Heat transfer and mixture vaporization in intake systems of spark ignition engines

by

Wolf-Dietrich Bauer

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Submitted to the
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at the
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June 1997

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Signature of Author

Department of Mechanical Engineering
May 1997

Certified by

John B. Heywood
Professor, Department of Mechanical Engineering
Thesis Supervisor

Accepted by

Ain A. Sonin
Chairman, Departmental Committee on Graduate Studies
Mechanical Engineering

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May 26, 1997

Abstract

Heat transfer and mixture vaporization in the intake port of port-fuel injected spark-ignition engines have important impacts on engine performance and emissions. Via the initial charge temperature, intake heat transfer affects engine volumetric efficiency, engine knock, and NOx emissions. Additional interest arises through the analogy between heat and mass transfer, by which fuel vaporization potential off wall fuel films can be inferred from knowledge of heat transfer coefficients.

Heat transfer and gas temperature measurements in the intake port and manifold of a firing spark-ignition engine are presented for motoring and propane fired operation. These are complemented by modeling predictions in 1 and 3-D. Heat transfer coefficients are computed from the wall heat flux and gas temperature data condensing these to a compact format. Impact of manifold geometry, flow pulsation and jet backflow phenomena from the cylinder into the intake port are investigated for a typical four valve engine. The heat transfer results for dry operation are expected to carry over to port-fuel injected cases where thin liquid fuel films are present on the port walls.

Measurements were also pursued for liquid fuel operation. Mixture vaporization rates are inferred from the difference between liquid fuel injected and propane fired heat flux measurements. Fuel spray targeting into the intake port is clearly detected by this method of investigation. Measured fuel vaporization rates are compared to the potential for vaporization as inferred from the analogy between heat and mass transfer.

Thesis Advisor: John B. Heywood

Title: Professor of Mechanical Engineering
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This work has been sponsored by the Ford Powertrain Research Laboratory in Dearborn, MI. Very special thanks to Dr. Stephen Russ, William Stockhausen, and Dr. Charles Aquino, all at Ford, for their inputs and constant review of the work. Our discussions greatly helped to focus the research and highlighted critical points. I would like to thank Oshin Avanessian and Derlon Chu, also at Ford, for making the 3-D computations of Chapter 2 available to us. The 3-D computations of Chapter 4 were conducted by Volvo Car Corporation in Sweden. I would like to thank Dr. Jonny Nisbet and Professor Ingmar Denbratt for discussions and the permission to include their results in this work.

The experimental work would not have been possible without the assistance of many students. Stefan Grönniger and Christian Ziegler came to our laboratory as visiting students from the RWTH in Aachen, Germany. Both of their work made the engine experiments of Chapter 4 possible. I had also the pleasure to work with undergraduate students from M.I.T.: Neeta Verma, Joseph Wenisch, Charles Tam, Paul Balun, and Kevin Trexler, who all contributed significantly. Joseph, Stefan, and Paul each wrote a thesis on experimental aspects of the work. As seems inevitable when conducting research, nothing works out from the beginning and everything falls apart in the end. The students doing thesis research had to absorb a lot of these troubles and did a great job in addressing them. And of course when it comes to engines, Brian Corkum and Peter Menard are the undisputed kings of the laboratory. Nancy Cook expertly manages the lab secretarial duties, and we love her as the sunshine of our lab.

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1 Introduction

Heat transfer the intake system of spark ignition engines is of great interest due to its impact on engine performance and mixture preparation. Any rise in initial charge temperature limits the engine’s volumetric efficiency and influences the chemical reaction rates leading to NOx formation and engine knock. The thermal environment in the intake port strongly affects fuel evaporation, where fuel which does not vaporize may eventually lead to the emission of unburnt hydrocarbons.

Investigation of heat transfer and mixture vaporization is made difficult by the unsteady nature of the flow processes in the intake manifold and port of the engine. Away from the intake valve, flow is approximately a sine half wave for one quarter of the engine cycle and imposed by the piston displacement while the intake valve is open. This phase is referred to as induction or forward flow. Two backflow phases are also observed. The valve overlap backflow which occurs primarily for throttled engine operation, where the residual burnt charge from the cylinder expands into the intake manifold after intake valve opening. This process occurs in a stronger manner if exhaust valve closing is significantly later than intake valve opening causing additional flow from the exhaust into the cylinder thus keeping the cylinder pressure from more rapidly dropping to the intake pressure. The displacement backflow occurs after the induction flow and is caused by late intake valve closing during the compression stroke (which is beneficial at high engine speeds where the inertia of the induction flow causes additional charging through the so called ram effect). The nature of these backflow processes were determined in an combined experimental and computational fluid dynamics study, Chapter 2. (The 3-D computation which also incorporated mesh generation was performed by Ford). Based on gas temperature measurements and predictions it was found that backflows are primarily confined to the engine intake port region. Subsequent experiments to quantify heat transfer in the intake system were therefore split between the intake manifold runner and the intake port.

A manifold runner experiment, Chapter 3, investigated the impact of pipe geometry and the pulsating nature of the flow by an imposed wall heat flux experiment. Two manifold runners were employed—a straight and a highly curved pipe—to provide a bracket for practically encountered geometries. Steady and pulsating (engine) flow
was set up to investigate the impact of flow pulsation. Heat transfer coefficients were correlated for both geometries and flow configurations. Additional time resolved heat flux measurements at the exit of the intake manifold runner to the cylinder head investigated the physics of heat transfer in pulsating flow in a detailed manner.

Heat transfer investigation in the intake port required time and spatially resolved measurements due to the transient backflow processes with rapidly changing gas temperatures, Chapter 4. The transfer function of a commercially available thin film heat flux probe was experimentally determined to allow compensation of the signal which otherwise showed poor temporal resolution, Appendix A. An intake port of a typical engines was densely instrumented with these heat flux probes and additionally with fast thermometers for gas temperature measurements. Heat flux probes were also mounted on the back of the intake valve. This was of particular interest since the CFD study of Chapter 2 predicted attachment of the valve overlap backflow to the back of the intake with potential for high heat flux and mixture vaporization rates. Heat transfer was first quantified for motored and propane fired operation. Heat transfer coefficients were obtained from the heat flux and gas temperature measurements for all but the valve overlap backflow period. For this time, heat transfer coefficients were based on a model prediction for gas temperature (and experimental heat fluxes) since measured gas temperatures showed high spatial gradients. The 1-D model is described in Appendix B.

Eventually, heat transfer rates for port fuel injected cases were measured for thermal steady state and warm-up transient operating conditions, also Chapter 4. Fuel related heat fluxes were obtained from the difference of liquid and dry heat flux data. The difference was attributed to the sensible and latent heat required to heat and vaporize the fuel. Thereby it was assumed that heat transfer from the wall to the gas is essentially unaffected by the presence of liquid fuel. The fuel related heat fluxes clearly showed the presence or absence of fuel films at various locations in the port, which were then compared to the static fuel spray targeting into the port. Finally, measured vaporization rates were compared to the vaporization potential predicted by the analogy between heat and mass transfer based on the experimentally determined heat transfer coefficients.
2 Intake flow process characterization

A computational fluid dynamics (CFD) prediction of the transient flow in the intake system of a spark ignition engine was compared to experimental data. The calculation was performed for a single cylinder version of a pre-1995 Ford two-valve production engine, while experiments were carried out on a single cylinder Ricardo Mark 3 research engine with similar overall geometric parameters. While the two engines have somewhat different geometries, this was not considered to be a significant problem for the study of flow features. Both set-ups employed gaseous fuel. The calculation was performed at Ford by Oshin Avanessian and Derlon Chu using the commercially available Star-CD code incorporating the complete intake manifold runner and cylinder into the mesh.

Computed cylinder pressures of the 3-D code were in good agreement with experiment indicating that wave dynamics were well captured. Comparison was made to measured instantaneous gas temperatures along the intake system. Good agreement was found for the detected penetration depths of the backflow from the cylinder into the intake port as it occurs during part throttle operation.

A particularly interesting flow feature was predicted by the CFD calculation. The high speed jet which emanates into the intake port from the cylinder attached to the back of the intake valve. This produced a region of very high heat transfer, which may have a strong impact on liquid fuel evaporation during engine warm-up.

A discrepancy was observed for the forward flow phase of the intake process. CFD predicted that the forward flow quickly entrains all the burnt gas in the intake port and manifold from the preceding backflow. The experimental data on the other hand, indicated that traces of residual gas are present during the entire forward flow phase of the intake process.

2.1 Background

According to Cheng et al. [2-1] the intake process may be characterized by three distinct flow phases. A first backflow into the intake port occurs at intake valve opening, when the cylinder pressure during valve overlap is
higher than the intake manifold pressure, a phenomenon which occurs especially strongly at part-throttle conditions. This flow phase is often referred to as overlap backflow. The main or forward flow phase comes next as the piston draws mixture into the cylinder. A second backflow follows at low and intermediate crankshaft rotational speeds during the compression stroke when the intake valve closes too late to prevent mixture from being displaced back into the intake port. This flow is usually referred to as the displacement backflow. The present investigation will focus on a part-throttle operating condition where all three flow regimes are prominent.

Previous CFD investigations of intake flows have often been limited to steady flow predictions, e.g. [2-2, 2-3]. While these help in understanding the forward flow part of the intake process, they will not address phenomena associated with the backflows. Tatschl et al. [2-4] performed a calculation over significant parts of a cycle for a part-load operation at 1000 rpm/0.5 bar intake manifold vacuum. Their computational mesh included the cylinder and the intake port up to the head flange. It was found that fuel vapor, initially deposited in a region behind the intake valve, filled most of the port after the completion of overlap backflow.

Transient intake flows in particular were numerically investigated by Kuo and Chang. Together, they reported results for the forward and backward flow in a simplified geometry without valve stem [2-5]. Kuo’s subsequent study incorporated a full valve model and the cylinder [2-6]. This report predicted for a 2200 rpm/0.68 bar intake pressure condition a backflow length of roughly 10 cm. It also predicted that all backflowing residual gas is drawn back from the intake port into the cylinder during forward flow, a conclusion which will be of interest in comparison to the experimental data presented here.

In this chapter, both a detailed CFD intake flow solution and experimental results are reported in an effort to deepen our understanding primarily of the backflow processes. One may question the accuracy of both approaches: CFD calculations will be influenced by the need to reproduce turbulence effects by relatively crude models while the experiment can only provide limited spatial resolution. Hence, a comparison of results obtained by these two fundamentally different approaches to identify intake flow features seems especially worthwhile. Both parts of the work employed single cylinder two-valve-per-cylinder engines with similar geometries, see Tables 1 and 2. While it would be preferable to compare identical geometries it is argued that the current approach serves the purpose of obtaining a better qualitative understanding.

The experiment investigated the transient intake port gas temperature behavior in a set up similar to Cheng’s [2-1]. A fast-response thin-wire resistance thermometer was placed at five locations along the intake port to measure the time-resolved temperature. These are compared to the CFD predictions carried out on a mesh which included the cylinder, the intake port, and the complete intake manifold runner.

2.2 CFD Calculation

Setup: A full cycle CFD calculation was carried out, employing a single cylinder geometry for the pre-1995 production version of Ford’s 4.6 liter V8 2 valve engine with masked intake valve. Figure 2-1 shows the mesh
which encompasses the full length of the intake manifold runner. Engine geometry and operating condition are summarized in Table 2-1; the valve timings are given in Table 2-2.

The calculation was performed using the Star-CD code version 2.210. It was initiated before intake valve opening at $322^\circ$CA with an initial cylinder temperature and pressure of approximately 800 K and 1 bar, respectively. At the intake manifold entry, a constant pressure boundary condition was imposed (0.44 bar, see Table 3). The calculation was carried out for stoichiometric pre-vaporized isoctane-air mixture.

The instantaneous mass flow rate through the intake valve shows the usual back- and forward flow pattern, Fig. 2-2. There is a large dip during mid-stroke caused by a rarefaction wave traveling from the intake port into the cylinder. This wave was indentified on pressure vs. distance plots of the intake manifold as well as in the in-cylinder pressure diagram, where it is a region of negative $\frac{dp}{dt}$ centered around $450^\circ$CA. (See Fig. 2-3, which is described more fully later.) A series of pressure, fuel distribution, and velocity vector plots at different crank angle positions during the intake process are now presented and discussed, obtained from the CFD calculation.

**360^\circ$CA (Top Dead Center, intake stroke):** Figure 2-4 shows the pressure drop across the valve gap and Fig. 2-5a depicts the corresponding velocity predictions. In the cylinder it is seen that the pressure drops and the velocity increases over a distance of the order of the throat height, as it would be anticipated from dimensional arguments. The prediction within the throat at early times has to be viewed with some caution. The cylinder to intake pressure ratio is critical \[ \left( \frac{P_{\text{intake}}}{P_{\text{cylinder}}} \right)_{\text{critical}} = 0.537 \] for $\gamma = 1.35$, and sonic velocities at the valve throat are therefore anticipated. It is seen that the pressure drops continuously in the valve gap from 0.7 bar to 0.4 bar for nearly constant cross section. The physically plausible boundary layer thickness to gap height ratio was estimated to be 0.1. At this relative boundary layer thickness, the velocity and pressure would be approximately constant along the gap. However, using only one computational cell in the perpendicular direction across of the gap (This applies for small valve lifts) extends the impact of viscosity over the whole cross section and thus probably caused the observed behavior. This would explain why the velocity remains subsonic (480 m/s instead of an estimated velocity of sound of 560 m/s).

As the backflowing jet emerges from the valve gap, a small pressure recovery of 0.06 bar occurs while velocities drop from 480 m/s to about 200 m/s. The jet attaches to the back of the valve and flows along the valve stem. Two recirculating zones are visible between the jet and the port wall. Away from the valve, the jet propagates towards the upper wall. The fuel-to-stoichiometric-fuel mass ratio contour plot, Fig. 6a, is consistent with the flow pattern observed.

**385^\circ$CA:** Flow reversal occurred just prior to this figure. The 0.95 fuel-to-stoichiometric-fuel contour is 14.5 cm away from the valve, indicating the penetration depth of the backflow into the intake in Fig. 2-6b. Also visible is a fresh gas pocket near the valve stem close to the upper wall. An additional view in Fig. 2-6c shows that this feature is caused by the non-symmetrical geometry of the intake port, which is designed to set-up a swirling in-
cylinder motion during charging. This geometry causes the backflow to be displaced from the top to the left wall of the port. The plot in Fig. 2-6b on the other hand is a cut closer to the right wall where more fresh mixture is present.

415°CA: Figure 2-6d is the last frame where residual gas is observed in the port (the frames were examined in 5°CA steps). For the same time, Fig. 2-5b shows the velocities in the complete intake manifold. In the bends, velocities are elevated at the inner radius, as expected from a primarily intertia controlled flow, while the profiles straighten out again at the exits of the bends with a small defect at the inner wall. Following a path of least curvature, the left side of the valve gap shows considerably higher gas velocities.

600°CA: This frame in Fig. 2-6e depicts the end of displacement backflow, which penetrated approximately 7 to 8 cm into the port. There is little mixing with the gas resident in the port. This contrasts with the overlap backflow behavior and is attributed to the lower prevailing gas velocities.

Looking at the volume flow of both phases reveals the different extent of mixing (heat transfer effects neglected). During overlap backflow, a volume of burnt gas corresponding to 10 cm of intake pipe length is expelled, while the actual (mixed) backflow length is 15 cm. Hence, about 5 cm of fresh charge are mixed into the overlap backflow. During displacement backflow, on the other hand, the equivalent of 7 cm of cylinder content is expelled. This length equals the penetration depth shown on the fuel contour plot, indicating that hardly any mixing with the fresh charge in the intake port occurred, see Fig. 2-6e.

2.3 Experiment

Experiments were conducted on a single-cylinder Ricardo Mark 3 research engine, with 2 valves per cylinder and 0.5 liter displacement. Engine geometry, operating condition, and valve timing are listed in Tables 2-1 and 2-2. Intake port and manifold together built an S-shaped intake duct as depicted in Fig. 2-7. The manifold was connected with rubber hose to a tank in order to simulate a realistic manifold and port unit with a combined length of 0.5 m.

Figure 2-7 also depicts the locations where transient temperature measurements were taken. There, a thin wire resistance thermometer (TSI probe 1220) was inserted in a thin probe holder through the injector hole (locations 1 to 4, distance to intake valve: 5, 8.5, 12.5, 15 cm, respectively). Location 5 was accessed through a separate hole farther upstream (22 cm distance to the valve). The thermometer wire thickness is several microns and the upper frequency limit is estimated to be 150 Hz. The probe was calibrated in the 20 to 100°C range. Time averages of the measurements were compared to thermocouple readings at the same locations and found to agree usually within three degree Celsius.

A typical ensemble average over 100 cycles of the gas expected, the closer the location is to the valve, the higher are the temperatures. The different flow phases are readily detected: overlap backflow corresponds to a hot peak, followed by the forward flow of cold fresh gas, and then the displacement backflow of air-fuel-residual mixture from the cylinder. During the period when the intake valve is closed, gas temperatures in the port are nearly
constant. There, temperature fluctuations compare nicely with the theoretical value of the eigenfrequency of pressure fluctuations in a pipe with one open and one closed end.

2.4 Comparison of CFD results to Experimental Data

Along with the computationally predicted pressure Fig. 2-4 also shows the experimental data. The computational 1500 rpm/0.44 bar trace is compared to experimental 1000 rpm/0.4 bar and 1500 rpm/0.6 bar traces as no direct equivalent was available.

The drop in cylinder pressure due to the initial high-speed backflow compares well with the experimental trace of similar intake pressure indicating that the mass flow rate should also agree. The 1500 rpm traces on the other hand compare well in terms of shape and propagation speed of pressure disturbances: The intake pressure elevation due to the opening of the intake valve travels multiple times back and forth the intake runner, each time changing the sign of its amplitude upon reaching the plenum while keeping it when reflected at the cylinder end.

Figures 2-9 and 2-10 compare the gas temperature along the intake port and manifold runner (the vertical bars for the CFD prediction indicate the range of temperatures encountered over the cross-section). The plots are taken at the time of maximum overlap and displacement backflows, respectively. Good agreement is achieved for the penetration depth indicating that the code is able to predict an important feature of the compressible turbulent jet. The absolute values of the temperature predictions are somewhat low, which is consistent with the following: The initial in-cylinder residual gas temperature was set to 817 K, while the prediction one cycle later was some 400 K higher. However, at that time the calculation was stopped because of runtime limitations and no second backflow prediction reflecting the new in-cylinder temperature could be obtained.

Figure 2-11 shows experimental temperature data at 530ºCA during forward flow. A temperature rise is apparent as the valve is approached, which might be attributed to heat transfer or to the continuing presence of residual gas. Separation of these effects is possible by varying the initial fresh gas temperature to the level of the wall temperature, thus greatly reducing heat transfer, Fig. 2-12. The gas temperature elevation is still encountered, most likely to be caused by mixing with residual gas from the earlier backflow Also considered as causes for the temperature elevation were: i) local heating due to a recirculating zone near the hot intake valve stem, or ii) radiation from valve stem to the probe. The former was rejected since the phenomenon is encountered quite far away from the valve and no such zone was observed in the CFD solution, while radiation can be excluded by conservative estimates based on black body radiation.

To quantify this phenomena, which was not predicted by the CFD solution, a linear regression on the experimental data was performed. Included in the correlation were the two runs of Figs. 11 and 12 and an additional trace, all for the same crankshaft rotational speed and intake pressure and, consequently, similar mass flow rates. The model equation is given by
\[
\left[ \frac{dT_{\text{gas}}}{dx} \right]_{\text{exp}, i} = h_{\text{fit}} \times \left[ \frac{P}{mC_p} (T_{\text{wall}} - T_{\text{gas}}) \right]_{\text{exp}, i} + \left[ \frac{dT_{\text{gas}}}{dx} \right]_{\text{burnt gas}, \text{fit}},
\]

where \( P \) is the perimeter of the pipe. For each experimental trace, mean values of gas and wall temperatures, and the mean gas temperature gradient were calculated. The regression was then performed to determine the common heat transfer coefficient and the burnt gas contribution to the temperature gradient. The best fit heat transfer coefficient was 56 W/m²-K while the temperature gradient due to mixing was found to be 46 K/m. In the absence of heat transfer to the burnt gas, this would correspond to a change in burnt gas mass fraction of 1% over the length of the test section.

These numbers compare quite well to results from other researchers. Zapf's correlation [2-7] for steady flow in intake ports and the commonly used Dittus-Boelter correlation for steady fully developed flow, e.g. [2-8], bracket the results for \( h \) with values of 70 and 20 W/m²-K, respectively. Cheng’s report [2-1] included fast FID hydrocarbon concentration measurements which were used to estimate the burnt gas fraction during forward flow.

These estimates are of the same order as the above estimates.

2.5 Conclusions

The intake flow of two similar engines has been investigated experimentally and numerically. Flow features were identified and comparison of cylinder pressures and instantaneous gas temperatures along the intake system were made. It can be expected that many of the reported results are only weakly dependent on the particular geometric details of an engine. Therefore, the comparison seems valid even though the study was pursued on non-identical different engines.

Comparable magnitude and frequency of in-cylinder pressure fluctuations suggested that wave dynamics were well captured by the CFD solution and hence instantaneous mass flowrate predictions should be reliable. The gas temperature measurements related directly to the penetration depth of the backflowing gas into the intake port. The backflow during valve overlap penetrated some 15 cm into the port, a result obtained from both the CFD calculation and the experiment.

A temperature rise during forward flow was experimentally detected in the intake port as the valve is approached by the fresh gas. This occurred for all test cases measured even if wall temperatures were held below gas temperatures. It seems likely that this was caused by the continuing presence of burnt gas in the intake port during forward flow. The experimental data were correlated to a model equation which included a term for both heat transfer and mixing with burnt gas. The correlated heat transfer coefficient and the residual gas concentration in the port during forward flow compared well with experimental results of other researchers. However, neither the present CFD calculation nor previous ones predicted such an effect.
A particularly interesting flow feature was observed in the CFD prediction: During the valve overlap backflow of residual that occurs immediately after inlet valve opening, it was found that the high velocity jet emanating from the cylinder into the intake port attaches to the back of the valve. This phenomenon has a potential impact on fuel evaporation off the back of the intake valve, which in turn impacts on cold start hydrocarbon emissions. Experimental measurement of the heat transfer from the intake valve seems thus a worthwhile goal.

2.6 References


## 2.7 Tables

**Table 2-1: Engine geometries and operating conditions:**

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<tr>
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<th>CFD</th>
<th>Experiment</th>
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<td>Ford 2V</td>
<td>Ricardo Mk 3</td>
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<td>Bore</td>
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</tr>
<tr>
<td>Stroke</td>
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<td>8.6 cm</td>
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<tr>
<td>Compression Ratio</td>
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<td>1000/1500 rpm</td>
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<tr>
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<td>0.4/0.6 bar</td>
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<td>-</td>
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<td></td>
<td>Iso-Octane Air</td>
<td>Propane Air</td>
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**Table 2-2: Valve timing**

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<th>EVC</th>
<th>IVC</th>
<th>EVO</th>
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<td>604</td>
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**Table 2-3: Computational parameters**

<table>
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<th>Code</th>
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<tr>
<td>Mesh</td>
<td>120,000 computational cells</td>
</tr>
<tr>
<td>Start and End of calculation</td>
<td>322 - 1042° CA (1 cycle)</td>
</tr>
<tr>
<td>Initial Condition</td>
<td>$T_{cyl} = 817 \text{ K}, p_{cyl} = 1.02 \text{ bar}$</td>
</tr>
<tr>
<td></td>
<td>$T_{port} = 300 \text{ K}, p_{port} = 0.44 \text{ bar}$</td>
</tr>
<tr>
<td>Upstream condition</td>
<td>Boundary 0.44 bar = const (open end)</td>
</tr>
</tbody>
</table>
2.8 Figures

Figure 2-1: Computational mesh.

Figure 2-2: CFD prediction of mass flow rate through intake valve curtain, 1500 rpm / 0.44 bar inlet pressure.
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(b) 385° CA, maximum extent of backflow. Line is section indicator of slide (c). Fresh gas pocket visible as burnt gas flows closer to left wall.

(c) 385° CA, side view with section indicator of (b).

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3 Intake manifold experiment

Entry flow heat transfer experiments are reported in steady and pulsating flows of air at mean Reynolds numbers of 3,000–35,000. The ducts were chosen to match typical spark-ignition engine manifolds: a highly curved and a straight pipe, which together bracket most practical geometries. The pulsating engine intake flow encompasses an approximate sine half wave followed by a stagnant phase three times as long, all at a repetition rate of 16–50 Hz (Reₐ = 730–2200, Λ = 25–75).

Heat flow is imposed through electric heating of the outsides of the manifolds. Heat transfer in the curved pipe is closely comparable to the straight geometry, and it is concluded that entrance effects dominate. The unsteady nature of the intake flow enhances heat transfer significantly, between 50% and 100%. Time-averaged axially-resolved and axially-averaged Nusselt numbers are reported. Time-resolved wall heat flux measurements at a location close to the exit of the system show that heat transfer is delayed relative to the onset of the flow. During the stagnant flow phase, heat transfer decays slowly from its high level during the induction flow phase.

3.1 Introduction

This study concerns itself with the heat transfer of a pulsating entry flow into curved and straight pipes. Among other applications, such flows occur in the intake system of internal combustion engines and in the corresponding components of compressors. Heat transfer in the intake system is detrimental as it lowers the breathing efficiency. For internal combustion engines, an additional interest arises: increased charge temperature causes higher chemical reaction rates leading to increased nitric oxide emissions and less tolerance to engine knock. Heat transfer in the intake system is also of interest through its analogous relation to mass transfer, which determines vaporization rates from liquid fuel wall films. The temporal velocity distribution in the intake manifold of an internal combustion engines resembles an approximate sine half wave followed by a stagnant phase three times as long (in total corresponding to 720° crank angle or two revolutions of the crankshaft). The magnitude of the flow is such that a
manifold runner of 10 pipe diameters length is approximately filled with fresh fluid over one engine cycle. Various features of the investigated system may influence heat transfer rates: entrance effects, pipe curvature, the unsteady flow, and pressure fluctuations. These have been investigated individually by many researchers.

Boelter et al. [3-1] and Mills [3-2] investigated entrance flows into straight pipes. Boelter controlled the wall temperature while Mills imposed the wall heat flux. Otherwise both experiments shared many features. Various entrance configurations such as sharp edge, bellmouth, bend, and orifice plate were investigated. They found an approximate doubling of heat transfer close to entrance which then relaxed to the fully developed values for locations ten or more pipe diameters downstream.

Curvature induces secondary flow patterns with the potential to increase heat transfer rates. Pratt [3-3] and Schmidt [3-4] correlated heat transfer as a function of the Dean number $Dn = \frac{Re R}{R_{pipe}}$, where $Re$ is the Reynolds number, $R$ is half of the hydraulic diameter, and $R_{pipe}$ is the pipe radius of curvature. Extrapolating their correlations for steady, fully developed flow to a lower value of $\frac{R}{R_{pipe}}$, as is employed here, suggests that twice as much heat is transferred in a typical curved manifold pipe as compared to a straight duct.

A number of researchers have investigated pulsating flow. Two parameters were found to be important for characterization: the Stokes number $\Lambda = \frac{R}{\delta}$ and $Re_\delta$, where $\delta = \left(\frac{2v}{\omega}\right)^{\frac{1}{2}}$ is the Stokes layer thickness and $Re_\delta$ is based on the amplitude of the velocity fluctuation. For large $\Lambda$, all researchers consistently found a transition criterion of $Re_\delta \geq 500$, e.g. Akhavan et al. [3-5], above which turbulence will appear during part of the cycle. Akhavan also report that transition occurs late during the acceleration phase with turbulence production persisting throughout the subsequent deceleration phase. Wall shear stresses in the turbulent part of the main flow phase are close to quasi-steady values. Departure of the radial velocity profile from the laminar solution is especially apparent at the time of flow reversal. A laminar solution would exhibit regions of both positive and negative flow in the axial direction. These are no longer observed in the turbulent case due to stronger radial momentum exchange. Early flow acceleration prior to the onset of turbulence production then resembles a laminar start-up solution rather than a fully developed laminar oscillatory flow.

The present case differs in that flow deceleration is followed by a stagnant phase, where one expects the transient growth of a shear free turbulent boundary layer. Malan and Johnston [3-6] investigated heat transfer during the growth phase of such a boundary layer by direct numerical simulation. Their results showed that heat transfer enhancement is negligible during the initial growth phase and approximately approaches an asymptotic value for longer times. This increase in heat transfer rate ranged from 0.2 to 0.55 for initial turbulent Reynolds numbers ($Re_T$) of 60 to 380, respectively. For the present study, $Re_T$ values of that order are expected. Accordingly, heat transfer during the stagnant portion of the flow should be elevated over purely diffusive heat transfer.
Typical internal combustion engine intake systems are designed to benefit from pressure wave dynamics at higher engine speeds. Relatively small velocity fluctuations accompany these waves, hence little impact on convective heat transfer may be expected. However, they also cause temperature fluctuations even within the thermal boundary layers. These induce heat flux fluctuations which may be estimated by Pfriem's [3-7] solution for small harmonic pressure fluctuations:

$$
\dot{q}_k = \left( k \rho c_p \omega_k \right)^{\frac{1}{\gamma - 1}} T_0 \rho_k \sin \left( \omega_k t + \varphi_k + \frac{\pi}{4} \right)
$$

(1)

where $k$, $\rho$, $c_p$, and $\gamma$ are the thermal conductivity, density, specific heat at constant pressure, and the ratio of specific heats, respectively, of the gas, and $T_0$ is the mean gas temperature. For a periodic signal, this formula applies for each individual harmonic, with $\omega_k$ and $\varphi_k$ being the angular frequency and the phase angle, respectively. Eventually, $p_k$ is the corresponding amplitude of the pressure fluctuation. While these heat fluxes may be quite significant at any given time within the cycle, integration suggests that their net contribution over one cycle is zero.

Flows which simultaneously exhibit all these features have received relatively little attention. Borman and Nishiwaki [3-8] reviewed the treatment of heat transfer in the intake system of internal combustion engines. They reported that standard pipe correlations were typically employed. Enhancement factors based on the intuition of the researchers were included to account for all of the above effects. The above review indicates the potential of individual flow features to enhance heat transfer, yet their possible interactions are not known. The review of pulsating flow gave a criterion for the occurrence of turbulence during parts of the cycle. However, it is not known when during the cycle this transition takes place. Due to this ambiguity, steady state heat transfer correlations cannot be integrated to give the total heat transfer even if it behaved quasi-steadily. Similarly, it may be expected that a diffusive solution with some turbulently enhanced mixing is appropriate during the stagnant portion of the cycle. However, the initial conditions are not obvious. An experiment was therefore conducted, which, for validation purposes, first aims at reproducing known results (steady entrance flow in a straight pipe) and then gradually adds complexity to the flow phenomena (curved pipe, and pulsating flow).

The experimental technique employed here resembles that of Condie and McEligot [3-9]. They determined heat transfer coefficients in the exhaust manifold of a motored engine modified to work as a reciprocating pump. Heat fluxes were imposed by an electric heater and heat transfer coefficients were inferred from the wall-to-gas temperature difference. The current work also addresses the intra-cycle resolution of heat transfer via time-resolved measurements of heat flux and other relevant quantities.
3.2 Experimental Apparatus

The experiments include both steady and pulsating flows employing a straight and a curved pipe. The steady flow experiment utilizes a pump to provide airflow. Mounting the manifold on a single-cylinder research engine sets up the pulsating flow. Table 3-1 summarizes geometric details and the valve timing events of the engine.

Both pipes have a length of approximately 35 cm, corresponding to the dimensions of the manifolds of many production engines. The experimental setup employing the straight pipe is shown in Fig. 3-1, while the curved manifold is shown separately in Fig. 3-2. Table 3-2 lists their geometric parameters. Using either pipe, their entries are connected to a tank in order to create a boundary condition that suppresses pressure wave dynamics upstream of that position. The tank has a hole of radius 1.25 cm larger than the outer radius of the upstream pipe flange. The flange is inserted into the opening such that it forms a flush surface with the tank side wall. A 3 mm thick silicone rubber sheet, serving as a vacuum seal as well as a thermal and mechanical insulator, bridges the remaining gap. A 1 cm thick Teflon gasket insulates the downstream end of the pipe. Electrical resistance heaters 1.5 m long, wrapped at constant pitch along the axial length of the pipes, provide heat input. A 3 mm layer of glass cloth and a 2.5 cm cotton layer provide insulation around the outside.

3.3 Instrumentation

Twenty K-Type thermocouples measure the wall temperature. They are glued with thermally conductive adhesive to the outer surface of the manifold walls in five grouped locations along the manifold length. Each group consists of four thermocouples spaced evenly around the circumference, matching the pitch of the strap heater. Close to the exit, the straight pipe is fitted with an access hole for instrumentation, to accommodate—one at a time—a Vatell heat flux sensor, a Data Instruments pressure transducer, or TSI hot and cold wire probes. An adjacent surface mounted thin film heat flux sensor from RdF Inc. complements the instrumentation. This square shaped probe has a side length of 12 mm. The Vatell probe is a HFM 6 fast response thermopile (10 μsec for 63% response to step input) deposited on an aluminum nitride substrate of approximately 4 mm diameter and 25 mm length. The DC components of the heat flux measurements reported here are taken from the RdF probe, while the AC component is measured with the Vatell gage. The signals of both heat flux probes lead into a Vatell amplifier. The voltage of the Vatell probe is further amplified by an Analog Devices unit to lift it well above the discretization floor of the data acquisition system. A PC based data acquisition system logs all signals.

The cold and hot wires are TSI standard units, 1220 PI 2.5 and 1210-20, respectively. The former is used as a resistance thermometer, calibrated by a line fit to two points: ambient air and 100 °C water vapor. The signal is 1st order response corrected using a time constant of 2 msec. This value is derived from the energy balance and a standard heat transfer correlation for the wire filament. Below it will be seen that this time constant matches the measured temperature signal sufficiently close for adiabatic temperature fluctuations, which will be computed from
measured pressure data. The hot wire is operated at constant temperature using a TSI 1750 unit. This system is calibrated for velocities higher than 3.5 m/sec.

3.4 Data Evaluation Procedure and Error Propagation

Heat transfer coefficients are calculated from their definition as the ratio of heat input per unit area to the gas-to-wall temperature difference. However, perfect insulation of the heated pipe along its outside surface cannot be achieved as spatial constraints limit the amount of wrapping. Opportunities to insulate both ends are also restricted by choosing to mount the pipes to the cylinder head of the test engine in a realistic manner. Finally, significant heat redistribution due to axial conduction is encountered as a result of using aluminum pipes of realistic wall thicknesses of 3 mm or more.

The loss and redistribution mechanisms are included into the energy balance equation used to evaluate the inside surface heat transfer coefficient, \( h_{\text{pipe}} \):

\[
\dot{q}_{\text{heater}} + k_{\text{wall}} A_{\text{wall}} \frac{d^2T_{\text{wall}}}{dx^2} = h_{\text{pipe}} P (T_{\text{wall}} - T_{\text{gas}}) + h_{\text{outside}} P (T_{\text{wall}} - T_{\text{ambient}}),
\]  

(3-2)

where \( P \) is the inside pipe perimeter. The wall temperature is fitted to a fourth order polynomial, which allows computation of the second order wall temperature derivative. By concurrently fitting appropriate wall temperature gradients at the upstream and downstream boundaries, conductive losses are incorporated into the analysis. In the presence of heat transfer to the interior, it is not possible to accurately measure the wall temperature gradients at the boundaries due to strong axial variation. However, they can be measured when eliminating this heat transfer mechanism. In both cases, the wall temperature gradients have to satisfy:

\[
k_{\text{wall}} A_{\text{wall}} \left. \frac{dT_{\text{wall}}}{dx} \right|_i = \frac{\Delta T_{\text{range}, i}}{R_{\text{range}, i}},
\]  

(3-3)

where the \( R_{\text{range}, i} \) and \( \Delta T_{\text{range}, i} \) are the heat flow resistances and temperature differences across the insulations at the upstream and downstream ends of the pipe (silicone rubber shee: and teflon gasket, respectively). Eliminating heat transfer to the inside by blocking the air passage, Eq. 3-3 can be used to compute the heat flow resistances from the wall temperature gradients and the temperature differences across the resistances. For regular operation, the wall temperatures gradients are then computed from the measured temperature differences and these resistances. Elimination of the heat transfer to the inside and the losses at both ends also enable calculation of \( h_{\text{outside}} \), the heat transfer coefficient to the outside. It is computed (Eq. 3-2) from the heat input and the fitted wall temperature distribution along the disconnected test section with insulated ends and blocked inside air passage. Generally, the gas temperature in the pipe can be calculated from the energy balance for a control volume extending from the pipe entry to the \( x \)-position in question:
\[ mc \left[ T_{\text{gas}}(x) - T_{\text{gas}}(x_0) \right] = \int_0^x \left[ \dot{q}_{\text{heater}} - h_{\text{outside}} P(T_{\text{wall}} - T_{\text{ambient}}) \right] dx + \left[ k_{\text{wall}} A_{\text{wall}} \frac{dT_{\text{wall}}}{dx} \right]_{x=0}^x. \]  

(3-4)

Equation 3-2 is evaluated at five axial locations where the resolution is limited by the number of thermocouples in the axial direction. Axially resolved Nusselt numbers are obtained from:

\[ Nu_x = \frac{h_{\text{pipe}} D_{\text{hyd}}}{k}, \]

(3-5)
in which property values are evaluated at the mean film temperature (arithmetic mean of wall and bulk gas temperature). A standard linearized error analysis was undertaken employing the root-mean-sum criterion, e.g. Tse and Morse [3-10]. The dominant error source in the axially-resolved data stems from the heat redistribution due to conduction along the manifold. This is because it is difficult to determine the second derivative of the wall temperature. Its local uncertainty is set to half the maximum value encountered along the complete axial length of the manifold. This error bound is chosen based on the robustness of the second temperature derivative to changes in the method used to fit the wall temperature.

Correlations are presented for both straight and curved manifold geometries at steady and pulsed flow. Within each data group, experimentally determined local \( Nu_x \) at all five axial locations and all airflow rates are curve fitted using the local Reynolds number and an expression in \( x/D_{\text{hyd}} \). Axially averaged Nusselt numbers

\[ \overline{Nu} = \frac{1}{L} \int_0^L Nu_x dx, \]

(3-6)
are also reported, where the integral is performed using the five local values for \( Nu_x \). The error bounds for this integral are taken as the mean of the local error bounds except for the conduction term. Different treatment is appropriate as errors in heat redistribution cancel when axially integrated. They are replaced by the uncertainty in the conduction losses at the ends. Uncertainty in any other quantity, e.g. the perimeter, tends to be the same over the whole axial extension and hence adds up. The following expression is therefore applied to estimate the uncertainty in the axially averaged Nusselt number:

\[ \Delta \overline{Nu} = \left( \sum_{j=1}^5 \frac{\partial Nu}{\partial x_j} \Delta x_j \frac{L_j}{L} \right)^2 + \left( \frac{D_{\text{hyd}}}{k_{\text{gas}}} \frac{A_{\text{wall}} k_{\text{wall}}}{P(T_{\text{wall}} - T_{\text{gas}})} \frac{1}{L} \right)^2 \left( \left( \frac{\partial T_{\text{wall}}}{\partial x} \right) \right)^2_{x=0}^{x=L} \left( \frac{1}{2} \right)^2, \]

(3-7)

where \( j \) is the section index running from one to five and \( \alpha \) denotes any quantity entering Eq. 3-6 other than axial conduction.
3.5 Time-Averaged Results

Experimental Nusselt numbers for steady flow in the straight duct geometry are compared to values from the literature in Fig. 3-3. All data are shown as a multiple of the Nusselt number for fully developed pipe flow, as given by

\[ \text{Nu} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4}, \]  
(3-8)

The data compare well with these previous findings for similar entrance configurations. Hence it is concluded that the apparatus and data evaluation procedure are adequate for investigating the other configurations where no literature data are available.

The data are correlated individually for each case (straight and curved manifold, steady and pulsed flow) with non-dimensional distance from the entrance and Reynolds number:

\[ \text{Nu}_x = a \text{Re}_x^b \left[ 1 + c \exp \left( \frac{d}{D_{\text{dia}}} \right) \right], \]  
(3-9)

with property values evaluated at the local mean film temperature. Table 3-3 lists the parameters \( a \) to \( d \) for each correlation. The deviation between correlation and data is well within the error-bars on the data and is not specified individually. The uncertainty is less than 40% for high Reynolds numbers or near the entrance of the system, see Table 3-4. As some of the loss terms are approximately constant, the relative uncertainty increases to 50% - 100% (160% in worst case) for low heat transfer rates. These are encountered at downstream locations with low flow rates. Comparison between heat transfer rates in the different configurations reduces the uncertainties, since experimental procedures were kept the same. As the findings compare well to literature values where available, it is assumed that the results are more precise than suggested by the above uncertainties.

The axially averaged Nusselt numbers are correlated by:

\[ \overline{\text{Nu}} = e \text{Re}^b, \]  
(3-10)

employing axially averaged values of the Reynolds number. The coefficient \( e \) is also given in Table 3-3. Figure 3-4 plots the correlations along with Mills’ [3-2] results for a sharp 90 degree entrance. Mills’ data were for a straight manifold geometry and steady flow; the results for the corresponding current configuration agree well. Heat transfer in the curved pipe for steady flow is somewhat higher in comparison to the straight pipe, while it approximately doubles due to the engine intake flow pulsation for all Reynolds numbers with both manifolds. It is inferred that the entry and the flow pulsation generally have a stronger effect on heat transfer rates than the curvature. The uncertainty of the axially averaged data (and correlation) is typically less than 25% (with values up to 40% for the lowest Reynolds number), see Table 3-5.

Changing the engine crankshaft rotational speed and adjusting the intake pressure varies pulsation frequency at constant mean mass flow rate. Following this procedure, \( \text{Re}_e \) and \( \Lambda \) are virtually constant. Possibly due to this fact, no measurable impact of pulsation frequency on heat transfer is found. Note that firing as opposed to
motoring operation of the engine has little impact on the heat transfer in the intake manifold runner. Potentially, hot gas jets are encountered at locations close to the valve due to backflow from the cylinder into the intake port after intake valve opening. However, they do not penetrate more than 10 to 15 cm upstream of the valve, Cheng et al. [3-11]. Only modest bulk flow is induced by the backflowing jet, and heat transfer effects are expected to be local to the regions where the jet actually impinges on the walls.

3.6 Time Resolved Results

Figure 3-5 plots the phase-locked ensemble average of the velocity on the centerline for a 2750 rpm/1 bar mean intake pressure operating condition, MAP. Note that the hot-wire measurements do not give directional information. Also, values below the calibration limit of 3.5 m/sec are arbitrarily set to zero. Zero crank-angle corresponds to the bottom dead center position of the piston before the compression stroke. The induction flow is apparent from about 0°CA (crank angle) to 180°CA. The hump between 180°CA and 240°CA corresponds to a short backflow phase during the early part of the compression stroke caused by late intake valve closing. The smaller amplitude fluctuations during the rest of the cycle are induced by pressure waves in the intake.

Turbulence is produced close to the wall in a turbulent boundary layer, which then diffuses away from the wall. Turbulence measurements are shown in Fig. 3-5 for a location 2 mm away from the wall. (For quasi-steady conditions, \( y_+ \) would take values of 6, 60, and 300 at that location, for \( u \) equal to 1, 10, and 50 m/sec, respectively. Peak turbulence intensities are typically 10% of the bulk velocity very close to the wall and drop to about 5% for \( y_+ = 500 \) and larger. Hence, this position is appropriate for discerning turbulence production for all encountered velocities.) The root-mean-square velocity fluctuation increases after intake valve opening as free-stream turbulence is convected into the test section. It further increases later during the induction flow. The most likely cause of this is the onset of turbulence production near the wall, which for pulsating flow conditions occurs late in the acceleration phase of the flow, Akhavan et al. [3-5]. With the deceleration of the flow, turbulence persists and decays slowly during the remainder of the cycle approximately following Malan and Johnston's [3-6] solution, again see Fig. 3-5.

Figures 3-6 to 3-8 show the experimental pressure, temperature, and heat flux traces (an adiabatic gas temperature is also plotted in Fig. 3-7 for comparison, based on the measured mean gas temperature and the experimental pressure data). During the closed portion of the cycle from 240° to 720°CA, pressure and temperature fluctuations are apparent with a frequency coinciding to a wave in a pipe of length 40 cm with an open and a closed end, here approximately 200 Hz. As discussed in the above, these fluctuations induce significant heat flux fluctuations since temperature changes also occur within the thermal boundary layers. Figure 3-9 compares a heat flux prediction based on Pfriem's [3-7] solution for periodic pressure fluctuations to the time-varying component of the measured signal. It is apparent that many features of the measured signal are attributable to the pressure fluctuations.
The pressure induced part of the signal can be subtracted from the total measured signal, and the difference should then constitute the convective and conductive heat transfer. Figure 3-10 compares that difference to the turbulent steady flow correlation Eq. 3-8, and Malan and Johnston's [3-6] numerical results. The latter are for heat transfer in a transiently growing shear free boundary layer. Their Re \(_T = 380\) case is chosen for the comparison, to match the estimated turbulent Reynolds number based on the measured turbulent intensity and a lengthscale of one tenth of the hydraulic diameter. The heat flux increase is somewhat delayed compared to the onset of forward flow. It eventually reaches values comparable to the steady situation. During flow deceleration, the heat fluxes again depart from quasi-steady behavior in that they relax more slowly. There, they resemble Malan and Johnston's solution. The stagnant phase obviously contributes strongly to the total heat transfer, explaining why heat transfer is elevated for pulsating flow above the steady state values. Integrating the ratio of the transient heat flux over the difference of the wall to the transient gas temperature yields the time-averaged heat transfer coefficient. This mean heat transfer coefficient is 61±23 W/m\(^2\)-K, which compares quite well to the corresponding time-averaged value obtained in the experiment described previously, 89±25 W/m\(^2\)-K.

3.7 Summary and Conclusions

Experiments were conducted to determine heat transfer rates in the intake pipe (manifold runner) of internal combustion engines. It is expected that the results also apply to the corresponding components of compressors and other devices where pulsating flows with stagnant phases occur. Two ducts, one straight and one highly curved, were employed which together bracket most realistic geometries. Heat transfer results for the straight pipe and steady flow compared well to previous findings. Heat transfer in the highly curved duct was comparable to that in the straight one, suggesting that entrance effects dominate over secondary flow phenomena. Heat transfer under real (pulsating) flow conditions was found to double compared to the steady flow case, which was further investigated by time-resolved measurements. Time-averaged, axially-resolved and axially-averaged Nusselt numbers were reported.

Time resolved measurements showed that wave dynamics induced significant heat fluxes. These were closely comparable to predictions employing Pfriem's [3-7] solution for heat fluxes induced by periodic pressure variations. As their net contribution cancels over one cycle, they were subtracted from the original heat flux signal. The time resolved measurements then showed that heat transfer rates were on the order of quasi-steady values during the main flow phase. The stagnant phase (three quarter of the cycle) contributed just as much to the total heat transfer as the main induction flow. This is attributed to the slow decay of the turbulence generated during the induction flow. Heat transfer rates in the stagnant phase were similar to Malan and Johnston's [3-6] prediction for a shear free turbulent boundary layer. For applications which require temporal resolution, it is clearly not appropriate to assume quasi-steady heat transfer behavior.
Little heat transfer enhancement was found for the curved vs. the straight pipe, and it is concluded that entrance effects dominate. The experiments did not show differences in heat transfer between motored and fired engine operation. Hot backflow from the cylinder into the intake, as it occurs especially during engine operation below atmospheric intake manifold pressures, did not reach the manifold test section as it was sufficiently remote from the intake valve (10 – 45 cm).

3.8 Notation

\begin{align*}
a & \quad \text{fitting parameter} \\
A & \quad \text{area} \\
b & \quad \text{fitting parameter} \\
c & \quad \text{fitting parameter} \\
c_p & \quad \text{specific heat at constant pressure} \\
d & \quad \text{fitting parameter} \\
D & \quad \text{diameter} \\
D_n & \quad \text{Dean number, Re } R/R_{pipe} \\
e & \quad \text{fitting parameter} \\
h & \quad \text{heat transfer coefficient [W/m}^2\text{-K]} \\
k & \quad \text{thermal conductivity [W/m-K]} \\
L & \quad \text{length} \\
m & \quad \text{mass} \\
\text{MAP} & \quad \text{intake manifold absolute pressure (temporal average) [bar]} \\
\text{Nu} & \quad \text{Nusselt number} \\
p & \quad \text{pressure} \\
P & \quad \text{perimeter} \\
Pr & \quad \text{Prandtl number} \\
R & \quad \text{half the hydraulic diameter; heat flow resistance} \\
\text{Re} & \quad \text{Reynolds number} \\
t & \quad \text{time} \\
T & \quad \text{temperature} \\
u & \quad \text{velocity} \\
x & \quad \text{axial position} \\
y & \quad \text{distance from wall} \\
\alpha & \quad \text{variable}
\end{align*}
\( \delta \)  
Stokes layer thickness, \((2\nu/\omega)^{\frac{1}{2}}\)

\( \Delta \)  
increment

\( \gamma \)  
ratio of specific heats

\( \varphi \)  
phase angle

\( \Lambda \)  
Stokes number, \( R/\delta \)

\( \nu \)  
kinematic viscosity [m²/sec]

\( \rho \)  
density

\( \omega \)  
angular frequency

Subscripts

\( \text{hyd} \)  
hydraulic

\( i \)  
index

\( j \)  
index

\( k \)  
wave number

\( t \)  
turbulent

\( x \)  
axially resolved

\( \delta \)  
Stokes layer

\( + \)  
inner variable, turbulence

Superscripts

\( , \)  
per unit length

\( ^\prime \)  
per unit area

Overscores

average

per unit time
3.9 References


### 3.10 Tables

**Table 3-1: Engine Data**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>Stroke (mm)</td>
<td>90.0</td>
</tr>
<tr>
<td>Bore (mm)</td>
<td>83.0</td>
</tr>
<tr>
<td>Connecting Rod Length (mm)</td>
<td>158.0</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>10.14</td>
</tr>
<tr>
<td>IVO- 0.1 mm lift</td>
<td>-4°</td>
</tr>
<tr>
<td>IVC- 0.1 mm lift</td>
<td>236°</td>
</tr>
<tr>
<td>EVO- 0.1 mm lift</td>
<td>492°</td>
</tr>
<tr>
<td>EVC- 0.1 mm lift</td>
<td>12°</td>
</tr>
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</table>

**Table 3-2: Geometric Parameters of Manifolds**

<table>
<thead>
<tr>
<th>Manifold</th>
<th>Straight</th>
<th>Curved</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hyd. Diameter</td>
<td>38 cm</td>
<td>~40 cm</td>
</tr>
<tr>
<td>Length</td>
<td>35.3 m</td>
<td>~34.5 cm</td>
</tr>
<tr>
<td>Wall thickness</td>
<td>0.7 cm</td>
<td>~0.3 cm</td>
</tr>
<tr>
<td>Radius of Curvature</td>
<td>-</td>
<td>5 - 25 cm</td>
</tr>
</tbody>
</table>

**Table 3-3: Coefficients of Nusselt number fit**

<table>
<thead>
<tr>
<th>Manifold/Flow</th>
<th>a</th>
<th>b</th>
<th>c</th>
<th>d</th>
<th>e</th>
</tr>
</thead>
<tbody>
<tr>
<td>straight/steady</td>
<td>0.043</td>
<td>0.73</td>
<td>2.9</td>
<td>-0.68</td>
<td>0.062</td>
</tr>
<tr>
<td>straight/pulsed</td>
<td>0.051</td>
<td>0.77</td>
<td>1.95</td>
<td>-0.49</td>
<td>0.072</td>
</tr>
<tr>
<td>curved/steady</td>
<td>0.13</td>
<td>0.66</td>
<td>0.69</td>
<td>-1.05</td>
<td>0.14</td>
</tr>
<tr>
<td>curved/pulsed</td>
<td>0.077</td>
<td>0.74</td>
<td>1.0</td>
<td>-0.42</td>
<td>0.10</td>
</tr>
</tbody>
</table>
Table 3-4: Uncertainty associated with axially resolved Nusselt number correlation of Table 3-3. The larger uncertainties correspond to the lower Reynolds number limits.

<table>
<thead>
<tr>
<th>Manifold/Flow</th>
<th>Re: 3,000 - 9,000</th>
<th>Re: 17,500 - 35,000</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Entrance</td>
<td>Exit</td>
</tr>
<tr>
<td>straight/steady</td>
<td>&lt; 40 %</td>
<td>160 % - 60 %</td>
</tr>
<tr>
<td>straight/pulsed</td>
<td>&lt; 40 %</td>
<td>75 % - 50 %</td>
</tr>
<tr>
<td>curved/steady</td>
<td>60 % - 40 %</td>
<td>90 % - 50 %</td>
</tr>
<tr>
<td>curved/pulsed</td>
<td>60 % - 40 %</td>
<td>90 % - 50 %</td>
</tr>
</tbody>
</table>

Table 3-5: Uncertainty associated with axially averaged Nusselt number correlation of Table 3-3. The larger uncertainties correspond to the lower Reynolds number limits.

<table>
<thead>
<tr>
<th>Manifold/Flow</th>
<th>Re: 3,000</th>
<th>Re: 8000 - 35,000</th>
</tr>
</thead>
<tbody>
<tr>
<td>straight/steady</td>
<td>50 %</td>
<td>&lt; 25 %</td>
</tr>
<tr>
<td>straight/pulsed</td>
<td>&lt; 25 %</td>
<td>&lt; 25 %</td>
</tr>
<tr>
<td>curved/steady</td>
<td>40 %</td>
<td>&lt; 25 %</td>
</tr>
<tr>
<td>curved /pulsed</td>
<td>30 %</td>
<td>&lt; 25 %</td>
</tr>
</tbody>
</table>
3.11 Figures

Figure 3-1: Straight manifold mounted to test engine
Figure 3-2: Curved manifold pipe, cross-section
Figure 3-3: Local Nusselt number divided by fully developed value: present data for straight manifold and steady flow in comparison to literature data.

Figure 3-4: Plot of measurements and correlations for axially averaged Nusselt number: steady and pulsating flow in curved and straight duct, comparison to literature value from Mills [3-2]
Figure 3-5: Velocity measurement on centerline of straight pipe, turbulence intensity measurement 2 mm from the wall, 2750 rpm/1 bar MAP

Figure 3-6: Measured intake manifold pressure, 2750 rpm/1 bar MAP
Figure 3-7: Measured centerline gas temperature and calculated adiabatic gas temperature based on the measured mean temperature and the pressure fluctuation

Figure 3-8: Heat flux measurements 5 cm upstream of the exit of the straight manifold (positive sign into the wall), 2750 rpm/1 bar MAP
Figure 3-9: Comparison of heat flux prediction using Priem's [3-7] solution for periodic pressure fluctuations to the time-varying component of the measured signal, 2750 rpm/1 bar MAP

Figure 3-10: Measured heat flux after subtraction of pressure wave induced component in comparison to steady state correlation (−0° - 180° CA), and Malan and Johnston's [3-6] solution (−180° - 720° CA), 2750 rpm/1 bar MAP
4 Intake port experiment

Time-resolved heat flux and gas temperature measurements in the intake port of a spark ignition engine are presented. Experiments were pursued for motored, propane fired, and liquid fuel operation. Heat transfer coefficients were built from the dry data. Also, heat transfer rates in the port and off the back of the intake valve were integrated over the main flow phases.

While the heat flux instrumentation was primarily intended for dry operation of the engine, liquid fuel experiments were also pursued. Liquid fuel vaporization was assessed for isooctane and indolene fuels at thermal steady state and for engine warm-up transients. Open valve injection showed significantly less wall impingement, which is consistent with previous findings. For low wall temperatures, vaporization rates inferred from the heat flux measurements compared well to the mixture vaporization potential predicted by analogy to heat transfer. For high wall temperatures, the analogy between heat and mass transfer indicated that all injected fuel could vaporize within one cycle if the fuel film evenly covered the spray targeting area. However, this was neither consistent with the data nor with known engine behavior for injection rate transients, in which a time of several cycles is required for mixture preparation to adjust to changing fueling rates. It is thought that the fuel film does not spread out in a homogenous manner. Rather, discrete fuel droplets or rivulets built up until sufficient surface area allows vaporization at a rate equal to the fuel deposition.

4.1 Introduction

Heat transfer and mixture vaporization in the intake port have an important impact on engine performance and emissions via the initial charge temperature. Thus intake heat transfer affects engine volumetric efficiency, engine knock, and NO\textsubscript{x} emissions. Additional interest arises through the analogy between heat and mass transfer, by which fuel vaporization rates off wall fuel films can be inferred from knowledge of heat transfer coefficients.
Heat transfer in the intake port of spark ignition engines has been reviewed by Borman and Nishiwaki [4-1]: For steady flow, Zapf [4-1] reported a heat transfer correlation for the port wall and intake valve together. Most other researchers employed standard pipe correlations. Enhancement factors were chosen based on the intuition of the researchers to account for flow unsteadiness and other effects such as duct curvature, entry flow behavior, etc.

A large body of work exists investigating mixture preparation by experimental or modeling approaches. Shayler et al. [4-4 to 4-6] presented time-resolved measurements of intake port wall heat fluxes. Primarily liquid fuel injected experiments were conducted for wall temperatures ranging from 0 to 90°C. Shayler et al. [4-7] also reported engine hydrocarbon emissions data for different injector types and related them to port heat flux measurements. A different approach to mixture preparation modeling comes from an engine controls focus: Shayler et al. [4-8] and Almkvist and Erikson [4-9] fitted wall wetting and fuel vaporization models (commonly known as \( \tau \)-X models) to engine data on fuel-to-air mixture ratio excursions induced by fast changes in air or fuel flow rates. While this model was primarily developed for transient fueling compensation, it also has a physical interpretation. The X parameter of the model stands for the fraction of fuel which deposits onto the intake port wall, while \( \tau \) represents the time scale for vaporization off the walls. Typical values for X range from 0.6 to 0.8 and \( \tau \) values from 0.3 to 2.5 s [4-8], where the higher values correspond to lower wall temperatures.

Brown and Ladommatos [4-10], among other researchers, presented a mixture vaporization model where the spray was modeled in a Lagrangian manner. The spray impinging onto the wall formed the fuel film. This film was then vaporized employing a heat and mass transfer analogy together with a standard pipe flow heat transfer correlation (for wall temperatures above the fuel’s saturation point, a nucleate boiling mechanism was included). Consistent with Shayler’s [4-8] X value, Brown and Ladommatos found that a small portion of the fuel directly vaporizes in the air (approximately 5%). They also found that film flow into the cylinder is quite high for moderate wall temperatures (at 60°C wall temperature, 60% of the fuel film flowed into the cylinder with heptane fuel with \( T_{sw} \sim 95°C \)). Martins and Finlay [4-11] reported heat transfer measurements for a heated intake valve. Their rig comprised a engine cylinder head, and the intake valve was moved in a realistic manner. Experiments were conducted for alcohol fuels and gasoline. The fuel spray was directly targeted onto the intake valve and complete vaporization was achieved for valve temperature above approximately 100°C.

This chapter aims to provide more detailed quantitative knowledge of heat transfer rates in the intake system of an operating spark ignition engine. In chapter 2, flow processes in the intake port were investigated by CFD and experiments which focused on intake gas temperatures. It was found that backflow during valve overlap from the cylinder into the intake port under typical part-load conditions reaches approximately 10 cm into the intake port, which coincides with the location of the intake manifold head flange. Two sets of experiments were therefore pursued: the intake manifold experiment of chapter 3 which investigated entrance, flow pulsation, and pipe curvature effects, and the current intake port experiment which also allows investigation of the backflow processes which have inherently shorter time scales.
The steady and pulsating engine flow heat transfer experiments pursued in chapter 3 showed that heat transfer rates typically doubled for the pulsed induction flow of the engine. Little difference was found between a straight and a curved manifold geometry, and entrance effects dominate those caused by pipe curvature. The time-averaged results were obtained from an imposed wall heat flux experiment. Additionally, time-resolved heat flux, gas temperature, pressure, and velocity measurements were taken at the exit of the manifold adjacent to the cylinder head. It was found that wall heat fluxes during the intake valve closed period of the cycle contributed significantly to the cycle average. This was attributed to the effect of decaying turbulence generated during the previous induction flow. It was also found that heat flux oscillation induced by acoustic wave action were well predicted by Pfiem's [4-3] solution.

The work described in the current chapter quantified intake port heat transfer by time-resolved wall heat flux and gas temperature measurements at representative locations in the intake port. The CFD prediction of chapter 2 suggested that valve overlap backflow into the intake port attached to the back of the intake valve, potentially causing high heat transfer and mixture vaporization rates. We investigated this phenomenon by also taking heat flux measurements at the back of the intake valve.

The heat flux and gas temperature measurements were correlated through heat transfer coefficients. Consistent with CFD predictions, gas temperature measurements close to the valve during valve overlap backflow exhibited strong gradients in axial and radial direction. Heat transfer coefficients during this flow phase were therefore based on a 1-D model solution for cross-sectionally averaged gas temperature (and the experimental heat fluxes).

Heat flux measurements were taken for liquid fuel operation. Vaporization rates for port fuel injected operation were estimated from the difference of heat flux signal for liquid fuel injected and propane fired cases. This heat flux was then compared to the predicted convective heat transfer employing the measured wall temperature and the experimental heat transfer coefficient for dry operation. The fraction of the injected fuel which vaporized in the port was determined for steady state operation at coolant temperatures from well below to close to the fuel's saturation temperature. These experiments were complemented by start-up experiments, in which significantly lower valve temperatures were encountered.

4.2 Engine

The experiments were conducted on a Ricardo Hydra single-cylinder research engine fitted with a cut version of a Volvo four-valve cylinder head with pent-roofed combustion chamber. Table 4-1 summarizes the engine's characteristics. The air intake system of the engine consisted of a custom made straight intake manifold runner of 0.35 m length, a buffer volume which simulated the plenum of a normal multi-cylinder engine, Fig. 4-1, and a throttle valve placed upstream of the buffer. The engine was equipped with a Bosch single-pintle injector, type EV 11 A [4-12]. A laboratory controller was employed to set the injection timing.
4.3 Instrumentation

The engine intake port was fitted with commercially available heat flux probes: thin film thermopiles from RdF Inc. containing 20 thermocouple pairs. They were square probes with 1 cm sides. The transfer function of the heat flux probe had been determined previously (see Appendix A) and was employed here to enhance the frequency response of the sensors. Since the heat flux probes were used at high-temperature conditions, a number of different adhesives were tested. Permabond 922 from Permabond International was found suitable and subsequently used to bond the probes on the port walls and the intake valve. Additional instrumentation included cold wire resistance thermometers, which were TSI standard units 1220 PI 2.5. These were employed to measure time-resolved gas temperatures and were calibrated by a line fit to two points: ambient air and 100°C water vapor. The temperature signals were processed employing a first order response correction with a time constant of 2 msec. This value was derived from the energy balance and a standard heat transfer correlation for the wire filament. This time constant matched the measured temperature signal sufficiently closely to adiabatic temperature fluctuations calculated from the measured time-resolved intake pressure.

Seven locations in the intake port were chosen for instrumentation with heat flux probes, Fig. 4-2. The probes were evenly distributed along the top and bottom walls of the port, and on the intake valve. Probe numbering reflected the position of the probe on the top or bottom wall and the upstream position relative to the intake valve seat. For example, probe \( q_{\text{top,13mm}} \) was on the top wall right next to the valve, and probe \( q_{\text{top,59mm}} \) was 59 mm upstream, between the valve guide and the intake manifold head flange. Three heat flux probes \( q_{\text{bot,xxmm}} \) were placed on the bottom wall, 13, 42 and 63 mm upstream of the valve seat. A thermocouple was mounted to the wall alongside each heat flux probe. Additionally, two pairs of the thermocouples and heat flux probes were mounted on the back of the intake valve, with an additional thermocouple in the center of the valve. These probes were labeled \( q_{\text{valve,bot}} \) and \( q_{\text{valve,top}} \) depending on their location being closer to the top or the bottom wall of the port. Gas temperature measurement were taken through two access holes into the port, which are also shown in Fig. 4-2. The first probe location \( T_{\text{gas,perp,15mm}} \) is 15 mm upstream of the intake valve and accesses the port in the direction perpendicular to the valve stem and port axis. \( T_{\text{gas,axial}} \) provided measurements in the axial direction approximately along the port center line from 13 mm to 150 mm upstream of the valve seat.

4.4 Correlation of wall temperature

This section reports wall temperature data for motored operation and propane, isooctane, and indolene fired conditions. These data are correlated individually for each fueling mode as a function of position in the port, coolant temperature, engine speed, and intake manifold pressure. The range of these variables examined is given in Table 4-2, where \( x \) is the distance from the valve seat upstream into the port and \( \alpha \) denotes vertical elevation (0° bottom
wall, 180° top wall). Valve temperatures are correlated separately because they exhibit different dependencies on most of the independent variables of the fit.

The wall temperature fit is of the form:

\[
T_{\text{wall}}(°C) = a + b \times \text{rpm} + c \times p_{\text{intake}}(\text{bar}) + d \times T_{\text{cool}}(°C) + e \times x(\text{mm}) + f \times \alpha(\text{deg}) + g \times x\alpha + h \times x^2
\]  \hspace{1cm} (4-1)

The parameter and the quality of the fit are given in Table 3, where \( \sigma \) is the standard deviation in degrees Celsius, and \( e_{\text{max}} \) is the maximum residual (deviation of the data from the fit), also in degrees Celsius. The valve temperature was fitted to:

\[
T_{\text{valve}}(°C) = a + b \times \text{rpm} + c \times p_{\text{intake}}(\text{bar}) + d \times T_{\text{cool}}(°C)
\]  \hspace{1cm} (4-2)

and it is described by Table 4-4.

4.5 Gas temperature, propane fired

In this section gas temperature measurements are discussed for a particular operating condition: 1500 rpm/0.5 bar intake manifold absolute pressure (MAP), propane fired operating condition at high coolant temperature and ambient fresh gas temperature. Next, they are compared to temperatures obtained from a CFD solution in order to complement the conceptual picture of flow processes in port. Finally, a one-dimensional prediction is presented, which solves the mass and energy balance in the intake port based on the experimental heat fluxes and a cycle simulation code prediction for mass and enthalpy flow rate in the intake valve gap. The gas temperature prediction from this code was employed to build heat transfer coefficients during the valve overlap backflow phase. This became necessary since experimental gas temperature measurements did not show enough spatial resolution during the overlap backflow phase to base heat transfer coefficients reliably on them.

The different flow phases during the engine intake processes can be seen in the gas temperature measurement along the axial extent of the port, Fig. 4-3. The main flow inducted fresh mixture at ambient temperature which quickly convect burnt gas from previous backflow events forward into the combustion chamber (the magnitude of the induction flow was such that the manifold runner and intake port were approximately filled with fresh mixture over one engine cycle). Some elevation of the gas temperature along the port was seen during that flow phase, which is attributed to heat transfer. Following BC of the intake stroke at 720°CA, in-cylinder mixture and residual gas were displaced backwards into the intake port due to intake valve closing during the compression stroke. This flow had mostly plug flow characteristics and a volume corresponding to approximately 5 cm of port length was displaced into the port, chapter 2. Because the gas from the cylinder is at elevated temperature due to its residual burnt gas fraction, higher gas temperatures were encountered in the intake port after this displacement backflow phase. The temperature further upstream in the port was also elevated due to backward convection of the temperature gradient along the port, which originated from heat transfer in the previous forward
flow phase. Gas temperatures at all measured locations were approximately constant during the stagnant phase following intake valve closing, possibly caused by the measurement locations being sufficiently removed from the wall so that wall heat transfer effects did not reach them. Valve overlap backflow started with intake valve opening at 0°CA (0.1 mm valve lift). At the end of the exhaust stroke, burnt charge in the cylinder was approximately at atmospheric pressure. This gave rise to backflow into the intake port as the cylinder charge expands into the intake, particularly for throttled operation. Closing the exhaust valve later than intake valve opening also generated flow from the exhaust into the cylinder and prevented cylinder pressure from dropping in a more rapid manner. Similarly to the displacement backflow process, higher gas temperatures were encountered further upstream in the port due to backwards convection of the existing axial gas temperature gradient.

Figure 4-4 shows measurements along a line perpendicular to the centerline of the port. The vertical elevation of this measurement location was approximately halfway between the top and the bottom wall of the port, and the axial distance to the valve seat was approximately 15 mm. This axial position coincided approximately with that of the axially variable probe moved closest towards the intake valve. During the flow phases which showed approximately plug flow characteristics (induction, displacement backflow, stagnant phase), similar temperatures were measured by the radial probe in any position and the axial probe in its foremost position. However, for valve overlap backflow, higher temperatures were measured by the former probe, which resulted from its higher elevation over the intake port floor.

CFD allowed a closer look at the valve overlap backflow. The CFD calculation became available through Volvo and was performed for a similar set of initial and boundary conditions as the above measurements: 1800 rpm/0.44 MAP. Fuel preparation simulated by initializing the calculation with a rich fuel air mixture cloud next to the intake valve, initial port gas temperature was taken from a previous one-dimensional gas dynamics calculation, the port wall temperature reflected warmed-up conditions (valve temperature was at 100°C, which was approximately 100°C lower than the experimental value). Figures 4-5 and 4-6 show the CFD prediction of the spatial gas temperature distribution at 15°CA after TC intake stroke. The section plots include the location of the experimental measurements along lines in the axial and radial directions. The experimental findings that gas temperatures increased from bottom towards the top of the port and from the port wall towards the valve stem are repeated. The backflowing jet apparently attached to the valve stem and then propagated towards the top wall (this phenomenon had already been observed in a previous CFD prediction of chapter 2). The gas temperature measurements are compared to the CFD prediction in Figs. 4-7 and 4-8. While CFD initial conditions and wall temperature boundary conditions varied somewhat from the experimental ones, it is apparent that the temperature distributions in the axial and radial directions agree qualitatively. Heat flux measurements complemented experimental gas temperature measurements. The probes on the valve back consistently showed the highest heat fluxes during valve overlap backflow (peak of 20 W/cm² from the gas into the wall for 1500 rpm/0.5 MAP, propane fired), confirming attachment to valve back. On the port wall next to the valve, peak heat fluxes were lower, on the
order of 5 W/cm². These values were typically of those encountered on the top wall. However, for some operating conditions, the sensor on the bottom wall records similar values. The presence of recirculating zones between the backflowing jet and the port wall is hypothesized, though it may not always form in the same manner.

A one-dimensionally resolved prediction for gas temperature in the intake port was obtained by solving the mass and energy balances in the intake port. (For a description of the model see Appendix B.) The in-cylinder temperature and the mass flow rate through the intake valve gap were input into the code and were generated from a previous run of an engine cycle simulation code. The resulting 1-D model prediction for locations corresponding to the heat flux probes is shown in Fig. 4-9, which, for comparison, also shows measured gas temperatures. It is clear that measured temperatures agree quite well with predicted ones. For all but the valve overlap backflow, measurements at the axial location closest to the valve seat (13 mm) and in perpendicular direction (15 mm to the valve seat, 5 mm to the valve stem) are closely comparable. During valve overlap backflow, their different vertical elevation over the bottom port wall becomes significant (approximately 5 mm and 15 mm for $T_{axial,13mm}$ and $T_{perp,15mm}$ respectively). CFD predicted that valve overlap flow preferentially moves to the top wall and close to the valve, which is consistent with these measurements. The 1-D code prediction for gas temperature during valve overlap backflow falls somewhat below the measured peak temperatures (from $T_{perp,15mm}$), which is expected since the code predicts the cross sectionally averaged temperature. Overall, comparison of 1-D prediction and experiment encourages use of the model. The model complements the measurements by also providing gas temperature predictions next to the back of the valve, which will be employed for that location to compute heat transfer coefficients during valve overlap backflow.

### 4.6 Heat transfer, motored and propane fired

Figure 4-10 shows time resolved wall heat flux measurements at six different locations for the baseline of 1500 rpm/0.5 bar MAP propane fired operating condition. Our sign convention defined heat transfer from the gas to the wall as positive, which has the advantage that gas temperature and heat flux measurements have approximately the same shape. During the induction flow phase, heat fluxes varied due to different wall temperatures in the port, which increased closer to the combustion chamber. Potentially, they also varied due to differences in convective heat transfer rates, which is later investigated by computing heat transfer coefficients. Displacement backflow heat transfer was generally low, since the gas displaced back into the intake port was at temperatures close to that of the walls. Valve overlap backflow appeared as a significant spike at 540°CA (TC intake stroke). The backflow was most visible at the valve back, and the port wall locations next to the valve, then it decreased rapidly. The impact of intake pressure is shown in Fig. 4-11. Backflow was clearly strongest for the lowest intake pressure, reaching furthest upstream for 0.3 bar. Generally, heat fluxes were stronger at the top wall, which is consistent with the CFD solution. However, the difference between top and bottom wall was less than what would be expected from the gas temperature prediction of the CFD solution. The impacts of wall and gas temperature variations are shown in the

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example of a motored operation condition, 1500 rpm/0.5 bar MAP, Fig 4-12. One of the probes on the valve back \( q_{\text{valve,bot}} \) and the probe farthest upstream in the port \( q_{\text{bot,63mm}} \) were chosen for this comparison. The magnitude and sign of the measured heat transfer rates changes clearly correlates with the difference between the coolant and the fresh gas temperatures. For probes close to the combustion chamber, and especially for the valve, the elevation of wall temperature over the coolant temperature became significant causing a negative offset in the heat flux data. However, computing heat transfer coefficients based on wall temperature data will collapse theses data to one curve. For motored operation, valve overlap backflow appeared as a negative spike, as the charge was overexpanded during the previous expansion process. This caused in-cylinder gas temperatures which were below the fresh gas temperature and, ultimately, the observed phenomenon.

The heat transfer signals were spatially and temporally averaged either over the main flow phases or the whole cycle. Spatial averaging was facilitated by the fact that the heat flux signals did not vary greatly over the port surface. Thus, it was not critical how representative areas for each probe are chosen as long as all areas sum to the total port area. The averaging procedure was carried out for 1500 rpm/0.5 bar and 1 bar MAP operating conditions. The temperatures were chosen to be most applicable for regular engine operation: ambient fresh gas temperatures and high wall temperatures. The time and area integrated heat fluxes were weighted by the mean enthalpy capacity flux through the intake system, \( \overline{\dot{m}_h c_p} \), giving a mean temperature increase of the charge at the exit of the intake system due to heat transfer from the surface(s) in question. Areas employed for averaging the heat flux traces are 12, 12, 21, 17, 17, 23, and 32 cm² for \( q_{\text{valve,bot}} \), \( q_{\text{valve,top}} \), \( q_{\text{bot,13mm}} \), \( q_{\text{bot,42mm}} \), \( q_{\text{bot,63mm}} \), \( q_{\text{top,13mm}} \), and \( q_{\text{top,59mm}} \), respectively. Following this procedure, the mean gas temperature increase at the exit of the intake system due to heating in the intake port was approximately 9°C for both 1500/.5 and 1 bar mean intake pressure. The increase during forward flow alone was 8°C and 6°C for 0.5 and 1 bar intake pressure, respectively. Displacement backflow and the subsequent stagnant phase caused an increase of approximately 3°C for both manifold pressures. The valve overlap period only showed significant backflow for throttled operation. This is clearly seen in the heat flux data, in which negligible heat transfer during the valve overlap period occurred for 1 bar intake manifold pressure (see Fig. 4-11). On the other hand, the equivalent of a 2°C average charge temperature decrease (mean temperature of charge leaving intake system into the cylinder) occurred due to valve overlap backflow heat transfer for throttled operation. For both intake pressures, the majority of the heat transfer was from the back of the intake valve: mean charge heating effect over a cycle was 5°C and 6°C, respectively, for 1500 rpm/0.5 bar and 1 bar MAP.

4.7 Heat transfer coefficients, motored and propane fired

Heat transfer coefficients were determined as a means of connecting all the heat flux data for motored or propane fired operation. They are useful for model implementation as well as for applying the analogy between heat and mass transfer for predictions of convectively enhanced fuel film vaporization. Heat transfer coefficients based on
the experimental gas temperatures. These yield adequate results for all but the valve overlap backflow phase. At that time in the cycle, gas temperature gradients will be seen to be too steep to compute meaningful heat transfer coefficients based on a locally measured gas temperature. We therefore replaced the measured gas temperature with a gas temperature prediction from a one-dimensional code.

Heat transfer coefficients were calculated according to their definition:

\[ h_i = \frac{\dot{q}_i''}{(T_{\text{gas}}(x_i) - T_{\text{wall},i})}, \]  

(4-3)

where \( \dot{q}_i'' \) is the experimental heat flux at probe \( i \), and \( T_{\text{wall},i} \) is the experimentally measured wall temperature, and \( T_{\text{gas}}(x_i) \) is either the experimental gas temperature at the axial position of \( i \), or the prediction by the 1d code. Error propagation shows that the higher amplitude heat fluxes caused by large gas to wall temperature gradients results in the most accurate determination of heat transfer coefficients:

\[ \frac{\Delta h}{h} = \frac{\Delta \dot{q}''}{\dot{q}''} - \frac{\Delta (T_{\text{gas}} - T_{\text{wall},i})}{(T_{\text{gas}} - T_{\text{wall}})^2}. \]  

(4-4)

Temperature difference is maximized by heating the coolant to 75°C while using unheated fresh charge air. Heat transfer coefficients for all flow phases except the valve overlap backflow coincide for motored or fired operation within the error bars. However, uncertainties during displacement backflow are lower for motored operation because the gas-to-wall temperature gradient was higher (due to lower temperature of the cylinder gas displaced back into the intake port). Therefore heat transfer coefficients are computed from the motored data. Figure 4-13 shows Nusselt numbers (based on the mean hydraulic diameter of the port, 4.2 cm) for all motored experimental runs and all probe locations. Gas temperature for all crank angles is the experimentally determined value. Speed and load ranged from 1000 rpm/0.5 bar MAP to 4500 rpm/1.0 bar MAP corresponding to Reynolds nos. ranging from 3,800 to 45,000. The plotted Nusselt nos. are divided by \( \text{Nu}_{\text{scale}} = \text{Re}^{0.5} \), which trial and error found to collapse best all data to approximately the same shape and magnitude. For valve overlap backflow, data in Fig. 4-13 (which are based on the experimentally measured gas temperatures) do not collapse to one curve. This is primarily due to the fact that the employed experimental local gas temperature from the axial measurement location closest to the valve is no longer representative for the entire region close to the valve. Figure 4-14 compares the time-averaged Nusselts numbers of the port excluding the valve overlap phase to the fit to the manifold data Chapter 3. (The fit is evaluated for the straight manifold, location close to manifold exit to head, with pulsed flow). It is seen that heat transfer magnitude in the port is higher or equal to that in the manifold with highest values recorded at the back of the intake valve.

For the flow phases other than displacement backflow, a function \( s_i \) is tabulated which fits the \( \text{Nu}/\text{Re}^{0.5} \) data. In time, the function is scaled into the interval 0 to 720°CA. The magnitude of the function differs between the probes on the port wall and the valve. It is scaled such that it peaks at approximately 2 and 4 during the middle of
the induction flow for the probes in the port and on the valve, respectively. Figure 4-15 plots the shape function on arbitrary scales.

Even when employing calculated gas temperatures, temporal resolution of the heat transfer coefficients is limited by the resolution of the heat transfer data. Heat transfer coefficients are most accurately calculated at the time of maximum heat flux which approximately coincides with maximum backflow rate into the intake port. At this time temporal gas temperature gradients (prediction) are also flat. Thus determined heat transfer coefficients are listed in Table 4-5 for all probes. It is found that the peak value does not appreciable change with intake pressure nor engine speed. Rather, duration of backflow changes. So does the gas temperature, which for longer backflow duration reaches higher values due to higher burnt gas concentrations in the port. Heat transfer is highest during the actual jet backflow phase and relaxes to lower values during the reverse induction of high temperature burnt gas back into the cylinder. Beginning of backflow (IVO) and termination of reverse induction of burnt gas into the cylinder are given in Table 4-6. Heat transfer coefficients for the whole backflow period which fit the heat flux data are obtained through a second shape function \( s_2 \). It is tabulated on an arbitrary but finite time interval and has a peak value of one. To obtain time resolved heat transfer coefficients during backflow, it is scaled in time onto the tabulated interval for start and end of backflow (reverse induction) and in magnitude to the tabulated \( h_{\text{max}} \) at the time of maximum heat flux. Figure 4-15 also shows \( s_2 \).

Fitted heat transfer coefficients for the whole cycle are obtained through the two shape functions. Scaling requires two inputs: the cycle averaged Reynolds number and the intake pressure. Heat transfer coefficients not considering valve overlap backflow are obtained from \( h_1(\text{CA}) = s_1(\text{CA}) f(\text{location}) \frac{k}{D_{\text{hyd}}} \Re^{0.5} \). For manifold absolute pressures less than one, heat transfer coefficients during valve overlap backflow are determined through \( h_2(\text{CA}_{\text{back}}) = s_2(\text{CA}_{\text{back}}) h_{\text{max}}(\text{probe}) \). At the beginning or end of the valve overlap backflow period, heat transfer coefficients \( h_2(\text{CA}) \) may be smaller than those predicted through \( s_1(\text{CA}) \). The actual \( h \) value is taken as the larger of the two. Fitted heat transfer coefficients for various mean flow rates and intake pressures are shown in Fig. 4-16. Predicted heat fluxes employing the fitted heat transfer coefficient are seen to compare quite well to experiment, Fig. 4-17.

4.8 Spray targeting

With liquid fuel injection, the injector was mounted into the intake manifold of the test engine, with the injector tip located approximately 15 cm upstream of the intake valve seat. This location was approximately 2 cm further upstream than the production engine's configuration, but the basic spray targeting was retained. The spray targeting is shown in Fig. 4-18. 80% percent of the mass flux of the spray was contained within a hollow cone whose opening angle was 4 to 12 degrees from the spray centerline [4-12]. (The Sauter mean radius of the droplets in the spray was
approximately 100 μm as calculated from [4-12]). In the top view, the solid lines depict the intersection between the port wall and the inner and outer envelope of the hollow cone. The different angles at which the spray impinged onto the walls caused varying mass deposition rates per unit area. The numbers in the graph indicate the fractions of total spray deposited in the control areas enclosed by the dashed lines. Approximately 30% of the spray impinged on the intake valve, which was in a plane perpendicular to the spray axis. The area on the bottom of the port close to the valve was approximately parallel to the spray axis, hence deposition was small, about 5% of the spray. The bifurcation wall of the port had an angle similar to the spray, and it was estimated that at most 15% of the spray was deposited there. The remainder of the spray, 50%, deposited quite far upstream of the valve on the port floor.

Probes on the top wall, \(q_{\text{top,13mm}}\) and \(q_{\text{top,59mm}}\), were targeted by the spray. The second probe on the valve back was not targeted because the valve stem obstructed the spray. The other probe on the valve, \(q_{\text{valve,bot}}\) was fully within the spray targeting region, as was the most upstream probe on the bottom wall, \(q_{\text{bot,63mm}}\). The two probes further downstream on the port wall are at the edges of the targeting area: probe \(q_{\text{bot,42mm}}\) was on the inside, probe \(q_{\text{bot,13mm}}\) was on the outside. Given the above described spray deposition characteristics relative to the probe location, it was expected that probe \(q_{\text{bot,13mm}}\) would show the fuel related signal. Additionally, the wall was sloping upwards to the side of the wall, where the sensor was mounted at 8 o’clock position (viewed from the manifold flange into the port, 6 o’clock was the bottom 12 o’clock was the top wall). Film flow down the intake port will therefore not reach this sensor. While probe \(q_{\text{bot,42mm}}\) was also not within the direct targeting area, its location was such that film flow may transport fuel onto it from upstream deposition areas. Film flow would increase the amount of fuel reaching the valve, either from the bottom of the port, or from the bifurcation wall in the middle of the port. Significant transport by fuel films was primarily expected at low port wall temperatures where fuel vaporization rates are slow.

### 4.9 Heat transfer for port fuel-injected cases

Liquid fuel experiments were conducted for 1500 rpm/0.5 and 1 bar MAP, closed and open valve injection, isooctane and indolene fuel. Thermal steady state results were obtained for coolant temperatures ranging from 25°C, 40°C to 75°C. Additionally, heat transfer rates were investigated for a simulated engine start-up. This experiment was started at ambient engine temperature and terminated once heat fluxes reached quasi-steady values.

Heat flux instrumentation for the current experiment was chosen primarily for gaseous fuel experiments with moderate heat flux levels, which requires high gain sensors. However, the heat transfer resistance of heat flux sensors increases with sensitivity when the temperature differential over a thermal barrier is used as the measurement effect. This is the case for the employed thermopile probes which have a steady state heat transfer coefficient of approximately 2,500 W/m²-K (Appendix A). For liquid fuel injection onto surfaces close to or above the fuel’s saturation temperature, very large heat fluxes are expected with strong dependence on wall temperature. This is not only the case for nucleate boiling heat transfer which is significant for wall temperatures exceeding the fuel’s saturation temperature by more than approximately 10°C (depending on surface conditions). Convectively
enhanced vaporization rates off liquid fuel surfaces close to the saturation temperature tend to infinity according to commonly employed predictions [4-13]. For very thin fuel films, the sensor's heat flow resistance may exceed that of the fuel film thereby limiting for free surface vaporization. For thick fuel films, nucleate boiling would be expected to be the dominating vaporization mechanism. Again, the sensor interferes by limiting heat supply from the wall. Additional intrusions into the system are steps in the form of glue layers of about 0.5 mm thickness at the edges of the sensors, which were necessary to prevent the sensors from peeling off the wall. These prevent thin fuel films, expected at high wall temperatures, from flowing onto the sensor.

While for these reasons quantitative interpretation of measurements may be biased at high wall temperatures, several features of the fuel preparation process can be investigated. The presence or absence of fuel is clearly detected from the heat flux data thereby defining spray targeting and secondary fuel deposition locations. Additionally, the measurements are considered to be quantitative for wall temperatures up to approximately 60°C. This will be discussed more fully in the section Fuel Vaporization Rates Inferred From Heat Transfer Data.

Figures 4-19 through 4-21 show steady state results for coolant temperatures of 25°C, 40°C, and 75°C in comparison to measurements from propane fired operation. Operating condition is 1500 rpm/0.5 MAP, with isoctane and indolene fuel, with open and closed valve injection (start of injection [SOI] 120° and 580°, respectively). Figure 4-22 corresponds to the 40°C coolant temperature case but intake pressure was elevated to 1.0 bar MAP. Spray impingement is seen in the heat flux signals as a negative hump, stemming from the sensible heat required to bring the fuel film to the wall temperature and from fuel vaporization at the time of spray impingement on the probe surface. For closed valve injection, spray arrival was seen around 200°CA while open valve injection showed spray arrival around 650°CA. The spray arrival times convert to a mean spray speed of 17 m/s with little impact of intake pressure.

For all wall temperatures and intake pressures with closed valve injection (SOI 120°), spray impingement was seen in a similar manner for probes \( q_{\text{bot,42mm}} \) and \( q_{\text{bot,63mm}} \) (bottom of port, 42 and 63 mm upstream of valve seat). It was stronger for \( q_{\text{bot,63mm}} \), which lay more directly in the targeting area. The heat fluxes did not approach the dry values (propane) prior to intake valve opening, indicating the continued presence of a liquid film. Valve overlap backflow was not detected since the probes were too far upstream of the intake valve. With the onset of the induction flow, the heat flux magnitude was elevated over the propane fired case indicating continuous presence of fuel. Towards the end of induction flow, the upstream probe \( q_{\text{bot,63mm}} \) appeared to dry for the highest coolant temperature case (0.5 bar MAP), which will be discussed later. For open valve injection and 0.5 bar MAP spray arrival occurred after approximately the same time of flight. The magnitude of spray impingement was reduced for 0.5 bar MAP and further reduced for 1.0 bar MAP, possibly because the higher density flow was more effective in deflecting the spray. Injection duration at 1 bar MAP and SOI 580° was long enough to cause spray arrival after the induction flow ceased, then causing a significant arrival signal. The upstream probe on the top wall, \( q_{\text{top,50mm}} \), was essentially dry throughout the cycle for both injection timings.
Spray impingement at locations further downstream was again consistent with the targeting and only a small amount of impingement was seen at $q_{\text{boc,13mm}}$. The valve overlap backflow (0.5 bar MAP) was significantly reduced, particularly for low wall temperature fuel injected cases. This is thought to be an interaction of backflow with liquid fuel puddles, which have been observed at low wall temperatures near the valve seat.

Qualitatively, the behavior for the valve was similar to the port. A pronounced spray arrival signal was consistent with spray targeting. Closed valve injection showed again more spray impingement than open valve injection. Higher valve temperatures for the latter case confirm that less spray impingement occurred. Valve overlap backflow for the throttled cases was significantly reduced at low wall temperature as was observed in the port close to the valve. One difference was apparent when comparing port and valve data: the valve, particularly for indolene injection at low coolant temperatures, showed elevated (more negative) heat transfer early during the induction flow. As this phenomenon decreases with increasing coolant temperatures, it is assumed that fuel impacted onto the valve from further upstream locations with less fuel being available at higher coolant temperatures.

Valve temperatures were quite high for all steady state cases, while engine warm-up transients start with ambient wall temperatures. Hence fuel related heat transfer rates off the valve during warm up are expected to be lower. Experiments were therefore conducted by motoring up the engine to 1500 rpm/0.5 bar MAP and then starting firing with injection at the stoichiometric mixture ratio. The secondary coolant circuit of the engine was shut off during the experiment which allowed the coolant temperature in the primary circuit to heat up in a manner quite similar to that of a production engine’s configuration. The fuel used was indolene, with open and closed valve injection, and propane was again used as a reference fuel. Wall and valve temperatures, and heat fluxes were recorded. Figure 4-23 shows the valve temperature increase with time. The corresponding heat fluxes are shown in Fig. 4-24, where a moving average over two cycles was applied. Wall temperatures which correspond to the 25°C coolant temperature steady state case were reached after 105 s, while 40°C coolant temperature was measured at the end of the experiment at 270 s. Steady state heat fluxes for both 25°C and 40°C coolant temperature compared well with the warm-up experiment, see circles in Fig. 4-24. This indicates that the system behaved in a quasi-steady manner from 100 s on, and possibly earlier. Heat fluxes prior to 100 s increased in a continuous manner, which was attributed to increasing fuel effect (dry heat fluxes are quite low for all measurements as indicated by the propane fired measurements).

To summarize the above, measured heat fluxes agreed well with expectations based on the static spray targeting for closed valve injection. Spray arrival was seen in a manner fully consistent with injection timing. Flow is known to affect the trajectories of small droplets. Consistently, less fuel was seen on the probes on the bottom wall and the valve when injecting into the induction flow. In the next section, we will compare the measured heat fluxes to the sensible and latent heat necessary to vaporize the injected fuel mass.
4.10 Fuel vaporization rates inferred from heat transfer data

Liquid fuel effects were approximately separated from gas flow heat transfer by subtracting liquid and propane fueled data:

\[ \dot{q}''_{\text{fuel},i} = \dot{q}''_i \text{ (liquid fuel operated) - } \dot{q}''_i \text{ (propane operated)}, \quad (4-5) \]

where index \( i \) stands for the individual heat flux probes. This difference reflects the sensible and latent heat necessary to heat and vaporize the impinging liquid fuel. Implicitly, it was assumed that the gas flow heat transfer for liquid fuel operation was essentially unchanged relative to the propane case.

Researchers are interested in relating the measured heat fluxes to the fraction of the injected fuel which vaporizes in the port and off the back of the intake valve. Representative areas \( A_i \) were assigned to each of the heat flux probes in a manner such that the summation of these areas equaled the wetted area based on the spray targeting. Relative heat flux magnitudes of the different transducers for liquid fuel operation were found consistent with the amount of fuel deposited onto the transducer as inferred from the spray targeting (for closed valve injection where no mean flow modified the targeting). The probe specific areas of spray impingement are shown in Fig. 4-18. The deposition rate into the areas \( A_i \) varied with distance from the injector and the angle of the areas relative the impinging spray. We estimated the fraction \( f_i \) of the fuel spray which impinged onto the control area \( A_i \) from geometric considerations. The areas and impinging fractions are given in Table 7.

The mean latent heat required for vaporization was calculated from:

\[ \bar{\dot{q}}''_{\text{latent},i} = \bar{m}_{\text{target},i} h_{fg} / A_i, \quad (4-6) \]

with \( \bar{m}_{\text{target},i} = m_{\text{fuel per cycle}} / t_{\text{cycle}} \), \( m_{\text{fuel per cycle}} \) is the injected fuel mass per cycle, and \( t_{\text{cycle}} \) the time for one cycle. \( \bar{\dot{q}}''_{\text{latent}} \) computed to 4.0, 0.65, 3.5, and 3.8 W/cm² for probes \( \dot{q}_{\text{top,13mm}}, \dot{q}_{\text{bot,13mm}}, \dot{q}_{\text{bot,62mm}}, \) and \( \dot{q}_{\text{bot,63mm}} \), respectively. The mean heat transfer rate to heat the fuel to the wall temperature is calculated from

\[ \bar{\dot{q}}''_{\text{sensible},i} = \bar{m}_{\text{target},i} c_{\text{fuel}} (T_{\text{fuel}} - T_{\text{wall},i}) / A_i, \quad (4-7) \]

where \( c_{\text{fuel}} \) is the specific heat capacity, and \( T_{\text{wall},i} \) is the wall temperature which was replaced by the fuels' saturation temperature if the former exceeded the latter.

The potential for fuel vaporization was assessed by employing the well known analogy between heat and mass transfer [4-13] using our experimentally determined heat transfer coefficients. The mass fluxes predicted by analogy should constitute an upper limit for vaporization, i.e. if the control area is fully covered by a fuel film. We converted these mass fluxes predicted by the analogy to heat fluxes:

\[ \bar{\dot{q}}''_{\text{analogy},i} = \bar{m}''_{\text{analogy},i} h_{fg}. \quad (4-8) \]

Figures 4-25 to 4-27 plot measured fuel related heat fluxes (Eq. 4-4) together with the fuel vaporization potential as predicted by the analogy between heat and mass transfer (Eq. 4-7) for different coolant temperatures.
with isoctane fuel closed valve injection (SOI 120°). Predictions by analogy are not plotted if the measured wall temperature was close to or above the fuel's saturation point. Then, vaporization fluxes (either from the liquid-gas interface or by nucleate boiling from the wall-liquid interface) are expected to increase strongly and the detailed heat transfer mechanism to and at the vaporization site becomes important. These mechanisms depend on many unknown parameters so that we could not include any simple prediction for comparison to the experimental data. Additionally, the heat transfer resistance of the employed sensors will become important for the expected high heat fluxes.

Heat fluxes during fuel arrival were seen to increase for all probe locations with increasing wall temperature. A prediction of the sensible energy required to heat all impinging fuel (Eq. 4-6) is also depicted in Figs. 4-25 to 4-27, where the shape of the curve is chosen to be similar to the spray arrival signal (and similar to the prediction by [4-10]). While the measured spray arrival signal is of somewhat lower amplitude, it correlates with the calculated sensible heat (impingement on probes \( q_{bot,13mm} \) and \( q_{bot,42mm} \) is probably less than expected due to the uncertainties in the spray targeting analysis causing a smaller spray arrival signal in comparison to the other probes).

Prior to fuel arrival, heat fluxes during the stagnant portion compare quite well to the analogy if fuel is present. However, it is apparent that during the induction flow phase, particularly for high wall temperatures, measured values fall below the prediction by analogy. The measured fluxes first rise with the onset of flow, but continuously decrease later into the induction flow phase. We conclude that fuel is transported off the sensor in a manner different from the adjacent wall. This is discussed in more detail in the following sections. The required heat for fuel vaporization was obtained from Eq. 4-5. For 25°C coolant temperature, the values measured or predicted by analogy indicate that approximately 1/20th of the injected fuel could be vaporized in the port. Based on the analogy, this value doubles to 1/10th for the 40°C coolant temperature case. Analogy of heat and mass transfer indicates that wall temperature has to approach the saturation temperature to allow complete vaporization. Probe \( q_{bot,63mm} \) has a wall temperature of 61°C for 75°C coolant temperature, and analogy between heat and mass transfer indicates that only one half of the impinging fuel could be vaporized. As film flow will transport liquid fuel to areas of wall temperature close to and above the saturation point, it is assumed that all the wall deposited liquid fuel vaporizes for this coolant temperature.

Measured heat fluxes for high wall temperatures fall below the potential for vaporization and also below what is required to vaporize the fuel impinging onto the sensors. We believe that fuel flow phenomena on our sensors are different from the surrounding port wall. This may be due to the fact that the surfaces are significantly different: smooth plastic of the sensor compared to a rough aluminum wall. Additional intrusions are steps in the form of glue layers of about 0.5 mm thickness at the edges of the sensors, which were necessary to prevent the sensors from peeling off the wall. We believe for that for high wall temperatures, during the main induction flow, fuel on our sensors is quickly dragged in the downstream direction off the sensor center were the heat fluxes are actually measured. However, fuel from further upstream locations is prevented from flowing onto the sensor by the
finite step at the sensor edges (the different surface roughness may also play a role). The decrease in signal during forward flow of our fuel related signal provides support for this conclusion. We expect that this phenomenon becomes more pronounced at higher wall temperatures, where a typical length scale of fuel film or fuel cluster thickness is likely to decreases. Disturbances brought into the system by our instrumentation become thus more important.

If fuel films where evenly distributed over the wall and transported by Couette flow, we estimate that the films moves about 0.5 mm per cycle in downstream direction. As analogy between heat and mass transfer indicates that uniformly distributed fuel could vaporize within one engine cycle for wall temperatures reasonably close to the fuel’s saturation point, differences in film flow phenomena on our sensors should not impact strongly. Even when allowing that the sensor represents a additional heat flow resistance in the system, all impinging fuel should vaporize within one cycle given high enough wall temperature. However, mean heat fluxes of approximately 3 W/cm² are not measured on any of our sensors. We therefore conclude that fuel films do not build in a uniformly distributed manner but deposit and transport as droplets or rivulets.

It is well known that fuel metering transient cause fuel-air mixture ratio excursions. The engine needs of order 5 to 10 cycles (for Tₜ₀₀ = 80 and 40°C, respectively) to equilibrate to the new operating condition. We apply our conceptual model of the vaporization process to the example of a fueling transient where the fueling for a warm engine is switched off until the engine is dried out, and then fueling is resumed at stoichiometric rate. The model holds that fuel of one injection event is deposited in a discrete manner onto the walls at locations where droplets impinge. Based on measured size distributions in the spray [12], we estimate that fuel of one injection event covers approximately 10% of the target area. This area is not sufficient to vaporize all the injected fuel. Subsequent injection events deposit more fuel at random locations within the targeting area thus increasing the fuel surface area for vaporization. Fuel surface area is lost once previously deposited droplets vaporize completely. This destruction term increases with increasing number of droplet clusters. Equilibrium is reached, when freshly created and destroyed area balance. The length scale for fuel wall film flow increases considerably for the fuel deposition pattern our conceptual model suggests when compared to an evenly spread film. Firstly, the mean residence time is 5 to 10 time longer, which enters the characteristic distance in a proportional manner. Secondly, wall fuel droplet clusters have larger characteristic thickness increasing for given wall shear stress (velocity gradient in the fuel film) the mean velocity. Finally, the deposited fuel is likely to experience form drag from the air flow further increasing transport.

While our data are not quantitatively valid for high wall temperatures, the conceptual model suggested is consistent with our data. Mixture vaporization modeling may incorporate the salient feature of this model: deposition of fuel at discrete locations. This reduces the surface area available for vaporization as well as increases the conduction resistance through the liquid fuel for heat to reach the film surface. The latter effect decreases surface temperatures which in turn strongly decreases vaporization rates. For low wall temperatures, measurements
compare well to the analogy between heat and mass transfer. This indicates that the droplet vaporization is sufficiently slow so that the wall is fully covered by fuel under these conditions. Finally, we found that spray impingement on the wall causes a characteristic arrival signal which scaled with the sensible heat required to bring the fuel to the wall temperature. The scaling factor was below one indicating that the fuel film temperature may be less than the wall temperature, which is again consistent with our modeling assumption of discrete fuel deposition with significant heat conduction resistance through the liquid fuel film (droplet and/or rivulet).

4.11 Summary

In this chapter, gas and liquid flow processes in the intake port were investigated. Gas temperature measurements in comparison to a CFD prediction reconfirmed the conceptual picture of Chapter 1 attained for the intake processes. A 1-D model was presented which captures most of the experimentally observed phenomena except the valve overlap backflow’s tendency to propagate along the top wall of the port. This was not considered to be of major importance, as measured heat fluxes during valve overlap backflow did not differ strongly between the top and bottom port wall.

Heat flux measurements showed that for a typical low-load propane-fired operating condition, heat transfer in the port caused a mean intake air temperature increase of approximately 10°C. The main different intake flow phases, induction or forward flow, displacement backflow, and valve overlap backflow, contributed approximately 10°C, 3°C, and negative 3°C, respectively. These mixture temperature changes are expected to be also applicable for liquid fuel injected cases.

Heat transfer coefficients based on the experimental heat flux measurements were reported for all flow phases. The gas temperature used to compute the heat transfer coefficients was either the experimentally measured one, or one obtained from a 1-D prediction. The latter was employed for the valve overlap backflow phase, where experimental gas temperature data showed too large gradients to compute meaningful heat transfer coefficients. Time-averaged heat transfer coefficients for the port wall compared well to those of the manifold experiment (excluding the valve overlap backflow phase which did not reach the manifold).

Heat transfer measurements for liquid fuel injected cases gave insight into the mixture vaporization process of port fuel injected engines. Spray targeting was clearly detected by this measurement method through a characteristic spray arrival signal. The magnitude of this signal correlated with the sensible heat required to bring the fuel to the wall temperature. From the arrival time of the impinging spray, a spray propagation speed of 17 m/s was computed. The spray speed did not change appreciably between closed and open valve injection indicating little interaction presumably of the larger droplets in the spray with the induction flow. However, some spray deflection into the flow direction was apparent probably through reduced wall impingement of the small droplets for open valve injection.
Fuel vaporization rates were inferred by subtracting liquid fuel heat transfer data from propane fired data at the same operating condition (load, speed, and coolant temperature). These were compared to the fuel vaporization potential as predicted by the analogy between heat and mass transfer employing the experimental heat transfer coefficients. For isooctane and low wall temperature, heat transfer rates were similar to the prediction by analogy. For high wall temperatures, it was concluded that the fuel is transported (sheared) off the sensor during the induction flow. A conceptual model for fuel deposition onto the walls was presented which suggested that fuel spray deposition onto the wall causes discrete patches of wall fuel film (droplets or rivulets). While this model was conceived to help explain the experimental data, it would have impact for fuel vaporization modeling.

Heat transfer data taken on the back of the intake valve showed liquid fuel related heat fluxes for the time of early induction flow. As this occurred primarily at low coolant temperatures where fuel vaporization potential in the port does not suffice to vaporize all the injected fuel, fuel transport onto the valve from upstream locations was thought to be cause. This phenomenon occurred more strongly for indolene injection indicating that less vaporization in the port is taking place for this fuel.

4.12 References


### 4.13 Tables

**Table 4-1: Engine data**

<table>
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<th>Parameter</th>
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<td>Connecting Rod Length (mm)</td>
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<td>EVO-0.1 mm lift (°BBDC)</td>
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<td>EVC-0.1 mm lift (°ATDC)</td>
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**Table 4-2: Variable range for wall and valve temperature fit. Liquid fired cases are only for 1500 rpm and intake pressures from 0.5 to 1 bar.**

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<thead>
<tr>
<th>Variables</th>
<th>rpm</th>
<th>( P_{\text{intake}} ) (bar)</th>
<th>( T_{\text{cool}} ) (°C)</th>
<th>x (mm)</th>
<th>( \phi ) (Deg)</th>
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<td>lower limit</td>
<td>1000</td>
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**Table 4-3: Parameters of wall temperature fit.**

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<tbody>
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<td></td>
<td>Offset</td>
<td>rpm</td>
<td>( P_{\text{intake}} )</td>
<td>( T_{\text{cool}} )</td>
<td>( x )</td>
<td>( \alpha )</td>
<td>( x\alpha )</td>
<td>( x^2 )</td>
<td>( \sigma ) (°C)</td>
<td>( \varepsilon_{\text{max}} ) (°C)</td>
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<tr>
<td>Propane</td>
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<td>Isooctane</td>
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<td>3.0</td>
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<td>5.0 ( 10^{-2} )</td>
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**Table 4-4: Parameters of valve temperature fit.**

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</thead>
<tbody>
<tr>
<td></td>
<td>Offset</td>
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<td>( P_{\text{intake}} )</td>
<td>( T_{\text{cool}} )</td>
<td>( \sigma ) (°C)</td>
<td>( \varepsilon_{\text{max}} ) (°C)</td>
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<tr>
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66
Table 4-5: Heat transfer coefficients at time of maximum heat flux during valve overlap backflow.

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<thead>
<tr>
<th>Probe ID</th>
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<tbody>
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<td>( Q_{\text{valve,bot}} )</td>
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<tr>
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<td>300</td>
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<tr>
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<td>500</td>
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<tr>
<td>( Q_{\text{bot,42mm}} )</td>
<td>50</td>
</tr>
<tr>
<td>( Q_{\text{top,59mm}} )</td>
<td>30</td>
</tr>
<tr>
<td>( Q_{\text{bot,63mm}} )</td>
<td>30</td>
</tr>
</tbody>
</table>

Table 4-6: Duration of valve overlap backflow.

<table>
<thead>
<tr>
<th>MAP [bar]</th>
<th>CA begin</th>
<th>CA end</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.3</td>
<td>540</td>
<td>561</td>
</tr>
<tr>
<td>0.5</td>
<td>540</td>
<td>558</td>
</tr>
<tr>
<td>0.7</td>
<td>540</td>
<td>555</td>
</tr>
<tr>
<td>1.0</td>
<td>540</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 4-7: Spray targeting: \( A_i \) is the probe specific area (x2 for twin port) and \( f_{\text{target},i} \) is the fraction of the spray impinging at that location.

<table>
<thead>
<tr>
<th>Probe ID</th>
<th>( A_i ) [cm(^2)]</th>
<th>( f_{\text{target},i} ) [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>( Q_{\text{valve,bot}} )</td>
<td>0.94 x 2</td>
<td>0.25</td>
</tr>
<tr>
<td>( Q_{\text{bot,13mm}} )</td>
<td>2.0 x 2</td>
<td>0.04</td>
</tr>
<tr>
<td>( Q_{\text{bot,42mm}} )</td>
<td>1.5 x 2</td>
<td>0.17</td>
</tr>
<tr>
<td>( Q_{\text{bot,63mm}} )</td>
<td>2.7 x 2</td>
<td>0.33</td>
</tr>
</tbody>
</table>
Figure 4-1: Engine intake system.
Figure 4-2: Instrumentation in intake port of test engine.
Figure 4-3: Gas temperature measurement in axial direction of port, propane fired 1500 rpm/0.5 bar MAP. The x-axis plots the axial position (0 mm at valve seat), the y-axis the engine crank angle (0 at BC compression stroke), and the z-axis the gas temperature. The measurement position closest to the valve is at a distance of 13 mm to the valve seat.
1500 rpm/0.5 bar MAP, propane, $T_{\text{gas, fresh}} = 28 \, ^\circ\text{C}$, $T_{\text{cool}} = 76 \, ^\circ\text{C}$

**Figure 4-4:** Gas temperature measurement in perpendicular direction to the centerline of the intake port. The measurements are taken between the wall and the valve stem. The x-axis plots the engine crank angle and the y-axis the radial distance from the valve stem (0 mm centerline, 3.5 mm radius of stem) to the wall (11 mm). Vertical elevation is approximately halfways between the bottom and the top of the port. Propane fired 1500 rpm/0.5 bar MAP. Axial distance to the valve seat is approximately 15 mm.
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Figure 4-11 continued
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Figure 4-13 continued
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Figure 4-19 continued
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Figure 4-20 continued
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1500 rpm / 0.5 bar MAP, Tcool 75 C

Figure 4-21 continued
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1500 rpm/1.0 bar MAP, Tcool 40 C

Figure 4-22 continued
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Figure 4-26: $T_{\text{cooler}} = 40^\circ C$. Measured fuel related heat flux in comparison to prediction by analogy, 1500 rpm/0.5 MAP.
Figure 4-27: \( T_{\text{co oil}} = 75^\circ \text{C} \). Measured fuel related heat flux in comparison to prediction by analogy, 1500 rpm/0.5 MAP.
5 Summary and conclusions

Heat transfer and mixture vaporization in the intake system of port fuel-injected engines was investigated. The unsteady nature of the flow processes impacts on many features of the investigated system. Away from the intake valve, the gas flow is approximately a half sine wave for one quarter of the engine cycle and imposed by the piston displacement while the intake valve is open. Two backflow phases into the intake port are observed. The valve overlap backflow which occurs primarily for throttled engine operation, where the residual burnt charge from the cylinder expands into the intake port. The displacement backflow occurs after the induction flow, and is caused by late intake valve closing during the compression stroke.

The nature of these backflow processes was investigated by gas temperature measurements and CFD predictions for 2 and 4 valve per cylinder geometries. From the initial experiment on a 2 valve geometry, it was found that the valve overlap backflow has jet-like character. It attaches to the back of the intake valve and then propagates preferentially towards the top wall of the intake port. It’s impact, which depends on intake system pressure, is confined to the intake port region up to the manifold head flange. Displacement backflow behaves more like a plug flow and propagated less deep into the intake system.

Subsequent experiments to quantify heat transfer in the intake system were therefore split between the intake manifold runner and the intake port. A manifold runner experiment investigated the impact of pipe geometry and the pulsating nature of the flow by an imposed wall heat flux experiment. Two manifold runners were employed—a straight and a highly curved pipe—to bracket practically encountered geometries. Little impact of geometry on heat transfer was found since entry effects dominated. Steady and pulsating (engine) flow was examined. It was found that heat transfer approximately doubles over the steady flow level due to the pulsating engine flow. Heat transfer coefficients were correlated for both geometries and flow configurations. Additional time resolved heat flux measurements at the exit of the intake manifold runner to the cylinder head investigated the physics of heat transfer in pulsating flow in a detailed manner. It was found that heat transfer during the intake valve
closed period slowly relaxed from its level during the forward induction flow. This was attributed to the slow decay of turbulence generated during the earlier induction flow.

Heat transfer investigation in the intake port required time and spatially resolved measurements due to the transient backflow processes with rapidly changing gas temperatures. The transfer function of a commercially available thin film heat flux probe was experimentally determined to allow compensation of the signal which otherwise showed inadequate temporal resolution. An intake port and valve of a typical four valve per cylinder engine was densely instrumented with these heat flux probes and additional fast gas temperature sensors.

The flow phenomena for the four valve cylinder head were again investigated by gas temperature measurements and CFD, and essentially the same flow phenomena were observed as in the two valve geometry. A 1-D model was developed which related the gas temperature and heat flux measurements to the mass and energy balance in the intake port.

Heat transfer for propane fired operation was quantified. Heat transfer coefficients were correlated based on the heat flux and gas temperature measurements. During valve overlap backflow, measured gas temperatures showed strong gradients and were not suitable for heat transfer coefficient calculation. They were therefore replaced by the 1-D prediction for this flow phase. To the knowledge of the author, this is the first study which provided detailed insight into the heat transfer related to backflow.

Heat transfer measurements for liquid fuel injected cases gave insight into the mixture vaporization process of port fuel injected engines. Spray targeting was clearly detected by this method. The spray speed based on the arrival of liquid fuel at the probe locations did not change appreciably for closed or open valve injection indicating little interaction of at least the larger droplets in the spray with the induction flow. However, some spray deflection probably of smaller droplets into the flow direction was apparent through reduced magnitude of the spray arrival signal for open valve injection.

Fuel vaporization rates were inferred by subtracting liquid fuel heat transfer data from propane fired data at the same operating condition (load, speed, and coolant temperature). These were compared to the fuel vaporization potential as predicted by the analogy between heat and mass transfer based on experimental heat transfer coefficients. For isoctane and low wall temperature, fuel related heat flux was comparable to the vaporization flux predicted by the analogy. For high wall temperatures, it was concluded that the fuel is sheared off the sensor during the induction flow. Measured heat fluxes for indolene injection closely resembled those of the isoctane case. A conceptual model for fuel deposition onto the walls was presented which holds that fuel spray deposition onto the wall causes discrete patches of wall fuel film (or droplets). While this model was conceived to help explain the experimental data, it may well have impact for fuel vaporization modeling.
Appendix A: Transfer function of thin film heat flux sensors

Two heat flux transducers of different frequency response characteristics are compared. The sensors are evaluated by mounting them adjacent to one another in the heated intake manifold runner of a motoring spark ignition engine. Their ensemble averaged signals are transformed into the frequency domain. The transfer function for the slower of the two probes is derived from the amplitude and phase relation between the two signals.

A.1 Heat flux sensors

This appendix investigates the behavior of a thin film heat flux probe for use in time varying low heat flux applications. While there are other kinds available (see e.g. [A-1]), the two most promising commercially available probes are of the thermopile type. Each of these devices consists of an array of thermocouples located on opposite sides of a thermal barrier. Each individual thermocouple pair measures the temperature difference across the barrier and returns a small voltage signal proportional to it. Because the thermocouples are connected in series, their signals add.

Signal strength increases proportionally to the number of thermocouples added. Another way to enhance the sensitivity is to increase the thickness of the thermal barrier, which has the advantage of maintaining constant sensor dimensions. However, the probe response time, based on the diffusion time scale, decreases proportionally as the square of the thickness, $t \sim l^2/\alpha$.

The heat flux range of interest are low signals between 0.1 and 1 W/cm² with a frequency content up to approximately 300 Hz. Two probes are commercially available in the United States with specifications in or near that range:

- **RdF Corporation**, based in Hudson, NH, offers custom made thin film thermopiles. These are manufactured using Kapton (polyamide) sheets of thickness as little as 12 μm, where three sheets are necessary to build a probe. The fastest of these probes has a calculated 63% response time of 8 milliseconds. Such a probe is employed with 20 thermocouple pairs resulting in an approximate gain of 660 μV/(W/cm²). The probe has a square shape with 1 cm sides.

- **Vatell Corporation** of Christianburg, VA, manufactures the HFM 6 thermopile by depositing thin films on an aluminum nitride substrate. The response time is on the order of 10 μs and its gain is approximately 15 μV/(W/cm²) [A-2]. The sensor comes in a 6 mm diameter housing of 2.5 cm length. A special amplifier which is offered along with the probe is employed for the current investigation.

A resistance thermometer is built into the Vatell probe and the Vatell amplifier provides a temperature signal. If required, a thermocouple can be embedded into a RdF probe. The temperature readings may be used to compensate the heat flux signals as well as to assess the thermal grounding of the probes.
Measuring signals of the order of 1 μV to resolve 0.1 W/cm² can be a daunting task. Even moving the connecting coaxial cables (BNC) will cause disturbances on the order of the main signal. This shows possible limitations of the Vatell probe. Low frequency noise may be expected, making DC measurements with this probe more difficult. On the other hand it is clear that the RdF probe will only provide limited temporal resolution. Its exact limits are investigated in this appendix and a method is devised to enhance its frequency response. This allows use of the RdF probe for a wider scope of applications, which is desirable due to its price, flexibility, ease of mounting, and the ready detection of small DC signals.

This appendix determines the transfer function of the RdF sensor. A similar work was recently published by Shayler et al. [A-3] utilizing the analytical approach of Epstein et al. [A-4]. They corrected the heat flux signals based on the solution for transient conduction in a slab. A prediction of the sensor response was experimentally verified using a laser square pulse as test signal. In the present approach, the transfer function of the RdF sensor is determined from the comparison of its signal to the much faster output of the Vatell probe. This is accomplished by subjecting both probes to the same broad band convective heat input. Besides simplicity, this procedure has the advantage of not requiring surface coating of the probe, which becomes necessary for a radiative calibration to minimize reflection. Also given is a simple analytical solution based on transient conduction in a semi-infinite body. The algebraic structure of this solution serves as a model expression to fit the experimental data of the transfer function into a compact expression.

A.2 Approach

A thermopile such as the RdF probe behaves to a good approximation as a linear system. Knowing the transfer function of the sensor, it is possible to reconstruct the heat flux at the surface of the sensor \( \dot{q}(t) \) given the output signal \( R(t) \). The transfer function is a function of the frequency and describes the attenuation and phase shift of the output signal for any sinusoidal heat input. With the chosen notation for the amplitude and phase shift of the transfer function, \( A(f) \) and \( \Theta(f) \), the sensor response to a heat input \( \sin(2\pi f t) \) is:

\[
R(t) = A(f)\sin(2\pi f t + \Theta(f)). \tag{A-1}
\]

One can now examine periodic heat flux variation \( \dot{q}(t) \) with a fundamental circular frequency \( f_0 \). Given the Fourier Transform of the output signal \( R(t) \)

\[
R(t) = \sum_{n=0}^{\infty} a_{\text{RdF,n}} \sin(2\pi f_0 t + \phi_{\text{RdF,n}}),
\]

it is possible to reconstruct the heat flux at the surface of the probe.
\[ \dot{q}(t) = \sum_{n=0}^{\infty} \dot{q}_n = \sum_{n=0}^{\infty} \frac{a_{Rd,F,n}}{A(f_n)} \sin\left(2\pi f_n t + \phi_{Rd,F,n} + \Theta(f_n)\right). \tag{A-3} \]

An analytical solution of the transfer function is given in the next section, while its experimentally determined counterpart is developed later.

### A.3 Model transfer function

A closed form solution of the transducer response is readily obtained if the transducer is modeled as a semi-infinite solid with constant thermal properties. Inserting \( e^{-r \sqrt{\alpha t + k x}} \) into the field equation for the transducer

\[ \frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2} \]

gives the well known solution \( e^{-x/X} e^{-r(2\pi f - x/X)} \), which satisfies harmonic boundary conditions, see e.g. [A-5]. The term \( X = \sqrt{\alpha / (\pi f)} \) appears twice, where \( f \) is the frequency of the sinusoidal input, \( x \) denotes the spatial coordinate from the surface into the transducer, and \( \alpha \) is the thermal diffusivity of the transducer material Kapton \((\alpha \approx 10^{-7} \text{ m}^2/\text{s})\). The response at the center of the sensor approximates the output signal of the transducer. Thus, \( x \) in the solution is set to \( l/2 \), where \( l \) is the thickness of the transducer \((l \approx 45 \mu \text{m including two layers of glue})\). The attenuation of the input signal then becomes

\[ A_{\text{model}}(f) = e^{-\frac{l}{2(\pi f / \alpha)^0.5}}, \tag{A-4} \]

and the phase shift is

\[ \Theta_{\text{model}}(f) = -\frac{l}{2} \left( \frac{\pi f}{\alpha} \right)^{0.5}. \tag{A-5} \]

### A.4 Test apparatus

The sensors are installed inside the intake manifold pipe of an experimental one-cylinder engine. The hydraulic diameter of the manifold is approximately 3.5 cm and its length is 35 cm. The sensors are mounted with 1 cm separation on the side wall of the manifold, approximately 5 cm upstream of the exit to the cylinder head. The Varion sensor fits into a specially made hole through the wall while the RdF probe is surface mounted using a low viscosity glue (Permabond 922 of Permabond International). A wall-to-gas temperature gradient is produced by electrically heating the manifold.

The engine is driven by a dynamometer at fixed speeds from 1,000 to 4,500 revolutions per minute crankshaft speed (rpm) with intake manifold absolute pressure (MAP) of 0.5 and 1 bar. The speed range translates
to fundamental circular frequencies from 8.3 to 37.5 Hz (note that one engine cycle encompasses to two crankshaft revolutions). The periodically varying flow in the manifold causes a convective heat flux into the wall with a frequency content well beyond the first harmonic. The heat flux signals are recorded for 25 cycles and subsequently ensemble averaged.

A.5 Experimental transfer function

The Vatell probe has sufficient response characteristics to accurately measure the heat flux data over a range of frequencies well beyond those covered in the experiments. Because the Vatell probe lies adjacent to the RdF sensor, it is assumed that they receive nearly the same input. The Vatell signal thus provides a good reference to approximate the heat input \( \tilde{q} \) into the RdF probe. Using the signals from both probes, the transfer function for the RdF probe can be constructed experimentally.

In the current approach any periodic broad band heat input, such as the heat flux encountered in the intake manifold at any particular operating condition, is sufficient to generate a discrete approximation of the transfer function. In order to accomplish this the signal of the Vatell (reference) and RdF probes are Fourier transformed. The attenuation components of the transfer function are then computed according to:

\[
A_{\text{exp}}(f_n) = \frac{a_{\text{RdF},n}}{a_{\text{Vat},n}}, \quad n = 0, 1, 2, \ldots
\]  

(A-6)

and the phase shifts from:

\[
\Theta_{\text{exp}}(f_n) = \eta_{\text{RdF},n} - \eta_{\text{Vat},n}, \quad n = 0, 1, 2, \ldots
\]  

(A-7)

\( a_{\text{RdF},n}, a_{\text{Vat},n}, \eta_{\text{RdF},n}, \) and \( \eta_{\text{Vat},n} \) are the Fourier coefficients of the signals, whose notation follows the convention in Eq. A-2. Index one corresponds to the frequency of the first harmonic \( f_1 \), which is determined by the duration of one cycle.

This procedure is repeated at various engine speeds to obtain values of the transfer function at different sets of frequencies. Additionally, data are included with manifold pressure reduced from 1.0 to 0.5 bar to obtain more independent data sets. All data are fitted to two general expressions representing the transfer function of the RdF probe at all frequencies. These expressions resemble the model solutions given in Eqs. A-4 & A-5:

\[
A_{\text{fit}}(f) = e^{af^{0.5}}, \quad (A-8)
\]

\[
\Theta_{\text{fit}}(f) = af^{0.5}, \quad (A-9)
\]

where \( a \) equal to -0.14 yields the best fit in a non-linear least square sense. Figure A-1 plots the fit of the transfer function to the experimental data. It also shows the subtraction of the experimental data from the fit (which is zero in the ideal case), and the standard deviations of the two transfer function components. The standard deviations are 0.127 (non-dimensional) and 1.09 (radian) for gain and phase, respectively, with approximately 75% of the data
scatter within one standard deviation. Note that the aggregate standard deviations are adequate, as the scatter appears to be independent of frequency.

Electric noise could be one error source. Its magnitude is assessed by covering the sensor with thick, insulating tape to shield the probe from heat input and pursuing an otherwise regular experiment. Data obtained in this manner exhibit very little noise. The most likely cause for data scatter is therefore due to non-identical heat input signals into the two probes, which may be caused by their finite separation. A set of data from any particular operating condition not only describes the transfer function of the RdF sensor, but also the component relating the difference in heat inputs. It is very probable that this component changes in an arbitrary manner with changes in the operating condition. The fitted value built from all repetitions of the experiment is therefore a much better estimate of the RdF sensor transfer function.

The individual data points for amplitude and phase lag may be thought of as members of two populations whose expectation values constitute the transfer function of the RdF sensor. The uncertainty in the transfer function will be required to estimate the error induced by its application. It is assumed that the transfer function has a limited number of degrees of freedom, in this case one, which is the number of constants necessary to obtain an apparently unbiased fit. The total number of experimental gain/phase lag data pairs minus the number of degrees of freedom may then be considered as the number of repetitions of the experiment, \( m = 110 \). The uncertainty in the transfer function decreases to \( \frac{\sigma_t}{\sqrt{m}} \), where the \( \sigma_t \) are the respective standard deviations of the data for gain and phase lag. Note that the undivided standard deviations reflect the difference in heat inputs into both probes for any particular operating condition. The entire values are therefore employed as error bounds on the transfer function when comparing a frequency corrected RdF trace to its corresponding Vatell signal.

### A.6 Comparison of transfer functions

The fit to the experimental data systematically deviates from the model in that the gain is lower and the phase lag is larger. This might be due to the simplistic modeling assumption of semi-infinite transducer dimensions, which is not valid for low frequency components. At low frequencies, the signal diffuses through the entire thickness of the transducer and further propagates into the different base material. This poses no problem for practical applications as the fit may be employed as transfer function rather than the model.

### A.7 Application of experimental transfer function

Figures A-2 and A-3 show the corrected as well as the uncorrected RdF signal and the Vatell signal (fast probe). The figures are plotted for the lowest and highest experimental frequencies and mass flow rates, respectively. The response-corrected RdF traces using the fitted transfer function (Eqs. A-8 & A-9 inserted into Eq. A-3) agree well
with the Vatell traces and reveal considerably more than the original signals. The signal correction employing the semi-infinite body solution also fares well given that only the readily available thermal diffusivity of the transducer material and the sensor (half-) thickness enter into that calculation.

The response-corrected signal depends (a) on the amplitude and phase of the measured RdF signal and (b) the amplitude ratio and phase lag of the transfer function for each individual frequency in the band of interest. Clearly, errors in (a) and (b) will propagate into the final results.

Conceivable errors affecting (a) are: electric noise or modified heat input due to the presence of the probe. Electric noise is deemed negligible because of ensemble averaging and for the reasons discussed in the Experimental Transfer Function section. Modification of heat input is assessed through the steady-state heat transfer coefficient of the sensor: 
\[ h_{\text{sensor, steady state}} = \frac{k}{l} \], where \( k \) is the thermal conductivity of the transducer material and \( l \) the thickness of the transducer. \( h_{\text{sensor}} \) should exceed \( h_{\text{external}} \) (heat transfer coefficient associated with the source driving heat into the sensor) by a factor of ten or more, thus ensuring that the relative temperature difference over the sensor will be small. Under transient conditions the sensor will have a larger heat transfer coefficient:
\[ h_{\text{sensor, transient}} \approx \sqrt{\frac{k \rho c_p}{l}} \geq \frac{k}{l} = h_{\text{sensor, steady state}} \], where the inequality is achieved through insertion of the diffusion time scale of the probe (see e.g. [A-5]) as an upper time limit. The steady state sensor heat transfer coefficient is approximately 2350 W/m²K while external heat transfer coefficients of the current flow configuration are typically 150 W/m²K. Modifications in heat input into the probe are therefore deemed negligible for the present application.

The uncertainty in the amplitude and phase of the experimental transfer function (b) has been quantified in the above. Error propagation analysis gives a term \( \Delta \hat{q}_n \sim a_{\text{RdF, n}} \Delta A(f_n) / A^2(f_n) \), which demonstrates how an error in the transfer function propagates into the prediction. It is clear from Fig. A-1 that higher frequencies components \( n \) will be associated with larger uncertainties because the uncertainty in the gain \( \Delta A(f_n) \) is approximately constant while the gain \( A(f_n) \) diminishes with increasing frequency. The total error will be bounded if the heat input \( |\hat{q}_n| \approx |a_{\text{RdF, n}} / A(f_n)| \) decays faster than \( A(f_n) \) with increasing frequency. If the high frequency behavior of the input is unknown, bandwidth limitations may be imposed. This approach will not approximate the full input signal, but its Fourier-filtered representation. For the current work, the 0 to 300 Hz range is chosen, where the application of the transfer function promises to be most successful.

In summary, the uncertainties in the transfer function are considered the dominant error sources when correcting an experimental RdF signal. They have been quantified in Experimental Transfer Function section as \( \sigma_i / \sqrt{m} \), where the \( \sigma_i \) are the standard deviations of the experimental gain and phase lag data for the transfer function, and \( m \) is the number of data pairs employed in the fitting procedure. The uncertainties are propagated in a standard way by building partial derivatives with respect to each model amplitude ratio and phase lag. The
individual error contributions are combined for all frequencies by the root-mean-sum criterion, see e.g. [A-6]. The maximum magnitude of the one standard deviation confidence interval is less than 0.02 W/cm² for a 2750 rpm / 1 bar intake pressure operating condition. Its distribution within a cycle is plotted in Fig. A-4 alongside the corrected RdF signal.

Figure A-4 also shows the corresponding Vatell signal. As was previously mentioned, one has to employ uncertainty bounds for the transfer function which are approximately ten times (\(\sqrt{m}\)) larger when comparing a corrected RdF signal to a Vatell trace. The error bound for the corrected RdF trace increases proportionally. This bound readily captures the deviation of the Vatell signal from the RdF trace over the complete cycle. However, it is sufficient to capture 69% of the Vatell trace within the one standard deviation confidence interval of the RdF trace (for a Gaussian distribution). An empirical reduction of the error bound by a factor of two generates this result, while retaining the error distribution within the cycle.

A.8 Conclusions

Two commercial fast response heat flux sensors have been compared experimentally. They were mounted adjacent to one another in the heated intake manifold of a motored spark ignition engine.

Transfer functions for six individual loads and speeds were built by assuming that the faster of the two probes measures the instantaneous heat flux which the slower probe is subjected to. An expression with one parameter was fitted to these data. This expression was also compared to a simple model and both agreed quite well.

The consistent results between the two probes encourage the use of either of them. One probe was found to be superior in measuring low-amplitude low-frequency heat fluxes. It may be used to measure signals up to about 300 Hz if its response is corrected using the experimentally determined transfer function. Where possible, application of this probe is desirable as it is of low cost and easy to install.
A.9 Nomenclature

\( a \) amplitude of Fourier component; parameter of data fit
\( A \) attenuation of RdF transducer as function of frequency
\( AC \) alternate current, generalized to time-varying component of heat flux signal
\( BDC \) bottom dead center piston position
\( c \) specific heat
\( CA \) crankshaft angular position, degrees
\( DC \) direct current, generalized to mean component of heat flux signal
\( f \) frequency
\( h \) heat transfer coefficient, W/m\(^2\)K
\( k \) wave number; thermal conductivity, W/mK
\( l \) thickness of transducer
\( m \) number of experimental gain/phase lag data pairs for transfer function
\( MAP \) intake manifold absolute pressure
\( \dot{q} \) heat flux, also referred to as heat input
\( \text{rpm} \) crankshaft rotational speed, min\(^{-1}\)
\( R \) output signal of RdF probe
\( t \) typical response time of transducer; time
\( T \) sensor temperature
\( x \) distance from surface into the RdF transducer
\( X \) intermediate variable in model for sensor response time, m
\( \alpha \) thermal diffusivity, m\(^2\)/s
\( \theta \) phase lag of Fourier component
\( \Theta \) phase lag of RdF transducer as function of frequency
\( \rho \) density
\( \sigma \) standard deviation

Prefixes
\( \Delta \) difference; increment of variable in uncertainty analysis

Subscripts
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>expt</td>
<td>experimentally determined data</td>
</tr>
<tr>
<td>external</td>
<td>refers to heat transfer mechanism which drives heat into the transducer</td>
</tr>
<tr>
<td>fit</td>
<td>fit to experimentally determined data</td>
</tr>
<tr>
<td>i</td>
<td>index, gain or phase lag of transfer function</td>
</tr>
<tr>
<td>n</td>
<td>nth Fourier component</td>
</tr>
<tr>
<td>RdF</td>
<td>pertaining to slower probe</td>
</tr>
<tr>
<td>Vatell</td>
<td>pertaining to faster probe</td>
</tr>
</tbody>
</table>
A.10 References


A.11 Figures

Figure A-1: Transfer function (gain and phase) of the RdF probe as function of frequency: fitted expression to data, subtraction of data from the fit (zero in the ideal case), the standard deviation of the data, and semi-infinite body model.

Figure A-2: 1000 rpm / 0.5 bar MAP, comparison of Vatell signal (fast gage) to response corrected signal of the RdF probe using either the fitted or the model transfer function. Also shown is the RdF probe output signal.
Figure A-3: 4500 rpm / 1 bar MAP, comparison of Vatell signal (fast gage) to response corrected signal of the RdF probe using either the fitted or the model transfer function. Also shown is the RdF probe output signal.

Figure A-4: 2750 rpm / 1 bar MAP, heat flux into RdF probe and its uncertainty, both calculated employing experimental transfer function. Also shown is the Vatell signal (fast probe).
Appendix B: 1-D intake port model

A 1-D model of the intake port and manifold was developed to assess the impact of intake heat transfer on gas temperature in the intake port. Of particular interest is the model prediction very close to the intake valve. During valve overlap backflow, spatial gas temperature gradients at that location are very large. Thus it is difficult to relate local measurements to an internal energy averaged temperature over the cross section. The model prediction of gas temperature allows calculation of heat transfer coefficients during the valve overlap backflow phase.

The model solves the mass and energy balance in the intake port, employing experimental heat fluxes as boundary conditions. The mass and enthalpy fluxes at the valve gap are input from a previous calculation of a cycle simulation code. Intake pressure is assumed to be constant during the intake process, which reduces the number of model equation. This assumption has only limited impacts. Heat transfer phenomena due to pressure waves are not captured. However, those heat fluxes cancel over the period of a pressure wave, which is a fraction of the cycle, see Chapter 3. Similarly, velocities are actually determined by wave dynamics and need finite time to adjust to flow rate and pressure changes at the valve gap. However, this time is short for low engine speeds and measured velocities (Chapter 3) agree quite well with velocities derived from mass flow predictions of cycle simulation codes.

B.1 Model equations

Assuming constant pressure, the momentum equation vanishes. The energy equation becomes

\[
\frac{\partial}{\partial \xi} \left( \rho A c_v T \right) = - \frac{\partial}{\partial \xi} \left( \rho A u c_p T \right) + \dot{q}'' P + \frac{\partial}{\partial \xi} \left( A k \frac{\partial T}{\partial \xi} \right)
\]  

(1)

where \( A \) and \( P \) are the pipe cross sectional area and perimeter, respectively. \( k \) is the thermal conductivity or its turbulent equivalent \( k_t \). Using the ideal gas law and the constant pressure assumption, Eq. (1) is rewritten as:

\[
\frac{\partial u}{\partial \xi} = \frac{\dot{q}'' P R}{A p c_p} + \frac{R}{p c_p} \frac{\partial}{\partial \xi} \left( k \frac{\partial T}{\partial \xi} \right),
\]  

(2)

where \( R \) is the specific gas constant. The intake manifold runner and port cross sections are approximately constant. Their axial derivatives are therefore neglected in the continuity equation:

\[
\frac{\partial p}{\partial \xi} + \frac{\partial pu}{\partial \xi} = 0,
\]  

(3)

which is rewritten using the above assumptions:

\[
\frac{\partial}{\partial \xi} T = -u \frac{\partial T}{\partial \xi} + T \frac{\partial u}{\partial \xi}.
\]  

(4)
B.2 Solution procedure

Equations 2 and 4 are discretized using second order accurate finite differences. It is seen that the velocity time derivative vanishes under the assumptions made. Instead, its value is imposed through the boundary condition since constant pressure implies infinite propagation speed of characteristics. The equation without time derivative (energy equation) becomes a constraint requiring modifications to available solution methods.

Temperature at time level $n+1$ is determined from Eq. 4 with Eq. 2 replacing the spatial velocity gradient on the right hand side. The method of lines [1] is employed by first discretizing space. Operator splitting is then applied for the time integration [1]. Time derivative in either case is the second order accurate central difference

$$
\frac{T_{j}^{n+1} - T_{j}^{n-1}}{2\Delta t},
$$

where $n$ and $j$ denote the indexes for time and space, respectively, and $\Delta t$ is the time step. The procedure to evaluate the right hand side at time level $n-1$, $n$, or $n+1$ characterizes the total scheme. The convective term

$$
u_j^n \frac{T_{j+1}^{n} - T_{j-1}^{n}}{2\Delta x},
$$

($\Delta x$ is the space increment) and the forcing function $\dot{q}''$ are evaluated at time level $n$ making this part of the scheme explicit (leap-frog scheme [1]). The diffusive part is evaluated at time levels $n-1$ and $n+1$, which is the Crank-Nickolson scheme [1]:

$$
\frac{1}{2} \frac{k_j^n}{\Delta x^2} T_{j+1/2}^{n+1} - 2k_j^n T_j^n + k_j^n T_{j+1/2}^{n-1} + \frac{1}{2} \frac{k_j^{n+1}}{\Delta x^2} T_{j+1/2}^{n+1} - 2k_j^{n+1} T_{j+1/2}^{n+1} + k_j^{n+1} T_{j+1/2}^{n+1}.
$$

$k^n$ is calculated at time $n$ which only introduces a higher order error (proof: Taylor series expansion). Then, the Crank-Nickolson discretization yields an implicit but linear system of finite difference equations which for 1-D can be solved by a tri-diagonal solver. Without operator splitting the system had been either very unstable (explicit discretization of diffusion) or non-linear and thus computationally more expensive to solve (implicit discretization of convection). $k_{j+1/2}$ is evaluated as the arithmetic mean of $k$ at the nodes $j$ and $j+1$, which yields the desired second order accuracy (proof: Taylor series expansion). In the valve gap, axial turbulent mixing is negligible compared to convection (i.e. no axial conduction into or out of the cylinder). $k_r$ one the last grid point is therefore set to zero and careful discretization of the conductive flux at the boundary is necessary to prevent unphysical heat loss out of the domain.

After solving the finite difference system in time, temperature at time level $n+1$ is available. Velocity at $n+1$ is calculated starting at the valve gap with the value from the cycle simulation code. Values upstream of the valve are calculated by backward stepping into the field using the discretized form of the velocity gradient Eq. 2 (evaluated with $n+1$ values). At the boundaries, first order accurate forward and backward euler discretizations are used, which do not destroy the total second order accuracy of the scheme.
To simulate the turbulent jet mixing during the valve overlap backflow period, axial diffusivity $k_r$ is increased by a factor of 1000 over the laminar value. This value is found through parameter variation to achieve penetration depth of the backflow consistent with experiment. For the remainder of the cycle, the prediction is not sensitive to the value of the enhancement factor, which is chosen to be 150 times the laminar conductivity to represent the impact of turbulent mixing.

B.3 References

B.4 Source code of 1-D model

```
&geometry
    din    = .043
    xmin   = -.15
    xmax   = 0.

&opcond
    pin    = .5e+5
    rpm    = 1500.
    tfresh = 300.
    tinitialize = 300.
    cstrace = 'esim\c0507_4.tx2'
    qtrace = 'q0128_22.txt'
    qnotzero = .true.
    restart = .true.
    restarttrace = 'restart.4'

&grid
    jmax    = 256
    courant = .5
    umax1500 = 50.

&dxgradients
    lvport  = .005
    lback   = .004
    xqtrans = .004
    f_kax_volap = 1000
    f_kax_fflow = 150
    f_kax Disp = 150
    f_kax stag = 20

&output
    methodstartend = 2
    startcalc       = 525.
    endcalc         = 1360.
    methodnout8     = 2
    rincrementnout8 = 2
    jout8           = 8
    methodnout10    = 2
    rincrementnout10= 1
    fort8           = 'c:\users\wolf\fort.8'
    fort9           = 'c:\users\wolf\fort.9'
    fort10          = 'fort20.4'
```

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real mu, k, mdot_bc

dimension & qloc(nqmin:nqmax), aqdry(nqmin:nqmax)
dimension & jperi(nqmin:nqmax,1:2)
dimension & peri(nqmin:nqmax,0:jdim), twall(nqmin:nqmax,0:jdim)
dimension & t_bc(0:720), mdot_bc(0:720), rejs_bc(0:720), vliftod_bc(0:720)
dimension & qm(0:720,nqmin:nqmax), q(nqmin:nqmax)
dimension & x(0:jdim)
dimension & t(-1:1,0:jdim), u(-1:1,0:jdim)
dimension xsmooth(1:jmaxsmooth), ysmooth(1:jmaxsmooth)

common /c1/ pi
common /c2/ rgas,mu,k,pr
common /c3/ pin, tfresh
common /c4/ din, qloc, aqdry, jperi
common /c5/ peri, twall
common /c6/ t_bc, mdot_bc, vliftod_bc, rejs_bc
common /c7/ qm, q
common /c8/ dtau, dx, taumax, tautcycle, xmin, xmax, jmax
common /c9/ x, t, u
common /c10/ xsmooth, ysmooth

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open(30, file=' ', status='old')

! hydraulic intake pipe diameter
! usually inflow (left) boundary
! usually outflow (right) boundary
read(30, geometry)

pin = .5e+5; ! p intake [bar]
rpm = 1500.; ! rpm
tfresh = 300.; ! inflow temperature [K]
tinitial = 300. ! initial temperature in port [K]
cstrace = 'esimt0507_4.txt2' ! cyclesim trace (mdot and Tcyl)
qtrace = 'g0128_22.txt' ! experimental heat flux trace
qnotzero = .true. ! .true. uses experimental q as wall heat flux
restart = .true. ! .true. reads restarttrace to initiate temperature
restarttrace = 'restart.4' ! trace with temperature values for all nodes (if restart)
read(30, opcond)

jmax = 2**8 ! is no. of grid points - 1
courant = .5 ! target max courant number
umax1500 = 50 ! estimated peak velocity for 1500 rpm
read(30, grid)
f_kax_volap = 1000! valve overlap enhancement factor for axial diffusion
f_kax_flow = 150 ! forward flow enhancement factor for axial diffusion
f_kax_disap = 150 ! displacement backflow enhancement factor for axial diffusion
f_kax_stag = 20 ! stagnant phase enhancement factor for axial diffusion
lvport = .008 ! distance to distribute valve in port
lbback = .004 ! distance to feed backflow into port
xqtrans = 7e-3 ! axial smoothing distance for heat fluxes
read(30, dxgradients)

c realtime, CA, or steps in time used to determine start and end of calc?

methodstartend = 2 ! time | CA | timesteps

! 1 | 2 | 3

startcalc = 650. ! start of calculation
endcalc = 1540. ! end of calculation

c
methodnout8 = 3 ! time | CA | timesteps

! 1 | 2 | 3

rcincrementnout8 = 20

c
jout8 = 4 ! output to file 8 every jout8 points, jmax should be

! integer multiple, jout8 will be changed otherwise

methodnout10 = 2 ! time | CA | timesteps

! 1 | 2 | 3

rcincrementnout10 = 1

fort8 = 'c:users\wolf\fort.8' ! output T, coarse matrix time and space
fort9 = 'c:users\wolf\fort.9' ! output T, dense in time, sparse in space
fort10 = 'fort20.4' ! output T, at locations of heat flux probes
read(30, output)
close(30)


c... Spatial discretization

if (jmax .gt. jdim) then
   write (*,*) 'increase jdim to: ', jmax + 1
stop
endif

xtotai = xmax - xmin
dx = xtotal / real(jmax)
jmax = int(lbback/dx)

write(*,*) 'No. of nodes in x (jmax+1): ', jmax + 1
write(*,*) 'No. of nodes for backflow (jmax+1): ', jmax+1


c... Target maximal courant number and determine time-stepping

tau_cycle = 120./rpm
umod = umax1500*rpm/1500.
dtau0dx = courant / umod
dtau = dtau0dx * dx
npertypecycle = int(taucycle/dtau)
dtau = taucycle/real(npertypecycle)

if (methodstartend .eq. 1) then
  taumin = startcalc
  taumax = endcalc
elseif (methodstartend .eq. 2) then
  taumin = startcalc / 720. * taucycle
  taumax = endcalc / 720. * taucycle
elseif (methodstartend .eq. 3) then
  taumin = startcalc * dtau
  taumax = endcalc * dtau
else
  write(*,*) 'STOP: Wrong input for methodstartend'
  stop
endif

nmax = int((taumax-taumin) / dtau)

write(*,*)
write(*,*) 'Time steps per CA: ', real(npertypecycle)/720.
write(*,*) 'CA start: ', taumin / taucycle*720.
write(*,*) 'CA stop: ', taumax / taucycle*720.

c... calculate how often to output to file 8, f(x,t)
if (methodnout8 .eq. 1) then
  nout8max = int(rincrenentnout8/dtau)
elseif (methodnout8 .eq. 2) then
  nout8max = int(rincrenentnout8/720.*taucycle/dtau)
elseif (methodnout8 .eq. 3) then
  nout8max = int(rincrenentnout8)
else
  write(*,*) 'STOP: Wrong input for methodnout8'
  stop
endif

c... check space increemnt to write to file 8
if (jout8 .lt. 1)
  jout8 = 1
endif
jout8max = floor(jmax/jout8+1.e-6)*jout8

c... calculate how often to output to file 10, f(t, qloc)
if (methodnout10 .eq. 1) then
  nout10max = int(rincrenentnout10/dtau)
elseif (methodnout8 .eq. 2) then
  nout10max = int(rincrenentnout10/720.*taucycle/dtau)
elseif (methodnout8 .eq. 3) then
  nout10max = int(rincrenentnout10)
else
  write(*,*) 'STOP: Wrong input for methodnout10'
  stop
endif

c... call subroutine which reads in cycle sim data for boundary condition
to cyclinder
call cycsim(cstrace)

c... call subroutine which reads in heat fluxes
   call qinput(qtrace)
c   qem=0.

c... call subroutine to smoothing function
call init_smooth : call before init_ht

c... call subroutine which set heat transfer areas and wall temperatures
call init_ht(x_geo, lvport)

c... determine locations of heat flux probes for output to file 10
cjmax = nint(devider/dx)
do nq = nqm,nqmax
  jperi10(nq) = jmax - nint(qloc(nq)/dx)
enddo

c... output some physical and computational parameters to file
open(4, file='out1d.m', status='unknown')
  write(4,*)'rpm = ', rpm, ',;'
  write(4,*)'pin = ', pin, ',;'
  write(4,*)'taumin = ', taumin, ',;'
  write(4,*)'taumax = ', taumax, ',;'
  write(4,*)'deltai = ', deltai, ',;'
  write(4,*)'npercyle = ', npercyle, ',;'
  write(4,*)'xmin = ', xmin, ',;'
  write(4,*)'xmax = ', xmax, ',;'
  write(4,*)'lvport = ', lvport, ',;'
  write(4,*)'lback = ', lback, ',;'
  write(4,*)'dx = ', dx, ',;'
  write(4,*)'jmax = ', jmax, ',;'
  write(4,*)'noutmax = ', noutmax, ',;'
  write(4,*)'jout8 = ', jout8, ',;'
  write(4,*)'joutmax = ', joutmax, ',;'
  write(4,*)'f_kax_volap= ', f_kax_volap, ',;'
  write(4,*)'f_kax_flow= ', f_kax_flow, ',;'
  write(4,*)'f_kax_disp= ', f_kax_disp, ',;'
  write(4,*)'f_kax_stag= ', f_kax_stag, ',;'
close(4)

c... open output files
open (8, file = fort8, status = 'unknown')
open (9, file = fort9, status = 'unknown')
open(10, file = fort10, status = 'unknown')

c... set temperature to initial distribution
cnote: taumin will be correspond to n=1 upon entering big loop in tau
do j = 0, jmax
  x(j) = xi(j,xmin,dx)
  t(-1,j) = tinitial ! fn_t0(x)
  t(0,j) = tinitial ! fn_t0(x)
  u(-1,j) = 0
  u(0,j) = 0
  hpt(-1,j) = 0
  hpt(0,j) = 0
  mdotp(-1,j) = 0
  mdotpc(0,j) = 0
  kjmp5(-1,j) = 0
  kjmp5(0,j) = 0
endo
tcyl(-1) = 0
tcyl(0) = 0
ujmaxp1(-1) = u(-1,jmax)
ujmaxp1(0) = u(0,jmax)
tjmaxp1(-1) = t(-1,jmax)
tjmaxp1(0) = t(0,jmax)

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tau = taumin
cap = fn_cap(tau)  ! periodic real ca

do nq = nqmin, nqmax
   q(nq) = 0.
enddo

if (restart) then
   open(80, file = restarttrace, status = 'old')
   do j = 0, jmax
      read(80, *) tmp
      t(-1,j) = tmp
      t(0,j) = tmp
   enddo
   close(80)
endif

c... output u(x), T(x), hpt(x) at taumin
   do j = 0, jout8max, jout8
      write(8,900) x(j), u(-1,j), t(-1,j), hpt(-1,j)
   enddo
   nout8 = 0

c... output in- and outflow boundary at taumin
   write(9,900) tau, u(-1,0), t(-1,0), u(-1,jmax), t(-1,jmax),
            u(-1,jmax-jbmax), t(-1,jmax-jbmax)

c... output in- and outflow boundary at tau(n)
   write(10,900) cap, t(1,jperi10(nq)), nq=nqmin, nqmax
   nout10 = 0

   onedox = 1./dx
   onedox2 = onedox/dx

   tau = tau + dtau

c  ******************* propagate solution in time *******************
c  ********** this loop calculates u(n+1,j) and t(n+1,j) **********

do n = 1, nmax - 1

tau = tau + dtau  ! it is tau(n+1)
cap = fn_cap(tau)  ! periodic real ca

if (qnotzero) then
   do nq = nqmin, nqmax
      q(nq) = cainterpl(qm(0:720,nq), cap)
   enddo
endif

-c... calculate t(n+1,j)

c *** build up matices for tridiag solve of CN

c_q = rgas / pin / cp / area(dum)

c... coefficients for diffusion
rkjmp5 = onedox2 * rgas / pin / cp * k
mdotvg = cainterpl(mdot_bc, cap)

if (mdotvg.lt.-1e-5) then
   if ((cap.gt.500.).and.(cap.lt.650.)) then
      f_kax = f_kax_volap
else
  f_kax = f_kax_disp
endif
else
  if ((cap.gt.540.) .and. (cap.lt.720.)) then
    f_kax = f_kax_fflow
  else
    f_kax = f_kax_stag
  endif
endif
rkjmp5 = rkjmp5 * f_kax

kjmp5(1,1) = 0.
do j = 2, jmax-1
  kjmp5(1,j) = rkjmp5
endoj
kjmp5(1,jmax) = rkjmp5

!... build right hand side: f
if (u(0,0) .gt. le-5) then
  f(0) = tfresh
else
  ! fishy: diffusion not coded on outmost elements
  j = 0
  f(j) = t(0,j) + dtau * ( & + t(0,j) * c_q_*(hpt(0,j) + mdotpc(0,j)*(tcyl(0)-t(0,j))) & - oneodx * u(0,j) * (t(0,j+1) - t(0,j))
) endif

do j = 1, jmax - 1
  f(j) = t(-1,j) + dtau * ( & + t(0,j) * ( kjmp5(-1,j+1) * t(-1,j+1) & - (kjmp5(-1,j+1)-kjmp5(-1,j)) * t(-1,j) & + kjmp5(-1,j) * t(-1,j-1) & + 2.*c_q_*(hpt(0,j) + mdotpc(0,j)*(tcyl(0)-t(0,j))) ) & - oneodx * u(0,j) * (t(0,j+1) - t(0,j-1))
) enddo

f(j) = t(0,j) + dtau * ( & + t(0,j) * ( kjmp5(-1,j) * t(-1,j) * dx / 2. & + kjmp5(-1,j) * t(-1,j-1) * dx / 2. & + c_q_*(hpt(0,j) + mdotpc(0,j)*(tcyl(0)-t(0,j))) ) & - oneodx * u(0,j) * (t(0,j) - t(0,j-1))
)
c
j = jmax
c
f(j) = t(-1,j) + dtau * ( & + t(0,j) * ( kjmp5(-1,j) * t(-1,j) * dx & + kjmp5(-1,j) * t(-1,j-1) * dx & + 2.*c_q_*(2.*hpt(0,j) + mdotpc(0,j)*(2.*tcyl(0)-t(0,j))) ) & - oneodx * (ujmaxpl(0)*jmaxpl(0)-u(0,j-1)*t(0,j-1))
)
c... fill triidiag matrix: a, b, c
b(0) = 1.
c(0) = 0.
do j = 1, jmax - 1
  a(j) = - t(0,j) * dtau * kjmp5(1,j)
  b(j) = 1. + t(0,j) * dtau * (kjmp5(1,j)+kjmp5(1,j+1))
  c(j) = - t(0,j) * dtau * kjmp5(1,j+1)
endoj
j = jmax
a(j) = - t(0,j) * dtau * kjmp5(1,j) * dx / 2.
b(j) = 1. + t(0,j) * dtau * kjmp5(1,j) * dx / 2.
c
a(j) = - t(0,j) * dtau * kjmp5(1,j) * dx
b(j) = 1. + t(0,j) * dtau * kjmp5(1,j) * dx
c... LU decomposition
    up(0) = b(0)
    do j = 1, jmax
        low(j) = a(j) / up(j-1)
        up(j) = b(j) - low(j) * c(j-1)
    enddo

c... forward solve
    y(0) = f(0)
    do j = 1, jmax
        y(j) = f(j) - low(j) * y(j-1)
    enddo

c... backward solve
    t(1,jmax) = y(jmax) / up(jmax)
    do j = jmax-1, 0, -1
        t(1,j) = (y(j) - c(j) * t(1,j+1)) / up(j)
    enddo

c
    c *** calculate u(n+1,j)
    hpt(1,jmax) = fn_hpt(1,jmax)

c... distribute backflow using smoothing function
    if (mdotvg .gt. 1e-5) then
        tcy1(1) = 0.
        rmdotpc = 0.
        uotjmaxpl = mdotvg*rgas/pin/area(xmax)
    else
        tcy1(1) = cainterpl(t_bc,cap)
        rmdotpc = - cp * mdotvg / dx
        uotjmaxpl = 0.
    endif

    uotjmaxpl = max(1e-6, mdotvg*rgas/pin/area(xmax))
    sum_ds = 0
    do j = 0,jbmax
        ds = sinterpl(real(j)/real(jbmax))
        sum_ds = sum_ds + ds
        mdotpc(1,j-jbmax+jmax) = rmdotpc * ds
    enddo
    do j = 0,jbmax
        mdotpc(1,j-jbmax+jmax) = mdotpc(1,j-jbmax+jmax) / sum_ds
    enddo

    j = jmax
    dudxjmax =
    & c_q * ( hpt(1,j) + mdotpc(1,j)*tcyl(1))
    & + kjmp5(1,j) * dx * t(1,j)
    & + kjmp5(1,j) * dx * t(1,j-1)
    j = jmax - 1
    dudxjmaxml =
    & c_q * ( hpt(1,j) + mdotpc(1,j)*tcyl(1))
    & + kjmp5(1,jpl) * t(1,jpl)
    & - (kjmp5(1,jpl)+kjmp5(1,j)) * t(1,j)
    & + kjmp5(1,j) * t(1,jml)

c
    a1 = -u(1,j)/uotjmaxpl
    a2 = (-2.*dx*t(1,j)+dudxj+(u(1,j)-u(1,j-1))*t(1,j-1))/uotjmaxpl
    tmaxpl(1) = -a1/2. + sqrt(a1**2/4. - a2)
    ujmaxpl(1) = uotjmaxpl * tmaxpl(1)

    tmaxpl(1) = t(1,jmax)
    ujmaxpl(1) = uotjmaxpl * t(1,jmax)

129
u(l,jmax) = ujmaxpl(l)
u(l,jmax-1) = ujmaxpl(l) - dx * dudxjmax

c unstable u(l,jmax) = ujmaxpl(l) - .5 * dx * dudxjmax
c unstable u(l,jmax-1) = ujmaxpl(l) - dx * dudxjmax - .5 * dx * dudxjmaxml

do j = jmax - 1, 1, -1
   jp1 = j+1
   jml = j-1
   hpt(1,j) = fn_hpt(1,j)
   u(l,jml) = u(1,jp1) - 2. * dx * (c_q * (hpt(1,j) + mdotpc(1,j) * tcyl(1))
   & + kjmp5(1,jp1) * t(1,jp1)
   & - (kjmp5(1,jp1)+kjmp5(1,j)) * t(1,j)
   & + kjmp5(1,j) * t(1,jml))
endo

hpt(1,0) = fn_hpt(1,0)

c... output of u(x), T(x)
nout8 = nout8 + 1

if (nout8 .eq. nout8max) then
   do j = 0, jout8max, jout8
      write(8,900) x(j), u(1,j), t(1,j), hpt(1,j)
   enddo
   nout8 = 0
endif

c... output in- and outflow boundary at tau(n)
write(9,900) tau,u(1,0),t(1,0),u(1,jmax),t(1,jmax),
      & u(1,jmax-jbmax),t(1,jmax-jbmax)

c... output in- and outflow boundary at tau(n)
nout10 = nout10 + 1
if (nout10 .eq. nout10max) then
   write(10,900) cap,(t(1,jperi10(ng)), ng=ngmin,ngmax)
   nout10 = 0
endif

c... reshuffle arrays for next time step
do j = 0, jmax
   t(-1,j) = t(0,j)
   t(0,j) = t(1,j)
   u(-1,j) = u(0,j)
   u(0,j) = u(1,j)
   hpt(-1,j) = hpt(0,j)
   hpt(0,j) = hpt(1,j)
   mdotpc(-1,j) = mdotpc(0,j)
   mdotpc(0,j) = mdotpc(1,j)
   kjmp5(-1,j) = kjmp5(0,j)
   kjmp5(0,j) = kjmp5(1,j)
endo
tcyl(-1) = tcyl(0)
tcyl(0) = tcyl(1)
ujmaxpl(-1) = ujmaxpl(0)
ujmaxpl(0) = ujmaxpl(1)
tjmaxpl(-1) = tjmaxpl(0)
tjmaxpl(0) = tjmaxpl(1)

enddo
900  format(10e14.5)
    end i of main program
    ccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccc
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ccc
i0 = floor(xi);
if (abs(xi-real(i0)) .lt. 5.*eps) then
  sinterp1 = ysmooth(i0);
else
  sinterp1 = ysmooth(i0) + (ysmooth(ii)-ysmooth(i0)) &
            *(xi-real(i0)) ! / 1
endif
return
end

function xi(j,xmin,dx)
c    spatial position of grid points in [m]
x = float(j) * dx + xmin
return
cend

function area(dumx)
c    cross section of pipe (flow area)
include 'idi.fi'
dum = dumx
area = pi * din**2 / 4.
return
cend

function fn_cap (tau)
c    converts time to ca in [0 720]
include 'idi.fi'
pos = tau/taucycle
fn_cap = (pos - floor(pos)) * 720
return
cend

function cinterp1 (trace, cap)
c    interpolates 0:720 array to time of calculation
dimension trace(0:720)
ical = ceiling(cap)
ica0 = floor(cap)
if (abs(cap-real(ica0)) .lt. 1e-5) then
  cinterp1 = trace(ical)
else
  cinterp1 = trace(ica0) + (trace(ical) - trace(ica0)) &
            *(cap - real(ica0)) ! / 1.
endif
return
cend

function fn_t0 (dumx)
include 'idi.fi'
dum = dumx
fn_t0 = tfresh
return
cend

function cycsim(cstress)
include 'idi.fi'
character*30 cstress
open (20,file=cstress,status='old')
c... read data
ica = 0 ! cyscim2.m interpolates to [1,720]
10  ica = ica + 1 ! ica = [1 .. icamax]
read(20,*,end=20) p_dum, t_bc(ica), mdot_bc(ica), vliftod_bc(ica)
goto 10
20  icamax = ica - 1

... build rejet_star (star because it could be modified by a
temperature ratio), see caton source code
c... ctmp do itau = 1, 720
c  ctmp if (mdot_bc(itau) .lt. 0.) then
c  ctmp rejs_bc(itau) = -0.357 * mdot_bc(itau) * din
ctmp &  / area(dum) / mu / vliftod_bc(itau)
ctmp endif
ctmp enddo

t_bc(0) = t_bc(720)
mdot_bc(0) = mdot_bc(720)
vliftod_bc(0) = vliftod_bc(720)
rejs_bc(0) = rejs_bc(720)

open(10, file='fort.10', status='unknown')
do itau = 0, 720
  write(10,700) t_bc(itau), mdot_bc(itau), vliftod_bc(itau),
  &  rejs_bc(itau)
dendo
close(10)

700  format(100e14.5)
close(20)
return
c
end

... c ycsim ccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccc
... c.... qtrace is experimentally measured wall heat flux at valve and
c and port
c
subroutine qinput(qtrace)
include 'idl.fi'
character*30 qtrace
open (20,file=qtrace,status='old')

... c... read data
ica = -1
10  ica = ica + 1 ! ica = [0 .. icamax], first exptl point at BDC = 0
read(20,*,end=20) (qm(ica,nq),nq=nqmin,nqmax)
goto 10
20  icamax = ica - 1
qm = qx*le4
do nq = nqmin,nqmax
   qm(720,nq) = qx(0,nq)
endo
c
open(10, file='fort.10', status='unknown')
c do ica = 0, 720
c    write(10,700) (qm(ica,nq),nq=nqmin,nqmax)
c enddo
c close(10)
c 700  format(100e14.5)
close(20)
return
c
end

... cs init_ht 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include '1d1.f9'
real lport, l1vport
dimension periq(nqmin:nqmax), twallq(nqmin:nqmax)
parameter (jtdim = 100)
dimension trans(0:jtdim)
data twallq /0.,0.,0.,0.,0./
twallq = twallq + 273.

lport = .08 ! length of port
peri = 0
twall = 0

c... jperi(nqmin:nqmax,1:2): 1 farther away from valve
    i.e. j is smaller

c... no. 0 = valve
  jperi(0,1) = jmax - nint(l1vport/dx)
  jperi(0,2) = jmax

c... no. 1 = bottom one
  devider = (qloc(1)+qloc(3))/2.
  jperi(1,1) = jmax - nint(devider/dx)
  jperi(1,2) = jmax

c... no. 2 = top one
  devider = (qloc(2)+qloc(4))/2.
  jperi(2,1) = jmax - nint(devider/dx)
  jperi(2,2) = jmax

c... no. 3 = bottom two
  devider = (qloc(3)+qloc(5))/2.
  jperi(3,1) = jmax - nint(devider/dx)
  jperi(3,2) = jperi(1,1)

c... no. 5 = bottom three
  jperi(5,1) = jmax - nint(l1port/dx)
  jperi(5,2) = jperi(3,1)

jmax = int(nxqtrans/2/dx)*2.+1 ! must be odd number
if (jmax .gt. jtdim ) then
  write(*,*) 'Stop: increase jtdim in init ht'
  stop
endif
do j = 1,jmax
  trans(j) = sinterpl(real(j)/real(jmax))
endo

jhalf = (jmax-1)/2

do nq = nqmin,nqmax
  periq(nq) = agdry(nq) / ((jperi(nq,2)-jperi(nq,1))*dx)
  jt = -1
  do j = jperi(nq,1)-jhalf,jperi(nq,1)+jhalf
    jt = jt + 1
    periq(nq,j) = periq(nq) * trans(jt)
    twall(nq,j) = twallq(nq)
  enddo
  do j = jperi(nq,1)+jhalf+1,jperi(nq,2)
    periq(nq,j) = periq(nq)
    twall(nq,j) = twallq(nq)
  enddo
  if (jperi(nq,2)+jhalf .le. jmax) then
    jt = jmax+1
    do j = jperi(nq,2)-jhalf,jperi(nq,2)+jhalf
      jt = jt - 1
      periq(nq,j) = periq(nq) * trans(jt)
    enddo
  endif
endif
enddo

open(10, file='fort.10', status='unknown')
do j = 0, jmax
  write(10,700) (peri(nq,j), nq=nqmin,nqmax)
enddo
close(10)

700 format(100e14.5)
return
end

cf fn_hpt
function fn_hpt(dumm,j)
  c arguments: n = [-1,1] from time stepping,
  c j = [0..jmax] spatial location,
  include 'id1.fi'
dum = dumm
res = 0
  do nq = nqmin,nqmax
    res = res - peri(nq,j) * q(nq)
  enddo
fn_hpt = res
return
end
THEESIS PROCESSING SLIP

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YEAR: 1997 DEGREE: Ph.D.

NAME: BAUER, Wolf-Dietrich