A THERMOELECTRIC COOLING/HEATING SYSTEM
FOR A HOSPITAL THERAPY PAD

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INTRODUCTION

A new cooling and heating unit has recently been developed for use with hospital therapy pads. These units employ thermoelectric (TE), or solid state, heat pumps providing significant design improvements over existing compressor driven freon systems for cooling and resistance heaters for heating.

Thermodynamically reversible solid state heat pumps are used to cool and heat a heat exchanger in a closed loop liquid system in this application. The resulting circulating liquid is used to temperature control a topically applied hospital therapy pad.

Design advantages of thermoelectrics are derived from the small size and weight and the inherent reliability of the no-moving-parts solid state configuration. However, the sensitivity of TE cooling efficiency to design variables necessitated a careful system integration and optimization before a practical unit could be derived.

SYSTEM DESCRIPTION

A block diagram depicting the operation of the entire heating and cooling system is shown in Figure 1. The heart of the system is the thermoelectric sub-assembly heat exchanger unit composed of a liquid exchanger, TE cooler, air exchanger and fan. The design optimization of this sub-assembly is the subject of this paper and will be discussed in detail later.

Basically, this unit cools or heats a liquid which is circulated through a cold/hot therapy pad by a pump. The temperature of the therapy pad is selected by the operator and controlled by sensing the temperature of the liquid and feeding back to the electronic temperature controller/power supply. The temperature controller/power supply provides and regulates the correct polarity of direct current power to the TE heat exchanger assembly to maintain the selected temperature.

Discussions to follow focus on the cooling aspect of the system. This is thermally much more difficult and thus is the overriding design criteria. Nevertheless, it should be remembered that heating is also intended and achievable by simply reversing the polarity of applied power.

THERMOELECTRIC COOLER THEORY OF OPERATION

The critical element of the system is the TE cooler schematically shown in Figure 2. TE coolers are actually small heat pumps which operate on physical principles established over a century ago. As electrical current is applied the resulting motion of electronic carriers transport heat via the Peltier effect. N and P-type semiconductor materials are connected electrically in series and thermally in parallel so that the carrier movement in both legs are away from the isolated cold surface. Both electric power connections are made at the hot junctions and thus do not contribute to the thermal heat loading of the cold surface.

TE coolers obey the laws of thermodynamics in that thermal energy flows from hot to cold. Indeed, conduction, convection, joule and radiation heat loads each tend to degrade the cooling effect. Consequently, system optimization as well as the TE material optimization is predicated on minimizing these effects.

Although the other effects are roughly linear, joule heating varies as the square of applied current and thus dominates at high currents. Consequently, there exists a maximum cooling effect at some applied power for each TE system. Cooling performance simply degrades as applied power is increased above this "maximum".

Similarly, there exists an optimum power level for each achievable cold side temperature for which the maximum net heat is pumped from the cold surface per unit of applied power. This condition is referred to as the maximum Coefficient of Performance (COP) defined by the dimensionless ratio of heat pumped ÷ applied power. The input power for maximum COP is below the "maximum" power for small ΔT's but approaches this power level as ΔT is increased. Generally, the maximum COP condition is applied in thermoelectric systems design to not only minimize power for the specified cooling requirement but to minimize size, complexity, and cost of heat sinking.

The energy transported from the cold junction to the hot junction by the electronic carriers will naturally tend to flow back to the cold junction. A means of heat sinking or dissipating the energy at the hot junction is imperative. In short, the cooler does not eliminate heat but simply moves it to a heat sink where it can be dissipated via normal thermodynamic effects.

The heat sink design is of equal importance with the design of the TE cooler. Ideally, the heat sink should carry heat away with a minimum of temperature rise. Practically, this must be weighed against the size, weight, cost and complexity. The same philosophy applies to the cold side if fluid cooling/heating is anticipated.

Thermal connection at both hot and cold sides is also a very important factor in system design. A typical system thermal profile is illustrated in Figure 3. The important conclusion is that the ΔT of the system is always less than the ΔT across the TE pellets to varying degrees depending on magnitude of the ΔT's, heat load, hot and cold sink effectiveness, and the many thermal interfaces involved.

It is important to minimize these "losses" in order to minimize the ΔT across the TE pellets. This is because the maximum COP is extremely sensitive to the pellet ΔT as illustrated in Figure 4. Improvements in system COP will manifest itself in improved performance and/or cost.

Finally, as implied by Figure 4, TE devices are also effective heating units. Indeed, systems with heating mode COP's above 1.0 are more efficient than simple electrical resistance heating. This "added" energy is a consequence of the Peltier heat pumping extracting energy from ambient and depositing it plus the input power to the item to be heated. The relative difference in cooling versus heating COP illustrates why system design is almost always dominated by the cooling requirement.

SYSTEM DESIGN

System design optimization consisted of minimizing the cold sink resistance (CSR), and heat sink resistance (HSR), while maximizing the thermoelectric COP.

The most important performance requirement of the system was to produce low liquid outlet temperatures and rapid cool-down. Specifically, the goal was to cool the pad to 7.2°C from an initial pad temperature of 40.6°C within 15 minutes. A preliminary study of the interaction of these variables was made by parametrically varying CSR and HSR using the thermoelectric COP for each case. The results are illustrated in Figures 5 and 6. These figures not only illustrate the relative sensitivity of system variables but also illustrate where design effort should be concentrated. That is, optimization of HSR is much more critical than CSR. In fact, the cold sink was designed primarily from a mechanical standpoint yielding a calculated value of 0.02°C/watt for CSR. A more detailed discussion for optimization of HSR follows and illustrates the general optimization procedure for forced fluid heat sinks.

HEAT SINK OPTIMIZATION

The design of the heat sink began with the selection of a fan. Heat sink designs were developed for several different fans selections before the ideal combination of cost, heat sink performance, and noise level was achieved. Next, the fan curve or characteristic flow versus pressure drop was plotted. This curve is typified by high flow rate at zero pressure to zero flow rate at some high pressure drop (See Figure 7).

The next step was to pick a starting mechanical design for the air heat exchanger. This initial design was influenced by size restrictions, material availability and past experience. The pressure drop was subsequently calculated for sequential steps in flow rate defining the characteristic curve for that particular heat exchanger. This curve starts at the origin and monotonically increases. Necessarily, then, it intersected with the fan curve representing the operating point of the fan in combination with the air heat exchanger. This condition is illustrated in Figure 7.

Optimization of the heat sink was achieved by successively optimizing each variable of the air heat exchanger. For example, as the fin density was increased the added fins provided more heat sinking surface area but the restriction produced higher pressure drop diminishing air flow. Consequently, a new characteristic curve was established and a new intersection point defined. This process was applied sequentially, each case yielding a unique value for HSR. These intersection points are indicated in Figure 7 and summarized in Figure 8. It is clear from this graph that the optimum fin density was 10 fins/inch, yielding a HSR of 0.065°C/watt.

The curve in Figure 8 is actually the culmination of many such curves generated by sequential steps in other air heat exchanger variables, such as fin height, fin thickness, and fin length. In fact, most of these variables were interactive and numerous iterative steps were required.

THERMOELECTRIC OPTIMIZATION

The final step in the design process was to quantify the cold side heat load which, in combination with CSR, yielded the TE cold side temperature. The optimum thermoelectric COP and corresponding heat delivered to the hot side was parametrically calculated with hot side temperature. This data is illustrated in Figure 9. HSR is plotted together with the TE performance. The design point is the intersection where the heat sink dissipation is exactly matched by the net heat delivered by the TE cooler.

The optimized TE cooler design parameters associated with this design point were used to define TE cooler geometry and DC power supply requirements. Some concessions were made in order to accommodate cost effective final TE cooler designs. This was accomplished, however, with minimum deviation from true optimum. The impact of these practical design modifications was to move the design point in Figure 9 slightly higher on the HSR curve.

THERMAL MODELLING

A thermal model was generated using data derived from details of the system final design. This model is illustrated in Figure 10. Each important node, or significant temperature point, of the system is indicated as a circle. The thermal mass of these key nodes was calculated and stored in the computer. The major thermal interactions between the nodes are represented by thermal resistors. The thermal resistance of each branch indicated in the network was calculated and stored in the computer. These calculations represented the essential elements of the thermal model in the computer software.

Transient performance calculations were subsequently generated by selecting initial thermal conditions and applied power to the thermoelectrics. This data was used to predict and analyze system performance under various environmental conditions. Calculations were made to not only verify design point compliance, but also to establish performance conditions for room temperature acceptance tests.

EXPERIMENTAL RESULTS

The TE coolers, cold sinks, and heat sinks were fabricated and the thermoelectric heat exchanger system was assembled. Water lines, pump, reservoir, and therapy pad were installed to simulate the final assembly. Test data was collected from the room temperature acceptance test and compared with calculations derived from thermal modelling. The results are shown in Figure 11. The excellent agreement observed in this graph was also consistent with subsequent tests performed on the completed prototype unit.

CONCLUSIONS

The design of the cold sink, heat sink, and thermoelectric cooler is a relatively complicated, iterative process involving optimization of several very sensitive, interactive variables. However, accurate modelling and computerized calculations methods have been developed and applied yielding effective water cooling and heating sub-systems.

This process has been applied to the design of a thermoelectric unit for use in a system for cooling and heating topically applied hospital pads. The resultant was the creation of a very effective and unique product offering all the advantages of thermoelectric systems; small size, weight, reliable and quiet operation.

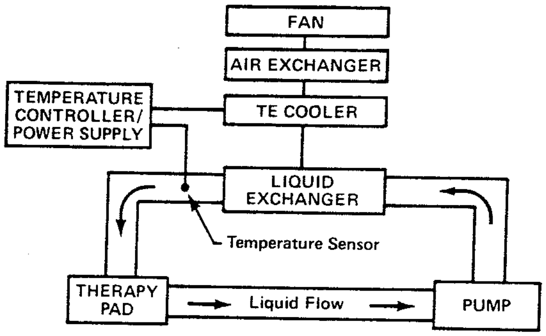

Figure 1 System Block Diagram

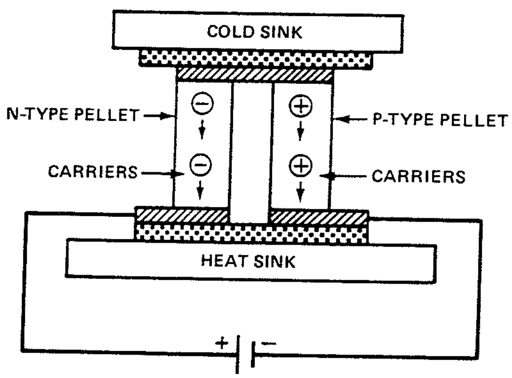

Figure 2 Thermoelectric Cooler Configuration

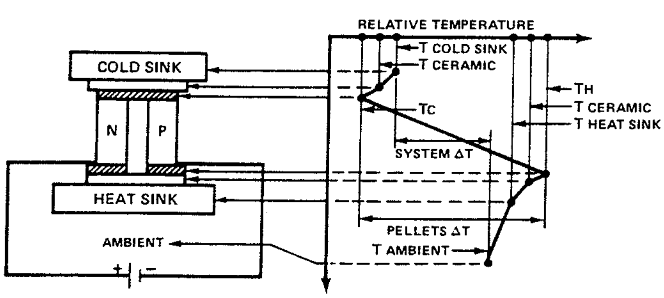

Figure 3 Thermoelectric System Thermal Profile

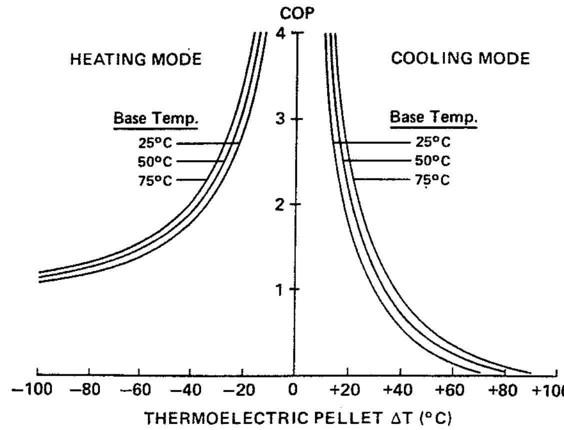

Figure 4 Thermoelectric Coefficient of Performance

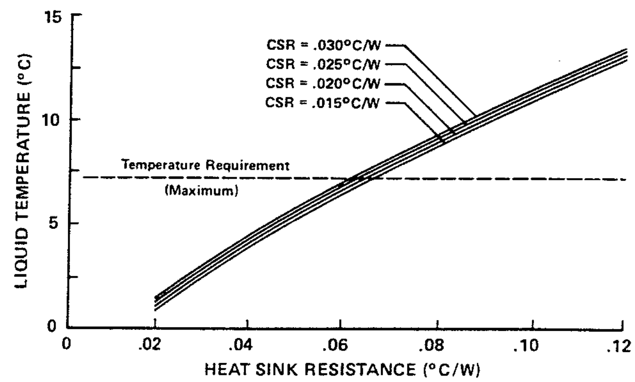

Figure 5 Steady-State Performance

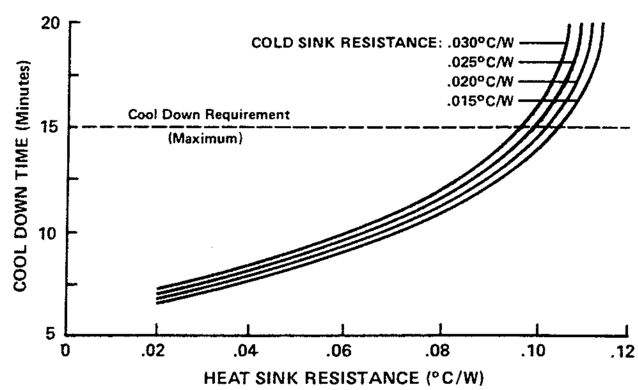

Figure 6 System Cool Down Time

$T_{initial} = 40.5°C$
$T_{final} = 7.22°C$

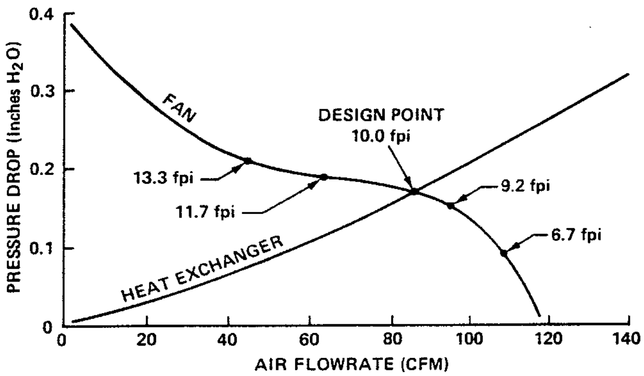

Figure 7 Heat Exchanger/Fan Integration

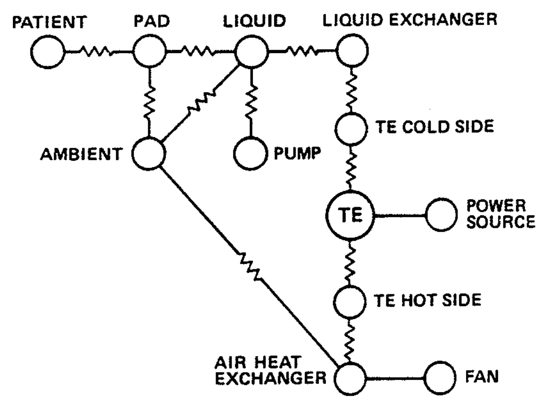

Figure 10 System Thermal Computer Model

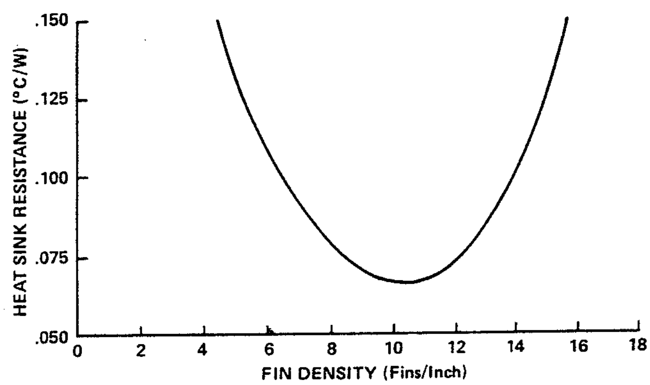

Figure 8 Fin Density Optimization

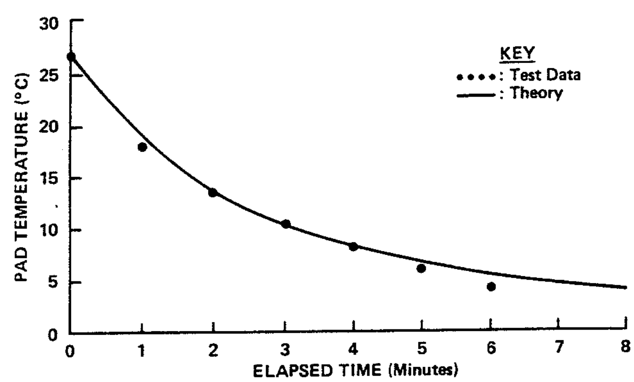

Figure 11 Thermoelectric Heat Exchanger Performance

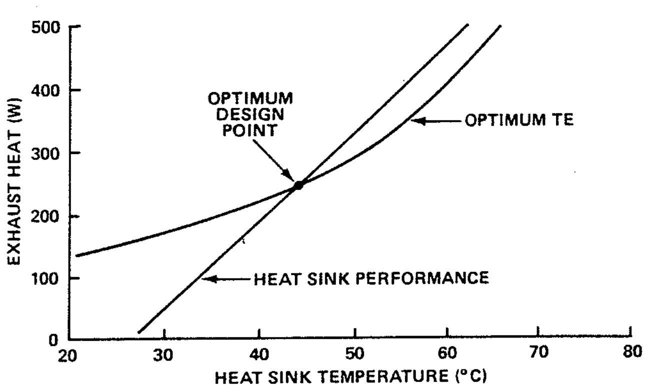

Figure 9 Thermoelectric/Heat Sink Optimization