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Measurements with reticulated vitreous carbon stacks in thermoacoustic prime movers and refrigerators[a)]

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Reticulated vitreous carbon has been successfully used as a stack material in thermoacoustic prime movers and refrigerators. It is a rigid glassy carbon material, with a porous spongelike structure. Test results indicate peak pressure amplitudes of up to 32% in a prime mover, and refrigeration performance comparable to that of a traditional plastic roll stack. © *1998 Acoustical Society of America.* [S0001-4966(98)05505-2]

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INTRODUCTION

he stack is considered to be the most important part of the thermoacoustic engine. However, until now, those wishing to build a practical thermoacoustic prime mover or refrigerator have had a limited choice of stack materials; the plastic roll stack, such as developed by Hofler[1] as shown in Fig. 1, the wire mesh stack described by Reed,[2] and metal or ceramic honeycombs having square[3] and hexagonal[4] channel sections. These stacks, while simple in concept, can be very labor intensive or costly for the casual low budget researcher to fabricate. Recently, a new stack geometry, the ''pin stack,'' was introduced by Swift and Keolian.[5] The pin stack has, possibly, the ideal geometry for a thermoacoustic engine. Unfortunately, it is currently the most difficult stack to fabricate and may have some other potential disadvantages mentioned below.

Reticulated vitreous carbon (RVC) was suggested as a stack material by one of the authors (Moss), and has been found to have many advantages and very few disadvantages. RVC is manufactured by Energy Research and Generation (ERG), Inc.[6] It is an open pore foam material composed solely of vitreous carbon as can be seen in Fig. 2. In addition to being relatively inexpensive, light weight, and easy to machine, it has a temperature limitation of 300 °C in air and 3500 °C in an oxygen-free atmosphere, and has low thermal conductivity and a reasonably high specific heat. ERG also claims that RVC has a very high ''void volume'' or porosity of about 97%. Neither the plastic or metal roll stack nor the wire mesh stack possess all of these properties. Because of its random fibrous structure, RVC has no ''straight through'' optical paths. This serves to block, or at least reduce, infrared radiation back to the cold end and may also reduce the heat leak caused by acoustic streaming; another claim that cannot be made for stacks having a regular and periodic geometry.

Design and construction of a thermoacoustic engine is not a trivial task. It was therefore decided that instead of designing a prime mover and refrigerator specifically for RVC stack measurements, existing devices could be used to test the suitability of RVC as a thermoacoustic stack material. Because of the difficulty in developing theoretical models for RVC stacks, the approach here is strictly empirical. We will report the results of measurements of the performance of several RVC stacks with different pore sizes.

A most elementary method was employed for picking an appropriate RVC material for use in our thermoacoustic engines. The manufacturer grades the material according to the average number of pores per linear inch, or ppi. In other words, if one looks at the surface of an RVC block and counts the number of pores in 1 in. in any given direction, this is the ppi value. RVC was available to us in 100, 80, 60, 45, 30, 20, 10, and 5 ppi. The same measurement of effective pores per inch can be made with a parallel plate stack simply by counting the number of channels per inch across the end of the stack, or it can be approximated by taking the reciprocal of the sum of the plate thickness and plate separation in inches. For example, if a parallel plate stack has a plate thickness of 0.003 in. and a plate separation of 0.015 in., the corresponding porosity would be 1/0.018 ppi, or 56 ppi.

As a starting point, an RVC stack was chosen with a ppi value roughly equal to the effective ppi value of a parallel plate or roll stack known to have good performance. Given the lack of a computational model for the RVC structure, it is valuable to have a simple and relatively effective (as we shall see) means to determine the correct sizing. This is in contrast to the empirical results for stainless steel mesh stacks in thermoacoustic engines[1,2] where the optimal geometry values are somewhat surprising.

I. EXPERIMENTAL APPARATUS

The first set of measurements taken were in a sealed prime mover, illustrated in Fig. 3, which was designed by Hofler and Gardner, and first described by Castro.[7] It was originally designed to test the effectiveness of heat exchangers in a very high amplitude acoustic environment. This device is a half-wavelength sealed resonator that is designed to use low variable pressure neon gas. This allows for relatively

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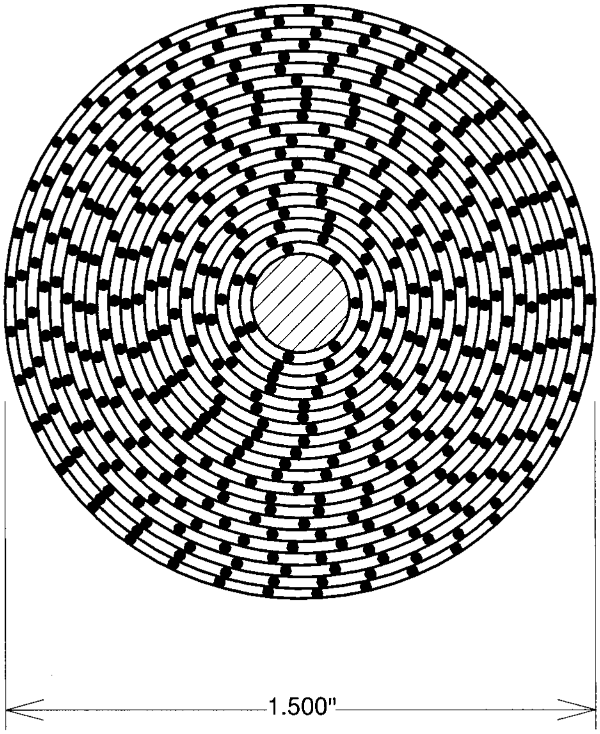

FIG. 1. The plastic roll stack seen edge-on and backlit.

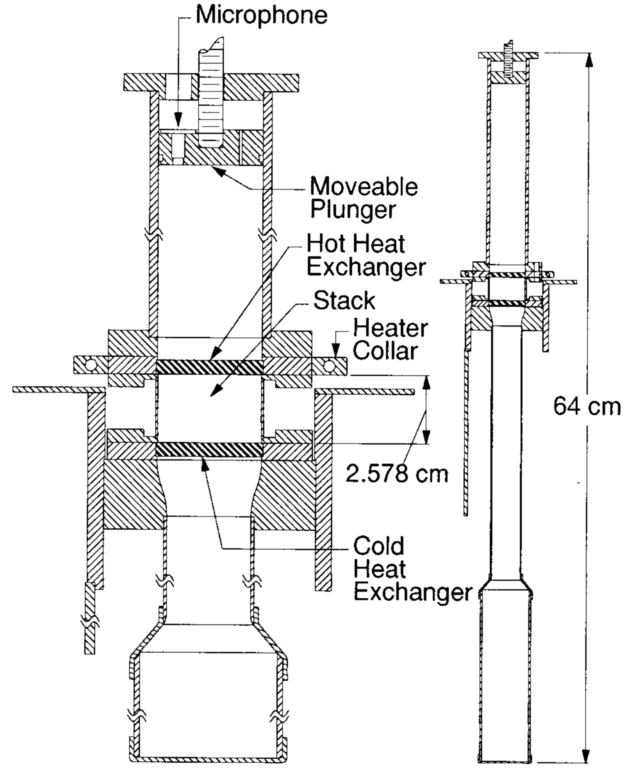

FIG. 3. Section drawing of the sealed prime mover apparatus, designed for high amplitude and relatively low heat power.

low input heater power with high acoustic amplitudes, as well as adjustable penetration depth. In addition, a moveable plunger is located at the pressure antinode to allow the stack's location in the standing wave to be adjusted. Measurements were taken with the warm end of the stack at $kx = 0.070$, 0.137, and 0.174 rad. An internal pressure transducer, located on the plunger, was used for measuring both the internal mean pressure and the dynamic acoustic pressure amplitude. Thermocouples were placed internally on the heat-exchanger fins and at various points on the outside of the resonator. The cold end of the resonator is placed in a liquid nitrogen bath to anchor the cold heat-exchanger's temperature at 78 K, while the hot heat exchanger is heated by two electrical resistive heater cartridges. The actual internal cold heat-exchanger temperatures were generally considerably higher than that of the liquid nitrogen, because the design did not anticipate the magnitude of the acoustic pressure amplitudes and resulting heat flows through copper parts having limited thermal conductance. Control of the experiment is semiautomated by using a temperature controller to operate the heater cartridges and maintain a constant temperature at the hot heat exchanger. As the mean pressure is gradually increased from about 2 kPa, the device reaches ''onset'' and spontaneously begins to oscillate. As the mean pressure is increased further, the increasing intensity of the acoustic pressure oscillations draws more heat from the hot heat exchanger, which is rejected at the cold heat exchanger. Thus the heat supplied by the heater cartridges must increase in order to maintain the constant temperature difference.

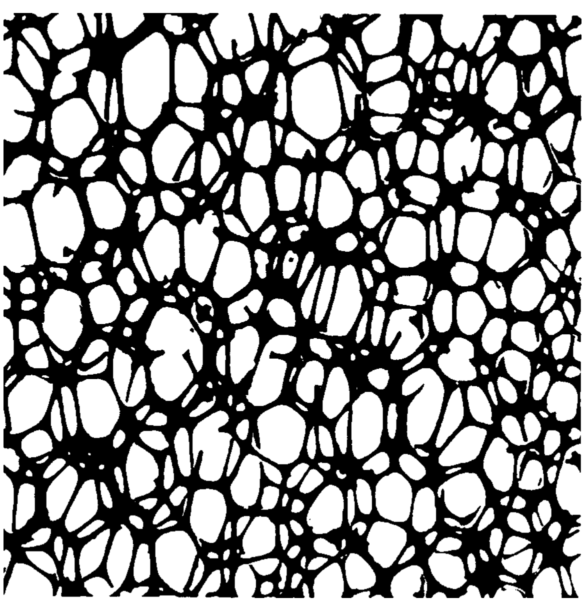

FIG. 2. Enlarged view of RVC showing the open cell reticulated structure.

The second set of measurements taken were in a small sealed refrigerator, illustrated in Fig. 4, which is a modification of Hofler's original refrigerator.[1] This refrigerator, first described by Brooks,[8] was designed for small temperature spans with the aid of a computer model for thermoacoustic engines, developed by one of the authors (Hofler). This device consists of a quarter wavelength resonator, driven by an electrodynamic loudspeaker at the pressure antinode. It normally operates with a plastic roll stack and is pressurized with 620 kPa of helium. It is typically capable of providing up to 10 W of cooling power at 0 °C with a temperature span of 35 °C and a coefficient of performance (COP) of approximately one.[9]

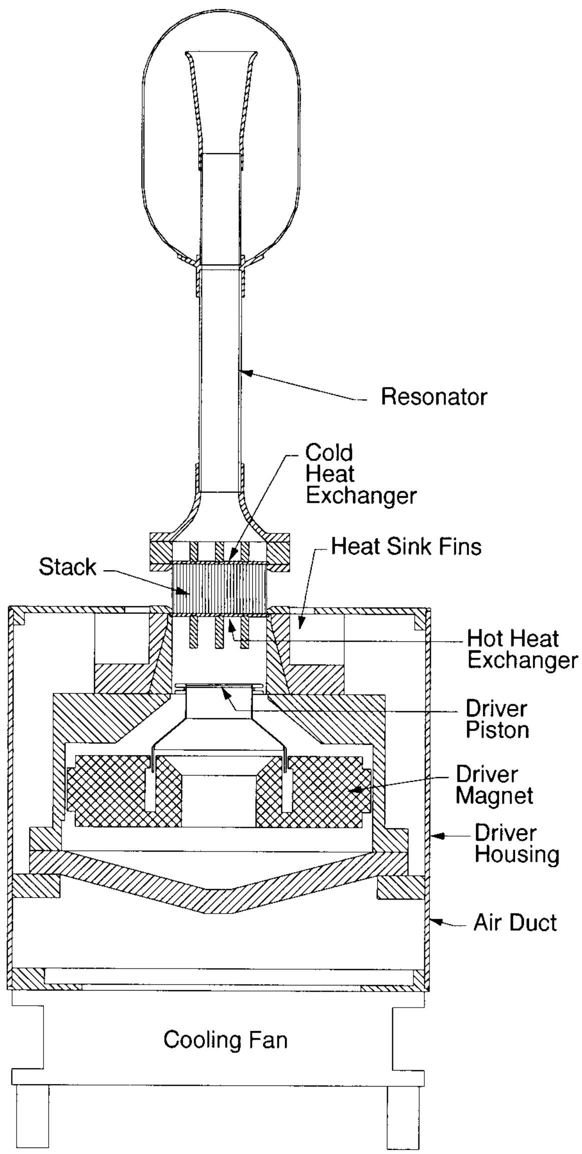

FIG. 4. Section drawing of the demonstration refrigerator.

II. PRIME MOVER MEASUREMENTS

For the prime mover, the plastic roll stack used by Castro[7] had a plate thickness of 0.102 mm and a plate separation of 0.711 mm for a sum of 0.813 mm. The reciprocal of this is 31 pores per inch. We therefore chose to begin our measurements with a 30-ppi RVC stack. Later, we tried a 20-ppi RVC stack because Reed's measurements with wire mesh stacks[2] indicated that more open stacks work better in this apparatus.

The procedure for these tests was to place an RVC sample stack into the resonator, purge the vessel of any air, and then fill it with pure neon to a pressure of about 2 kPa. The cold end was then submerged in liquid nitrogen, and the heater cartridges in the hot end were turned on and placed under the control of an analog temperature controller to maintain the hot heat-exchanger's temperature at approximately 300 K. The mean pressure in the resonator was then slowly increased until onset was reached. The system was allowed to equilibrate for about 10–15 min at which time the hot and cold heat-exchanger temperatures, acoustic and mean pressure, frequency, and heater input power were measured. The mean pressure was then increased slightly, the system allowed to equilibrate again, and another set of data recorded, and so on up to a mean pressure of about 80 kPa. The same test was conducted with the plunger at different positions. In addition, a set of data points was taken holding the mean pressure constant at 50 kPa, but varying the plunger position.

In characterizing the performance of the RVC prime mover stack, we examined two important parameters, the acoustic pressure amplitude and the ''efficiency.'' The acoustic pressure amplitude P_0 is expressed in rms values and given relative to the mean pressure P_m in the resonator, or P_0/P_m. We use rms rather than the usual peak values here because of the occasional presence of some moderate distortion in the acoustic pressure waveform which was measured with a true rms voltmeter. The issue of efficiency is more complicated, however, because the only acoustic load is the wall loss inside the resonator, which we were not able to measure. Therefore, a value for quasiefficiency, $\mathrm{Eff}_{\mathrm{Quasi}}$ derived by Hofler and Reed[1,2] is used which assumes similar loading for each of the different stacks and which is intended for comparison purposes only.

The following derivation for the quasiefficiency follows Reed[10] exactly and begins with Swift's equation for the total resonator dissipation power $\dot{E}$ for a plane-wave resonator:

$$\dot{E}=\frac{1}{4}\,\frac{P_0^2}{\rho_m a^2}\,\omega\pi RL\left[\delta_\kappa\,\frac{\gamma-1}{1+\epsilon_s}\left(1+\frac{2R}{L}\right)+\delta_\nu\right]. \quad (1)$$

Where P_0 is the rms dynamic pressure amplitude, ρ_m is the mean density, a is the speed of sound, ω is the angular frequency, δ_κ and δ_ν are the thermal and viscous penetration depths, γ is the ratio of specific heats, and the radius and length of the resonator are given by R and L. The available surface heat capacity is ϵ_s, which for these experiments is very small compared to 1 and can be ignored. Then, by using $\rho_m a^2=\gamma P_m$, Eq. (1) becomes

$$\dot{E}\approx\left(\frac{\pi R}{4\gamma}\right)\left(\frac{P_0}{P_m}\right)^2 P_m\omega L\left[\delta_\kappa(\gamma-1)\left(1+\frac{2R}{L}\right)+\delta_\nu\right]. \quad (2)$$

The geometry of the prime mover is such that the term $2R/L$ is much less than 1 and can be ignored. Using the definition of the Prandtl number, Pr,

$$\delta_\kappa=\frac{\delta_\nu}{\sqrt{\mathrm{Pr}}}, \quad (3)$$

Eq. (2) can be rewritten as

$$\dot{E}\approx\left(\frac{\pi R}{4\gamma}\right)\left(\frac{P_0}{P_m}\right)^2 P_m\omega L\delta_\nu\left[\left(\frac{\gamma-1}{\sqrt{\mathrm{Pr}}}\right)+1\right]. \quad (4)$$

Collecting all the constants, Eq. (4) can be written as a proportionality:

$$\dot{E}\propto\left(\frac{P_0}{P_m}\right)^2 P_m\omega L\delta_\nu . \quad (5)$$

Since the dynamic viscosity is nearly independent of the pressure, the viscous penetration depth can be expressed as

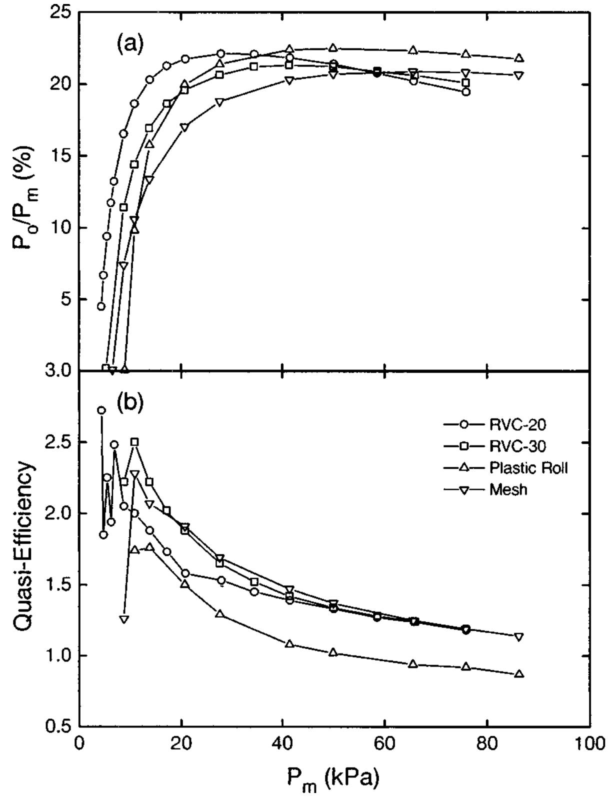

FIG. 5. Measured performance of the thermoacoustic prime mover comparing four different stacks at position $kx=0.070$. (a) Pressure ratio and (b) quasiefficiency plotted against mean pressure.

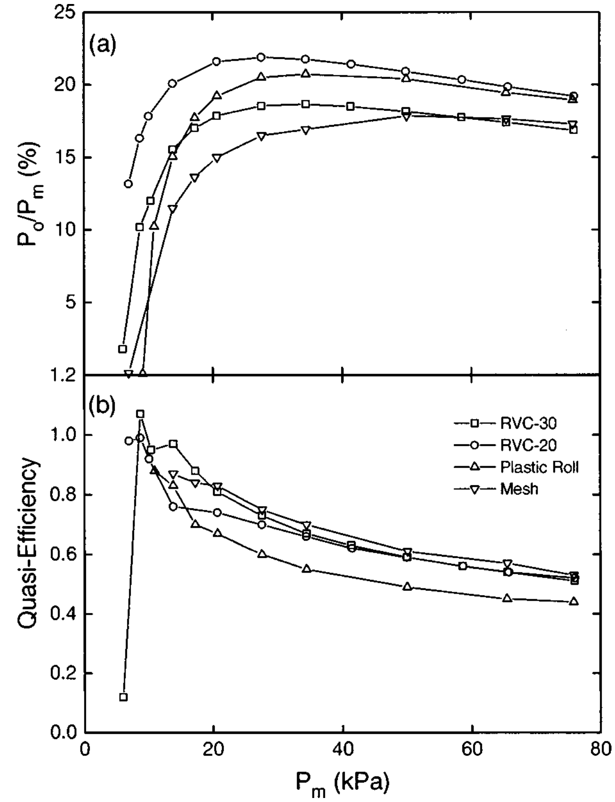

FIG. 6. Measured performance of the thermoacoustic prime mover comparing four different stacks at position $kx=0.137$. (a) Pressure ratio and (b) quasiefficiency plotted against mean pressure.

$$\delta_\nu=\sqrt{\frac{2\eta}{\omega\rho}}\propto(\omega\rho_m)^{-1/2}\propto(\omega P_m)^{-1/2}. \tag{6}$$

Proportionality (5) then becomes

$$\dot{E}\propto\left(\frac{P_0}{P_m}\right)^2 L(\omega P_m)(\omega P_m)^{-1/2}=\left(\frac{P_0}{P_m}\right)^2 L\sqrt{\omega P_m}. \tag{7}$$

Note that the L dependence is for a constant diameter resonator at uniform temperature. In this case $\omega\propto 1/L$, so then

$$\dot{E}\propto\left(\frac{P_0}{P_m}\right)^2 \frac{1}{\omega}\sqrt{\omega P_m}=\left(\frac{P_0}{P_m}\right)^2\sqrt{\frac{P_m}{\omega}}. \tag{8}$$

The efficiency, Eff, is defined as the resonator losses divided by the heat input

$$\mathrm{Eff}=\frac{\dot{E}}{Q_{\mathrm{Hot}}}. \tag{9}$$

Therefore a quasiefficiency, $\mathrm{Eff}_{\mathrm{Quasi}}$, can be defined as

$$\mathrm{Eff}_{\mathrm{Quasi}}=\frac{(P_0/P_m)^2\sqrt{P_m/2\pi f}}{Q_{\mathrm{Hot}}}, \tag{10}$$

where f is frequency. This quasiefficiency, as shown in Figs. 5(b), 6(b), and 7, has units of $\sqrt{\mathrm{kPa/Hz}}/\mathrm{W}$. This expression assumes a linear system where the resonator dissipation has a quadratic amplitude dependence, and it accounts for pressure and resonator length variations.

It should be noted that we expect that nonlinearities such as flow separation do occur at higher amplitudes and therefore the resonator dissipation increases more rapidly than quadratic at the higher amplitudes. While caution should be exercised in comparing quasiefficiencies at dramatically different amplitudes, a stack that produces higher quasiefficiency values at an equal or higher amplitude is truly more efficient.

The value Q_{Hot} in Eq. (10) is formed by subtracting the heater power measured below acoustic onset, which is essentially an apparatus heat leak, from the heater power measured above onset. The heat leak is generally much smaller than the typical heater power above onset, except at the lower mean pressure and amplitude combinations. Typical heat

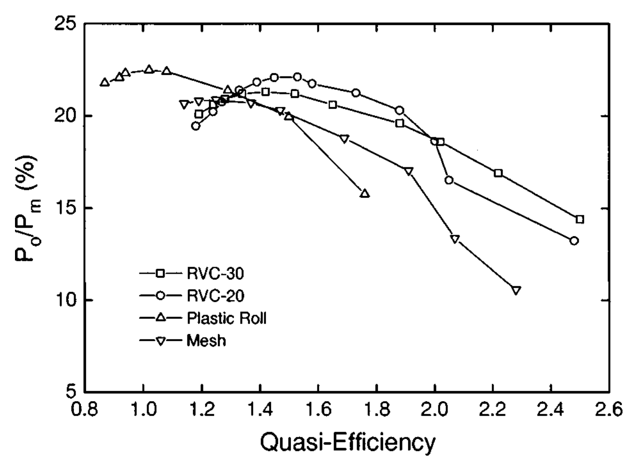

FIG. 7. Comparison of measured P_0/P_m with respect to quasiefficiency for the prime mover with the stack at position $kx=0.07$, for all four stacks.

leaks are in the 15–20 W range, whereas the highest heater powers were in the 250 W range. Estimated uncertainty in the heater power measurements near the heat leak level were between 0.5 and 1.0 W.

The pressure amplitude performance of the RVC stack was very encouraging, as may be seen in Figs. 5(a) and 6(a), showing the ratio P_0/P_m versus mean pressure for two different plunger positions. A comparison is made to the original plastic roll stack used by Castro[7] and the best stainless steel mesh stack used by Reed.[2] At the closest plunger position and at lower mean pressures, the 20-ppi RVC stack performed the best, followed by the 30-ppi RVC stack, then the stainless mesh stack, and finally the plastic roll stack. Only at the higher mean pressures did the plastic roll stack begin to surpass the other stacks. However, as the plunger was pulled back, effectively moving the stack farther from the pressure antinode, the superiority of the 20-ppi RVC stack became quite pronounced. Its performance at low mean pressures surpassed the other stacks considerably, and slightly surpassed the plastic roll stack at the higher mean pressures. The trend continued as the stack was moved farther away from the pressure antinode. The conclusion from these graphs is that under many of the conditions tested the 20-ppi RVC stack offered higher acoustic pressure amplitudes than the other stack materials. Only at the closest stack positions with higher mean pressures was the RVC stack clearly inferior to the plastic roll stack and the wire mesh stack.

Even more interesting are the results of the quasiefficiency plots in Figs. 5(b) and 6(b). The data points at low mean pressure are not very reliable because of the heat leak subtraction discussed above. This was exacerbated by the extremely low onset pressures achieved with the RVC stacks and the correspondingly low heater cartridge power levels. For mean pressures above 10 kPa, however, the data becomes acceptable. With the plunger all the way in, and the stack closest to the pressure antinode, we can see in Fig. 5(b) that the RVC and stainless mesh stacks were clearly superior to the plastic roll stack. Because some of these comparisons are being made at different pressure amplitudes, we can expect that a stack that has both higher quasiefficiency and higher pressure amplitude should have higher true efficiency than a stack with lower quasiefficiency and lower pressure amplitude. We therefore concluded that the 20-ppi RVC stack should have slightly higher true efficiency at the lower mean pressures, while the plastic roll stack should have slightly higher true efficiency at the higher mean pressures relative to what is shown in Fig. 5(a). This gave us a confident assessment that the 20-ppi RVC stack was slightly better than the plastic roll stack, and at least as good as the wire mesh stack.

With the plunger pulled back to the second position, the same behavior was noted as in the closest position. Because the 20-ppi RVC stack always had the highest acoustic pressure amplitudes at this plunger position, we concluded that the true efficiency for this stack was even higher relative to the plastic roll stack than is already seen in Fig. 6(b), while the 30-ppi RVC stack and the mesh stack had lower true efficiencies because they always produced lower acoustic pressure amplitudes. Again, we were confident that the 20-ppi RVC stack was superior to the plastic roll stack, and at least as good as the wire mesh stack. The results were similar for the farthest stack position, even though we do not present them here.

Another interesting way to summarize the performance of these stacks is illustrated in Fig. 7 showing P_0/P_m versus quasiefficiency for the closest plunger position. Ideally, one would want a stack that delivers a high acoustic pressure amplitude ratio, P_0/P_m, as well as high efficiency. In fact, this graph shows that the 20- and 30-ppi RVC stacks delivered higher pressure amplitudes at higher efficiencies than the plastic roll stack.

III. REFRIGERATOR MEASUREMENTS

The procedure used for quantifying the performance of the thermoacoustic refrigerator was to first run the device with no externally applied heat load until it reached its ultimate cold temperature. A data point was taken consisting of hot and cold heat-exchanger temperatures, and the acoustic pressure amplitude and driver volume velocity along with the phase between them. A known heat load was then applied to the cold heat exchanger and the refrigerator's temperatures were allowed to equilibrate again, after which the same data points were recorded with the addition of the external heat load. The refrigerator was operated at a constant drive current level of 1.85 A and a constant frequency of 645 Hz. This was done in order to duplicate the conditions under which the unit was originally tested by Berhow.[9]

The previously mentioned data was reduced down to two important parameters, the temperature ratio T_c/T_h, and the coefficient of performance (COP), both as a function of the total heat load on the cold end of the refrigerator. The total heat load included the heat leak from the hot end of the refrigerator to the cold end of the refrigerator due to conduction by the walls of the resonator vessel, the stack, and the helium gas, the insulation heat leak, and any external heat load added to quantify its performance. The external heat load was applied via a resistive electrical heater and was very simple and straightforward to measure. The nuisance heat leak was measured by running the refrigerator until it reached its ultimate cold temperature and then shutting it off and measuring the warm up rate, which if the resonator is well insulated, is an exponential function. From this data a thermal time constant is derived which, along with the heat capacity of the cold end, gives a heat leak in W/°C. The temperature ratio is also very straightforward to measure, while the COP can be obtained from the simple relation:

$$\mathrm{COP} = Q_{\mathrm{total}}/W,$$

where Q_{total} is the total heat load, in watts, on the refrigerator and W is the acoustic power, or work flow, into the resonator. The rms P_o/P_m values were typically between 2.5% and 3.5% for these tests.

The RVC results for our low-powered refrigerator were not as favorable as they were for the prime mover, and might reflect the fact that the RVC stacks were simply inserted into an engine that had been optimized for use with a particular parallel plate geometry stack, and which was designed for use as a demonstration aid rather than a solid thermoacous-

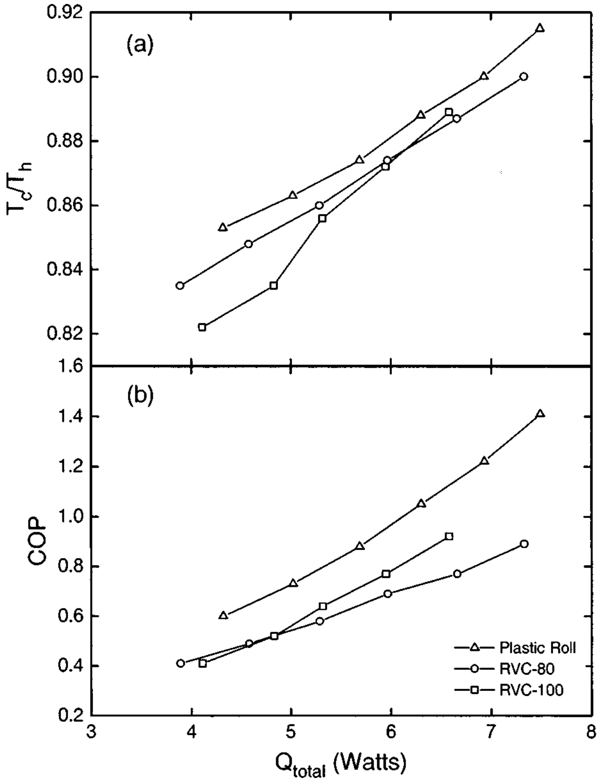

FIG. 8. Measured performance of the thermoacoustic refrigerator at a mean pressure of 620 kPa, comparing three different stacks. (a) Temperature ratio and (b) coefficient of performance against total refrigerator load Q_{total}.

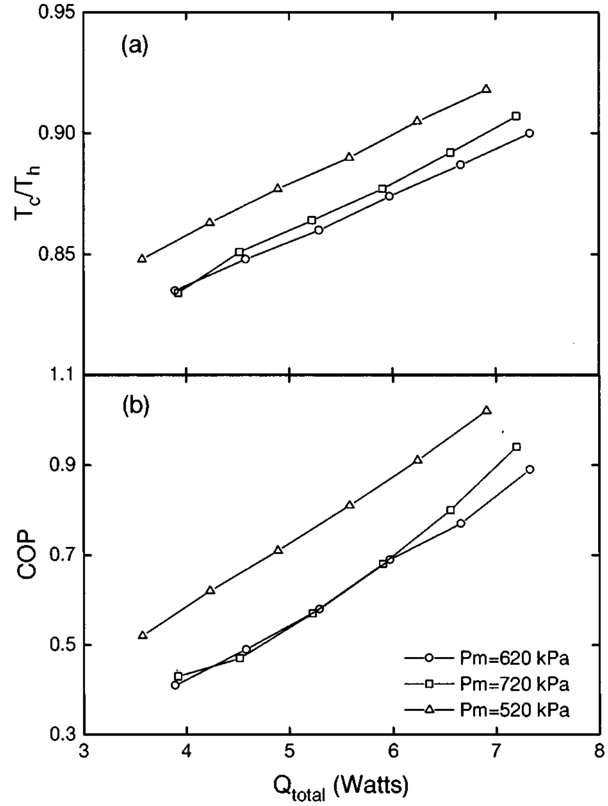

FIG. 9. Measured performance of the thermoacoustic refrigerator with an 80-ppi RVC stack, comparing three different mean operating pressures. (a) Temperature ratio and (b) coefficient of performance plotted against total refrigerator load Q_{total}.

tics test engine. Also, the refrigerator lacked the adjustment provided by the plunger in the prime mover apparatus.

The first set of measurements were taken at a mean pressure of 620 kPa with pure helium, which is the normal operating pressure with the plastic roll stack. The roll stack used in this engine is made from polyester film with a thickness of 0.051 mm, a layer separation of 0.254 mm, and a spacer interval of 5 mm. An RVC porosity of 80 ppi was initially chosen based on these numbers, with 60 and 100 ppi being the secondary choices. Figure 8 shows the results for the temperature ratio (T_c/T_h) and COP as a function of total heat load Q_{total} for the plastic roll stack and for the 80- and 100-ppi RVC stacks. The 60-ppi stack is not included here because it produced negligable refrigeration power, and no data was taken with an applied heat load. Our measurements indicate that while the temperature ratio for a given heat load was slightly lower with both RVC stacks, indicating higher available cooling powers, the plastic roll stack still held the edge in efficiency, giving higher COP's. However, based on this data, it is not clear if one of the two RVC stacks was generally superior to the other.

The second set of measurements taken with the refrigerator consisted of varying the mean pressure up and down by 100 kPa from the normal 620-kPa operating pressure, and taking the same data as before. Unfortunately, due to a sudden driver failure, we were only able to obtain this data for the 80-ppi RVC stack. Figure 9(a) shows that the 80-ppi stack has higher cooling power available at the higher mean pressure of 720 kPa, but Fig. 9(b) shows that it is more efficient at the lower mean pressure of 520 kPa, as evidenced by the higher COP. In fact, when we looked at the COP relative to Carnot (COPR) we found that the refrigerator with the 80-ppi stack had a COPR 1.05 times higher at 520 kPa than at 720 kPa at the lower heat loads, while at the higher heat loads the COPR's were the same. The higher efficiency at lower mean pressure is probably an indication that the 80-ppi stack has too large of a pore size to be operated efficiently at 620 kPa, since lowering the mean pressure to 520 kPa increases the thermal penetration depth and makes more of the gas in the channels productive. The increment of 100 kPa was also perhaps too large to allow us to find an ideal operating pressure, but it does serve to point out the performance trends associated with varying penetration depth.

IV. CONCLUSIONS

Based on these measurements, we believe that RVC has a definite potential as a low cost, easy to fabricate material suitable for use in all types of thermoacoustic prime movers and refrigerators. Further investigation is still needed to determine exactly which combinations of stack and resonator dimensions and gas parameters, including the use of gas mixtures, would yield optimum performance. Currently, measurements are being taken on precompressed samples of RVC in a high-temperature and high-pressure prime mover, in an attempt to decrease the extremely high porosity of the RVC and thereby increase the heat capacity of the stack. The only drawback we have found to RVC, thus far, is that because it is extremely brittle, a loose fitting stack can vibrate against the heat exchangers enough to cause the filaments to break. However, even if the RVC stack never substantially surpasses the overall performance of the parallel plate stack, it has other advantages that still make it an attractive material. RVC could also become potentially more cost competitive compared to other materials like plastic roll and stainless steel mesh, given enough demand for the product.

ACKNOWLEDGMENTS

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