AN ULTRA-COMPACT LAMINAR-FLOW CRYOGENIC HEAT EXCHANGER

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ABSTRACT

A unique header arrangement and exchanger manufacturing process are being developed that allows economic parallel manifolding of millions of short stainless steel microtubes with outside diameters below 0.7 mm in compact counterflow exchangers. Flow inhomogeneity and axial conduction losses are expected to be sufficiently low for effectiveness above 99.5% for both nitrogen and helium. Relative pressure drops less than 0.02% have been measured under moderate-pressure, laminar-flow conditions. The modular microtube design is shown to offer advantages in diverse applications from 10 K to 1000 K. Preliminary test data is reported.

The steps involved in the novel manufacturing process include precision fineblanking, high speed automation, sacrificial tooling, and diffusion welding. Progress in the manufacturing process development is reported, and the prospects for economic production are discussed.

The intended initial application for this compact heat exchanger is in reverse Brayton cycle cryocooler recuperators. Preliminary experiments with an RBC cryocooler using novel micro-turbine expanders with hydrostatic gas bearings and micro-generators in the 10-30 W range are reported.

INTRODUCTION

Heat exchanger design always involves a compromise between at least three objectives: maximizing the heat exchanger effectiveness; minimizing the work required to overcome fluid friction in the heat exchanger; and minimizing the manufacturing and material costs of the exchanger. Often a fourth objective, minimizing the heat exchanger mass, is crucial. In the past the third objective has pushed design in the direction of high turbulence heat exchangers, to take advantage of the high heat transfer coefficient associated with turbulent flow. But with respect to the first two objectives this is a poor choice: turbulent flow extracts a disproportionately high penalty in pumping work.

We will often address applications in which the fourth objective - minimizing mass - is of paramount importance. Hence, we define a critical parameter to be maximized: specific conductance, or power density. We define specific conductance (W/kgK) as the exchange power per mean degree difference of the counterflowing streams per exchanger mass for a laminar flow gas. We show elsewhere that this parameter is essentially independent of both flow velocity and pressure for fully laminar conditions.[1] For normalized comparisons by simple laboratory techniques, we use helium gas at 350 K. The specific conductance then scales linearly with the gas thermal conductivity, which generally is proportional to the two-thirds power of temperature. Although specific conductance has not customarily been reported in the literature, we deduce that the best prior compact recuperators achieve normalized specific conductance near 60 W/kgK, and typical values for compact exchangers have often been below 10 W/kgK. Our prototypes now exceed 300 W/kgK, and we expect major further improvements will be possible.

Deriving laminar flow heat transfer theory with the perspective of modern manufacturing technology has resulted in two theoretical conclusions along with a technologically revolutionary design concept. Specifically, (1) An optimum heat exchanger must have fully laminar and extremely low velocity flow - typically 2% of the speed of sound; (2) The flow path length should be minimized, subject to the constraints of manufacturability, flow uniformity, and longitudinal conduction losses; (3) Precision micro-tubing, under 0.7-mm outside diameter and 200-mm length, can be automatically assembled to produce low cost heat exchangers with 5 to 50 times the specific conductance of current, compact designs. By taking advantage of modern manufacturing techniques, heat exchanger designs which were impossible to produce at the time current design practices evolved are now potentially feasible, but some manufacturing practices must be pressed somewhat beyond the current state of the art.

THE MICRO-TUBE STRIP (MTS) HEAT EXCHANGER DESIGN

Both the viscosity and the specific heat of a gas are essentially independent of the static pressure. Thus, it is desirable to operate at high static pressures (1 to 5 MPa) to minimize flow velocities for low pumping losses. Mechanical stress considerations clearly dictate a tubular design.

The primary feature of our work that allows order-of-magnitude increases in specific conductance is the understanding (not novel, but often forgotten) that the heat exchange capacity of a tube under laminar flow conditions is independent of the tube diameter and gas velocity.

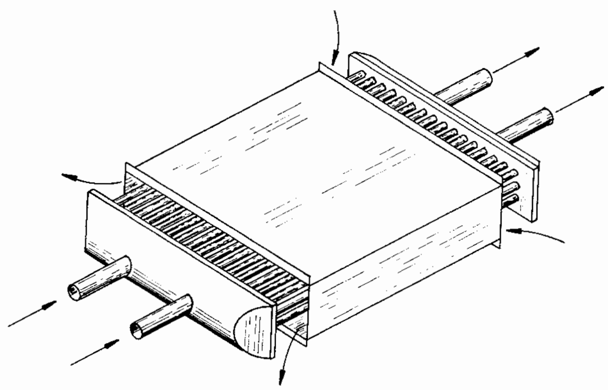

Figure 1. Schematic representation of the counterflow MTS module.

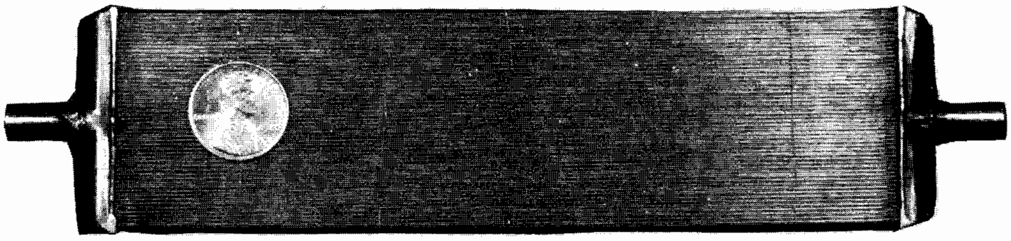

Figure 2. A typical stainless steel counterflow heat exchanger module as shown (without counterflow shell) contains 408 parallel microtubes, each 150 mm long, with 0.55-mm outside diameter and 0.8 mm between centers.

We approached the analysis of the counterflow gas-to-gas MTS module as shown in Figures 1 and 2 as follows. Assume that the thermal conductivity of the tube material is large compared to the thermal conductivity of the gas (this is easily the case for all gases, even for hydrogen). The heat transfer power W_X for this module under laminar flow conditions can be shown to be[1]

$$W_X = 4\pi nLk_1k_2T_\delta/(ak_1+bk_2), \quad (1)$$

where k_1 and k_2 are the thermal conductivities (W/mK) of the inner (tube-side) and outer (shell-side) gases respectively, n is the number of small tubes of length L, T_δ is the mean temperature difference between the counter-flowing streams, and a and b are dimensionless coefficients on the order of unity that are functions of tube inner and outer diameters and tube spacing. For tube inner diameter d and tube centers spaced 2d with tube wall w = 0.2 d, a is approximately 0.7 and b is unity. We will assume these geometric relationships and will assume $k_1 = k_2$.

The tube-side pumping power loss W_P can be expressed in terms of the tube-side mass flow rate G (kg/s) or velocity v (m/s) as follows:

$$W_P = 128\mu L(G/\rho)^2/(n\pi d^4) = 8\pi\mu nLv^2, \quad (2)$$

It is straightforward to show that the pumping power W_P is proportional to v^2W_X/T_δ, where v is the flow velocity, W_X is the heat exchange power, and T_δ is the mean temperature difference between counterflowing streams. As the tube diameter is reduced, the velocity must be kept small by increasing the number of parallel tubes by the square of the diameter ratio. The tube length and the total exchanger volume are reduced by the same factor (d_2^2/d_1^2) to maintain constant thermal and pumping powers.

In most cryogenics and power generation applications, it is critical that high effectiveness be achieved. This further imposes the requirements of (a) low longitudinal conduction losses and (b) uniform flow - both shell-side and tube-side.

As the tube length L is reduced, the longitudinal thermal conduction loss of the tube metal from the hot end to the cool end becomes more significant. It is predominately this loss mechanism which establishes the theoretical limit to specific conductance (by limiting minimum tube diameter) in high efficiency counterflow exchangers. For 95% effectiveness in a typical stainless steel helium recuperator between 300 K and 700 K, the minimum tube length is about 140d. For 99% effectiveness between 10 K and 50 K, the minimum tube length is about 1000d. It is desirable to minimize d until pumping power losses and axial conduction losses become unacceptable.

To facilitate uniform shell-side flow, it is necessary to depart from the disc shaped header tubesheet normally used in shell-and-tube heat exchangers and instead to use a rectangular header tubestrip with fewer than ten rows of tubes as shown in Figure 1.[2] Thus, we refer to the design as the Micro-Tube Strip, or MTS, design. Uniform tube spacing is maintained by including several spacer strips with triangular holes to position the tubes while allowing adequate shell-side flow around the tubes. Shell-side flow enters in cross-flow fashion between the headers and is then turned and constrained to flow axially over the tubes before exiting in like manner. The resistance to cross flow is much lower than the resistance to longitudinal flow. Hence, uniform shell-side axial flow is quickly established, and highly uniform counterflow conditions exist over about 90% of the tube length.

For a stainless-steel heat exchanger with helium as the working fluid at 1 MPa, operating between 400 and 1000 K, the lower limit on the tube inner diameter can be shown to be approximately 0.2 mm for 5% total losses (the sum of pumping power, longitudinal conduction, flow inhomogeneity effects, and Gc_pT_δ). For a cryogenic recuperator, inner diameters as small as 0.1 mm (the smallest size currently available at low cost) can be used - even near atmospheric pressure.

Parallel and Series Manifolding

To realize the parallel manifolding of hundreds of thousands of microtubes, a number of identical modules - each containing typically 400 tubes - must be manifolded together. These modules can be assembled into heat exchanger "banks" consisting of perhaps ten identical modules as shown in Figure 3. Obtaining uniform flow between the parallel modules within a bank and between parallel banks is simplified to some degree by virtue of the laminar flow conditions within the modules, both tube-side and shell-side. Velocities are low enough to eliminate many inertial effects.

In cryogenic applications, where viscous losses are small and extremely high effectiveness is required, a number of banks must be manifolded in series to achieve sufficiently low longitudinal conduction losses and to achieve adequate flow homogeneity by inserting flow mixers into the connections. For example, a typical cryogenic recuperator requiring 99.5% effectiveness between 50 K and 300 K with a helium mass flow rate of 1 g/s (with tube-side mean pressure of 0.3 MPa and shell-side mean pressure of 0.12 MPa) could easily be produced from the MTS modules shown in Figure 2 by connecting the following four MTS exchanger banks in series: a 10-module bank (4080

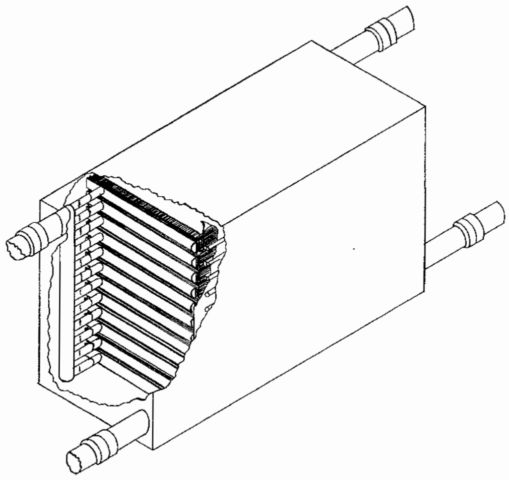

Figure 3. Schematic drawing of 10 modules combined in parallel.

microtubes) at the 300-K end, followed by an 8-module bank, followed by a 6-module bank, followed by a 4-module bank at the 50-K end. While the resulting arrangement is about three times longer than comparable wire-mesh regenerators, it is less massive and has much lower pressure losses for similar operating conditions - and of course it has isolation between the two flows.

The use of a lower conductivity alloy and thinner-wall tubing would allow shorter modules and reduced mass. Increasing the mean pressures would allow the use of finer tubing and a substantial reduction in mass. The ability to use identical modules for temperatures from 10 K to 1000 K, regardless of the system size, results in substantial savings in manufacturing costs.

MANUFACTURING CONSIDERATIONS

The very small stainless microtubes which are required may be produced at low production costs - less than $0.14 per meter - due to advances in high speed laser welding technology and diamond drawing dies.[3] Successful utilization of these microtubes depends on technological advances in the assembly, welding, and manifolding processes.[4]

Diffusion-Welded Tube Joints

The extremely small size and large number of tubes makes conventional welding methods unsuitable, but an option of demonstrated viability for the elongated tube-strip geometry is diffusion welding.

Diffusion welding occurs when clean metal surfaces are held together under pressure at high temperatures - but well below the melting point.[5] It is a solid state metallurgical process in which the combined action of solid-state diffusion mechanisms and solid-state surface tension result in (1) recrystallization or grain growth across an interface and (2) the solution or dispersion of interfacial contaminants. The time required to form the bond is generally an inverse exponential function of temperature and a quadratic function of surface finish and interfacial gaps. For most nickel-chromium wrought alloys with precision surfaces (0.4 μm rms) under moderate pressure (5 MPa, or 700 psi), high quality welds (90% of the base metal strength) can be obtained in several seconds at 1230°C. The rate decreases by an order of magnitude for a 100°C decrease in temperature.

Suitable weld conditions are readily achieved if the tube diameter and hole size can be held to very tight tolerances. The use of hardened tubes and annealed tubestrips then makes it possible to press the tubes into slightly undersized (interfering) holes in the thin, rectangular, header tubestrip. With hard, straight tubes, of length less than 300 times their outside diameter, it is possible to press them into soft tubestrips with up to 3% interference without serious difficulty. With proper attention to surface quality and a minimum of 0.4% interference press fit, the conditions required for diffusion welds are readily achieved. Surface finishes of about 0.4 μm rms in the area of the diffusion weld have proven to be leak tight within 10^{-6} standard mm^3/s to hydrogen at one atmosphere.

After the header tubestrips have been pressed onto the microtubes and the assembly of microtubes and header tubestrips is thoroughly cleaned, the assembly must be heated to effect the diffusion weld to the tubes. Most of our diffusion welding thus far has been done in inert or reducing atmosphere ovens and consequently has been limited to slow cycles.

Fineblanked Tubestrips

The MTS requirement of low production costs in the hard-drawn microtubing imposes a tolerance limit of $\pm 0.4\%$, which then leaves a $\pm 0.9\%$ tolerance requirement for the hole diameters in the tube-strip. Fortunately, the hole diameter need not be constant over the majority of its length, and a slight taper is in fact beneficial from an assembly standpoint. Punching consistently suitable, closely spaced microholes in superalloys does represent a technical challenge. For large scale production the most promising technique is Swiss fineblanking - a controlled cold-flow blanking (punching) process that includes the use of a counterpunch and a high pressure ring indenter (stripper) which applies sufficient pressure to the metal surfaces near the punch edges to prevent normal and planar deformation of the material during punching. The technique requires compound dies and triple action presses, but it results in minimal edge fracturing and deformation, as illustrated in Figure 4.

No other technique comes close to competing on a cost basis in large scale production with Swiss fineblanking - less than $1 per tubestrip. Preliminary diffusion welding experiments have demonstrated the fineblanking feasibility of MTS header tubestrips with eight rows of 0.53 mm holes spaced about 0.8 mm apart with 50 holes per row. Electrochemical finishing techniques will be used in conjunction with the fineblanking technique.

The major limitation with the fineblanking technique is the extreme die stress that ultimately results in die rupture as the relative hole spacing is reduced or the number of rows is increased - because of the L/d limitation of the precision holes in the die. Powder metallurgically produced die blanks of alloy T-15 appear to be most promising. The dies are quite expensive, but hundreds of thousands of parts per die are expected with minimal die maintenance.

High-Speed, Precision Assembly Technique

Our ultimate objective is assembly at rates up to 1.5 million microtubes per day per production line or about 2000 modules per day. Whether or not this can be achieved yet remains to be seen. A complex sequence of operations is being developed, which briefly comprises the following steps:

The microtubes are finished to the required length with electrochemically radiused ends. A hopper and Gatling-gun arrangement fires the microtubes into an alignment fixture, positioned with an x-y-z table, using bursts of compressed air. Then the tube array (except for the tube ends) is encapsulated in a low-melting-point alloy using vacuum injection techniques. This

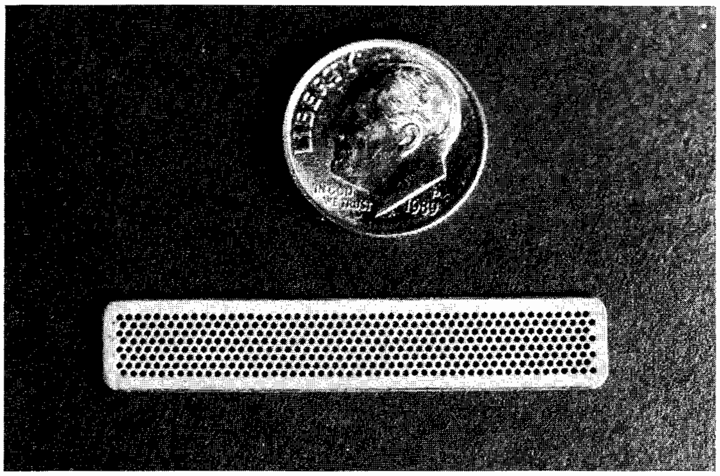

Figure 4. Shown above is a photograph of the current 408-hole tube-strip.

provides the rigid support necessary to enable the header tubestrips to be pressed into position without buckling the fine tubes. The fusible alloy is melted and cleaned from the subassembly. An oven cycle then effects the diffusion welding of the tubes-to-tubestrips. Finally, the appropriate baffles, caging, and manifolding is attached, and different bank combinations are constructed depending upon the application.

EXPERIMENTAL TEST RESULTS

Some data on the initial prototype 300-tube bank of microtubes was previously reported which showed good agreement with theory for flow rates that yielded effectiveness below 90%, but the errors increased rapidly for effectiveness up to the maximum reported value of 96%. We attributed this behavior predominately to shell-side flow inhomogeneity of about 3% in the first prototype.[1]

The steps taken to maintain improved uniformity of tube spacing in our latest 4080-tube bank are expected to reduce the total flow inhomogeneity to approximately 0.5%. We expect to complete this testing within four weeks and report results shortly thereafter.

APPLICATIONS IN REVERSE BRAYTON CYCLE (RBC) CRYOCOOLERS

The RBC has often been used at cooling powers above 200 W because it allows high efficiency over a wide range of load conditions and permits very compact expanders. Moreover, the expanders and compressors can be designed for virtually zero vibration and noise. These appealing features have lured researchers in many labs over the past three decades to attempt to develop a commercial RBC cryocooler, but the dream has always been beyond their grasp. Being a steady-state process, the RBC requires separate-stream recuperators, rather than cyclic regenerators. It would appear that the RBC cannot be scaled down below several hundred watts at 70 K, or below several tens of watts at 12 K - partially because of the severe recuperator requirements and partially because of the difficulties encountered in producing acceptable microturbine expanders. The promise of a more suitable recuperator - the MTS design - suggests that we once again turn our attention to this cycle.[6,7]

The dominant limitation in most cryogenic systems is the recuperator cost, size, and performance. Viscous losses at the high temperature end of the recuperator are always a major factor limiting cycle efficiency. Fortunately, the MTS exchangers can be scaled so that these viscous losses remain small even at very low system mass.

One of the most convincing arguments against the RBC is usually the cost of its cryogenic turbine expander, since the J-T cryocooler needs only a simple orifice and the piston displacers of the Gifford-McMahon cycle are well-developed. Available microturbine expanders are often priced near $100,000 each in prototype production. Our recent turbine designs promise high-efficiency cryogenic expanders from several watts to hundreds of kilowatts at very low cost. We have recently developed a highly stable air bearing for surface mach numbers from 0.3 to 1.1 on journal diameters above 3.5 mm that solves the whirl instabilities of earlier designs without the complex manufacturing requirements of alternative approaches.[8,9] We have also demonstrated a novel microturbine design that appears suitable for 2-W helium expanders at 12 K. Efficiency above 45% has been demonstrated in a similar 20-W air turbine at room temperature with blade Reynolds numbers below 4000.[10] More details on our novel gas bearing and microturbine designs will appear in subsequent papers in the near future.

The Figure of Merit (FOM) of small cryocoolers is typically less than 0.08, i.e., they operate at less than 8% of the Carnot efficiency limit. We believe practical RBC cryocoolers using the MTS recuperators can achieve FOM of 0.3.

CONCLUSIONS

We have demonstrated a recuperator with specific conductance above 300 W/kgK - which we believe is higher power density by a factor of five than has previously been obtained for highly effective recuperators above 100 K. We expect this advance in heat exchanger technology will effect substantial changes in cryogenic refrigerator system designs.

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