Solar-Powered Refrigeration System for Railway Refrigerator Cars

David R. Conover,* Fruit Growers Express Company, Alexandria, Virginia

The potential of an available method to reduce the fuel consumption of the mechanically refrigerated railcars used for transporting perishable commodities has been investigated. An energy-oriented engineering analysis showed that a compact refrigeration system deriving its power from photovoltaic cells could be used in the design of railcar equipment for transporting perishable produce. Maximum load-respiration data and a heat-loss analysis for mechanically refrigerated railcars in service today were used to develop the energy design requirements of an alternative system. A solar-energy system was used to eliminate the need for fossil fuels, which also eliminates the emission of air pollutants and reduces acoustical emissions. The results reflect the present state of the art for designing and supplying power to railcar refrigeration systems and the way in which these systems could be used to alleviate projected energy problems in refrigerated transport.

Perishable produce has been shipped in refrigerated railcars since 1851 when ice was first used to cool a load of butter in shipment from upstate New York to Boston. Six years later, 30 insulated cars with ice-boxes in the doorways were constructed for use on the Pennsylvania Railroad. In 1868, patents for the ice tank car, which had iron tanks filled with ice and salt, were issued, and this design led to the Hutchins wool-insulated cars, which established the usefulness and practicability of railway refrigerator cars.

In the early twentieth century, research and development efforts in the area of refrigerated transport by rail were intensified. Numerous arrangements were designed and tested, some of which proved to be of great importance. The most important advance in the pre-World War II era was the interior fans, driven by the movement of the railcar wheels, which circulated ice-cooled air throughout the load. As the shipment of perishable produce by highway increased, so too did the activities by the railroads to develop new equipment and methods for the transport of perishable produce.

Mechanical refrigeration equipment, commercially introduced to railcar design in 1949, uses a separate diesel-engine power plant in each car to power a mechanical refrigeration system. In the past 25 years, both the equipment and the general design of the railcar have been improved, and together they now provide a reliable way of cooling perishable and frozen produce during transit; however, further development is needed. With the cost of diesel fuel and machinery escalating, and the share in the transport of perishable shipments carried by the railroads decreasing, there is a great need for a new energy-conserving railcar design for shipping items requiring refrigeration.

Solar energy, which is considered our ultimate energy resource at present, is a perpetual source that can be converted directly into electrical power by photovoltaic cells. At present, research in the area of photovoltaic cells is being increased; the proposed spending for fiscal year 1977 is more than $60 million. With increased research, the cost and availability of these cells is expected to become comparable to that of other energy sources, while the technology of the product itself will increase. It is therefore pertinent to investigate the possibility of using solar cells for the bulk generation of electricity in railcars.

This paper presents an analysis of the use of these cells mounted on a railcar roof to produce electricity to power a refrigeration system. Although many limitations exist today, the theoretical analysis shows that the concept is sound and possesses the capability of reducing the dependence of the rail industry on fossil fuels.

CAR DESIGN AND USE

The railcar designed for use with the solar-panel refrigeration design would be similar in construction to the refrigerated railcars presently manufactured (Figure 1). The specific car would be a 91-Mg (100-ton) capacity refrigerator car with a 50.8-cm (20-in) travel hydraulic cushioning device incorporated in the underframe. The interior dimensions are given below (1 m = 3.3 ft):

Dimension	Value
Inside length, m	16.0
Inside width, m	2.7
Inside height, m	2.9
Interior volume, m^3	128.2

The exterior sides, ends, roof, and doors would be constructed of steel design standard to present railcar technology and in compliance with all Association of American Railroads and Federal Railroad Administration specifications. The door and side-wall lining would be reinforced fiberglass, the ceiling aluminum sheets, and the ends and floors plywood. Alternative materials, such as high-density foam, could be considered for interior construction. Wood would be used for structural members between the interior lining and the exterior steel shell. The entire railcar would be insulated with 38.4 kg/m^3 (2.4-lb/ft^3) density polyurethane foam. Railcars of this design have been built with tested heat losses of 47.5 J/s·°C (90 Btu/h·°F). The design car would be heavily insulated, using alternative materials wherever possible, and built to eliminate hot spots in the interior. The calculation of the theoretical heat loss of the railcar designed is described below.

$$Q_c/dT = kA/dx \tag{1}$$

where

- Q_c = rate of heat flow by conduction in material,
- dT = temperature difference,
- k = thermal conductivity of material [(for polyurethane foam, k = 0.025 J/s·m·°C (0.015 BTU/h°·ft°·F)],
- A = area of section through which heat flows by conduction perpendicular to the heat flow, and
- dx = distance or thickness of insulation in direction of heat flow.

The following assumptions are made:

1. Heat flow is one dimensional through the car body,
2. Conduction is in the steady state,
3. Thermal and mechanical properties of the car-

body construction are constant,

4. No heat is generated in the car-body structure,

5. The metal exterior car structure and interior lining are neglected,

6. The car structure is all foam insulated; there is an additional heat loss of 15 percent for areas around the door and interior hot spots, and

7. The corner and edge sections of the car structure are neglected.

The data used in the calculations are given below (1 m = 3.3 ft).

Term	Definition	Value
A_1, m^2	Ceiling area	43.9
A_2, m^2	Interior floor area	43.9
A_3, m^2	Area of B-end interior	8.36
A_4, m^2	Area of A-end interior	8.36
A_5, m^2	Area of interior sides and doors	97.6
dx_1, m	Thickness of roof insulation	0.203
dx_2, m	Thickness of floor insulation	0.178
dx_3, m	Thickness of B-end wall insulation	0.178
dx_4, m	Thickness of A-end wall insulation	0.305
dx_5, m	Thickness of side and door wall insulation	0.178

By modifying Equation 1, the total heat loss for the entire design car can be calculated. Thus,

$$
\begin{aligned}
Q_c/dT &= \text{heat loss in (roof + floor + A-end + B-end} \\
&\quad \text{+ side and door walls)} \\
&= kA/dx(\text{total}) \\
&= k[(A_1/dx_1) + (A_2/dx_2) + (A_3/dx_3) + (A_4/dx_4) + (A_5/dx_5)] \quad (1a)
\end{aligned}
$$

= 0.025 [(43.9 ÷ 0.203) + (43.9 ÷ 0.178) + (8.36 ÷ 0.178) + (8.36 ÷ 0.305) + (97.6 ÷ 0.178)] = 28.9 J/S·°C (53.4 Btu/h·°F). To add the 15 percent heat loss allowed in assumption 6, this becomes Q_c/dT = 32.4 J/s·°C (61.5 Btu/h·°F). Thus, for the railcar design analysis, the total heat loss by conduction through the railcar structure can be considered as 32.7 J/s·°C (62.0 Btu/h·°F).

To design the system for the test car, it was necessary to determine a theoretical maximum for the heat generation by the load. Even after harvest, all fresh commodities respire or breathe, i.e., take in oxygen and give off carbon dioxide. This phenomonon produces heat that must be removed by the refrigeration system to maintain the product at the proper temperature until it reaches its destination. All types of produce have different respiration rates and, therefore, the maximum amount of heat produced will be determined by the commodity shipped. The maximum possible respiration heat is that of sweet corn. This value, which was used in the refrigeration-system design calculations, is calculated below. A frozen load would generate a negligible amount of heat.

$$Q_{gen} = \text{Resp} \times {}^1\!/_{24} \times {}^1\!/_{3600} \times \text{Den} \times \text{Cap} \quad (2)$$

where

Q_{gen} = rate of heat produced by load due to respiration,
Resp = maximum respiration rate for all U.S. Department of Agriculture commodities (1),
$^1/_{24}$ = conversion factor days to hours,
$^1/_{3600}$ = conversion factor hours to seconds,
Den = maximum density of commodity, and
Cap = volume capacity of railcar.

The following assumptions are made:

1. The entire load is precooled before shipment,
2. Heat within the load is generated in the steady state and uniformly, and
3. Heat generated by boxes, pallets, or shipping containers is neglected.

The data used in this calculation are given below [1 (J/d)/kg = 0.86 (Btu/d) ton, 1 kg/m^3 = 0.062 lb/ft^3, and 1 m^3 = 35.3 ft^3].

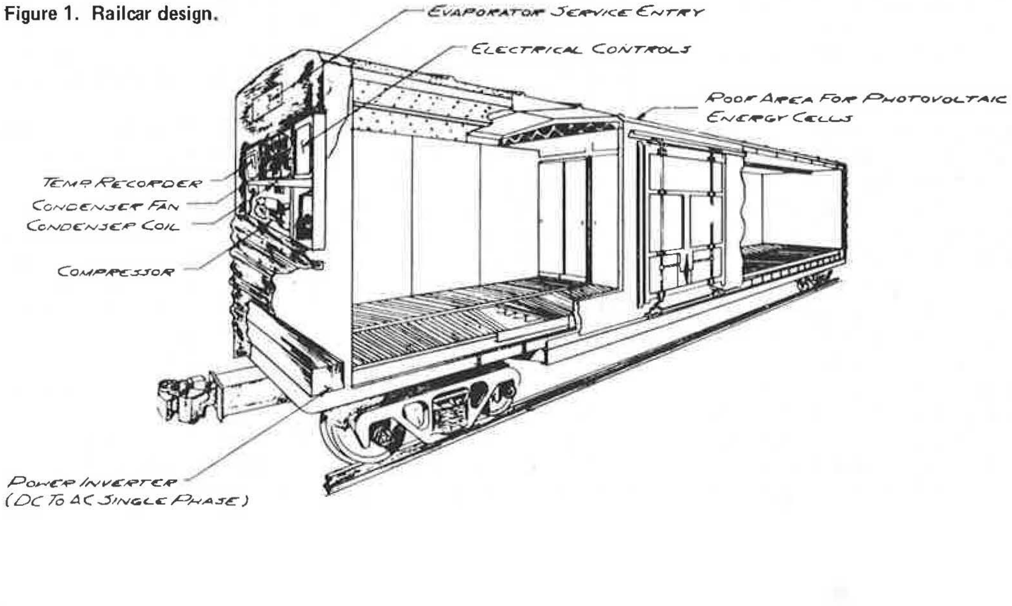
Term	Value
Resp, (J/d)/kg	8140
Den, kg/m^3	537
Cap, m^3	128

By substituting the above data into Equation 2, the maximum heat generated by the load can be calculated: Q_{gen} = 8140 × $^1/_{2000}$ × $^1/_{24}$ × $^1/_{3600}$ × 537 × 128 = 6482 J/s (22 121 Btu/h). Thus, for the calculation of maximum refrigeration load in the design railcar, the maximum heat generated by the load is 6475 J/s (22 100 Btu/h) at 0°C (32°F).

The photovoltaic-cell panels would be secured on a roof area of 55.7 m^2 (600 ft^2), arranged transversly in 1.52 by 0.61-m (5 by 2-ft) sections. The refrigeration system would be located on the A-end of the railcar.

Initially a prototype should probably be limited to a

Figure 1. Railcar design.



private operation, which would permit closer evaluation under controlled movement conditions. The train operation should be planned to achieve the maximum travel distance during the day, when the solar system could be in operation, and eliminate the need for large numbers of batteries. With a 12-h daylight period during the summer and a projected speed of 113 km/h (70 mph) average, the operating distance would be 1352 km (840 miles). The shorter winter daylight or some night operation would not be a problem because of the lower ambient temperatures, lower car heat losses, and lower refrigeration loads under those conditions. (Specific details of train operation were not developed here and would require further study.)

REFRIGERATION SYSTEM

Heat is defined as a quantity of energy transferred across boundaries because of a temperature difference between those boundaries. Because heat moves from an area of high temperature to one of low temperature, a low-temperature substance must be introduced into the rail-car interior sufficient to remove the heat inside. The specific purpose of the refrigeration or heat-pump system in a railcar is to create a low-temperature heat sink capable of removing the railcar heat.

The refrigeration system most commonly used today (Figure 2) uses a system of an evaporator, a compressor, a condenser, and an expansion valve to circulate the refrigerant fluid. The refrigeration cycle begins when gaseous refrigerant at a low temperature and a low pressure is introduced into the compressor. The temperature and pressure of the refrigerant are increased by the operation of the compressor. The refrigerant then flows through the condenser coils where it is liquefied, giving up heat to the environment. Still at a high pressure, but now at a lower temperature, the liquid refrigerant flows to an expansion valve where its pressure and temperature are reduced. The low-temperature refrigerant next flows into the evaporator, where air from inside the car is blown over the cooling coils. The refrigerant absorbs the heat from this interior air and carries it to the compressor where the cycle begins again.

The particular design selected for this study was an 8790-J/s (2.5-ton) refrigeration system similar to that used in refrigerated containers, adapted to fit on the top end section of the railcar (Figure 3). A flue would be constructed at this end of the car to allow air to circulate from the ceiling, through the evaporator, and down along the floor area. A plan of the system, shown in Figure 4, would have cooling capacities of 8790 J/s (30 000 Btu/h) for a perishable load at 35°C (95°F) ambient temperature and 2930 J/s (10 000 Btu/h) for a frozen load at 37.8°C (100°F) ambient temperature respectively.

This system using R-22 refrigerant is capable of refrigerating the maximum possible heat load under the most extreme environmental design conditions. The arrangement has an 8790-J/s (30 000-Btu/h) cooling capacity for a perishable load at 35°C ambient temperature and a 2930 J/s (10 000 Btu/h) capacity for a frozen load at 37.8°C ambient temperature. For the frozen load, where minimal heat is generated within the load, the system will have more than ample refrigerating capacity. At ambient temperatures lower than those used in the design calculations, the efficiency of the system would be higher, and the refrigeration load would be lower. When all environmental and perishable-loading factors are considered, the system as designed is capable of refrigerating most loads under normal ambient temperatures. The calculations below present an analysis of this system and its thermodynamic states for the most extreme conditions of temperature and pressure. (The numbered points refer to Figure 4.) (SI units are not given for these calculations because they were developed for U.S. customary units.)

$$Q_{tot} - W_{tot} = \Delta H + \Delta u^2/2g_c + \Delta z(g/g_c) \qquad (3)$$

where

Q_{tot} = all heat transfered into or out of system,
W_{tot} = amount of work or energy put into the system,
ΔH = enthalpy change of system,
$\Delta u^2/2g_c$ = kinetic energy of system, and
$\Delta z(g/g_c)$ = potential energy of the system.

The following assumptions are made:

1. The system is steady state and steady flow;
2. Compression and expansion of the refrigerant fluid is adiabatic, but not reversible;
3. Kinetic energy = 0; and
4. Potential energy = 0.

(Other equations and assumptions will be introduced as required.) The data used are given below:

Term	Definition	Value (Btu/h)
Q_{gen}	Respiration heat generated by commodity	22 100
Q_c	Rate of heat flow into interior loading space of car	3 906
Q_e	Heat generated by 0.75-bhp evaporator blower model	1 909
Q_{tot}	$Q_{gen} + Q_c + Q_e$	27 915

At point 1, the evaporator temperature = 28.00°F, the amount of superheat in refrigerant = 10°F, and therefore the saturated vapor temperature = 18.00°F. The corresponding pressure from the pressure-enthalpy (P-H) diagram (Figure 5) for 18.00°F saturated vapor = 55.551 lbf/in^2 (absolute). Interpolation of tables of properties of refrigerant 22 in the superheated region at 28.00°F and 55.551 lbf/in^2 (absolute) gives the following results: H_1 = enthalpy at point 1 = 108.4400 Btu/lb (mass) and S_1 = entropy at point 1 = 0.229 64 Btu/lb (mass)·°R.

At point 4, the condenser temperature = 95.00°F (from the maximum ambient temperature), and the state is saturated liquid. The corresponding pressure from the P-H diagram for saturated liquid at 95.00°F = 196.51 lbf/in^2 (absolute). Interpolation of the R-22 tables for the saturated liquid at 95.00°F and 196.51 lbf/in^2 (absolute) gives the following results: H_4 = enthalpy at point 4 = 37.7050 Btu/lb (mass) and S_4 = entropy at point 4 = 0.076 68 Btu/lb (mass)·°R.

At point 2′, there is isentropic compression from point 1 to a pressure of 196.51 lbf/in^2 (absolute), which is equal to P_4, the pressure at point 4—because the compression is isentropic, the entropy at point 1 must be equal to the entropy at point 2′, or $S_2' = S_1$ = 0.229 64 Btu/lb (mass)·°R. From S_2', interpolation of the R-22 tables in the superheated region gives the following results: H_2' = enthalpy at point 2′ = 122.6970 Btu/lb (mass), and T_2' = temperature at point 2′ = 148.23°F.

At point 2, there is compression from point 1 to a pressure of 196.51 lbf/in^2 (absolute) in the superheated region. This calculation requires the following assumptions: (a) compressor efficiency = 70 percent = 0.70, and (b) the compressor is adiabatic or Q_c from the compressor is 0. From Equation 3,

Figure 2. Operation of refrigeration system.

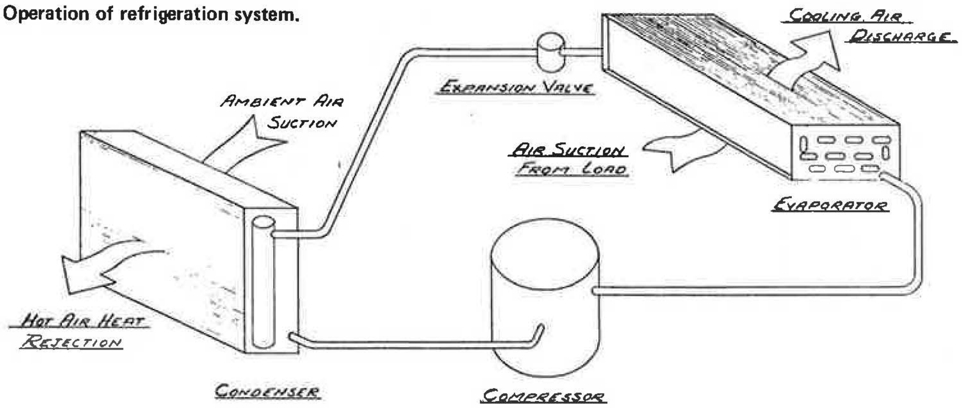


Figure 3. Arrangement of refrigeration system.

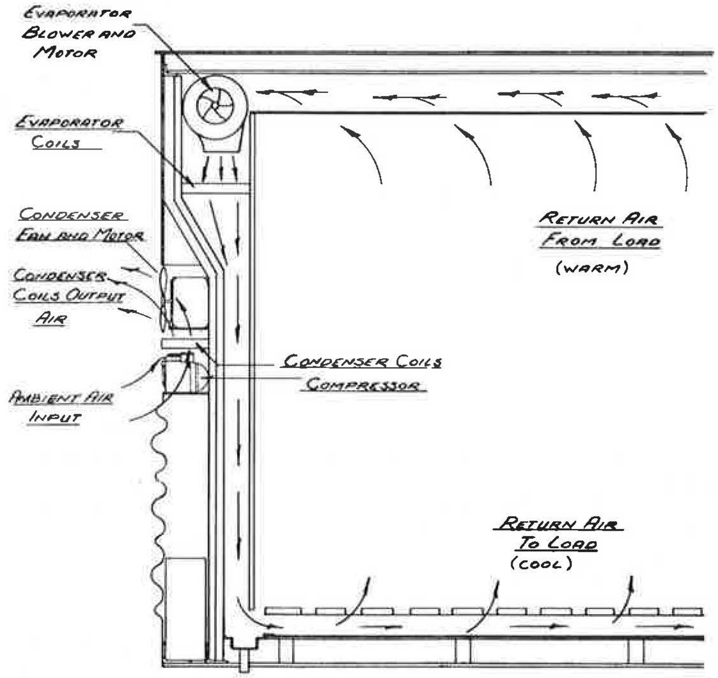


Figure 4. Plan of refrigeration system.

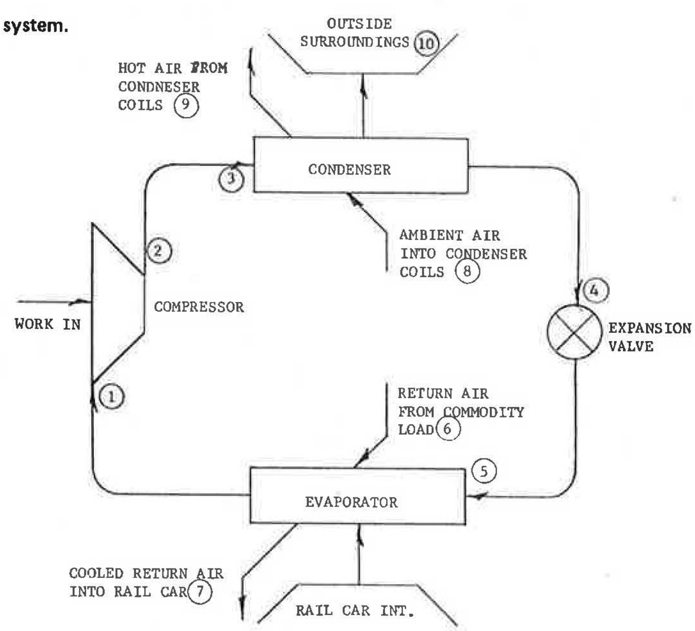
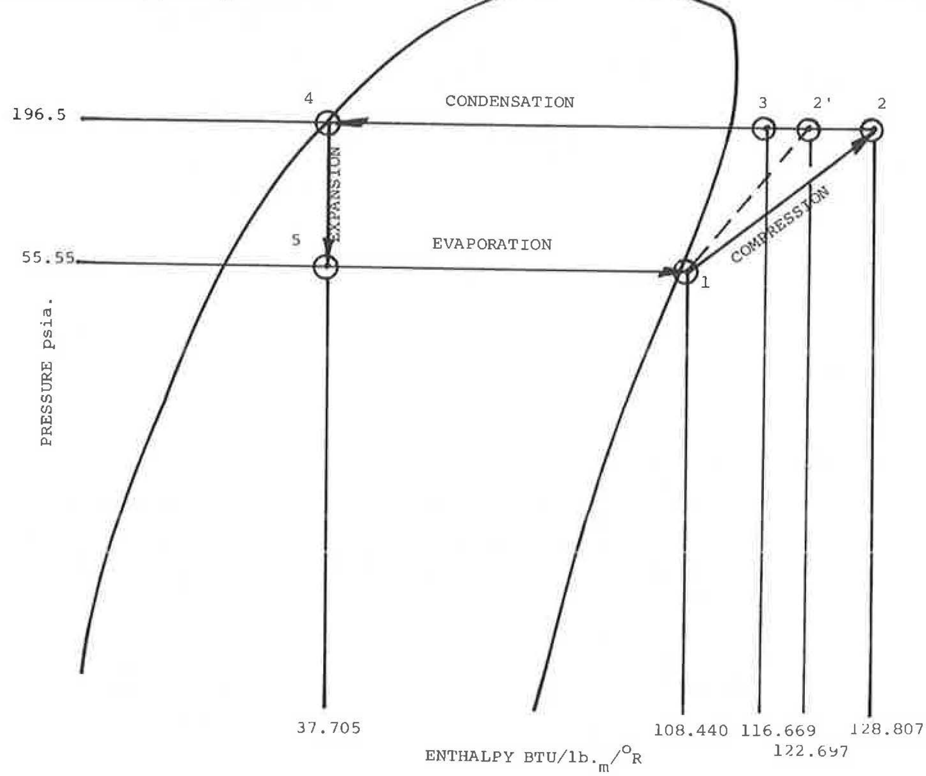


Figure 5. Pressure versus enthalpy: refrigerant 22.



$$W_{c_{rev.}} = H_2' - H_1 = (H_2 - H_1)n_c \quad (3a)$$

The reversible work of the compressor = actual work × efficiency. From Equation 3a, H_2 = enthalpy at point 2 = $[(H_2' - H_1)/n_c] + H_1$ = [(122.6970 - 108.4400) 0.70] + 108.4400 = 128.8070 Btu/lb (mass). Interpolation of the R-22 tables for superheated vapor at 196.51 lbf/in^2 (absolute) and H_2 = 128.8070 Btu/lb (mass) gives the following results: T_2 = temperature at point 2 = 181.95°F, and S_2 = entropy at point 2 = 0.239 49 Btu/lb (mass)·°R.

At point 3, because of cooling of the refrigerant between the compressor and the condenser, the temperature decreases. T_3 = temperature at point 3 = T_4 + 20°F = 115.0°F, and P_3 = pressure at point 3 = pressure at point 2 = 196.51 lbf/in^2 (absolute). Interpolation of the R-22 tables for superheated liquid at 115.00°F and 196.51 lbf/in^2 (absolute) gives the following results: H_3 = enthalpy at point 3 = 116.6690 Btu/lb (mass), and S_3 = entropy at point 3 = 0.219 42 Btu/lb (mass)·°R.

At point 5, the refrigerant has been expanded isentropically from point 4; i.e., there is no work or heat exchange in the process. H_4 = enthalpy at point 4 = H_5 = enthalpy at point 5 = 37.705 Btu/lb (mass), P_5 = pressure at point 5 = 55.551 lbf/in^2 (absolute), and T_5 = temperature at point 5 = 18.00°F. For a second-law analysis of the dynamics of wet refrigerant,

$$H_5 = 37.705 \text{ Btu/lb (mass)} = H_{f5} + x_5(H_{fg5}) \quad (4)$$

H_{f5} = tabulated value of enthalpy for dry refrigerant = 15.44 Btu/lb (mass), H_{fg5} = enthalpy for dry refrigerant - enthalpy for saturated refrigerant = 91.48 Btu/lb (mass), and x_5 = degree of saturation = $(H_5 - H_{f5})/H_{fg5}$ = (37.705 - 15.440)/91.480 = 0.2434,

$$S_5 = \text{entropy at point 5} = S_{f5} + x_5(S_{fg5}) \quad (5)$$

S_{f5} = tabulated value of entropy for dry refrigerant = 0.0341 Btu/lb (mass)·°R, and S_{fg5} = entropy for dry refrigerant - entropy for saturated refrigerant = 0.1916 Btu/lb (mass)·°R. Thus, S_5 = 0.0341 + 0.2434 (0.1916) = 0.080 74 Btu/lb (mass)·°R.

The following calculations of sensible and latent heat are required to find the properties of air at points 6 to 9.

$$q_s = \text{sensible heat change} = 1.08 \times \text{air flow} \times TD_{db} \quad (6)$$

$q_s = 0.25Q_{gen} + Q_c + Q_{em}$ = (0.25 × 22 100) + 3906.0 + 1908.75 = 11 339.75 Btu/hr, air flow = 2500 ft^3/min, and TD_{db} = temperature difference (dry bulb) between air entering and leaving the evaporator coils = q_s/(1.08 × 2500 = 11 339.75/(1.08 × 2500) = 4.20°F (dry bulb). T_7 = temperature of air entering interior of car from evaporator = 29.0°F, T_6 = temperature of air entering evaporator from car interior = 29.0 + 4.2 = 33.2°F, and P_6 = P_7 = pressure of air at evaporator = 41.697 lbf/in^2 (absolute).

$$q_l = \text{latent heat change} = 4.5 \times \text{air flow} \times \Delta H_L \quad (7)$$

$q_l = 0.75 \times Q_{gen}$ = 0.75 × 22 100 = 16 575 Btu/h, and ΔH_L = change in enthalpy of air (latent heat) = q_l/(4.5 × 2500) = 16 575/(4.5 × 2500) = 1.4733 Btu/h.

$q_t = q_s + q_l$ = total heat change = 11 339.75 + 16 575.0 = 27 914.75 Btu/h = 4.5 × air flow × ΔH_t. Thus, ΔH_t = total enthalpy change of supply air to the evaporator = q_t/(4.5 × 2500) = 27 914.75/(4.5 × 2500) = 2.4813 Btu/lb (mass). SHR = sensible heat ratio = q_s/q_t = 11 339.75/27 914.75 = 0.4062.

At point 6, air at 80 percent degree of saturation and 33.2°F is going from the car interior to the evaporator coils. H_t = total enthalpy change in air = 2.4813 Btu/lb (mass) = $H_6 - H_7$, and H_6 = enthalpy at point 6 = H_{a6} + $x_6(H_{as6})$ (from Equation 4). From the American Society of Heating, Refrigerating, and Air-Conditioning Engineers air tables, H_{a6} = tabulated value of enthalpy of dry

Table 1. Thermodynamic properties of refrigeration system.

Point	State	Temperature (°F)	Pressure [lb/in² (absolute)]	Enthalpy [Btu/lb (mass)]	Entropy [Btu/lb (mass)°R]
1	Superheated vapor	28.00	55.551	108.4400	0.229 64
2	Superheated vapor	181.95	196.510	128.8070	0.239 49
2	Superheated vapor	148.23	196.510	122.6970	0.229 64
3	Superheated vapor	115.00	196.510	116.6690	0.219 42
4	Saturated liquid	95.00	196.510	37.7050	0.076 68
5	Saturated liquid	18.00	55.551	37.7050	0.080 74
6	Moist air	33.20	14.697	11.3968	0.024 15
7	Moist air	29.00	14.697	8.9155	0.018 99
8	Dry air	95.00	14.697	22.8270	0.045 13
9	Dry air	108.32	14.697	26.0301	0.050 84
10	Dry air	95.00	14.697	22.8270	0.045 13

air = 7.975 Btu/lb (mass), and H_{as6} = enthalpy of dry air - enthalpy of saturated air = 4.2772 Btu/lb (mass). x_6 = degree of saturation = 80 percent = 0.80, and thus H_6 = 7.975 + 0.80(4.2772) = 11.3968 Btu/lb (mass). S_6 = entropy of air at point 6 = $S_{a6} + x_6(S_{as6}) + S_{cl}$ (from Equation 5), S_{a6} = tabulated value of entropy for dry air = 0.016 764 Btu/lb (mass)·°R, S_{as6} = entropy for dry air - entropy for saturated air = 0.009 112 Btu/lb (mass)·°R, and S_{as6} = entropy correction factor = 0.000 10 Btu/lb (mass)·°R. Thus, S_6 = 0.016 764 + 0.80 (0.009 112) + 0.000 10 = 0.024 153 6 Btu/lb (mass)·°R.

At point 7, air at 29.0°F is being circulated from the evaporator to the railcar interior. H_7 = enthalpy at point 7 = H_6 - 2.4813 = 11.3968 - 2.4813 = 8.9155 Btu/lb (mass). $H_{a7} = H_{a7} + x_7(H_{as7})$ (from Equation 4). H_{a7} = tabulated value of enthalpy of dry air = 6.966 Btu/lb (mass), and H_{as7} = enthalpy of dry air - enthalpy of saturated air = 3.540 Btu/lb (mass). x_7 = degree of saturation = $(H_7 - H_{a7})/H_{as7}$ = (8.9155 - 6.9660)/3.540 = 0.5507. S_7 = entropy of air at point 7 = $S_{a7} + x_7(S_{as7}) + S_{cl}$. S_{a7} = tabulated value of entropy of dry air = 0.014 70 Btu/lb (mass)·°R. S_{as7} = entropy of dry air - entropy of saturated air = 0.007 61 Btu/lb (mass)·°R, and S_{cl} = entropy correction factor = 0.000 10 Btu/lb (mass)·°R. Thus, S_7 = 0.014 70 + 0.5507 (0.007 61) + 0.000 10 = 0.018 99 Btu/lb (mass)·°R.

The air-flow and refrigerant-flow requirements are $q_t = \dot{m}_{air}(H_6 - H_7)$ (from Equation 3) = total heat change = 27 914.75 Btu/h. $H_6 - H_7$ = enthalpy change of air over evaporator = 2.4813 Btu/lb (mass), and $\dot{m}_{air}$ = air mass required to remove heat q_t = 27 914.75/2.4813 = 11 250.05 lb (mass)/h. Air flow required at evaporator = $m_{air}/(p_{air} \times 60)$, where p_{air} = air density at 32°F, = 11 250.05/(0.081 × 60) = 2314.80 ft³/min. For an energy balance at the evaporator, the heat leaving the air must be equal to the heat transfered to the refrigerant. Therefore, $q_t = \dot{m}_r(H_1 - H_5)$ (from Equation 3) = total heat change = 27 914.75 Btu/hr. H_1 = enthalpy at point 1 = 108.440 Btu/lb (mass), H_5 = enthalpy at point 5 = 37.705 Btu/lb (mass), $\dot{m}_r$ = mass-flow rate of refrigerant = $q_t/(H_1 - H_5)$ = 27 914.75/(108.440 - 37.705) = 394.64 lb (mass)/h.

For an energy balance, the condenser coils must remove to the environment the heat transferred by the evaporator and the energy put into the compressor. $q_t + W_c = TD_{db} \times 1.08 \times$ air flow (from Equations 6 and 7). q_t = total heat change = 27 914.75 Btu/lb (mass), W_c = work put into compressor = $\dot{m}_r(H_2 - H_1)$ = 394.64 (128.807 - 108.440) = 8037.63 Btu/h. TD_{db} = temperature difference dry bulb through the condenser = $(q_t + W_c)/(1.08 \times 2500)$ = (27 914.75 + 8037.63)/(1.08 × 2500) = 13.32° F.

At point 8, ambient air enters the condenser coils at a temperature of 95.0° F and atmospheric pressure. H_8 = enthalpy of dry air at point 8 = 22.827 Btu/lb (mass) (from the air tables), and S_8 = entropy of dry air at point 8 = 0.045 13 Btu/lb (mass)·° R.

At point 9, air leaves the condenser coils with added heat from the refrigerant. T_9 = dry air temperature at point 9 = T_8 + 13.32°F = 108.32°F, H_9 = enthalpy of dry air at point 9 = 26.0301 Btu/lb (mass), and S_9 = entropy of dry air at point 9 = 0.050 84 Btu/lb (mass)·°R. These results are summarized in Table 1.

The power requirements of the system are calculated below. $Comp_{bhp}$ = power requirement of compressor = $W_s \times \dot{m}_r = (H_2 - H_1) \times \dot{m}_r$, where heat loss from the compressor = 0 (from Equation 3), = (128.807 - 108.440) × 394.64 = 8037.6328 Btu/h = 3.1582 brake horsepower (bhp). The closest commercially available compressor is a 3⅓-bhp model having 2 cylinders, which would be the type used to run the refrigeration system.

$Evap_{bhp}$ = power requirement of evaporator fan motor = (air flow × P_t)/(6356 × n_{er}). Air flow = 2314.80 ft³/min, n_{er} = assumed motor efficiency = 80 percent, and P_t = maximum static pressure through evaporator coils = 1.5 in of mercury. Thus $evap_{bhp}$ = (2314.80 × 1.5)/(6356 × 0.80) = 0.6829 bhp. The closest commercially available motor is a ¾ bhp model, which would be used to run the blower for the evaporator air.

$Cond_{bhp}$ = power requirement of condenser fan motor = (air flow × P_t)/(6356 × n_{con}). Air flow = $\dot{m}_r(H_3 - H_4)/[(H_9 - H_8)/(p_{air} \times 60)]$, where p_{air} = air density at 95°F, = 394.64(116.6690 - 37.7050)/[(26.0301 - 22.8270)/(0.0684 × 60)] = 2370.57 ft³/min, P_t = maximum static pressure through condenser coils = 1.0 in of mercury, n_{con} = assumed motor efficiency = 80 percent. Thus, $cond_{bhp}$ = (2370.57 × 1.0)/(6356 × 0.80) = 0.4662 bhp. The closest commercially available motor is a ½-bhp model, which would be used to run the blower for the condenser air.

ANALYSIS OF ENERGY AVAILABILITY

By using an energy-availability analysis of a system, it is possible to establish a program of energy management for it. This type of analysis, which is currently needed in all areas of energy use, will be used to show how fuel oil is consumed in the present railcar environment and the advantages in energy consumption of the smaller refrigeration system powered by solar cells.

Available energy is defined as that portion of energy added to a system that is converted directly to work by a reversible process (i.e., a process in which the total amount of energy put in can be recovered at any time). Unavailable energy is defined as that portion of energy added to a system that cannot be converted directly into work, but is lost to the environment because of friction, heat loss, or inefficiency of the components of the system.

These two definitions are the basis for the thermodynamic energy analysis of any system using energy or transferring heat. According to the first law of thermodynamics, mechanical systems convert fuel into useful work, and this useful work is recoverable at any time. According to the second law of thermodynamics, of the work produced by a fuel, some is transformed into unrecoverable energy or work, and the remainder is used for the final goal of the system. The calculations below show how much of the total energy input will be used for the refrigeration itself in the new design as compared with that used at present in railcars.

e = effectiveness
= ideal power input to system/actual power input to system (8)

b = availability function = $(H - T_o S)$ (9)

$\dot{I}$ = irreversibility = maximum work - actual work
= amount of heat or work lost because of effects such as friction
= $\dot{m}_r \Delta b$ (10)

The operating data are shown below.

Item	Value
Refrigerant	R-22
$\dot{m}$, lb(mass)/h	394.64
Electric input to compressor, Btu/h	8 483.32
n_c	0.80
Compressor heat loss, Btu/h	0
Condenser air-flow rate, ft³/min	2 370.57
Electric input to fan motor, Btu/h	1 272.50
n_{con}	0.80
Evaporator air-flow rate, ft³/min	2 314.80
Electric input to evaporator fan motor, Btu/h	1 908.75
n_{ev}	0.80
Maximum energy absorbed from railcar interior, Btu/h	27 914.75
Interior temperature	491.67°R
Ambient temperature	554.67°R

b_1 = availability function at point 1 = $H_1 - T_0S_1$ = 108.440 - (554.67 × 0.229 64) = -18.934 41 Btu/h, b_2 = availability function at point 2 = $H_2 - T_0S_2$ = 128.8070 - (554.67 × 0.239 49) = -4.030 91 Btu/h, b_3 = availability function at point 3 = $H_3 - T_0S_3$ = 116.6690 - (554.67 × 0.219 42) = -5.036 69 Btu/h, b_4 = availability function at point 4 = $H_4 - T_0S_4$ = 37.7050 - (554.67 × 0.076 68) = -4.827 095 Btu/h, and b_5 = availability function at point 5 = $H_5 - T_0S_5$ = 37.7050 - (554.67 × 0.080 74) = -7.079 055 Btu/h.

The actual power input into the system = compressor power input + condenser-fan motor input + evaporator-fan motor input = 8483.32 + 1272.5 + 1908.75 = 11 664.57 Btu/h. The ideal power input into the system = $\dot{m}_{air} T_0 (S_7 - S_6)] - (H_7 - H_6) = IP_{in}$ = 11 250.05 × 554.67 (0.018 99 - 0.024 15) - (8.9155 - 11.3968) = -4283.9875 Btu/h.

e = effectiveness of system = IP_{in}/actual power = 4283.9875/11 664.57 = 0.3673. $\dot{I}_{comp}$ = irreversibility of compressor = $\dot{m}_r(H_2 - H_1) - \dot{m}_r(b_2 - b_1)$ = 8037.6328 - 5881.9603 = 2155.6725 Btu/h, $\dot{I}_{e\,fan}$ = irreversibility of evaporator fan motor = T_L/T_H × electric input to evaporator motor = (491.67/554.67) × 1908.75 = 1691.952 Btu/h, $\dot{I}_{cond}$ = irreversibility of condenser = $\dot{m}_r(b_5 - b_1)$ = 394.64[-5.036 69 - (-7.8271)] = 1101.2911 Btu/h, $\dot{I}_{con\,fan}$ = irreversibility of condenser fan motor = 0 Btu/h (because no energy from the fan motor is available to the system), $\dot{I}_{ev}$ = irreversibility of expansion valve = $\dot{m}_r(b_5 - b_4)$ = 394.64[-7.07906 - (-7.82710)] = 295.2065 Btu/h, $\dot{I}_{evap}$ = irreversibility of evaporator = $\dot{m}_r(b_5 - b_1) - (1 - T_L/T_H)\dot{m}_r(H_5 - H_1)$ = 394.64[-7.079 06 - (-18.934 41)] - [1 - (491.67/554.67)] × 394.64(37.7050 - 108.440) = 1478.6313 Btu/h, $\dot{I}_{dl}$ = irreversibility of discharge line from compressor = $\dot{m}_r(b_2 - b_3)$ = 394.64 [-4.030 91 - (-5.036 69)] = 396.9210 Btu/h. The ideal power input into the system + the total of all $\dot{I}$ = actual power into system = 4283.9875 + 7119.6744 = 11 403.662.

The present railcar refrigeration system hourly converts an average of 3.2 L (0.85 gal) of diesel fuel into the energy necessary to power a 35.2-kJ/s (10-ton) refrigeration system. The effectiveness of this system is about 15 percent of the actual power available from the fuel. The remaining 85 percent of the fuel is eventually converted into wasted heat and cannot be recovered. This can and should be improved on. One solution would be the use of smaller refrigeration systems, incorporated in better designed railcar structures, and deriving power from solar energy.

The refrigeration system proposed here would require no diesel fuel. Its effectiveness would be 37 percent, which is an increase of over 200 percent in the use of the energy delivered to the system.

SOLAR CELL SYSTEM

A railcar refrigeration system can operate on electricity produced by a diesel engine, drawn from storage batteries, taken from the electric wires above the train, or produced by photovoltaic energy cells. The first three alternatives all rely on petroleum resources that at the present rate of growth in use of 7.3 percent will be exhausted in 31 years (3). Solar energy as an alternative energy source could also be used to provide electricity to refrigerate the railcar. The amount of solar energy falling on the ground is 10 000 times the present world consumption, which establishes the sun as a major energy source.

The photovoltaic effect is the direct production of electricity through absorption of sunlight by semiconductors that can convert light energy into electrical energy.

The annual average incidence of solar energy on the ground in the United States is 720 W/m² (67 W/ft²) (3). Photovoltaic cells are currently available made from silicon or cadmium sulfide having efficiencies of 14 percent. A system of photovoltaic cells operating at this efficiency and receiving the average solar power would produce 101 W/m² (9.38 W/ft²). By using cells having a 12.5 percent efficiency and a roof area of 55.74 m² (600 ft²), the average power output available from a railcar solar system would be 5 kW, which would be sufficient to power its refrigeration system.

Although these theoretical figures indicate that the solar array is capable of supplying the needed power, the present cost of $4000/kW would put the cost of the array at $20 000. Cost reduction in this area can be expected: The Energy Research and Development Administration has set goals of $2000/kW for 1980, $500/kW for 1985, and $100/kW for 2000. If the refrigeration system and the railcar design described above prove to be adequate, the 1980 cost of a 5-kW array would be $10 000. By eliminating the present costs of $6100 and $900 for the diesel engine and the fuel tank respectively and $900/year for the fuel, the cell cost would be offset in 3 years. The following (2) summarizes the position that has been taken on the possibilities of solar cells to produce large quantities of electrical power.

> In order to justify these (photovoltaic) expenditures, however, it is necessary to demonstrate that, when these array-cost goals are reached, photovoltaic solar-energy systems will have a good chance of competing successfully for a significant place in the nation's energy market.

Preparing for the future needs of refrigerated transport is vital to the railroad industry. The power capabilities of photovoltaic cells will provide an alternative power source as sources of diesel fuel diminish and costs of operation increase.

ENVIRONMENTAL CONSIDERATIONS

Air and noise pollution have recently become major concerns in present and projected transportation operations.

New federal and state air pollution standards have been set and are being enforced. The diesel engines powering the present railcar refrigeration systems are contributors to air pollution, control of which will be very costly. The elimination of the diesel engine by using solar panels would not only eliminate chemical air pollutants, but could also save the costs of pollution-control equipment, which would offset a portion of the first cost of the solar panels.

Noise levels, recently regulated by the Environmental Protection Agency, are another major concern. Certain projected state noise-emission controls now under consideration could force the industry into costly and time-consuming programs of redesigning or retrofitting existing equipment. The elimination of the diesel engine as a power source would therefore provide additional environmental-control benefits.

DISCUSSION AND CONCLUSIONS

The refrigeration of a perishable load can be theoretically accomplished with an 8.8 kJ/s (2.5- ton) refrigeration system powered by the direct current generated by photovoltaic cells. This affords the operator of this type of equipment almost total independence from energy-oriented problems.

However, there are limitations that must be considered because they dictate the circumstances under which such a system could be used and the extent of engineering technology that must be perfected.

1. At present, only daytime operation is feasible. Further advances in fuel-cell or storage-battery technology are required before continuous day-night operation could be used.
2. Some loads would still require precooling to a predetermined temperature level, mainly because of their respiration rates. The system as designed could precool an item with a low respiration rate such as oranges or cantaloupes. Corn or peas, which have higher respiration rates, would require precooling before release for shipment and probably top icing to provide the proper humidity.
3. Along with precooling before shipment, the railcar would require a storage battery to power the system until the solar energy was sufficient to operate it.
4. Maximum efficiency would be obtained only if the railcars were operated as a unit train, thereby benefiting from maximum travel distance during daylight operation.
5. If for some reason the railcars were stopped in a classification yard for any length of time, standby power sources would be necessary for continuous operation.
6. The solar panels would require careful and scrupulous maintenance.

RECOMMENDATIONS

Railroad refrigerator cars operating independently of fossil fuels may be an immense asset in the near future. An existing mechanically refrigerated railcar could be modified in construction to lower the structural heat loss. A smaller, more efficient refrigeration system could then be incorporated into the railcar for analysis and testing, using standby electrical power, to achieve maximum efficiency and system effectiveness. When the costs of photovoltaic-cell arrangements become lower and their technology is improved, a solar-assisted system could be attached to the roof of the prototype.

A model of this type of system could be operational by 1980. The tempo of technological advancement and our dwindling energy supplies make it mandatory for the transportation industry to play a leading role in the development of new methods for storing and transporting perishable commodities.

ACKNOWLEDGMENT

I wish to express my appreciation to the following individuals who gave freely of their time and energies toward the preparation of this paper: Albert L. Creel, Jr., Lydia V. McClary, and Charles C. Necessary of the Fruit Growers Express Company and James B. Kelly, Ivor S. Pelsue, and Edward F. Russ of the Carrier Transicold Company.

REFERENCES

1. W. H. Redit. Protection of Rail Shipments of Fruits and Vegetables. Agriculture Handbook Number 195, July 1969.
2. S. S. Leonard. Mission Analysis of Photovoltaic Conversion of Solar Energy. 11th IEEE Photovoltaic Specialists Conference, 1975.
3. J. H. Krenz. Energy Conservation and Utilization. Allyn and Bacon, Boston, 1976.
4. J. A. Merrigan. Prospects for Solar Energy Conversion by Photovoltaics. MIT Press, Cambridge, 1975.

*Mr. Conover is now with the American Gas Association, Arlington, Virginia.