UMTA-MI-11-0006-02

Incipient Fault Detection System Study

≫ Technical Report ≪

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William B. Ribbens

Center for Transit Research and Management Development
University of Michigan Transportation Research Institute
2901 Baxter Road
Ann Arbor, MI 48109

MAY 1985
FINAL REPORT

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Prepared for
U.S. DEPARTMENT OF TRANSPORTATION
URBAN MASS TRANSPORTATION ADMINISTRATION
Office of Technical Assistance
Washington, D.C. 20590

NOTICE

Technical Report Documentation Page

1. Report No. UMTA-MI-11-0006-02	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle INCIPIENT FAULT DETECTION SYSTEM STUDY: TECHNICAL REPORT		5. Report Date May 1985
		6. Performing Organization Code
7. Author(s) William B. Ribbens		8. Performing Organization Report No. UMTRI-85-57-2
9. Performing Organization Name and Address Center for Transit Research & Management Dev. Univ. of Michigan Transportation Research Inst. 2901 Baxter Road Ann Arbor, Michigan 48109		10. Work Unit No.
		11. Contract or Grant No. MI-11-0006
12. Sponsoring Agency Name and Address U.S. Department of Transportation Office of Technical Assistance Washington, D.C. 20590		13. Type of Report and Period Covered TECHNICAL FINAL REPORT March 1983 - Sept. 1984
		14. Sponsoring Agency Code URT-33
15. Supplementary Notes		

16. Abstract

This technical report presents the results of the first phase of a project intended to detect incipient failure in bus components. This phase of the study has developed methods and instrumentation for measuring engine torque and torque non-uniformity. The next phase will involve collection of data from which a statistical model will be constructed. The statistical model will be used to estimate a time-to-failure for individual components, given the level of degraded performance.

This technical report contains details of the theory and methods used to measure degraded performance and the results of a test of the instrumentation in diesel-powered vehicles driven on a chassis dynamometer and on city streets. For a summary of the theory, methods, and results see *Incipient Fault Detection System Study: Executive Summary*.

17. Key Words Failure detection, Continuous performance measurement, Engine torque, Torque non-uniformity, Non-contacting measurements, Diesel engine, Bus maintenance		18. Distribution Statement This document is available to the U.S. public through the National Technical Information Service, Springfield, Virginia 22161	
19. Security Classif. (of this report) Unclassified	20. Security Classif. (of this page) Unclassified	21. No. of Pages 79	22. Price

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I. INTRODUCTION

This report summarizes technical progress in a study which has the goal of detecting incipient failures in certain critical components in buses. The present study has considered the feasibility of predicting bus component failures from optimal electronic instrumentation. The concept for this study involves direct measurements of bus component performance. Degradation in performance is an early indicator of incipient failure and can be used to predict failures before they occur. The specific bus components for this study include:

- Engine
- Transmission
- Brakes

The present study has great potential for reducing maintenance costs for bus fleets.

The potential areas for maintenance cost reduction and improved efficiency are:

- reduction of road calls by early detection of defects,
- reduction in incorrect diagnosis, and
- reduction of diagnostic time.

Each of these areas has a great impact on repair costs for bus fleets of all sizes.

A system for detecting incipient bus system failures has been relatively expensive in the past compared with costs of ground-based transportation. However, the commercial availability of relatively low cost sensors and microprocessor-based electronics raises the possibility of applying this failure detection concept to bus or truck fleets.

There are several issues affecting the feasibility of this concept in bus fleets. For example, the size of the system and the maintenance costs in total and per bus are important. In addition, the costs involved in bus in-service breakdowns greatly influence this feasibility because this failure detection scheme has the potential to nearly eliminate, or at least greatly reduce, the number of such breakdowns.

The proposed system is significantly different from existing fleet maintenance aids or automatic diagnostic instrumentation. In the latter case, computer-aided testing is performed to identify the existing status of individual components in an attempt to detect existing failures. Such tests are normally performed at test stations (e.g. fuel islands) and the monitoring results in a sample of the system's status at the time that measurements are made.

In the incipient failure detection, the vehicle's overall performance is continually monitored during normal operation. Performance degradation can be detected at an early stage. Whenever performance degradation is detected, an on-board diagnostic routine can be called upon to isolate the degraded component(s). Appropriate warning messages, calling for specific maintenance action, can be displayed or stored.

The concept of the incipient failure detection system can be understood with respect to figure 1 which is a block diagram for the instrumentation.

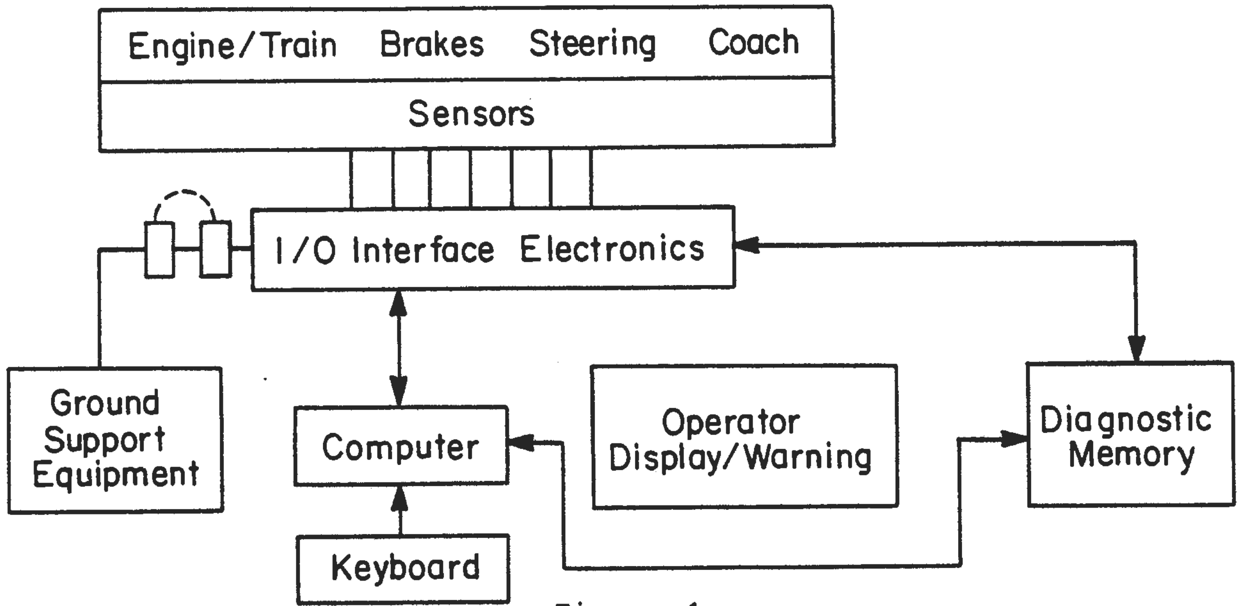

Figure 1

In this figure a group of sensors are schematically indicated which measure/monitor various vehicle variables or parameters. The electrical signals from these sensors are coupled to a dedicated on-board computer. This computer performs all computations or logical operations required to determine whether the vehicle subsystem is within allowable bounds. It also performs decision logical operations as required to issue a warning message. If this is a short-term, action-required issue the warning is given to the operator via the display. If this message is a longer-term maintenance item it might be stored in a non-volatile memory.

The longer-term action items (e.g., at end of in-service period or at routine maintenance period) messages are obtained from memory by Ground Support Equipment (GSE). This equipment can, for example, be connected to the vehicle during normal maintenance or it might be connected briefly when the vehicle is removed from service log at end of day.

In addition to alerting maintenance personnel to incipient component/subsystem failure, this GSE can assist in maintenance diagnostics. In this application each vehicle electrical, electronic, or electromechanical component can be performance tested by a computer routine. This feature has the benefit of reducing diagnostic time and can thereby reduce maintenance costs.

It is helpful to consider a few examples of vehicle performance monitoring to illustrate the system concept. One of the most significant performance indicators of any vehicle is the engine brake torque or output power. A simple, inexpensive, non-contacting sensor for torque measurement has recently been reported [Ref. 1]. Although the sensor concept as explained there is applicable only to vehicles equipped with automatic transmissions, recent improvements have yielded a system equally applicable to vehicles with a manual transmission.

Engine output power can be obtained simply by multiplying the torque by crankshaft angular speed. This speed measurement may be obtained from the same sensor used to measure torque. Thus, a simple and inexpensive sensor exists for continuously monitoring engine performance. This sensor, in conjunction with a simple computer algorithm, can essentially instantaneously detect degradation in engine torque/power.

Another major engine performance variable is torque non-uniformity. Although engine torque is inherently non-uniform, excessive non-uniformity is a direct indication of engine performance degradation.

Another important performance variable of a highway vehicle is braking effectiveness. This performance variable is particularly important for buses as a safety issue--a large proportion of urban bus fatal accident involvements in the U. S. involve pedestrians and other vehicles being struck by the front of the bus. Braking effectiveness involves the relationship between brake line hydraulic pressure and braking force for a given vehicle. Braking force can be measured using sensors at the wheel, and brake line pressure may be estimated closely from sensors in the vehicle suspension system. From such measurements, braking effectiveness can be computed and presented or recorded each time brakes are used during normal operation of the vehicle. Degradation of the braking system thus can be identified as a continuous rather than a pass/fail measure, and the results can alert maintenance personnel for early repair or replacement.

The procedures for the present study involve several key steps. The initial step has been to develop relatively low cost electronic instrumentation for monitoring the performance of the bus components listed above. This instrumentation can detect degradation in performance which serves as the first indication of incipient failure (except for the case of sudden catastrophic failure). The next phase of the study involves development of a statistical model for the relationship between time to failure and a given level of degradation. This has been the most difficult portion of the study because of the unavailability of suitable data bases upon which the statistical model can be formed. The final aspect of the study has been to synthesize a suitable maintenance strategy which optimally utilizes the information from the performance instrumentation. Owing to limited funds for this project, not all of the intended studies could be completed.

II. PERFORMANCE MEASURING INSTRUMENTATION

The present study is concerned with measurements of performance of 3 primary bus components: the engine, transmission, and brakes. Performance measurements are distinctly different for each of these components. The primary performance measurement for the engine is the torque (or horsepower). Accurate measurements of engine torque and torque non-uniformity can be made using instrumentation developed during this study. These measurements can give engine performance degradation directly.

Brake performance can also be represented directly by braking efficiency or by the hydraulic pressure required for a given vehicle deceleration indirectly. The sensors required for either direct or indirect brake performance measurements are currently available and require no additional development.

The instrumentation for each of the bus components is discussed separately in this report. We begin with the instrumentation for torque non-uniformity measurements.

Torque Non-Uniformity Measurement

Ideally, the torque produced by an internal combustion engine should be maximally uniform. That is, the torque produced during steady state operation by each cylinder should be identical. In addition, the torque should be identical for each engine cycle during steady state operation.

In practice the torque production varies from cycle to cycle and from cylinder to cylinder. However, in a normal engine the torque variation from cylinder to cylinder is relatively small.

On the other hand, the variation in torque production for an engine with degraded performances is relatively large. For example, a diesel engine having one partially malfunctioning fuel injector has a larger variation in cylinder-to-cylinder torque production than for a normal engine.

There are several types of component malfunctions or degradation which result in engine performance degradation. For example, the fuel injector performance is influenced by wear, dirt in the injector nozzle assembly, incorrect pressures (line, opening, closing), and many other non-catastrophic failures. Any such change in the injector causes the fuel injected into the cylinder to be incorrect for any particular operating condition.

The torque produced by the affected cylinder will be different from normal torque production and, in particular, different from the other cylinders. That is, the non-uniformity in torque production will be larger than for normal cylinder operation. A measurement of torque non-uniformity serves as an indicator of degraded (non-standard) performance of the components associated with that cylinder.

The instrumentation developed on this project is capable of measuring such torque non-uniformity. Moreover this instrumentation is capable of identifying the affected cylinder(s) causing the non-uniformity.

Once it has been determined that a given cylinder has degraded performance there are several standard procedures for identifying the component which is responsible for the degraded performance. For example, compression measurements can identify poor valves or worn/broken rings. In addition, various pressure measurements can isolate malfunctioning injectors (ref. Henein 1).

Other fuel system components can similarly cause an increase in torque non-uniformity. Depending upon the type of fuel system used, there can be failures in the pump or the rack as well as individual fuel injectors. Moreover, poorly seating valves, worn or broken piston rings, and leaky cylinder head gaskets can cause severe non-uniformity in the torque which is generated by a diesel engine. Thus, a measurement of torque non-uniformity serves as the basis for detecting degradation in various components of the diesel engine and can be used to predict bus failure.

In addition, the present method of monitoring torque non-uniformity can identify an individual malfunctioning cylinder.

The instrumentation system has sufficient information to detect degraded torque generation in a cylinder and to issue an alarm indicating a potential failure in a given cylinder. This information is useful for preventative maintenance and can significantly shorten diagnostic time and repair time thereby improving maintenance efficiency.

The present research project has developed a scalar index for torque non-uniformity which can distinguish between normal and degraded torque production. This study has also developed relatively low cost non-contacting instrumentation for measuring this torque non-uniformity. This instrumentation, which is discussed in the next section of this report, requires minimal engine modification.

Theory of Method

The present non-contacting method of measuring IC engine torque non-uniformity is based upon the relationship between torque and angular velocity of the crankshaft. Although this relationship is complex, it can be expressed approximately by a relatively simple model. In presenting this model it is convenient to represent the engine load in terms of an approximate equivalent electrical circuit such as is depicted in figure 2.

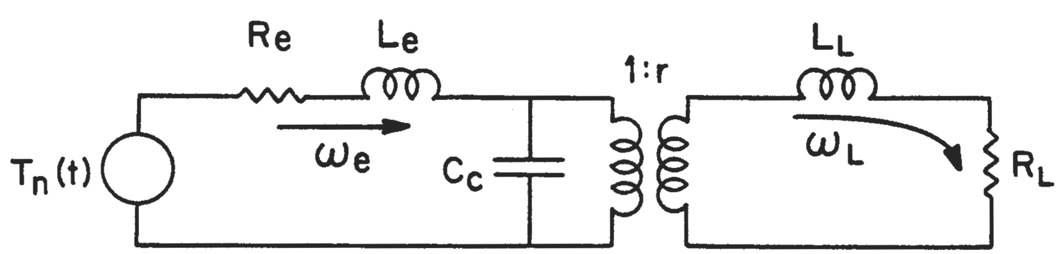

Figure 2

In this equivalent circuit torque T is analogous to voltage and angular velocity ω_e is analogous to current. Furthermore, friction is represented by resistance, moment of inertia by inductance, the angular elastance of the clutch torsional damper springs by capacitance, and the transmission is represented by an ideal transformer.

For the purposes of this report it is convenient to denote the net torque which is applied at the crankshaft as $T_n(\theta)$. This torque is produced by the cylinder pressure during power stroke. The force resulting from this pressure is applied through the connecting rod to the crankshaft.

However, there are several losses in the mechanism which couples the cylinder pressure to the crankshaft. There are losses due to friction, pumping of intake and exhaust gases, accessory loads, camshaft loads. Moreover, there are forces associated with the acceleration of the piston connecting rod assembly. The superposition of the torques from all of these forces produces a torque T_n which is applied to the crankshaft and which we have denoted T_n.

Also shown in figure 2 are a pair of currents ω_e and ω_L denoting:

ω_e = instantaneous crankshaft angular speed

ω_L = instantaneous transmission output speed.

The instantaneous crankshaft angular speed is represented by $\omega_e(t)$ where

$$\omega_e = \frac{d\theta_e}{dt}$$

θ_e = instantaneous angular position of crankshaft (rel. to TDC of #1 cyl. on compression/power).

The component of friction torque, which is proportional to crankshaft angular speed, is represented by the 'torque drop across' resistance R_e. The time varying frictional torque is included in $T_n(\theta)$.

The moment of inertia of the crankshaft flywheel and clutch pressure plate is represented by inductance L_e. This parameter is fixed for any given engine configuration.

The so-called clutch torsional damper springs have the greatest compliance of any driveline component. The purpose of the clutch torsional damper is to isolate the load from the torque variations which are a concommitant part of any reciprocating I-C engine. The torsional elastance is analogous to shunt capacitance in the equivalent circuit. This elastance is represented by capacitor C_c in figure 2.

It has been presumed implicitly that the mechanical configuration under discussion includes a manual transmission. This transmission is represented by an ideal transformer (d-c coupled) having turns ratio 1:r. That is, the average load torque $\langle T_L \rangle$ is given by

$$\langle T_L \rangle = r \langle T_b \rangle$$

where

$\langle T_b \rangle$ = brake torque.

The load is represented in figure 2 by an inductance L_L and resistance R_L combination. This representation is applicable to all studies reported in this paper, some of which were conducted with a water brake dynamometer. The moment of inertia of a water brake varies somewhat with the total water content. In the present case the full water inertia is 12% more than the empty brake inertia. The moment of inertia of the engine rotating components combines with the load moment of inertia yielding one of the calibration parameters for the measurements reported in this paper. The variation in load moment of inertia could potentially result in a calibration error (or uncertainty). However, it can be shown that the clutch torsional damper springs provide sufficient isolation of the crankshaft from load variations, that the calibration constant variation is negligible.

The load angular speed is represented by $\omega_L(t)$ and the instantaneous load torque is represented by $T_L(t)$. Power absorption in the load is represented by resistor R_L, the resistance of which depends upon the operating condition.

The present study and its conclusions are also applicable to a system having an automatic transmission. In this case the shunt capacitor in the equivalent circuit can be replaced by a shunt resistance. This resistance is nonzero at zero frequency representing torque converter slip and is essentially zero resistance for the time varying components of torque (i.e., harmonics of cylinder firing frequency). That is, the torque converter provides extremely high isolation of the crankshaft from load variations.

The torque acting on the clutch pressure plate (which can properly be termed the instantaneous brake torque) is denoted $T_b(t)$. The torque multiplication in the transmission is represented by the turns ratio 1:r of the transformer in figure 2.

The above lumped parameter equivalent circuit and the associated mathematical model are approximations to the actual dynamic system. However, these approximations are sufficiently accurate to explain the present method of measuring torque non-uniformity.

The present study is concerned with measurements of non-uniformity of torque T_n under steady state conditions. In this case it is possible to separate each of the variables in the equivalent circuit into d-c and a-c components:

variable		d-c		a-c
$T_n(t)$	=	T_n	+	$\tau_n(t)$
$\omega_e(t)$	=	Ω	+	$\eta_e(t)$

Non-uniformity occurs in the time varying components of torque. An equivalent circuit with respect to d-c components is shown in figure 3 and a corresponding equivalent circuit for time varying components is depicted in figure 4.

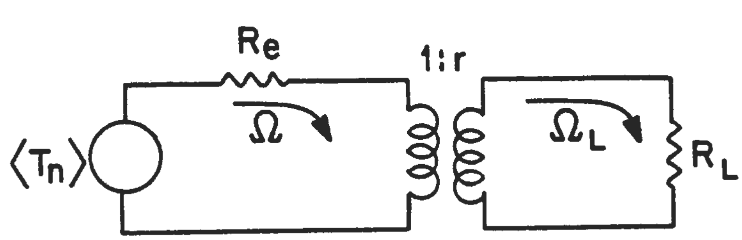

Figure 3

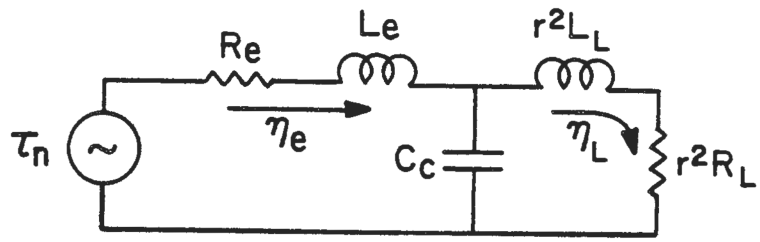

Figure 4

In this figure the load parameters have been referred to the primary of the transformer (i.e. at the transmission input). The angular speed η_L' is the time varying component of the transmission input shaft.

In certain configurations (depending upon system parameters) the complex load impedance Z_L satisfies the following inequality:

$$|Z_L| = |j\lambda r^2 L_L + r^2 R_L| >> \frac{1}{\lambda C_c} \qquad \lambda \epsilon \Lambda$$

where

λ = radian frequency

Λ = set of all frequencies for which $T_n(t)$ has non-negligible power).

In particular, it is shown later that this inequality is satisfied for experimental studies reported in this report. For such cases the a-c equivalent circuit of figure 4 can be further simplified as depicted in figure 5.

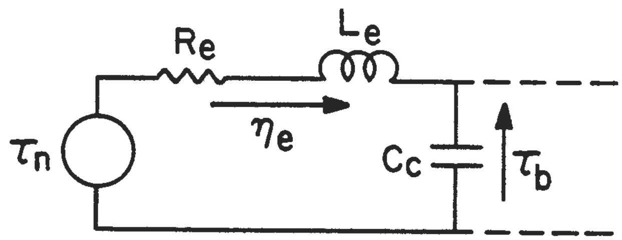

Figure 5

The circuit of figure 5 is equivalent to the condition in which the clutch torsional damper has effectively caused total isolation of the load from torque fluctuations $T_\eta(t)$.

Within the validity of a linearized model the crankshaft instantaneous angular speed $\eta_e(t)$ is given by the convolution of the torque $\tau_n(t)$ with the impulse response of the equivalent circuit:

$$\eta_e(t) = \tau_n(t) * h_c(t)$$

where

$h_c(t)$ = impulse response of equivalent circuit.

This model is approximately valid with respect to torque variation $\tau_n(t)$ at any steady state operating condition.

Alternately the relationship between torque and crankshaft angular velocity can be expressed in terms of the sinusoidal frequency response $H_c(j\lambda)$

$$H_c(j\lambda) = \frac{\eta_e(j\lambda)}{\tau_\eta(j\lambda)}$$

where

$$\tau_n(t) = \text{Re}\ \tau_n(j\lambda)e^{j\lambda t}$$

$$\eta_e(t) = \text{Re}\ \eta_e(j\lambda)e^{j\lambda t}$$

$$\lambda = \text{radian frequency}$$

For the approximate linear equivalent circuit of figure 5 we have

01 $$H_c(j\lambda) = \left[R_e + j(\lambda L_e - \frac{1}{\lambda C_c})\right]^{-1}$$

Moreover as the frequencies of the Fourier components of τ_n are proportional to RPM (steady state condition), this sinusoidal frequency response is a function of RPM.

Thus, non-uniformity in torque $T_n(t)$ gives rise to an associated non-uniformity in $\eta_e(t)$ and this latter variable serves as a measurement of the non-uniformity of the torque. However, the relationship between the two measures of non-uniformity depends upon RPM and upon the spectral distribution of $\tau_n(t)$.

In certain engine drive train configurations the system parameters are such that the equivalent circuit of figure 5 is reduced to that depicted in figure 6.

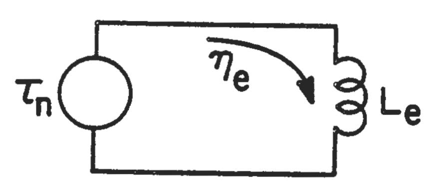

Figure 6

This system is equivalent to a purely Newtonian mechanical system in which the torque τ_n is balanced by inertial torque of the rotating components of the engine and the clutch pressure plate

$$\tau_n(t) = L_e a(t)$$

where

$$a(t) = \text{crankshaft angular acceleration}$$
$$= \frac{d\eta_e(t)}{dt}$$

This situation involves the mathematically simplest relationship between non-uniformity in $\tau_n(t)$ and in $\eta_e(t)$:

$$\eta_e(t) - \eta_e(0) = \frac{1}{L_e} \int_0^t \tau_n(t')dt'$$

Regardless of the complexity of the model which is required to represent the system, it is clear that non-uniformity in torque results in non-uniformity variation in crankshaft angular velocity. This relationship between $\tau_n(t)$ and $\eta_e(t)$ raises the possibility of measuring non-uniformity in torque τ_n from a measurement of the corresponding non-uniformity in $\eta_e(t)$. This paper presents a non-contacting method of measuring the non-uniformity in $\eta_e(t)$.

Ideally the torque generated by a reciprocating I-C engine at constant load and RPM would be periodic in time. However, in practice, this torque is a random process which is not necessarily stationary. An earlier paper (ref. 1) presented a quantitative index for the torque non-uniformity based upon measurements of driveline torque. The present paper extends this concept of representing non-uniformity to non-contacting measurements of crankshaft angular speed and to an equivalent computation of a non-uniformity metric.

The concept of measuring torque non-uniformity which is presented in (ref. 1) is based upon the presumption that the essential non-uniformity is represented by relative maxima and relative minima of the torque random process. Similarly the torque non-uniformity is representable by the corresponding relative maxima and minima of angular speed η_e. The angular speed can be measured using a non-contacting sensor and straightforward electronic signal processing.

In method of ref. 1 the instantaneous torque is sampled at these extrema and a 2N dimensional vector is obtained for an N cylinder engine. From this vector a scalar metric is computed by elementary arithmetic operations. This scalar is identically zero for maximally uniform torque and increases monotonically with non-uniformity.

In this report a non-uniformity metric is introduced which is somewhat modified from ref. 1. However, this metric is also based upon extremal values of the crankshaft angular speed. Figures 7 and 8 are records of an analog of crankshaft angular speed (at constant RPM) for two separate operating conditions illustrating the instantaneous variation due to each cylinder firing event. The non-uniformity for this operating condition, though relatively small, can also be seen. In figures 7 and 8 the ordinate is proportional to the negative of angular speed.

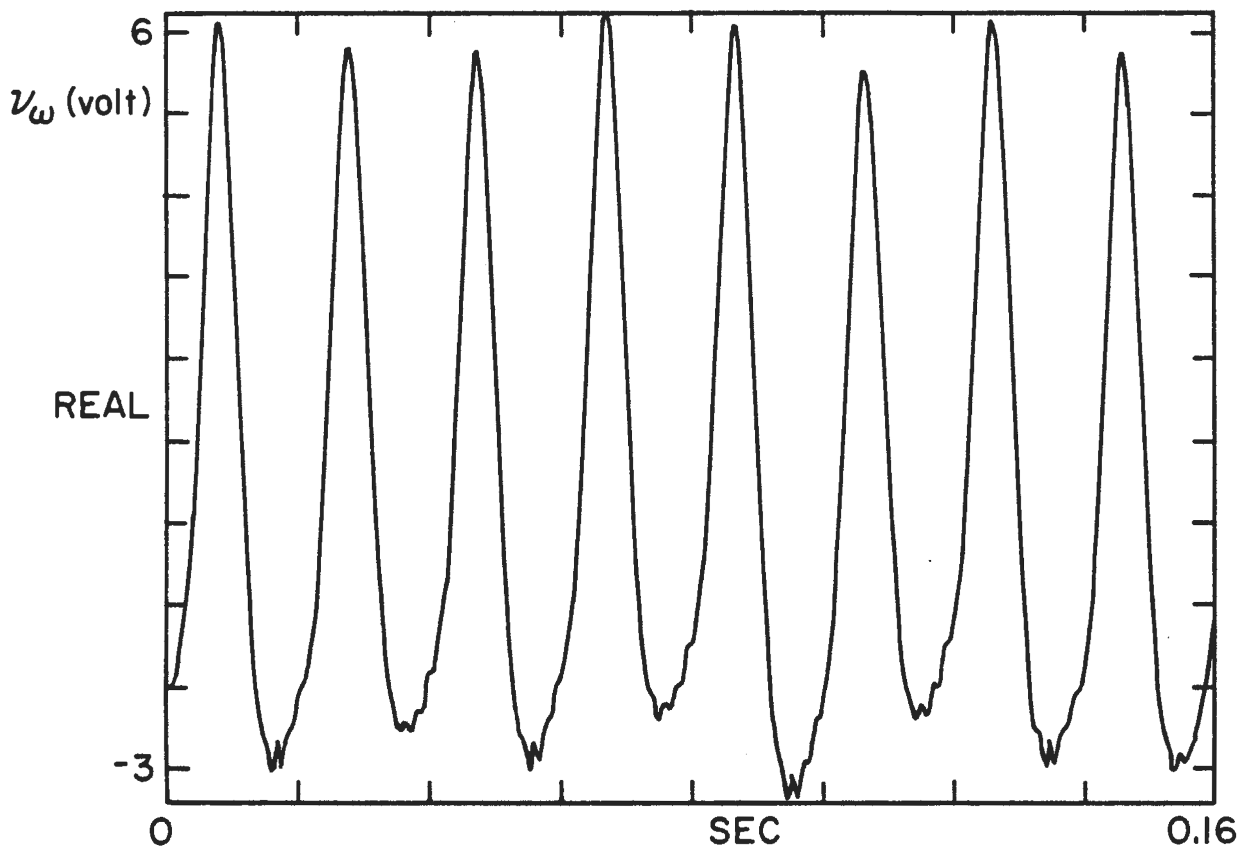

Figure 7. 1500 RPM, Average Load Torque 58 N·m

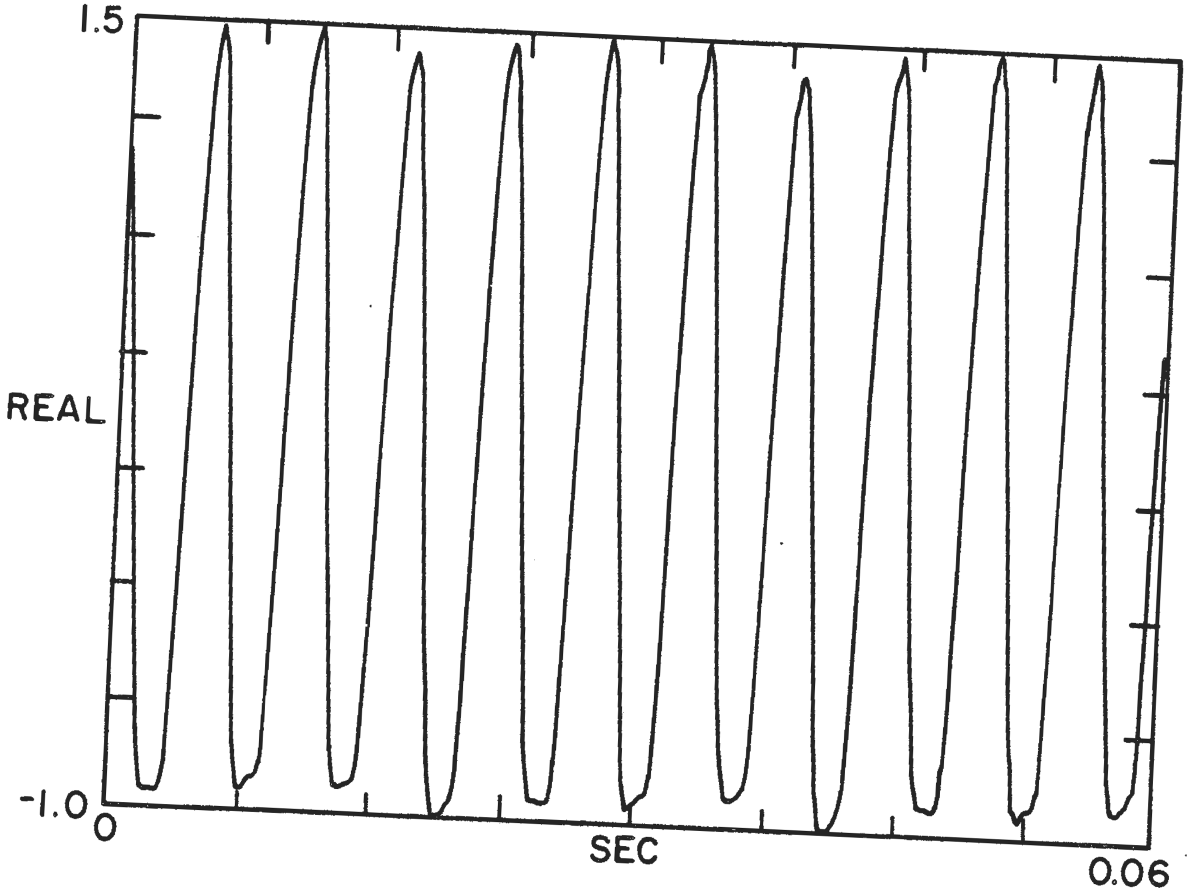

Figure 8. 4000 RPM, Average Load Torque 92 N·m

Consequently relative maxima of the analog signal represent relative minima of angular speed and vice-versa.

The non-uniformity metric for torque is obtained from an electrical analog of crankshaft angular speed which is denoted $v_\omega(t)$:

$$v_\omega(t) = -k\eta_e(t)$$

The instrumentation for generating this analog is explained in a later section of this paper. The essential torque non-uniformity information is contained in the relative maxima and minima of $v_\omega(t)$. The following notation is used throughout this paper:

$$v_n(k) = v_\omega[t_n(k)] \qquad n = k,\ k+1 \ldots k+N-1$$
$$k = 0,\ N,\ 2N \ldots$$

where

$t_n(k)$ = time of nth relative maxima of $v_\omega(t)$ for the kth block of data.

Often N is chosen to be an integral multiple of the number of cylinders and k denotes the number of blocks of sampled data each having length N. This is a convenient choice for studying non-uniformity which is repeated (at least approximately) each engine cycle. However, it should be emphasized that for other applications other choices for N are perhaps desirable.

An N dimensional vector $\underline{v}(k)$ is then formed:

$$\underline{v}'(k) = [v_1(k),\ v_2(k) \ldots v_N(k)]$$

The average value v(k) of the components is obtained:

$$v(k) = \frac{1}{N} \sum_{n=1}^{N} v_n(k)$$

Next, the so-called non-uniformity vector $\underline{n}(k)$ is formed:

$$\underline{n}(k) = \underline{v}(k) - \underline{u}v(k)$$

where

$$\underline{u}' = [1,1 \ldots 1] \qquad \text{N dimensional}$$

We define the non-uniformity metric n(k) for the kth interval in terms of the ℓ_1 norm of the vector n(k):

$$n(k) = \frac{1}{N} \, || \, \underline{n}(k) \, ||_1.$$

In addition, the average non-uniformity metric can be computed for K intervals

$$\langle n \rangle = \frac{1}{k} \sum_{k=1}^{K} n(k)$$

In the case of maximal uniformity over the kth interval all components of vector $\underline{v}(k)$ are equal and $\underline{n}(k)$ is a null vector. The non-uniformity metric for the kth interval is also null:

$$n(k) = 0$$

As the torque (and consequently the angular speed) deviates from perfect uniformity, the non-uniformity metric increases from zero.

The results of the computation for $\underline{n}(k)$ and $n(k)$ depend upon operating conditions. In particular these results depend upon RPM. This occurs because the relationship between torque τ_n and crankshaft angular speed η_e is dependent upon RPM and load (i.e., $H(j\lambda)$ depends upon RPM and load).

The numerical results for $n(k)$ depend upon the number N of samples which are used to construct vector $\underline{v}(k)$. The choice of N depends largely upon the application of the non-uniformity metric. If the non-uniformity data is to be synchronized with each engine cycle then N should be twice the number of cylinders. For normal operation of an engine the non-uniformity is roughly synchronous with each engine cycle. In this case it is advantageous (though not necessary) to choose N as an even integral multiple of the number of cylinders. However, as N increases, the time delay for computing non-uniformity is also increased.

In addition, the computation of $n(k)$ is highly tolerant of sampling errors. That is, for certain operating conditions, certain relative maxima are sufficiently small that they are below detectability (as explained in the instrumentation section). A condition such as this is easily detected by means of a real time clock and a simple algorithm in the computer. In the case of a missing data point in a block of data, the computation proceeds normally with N replaced by N-1. This produces a negligible change in the results as will be demonstrated in a later section of this report.

It should be emphasized at this point that the electrical signals required to obtain the non-uniformity vector come from an analog of crankshaft angular speed. It is shown in the next section of this report that these signals are obtained using a single non-contacting, inexpensive sensor which can be installed on the engine with minimal engine modification.

The instrumentation for measurements of crankshaft angular speed and for computing torque non-uniformity is depicted in figure 9:

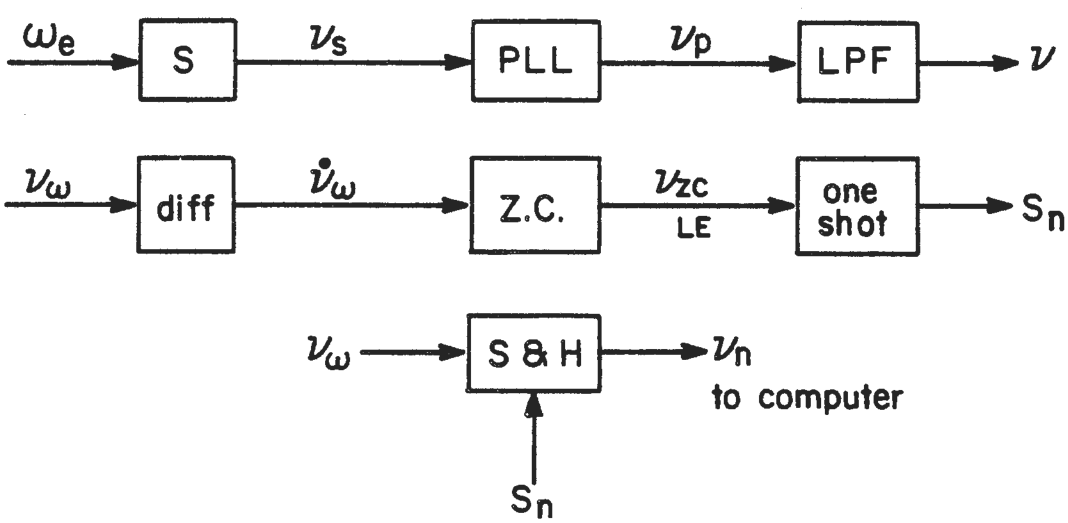

Figure 9. Instrumentation

The measurements of crankshaft angular speed are based upon a non-contacting magnetic field system sensor. This sensor generates an output voltage the frequency of which is proportional to instantaneous crankshaft angular speed.

A phase locked loop (PLL) acts as a frequency to voltage converter generating an output voltage v_p which has a component proportional to ω_e. A low pass filter (LPF) separates the desired analog of crankshaft angular speed from other undesirable output components which are produced by the PLL.

A standard extremal sampler circuit generates sampling pulses S_n at times t_n, respectively. This circuit includes a differentiator and a zero crossing detector. The latter device has binary output which changes state as the time derivative of v_ω changes polarity.

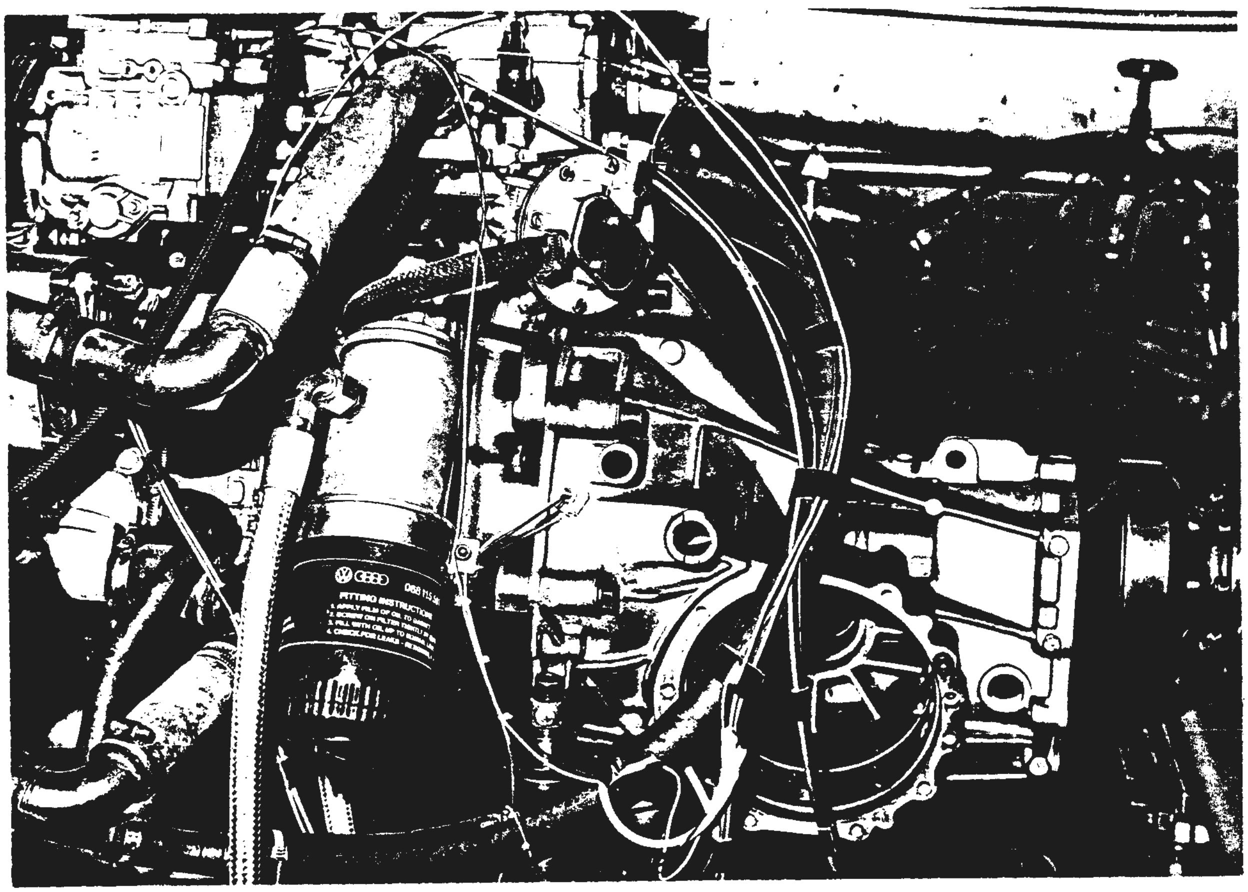

Figure 10

A pair of one shot circuits generate relatively short duration pulses at the desired sampling times. A sample and hold circuit (S-H) generate the corresponding sampled values of v_ω at t_n.

Figure 10 is a photograph of the sensor as mounted in an experimental configuration. The sensor is mounted on the bell housing directly over the starter ring gear.

The sensor consists of a permanent magnet around which a coil is wound. The magnetic field of the permanent magnet couples to the steel starter ring gear. The magnetic flux linkage of the coil varies with crankshaft angular position being influenced by the spacing between the magnet and ring gear and by the ring gear profile.

Figure 11 is a sketch of the sensor-starting gear configuration.

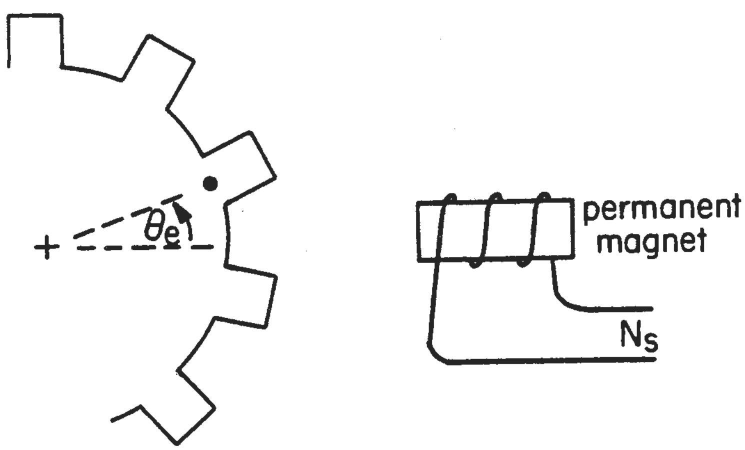

Figure 11

As the crankshaft rotates the sensor generates output voltage $v_s[\theta_e(t)]$.

This voltage can be approximately modeled as follows:

$$v_S = V(\theta_e,\Omega)f_S\{M\theta_e(t) + \psi_e[\theta_e(t)]\}$$

where

$$M = \text{\# teeth on ring gear}$$

$$V = \text{amplitude function}$$

$$\psi[\theta_e(t)] = \text{pseudo-random phase due to tooth to tooth aperiodicity.}$$

The instantaneous sensor frequency $\omega_S(t)$ is given by:

$$\omega_S(t) = M\omega_e(t) + \dot{\psi}_e(t)$$

The phase locked loop output voltage is given by:

$$v_p = v_o - k_p\omega_S(t) + r(t)$$

where

v_o = d-c component

$r(t)$ = undesirable components from PLL which are at harmonics of frequency

k_p = constant for PLL.

The low pass filter output is given by the convolutin of v_p and the filter impulse response

$$v_\omega(t) = h_{LPF}(t) * v_p(t)$$

where

$h_{LPF(\cdot)}$ = filter impulse response.

For the experimental measurements which are reported in this paper the filter bandwidth is sufficiently large relative to the important Fourier coefficients of $n_e(t)$ that we have the following approximate result.

$$v_\omega \simeq v_o - k_a\omega_e(t) + \varepsilon(t)$$

where

$$k_a = .074 \text{ volt/rad/sec}$$

$$\varepsilon(t) = \text{error due to } \psi(\theta_e) \text{ and amplitude fluctuation } V(\theta_e,\Omega)$$

The error term $\varepsilon(t)$ is primarily a pseudo-random process which is synchronous with θ_e. However, there may also be a non-negligible electrical noise for certain operating conditions.

Experimental Study

Experimental measurements have been made of the non-uniformity metric for a 4 cylinder 1.5 L VW Diesel engine. Separate sets of measurements were made with the engine in a test cell and in a vehicle which was driven on various streets. The results of these studies are presented separately.

Test cell measurements were made with the engine driving a water brake load through an elastic link coupling. Figure 12 is a photograph of the installation showing the major components. Figure 13 is a schematic drawing of the experimental configuration.

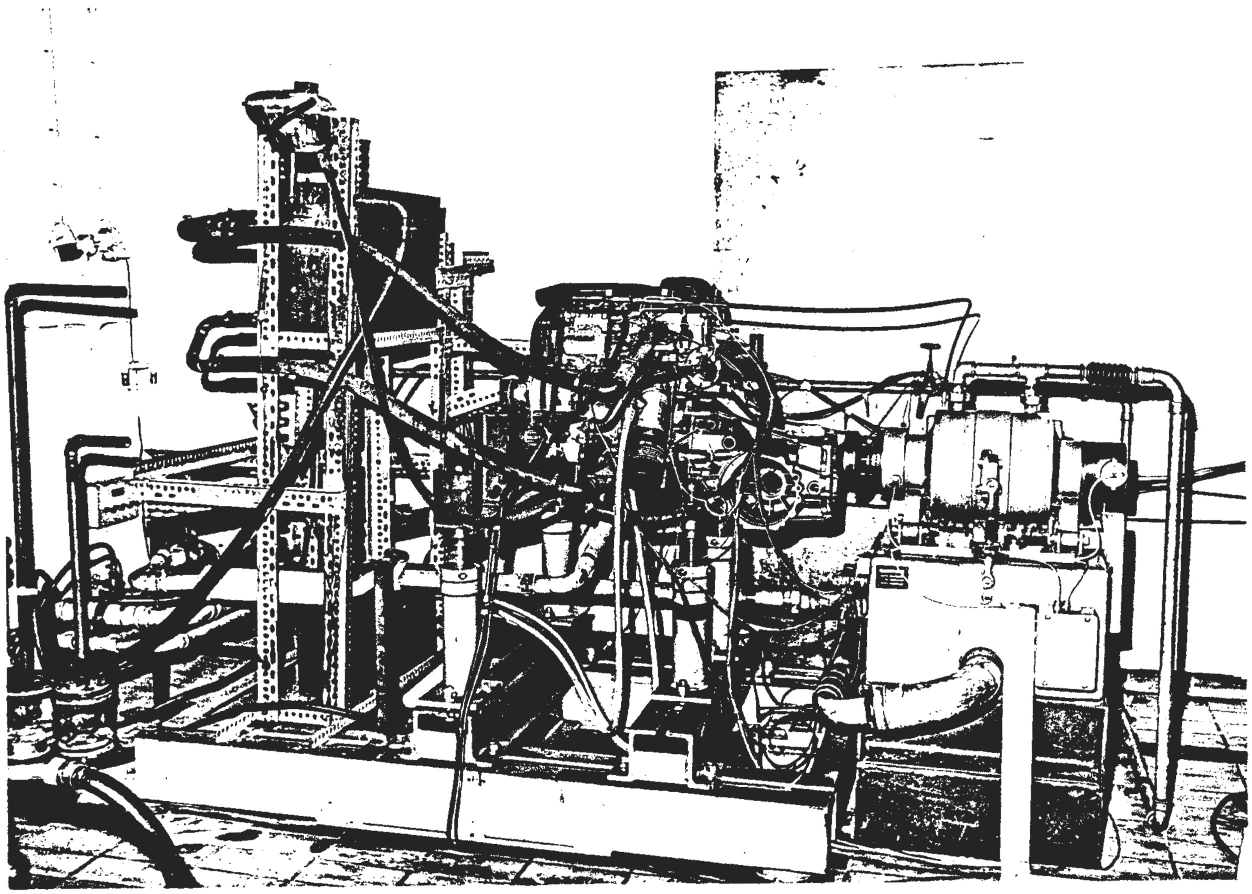

Figure 12

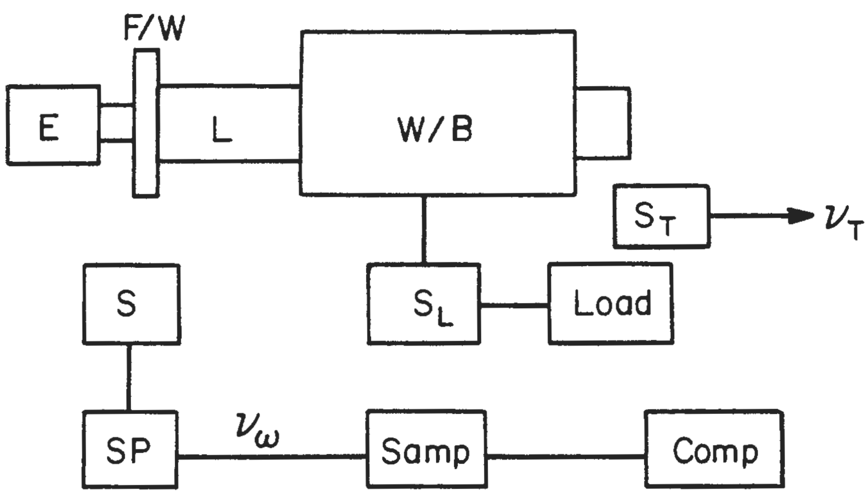

Figure 13

This figure depicts the engine (E), the flywheel (F/W), the elastic coupling link (L), the water brake (WB), the angular speed sensor S, a sensor S_L for measuring the WB reaction torque and another S_T at the rear of the WB. This latter device is an electro-optical sensor which generates a single output pulse each revolution which occurs essentially at TDC of #1 cylinder (except for torsional displacement along link L). The electronic signal processing SP for generating v_ω. The extremal sampling circuit (samp) and the computer (comp) are also depicted.

The experiments were made using a standard production VW Diesel engine running at a variety of conditions consisting of steady RPM and various load torques. In a separate set of measurements the engine was operated with one fuel injector disconnected in order to achieve a condition of extreme torque non-uniformity. For each operating condition a permanent record was obtained of a sample of the analog waveform. In addition the torque non-uniformity metric was obtained for K successive engine cycles. The number of cycles varied somewhat with the choice of operating conditions but generally ranged from about 600 to over 1000 cycles.

The torque non-uniformity metric n(k) is, of course, a non-negative random variable for which estimates have been computed of the mean $\langle n(k) \rangle$ and standard deviation $\sigma(n)$ from the K measurements of n(k):

$$\langle n \rangle = \frac{1}{K} \sum_{k=1}^{K} n(k)$$

$$\sigma(n) = \{ \frac{1}{K} \sum_{k=1}^{K} (n(k) - \langle n(k) \rangle)^2 \}^{1/2}$$

In reporting the results of measurements it is convenient to express the ratio σ<n>:

$$\rho = \frac{\sigma(n)}{\langle n\rangle} 100\%$$

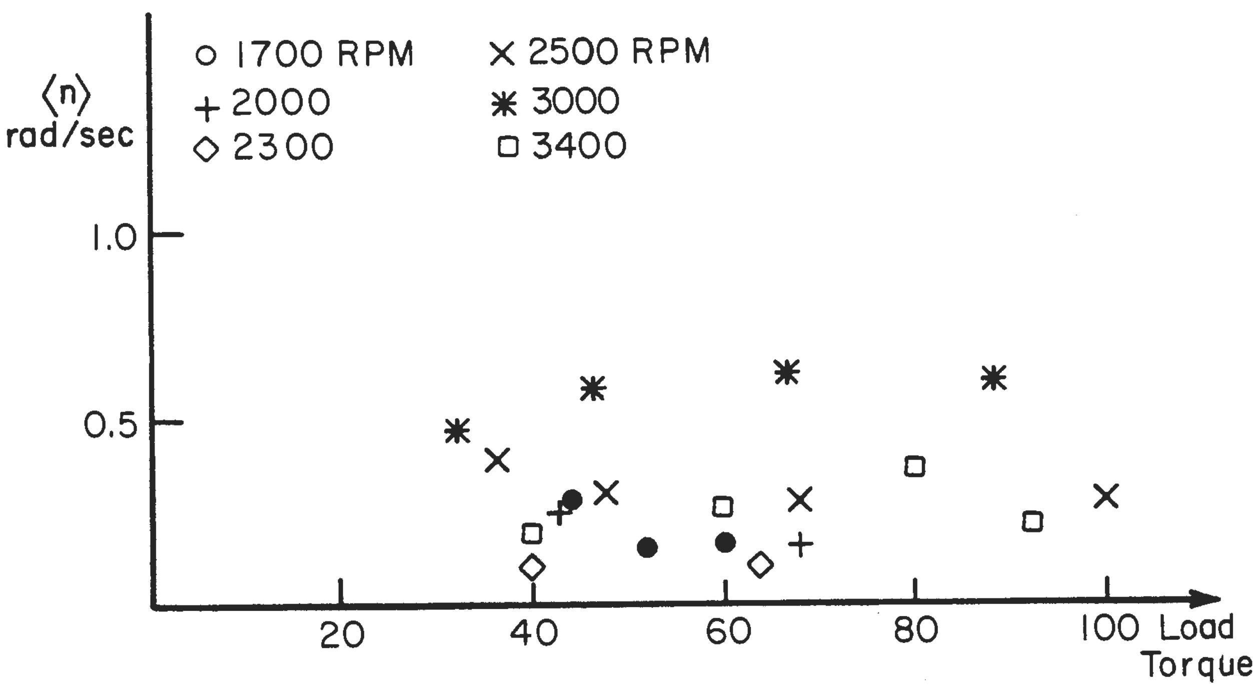

Figure 14

Figure 14 is a graph of the measurement results for the mean value <n(k)> for several operating conditions for the standard production engine. The non-uniformity metric varies with operating condition but for the particular experimental test engine <n> is generally less than about .6 rad/sec.

The standard deviation varies with load and RPM. Table 1 presents the largest fractional standard deviation for each RPM.

Table 1

RPM	max ρ %
1700	14
2000	15
2500	63
3000	20
3400	8

It is, perhaps, instructive to examine the actual wave form v_ω for a few representative example operating conditions. Figure 15 is a set of recorded samples of this waveform.

Several interesting features of the crankshaft angular speed are evident from figure 15. For example, it can be seen that the torque non-uniformity is relatively small for this production engine except for the region near 3000 RPM.

The experimental test engine was also operated with one cylinder disabled. The fuel injector was disconnected and the engine operated on 3 cylinders. This yields a condition of extreme torque non-uniformity, far worse, in fact, than would be encountered in any normal engine operation. The extreme non-uniformity in v_ω results because one cylinder is not firing at all although it is still compressing the intake air.

The angular velocity similarly has extreme non-uniformity as can be seen from figure 16. This figure presents the waveform $v_\omega(t)$ for several operating conditions with 3 cylilnder operation. The waveforms of figure 16 can be compared with the corresponding waveforms of figure 15. The large excursions in crankshaft

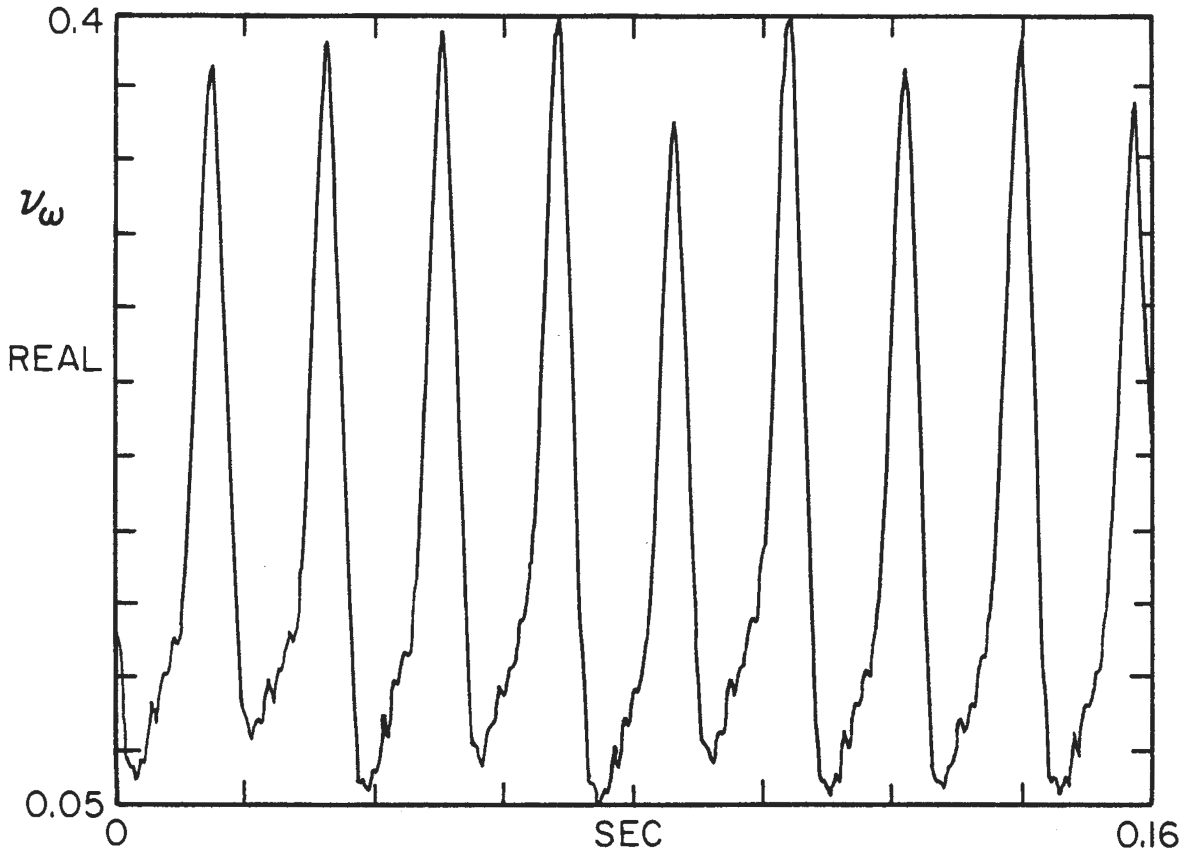

Figure 15a. 1700 RPM, Average Load Torque 52 N·m

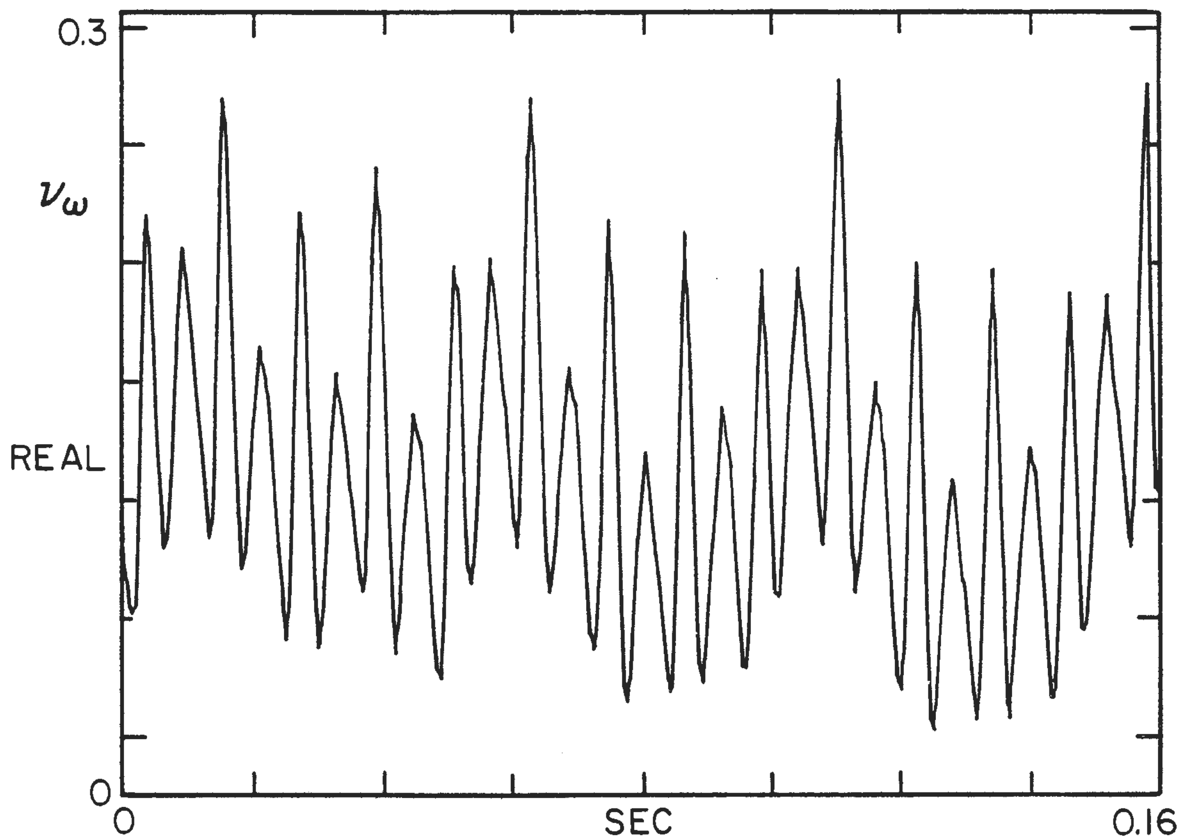

Figure 15b. 2500 RPM, Average Load Torque 48 N·m

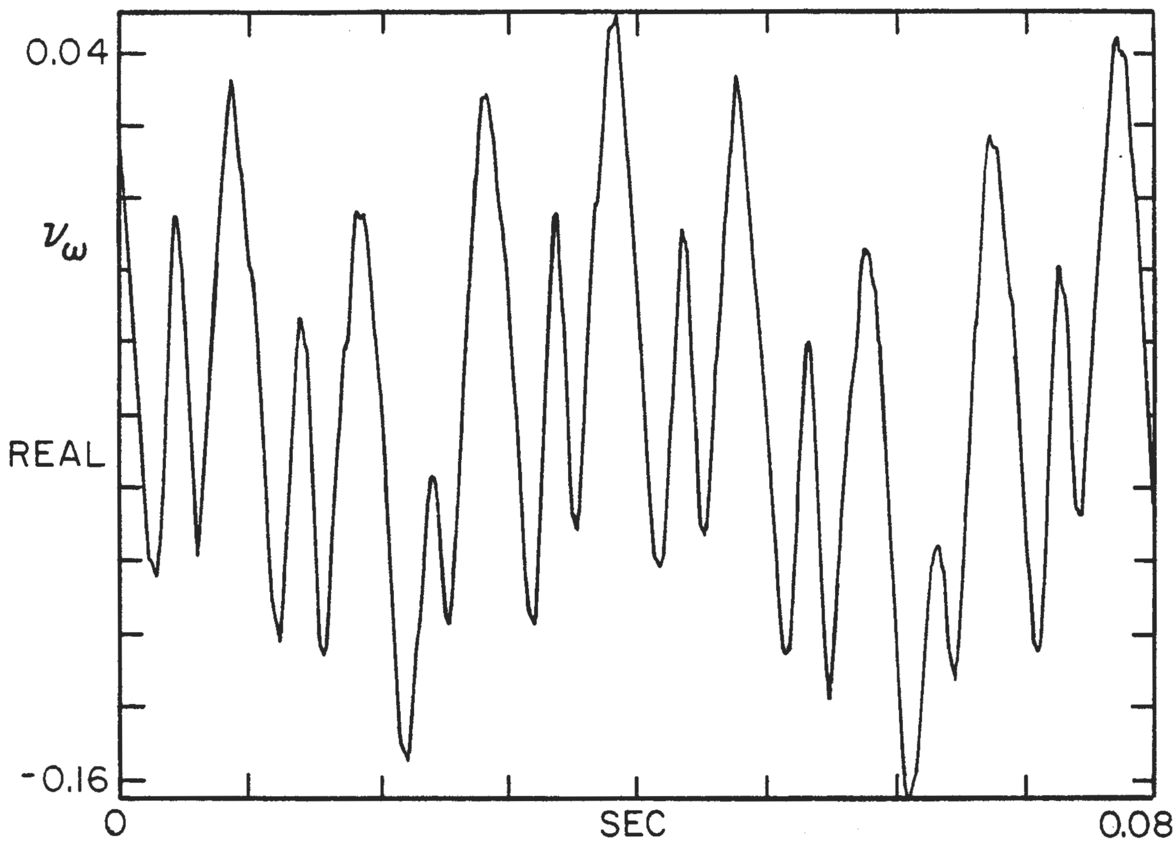

Figure 15c. 3000 RPM, Average Load Torque 37 N·m

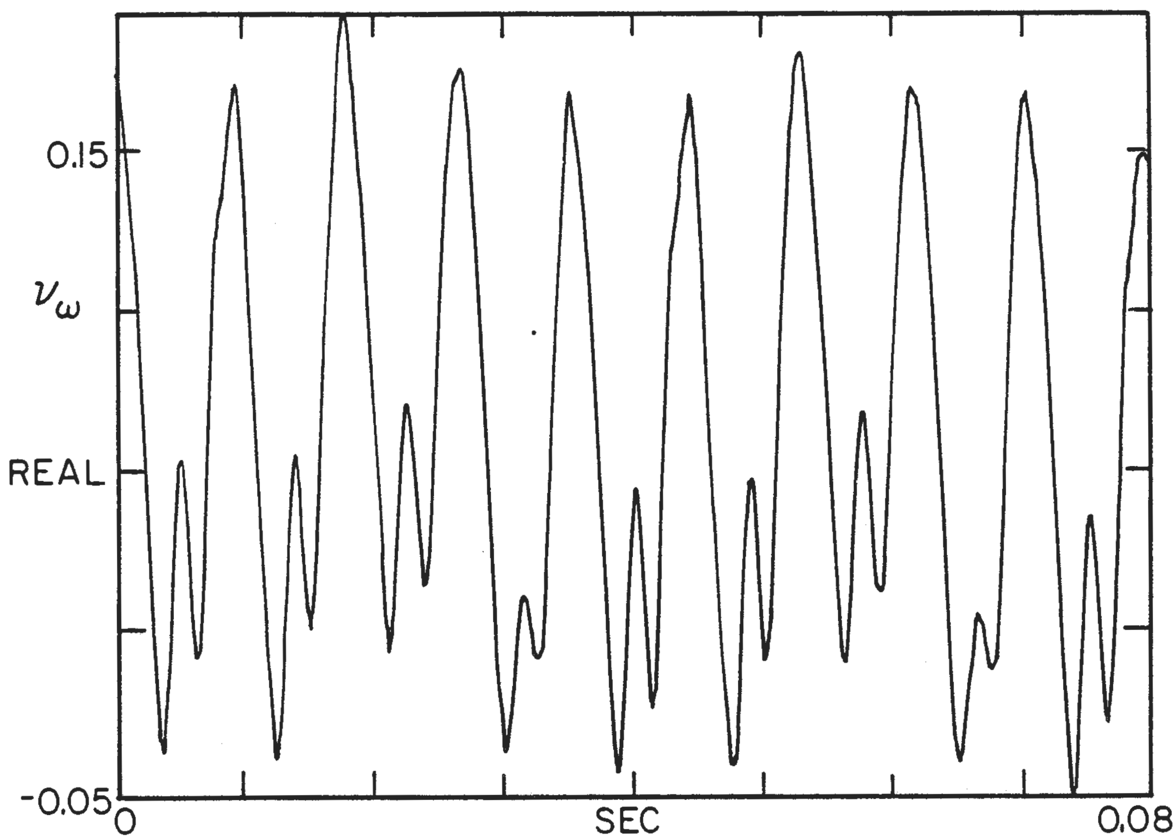

Figure 15d. 3400 RPM, Average Load Torque 39 N·m

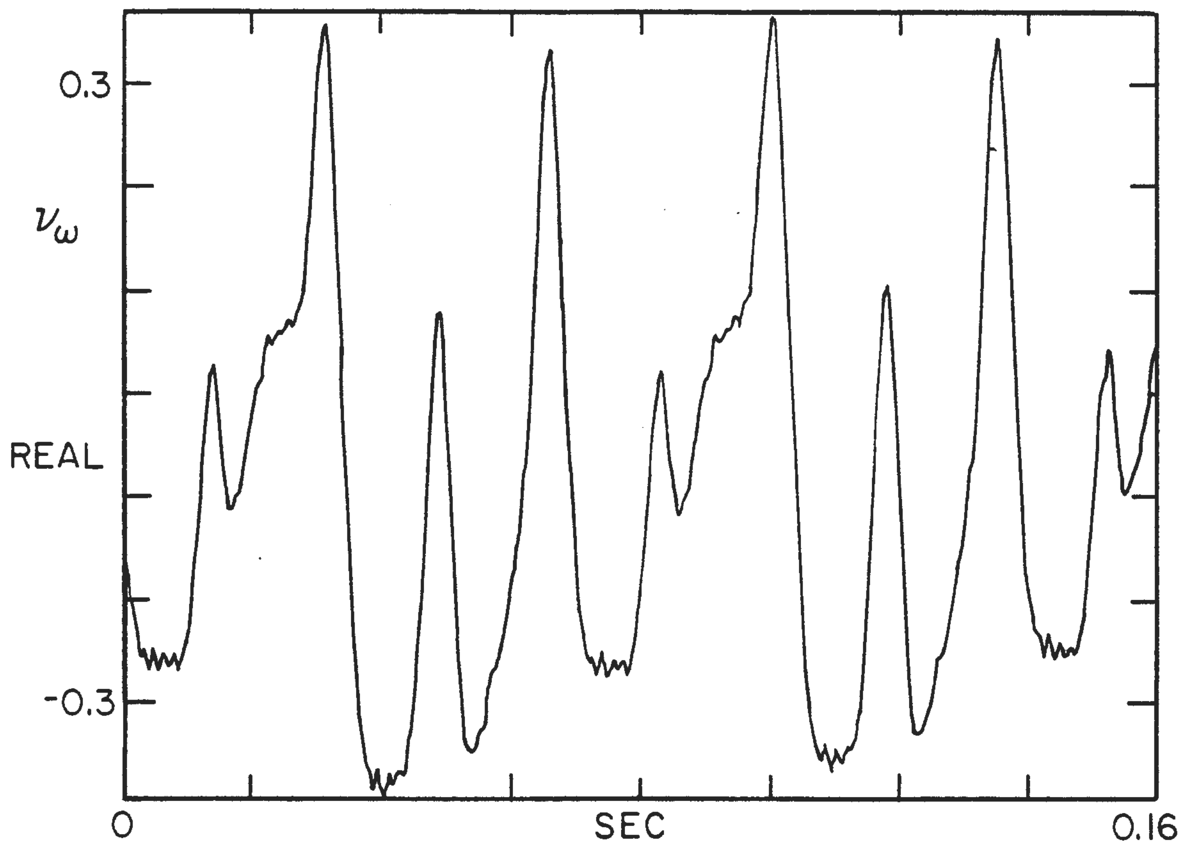

Figure 16a. 1700 RPM, Average Load Torque 32 N·m, 3 cylinder operation.

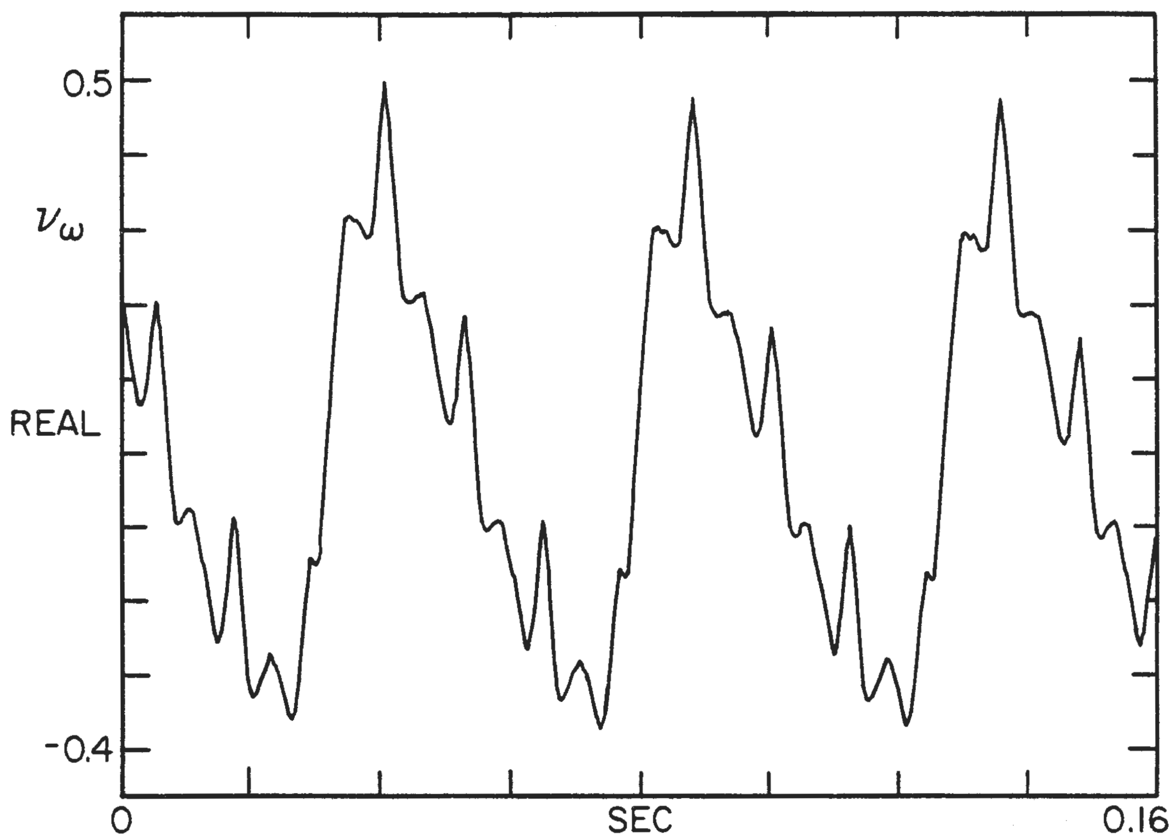

Figure 16b. 2000 RPM, Average Load Torque 38 N·m,
3 cylinder operation.

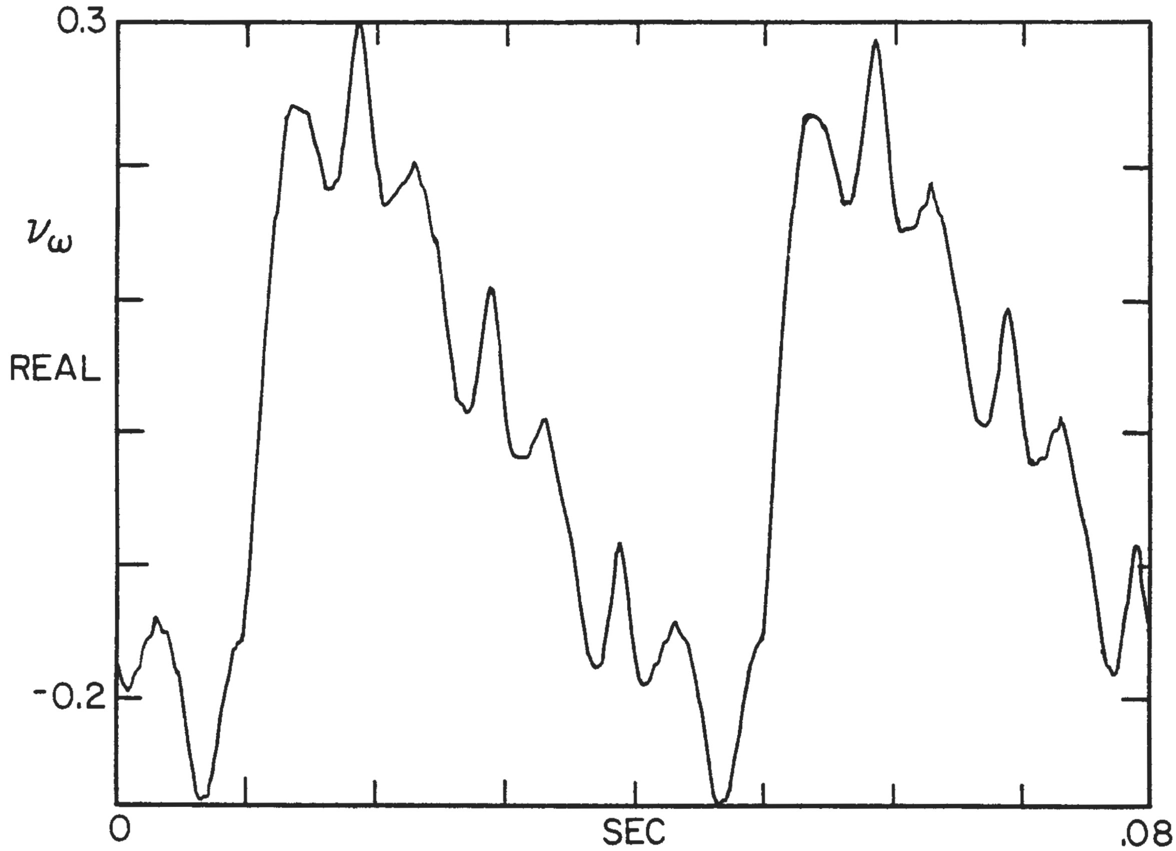

Figure 16c. 3000 RPM, Average Load Torque 28 N·m, 3 cylinder operation.

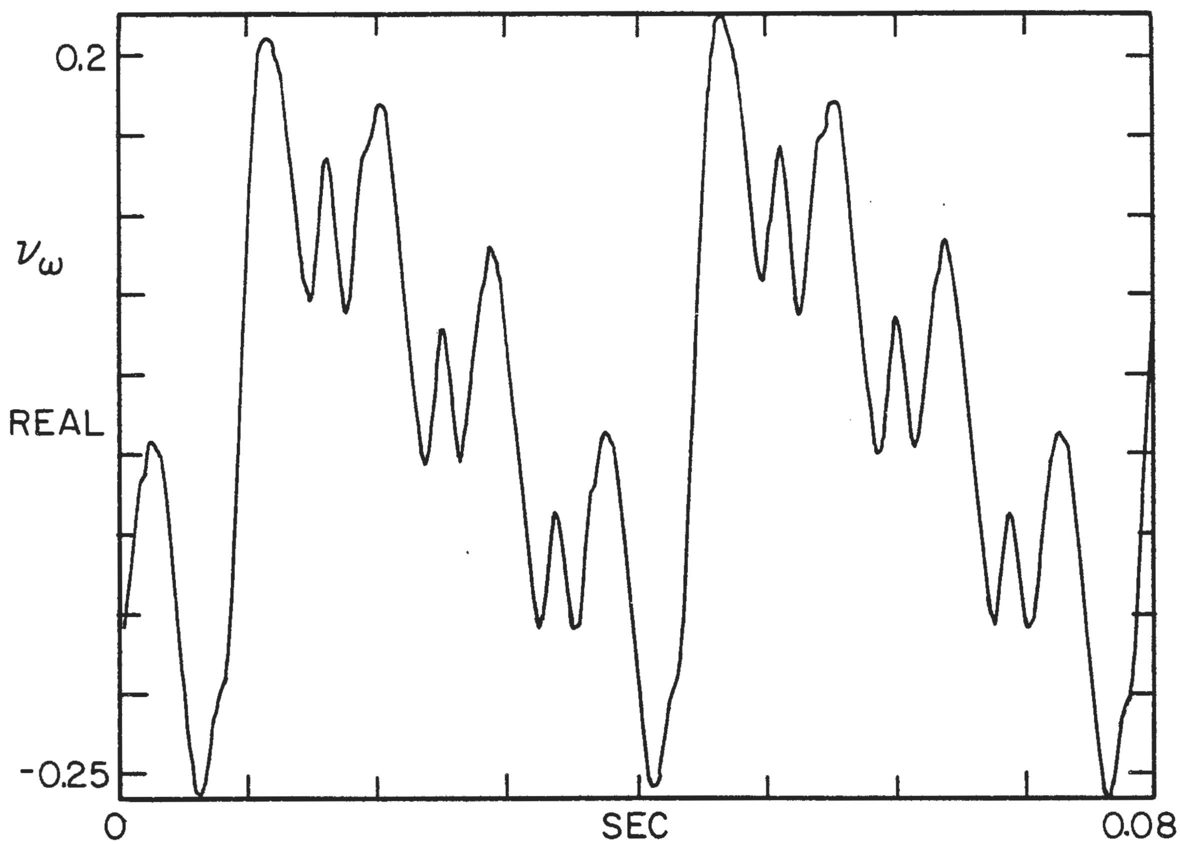

Figure 16d. 3400 RPM, Average Load Torque 30 N·m, 3 cylinder operation.

angular speed under 3 cylinder operation are clearly visible particularly in comparison to normal operation.

The non-uniformity metric for 3 cylinder operation is significantly larger than for normal operation. Figures 17 and 18 present the non-uniformity metric for 3 cylinder and normal operation for each RPM separately and for a variety of load torques.

The relationship between normal and 3 cylinder operation is relatively RPM dependent and relatively independent of load. At low RPM the non-uniformity metric is largely due to the relative maxima associated with cylinder pressure. At high RPM the various torque losses yield a relative maximum which tends to predominate. Nevertheless, the non-uniformity associated with the 3 cylinder operation is clearly evident at each RPM.

The relationship between the non-uniformity metric n for normal and 3 cylinder operation is particularly significant in comparison to the variation in n(k) at each operating condition. The ratio of n for 3 cylinder to normal operation is many times larger than the standard deviation of n(k) at each operating condition. This result suggests that intermediate levels of abnormal non-uniformity between the two extreme conditions can readily be detected. There are several potential applications for the present non-uniformity metric. For example, at each RPM a threshold level of non-uniformity could be established. Abnormal non-uniformity at that RPM corresponds to n(k) exceeding the threshold. This information is, of course, directly applicable for bus maintenance in that it provides a direct measure of degraded performance.

EXPERIMENTAL RESULTS, LOW RPM

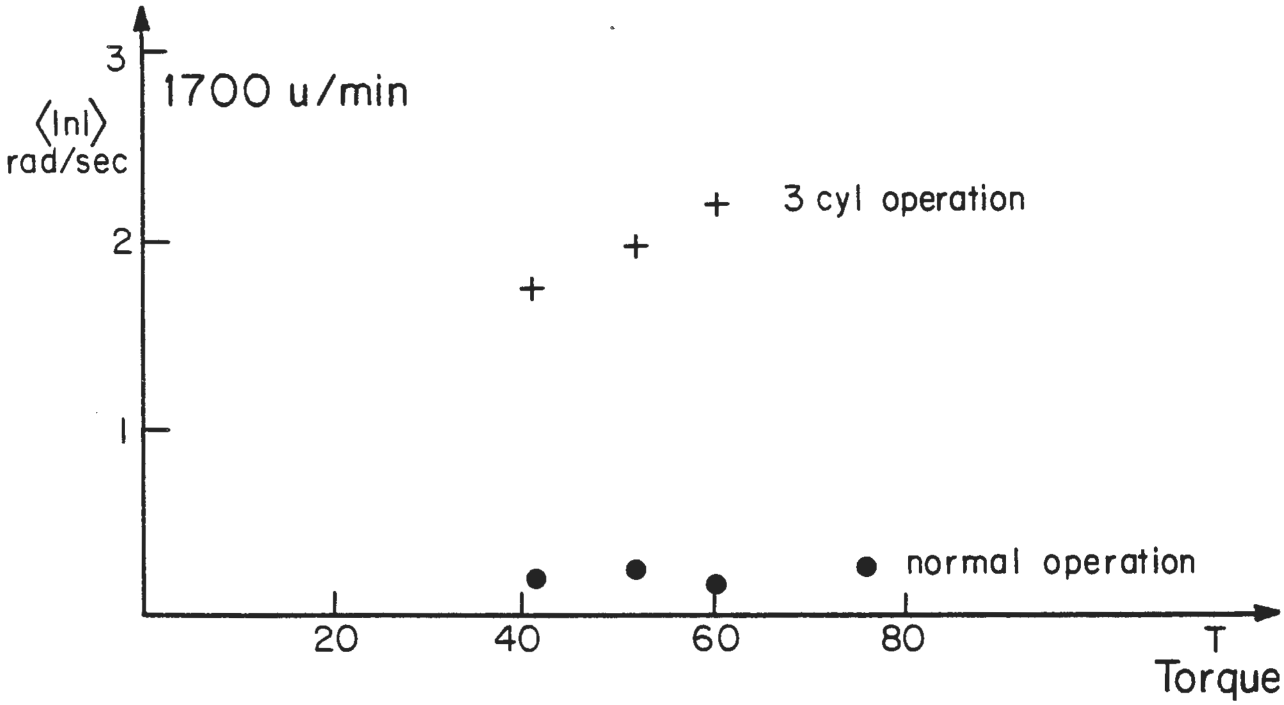

Figure 17a

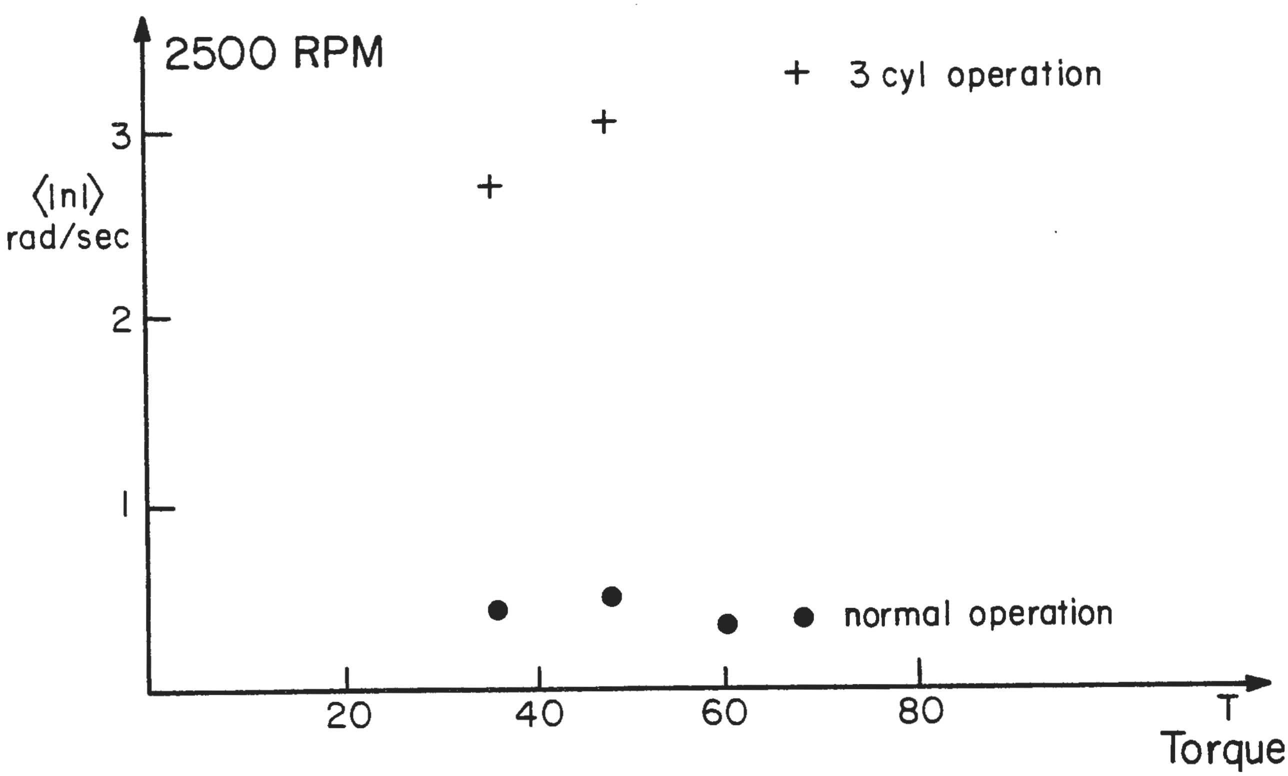

Figure 17b

EXPERIMENTAL RESULTS, HIGH RPM

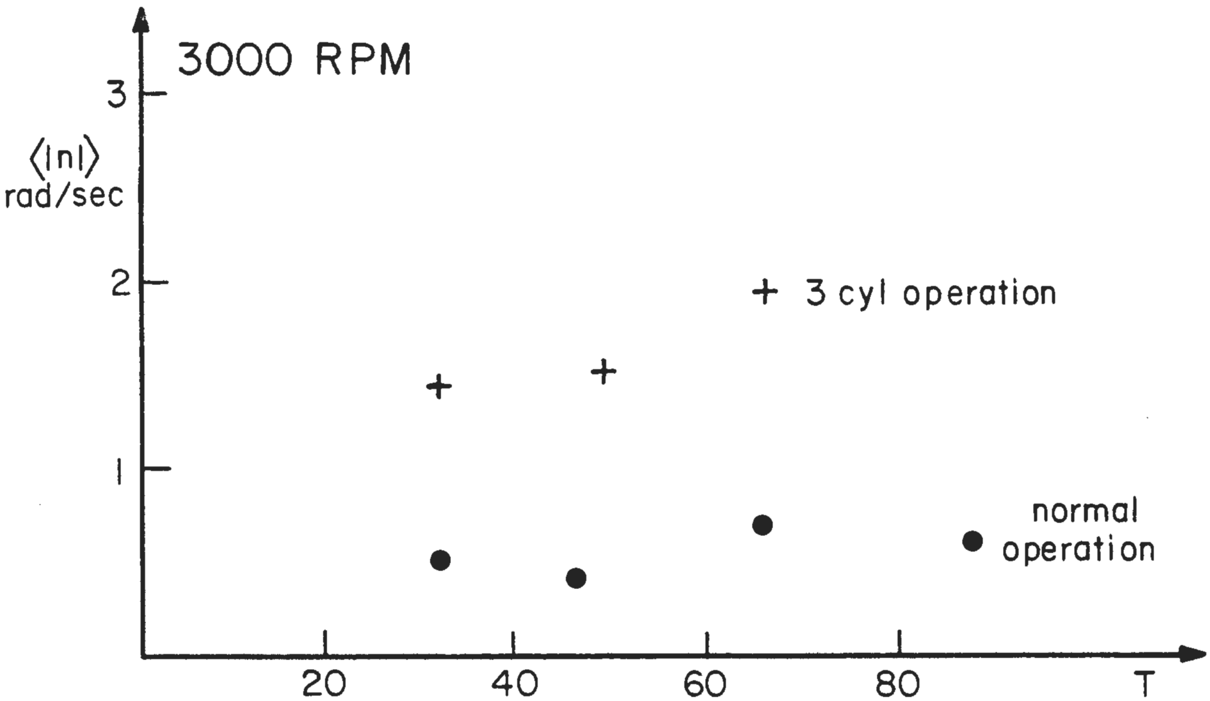

Figure 18a

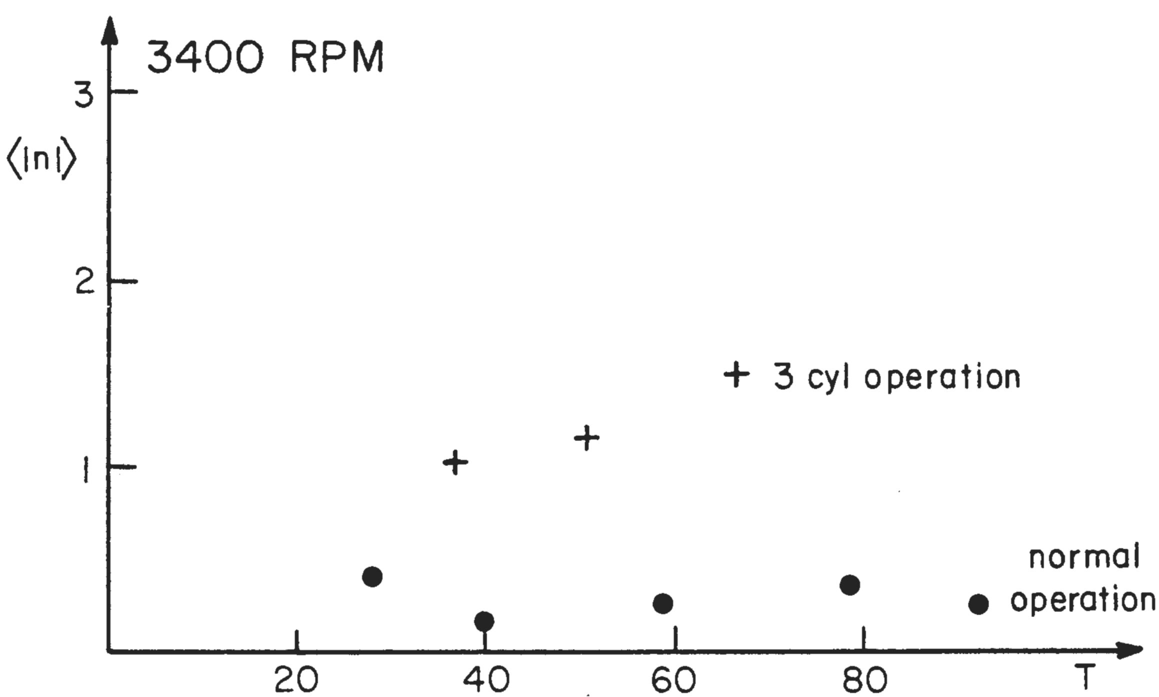

Figure 18b

Furthermore, it is possible to identify an individual malfunctioning cylinder if a separate timing reference is available. In the case of a gasoline fueled, spark-ignited engine, a spark sensor on any cylinder provides the required information. For a Diesel engine an injector pressure sensor could potentially provide this same information.

In addition to experimental studies of this concept as applied to an engine on a test stand, other studies have been made in a vehicle. For these studies a sensor which is identical to that used in the test stand studies has been installed in a VW Passat. This car has a 1.5L Diesel engine, 5 speed manual transmission and is front wheel drive.

The instrumentation for these studies is depicted in figure 19.

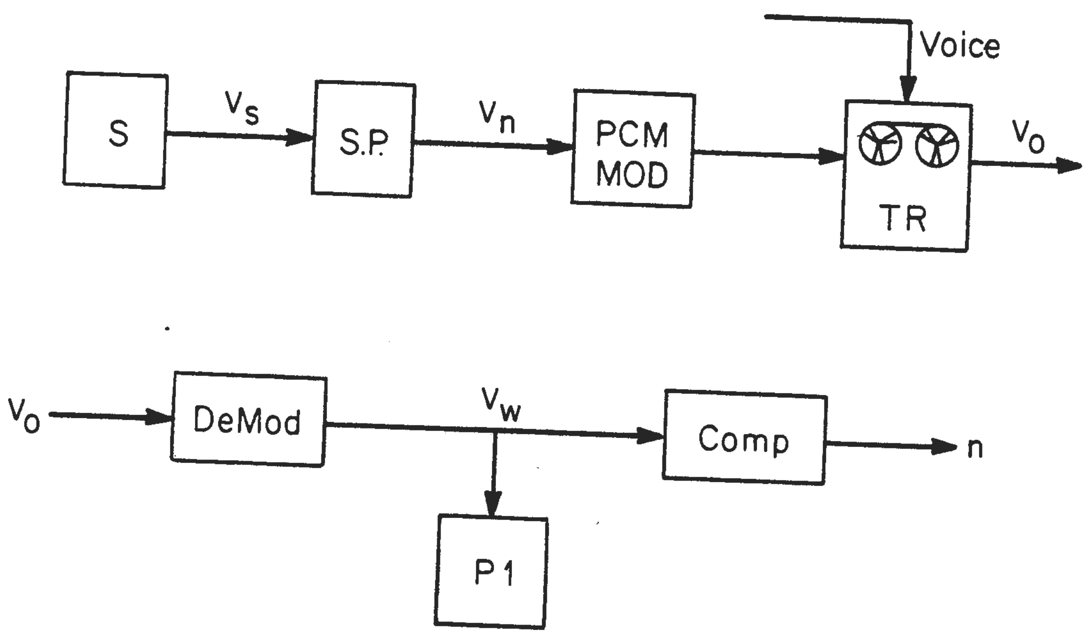

Figure 19

For the vehicular tests the analog signal v_ω was recorded using an instrumentation tape recorder and later replayed for signal processing. Toward this end the electronic signal processing for generating the analog of crankshaft angular speed v_ω was installed in the vehicle and operated from the 12 volt vehicle power supply system. For recording purposes an 8 channel pulse code modulator (PCM MOD) and a 4 channel tape recorder were also installed and configured for 12 volt operation.

The data was recorded for a variety of vehicle operating conditions. Then the recorded data was played back through a PCM demodulator (DeMod) which yielded output $v_\omega(t)$. Several samples of the waveform $v_\omega(t)$ for various operating conditions were plotted using an x-y plotter. In addition, the analog signal was the input for a computer based system from which the non-uniformity metric was obtained.

Experimental studies were conducted with the vehicle on a chassis dynamometer and with the vehicle on various streets. The chassis dynamometer provides an essentially steady load similar to the test stand dynamometer.

On the other hand, street operation introduces random variations in the driveline angular speed. Some of these random variations are coupled through the clutch to the engine. The random variations thereby introduced into the crankshaft angular speed are of the form of an additive random process in the signal being measured. That is, the instantaneous variation in crankshaft angular speed results from torque non-uniformity and from the road induced random process.

One of the issues considered in the present study is the relationship between the non-uniformity of crankshaft angular speed due to the presence of road induced noise and that which is due to the torque non-uniformity. Of course this relationship depends upon road quality.

Experimental studies were conducted on a typical Berlin (W. Germany) road (Straße des 17 Juni), on an inner city expressway (AVUS) and on an old cobblestone street (Altvater Straße). The quality of a typical Berlin street is excellent as is the quality of the AVUS. However, the cobblestone street is sufficiently rough to represent an extreme case of road induced crankshaft angular speed non-uniformity.

It is reasonable to expect that the crankshaft speed non-uniformity would be higher for street operation than for the chassis dynamometer. However, it is reasonable to expect the non-uniformity metric obtained for the cobblestone street would be higher than for the good quality streets. Experimental results support this conjecture.

In addition to the above street operation, the vehicle was operated with one cylinder disabled by disconnecting the fuel line from one fuel injector. Three cylinder operation provides a case of extreme non-uniformity and is a convenient operating reference for interpreting the results obtained in street operation. Such operation probably involves a worse degree of engine roughness than would be tolerated by a motor vehicle operator without obtaining repairs and can be considered at the upper level of engine roughness to be encountered in vehicle use. The relationship between torque non-uniformity for normal operation and 3 cylinder operations is significant for the application of the concept which is presented in this paper.

The measurements on the street were made while operating the vehicle at essentially constant speed. During steady state operation samples of the data were recorded for approximately 2 min. intervals (traffic permitting).

The recorded data were then played back through the demodulator yielding the analog waveform $v_{\omega}(t)$. Samples of the analog waveform were obtained and are presented in figures 20a through 20d for several operating conditions.

It is instructive to compare the waveforms of figures 20a through 20d with the corresponding waveforms of figures 15 and 16. The characteristic shape of the waveform associated with the torque produced by each cylinder firing event is essentially the same for the vehicle engine as for the test stand engine.

The non-uniformity for any given condition is least for the operation on the chassis dynamometer, somewhat greater for the operation on Straße des 17 Juni or AVUS and yet worse for operations on Altvater Str. Note that the non-uniformity is the worst for 3 cylinder operation.

A quantitative study of the non-uniformity of the crankshaft angular speed has been made by computing the average non-uniformity metric n for each operating condition. The computation of this metric is identical to the method which has been explained earlier for the test stand engine studies. The parameter N has been 8 for all of these studies. The number of cycles K varies somewhat with operating condition but was generally in the range from 150 to 400.

Figures 21 and 22 present the results of the study of the vehicle non-uniformity study. Figure 21 is for operation at steady state in 5th gear. Figure 22 is for corresponding studies in the 4th gear.

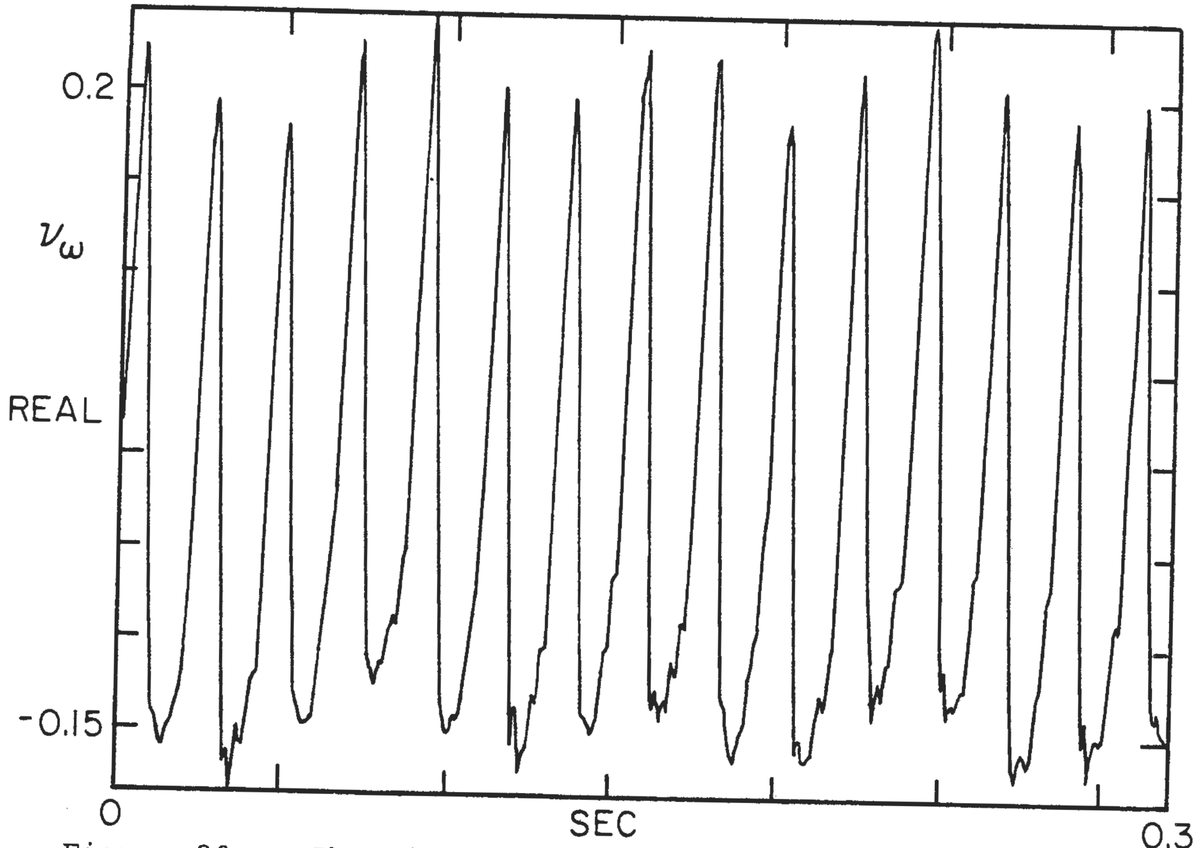

Figure 20a. Chassis dynomometer, 5th gear, 50 km/hr

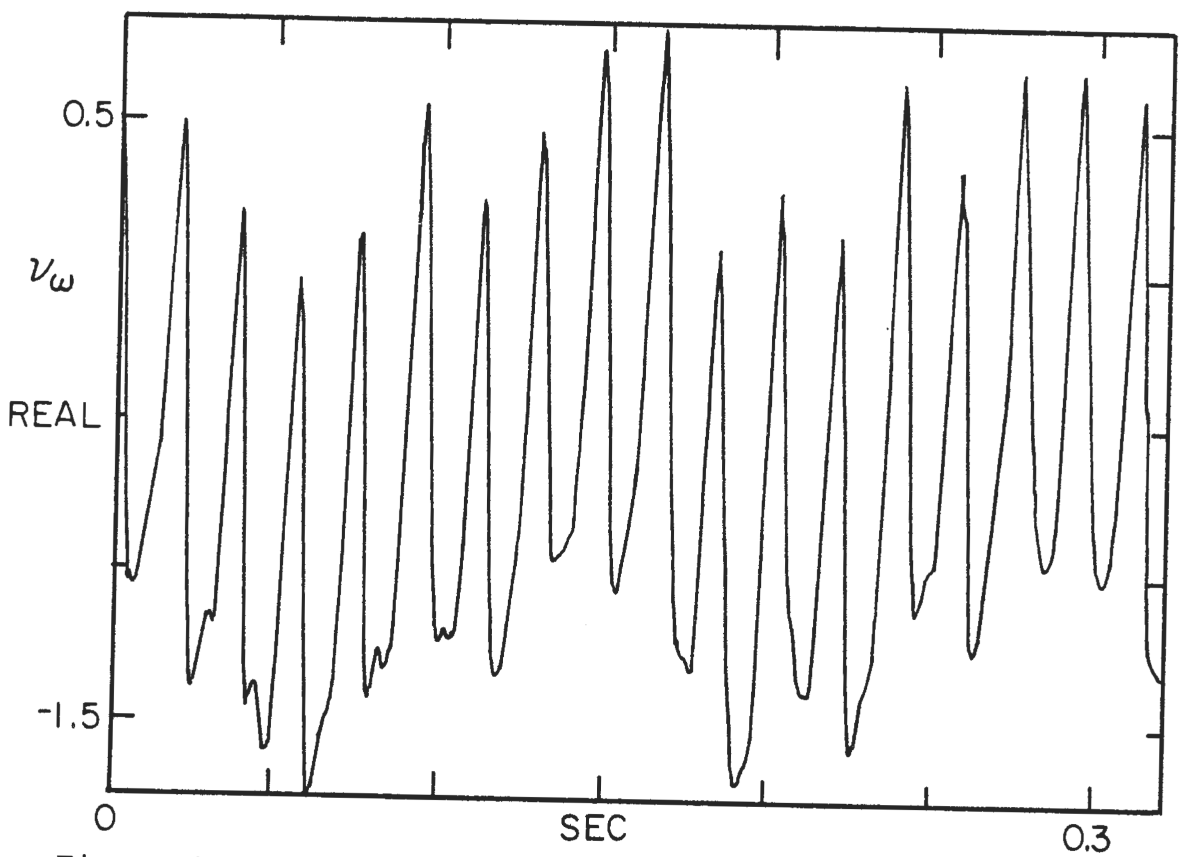

Figure 20b. Str. des 17 Juni, 4th gear, 50 km/hr

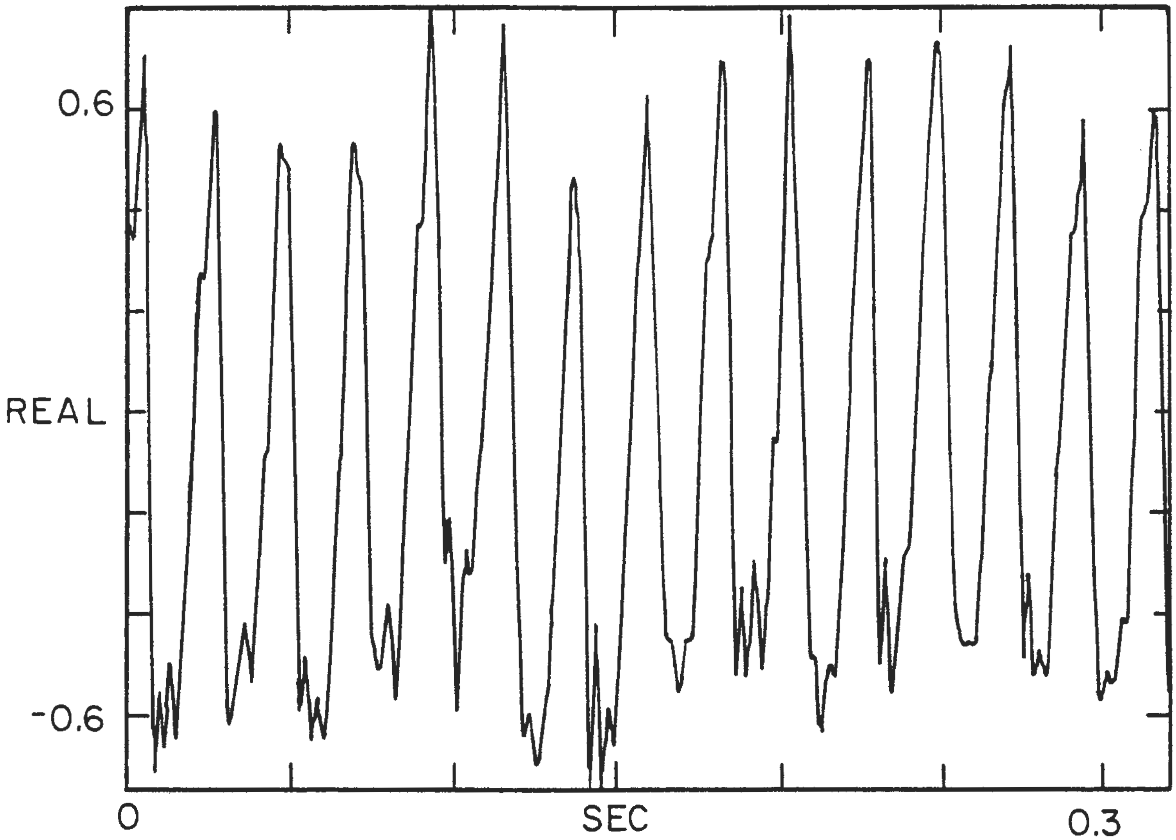

Figure 20c. Altvater Str., 4th gear, 50 km/hr

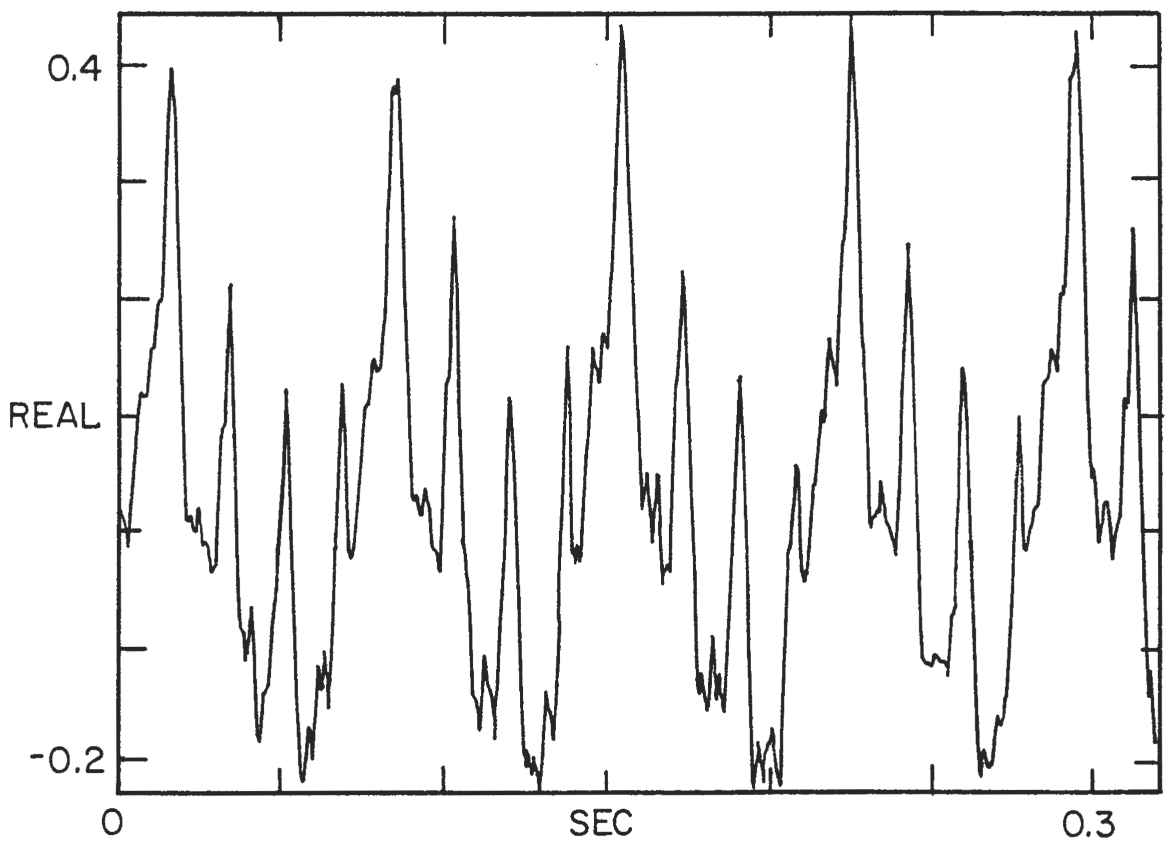

Figure 20d. 3 cylinder operation, 4th gear, 50 km/hr

These latter figures illustrate the influence of road surface upon measured non-uniformity. Generally speaking, the non-uniformity increases with increasing road roughness. However, in none of the normal 4 cylinder road operations is the non-uniformity as large as for 3 cylinder operation.

These results suggest that levels of torque non-uniformity can be measured using the present inexpensive non-contacting method for actual vehicle operation on the street. The concept reported in this paper is applicable for measurement of torque non-uniformity resulting in n greater than about 1 rad/sec. This latter value is significantly less than the value associated with 3 cylinder operation. Thus the present method has potential application for electronic engine control and for system maintenance monitoring.

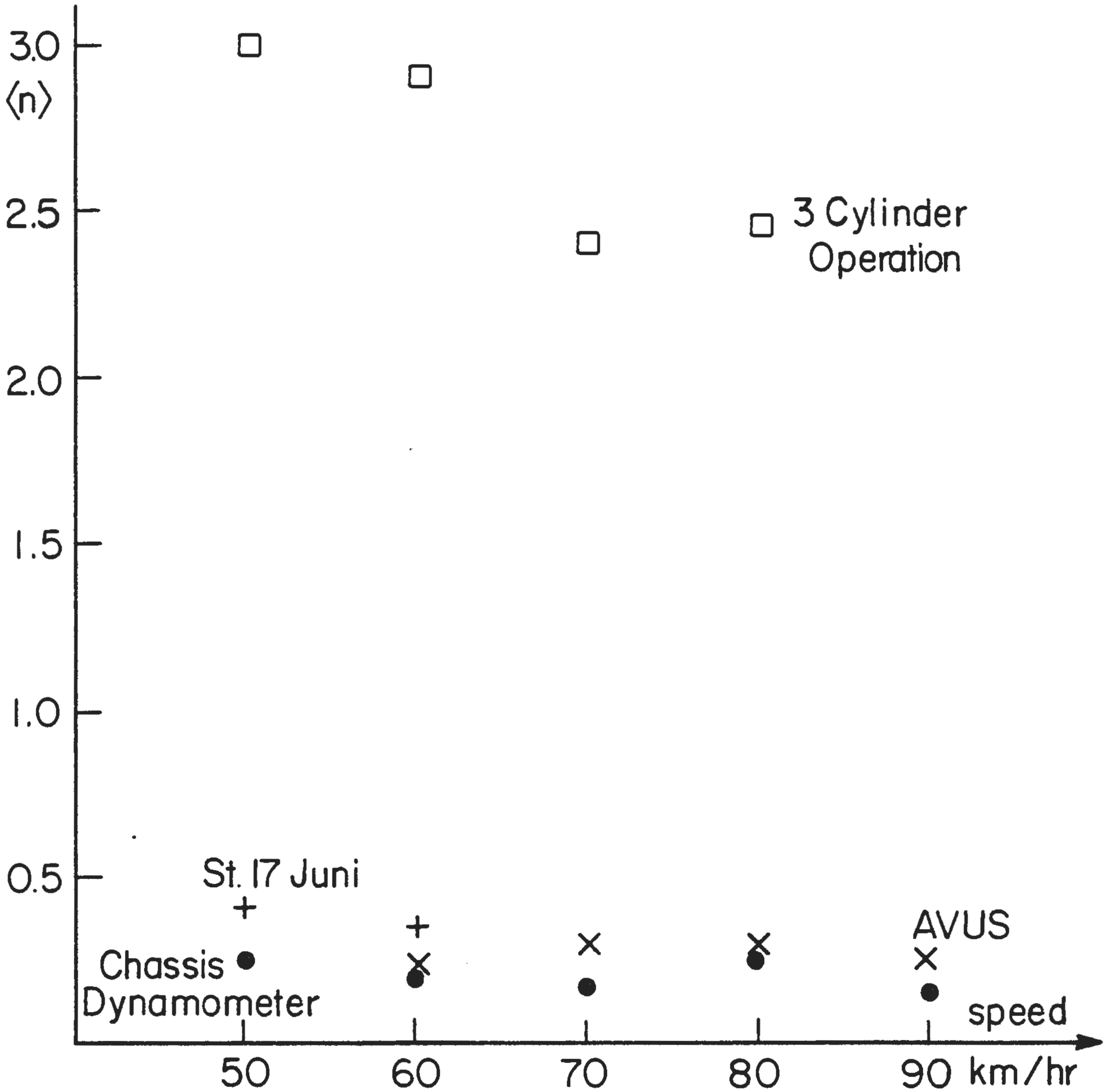

Figure 21. Experimental Road Test Results, 5th Gear.

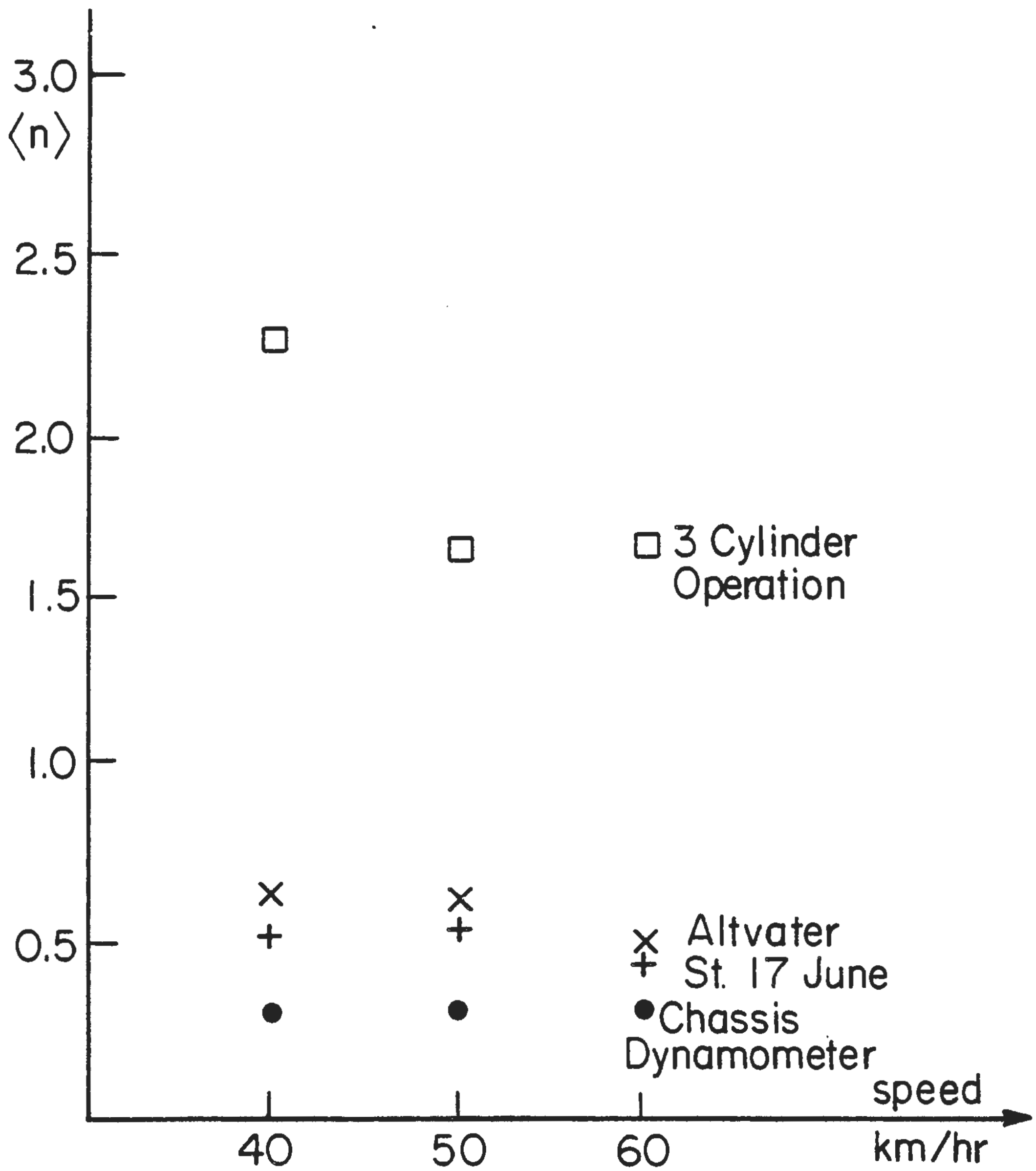

Figure 22. Experimental Road Test Results, 4th Gear.

Summary and Conclusions

This report has presented a method of measuring torque non-uniformity in reciprocating IC engines. It utilizes a relatively inexpensive sensor which can be installed in production engines with minimal modification and relatively inexpensive electronic signal processing. This report has experimentally demonstrated a method of detecting extreme torque non-uniformity resulting from a malfunctioning cylinder. Lower levels of non-uniformity can similarly be detected because of the extreme sensitivity of the present non-uniformity signal.

Interpretation of Results

The preceding results have great significance for failure detection in buses. Measurements of torque non-uniformity are a direct engine performance indicator and can lead to rapid diagnosis of engine component failure.

A system for such diagnosis is depicted in Figure 23.

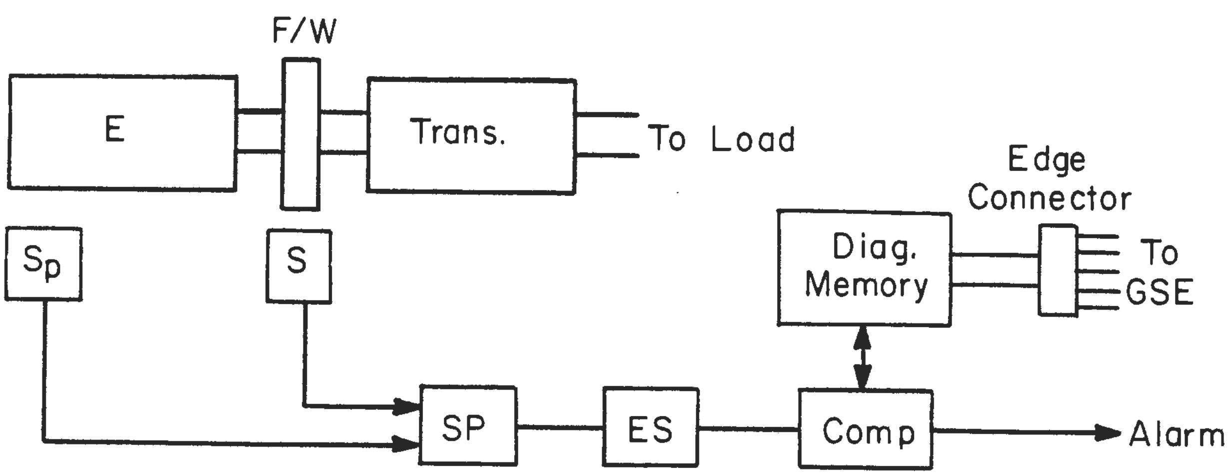

Figure 23

A sensor S measures crankshaft angular speed. The output of the electronic signal processing v_ω is an analog of instantaneous angular speed $\omega_e(t)$. An extremal sampler circuit ES samples v_ω at the relative minimum values of angular speed. The computer calculates the non-uniformity metric n from the sampled relative minima of $\omega_e(t)$. Whenever this non-uniformity exceeds an acceptable level there is a possibility of degraded engine performance. Should this degradation continue for a longer period than is statistically expected for a normal engine then degraded engine performance has occurred. The computer then generates an alarm message which is stored in a non-volatile memory.

At the same time, a sample of the non-uniformity vector $\underline{n}(k)$ can be stored in memory for diagnosis purposes. The affected cylinder(s) can be identified by comparing the time of occurrence of the degraded performance with the time of occurrence of #1 fuel injector operation. A sensor s_p is provided for this purpose. This sensor can be, for example, an injector lift sensor or an injector pressure sensor. The details of this sensing operation are dependent upon the nature of the injection system.

The optimal format for data storage in the diagnostic memory has not yet been determined on this project owing to relatively limited funds and time. However, there are many possibilities which can greatly facilitate detection of incipient failure and diagnosis of component degradation.

Standard signal detection theory can be applied to these probabilities to evolve an optimum strategy for detecting degraded performance.

The concept of the statistical processing of torque non-uniformity metric can be understood from figure 24. This figure depicts the conditional probability density function for the random variable n under two hypotheses H_o and H_1:

$H_o \rightarrow$ normal engine

$H_1 \rightarrow$ degraded performance engine

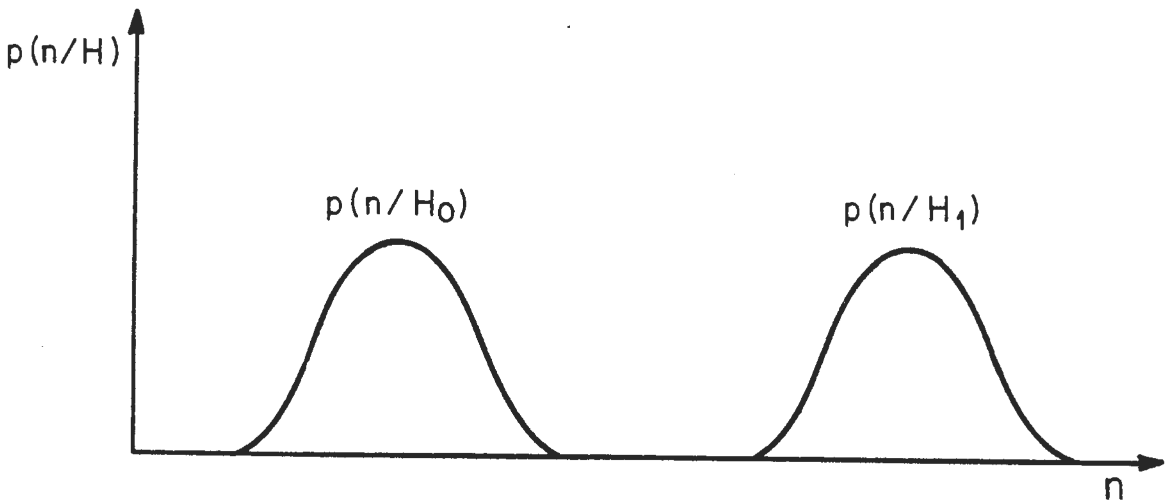

Figure 24

The following definitions apply:

$p_n(n/H_o)$ = probability density function for n under hypothesis H_o

$p_n(n/H_1)$ = probability density function for n under hypothesis H_1.

III. CONCEPT OF DETECTING ENGINE PERFORMANCE DEGRADATION

The present scheme for detecting incipient engine failure is based upon measurements of engine performance. Torque non-uniformity measurements are a significant performance index which are used to determine that degradation has taken place. The torque non-uniformity metric n is the index which is to be used for failure detection.

However, for a normal engine running at a constant RPM the torque non-uniformity metric is a random variable. Consequently statistical means are required to determine degradation. A goal of this project is to find a statistical sampling scheme from which performance degradation can be inferred with a given confidence level.

This determination of an optimum sampling strategy is beyond the scope of the present study owing to the relatively limited funds which were provided. The present project has concentrated on collecting the statistical data from which the required conditional probability density functions $p(n/H_o)$ and $p(n/H_1)$ can be found.

For the purposes of experimentally obtaining those statistical functions measurements were made of the non-uniformity index n(k) for an 8 cylinder gasoline-fueled spark-ignited engine. Unfortunately no diesel engine was available for study for this portion of the project. However, the theory of detecting engine performance degradation from the statistics of the non-uniformity index n applies equally well to gasoline and diesel engines.

Consequently the feasibility of detecting incipient engine failure from measurements of n can be assessed from measurements which are made on a gasoline engine.

The instrumentation for these statistical studies is explained in Appendix A and is similar to that used for the diesel engine studies. The extremal values for $v_\omega(t)$ were obtained and n(k) computed for a sequence of engine cycles k = 1, 2 .. k. Estimates were then obtained of $p(n/H_o)$ and $p(n/H_1)$ using a histogram algorithm in the PDP 11-23 computer system. The degraded engine performance (i.e., H_1 condition) was obtained by disconnecting spark plug leads from the engine. Measurements were made with 8 cylinder (normal operation) and 7 or 6 cylinder operation (simulating degraded performance).

Figures 25 and 26 present the results of measurements of $p(n/H_o)$ and $p(n/H_1)$ for various operating conditions. These figures clearly show that the $p(n/H_1)$ is shifted to significantly higher values than $p(n/H_o)$.

Although 6 or 7 cylinder operation represents an extreme case of degraded engine performance, these measurements establish the feasibility of detecting intermediate levels of degradation using measurements of n(k) and statistical decision theory. The development of optimum statistical sampling and the relationship between sample size and confidence limits in detecting intermediate degradation levels belong to a second phase of this study.

A second phase of this study should involve statistical measurements of n(k) on buses. The instrumentation for such

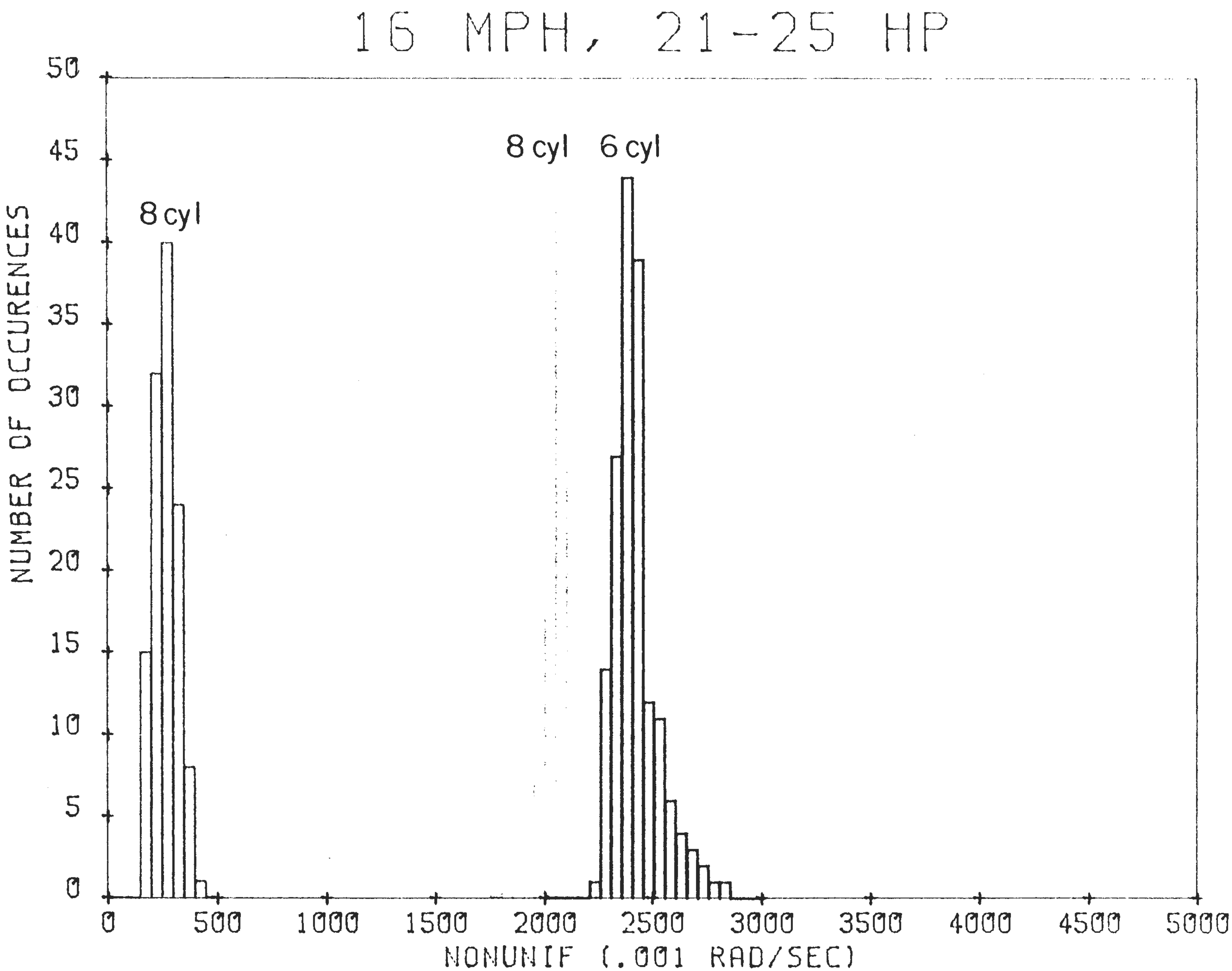
16 MPH, 21-25 HP
NUMBER OF OCCURENCES
NONUNIF (.001 RAD/SEC)
8 cyl
8 cyl
6 cyl
0
5
10
15
20
25
30
35
40
45
50
0
500
1000
1500
2000
2500
3000
3500
4000
4500
5000

Figure 25

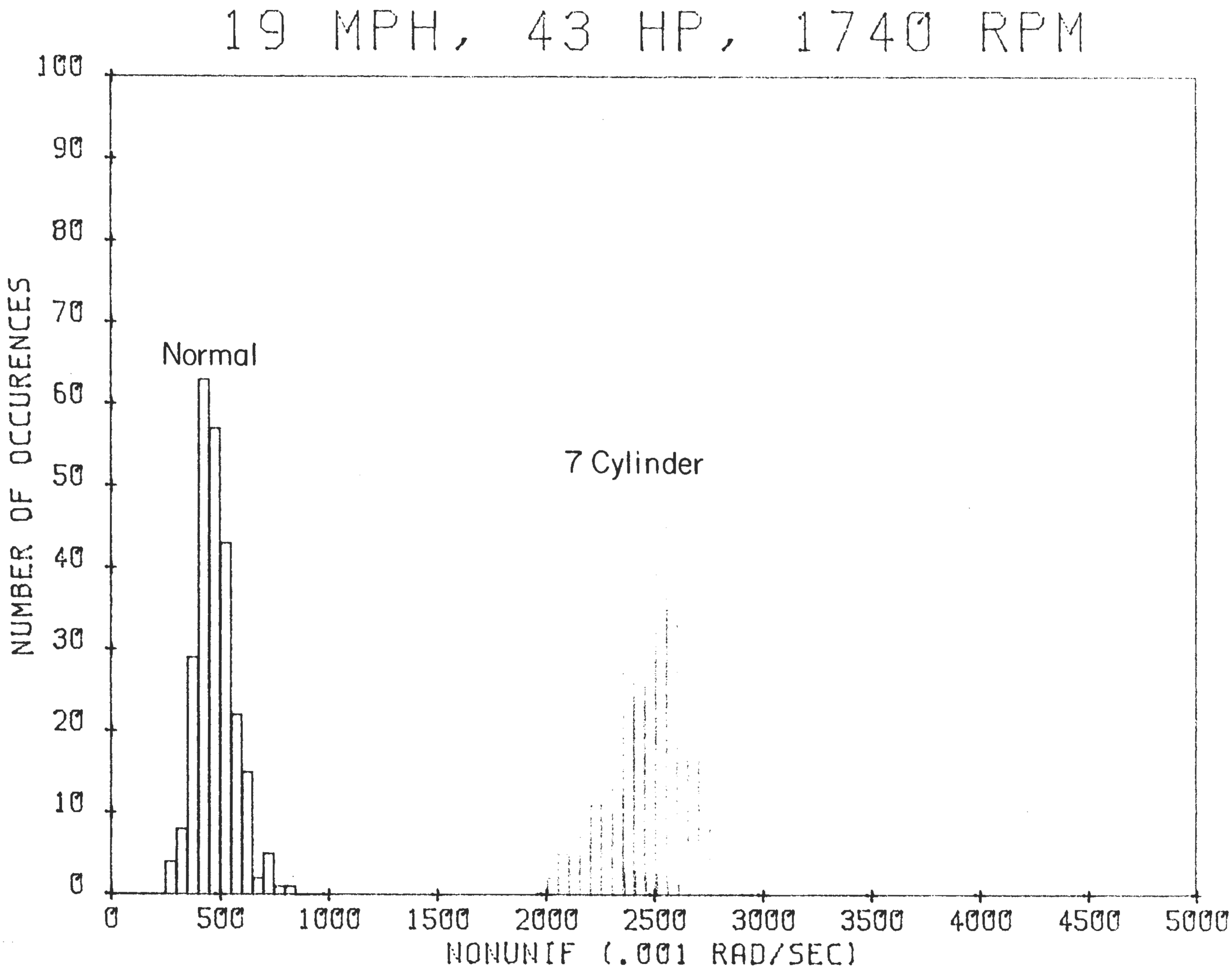
19 MPH, 43 HP, 1740 RPM
NUMBER OF OCCURENCES
0
10
20
30
40
50
60
70
80
90
100
Normal
7 Cylinder
0
500
1000
1500
2000
2500
3000
3500
4000
4500
5000
NONUNIF (.001 RAD/SEC)

Figure 26

studies has been developed and can be installed on buses with minimum modification (typically 4-6 hrs. installation time). The buses should then be operated in normal service for a period of time during which data can be collected to establish a data base from which optimum strategy for detecting degraded performance can be developed. This second phase of the present study should be conducted with cooperation of an existing bus authority.

A continuing degradation in engine performance as represented by a monotonic increase in non-uniformity index indicates potential component failure. Software tests in the computer can easily determine whether a given non-uniformity level occurs on a continuing basis indicating degradation or whether it is a relatively infrequent occurrence which is associated with normal engine operation. This aspect of the detection of incipient failure has not been pursued further owing to limited funds for this project.

Average Brake Torque Measurement

Another important measure of engine performance is the average brake torque. This important variable can also be obtained from crankshaft angular speed measurements. The theory of the present method of measuring average brake torque for a gasoline fueled engine was presented in reference [Ribbens-2). This theory is applicable to diesel engines as well as has been demonstrated in this project.

It can be seen from the theory which was presented in the previous section that the variation in crankshaft angular speed is proportional to the torque variations over each engine cycle at any steady state operating condition. In addition, the average brake torque is proportional to the torque variations. Thus variations in instantaneous crankshaft angular speed are proportional to average brake torque at any constant RPM.

The signal processing for the average brake torque measurements involves calculating the rms value $\tilde{\omega}_e$ of the crankshaft instantaneous angular speed:

$$\tilde{\omega}_e = \left[\frac{1}{T} \int_0^T \omega_e^2(t)\, dt\right]^{\frac{1}{2}}$$

The instrumentation for these measurements is depicted in figure 27.

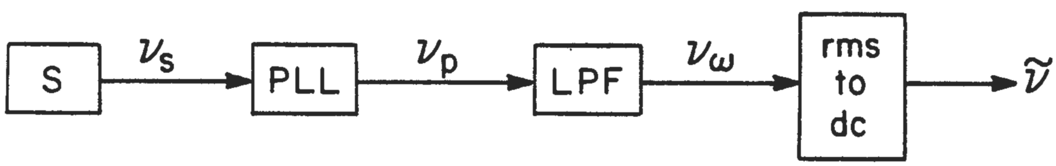

Figure 27

This instrumentation has already been explained up through the generation of the analog v_ω for speed $\omega_e(t)$. The rms value $v_\omega(t)$ is compared to using a commercial rms to dc converter circuit. According to theory $\tilde{v}$ is proportional to average brake torque T_b at any RPM. This conjecture has been tested experimentally.

The experimental measurements were made using a 1.5L 4 cylinder VW Diesel engine driving a water brake dynamometer. This experimental configuration is identical to that reported above for torque non-uniformity measurements.

The results of measurements of $\tilde{v}_\omega$ vs T_b are presented in figure 28.

Results

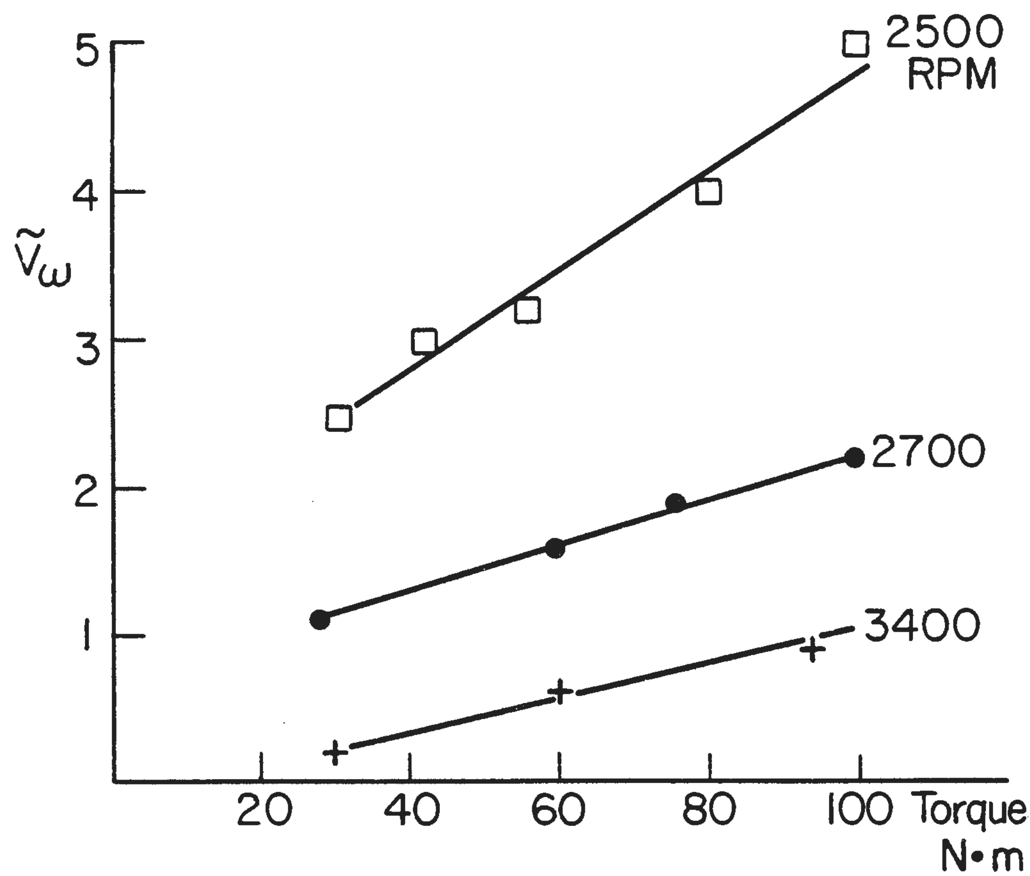

Figure 28. Experimental results for torque estimate.

It can easily be seen from figure 28 that the conditional probability density functions for hypotheses H_0 and H_1 are widely separated. The detection of hypothesis H_1 is straightforward for severe degradation which is associated with 3 cylinder operation.

Furthermore, detection of intermediate degradation between normal operation and 3 cylinder operation is possible. The relative positions of $p(n/H_0)$ and $p(n/H_1)$ depend directly upon the level of degradation. For smaller degradation than that which is associated with 3 cylinder operation $p(n/H_1)$ will be displaced to larger values of n than $p(n/H\ )$.

Thus, statistical processing of samples of n can detect degraded performance at lower levels than 3 cylinder operation. The determination of levels of degradation which can be detected through optimal statistical signal processing is beyond the scope of the present project.

It is recommended that a second phase of this project be funded which can experimentally determine the minimum levels of degradation which can be detected. This study can also explore confidence limits which are associated with given statistical sampling of the random variable n.

IV. BRAKES

Although the major emphasis of this project has been on engine/drive-train components, there are other bus components for which the present concept of detecting incipient failure is applicable. These include the brakes, transmission and possibly the suspension systems. The present concept of detecting incipient failures is to measure system or subsystem performance directly. Any degradation in performance serves as an indication of incipient failure.

One of the primary measures of braking is a quantity which we have termed the braking efficiency. Essentially the braking efficiency expresses the relationship between braking force and hydraulic line pressure. For the purposes of the present discussion we assume a linearized relationship of the form:

$$T = \eta p$$

where

T = braking torque

p = hydraulic pressure

η = efficiency

The parameter which we are calling braking efficiency is determined by the shoe-factor and geometry of the brake system. The linear relationship is assumed for convenience in the present discussion but is unimportant for the validity of the concept.

The braking torque acts on the wheels causing the vehicle to decelerate within the validity of our linear model. This braking force is proportional to the pressure p which is controlled by the operator foot force on the brake pedal. Normally, the hydraulic pressure is augmented by some sort of boost system (e.g. vacuum operated cylinder).

The braking efficiency is determined by several factors including:

- configuration of brake system
- condition of brake linings and wheel cylinder (drum brakes)
- brake temperature
- foreign substances on brake lining.

As the brakes are used the braking efficiency decreases due to wear and the resulting change in brake shoe surface condition and geometry. At some point the braking efficiency is sufficiently low that it is desirable to replace the brake linings and resurface the brake drums.

A corresponding degradation occurs with disk brakes. In this case the brake pad surface deteriorates through wear and it is desirable to replace them.

Normally the braking efficiency remains essentially constant for the life of the brakes. However, as the linings wear and accumulate metal particles and other foreign material near the end of the period in which the brakes are functional, there is a relatively sharp decrease in η.

This project proposes the braking efficiency as the appropriate performance measure for the brakes. This quantity is to be continually measured by the on-board instrumentation and examined for degradation. Whenever degradation in η occurs for any of the vehicle brakes, a suitable warning message is issued to the maintenance/diagnostic memory that it is time to replace the affected brakes. Unfortunately, this concept could not be tested on the present project owing to the relatively limited funds. Moreover, that database from which a computer model could be constructed does not appear to exist.

Braking efficiency can be determined from measurements of braking torque and hydraulic pressure. Then, the braking efficiency can be computed:

$$\eta = \frac{T}{p}$$

This computation can be accomplished using the onboard microcomputer.

The hydraulic pressure measurement can be accomplished using a commercial pressure sensor. Of course the dynamic range for this sensor must include the range of pressures which are encountered in the hydraulic brake lines. Moreover, the installation of a pressure sensor must not compromise the safety of the brake system in the event of a pressure sensor failure.

One such sensor is depicted in figure 29. This sensor consists of a section of hydraulic line to which a pair of strain gauges have been attached. The sensitive axis S for each gauge is shown on figure 29.

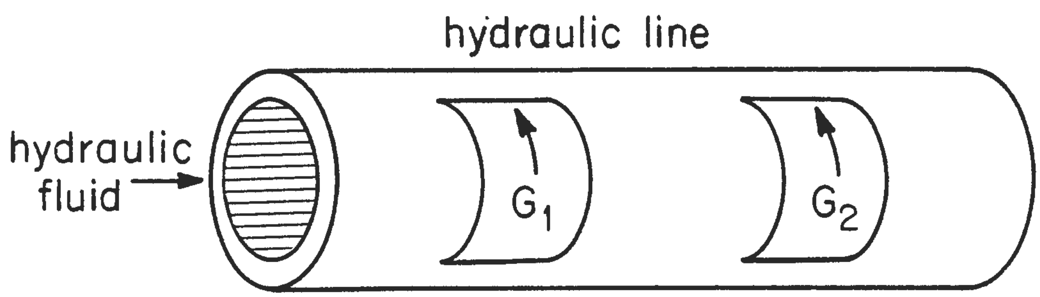

Figure 29

An increase in pressure p causes a proportional increase ΔC in the circumference C of the hydraulic line. This increase in circumference is equivalent to a strain ε along the circumferential direction (i.e., along strain gauge sensitive axis):

$$\varepsilon = \frac{\Delta C}{C} = kp$$

The resistance of each strain gauge varies linearly with strain

$$R(\varepsilon) = R_o[1 + G\varepsilon]$$

where

R_o = zero strain resistance

G = gauge factor for strain gauge

The strain gauges are connected into a Wheatstone bridge as depicted in figure 30.

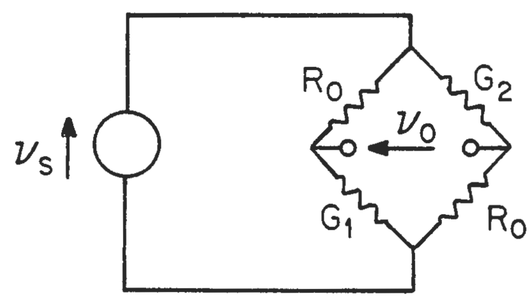

Figure 30

In addition to the two gauges (G_1 and G_2) a pair of precision resistors having resistance R_o complete this circuit. The open circuit voltage v_o is linear in strain ε and correspondingly linear in pressure:

$$v_o = v_S G k p$$

where

v_S = supply voltage (regulated).

Thus, this configuration generates a linear electrical analog of hydraulic pressure and provides a technique for measuring p.

The technique for measuring braking torque also involves strain gauges. For convenience we presume that the bus is equipped with disk brakes. The location of the strain gauges is depicted in figure 31.

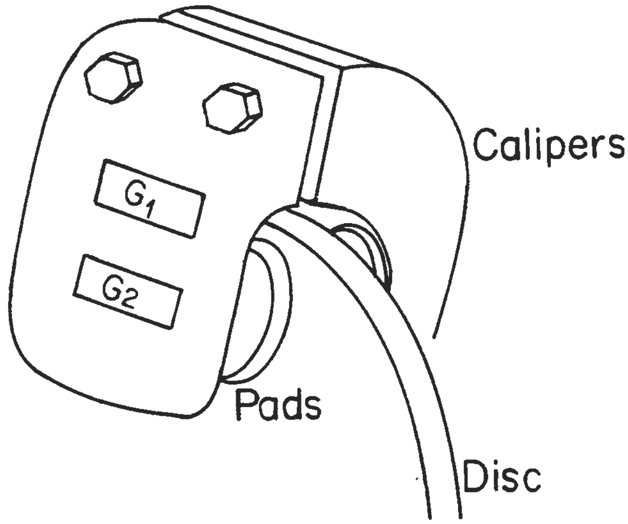

Figure 31

The strain gauges G_1 and G_2 are attached to the calipers of the disk brake system. As the brakes are applied the braking torque is transmitted through the calipers to the vehicle frame. The internal stresses in the calipers which result from braking torque cause proportional strain. The strain at locations 1 and 2 are given respectively by

$$\varepsilon_1 = k_1 T$$
$$\varepsilon_2 = k_2 T$$

These strain gauges are connected in a Wheatstone bridge circuit as shown in Figure 30. The open circuit voltage v_o is given by

$$v_o = v_S G(k_1 + k_2)T$$

Thus this configuration provides a measurement of braking force.

The complete instrumentation (block diagram) for determining braking efficiency is shown in figure 32.

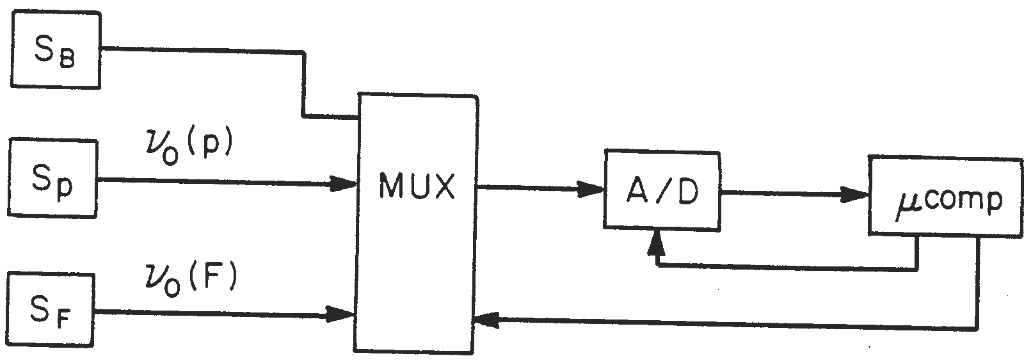

Figure 32

In this figure the data from the two sensors for measuring p and T is multiplexed (mux) and converted to digital format (A/D converter) and then is sent to the onboard computer. The computer can then determine η using ordinary computer arithmetic.

The degradation in η is sufficiently slow that extremely low sampling rates can be used. However, measurements of T and p can only be made when brakes are applied. This condition is sensed by the operation of the brake light switch.

Each time the brakes are applied a measurement of η can be made. The computer can compare the measured value for η with the normal expected value. Whenever η is out at allowed tolerance limits a warning message can be sent to the diagnostic memory. During routine reading of this memory the brake warning message is available to maintenance.

V. TRANSMISSION

Owing to the limited funds available for this project there was no opportunity to specifically assess the feasibility of performance monitoring of transmission variables. However, a concept for transmission performance monitoring has evolved.

A very high percentage of buses are equipped with automatic transmissions. Consequently the present project has concentrated only upon automatic transmission performance monitoring.

One of the important performance variables for an automatic transmission is torque converter slip. Slip S which is defined:

$$S = \frac{\omega_i - \omega_o}{\omega_i}$$

ω_i = driving element angular speed

ω_o = driven element angular speed.

The slip for a normal torque converter is a known function of load and speed. Slip can be measured for a transmission using a pair of sensors such as were used for the crankshaft angular speed measurement. That is, one sensor will measure ω_i and another will measure ω . Slip is then computed in the onboard bus digital processor.

The digital processor maintains a table of values of slip as a function of engine torque and speed. The bus diagnostic computer has instantaneous measurements of torque and crankshaft speed. The digital processor can determine the normal level of slip from the look-up table based upon known torque and speed. This level of slip can then be compared with measured slip.

Whenever slip deviates excessively from the normal value for any given operating condition, transmission performance is degraded. Degraded performance is an indication of incipient failure and a warning message should be generated.

It is recommended that the transmission performance monitoring concept be explored experimentally in a second phase of the present project. The instrumentation for transmission performance monitoring is already in existence. Hence, the feasibility of the above concept can be assessed at relatively low cost.

VI. SUMMARY AND CONCLUSIONS

This report has presented the results of a study concerning the feasibility of detecting incipient failure in various bus components. The concept for this study is based upon continuous monitoring of the performance of the various bus systems and subsystems.

This study has also developed relatively low cost instrumentation for the purpose of monitoring performance of selected subsystems. This instrumentation can be installed with negligible bus modification in a relatively short time.

This study has experimentally demonstrated the feasibility of performance monitoring an engine/drivetrain.

Statistical processing of the measured variable has firmly established the feasibility of detecting degraded engine performance which provides the necessary basis for detecting incipient engine failure.

In addition instrumentation has been described which has potential for detecting incipient brake failure through measurements of "braking effectiveness". The extension of this study to experimental tests and to performance monitoring of other subsystems was limited by available funds for this project.

The present study has established the feasibility of low-cost performance monitoring. A second phase of this study is recommended in which performance monitoring instrumentation is installed on buses. A data base could then be established of subsystem performance for buses in actual service. This data base would provide the framework for establishing an optimal maintenance strategy given the performance measurements for various subsystems.

APPENDIX A

Instrumentation for Histogram Studies of ncb

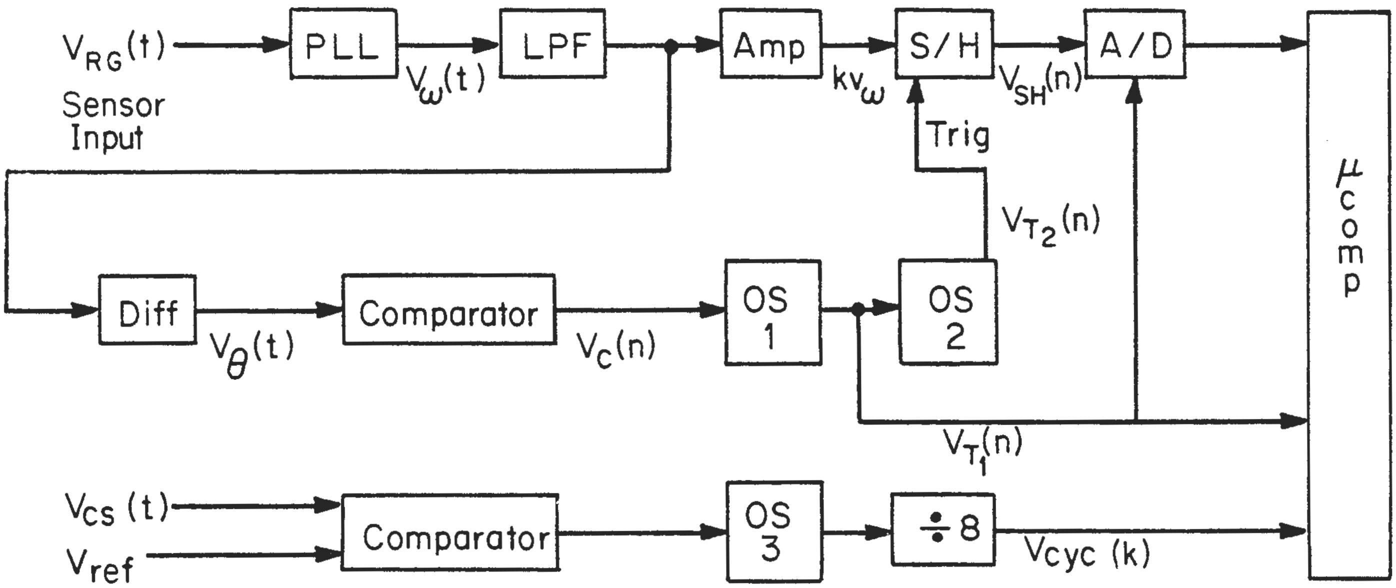

Figure A-1

The instrumentation for our measurements of velocity minima is illustrated in the block diagram of Figure A -1. A magnetic sensor is mounted near the ring gear of an Oldsmobile Vista Cruiser engine. The instantaneous frequency of the output, $v_{RG}(t)$, is the frequency with whcih the gear teeth pass the sensor. The phase-locked loop, PLL, frequency demondulates this signal, yielding an analog, $v_\omega(t)$, of the instantaneous angular velocity of the engine crankshaft. A low pass filler LPF suppresses undesirable frequency compontnts which are generated by the PLL. The desirable PLL output component (i.e., the analog of crankshaff angular speed) is passed by the filler with negligible attenuation. Because of the nature of the phase-locked loop, $v_{\omega(t)}$ is inverted. The signal is then differentiated to give accelerattion, v_a (t), which is compared with zero to find the relative extrema of the velocity signal. The monostable multivibrators (OS, and OS_2) generate short duration pulses at the signal relative maxima. These trigger the sample and hold circuit, S/H, and the A/D converler, so that the computer stores the signal values at the signal maxima (the inverted minima of the true velocity). The signal is amplified in order to make to maximum use of the range of the A/D converter.

A once per-cycle signal, Vcyc (k), is obtained from another magnetic sensor. This sensor picks up pulses from four metal pieces mounted on the crankshaft. The sensor output, $v_{cs}(t)$, is compared to a reference voltage, and the comparator output triggers a third monostable multivibrator. The resulting pulges, four per crankshaft revolution, are divided by eight to give one trigger pulse, Vcyc (k), per cycle of two revolutions (eight cylinders).

The computer checks for the proper number of V_{T_1} (n) triggers per cycle - eight for normal operation, seven with one sparle plug disable - and stores the A/D value for each trigger pulse. Figure A-2 is a copy of oscilloscope photographs of some representative wave forms. The continuous waveform is the analog ($v_\omega(t)$) waveform and the step wise continuous waveform is the sample relative maxima.

Although there is a possibility of errors in detecting relative maxima, experimental measurements have yielded no errors for sampling intervals involving over 200 engine cycles.

The computer then calculates $\underline{n}(k)$ and non uniformity index n (k) as explained in the body of the report. This non-uniformity index is computed once for each engine cycle. A commercial computer program (available from DEC) is used to find the higtogram for the random variable n(k).

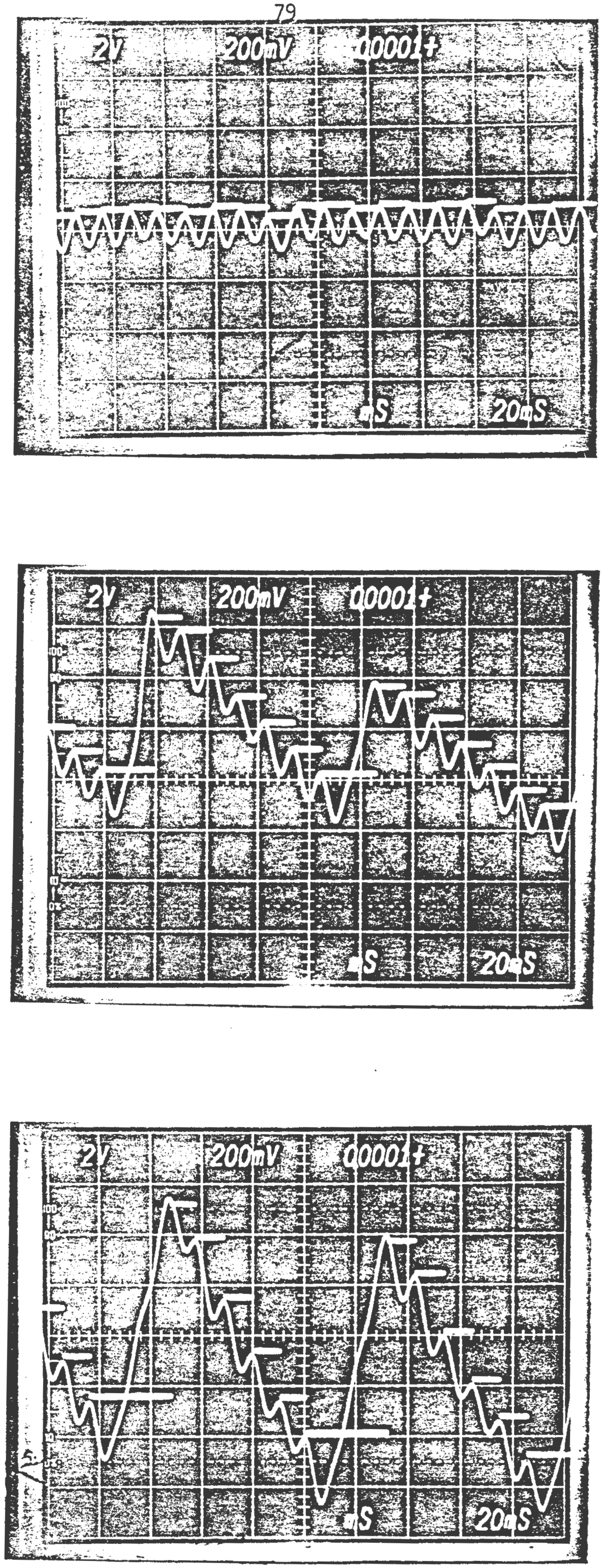
2V
200mV
0.0001 t
mS
20mS
2V
200mV
0.0001 t
mS
20mS
2V
200mV
0.0001 t
mS
20mS

Figure A-2

REFERENCES

1. Ribbens, W.B. "Experimental Road Test of a Noncontacting Method of Measuring I-C Engine Torque Nonuniformity," SAE International Congress and Exposition, Cobo Hall, Detroit, Michigan February 25 - March 1, 1985. SAE No. 850454

2. Ribbens, W.B. "A Noncontacting Torque Sensor for the Internal Combustion Engine," SAE International Congress and Exposition, Cobo Hall, Detroit, Michigan February 23-27, 1981. SAE No. 810155