HEAT FLOWS IN PISTON COMPRESSORS

U. Lekić, J.B.W. Kok
University of Twente,
Faculty of engineering technology,
Enschede, The Netherlands

Abstract

Piston compressors are widely used in today's engineering applications. Among the most important applications is however the compression of thermal carrier gas in Rankine and Stirling refrigeration cycles. Fluids used in these cycles are commonly Ammonia and Helium. In order to improve the design and reliability of these compressor designs, transient thermal and fluid flow phenomena and thermo-mechanical processes in an unlubricated piston-cylinder gas spring are investigated. Conventional Nu-Re correlations, as derived for steady-state developed boundary-layer flows, are not applicable for the transient heat transfer between the gas undertaking rapid compression and expansion and the surrounding walls. In this paper results are presented of CFD simulations of laminar and turbulent flows inside the compressed volume and corresponding heat flows. Parameters varied in the present work are the compression ratio and operating frequency. The CFD results are compared with heat flux data post processed from pressure recordings performed at a test rig at MIT. In the post processing essential thermodynamic assumptions were made. It is shown that at low RPM and hence laminar flow, CFD results on heat flux compare well with these measurements. At more practical high RPM and turbulent fluid flow the CFD result differ significantly from the post-processed data. This can be explained by the break down of the thermodynamic assumptions in the measurement data processing. In the new and highly advanced test rig built at the University of Twente, heat fluxes will be measured directly.

Keywords: unlubricated PCC, transient heat flux, CFD modelling

1 Introduction

Conventionally designed piston-cylinder based compressors require the use of a lubricant for efficient and long-time performance. Lubrication presents a source of pollution and contamination of both the working medium and the cooling system, accompanied by the compatibility issues and diffusion effects of the lubricant on the interacting parts, thus diminishing the overall cooling performance. Effort is put into development of novel friction-wear control and lubrication free solutions for future compressors. With advanced steel grades it is already possible to prevent cold welding occurrence and significantly reduce delamination.

Transient flow and heat transfer phenomena occurring in a closed piston-cylinder gas spring are being explored in the research presented here. Established Nu-Re correlations for determining the gas to wall heat transfer can not be used here. These correlations are developed for steady state developed boundary layer flow in long pipes. These conditions are far from the flow situation encountered in a piston cylinder gas spring. Here the flow is oscillatory in character: almost arrested in top and bottom dead position and at maximum speed in between. There are also strong axial and radial velocity gradients. Axial gas velocity equals the piston velocity at its surface, and vanishes at the cylinder head; the axial motion excites radial motion and turbulence. Hence complicated transient and local fluid mechanical and thermodynamic phenomena have to be taken into account. At present no universal model for accurate heat transfer predictions exists for this application. An overview of available models is presented in Section 2. CFD and CSD techniques are used for numerical modelling. Problems discussed in this paper are suitable turbulence models, near wall heat transfer modelling and modelling of the fluid flow structures developing in the gas flow above the piston. These analyses are reported in Section 3. In section 4 the CFD results for heat flux and pressure will be compared with experimental data. Crucial in this is the indirect way the heat flux is post processed from measured pressure data. For low piston speed and laminar flow excellent comparison is obtained of the predicted and measured data. At high piston speeds and turbulent flow this comparison fails. This is not due to errors in prediction or measurement but due to the post processing and the thermodynamic assumptions involved. It is concluded that the validation of the presented models is hampered by the lack of consistent and detailed experimental data on high piston speed flows. In order to provide good data a novel experimental setup is built and operated at the University of Twente. It will enable precise measurements of various relevant properties, including pressure, surface temperature and heat-flux measurements. The influence of different compression ratios and piston geometries on the development of fluid-flow structures will be possible investigate. The construction of the experimental setup is described in short in Section 5.

2 Heat transfer predictions and phase shift

In a piston gas compressor, heat transfer is a combination of forced convection and conduction, as radiation can be neglected. With the piston reciprocating in axial direction, transient velocity and thermal boundary layers occur. The time dependent characteristics like thickness and structure of the boundary layers, turbulence intensity, density and temperature of the enclosed gas determine the heat transfer to the solid walls. A phase shift can be observed between the heat flux at the cylinder walls and the peak temperature instant of the bulk gas volume. Heat transfer peaks can precede or lag behind the driving potential known from steady state heat transfer: the difference between the bulk gas and bounding walls temperatures. This was reported from experiments for both motored (Lawton [6]) and fired piston engines (Oude Nijeweme [10]). The phenomenon can be explained by the phasing property of the work term in the simplified one-dimensional energy transport equation:

$$\rho C_p \frac{\partial T}{\partial t} + \rho v C_p \frac{\partial T}{\partial y} = \frac{\partial q_y}{\partial y} + \frac{dp}{dt}, \tag{1}$$

with y the coordinate measuring the distance from the wall.

Equation 1, when analyzed, indicates that the pressure work term induces the phasing of the heat transfer. The measured peak heat flux shift ranges from approximately 8° before Top-Dead-Centre - TDC (Lawton [6]), to as much as 45° (Pfriem [11] and Lee [7]). Phase shift phenomenon will be predicted and studied in the present work.

While pressure work induces a phase shift in the heat flux, the main term that quantifies the transient heat transfer, is the time-dependent and oscillatory convection. Schlieren photos taken in a boundary layer study, performed by Lyford-Pike and Heywood [8], show that the boundary layers on the crown of the piston and the cylinder head are two to three times thicker than the layer on the cylinder-liner. This illustrates the likeliness of non-uniform and transient surface heat fluxes. Conventional Nusselt-Reynolds wall heat transfer models developed for steady state pipe flow can not be applied here. Efforts were made to have them adapted by a range of constants to fit a specific compressor, but that would of course not lead to a generic relation.

Kornhauser [5] defined a “complex Nusselt number”, correlation. This expressed the heat flux as a sum of a part proportional to temperature and a second part proportional to the rate of change of temperature. This correlation is presented in equation 2.

$$\dot{q}'' = \frac{k}{D_h} Nu_c (T_c - T_w) = \frac{k}{D_h} [Nu_r (T - T_w) + \frac{Nu_i}{w} \frac{dT}{dt}] \tag{2}$$

He reported very good agreement with measured values over a range of operating conditions. Still, the correlation broke down for flushed cylinders operating in the range $10<Pe_\omega<100$, which is the most-common operating range in reciprocating machinery. Another issue here is that in Kornhauser's work, the measured heat flux, which is a local phenomenon, is computed on the basis of the First Law of Thermodynamics, in correlation to the work loss. In this method the temperature term in the internal energy formulation is calculated from the measured pressure and thus a volume-averaged temperature. This is a questionable application of global thermodynamic laws on the local non-equilibrium heat and fluid flow phenomena. Also Pfriem and Lee derived complex Nusselt numbers, using in their correlations oscillating Peclet numbers rather than Reynolds. In sections 3 and 4, experimental results of Kornhauser will be compared with the results obtained in the present study, by the 3D numerical simulation package CFX11.0.

3 Computational Fluid Dynamics predictions

3.1 Isentropic compression and expansion

To model the fluid and energy flows in the piston cylinder gas spring, in this project, the ANSYS CFX11.0 CFD package has been used. Three typical cases are modelled: a gas spring simulated as an adiabatic gas volume, for the validation of thermodynamic conservation laws, followed by two cases with heat transfer taken into account. These two non-adiabatic cases comprise a laminar and a turbulent run at 2 and 1000 RPM operating frequency respectively. The turbulence is modelled using models available in CFX: k-ω based SST , SAS-SST and Smagorinsky LES. Conceptual differences of the turbulence models will not be discussed here.

The experimental data-base made available by Kornhauser is used here as the only available, but carefully set up and well documented, validation source. Kornhauser's setup was a single-cylinder piston-cylinder, crankshaft driven, apparatus, with piston diameter 50.8 mm and 76.2 mm stroke. The CFX model geometry and mesh were dimensioned accordingly. For computational time reduction, and assuming the flow to be axi-symmetric, modelled is only a 5° slice of this computational domain. The moving mesh is uniform, the piston is the only moving boundary and the mesh elements are allowed to deform and redistribute in axial direction. After consecutive spatial and temporal refinements, the model is optimized to 6363 nodes (5400 elements) and 200 time steps per cycle. Every node displacement is specified through an explicit model in order to ensure uniform cell-distortion. The compressed gas is specified as Helium, following the perfect gas equation of equation.

The simplest case is presented first: the gas spring enclosed by adiabatic walls. The computed pressure along the cycle needs to comply with the perfect gas Clapeyron equation (pV=mRT) and show the model's energy conservation and convergence capabilities. The results obtained for all 3 turbulence models (k-ω, SAS, LES) are observed to be exactly coinciding with the theoretical isentropic correlation. The pressure coincides with the analytical solution and the energy is conserved over the simulated period (10 cycles). As no difference can be observed between simulations and analytical results, these are not shown here. More challenging and interesting after this check of conservation of the CFD method will be the cases taking into account heat loss in laminar and turbulent flows.

3.2 The laminar flow case

Identical geometry and mesh are used for both laminar and turbulent runs. The laminar run is characterized by a low operating frequency of 2 RPM to ensure absence of turbulence. The walls have as boundary conditions no-slip of velocity. The piston surface (moving boundary) is set as an adiabatic wall and the cylinder and top (head) walls are defined to have a constant temperature of 295K, equal to the temperature of the coolant on the other side of the gas-spring walls. This assumption was made due to the very small actual oscillations in the wall temperature. This was measured for motored piston engines by Adair et al [1], Anand and Pinfold [2], Boroman and Nishiwaki [3], to be lower than ±3K), due to the large thermal time constant of the wall. Validity of this assumption was examined by Jak [4], and he also showed that the wall temperature can be assumed to be constant with a view to the effect on the gas heat flux. Analysis of the oscillating surface temperature and heat penetration through the wall are to be reported in a subsequent paper.

Initial conditions are set to comply with Kornhauser's measured data; the runs are started from the Bottom-Dead-Centre – BDC position. Kornhauser's measurements were set to start when the gas spring has reached the cyclic steady state, i.e. when the mean cycle pressure remained constant (less than 1% deviation). Results obtained are presented in Figure 1.

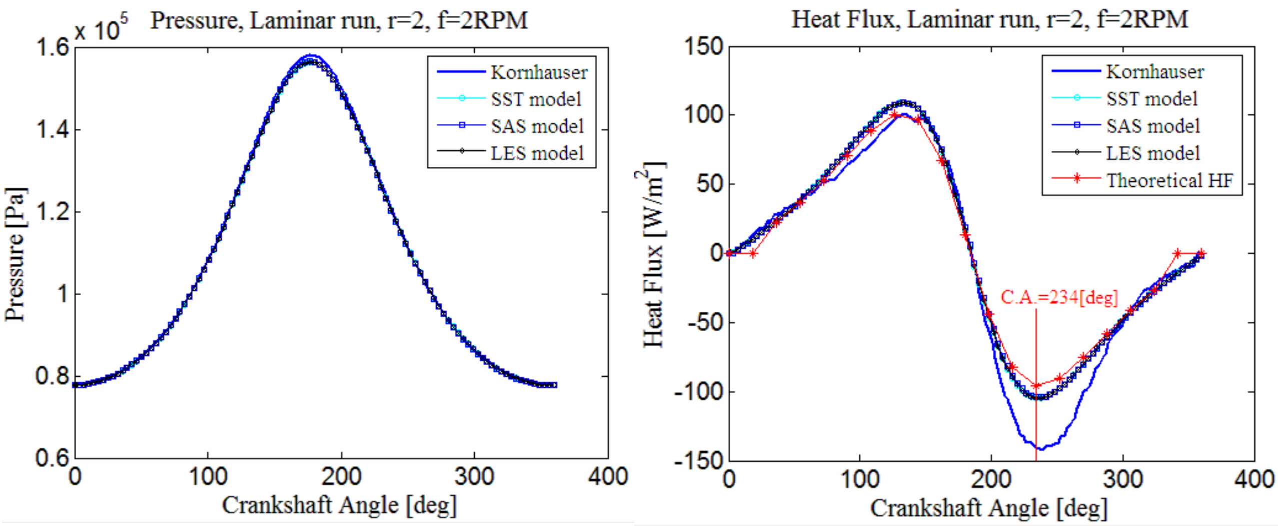

Figure 1: *Laminar run - pressure of the compressed gas (left) and the heat transfer to the walls(right)*

Figure 1 left shows the pressure as a function of crankshaft angle to oscillate periodically between 80 kPa and 160 kPa. The maximum pressure is reached at top dead centre, at 180 degrees crank angle. There is a very good comparison between the Kornhauser experimental data and all 3 different CFD simulations. It was to be expected that all 3 turbulence models are able to predict the laminar flow well. Figure 1 right shows the transient heat flux to be positive in the compression phase of the cycle. Past top dead centre the expansion starts and the heat flux becomes negative, with a maximum negative heat flux at 234 degrees crank angle. The CFD predictions compare again very well with the experimental data. Interesting curve is the one denoted with "theoretical heat flux". This is the heat flux derived via the Kornhauser method using thermodynamic conservation laws (see eq's 3-8 below), but on basis of the CFD predicted pressure. This is the only discrepancy and over-predicts the negative heat flux minimum by 50%. Still it can be concluded that CFD is very well able to predict the transient pressure. The transient heat flux predicted by CFD compares well with the heat flux as determined by global thermodynamic variables and conservation laws of the gas volume on basis of the measured or predicted pressure.

3.3 The turbulent flow case

The measurements of fast runs done by Kornhauser are at 1000 RPM and report turbulent flow inside the spring. The boundary conditions in the CFD simulation are identical to those in the laminar runs. The initial conditions are adapted to the experimental data. The results obtained are presented in Figure 2.

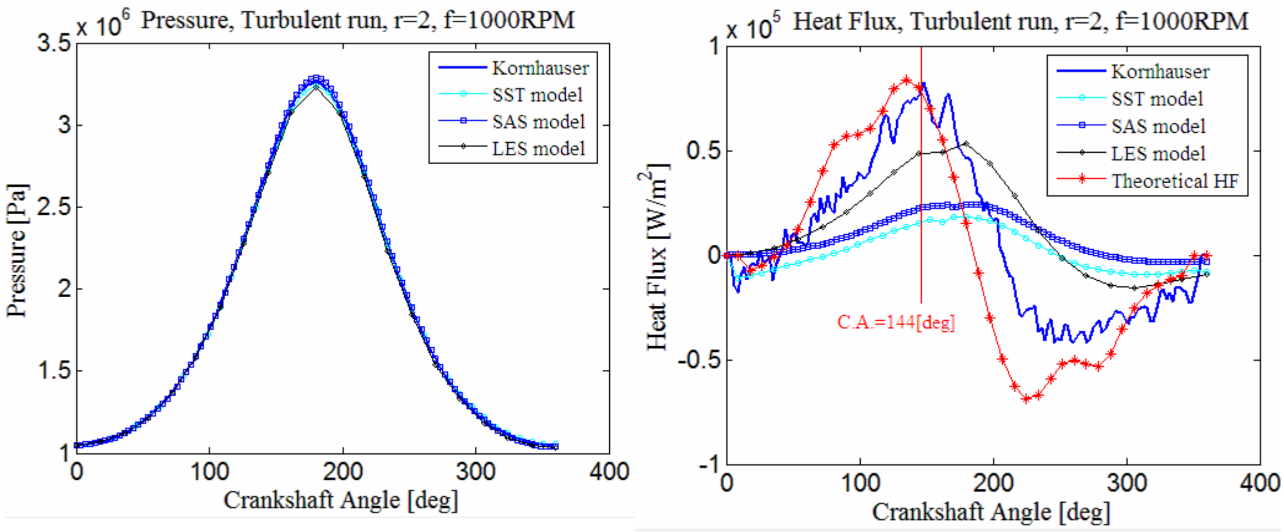

Figure 2: *Turbulent run - pressure of the compressed gas (left) and the heat transfer to the walls(right)*

Figure 2 left shows the predicted and the measured pressure as a function of crank angle. The pressure is observed to vary between 1 and 3.3 MPa, with the maximum at top dead centre, 180 degrees crank angle. Similar to the laminar case, the CFD predictions and Kornhauser measurements compare very well for the pressure data. The measured and predicted heat flux as a function of crank angle is depicted in figure 2 right. This shows an interesting difference between measurements and CFD predictions. The heat flux, as determined by Kornhauser from the measured pressure on basis of thermodynamics, is positive during compression and just after top dead centre. Past 210 degrees crank angle Kornhauser finds the heat flux to become negative. The amplitude varies from plus 80 kW/m^2 to minus 40 kW/m^2. With very little error this is reproduced, when Kornhauser's thermodynamic method is used, by the heat flux on basis of the CFD predicted pressure (curve with asterisks, using any turbulence model).

These results differ very much from the transient heat flux as predicted by CFD on basis of the enthalpy conservation equation and fluid flow. The LES model, that should be expected to be the most accurate, predicts a positive heat flux during compression with a maximum of 55 kW/m^2, a change to negative heat flux at 240 degrees crank angle, leading to a maximum of minus 20 kW/m^2. The other two turbulence models predict a sign change at similar values, but even lower amplitudes of about 10 kW/m^2.

In view of the earlier good comparison in the case of laminar flow heat flux and the good comparison of the mean turbulent pressure, this deviation of predicted heat flux is remarkable. In the next section this will be explained, arguing the pressure based thermodynamic post processed heat flux prediction to be crucial.

4 Discussion

In this section the results on heat flux in turbulent flow, as derived by the CFD method and by the thermodynamic/pressure based method, will be explained. Essential here is, that the "measured" heat flux values, as reported by Kornhauser, are actually values derived from the measured instantaneous gas domain pressure and volume, Ideal Gas Law and the First Law of Thermodynamics. The time derivatives are evaluated using a five-point least squares fit on the time dependent pressure data, and the heat flux is averaged over the enclosing heat-transfer surface.

$$T_n = \frac{p_n \cdot V_n}{\frac{p_{init} \cdot V_{init}}{T_{init}}}; \tag{3}$$

$$\left(\frac{dT}{dt}\right)_n = \frac{5 * \sum_{n-2}^{n+2} T * t - \sum_{n-2}^{n+2} T * \sum_{n-2}^{n+2} t}{5 * \sum_{n-2}^{n+2} t^2 - \left(\sum_{n-2}^{n+2} t\right)^2}; \tag{4}$$

$$\left(\frac{dV}{dt}\right)_n = \frac{5 * \sum_{n-2}^{n+2} V * t - \sum_{n-2}^{n+2} V * \sum_{n-2}^{n+2} t}{5 * \sum_{n-2}^{n+2} t^2 - \left(\sum_{n-2}^{n+2} t\right)^2}; \tag{5}$$

$$\Delta U_n = \left(\frac{dT}{dt}\right)_n * c_v * m; \tag{6}$$

$$W_n = p_n * \left(\frac{dV}{dt}\right)_n; \tag{7}$$

$$Q_n = -\Delta U_n - W_n; \quad q_n = \frac{Q_n}{A_n}. \tag{8}$$

Indexes *n* and *init* denote the instantaneous and initial values.

It is clear that for both laminar and turbulent runs, simulated pressure values match perfectly the measured ones. This is achieved regardless the turbulence model that is used. This proves the correct time- and space-discretization in the simulations and the consistence of the CFX 11.0 numerical schemes. The computed heat flux on basis of the time dependent (measured or computed) pressure data is given by equation 8. Crucial in the heat flux analysis is the comparison of the predicted data on basis of measured direct data, rather than on indirect and post processed data.

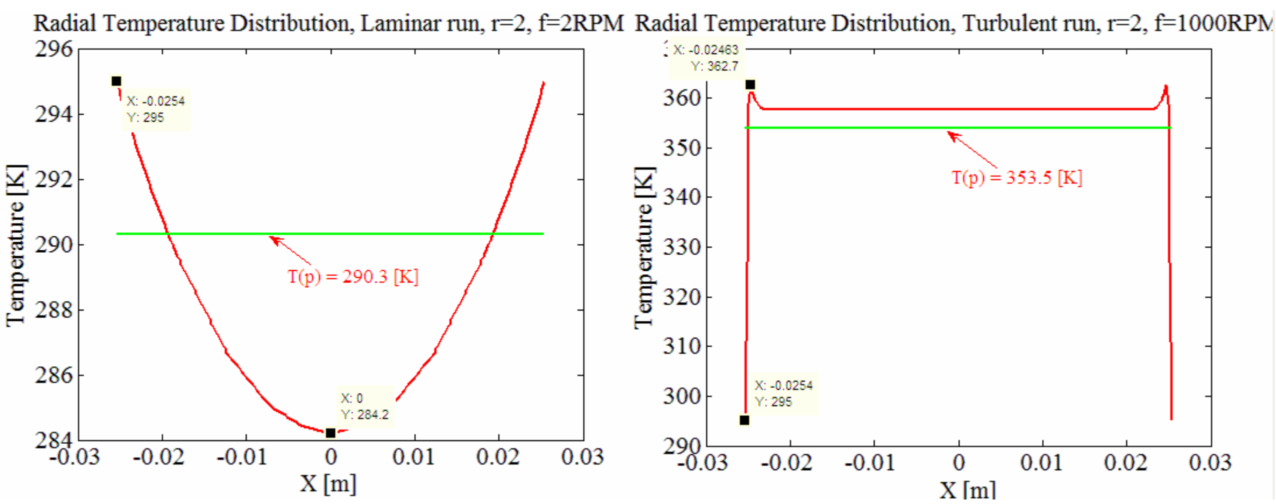

Figure 3: *CFD predicted radial profiles of temperature - laminar (left) and turbulent case (right)*

In order to analyse the difference in prediction of heat flux on basis of thermodynamic global averaged variables and in detailed CFD computation, the computed radial temperature profiles are analysed. This because the radial temperature gradient connected to convection will be driving the heat flux. The thermodynamic approach of Kornhauser, as represented in eq's 3-8, will recognize only a volume averaged thermodynamic temperature – T(p) on Figure **3**. A typical radial temperature profiles in the moments close to BDC for laminar and TDC for turbulent (as the points of the largest discrepancy of the heat fluxes, also indicated on Figure 1: *Laminar run - pressure of the compressed gas (left) and the heat transfer to the walls(right)* and Figure **2**: *Turbulent run - pressure of the compressed gas (left) and the heat transfer to the walls(right)*), is computed with CFD and presented in Figure **3**: *CFD predicted radial profiles of temperature - laminar (left) and turbulent case (right)* and compared with the thermodynamic temperature. In Figure 3 left the thermodynamic temperature is 290 K, while the CFD computed temperature shows a typical laminar quadratic radial profile of temperature. A clear connection is observed between the mean/maximum temperature and the driving temperature gradient. This explains why the CFD based heat flux prediction compares well with the thermodynamic temperature based approach.

Figure 3 right shows a typical radial temperature profile as computed by CFD, compared with the thermodynamic temperature, for the turbulent flow case. The thermodynamic temperature can be observed to be 354 K, and close to the radially independent flat temperature profile of the bulk gas. In a turbulent flow the bulk gas is well mixed and of uniform temperature, due to the turbulence. The heat flux however is determined by the temperature gradient in the very thin boundary layer. This temperature gradient (and hence the heat flux) is independent of the bulk gas temperature, but depends mainly on the structure of the turbulent flow in the boundary layer. This explains the break down of the good comparison between the CFD based heat flux prediction, and the thermodynamically based heat flux calculation. The CFD based heat flux is correct, allowing errors in turbulence and other modelling approaches. This is also illustrated by the fact that the only available direct measurement data, namely the pressure, is comparing well with CFD. In the turbulent case, the thermodynamics based heat flux calculation from pressure data is not correct.

5 Perspectives - The new experimental setup

For a piston gas spring, directly measured data are available in the literature with regard to the gas pressure as a function of time. Directly measured data for the heat flux are not available at the moment, to the knowledge of the authors. In sections 3 and 4 it has become clear that for the most common operation of piston compressors hence the high speed turbulent flow regime, directly measured data for the heat flux are essential for validation of CFD simulations. To this end, at the University of Twente a newly developed experimental setup is built and operated. It is a single cylinder, valve-less, oil-free piston gas spring. The thermal carrier fluid that is compressed and expanded is Helium gas. Several combinations of piston/cylinder materials are investigated for tribological and thermo-mechanical properties under operation. The adjacent surfaces are made with very narrow tolerances (H5/g5; clearance 10μm).

The main instrumentation on the test rig is listed below:

- One extremely fast - 8μs resolution surface temperature sensor and instantaneous heat flux - measuring element.
- Two bulk gas thermocouples (less than 1ms time response).
- Piezoelectric pressure transducer for the compressed gas.
- Infrared temperature sensor for the outer cylinder surface.
- High-resolution crank-angle transducer (3600 pulses/cycle).

6 Conclusions

By method of CFD, predictions were made for the instantaneous pressure and heat flux in a piston cylinder gas spring. It was found that unambiguous experimental data are available from Kornhauser for the pressure as a function of time. This applies for both low speed laminar flow as for high speed turbulent flow operation. These experimental pressure data compare very well with the CFD predicted data. For the heat flux between gas and cylinder walls, there are not any directly measured data available. Indirect experimental data on heat flux obtained by Kornhauser's method on basis of thermodynamics and volume averaged gas temperature, was shown to compare very well with the CFD predictions in case of low speed laminar flow operation. At high speed and turbulent flow operation, the CFD predictions do not compare well with the heat flux, as calculated with the use of thermodynamics, as based on either measured or CFD predicted pressure. This was shown to be caused by the break down of the correlation between heat flux and volume averaged temperature. For validation of the CFD model an experimental setup is constructed to produce a complete, consistent and representative set of data.

7. Acknowledgment

The authors would like to thank Drs. I.P. Jones and Ph. Stopford and ANSYS-CFX for their support and the use of the code. Also many thanks to Prof. Kornhauser presently at Virginia State University for making available his measurement data.

8 References

[1] Adair et al., 1972, "Instantaneous heat transfer to the cylinder wall in reciprocating compressors", Proceedings of the 1972 Purdue Compressor Technology Conference, Purdue University.

[2] Anand, W.J.D. and Pinfold, D., 1980, "Heat Transfer in the Cylinder of a Motored Reciprocating Engine", SAE Paper 800457, Society of Automotive Engineers.

[3] Borman, G. and Nishiwaki, K., 1987, "Internal Combustion Engine Heat Transfer", Progress in Energy nad Combustions Science, Vol 13, No 1, pp 1-46.

[4] Jak, H.G., "Heat Transfer in a Gas Spring", 2001, M.SC.Thesis, University of Twente.

[5] Kornhauser, A.A., 1989, "Gas-Wall Heat Transfer during Compression and Expansion", PhD Thesis, Massachusetts Institute of Technology.

[6] Lawton, B., 1987, "Effect of compression and expansion on instantaneous heat transfer in reciprocating internal combustion engines" MIMechE Paper Vol 201, No. A3.

[7] Lee, K.P., 1983, "A Simplistic Model of Cyclic Heat Transfer Phenomena in Closed Spaces", Proc. 18th IECEC, pp. 720-730.

[8] Lyford-Pike, E.J. and Heywood, J.B., 1984, "Thermal Boundary Layer Thicknesses in the cylinder of a Spark Ignition Engine," M.S. Thesis, Dept. of Mech. Eng., Massachusetts Institute of Technology.

[9] Morton, K.W. and Mayers, D.F., 2005, "Numerical Solution of Partial Differential Equations", 2nd Ed., Cambridge University Press.

[10] Oude Nijeweme, D.J., 2000, "Unsteady in-cylinder heat transfer in a spark ignition engine: experiments and modelling", M.SC.Thesis, University of Twente.

[11] Pfriem, H., 1943, "Periodic Heat Transfer at Small Pressure Fluctuations", NACA TM-1048