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# LOCAL ENTROPY GENERATION ANALYSIS OF A ROTARY MAGNETIC HEAT PUMP REGENERATOR

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## ABSTRACT

The rotary magnetic heat pump has attractive thermodynamic performance but it is strongly influenced by the effectiveness of the regenerator. This study uses local entropy generation analysis to evaluate the regenerator design and to suggest design improvements. The results show that performance of the proposed design is dominated by heat transfer related entropy generation. This suggests that enhancement concepts that improve heat transfer should be considered, even if the enhancement causes a significant increase in viscous losses (pressure drop). One enhancement technique, the use of flow disrupters, was evaluated and the results showed that flow disrupters can significantly reduce thermodynamic losses.

## NOMENCLATURE

| | |
|---|---|
| A | flow area, $m^2$ |
| C | ratio of thermal capacities ($\dot{m}_{mm}Cp_{mm}/\dot{m}_{rf}/Cp_{rf}$) |
| Cp | specific heat, J/K kg |
| d | flow path width (m) |
| D | hydraulic diameter, m |
| $\mu$ | effective viscosity, Pa s |
| f | friction factor |
| h | disruptor height (m) |
| k | thermal conductivity, W/K(m) |
| L | regenerator length (m) |
| $\dot{m}$ | mass flow rate, kg/s(m) |
| N | augmentation entropy generation number |
| $\dot{q}_o$ | surface heat flux, $W/m^2$ |
| r | disruptor spacing (m) |

(a) Operated for the U.S. Department of Energy by Battelle Memorial Institute under Contract DE-AC06-76RLO 1830.

| | |
|---|---|
| $R_e$ | Reynolds number based on flow path width |
| $S_{total}$ | total entropy generation rate, W/K($m^2$) |
| $S_{Thmm}$ | entropy generation rate caused by thermal conduction in the magnetic material, W/K($m^2$) |
| $S_{Thrf}$ | entropy generation rate caused by heat transfer between magnetic material and the regenerator fluid, W/K($m^2$) |
| $S_{vis}$ | entropy generation rate caused by viscous dissipation, W/K($m^2$) |
| $S_{Lostw}$ | entropy generation rate due to residual losses, W/K($m^2$) |
| St | Stanton number |
| t | thickness of an individual sheet of magnetic material (m) |
| T | temperature, K |
| $T_o$ | reference temperature, K |
| $t_d$ | thickness of an individual disruptor |
| U, V, W | x, y, z direction velocities |
| $\phi$ | irreversibility distribution ratio |
| $\rho$ | density (kg/$m^3$) |
| Subscript | |
| e | related to performance with enhancement |
| p | related to performance without enhancement |
| mm | related to magnetic materials |
| rf | related to regenerator fluid |

## INTRODUCTION

Barclay (1987) and Kirol et al (1984) have shown that the magnetic heat pump has attractive thermodynamic performance but its performance is strongly influenced by the effectiveness of the regenerator. The work reported in this paper uses local entropy generation analysis to evaluate a regenerator design proposed by Kirol et al. (1984) for a rotary magnetic heat pump.

A magnetic heat pump uses the entropy changes associated with the magnetization and demagnetization of a magnetic material to transfer thermal energy from a low-temperature reservoir to a high-temperature reservoir. A number of heat pump configurations have been investigated, but Kirol et al. (1984) concluded that a rotary design appears to be most attractive. In the

rotary heat pump design, thin ring-shaped sheets of magnetic material are rotated through fixed magnetic fields. A schematic arrangement of the magnetic heat pump is shown in Figure 1. A regenerator fluid (such as water) is pumped in the direction opposite to the rotation of the magnetic material because efficient operation of the heat pump requires regeneration. When the material enters a magnetic field (state point 2 on Figure 1), the temperature of the material increases and a fraction of the thermal energy is extracted ("heat to high temperature sink" on Figure 1); the balance is used to heat the regenerator fluid (process occurring from state point 2 to 3 on Figure 1). When the material is demagnetized, its temperature drops (state point 3) and heat is added to the magnetic material from both the low-temperature reservoir ("heat from low temperature source") and the regenerator fluid (process occurring from state point 4 to 1). A more detailed description of the concept is presented in Kirol et al. (1984).

Regeneration occurs at two locations in the cycle: between state point 2 and 3, hot magnetic material is used to heat regenerator fluid while the regenerator fluid is used to heat the magnetic material between state points 4 and 1. Studies of the magnetic heat pump concept show that an efficient design must have a high performance regenerator. This has led researchers to apply advanced analytical techniques, such as second law analysis, to the evaluation of the regenerator. Several researchers have used a lumped parameter second law analysis to investigate a magnetic heat pump regenerator (Kirol et al. 1984; Barclay 1984; and Barclay and Sarangi 1984) and the results have been used to identify optimum regenerator characteristics, resulting in a design that minimizes second law losses.

The research reported in this paper involves the application of local entropy generation analysis to a rotary magnetic heat pump regenerator design described in Kirol et al. (1984). Local entropy generation analysis determines the local generation of entropy in solids and convecting fluids based on the temperature and flow fields (as predicted by a computational fluid dynamics simulation code). This approach allows the explicit investigation of trade-offs between sources of entropy generation and design optimization (White and Drost 1989).

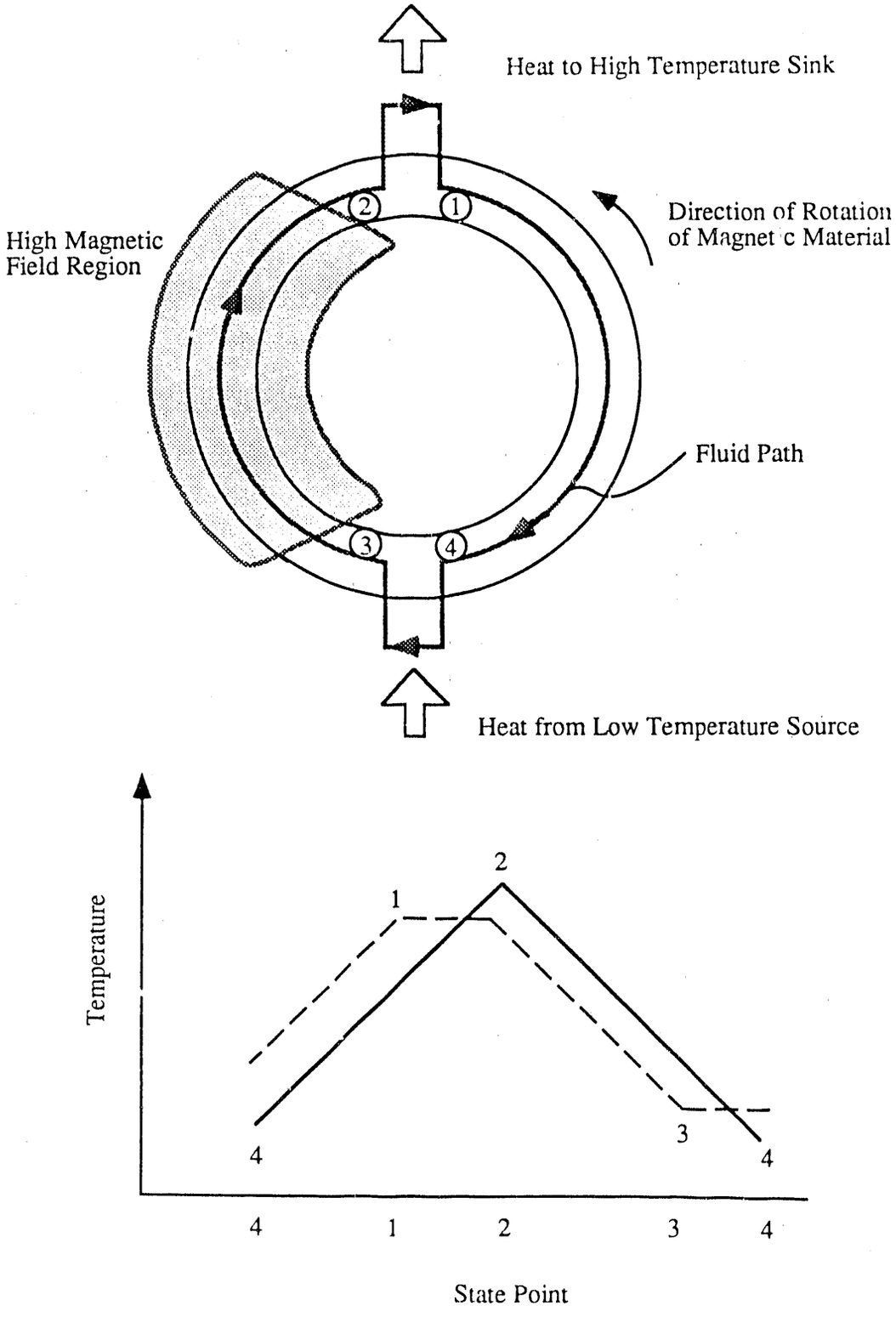


FIGURE 1. Rotary Magnetic Heat Pump (Kirol et al. 1984)

## OBJECTIVE

The research included in this study of local entropy generation in a rotary magnetic heat pump regenerator has two objectives.

1. Demonstrate the usefulness of local entropy generation analysis in the design of thermal system components.

2. Identify design modifications that improve the performance of the rotary magnetic heat pump regenerator.

## METHODOLOGY

One goal of second law analysis is to identify and minimize the thermodynamic losses (irreversibilities) associated with a process or component. It can be shown that the minimization of entropy generation results in irreversibility minimization (Bejan 1982). For a convecting isotropic and Newtonian fluid, entropy generation is given by Equation (1) for three-dimensional rectangular coordinates (Bejan 1982).

$$\dot{S}''_{gen} = \dot{S}''_{Th} + \dot{S}''_{vis} \tag{1}$$

where

$$\dot{S}''_{Th} = \frac{k}{T^2}\left[\left(\frac{2T}{2x}\right)^2 + \left(\frac{2T}{2y}\right)^2 + \left(\frac{2T}{2z}\right)^2\right] \tag{2}$$

$$\dot{S}''_{vis} = \frac{\mu}{T}\left\{2\left[\left(\frac{2u}{2x}\right)^2 + \left(\frac{2v}{2y}\right)^2 + \left(\frac{2w}{2z}\right)^2\right] + \left(\frac{2u}{2y} + \frac{2v}{2x}\right)^2 + \left(\frac{2u}{2z} + \frac{2w}{2x}\right)^2 + \left(\frac{2v}{2z} + \frac{2w}{2y}\right)^2\right\} \tag{3}$$

Equation (2) is entropy generation caused by heat transfer across a finite temperature difference, whereas Equation (3) is the local entropy generation caused by viscous dissipation. When analytical expressions are available for the temperature and velocity fields, Equation (1) can be evaluated to yield the

local entropy generation. Total entropy generation can be determined by integrating Equation (1) over the region of interest. Most practical problems are sufficiently complex that analytical solutions do not exist. Alternatively, a computational fluid dynamics (CFD) computer code can be used to predict the velocity and temperature fields. This information can then be used to numerically calculate entropy generation. The details of incorporating Equation (1) in an existing CFD code and the subsequent benchmarking are reported in White and Drost (1989).

The analysis of the magnetic heat pump regenerator consisted of two evaluations. First, local entropy generation in a typical flow path in the regenerator was evaluated using the TEMPEST CFD code (Trent, Eyler and Budden 1989) to predict velocity and temperature fields. Local entropy generation was then integrated over the flow path, resulting in an estimate of total entropy generation for that specific regenerator design. Subsequently, integrated local entropy generation was used as the figure-of-merit for parametric evaluations of design variations. The second evaluation concentrated on investigating the use of flow disrupters for heat transfer enhancement. In this case, local entropy generation was evaluated for a small section of the regenerator including several flow disrupters.

While this study focused on the minimization of thermodynamic losses rather than using an economic figure-of-merit, local entropy generation is also the source of local cost generation. If the cost of the various sources of entropy generation was available, it is possible to determine local cost generation and the optimum regenerator configuration based on minimum cost generation.

## Parametric Analysis

The parametric analysis was conducted by calculating integrated local entropy generation for a typical flow path in the regenerator during one phase of the regeneration cycle during which heat is transferred from the hot magnetic material to the regenerator fluid. The regenerator flow path model is shown in Figure 2. The regenerator flow path consists of two sheets of magnetic material separated by the regenerator fluid. Hot magnetic material

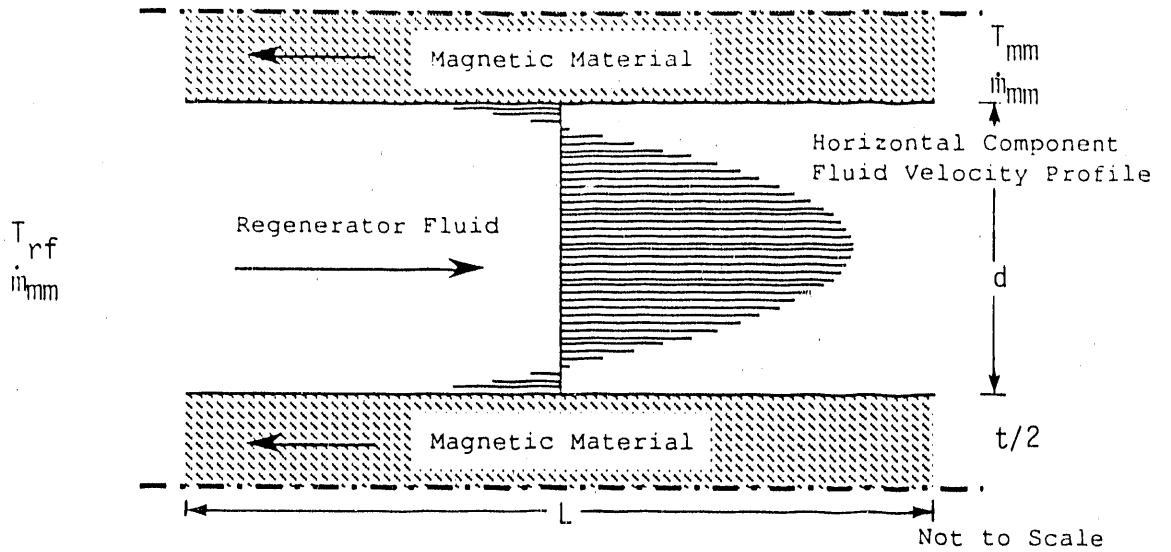


FIGURE 2. Regenerator Flow Path Model

enters the regenerator and is assumed to be moving to the left. Cool regenerator fluid enters the regenerator and moves to the right while thermal energy is transferred from the magnetic material to the regenerator fluid. The incoming regenerator fluid was assumed to have a constant inlet velocity profile. Figure 2 shows the fully developed velocity profile that would be obtained as the regenerator fluid flows through the regenerator.

The regeneration process involves four thermodynamic losses: 1) entropy generation caused by heat transfer across the finite temperature difference between the magnetic material temperature and the regenerator fluid, 2) entropy generation caused by conduction of thermal energy along the magnetic material, 3) entropy generation caused by viscous dissipation in the regenerator fluid, and 4) residual losses (losses associated with the incomplete cooling of the magnetic material).

The fourth loss mechanism assigns a penalty for a regenerator design that does not extract all of the available thermal energy in the magnetic material. This loss can also be viewed as a source of entropy generation if we consider the fate of the thermal energy remaining in the magnetic material after it leaves the regenerator. The entropy generation can be calculated if it is assumed that the flow of the heat source fluid is sufficiently high to cool the magnetic material to the temperature of the regenerator fluid as it

enters the regenerator, in which case the entropy generation associated with residual losses is given by Equation (4).

$$S_{Lostw} = \dot{m}Cp(T - T_o) - T_o\dot{m}\ Cp\ \ln \frac{T}{T_o} \tag{4}$$

Parametric investigations were conducted by selecting a set of design parameters and using the TEMPEST CFD code to determine the temperature and velocity profiles in the regenerator fluid, the temperature field in the magnetic material, the exiting temperature of the magnetic material, and the local entropy generation. The regenerator was a particularly attractive application for the numerical evaluation of local entropy generation because most simulations involved laminar flow. This eliminated concerns about appropriate turbulence modeling and modeling turbulence generation near solid boundaries. Some flow disrupter simulations probably involved turbulent flow, but we assumed laminar flow for all cases because this assumption would result in a conservative estimate for heat transfer enhancement. Therefore, all cases were modeled as being laminar.

The total entropy generation for a specific design was determined by integrating the local entropy generation over the magnetic material and regenerator fluid and then adding the integrated local entropy generation to the entropy generation associated with remanent losses. This resulted in one figure-of-merit that included the impact of the four relevant thermodynamic loss mechanisms described above.

The base case regenerator geometry and bulk flow characteristics were taken from the optimum design, as identified by Kirol et al. (1984). $Y_2(Fe_xCo_{1-x})_{17}$ was used for the magnetic material and water was assumed to be the regenerator fluid. Material properties for the magnetic material are presented in Kirol et al. (1984). Key base case parameters are defined on Figure 2 and are presented below.

- regenerator length (L) = 0.28 m

- magnetic material thickness (t) = $2.6 \times 10^{-4}$ m

- flow path width (d) = $2.6 \times 10^{-4}$ m

- magnetic material inlet temperature ($T_{mm}$) = 160°C

- regenerator fluid inlet temperature ($T_{rf}$) = 104°C

- magnetic material mass flow rate ($\dot{m}_{mm}$) = 1.466 Kg/s(m)

- regenerator mass flow rate ($\dot{m}_{rf}$) = 0.1992 Kg/s(m)

The parametric study investigated the impact of three design parameters on regenerator performance.

- Relative velocity between the regenerator fluid and the magnetic material: Entropy generation and regenerator capacity will be a strong function of the relative velocity between the fluid and the magnetic working material. Usually, entropy generation and the capacity of the regenerator will increase monotonically with velocity because increased velocity will result in increased heat transfer (and increased entropy generation caused by heat transfer).

- Length of the flow path: Entropy generation can be determined as a function of the length of a flow path. As the length of the flow path increases, heat transfer entropy generation and remanent losses will decrease, while viscous losses increase. Consequently, there will be one regenerator length that results in minimum entropy generation.

- Ratio of thermal capacity: The regenerator can be viewed as a counterflow heat exchanger. As with any counterflow heat exchanger, the ratio of the thermal capacities (the product of mass flow rate and specific heat) of the two fluids will affect entropy generation. Based on earlier evaluations of entropy generation in counterflow heat exchangers, it was anticipated that balanced flow would result in minimum entropy generation (Bejan 1982).

## Evaluation of the Flow Disrupter Concept

The results of parametric evaluations showed that the base case design for the regenerator was not the optimum based on thermodynamic performance. The thermal losses were several orders of magnitude larger then the viscous losses. This suggested that the design should include enhancement techniques that improve heat transfer, even at the expense of a substantial increase in viscous dissipation losses. The inclusion of enhancement techniques is difficult because the magnetic material moves past seals as it rotates inside of the heat pump casing. The seals prevent the use of typical enhancement techniques, such as surface roughening. An alternative approach is to use a stationary flow disrupter located between the plates of magnetic material and attached to the casing. In this case, both the magnetic material plates and the regenerator fluid move past the stationary flow disrupter. The flow disrupter is shown in Figure 3.

Optimization of the flow disrupter design involved the trade-off of several sources of entropy generation. Local entropy generation analysis was used to evaluate the sources and distribution of entropy generation associated with the disrupter. It was anticipated that a number of flow disrupters would be located along the regenerator but a tractable simulation required that

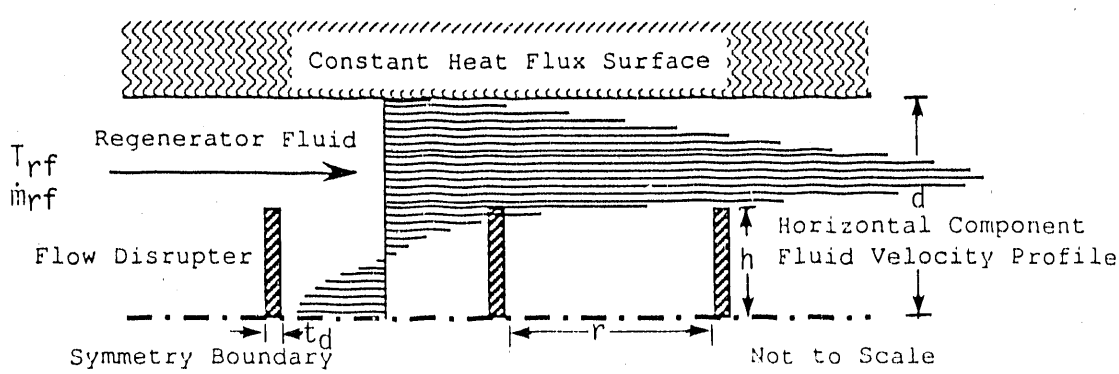


FIGURE 3. Flow Disrupter

only a small segment of the regenerator be evaluated. Figure 3 shows the model used for numerical simulation. The magnetic material was not included in the simulation; instead a constant heat flux boundary condition was imposed, with the heat flux being representative of the heat fluxes encountered in the parametric studies described above. The inlet velocity was the fully developed velocity profile obtained for a numerical simulation of an infinite series of flow disruptors. As shown in Figure 3, three disrupters were included in the simulation. The major design variable included in the study was disrupter height. The impact of disrupter spacing or shape on the device's thermodynamic performance could also be investigated but these evaluations were beyond the scope of the study. The geometric and flow characteristics of the disrupter model include:

- flow path width (d) = $2.6 \times 10^{-4}$ m
- disrupter thickness (t) = 5 $\mu$m
- disrupter spacing (r) = 150 $\mu$m
- disrupter height (h) = variable
- inlet water mass flow rate ($\dot{m}_{rf}$) = 75 gm/s(m)
- inlet water temperature ($T_{rf}$) = 130°C
- wall heat flux ($\dot{q}_w$) = $6.675 \times 10^5$ W/m$^2$

## RESULTS

The parametric evaluations consisted of three sensitivity studies. Table 1 presents the results of varying the length of the regenerator while holding inlet temperatures, ratio of $\dot{m}Cp$, and magnetic material and fluid velocities constant.

TABLE 1. Integrated Entropy Generation as a Function of Regenerator Length

| Case | Regenerator Length, m | C | Re | $S_{Vis}$, w/K | $S_{Thm}$, w/K | $S_{Thrf}$, w/K | $S_{Lostw}$, w/K | $S_{Total}$, w/K |
|---|---|---|---|---|---|---|---|---|
| 1 (Base) | 0.28 | 1 | 915 | 0.011 | 0.394 | 0.958 | 0.158 | 1.51 |
| 2 | 1.[illegible] | 1 | 915 | 0.0539 | 0.1040 | 0.250 | 0.024 | 0.431 |
| 3 | 2.80 | 1 | 915 | 0.108 | 0.0541 | 0.130 | 0.023 | 0.315 |
| 4 | 3.80 | 1 | 915 | 0.147 | 0.0410 | 0.0981 | 0.008 | 0.294 |
| 5 | 8.80 | 1 | 915 | 0.337 | 0.0197 | 0.0472 | 0.003 | 0.407 |
| 6 | 0.028 | 1 | 915 | 0.001 | 0.728 | 1.66 | 1.85 | 4.24 |

A review of Table 1 shows several interesting results. First, the base case design is far from optimum. The optimum regenerator length, based on minimum entropy generation, is approximately 3.8 m as compared to the 0.28 m length used in the base case. This does not necessarily imply that the base case design is a poor selection because the use of a longer regenerator would substantially increase the size (and cost) of the magnetic heat pump. However, significant improvements in performance could be obtained by lengthening the regenerator.

Assuming that the physical dimensions of the regenerator are limited to the base case length of 0.28 m, the results show that the viscous losses are approximately 1% of the thermal losses. This suggests that enhancements that reduce heat transfer related entropy generation should be considered even if they result in a substantial increase in viscous losses. As an example, an enhancement technique that reduces heat transfer losses by 10% could cause a 1000% increase in viscous losses and still result in improved thermodynamic performance. The low viscous losses also indicate that the designer need not consider techniques for reducing viscosity related losses. An example would be the use of viscosity reducing additives. The benefit of using an additive would be insignificantly small compared to the thermal losses.

Finally, the results show that entropy generation caused by conduction in the magnetic material is a significant fraction of the total losses. Kirol et al. (1984) concluded that conduction losses could be ignored. Our results suggest that this is not the case. Conduction entropy generation contributes between 5% and 25% of the total entropy generation.

Table 2 presents the results of varying C, the ratio of thermal capacities ($\dot{m}_{mm}\ Cp_{mm}/\dot{m}_{rf}\ Cp_{rf}$) while holding inlet temperatures, regenerator length, and magnetic material and fluid velocities constant.

TABLE 2. Integrated Entropy Generation as a Function of the Ratio of $\dot{m}Cp$

| Case | Regenerator Length, m | C | Re | $S_{vis}$, w/K | $S_{Thmm}$, w/K | $S_{Thrf}$, w/K | $S_{Lostw}$, w/K | $S_{Total}$, w/K |
|---|---|---|---|---|---|---|---|---|
| 1 | 0.28 | 0.25 | 2410 | 0.054 | 1.38 | 2.15 | 0.00 | 3.59 |
| 2 | 0.28 | 0.50 | 1413 | 0.019 | 0.830 | 1.67 | 0.007 | 2.53 |
| 3 | 0.28 | 0.75 | 1081 | 0.013 | 0.537 | 1.20 | 0.045 | 1.80 |
| 4 (Base) | 0.28 | 1.00 | 915 | 0.011 | 0.394 | 0.958 | 0.158 | 1.52 |
| 5 | 0.28 | 2 | 666 | 0.008 | 0.253 | 0.778 | 1.15 | 2.19 |
| 6 | 0.28 | 4 | 541 | 0.008 | 0.122 | 0.463 | 2.46 | 3.05 |
| 7 | 0.28 | 8 | 479 | 0.008 | 0.043 | 0.191 | 3.30 | 3.54 |

A review of Table 2 emphasized the importance of maintaining balanced flow (when $\dot{m}Cp$ of the fluid equals $\dot{m}Cp$ of the magnetic material) in the regenerator. A small flow imbalance (C = 0.75) results in an 18% increase in total entropy generation. A larger flow imbalance (C = 0.5) can increase total entropy generation by 66%. Earlier magnetic heat pump regenerator design studies did not identify the importance of maintaining balanced flow in the regenerator. This will impose another design constraint on the concept.

Table 3 presents the results of varying the velocity of the magnetic material and regenerator fluid while holding inlet temperatures, ratio of $\dot{m}Cp$, and regenerator length constant.

TABLE 3. Integrated Energy Generation as a Function of Regenerator Length

| Case | Regenerator Length, m | C | Re | $S_{vis}$, w/K | $S_{Thmm}$, w/K | $S_{Thrf}$, w/K | $S_{Lostw}$, w/K | $S_{Total}$, w/K |
|---|---|---|---|---|---|---|---|---|
| 1 | 0.28 | 1 | 250 | 0.00 | 0.038 | 0.091 | 0.009 | 0.138 |
| 2 | 0.28 | 1 | 500 | 0.00 | 0.138 | 0.332 | 0.047 | 0.517 |
| 3 | 0.28 | 1 | 915 | 0.011 | 0.394 | 0.958 | 0.158 | 1.52 |
| 4 | 0.28 | 1 | 2000 | 0.051 | 1.33 | 3.26 | 0.940 | 5.58 |

The results presented in Table 3 show that entropy generation increases monotonically with increasing Reynolds number. This was expected because increasing Reynolds number while the ratio of $\dot{m}Cp$ is held constant results in an increased velocity for the magnetic material. This increases the heat transferred in the regenerator. Therefore, the magnitude of $S_{Thmm}$ will be the result of two competing effects; the large increase in heat transfer will increase entropy generation while a small decrease in entropy generation will result from the velocity related increase in the convective heat transfer coefficient. The net result is a monotonic increase in $S_{Thmm}$.

The evaluation of the flow disrupter consisted of two studies. First, a simple flow disrupter was evaluated by determining the integrated local entropy generation as a function of the fraction of flow area blocked by the disrupter. The second evaluation investigated the distribution of local entropy generation to determine if the simple flow disrupter geometry could be improved.

The results of the evaluation of integrated local entropy generation for a simple flow disrupter design are presented in Table 4. While the fraction of the flow area blocked by the disrupter was allowed to vary, the mass flow rate of the regenerator fluid (assumed to be water) and the wall heat flux were held constant.

TABLE 4. Integrated Local Entropy Generation for a Simple Flow Disrupter Design as a Function of Blocked Flow Area

| Case | Flow Area Blocked by Disrupter, % | $S_{Vis}$, w/K x $10^{-3}$ | $S_{Thrf}$, w/K x $10^{-3}$ | $S_{Total}$, w/K x $10^{-3}$ | $S_{Total}$ Near Wall, % |
|---|---|---|---|---|---|
| 1 | 0 | 0.004 | 3.939 | 3.943 | 33 |
| 2 | 25 | 0.045 | 3.50 | 3.545 | 37 |
| 3 | 50 | 0.092 | 2.350 | 2.442 | 49 |
| 0 | 75 | 0.461 | 0.892 | 1.353 | 72 |

Table 4 shows that the disrupter achieves the desired reduction in total entropy generation. When 75% of the flow area is blocked, the integrated local entropy generation is approximately 1/3 of that encountered without a disrupter (case 1). The improvement was achieved by a 78% reduction in heat transfer related entropy generation while viscous dissipation related entropy generation increased by 12,000%. It is anticipated that additional improvements in performance could be achieved by further increases in the blocked flow area.

A second interesting item is noted in the last column on Table 4. This reports the percent of the total entropy that is generated in the row of computational cells closest to the wall. As the disrupter size is increased, a progressively larger fraction of the entropy is being generated close to the wall. This is caused by thinning of the thermal boundary layer resulting in large thermal gradients close to the wall. The concentration of entropy generation close to the wall suggests that finer noding close to the wall may be required to attain a firm understanding of the structure of the local entropy generation.

A review of the distribution of local entropy generation showed that entropy generation caused by heat transfer is concentrated close to the wall in the thermal boundary layer. Total local entropy generation is dominated by heat transfer related entropy generation, resulting in total entropy generation also being concentrated in the thermal boundary layer. Local entropy generation caused by viscous dissipation is more evenly distributed throughout the flow field with a significant concentration in the region immediately upstream of the flow disruptor. In this region, the fluid is rapidly decelerating and the strong velocity gradients caused a concentration in local entropy generation. The identification of the upstream concentration in local entropy generation suggested that a more streamlined design should be considered.

While the simulation results discussed above indicate that the flow disrupter may be an attractive heat transfer enhancement technique, simulation is no substitute for experimental confirmation. It is recommended that an experimental proof-of-concept test be conducted on the flow disrupter concept.

## CONCLUSIONS

Based on the results of this study, we have drawn the following conclusions.

- In the base case design, heat transfer related entropy generation dominates viscous dissipation losses. This suggests that the design should be altered to improve heat transfer, even if the alteration results in a substantial increase in viscous dissipation losses.

- It is important to maintain balanced flow (where $\dot{m}Cp$ of the magnetic material is approximately equal to $\dot{m}Cp$ of the regenerator fluid) in the regenerator. Substantial deviation from balanced flow conditions can result in a significant increase in thermodynamic losses.

- Entropy generation caused by conduction in the magnetic material is a significant loss mechanism.

- The addition of a flow disrupter can significantly improve the performance of the regenerator. The optimum disrupter design will block more than 75% of the flow area.

- Local entropy generation analysis proved to be a useful method for evaluating heat exchanger designs. This approach identified the dominant thermodynamic losses in the rotary magnetic heat pump regenerator and allowed the efficient evaluation of heat transfer enhancement devices.

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## APPENDIX A - FIGURES-OF-MERIT FOR HEAT EXCHANGER ANALYSIS

Researchers involved in evaluating heat transfer enhancements are often faced with the problem of combining pressure drop and heat transfer losses in one figure-of-merit. This has led to the development of several methods for measuring thermodynamic performance of heat transfer enhancements. An example is the efficiency index presented by Webb, Eckert and Goldstein (1971). The definition of the efficiency index is given in Equation A.1.

$$\eta = \frac{St_e/St_p}{f_e/f_p} \tag{A.1}$$

If the efficiency index is greater than 1.0, the enhancement is attractive because the improvement in heat transfer ($St_e/St_p$) is larger than the increase in pressure drop ($f_e/f_p$). Unfortunately, a thermodynamic figure-of-merit that does not include second law effects will not, in many cases, give appropriate design guidance.

When we consider internal duct flow, one second law measure of the performance of an enhancement is given by the augmentation entropy generation number N (Bejan 1982) where N is given by Equation (A.2). An enhancement is attractive when $N < 1.0$.

$$N = \frac{\left(\frac{St_p}{St_e}\right)\left(\frac{D_e}{D_p}\right) + \phi\left(\frac{f_e}{f_p}\right)\left(\frac{D_p}{D_e}\right)\left(\frac{A_p}{A_e}\right)^2}{1 + \phi} \tag{A.2}$$

The expression for N given in Equation (A.2) has two interesting features; the impact of the enhancement concept on heat transfer losses (the term containing St) and pressure drop losses (the term containing f) are additive and the relationship between the two terms depends on the irreversibility distribution ratio $\phi$, which is defined by Equation (A.3).

$$\phi = \frac{S_{vis}}{S_{Thmm}} \tag{A.3}$$

$\phi$ gives an indication of the relative importance of the two loss mechanisms. When pressure drop losses are small, $\phi = 0$; N is only influenced by heat transfer related losses. The major limitation on the efficiency index [Equation (A.1)] is that it does not include a measure of the relative importance of heat transfer and pressure drop related losses. The evaluation of entropy generation in the regenerator of a magnetic heat pump is a good example. We will show that Equation (A.1) predicts that the inclusion of flow disruptors is not attractive while, as we have reported, a second law approach clearly indicates that flow disruptors will result in a significant reduction in thermodynamic losses.

Bejan (1982) shows that for duct flow, Equations (A.4) and (A.5) can be written for entropy generation caused by heat transfer ($S_{Thrf}$) and viscous dissipation ($S_{vis}$).

$$S_{Thrf} = \frac{\dot{q}_o'^2}{4T^2\dot{m}Cp} \frac{D_e}{St_e} \tag{A.4}$$

$$S_{vis} = \frac{f_e}{D_e A_e^2} \frac{(2\dot{m}^3)}{\rho^2 T} \tag{A.5}$$

Equations (A.4) and (A.5) can be written for the base and enhanced cases, assuming that $\dot{q}$, T, $\dot{m}$, Cp and $\rho$ are not influenced by the enhancement, they can be combined to give Equations (A.6) and (A.7).

$$St_p/St_e = (D_p/D_e)(S_{Thrf_e}/S_{Thrf_p}) \tag{A.6}$$

$$f_e/f_p = (D_e/D_p)(A_e/A_p)^2(S_{vis_e}/S_{vis_p}) \tag{A.7}$$

It is now possible to use the entropy generation results for the flow disruptor to calculate the ratio of Stanton number and friction factor. We will consider two cases. First, the flow area and hydraulic diameter will be based on the flow

path around the disruptor. In the second case, the flow areas and hydraulic diameter will be based on the full width of the flow path without disruptors. In both cases the entropy generation results were taken from Table 4 for a disruptor design that blocks 75% of the flow area.

Case 1 $\quad f_e/f_p = 1.80 \qquad St_e/St_p = 1.10$

$$\eta = 1.10/1.80 = 0.611$$

Case 2 $\quad f_e/f_p = 115.2 \qquad St_e/St_p = 4.419$

$$\eta = 4.419/115.2 = 0.0383$$

Augmentation Entropy Generation Number

$$N = 1.353/2.942 = 0.343$$

In both cases, the efficiency index indicates that the use of flow disruptors is not attractive, while N indicates that this enhancement technique is quite attractive. The reason for the different conclusions is related to the distributions of losses. In the base magnetic heat pump regenerator design, thermal losses are two orders of magnitude larger than the pressure drop losses. Consequently, an enhancement technique that reduces heat transfer losses is attractive, even if it results in a substantial increase in pressure drop losses. The efficiency index does not account for the difference in relative importance between the heat transfer losses and pressure drop losses.

## References

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