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DESIGN AND OPERATION OF CHILLED SEA WATER SYSTEMS

by

Frank N. Lee

Department of Fisheries and Oceans
Fisheries Research Branch
6640 N.W. Marine Drive
Vancouver, British Columbia
V6T 1X2 Canada

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ABSTRACT

Lee, F.N. 1985. Design and operation of chilled sea water systems. Can. Tech. Rep. Fish. Aquat. Sci. 1363: 31 p.

Chilled sea water (CSW) systems, or better known as champagne systems in the Canadian west coast fishing industry, deliver fish in prime condition due to the rapid and uniform chilling that can be achieved in the ice-sea water medium agitated by air. Another attractive feature of the system is the relatively low initial installation cost although the cost of ice to run it has to be taken into consideration. High volume of low pressure air from an air blower is distributed in the fish holding tank with the slush ice mixture uniformly by small air orifices distributed along branch pipes the separation of which is less than 90 cm. When a branch pipe is located next to a solid boundary, the separation between the two should be no more than 45 cm. Air flow requirement for CSW systems is 9.14 m^3 per square meter of deck head area per minute, measured at 38.9°C and 101 kpa. The approximate time required for chilling fish weighing up to 3.5 kg (salmon or groundfish) is 6 hours. For smaller fish such as herring, this time can be reduced to 2 hours. Methods for estimating ice requirement for the operation of the system and pressure loss in the air pipes are given in the appendices.

Keywords: fish refrigeration, chilled sea water system, design, operation

RÉSUMÉ

Lee, F.N. 1985. Design and operation of chilled sea water systems. Can. Tech. Rep. Fish. Aquat. Sci. 1363: 31 p.

Les systèmes de réfrigération à eau de mer, plus connus sous le nom de systèmes au champagne dans l'industrie halieutique de la côte ouest canadienne, donnent du poisson de qualité supérieure grâce à un refroidissement rapide et uniforme dans l'eau de mer glacée et aérée. Le coût assez modique d'installation initiale est un autre avantage du système, quoique le coût de la glace nécessaire pour le fair fonctionner doive être considéré. Des volumes élevés d'air à basse pression produit par un ventilateur sont injectés dans le vivier contenant le mélange d'eau et de glace concassée; l'air est distribué uniformément par de petites ouvertures réparties le long de tuyaux séparés de moins de 90 cm. Quand un tuyau est situé près d'une paroi, la séparation entre les deux ne devrait pas être supérieure à 45 cm. Le débit d'air requis pour de tels systèmes s'élève à 9,14 m^3 par m^2 de superficie du devant de pont et par minute, mesuré à 38,9°C et 101 kpa. Le temps approximatif requis pour la réfrigération de poissons pesant jusqu'à 3,5 kg (saumon ou poisson démersal) est de 6 heures. Pour de plus petits poissons comme le hareng, ce temps peut être réduit à 2 heures. On présente en annexe des méthodes d'estimation des besoins en glace pour l'exploitation d'un système et de la perte de pression dans les tuyaux d'air.

Mots-clés: réfrigération du poisson, système de réfrigération à l'eau de mer, conception, exploitation

I. INTRODUCTION

The chilled sea water (CSW) system was initially designed in 1976 for the chilling of salmon on board small collectors in which there is inadequate space to accommodate the relatively large refrigeration compressor and auxillary engine needed for refrigerated sea water (RSW). This system can be considered as a combination of the slush ice system in which fish were chilled in bulk in an ice-sea water mixture circulated by a pump, and the air-agitated containerized fish chilling system developed in the UK (Eddie and Hopper, 1975). The slush ice system failed to chill the fish uniformly and therefore is now seldom used. This shortcoming is overcome by air agitation in the CSW system.

CSW systems are better known as "champagne systems" on the Canadian west coast. During the initial period when the CSW system was introduced, a salmon packer delivering fish to a processing plant in Victoria, B.C. operated the system throughout the trip back, even when they were in the harbour. The continuous injection of air into sea water mixed with fish blood and slime created a large quantity of foam overflowing the hatch onto the deck of the vessel. Curious bystanders asked what that was and the skipper jokingly said "Champagne!".

Simple to operate, CSW systems deliver fish in prime condition to the processing plants due to rapid and uniform chilling. This system is more compact than RSW system and requires less power to operate. The initial capital cost is low although the cost of ice to run the system has to be taken into consideration. For these reasons, the number of installation of CSW systems mushroomed in the ensuing years in vessels that encompass small trollers, seiners, draggers, salmon packers and collectors, with capacities ranging from a few tonnes to 500 tonnes, participating in the catch or transport of salmon, groundfish or herring. However, the scope of CSW application is not limited to these fisheries.

One unique feature of the system is the large, almost instantaneous refrigeration capacity available from the melting of ice. This feature differs from RSW systems in which only certain refrigeration capacity per unit time is available depending on the size of the compressor. In fisheries in which high catch rates are prevalent, the application of CSW system should certainly be considered.

This report describes the design, specifications and operation of CSW systems for typical conditions on the B.C. coast. In the appendices, methods for calculating pressure loss and heat load in CSW systems are presented.

II. RECOMMENDED CSW SYSTEM DESIGN

A) Piping installation

A CSW system is essentially an ice-sea water mixture agitated by air to chill the fish efficiently and uniformly down to the equilibrium temperature. Compressed air from an air blower is distributed evenly in the holding tank by a distribution piping system as shown in Fig. 1, 2 and 3. The air blower is usually installed in the engine room with the suction located in the ventilation shaft or other locations where cool air can be obtained. It is important that the temperature of the air be as low as possible because it can increase by as much as 60°C due to compression in the air blower. The cool air is led down to the air blower, compressed to the operating pressure and delivered to the fish holding tanks through an inverted "U" section and some shut off valves. There should be a discharge pressure gauge immediately downstream from the air blower, 0 to 100 kpa range and a safety pressure relief valve set to the maximum operating pressure of the air blower or lower. The inverted "U" section is there to prevent sea water from flooding back to the air blower and should be at least 1 m above the highest water level in the tanks. A check valve can be used in conjunction with this but not to replace the inverted "U". Experience has shown that a check valve can easily be rendered nonfunctional by fish scales or debris.

The size of pipe depends on air flow rate and the maximum operating pressure of the blower. Large size pipes should be used for high flow rates to avoid excessive friction losses the total of which must be lower than the maximum operating pressure. However, once the size of the pipe at the discharge of the blower has been determined, the sizes of feeder and branch lines and air orifice should be determined by the so-called "manifold principle", ie. the total cross-sectional area of the branch pipes should be equal to the cross-sectional area of the feeder line and similarly for the air orifices. For easy reference, the approximate number of smaller diameter pipes or orifices required for this purpose is given in Table 1.

Table 1. Approximate number of pipes required to obtain equal cross-sectional area to a larger pipe (Sch #40 pipes only).

		Diameter of Discharge or Feeder Line (mm)					
Nominal Diam.		100	50	38	25	19	13
Internal Diam.		102.3	52.5	40.9	26.6	20.9	15.8
Diameter of Feeder or Branch Line (mm)	50	4	1				
	38	6	2	1			
	25	15	4	2	1		
	19	24	6	4	2	1	
	13	42	11	7	3	2	1
Orifice	4.8 mm	454	120	73	31	19	11
	3.2 mm	1021	269	163	69	43	24

For example, if the discharge line is 50 mm, and the size of the feeder line is 25 mm, under "Diameter of Discharge or Feeder Line", 50 mm is located. Then under "Diameter of Feeder or Branch Line" on the left hand side, the row 25 mm is found. The intersection of the two gives that four 25 mm pipes should be used. Or conversely, if four pipes are required to span the tank to give a spacing of 90 cm, the size of the branch line should be 25 mm. Sometimes it may be found that four pipes are not sufficient to satisfy the requirements of Section IIA; ie. the separation between pipes may be greater than that recommended. To satisfy the requirement, maybe six 19 mm pipes will have to be used. In a situation like this, it is recommended that no pipes smaller than 2.5 cm should be used as they can be damaged easily. However, the number of air orifices in the branch line should be determined on the basis that the 1.9 cm pipe has been used. For example, if the size of the branch line is calculated to be 1.9 cm but 2.5 cm pipes are used, the number of 0.48 cm holes drilled in the 2.5 cm pipe should be 19 and not 31.

For practical considerations, PVC pipes are not recommended for installations where the pipes are exposed such as shown in Fig. 2. Experience shows that such pipes can easily be broken by fish unloading devices such as fish pumps. Normally aluminum pipes are used in fibreglass lined tanks. If metallic lining material is used, corrosion of the dissimilar metals in sea water has to be considered (Bosich, 1970). In the engine room, schedule #40 PVC pipes are not recommended either as they cannot maintain their rigidity at high temperatures. Either galvanized or aluminum pipes are acceptable as they do not have the same shortcoming.

As for all refrigeration systems in which fish are held in a liquid medium such as CSW, special attention has to be directed to cleaning and sanitizing. Such procedures can be found in Kolbe and Lee (1980). As far as CSW systems are concerned, it is important to provide a connection to allow flushing of the air distribution piping system with clean water as shown in Fig. 1.

There are various ways to install air distribution pipes in the fish holding tank. The primary objective of the system is to distribute air bubbling evenly in the tank. Two common types of this are shown in Figs. 2 and 3. In Fig. 2, the air distribution branch pipes are installed on top of fibreglass covered wooden blocks, 5 to 7 cm high. The separation between the branch lines should be no more than 90 cm apart, a figure arrived at empirically to be used with the air flow rate specified later in this report. Air orifices can be drilled underneath or on the top of the pipes. This method of installation is suitable for existing tanks with proper insulation and fibreglass lined. For other lining materials, different methods may be required to secure the branch lines.

In Fig. 3, the air distribution branch lines are embedded in the insulation, resulting in a smooth fibreglass surface inside the tank free of obstructions and easily cleanable. Air orifices are drilled from the top after the fibreglass is set. The branch lines end in a small trough where they are capped off during normal operation. The end openings provide cleaning access to the pipes at the end of a trip. This type of construction can be used in new installations. The separation between branch lines is the same as in the previous case, i.e. no more than 90 cm.

In both cases, the feeder line should go through the engine room bulkhead at the highest possible location.

B) Air blower specification

The air blower provides the pressurized air needed for agitation of the ice-sea water-fish mixture in the holding tank. With the air distribution pipes in configurations shown in detail in Figs. 2 and 3, it was found empirically that 9.14 m^3 standard air/min per $meter^2$ of deck head area (from 0.5 ft^3/min per $foot^2$) is adequate (standard air refers to air at 38.9°C and 101 kpa). This rate does not cause excessive turbulence when agitating and yet is sufficient to chill the fish uniformly and quickly.

For a large vessel, the capacity of the air blower required may be unnecessarily high if the above rate is used. In cases like this, other practical considerations will have to be taken into account. For example, a six-tank vessel of total deck head area of 85 m^2 will require a capacity of 12.9 m^3/min according to the rate specified. This requires a high capacity air blower with the attendant high power requirement, large suction and discharge lines. However, if one looks at the simultaneous air requirement, this capacity may be reduced by more than 60%.

Whether the vessel is involved in packing or fishing, it takes time to load the tanks. If the time it takes to do that is about the same as or longer than the time to chill the fish, then only one or two tanks need be agitated at any one time. (The two tank requirement arises from the need to load two tanks at the same time for balancing of the vessel or faster loading.) Therefore capacity for only one or two tanks is needed.

In the design of CSW systems for large vessels like the one under consideration, it is advisable to install two smaller air blowers to agitate one tank each instead of one of higher capacity to agitate two tanks at the same time. The initial cost is higher but one unit will always back up the other with a crossover arrangement as shown in Fig. 1 in case of breakdown or malfunction.

With the capacity determined, the operating pressure has to be known to specify the air blower completely. Air is compressible, ie. its volume depends on pressure. This property makes it difficult to calculate pressure losses in pipes and fittings as they depend on flow velocity. A method for estimating pressure loss in the entire piping system is given in Appendix 1. However for typical B.C. vessels in which the total length of pipes is less than 15 m, feeder line not less than 5 cm in diameter, deck head area not more than 14 m^2 and maximum water depth less than 3.5 m, generally the air pressure required will be less than 55 kpa.

III. OPERATION OF CSW SYSTEMS

The main objective of operating the CSW system is to bring the temperature of the fish down to the equilibrium temperature as quickly and uniformly as possible. Uniformity in temperature is achieved through good design and, to a lesser extent, even distribution of ice in the tank. The rate of chilling is greatly enhanced by agitation of the mixture with air provided sufficient chilling capacity is available.

Chilling capacity in CSW systems is generated by the melting of ice. For each kilogram of ice melted, the chilling capacity released (or the heat absorbed) is sufficient to lower 88.9 kg of fish by 1°C. Ice is therefore the single most important ingredient in the CSW system and sufficient amount must be loaded at the beginning of a trip. The amount of ice required depends on several parameters: the initial fish and sea water temperatures, ambient temperature, tank insulation and length of trip. How these parameters affect the amount of ice needed is shown in the calculation in Appendix 2. However, for conditions found typically on the B.C. coast, a proven rule of thumb can be used.

In the summer time on the B.C. coast, sea water temperature is about 13°C and the ambient temperature is in the neighbourhood of 18°C. For a trip lasting 7 days with initial fish temperature at 13°C and tanks insulated with 7.5 to 10 cm polyurethane insulation, a fish to ice ratio of 4:1 by weight is conservatively sufficient. For example, a 30 tonne capacity tank will require 7.5 tonnes of ice. At the end of the first trip, the excess amount of ice will be noted and the quantity of ice will be adjusted accordingly in the following trip. Repeat this procedure until an optimal amount is found.

Once ice is loaded, it should be spread evenly on the bottom of the tank for two reasons. As mentioned earlier, if ice is not distributed in this manner, chilling may not be uniform. In a CSW system, the vertical movement of the cold sea water is assisted by air bubbles while the lateral movement is blocked by the presence of fish. Therefore it takes time for cold sea water to be transported laterally from where the ice is to where there is no ice, resulting in delayed chilling or hot spots in the tank. The other reason is that in the event the ice freezes up, a sheet of ice is easier to break up than an iceberg.

To avoid freezing up of ice, some operators prefer to add some sea water to make a slushy mixture as soon as the boat is in an area where good (in terms of salt content), clean sea water is available. Other operators may not be able or willing to do so because of stability or other problems.

As soon as the vessel arrives at the fishing ground, fish can be loaded into the tanks. If the ice has been made slushy already by the addition of sea water, just simply turn the air blower on to agitate. More sea water may be required as more fish are added to ensure that they are totally submerged for good heat transfer. However, it must be remembered as heat is extracted from the fish, ice will melt to supply part of the water required. This source of fresh water is important for maintaining fish temperature above their freezing point as will be discussed later. Therefore avoid the spillage of water. The optimal amount of sea water to add can best be obtained through practice. However, the example in Appendix 2 in which 12.8% of the tank capacity is sea water can serve as a guide for operation on the B.C. coast.

If the ice has not been made slushy beforehand, most likely it will have been frozen up by the time the vessel arrives at the fish ground. In that case, enough fish should be loaded on top of the ice before sea water is pumped into the tank to prevent floating. The heat from fish and air agitation should break up the sheet of ice in a reasonably short time. Some operators prefer to break up the ice with an ice pick or other tools before the tank is loaded.

It cannot be over emphasized that any fish loaded into a holding tank should be clean and no mud or sand should be mixed especially for ground fish operations. Cases have been reported where branch lines are totally blocked by mud and fish debris.

The time required to chill the fish thoroughly in CSW systems depends on the size of the fish. In a properly designed system with a 72% loading factor (720 kg of fish/m^3), the time required for chilling salmon and ground fish up to 3.5 kg/fish should be no more than 6 hours. In the case of herring, this time can be reduced to two hours. The air supply can be turned off after the load has been chilled as the sole function of the air is to accelerate heat transfer and prevent the formation of hot spots.

While the mixture is not agitated, due to heat leakage into the tank through the insulation, the temperature of the outer region of the mass of fish next to the tank lining will increase and this increase will propagate toward the thermal centre of the mass. In a well insulated tank, the temperature increase is only a fraction of a degree in a 24 hour period. This temperature difference can be removed by a short period of agitation. Depending on the distance D as shown in Fig. 2, 15 to 30 minutes will be sufficient. It should be noted that the temperature of the mixture can only increase in the direction as described, ie. from the outer regions toward the thermal centre. Only when fish are not properly chilled temperature can increase from the centre out with the fish that are still warm providing the source of heat.

It is important that every vessel should be equipped with a remote reading electronic or other type of thermometer suitable for CSW to guide the operator. The range of temperature should cover -5.0 to 20°C in at most 0.2°C divisions and the accuracy should be within ±0.3°C. These thermometers are available from electronics or refrigeration outlets or can be made by electronics hobbyists by using simple components and circuitry (Lee and Roach, 1976).

IV. OPERATING TEMPERATURES IN CSW SYSTEMS

The operating temperature in a CSW system is dependent on the salinity of sea water and is the same as its freezing point or equilibrium temperature at which ice and sea water co-exist. Along the B.C. coast, sea water with a specific gravity of 1.025 (3.4% salt) is available. The freezing point of this sea water is -1.94°C according to Fig. 4. It can be seen from the same figure as specific gravity decreases from 1.0283 to 1.0000, the freezing point rises from -2.2 to 0°C. In a CSW system ice melts continually because of the heat from the fish, the sea water, the fish holding tank and infiltration through the insulation and hatch openings. The fresh water from melted ice dilutes the sea water and therefore the freezing point rises continually. The final operating temperature is usually in the range from -0.5 to -1.1°C. However, there are cases in which the temperature can be lower than this.

As shown in Appendix 2, about 74% of the ice is used to chill the load of fish. If the heat load from the fish is small, only a small amount of ice is melted. The freezing point of this mixture is then low and temperatures as low as -1.94°C have been measured. There are two cases in which special attention of the operator is required to avoid the partial freezing of valuable roe or fish.

When packing salmon from seiners or collectors, very often the fish are already chilled. There is only a small heat load, if at all, associated with these fish and therefore practically no ice will melt. Thus the freezing temperature will be almost the same as that of pure sea water resulting in a potential partial freezing environment. Similarly if only a few fish are loaded into a tank designed for a much higher loading rate, the heat load is again small and no significant amount of ice is melted to dilute the sea water. Once again partial freezing may take place.

Under these circumstances, it is advisable to partly fill the tank with fresh water instead of totally with sea water to raise the operating temperature. The use of thermometer and hydrometer in association with Fig. 4 will guide the operator as to the amount of fresh water required. Partial freezing of roe has been observed at a temperature as high as -0.6°C, corresponding to a specific gravity of 1.0080 (Gibbard, Tech. Services Div., FRB, pers. comm.).

V. SOME COMMON MISCONCEPTIONS

Since the introduction of CSW in the summer of 1976, a number of misconceptions have been formed about the principle and the operation of CSW systems. In the following, some of the common ones will be discussed.

1) The longer the air is turned on, the lower the temperature of the mixture falls.

This is not true at all. Air bubbling only provides agitation to the mixture but not refrigeration capacity which comes solely from the melting of ice. For each kilogram of ice melted, the temperature of 89 kg of fish will by lowered by 1°C. Therefore no matter how long the mixture is agitated, the temperature stays at the freezing point and no lower. As a matter of fact, the temperature of the air is usually 35 to 60°C higher than the ambient temperature due to compression. Some heat is transferred to the mixture and therefore some ice is used up to absorb this heat (about 20 to 25 kg/hour at a flow rate of 1.7 m^3/min at 71.1°C). Theoretically this will raise the freezing point of the mixture due to the additional dilution of sea water. However this increase in temperature is so minor that probably it will never be noticed.

2) The more ice is left in the tank, the better the system is working.

Too much ice left in the tank is wasteful as it is not doing any useful work. It absorbs heat only when it is melted. This heat may come from the fish, sea water or infiltration through the insulation or openings. If there is an excessive amount of ice in the tank, it indicates either the load is not properly chilled or too much ice has been taken. Fish temperature should be checked to ensure that they are properly chilled. Of course there should be a certain amount of ice left in the tank to ensure that the temperature is kept below 0°C but excessive amount is unnecessary and wasteful.

3) If sea water is pumped into the tank to make the ice slushy on the way to the fishing ground, ice will melt faster than taking sea water on arrival at the fishing ground.

The amount of ice melted does not depend on at which point of the trip ice is made slushy. Therefore adding sea water to the ice on the way will not use any more ice than at any other time. As mentioned in "The Operation of CSW Systems", it is recommended that ice be made slushy as early in the trip as possible to avoid freezing up.

4) Slush ice systems without air bubbling work as well as CSW systems.

A slush ice system without air bubbling does not work as well as CSW systems. If there is no air to agitate the mixture, cold as well as hot pockets or temperature stratification will form as learned in the old slush ice system in which agitation was provided by a sea water circulation pump. The discharge of the pump was located higher than the water level in the tank. Obviously the sea water dumped into the tank this way did not create enough agitation to overcome the formation of hot and cold spots. In regions where it is too cold, partial freezing of fish and roe may occur and in regions where it is too hot, spoilage of fish will be accelerated. The slush ice system has not been used in the B.C. fishing industry for a number of years.

5) The motion of the vessel will serve to agitate the tank and therefore there is no need for an air blower.

While it is true that some mild agitation results from the motion of the vessel, this gentle action is far from adequate for the efficient and uniform chilling of fish. A good example to look at is a cup of coffee with cream and sugar. It will take a long time to mix the three components together uniformly by rocking the cup back and forth. With a spoon providing the stirring motion, this can be accomplished in seconds.

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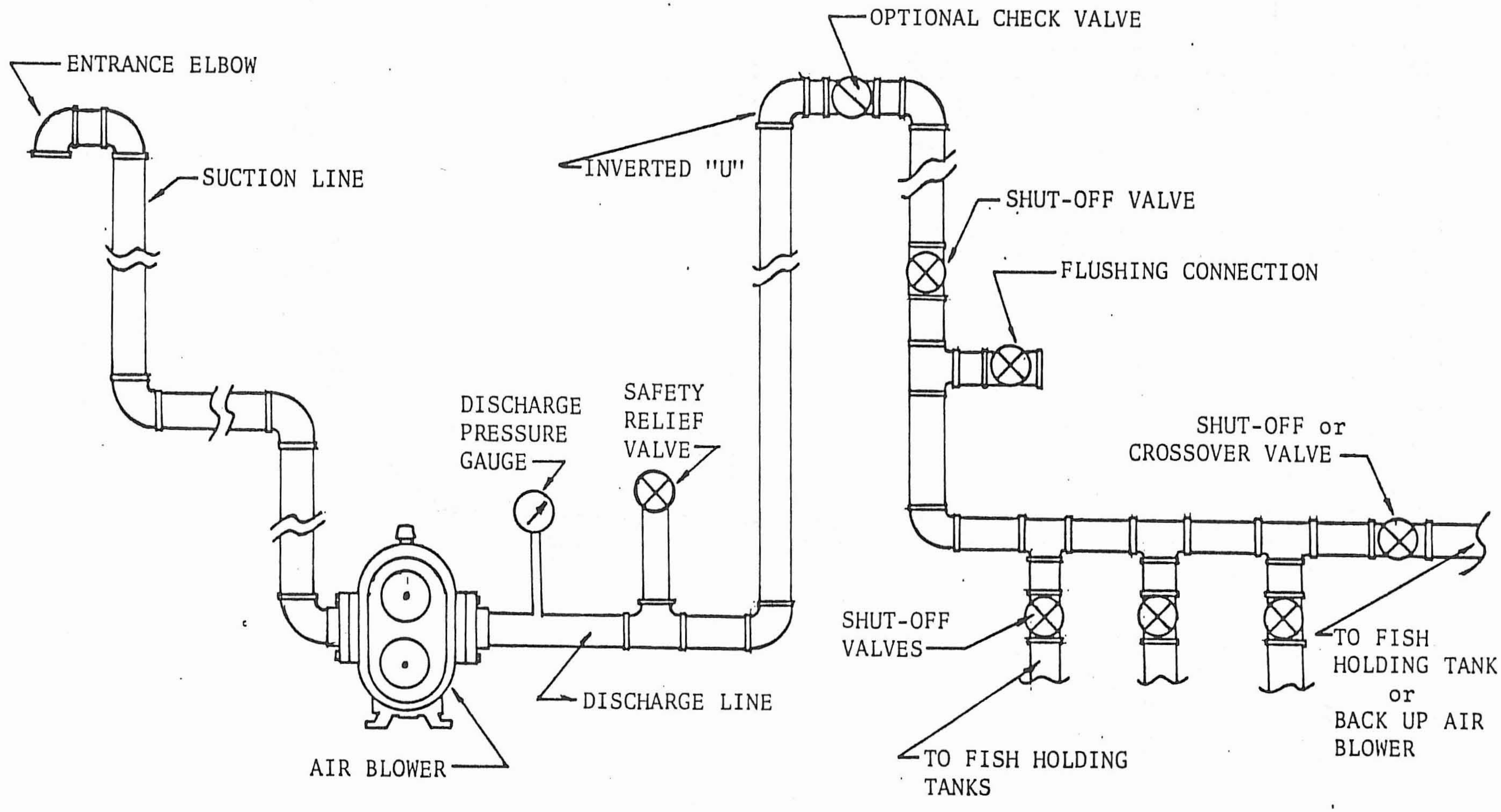

FIGURE 1. CSW piping arrangement outside of fish holding tank.

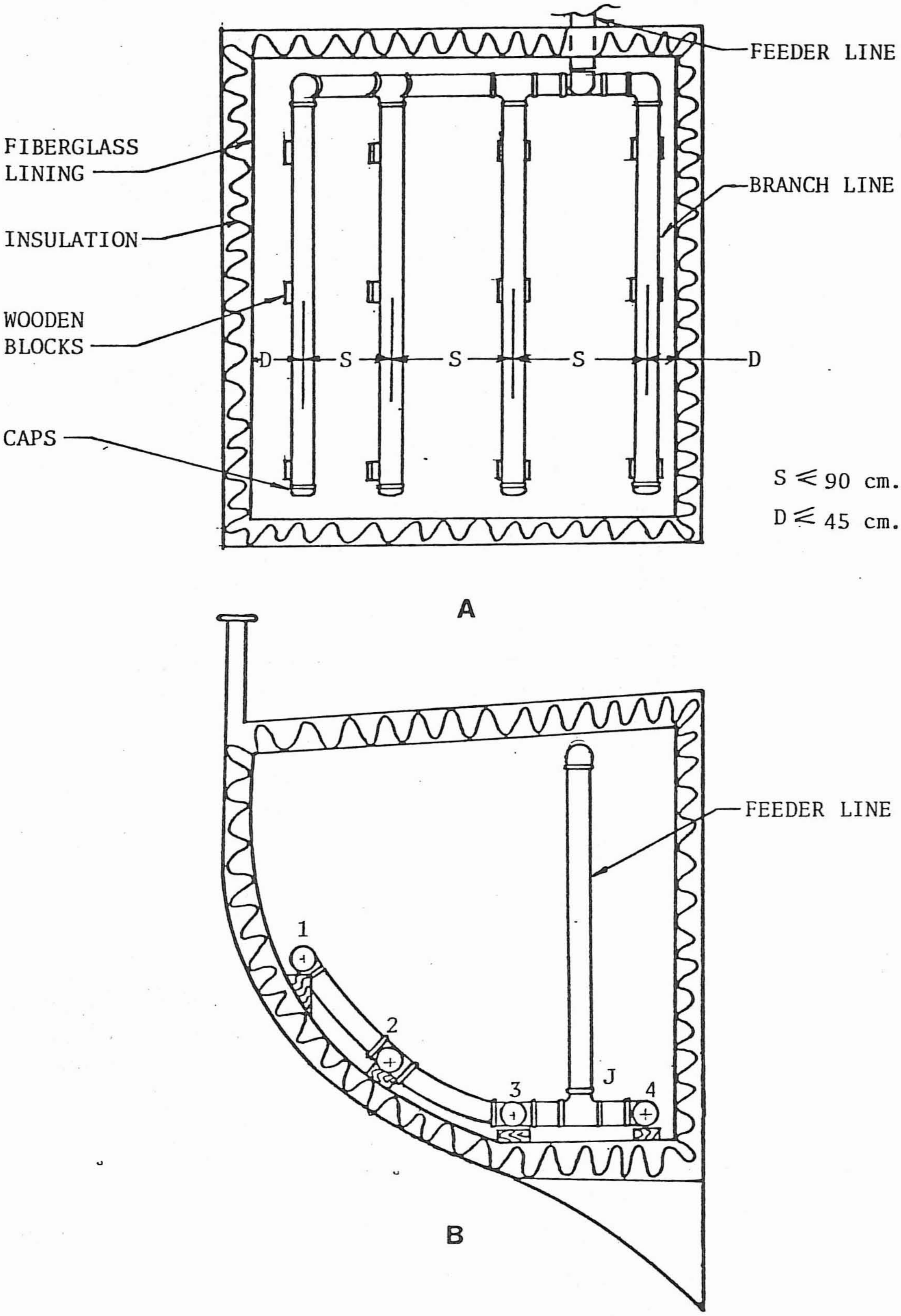

FIGURE 2. Schematic of CSW piping inside the fish holding tank. A=Plan View, B=Elevation.

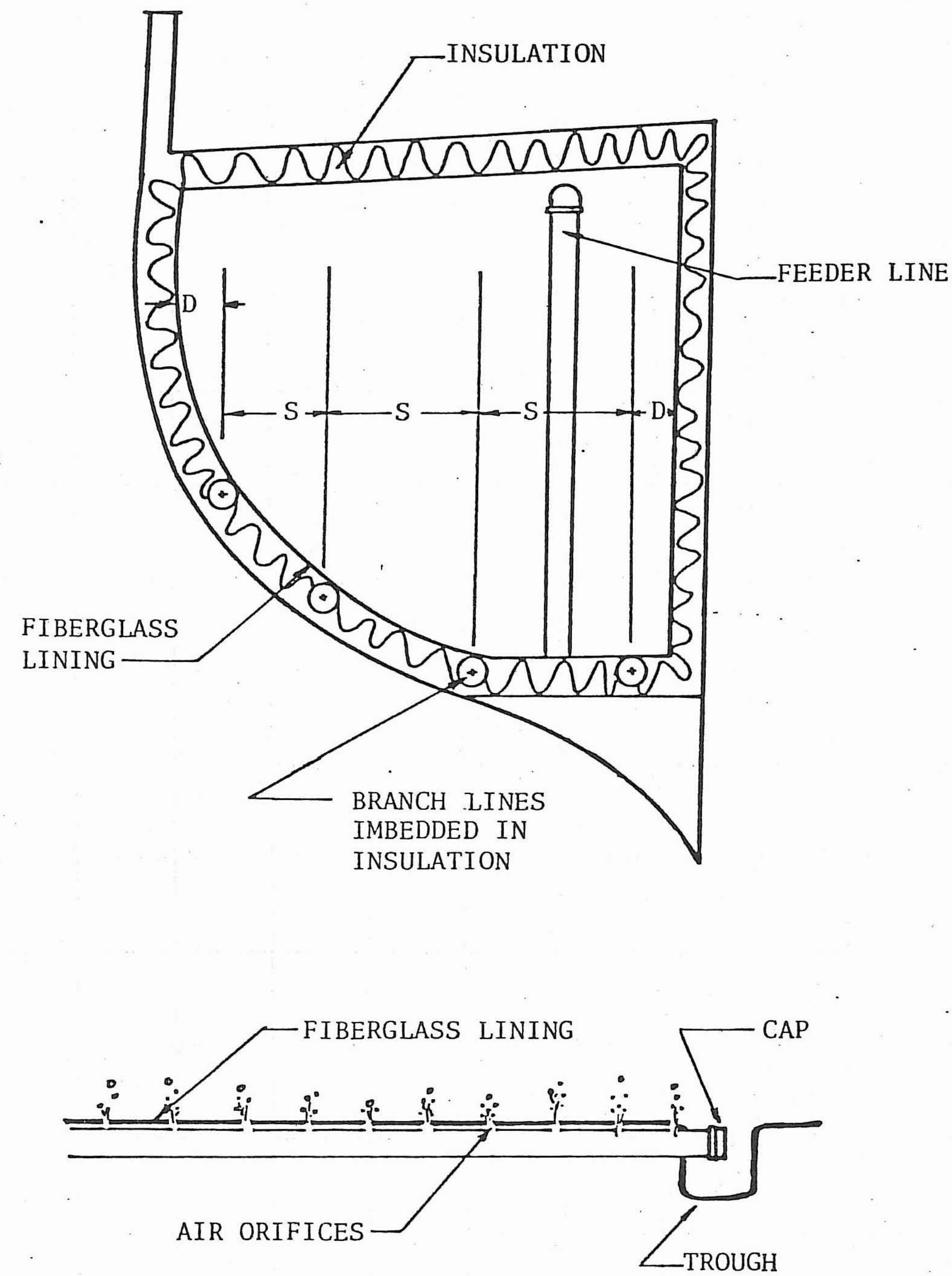

FIGURE 3. Schematic showing branch lines imbedded in insulation of fish holding tank.

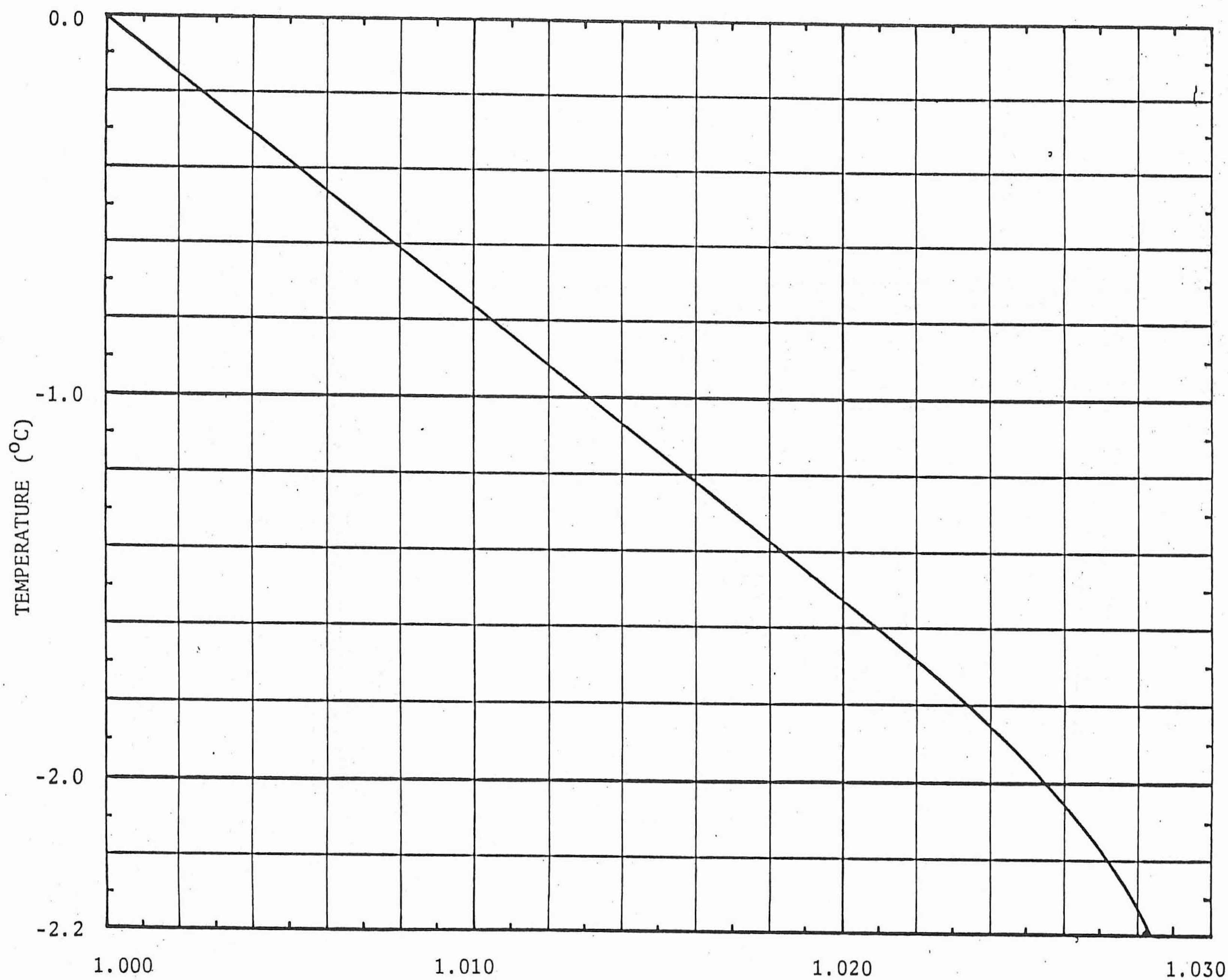

FIGURE 4. Freezing point vs. specific gravity for brine at 15.6°C.

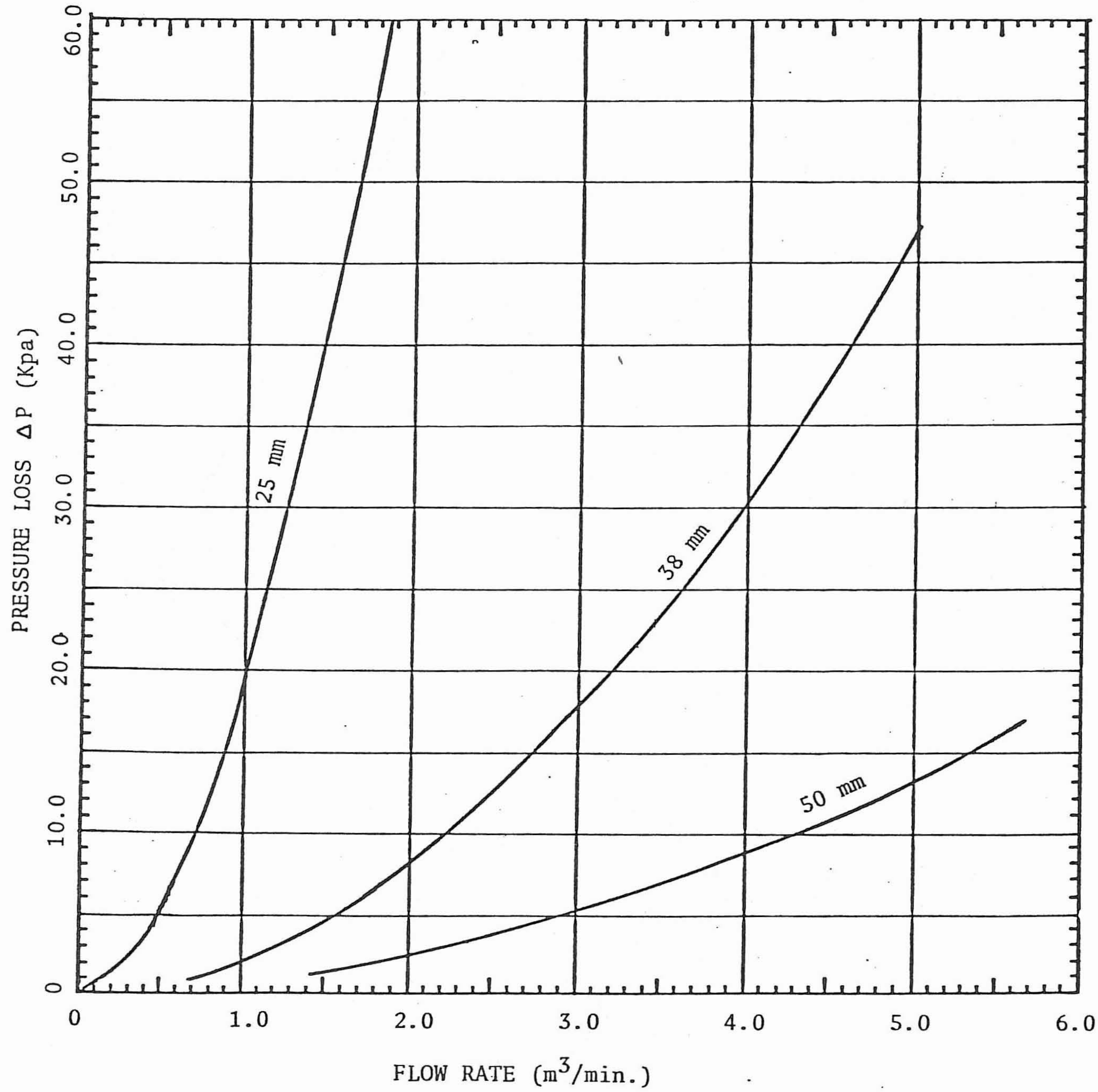

FIGURE A1. Pressure loss across 30.5 m of pipe with air flow at atmospheric pressure and 71.1°C.

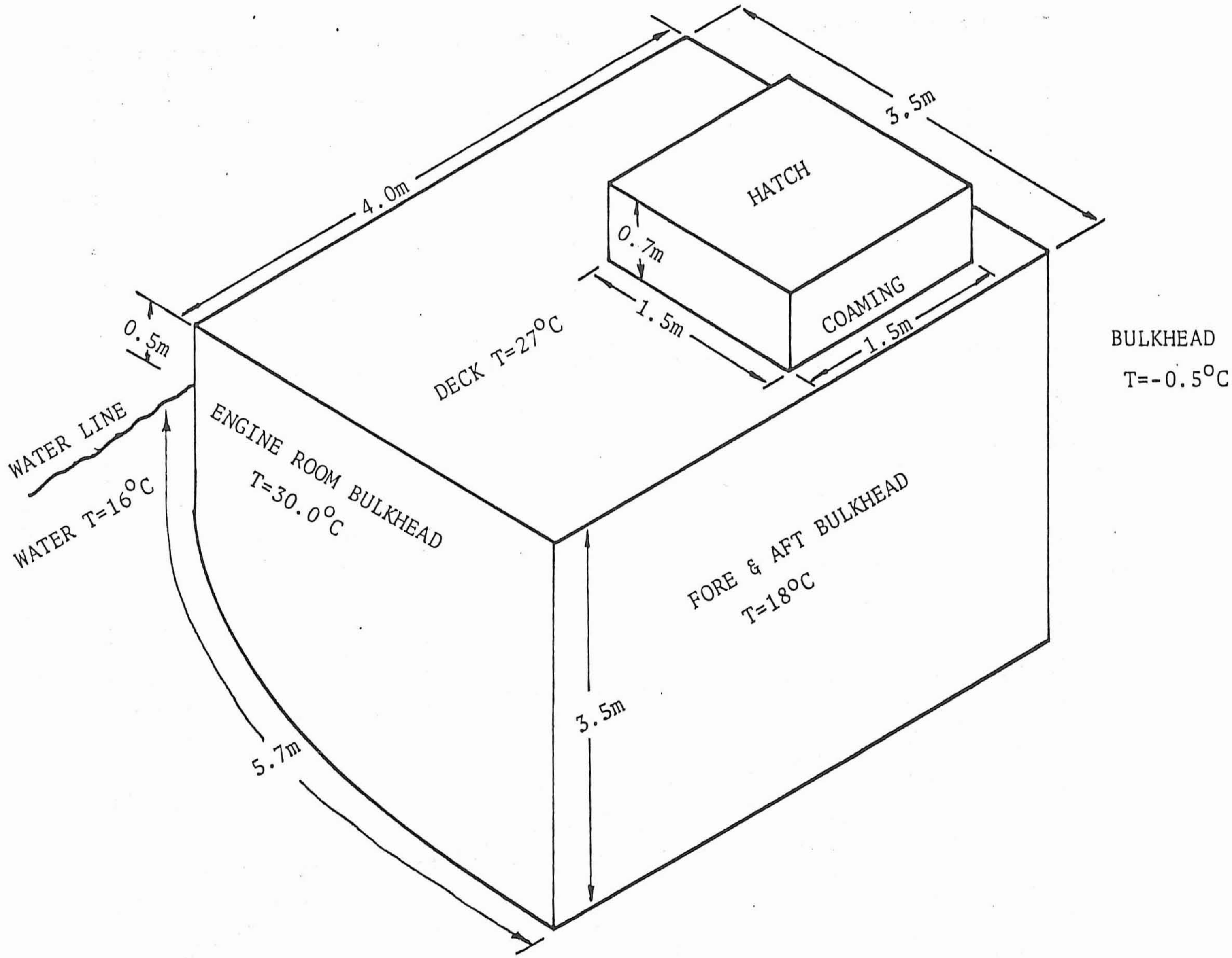

FIGURE A2. Typical temperature conditions on outer surfaces of a fish holding tank.

APPENDIX 1. CALCULATION OF PRESSURE LOSS IN CSW SYSTEMS

Air is a difficult fluid to measure the volume of because it is compressible and temperature sensitive. Complex mathematical expressions have been developed to calculate pressure losses in compressible flow but generally the range of losses in CSW systems is small enough (usually less than about 40% of discharge pressure) so that the well known Darcy equation for incompressible flow

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

where ΔP is the pressure loss, f is a friction factor of the pipe with length L and diameter D, v is the velocity of air flow and $g=9.8 m/sec^2$ is the gravitational constant, can be used (Crane Canada Ltd., 1968). The velocity is usually obtained by dividing the flow rate Q by the cross-sectional area A of the pipe under consideration, ie.

$$V = \frac{Q}{A}$$

The procedure to estimate ΔP in a CSW system is to assume first a realistic value for pressure drop in the entire piping system to which the hydrostatic head of the water in the holding tank is added to give an estimated discharge pressure at the air blower. The sum of this discharge pressure and the pressure at the end of the piping system, the branch pipe, is then divided by 2 to give an average pressure in the piping system to be used to correct the pressure losses obtained by assuming incompressible flow or flow at standard atmospheric pressure.

In order to use the Darcy equation in which only the length of the pipe appears, losses in fittings have to be expressed in terms of equivalent lengths. For commonly used fittings in CSW systems, the values are given in Table A1.

Table A1. Equivalent lengths of pipe fittings in meters

Type of Fitting	Pipe Diameter		
	25 mm	38 mm	50 mm
90° standard elbow	0.75	1.14	1.50
90° street elbow	1.25	1.90	2.50
45° standard elbow	0.40	0.61	0.80
45° street elbow	0.65	0.99	1.30
tee - straight through	0.50	0.76	1.00
tee - branch	1.50	2.28	3.00

For easy reference, the pressure losses for air at atmospheric pressure and 71.1°C in 30.5 m of schedule #40 steel pipe of 25, 38 and 50 mm diameters are given in Fig. A1. For the purpose of CSW pressure loss estimation, these values can be used for aluminum pipes as well.

To illustrate the procedure, an example for a typical installation will be given here. It is assumed in this example the size of the holding tank is 3.05 m x 3.66 m and two such tanks are to be agitated at the same time. The air flow requirement Q is then

$$Q = 2 \text{ tanks} \times \frac{0.1524 \text{ m}^3}{\text{min m}^2} \times \frac{(3.05 \text{ m} \times 3.66 \text{ m})}{\text{tank}}$$

$$= 3.40 \frac{\text{m}^3}{\text{min}} \text{ at } 21.1°\text{C and atmospheric pressure of 101 kpa}$$

A schematic of the piping system is shown in Fig. 1 and the suction side will be considered first. The suction line is usually 50 mm schedule #40 pipe (I.D. = 52.5 mm) whose cross-sectional area A is

$$A = \frac{\pi}{4}(0.0525\ M)^2 = 0.0022\ m^2$$

$\therefore$ Average air velocity V_s in the suction line is

$$Vs = \frac{3.40\ m^3/min}{0.0022\ m^2} = 1545\ \frac{m}{min}\ \text{or}\ 25.8\ \frac{m}{sec}$$

In some installations, a suction filter is mounted at the open end of the line. In that case, the pressure loss across this filter should be obtained from the manufacturer. In this calculation, no such filter will be considered as in most installations, the intakes are located in the ventilation of the engine room where such a device is not necessary. As air enters the entrance elbow, eddies are formed and therefore there is an entrance loss associated. Entrance losses ΔP_e are usually expressed by

$$\Delta P_e = K\,\frac{Vs^2}{2g}$$

where K is a constant whose value depends on the entrance geometric shape. For the abrupt entrance shape as in an elbow, K = 1.0

$$\therefore\ \Delta P_e = 1.0 \times \frac{(25.8\ m/sec)^2}{2 \times 9.8\ m/sec^2} = 34.0\ \text{m of air}$$

Since 1 m of air = 1.18×10^{-2} kpa

$$\Delta P_e = 34.0\ m \times 1.18 \times 10^{-2}\ \frac{kpa}{m} = 0.40\ kpa$$

Next, the pressure loss along the 50 mm suction line, ΔP_s, will be estimated. Suppose the total length of the line is 5.0 m, and with the equivalent lengths of the four 90° elbows, the total equivalent length L_{es} is then, with reference to Table A1

$$L_{es} = 5.0 \text{ m} + 4 \text{ elbows} \times \frac{1.5 \text{ m}}{\text{elbow}} = 11.0 \text{ m}$$

From Fig. A1, for air flow of $3.4 m^3/min$ at 71.1°C P per 30.5 m of 50 mm schedule #40 pipe is 6.5 kpa. The loss has to be corrected to 21.1°C by multiplying by the factor

$$F_t = \frac{273 + T}{344}$$

where T is the temperature under consideration in °C and in this case

$$F_t = \frac{273 + 21.1}{344} = 0.85$$

$$\therefore \Delta P_s = 6.5 \text{ kpa} \times 0.85 \times \frac{11.0 \text{ m}}{30.5 \text{ m}} = 1.99 \text{ kpa}$$

Inside the air blower, air is compressed, raising the pressure and temperature at the discharge. Before any calculation is done, a value for pressure drop in the entire system will be assumed as mentioned earlier. For most installations on the B.C. coast, any value in the range of 15-30 kpa is realistic.

At the branch line, a typical hydrostatic pressure is about 27.6 kpa, the equivalent of about 3 m of water. Assuming that the pressure loss along the entire distribution system is also 27.6 kpa, the discharge pressure at the airblow is then

$$(27.6 + 27.6) \text{ kpa} = 55.2 \text{ kpa}$$

The average pressure P_{ave} in the piping system from the air blower discharge to the orifices in the branch line is

$$P_{ave} = \frac{27.6 \text{ kpa} + 55.2 \text{ kpa}}{2} = 41.4 \text{ kpa}$$

In Fig. A1, pressure drop in a 30.5 m length of pipe of various diameters with air flowing at atmospheric pressure can be found. To convert the pressure loss to any other pressure P, correction factor F_p has to be used as in the case of temperature, and

$$F_p = \frac{101}{101 + P}$$

For example, it is required to calculate the loss in a 30.5 m length of 25 mm pipe with an air flow rate of 1.0 m^3/min at 71.1°C and 20 kpa. From Fig. A1, the pressure loss is 19.4 kpa at atmospheric pressure. Therefore at 20 kpa

$$\Delta P = 19.4 \text{ kpa} \times \frac{101}{101 + 20}$$

$$= 16.2 \text{ kpa}$$

Now the pressure loss along the 38 mm discharge line will be estimated. Assuming in the general case a total pipe length of 5 m and six 90° elbows (two more than what is shown in Fig. 1), the total equivalent length L_{ed} is then

$$L_{ed} = 5 \text{ m} + 6 \text{ elbows} \times 1.14 \frac{\text{m}}{\text{elbow}}$$

$$= 11.8 \text{ m}$$

From Fig. A1 ΔP for 30.5 m of 38 mm pipe with a flow rate of 3.4 m^3/min is 22.5 kpa

$$\therefore \Delta P_{dis} = 22.5 \text{ kpa} \times \frac{11.8 \text{ m}}{30.5 \text{ m}} = 8.70 \text{ kpa}$$

At the end of the 38 mm pipe, there is an air distribution manifold by which air is directed to the various holding tanks. Several methods are available for constructing such devices but the one shown in Fig. 1 using tees is the most common. It is obvious that the pressure loss in the manifold is essentially that of the tee and it can be calculated together with that of the feeder line. Before this is carried out, some comments will be made on the shut off valve which is located immediately downstream of the tee.

Of all types of valves available, ball valves are probably the most convenient one to use as between the fully open and fully closed positions, only a 90° turn is required. When ball valves are used, the type with the port size the same as the internal diameter of the pipe should be selected over the type with smaller openings which presents unnecessary restrictions to the flow. When ball valves with the appropriate port size are used, practically no losses are caused in the fully open position and therefore it will be neglected in this estimation.

The size of the feeder line is usually 25 mm in diameter and 3 m long with one 90° elbow. Together with the 1.5 m for the branch tee, the equivalent length of this line L_{ef} is

$$L_{ef} = 1.5 \text{ m} + 3.0 \text{ m} + 1 \text{ elbow} \times \frac{0.75 \text{ m}}{\text{elbow}}$$

$$= 5.25 \text{ m}$$

From Fig. A1, ΔP for 30.5 m of 25 mm pipe at a flow rate of 1.7 m^3/sec is 52.5 kpa. Therefore the pressure drop for 5.25 m is

$$\Delta P_f = 52.5 \text{ kpa} \times \frac{5.25 \text{ m}}{30.5 \text{ m}} = 9.04 \text{ kpa}$$

As can be noted from above, air flow rate in the feeder line is only half of the total.

Next, the pressure loss in the header and branch lines will be estimated. As can be seen from Fig. 2B, the hydrostatic head at line #1 should be lower than those in #2, 3 or 4. Therefore there is a corresponding higher flow rate in line #1. For the sake of simplicity, it is assumed that air flow divides at junction J in a 75%-25% manner according to the number of branch lines fed. Therefore the flow rates in the two arms of the header are 1.28 m^3/min and 0.42 m^3/min. The flow rate through the arm feeding three branch lines is not the same throughout the length as air is fed off to the branch lines. To account for this, either an average flow or length can be used. In this calculation, an average length of 1.0 m, equal to one half of the length of the branch will be used.

The equivalent length of the tee at J is 1.50 m and therefore the header total equivalent length L_{eh} is

$$L_{eh} = 1.50 \text{ m} + 1.0 \text{ m} = 2.5 \text{ m}$$

From Fig. A1, Δp for a flow rate of 1.28 m^3/min in 30.5 m of 25 mm pipe is 30.8 kpa.

$$\therefore \Delta P_h = 30.8 \text{ kpa} \times \frac{2.5 \text{ m}}{30.5 \text{ m}} = 2.52 \text{ kpa}$$

The pressure loss through the other branch feeding only one line will not be used as it will be lower. However since physically the pressure at junction J must be equal, in reality more air will flow through line #4 to reach this equilibrium. Therefore the actual pressure loss should be lower than 2.52 kpa. However in this estimate, this will be regarded as the answer.

In the branch lines, air is discharged through air orifices evenly distributed along the pipe, causing flow rate to decrease along the pipe. Again, as in the previous case, average pipe length will be used to account for this effect. Since the flows through the branch lines are assumed to be the same, the pressure losses will also be the same in all four lines.

The equivalent length of a 25 mm tee is 1.5 m, and assuming one half of the length of the branch line is 1.5 m, the total equivalent length L_{eb} is

$$L_{eb} = 1.5 \text{ m} + 1.5 \text{ m} = 3.0 \text{ m}$$

From Fig. A1, ΔP for 0.425 m^3/min in 25 mm pipe is 3.8 kpa

$$\therefore \Delta P_b = 3.8 \text{ kpa} \times \frac{3 \text{ m}}{30.5 \text{ m}} = 0.37 \text{ kpa}$$

As air is forced out through the orifice, eddies are formed because of the sharp corners and therefore there should be a loss ΔP_o associated. A procedure to estimate this loss can be found (Perry, 1963) but, for the sake of simplicity, it suffices to assume

$$\Delta P_o = 0.5 \text{ kpa}$$

The total pressure loss ΔP_{total} is then the sum of all the components that have been calculated.

$$\therefore \Delta P_{total} = \Delta P_e + \Delta P_s + \Delta P_{dis} + \Delta P_f + \Delta P_h + \Delta P_b + \Delta P_o$$

However, ΔP_{dis}, ΔP_f, ΔP_h and ΔP_b are subjected to compression effects and therefore have to be corrected.

$$\therefore \Delta P_{total} = \Delta P_e + \Delta P_s + \Delta P_o + (\Delta P_{dis} + \Delta P_f + \Delta P_h + \Delta P_b) \times F_p$$

$$= 0.40 + 1.99 + 0.5 + (8.70 + 9.04 + 2.52 + 0.37) \times \frac{101}{101 + 41.4} \text{ kpa}$$

$$= 2.89 + 20.6 \times 0.71 \text{ kpa}$$

$$= 17.5 \text{ kpa}$$

Suppose the assumed pressure drop was 17.5 kpa instead of 27.6 kpa, then

$$P_{ave} = \frac{17.5 + (17.5 + 27.6)}{2} \text{ kpa}$$

$$= 31.3 \text{ kpa}$$

and

$$\Delta P_{total} = 2.89 + 20.6 \times \frac{101}{101 + 31.3} \text{ kpa}$$

$$= 18.6 \text{ kpa}$$

It can be seen that there is only a 6% difference in the final answers and therefore either answer is acceptable.

Finally, the pressure loss in the piping system is

$$\frac{17.5 \text{ kpa}}{(17.5 + 27.6) \text{ kpa}} \times 100\% = 38.8\%$$

of the discharge pressure, within the range in which the Darcy equation is still valid.

APPENDIX 2. HEAT LOAD CALCULATIONS

To estimate the amount of ice required for the proper operation of CSW systems under different conditions, a heat load calculation will be carried out in the following example with temperatures as specified in Fig. A2. Similar calculations can be made for other conditions to determine the amount of ice required.

The size and shape of the fish holding tank are shown in Fig. A2. It is assumed that the hatch cover is as well insulated as the tank and there is no significant heat leakage through the opening. Therefore the deck can be assumed as continuous. Disregarding the variation between night and day, temperature on deck is taken to be 27°C and sea water temperature 16°C. It is further assumed the part of the hull under the waterline is at sea water temperature while the part not submerged is at deck temperature. The temperature of the engine room bulkhead is usually high at 30°C while at the opposite end, the bulkhead separating the fish holding tanks is at -0.5°C. The fore and aft bulkhead is at 18°C. The holding tank is insulated with 10 cm of 30 kg/m^3 polyurethane foam.

The thermal conductivity K for this type of foam is 1.36 to 3.1 kcal cm/hr m^2 °C (Reichhold Chemicals Inc.) and in the following calculation 3.1 kcal cm/hr m^2 °C will be used. Insulating contributions from other components of the tank such as fibreglass lining or plywood lining are excluded in this calculation.

Heat infiltration Q is estimated by using the steady state heat conduction equation in one dimension.

$$Q = - K A \frac{\Delta T}{\Delta x}$$

where A is the area, ΔT is the temperature difference and Δx the thickness of insulation. Accordingly, heat load Q_1 from the deck area and the non-submerged part of the hull is

$$Q_1 = 3.1 \frac{\text{kcal cm}}{\text{hr m}^2\ ^\circ\text{C}} \times [4.0\text{ m} \times 4.0\text{ m} + 6\text{ m} \times 0.7\text{ m}] \times [27^\circ\text{C} - (-0.5^\circ\text{C})] \times \frac{1}{10\text{ cm}}$$

$$= 172 \frac{\text{kcal}}{\text{hr}}$$

Similarly, heat load from the other areas are:

Heat from submerged part of hull = Q_2 = 95.4 kcal/hr
Heat from engine room bulkhead = Q_3 = 89.8 kcal/hr
Heat from fore and aft bulkhead = Q_4 = 80.3 kcal/hr

Heat from bulkhead separating the holding tanks is zero since there is no temperature difference.

$$\begin{aligned} \text{Total heat infiltration load } Q_I &= Q_1 + Q_2 + Q_3 + Q_4 \\ &= (172 + 95.4 + 89.8 + 80.3)\text{ kcal/hr} \\ &= 438\text{ kcal/hr} \\ &= 10500\text{ kcal/day} \end{aligned}$$

Since the fusion of one kilogram of ice will absorb 79.7 kcal this heat will require about 132 kg of ice per day assuming ice temperature = -0.5°C and therefore for a seven day trip, the ice required W_I = 921 kg.

Next it is necessary to estimate the amount of ice required for chilling of fish. The specific heat of fresh fish is approximately 0.9 kcal/kg °C. Assuming the tank can hold 30 tonnes of fish at an initial temperature of 13°C, the heat to be removed from fish Q_F to lower the temperature to -0.5°C is

$$Q_F = 0.9 \frac{\text{kcal}}{\text{kg }^\circ\text{C}} \times (30\text{ tonnes} \times \frac{1000\text{ kg}}{\text{tonne}}) \times [13^\circ\text{C} - (-0.5^\circ\text{C})]$$

$$= 364{,}500\text{ kcal}$$

The amount of ice required W_F = 4573 kg.

Finally, the heat load of the sea water added to the mixture will be estimated. Assuming no spillage of chilled water and typically 25% of the total load is sea water and ice, the total weight of ice and sea water W_{ISW} is

$$W_{ISW} = \frac{30 \text{ tonnes}}{75\%} \times 25\% = 10 \text{ tonnes} = 10{,}000 \text{ kg}$$

It has been obtained earlier that the ice required to chill the fish and to absorb the heat infiltrated is (4573 + 921) kg = 5,494 kg. Therefore the weight of the sea water and the ice required to chill this sea water to -0.5°C is

$$(10{,}000 - 5{,}494) \text{ kg} = 4506 \text{ kg}$$

Let W = weight of sea water and I = weight of ice. A pair of simultaneous equations can be set up to find W and I. Assuming the specific heat of sea water = 1 kcal/kg °C,

$$I + W = 4506.$$

Heat to be removed from the sea water = Heat of fusion of ice

$$\therefore 1.0 \frac{\text{kcal}}{\text{kg °C}} \times W \times [13\text{°C} - (-0.5\text{°C})] = I \times 79.7 \frac{\text{kcal}}{\text{kg}}$$

Solving for I and W, it can be shown that I = 653 kg and W = 3853 kg

$$\begin{aligned} \therefore \text{ Total amount of ice required } &= W_I + W_F + I \\ &= (921 + 4573 + 653) \text{ kg} \\ &= 6{,}147 \text{ kg} \end{aligned}$$

In the above calculation, no allowance has been made for extra ice requirement arising from various situations. For example, it is necessary to have some excess ice on board the vessel as the trip length may be extended by bad weather or other unforeseen circumstances. In actual operations, spillage of chilled sea water may be necessary due to excess water and slime from fish or operational requirements. Finally, heat from hot air bubbling through the mixture has not been accounted for. Therefore a good practice is to carry 10 to 15% more ice, raising the total amount of ice required to

$$6{,}147 \text{ kg} \times 1.15 = 7{,}069 \text{ kg} \simeq 7.0 \text{ tonnes}$$

This compares well with the 1:4 ice to fish ratio which will require 7.5 tonnes of ice.

It is interesting to note the relative amount of ice used for the various functions - 15.0% to absorb heat through insulation, 10.6% to chill the sea water needed for the operation of CSW systems and 74.4% to chill the load.