CHARACTERIZATION OF 350 HZ THERMOACOUSTIC DRIVEN ORIFICE PULSE TUBE REFRIGERATOR WITH MEASUREMENTS OF THE PHASE OF THE MASS FLOW AND PRESSURE

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ABSTRACT

The world's first 350 Hz thermoacoustic driven orifice pulse tube refrigerator (TADOPTR) has been designed and built by Tektronix, Inc., in cooperation with Los Alamos National Laboratories (LANL) and the National Institute of Standards and Technology (NIST). This highly instrumented system includes hot wire anemometers and pressure sensors for measuring the phase of the mass flow and pressure at all key locations in the TADOPTR, permitting for the first time detailed comparison to analytical models developed by LANL and NIST. Characterization results for velocity and pressure phase, pressure amplitude, and enthalpy flow show good agreement with the simulations. We have also demonstrated a new design method that uses the inertance of the pulse tube at 350 Hz to achieve the desired phase between the mass flow and pressure, rather than the usual double inlet design. We have designed and characterized single stage and two stage 350 Hz TADOPTRs.

INTRODUCTION

TADOPTR's are of interest because they have no moving parts and thus should have a long lifetime. The first TADOPTR was built by NIST and LANL and operated at 27 Hz[1]. Tektronix, LANL, and NIST have jointly designed and constructed 100 Hz and 350 Hz TADOPTRs.

We describe here the design and characterization of single stage and two stage 350 Hz OPTRs. The design of the both OPTRs uses the inertance of the pulse tube to control the phase angle of the velocity at the cold end of the pulse tube. We are able to design the OPTR to obtain phasing of the mass flow similar to that found in a Stirling cycle[2,3]. The two stage system is designed to be a testbed for validating our design methodology. In order to verify the design prediction for velocity phasing, we included hot wire anemometers at key locations in the two stage TADOPTR. Temperature, pressure and heat rejection are also measured, allowing detailed comparison to analytical models developed by LANL and NIST.

DELTAE[4,5] and REGEN3.1[6] are modeling tools developed by LANL and NIST, respectively. DELTAE (Design Environment for Linear ThermoAcoustic Engines) can simulate thermoacoustic systems that are combinations of segments such as ducts, heat exchangers, regenerators or thermoacoustic stacks, and impedances. Among its predictions at each segment are: pressure, volume velocity, enthalpy flow and work flow. REGEN3.1 solves for the performance of many types of regenerators. Since the stacked screen regenerator segment for DELTAE was only recently completed, REGEN3.1 was used for the initial design of the stacked screen regenerators, and was used to verify the results of the new DELTAE segment.

A 350 Hz thermoacoustic driver (TAD) designed and constructed by LANL is used to drive the OPTRs. All characterization has been done with helium as the working fluid; most tests have been done at a mean pressure of 3.1 MPa, but some experiments have been performed at lower mean pressures.

SINGLE STAGE 350 HZ OPTR

The design of the 350 Hz single stage pulse tube is based on simulation results from DELTAE and REGEN3.1. The DELTAE simulation shows that in the design optimized for the best coefficient of performance, the phase angle of the mass flow at the two ends of the regenerator is shifted by about 90 degrees, and this shift is approximately evenly split around the pressure oscillation, as seen in Figure 1. This arrangement is very similar to that of Stirling machines, where one has the freedom to adjust the mass flow phase angles for optimal performance. It is possible to obtain this phasing relationship without pistons by making use of the inertance of the pulse tube. This is feasible at 350 Hz due to the shorter wavelength of sound in helium at this pressure.

A schematic of the single stage OPTR is shown in Figure 2. The regenerator is 3.6 cm in diameter and 5.5 cm long; it is packed with #325 stainless steel mesh and has a

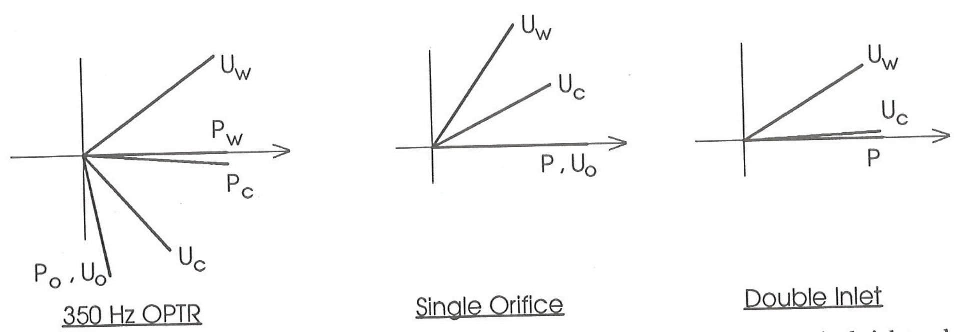

Figure 1. Phasor diagrams for the 350 Hz single stage OPTR design, and for conventional single inlet and double inlet orifice pulse tubes. U denotes the velocity, P denotes the pressure, and the subscripts w, c, and o denote the warm end of the regenerator, the cold end of the regenerator, and the orifice (the warm end of the pulse tube). Phases are relative to the pressure at the warm end. In the 350 Hz OPTR, the velocity lags the pressure at the cold end; the phase of the pressure changes also, whereas in conventional low frequency OPTRs, the change in the phase of the pressure is usually negligible.

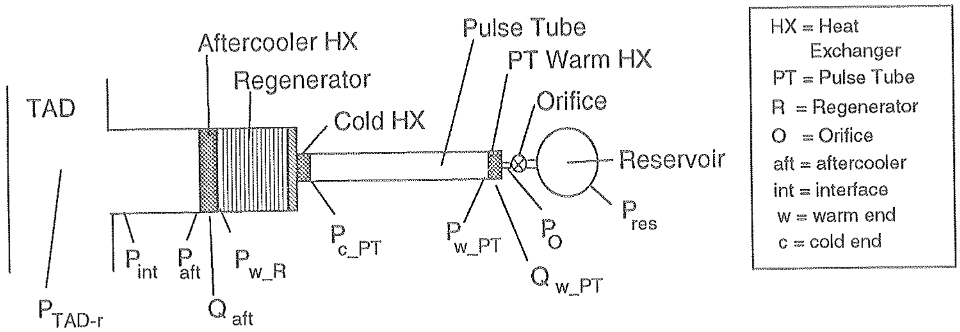

Figure 2. Schematic of the single stage OPTR, with locations of components and pressure sensors indicated.

measured porosity of 73%. The pulse tube is 0.84 cm in diameter and 20.0 cm long. The reservoir is 150 cc, and the orifice valve is 7.1 mm in diameter. The orifice valve is completely open for these experiments. The connecting duct between the TAD and OPTR is 20 cm long and 3.6 cm in diameter.

The single stage OPTR reached a cold temperature of 147 K with a mean pressure of 3.1 MPa and a pressure amplitude of 0.190 MPa (6.1%) at the entrance to the OPTR. The heat rejected to the aftercooler heat exchanger was 494 W, which is a measure of the power input to the OPTR. Figure 3 shows the cold temperature of the OPTR as a function of the normalized pressure amplitude squared at the TAD. Although we were limited in the pressure amplitude obtainable by limitations in the hot end temperature of the TAD heaters, we came close to achieving the desired performance. This result is a very strong confirmation of the utility and power of the design tools, and the overall design approach to this new concept for high frequency orifice pulse tube refrigerators.

The pressure amplitude at several locations in the OPTR is shown in Figure 4 as a function of the pressure amplitude at the warm end of the regenerator. Also shown is the DELTAE model prediction. The simulation uses the geometry of the OPTR and uses the observed pressure amplitude at the warm end of the regenerator as an initial condition. The impedance between the pulse tube warm end and the reservoir is modeled as an 8 cm long 7.6 mm diameter duct, which is approximately the total length of the duct between the reservoir and pulse tube. There are no adjustable parameters in the model aside from the initial condition. The observed pressure amplitude agrees very well with the model simulation.

The simulation also gives the enthalpy flow through the regenerator, and the heat rejected at the aftercooler heat exchanger. For the operating point mentioned above (6.1% pressure amplitude at the TAD), the measured regenerator loss is 32 W, and the simulation result is 36 W. The measured heat rejected at the aftercooler was 494 W and the prediction was 288 W (42% discrepancy). The system was designed for the velocity phase at the cold end to lag the pressure by 41^{o}, and the simulation indicates that the velocity lags the pressure by 29^{o}. Although the basic performance agrees well with the predictions, several factors need more investigation. The temperature at the exit of the aftercooler heat exchanger was lower than its cooling water temperature. This behavior was unexpected, and the disagreement between the predicted and observed aftercooler heat is puzzling given the agreement for other parameters. The system was also designed to reach 110 K with 5 W cooling power, at 10% pressure amplitude at the TAD; extrapolations of the data shown in Figure 3 to higher pressure amplitude indicate that this performance would probably not be realized. Although the observed performance is consistent with a simulation that uses the observed initial conditions for pressure amplitude, in order to efficiently design the final system it is necessary to be able to construct a system that reaches its targeted operating condition.

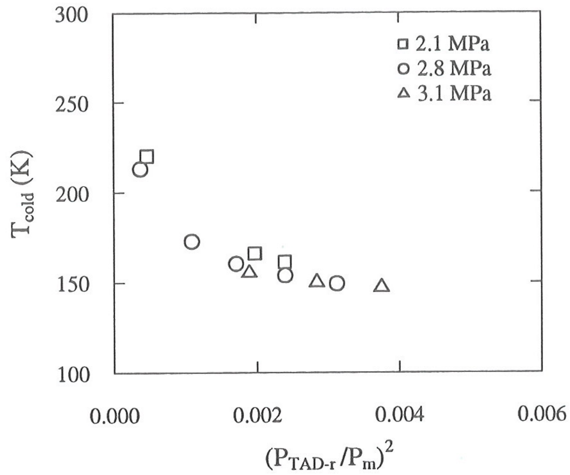

Figure 3. The cold temperature for the single stage OPTR as a function of the square of the normalized pressure amplitude at the TAD. Data are shown for three different mean pressures. We are prevented from going to higher pressure amplitude due to limitations on the hot end temperature of the thermoacoustic driver.

350 HZ TWO STAGE OPTR

We applied the design method developed for the single stage OPTR to the design of the two stage OPTR. The two stage system was designed primarily to give a detailed evaluation of our design methods and tools, in preparation for design and construction of the final system. As a result, instrumentation rather than absolute performance was of primary importance. A more in depth characterization and comparison to theory was desired for the two stage device than was obtained for the single stage device. This was due to the added complication of designing the branch between the two OPTRs to preserve the desired phasing scheme, and because of the unexpected behavior of the aftercooler heat exchanger in the single stage OPTR. As mentioned in the introduction, hot wire anemometers were included to measure the phase of the velocity, allowing us to make direct comparisons between the prediction and results for the velocity phasing.

The design of the two stage OPTR was again based on DELTAE and REGEN3.1. Two separate models for the first and second stages were used. The branch connecting the two stages was modeled as a load on the first stage, with the branch impedance being

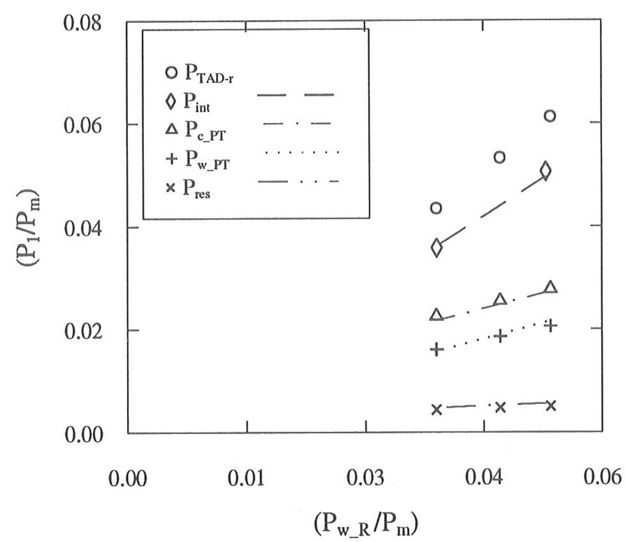

Figure 4. The pressure amplitude P_1 at various locations in the single stage OPTR as a function of the pressure amplitude at the warm end of the regenerator. The mean pressure P_m is 3.1 MPa. Experimental data are shown with symbols, and DELTAE model predictions are shown with lines. The model uses the observed pressure amplitude at the warm end of the regenerator as an initial condition, but requires no other adjustable parameters.

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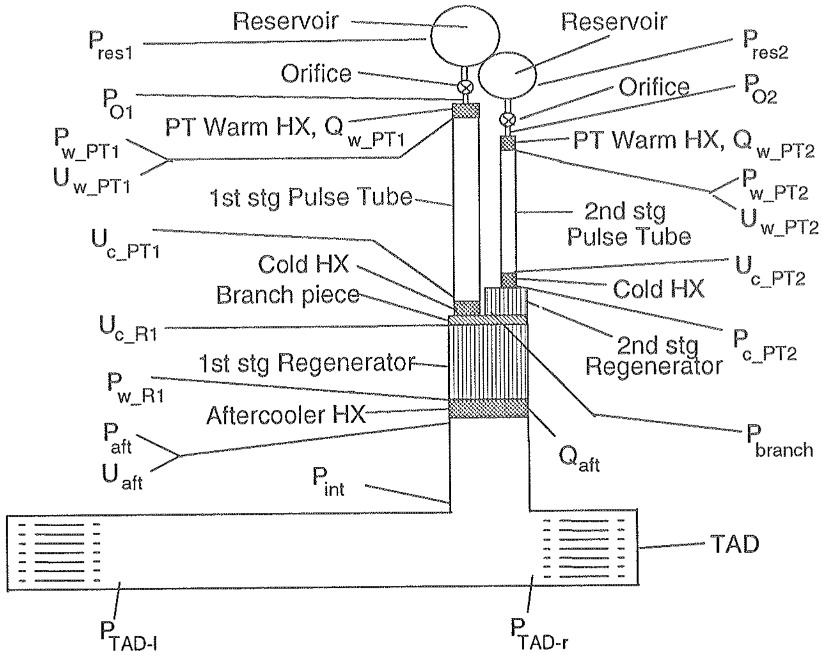

Figure 5. A schematic of the 350 Hz two stage TADOPTR system, showing location of pressure sensors (P), and anemometers (U). The anemometers are located at the entrance to the aftercooler heat exchanger, at the cold end of the first stage regenerator (before the branch to the second stage), at the cold end of the first stage pulse tube (after the branch to the second stage), at the warm end of the first stage pulse tube, and at the cold and warm ends of the second stage pulse tube. The heat Q rejected at the aftercooler and first and second stage warm end heat exchangers is determined by measuring the cooling water flow rate and temperature rise.

determined from the second stage model. A schematic of the two stage OPTR is shown in Figure 5. The first stage regenerator is 4.58 cm in diameter and 3.0 cm long; it is filled with #325 stainless steel mesh with 0.028 mm wire diameter, and has a measured porosity of 71%. The second stage regenerator is 3.19 cm in diameter and 3.0 cm long; it is filled with #325 bronze mesh calendered to 0.061 mm thickness, and has a measured porosity of 61%. The first stage pulse tube is 0.93 cm in diameter and 22.0 cm long; the second stage pulse tube is 0.57 cm in diameter and 11.0 cm long. The first and second stage reservoirs are 300 cc and 150 cc, respectively. The orifice valves are 7.1 mm and 4.4 mm in diameter for the first and second stage, respectively. The duct between the TAD and OPTR is 13 cm long.

The design of the first stage regenerator emphasized a low pressure drop in order to insure that adequate pressure amplitude was available at the second stage. As a result of this, the performance of the first stage is poor, and we only reached a cold temperature of 255 K at the first stage for a 4.30% pressure amplitude at the TAD. The second stage reached a cold temperature of 180 K at this operating point.

The anemometers used are TSI 1260-T1.5[7]. Since we only desire phase information, it was not necessary to calibrate the anemometers. Figure 6 shows typical oscilloscope traces for one pressure sensor and the six hot wire anemometers in the system. The anemometer signal is at 2ω, where ω is the frequency of the pressure oscillation. A lockin amplifier referenced to the pressure oscillation in the TAD is used to measure the phase of the anemometer signal.

Model comparisons are made by first fixing the value of the pressure amplitude in the model at the warm end of the regenerator to be the same as the experimental data, and then adjusting the model value for the orifice impedance until the simulated pressure amplitude in the reservoir matches the data. This sets the mass flow at the warm end of the pulse tube to be the same as observed in the experiment. The second stage data are analyzed first, and the impedance at the warm end regenerator is calculated from the pressure and mass flow. The second stage impedance is then entered into the branch segment for the first stage model, and the first stage data are analyzed with the above method.

Pressure amplitude, pressure phase and velocity phase data for the first stage and second stage are shown in Figure 7 as a function of position in the OPTR, for a pressure amplitude of 4.30% at the TAD. The agreement between observed and predicted pressure amplitude is good, as seen in (a) and (b) of Figure 7. For the phase relationships in the first stage, the mass flow precedes the pressure amplitude at the warm end of the regenerator, is in phase with the mass flow just before the branch to the second stage, and lags the pressure amplitude at the cold end of the first stage pulse tube (just after the branch to the second stage). This is what we expected to see based on our original design. The agreement

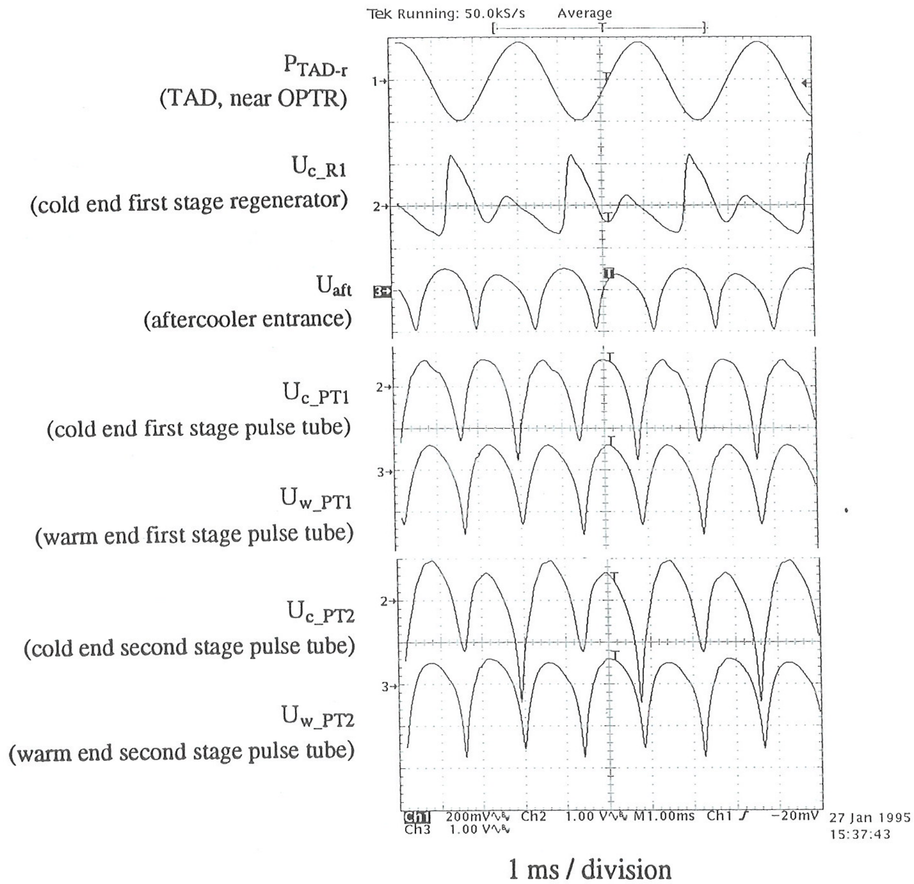

Figure 6. Oscilloscope traces for the anemometers in the 350 Hz two stage system. The locations of the anemometers are shown in Figure 5. The asymmetric traces for the cold end first stage regenerator and cold end second stage pulse tube indicate differing temperatures for the incoming and outgoing gas. The vertical scale is for reference only.

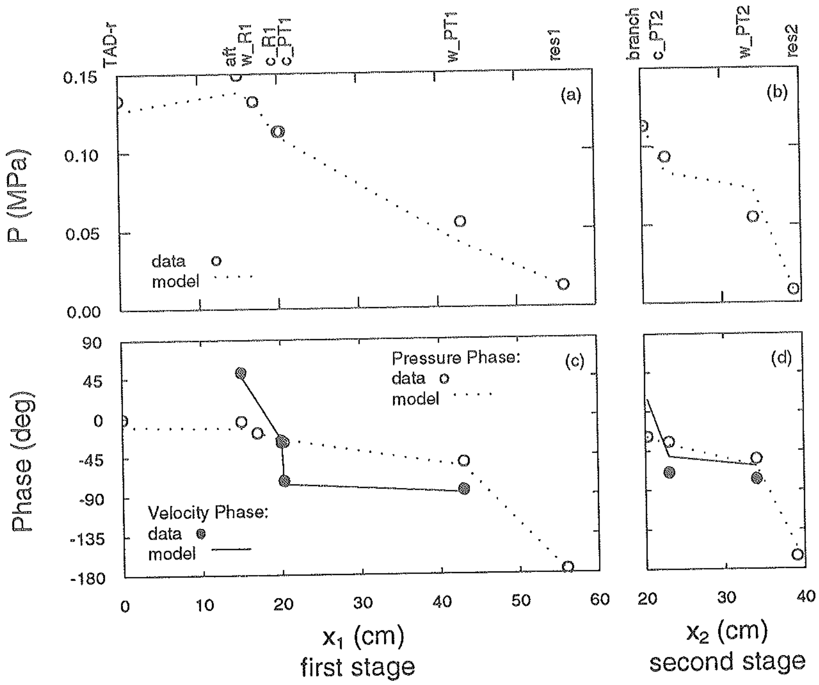

Figure 7. The pressure amplitude as a function of position in the OPTR for the first stage (a), and second stage (b). The model prediction is shown with a dotted line. The phase of the pressure and velocity are shown in (c) and (d) for the first and second stage, respectively. The dotted and solid lines give the model predictions. The top axis gives the location of the OPTR components, and is keyed to Figure 5. As seen in Figure 5, the locations c_R1 and c_PT1 are on either side of the branch, and therefore have the same pressure amplitude but different velocity phases. Mean pressure $P_m = 3.1$ MPa.

between observed and predicted phases is very good for the first stage, as seen in Figure 7c. The agreement for the velocity phase is not quite as good for the second stage. This may be due to sensitivity of the system to the geometry between the warm end of the pulse tube and the reservoir. A significant result is our confirmation that the mass flow lags the pressure at the cold end of the second stage pulse tube. These measurements provide an important quantitative verification of the design method and the concept of using the inertance of the pulse tube to shift the phase of the mass flow.

The regenerator loss for the first stage regenerator was 45 W when the pressure amplitude at the TAD was 4.30%; the model prediction was 28 W. The agreement for the regenerator loss is much better for a low pressure amplitude point: 19.4 W measured regenerator loss and 19.5 W predicted at a 2.84% pressure amplitude at the TAD. The regenerator loss for the second stage regenerator was 11.3 W at the 4.30% operating point, and the model prediction was 9.5 W. The heat rejected to the aftercooler at this point was 269 W, and the prediction was 192 W (29% discrepancy). The mass flow into the first stage reservoir was 8.8 g/s, and 2.7 g/s into the second stage reservoir.

In the results for the single stage 350 Hz OPTR, we mentioned that we saw unexpected behavior at the aftercooler heat exchanger, i.e. the gas temperature at the exit of the aftercooler was lower than the cooling water temperature. This behavior was not seen in the two stage OPTR. We observed instead that the exit gas temperature from the aftercooler was slightly above the cooling water temperature, as expected. The two systems used the same aftercooler heat exchanger, so other system parameters are affecting the aftercooler performance.

The poorer agreement between simulation and experiment for the phase in the second stage OPTR may be due to sensitivity of the system to the geometry at the orifice valve. The best design would eliminate any ducts going to and from the orifice valve, or use a carefully characterized capillary tube or porous plug as the impedance. Since it is difficult in practice to eliminate ducts going to and from a valve, the capillary or porous plug might be the best choice. The system is also sensitive to the design of the heat exchangers at either end of the pulse tube. Typical OPTR designs have used copper screen heat exchangers which are the same diameter as the pulse tube. We found that the small diameter of the pulse tubes in these designs caused an unacceptably high pressure drop in the heat exchangers. In order to reduce the pressure drop, we used copper screen heat exchangers of a diameter larger than the pulse tube, in combination with flow straighteners.

CONCLUSION

We have presented experimental results from single stage and two stage 350 Hz TADOPTRs that use the inertance of the pulse tube to shift the phase of the mass flow. In addition to pressure sensors, the two stage system included anemometers at key locations in the TADOPTR for measurement of the phase of the mass flow. The comparison of experimental results to simulations indicated good agreement in general, especially at low pressure amplitude. The agreement for the pressure amplitude, pressure phase and velocity phase are very good, especially for the first stage of the two stage system, and demonstrated that the design method using the inertance of the pulse tube to shift the phase of the mass flow is successful. This quantitative comparison to simulation results gives us confidence that other parameters determined by the simulation, such as mass flow, are also correct. As far as we know this is the first direct confirmation of the internal operation and phase relationships in an OPTR at key locations along its length. At this point we have confidence in our design tools and methods for designing orifice pulse tubes at frequencies up to 350 Hz.

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REFERENCES

1. R. Radebaugh, K.M. McDermott, G.W. Swift, and R.A. Martin, Development of a Thermoacoustically Driven Orifice Pulse Tube Refrigerator, Proc. of Fourth Interagency Meeting on Cryocoolers, 205 (1991).
2. A. Kirk, On the Mechanical Production of Cold, Proc. Ins. Civil Eng. **37**, 244, London (1874).
3. J.W.L. Kohler, The Stirling Refrigeration Cycle, Sci. Am. **212**(4), 119 (1965).
4. W.C. Ward and G.W. Swift, Design environment for low amplitude thermoacoustic engines, J. Acoust. Soc. Am. 95, 3671 (1994). Fully tested software and users guide available from Energy Science and Technology Software Center, US Department of Energy, Oak Ridge Tennessee. For a beta-test version, contact ww@lanl.gov (Bill Ward) via Internet.
5. G.W. Swift and W. C. Ward, Simple harmonic analysis of stacked-screen regenerators, submitted to J. Thermophys. Heat Trans.
6. J. Gary and R. Radebaugh, An improved numerical model for calculation of regenerator performance (REGEN3.1), Proc. of Fourth Interagency Meeting on Cryocoolers, 165 (1991).
7. TSI Inc., 500 Cardigan Road, P.O. Box 64394, St. Paul, MN 55164.