

High Resolution Scalar Dissipation and Turbulence Length Scale Measurements in an Internal Combustion Engine

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ABSTRACT

High resolution planar laser-induced fluorescence (PLIF) measurements were performed in an optically accessible internal combustion (IC) engine to investigate the behavior of scalar dissipation and the fine-scale structures of the turbulent scalar field. The fluorescent tracer fluorobenzene was doped into one of the two intake streams and nitrogen was used as the carrier gas to permit high signal-to-noise ratio fluorescence measurements without oxygen quenching effects. The resulting two-dimensional images allowed for an analysis of the structural detail of the scalar and scalar dissipation fields defined by the mixing of the two adjacent intake streams. High levels of scalar dissipation were found to be located within convoluted, sheet-like structures in accordance with previous studies. The fluorescence data, which were acquired during the intake stroke, were also used to examine the scalar energy and dissipation spectra. The measured spectra were found to closely match model turbulent energy and dissipation spectra over the full range of wavenumbers for all conditions. The Batchelor length scale was directly measured using the point where the dissipation spectrum fell to 2% of its peak value and was found to range from 29 to 34 μm . The integral length scale and Taylor microscale, were determined from the spatial autocorrelation of the scalar field, and were found to range from 3.03 to 7.95 mm and from 0.54 to 1.30 mm, respectively. In general, the turbulence length scales were slightly different in the two dimensions captured in the fluorescence images, suggesting the turbulent scalar field is slightly anisotropic during the intake stroke. The length scales were found to vary only

slightly with intake manifold pressure (60 - 109 kPa) and engine speed (600 - 1200 RPM).

INTRODUCTION

Engine manufacturers have long concentrated on reducing harmful emissions such as soot and NO_x while maintaining high fuel efficiency to limit operating costs and carbon dioxide (CO_2) output. Combustion strategies collectively referred to as low temperature combustion (LTC) have been developed to limit the formation of soot and NO_x by decreasing the temperatures and local equivalence ratios achieved within the combustion chamber. These strategies rely heavily on controlling the mixing process between the fuel and the in-cylinder mixture prior to the combustion event. The pre-combustion mixture preparation affects combustion parameters such as temperature distributions, local equivalence ratios, reaction rates, and ultimately which product species are formed. Inherently, turbulence plays a significant role in the mixing process and an increased understanding of the turbulence properties of in-cylinder flows is important to the advancement of LTC in internal combustion (IC) engines.

In this study, the turbulent scalar field is investigated to exploit the high spatial resolution capabilities of scalar measurement techniques. Planar laser-induced fluorescence (PLIF), which can achieve higher spatial resolution than traditional velocity measurement methods, was utilized to resolve details of in-cylinder turbulence down to the dissipative length scales. The scalar field, made random by the turbulent nature of the velocity field, can be described

using many of the concepts developed for the turbulent velocity field. These concepts include the ideas of an energy cascade and the hypotheses of Kolmogorov's universal equilibrium theory. By defining a turbulent pseudo-scalar energy as $(\xi')^2/2$, where ξ is a scalar variable such as concentration and ξ' is the fluctuating component of the scalar variable, a relation for the dynamics of the turbulent scalar energy can be developed analogous to the relation for the turbulent kinetic energy of the velocity field [1]. An important parameter in the relation for the turbulent scalar energy has been identified as the turbulent scalar dissipation, χ , and is defined here as:

$$\chi = 2\Gamma \nabla \xi' \cdot \nabla \xi'(\mathbf{x}, t) \quad (1)$$

where Γ is the mass diffusivity. The scalar dissipation acts to decrease gradients in scalar energy by the action of diffusional processes and, therefore, can be thought of as a molecular mixing rate. Molecular mixing rates are important in engine applications where the combustion is mixing-rate limited (e.g. LTC strategies incorporating mixture stratifications to modify combustion behavior). Engine combustion Large Eddy Simulations (LES) also rely on scalar dissipation models to properly describe in-cylinder mixing and, therefore, accurate fine-scale measurements of χ are important in the advancement of such models [2].

Previous investigators have made two-dimensional (2D), high resolution scalar dissipation measurements in turbulent jets [3,4,5,6,7,8,9,10,11] and jet flames [5,12,13,14,15,16] and to a lesser extent in IC engines [17-18]. The scalar dissipation in all cases was found to be located within convoluted, sheet-like structures with the probability distribution of the one-dimensional (1D) scalar dissipation approaching a log-normal shape. A number of studies have also investigated the effects that resolution [15,17,18,19,20,21], noise [12,14,20,22], and data processing techniques [16,22] have on the accuracy of scalar dissipation measurements. These studies have concluded that accurate measurements of χ require data processing techniques with little-to-no inherent filtering and high spread-noise factors, κ_{noise} , defined as:

$$\kappa_{noise} = \frac{\Delta \xi}{\delta \xi} \quad (2)$$

where $\Delta \xi$ is the difference between the highest and lowest scalar values and $\delta \xi$ represents the noise level. Authors [19,23] have found that experimental resolutions approaching the Batchelor scale were found to be a good compromise to avoid under-sampling the scalar dissipation field and to also avoid over-sampling, which makes the measurements more susceptible to noise. The scaling relation for the Batchelor length scale, λ_B , is given in Eq. (3):

$$\lambda_B = \eta Sc^{-1/2} = \Lambda_B \delta Re^{-3/4} Sc^{-1/2} \quad (3)$$

where η is the Kolmogorov length scale, Re is the characteristic Reynolds number, Sc is the Schmidt number ($Sc \equiv \nu/\Gamma$ with kinematic viscosity ν), δ is the integral length scale, and Λ_B is a constant. A more detailed discussion on resolution requirements of scalar dissipation measurements can be found in refs [22, 25].

The scaling relation in Eq. (3) is useful for flows that have well defined integral scales and Reynolds numbers and when the value of Λ_B is relatively well known. For engine flows, all three parameters are unknown and Eq. (3) cannot be used to determine the required resolution for scalar dissipation measurements prior to engine experiments.

Recent measurements by Wang and coworkers in turbulent non-premixed flames have utilized scalar energy and dissipation spectra to estimate the Batchelor length scale [13,14,15,22]. They demonstrated that measured 1D scalar energy and dissipation spectra matched the 1D form of Pope's model energy and dissipation spectra [1], which were developed from turbulent kinetic energy measurements (velocity data). Further evidence supporting the use of a model for turbulent kinetic energy spectra to model scalar spectra has been provided by Antonia et al. in [24]. They analyzed Direct Numerical Simulations (DNS) of a turbulent channel flow and found that the 3D turbulent kinetic energy spectra were similar to the corresponding spectra of the scalar variance, θ ($\theta \equiv (\xi')^2$ = twice the turbulent scalar energy as defined above). These results suggest measurements of scalar energy and dissipation spectra, which can be made at very high resolutions, can be used to infer information about the turbulent velocity field.

In Pope's model dissipation spectrum, the high-wavenumber point where the spectrum is 2% of its peak value corresponds to the wavenumber given by $\kappa = 1/\eta$. Wang et al. [14] used the 2% point in their highly resolved scalar dissipation spectra from jet-flame measurements to determine the Batchelor scale and defined their measurements as λ_β in Eq. (4):

$$\lambda_\beta = \frac{1}{\kappa_{2\%}} \quad (4)$$

where $\kappa_{2\%}$ is the wavenumber at which the scalar dissipation spectrum is 2% of its peak value. This *a posteriori* method of spectrally estimating the Batchelor scale is useful in the design of high resolution scalar measurements, especially in flows where the parameters of turbulence are not well defined. Note that the physical length scale corresponding to $\kappa_{2\%}$ is a factor of 2π larger than λ_β . This length scale has

previously been termed the cutoff length scale, $\lambda_c = 2\pi\lambda_\beta$, in [16].

In this study, high resolution tracer-based PLIF measurements were performed in an IC engine to investigate the fine-scale properties of the scalar field and the statistical distribution of the scalar dissipation over a range of conditions. The results build upon previous 1D LIF measurements made in the same engine [25]. This previous work demonstrated the ability to resolve the 2% level in the measured scalar dissipation spectrum in an IC engine during the intake stroke, exclusively. The corresponding Batchelor scale measurements, λ_β , varied from 20 to 40 μm for engine speeds ranging from 300 to 1800 RPM and intake pressures ranging from 45 to 109 kPa. High number densities of the tracer, fluorobenzene, and high laser intensities were required to achieve sufficient signal levels causing significant laser energy attenuation and saturation of the fluorescence, respectively. The measured scalar dissipation distributions were found to have a log-normal shape and did not follow any significant trends with respect to two Reynolds numbers describing the turbulence of different in-cylinder flows.

The current work expands on the previous study [25] by acquiring measurements in two dimensions, allowing for an analysis of the structures within the scalar and scalar dissipation fields. High resolution fluorescence images were acquired at a single crank angle (240° bTDC) within the intake stroke at several engine speed - intake pressure combinations. Two-dimensional scalar dissipation fields were calculated from the fluorescence data and revealed a significant amount of detail at the smallest length scales of the in-cylinder turbulence. One-dimensional scalar energy and dissipation spectra were also generated from the data in each dimension captured in the measurements. In addition to measuring the Batchelor scale, the integral and Taylor micro-length scales were determined for each direction as well. This allowed for an investigation into the isotropy of the in-cylinder turbulence by comparing the one-dimensional scalar energy and dissipation spectra and turbulence length scales from each dimension. The result is a detailed description of the scalar dissipation and turbulence length scales within the turbulent engine flow field.

EXPERIMENTS

ENGINE

The two-dimensional fluorescence experiments were performed in a motored optical engine used previously for in-cylinder mixing studies [26,27]. The main geometric specifications and valve timings are given in Table 1 and a cross section of the piston-combustion chamber portion of the engine is provided in Fig. 1. Optical access to the pent-roof combustion chamber was achieved through a sapphire window ($d = 58.3 \text{ mm}$) in the top surface of the Bowditch-

type piston. The combustion chamber was imaged via a 45 degree plane mirror (UV anti-reflection coated) at the bottom of the extended piston. The laser sheet was admitted to the combustion chamber through a UV-grade fused silica (UVFS) ring window which comprises the top 20 mm of the engine cylinder.

Table 1. Main geometric specifications and valve timings of the single-cylinder optical engine.

Engine Geometry	
Bore	86.0 mm
Stroke	94.6 mm
Displ. Vol.	550 cm ³
Comp. Ratio	11
Clear. Vol.	55.23 cm ³
Con. Rod Length	152.5 mm
Valves	
Intake Valves	2
Intake Valve Diam.	35.1 mm
IVO	360° aTDC
IVC	540° aTDC
Exhaust Valves	2
Exhaust Valve Diam.	30.0 mm
EVO	131° aTDC
EVC	375° aTDC

The intake manifold of the engine was divided into two flow streams such that each of the two intake valves was fed independently. Each intake valve is connected to its own intake runner and surge tank. Fuel was injected upstream of one of the surge tanks and a valved flow path could be configured to feed each intake stream with either similar or different air/tracer/fuel mixtures. The intake system used six heating tapes to allow the temperature of the intake charge to be controlled. The exhaust system consisted of a single stainless steel runner connected to a mixing diffuser and surge tank.

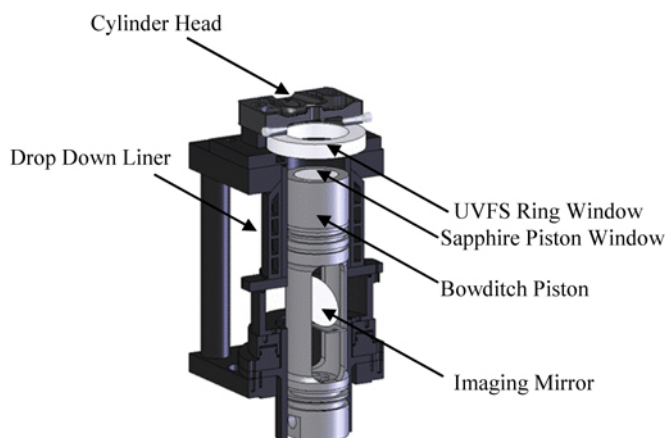


Figure 1. Model of Bowditch piston and pent-roof combustion chamber of the single-cylinder optical research engine.

A mixture of spectroscopic-grade iso-octane and the fluorescence tracer, fluorobenzene, was delivered to the engine using a Mercury Marine Optimax air-assisted fuel injector. The air-assisted injection provides a well atomized spray to aid vaporization and the fuel was injected against the intake flow ~ 1 m upstream of the intake valve to ensure complete mixing of the injected fuel.

OPTICAL EQUIPMENT

The fluorescence was excited by the 266 nm output of an EKSPLA model NL303D Nd:YAG laser. This module has two independent laser heads whose output beams were combined to increase the overall laser energy and effective pulse width. The two laser pulses were separated temporally by 200 ns. After harmonic generation, the two 266 nm beams were aligned and directed toward the engine via four high energy laser mirrors. The beams were first expanded with a 200 mm focal length UVFS cylindrical lens and then focused in the center of the cylinder with a 500 mm focal length UVFS spherical lens. The positioning and the field of view of the fluorescence measurements are shown in Fig. 2.

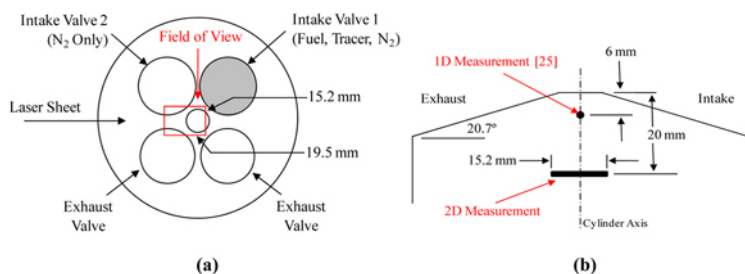


Figure 2. Schematics of fluorescence measurement location and field of view (a) looking up into the combustion chamber and (b) side view of the pent-roof combustion chamber.

The fluorescence was imaged through the 45° piston mirror by an Andor DV437-BU2 CCD camera with a 105 mm f/4.5 UV Nikkor camera lens. The CCD array has $13 \mu\text{m}$ square pixels and was cooled to -40°C for all experiments. Spurious reflections of the incident laser light were filtered out with a WG295 Schott glass filter prior to imaging. Fluorobenzene was chosen as the tracer species for the current experiments based on its superior fluorescence yield compared to common fluorescence tracers such as toluene, 3-pentanone, and triethylamine (TEA) at the conditions used in this study [27]. Nitrogen was used exclusively as the bath gas throughout the operation of the engine to reduce the strong collisional quenching of fluorobenzene fluorescence by oxygen molecules. The nitrogen was supplied to the engine via high-pressure tanks and was metered with a critical flow orifice.

Resolution, Noise, and Data Processing

The effective resolution of LIF measurements is defined by two components of the fluorescence/optical system: (1) the sheet thickness within the region of interest and (2) the resolution of the imaging system (camera and camera lens).

The sheet thickness (beam size) was approximated by assuming the beam propagation factor, M_{sq}^2 , of both laser beams was 3.4, which was measured for a similar laser in our laboratory [28,29]. This results in a theoretical beam waist (full width at $1/e^2$ point of maximum intensity) of approximately $24 \mu\text{m}$ at the focus and $254 \mu\text{m}$ at the extents of the measurement volume.

The resolution of the imaging system was determined by measuring the line spread function (LSF) of the camera and UV camera lens with the scanning knife-edge technique described in [30]. The standard deviation, σ , of the LSF was estimated to be $35.4 \mu\text{m}$ at the magnification of 0.32 used in this study by fitting a Gaussian function to the measured LSF data. The pixel pitch at this magnification is $40 \mu\text{m}$. The field of view of the LIF images was 15.2 mm wide and 19.5 mm in the beam propagation direction as shown in Fig. 2.

The estimates of the beam size are based on laser beam propagation theory and do not incorporate the effects of beam steering introduced by density gradients within the combustion chamber. The true beam width in the measurement volume is, therefore, larger than the above estimates. Similar 1D measurements resulted in a beam width of approximately $400 \mu\text{m}$ [25]. Although the beam width is the largest probe dimension, previous authors [13] have suggested that the resolution in the out-of-plane direction (the laser beam width in this case) is less critical to the overall resolution of one-dimensional gradient measurements.

The entire set of image data was background and flatfield corrected to suppress spurious background signal and to

eliminate striations and other irregularities in the laser sheet excitation. The average background signal, $I_{i,j}^{background}$, was first subtracted from the raw data images, $I_{i,j}^{data}$. The result was divided by the average background-corrected flatfield image, $I_{i,j}^{flatfield}$, according to the image processing algorithm in Eq. (5):

$$Fl_{i,j}^{2D} = \frac{I_{i,j}^{data} - \frac{1}{N_b} \sum_{k=1}^{N_b} I_{i,j,k}^{background}}{\frac{1}{N_{ff}} \sum_{k=1}^{N_{ff}} I_{i,j,k}^{flatfield}} \quad (5)$$

where i and j represent the vertical and horizontal pixel locations, $Fl_{i,j}^{2D}$ is the two-dimensional background and flatfield-corrected fluorescence image, N_b is the number of acquired background images, and N_{ff} is the number of flatfield images taken for each data set. The flatfield data were acquired with a homogenous in-cylinder mixture at the same crank angle as the raw image data. The flow field was made homogeneous by introducing the fluorobenzene tracer to both of the intake manifolds.

Experimental noise also affects the accuracy of scalar dissipation measurements [22]. The effect of the experimental noise, mainly shot-noise incurred during image collection, was minimized in these experiments by selecting a fluorescence tracer (fluorobenzene) with a high fluorescence yield ($\phi \sim 0.1-0.3$), using high laser energies ($E_{p,total} = 10$ mJ, split between two pulses), and high tracer number densities. Data processing methods can also reduce the effective resolution of measurements by introducing unwarranted filtering. Therefore, all presented data were unfiltered and a description of the data processing techniques are included in the Results section. Reference [22] provides a comprehensive review of error sources in scalar dissipation measurements.

ENGINE CONDITIONS

A parametric study was conducted to investigate how different engine flows affect the scalar energy and dissipation spectra and the estimated Batchelor length scale. Due to a scalar field homogeneity constraint determined in previous work [25], data were only collected in the middle of the intake stroke (240° bTDC) when the piston is near its maximum velocity. The flow field at this crank angle is assumed to be a combination of the annular jets issuing from the adjacent intake ports and the flow induced by the motion of the piston. Image data was acquired in a plane between the two intake valves as diagramed in Fig. 2. Multiple Reynolds numbers can be defined to describe different aspects of this complex flow field. The two Reynolds numbers that will be used in the subsequent discussion include: an intake Reynolds

number, Re_{intake} , that is based on the flow through the intake valves, and a piston-induced flow Reynolds number, Re_{engine} , that is based on bulk flow arguments [31]. The motivation for investigating the dependence of the scalar field on these two Reynolds numbers is to determine which flow-type dominates the turbulent mixing within the in-cylinder flow field.

The Reynolds number used here to describe the intake flow uses an integrated approach to deal with the transient nature of the complex flow field. Averaged or cumulative length and velocity scales are used in the definition of the intake Reynolds number in Eq. (6):

$$Re_{intake} = \frac{\rho \langle uh \rangle}{\mu} \quad (6)$$

where the triangular brackets $\langle \rangle$ represent crank angle averaging, ρ is the density of the intake flow, μ is the dynamic viscosity, u is the characteristic velocity, and h is the valve lift. The quantity $\langle uh \rangle$ can be related to the average mass flow rate through the valve during intake, $\dot{m}_{intake}$, as in Eq. (7), by assuming the flow area, A , is characterized by the valve lift, h , and the valve diameter, D_v :

$$\dot{m}_{intake} \cong \frac{m_{cycle}}{\tau_{intake}} = \langle \rho u A \rangle = \rho \langle uh \rangle \pi N_v D_v \quad (7)$$

where m_{cycle} is the mass delivered to the engine per intake stroke, τ_{intake} is the time the intake valve is open, and N_v is the number of intake valves. Substituting Eq. (7) into Eq. (6) results in the averaged intake Reynolds number given in Eq. (8).

$$Re_{intake} = \frac{m_{cycle}}{\pi N_v D_v \tau_{in} \mu} \quad (8)$$

This averaged intake Reynolds number describes the pressure-driven flow through the intake valves and incorporates the effect that the intake manifold pressure has on the turbulent properties of the jet flow.

The other Reynolds number used here describes the flow induced by the motion of the piston. It was originally developed from turbulence measurements near TDC of the compression stroke and is given in Eq. (9) [31]:

$$Re_{engine} = \frac{(0.21\delta_{cl}) \cdot (S_p/2) \rho}{\mu} \quad (9)$$

where S_p is the mean piston speed, δ_{cl} is the instantaneous clearance height, and μ is the dynamic viscosity. This Reynolds number relation may not be applicable throughout the intake stroke (especially the 0.21 scaling factor), however, it does incorporate the physical flow characteristics (mean piston speed and clearance height) of the piston-induced flow motion.

In the proceeding experiments, the dependence of the scalar and scalar dissipation fields on the Reynolds number of the flow through the intake valves, Re_{intake} , and the flow developed by the motion of the piston, Re_{engine} were studied by holding the engine speed constant as Re_{intake} was varied. The engine conditions were similar to those used in [25] and are provided in Table 2. The interdependence of Re_{intake} and Re_{engine} causes both Reynolds numbers to decrease due to the drop in the in-cylinder mixture density as the intake pressure is reduced. However, the effects of varying Re_{intake} and Re_{engine} are investigated over a range of conditions.

<table 2 here>

RESULTS

SAMPLE IMAGES

For each condition, 50 data, background, and flatfield images were acquired sequentially. Sample background- and flatfield-corrected fluorescence and calculated scalar dissipation images acquired at two different intake pressures at engine speeds of 600 and 1200 RPM are shown in Figs. 3 and 4. The average signal properties (signal-to-noise ratio (SNR) and spread-noise factor, κ_{noise}) of each data set are provided in Table 2 above. The spread-noise factor, κ_{noise} , was determined using Eq. (2) and approximating the noise level as the theoretical shot-noise, $\delta I_{shot,ave}$, from the average signal level, I_{ave} , as shown in Eq. (10):

$$\delta I_{shot,ave} = \sqrt{C_{A/D} I_{ave}} \quad (10)$$

where CA/D is the gain of the camera digitizer (counts/e-). The data images show a significant amount of detail in both the scalar and scalar dissipation fields at small length scales. The calculated dissipation fields are consistent with previous measurements in that high levels of χ are isolated to thin sheet-like structures with a large fraction of the flow field containing low- χ regions. Note that the width of the sheet-like structures appear to be very similar for all of the conditions presented in Figs. 3 and 4. The effects of shot-noise can

clearly be seen in the low- χ areas and the horizontal striations observed in the fluorescence data are believed to be a result of imperfections in the flatfield correction due to engine vibration. Note that the fluorescence and dissipation data are not spatially filtered.

<figure 3 here>

<figure 4 here>

SCALAR ENERGY AND DISSIPATION SPECTRA

Scalar energy and dissipation spectra were calculated for each individual horizontal row and vertical column of data in the acquired data images, essentially breaking the images down into a collection of individual 1D measurements. The interlacing method [16] of determining the scalar energy spectra, $E(\kappa)$, was used to suppress the noise contribution in the measured spectra:

$$E(\kappa) = \Im(\xi'_i(x)) \Im^*(\xi'_{i+1}(x)) \quad (11)$$

where $\Im$ represents the Fourier transform operator, $\Im^*$ represents the complex conjugate of the Fourier transform, ξ'_i is the fluctuating scalar profile along row i , and ξ'_{i+1} is the fluctuating scalar profile along row $i + 1$. The spectra were then averaged over all the rows or columns of each image and again over the 50 images in each data set. The result is average scalar energy and scalar dissipation spectra in the horizontal and vertical directions. Scalar dissipation spectra, $D(\kappa)$, were calculated directly from the energy spectra as shown in Eq. (12):

$$D(\kappa) = 2\Gamma \kappa^2 E(\kappa) \quad (12)$$

where κ is the wavenumber corresponding to a length scale, $l = 2\pi/\kappa$. The scalar energy and dissipation spectra determined from the horizontal rows of the fluorescence data are provided in Figs. 5 and 6. Since the data were taken at a constant crank angle of 240° bTDC, the integral scale, assumed to scale with the intake valve lift, is predicted to remain constant. The reduction of the defined Reynolds numbers is then expected to modify the high wavenumber range of the measured spectra. However, the dissipation spectra vary only slightly across the range of both Re_{intake} and Re_{engine} in each data set. The 600 RPM data (Fig. 5) do show a slight trend of shifting towards lower wavenumbers (larger length scales) as the Reynolds numbers are decreased, specifically in the dissipation range of length scales. The 1200 RPM data (Fig. 6) show almost no variance with respect to either of the defined Reynolds numbers. The spectra at both engine speeds deviate from one another at low