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Overview of NASA Supported Stirling Thermodynamic Loss Research

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ABSTRACT

The National Aeronautics and Space Administration (NASA) is funding research to characterize Stirling machine thermodynamic losses. NASA's primary goal is to improve Stirling design codes to support engine development for space and terrestrial power. However, much of the fundamental data is applicable to Stirling cooler and heat pump applications. The research results are reviewed. Much has been learned about oscillating-flow hydrodynamics, including laminar/ turbulent transition, and tabulated data has been documented for further analysis. Now, with a better understanding of the oscillating-flow field, it is time to begin measuring the effects of oscillating flow and oscillating pressure level on heat transfer in heat exchanger flow passages and in cylinders. This critical phase of the work is just beginning.

INTRODUCTION

NASA is funding research to characterize Stirling machine thermodynamic losses. This work is being accomplished via university grants and Small Business Innovation Research contracts. NASA's primary goal is to improve Stirling design codes to support engine development for space and terrestrial power.

Mechanical Technology, Inc. (MTI) built and tested the nominally 25 kWe, two module, Space Power Demonstrator Engine (SPDE). The SPDE was then divided into two 12.5 kWe Space Power Research Engines (SPRE) for testing at MTI and NASA Lewis. One of the SPRE's is still under test at NASA Lewis. Now, MTI is developing a second generation Stirling space engine, the Component Test Power Converter (CTPC). NASA also manages two DOE development contracts for Stirling solar terrestrial engines; the two contractors are the Stirling Technology Company and the Cummins Engine Co.

The areas of experimental research are: (1) Oscillating-flow (zero-mean) hydrodynamics and heat transfer; (2) oscillating-flow and oscillating-pressure level heat transfer; (3) oscillating-flow viscous losses and heat transfer in porous materials with axial temperature gradient; and (4) cylinder heat transfer with oscillating inflow/outflow. Two-dimensional (2-D) computations are supporting the experiments. Two-dimensional models can extrapolate data and provide insight beyond the limited experimental data base. An overview of this research was last reported at the 1990 IECEC [1]. This paper updates the research results.

NEED FOR STIRLING THERMODYNAMIC LOSS RESEARCH

Engines that are to compete for space power missions, and in the marketplace, need superior performance. Since development is so expensive, accurate design procedures are needed to minimize the hardware modifications required to achieve this performance.

HFAST and GLIMPS are the two major Stirling design codes used by NASA and its contractors. HFAST, developed by MTI, is used to develop space power engines. GLIMPS, developed by Gedeon Associates, is being used to develop the solar terrestrial designs.

NASA's experience indicates that Stirling thermodynamic losses are still poorly understood. Recent practice has been to use 20 percent design margins on engine power. However, these margins appear to have a strong tendency to shrink as engine and code development continues.

DIFFERENCES IN HFAST AND GLIMPS LOSS PREDICTIONS – Thermodynamic second law analysis has recently been included in HFAST and GLIMPS. In such codes, second law analysis is required to quantify the irreversibilities due to heat transfer in various components, mixing losses, fluid viscous losses, etc.

During 1991, Geng [2] compared predictions of HFAST and GLIMPS for the CTPC design. Important "available power" losses predicted by these codes are compared in Table I. The CTPC design operating conditions were: Nominal power = 12.5 kWe, Hot end temperature = 1050 K, Cold end temperature = 525 K, Mean pressure = 15 MPa, Frequency = 70 Hz. Predicted PV-power and efficiency were very close as shown in Table I. However, the major concern here is the large differences in losses.

Table I.— HFAST and GLIMPS Loss Predictions for CTPC.

	HFAST V2	GLIMPS V3
PV-Power, kW	14.87	14.58
PV-Efficiency	0.27	0.25
Available power losses, kW		
Viscous loss	2.64	1.65
Gas/Wall heat transfer		
Convection-heat exchangers	4.41	4.48
Hysteresis-heat exchangers	0.57	0
Hysteresis-cylinders	0.44	3.1
Total	5.42	7.58
Gas/matrix conduction (regenerator)	0.12	2.00
Mixing	1.26	0
Regenerator "heat leak"	1.2	5.3

Major differences in Table I are: (1) HFAST predicted 1 kW more viscous loss than GLIMPS; (2) GLIMPS predicted 2.5 kW more cylinder heat transfer or hysteresis loss than HFAST; (3) The 0.57 kW heat exchanger hysteresis loss calculated by HFAST is pressure-driven heat transfer loss (GLIMPS heat exchanger heat transfer is determined solely via the standard incompressible-flow correlation); (4) GLIMPS accounts for "enhanced conductivity" in the regenerator (HFAST does not); the result is almost 2 kW more of regenerator conductivity loss than HFAST (Also, the "total regenerator heat leak" predicted by GLIMPS is about 4 kW more than predicted by HFAST; part of this is due to the larger GLIMPS effective axial conductivity and the rest is due to larger "integrated enthalpy flux" over the cycle); (5) HFAST accounts for a 1.26 kW mixing loss, while GLIMPS accounts for no such loss (HFAST calculations imply a mixing loss because of the discontinuity in gas temperatures between adjacent control volumes; GLIMPS calculations imply no mixing loss because the fluid temperature is assumed to be continuous from one control volume to the next).

The above differences are significant. Since optimization trades off the various losses to arrive at a minimum total loss (if optimizing efficiency), it is likely that the two codes would arrive at significantly different CTPC geometries if used to optimize that design (The CTPC was designed with an earlier version of HFAST).

A more comprehensive breakdown of these loss predictions was presented to MTI and Gedeon at a meeting at NASA Lewis in Oct. 1991. After subsequent checking into their codes, both code developers found some errors that needed correction. GLIMPS modifications, for example, have resulted in a reduction in predicted cylinder heat transfer loss (by roughly 40 to 50 percent). However there are still major differences in this and other losses. More recent code comparisons were made in March 1992 for the RE-1000 and SPRE engines [3].

IMPACT OF REVISED TRANSITION CRITERION ON HFAST PREDICTIONS – HFAST normally uses steady-flow friction factor and heat transfer correlations, including a steady-flow transition criterion. Following University of Minnesota (U. of Minn.) oscillating-flow experiments, Seume suggested a "first cut" oscillating-flow transition criterion [4]. Gedeon implemented a version of this revised criterion in the GLIMPS code [5]; it is now the standard GLIMPS criterion. Huang of MTI recently implemented the GLIMPS transition criterion in HFAST to check its impact on engine predictions [6].

The differences in the two transition criteria are shown in Fig. 1. The steady-flow criterion implies laminar flow below Reynolds number, Re = 2000; transition occurs linearly with Re between 2000 and 10 000; above Re = 10 000, the flow is turbulent. In contrast, for the GLIMPS oscillating-flow criterion, the "all laminar" and "all turbulent" lines are seen to be a function of Valensi number. Also, between these two limiting lines, turbulence is solely due to "convective triggering." That is, at flow reversal, all fluid in a tube becomes laminar. As fluid flows into the tube, flow separation is assumed to trigger transition to turbulence. This turbulent "front" travels down the tube until flow becomes zero again. Then, after flow reversal, the process repeats from the other end of the tube.

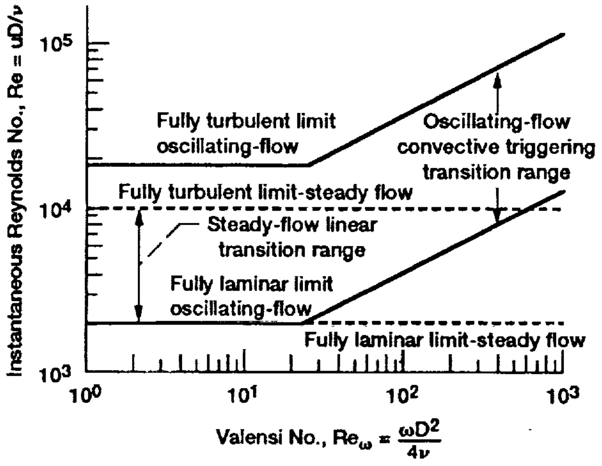


Figure 1.—Comparison of steady-flow and oscillating-flow transition criteria.

Summary charts from Huang's sensitivity study are shown in Tables II and III (The versions of HFAST used for the calculations in Tables I and II were not identical). Table II shows that the change in transition criterion had little effect for the CTPC design. However, Table III shows the change had a major impact on SPRE performance. The difference in

Table II.—Sensitivity of HFAST Predictions to Transition Criterion for CTPC.

CTPC engine conditions: 15 MPa, 70.3 Hz, helium heater at 1050 K, cooler at 525 K	Steady-flow transition model	Oscillating-flow transition model
Piston cycle power, kW	13.274	13.068
Piston cycle efficiency	0.246	0.243
Heater		
Maximum Reynolds number	6951	7002
Valensi number	19.1	19.2
Heat transfer, kW	52.858	52.752
Heat transfer available power loss, kW	1.392	1.488
Cooler		
Maximum Reynolds number	10 082	10 134
Valensi number	31.4	31.5
Heat transfer, kW	32.221	32.139
Heat transfer available power loss, kW	1.110	1.045

Table III.—Sensitivity of HFAST Predictions to Transition Criterion for SPRE.

SPRE engine conditions: 15 MPa, 101.23 Hz, helium heater at 642 K, cooler at 323 K	Steady-flow transition model	Oscillating-flow transition model
Piston cycle power, kW	12.227	9.435
Piston cycle efficiency	0.222	0.176
Heater		
Maximum Reynolds number	15 819	17 086
Valensi number	95.1	105.1
Heat transfer, kW	53.842	51.457
Heat transfer available power loss, kW	1.336	2.124
Cooler		
Maximum Reynolds number	35 741	34 262
Valensi number	379	357
Heat transfer, kW	37.845	38.603
Heat transfer available power loss, kW	1.836	3.323

sensitivity is related to the higher Valensi numbers for the SPRE heater and cooler (due to differences in tube size, frequency, and temperature level). The CTPC heater and cooler Valensi numbers are near the "knees" of the limiting curves, while the SPRE Valensi numbers are well up on the nonzero slope portion of the limiting curves. The nonzero slope portion of the curve reflects experimental findings that fluid acceleration delays transition from laminar-to-turbulent flow. Thus, for the SPRE, the GLIMPS oscillating-flow criterion predicts laminar flow over a significantly larger portion of the high Reynolds number part of the cycle, than the steady-flow criterion.

SPRE fluid displacement ratios (defined in the paragraph below) were also somewhat smaller than for the CTPC. Thus, in the region between the two limiting curves (Fig. 1), convective triggering was not as effective in triggering turbulence for the SPRE.

OVERVIEW OF LOSS RESEARCH EFFORTS

UNIVERSITY OF MINNESOTA OSCILLATING-FLOW RIG: EXPERIMENTS AND 2-D MODELING – The design, construction, and use of a large-scale, low-frequency, oscillating-flow rig for research into oscillating-flow hydrodynamics, has been guided by Simon. First, the literature was reviewed. Preparatory work established that appropriate dimensionless parameters for characterizing the hydrodynamic operating conditions of Stirling heat exchangers are maximum Reynolds number, Valensi number (or dimensionless frequency), and fluid displacement ratio, as follows:

$$Re_{max} = \frac{u_{max} D}{\nu} \qquad Re_{\omega} = \frac{\omega D^2}{4\nu} \qquad A_r = \frac{1}{2}\frac{D}{L}\frac{Re_{max}}{Re_{\omega}}$$

The dimensionless operating conditions for the heat exchangers of many Stirling engines were calculated and plotted [7].

Results from this test rig have recently been documented in a two-volume NASA Contractor report. The first volume is the written report [8] and the second volume contains the tabulated data. Early results showed: (1) In general, oscillating flows in the range of Stirling engine dimensionless parameters undergo transition from laminar-to-turbulent and back, twice per engine cycle; however, a few test conditions showed all laminar or all turbulent conditions over the entire cycle; (2) fluid acceleration delays transition from laminar-to-turbulent flow, while fluid deceleration delays relaminarization; the net effect is the flow stays laminar over a larger portion of the high Reynolds number part of the cycle than implied by the steady-flow transition criterion; (3) turbulence generated in test rig exit plenums during outflow persisted and was ingested into the test section as a turbulent slug at flow reversal.

Seume hypothesized that in Stirling heat exchangers, where sudden area changes occur at tube inlets/outlets, convective triggering of turbulence might be the most important path to transition; it's relative importance would be strongly dependent on the fluid displacement ratio, A_r. There would be no convective triggering, except very close to the ends, with infinitely small displacement ratios ($A_r \sim= 0$). Heater and cooler A_r's for the SPRE and CTPC are in, or near, the 1 to 3 range. For the GLIMPS transition criterion (Fig. 1), convective triggering is the sole basis of transition in the region between the limiting curves; crossing of the upper limiting curve implies boundary-layer transition.

More recent U. of Minn. test results [8] are: (1) The first experimentally determined, instantaneous, oscillating-flow friction factors; examples of these are shown in Fig. 2 for several test-section locations. Measured oscillating-flow values are compared with steady, fully-developed flow correlations in Fig. 3 for two test-section locations; during the initial laminar-flow portion of the cycle, the test values are substantially higher than those determined from steady-flow correlations. Two-dimensional unsteady laminar calculations are consistent with the test values if transient, developing flow is assumed [8]. It appears that turbulence persists until the flow is zero and helps in creating a near uniform velocity as the flow accelerates from zero-flow in the following cycle. Thus, during the laminar portion of the cycle, the fluid is apparently

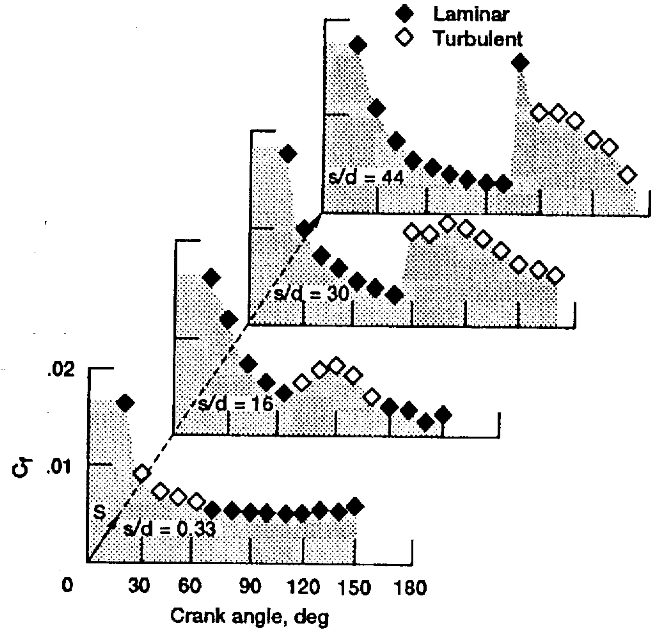
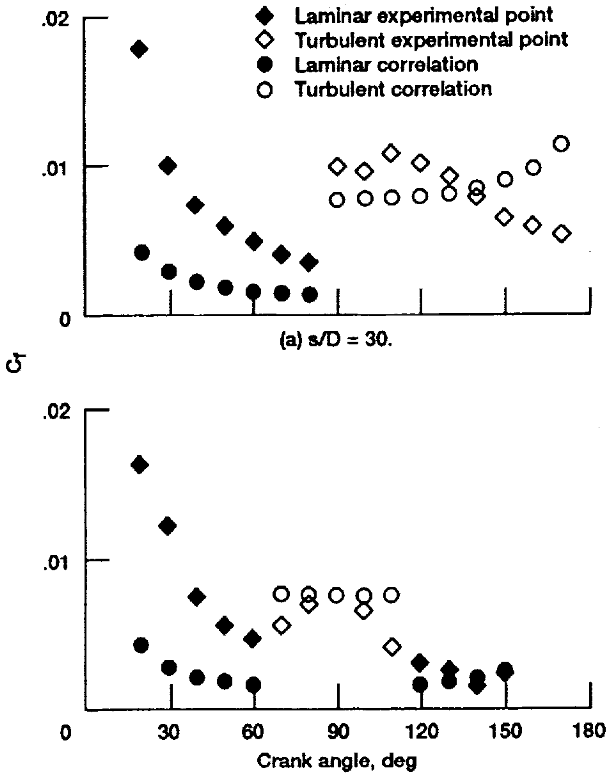


Figure 2.—Experimental, instantaneous, oscillating-flow friction factors (inlet at s/D = 0, outlet at s/D = 60, for half-cycle shown).



(a) s/D = 30.

(b) s/D = 16.

Figure 3.—Comparison of experimental, oscillating-flow, friction factors and steady-flow friction factors, when the measured transition points are used with the steady friction-factor correlation.

undergoing a transient from zero-flow (until transition occurs)—as opposed to executing part of a steady-periodic cycle; (2) a procedure for developing an empirical transition model for use in 2-D calculations has been outlined; the U. of Minn. and Cleveland State University (CSU) have been collaborating on implementation of such a model [9].

The latest U. of Minn. testing involved flow visualization in the vicinity of tubular end geometries [10]. Oscillating-flow heat transfer testing will soon be underway. A new technique for determining wall-temperature, by extrapolation of fluid temperatures, is expected to permit determination of instantaneous surface temperature and heat transfer coefficients.

Patankar is guiding the use of a low-Reynolds number k-epsilon turbulence model in developing a turbulence/transition model [11]; test rig turbulence measurements are providing the data to check this model. Early problems were: (1) transition was predicted too early (acceleration delay apparently not accounted for); (2) turbulent slugs which entered the tube from adjacent plenums dissipated much too quickly (it appeared that assumed boundary values of turbulent dissipation were inappropriate). If successful, the 2-D transition model will be used to generate new friction-factor and heat transfer correlations for use in Stirling one-dimensional (1-D) design codes.

MASSACHUSETTS INSTITUTE OF TECHNOLOGY (MIT) OSCILLATING-FLOW AND OSCILLATING-PRESSURE LEVEL TEST RIG – Smith has guided the design and construction of a new large-scale, low-frequency, oscillating-flow and oscillating pressure level test rig at MIT, with financial support from DOE and NASA. Heat transfer measurements will be made in a tubular test section.

The U. of Minn. tests have been at nearly constant temperature and pressure and are, therefore, essentially incompressible flow tests (air was the working fluid). Also, steady-flow heat transfer correlations, of the type that are used in Stirling design codes, are based on incompressible flow test results; they do not properly account for pressure-driven heat transfer.

However, the HFAST code, does separately account for pressure driven heat transfer in the heat exchangers, using an equation derived from MIT gas spring testing; this pressure-driven heat transfer is then superimposed on the flow-driven heat transfer (calculated with a standard steady-flow heat transfer correlation). This appears to be an improvement but, due to the linear superposition, may be "double accounting" for some heat transfer. The HFAST heat exchanger hysteresis loss in Table I is such a calculation for the CTPC design. GLIMPS does not account for any heat transfer in the heat exchangers, beyond that calculated with standard steady-flow heat transfer correlations.

Earlier gas spring tests at MIT [12] showed that, in general, heat transfer leads the wall-to-mean-gas temperature difference. To account for this in 1-D models, a complex heat transfer coefficient was proposed; the real part is proportional to the temperature difference and the imaginary part is proportional to the rate of change of gas temperature. Gas spring heat transfer, primarily pressure driven, typically leads to a "hysteresis" power loss (only adiabatic or isothermal processes avoid this loss).

A temporary test section, derived from earlier gas spring testing, was initially used in the new MIT rig. Initial test results [13] were taken with the two "opposed" pistons moving exactly out of phase. So there was a relatively large variation in pressure level with relatively small gas flow in the tubular test section, similar to the conditions in a gas spring. The results were in qualitative agreement with gas spring tests; that is, the heat transfer was found to lead the wall-to-mean-gas temperature difference. The next phase of the work is to install the primary "wide range" test section together with appropriate instrumentation (including a laser doppler velocimeter for gas velocity measurements) and measure heat transfer with various phase angles between the two pistons [14].

Jeong [15] (not supported by NASA) did a laminar-flow analysis of gas spring phenomena and reached a number of interesting conclusions: For sufficiently large dimensionless frequencies so that the Stokes layer is small relative to the cylinder diameter: (1) A steady large-scale recirculating flow moves from the piston to the closed end, near the wall, and in the opposite direction in the core; (2) this recirculating flow implies the likelihood of shear layer instability and turbulence; (3) analytically, the recirculating flow is due to three steady vorticity generation terms, two of which are due to gas compressibility effects (density variations due to time varying pressure and spatially varying temperature); (4) time average gas temperature peaks at a location between the core and the cylinder wall. Increasing frequency moves the peak closer to the wall. The peak is caused by oscillating-velocity normal to the wall (driven by oscillating pressure level and, corresponding, oscillating gas temperature and radial gas temperature gradient); this results in a net transfer of energy normal to the wall via mechanical energy flux, due to the phase shift between pressure and normal velocity (and conversion to heat near the wall). This temperature peak between the core and the wall helps explain why heat transfer should not be proportional to the wall-to-mean-gas-temperature difference (or the need for a complex heat transfer coefficient in 1-D models).

Jeong also found that, in the mid-frequency range, cylinder heat transfer was sensitive to Mach number changes in the range from 0.005 to 0.1. This seemed related to the recirculating flows and, perhaps, turbulence. Therefore, transition to turbulence in closed gas cylinders may be sensitive to gas-compressibility effects.

VIRGINIA POLYTECHNIC INSTITUTE (VPI) CYLINDER HEAT TRANSFER TEST RIG – GLIMPS has been shown to calculate much larger cylinder heat transfer losses than HFAST. HFAST uses equations derived from MIT gas spring data for cylinder and heat exchanger hysteresis losses; the data upon which these equations are based did not involve inflow and outflow. The GLIMPS calculation is based on a cylinder heat transfer analysis derived by Gedeon which accounts for turbulence [16]. Kornhauser of VPI has examined another model which was similar to the GLIMPS model, in concept, but used different values for many of the equivalent parameters [17]; it also tends to calculate smaller cylinder losses. It seemed apparent that experimental data was needed to define appropriate values of the model parameters. A test rig for measuring cylinder heat transfer over a range of operating conditions is now being developed at VPI [18], under Kornhauser's guidance.

SUNPOWER/OHIO UNIVERSITY (OHIO U.) OSCILLATING-FLOW TEST RIG – This rig was originally designed to investigate oscillating-flow viscous flow losses in tubes and porous materials. The results are documented in terms of ratios of measured oscillating-flow viscous losses to calculated steady-flow viscous losses for combined end-effects and core-friction [19]. These results should be useful for estimating the effect of oscillating-flow on viscous losses in tubes and matrices.

The results also suggest a tentative design guideline for Stirling heat exchanger design. That is, if fluid displacement ratio is maintained sufficiently large (>2 may be adequate) then it appears that the flow is mostly turbulent and the steady-flow turbulent friction-factor correlation may be adequate for predicting viscous losses (no experimental information is available for heat transfer). These results seem consistent with Seume's hypothesis that laminar oscillating flow is "convectively triggered" to turbulence as it enters the tube of a Stirling engine heat exchanger. Small fluid displacement ratios (<1) resulted in viscous losses that were less than predicted by steady-flow correlations; this result is consistent with delay of laminar-to-turbulent transition during fluid acceleration.

The test rig, now loaned by NASA to Ohio U., has been rebuilt to measure heat transfer in regenerators under oscillating-flow conditions [20]. Since heat stored, and removed, from the regenerator matrix each half-cycle is typically four or five times the amount of heat entering the engine per cycle, regenerator effectiveness has a major impact on engine efficiency. The GLIMPS and HFAST codes show major disagreements in predictions of regenerator effectiveness. For example, Table I indicates that GLIMPS calculated a large enhanced conductivity which significantly increased the net regenerator "heat leak" in the CTPC design. In contrast HFAST, which does not account for any "enhanced conductivity," predicted 4 kW less heat leak than GLIMPS.

CLEVELAND STATE UNIVERSITY COMPUTATIONAL EFFORT – Ibrahim of CSU is guiding development of 2-D Stirling component models. Heaters, coolers, and flat-plate regenerators were initially modeled assuming pulsating (nonzero mean), incompressible, laminar, fully developed flow and heat transfer. The models have been generalized to account for oscillating (zero-mean) flow, conjugate heat transfer, and developing flow and heat transfer. Density variations due to temperature changes are now accounted for (thermal expandability) and density variation due to pressure has been introduced for steady and pulsating flow. A high Reynolds number turbulence model is coded but has not yet been "turned on." Full compressibility has not yet been used for oscillating-flow conditions.

CSU and the U. of Minn. worked together to demonstrate that the laminar, accelerating flow portion of the Minnesota test data is transient developing flow [8] instead of steady-periodic flow; they are now working on development of an empirical transition model for use in a 2-D oscillating-flow code [9]. CSU has also developed a sudden expansion/contraction model for laminar oscillating flow that has been used to study the impact of area changes on the flow field and on heat transfer [21].

NASA LEWIS IN-HOUSE SUPPORT FOR THE STIRLING LOSS RESEARCH – Mankbadi of the Lewis Research Academy has assisted in monitoring the turbulence modeling portion of the oscillating-flow work. He has warned that quasi-steady turbulence models do not strictly apply to oscillating flow in the vicinity of zero flow [22,23]. However, it has not yet been determined whether the deviation from quasi-steadiness has a significant impact on predictions of turbulent fluid flow and heat transfer in Stirling engines. Mankbadi also suggested consideration of an empirical transition model for 2-D component simulation purposes. He also supervised a preliminary look at improving the accuracy of the Rapid Distortion Theory turbulence model in the low dimensionless frequency range (it has good accuracy in the high frequency range).

Kim of the Computational Methods for Space Branch has applied a multiple-scale k-epsilon model to a wide range of dimensionless frequencies for pulsating flow [24]. These results indicate such a model can give good accuracy over a wide frequency range. The multiple-scale approach permits representation of time delays required for turbulence to cascade from larger to smaller scales. It seems to offer hope of improving representation of phase lags in calculation of oscillating-flow turbulence.

CONCLUDING REMARKS

Tests have provided a good qualitative understanding of the hydrodynamics of oscillating-flow, including laminar/turbulent transition in tubes, and much tabulated data; efforts to produce practical quantitative descriptions based on the data are underway. Viscous loss tests have indicated that continuous turbulence may be maintained in tubes by "convective triggering" for oscillating-flow conditions, if $A_r >$ about 2; under these conditions, steady-flow turbulent correlations might be adequate for predictions. A sensitivity study with a "first cut" oscillating-flow and a steady-flow transition model showed that (1) CTPC performance was about the same with either transition model, but (2) SPRE performance dropped dramatically when a switch was made from the steady- to the oscillating-flow transition model. HFAST and GLIMPS loss comparisons have shown major disagreements in magnitudes and types of losses.

The critical heat transfer testing has just begun. The regenerator oscillating-flow rig at Ohio U. has begun to produce data. The U. of Minn. is ready to begin oscillating, incompressible flow heat transfer testing. The MIT test rig is about ready to use a "wide range" test section to measure the effects of oscillating flow and pressure level on heat transfer. The VPI cylinder heat transfer rig is scheduled to begin producing data in the fall of 1992. These heat transfer measurements are necessary for proper characterization of Stirling thermodynamic losses.

Stirling design codes are evolving along with engine development and testing. The thermodynamic loss research, by providing a source of new fundamental data, has also begun driving code evolution. Measurements of the fundamental physical phenomena which occur inside Stirling devices, offer the best hope of adequately characterizing Stirling thermodynamics.

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

The National Aeronautics and Space Administration (NASA) is funding research to characterize Stirling machine thermodynamic losses. NASA's primary goal is to improve Stirling design codes to support engine development for space and terrestrial power. However, much of the fundamental data is applicable to Stirling cooler and heat pump applications. The research results are reviewed. Much has been learned about oscillating-flow hydrodynamics, including laminar/turbulent transition, and tabulated data has been documented for further analysis. Now, with a better understanding of the oscillating-flow field, it is time to begin measuring the effects of oscillating flow and oscillating pressure level on heat transfer in heat exchanger flow passages and in cylinders. This critical phase of the work is just beginning.

14. SUBJECT TERMS
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