DESIGN OF COLD WATER PIPE
FOR SEA THERMAL POWER PLANTS

Progress Report
for Period 1 May 1975 - 30 May 1976

J. Hilbert Anderson

June 1976

NOTICE
This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

Prepared For
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
UNDER CONTRACT NO. E(11-1)-2691

SEA SOLAR POWER, INC.
York, Pennsylvania 17402

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

ABSTRACT

This report covers the preliminary analysis of design conditions for a 40-ft. diameter, 4000 ft. long, cold water supply pipe for a 100 MW sea thermal power plant. The pipe is assumed to be freely suspended from a floating platform. The design is based on a circular row of tubes with spacers between to form the pipe wall. Internal pressure conditions are calculated for maximum assumed flow rates in the pipe. External pressure distribution is calculated for maximum assumed ocean current velocity. Drag and moment distributions are calculated for the pipe loaded with an assumed current velocity profile and buoyancy distribution. Collapse stability calculations are made for the pipe and for the individual tubes. Tube and spacer interaction stresses are calculated for the combined pressure, bending moment, and tensile loads imposed on the pipe. Preliminary analysis is performed on a flexible pipe support system capable of isolating the pipe from the platform during any sea state likely to be encountered by a sea thermal power plant. It is concluded that the basic design is feasible and justifies more precise analysis.

TABLE OF CONTENTS

	<u>Page No.</u>
INTRODUCTION	1
TECHNICAL SECTIONS	2
A. REQUIREMENTS	2
B. CONCEPTS	5
C. ANALYTICAL PROCEDURE	9
1. Tube Stress and Buoyancy	11
2. Pipe Compression	14
3. Tension Loads	17
4. Drag Forces and Resulting Bending Stresses	17
5. Shear Stresses	20
6. Pipe Element Clamping	21
7. Tube-Spacer Interaction Forces	23
8. Pipe Wall Circumferential Bending Moment	25
9. Buckling of the Pipe as a Beam	30
10. Heat Leaks	31
11. Pipe Support	33
12. Vibration and Vortex Shedding	36
CONCLUSIONS	39
RECOMMENDATIONS	41
ILLUSTRATIONS	
Fig. 1 - Plant Water Requirements	43
Fig. 2 - Diagram of Pipe Construction	44
Fig. 3 - Tube Spacer Arrangement	45
Fig. 4 - Tube Joint	46
Fig. 5 - Pipe Support	47
Fig. 6 - Horizontal Load Diagram	48
Fig. 7 - Pipe Support Spring Characteristic	49
Fig. 8 - Vertical Load Diagram	50
REFERENCES	51
APPENDIX	
Drag Force Caused by Flow Past a Long Cylinder	52

INTRODUCTION

A sea thermal power plant requires huge quantities of cold water for condensing purposes, brought up from ocean depths of approximately 2,000 to 5,000 ft. Typically, for a 100 MW plant, volume flows of 8,000 to 25,000 cubic ft. per second might be required. The exact quantity for any particular design plant depends on optimization of all of the other equipment involved, in order to make the most effective economic cycle for power generation, and/or production of fresh water, chemicals, and food. From earlier work, we have chosen a pipe design diameter of 40 ft. as being typical of what might be used to supply cold water to a 100 MW power plant.

Any large cold water pipe suspended from a vessel in the ocean must be attached to the vessel. Since a floating power plant is subject to ocean forces such as waves and winds, it is almost certain to be subjected to forces large enough to move the vessel in the usual modes of heave, yaw, pitch, and roll. Since a suspended pipe up to 4,000 ft. in length becomes a large mass, then any connection between the pipe and the vessel must necessarily allow for the motions and forces between the vessel and the pipe. Early failures by Claude in trying to make a floating power plant with a suspended cold water pipe were failures simply because the cold water pipe broke away from the vessel. This is a most serious problem and must be recognized in any potential cold water pipe design.

This report discusses in detail the various problems that must be solved in connection with the cold water pipe design, and presents some conclusions as to their solution.

TECHNICAL SECTIONS

A. REQUIREMENTS

The first requirement for any cold water pipe is that it must be strong enough to resist all of the forces involved in its association with transporting the water and the ocean climate surrounding it. The pipe must accept the drag forces produced by the ocean currents surrounding the pipe. These vary over the depth of the pipe, as well as with time and location. The pipe must also be strong enough to resist any forces imposed on it by the platform supporting it. Since the platform is subject to wave forces in the upper levels of the ocean, then the movement of the platform will normally produce variable forces on the pipe; which can be vertical, horizontal, or angular motion.

Since the pipe is carrying water which must be lifted from deep in the ocean, there is necessarily a lower pressure inside the pipe than external to the pipe. This pressure difference is partly due to the conversion of potential energy into kinetic energy of flow, partly due to friction losses, and partly due to the difference in density between the cold water on the inside of the pipe and the warmer surrounding waters. Since the pressure inside the pipe is lower than the pressure surrounding the pipe, then the pipe must be designed so that it will not collapse due to the external pressure difference across the wall. In any pipe that has a small thickness-to-diameter ratio, the unstable collapse problem is a serious one. Even though the pressure difference across the wall in this case is quite small by ordinary standards, it can become a very important problem in the design of this pipe. Any bending stresses on the pipe due to currents also increase the tendency to collapse, and must be taken into account in the pipe structure design.

Ocean currents flowing around a circular pipe create unstable vortices, which generate vibrating forces on the pipe. This is a familiar effect on long electric cable transmission lines, where failures have occurred due to cross winds causing the lines to vibrate. This type of self-excited vibration has not only caused failures in transmission lines, but in a more dramatic case, actually caused the collapse of the long, narrow Tacoma suspension bridge. The design of the cold water pipe must take account of these self-excited vibration forces, and make the pipe resistant to failure from this cause.

Since there will inevitably be some pulsating forces on the pipe due to the self-excited vibration, then the design should be such that it will, as much as possible, damp out the stresses generated by these vibrating forces and have fatigue resistance to the application of repetitive forces.

In order to get damping effects and reduce stress concentration prob-

lems that might possibly be caused by vibrating effects, or sudden local stress changes, it is extremely desirable to make the pipe flexible in design. By making the pipe flexible, we adopt the principles long established in nature that show that stress problems are reduced by allowing considerable movement in any object, thereby allowing greater damping in the structure as well as reducing the rate of transmission of stresses from one part to another.

The very nature of this system requires that a long pipe be suspended from the floating platform. Since the platform is going to move when wind and wave forces operate on it, and the pipe is a large mass connected to it, then any motion of the platform tends to produce extremely large forces on the pipe mass. For this reason, it appears logical to allow the platform to move at least partially independently of the pipe. We therefore need to design a connection between the pipe and the platform that has the lowest possible force transmission between the two when the platform moves. Ideally, we would like to have a connection which transmits only a constant force, sufficiently large to support the negative buoyancy of the pipe hanging from the platform.

The force required to support the pipe is a function not only of the mass of the pipe, but also of its buoyancy. We cannot make the pipe completely self-buoyant, because it would then not sink to the desired depth. However, by building in a certain amount of buoyancy in the pipe, we can reduce the force required to support it from the structure. Therefore it is desirable to build buoyancy into the pipe itself, rather than have the supporting buoyancy provided by the platform. This will reduce the necessary force in the connection between the two.

The connection between the pipe and the platform must allow for vertical movement of the platform with respect to the pipe; it must allow for angular motion to take care of pitching and rolling of the platform, and it may even be required to allow for some yaw in the platform. The problem, then, is to allow all of these relative motions without producing excessive forces at the connection between the pipe and the platform.

The pipe connection should also, desirably, be in operation as a flexible support during construction of the pipe. Any of the forces mentioned might occur during construction, and it is just as important to protect the connection between pipe and platform during construction as it is following construction.

Another possible force problem in connection with the pipe is the inertia of the flowing water. Since we have a very large volume of water flowing at an appreciable velocity, then any sudden variations in flow will potentially produce stresses on the pipe or on its connection to the vessel. This is a familiar problem in hydroelectric water transmission lines.

Since a large quantity of water must be pumped, and the power required to pump it subtracts directly from the usable power for sale from the plant, then pumping losses must be assessed at a value of the total plant cost to produce the power required to overcome those losses. It becomes important, then, to design a cold water pipe which has low inherent friction losses. The inlet to the pipe must also be carefully designed so that high losses do not occur. For example, a straight pipe end would incur a direct loss of about 40% of the kinetic energy in the pipe at the inlet. The exit losses from the pipe also need to be carefully controlled. The total kinetic energy in the pipe can become the exit loss if the discharge is not carefully designed to recover the kinetic energy as the water leaves the pipe.

It appears advisable to put a screen at the inlet of the cold water pipe. While we cannot hope to design a screen for the cold water inlet that is sufficiently fine to protect the heat exchanger passages, it is logical to keep out any large animals.

The large size and large quantity of water being handled make heat leakage from the surrounding warm water into the cold water not too much of a problem. However, when you realize that 1°F. will make approximately 5% difference in plant output, then it is certainly desirable to design a pipe which does have a minimum flow of heat into the cold water from the surrounding warm water.

Since the pressure in the pipe is lower than the surrounding water pressure, there is a tendency for leakage of warm water into the pipe as the water flows upward. This possibility must be recognized, and the pipe should be designed to minimize any leaks from the outside. It should also, preferably, be possible to repair leaks if they do occur.

It has been proposed by some to construct the cold water pipe at a shore base and transport it to the ocean platform, then sink it into position. This is pretty obviously a stupendous, and difficult, and costly task. We believe it to be far more desirable to be able to transport easily transportable materials to the plant platform, and then to be able to assemble these parts and construct the pipe on the platform at sea. In order to do this, we need a design which can be constructed on the platform at sea, and can be so constructed that it is lowered from the platform, one section at a time, as the pipe is being constructed. By its very nature, this requires that the supporting structure be usable as construction proceeds, as well as after the pipe is finished and ready for service.

As another desirable, although not completely necessary, feature, the pipe should be retrievable from the ocean after long-time service. Since an appreciable amount of material will be used, it would be desirable to be able to retrieve this material after the pipe has possibly deteriorated to the point of requiring replacement.

While it is desirable that the pipe be fouling-resistant, this is probably not a very serious problem. It has already been found from experience at St. Croix that the inside of a pipe carrying cold water from deep in the ocean does not foul. (13) At St. Croix, the Lamont-Doherty Geological Observatory is using 3-inch diameter polyethylene tubing to pump cold, nutrient-rich sea water from an ocean depth of 2700 feet. The water is being used in a mariculture project. After three years of operation, the entire length of tubing was inspected, and it was learned that:

1. No scaling or fouling occurred on the inside of the tube.
2. Sections of the tube that were located in the surface waters had considerable scaling and fouling only on the outside of the tube.

Since a sea thermal power plant will use similar cold, deep ocean water, there is very little chance of any fouling occurring inside the cold water pipe. In the upper ocean layers close to the platform, there is the possibility of fouling of the external structure, but this is not likely to be a problem over the greater part of the pipe length.

The problem of corrosion is always with us in the ocean. Therefore the materials used should be chosen to minimize the possibility of corrosion. However, since the water from deep in the ocean is quite cold, there is a little less likelihood of corrosion problems in the cold water pipe than there is in the other parts of the structure. Despite this favorable condition, protection methods should be provided, if at all possible.

Many materials for a cold water pipe have been suggested, ranging from wood, to plastic, to aluminum, to concrete. All of these materials are potential candidates, and all have their advantages and disadvantages.

The final requirement for the cold water pipe is that it should be low in cost. Since we must handle vast quantities of water over a fairly long distance to produce a relatively small amount of power, then the cost of materials and building the cold water pipe does become an important factor in the total plant cost. However, it is interesting to note from Figure 1 that the amount of water that needs to be handled in a sea thermal plant per unit of power produced is of the same order of magnitude as the amount of power that flows through a hydroelectric plant. Therefore it is not likely that the cold water pipe will be a limiting factor in the cost of sea thermal power.

B. CONCEPTS

After some 10 years of study and investigation, looking for the ideal design of a cold water pipe and making preliminary evaluations of many types of construction, we tentatively decided that the design that would fulfill

all of the foregoing requirements with the greatest probability of success is a construction as depicted in Figure 2. This consists of a multiplicity of tubes, arranged in a circle 40 feet in diameter, with interconnecting spacers between the tubes as is tentatively shown in Figure 3. The tubes could be joined to each other and, being hollow, could provide their own buoyancy.

Most of the concepts that have been suggested by different people do satisfy some of the requirements, but little by little, we have become convinced that this construction is most likely to satisfy all of the requirements.

The multiplicity of hollow tubes provides the flexibility in the system corresponding to each individual tube, thereby making a much more flexible structure than would be the case with a single-wall circular pipe. The multiple tube elements, being clamped to each other, also provide a great amount of damping friction in the structure, so that any vibration forces tend to be damped out in friction of the individual parts on each other. The principle is rather similar to that which has been successfully used to construct silos, which are rather high, circular structures with high wind loading. The hollow wall raises the thickness-to-diameter ratio of the structure so that the unstable collapsing problem is minimized. The high damping factor of individual elements should reduce tremendously the possibility of self-excited vibrations from currents being built up to any dangerous level. This might not be the case with any single-wall structure. The individual hollow tubes create a roughening effect on the exterior, which should develop small, individual eddies as the currents flow past the pipe, and this should tend to minimize very large eddies that cause self-excited transverse vibrations. This effect must finally be analyzed by actual testing.

This type of pipe construction should have a very low friction loss, since the inside of the pipe can be perfectly smooth in the direction of flow. In the construction as conceived, the joints between individual tubes will be purely mechanical, expanded joints, with their external diameter equal to the tube diameter. Therefore there is no obstruction inside in the flow stream.

Since the hollow wall construction is relatively thick, this almost eliminates any measurable heat leakage from the water outside to the cold water inside the pipe.

This pretty well fills the desirable requirement that materials can be transported easily at sea and constructed on the platform. For the most part, all parts are reasonably small, being pieces of ordinary pipe. These can easily be transported by ship to the platform and stored on the platform during the construction period. The hollow tube construction permits fairly accurate buoyancy control. For example, the lower tubes can be filled with water, or even with rock, so as to produce high-density pipe at the bottom, whereas the upper part of the hollow wall can be filled with gas, so that it provides

a very large share of the total buoyancy required. This means that the support at the platform does not need to carry a great deal of weight, but must allow for the inertia of the entire pipe mass.

By using entirely mechanical joints expended into the tubes (as shown on Fig. 4), this makes it possible to use coated pipe materials, if so desired. This means that handling at sea is much less of a chore, since no welding is required that would obviously destroy any coating of the pipes that was used in the assembly.

The construction lends itself to the possible use of several different materials. For the tubes, there is a possibility of using either steel, aluminum, or fiber glass plastic. Tubes from any of these materials are already in production, and the construction should lend itself to the use of any of these materials.

The pipe support construction is illustrated diagrammatically in Figure 5. The cold water pipe is supported in the well by chains attached to the top of the pipe. The chains pass over sprockets located on the support carriage, the platform, and the pneumatic spring system. The end of the chain is attached to a winch located on the platform.

During plant operation, the support carriage is locked in position to obtain the desired non-linear spring characteristic illustrated in Figure 7. The spring system produces the extremely low natural frequency required to isolate the cold water pipe from the platform during all conditions of motion encountered by the sea thermal power plant. The spring system is also capable of producing the larger support force that will be required during construction of the cold water pipe. The support force required during part of the construction operation will be larger than the final support force due to the buoyancy distribution built into the cold water pipe. The largest negative buoyancy is located in the bottom portion of the pipe. The largest positive buoyancy is located in the upper portion of the pipe and the top portion of the pipe will be neutrally buoyant as illustrated in Figure 8.

A rubber ring can seal the space between the pipe support sleeve and the pipe support carriage. This ring also transmits side thrust from the pipe and can accommodate a 5-degree tilt of the pipe relative to the platform. Friction and wear on the ring are reduced by lubricant injection into the rubber joint through holes in the rubber bushing. The pipe support carriage is sealed into the open well with a seal plate.

The pipe support carriage has rolling supports at the outside diameter to provide smooth up and down operation during construction of the cold water pipe. Note that during pipe construction, the spring isolation system is effective. The allowable movement of the pipe support sleeve relative to the pipe support carriage must be sufficient to allow for any amount of ocean-induced vertical movement of the platform with respect to the pipe. The allowable movement of the pipe support carriage relative to the platform

must be sufficient to allow for the construction of one section of pipe.

During construction of the cold water pipe, the winch is used to lower and raise the pipe support carriage and to lower the pipe, one section at a time, as each section is constructed. The gas pressure in the pneumatic spring is varied during the pipe construction period to maintain the proper tension in the chain and support force on the partially constructed pipe. The desired spring rate for the pipe support system is obtained by variation of the volume of the static gas chamber during construction of the pipe.

Assembly fixtures will be required to position and hold the tubes and beam spacers apart at a diameter greater than the 40-foot pipe diameter to provide ample space for assembly of random length tubes and beam spacers without interference from adjacent pieces. Other pipe assembly fixtures will be required to bring the tubes and spacers together to form a 40-foot diameter pipe and to apply the proper preload compression force to the tubes and spacers before the tension bands and retention devices are assembled to the pipe. The assembly will take place as a continuous flow process with fixture operations occurring in a continuous sequence.

With this system, construction of the cold water pipe can proceed above deck, section by section, and the pipe support carriage can be used to lower the pipe as it is being constructed. This system provides isolation of the cold water pipe from the platform as the pipe is being constructed.

The cold water pump can advantageously be supported from the platform at the top of the pipe as shown in the diagram. This has the advantage that there is no head loss between the pipe outlet and the pump inlet. The discharge from the pump can then be designed as a diffuser, so that the velocity can be decreased, at the same time converting the kinetic energy of the high-velocity water into pressure energy in the well.

If the well is approximately twice the diameter of the cold water pipe, then the velocity is approximately one-fourth that in the cold water pipe, and the kinetic energy of flow is reduced to approximately 1/16 of that in the cold water pipe. This, then, means that the large open well permits easy access from its openings into passages that lead to the condensers and condenser manifolds. With this arrangement, a very, very low friction loss and loss due to turning the water are achieved.

Since the pressure at the pump outlet is necessarily higher than the surrounding ocean pressure, in order to force the water through the condensers with their frictional resistance, then the water level in the open well is shown above the normal deck level of the supporting platform. The open well has the additional advantage that, if any disturbance shuts off the water, the inertia of the water itself is easily absorbed merely by a level rise in the open well. It has no appreciable effect on the dynamics of the platform itself. It also should produce no additional forces on the platform.

From a preliminary evaluation of the proposed pipe construction concept, we decided that this concept has an excellent chance to meet all of the requirements of the cold water pipe. No other concept that we are aware of comes so near to having the potential to fulfill all of the desirable requirements for such a pipe.

C. ANALYTICAL PROCEDURE

In trying to design any structure with the complex requirements of the cold water pipe, it is difficult to evaluate the most critical design problems and to know exactly where to start. We therefore decided to start with a more or less logical pipe design, then evaluate its compatibility with all design requirements. The information we obtained from this should provide direction toward a more nearly optimized design.

At the start, we assumed a mean pipe diameter of 40 ft. and a length, from support to bottom, of 4000 ft. (1219 M.). This length is probably longer than the average that might be used, but would make the design problem a little more difficult. We assumed a current velocity profile that is the most severe that any sea thermal power plant will encounter. Our calculations indicate that the current loading is the most difficult parameter to overcome in the cold water pipe design. If the design proved feasible for this depth and current profile, then it should also be feasible for shallower depths and less severe currents.

In the preliminary report (10), the potential of using 6, 8, and 10-inch diameter steel tubes as elements in the pipe wall was investigated. The possibility of using aluminum tubes in place of steel tubes was investigated after the preliminary report was published. Design methods similar to those performed in the preliminary report were executed in the aluminum tube investigation. A summary of the results of the investigation is presented in the following comparison table. (Table I)

The table indicates that the material costs for an aluminum design are 2.6 times the cost of a steel design. If a coated steel were used in the pipe construction, the aluminum design would cost 1.8 times the coated steel design.

This material comparison is shown for only 6" nominal pipe size. It indicates that aluminum would be more expensive. However, its overall lighter weight means a lower overall cost of the pipe support system and less buoyancy needs to be built into the floating structure to support the pipe. Therefore our most important immediate conclusion from this comparison is that aluminum tubing is a feasible candidate material for a stockade pipe design. Later, more detailed cost analyses will determine whether steel or aluminum is more practical.

TABLE I

STEEL VS. ALUMINUM TUBE MATERIAL DESIGN

<u>Steel</u>	<u>Al</u>		
6.625	6.625	in.	O. D. of tube
.188	.344	in.	Wall thickness of tubes
60,000	35,000	psi.	Tensile strength of tubes
12.89	7.82	lb. /ft.	Weight of tube per ft. length
15.32	15.32	lb. /ft.	Weight of sea water displaced/ ft. length
2.43	7.50	lb. /ft.	Buoyancy of tube/ft. length
1506.	3147.	psi.	Critical buckling pressure of tube caused by external pressure on tube
4.27	4.27	psi.	Pressure difference across pipe wall, at top of pipe
7.39	4.61	psi.	Critical buckling pressure of pipe caused by external pressure on pipe
2725.	1490.	psi.	Tube wall compressive stress caused by pressure difference across pipe wall
12.89	7.82	lb. /ft.	Tube mass
1.685	1.685	lb. /ft.	Water displaced
44,820	24,540	lb.	Net weight of tube in water
11,819	2,839	psi.	Tensile stress caused by weight of tube
.20	.84	\$/lb.	Material cost, Nov., 1975
2.578	6.569	\$/ft.	Tube cost, Nov., 1975

1. Tube Stresses and Buoyancy

The preliminary report indicates that, as a trial design, we should use tubes made of nominal 12" standard schedule steel pipe. The O. D. = 12.750" and wall thickness = .358". For grade "B" pipe, mill test pressure is 1200 psi. and tensile strength is 60,000 psi.

Assuming 1" spacers between tubes, then

$$\text{No. of tubes} = \frac{(40 \times 12 + 12.750)\pi}{12.750 + 1} = 113 \text{ tubes.}$$

Weight of tube per ft. length = 47.35 lbs. /ft.

Weight of sea water displaced by tube

$$= \frac{12.750^2 \pi \times 64}{4 \times 144} = 56.75 \text{ lbs. /ft.}$$

Therefore, each tube would be buoyant and would produce $56.75 - 47.35 = 9.40$ lbs. buoyancy per ft. of length, if filled with gas or air.

Weight of spacer per ft. length = 19.39 lbs. /ft. tube.

Weight of sea water displaced by spacer = 2.54 lbs. /ft. tube.

Weight of spacer in sea water = 16.85 lbs. /ft. tube.

Weight of tube connector = 3.55 lbs. /ft. tube.

Weight of tension band (average) = 1.56 lb. /ft. tube.

Weight of sea water displaced by band = .20 lb. /ft. of tube.

Weight of tension band in sea water = 1.36 lb. /ft. of tube.

Buoyancy of pipe in sea water

$$113 \times (9.40 - 16.85 - 3.55 - 1.36) = -1397 \text{ lbs. /ft. of pipe.}$$

The downward load on the pipe support is:

$$1397 \times 4000 = 5,588,000 \text{ lbs.}$$

A pipe made with aluminum parts would provide greater buoyancy and will be investigated later. Fiber glass-reinforced plastic parts would also provide greater buoyancy. However, the long-term strength characteristics of fiber glass-reinforced plastic are not suitable for this application. (11)

Typical properties of fiber glass-reinforced pipe are: 10,300 psi. tensile strength and 1.4×10^6 modulus of elasticity at 75°F. (12)

The buoyancy of the pipe can be controlled by filling the tubes with gas or liquid or solids. There are several possible alternatives for filling the tubes to produce controlled buoyancy:

1. The tubes could be entirely filled with air or gas at atmospheric pressure. Then, at 4000 ft. depth, the external pressure on the tube would be

$$\frac{4100 \times 64}{144} = 1822 \text{ psi.}$$

Note: We assume pipe support is 100 ft. below sea level. This would produce a compressive stress of approximately

$$\frac{1822 \times 12.750}{2 \times .358} = 32,445 \text{ psi. on the tube wall.}$$

At this value of external pressure, the tube wall is subject to collapse due to instability of the circular section. Ref. 1 provides a convenient means for evaluation of the critical pressure that might cause buckling of a circular tube.

Using Eq. (6) of Ref. 1 for the tubes and assuming a tube length of 40 ft. between joints, we have:

$$L/D = 40 \times 12/12.750 = 37.6$$

$$T/D = .358/12.750 = .0281$$

From Fig. 1, this falls into the range of infinitely long cylinders, and Eq. (7) can be used to determine critical buckling pressure.

$$P_{CR} = \frac{ET^3}{4(1 - \mu^2) R^3}$$
$$P_{CR} = \frac{30 \times 10^6 \times .358^3}{4 \times .91 \times 6.375^3} = 1460 \text{ psi.}$$

Since this value is less than the external pressure of 1822 psi., we cannot use these tubes at this depth filled with air at atmospheric pressure.

2. The tubes could be filled with water. In this case, the internal pressure would be approximately equal to the external pressure, and there would be almost no wall stress from pressure difference across the

tube wall. This scheme has the obvious disadvantage that the buoyancy corresponding to this weight must be provided in the platform.

3. The tubes could be filled, or partially filled, with a low density, volatile liquid, such as propane. At 40°F., the liquid density is 32.7 lbs./cu. ft. The vapor pressure at 60°F. is 93 psig. If the tubes were filled with liquid to within 100 ft. of the top, then the internal pressure at the bottom would be

$$P = 93 + \frac{(4000 - 100) 32.7}{144} = 979 \text{ psig.}$$

The pressure difference across the tube wall at the bottom would be $1822 - 979 = 843$ psi. Since this is less than the critical buckling pressure, it would be possible to use the tube under these conditions. By pressurizing the upper part of the tube with a gas such as methane, the internal pressure could be increased so that pressure difference would be acceptably low.

4. The tubes could be partially filled with water and pressurized with gas at the top. For example, if half-filled with water, the internal water pressure at the bottom would be $1822/2 = 911$ psi. If gas pressure in the upper half is raised to 450 psi., the wall pressure differential would then be $1822 - 911 - 450 = 461$ psi. at the bottom and at the middle. The internal pressure would be approximately $450 - 44 = 406$ psi. above external pressure at the top of the tube.
5. The tubes could be partially filled with water, with selective gas pressurization in the upper sections. This would create approximately equalized pressures across the tube wall throughout its length.
6. The tubes could be partially filled with high density rocks at the bottom and selectively pressurized with gas from the bottom to the top. This scheme has the advantage of producing maximum vertical moment with minimum negative buoyancy. The vertical moment is required to compensate for net horizontal current drag moments on the pipe.

From the above considerations, it is clear that depth and water pressures would not prevent us from using 12" standard schedule steel pipes for tubes in the pipe wall. Aluminum alloy tubes would potentially provide greater buoyancy, but the steel tubes can provide a reasonable amount of buoyancy.

The use of positive buoyancy in the tubes reduces the amount of buoyancy that must be built into the platform and reduces the supporting force required between platform and pipe. Both factors are important.

$$\text{Buoyancy} = \text{Mass of water displaced} - \text{mass of pipe.}$$

The following table shows a crude estimate of weights and buoyancy for the 4000 ft. pipe, using 12.750 in. diam. steel pipe with .358-inch wall thickness.

TABLE II
PIPE WEIGHTS AND BUOYANCY

	<u>Mass/Ft.</u>	<u>Water Displacement Per Ft.</u>
Tubes	5351.	6413
Spacers	2191.	287
Tube Connectors	401.	0
Tension Bands	<u>176.</u>	<u>23</u>
TOTAL:	8119 lbs. /ft.	6723 lbs. /ft.
Total for 4000 ft.	32, 476, 000 lbs.	26, 892, 000 lbs.

The net force downward on the platform would be 5, 584, 000 lbs. If the pipe were made of aluminum, the total pipe mass would be reduced to approximately 19, 702, 000 lbs. (See Table I). The force on the platform could then be reduced to zero, or nearly so, by partially filling the tubes with water. It is obviously desirable to use aluminum. However, it is also possible that thinner wall steel tubes could be used over the lower stressed lengths of the pipe. This should be investigated further and will relate closely to a more accurate determination of drag forces and their distribution.

2. Pipe Compression

The next design consideration is the buckling stability of the pipe against differential pressures created by internal water pressure being less than external water pressure surrounding the pipe. This reduction in internal pressure is caused by several factors:

1. The 40°F. water flowing through the pipe is denser than the surrounding warm water. This results in a lower pressure at the pump inlet, in order to lift this denser water.
2. The static pressure inside the pipe is lower than pressure outside because the water is flowing at relatively high velocity. The kinetic energy of the water must be generated by a pressure difference corresponding to the velocity head.
3. The inlet loss to the pipe contributes to the pressure difference.
4. The friction pressure drop in the pipe reduces the pressure, so that the maximum difference between external and internal pressure occurs immediately before the pump inlet at the upper end of the pipe.

The pressure difference due to lifting higher density water through the pipe is approximately 0.9 psi.

The internal area of the pipe is approximately calculated as:

$$A = 40^2 \frac{\pi}{4} + \frac{113}{144} \left(\frac{12.750 (1 - \sin 45)(12.750 + 1)}{2} - 12.750^2 \frac{\pi}{4} \frac{(90 - 45)}{180} + \frac{12.750^2}{4} \sin 45 \cos 45 \right)$$

$$A = 1268. \text{ ft.}^2$$

At a nominal flow of 10,000 cfs., the mean flow velocity is:

$$V = \frac{10,000}{1268} = 7.89 \text{ ft./sec.}$$

Since we do not know the best or most economic velocity for this size pipe, it is advisable to assume a higher velocity for these first calculations. We will assume 10 fps. as a possible maximum economic velocity. The pressure difference corresponding to the velocity head is:

$$\Delta P_V = \frac{\rho v^2}{2g} = \frac{64 \times 10^2}{64.3 \times 144} = .691 \text{ psi.}$$

If we assume an inlet loss of 5% of the velocity head, then

$$\Delta P_I = .05 \times .691 = .035 \text{ psi.}$$

The Reynolds number for flow through the pipe is found from the equation:

$$Re = \frac{VD}{\nu}$$

where V = velocity in fps.

D = diameter in ft.

ν = kinematic viscosity in ft.²/sec.

$$Re = \frac{10 \times 40}{1.72 \times 10^{-5}} = 2.33 \times 10^7$$

Since this is in the high turbulent flow range, we can compute friction loss from the familiar Darcy formula.

$$\Delta P = f \frac{L}{D} \frac{\rho v^2}{2g}$$

A conservative high absolute roughness value should be .01, with a corresponding value of $f = .015$. Then

$$\Delta P_f = .015 \times \frac{4000}{40} \times \frac{64 \times 10^2}{64.3 \times 144} = 1.037 \text{ psi.}$$

The total value of ΔP near the upper end of the pipe would then be

$$\Delta P = .9 + .69 + .04 + 1.04 = 2.67 \text{ psi.}$$

The pipe must then withstand this much pressure difference without buckling. At the lower end, near the inlet, it must withstand a pressure difference of .73 psi. Recent experience at St. Croix (13) shows that the inside of the cold water pipe does not foul with marine organisms. This means that the flow resistance of the pipe and therefore the maximum pressure difference across the pipe wall will remain constant at the foregoing value during the life of the pipe.

Since the pipe wall is hollow, the effective wall thickness to resist buckling is much greater than the tube wall thickness and would be fairly close to the tube diameter. From Ref. 3, one can determine an approximation of effective wall thickness to be $12.750 \sin 45^\circ = 9.02''$.

Then, using Eq. (7) from Ref. 1, we can get an approximation to the critical buckling pressure of the pipe. This approximation assumes that no stiffening rings are used and that tubes are clamped to spacers tightly enough to prevent rotational slipping. Since this is not a solid wall, we must adjust the effective modulus of elasticity in the ratio of actual metal wall thickness to assumed hollow wall thickness.

$$\text{Then } P_{CR} = \frac{2 \times .358}{9.02} \times \frac{30 \times 10^6 \times 9.02^3}{4 \times .91 \times 240^3} = 34.73 \text{ psi.}$$

From this, we can assume that the 12.750" tubes would form a wall thick enough to prevent buckling collapse, even though there were no stiffening rings. Note, however, that this assumes sufficient clamping force or shear bonding at the spacers to prevent rotational slipping of the spacers around the tubes.

The compressive stress on the tube wall must also be checked.

$$S = \frac{Pd}{2t} = \frac{2.67 \times 40 \times 12}{2 \times 2 \times .358} = 895 \text{ psi.}$$

This is acceptable.

3. Tension Loads

The weight of the pipe produces a direct tension load up to the value of force supplied by the pipe support. As a minimum, this must equal the total negative buoyancy of the pipe. As a maximum, it will be the maximum force applied to the mass of the pipe when it is being accelerated upward during heave conditions of the platform. As a starting assumption, we can assume the downward force of an individual tube in water.

Tube mass = 47.35 lbs. /ft.

$$\text{Water displaced} = (12.750^2 - 12.034^2) \frac{\pi \times 64}{4 \times 144} = 6.194 \text{ lbs. /ft.}$$

$$\text{Net weight in water} = (47.35 - 6.194)4000 = 164,623 \text{ lbs.}$$

$$\text{Cross sectional area} = (12.75^2 - 12.034^2) \pi/4 = 13.937 \text{ sq. in.}$$

$$\text{Tensile stress} = 164,623/13.937 = 11,812 \text{ psi.}$$

This value appears to be reasonable, but it must be kept in mind that bending forces will add to this. Also, extraneous masses, such as spacers, clamping rings, and tube connectors will add to the mass and the tensile loading.

From this, we can conclude that it is highly desirable to reduce this load by using the buoyancy of gas-filled tubes to reduce the tensile load on the pipe. This load would also be reduced with aluminum tubes.

4. Drag Forces and Resulting Bending Stresses

Currents will produce horizontal forces on the pipe. These currents will vary with time and depth.

As a first assumption, we can assume drag forces to be like those about a true cylinder. Data for drag forces generated by flows across a cylinder are shown in Ref. 4, p. 7-102.

If we assume a maximum velocity of 8.2 ft. per sec. or 4.86 knots, then the Reynolds number is:

$$Re = \frac{VD}{\nu} = \frac{8.2 \times 40}{1.72 \times 10^{-5}} = 19.1 \times 10^6$$

From Fig. 39 on p. 7-102, the probable drag coefficient $C_D = 0.35$.

$$\text{Drag force} = C_D LD \frac{\rho v^2}{2g} = \frac{.35 \times 1 \times 40 \times 64 \times 8.2^2}{64.3} = 937 \text{ lbs.}$$

This is the drag force per ft. length of pipe. From Ref. 5, the maximum velocity at a depth of 100 meters is likely to be about 140 cm. /sec. or 3.9 ft. /sec. in the Florida current, and less than half that value at 200 meters depth. It is not likely that currents faster than this will be found in usual locations for sea thermal power plants.

Drag coefficients are likely to be higher for rough circular cylinders, although no test data is available or known to us at these high Reynolds numbers. There is some indication from drag tests on rough cylinders at lower Reynolds numbers that drag coefficients may be as high as 1.0. If this is the case, then the velocity, which would produce a drag force of 937 lbs. /ft., would be

$$8.2 \times \sqrt{\frac{.35}{1.0}} = 4.85 \text{ ft. /sec.}$$

An attempt at estimating pipe forces is shown in Fig. 6. A hypothetical current velocity profile has been established which is the most severe profile that will be encountered by any sea thermal power plant. Transverse shear, bending moment and longitudinal stress for the pipe were calculated based on this profile, an estimated drag coefficient, and an assumed pipe buoyancy distribution. More information is needed about actual velocity profiles and actual drag coefficients before an actual loading diagram can be determined.

In order for the pipe to be in equilibrium with a pivot support, which produces almost no reaction moment, the pipe must slope in the water, with a downward force. This downward force produces an opposing moment that must balance the moment created by the current drag forces. The calculated moments in Fig. 6 were based on an assumed constant negative buoyancy along the length of the pipe. While the moments shown are probably not correct for an actual situation and must wait for better data before making

a final determination, at the same time they do illustrate some possibilities and probably show higher moments than those that will be reached in the actual cases.

Figure 8 shows a vertical loading diagram that will probably be preferred over a constant negative buoyancy along the length of the pipe. By concentrating the negative buoyancy near the bottom of the pipe and positive buoyancy in the upper portion, a greater compensating moment is produced for a given pipe slope. This is desirable because less total negative buoyancy is required for any limiting pipe slope. By making the top section of the pipe neutrally buoyant, the combined bending moment stress and tensile stress in the pipe members can be minimized.

From Fig. 6, the maximum bending moment required for the pipe design is 124.2×10^6 lb.ft. This loading is probably unrealistically severe, but should serve to point out possible limitations in the design.

It is of interest to note how much moment the pipe could withstand as a summation of the moments on all the individual tubes and spacers. This would then require no vertical shear loading between tubes.

$$I/C = 42 \text{ in.}^3 \text{ for each tube}$$

$$I/C = 113 \times 42 = 4746 \text{ in.}^3 \text{ for all tubes}$$

Assuming a limiting yield stress of 30,000 psi.

$$M = \frac{30,000 \times 4746}{12} = 11.87 \times 10^6 \text{ lb.ft. for the tubes only}$$

$$I/C = 1,113 \text{ for the I beam spacers arranged in a circle}$$

The combined tubes and spacers have an $\frac{I}{C}$ of

$$I/C = 1,113 + 4746 = 5859 \text{ in.}^3$$

The combined tubes and spacers without shear loading can withstand an M of:

$$M = 30,000 \times \frac{5859}{12} = 14.65 \times 10^6 \text{ lb.ft.}$$

This is much less than the 124.2×10^6 lb.ft. design requirement, and it becomes evident that spacers between tubes must transmit vertical shear loads. This can be transmitted either along the spacer or at an interlocking joint between tube connecting joints.

If we assume that vertical shear loads are transmitted between tubes, then we can assume the pipe to be a beam of circular cross section, with

the equivalent wall thickness equal to the total cross sectional area of the individual tubes divided by the mean circumference of 480". If we assume spacers 1" thick between tubes, then equivalent wall thickness becomes:

$$t = \frac{\pi dt}{d + 1} = \frac{12.750\pi \times .358}{12.750 + 1} = 1.043 \text{ inch}$$

$$\frac{I}{C} = \frac{\pi}{32} \left(\frac{d_o^4 - d_i^4}{d_o} \right) = \frac{\pi}{32} \left(\frac{(480 + 1.043)^4 - (480 - 1.043)^4}{480 + 1.043} \right)$$

$$\frac{I}{C} = 1.8833 \times 10^5 \text{ in.}^3$$

$$S = \frac{MC}{I} = \frac{124.2 \times 10^6 \times 12}{1.8833 \times 10^5} = 7,914 \text{ psi.}$$

This stress is acceptable.

5. Shear Stresses

The maximum horizontal shear load is most likely to occur at the top of the pipe, near the pipe support. The total shear load at the top from Fig. 6 is:

$$V = 454,000 \text{ lbs.}$$

$$\text{The tube cross section area} = 13.937 \text{ in.}^2/\text{tube}$$

$$\text{The spacer cross section area} = 5.711 \text{ in.}^2/\text{spacer}$$

$$\text{The combined cross section area} = 19.648 \text{ in.}^2/\text{section}$$

$$\text{Shear stress} = \frac{V}{A} = \frac{454,000}{113 \times 19.65} = 205. \text{ psi.}$$

The maximum vertical shear stress in a pipe is given by the equation:

$$S_V = \frac{2V}{A} = 2 \times 205 = 410 \text{ psi. (Ref. 8, p. 166)}$$

This would be the case for a constant thickness pipe, but since our pipe has a minimum wall thickness of $2t = 2 \times .25 = .5$, and the mean pipe wall thickness is 1.043, then the maximum shear stress will be

$$S_V = \frac{410 \times 1.043}{.5} = 855 \text{ psi.}$$

This vertical shear stress is called shear flow when translated into vertical shear force per unit length of pipe.

Let q = shear force per ft. of pipe

$$\text{Then } q = S_V \times 2t \times 12 = 855 \times .5 \times 12 = 5,130 \text{ lbs./ft.}$$

6. Pipe Clamping

In order to resist the vertical shear force between tubes, we can possibly clamp the tubes tightly enough against the spacers so that friction can resist the shear force.

Assuming that coefficient of friction, $f = .25$

$$\text{Then tension rod force} = \frac{q}{f} = \frac{5,130}{.25} = 20,520 \text{ lbs.}$$

The cross sectional area of the tension rod at 20,000 psi. is:

$$A = \frac{20,520}{20,000} = 1.026 \text{ sq. in.}$$

This would mean that one tension rod of this area would be required in each foot of pipe length. This is obviously not very desirable, and we would therefore be well-advised to design interlocking shear joints as a part of the tube connector. However, if further tests show that shear forces are much less than we have assumed, then the frictional clamping may prove to be feasible.

Note that near the top of the pipe, there is a suction of up to 2.67 psi. This produces an inward force, just as the clamping bands would do. In each wall, the force per ft. of length would be:

$$F = \frac{480'' \times 12'' \times 2.67 \text{ lbs./sq. in.}}{2} = 7,690 \text{ lbs.}$$

This much clamping force is furnished by the pressure, independently of the tension bands.

If it turns out that the total shearing force is less than what we have assumed, then there is an excellent possibility of reducing clamping force requirements, and it is quite possible that independent shear joints will not be required over much of the pipe length. This would have the advantage of increasing frictional damping between the tubes.

It is desirable to maintain compression force between the tubes and the spacers at all times. While the suction pressure inside the pipe automatically maintains this compressive force, this would only act when the cold water pump is operating. Therefore it is important to have tension bands to keep compressive force on the tubes, both when the pump is operating and when it is shut down.

The pipe, when compressed by the tension bands, deflects, becoming smaller at a rate determined by the load vs. deflection rate, or by the spring rate. The tension band also has a spring rate, and stretches as it is tightened on the pipe. When pump suction produces a compressive force on the pipe, it tends to reduce its diameter and therefore to loosen the tension bands. Because of this effect, it is important to have a greater stiffness rate or spring rate in the pipe than in the tension bands. For this reason, we found that the idea of using rubber spacers between tubes in compression, or thin-walled tubes, is not practical. The compressive spring rate would be much too low, and it would be difficult to reduce the spring rate of the tension band to a lower value than that of the pipe.

The situation is similar to that of a wood stave silo, or a wooden barrel. When there is moisture in the staves, they expand and are tightly compressed in the steel tension bands. When they dry out, the staves contract, but since the band stiffness or spring rate is much higher than that of the staves, the bands do not contract with the staves, and they become loose. This then causes the silo to lean, or to collapse.

For the foregoing reasons, we found that a stiffer "I"-shaped spacer piece between tubes would be important to keep the compressive spring rate of the pipe high. The spring rate for this construction can be calculated with reasonable accuracy from Ref. 3, p. 173, Case 8. By using a value of $\theta = 55^\circ$ to 60° , it turns out that the spring rate can be raised to quite a high value, nearly to that of a solid wall pipe.

The construction shown on Fig. 3 provides two lines of sealing to minimize leakage into the pipe. Calculations of potential leakage of surrounding warm water into the pipe show that this should be almost insignificant. The important factor is to provide tension bands of sufficiently lower spring rate than the pipe, so that they will maintain tightness under all conditions.

An alternative possible construction is to use extruded pipe with an integral formed spacer as a part of the pipe wall. It may be possible to manufacture such a design, particularly if aluminum is used, but we have not investigated this possibility.

7. Tube-Spacer Interaction Forces

The compressive force on the tube wall caused by the pressure difference across the pipe wall is:

$$F = \frac{2.67 \times 40 \times 12 \times 12}{2 \times 2} = 3845 \text{ lbs./foot of pipe}$$

The stress in the tube wall that is caused by the above load is calculated by the method described in Ref. 3, p. 173.

$$\theta = 45^\circ; \quad \phi = 135^\circ$$

For $z = 0$ to θ

$$M = WR \left[.3183 (\phi \sin \phi + \cos \phi - \theta \sin \theta - \cos \theta - \cos z \sin^2 \theta + \cos \sin^2 \phi) - \sin \phi + \sin \theta \right]$$

At $z = 0^\circ$

$$M = WR \left[.3183 (-.10496) \right]$$

$$M = WR \left[-.03341 \right]$$

For $z = \theta$ to ϕ

$$M = WR \left[.3183 (\phi \sin \phi + \cos \phi - \theta \sin \theta - \cos \theta - \cos z \sin^2 \theta + \cos z \sin^2 \phi) - \sin \phi + \sin z \right]$$

At $z = 90^\circ$

$$M = WR \left[.3183 (-.3035) - .70711 + 1 \right]$$

$$M = WR \left[.19629 \right]$$

The maximum pipe wall bending moment is at $z = 90^\circ$ and has a value of:

$$M = 3845 \times \frac{12.75}{2} \times .19629 = 4811.4 \text{ lb. in. /foot of pipe}$$

$$\frac{I}{C} = \frac{bt^2}{6} = \frac{12 \times .358^2}{6} = .25633 \text{ in.}^3$$

$$S = \frac{MC}{I} = \frac{4811.4}{.25633} = 18,770 \text{ psi. stress in the tube wall}$$

This stress is acceptable.

The stress in the flange of the I beam spacer caused by the above load is calculated next.

Web Stress

$$S = \frac{F}{A} = \frac{7690}{12 \times .250} = 2,563 \text{ psi.}$$

Shear Flow

$$q = 1.5 \frac{F}{A} = 1.5 \times 2,563 = 3,845 \text{ lb. /foot of pipe.}$$

Flange Stress

$$M = Fd = 2.367 \times 3845 = 9,101 \text{ lb. in.}$$

$$\frac{I}{C} = \frac{bh^2}{6} = \frac{12}{6} \times .4926^2 = .4853 \text{ in.}^3$$

$$S = \frac{MC}{I} = \frac{9,101}{.4853} = 18,753 \text{ psi.}$$

These stresses are acceptable.

8. Pipe Wall Circumferential Bending Moment

The pressure distribution around a cylinder is: $P - P_0 = .5 \rho U^2 (1 - 4 \sin^2 \theta)$ Ref. 9, p. 227. This equation applies for $\theta = 0$ to ϕ , where ϕ is determined by the drag force on the cylinder. For $\theta = \phi$ to π , $P - P_0$ is a constant.

To determine ϕ for the assumed drag, we will assume that the drag force is mainly a pressure-drag force, since it has been found experimentally that the friction-drag force can be neglected at large Reynolds numbers. (Ref. 9)

For the drag force $F = C_D \rho \frac{AV^2}{2}$. If we assume $C_D = .5$ for the top of the pipe.

$$F = .5 \times \frac{.03704 \times 1 \times 480 \times 98.4^2}{386 \times 2} = 111.37 \text{ lb. /inch of pipe}$$

The following force equation was derived from differential equations (See Appendix A).

$$\frac{F}{2} = R \rho \frac{U^2}{2} \left(\sin \phi - \frac{4}{3} \sin^3 \phi \right) - CR \sin \phi$$

Assume $\phi = 140^\circ$ and insert values into the above equation.

$$\frac{111.37}{2} = 240 \times \frac{.03704}{386} \times \frac{98.4^2}{2} \left(\sin 140 - \frac{4}{3} \sin^3 140 \right) - 240 C \sin 140$$

$$55.69 = 32.15 - 154.27C$$

$$C = -.15 \text{ psi.}$$

Check this pressure value by inserting numbers in the pressure equation to see if the calculated pressures agree.

$$P = .5 \rho U^2 (1 - 4 \sin^2 \phi) = .5 \times \frac{.03704}{386} \times 98.4^2 (1 - 4 \sin^2 140)$$

$$P = -.30 \text{ psi.}$$

This pressure does not agree with the value from the force equation, but it is close.

Determine ϕ with the pressure equation when the pressure is $-.15$ psi.

$$-.15 = .464 (1 - 4 \sin^2 \phi) = .464 - 1.856 \sin^2 \phi$$

$$\sin^2 \phi = \frac{.464 + .15}{1.856}$$

$$\phi = 145^\circ$$

This is close to the assumed value of ϕ . Therefore:

ϕ is between 140 and 145° ; use $\phi = 143^\circ$

P is between $-.15$ and $-.30$ psi.; use $P = -.2$ psi.

Calculate a triangular pressure load on the pipe that is equivalent to the non-circular variation in the calculated pressure distribution.

$$F = \int_0^{90} P d\theta = \int_0^{90} [.464 (1 - 4 \sin^2 \theta) + 1.39] d\theta$$

$$F = 1.235 = .5 W$$

$$W = 2.47 \text{ psi.}$$

Calculate the maximum positive circumferential pipe wall bending moment caused by the triangular pressure distribution.

See Ref. 3, p. 174.

$$M = .19139 W_1 R^2 - .5 W_2 R^2$$

$$M = (.19139 \times 2.47 - .5 \times .038) 240^2$$

$$M = 26,133 \text{ in. lbs. per inch of pipe}$$

This pipe wall circumferential bending moment produces tension on the I. D. of the pipe.

Calculate the maximum negative circumferential pipe wall bending moment caused by the triangular pressure distribution.

$$M = -.16847 W_1 R^2 - .5708 W_2 R^2$$

$$M = (-.16847 \times 2.47 - .5708 \times .038) 240^2$$

$$M = -25,217 \text{ in.-lbs. per inch of pipe.}$$

This pipe wall circumferential bending moment produces compression on the I. D. of the pipe.

Calculate the tube wall stress caused by the pipe wall circumferential bending moment. Assume a 1-inch thick tension band.

The moment arm is:

$$\frac{12.750 + 1}{2} = 6.875 \text{ inch}$$

The moment force on the tube wall from each spacer flange is:

$$\frac{26,133 \times 12}{2 \times 6.875} = 22,807 \text{ lbs. /foot of pipe}$$

$$S_t = 18,770 \times 22,807 / 3845 = 111,336 \text{ psi.}$$

This is too large. Determine the tube wall thickness required to reduce the tube wall stress to 30,000 psi.

$$\frac{I}{C} = \frac{22,807}{30,000} \times \frac{12.75}{2} \times .19629 = .9513 \text{ in.}^3$$

$$t = \sqrt{\frac{6}{12} \times .9513} = .690 \text{ inch}$$

Calculate the stresses in the I beam spacer.

$$\text{Web stress} = S_w = \frac{22,807 \times 2}{12 \times .25} = 15,205 \text{ psi.}$$

$$\text{Shear flow} = q = 1.5 \times 15,205 = 22,807 \text{ lb. /foot of pipe}$$

$$\text{Flange stress} = S_f = \frac{MC}{I} = \frac{2.367 \times 22,807}{.4853} = 111,239 \text{ psi.}$$

This is too large. Determine the I beam flange thickness required to reduce the stress to 30,000 psi.

$$\frac{I}{C} = \frac{2,367 \times 22,807}{30,000} = 1.7994 \text{ in.}^3$$

$$t = \sqrt{\frac{6}{12}} \times 1.7994 = .949 \text{ inch}$$

Calculate the force in the tension band. This force must be greater than:

(1) The force produced by the maximum negative pipe wall circumferential bending moment,

OR: (2) The force produced by the maximum positive pipe wall circumferential bending moment minus the force produced by the pressure difference across the pipe wall.

The force from the negative moment is:

$$I = \frac{25,217 \times 12}{6.875} = 44,017 \text{ lb./foot of pipe}$$

For a 20,000 psi. stress level, the required tension band area is:

$$A = \frac{44,017}{20,000} = 2.201 \text{ in.}^2/\text{foot of pipe}$$

The tube wall and spacer flange must be sized for the combined forces of the tension band and the pressure difference across the pipe wall. (See p. 16 for ΔP)

For the tube:

$$\frac{I}{C} = \frac{(.5 \times 44,017 + 3,845)}{30,000} \times \frac{12.75}{2} \times .19629 = 1.0784 \text{ in.}^3$$

$$t = \sqrt{\frac{6}{12}} \times 1.0784 = .734 \text{ inch}$$

For the spacer flange

$$\frac{I}{C} = \frac{2.367 \times (.5 \times 44,017 + 3,845)}{30,000} = 2.0398 \text{ in.}^3$$

$$t = \sqrt{\frac{6}{12}} \times 2.0398 = 1.010 \text{ inch}$$

The above calculations were repeated for 400 and 800 feet of depth and are based on the current profile shown in Fig. 6. The results are tabulated as follows:

Depth in Feet	Tension Band Area in. ² /ft. of pipe	Tube Wall Thickness Inch	Spacer Flange Thickness Inch
200	2.20	.734	1.010
400	1.20	.570	.784
800	.13	.308	.424

(Table based on pipe wall bending moment plus pump ΔP)

The information in the table indicates that the mass of the pipe is greater than the assumed mass between the pipe support and the 800 foot depth as shown in Table II. This means that the pipe support must support more than the 5.5 million pounds resulting from the assumed uniform pipe construction. However, the results show that the force between the tube and spacer can be accommodated.

The tension band area required to produce enough force to prevent the tubes from slipping due to the shear flow is calculated from the following equation:

$$A = \frac{V}{443,550} \text{ in.}^2/\text{foot of pipe}$$

Results of the equation are plotted on Fig. 6 for the pump stopped condition.

The tension band area required by the governing force is also plotted on Fig. 6. The plot indicates that the pipe wall circumferential bending moment is the governing force and determines the tension band area in the top 800 feet of pipe. The shear flow caused by current drag forces acting on the pipe as a beam determines the tension band area at depths from 800 to 2300 feet. At depths from 2300 to 4000 feet, the pipe wall circumferential bending moment again determines the tension band area. The area of the tension band is calculated on the basis that no shear joints are used in the pipe wall construction.

Also plotted on Fig. 6 is the longitudinal stress in the pipe wall, which is the sum of:

- (1) The longitudinal tensile stress caused by the weight of the pipe and a drag component resulting from the current force.
- (2) The pipe bending moment stress caused by the current and weight forces acting on the pipe as if the pipe were a beam.

The bending moment stress equation for the above calculation is:

$$S_B = M \frac{C}{I} = \frac{M}{1.8833 \times 10^5}$$

M is the bending moment from Fig. 6.

$\frac{I}{C}$ was calculated previously for the pipe.

The longitudinal stress equation for the above calculation is:

$$S_L = \frac{F_L}{A}$$

F_L is the longitudinal force calculated from the beam equilibrium equations.

A is the cross section area of the pipe material.

9. Buckling of the Pipe as a Beam

It is possible for a long thin wall cylinder to fail by buckling at a stress level lower than the yield stress of the material in the outer fibers of the pipe, which is acting as a beam with an applied load. A relationship for this kind of failure was developed by Timoshenko and presented in an article by T. J. Atterbury. (15)

$$\sigma = \frac{.4 Et}{D \sqrt{3(1 - \nu^2)}}$$

σ = psi., compressive stress at buckling failure

E = psi., modulus of elasticity

t = inches, wall thickness

D = inches, pipe O. D.

ν = Poisson ratio

$$\sigma = \frac{.4 \times 30 \times 10^6 \times 1.43}{480.72 \sqrt{3(1 - .28^2)}}$$

$$\sigma = 21,468 \text{ psi.}$$

This says that buckling would occur in a 40 foot diameter thin wall steel pipe of 1.43 inch wall thickness when the bending stress reaches 21,468 psi. We found that the maximum stress realized for the combined bending and tension in our pipe was about 10,000 psi. on the tension side and 6,000 psi. compressive stress on the concave side. This applied compressive stress is much lower than the stress required to buckle the thin wall 40 foot diameter pipe. Since our wall is much stiffer than the assumed 1.43 inch wall, we would assume that the stress required to buckle our pipe is much higher than 21,468 psi. But it is difficult to estimate values for wall thickness, modulus of elasticity, and Poisson ratio that should be used in the equation to determine maximum stress values for our pipe design.

10. Heat Leaks

The cold water, at 40°F., must pass through upper layers of warm water in its travel to the condenser. If the water temperature rises by as little as one degree, it reduces gross plant output by more than 5% and net plant output by an even greater amount, maybe up to 7%. Therefore it is important to minimize heat leaks through the pipe wall.

As a crude approximation, we can assume that we have 3000 ft. of pipe immersed in water at 70°F. We can also assume a flow of 10,000 cfs., with a velocity of 8.3 ft. per sec.

The mass flow rate of water:

$$W = 10,000 \times 64 \times 3600 = 2.3 \times 10^9 \text{ lbs. /hr.}$$

The area of the pipe is:

$$A = 40\pi \times 3000 = 377,000 \text{ sq. ft.}$$

$$\Delta t = 70 - 40 = 30 \text{ F.}$$

Heat flow through wall is:

$$Q = \frac{\Delta t \times A}{R}$$

where R is the heat transfer resistance through the wall and surface films.

$$R = \frac{1}{h_o} + \frac{t}{K} + \frac{1}{h_i}$$

$h_o = 100 \text{ Btu/sq. ft. hr. F.}$ for outer wall coefficient, assuming velocity = .35 ft./sec.

$h_i = 1300 \text{ Btu/sq. ft. hr. F.}$ for inner wall coefficient, assuming velocity = 8.3 ft./sec.

$t = 9.5''$, assumed thickness of pipe wall.

$k = 0.33 \text{ Btu in./sq. ft. hr. F.}$, assuming water filled wall.

$$\text{Then } R = \frac{1}{100} + \frac{9.5}{.33} + \frac{1}{1300} = 28.80$$

$$\text{Then } Q = \frac{30 \times 377,000}{28.80} = 392,727 \text{ Btu/hr.}$$

$$\text{Heat input per lb. of water} = \frac{Q}{W} = \frac{392,727}{2.3 \times 10^9} = .00017 \text{ Btu.}$$

$$\text{Temperature rise in water} = \frac{Q}{W \times \text{spec. ht.}} = \frac{.00017}{.95} = .00018^\circ \text{F.}$$

This is so small that it does not appear worthwhile to refine the calculations further. It should be noted that the upper part of the pipe wall is likely to be filled with gas instead of water. Since the conductivity of gas is far lower than that of water, the heat leak and water temperature rise should be insignificantly small.

As a comparison to the above, we can calculate the heat leak for a single wall steel pipe. If we assume 1" wall thickness with ring stiffeners, then the equation for resistance becomes

$$R = \frac{1}{100} + \frac{1}{314} + \frac{1}{1300} = .0139$$

$$Q = \frac{30 \times 377,000}{.0139} = 811 \times 10^6 \text{ Btu/hr.}$$

$$\text{Heat input per lb. water} = \frac{811 \times 10^6}{2.3 \times 10^9} = .352 \text{ Btu/lb.}$$

$$\text{Temperature rise} = \frac{.352}{.95} = .371^\circ \text{F.}$$

While we have probably been too conservative in assuming a very high mean value of Δt , it is apparent that a single wall pipe can have an appreciable heat leak and temperature rise. The tubular wall pipe has a definite advantage.

11. Pipe Support

If it is assumed that the cold water pipe support system must support a negative buoyancy of 8,560,000 lbs. as computed for the steel pipe and plotted on Fig. 6, then the support components can be evaluated with respect to this load. The load will be reduced by the friction drag of water flowing in the pipe when the cold water pump is running. The friction drag force in the pipe is:

$$F = \Delta P_f A = 1.037 \times 144 \times 40^2 \times \frac{\pi}{4} = 187,651 \text{ lbs.}$$

The negative buoyancy of the pipe when water is flowing in the pipe is 8,560,000 - 188,000 = 8,372,000 lbs. This is only about a two-percent reduction in the force required to support the pipe and is not considered significant for the pipe design.

An approximation of the variation in pipe support forces as caused by ocean conditions is calculated in the following paragraphs.

For the assumed conditions of a 60 knot wind speed, a 300 mile fetch length, and a minimum wind duration of 16 hours, the maximum significant wave period is about 16 seconds, and the significant wave height is 40 feet (14, p. 10-33).

The assumed current profile shown in Figure 6 causes the cold water pipe to tilt 1.8 degrees from vertical. A maximum of 5 degrees tilt of the pipe relative to the platform is assumed for the above storm conditions, which cause the platform to heave 7 feet maximum double amplitude. These conditions result in platform motion of:

$$\text{Average vertical velocity} = 2 \times 7/16 = .875 \text{ ft./sec.}$$

$$\text{Maximum vertical velocity} = 2\pi \times 7/(16 \times 2) = 1.374 \text{ ft./sec.}$$

$$\text{Maximum acceleration} = 1.374 \times 2\pi/16 = .540 \text{ ft./sec.}^2$$

$$\text{or} = .540/32.17 = .0168 \text{ times gravity.}$$

It is of interest to calculate the maximum vertical force that this acceleration would cause at the joint between the platform and the cold water pipe if the joint were a solid connection and the platform motion were sinusoidal. The mass of the cold water pipe is approximately 33,600,000 lbs. Therefore the maximum vertical force would be 33,600,000 x .01679 = 564,100 lbs. This is a small force relative to the 8,560,000 lb. support force and would not produce a significant increase in the stress level at the joint. However, it is well known that 40-foot ocean waves will not induce sinusoidal motion in a sea thermal power plant and that the actual acceleration on the plant caused by ocean wave forces is significantly larger than the indicated sinu-

soidal value. Therefore ocean wave forces can greatly increase the stress level above the static value of a rigid pipe connection. This then requires a flexible pipe support system to isolate the cold water pipe from the platform and to reduce the transmission of oscillating force between them.

A rubber ring could be used as a seal between the pipe support sleeve and the pipe support carriage. This ring must transmit a side thrust of 350,000 lbs. between the sleeve and the carriage. If the ring is 12 inches wide, the pressure on the rubber is $350,000 / (44 \times 12 \times 12) = 55.2$ psi. This is quite low and should be acceptable. The ring must also deflect to accommodate the 5-degree maximum tilt angle between the sleeve and the carriage. If the ring is 10 inches thick, the required ring deflection is $(12 \sin 5) / 10 = .105$ inches/inch of rubber thickness. This is acceptable.

The following sample selection of components for the pipe support system illustrates the feasibility of the pipe support design. This first trial selection will be refined and optimized along with future cold water pipe and platform design optimization, which should reduce the required support force.

It is good design practice to have the natural frequency of the system equal to about one-half or less of the minimum exciting frequency, in order to obtain good isolation between the cold water pipe and the platform at all exciting frequencies. The minimum exciting frequency is 1/16 Hertz. Therefore the desired natural frequency is 1/32 Hertz.

The spring constant required to obtain this natural frequency is determined from the following equation.

$$f = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad \text{or} \quad k = m(2\pi f)^2$$

$$k = \frac{33,600,000}{386} \left(2\pi \times \frac{1}{32} \right)^2 = 3356 \text{ lbs./inch of pipe displacement}$$

The support will be designed to produce a change in chain length equal to .7 times the change in pipe displacement when the pipe is at its extreme displacement. At the displacement limit, the value of k is:

$$3356 / .7 = 4794 \text{ lbs./inch of pipe displacement}$$

$$\text{or } 4794 \times 2 = 9588 \text{ lbs./inch of piston displacement}$$

Select a double strand, 3-inch pitch chain with 260,000 lb. strength and allow the design load to be $260,000 / 4 = 65,000$ lbs. Then the number of chains required is $8,560,000 / (.7 \times 65,000) = 189$ chains. The average linear spring constant for each spring is $9588 / 189 = 50.73$ lbs./inch of piston displacement. However, the pipe support spring isolator is not a linear spring.

It has a sinusoidal spring rate. Therefore the maximum spring rate equals the effective spring rate times the square root of two.

$$k_{\max} = 50.73\sqrt{2} = 71.74 \text{ lbs./inch of piston displacement}$$

If an air spring is attached to each chain and the air pressure in the spring is 500 psi. at the design load, then the effective area of the air piston is $2 \times 65,000/500 = 260$ sq. in. At 20,000 psi. piston rod stress, the rod area is 6.500 sq. in., and the rod diameter is 2.877 inches. The cylinder area is $260 + 6.500 = 266.5$ sq. in., and the inside diameter of the cylinder is 18.421 inches. These are reasonable diameters for a 500 psi. air cylinder.

Determine the volume required for the static air cylinder (air reservoir). Design the air spring isolator for isentropic expansion and a perfect gas relationship.

$$P_1 V_1^n = P_2 V_2^n; \frac{P_1}{P_2} = \left(\frac{V_2}{V_1} \right)^{1.406}; \frac{V_2}{V_1} = \left(\frac{P_1}{P_2} \right)^{.71124}$$

$$P_1 = 514.7 \text{ psia.}$$

$$\Delta p = 71.74/260 = .276 \text{ psi.}$$

$$P_2 = 514.7 - .28 = 514.42 \text{ psia.}$$

$$V_2 - V_1 = 260 \times 1 = 260 \text{ cu. in.}$$

$$V_2 = V_1 \left(\frac{514.7}{514.42} \right)^{.71124} = 1.0003871 V_1$$

$$.0003871 V_1 = 260$$

$$V_1 = 671,661 \text{ cu. in.} = 388.7 \text{ cu. ft.}$$

The length of an air reservoir made from 40 inch O.D. steel pipe will be:

$$\frac{4 \times 671,661}{12\pi \times 38^2} = 49.4 \text{ ft.}$$

This is a reasonable length.

The foregoing calculations illustrate that the pipe support system is feasible. The results of the calculations indicate that design refinements can be made to optimize the combined design of the cold water pipe and

pipe support system. Design work to optimize the components in this system should be performed.

12. Vibration and Vortex Shedding

During our analysis of the pipe, the question of forced vibrations due to vortex shedding phenomena has been raised. This is a logical design problem and should be examined prior to construction of the pipe.

$$Re = \frac{\rho v D}{\mu} = \frac{1.989 \text{ slug} \times 6 \text{ ft./sec.} \times 40 \text{ ft.}}{\text{ft.}^3 \times 2.246 \times 10^{-5} \text{ slug/ft. sec.}} = 21.2 \times 10^6$$

This Reynolds number is high and indicates turbulent flow around the pipe. There is very little, if any, quantitative data concerning drag and vortex effects at such high Reynolds numbers.

In some wind tunnel tests at the Guggenheim Aeronautical Laboratory of the California Institute of Technology, Anatol Roshko found that periodic vortex shedding did occur at Reynolds numbers greater than 3.5×10^6 . The dimensionless Strouhal number of the vortex shedding frequency was relatively constant over the range of Reynolds numbers 3.5×10^6 to 10^7 . The range of Strouhal number was

$$s = .26 - .28$$

$$s = \frac{nD}{v}$$

n = shedding frequency in cycles/sec.

D = pipe diameter

v = velocity of fluid past the pipe

Roshko's data goes up to Reynolds numbers as high as 10^7 . This would be the Reynolds number for a current of about 3.6 ft./sec. Solving the Strouhal equation for n :

$$n = \frac{sv}{D} = \frac{.28(3.6)}{40} = .0252 \text{ cycles/sec.}$$

$$2\pi \times .0252 = .1582 \text{ rad./sec.}$$

For $s = .26$

$$n = \frac{.26(3.6)}{40} = .0234$$

$$2\pi \times .0234 = .147 \text{ rad./sec.}$$

Then, according to this experimental data, a pipe of 40 ft. in diameter, if subjected to a transverse current along its entire length, would shed vortices at the predicted frequencies between .0252 and .0234 cycles/sec. However, we see that the shedding frequency n is directly proportional to the current velocity. This means that, as we go deeper and the currents gradually decrease in velocity, the vortex shedding frequency also decreases. It is extremely unlikely that a steady current would exist across the pipe from top to bottom and thus produce vortex shedding at a single frequency. In reality, the current velocity varies with the ocean depth, and therefore the vortex shedding frequency (the exciting frequency) varies along the length of the pipe. This means that the pipe will sense a white noise type of exciting frequency which cannot excite the natural vibration frequency of the pipe.

Let us examine the pipe and see what natural frequencies exist for the pipe acting as a beam.

We have designed the pipe so that it hangs beneath the hull structure with a single point attachment. The attachment permits angular movement of the pipe with respect to the hull structure and axial movement of pipe up and down.

Let us make the simple assumption that the pipe is simply supported at one end and free at the other. The bending natural frequency of the beam is

$$\omega_n = a_n \sqrt{\frac{EI}{\mu_1 l^4}} \text{ rad./sec.}$$

E = modulus of elasticity

I = cross-section moment of inertia

μ_1 = mass/unit length

l = length

a_n = numerical constant = 15.4

For a steel pipe, $E = 30 \times 10^6 \text{ lb./in.}^2$. If we make up the pipe using 12.750" O.D. tubing, wall thickness .358", weighing 47.35 lb./ft., and I-beam spacers weighing 19.39 lb./ft. of length, then the weight of steel for each ft. of pipe is

$$47.35 + 19.39 = 66.74 \times 113 = 7541.62 \text{ lb./ft.}$$

If this weight of metal were assumed to be smeared over the perimeter of the 40-ft. diameter pipe, then it would act as a large, thin wall cylinder of wall thickness 1.43 in. The moment of inertia would be

$$I = \frac{\pi(d_1^4 - d_2^4)}{64} = \frac{\pi(480.72^4 - 479.29^4)}{64} = 31.053 \times 10^6 \text{ in.}^4$$

μ_1 = mass per unit length

$$\mu_1 = \frac{\pi(480.72^2 - 479.29^2)}{4} \times \frac{.283 \text{ lb. /in.}^3}{386 \text{ in. /sec.}^2} = \frac{.790 \text{ lb. sec.}^2}{\text{in.}^2}$$

For a pipe 3000 ft. long

$$\omega_1 = .409 \text{ rad. /sec.}$$

For a pipe 4000 ft. long

$$\omega_1 = .230 \text{ rad. /sec.}$$

These natural frequencies are sufficiently higher than the vortex shedding frequencies calculated above at 3.6 ft. /sec. Furthermore, because of the stockade pipe design, there will be some vibration damping between tubes and spacers.

This analysis was based on the simplified beam end support. Actually, the upper end will be supported by a spring both in the angular mode and in an axial mode. It could be argued that maybe these springs would affect the natural frequency vibrations of the pipe as a beam. However, when we once know more precisely how vortex shedding frequencies are affected by the current distribution, then we can design the spring rate for whatever is desirable. This is another distinct advantage of the spring support system we have designed, because the spring rate can be varied.

CONCLUSIONS

The following modes of failure have been investigated and the results reported in this paper:

1. Transverse shear on the cold water pipe when the pipe acts as a beam. This mode would cause the pipe wall metal to yield. (p. 20 and Fig. 6)
2. Shear flow along the entire 4000 ft. of pipe. This mode would cause a tube to slip relative to an adjacent spacer. (p. 21 and Fig. 6)
3. Buckling collapse stress in the pipe caused by uniform differential pressure across the pipe wall. The pipe would collapse if the differential pressure was too large. (p. 16)
4. Compressive stress in the pipe wall caused by the uniform differential pressure across the pipe wall. (p. 17)
5. Bending moment caused by the current acting on the pipe when the pipe performs as a beam. The bending moment stress and the additive longitudinal load stress would cause the pipe wall metal to yield. (pp. 19, 29, and Fig. 6)
6. Circumferential bending moment in the pipe wall caused by the current-induced pressure variation around the cylindrical pipe. In this mode, the pipe becomes oval-shaped. Both the positive and the negative circumferential pipe wall bending moments were investigated. (p. 25)
7. Buckling stress in the tube caused by uniform differential pressure across the tube wall. (p. 12)
8. Shear flow in the tube. (p. 21)
9. Circumferential bending moment stress in the tube wall caused by interaction with the I-beam spacer. (pp. 23 and 26)
10. Bending moment stress in the I-beam spacer flange caused by interaction with the tube. (pp. 24 and 27)
11. Shear flow in the I-beam spacer. (pp. 24 and 27)
12. Tension band loading. (pp. 21 and 28)

Analysis of all the modes of failure indicates that the construction of the cold water pipe from a stockade-type pipe wall design is feasible. The

analysis indicates that vertical shear loads must be transmitted between spacers and tubes and that tension bands alone can provide all of the required prestressing and clamping force in the pipe wall members. Therefore shear joints between the pipe wall elements are not required, but interlocking shear joints could be used in the pipe construction.

An investigation of standard manufacturing tolerances for tubes and I-beams indicates that a device will be necessary to position and retain each pipe wall element. The manufacturing and assembly tolerances are so large that, unless a retention device is used, there is a statistical possibility that the interacting forces would not act to resist all destructive forces applied to the pipe. Design of a position and retention device is a simple engineering task and does not affect the feasibility of the stockade-type pipe wall design.

The material cost of a 4000 foot long, 40 foot diameter cold water pipe utilizing 12-inch diameter tubes made of steel weighing 32,476,000 lbs. is estimated to be \$6.5 million. If the cold water pipe is made of aluminum, it is estimated that the material would cost 2.6 times the cost of the steel pipe.

The labor cost required to construct the cold water pipe from the deck of the sea thermal power plant is estimated to be \$1.5 million. The labor cost is a smaller than expected fraction of the material cost because the amount of labor time has been reduced by the design of the pipe. For example: Time consuming welded tube joints are eliminated and replaced by expanded tube joints, which can be made quickly and easily and are more reliable than the welded tube joint.

The cold water pipe design which utilizes 12-inch tubes has not been evaluated to determine if it results in the optimum plant cost. Another tube size may yield a lower plant cost when all cost factors are evaluated. Some of the cold water pipe-related costs that must be evaluated in more detail to produce the optimum cost are:

1. Pipe material
2. Pipe construction labor
3. Pipe support system cost as a function of the supported pipe weight
4. Buoyancy that must be built into the platform to support the weight of the pipe.
5. Buoyancy that can be incorporated in the pipe.

Design methods can also be used to reduce the pipe cost. One method is to use more variation in the size of the pipe wall elements along the

length of the pipe. For example, thinner wall tubes and lighter weight I-beam spacers could be used at the deep end of the pipe. Another design method of cost reduction is to replace some of the tension bands with structural rings.

From the foregoing preliminary design analyses, it is feasible to use the tubular wall construction for the cold water pipe. It is probable that preliminary assumptions have been too severe and that further data and analyses should permit a lighter weight wall to be used. It would be desirable to use aluminum tubes to make the pipe lighter and have more buoyancy, as well as permit better distribution of buoyancy.

This study has enabled us to evaluate the feasibility of the stockade construction and to indicate the direction for improvements and lower overall costs.

RECOMMENDATIONS

This preliminary study has provided sufficient information to guide us toward what the next steps should be. The following recommendations are important steps toward optimizing a final pipe and support design:

1. We should get measured drag coefficients of fluted structures like this at suitably high Reynolds numbers.
2. We should get motion data on a floating platform in a suitable wave tank, with a simulated cold water pipe suspended from it.
3. We should collect data on ocean currents vs. depth at different typical locations so as to evaluate drag forces much more closely. This has a large effect on design requirements.
4. We should collect weather data and surface wave shape data so we can evaluate platform motion acceleration caused by any sea state.
5. We should continue to investigate the possibility of using aluminum tubes as well as steel.
6. We should take more detailed evaluation steps of current and drag information that is presently available. The assumptions used for this preliminary design are probably excessively high but have served to outline the design problems.
7. We should evaluate weight and buoyancy distribution of the vessel in the towed condition and in the operating condition. Storm and typical weather, wave, and current conditions will be required to make this analysis.

8. We must investigate pipe angle from vertical, caused by buoyancy vs. drag conditions.
9. We must go into more detail on pipe support design.
10. We must develop more detailed analysis of tube joint design, vertical shear analysis, tension band design, and retention device design.
11. We must confirm the cold water pipe design assumptions by extensive testing of a scale model of the pipe and pipe support.
12. We must perform an analysis to optimize cost of the cold water pipe.

While items 1 through 4 above are beyond the scope of the present investigation, the preliminary results of this study provide enough encouragement to justify such further investigations.

FIG. 1

FIG. 2

TYPICAL PIPE SEPARATOR

TUBES WITH I SPACERS

FIG. 3

SKETCH OF SSP COLD WATER PIPE SUPPORT

FIG. 6

FIG. 7

VERTICAL LOADING DIAGRAM

FIG. 8

REFERENCES

1. "An Analysis of Thin Shelled Cylindrical Bodies," by Albert J. Faulstich, Jr., Design News, October 27, 1969.
2. "Flow of Fluids," Crane Co. Paper No. 410.
3. "Formulas for Stress and Strain," by Roark; McGraw-Hill, p. 173.
4. "Handbook of Engineering Fundamentals," by Eshbach; John Wiley, p. 7-102.
5. "Heat Transport of the Florida Current," by Chew, Frank, Wagner, and Work; University of Miami Technical Report No. 57-21, 1957.
6. "Handbook for Mechanical Engineers," by Marks; McGraw-Hill, p. 5-23.
7. "Dynacat-semi-submersible for 4000 Ft. of Water," by Barnard Cintract, Ocean Industry, April, 1975, pp. 101-107.
8. "Resistance of Materials," by F. B. Seely and J. O. Smith, fourth edition; John Wiley & Sons, Inc., N. Y., 1956.
9. "Engineering Applications of Fluid Mechanics," by Hunsaker and Rightmire; McGraw-Hill, 1947.
10. "Design of Cold Water Pipe for Sea Thermal Power Plants," SSP-2691-1 Progress Report for Period 1 May 1975 - 15 September 1975 by J. Hilbert Anderson; Sea Solar Power, Inc., York, PA.
11. "Hydrostatic Design Basis for Techite RPM Type 1 Pressure Pipe," technical report no. ER-01033 Rev. B; Amoco Reinforced Plastic Company, Riverside, CA.
12. "General Specifications and Ratings," ST 100-2; A. O. Smith - Inland, Inc., Reinforced Plastics Division, Little Rock, AR.
13. J. H. Anderson personal visit of April, 1976, to Lamont-Doherty Geological Observatory at St. Croix, U. S. Virgin Islands.
14. "Handbook of Hydraulics," H. W. King and E. F. Brater, 5th ed.; McGraw-Hill, New York.
15. "Stress Problems with Thin Wall Pipe," T. J. Atterbury, Pipe Line Industry, Oct., 1962.

APPENDIX A

Drag force caused by flow past a long cylinder.

B = inches, width

C = psi., constant

F = lb. /inch of pipe

R = inches, radius

U = ips., velocity

θ = degrees, variable angle

ρ = slugs, density

ϕ = degrees, angle of flow separation

For a cylinder, half the force equation is:

$$\frac{F}{2} = BRP \cos \theta \, d\theta + BRC \cos \theta \, d\theta$$

$$\text{where } P = .5 \rho U^2 (1 - 4 \sin^2 \theta) \quad (\text{Ref. 9, p. 227})$$

$$.5F = .5BR\rho U^2 \left(\int_0^\phi \cos \theta \, d\theta - 4 \int_0^\phi \sin^2 \theta \cos \theta \, d\theta \right) + BRC \int_\phi^\pi \cos \theta \, d\theta$$

$$.5F = .5BR\rho U^2 \left(\sin \phi - \frac{4}{3} \sin^3 \phi \right) - BRC \sin \phi$$