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A Vibration Test Stand for a Thermal Systems Laboratory*

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This report documents our experience developing a novel test stand for a thermal systems laboratory. The laboratory is the Texaco Energy Systems Lab at the Mechanical Engineering Department of Penn State. This lab was conceived to help engineering students appraise the integrating nature of the design activity. Among other test stands, the lab has an instrumented IC engine. The level of instrumentation allows an energy balance on the engine, and computation of magnitudes such as volumetric and thermal efficiency. Vibration elimination or absorption has important consequences for product life and appeal. Thus, an identical engine, externally driven, was mounted on a specially designed base of variable stiffness. In this engine, valve and piston displacement can be monitored via accelerometers. The base displacement as function of time can be captured via an LVDT sensor. Analysis of the data allows determination of two dominant frequencies, and of the corresponding phase angle.

INTRODUCTION

DESIGN is an integrating activity, with team members contributing their talents under the direction of insightful professionals to achieve a given end. Hence, a multidisciplinary perspective, coupled with integration of knowledge throughout the curriculum, is deemed a desirable trait of engineering education [1]. One of the objectives of the Texaco Energy Systems Lab is to convey to the students that most thermal systems are a successful embodiment of the cycles studied in the classroom. By analyzing different aspects of each design, the students can gain a more vivid understanding of the utility and applications of many of the methods presented in the curriculum.

In addition to the vibration test stand subject of this paper, the lab currently has an IC engine, a steam engine, a refrigeration unit test stand, a gas turbine display stand, and a virtual energy lab (VEL) [2, 3]. The location of the components relative to each other is shown in Fig. 1. Both engines can be connected to a dynamometer to measure speed and torque. In addition to the test stands, a small chiller was designed and built by students to fulfill the dual purpose of removing waste heat from the room (1.5 kW), and of measuring overall heat transfer coefficients of an air-to-water finned coil. The vibration test stand and the gas turbine display (GT) can be rolled in and out of the room, and hence the GT is not shown. A complete description of the lab is given in [4], and a complete description of the VEL in [2].

A great deal of engineering effort is spent in minimizing IC engine vibration and also in absorbing or neutralizing the motion that otherwise would be transferred to its supports. Also, the relative timing of valve and piston motion is of crucial importance to engine efficiency. Highlighting these two aspects of engine design while teaching our dynamics course seems like a worthwhile way to enhance the curriculum. Hence, a test stand with the capabilities of monitoring piston and valve motion, and engine base displacement was developed. Undergraduate students taking a course on dynamics and vibrations used the test stand. Both the test stand and the experience of the students are described here.

VIBRATION TEST STAND

To realize compatibility with the rest of the Lab, the test stand was designed along the following criteria:

1. The subject engine (Robin EY08) is similar to the engine that will be analyzed thermally in subsequent courses.
2. The lab must be completed in one hour, although students are welcome to return in other time slots, if desired.
3. Teams of two are required to complete the lab.
4. The lab must show one or two fundamental aspects that can be readily related to classroom material.

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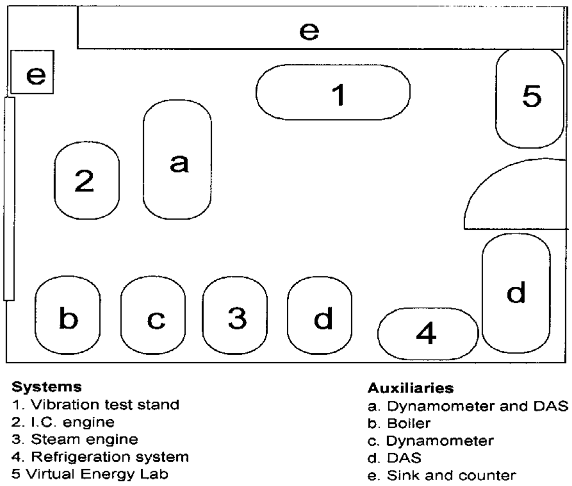

Fig. 1. Laboratory layout, with components.

5. Conceptual understanding of the phenomena at hand is crucial. Tedious or laborious computation that can obscure the issues must be avoided, relying on computer programs written by the designer to complete calculations.
6. Due to space constraints, the system must be self-contained and mobile, the only utility required being electrical power.
7. The system must be safe, reliable and easy to operate.

Consistent with those criteria, the test stand of Fig. 2. was designed and built.

From left to right, the stand has a personal computer (PC), signal processors and power supplies for instrumentation, a single cylinder IC engine mounted on a base of variable stiffness, and a variable speed drive for the engine. A printer, part of the PC subsystem, can be seen in the lower shelf along with the CPU case. For descriptive purposes, we will divide the stand into the engine/drive system, and the instrumentation/PC system.

Engine/drive system

It is generally known that adjusting the stiffness of the base of a vibrating object will alter the amplitude of motion and the vibrational modes. By attaching the engine to a base with adjustable stiffness, it is possible to analyze the force transmitted by the engine to the base, and the response of the latter. The method chosen for adjusting stiffness is to vary the unsupported length of a beam, upon which the engine is mounted. Figure 3 contains details of the mechanism to vary base stiffness, and Fig. 4 shows some details of the base construction. In what follows, the reader is asked to refer to those two figures.

The engine base unsupported length may be adjusted by the simple turn of a handle (Part 16 in Fig. 3.). This displaces an adjustable support (Part 7, Fig. 3). The base is attached to this support and to a twin one on the opposite end. As the supports are displaced, the unsupported length of the beam and hence the rigidity of the base can be varied. Figure 4 affords a view of the engine base and of one of the supports.

The adjustable supports run along two dovetail slots, visible on the base at each side of the crank. Two slots minimize twisting in the mechanism. The plate containing the dovetails is split along its center so that misalignments can be easily resolved by turning a screw to vary the plate separation.

The adjustable mechanism of Fig. 3 is attached to the split dovetail base (Part 2), which is then mounted on to the main base, (Part 1 in Fig. 5). The latter figure shows the base assembly. A rubber sheet (Part 17) and rubber feet are affixed to the main base to dampen vibration from the system that could be transmitted to the workbench.

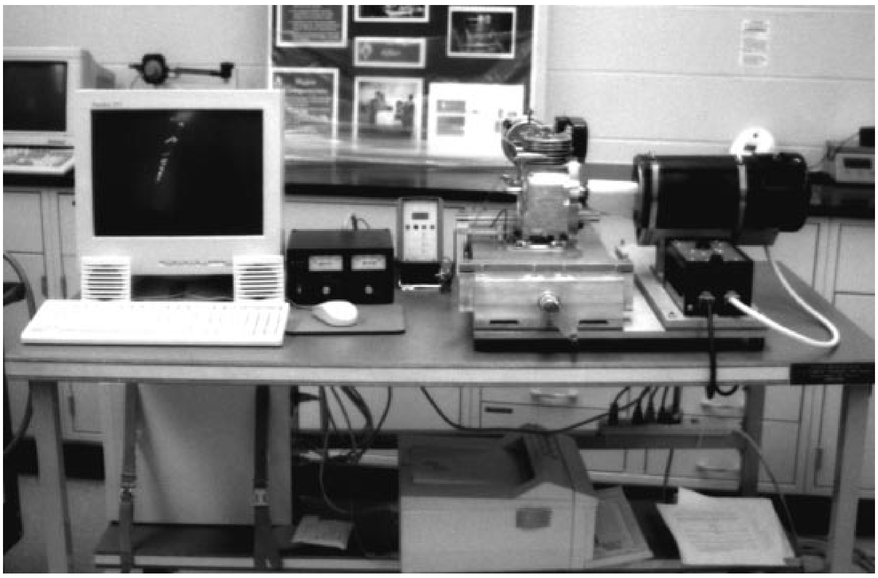

Fig. 2. Completed test stand.

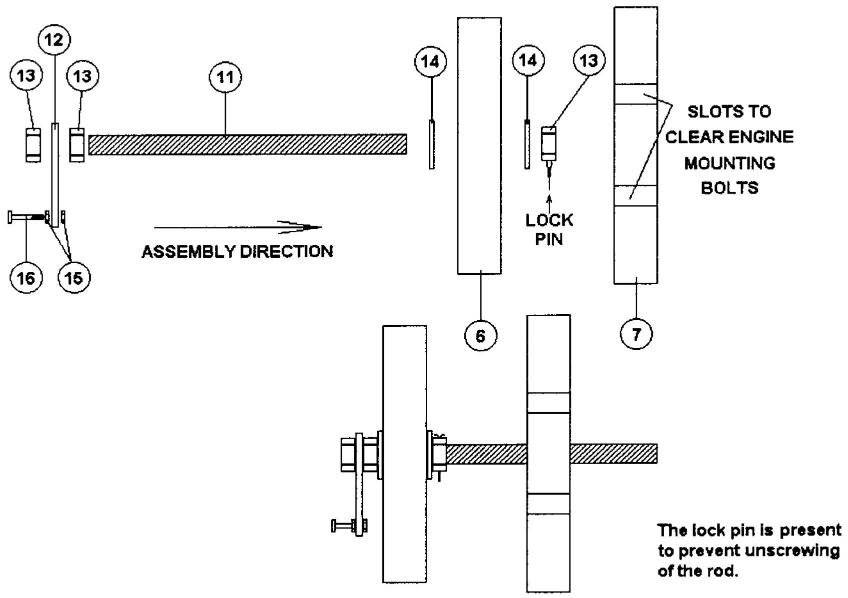

Fig. 3. Adjustable mechanism assembly diagram (top-view).

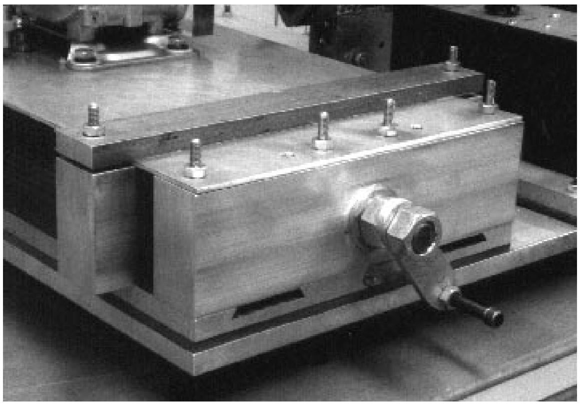

Fig 4. Adjustable support mechanism (side-view).

Supporting the assembly

The thickness of Part 5, the engine base, was determined by assessing experimentally the force required to deflect the base and comparing it with the force to be transmitted by the engine. This force was approximated analytically [5].

A variable speed electric motor drives the engine. A universal joint connects the IC engine shaft with the electric drive shaft. Since the IC engine is mounted on a flexible base, and the electric drive on a rigid one, the universal joint allows unrestricted engine vertical motion. Additional effects are minimized by separating the bases of the electric motor and the engine, and by mounting both on damping material. The

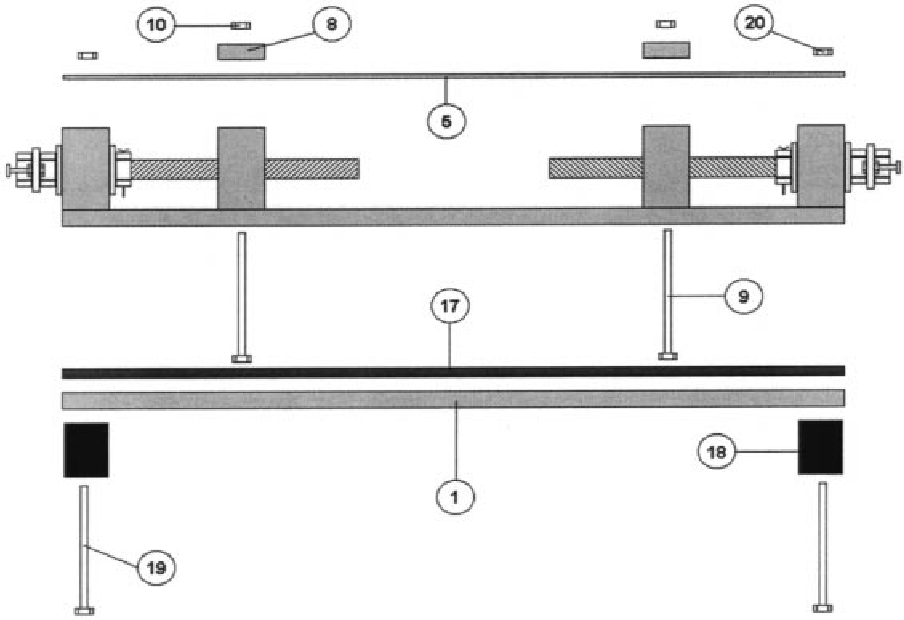

Fig. 5. Assembly base drawing, side view.

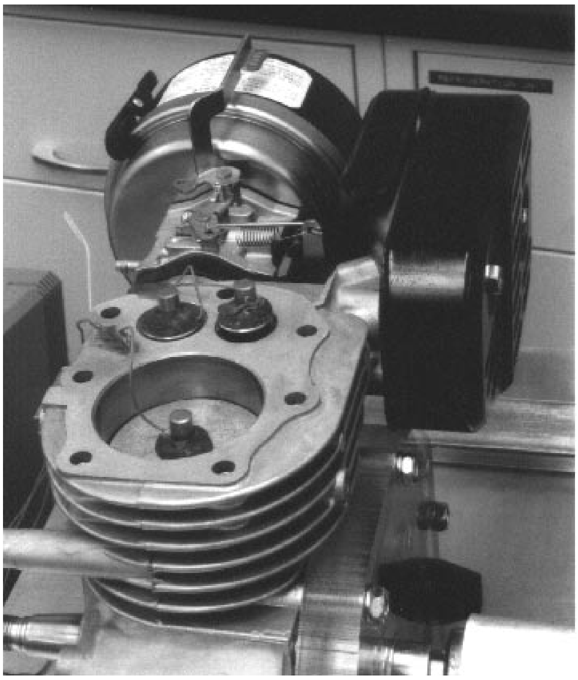

Fig. 6. Accelerometers mounted on piston, intake and exhaust valves.

thickness of the support of the electric motor is such that the universal joint is positioned horizontally when the electric motor is not rotating. All rotating parts are shielded for safety.

Instrumentation/PC system

Accelerometers are used to monitor the acceleration of the piston, intake and exhaust valves of the engine (Fig. 6, top of valves and piston), and the acceleration of the engine base. A linear voltage displacement transducer (LVDT) monitors the displacement of the engine base. As the accelerometer oscillates, its internal seismic mass exerts a shear force on a quartz piezoelectric crystal. This applied force causes the piezoelectric material to produce an electric charge. Since the seismic mass is constant, the charge is proportional to the sensor acceleration. Each sensor can detect motion perpendicular to its base. The accelerometers are connected to the data acquisition board via a charge coupler, which measures the charge produced by the sensor.

The linear displacement transducer operates on the LVDT principle, where movement of a core inside the transducer body is detected by a differential change in output on two secondary coils, the primary coil(s) being energized by an appropriate AC signal. With the core in the central position, the coupling from the primary to each secondary is equal and opposite and therefore the resultant output voltage is zero. As the core is displaced further into one secondary, its induced voltage increases proportionally and the other secondary voltage decreases, hence the output voltage responds in magnitude and phase to the core motion. The core follows the motion of the IC engine base. The signal from the transducer is conditioned, and sent to the PC A/D board.

TEST STAND APPLICATIONS

Two tests can be conducted on the stand. The first one focuses on timing, and the second one on vibrations.

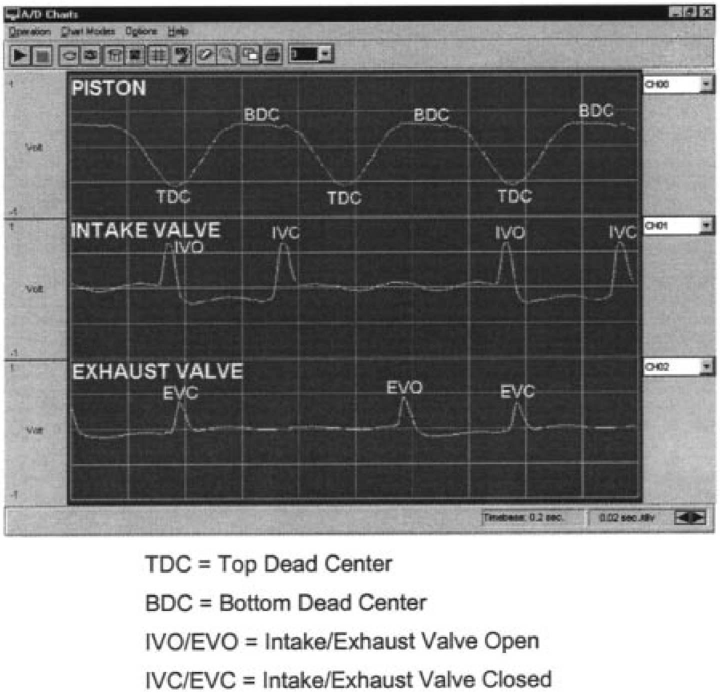

Fig. 7. Interpretation of peaks in accelerometer display.

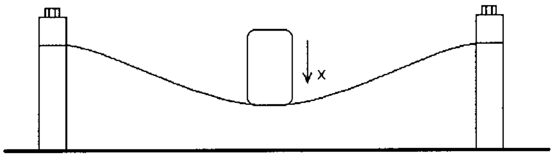

Fig. 8. Coordinate system.

Timing test

Changes in acceleration of the valves and piston allow monitoring of engine timing. An interesting comparison can be obtained in terms of engine rotational speed. The rotational velocity can be measured directly with a contact tachometer and it can also be calculated from the recorded output of the accelerometers. If the rotational velocity of the tachometer and that derived from the accelerometer match, the acquisition software can be assumed accurate.

With the engine driven at 1010 rpm (as measured by the tachometer), the output of Fig. 7 was obtained.

For each division along the x-axis equal to 0.02 seconds, the rotational velocity of the crankshaft may be calculated as follows:

1. Measure the length $T_{DIVISION}$ of one division in the x direction.
2. Measure the distance (Fig. 7) between two consecutive TDCs.
3. Calculate the time for one revolution via equation (1).

$$T_{ONEREVOLUTION} = \frac{T_{REV}\,(\text{mm})}{T_{DIVISION}\,(\text{mm})} \times 0.02\,(\text{sec/division}) \quad (1)$$

A rotational velocity of 1010 rpm (6060 deg/sec), which compares favorably with the value from the tachometer, is obtained in this way.

Other magnitudes of interest concerning the timing of several events can be deduced from the display. For instance, during a short interval both the intake and exhaust valves are open. The time preceding or lagging top dead center under which a valve prematurely, or belatedly, opens or closes, is normally expressed as degrees before, or after, top dead center. These timings may be readily found by simply relating the period (corresponding to 360°) to the time interval measured from the display.

Dynamic test

The system composed of the IC engine and base is approximated as a linear second-order dampened system. Since the base is constrained to vertical displacements, and it has been designed to operate in the elastic region, the only unknown parameter is the degree of damping in the system. Clearly, since energy continuously enters the IC-base system via the IC crank shaft, energy must either be stored or dissipated for the system to reach steady state. Enough energy is dissipated via absorption by the rubber base, air displacement and irreversibilities associated with the vibratory motion of the base, to allow steady state. For the system of Fig. 8, the equation of motion is:

$$m\ddot{x} + c\dot{x} + kx = F(t) \quad (2)$$

where, m is the equivalent mass of the system, c is the damping and k is the stiffness. Insights as to the nature of k, c and $F(t)$ can be gained via a number of tests. A brief description of the capabilities of the test stand and results follows here. An upcoming article, now being prepared, will furnish a more thorough description.

The value of stiffness k in equation (2) can

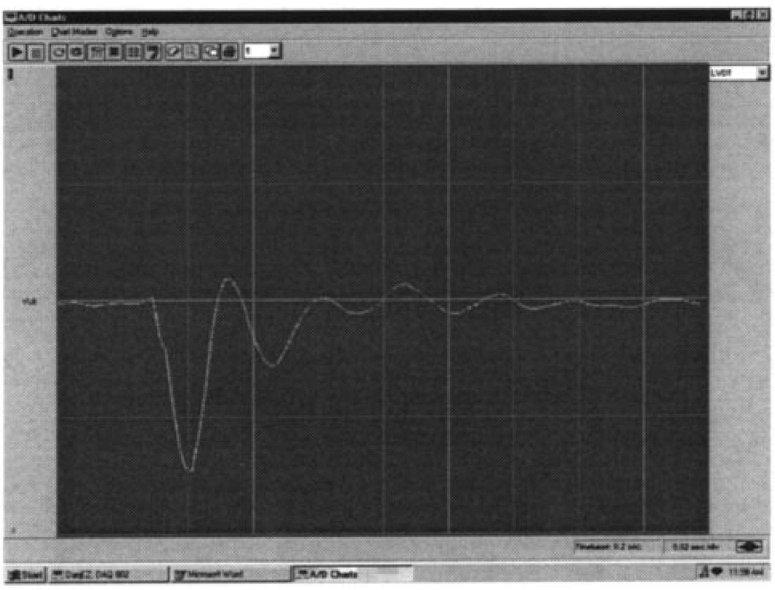

Fig. 9. LVDT signal of displacement in a free vibration.

be readily determined analytically from beam theory or via experimental measurements of the displacement during a static loading experiment.

The value of damping c is determined via experimental measurements of the displacement amplitude decay (log decrement) during free vibration (see Fig. 9). This experiment also yields the first natural frequency of the system, which in turn can be used to analytically determine the equivalent mass of the system.

When the electric motor drives the IC engine, the system becomes a second-order forced vibrational one. The moving piston and valves and the rotating shafts of the IC engine transmit inertial forces to the engine case and result in a periodic force acting on the base. IC engine analyses show this periodic force to have two harmonic terms with two frequencies. The first harmonic has the same frequency as the rotational velocity of the engine and the second is twice the first one [6].

Since the vibrational system is assumed to be linear, the base displacement responds to this excitation with two harmonic terms. The amplitude and phase of each harmonic displacement depend on the amplitude of the forcing and the frequency ratio of the forcing and natural frequency. For instance, when both forcing frequencies are below the natural frequency of the system engine-base, the displacement vs. time is shown in Fig. 10, upper half. A Fourier transform of this signal can be found, yielding two dominant frequencies, Fig. 10, lower half. A third frequency peak at zero in the figure is merely showing a zero offset of the displacement sensor. The two frequency peaks can be related to the excitation frequency of the engine. Operating the engine at different rotational speeds demonstrates the effects of frequency ratio on the amplitude and phase response of a forced vibration second-order system.

Students from a Penn State Dynamics course attended the lab. Feedback from the students on the effectiveness of the lab was received via a survey on the lab and course evaluations. On the survey, the students rated the overall educational value of the lab at 8.14/10 with a standard deviation of 1.68/10. On the evaluation of the course, 70% of the students added on their comments that they found the lab to enhance their understanding of the material. Their comments included: the labs 'allowed real life visualization of the material'; having the labs 'makes it easier to understand the concept'. Suggestions on the improvement included using better visualization software that includes screen capture and increasing the length of the lab.

CONCLUSIONS

An IC engine vibrational test stand has been developed. In the setting of a thermal systems laboratory, the test stand is aimed at illustrating the integrating nature of design. The stiffness of the engine base can be readily varied. Variable stiffness and variable speed allow studies of the vibrational response of the base to the excitation from the engine. Fourier transform of the displacement signals show the power spectrum and phase transitions of a second-order system. Students attending the lab indicated in subsequent surveys that the experience enhanced significantly their understanding of the dynamics of machinery

The valves and piston have been fitted with accelerometers that serve to illustrate engine timing. In this way, the relationships of piston

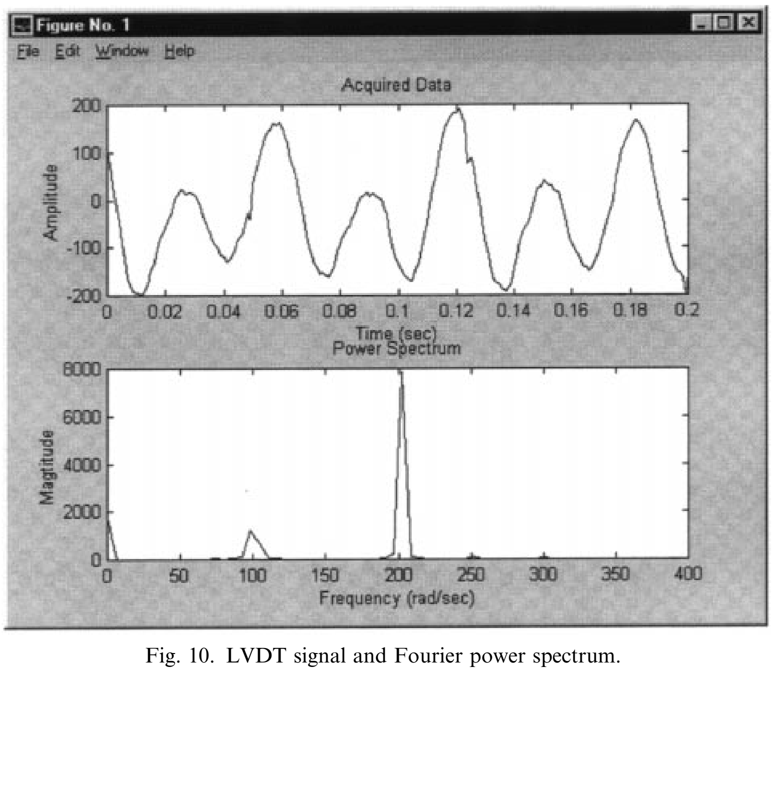

Fig. 10. LVDT signal and Fourier power spectrum.

and valve motion can be studied in an actual engine, and the approximations inherent in a real Otto cycle to the theoretical one can be best appraised.

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Matt Hall works for Jaguar in England. He was the main designer/builder of this test stand as an undergraduate at Leeds University, during a study period at Penn State.

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