Refrigeration Energy Prediction for Flooded Tanks on Fishing Vessels

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Abstract

Calculations combined with documented experience predict and compare direct energy consumption for the two onboard flooded-tank refrigeration systems in current use. These are RSW (refrigerated seawater) which uses mechanical refrigeration, and CSW (chilled seawater) which uses ice brought from shore. RSW is found to consume energy at roughly half the rate predicted for CSW; seawater temperature has a major influence on results.

Introduction

The passage of the Fishery Conservation and Management Act in 1976 gave the United States control of fisheries in a 200-mile zone adjacent to its territory. This has resulted in a rapid and dramatic development of the U.S. seafood industry. Five years ago in the Alaskan zone for example, fish were taken largely by foreign factory trawlers and/or U.S.-Foreign Joint Venture operations. In 1989, 85-90% of the fish taken in the Alaska zone would be caught and processed by domestic vessels and plants (personal communication, C. Pautzke, North Pacific Fisheries Management Council, 1989). Large volumes of landed cod, pollock, and flatfish are rapidly processed into frozen blocks or fillets, or into surimi – a minced washed fish protein from which many seafood analogs are subsequently manufactured.

Large vessels towing mid-water trawls can catch on the order of 40-50 metric tons per tow (Roach, 1981; Hilderbrand, 1986). This must be rapidly refrigerated prior to processing. Stowing the fish in large flooded tanks of water near 0° C is the most feasible option. There are two options for cooling these flooded tanks. The first, Refrigerated Seawater (RSW), uses mechanical refrigeration to maintain water temperatures at levels slightly less than 0° C (Gibbard and Roach, 1976). The second, Chilled Seawater (CSW), is a system which uses ice brought from shore to lower the water and fish temperature to about 0° C (Kolbe et al., 1985). Both are commonly used aboard large vessels (25-45 m) which might land high volume species such as Alaskan pollock and Pacific whiting.

Many factors affect decisions on design and application of onboard refrigeration systems. One of these is energy consumption, the concern for which has fluctuated, but which will undoubtedly assume a growing priority in the near future. This article addresses the direct energy costs of onboard refrigeration for both RSW and CSW systems.

Approach

The predictions are based on scenarios describing the volume of fish landed at a representative seafood processing facility. The production scenario assumes a plant which processes 150 metric tons of fish per day into surimi at an overall yield of 20% (AFDF, 1987; Holmes and Riley, 1987; Surimi Inc., undated; Hilderbrand, 1986; Talley, 1986; Sonu, 1986; Anon., 1988). Calculation of direct energy cost is based on the weight of final product.

The analysis of these flooded tank systems assumes that fully-flooded tanks are necessary for open-ocean vessel stability. This means that previously chilled water must be dumped as fish are stowed; thus, some energy must be wasted in the interest of operational safety.*

Landing an average of 150 metric tons of fish per day requires a fleet of typical West coast vessels fishing for 2-3 days at a time and making deliveries at staggered intervals. Table 1 shows one fleet scenario; additional assumptions include:

1. The fish hold or tank volume is related to vessel length by an equation developed from Merritt et al., (1983):

$$V = 0.242 - 1.854\,(L) + 0.252\,(L)^2$$

   where
   V = volume (m^3),
   L = length (m).
   (Note: In some cases, the actual part of the hold which is used could be less than the full volume available. Also, the hold volume in a particular vessel may be different than indicated by the formula.)
2. Density of fish in the hold when fully loaded will be 673 kg/m^3 (42 lbm/ft^3; Kolbe et al., 1985).
3. Some "percent loading capacity" must be chosen to account for the fact that the fisherman may fail (or decline) to fill the boat.
4. Some delivery schedule matched to the cruising/fishing time is specified for each vessel. Labeled "days fishing", this quantity will identify onboard refrigeration time and will assume short turn-around at the dock.

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* In protected waters, some larger vessels with several tanks may have the option of "saving" this cold water by dumping it into an empty tank and operating safely with one tank: having a free-surface. This is generally not recommended, however (personal communication, B. Wittemore, MARCO, 1988).

TABLE 1. Fish delivery scenario

Vessel Length (m)	Tank Size (m^3)	Days Fishing	Loading Capacity (%)	Landings/Trip (t)†	(t/da)
20	63	2	80	34	17
26	122	3	80	66	22
26	122	3	80	66	22
38	255*	3	80	137	45.5
38	255*	3	80	137	45.5
			Total	152 t/da	

* Although this value is slightly less than that given by the formula, it corresponds to the hold volume reported for a representative MARCO vessel recently landing Pollock for surimi in Dutch Harbor, AK (Roach, 1981; Miller, 1988).
† The symbol "t" represents metric tons.

CHILLED SEAWATER (CSW)

The direct energy cost of operating a CSW system will relate both to the quantity of ice required for the vessels specified in Table 1 and to the shore-based ice manufacturing cost.

Required ice quantities were calculated by a computer program ("SLUSH") described by Kolbe et al. (1985) assuming the following:

1. Ice will be loaded in anticipation of the vessel's returning with a full load of fish (loading capacity = 100%).
2. Operation will be under the "Fill/Spill" mode which means that the hold remains full; cold water is dumped as fish go below.
3. The fish-hold is well insulated. Heat leakage is calculated using conductance values given by Kolbe et al. (1985) and by Wang and Kolbe (1989).
4. Air bubbling to effect water circulation in the hold proceeds for 6 hr/day.

Two options are available to determine the required amount of ice for the boats listed in Table 1. One is to run the program ("SLUSH") for each vessel and set of conditions. Figure 1 is a sample of that program's output. The second option is to read this data from figure 2 which is a graphical representation of program results using a range of input conditions.

Table 2 gives the resulting CSW ice requirements for the fleet defined in Table 1 and for two different values of seawater temperature. The seawater temperatures chosen (10 and 15.5° C) are representative of summer surface temperatures off the coasts of Alaska and Oregon, respectively. The specific requirement was calculated by dividing the total ice required (t/da) by the total fish landed (152 t/da for the scenario of Table 1)†.

The energy cost of manufacturing ice in a shore plant depends to some extent on the type of ice (FAO, 1975). Table 3 lists documented energy costs of ice production.

Energy costs of flake ice, the type most commonly used in West Coast fisheries, shows a range of values from 44 to 80 kWh/t, a range that appears to depend somewhat upon the size of the machine. If we choose 60 kWh/t as a representative value, we can then determine energy cost per metric ton of final product by the following calculation.

† The symbol "t " represents metric tons.

TABLE 2. Ice requirements for 152 t/da

Vessel Length (m)	Tank Size (m^3)	Days Fishing	Max. Capacity Landing (tonnes)	Ice Required 10° C (t)	(t/da)	Ice Required 15.5° C (t)	(t/da)
20	63	2	42	11.8	5.9	16.7	8.4
26	122	3	82	23.2	7.7	32.7	10.9
26	122	3	82	23.2	7.7	32.7	10.9
38	255	3	172	48.3	16.1	68.4	22.8
38	255	3	172	48.3	16.1	68.4	22.8
		TOTAL	(t/da)		53.5		75.8
			(t/t fish)		.352		.499

For example, for 10° C seawater:

$$\text{Specific Energy} = \frac{(0.353\ t_{ice}/t_{fish})\ (60\ \text{kWh}/t_{ice})}{(0.2\ t_{product}/t_{fish})}$$

$$= 106\ \text{kWh}/t_{product}$$

REFRIGERATED SEAWATER (RSW)

The energy consumed by chilling fish in a mechanically refrigerated seawater system is considered to be the sum of four parts—i.e., four heat loads on the system:

1. Chilling the seawater from its original temperature to -0.6° C;
2. Chilling the fish from its original temperature to -0.6° C;
3. Absorbing heat which leaks through the fish hold boundaries;
4. Removing heat generated by the circulating pumps.

TABLE 3. Energy costs of manufacturing ice in temperate areas (in kWh/t)

Source	Flake	Tube	Block
FAO (1975)	50-60*	40-50*	40-50*
Dassow (1965)			55-58
A/S Atlas (Anon., 1975, '79)	44-75†		40-60
Matal (Anon., 1975)	70-80		
Galt (Anon., 1979)		64-75‡	
North Star	58**-63#		
Stal (Anon., 1979)		40	
Stal-Levin (Anon., 1975)		45-67	
Grenco (Anon., 1975)	48-53§		
Enriquez et al.. (1986)		96-105‖	

Notes:
* FAO (1975) gives the following values for "tropical areas":
flake: 70-85 kWh/t
Tube: 55-70 KWh/t
Block: 55-70 kWh/t
Figures include energy for ice making only; additional energy is needed for conveyors and storage.
Peak power demand: 1.5-3.8 kW per (tonne/da).
† Also called "slice" ice.
‡ Called "shell ice" by manufacturer.
§ Required peak power of 2-2.2 kW/t-day.
‖ These are actual plant figures which include kWh-equivalence of fuel oil to heat water for icemaker defrost cycle, plus other electric-power consumers in the ice plant.
\# From (Anon., 1979).
** Based on the "rule of thumb" that 1.3 refrigeration tons (TR) are required for each short ton of ice produced in 24 hours (personal communication and literature, North Star and Howe ice machine manufacturing companies). Kolbe (1988) describes further calculations accompanying this assumption.

The following summarizes the calculation procedure:

Chilling seawater from its original temperature to -0.6° C. Calculation procedures were based on those of Roach (undated), Gibbard and Roach (1976), and Kolbe (1979) and assumed the following:

- Each vessel's hold is divided into several 28.3 m^3 (1000 ft^3) tank units, plus a smaller tank to make up the total hold volume.
- The typical refrigeration unit consists of a Carrier Model 5H40 compressor using R-22 refrigerant, operating at 1750 RPM and 38° C (100° F) saturated discharge temperature, and paired with a set of seven 3.05-m (10-ft) barrel (shell and tube-type) chillers (Gibbard and Roach, 1976).
- Temperature differences between RSW and the refrigerant vary from 8.3° to 5.6° C (15° - 10° F) as recommended by Roach (undated) and by Gibbard and Roach (1976).
- A percentage loading capacity (this example will use 80%) is assumed common to all vessels.
- Ninety percent of the system's refrigeration capacity removes heat from the tank; the rest removes heat generated by pumping.
- The electric motor driving the compressor has an efficiency of 85%.

For a series of small finite increments of decreasing RSW temperature, calculations then balanced the rate of heat extracted with the rate absorbed by the refrigeration machinery. A summation was then made of energy required to power the refrigeration as RSW temperature fell from its initial (sea surface) temperature to -0.6° C (31° F).

Chilling fish from its original temperature to -0.6° C. The refrigeration system must chill fish from their original (sea surface) temperature within a given time period. Hilderbrand (1986) described a "typical" tow in the joint venture fishery for Alaskan pollock to be about two hours. If we add an hour for shooting and hauling the trawl, it means that a warm load of fish might be stowed every three hours. The fish mass that could be chilled in three hours, based on calculation procedures described by Gibbard and Roach (1976) was projected to be 25.5 metric tons. Energy cost was calculated based on the power rating of the system and the three-hour chilling period, assuming a refrigeration suction temperature of -6.7° C (20° F). The size of the tow, or mass of fish landed, can reach 35-45 metric tons (Hilderbrand, 1986; Roach, 1981). Thus, in some cases, additional refrigeration capacity must be installed to keep up with the rate of warm fish stowed in multiple tanks. Although this represents a much higher capital expense for the vessel owner, the energy cost per unit of fish chilled would be unchanged.

Absorbing heat leaking through the fish hold boundaries. Heat leakage calculations followed the same procedures reported above for CSW refrigeration and outlined by Kolbe et al. (1985). The total heat load due to leakage through external boundaries determined the required refrigeration system operating time, again assuming a system capacity at -6.7° C SST saturated suction temperature) and well-insulated fish hold boundaries.

Removing heat generated by the circulating pumps. Pumps circulating water through an RSW system will generate heat. which must be removed by the refrigeration. Previous sections (1-3) have already accounted for this by assuming a 10% reduction in effective refrigeration capacity.

An additional amount of energy is required to drive the circulating pumps during those periods in which the refrigeration machinery is operating. This energy was calculated separately.

Roach et al. (1967) recommended pump sizes for typical flooded tank RSW systems. Based on representative system loads, calculations showed that each refrigeration unit (as defined previously) required approximately 6 kW of power to operate circulating pumps.

Results and Discussion

Predicted energy required to refrigerate fish for the two flooded hold systems appears in Table 4.

Specific energy cost for the CSW system is approximately double that of the RSW system performing the same task. This results from the performance characteristics of a mechanical refrigeration system. As saturated suction temperature decreases, the power required per unit of refrigeration capacity will increase. The RSW system will operate with a refrigerant temperature (SST) of approximately -7° C. Ice machines producing flake ice will operate at an SST value closer to -32° C. With that difference in SST values, a unit of refrigeration generated by the ice machine will "cost" about twice as much power as a unit of refrigeration generated by the RSW machinery.

Energy required for both systems is highly dependent on seawater temperature. The major reason is that it represents the initial warm temperature of both water and fish that must be ultimately cooled to 0° C. A second reason is that it represents the driving temperature controlling heat leakage rate into the hold through the vessel walls. However, as shown by the "Heat Leak" quantities of Table 4, this makes an almost insignificant contribution to the total, especially when the hold is well-insulated, as assumed in these calculations. A well-insulated wall corresponds to the "MAX" value used in figures 1-3. It is defined by Kolbe et al. (1985) to represent a vessel wall section in which the foamed insulation depth covers the reinforcing stiffeners, a recommended and common industry practice (Wang and Kolbe, 1989).

Figure 3 shows how RSW energy will vary both with seawater temperature and insulation quality. Figure 2 shows similar effects for the CSW system. A change in the tank size (and thus surface-to-volume ratio) and the days stowed (affecting quantity of heat leakage) both have a minor effect on ice consumption and energy cost, compared with effects of seawater temperature.

TABLE 4. Onboard refrigeration specific energy (in $kWh/t_{product}$)

	10° C Seawater	15.5° C Seawater
RSW		
Water chill	33.7	47.0
Fish chill	14.6	22.3
Heat Leak	0.8	1.3
Pump drive	10.4	15.0
Total	59	86
CSW		
Total	106	150

```
CASE  1  FILL/SPILL

INITIAL TEMPERATURE                 50 F
TANK VOLUME                         2200 CUBIC FEET
FISH TO BE LOADED                  46.2 TONS
NUMBER OF DAYS TO BE HELD           2 DAYS
HOURS PER DAY OF AIR BUBBLING       6 HOURS
ASSUMED INSULATION LEVEL            MAX
ICE MELT FROM FISH                  5.2 TONS
ICE MELT FROM HEAT LEAK             0.2 TONS
ICE MELT FROM BUBBLED AIR           0.5 TONS
ICE MELT FROM WATER ADDED           7.1 TONS
TOTAL ICE REQUIRED                  13.0 TONS
APPROX PERCENTAGE OF TANK
  OCCUPIED BY DRY
  BULK-LOADED ICE                   40 %
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Figure 1–Sample output from computer program "SLUSH" (Kolbe et al., 1985).

Two other assumptions will have a direct effect on the absolute results in Table 4. The first is the loading capacity, assumed to be 80% (Table 1). If more or less fish were actually loaded into the flooded tanks, the energy cost to refrigerate the fish would change proportionally, but the effect on the total energy cost would be much less (Table 4). In addition, the relative results which compare systems and seawater temperature effects would be essentially unchanged.

The second assumption that will affect absolute results is the "overall plant yield" taken here as 20%. As this number varies, the results, which are based on the mass of final product, will vary proportionally.

Other major assumptions have related to the total daily plant capacity and the corresponding fleet scenario to deliver this capacity. For both the RSW and CSW systems, calculations not shown here indicated that variations in plant capacity will have a negligible effect on the per-unit energy costs.

Conclusions

1. Seawater temperature, as influenced by season and locale, has the most significant effect on energy consumed in refrigerating fish in flooded tank systems.
2. Effects of stowage time and insulation quality on refrigeration energy costs are relatively minor.
3. The energy required to operate Chilled Seawater (CSW) systems is roughly double that required for Refrigerated Seawater (RSW) systems.

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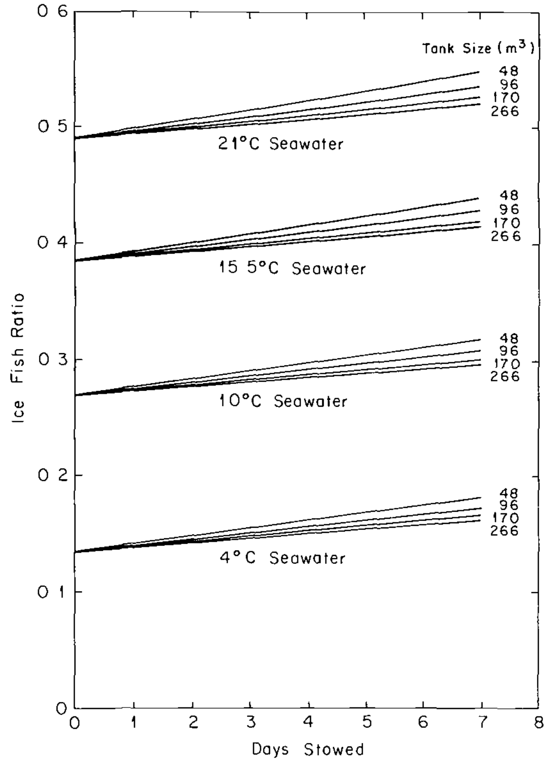

Figure 2–ICE:FISH ratio vs. time and seawater temperature in CSW system.
Ice: Fish ratios calculated from program "SLUSH" under the following assumptions:
1. **MAX Insulation.**
2. **Fill/Spill Strategy.**
3. **6 h/da Air Bubbling.**
4. **Maximum fish loading density ($673\ kg/m^3$, or $42\ lbm/ft^3$).**

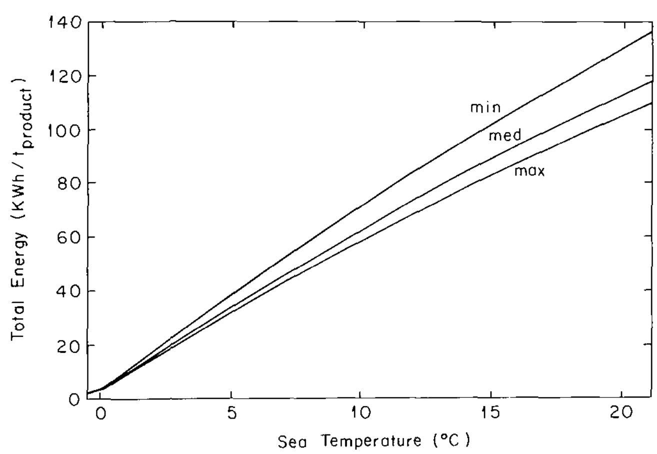

Figure 3–Total RSW energy consumed vs. seawater temperature and fish hold insulation.
Calculations assume.:
1. **MIN, MED, MAX insulation levels as defined by Kolbe et al., 1985.**
2. **Fish loading capacity of 80% ($545\ kg/m^3$)**
3. **Daily landings of 150 t/da.**
4. **Product yield of 20%.**

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