loop is added, the range increases to more than 1000 to 1 with an accuracy dependent on the tachometer accuracy rather than the motor characteristic. The lower limit of speed of operation is determined by the static load or breakaway friction, and careful design can let the minimum controllable speed approach zero. The speed is controlled by varying the motor voltage and the motor operates at the intersection of the voltage-limited torque curve and the load line.

The DC drive system is also ideally suited for braking the carriage and returning it to the starting position. If the low speed carriage is towed by the high speed carriage, the peak horsepower required is 318 to accelerate both carriages and drive load to 25 feet per second at .25 g's. In fact, the high speed carriage could tow a conventional carriage weighing up to 120,000 pounds.

(2) AC Induction Motors

The peak horsepower required for an AC drive system is 1450 short term, just as it was the for the DC system. However, an AC motor at best can deliver only about 250% of continuous torque for short term overloads. Therefore, a motor rated at 800 continuous horsepower would be required. The speed-torque characteristic of the induction motor shows that the only stable region of operation is at near synchronous speed, thus the only reasonable way to obtain a variable speed is to vary the applied frequency.

Several problems now become evident concerning the use of an induction motor to propel the carriage. The first is that in order to utilize the available 250% torque for acceleration, the slip must be sensed and a control loop included to limit slip to about 15 Hz. Secondly, the voltage must be made proportional to frequency except that resistance-limited current must not be less than the magnetizing current required by the primary. Finally, even doing these things, about the largest accurately controllable speed range that can be accomplished is 10 to 1 (without the additional loops, 3 to 1 is a practical limit).

It was concluded that the best means of propelling the high speed carriage under today's technology is to use an off-board DC motor-powered drive system to tow the carriage with cables. Since there is no reason to believe the linear induction motor will have a greater speed range than the conventional induction motor (about 10 to 1), there is little justification for designing the system to incorporate a LIM at some future stage of LIM development.

While it is possible to drive the conventional or low speed carriage with on-board motors, it is also possible to tow the conventional carriage with the high speed carriage, and the latter was selected as it results in the simplest system and keeps possible sources of acoustic and EMI noise away from the carriage.

2. Selected System

The propulsion system which was selected for the USNA towing tank facility is illustrated in Figure 5 . An off-board DC electric motor drives a continuous loop cable attached to the high speed carriage. During high speed operation, this carriage is propelled independently, while for low speed operation the high and low speed carriages are coupled, and the high speed carriage serves as a tow vehicle. The propulsion system operates both carriages using a single speed control and mechanical drive system. Furthermore, it has excellent acceleration and deceleration capabilities to maximize the data run, and it removes the primary source of noise and vibration from the carriages.

Power for this system is supplied by two 200 HP DC electric motors. The motors are air cooled and totally enclosed for maximum protection. The motors are coupled to a gear box having a 4.7/1 reduction. The output shaft of the gear box transfers power to the main propeller shafts through flexible couplings. The two electric motors and gear box are mounted on a

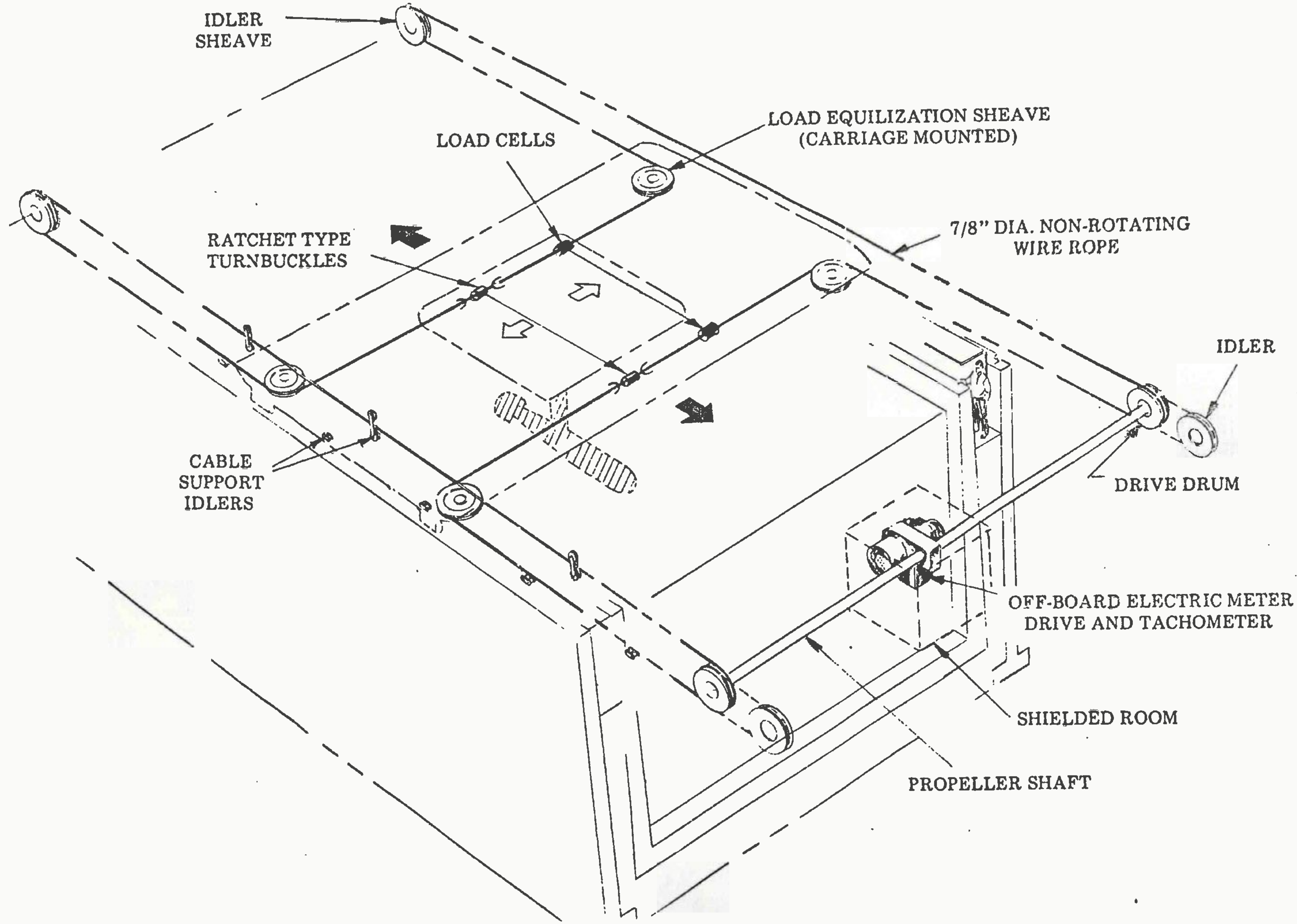


FIGURE 5 CARRIAGE PROPULSION SYSTEM

concrete foundation which is isolated from the towing tank foundation to minimize the transmission of noise and vibration. The electric motors and SCR controllers are housed in a screen room having a concrete block exterior to provide RFI shielding and acoustic isolation from the towing tank test environment.

The digital tachometer which is used to control carriage speed is driven by the main propeller shaft and is mounted near the gear box. Having a single propeller shaft to power both cable drive drums, and a single tachometer to monitor their rotational speed will provide a more simple and accurate system than could be achieved with separate motors, gear boxes and tachometers for each drive drum. A very stiff propeller shaft will be provided to avoid any significant windup and to assure that its resonant frequency is much higher than the cable.

A continuous loop drive system was selected in preference to a drum hoist type in order to provide the carriages with propulsion and speed control in both directions of travel with a single power source. A simple centerline tow cable system was impractical to support, except at the ceiling, and this would have introduced very large overturning loads into the carriage, A single, eccentric tow cable located on the side of the carriage would have introduced very large racking loads into the carriage and its suspension causing undesirable vibrations, and excessive suspension wear and friction. To circumvent these problems a twin loop drive (as suggested in Reference 1) has been selected with the loops running just inside the rails. One of the problems with this system is, unequal drive drum diameters can cause severe load differentials in the twin cables. This problem can be eliminated by either towing the carriage with a whiffletree, or by joining the twin cables at the carriages and anchoring them with load equalization sheaves. The latter arrangement was selected because it loads the carriage in a way that minimizes frame deflection and vibrations, is lighter in weight, and has greater stiffness.

Both cable and tape drives were evaluated for the propulsion system. The tape has the advantage of greater stiffness, increased aerodynamic damping to minimize catenary vibrations, and is not susceptible to unwinding. These characteristics will tend to minimize longitudinal loop vibrations and consequently improve speed control. However, providing adequate traction between the tape and drive drum requires excessive pretensioning in order to prevent slippage. This is due to the fact that is is impractical to provide more than a 180^{o} wrap over the drum drive based on practical tape dimensions for this application. The flexibility of a cable allows multiple wraps around the drive drum, and pretensioning can be kept to tolerable levels. The tendency for a cable to unwind can be minimized by using a torque-balanced wire rope. While the longitudinal stiffness of the cable is not as great as the tape, an effective Young's Modulus of 20×10^{6} psi can be obtained. Structural stretch of the cable can be effectively eliminated by pretensioning. A cable will possess much greater damping in the longitudinal mode of vibration than a solid metal tape, and the increased damping will have a greater effect in reducing the longitudinal vibration than decreased stiffness will have in increasing vibration.

As a result of these considerations a 3/4-inch diameter 6 x 25 Unitlay cable was selected. The cable will be fabricated from galvanized steel and it will be lubricated to promote long life, and quiet operation. The cable will be operated over drive drums and idler sheaves of 40-inch diameter to insure adequate service life. Five 180-degree wraps around the drive drum are provided to prevent slippage during the most adverse operation conditions. The cable is supported on approximately 25-foot centers by idler sheaves running on low friction roller bearings. The brackets for these sheaves are supported on vibration isolation mounts to minimize the transmission of cable vibration to the foundation. The spacing of these idler sheaves will be varied to avoid sympathetic vibration of the catenaries.

The cable system is pretensioned to prevent any part of it from becoming slack under the most adverse operating conditions. This will minimize the possibility of the cable unwinding, kinking or slipping on the drive drum. A cable pretensioning load of 7000 pounds will insure a minimal tension load of 1000 pounds at all times. The pretensioning load will be applied by ratchet type turnbuckles located between the carriage mounted load equilization sheaves. The magnitudes of the pretensioning load will be set and monitored by means of load cells permanently installed in series with the turnbuckles. A suitable type of load cell is the Dillon Series 200 load cell system. It is basically an induction-type load cell with a 15,000-pound capacity and a 4:1 safety factor. It is fabricated from stainless steel and has a very rugged and compact configuration. Either a digital or meter readout is available with an overall system accuracy of ± 1/2% of full range. A strip chart recorder is available and can be used in conjunction with the readout meter to obtain a continuous record of cable loading during the run.

B. Rail System

The primary requirement of the rail system is that it provide accurate alignment of the carriage relative to the water surface and furthermore that it maintain this alignment in the presence of operational loads and environmental conditions for long periods of time. Though the rail support foundation was designed to minimize misalignment attributable to long term settlement and other factors, the prudent approach was to make provision for periodically aligning the rails in a quick and inexpensive manner.

In order to arrive at an optimum design for the rail system, decisions had to be made concerning rail elevation, spacing, configuration, foundation and method of alignment. In order to provide a rational basis for making these decisions, the implications of varying these rail system parameters were investigated.

1. Rail Location and Spacing

The original concept design study by Compton, Dyer and Johnson (Reference 1) suggested 15 foot rail spacing using cantilevered rail supports. This would give the desired acoustic and vibration isolation from the tank walls since the supports and the basin were to be on separate pile systems.

The AAI design group considered other rail spacings, out to the maximum 33.5 foot spacing possible between the architecturally fixed 37 foot distance between the structural walls enclosing the basin. The influence of rail spacing on carriage weight is illustrated in Figure 6 and the relative merits of 15.0 and 33.5-foot rail spacings were summarized and used as the basis of selecting rail spacing. Comparisons were predicated on a 24-inch deep high speed carriage and a 48-inch deep low speed carriage.

Advantages of 33.5-Foot Rail Spacing

- Deflection of rail support structure is minimized.
- Size and cost of rail support structure are reduced.
- Provides continuous foundation support for rail system:
    - Eliminates need for rail support beam.
    - Provides better distribution of wheel loads to soil.
    - Minimizes differential settlement problems.
- Eliminates flexural vibrations in rail support foundation, reducing excitation of basin walls.
- Improves visibility from sides of tank and from carriage.
- Better accessibility for track alignment and maintenance.
- Better weight distribution on wheels of low speed carriage.
- Eliminates water contamination from rail lubricants and foreign matter on carriage suspension.
- Easier to shield rails from water spray.

Advantages of 15-Foot Rail Spacing

- High speed carriage is 1400 pounds lighter.
- Low speed carriage is 4500 pounds lighter.
- Drive system requires 18% less horsepower.
- Carriage velocity variations induced by drive system are reduced.
- Carriage frame need not be as deep, creating less drag and less disturbance of water surface.
- Carriage is smaller, lighter and easier to install.

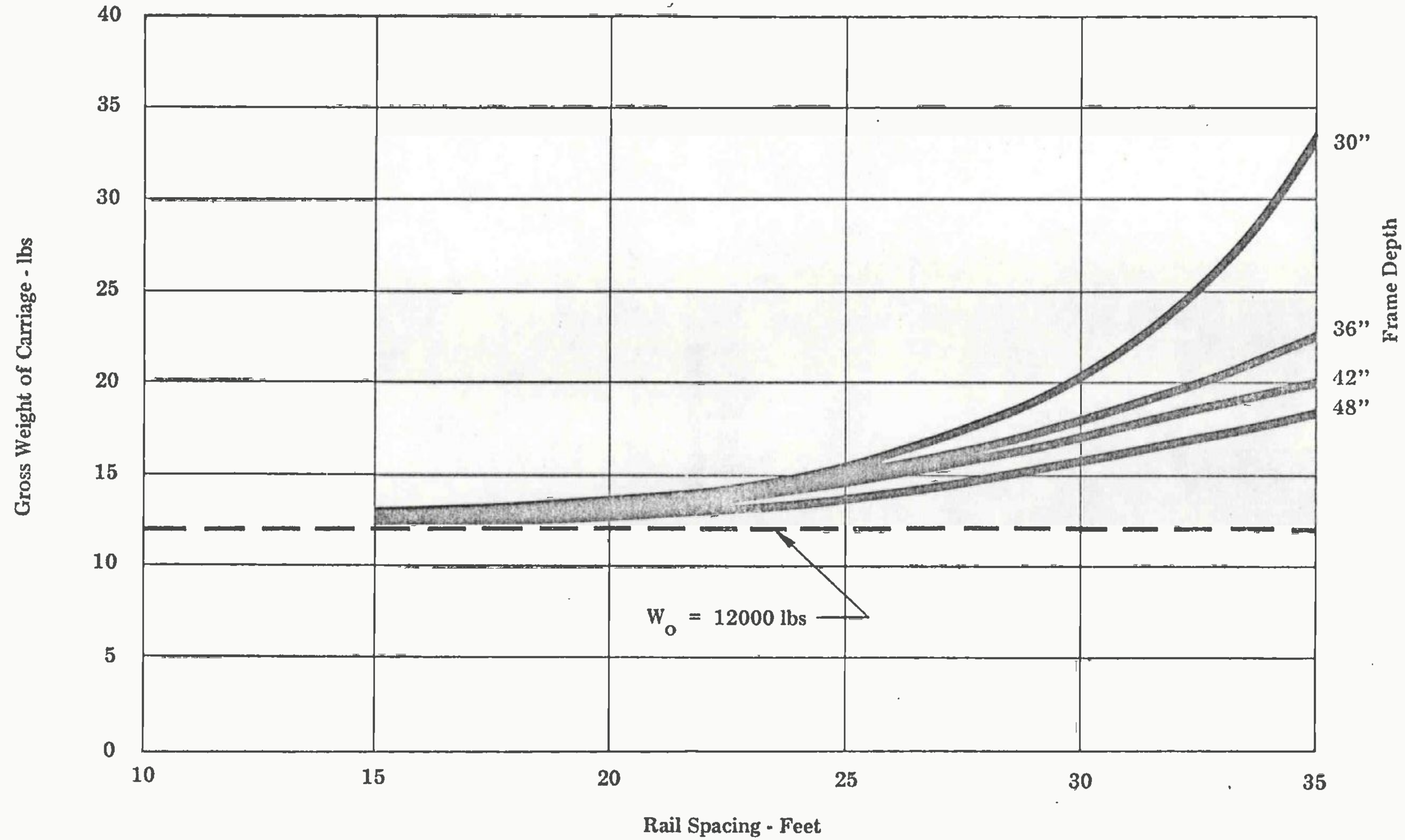


FIGURE 6 GROSS WEIGHT CAHARACTERISTICS OF LOW SPEED CARRIAGE

On the basis of this comparison, a 33.5-foot rail spacing was selected. Quiet operation, improved rail alignment, better visibility and more realistic rail support foundation requirements were the deciding factors.

2. Rail Configuration

Most towing tank facilities throughout the world use either a crane rail or rectangular bar as a means of guiding carriage suspension systems. Since many of these carriages have on-board propulsion, either a solid or pneumatic drive wheel is required to provide traction. In order to keep contact stresses and wear within reasonable limits, a flat rolling surface is required. However, the performance requirements for the USNA's towing tank facility are such that on-board propulsion is impractical for either the low speed or high speed carriages. Furthermore, on-board propulsion is not consistent with the requirement for quiet operation. When off-board propulsion systems are considered, the need for a driven wheel and hence a flat rail is eliminated. Other types of suspension systems such as slipper, air bearings, linear ball bushings, and linear roller bearings are compatible with a round rail system. The round rail has been successfully used in a number of installation with guidance requirements quite similar to the Naval Academy Towing Tank Facility. These include, the Daisy Impact Facility at Holloman AFB, the Terrain Simulator at Martin-Orlando and numerous space simulators at NASA-Houston.

Based on these considerations the relative merits of the flat and round rail systems were investigated, and preliminary rail configurations were defined to facilitate a meaningful comparison.

a. Flat Rail

For a suitable flat rail system five machined surfaces are required to provide the necessary alignment for the high and low speed carriage suspension systems. While this rail can be machined from soft, low carbon steel, the required tolerances and complexity of machining operations are much greater than for a comparable round rail. Because of the close tolerances required, it is very difficult to machine the rails in lengths greater than 8 feet. This means twice as many joints are required as compared to the number for the standard 16-foot length of round rail. Furthermore, if a wheeled suspension is used on conjunction with the flat rail, the joints should be welded and ground to prevent rail joint impacts. The cost of the flat rail is approximately $30 per foot. Alignment of the flat rail is considerably more difficult than the round rail since rotations as well as vertical and lateral adjustments are required. Furthermore, these adjustments cannot be made independently as for the round rail, thus greatly complicating the alignment operation.

The principal advantages of the flat rail are its compatibility with the wheeled suspension system and on-board power requirements and the flexibility in design to provide greater stiffness and consequently fewer supports than would be needed for a round rail.

b. Round Rail

The round rail is much easier to fabricate and can be machined to closer tolerances than a rectangular rail. Four-inch diameter rails of 1060 steel, 60 case hardened and ground can be provided as off-the-shelf items in 16-foot lengths. Diametric tolerances can be held to ± .0006 inches and straightness to .0005 inches per foot. The cost of these rails in the 4-inch diameter size is approximately $25 per foot. The extremely hard surface finish on these rails will contribute to their durability and long service life. The principal advantage of the round rail is their compatibility with off-the-shelf alignment mounts and the simple independent means of adjusting the rails to achieve the precise alignment required. Based on these considerations a round rail system was selected.

3. Rail Support Structure

A satisfactory rail support structure for the USNA Towing Tank must have sufficient rigidity to maintain carriage alignment and to prevent excitation of carriage vibration due to support deflections. To insure that this requirement is met, the deflection of the rail system was limited to .001 inches as a result of the most critical carriage suspension loading.

There are a number of advantages in using a large diameter rail with a minimum number of supports. Most importantly, the complexity of rail alignment is reduced as the number of support is reduced. Also, the larger the rail diameter, the greater the load capacity and life of the associated suspension system.

The load capacity of various rail systems having equivalent stiffness was investigated. By using 60 case hardened and ground steel rail having an ultimate tensile strength in excess of 300,000 psi, an allowable working stress of 60,000 psi would provide a safety factor of 5. Although even the 1-inch diameter rail has adequate strength, the increased load capacity of the larger diameter rails afford much greater flexibility in the future usage of the facility.

As a result of the considerations of stiffness, load capacity, total cost and east of alignment, a 4-inch diameter rail system was selected. The rail will be supported on approximately 21-inch centers by the standard off-the-shelf waymounts. The load capacity of this rail system will be in excess of 100,000 pounds. The waymount provides independent alignment of the rail in the vertical and horizontal planes by means of screw adjustments. A total adjustment of 1/8 inch can be accommodated in each direction. The waymounts will be bolted to a 1-inch thick steel plate which in turn will be firmly anchored on the concrete foundation.

4. Rail Alignment

During sea-keeping tests the low speed carriage frame will be used as a spacial reference to make wave height measurements to a desired accuracy of ± .010 inches. The following error sources contribute to the inaccuracy of these measurements.

Error Source		
Rail Alignment	.002 (min)	.008 (max)
Rail Deflection	.001	
Frame Deflection Due to Live Load	.001 (min)	.005 (max)
Mechanical Vibration of Frame	.001	
Accuracy of Measurement	.002	
RMS Error	.0035 (min)	.010 (max)

In order to obtain a probable wave height error of $\pm$ .010 inch, the rail alignment error could be as large as $\pm$ .008 inch. However, for a 99.7% probability (3σ) that the wave height measurement is accurate to $\pm$ .010 inch, the rail alignment error should not exceed $\pm$ .002 inch.

Alignment of the rails in the horizontal plane is not as critical. However, reasonable alignment must be maintained to prevent excessive carriage vibration and to minimize model angles of attack. Since the type of suspension system recommended for the high and low speed carriages only use one rail for lateral constraint , the alignment tolerances on the secondary rail can be relaxed in the lateral direction. Based on these considerations, the following rail alignment tolerances were selected.

Alignment Tolerances	Master Rail	Secondary Rail
Vertical	$\pm$ .002 in	$\pm$ .002 in
Lateral	$\pm$ .004 in	$\pm$ .010 in
Straightness	.0005 in/foot	.0005 in/foot

Several existing towing tank facilities have achieved rail alignment tolerances of $\pm$ .006 inch, and the 250-foot tank at the University of Glasgow held rail alignment to $\pm$ .004 inch. By utilizing a laser alignment system it appears feasible to achieve the alignment tolerances indicated for the USNA Towing Tank Facility.

The Laser Alignment System projects a low-power (under 0.5 milliwatt) coherent light beam along an optically straight path for operational distance of 300 feet and beyond. The horizontal and vertical displacement of the rail from the center of the laser light beam can be measured to an accuracy of 0.001 inch for the first 150 feet and to within .002 inch over the second 150 feet. Displacement is detected by photo-electric sensors in a detector target and displayed as horizontal and vertical displacement components on direct-reading meters of the readout unit. A water trough will be used in conjunction with this system in order to establish benchmarks at the four corners of the towing tank. Since the rails must be a constant height above the water, they will have a slight crown due to the curvature of the earth. The magnitude of this curvature will be calculated and compensated for during rail alignment.

C. Carriages

As previously stated two carriages will be provided to accomplish the wide range of testing envisioned for the facility. The high speed carriage will be used generally at velocities of from 15 to 30 knots (25 to 50 fps), while the low speed carriage is capable of operation in the range from 15 knots (25 fps) on downward. Testing with the high speed carriage will be primarily with submerged models while the low speed carriage will be used for traditional seakeeping and resistance testing.

1. Carriage Suspension

Of special note are the unique suspension systems selected for use with the towing carriages.

Since a primary mission of the high speed carriage is to conduct flow noise measurements, quiet operation is the principal requirement of its suspension system. Secondly, the suspension system must have sufficient stiffness to limit model deflection to tolerable angles of attack, and to prevent resonant excitation by the cable drive propulsion system. Thirdly, the suspension must have sufficient load capacity to carry the large overturning moments due to model drag, and to accommodate sizeable lift forces induced by hydrofoil testing. Fourthly, from an operational viewpoint, the suspension should provide minimum wear and maintenance. Finally, since the inertia of the high speed carriage dictates the power requirements for the tow cable propulsion system, it is desirable to have a lightweight suspension system for the high speed carriage.

The low speed carriage will be used primarily to conduct seakeeping and resistance-type tests. As a consequence, its suspension must have the capabilities for providing precise carriage alignment and for affording accurate speed control. This implies the need for a stiff suspension, free of mechanical vibration and having very low frictional resistance. Secondary considerations include long life, high load capacity and compactness so as not to interfere with the cable drive and secondary braking systems.

A qualitative comparison of candidate suspension systems for the high speed carriage is presented in Table 1 . Based on this comparison, a slipper suspension system was selected for flow noise test operations. Quiet, vibration free operation is the overwhelming consideration affecting this selection. It is recognized that poor stick/slip characteristics of the slipper suspension system precludes its use for low speed operation. To overcome this problem a secondary suspension system utilizing linear roller bearings was selected. A simple, quick method of changing suspensions is provided. The benefits which can be derived from a dual suspension system

TABLE 1 COMPARISON OF CANDIDATE SUSPENSION SYSTEMS FOR HSC

Suspension Characteristic	Suspension System				
	Rubber Wheels	Slippers	Linear Ball Bushings	Linear Roller Bearings	Air Bearings
Mechanical Vibration	Poor	Excellent	Good	Good	Very Good
Noise of Operation	Very Good	Excellent	Poor	Fair	Good
Stick Slip	Very Good	Poor	Very Good	Very Good	Excellent
Life	Very Good	Very Good	Good	Excellent	Good
Load Capacity	Fair	Very Good	Good	Excellent	Fair
Stiffness	Poor	Very Good	Good	Very Good	Fair
Constraint	Poor	Very Good	Good	Poor	Fair
Self Alignment	Good	Good	Good	Excellent	Good
Weight	Fair	Very Good	Very Good	Good	Fair
Volume	Poor	Very Good	Very Good	Good	Fair
Cost	Good	Good	Good	Good	Poor
Maintenance	Good	Good	Good	Very Good	Poor
Simplicity	Good	Good	Good	Very Good	Poor
Rail Requirements	Flat	Flat or Round	Round	Flat or Round	Flat or Round

outweigh the weight and cost penalties. An artist's sketch of the slipper is shown in Figure 7.

A qualitative comparison of candidate suspension systems for the low speed carriage is presented in Table 2. On the basis of this comparison a linear roller bearing suspension system was selected. Vibration-free operation, low friction characteristics, and suspension stiffness are the primary reasons for selecting this type of suspension system. Secondary, but important considerations are good load capacity, long life, self-aligning features, compactness, minimal maintenance and low cost. An artist's sketch of the Roundway bearing is shown in Figure 8.

2. High Speed Carriage

The high speed carriage has been designed for the purpose of conducting flow noise tests on submerged models at speeds up to 50 fps. It will also be used for tests on high performance surface craft which require speeds between 25 and 50 fps. Because of the high accelerations and decelerations associated with this carriage, it will be unmanned. In addition to its primary role as a test vehicle, it will also be used to tow the low speed carriage at speeds up to 25 fps. The principal requirement of the carriage in its primary role is quiet operation, whereas its secondary usage demands precise speed control.

An artist's sketch of the high speed carriage is shown in Figure 9. The basic structure consists of two transverse truss frames which are bolted to two longitudinal truss frames, all of which are constructed from aluminum extruded sections. The longitudinal and transverse

TABLE 2 COMPARISON OF CANDIDATE SUSPENSION SYSTEMS FOR LSC

Suspension Characteristics	Suspension System				
	Steel Wheels	Slippers	Linear Ball Bushings	Linear Roller Bearings	Air Bearings
Mechanical Vibration	Fair	Excellent	Good	Very Good	Very Good
Noise of Operation	Good	Excellent	Fair	Fair	Good
Stick Slip	Very Good	Poor	Very Good	Very Good	Excellent
Life	Good	Fair/Poor	Poor	Very Good	Good
Load Capacity	Good	Good	Poor	Very Good	Fair
Stiffness	Good	Fair	Fair	Very Good	Fair
Constraint	Good	Very Good	Good	Good	Fair
Self Alignment	Poor	Good	Good	Excellent	Good
Weight	Fair	Good	Good	Good	Poor
Volume	Poor	Very Good	Very Good	Good	Fair
Cost	Fair	Good	Good	Good	Poor
Maintenance	Good	Good	Good	Very Good	Poor
Simplicity	Good	Good	Good	Very Good	Poor
Rail Requirements	Flat	Flat or Round	Round	Flat or Round	Flat or Round

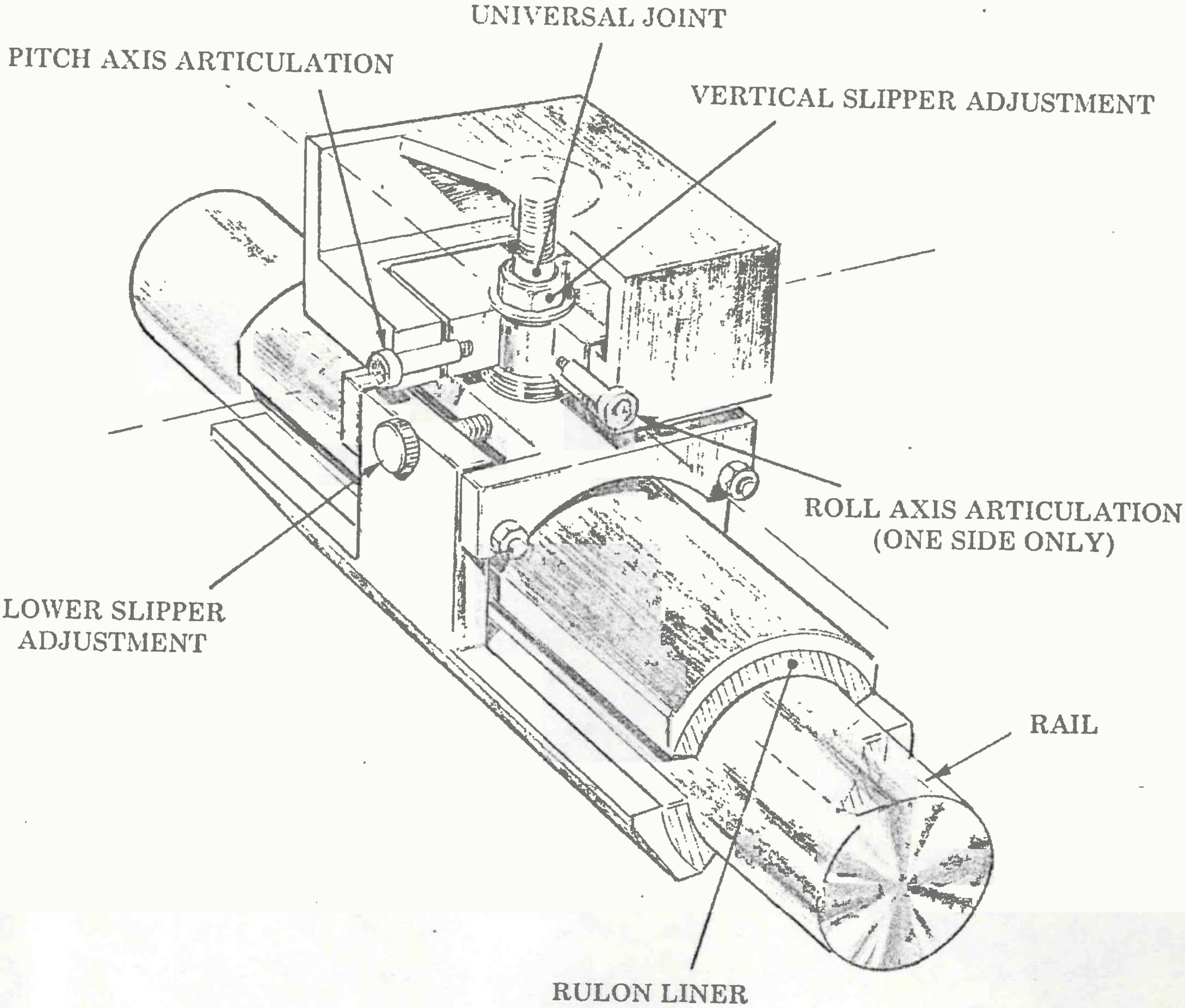


FIGURE 7 SLIPPER SUSPENSION SYSTEM FOR HSC

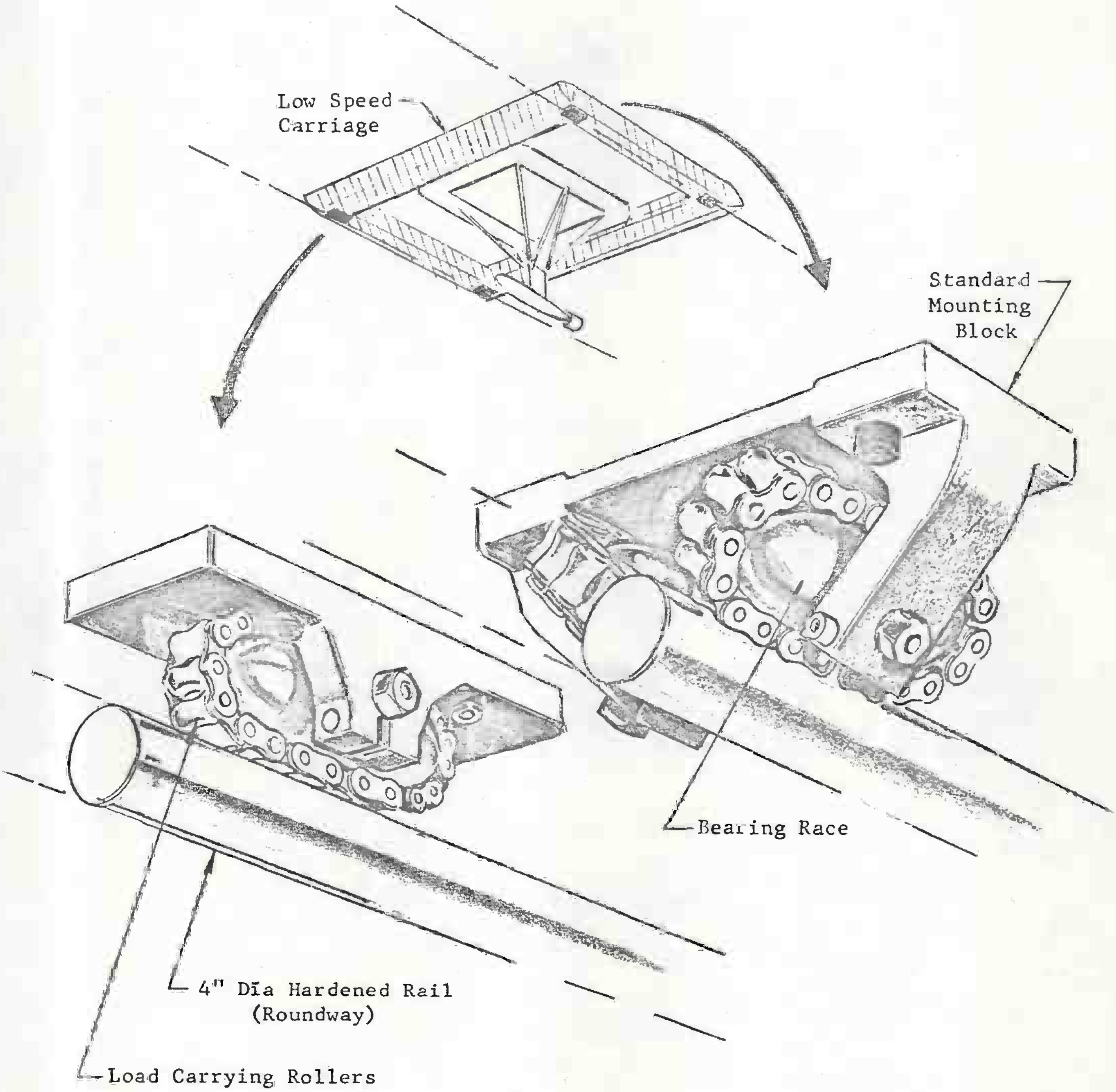


FIGURE 8 ROUNDWAY BEARINGS FOR LOW SPEED OPERATION

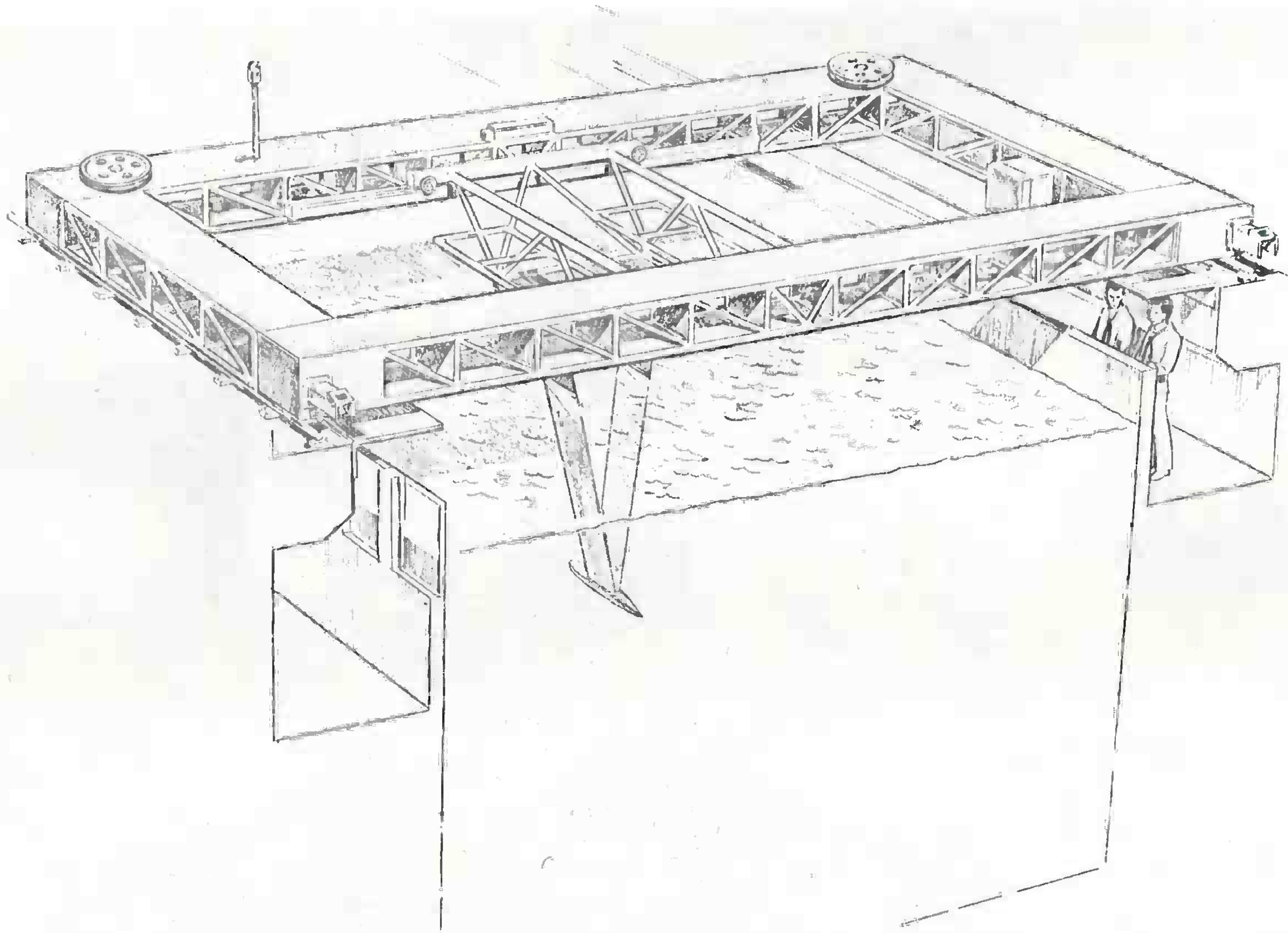


FIGURE 9 HIGH SPEED CARRIAGE

frames are of welded construction with cover plates to provide additional stiffness as well as to allow personnel to walk over to the instrumentation module. The original concept for the high speed carriage consisted of a monocoque structure typical of aircraft type construction with a wedge fairing along the front of the foremost transverse box beam. However, aerodynamic tests conducted in the wind tunnel at the U.S. Naval Academy with models of both carriages indicated that an overpressure* field is induced upstream of the leading carriage which tends to become an underpressure function beneath and downstream of the carriage. This was considered to be a critical condition for low speed carriage operation since a wave system produced by the induced pressure fields would appear in the area under the low speed carriage where the test items would be located. It was decided to change the high speed carriage construction to the open truss-type since tests run with only the low speed carriage in the wind tunnel indicated that the potential water surface disturbance would be lessened for this type of construction.

In order to meet the requirements of quiet operation during flow noise testing, and accurate speed control during low speed operation, the high speed carriage is equipped with a dual mode suspension system. Slippers with "Teflon" composition liners will be used during high speed operation while linear roller bearings will be used for low speed operation. The configuration of these suspension systems are shown in Figures 7 and 8 .

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* Overpressure is used here in the sense of positive pressure, i.e., a pressure that tends to depress the water surface. Underpressure or negative pressure thus tends to elevate the water surface.

Suspensions can be changed in a matter of minutes by means of four gear boxes which raise and lower the upper bearing surfaces of each slipper. Limit switches will be used to maintain carriage alignment. Both suspensions are self-aligning in pitch, roll and yaw, and have built-in adjustments for vertical positioning. All lateral loads are reacted by the suspensions on the right side of the carriage while the suspensions on the left side are articulated to accommodate variations in rail spacing.

Idler sheaves are provided at the four corners of the carriage as anchorages for the tow cables. These sheaves accommodate variations in cable length from side to side, and equalize cable loads. The idlers are located so as to take maximum advantage of carriage frame strength and stiffness.

The high speed carriage supports an instrumentation module which is readily removable to facilitate the installation and calibration of instrumentation. The module is a welded assembly, constructed from aluminum tubing. The height of the instrument module structure has been limited to five feet in order to provide head room clearance when handled by the monorail hoist. A construction joint is provided at the base of the module to attach and align the model strut support structure. The module is supported by a linear roller bearing suspension system which rides on one inch diameter stainless steel rails. The rails are supported by aluminum box beams which are attached to the inside face of the transverse frame members at each rib location. The rail system allows the model to be positioned laterally between the tank side walls. Clamps are provided at each corner of the instrument module to lock it in position and to carry the overturning moments induced by model drag.

3. Low Speed Carriage

The low speed carriage has been designed primarily for the purpose of conducting seakeeping and resistance tests at speeds up to 25 fps. The carriage will be used to support the Naval Systems Engineering curriculum, for midshipman and faculty research and for Navy sponsored research which is not formally curriculum-connected. In keeping with the first objective, the carriage will accommodate approximately 22 midshipmen. Model visibility was an important design consideration and led to a carriage with a large open center well, and a box beam-type structure which also serves as an observation platform. The instructor may stand on catwalks within the well while explaining the experiment under way. Since the carriage will be manned, maximum accelerations and decelerations will be limited to 0.25 g's in order to avoid elaborate restraint of passengers.

An artist's sketch of the low speed carriage (without instrumentation module) is shown in Figure 10 . The carriage frame consists of four rectangular box beams having welded tubular truss skeletons, with heavy aluminum plates bolted to their upper and lower faces. An open truss was selected in preference to a closed box beam in order to minimize aerodynamic interference with the water surface. The upper cover plate not only adds considerable stiffness to the box beams, but also serves as an observation platform. Bolting the plates to the welded truss will add considerable damping to the carriage frame. The transverse frame members are tapered in

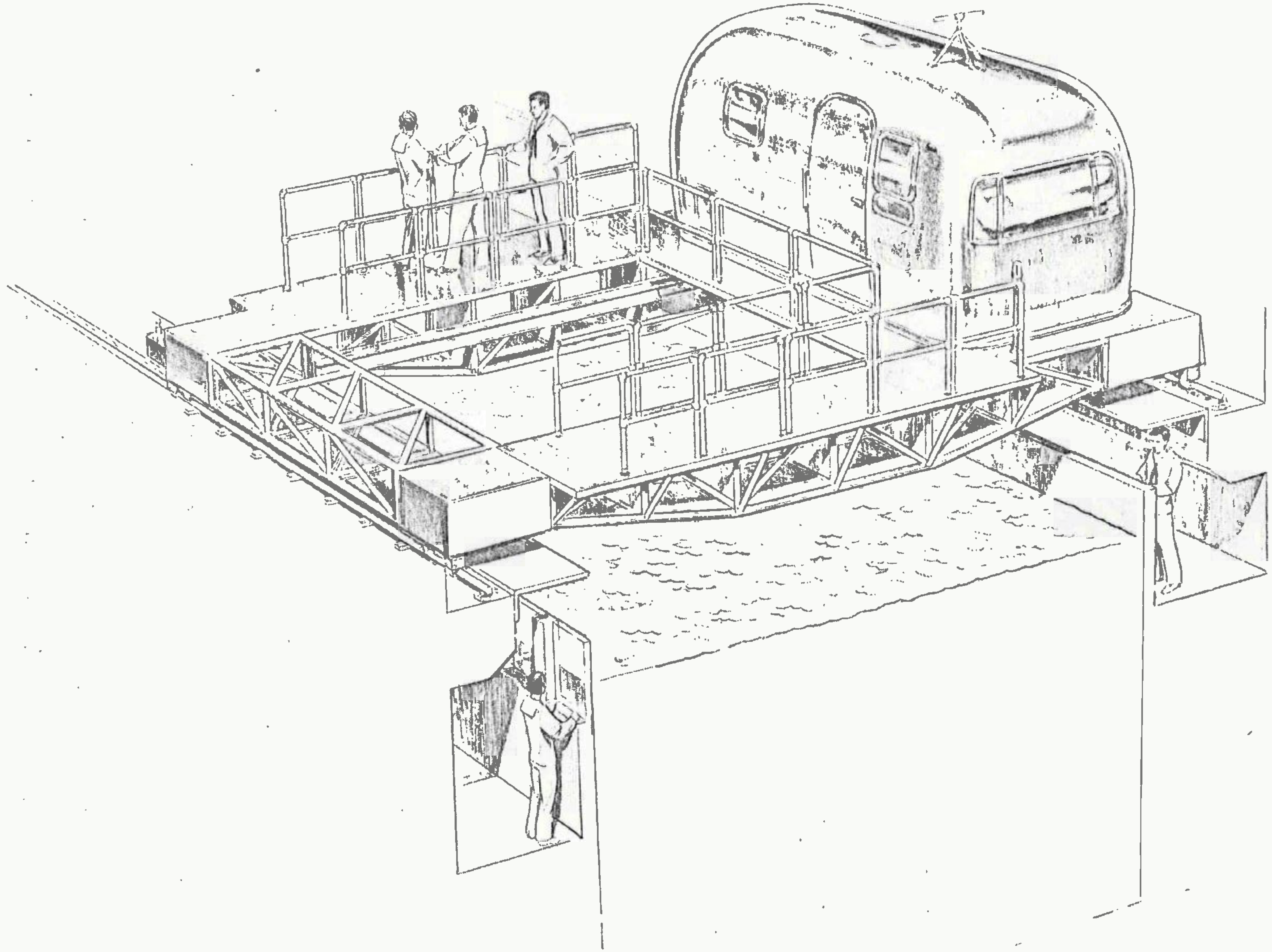

FIGURE 10 LOW SPEED CARRIAGE

order to provide maximum stiffness within the available envelope. Since the low speed carriage will be used as a spacial reference for wave height measurements frame stiffness is of paramount importance. Bolted construction joints are provided between the longitudinal and transverse frame members to facilitate installation and assembly within the towing tank building. A secondary platform is cantilevered from the front of the carriage frame to accommodate miscellaneous instrumentation without blocking the walkways. Removable handrails are provided around the center well and the outer perimeter of the carriage.

The low speed carriage is provided with a linear roller bearing suspension system illustrated previously in Figure 8 . This type of suspension provides the alignment, rigidity, and friction-free operation necessary for the type of tests conducted by this carriage. A Vee-type mounting arrangement on one side of the carriage carries both vertical (downward) and lateral loads. A single mounting arrangement on the opposite side of the carriage accommodates only vertical loads and allows for variations in rail spacing. While it is not anticipated that the low speed carriage suspension will experience vertical up-loads, hold-down slippers are provided for emergency conditions.

An 18-foot Airstream trailer body is mounted on one side of the low speed carriage to house the carriage speed control system and model instrumentation. The airstream trailer is fabricated from stretched aluminum skins to provide a lightweight, streamlined configuration. The good aerodynamic characteristics of this cab will minimize drag and tend to alleviate the blockage problem within the restricted cross-section of the towing tank facility. The trailer body is fully insulated and will be purchased with a built-in roof mounted air conditioner to provide environmental control for instrumentation. The wide safety glass windows on all sides of the trailer will provide the speed control operator with the necessary visibility for safe operation.

As in the high speed carriage, the low speed carriage has a rail system within its center well which can accommodate modular instrumentation packages, subcarriages, dynamometers and catwalks. Two one-inch diameter stainless steel rails on approximately 11-foot centers span the width of the carriages and obtain their support from the transverse frame members. All modular units are equipped with linear roller bearing suspension. The low frictional characteristics of this suspension will allow lateral positioning of the modules with a minimum of effort. The rigidity of the suspension will insure precise positioning of the modules at all times. The length of the rails will permit the positioning of the models 5 feet off center in order to allow clear photography and observation of models and wave patterns. The modules can be rigged in the shop area and transported to the towing tank for mounting and calibration, without interrupting operation of the high speed carriage.

Catwalks are provided on either side of the instrument modules to serve as observation and photography platforms and to provide access to the models. The catwalks will be suspended from the rail system on linear roller bearings, to allow lateral positioning. Lift devices are provided at either end of catwalks to obtain vertical positioning which is necessary when wave tests are being conducted and the carriage is operated in the drydock area.

Couplings are provided on the forward end of the carriage to permit it to be towed by the high speed carriage (See Figure 11). Hydraulic shock absorbers are built into the couplings to absorb low velocity impacts of the carriage. In addition, braking pads are provided on the aft end of the carriage which will engage stationary hydraulic shock absorbers in the event of a failure of the primary braking system during a return run.

D. Velocity Control System

Once the use of an off-board carriage propulsion system using a cable drive had been selected for the towing system it was possible to investigate the control systems to determine the best system capable of providing the required speed regulation. Some of the factors which had to be considered were the type and location of the carriage velocity feedback transducer, the optimum type of control loop and associated gains, the compensation network parameters, and the optimum command acceleration and deceleration profiles. In addition, the effects of varying the physical characteristics of the drive system were evaluated. This included motor horsepower, drive system inertia characteristics, cable damping and stiffness, model drag, and suspension system friction.

A manual solution of the control system differential equations is virtually impossible because of nonlinearities caused by significant changes in cable stiffness and damping with carriage travel. A digital computer solution

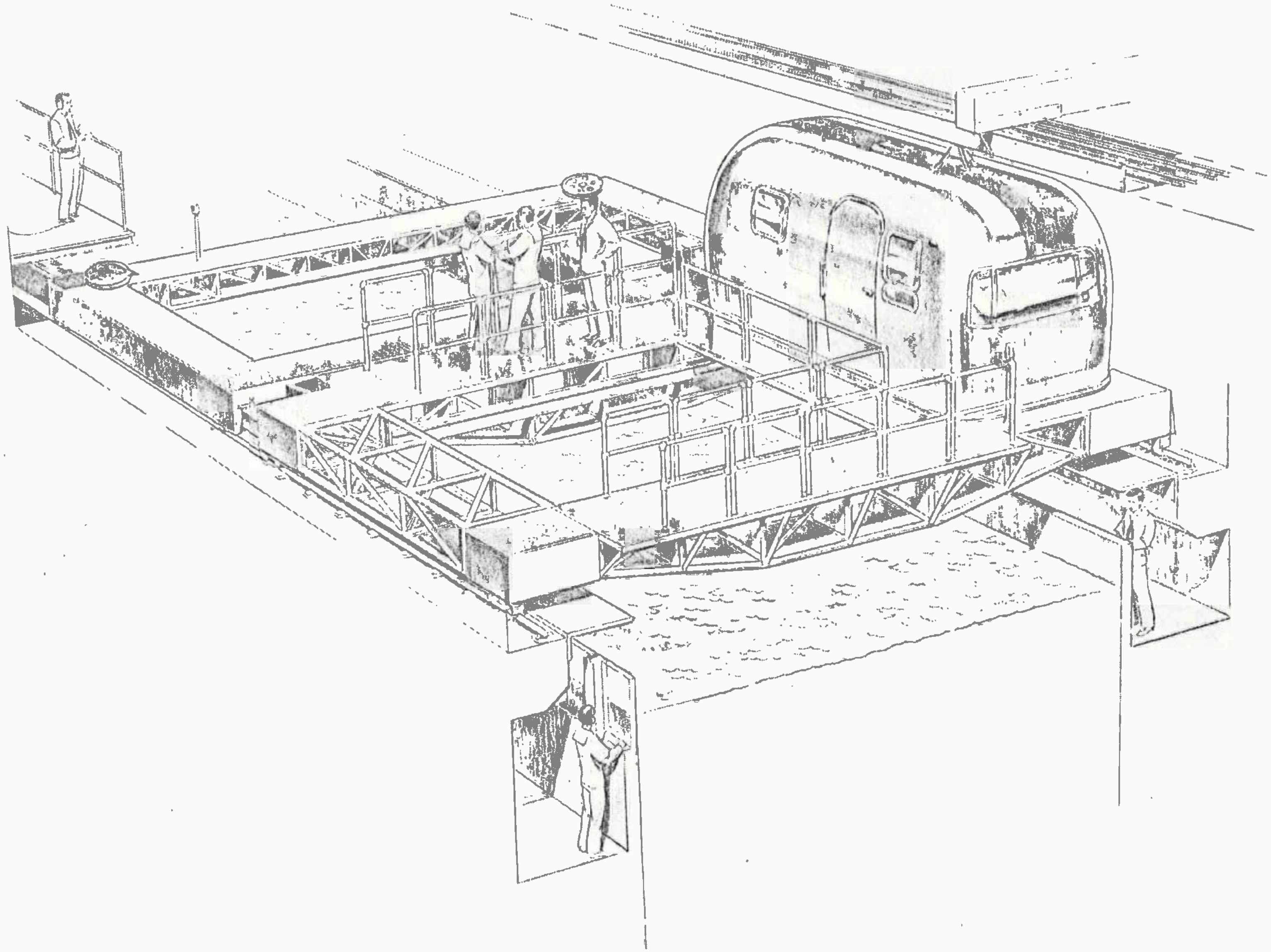

FIGURE 11 CARRIAGES COUPLED TOGETHER

would have been quite expensive, because of the large number of integrations to be performed and the amount of computer time involved. Since a EAI-231R analog computer was available at the Naval Academy, it was used to accomplish these studies.

Based on analog computer data covering a wide range of control system configurations and mechanical parameters, a system was formulated which yields a zero velocity error for the high speed carriage with transients damped, within 90 feet of carriage travel. Even with this control loop, certain restraints must be placed on inputs and loop gain in order to achieve the desired velocity control. The analog studies indicated that to avoid instabilities, the tachometer must be mounted on the motor rather than the carriage if a loop gain of 6 or more is used. It was shown that a command acceleration of 30 fps^2 will provide zero high speed carriage velocity error after 90 feet of carriage travel. It also shows that a command acceleration of 55 fps^2 will provide the minimum acceleration distance of approximately 70 feet. Similarly, it was shown that a command acceleration of 8 fps^2 will provide a zero velocity error for the low speed carriage after 75 feet of travel.

The principal advantage of the system finally selected is that the integration of the velocity error is performed in the digital part of the servo rather than in the analog amplifier. This eliminates the need for the time base and provides continuous rather than sampled velocity error data. As a consequence, carriage velocity error depends only on the accuracy of the input frequency source and excitation due to external disturbances such as variations in suspension friction, and longitudinal load fluctuations caused by testing in waves or using a planar motion mechanism. The requirement for a precision frequency source and more sophisticated circuitry makes this a more complex and costly system, but is much more likely to meet the velocity control requirements.

Based on the analog computer and mechanization studies, it was determined that a velocity control system capable of damping velocity transients to .1% of set speed within the available acceleration distances, and providing average speed regulation to .02% of set speed is possible. Furthermore, to obtain the .02% speed regulation a digital tachometer is required, with integration of the velocity error performed in the digital part of the servo rather than in the analog amplifier. Optimum performance is obtained with a gain of 12.5 for high speed operation, and 5 for low speed operation. Corresponding compensation network time constants are .1 for high speed operation and .25 for low speed operation. Two 200 HP DC electric motors operating at 400% overload will provide ample power.

Control of the high speed carriage is accomplished by an operator located in the main control room near the drydock end of the tank. The low speed carriage may be controlled from either the main control console or from an onboard console located in the instrumentation cab. The following is a list of the controls and displays on that panel and a description of what they indicate and/or control. Figure 12 shows the actual layout of the main control panel.

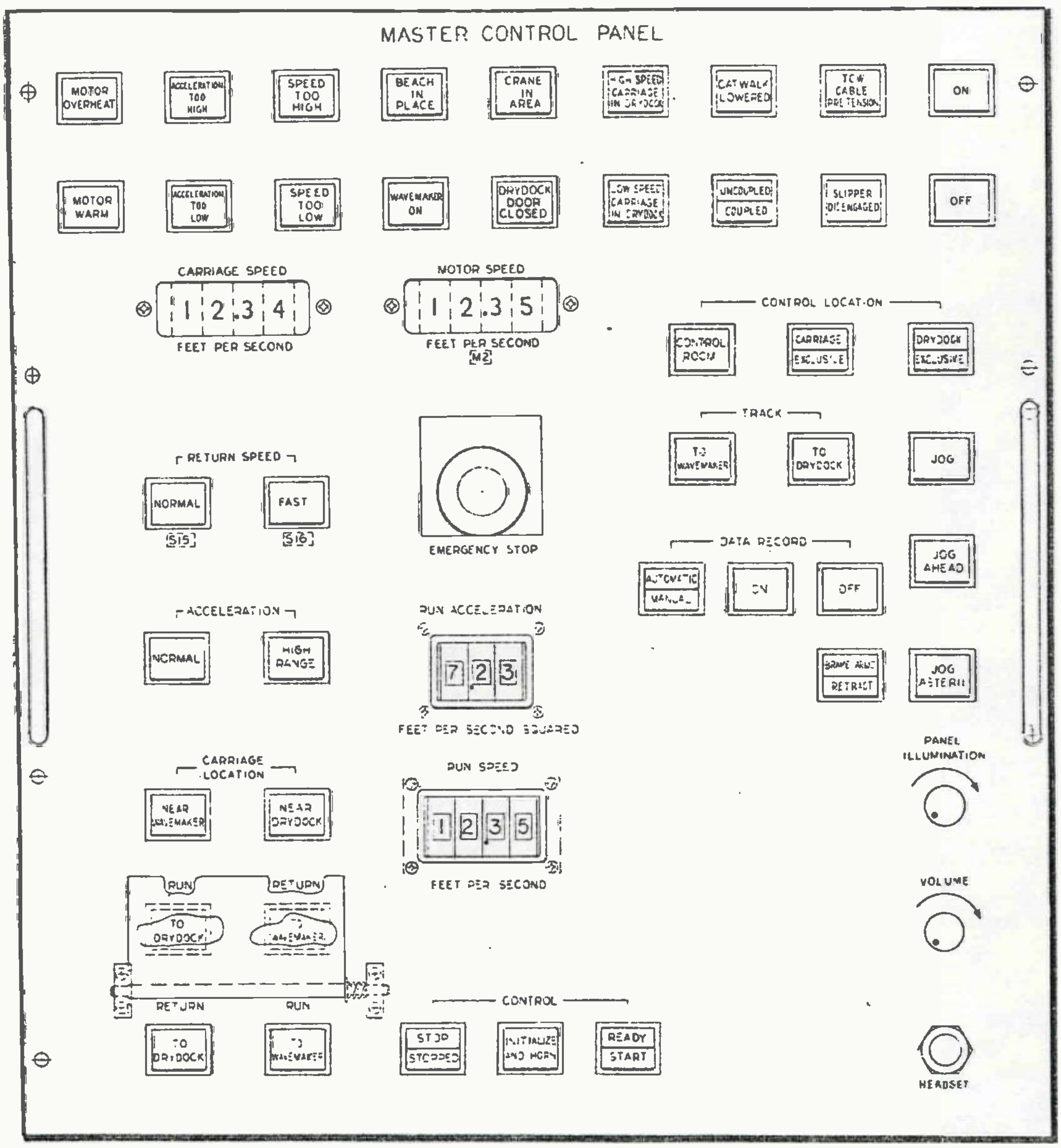

FIGURE 12 MAIN CONTROL PANEL

CONTROLS/DISPLAYS DESCRIPTION

NOMENCLATURE	DESCRIPTION
MOTOR OVERHEAT indicator	Lights red when drive motor overheats, out when drive motor not overheated.
MOTOR WARM indicator	Lights yellow when drive motor begins to overheat as a caution note.
ACCELERATION TOO HIGH indicator	Lights yellow to indicate too high acceleration value has been selected with the run acceleration switches. System will operate, but at permitted maximum acceleration. Out when proper acceleration value has been selected.
ACCELERATION TOO LOW indicator	Lights yellow if too low a value of acceleration has been entered with the run acceleration switches, as a caution note. Out when a proper acceleration value has been entered.
SPEED TOO HIGH indicator	Lights red when too high a run speed value has been entered. System will not operate until corrected. Out when a proper range speed value has been entered.
SPEED TOO LOW indicator	Lights yellow when too low a run speed value has been entered, as a caution note. Out when proper speed value has been entered.
BEACH IN PLACE indicator	Illuminates white to denote the beach has been put in place in the tank. Automatic stop initiated to not strike beach.
WAVEMAKER ON indicator	Lights white when the wavemaker is turned on as a status indication; when the wavemaker is off, the light is out.
CATWALK LOWERED indicator	Status display indicating white, red or out; lights white when the catwalk is lowered and no hazardous activity is going on, lights red when the catwalk is in a lowered position and the carriage is at the drydock. Out when the catwalk is not in the lowered position.

CONTROLS/DISPLAYS DESCRIPTION (CONT'D)

NOMENCLATURE	DESCRIPTION
DRYDOCK DOOR CLOSED indicator	Status indicator lighting white when the drydock doors are closed, lighting red as a danger indication when the interlock senses HSC in the drydock or carriages are coupled and either is in the drydock. Only Jog mode is active when red light is on. It is out when the gate is in an open position.
HIGH SPEED CARRIAGE IN DRYDOCK indicator	Status indicator illuminating white when the high speed carriage is in drydock, it is out when the high speed carriage is not in drydock.
LOW SPEED CARRIAGE IN DRYDOCK indicator	Status indcator lighting white when the low speed carriage is in the drydock, when the low speed carriage is not in drydock, indicator is out.
CRANE IN AREA indicator	A warning light that illuminates red when a crane is positioned over the tank. In this status, it also inhibits the run mode. If the crane is clear of the tank, the indicator is out.
LOW SPEED CARRIAGE COUPLED-UNCOUPLED indicator	Interlock indicator illuminating the appropriate section white then the low speed carriage is fully coupled to or completely uncoupled from the high speed carriage. One or the other must be accomplished to allow operation.
TOW CABLE PRE-TENSION indicator	Lights red if the cable pre-tension is too low or too high and operation is inhibited. This is checked prior to run but is disabled by the start signal. The indicator is out if the cable tension is within the proper range.
SLIPPER DISENGAGED indicator	Status indicator lights yellow if any slipper is disengaged and the speed selection is above the specified limit for roundway bearing operation.

NOMENCLATURE	DESCRIPTION
ON switchlight	Momentary switch, lights green when power is applied to the Master Control Console, out when power not applied, i.e., when the OFF switchlight is activated.
OFF switchlight	Momentary switch lights white when power is not applied to the Master Control Panel. Out when the power is applied, i.e., the ON switchlight is activated, or when power is turned off at the Master Power Panel.
CARRIAGE SPEED display	A 4 digit read out displaying carriage speed from 00.00 to 99.99 feet per second.
MOTOR SPEED display	A 4 digit readout displaying motor speed in terms of equivalent carriage speed.
RETURN SPEED - NORMAL and RETURN SPEED - FAST switchlights	Selector switches that illuminate white when selected and are out when not selected. Selects the normal or fast mode to return the carriage to its start position. Can be operated during return. Set to normal at end of return.
ACCELERATION - NORMAL and ACCELERATION - HIGH RANGE switchlights	Selector switches that illuminate white when selected, are out when not selected. Selects the normal or high range (if permitted) of acceleration for a run by conditioning the system circuitry and interlocks for the acceleration selection. Enables the RUN ACCELERATION switches in either the X.XX (normal) or XX.X (high range) configuration. Set at end of run to normal.
CARRIAGE LOCATION-NEAR WAVEMAKER indicator	Lights white when the carriage is near the wavemaker, out when it is not. This is a limit switch actuated status indication.

NOMENCLATURE	DESCRIPTION
CARRIAGE LOCATION - NEAR DRYDOCK indicator	Lights yellow to note caution when the carriage approaches the drydock, out when the carriage is not near the drydock. This is a limit switch actuated indicator.
RUN-RETURN switchlights	A series of four switchlights that illuminate white when activated, used to select the direction the carriage will move on a given run. There are two sets of switchlights labeled TO DRYDOCK and TO WAVEMAKER, one of which is covered (not applicable to a run.)
EMERGENCY STOP switch	This is a large red "push-to-activate" type switch that immediately stops carriage motion at maximum deceleration available for the operating carriage. Must be reset manually.
RUN ACCELERATION switches	Three thumbwheel switches used to select the run acceleration value. Values go up to 9.99 ft/sec^2 or 99.9 ft/sec^2 depending on whether the ACCELERATION-NORMAL or ACCELERATION HIGH RANGE is selected.
RUN SPEED switches	Four thumbwheel switches used to select the carriage speed for the run, expressed in hundredths of feet per second up to 99.99 ft/sec.
CONTROL-STOP/STOPPED switchlight	This is a momentary switch that, when activated, initiates the stop command to the carriage. The top half of the indicator, STOP, illuminates yellow when the switchlight is depressed to indicate the stop command has been initiated, or an automatic stop is initiated by the control system; the lower half of the indicator, STOPPED, indicates white after the carriage comes to a stop, when STOP goes out.

CONTROLS/DISPLAYS DESCRIPTION (CONT'D)

NOMENCLATURE	DESCRIPTION
CONTROL-INITIALIZED AND HORN switchlight	This is a momentary switch that illuminates white when depressed and for 15 seconds thereafter. It activates the initialization of the system logic, sounds the warning horn, and lights the flashing light. Prior to starting a run, this switchlight must be depressed.
CONTROL-READY/START switchlight	This is a momentary switch that, when enabled and activated, starts the carriage run. The top half of the switch, READY, is illuminated green when the carriage is ready to begin the run, i.e., after the initialization of the logic. When depressed the READY portion of the indicator goes out and the lower half, START, illuminates white to show that the carriage has started to move.
CONTROL LOCATION-CONTROL ROOM switchlight	Momentary switch that, when depressed, transfers the carriage control to the CONTROL ROOM. It is interlocked with the other CONTROL LOCATION switches and is lit white when selected.
CONTROL LOCATION-CARRIAGE/EXCLUSIVE switchlight	This is a split lens switchlight that, when depressed, illuminates the upper half, CARRIAGE, white to note that control has been transferred to the carriage. If control freeze is activated from the carriage, the bottom half of the switchlight, EXCLUSIVE, will illuminate yellow. Control cannot be removed from the carriage under this condition.
CONTROL LOCATION-DRYDOCK EXCLUSIVE switchlight	This is a split lens indicator switchlight that, when depressed, illuminates the upper half, DRYDOCK, white to note that control has been transferred to the drydock. If control is then frozen from the drydock, the bottom half of the switchlight, EXCLUSIVE, will illuminate yellow to note that control cannot be returned to the Master Control Room unless released at the drydock. This is done as a safety feature.

CONTROLS/DISPLAYS DESCRIPTION (CONT'D)

NOMENCLATURE	DESCRIPTION
TRACK-TO-WAVEMAKER switchlight	This is a momentary switch that, when depressed, illuminates white and enables carriage movement towards the wavemaker in the track mode of operation.
TRACK-TO-DRYDOCK switchlight	This is a momentary switch that, when depressed, illuminates white and enables carriage movement toward the drydock in the track mode of operation.
JOG switchlight	This is a momentary switch that, when activated, illuminates yellow to note this system is not in a normal condition and activates the jog mode of operation. It is used in conjunction with the JOG AHEAD and JOG ASTERN controls.
JOG AHEAD switchlight	This is a momentary switch that when enabled by jog mode and depressed, moves the carriage at the jog speed in a forward direction. It illuminates white when activated to denote a status condition.
JOG ASTERN switchlight	This is a momentary switch that, when enabled by jog mode and depressed, moves the carriage at the jog speed in a sternward direction. It illuminates white when activated to denote a status condition.
DATA RECORD-AUTOMATIC/MANUAL switchlight	This is a split-lens alternate action switch that has either the upper, AUTOMATIC, or lower, MANUAL, portion illuminated white depending on which mode of data recording is selected.
DATA RECORD-ON switchlight	This is a momentary switch and indicator that functions in 2 ways. In the AUTOMATIC mode it will illuminate green when the carriage automatically starts the data recording equipment, and in the MANUAL mode the switch must be depressed to begin data recording, again illuminating green when activated.

NOMENCLATURE	DESCRIPTION
DATA RECORD-OFF switchlight	This is a momentary switchlight and indicator that performs in 2 ways. In the AUTOMATIC mode it will illuminate white when data recording is automatically stopped and in the MANUAL mode it must be depressed to stop the recording function, again illuminating white.
PANEL ILLUMINATION rotary control	A potentiometer that control the level of illumination of the panel indicators.
BRAKE ARMS/RETRACTED switchlight	This is a split lens momentary switchlight which, while depressed, will retract the brake arms and permit the HSC to enter the drydock. The BRAKE ARMS legend is illuminated red when the return secondary brake has contracted the brake arms. The RETRACTED legend is illuminated red when the arms are retracted and the return secondary brake can be defeated. Releasing the switch will extend the brake arm.
VOLUME rotary control	This is a potentiometer that controls the volume of the intercom system.
HEADSET connector	A disconnect provided to plug the operators' intercom earphones and microphone into the carriage/master control communication system.

The method of changing from one mode of operation to another is by actuating the new mode, except as specifically described in the previous section under "Exclusive" controls. The STOP control is considered an operating control for this description. For example, if the system is in the "TRACK" mode with "TO WAVEMAKER" selected then the run must be completed to obtain an automatic stop or a manual stop operation has to be performed in order to set in a RUN or a RETURN mode of operation. After the system has stopped, selection of the next mode by operating the appropriate momentary pushbutton switch clears the previous mode.

CONTROLS/DISPLAYS DESCRIPTION (CONT'D)

A control panel is provided for an operator in the instrumentation cab of the low speed carriage. This panel is derived from the panel described above by deleting controls and indicators not essential to on-board operation. The essential controls and displays are in the same relative locations as those of the control room operator panel. The low speed carriage control panel is active only if the carriages are coupled and control assignment has been made from the control room.

E. Braking Systems

The operation of the USNA Towing Tank Facility entails the high-speed transport not only of expensive equipment but of human passengers; thus it is imperative that the design of the facility provide the maximum possible degree of safety. To assure that the moving carriage will always be brought safely to a controlled stop under any circumstances, the primary drive and braking systems will be supplemented by an emergency braking system designed to function automatically in the event of any failure of the primary system.

In order to attain the highest level of safety possible within the space available, the emergency braking system will consist of two independent systems - the secondary and tertiary braking systems. Each of these will be capable of stopping either carriage traveling at its maximum possible velocity with maximum payload, even if all of the other systems fail completely.

Normal braking of the carriages in both the forward and return modes of operation is accomplished through regenerative braking of the tow cable propulsion system. Normal decelerations for the coupled carriages are accomplished at .25g. In the event of a primary propulsion system failure, secondary brake systems are provided to decelerate the carriages from maximum speed in a manner that will not cause any equipment damage, or injury to unrestrained passengers. In the event of a malfunction of the secondary brake system, a tertiary brake system is provided for both the forward and return modes of operation. The tertiary brake systems are designed to decelerate the carriages from maximum velocity in a short distance without causing serious injury to unrestrained passengers, major equipment damage or flying parts.

In the forward mode of operation a cable arresting system is provided as the secondary emergency braking system, with energy dissapation provided by two hydraulic shock absorbers anchoring either end of the arresting cable.

The cable arresting system utilizes off-the-shelf hydraulic shock absorbers to absorb the kinetic energy of the carriages and drive system. In order to utilize existing hydraulic shocks having dimensions compatible with the towing tank facility, and at the same time provide low decelerations (0.6g), a 6:1 reeving system was designed. A 3/4" diameter 6 x 25 Unitlay cable will be used to engage two idler sheaves on the high speed carriage frame.

The cable arresting geometry allows the carriage to travel a maximum distance of 20'-0" before the hydraulic shock bottoms out. However, the tertiary braking system engages the HSC after traveling 16'-0" and brings it to a stop after 20'-0" of travel. This allows 6 inches of margin against snapping the cable during tertiary braking.

The hydraulic shock absorbers for the cable arresting system were designed to provide a nearly constant deceleration of the LSC and associated drive system, over a stroke corresponding to 20'-0" of carriage travel, (35" at the hydraulic shock). This will allow approximately 17'-6" of cable to pay out.

The hydraulic shock absorber will be required to have a free travel (i.e. F = 0) after 27" of travel so that the deceleration force of the cable arresting system is not added to that of the tertiary braking system, thus creating excessive carriage decelerations.

A tertiary arresting system is provided at the machinery room end of the towing tank to absorb the maximum kinetic energy of both carriages in the event of a failure of the cable arresting system. The system utilizes two hydraulic shock absorbers operating in parallel. The shock absorbers have 48 inch strokes, six inch bores, and are of the spring return type. The shock absorbers are engaged by bearing pads on either side of the high speed carriage, and in line with the main rail. Deceleration of the low speed carriage will be kept below 2.5g.

In the return mode of operation both the low and high speed carriages can reach a maximum speed of 15 fps. The secondary brake system in the return mode will also use two hydraulic shock absorbers operating in parallel. These shock absorbers also have 48 inch strokes, and can keep deceleration of the low speed carriage below 1.0g. The shock absorbers will be installed in front of the drydock to prevent the carriages from entering this area at high speed and possibly injuring personnel working in the drydock. Brake arm mechanisms are provided on either side of the high speed carriage frame to engage the hydraulic shock absorbers. The brake arms must be automatically retracted before the carriages can enter the drydock area, and then only at 1 inch per second (which may be changed based on operating experience).

Track side proximity switches prevent the main drive system from powering the carriages into these hydraulic shock absorbers, even though they have the capacity to stall the main propulsion system.

A tertiary emergency braking system is provided at the end of the drydock. It serves as a final bumper stop for the carriages and also has the capability of stopping both carriages from 15 fps in the event of a failure of the secondary return braking system. Deceleration of the low speed carriage will be kept below 2.5g for this braking condition.

F. Data and Power Transmission Systems

A number of data and power transmission methods were investigated during the study phase of the program. These systems are the subject of Report EW-71-3, Data Acquisition and Analysis Systems for the 380' High Performance Towing Tank, Reference 3.

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