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STUDY OF TWO-DIMENSIONAL COMPRESSIBLE NON-ACOUSTIC MODELING OF STIRLING MACHINE TYPE COMPONENTS

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ABSTRACT

A two-dimensional (2-D) computer code was developed for modeling enclosed volumes of gas with oscillating boundaries, such as Stirling machine components. An existing 2-D incompressible flow computer code, CAST, was used as the starting point for the project. CAST was modified to use the compressible non-acoustic Navier-Stokes equations to model an enclosed volume including an oscillating piston. The devices modeled have low Mach numbers and are sufficiently small that the time required for acoustics to propagate across them is negligible. Therefore, acoustics were excluded to enable more time efficient computation. Background information about the project is presented. The compressible non-acoustic flow assumptions are discussed. The governing equations used in the model are presented in transport equation format. A brief description is given of the numerical methods used. Comparisons of code predictions with experimental data are then discussed.

INTRODUCTION

Background

NASA Glenn Research Center (GRC) has been involved in development of Stirling engines for ~25 years. GRC began managing the Stirling Automotive Development Program for the Department of Energy (DOE) in ~1977. The DOE/NASA contractors were first Ford Motor Co. (for one year) and then Mechanical Technology, Inc. (MTI) of Albany, NY. These Stirling automotive engines used hydrogen as the working fluid and were required to operate over demanding automotive driving cycles. This work continued into the early '90's (Ernst and Shaltens [1]). Engines were demonstrated in automobiles and trucks and future development looked promising. However, due to decreases in oil and gasoline prices and improvements in spark ignition engines, the automotive manufacturers did not choose to develop the Stirling technology for the automotive market.

Development of Stirling engines for generation of space auxiliary power began in ~1983 with MTI as the primary contractor. These engines used helium as working fluid, were 25 kWe (12.5 kWe/cylinder) designs with electrical power produced by linear alternators and used gas bearings. Back-to-back cylinders with synchronized pistons damped engine vibrations.

One of the problems encountered in Stirling engine development by GRC and MTI was the limited accuracy of the 1-D engine design computer codes. Performance projections for new designs were usually optimistic. The last major design of a 25 kWe (12.5 kWe/cylinder) Stirling engine for space power was the Component Test Power Converter or CTPC (Dahr [2]). Because of previous power shortfalls with other new engine designs, MTI and NASA chose to design the CTPC with a 20% margin on power. When tested the engine produced slightly in excess of the 25 kWe power goal, meaning that it produced almost 20% less power than predicted by the computer code that was used to design it.

During this period of space power Stirling engine development at MTI (~1983 to 1993), concerns about 1-D design code accuracy led to fundamental research. A "loss understanding" program was started consisting of grants and small contracts for investigation of thermodynamic losses (Tew and Geng [3]). The participants were hosted by GRC at yearly Stirling Loss Understanding Workshops for several years. The long range goal was to improve the accuracy of 1-D Stirling engine design codes.

The SP-100 (or Space Power-100 kWe) reactor space power system program, for which these 25 kWe space power Stirling designs were intended, ended in 1993. Along with it, funding for the Stirling loss understanding program ended also.

From ~1993 to 1999, NASA support of Stirling engine development continued at a low level via NASA SBIR funding. Most of this funding was for continued development of free-piston engines at the Stirling Technology Company (STC) of Kennewick, WA based on flexural bearing technology. GRC and Cleveland State University (CSU) also continued low-level development of a multi-D Stirling code via the thesis project reported here.

Now Stirling power convertors are being developed by DOE, NASA and STC as a substitute for the less efficient radioisotope thermoelectric generators for deep-space missions (Thieme, Qui, and White [4]). The success of STC in accumulating years of reliable operation on its free-piston flexural bearing machines (over 6 years on a 10 kWe convertor with no maintenance and no degradation in performance) has been a key factor. In late 1999, a 55 We STC free-piston Stirling was subjected to a number of tests and evaluations to determine its suitability for a deep-space mission (For example, it passed a simulated launch vibration test, while operating). As a result, in 2000, the Stirling radioisotope power system, or RPS (Furlong and Shaltens [5]), was baselined as the advanced power system that may be used on NASA missions such as Europa and Solar Probe missions. It now appears that long-duration Martian rover missions will be the first application of Stirling RPS.

Stirling One-Dimensional Flow Design Codes

Two one-dimensional-flow Stirling machine codes have been in primary use by NASA's Stirling engine contractors in recent years. These are the HFAST code developed by Mechanical Technology, Inc. (Huang [6]) and the GLIMPS code (now Sage, (Gedeon [7])) developed by Gedeon Associates. HFAST was used by MTI in design of the large (25 kWe) Stirling convertors discussed earlier. The GLIMPS code and now its successor, Sage, have been the primary design codes used by the Stirling Technology Company. Sage is also used by Sunpower, Inc. While these codes have been used to develop excellent engine designs, 1-D codes assume uniform axial flow and are thus believed to be deficient in modeling those interfaces in Stirling engines where significant changes in area take place. Past comparisons of code predictions, such as the GLIMPS/HFAST study done by Geng and Tew [8], showed rough overall performance agreement but differences in predictions of individual losses. It would be of significant value to be able to accurately characterize all major losses so that the design process could do an adequate job of trading off the various losses against each other.

Objective of the Work

The immediate purpose of the 2-D code development reported here was to develop a time-efficient compressible code for study and definition of Stirling machine type cylinder heat transfer/power (or hysteresis) losses; see Tew [9] for more complete documentation. A long range purpose was to provide the basis for development of a 2-D model of a complete Stirling engine that could be used to study interaction of various thermodynamic losses. Even longer range, if computer software/hardware time efficiencies are increased sufficiently, it may be possible to use a 2-D code (perhaps even a 3-D code) for engine design. The previous IECEC paper by Makhkamov and Ingham [10] suggests such design use of a 2-D code is already underway at the Laboratory for Stirling Engines at the Physical-Technical Institute in Tashkent.

NOMENCLATURE

Latin letters

c_p	specific heat at constant pressure
d	denotes differential of variable
D_h	hydraulic diameter (=4 x wetted area/wetted perimeter)
$\overline{D}$	viscous force vector
$\bar{f}$	mass force vector (due to gravity, for example)
p	static pressure
P	mean spatial pressure
P_a	amplitude of mean spatial pressure
P_o	arithmetic mean of max. and min. mean spatial pressures
q_m	entropy source due to non-zero mass sources
Q	density of continuously distributed heat sources
R	gas constant
s	entropy per unit mass
t	time
T	temperature
$\overline{\overline{T}}_{ST}$	stress tensor
$\bar{u}$	velocity vector $(=\bar{i}U + \bar{j}V)$
U	axial, x-direction, velocity
V	radial, r-direction, velocity or volume
V_o	arithmetic mean of maximum and minimum volumes
$\overline{w} = \xi\rho\bar{u}$	rate of momentum change because of mass sources
$\hat{W}_{loss}$	non-dimensional work or hysteresis loss

Greek letters

γ	ratio of specific heats of fluid
ϕ	transport quantity per unit mass
λ	molecular thermal conductivity
μ	absolute viscosity
ρ	density, fluid mass per unit volume
ω	angular velocity (rad/sec)
ξ	mass source strength per unit mass

COMPRESSIBLE NON-ACOUSTIC FLOW

Assumptions

Fedorchenko [11] discusses a number of subsonic initial-boundary-value problems that cannot be solved using the classical theory of incompressible fluid motion, which involves the equation $\nabla \cdot \bar{u} = 0$ (where $\bar{u} = \bar{i}U + \bar{j}V$ and the unit vector $\bar{i}$ is the x-direction for both cartesian and axisymmetric coordinates; the unit vector $\bar{j}$ is the y-direction for cartesian coordinates and in the r-direction for axisymmetric coordinates.) Among these problems are: (1) flows in a closed volume initiated by blowing or suction through permeable walls, (2) flows in a closed volume initiated by a moving boundary such as in a piston-cylinder problem, (3) flows with continuously distributed mass sources, and (4) viscous flows with substantial heat fluxes. Fedorchenko notes that application of the most general theory of compressible fluid flow may not be best in such cases because of the difficulties in accurately resolving complex acoustic phenomena and in assigning proper boundary conditions.

Fedorchenko proposes a non-local mathematical model where $\nabla \cdot \bar{u} \neq 0$ in general, for simulation of unsteady subsonic flows in a bounded domain with continuously distributed sources of mass, momentum and entropy, also taking into account the effects of viscosity and conductivity when necessary. The exclusion of sound waves is one of the most important features of the model.

The most general form of Fedorchenko's compressible non-acoustic system of equations for simulation of unsteady subsonic, heat conducting viscous flows are the following forms of the momentum, continuity, energy and state equations:

$$\frac{\partial(\rho\bar{u})}{\partial t} + \nabla \cdot \rho\bar{u}\bar{u} + \nabla p = \rho\bar{f} + \overline{w} + \overline{D} \tag{1}$$

$$\frac{\partial\rho}{\partial t} + \nabla \cdot \rho\bar{u} = \xi\rho \tag{2}$$

$$\frac{\partial s}{\partial t} + \bar{u} \cdot \nabla s = \frac{R}{P}\left[\nabla \cdot \lambda\nabla T + Q\right] + q_m \tag{3}$$

$$F(s, P, \rho) = 0 \tag{4}$$

When $\nabla\mu = 0$ (as it is for constant viscosity problems) then

$$\overline{D} = \mu\Delta\bar{u} + \frac{1}{3}\mu\nabla(\nabla \cdot \bar{u}) \tag{5}$$

Note that the energy equation is written in terms of entropy, s, rather than in terms of internal energy, enthalpy, or temperature.

The simplifications in the Navier-Stokes equations used to eliminate acoustics and arrive at the above system of equations were: (1) Pressure at any time, t, and position, $\bar{r}$, is split into a mean spatial pressure level that varies only with time and a $\Delta(pressure)$ relative to a reference position that varies with position and time:

$$p(\bar{r},t) = P(t) + \Delta p(\bar{r},t) \tag{6}$$

(2) The pressure appearing in the equation of state is the mean spatial pressure that varies only with time. Therefore, from the ideal gas equation of state

$$\rho = \frac{P(t)}{RT(\bar{r},t)} \tag{7}$$

So density is a function of mean spatial pressure level and the temperature field. (and is independent of the spatial pressure drop).

Simplification of Equations for the Stirling Problem

For the Stirling piston-cylinder problem of interest here, there are no mass or heat sources distributed within the cylinder volume and the gravity force on the gas is not of interest. Therefore the variables defined above: $\bar{f}$, $\overline{w}$, ξ, Q and q_m are all zero. Also the ideal gas equation of state is sufficiently accurate for the helium gas that is used in most current Stirling engines of interest to NASA. Therefore the equations (1)-(4) reduce to the following set:

$$\frac{\partial(\rho\bar{u})}{\partial t} + \nabla \cdot \rho\bar{u}\bar{u} + \nabla p = \overline{D} \tag{8}$$

$$\frac{\partial\rho}{\partial t} + \nabla \cdot \rho\bar{u} = 0 \tag{9}$$

$$\frac{\partial s}{\partial t} + \bar{u} \cdot \nabla s = \frac{R}{P}\left[\nabla \cdot \lambda\nabla T\right] \tag{10}$$

$$\rho = \frac{P}{RT} \tag{11}$$

CAST AND MODIFIED CAST CODE

The incompressible flow CAST (Computer Aided Simulation of Turbulent Flow) code, as originally received by Cleveland State University and NASA is documented in Peric and Schuerer [12]. Changes made to CAST in developing the modified CAST compressible non-acoustic code are documented in Tew [9].

Transport Equation Format

The solution technique used in the CAST code is based on a transport equation formulation of the governing equations. A coordinate-free form of the general transport equation is:

$$\underbrace{\frac{\partial(\rho\phi)}{\partial t}}_{\text{Time rate of storage}} + \nabla \cdot \Big(\underbrace{\rho\bar{u}\phi}_{\text{convection}} - \underbrace{\Gamma_\phi\nabla\phi}_{\text{diffusion}}\Big) = \underbrace{S_\phi}_{\text{source}} \tag{12}$$

The text under equation (12) categorizes the terms in the equation.

Transport quantities, exchange coefficients and source terms for the continuity, momentum and energy equations used in Modified CAST are shown in Table I. Note that the energy equation is in enthalpy format, rather than the entropy format used by Fedorchenko. The turbulent kinetic energy and dissipation rate equations are not shown here due to space limitations. They are defined in Tew [9].

Table I: General Transport Equation—Transported Quantities, Exchange Coefficients, and Source Terms for Continuity, Energy and Momentum Equations

Equation Name	Transport Quantity/vol.	Exchange Coefficient	Source Term
Continuity	ρ (mass/vol.)	0	0
Momentum	$\rho\bar{u}$ (momentum/vol)	μ	$\nabla \cdot \overline{\overline{T_{ST,S}}}$
Energy	ρh (enthalpy/vol.)	$\frac{\lambda}{c_p} = \frac{\mu}{\Pr}$	$\frac{dP}{dt}$

Numerical (Finite-Volume) Methods

The numerical methods in modified CAST are almost the same as those used in CAST (Peric and Scheuerer [12]). An exception is the use of Leibniz's rule (Ferziger and Peric [13]) to account for the effect of the moving boundary (i.e., piston motion). Modifications to the governing equations themselves, to account for the change from incompressible to compressible non-acoustic flow, are discussed in Tew [9]. These included accounting for the non-zero $\nabla \cdot \bar{u}$ terms, adding the time derivative of the mean spatial pressure to the source terms of the energy equation, and using mean spatial pressure in the ideal gas equation so that the density field is a function of the temperature field and the mean spatial pressure. Spatial pressure drop still appears in the momentum equations to help determine the velocity field. Since the density is not affected by spatial pressure, the incompressible SIMPLE algorithm still applies.

The governing equations are solved with a conservative finite-volume method (see Patankar [14]). The basic principle is: (1) Discretize the solution domain by subdividing it into small axisymmetric (or rectangular) control volumes. Locate the numerical grid points in the center of the control volumes. (2) Discretize the transport equations.

Discretization is done by formally integrating the single terms in the equations over a control volume. Application of Leibniz's rule and Gauss's theorem yields an integro-differential equation relating the net increase in the transported quantity per unit time to the convective and diffusive fluxes across the control volume boundaries and to the source (or sink) terms within the control volume. This practice leads to a conservative method because boundary-fluxes leaving one control volume through its right boundary enter the neighboring control volume through its left boundary. Since this principle applies to all control volume faces, the scheme becomes overall conservative. The approach is described in Peric and Schuerer [12] and Tew [9] in more detail.

AVAILABLE DATA FOR CAST COMPARISONS

Rectenwald Computations and Kornhauser Experimental Data

Recktenwald [15] computed heat transfer between the walls and gas inside the cylinder of a reciprocating compressor. A compressor cylinder contains intake and discharge valves, unlike the cylinders of a Stirling engine. Recktenwald used 2-D, unsteady, compressible equations (acoustics included) to simulate the compressor. In order to validate his computer code, he simulated a gas spring for comparison with data generated by Kornhauser and Smith [16, 17]. The comparison between data and experiment was based solely on experimental and simulated values of non-dimensional hysteresis loss for the gas spring over an operating range. It should be noted that a gas spring is essentially a piston-cylinder which has no flow to or from the enclosed volume of the cylinder. Kornhauser [18] reported further details of these gas spring experiments. He also reported on tests made with a modification of the gas spring test rig, to include a heat exchanger mounted on top of the cylinder, such that flow could continuously pass between the cylinder and the heat exchanger as the piston expanded and compressed the gas. This "two-space test rig" operated more like a Stirling machine cylinder than either a gas spring or a compressor.

Kornhauser's gas spring test data was also used as a basis for validation of the 2-D Modified CAST compressible non-acoustic code. Hysteresis losses computed by CAST were compared with the experimental values of Kornhauser and the calculated values of Recktenwald, over a range of gas spring operation. Also, since Recktenwald published plots of his calculated velocity vectors and temperature contours within the gas spring for two operating points, these were compared with similar plots of CAST calculated values for one of the operating points. Thus it was possible to compare the calculations of a compressible model, which did account for acoustics in the computations against the compressible non-acoustic calculations of the Modified CAST code. Comparison of CAST and Reckenwald's temperature contours over the cycle for a 10 RPM gas spring showed excellent agreement. Agreement between velocity vector plots also appeared to be very good, although due to some difference in the way the plots were made only a qualitative comparison could be made. These 2-D comparisons are shown in Tew [9].

Gas Spring and Two-Space Test Rig Dimensions

The dimensions of Kornhauser's test rigs are shown in Tables I and II. Due to a limitation of the CAST code in simulating complex geometries, it was necessary to make a slight change in the annular heat exchanger geometry simulated for the two-space test rig. The annular heat exchanger, physically mounted on top of the cylinder, was moved slightly outward so that the outer wall coincided with the cylinder wall, and the heat exchanger volume was maintained the same in order to maintain the same volume ratio. Physical and simulated dimensions are shown in Table III (This small change in geometry likely tended to reduce the difference between test and data results, to be discussed below).

Table II: Gas Spring Dimensions

Physical Quantity	*Symbol*	*Value*
Cylinder Bore (Diam.)	D	50.80 mm (2 in.)
Piston Stroke	S	76.2 mm (3 in.)
Volume Ratio	r_v	2.0

Table III: Two-Space Test Rig Dim. (Physical and Simulated)

Physical Quantity	*Physical Value*	*Simulation Value*
Cylinder Bore	50.80 mm (2 in)	50.80 mm (2 in)
Piston Stroke	76.20 mm (3 in)	76.20 mm (3 in)
Volume Ratio	2.0	2.0
Annulus O.D.	44.5 mm (1.75 in)	50.80 mm (2 in)
Annulus I.D.	39.4 mm (1.55 in)	46.4 mm (1.83 in)
Annulus Gap	2.5 mm (0.10 in)	2.2 mm (0.09 in)
Annulus Length	445 mm (17.5 in)	445 mm (17.5 in)
Min Pist/Head Clr.	2.9 mm (0.11 in)	2.9 mm (0.11 in)

Both test rigs used helium gas and the walls of the cylinder and heat exchanger were at a constant temperature of approximately 294 K.

RESULTS: COMPUTATIONS VS. DATA

Gas Spring Hysteresis Losses

For a gas spring, the hysteresis loss is the work that is dissipated by the spring per cycle at steady operating conditions; it's also equal to the heat generated in and transferred out of the spring. A good way to compare computational and measured hysteresis losses is via plots of dimensionless work as a function of oscillating flow Peclet number. Dimensionless work and oscillating flow Peclet numbers are defined, respectively, as follows:

$$\hat{W}_{loss} = \frac{\oint P\, dV}{P_o V_o \left(\dfrac{P_a}{P_o}\right)^2 \left(\dfrac{\gamma - 1}{\gamma}\right)} \tag{13}$$

$$Pe_{\omega} = \frac{\rho_o c_p \omega D_h^2}{4\lambda} \tag{14}$$

Recktenwald had previously plotted his calculated dimensionless losses on a plot of Kornhauser's data. The Modified CAST dimensionless losses were superimposed on this plot and the result is shown in Figure 1. This figure shows dimensionless loss as a function of oscillating flow Peclet number. The CAST loss values were plotted at "uncorrected" Peclet numbers as the obviously handwritten "X" symbols. If plotted at the corrected values of the Peclet numbers they would be shifted slightly to the right and would fall on Recktenwald's values (the solid diamonds).

Five of the six pairs (of CAST and Recktenwald values) of calculated dimensionless losses agree well with Kornhauser's data. The one pair of calculated points that did not agree well with the data is the pair shown at the highest Peclet number and corresponds to a 1000 RPM, 1465 kPa gas spring operating condition. Recktenwald [15] discusses several plausible explanations for disagreement at high Pe_{ω}.

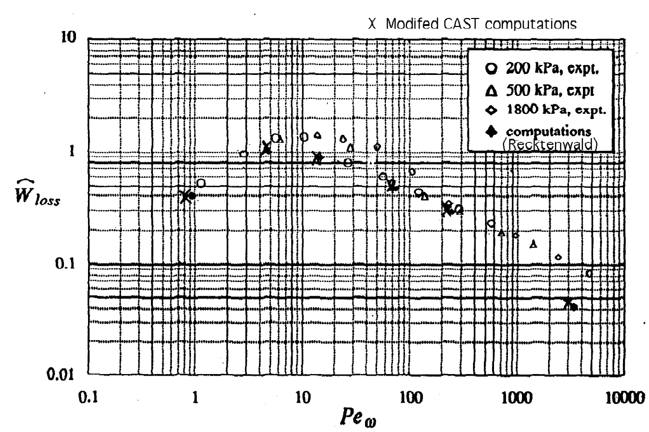

Figure 1: Modified CAST Dimensionless Losses Superimposed on Recktenwald [15] Plot of Recktenwald's Calculated Losses and Kornhauser's [18] Experimentally Derived Losses

Also in the mid-range of the data in the vicinity of Pe_{ω}=10, where there appears to be a higher and a lower data curve, the data agrees with the lower data curve. Kornhauser [18] found that the higher and lower data "curves" were related to differences between data taken at high-pressure/low-speed (higher curve) and that taken at low-pressure/high speed (lower curve) which had oscillating flow Peclet numbers in the same range. Thus he concluded there was some other dimensionless parameter needed to resolve the data in this range. His experiment also showed that adding fins within the clearance volume of the gas spring suppressed the difference in losses of the two types of data (high-pressure/low-speed and low-pressure/high-speed). See Kornhauser [18] for more discussion.

Comparison of modified CAST and experimental P-V diagrams are shown in Figures 2 and 3 for ~49 and 496 RPM respectively. Agreement is very good. When plotted in Microsoft Excel the curves were smooth. The 'jagged" nature of the curves appeared as an artifact of "pasting" the Excel plots into Microsoft Word.

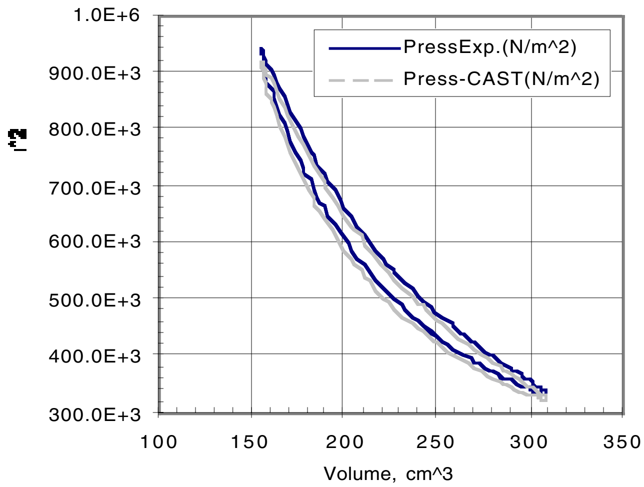

Figure 2: Comparison of Experimental and CAST P-V Diagrams. CAST used 18 x 12 grids, 120 time steps/cycle (Operating Conditions: 48.6 RPM, Mean Pressure = 555.7 kPa (80.6 psia), Wall Temp. = 294 K)

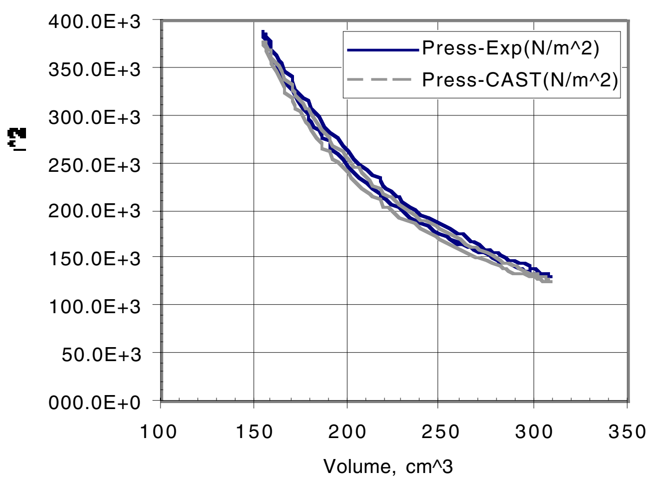

Figure 3: Comparison of Experimental and CAST P-V Diagrams. CAST used 18 x 12 grids, 120 time steps/cycle (Operating Conditions: 495.8 RPM, Mean Pressure=223.3 kPa (32.4 psia), Wall Temp. = 294 K)

Two-Space Test Rig Data and Calculation Comparisons

Figures 4 and 5 show experimental heat exchanger heat transfer per unit area and annulus center-to-wall temperature difference for Kornhauser's [18] two-space rig. Figures 6 and 7 show the corresponding Modified CAST calculated results for comparison with Figs. 4 and 5. The experimental and calculated heat transfer have different signs due to differences in definition of the positive heat transfer direction.

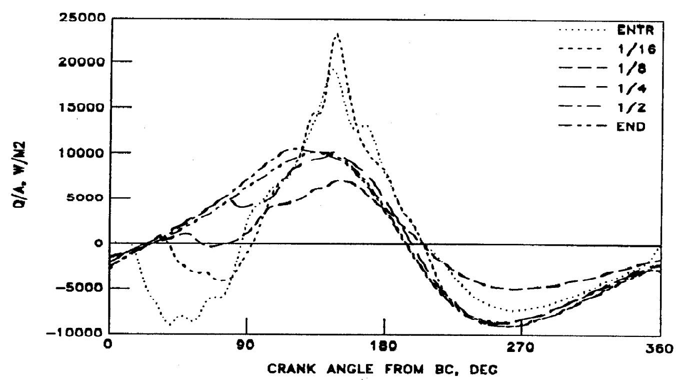

Figure 4: Heat Transfer per Unit Area at Various Positions in Heat Exchanger Relative to Entrance to Cylinder. Kornhauser [18] Exp. Data: Run #12071539, 201.7 RPM, 1.008 MPa Mean Pressure

Figures 4 and 6 show that peak experimental heat transfer/unit area near the entrance is 4000 to 5000 W/m^2 less than the corresponding calculated value; near the end of the heat exchanger the peak experimental value is about 9000 W/m^2 less than the corresponding calculated value. Thus even though the qualitative variations in heat transfer look very similar in the experimental and calculated plots, the quantitative agreement is not very good.

Comparison of the annulus center to wall temperature differences in Figs. 5 and 7 show that the calculated temperature differences are smaller that the experimental values. This is consistent with the calculated heat transfers/area being larger than the experimental values.

Two-dimensional plots of the calculated temperatures, velocities, pressures, etc. are given in Tew [9]. However, there are no experimental values available for comparison.

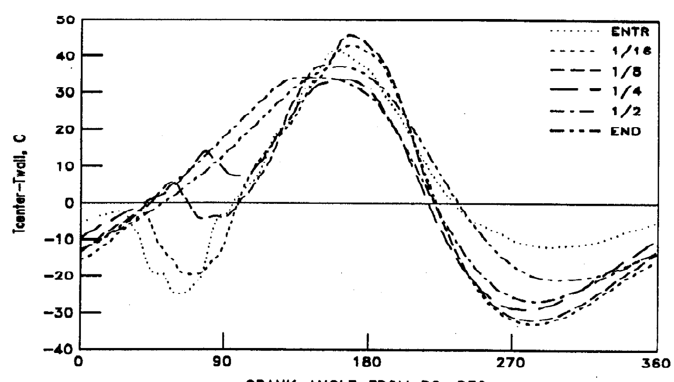

Figure 5: Temperature Difference from Heat Exchanger Center to Wall. Kornhauser [18] Exp. Data: Run #12071539, 201.7 RPM, 1.008 MPa Mean Pressure

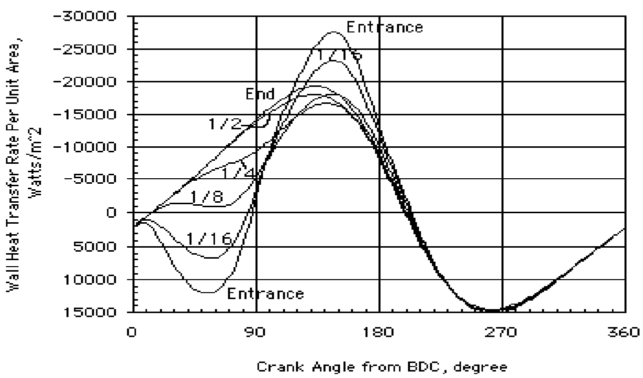

Figure 6: Modified CAST Calculation of Heat Transfer per Unit Area at Various Positions in Heat Exchanger Relative to Entrance to Cylinder (34x20 grids, 120 time steps/cycle). For Comparison to Kornhauser [18] Exp. Data: Run #12071539, 201.7 RPM, 1.008 MPa Mean Pressure

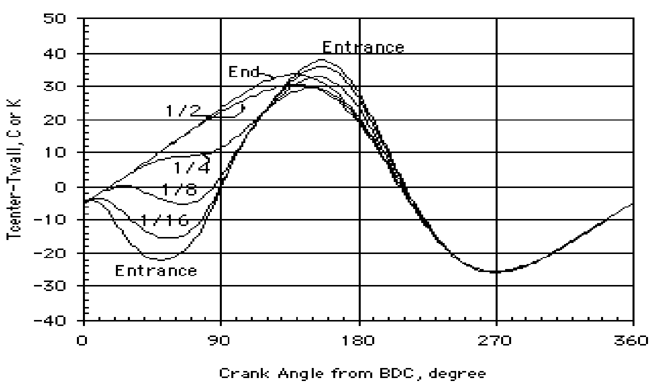

Figure 7: Modified CAST Calculations of Temperature Difference from Heat Exchanger Center to Wall (34x20 grids, 120 time steps/cycle). For Comparison with Kornhauser [18] Exp. Data: Run #12071539, 201.7 RPM, 1.008 MPa Mean Pressure

CONCLUDING REMARKS

The Modified CAST compressible non-acoustic model agreed well with 10 RPM gas spring hysteresis and P-V diagram data, and also with compressible acoustic calculations of two-dimensional velocities and temperatures. However, modified CAST calculations deviated from experimental values of two-space (piston/cylinder-heat exchanger) test rig data, although trends were predicted well.

Recent preliminary comparisons of the commercial CFD-ACE+ code compressible acoustic calculations with CAST, at CSU (Ibrahim, et. al. [19]), for the two-space rig suggests temperature agreement within about 2% for somewhat different grids. Therefore, data and 2-D computations do not agree, for some reason, for the two-space test rig. The reasons will be explored further. New data will likely be required for further multi-D Stirling code validation efforts.

When CAST and CFD-ACE+ calculations were made with the same type of grid and time step size, CFD-ACE+ was about 8% faster than CAST. Although not demonstrated here in comparisons against the highly developed commercial CFD-ACE+ code, Fedorchenko's compressible non-acoustic technique, may have the capability for reductions in simulation time for those transient situations where compressibility must be simulated but acoustics are not important.

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13. ABSTRACT ***(Maximum 200 words)***

A two-dimensional (2-D) computer code was developed for modeling enclosed volumes of gas with oscillating boundaries, such as Stirling machine components. An existing 2-D incompressible flow computer code, CAST, was used as the starting point for the project. CAST was modified to use the compressible non-acoustic Navier-Stokes equations to model an enclosed volume including an oscillating piston. The devices modeled have low Mach numbers and are sufficiently small that the time required for acoustics to propagate across them is negligible. Therefore, acoustics were excluded to enable more time efficient computation. Background information about the project is presented. The compressible non-acoustic flow assumptions are discussed. The governing equations used in the model are presented in transport equation format. A brief description is given of the numerical methods used. Comparisons of code predictions with experimental data are then discussed.

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