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Study of Solar-Assisted Thermoelectric Technology for Automobile Air Conditioning

An analytical study was conducted to determine the feasibility of employing solar energy assisted thermoelectric (TE) cooling technology in automobile air conditioners. The study addressed two key issues—power requirements and availability of thermoelectric materials. In this paper, a mathematical model was developed to predict the performance of TE air conditioners and to analyze power consumption. Results show that the power required to deliver a cooling capacity of 4 kW (13,680 Btu/h) in a 38°C (100°F) environment will be 9.5 kW electric. Current TE modules suitable for air conditioning are made of bismuth telluride. The element tellurium is expected to be in short supply if TE cooling is widely implemented for auto air conditioning; some options available in this regard were studied and presented in this paper. The photovoltaic (PV) cells, assumed to cover the roof area of a compact car, can only generate about 225 W. However, this is more than enough to power a fan to provide air ventilation to the car interior, which significantly reduces the peak cooling load when the car is parked in bright sunlight.

Introduction

Increased awareness of the impact of chloroflurocarbons (CFCs) on the global environment has become the impetus in searching for alternative refrigerants and cooling methods for automotive air conditioning. Automotive air conditioning is one industry that heavily uses CFC compounds, and the leakage of CFCs from such air conditioners is substantial compared to that from stationary air conditioners. The consensus of the industry is that the CFC–12 replacement refrigerant, HFC–134a, could still face future regulatory uncertainty because of its global warming potential. Therefore, the use of non-CFC air conditioners in automobiles is becoming very important.

There are a number of air conditioning technologies that are not CFC based. One of them is the well-known thermoelectric (TE) technology. The air conditioners employing this technology use some kind of specialized semiconductors instead of working fluids: thus the core of a TE air conditioner is a completely solid-state device. In general, TE technology appears to possess characteristics that are favorable for automotive applications. These include (a) a match in power source—d-c power is required to operate TE units, and the power system in automobiles is d-c; (b) high reliability and easy maintainability—important for consumer products such as automobiles; (c) compact size; (d) quietness; and (e) easy capacity modulation. Factors that are unfavorable to TE technology include (a) relatively low efficiency (TE units' efficiency is lower than that of CFC-based units, yet is very comparable to that of many other non-CFC units) and (b) availability of TE materials to meet the quantity of production required for auto air conditioning.

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A TE air conditioning system was used on railway coach (Stockholm et al., 1982). The railway coach has operated the TE system for over 20,000 hours without a TE failure. TE cooling has also been considered for submarine and surface ship applications (Blankenship et al., 1989; Stockholm and Schlicklin, 1988, 1989). However, most of them are for special applications where the advantages of TE systems outweigh their low efficiency. For mobile air conditioning systems, because of recent environmental concerns on refrigerant used in the cooling units, alternative cooling technology is needed. With better TE system design and improved TE materials, TE cooling could be a viable alternative to the existing vapor compression systems.

Adding solar PV cells to the TE cooling systems serves two purposes: to provide air ventilation to the car interior and to provide power to the TE system during operation. Ingersoll (1989) states that this concept could reduce the passenger car peak cooling load up to 40 percent. The product information on solar roof panels manufactured by Sekurit company (1987) indicates that 20 100-by-100 mm (3.94 by 3.94 in.) monocrystalline silicon cells could generate 25 W of power at full insolation, which is enough to power an air circulation fan. Assuming a compact car with a roof area of 1.81 m^2 (19.5 ft^2) (Ruth, 1979), there will be enough room for PV cells to produce about 225-W power at full insolation, which can be used to operate the air circulating fan to lower car interior temperature, to charge the battery, or to run the blower. This concept is particularly important to electric vehicles which have a limited power supply from battery packs and yet have to maintain the comfort of the passenger compartment.

Description of a Conceptual System

The conceptual solar-assisted TE system has six essential components: a d-c generator, a blower, a PV cell topped roof, an air circulating fan, a cooler, and a heat rejector. The schematic of the TE air conditioner system is shown in Fig. 1 (PV cell part and air circulation fan are not shown). The d-c generator is where the compressor of a conventional freon system would be located. This high-voltage d-c generator (voltage depends on the number of TE modules) is driven directly by the engine to supply the necessary d-c power to the TE modules. A blower is used, as in the conventional CFC system, to circulate the air to be conditioned; this stream of air is called "process air" in this study. The process air is either the returning interior air or a mixture of fresh and interior air. The blower draws process air, pumps it through the cooler, and finally sends it back to the car interior via several vents. The PV cells provides additional power that can be used to operate the air circulating fan when the car is parked under the sun, to operate the blower, or for any other uses such as charging the battery.

Figure 2 shows the possible design of TE heat exchangers. The cooler is located where the evaporator of the current CFC system would be located. This unit is a thermoelectric solid-state device having several TE modules. It removes heat and moisture from the process air using d-c power, and rejects heat to a transfer fluid to be used in the system. The transfer fluid is a mixture of ethylene glycol and water, which is the same as that used in automobile radiator fluid. The last component is the heat rejector. This unit is located where the condensing unit of the CFC system would be placed. The heat rejector is also a thermoelectric device capable of removing heat from the transfer fluid and rejecting it to the ambient air.

The flow rate of process air and the mix of interior/outside air can be controlled in the same manner as in the existing air conditioning system. The temperature of air exiting the vents, however, is controlled by regulating the d-c power applied to the cooler and/or the heat rejector.

Mathematical Model

The system schematic is shown in Fig. 1. The cooling system is composed of two cross-flow thermoelectric-liquid (water/ethylene-glycol mixture) heat exchangers, a heat rejector, and a cooler (heat absorber). A liquid circulating pump is also included. Based on the energy balance of the heat transfer medium, air, and liquid (water/ethylene-glycol mixture), the following mathematical model can be formed.

Heat Rejector:

- Liquid side—

$$-M_W C_{WR} L_{1R} \frac{\partial T_{WR}}{\partial X} = U_{WR}\left(T_{WR} - T_{CR}\right), \text{ and} \quad (1)$$

$$-M_W C_{WR} L_{1R} \frac{\partial T_{WR}}{\partial X} = N\left[\alpha T_{CR} I_R - \frac{0.5 I_R^2 R}{R} - K\left(T_{hR} - T_{CR}\right)\right], \quad (2)$$

where

$$K = k_p A_p / l_p + k_n A_n l_n, \text{ and}$$
$$R = \rho_p l_p / A_p + \rho_n l_n / A_n.$$

- Air side—

$$M_{aR} C_{aR} L_{2R} \frac{\partial T_{aR}}{\partial Y} = U_{aR}\left(T_{hR} - T_{aR}\right), \text{ and} \quad (3)$$

$$M_{aR} C_{aR} L_{2R} \frac{\partial T_{aR}}{\partial Y} = N\left[\alpha T_{hR} I_R + \frac{0.5 I_R^2 R}{R} - K\left(T_{hR} - T_{CR}\right)\right]. \quad (4)$$

Cooler (Heat Absorber):

- Liquid side—

$$M_W C_{WA} L_{1A} \frac{\partial T_{WA}}{\partial X} = U_{WA}\left(T_{hA} - T_{WA}\right), \text{ and} \quad (5)$$

$$M_W C_{WA} L_{1A} \frac{\partial T_{WA}}{\partial X} = N\left[\alpha T_{hA} I_A + \frac{0.5 I_A^2 R}{R} - K\left(T_{hA} - T_{CA}\right)\right]. \quad (6)$$

- Air side—

$$-M_{aA} C_{aA} L_{2A} \frac{\partial T_{aA}}{\partial Y} = U_{aA}\left(T_{aA} - T_{cA}\right), \quad (7)$$

$$-M_{aA} L_{2A} \frac{\partial h_{aA}}{\partial Y} = N\left[\alpha T_{cA} I_A - \frac{0.5 I_A^2 R}{R} - K\left(T_{hA} - T_{cA}\right)\right], \text{ and} \quad (8)$$

$$-M_{aA} L_{2A} \frac{\partial W_{aA}}{\partial Y} = J_{aA}\left(W_{aA} - W_{cA}\right). \quad (9)$$

Nomenclature

- A = TE element cross-section area
- C_a = specific heat of air
- C_w = specific heat of liquid
- COP_c = cooling coefficient of performance
- h_a = enthalpy of air
- I = electrical current flow
- J_a = product of air-side mass transfer coefficient and area
- K = thermal conductance of a TE couple
- l = TE element length
- L_1 = characteristic length of liquid channels in the TE heat rejector/cooler
- L_2 = characteristic length air passage in the TE heat rejector/cooler
- M_a = air mass flow rate
- M_w = liquid mass flow rate
- N = total number of thermoelectric couples
- P = power input to the TE system
- NTU = number of transfer units
- R = electrical resistance
- T_a = air temperature
- T_c = TE module cool-side temperature
- T_h = TE module hot-side temperature
- T_w = liquid temperature
- U_a = product of air-side heat-transfer coefficient and area
- U_w = product of liquid-side heat-transfer coefficient and area
- W_a = air moisture content
- W_c = air moisture content at temperature T_c
- X = space coordinate in the liquid flow direction
- Y = space coordinate in the air flow direction
- α = Seebeck effect of TE couple

Subscripts

- A = cooler
- n = TE module negative element
- p = TE module positive element or pump side
- R = heat rejector
- s = solar photovoltaic cells

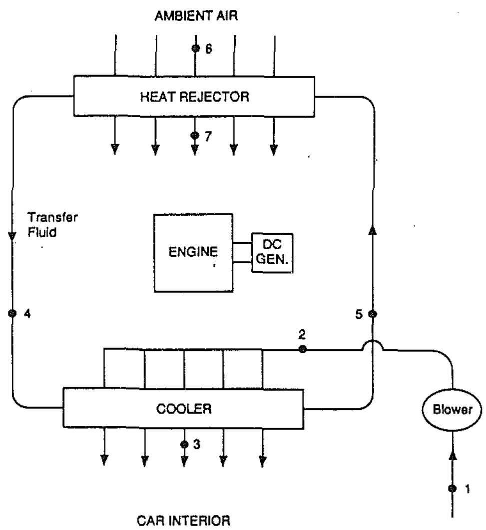

Fig. 1 Schematic of a TE automobile air conditioning system

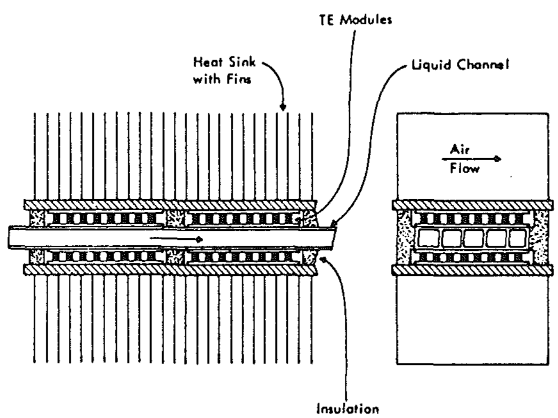

Fig. 2 Schematic of a TE heat exchanger

The system cooling COPs can be calculated by the following equation:

$$COP_c = \frac{\text{Cooling Capacity}}{P_A + P_R - P_s} \qquad (10)$$

where P_s is the power generated by PV cells.

The computer code was based on a previous study (Mei et al., 1989) modified for this application. Realistic TE material properties, as shown below, were used for the simulation.

Seebeck Coefficient:

$$\alpha = 1.8 \times (2T + 1985)\ 10^{-7}\ \text{volt}/^\circ\text{C}.$$

Electrical Resistance:

$$\rho = (6T + 1735)\ 10^{-6}\ \text{ohm-cm}.$$

Thermal Conductivity:

$$k = 0.0324\ \text{watt/cm}/^\circ\text{C}.$$

where

$$T = \frac{(T_h + T_c)}{2}.$$

The above TE property figures are for those typical off-the-shelf (not state-of-the-art) TE modules, including the semiconductors, copper straps, and thermal resistance of ceramic substrates, which are very realistic for system design purposes.

Design Conditions

Based on a literature review of the cooling requirements of a typical automobile, a set of design conditions was selected to estimate the power requirements of the TE air conditioner. The additional power provided by PV cell is not considered in the design conditions.

Cooling Load. The paper published by Ruth (1975) contains the cooling requirement data for various car types, ambient weather conditions, and driving conditions. His data, shown in Table 1, were considered for selecting the design cooling load. A compact car was also chosen for this analysis. We considered the worst-case cooling load for ambient conditions of up to 38°C (100°F); the cooling needs at 43°C (110°F) ambient were not considered for design since it was felt that the probability of an air conditioner operating under such ambient conditions would be extremely low. According to Ruth's data, a cooling requirement of 4.01 kW (13,680 Btu/h) is the worst case at 38°C (100°F) ambient; therefore, it was decided to choose a design cooling load of 4.01 kW (13,680 Btu/h) at 38°C (100°F) ambient. Because TE efficiency increases at a lower ambient temperature, the TE air conditioner designed to provide 4.01 kW (13,680 Btu/h) at 38°C (100°F) ambient will provide more than 4.17 kW (14,220 Btu/h) in a 32°C (90°F) environment.

State of Interior Air. Several investigators in the area of car air conditioning have used a design state of 25°C (77°F) at 60 percent relative humidity (RH) as the design value. This value seems very appropriate from the standpoint of the occupants' comfort. Therefore, we decided to use this state as our design state of interior air.

State of Process Air at Stage 1. Again, per Ruth's data, a cooling requirement of 4.01 kW (13,680 Btu/h) applies when the system is operating under a 100 percent recirculating mode. Under this condition, the state of process air at stage 1 (see Fig. 1) should be the same as the state of interior air (25°C (77°F) 60 percent RH). The humidity ratio of this stream is 0.0119 gr water/gr dry air (0.0119) lb water/lb dry air).

State of Process Air at Stage 2. This is the state of the air at the entrance to the cooler. Because of the blower heat and the heat transmitted from the engine and picked up by the panel, the temperature at stage 2 is expected to be about 8.3°C (15°F) more than at stage 1. As such, we chose a design temperature at stage 2 of 25 + 8.3 = 33.3°C (77 + 15 = 92°F). The humidity ratio at stage 2, however, will be the same as stage 1. The RH of air at 33.3°C (92°F) dry bulb and 0.0119 humidity ratio is 37 percent; thus, the state of process air at stage 2 will be 33.3°C (92°F) and 37 percent RH. The enthalpy of process air at stage 2 is 81.97 kJ/kg (35.24 Btu/lb).

Process Air Flow Rate. For current automotive air conditioners, the design flow rate of process air varies from 386 to 567 kg/h (850 to 1250 lb/h). For our design, we have chosen 454 kg/h (1000 lb/h), which seems reasonable for a compact car.

State of Process Air at Stage 3. Having known the total cooling load, the enthalpy of air at stage 1, and the mass flow rate, we calculated the enthalpy of air at stage 3, using

$$\text{Cooling} = \text{Mass flow rate} \times (\text{enthalpy at stage 2} - \text{enthalpy at stage 3}).$$

The enthalpy of process air at stage 3 was thus found to be 50.14 kJ/kg (21.56 Btu/lb). From psychrometrics, this relates to an air state of 11°C (52°F) and 100 percent RH.

Table 1 Automotive cooling requirements

Car type	Ambient condition °C (°F)/RH	City driving (cooldown) no outside air kW (Btu/h)	City driving 30 mph 100% outside air kW (Btu/h)	Highway driving 60 mph 100% outside air kW (Btu/h)
	32 (90)/50%	3.59 (12250)	3.49 (11910)	3.77 (12850)
Subcompact	38 (100)/20%	3.47 (11830)	3.20 (10930)	3.41 (11640)
	43 (110)/ 5%	3.86 (13170)	3.79 (12950)	4.08 (13940)
	32 (90)/50%	4.14 (14140)	4.17 (14220)	4.14 (14120)
Compact	38 (100)/20%	4.01 (13680)	3.76 (12840)	3.73 (12730)
	43 (110)/ 5%	4.42 (15100)	4.51 (15380)	4.48 (15280)
	32 (90)/50%	5.06 (17270)	5.16 (17620)	5.13 (17520)
Standard	38 (100)/20%	4.91 (16770)	4.64 (15830)	4.61 (15730)
	43 (110)/ 5%	5.37 (18320)	5.55 (18950)	5.52 (18850)

State of Ambient Air at Stage 6. Just as we did while selecting the cooling requirements, we chose an ambient temperature of 38°C (100°F) as the design value at stage 6.

Transfer Fluid Temperature Change. The fluid used as the transfer fluid is the ethylene-glycol water mixture. This fluid is heated while passing through the cooler and cooled while passing through the heat rejector. Because both of these units will have a cross-flow configuration, the first law of thermodynamics dictates that the temperature increase in the cooler (which is also equal to the temperature drop in the heat rejector, assuming no thermal losses in the line) should be as small as possible. Our estimate is that this increase in temperature should be limited to about 3 to 5.5°C (5 to 10°F). An increase of less than 3°C (5°F) will require an excessively large heat exchanger, and an increase of more that 5.5°C (10°F) will make the TE system inefficient.

Volumetric Flow Rate of Transfer Fluid. The volumetric flow rate of transfer fluid depends on the amount of heat removed from the cooler and the temperature rise in the cooler. For the purpose of power analysis, we used three different flow rate values: 1.51, 1.89, and 2.27 m^3/h (400, 500, and 600 gal/hr). We estimate that we can achieve a COP of 1.5 in the cooler based on our past experience; this means that the heat to be removed in the cooler will be 4.01 kW + 2.67 kW = 6.68 kW (13,680 + 13,680/1.5 = 22,800 Btu/h). The heat capacity of the 50/50 ethylene-glycol/water mixture is about 0.989 kW$\cdot$ $m^3\cdot$°C [7.11 Btu/(gal·°F)]. At 1.89 m^3/h (500 gal/h) flow rate, the temperature rise in the cooler will be. 3.6°C (6.4°F), which is well within the acceptable range.

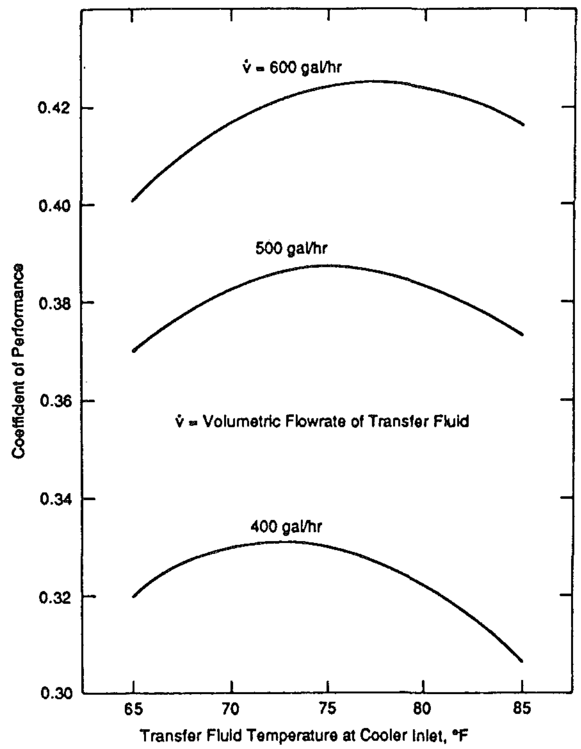

Fig. 3 System performance as a function of transfer fluid cooler inlet temperature

Number of Transfer Units (NTU) Values. NTU is defined as the ratio of the total heat transfer rate [in W/°C or Btu/(h·°F)] on any side of a heat transfer device to the heat capacity of the fluid [in W/°C or Btu/(h·°F)]. At this time, we estimate that we can design the cooler such that the NTU on the process air side will be about 6.0 and about 1.0 on the transfer fluid side. We also estimate that we can design the heat rejector such that the NTU on the transfer fluid side will be about 1.0 and about 0.9 on the ambient air side.

Power Analysis

A computer simulation of the TE air conditioner system was performed using the design conditions. The output from the computer included the cooler COP, heat rejector COP, overall COP (with and without solar PV cells), temperatures of transfer fluid at stages 4 and 5, and other information pertinent to the selection of TE modules.

The cooler COP is defined as the ratio between the useful cooling delivered to the process air and the electric power applied to the cooler. The heat rejector COP is defined as the ratio between the useful cooling delivered to the transfer fluid and the electric power applied to the heat rejector. The overall COP is defined as the ratio between the useful cooling delivered to the process air and the total electric power applied to both the cooler and the heat rejector minus the power generated by the roof-top solar PV cells.

Table 2 contains the values of different COPs mentioned above. Figure 3 shows the values of overall COP (not considering the power provided by PV cells) in graphical form. As can be seen from the table values, for a given flow rate of transfer fluid, the cooler COP decreases as the transfer fluid temperature at the cooler inlet is increased; at the same time, the rejector's COP increases. The overall COP attains a maximum value at some optimum value of transfer fluid temperature. In general, this optimum temperature is around 24°C (75°F).

Table 2 Coefficient of performance

Transfer Fluid		Cooler COP	Rejector COP	Overall	
Flow rate m^3/h (gal/h)	Cooler inlet temperature °C (°F)			COP	COP (solar energy assisted)
1.51 (400)	18 (65)	3.318	0.460	0.312	0.326
	21 (70)	2.363	0.546	0.330	0.336
	24 (75)	1.781	0.633	0.330	0.337
	27 (80)	1.389	0.721	0.322	0.328
	29 (85)	1.108	0.806	0.306	0.312
1.89 (500)	18 (65)	3.836	0.516	0.370	0.377
	21 (70)	2.686	0.613	0.383	0.391
	24 (75)	2.009	0.718	0.387	0.395
	27 (80)	1.564	0.833	0.384	0.392
	29 (85)	1.250	0.958	0.373	0.381
2.27 (600)	18 (65)	4.234	0.548	0.401	0.410
	21 (70)	2.925	0.653	0.417	0.427
	24 (75)	2.173	0.769	0.424	0.434
	27 (80)	1.687	0.901	0.424	0.434
	29 (85)	1.348	1.050	0.416	0.426

For a given transfer fluid temperature at the cooler inlet, the cooler COP, rejector COP, and overall COP all increase as the flow rate of transfer fluid is increased. This suggests that the flow rate of the transfer fluid should be as high as is practical. This will reduce the temperature change of the transfer fluid in the cooler (and in the rejector). Theoretically, a fluid with zero temperature change will offer the highest COP values. From this perspective, a phase change fluid (which will evaporate in the cooler and condense in the rejector) might be a better candidate than the chosen ethylene glycol.

At a flow rate of 2.27 m^3/h (600 gal/h) and 24°C (75°F), the analysis shows that an overall COP of 0.424 (without considering the power input from PV cells) is achievable employing the TE technology. Based on the limited data available on conventional air conditioning systems, we project the COP of such systems to be in the area of 1.8. As such, the COP of air conditioners using TE technology appears to be low. However, this COP will be competitive with other non-CFC-based systems. For a design cooling load of 4.01 kW (13680 Btu/h) at 38°C (100°F) and the associated COP of 0.4241 (or 0.434 with the power input from PV cells), the power consumption will be 9.451 kW (12.67 hp). In a car with an engine power of, say, 120 to 130 hp, this represents a ten percent power draw.

Materials Availability

Four materials are used to make up the basic TE components. Bismuth and tellurium are the primary ingredients, and small portions of selenium and antimony are added in different proportions to create the *N* and *P* polarity of the resulting semiconductor materials. Of these four materials, bismuth, selenium, and antimony are widely available from multiple sources with considerable excess capacity potential at no increase in price. Only tellurium has a finite limit on availability with some cost dependency.

Tellurium is not practical to mine directly and is a by-product of other mining operations. The primary source of tellurium is the refining of copper waste, although tellurium is also present and recoverable from lead, silver, and gold refining wastes. Only copper mining is of significant quantity to be used as a practical source for tellurium. Additional demands for tellurium above that available from these sources will not affect these mining activities. The availability of tellurium is thus closely related to copper industry demands.

The total production of tellurium from these sources amounts to about 363,000 kg (800,000 lb) of usable refined material per year. This amount has been extremely stable, and the level of production is not affected significantly by anything known to date. TE and other semiconductor producers consume about ten percent of this production, with the remainder of the production consumed by steel and other metals industries as well as other minor chemical processes. The ten percent consumed for semiconductors is largely for TE material production. The tellurium used for TE production must be further refined to increase its purity. The other 90 percent is consumed as is from the initial refining process.

If the maximum number of TE modules was limited to the ten percent of available tellurium, the number of automotive TE air conditioning units that could be built would be restricted. At this time, it appears that about 400 standard TE modules would be required per vehicle if the TE air conditioner were fabricated to provide the same level of cooling as current vapor-cycle AC units. If all eight million modules fabricated were exclusively available for this effort, about 20,000 vehicles could be equipped with TE air conditioners per year.

Conclusions

Two key issues were addressed in this paper. It was found that a solar energy assisted TE air conditioner, delivering a total cooling of 4.01 kW (13,680 Btu/h), will require a total d-c power of about 9.25 kW, which is equivalent to a cooling COP of 0.43. This power also represents about ten percent of the total power of the engine. The power consumption of TE air conditioners seems higher than that of conventional CFC-based air conditioning systems. The power generated by the rooftop PV cells can only improve the overall COPs by about two percent. However, the real contribution of the PV generated power is that it can be used to reduce the car interior temperature and thus the peak car air conditioning load. Because over 50 percent of driving lasts less than 20 minutes, a reduction of vehicle peak air condition load will be very helpful. This concept is particularly important for electric vehicles, which have limited power supply, because a reduction in peak air conditioning load means a longer driving range.

One the material side, it was found that one of the elements used to make TE modules—tellurium—will have availability problems. Current production of TE modules can be increased by about ten times without a price impact. Beyond that point, the cost of TE modules will be dependent on the quantity.

Wu (1992) has presented a TE heat pump model which indicated that designs of TE systems should be based on a minimum heat exchanger area per unit load or a minimum cost per unit load. This concept could lead us to realistic TE system designs. However, the design of TE cooling system discussed in this paper is very conservative to reflect what can be achieved with the current technology. The COP could be around 0.7 to 1.0 or higher as shown by Stockholm and Schlicklin (1989). Also, the power consumption for TE cooling drops exponentially when cooling capacity requirement decreases such as during part load conditions which are the major part of operation.

Currently, a 20 percent performance improvement in TE materials has already been obtained in industrial size samples, which means almost directly a 20 percent increase in cooling capacity or a 20 percent reduction in power consumption. Besides, TE systems do not contain any CFC-type fluids, they are very rugged and reliable, and they require almost no maintenance, they are also very flexible in operation. All these advantages together with further improvement of TE materials with a possible Figure-of-Merit of $3.4 \times 10^{-3} K^{-1}$, TE cooling could become a viable alternative to CFC or HFC vapor compression technology. With refrigerant R-134a, the replacement of R-12 for mobile air conditioning still faces future regulatory uncertainty due to its global warming potential. TE mobile air conditioning systems could very possibly become a reality.

Acknowledgments

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