GAS AND PRESSURE SAMPLING OF HIGH SPEED ENGINES

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PREFACE

The design of the internal combustion engine has progressed at an ever increasing rate through the years as we have developed new materials to work with and better fuels to use in these engines. As the development of these engines has progressed towards higher operating speeds, lighter construction, and increased power output, many problems pertaining to the functioning of the engine have been confronted. The solution of these problems lies in the development and use of test equipment.

Knowledge of the pressure and combustion phenomena within the engine cylinder have probably contributed more to the advancement in the internal combustion engine field than any other sources of information. The conversion of the stored energy in the fuel into useful work is the functibn of the engine. In the modern high speed engine the design and operating characteristics depends upon what happens within the engine cylinder during combustion and the resulting expansion stroke.

Devices for gas and pressure sampling have been built to give the desired information pertaining to combustión and pressure changes during the working stroke of the engine. Extensive design research is being made today at the Oklahoma Power and Propulsion Laboratory on the development of new high speed gas and pressure sampling devices which will make additional information available on combustion and pressure phenomena, thus paving the way to better engine design and performance.

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INTRODUCTION

The development and improvement of internal combustion engines from the first heavy slow speed and wasteful engines to the light high speed and economical engines of today have been made possible by the data obtained from a variety of engine tests. Information obtained from pressure recording and gas sampling devices have probably contributed more to the development of the present engines than any other one phase of engine testing. De Juhasz, states in his book on engine indicators that:

> "The theory of the steam engine, and also that of the gas and oil engines, as well as of pumps and compressors, has been largely built up on information obtained by means of the indicator".[1]

Essentially, the indicator is a recording pressure gage, by which changing cylinder pressures can be recorded against piston displacement, crank angle, or time.

If the indicator records pressure change against piston displacement an area is traced by the instrument which represents the work done by or upon the work medium. In internal combustion engines this indicator card represents the work done within the engine cylinder. With the information thus obtained from this card the mean indicated pressures and hence the indicated horsepower, of an engine cylinder can be obtained.

For some purposes it is desirable to record pressure changes against time, or crank angle. From characteristic points of the time card, the stresses in the engine parts, the timing of the

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[1] De Juhasz, Kalman J. The Engine Indicator, pp. 1.

valves, throttling effect through valve openings, leaking piston rings, ignition and combustion phenomena, intake and exhaust phenomena, and other data important from a design or performance point of view can be determined.[2] The ability to obtain gas sampling during the engine cycle gives us information pertaining to air-fuel ratio, exhaust gas analysis, contamination of fresh charge with exhaust gases, completeness of combustion, and other important information pertaining to design or performance. Another use to which the engine indicator can be put which will be only mentioned here to further enlighten the engineer as to its uses, is to measure variable forces in various forms of material testing equipment, as in the application of a hydraulic press, adapting for recording a certain motion (as lift on valves) or travel on a sliding mechanism.[3]

The foregoing discussion indicates the extreme importance of gas sampling and pressure recording devices in the design, development, and improvement of the modern internal combustion engine.

The first attempt to indicate changing pressures in an engine cylinder was made by James Watt, the great improver of the steam engine in the year of 1790. The instrument was conceived to show changing steam pressure in his vacuum type engine. It consisted of a cylinder and within it a closely fitting spring-loaded piston.

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2. Ibid., pp. 1.

3. Ibid., pp. 2.

The rod of the latter carried a pointer, movable over a fixed scale. Sometime later in about 1796 John Southern, one of the collaborators of Watt, made important improvements on this apparatus by substituting a pencil for the pointer and a piece of paper fastened to a movable tablet for the scale; thus the first pressure recording instrument was developed. Soon afterwards, however, Southern conceived the idea of connecting the card carrier with some reciprocating point on the balance beam of the engine; thus obtaining a closed card from which the work done by the engine piston could be determined. It is believed that the indicator diagram produced by the Watt-Southern indicator was the inspiration for Watt's great invention, the expansive working of steam.

Around 1825-1830 John McNaught of Scotland, began to build indicators on a small commercial scale and brought about the change from a sliding card carrier to a spring actuated drum. From this point on only minor refinements have been made to give us the indicators of today. [4] The following inventors and manufacturers are the most prominent in the field of research and development of engine indicators:

In the United States:

Crosby Steam Gage and Valve Company
J.W. Thompson
Trill Indicator Company

In Great Britian:

Budenberg Gauge Company
Dobbie-McInnes Ltd.
Elliot Bros.

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[4] *Ibid.*, pp. 3.

In Germany:

Dreyer, Rosenkranz and Droop
Lehmann and Mechels
Schaeffer and Budenberg.[5]

The slow-speed indicators have been developed to such a state of refinement that very reliable results can be attained with them if speeds of 250-350 rpm are not exceeded. These speeds are encountered in stationary steam, gas, and oil engines.[6]

With the augmentation of speed, pressure, and temperature, which characterizes the trend of the modern engine development, the error introduced by the normal pencil type indicator increased rapidly. This difficulty lead to the development of optical, electrical, and stroboscopic methods of determining the changing cylinder pressures. The reason for the error introduced in pencil type indicators is explained by K.J. De Juhasz, as follows:

> "Owing to the inertia of its moving parts the pencil of the indicator does not describe the true pressure line but superimposes upon it its own natural vibration. Due to this an error results, the magnitude of which is the greater the more rapid the pressure change and the longer the natural period of the moving parts.
>
> On a slow speed steam engine (with a low rate of pressure change) the superimposed vibration may be so small as to be hardly detectable. On a high speed engine, or on one with a high rate of pressure change, the vibrations may entirely distort the diagram and render it useless for evaluation.
>
> In order to reduce the error, it is necessary that the natural vibration period of the pressure measuring parts be small in comparison with the time of a given change in pressure".[7]

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5. Ibid., pp. 9.

6. Ibid., pp. 40.

7. Ibid., pp. 81.

In order to reduce the natural period of vibration of the indicators, stiffer springs and lighter weight moving parts had to be incorporated into their construction. The result was the development of the micro indicator which scribes indicator cards which must be viewed under a microscope, or photographed and enlarged in order to make use of them. These indicators give satisfactory results up to 1500 rpm. [8]

As the trends of engine design progressed toward higher speeds, temperatures, and pressure, the engine indicator as a pressure recording instrument reached its limit of dependability. It then became necessary to apply a different method to obtain a diagram of the changing pressures during the engine cycle because with the high speeds (above 1500 rpm.) the drawing of a complete cycle would have to be accomplished in such a short period of time that it becomes mechanically impossible. The Optical and Electrical indicators came into existence to give a means of recording the changing pressures over each cycle of operation of the engine, and are the most widely used today for high speed engine indicators. The principle of operation of the optical indicator is as follows: The minute motion of the measuring parts is recorded by photographic means. The mirror is given a rocking motion in one direction proportional to the changing cylinder pressure, and a rocking motion in the other direction proportional to the reciprocating motion of the engine piston.

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[8] Ibid., pp. 94.

The principle of operation of the electrical indicator is as follows: The changing pressure in the engine cylinder acts upon a diaphragm whose motion is translated into electrical terms; or a substance such as a carbon pile which changes its electrical characteristics with changing pressures. The electrical terms are recorded by means of an oscillograph which can be observed or photographed.

The electrical indicators are more popular than the optical ones because the optical indicators have the disadvantage of too much bulk and weight which gives them a long period of vibration, and the whole instrument is subjected to engine vibration because its principle of operation necessitates its close proximity to the engine cylinder. The electrical indicators, on the other hand, have some limitations. The pressure changes are recorded on a time basis because the cathode ray tube has only one axis and some error will result in replotting data to obtain pressure volume cards; also the electrical indicators are delicate and intricate instruments and must be handled by a skilled operator in order to obtain accurate results and to insure no harm comes to the instrument.

The indicators described thus far trace a complete diagram of each engine cycle. There is another principle used in indicators in which the diagram is built up, point-by-point, from a large number of engine cycles. This method can be applied only if the successive cycles are sufficiently alike during the taking of the diagrams. The fulfilling of this condition necessitates the maintaining of a constant load and speed on the engine during the taking of the data and, that a state of equilibrum is reached with the engine before the data is taken. K.J. De Juhasz, states:

> "If however, for the duration of taking a diagram sufficient constancy of the engine cycle can be assured, then this method offers a far greater convenience and accuracy than any of the single cycle indicators. A further advantage of this method is that the resulting diagram represents mean values, which, for many engineering purposes, are preferable to the record of one single cycle.[9]

As engine speeds and pressures increased still further we find phase-to-phase sampling devices coming into use. These instruments in general have the advantage of being adaptable to both pressure and gas sampling. The details of this will be shown later in the design of a phase-to-phase sampling device.

The cyclic pressure indicators are of two classes:

1. From pressure to pressure, that phase of the engine cycle is determined at which a given pressure exists.
2. From phase-to-phase, that pressure is determined which exists at a given phase of the cycle.

The first class of cyclic indicators is not adaptable to gas sampling because it operates on the following principle: That a storage bottle under a pressure or a vacumn operating against a piston connected to the engine cylinder balances the cylinder pressure; and when this occurs a spark is made to burn through a piece of paper on a rotating drum; and by balancing the pressures throughout the cycle a pressure time card is built up through a large number of cycles of operation.

The pressure time card type of indicator based on the press-

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[9] Ibid., pp.130.

ure balance principle was developed by Major Norman, and H. Wood, of the Royal Aircraft Establishment of England, and is being manufactured by Dobbie-McInnes of Glasgow, England. The Farnboro indicator has many uses in the testing labortories. [10]

The phase-to-phase class of indicators presents the most diversified test instrument. It is possible to adapt a slow speed single cycle indicator to the sampling device and record the phase-to-phase pressure changes as well as extract gas samples for analysis on a phase-to-phase basis. The phase-to-phase indicator operates on the following principle: A sampling valve is directly connected to the engine cylinder and is actuated by either electrical or mechanical means; the time of opening of the valve must be phased to open once each engine cycle and the period of sampling must be controlled, this also can be done mechanically or electrically. If pressure sampling is desired, the pressure is allowed to build up over successive cycles until an average pressure is reached for the time and point of opening of the valve. These pressures are recorded from phase-to-phase through the engine cycle to build up a pressure time card or pressure volume card. If gas sampling is desired, the sample extracted when the valve is open is collected until there is sufficient sample to analyse. This can be done on a phase-to-phase basis throughout the engine cycle.

Mechanical sampling valves have been developed which operate

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[10] Heldt, P.M. High-Speed Combustion Engines ., pp. 650

on the phase-to-phase principle by Professor K.J. De Juhasz, Pennsylvania State College; Professor H.M. Kanklin, Purdue University; Professor Ford L. Prescott, University of Florida; J.A. Sponogle, and E.C. Buckly, staff of the National Advisory Committee for Aeronautics at the Langley Memorial Aeronautical Laboratories; Claude E. Cox, Commercial Engineering Laboratories; Burgess-Manning Company; Withrow, Lovell, and Boyd, General Motors Research Laboratories.

The De Juhasz sampling valve was designed for a four-stroke cycle engine. The mechanical phase changer consisted of a planetary gear train with a 2:1 reduction. The sampling of cylinder pressures was accomplished with a reciprocating slide valve connected to cranks at almost 180° from each other. When the slots in the slide valves lined up, each cycle, a sample flowed to the engine indicator, connected to the sampling valve, which records the pressure changes throughout the engine cycle. This sampling valve has been used successfully on engine speeds up to 6000 rpm. The main source of error of this type of instrument is leakage which may occur in the various pipe unions and in the valve element. To eliminate these leaks all connections must be sealed and the valve surfaces lubricated with a very viscous oil.[11]

The Langley Memorial Aeronautical Laboratories have developed a sampling valve which has been used successfully for gas sampling to analyze the cylinder gases during combustion, which gave

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11. De Juhasz., Op. cit. pp. 143-148.

information pertinent to improve the design of combustion chambers on compression ignition engines. This instrument has a needle valve, sealed with a flexible steel diaphragm, with a lift of 0.004". The operation of the valve is as follows: A valve cam driven by a flexible shaft compresses a cam spring through a tappet arm; and upon release the inertia of the system lifts the valve stem from its seat against a stronger valve spring; which then returns the valve stem to its seat. This type of valve features a constant sampling period regardless of engine speed.[12]

The Cox, Burgess-Manning, General Motors, and Prescott, sampling valves are of the poppet valve type. The first two are opened by an electro-magnetic coil and closed by a spring. The Prescott valve is opened by a cam and rocker arm arrangement and closed by a spring. The Cox sampling valve features a poppet valve and is used for gas and pressure sampling. The Burgess-Manning valve features a combination spark plug and needle valve and is used for gas sampling of the exhaust gases of a spark ignition engine. The Prescott valve is of the poppet type and is used in combination with an engine indicator to record pressure changes within an engine cylinder. The General Motors valve is used for combustion research and is mechanically actuated.

The foregoing discussion has been given to stress the importance of gas sampling and pressure recording devices; and to present the development and improvement of these instruments in

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12. National Advisor Committee Aeronautics Technical, Note No. 454.

a logical order, as they were needed to give the necessary data for the design and improvement of the modern high speed internal combustion engine.

SELECTION OF SAMPLING VALVE TYPE

The testing of high speed engines requires test equipment which will give dependable results from the engines being tested. A phase-to-phase sampling valve will give information pertaining to both gas and pressure phenomena. There have been many different types of sampling valves made for both research and routine test work. The particular design selected depends to a great extent upon the engines to be tested as to physical structure, speed, maximum cylinder pressure, and temperature. The general requirements of sampling valves are given by K.J. De Juhasz as follows:

"(a) It is essential that the communication period be a small fraction of the engine revolution, so that the pressure change during the opening period be small;

(b) during this short period the valve must provide a large opening area of high orifice coefficient so as to give an unobstructed passage for the flow of the medium;

(c) the valve must be free from leakage under high pressure and high temperature conditions. Any leakage constitutes a serious source of error when it is considered that the time available for leakage is a large multiple of the duration of communication. Therefore, it is important to avoid distortion from heat which induces leakage;

(d) it has to be of small dimension and weight and of such a shape which permits a short connection to be made to the space to be investigated. For gas sampling this requirement is especially important."[1]

The design and construction of a sampling valve is something which requires considerable time and effort. Therefore, the type of device and principle of operation should be chosen which will best fit the immediate needs for such a device in testing

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1. K.J. De Juhasz, pp. 150.

the equipment available. Thus far the commercially available sampling devices have had such a high purchase price that it is not possible to obtain one for use in testing in our internal combustion engines laboratory. These commercial sampling devices are found mostly in internal combustion engine manufacturer's testing laboratories.

There has been a definite need for a sampling valve for some time to be used in graduate studies in connection with the internal combustion engine. Therefore the selection of the mechanical features for the sampling valve have been chosen which will best fit these needs. The sampling valve will be mechanically actuated by a direct gear drive from the engine. This feature gives durability and eliminates any error in the phase relationship between engine and sampling valve. The sampling valve will be of the poppet type because this means positive sealing of the valve against leakage when closed, and that the sample can be extracted directly from the engine cylinder without any intervening passageway. The poppet type sampling valve has the following disadvantages: sampling occurs over an increasing number of degrees of engine crank travel as the engine speed increases; and the pumping action of a poppet valve in the direction of greatest orifice coefficient will introduce an error when it is used for pressure sampling.[2]

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2. Ibid., pp. 151.

The engine speeds usually encountered are from 1200 to 1800 rpm. in most cases and the poppet type valve will function satisfactorily and has thus been selected as the type to be designed and constructed.

DESIGN OF GAS AND PRESSURE SAMPLING VALVE

The sampling valve was designed to satisfy the following donditions of operation:

1. The valve must be able to extract a sample in approximately 8° of engine crank travel at engine speeds up to 1800 rpm (this allows 5 to 6 representative samples during combustion which was sufficient for combustion analysis).
2. A means of cooling the sampling valve in the region where the sample is to be extracted is necessary to stop combustion of the gases immediately upon extraction, and to insure that the valve operates cool enough that the lapped surface between the valve and valve guide (acting as a gas seal) does not seize.
3. The valve must have a large enough flow area so that sufficient samples can be extracted for analysis within approximately a 3 to 5 minute period.
4. Because of the magnitude of the inertia forces encountered with a poppet type of valve, every effort must be made to reduce the lift and weight of the moving parts of the valve.
5. The cam should be in direct contact with the valve stem to eliminate weight and lost motion thus contributing to a faster valve action.
6. For convenience, compactness, and adaptability the phase changing device should be incorporated directly with the valve actuating mechanism.

7. Since the four-stroke cycle engine is more commonly encountered than the two-stroke cycle the phasing mechanism will have incorporated within it a 2:1 reduction thus giving the cam a cycle of 720° of crank travel which corresponds to the four-stroke engine cycle.

The calculations necessary for the design and construction of the described sampling valve have been set forth on the following pages.

The flow area around the valve was first investigated to determine whether the amount of sample desirable (300 cc in two minutes) was a limiting factor in the design of the valve. From the standpoint of design a small diameter light weight valve was essential to reduce the inertia forces and combustion gas forces to reasonable values, so that a light spring could be used, thus greatly reducing the operating stresses imposed on the valve and giving a long life valve mechanism.

The flow area determinations are as follows:

Volume of flow at 14.7 psia;

$$V = \frac{300 \text{ cc of sample}}{16.39 \times 2 \times 60} = 0.1525 \text{ in.}^3/\text{sec} \qquad (1)$$

Weight of gas flowing per second at 14.7 psia:

$$PV = MRT \qquad (2)$$

$$M = \frac{PV}{RT} = \frac{14.7\ (144)\ (0.1525)}{53.3\ (460 + 100)\ (1728)} = 0.00000626 \text{ lbs/sec}$$

Where;

P = the gas pressure in lbs/ft^2

T = approximate temperature of gas flowing into Orsat analyzer in degrees Rankine

V = volume of gas in ft^3/sec

R = gas constant for air

M = weight of gas in lbs per second.

The most critical condition of sampling occurs during the suction stroke of the four-stroke cycle engine. This occurs at approximately 10 psia. It is necessary to have a collector tube in which a pressure of approximately 3 psia can be maintained to collect the sample before pumping it through the Orsat apparatus when sampling below atmospheric pressure.

The volume of gas under suction conditions is as follows:

$$V = \frac{MRT}{P} \tag{2}$$

$$V = \frac{53.3 \times .00000626 \times 600 \times 1728}{10 \times 144}$$

$$V = .242 \text{ in.}^3/\text{Sec.}$$

The general formula for gas velocity valid for any pressure difference for an adiabatic flow as given by V.L. Maleev is:[1]

$$V = \sqrt{\frac{2g\ KRT_1}{K-1}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{K-1}{K}}\right]} \tag{3}$$

$$V = \sqrt{\frac{2 \times 32.2 \times 1.38 \times 53.3 \times 600}{1.38-1}\left[1-\left(\frac{3}{10}\right)^{\frac{1.38-1}{1.38}}\right]}$$

$$V = 1370 \text{ ft./Sec.}$$

Where;

G = acceleration of gravity (32.2) ft./Sec.^{2}

K = constant for a polytropic expansion

R = gas constant for air

T_1 = temperature of charge in cylinder during the

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1. V.L. Maleev, Internal Combustion Engines, pp. 321

suction stroke in degrees Rankine

P_1 = initial pressure psia

P_2 = final pressure psia.

Allowing for friction and change in velocity with change in flow area as the valve opens and closes, the average velocity of the gases would be approximately 1/4 of calculated value, thus:

$$\bar{V}_a = \frac{1370}{4} = 342.5 \text{ ft./Sec.}$$

The sampling time of valve is determined by:

$$\theta = \frac{1}{\frac{n}{60}} \times \frac{\alpha}{360} \tag{4}$$

$$= \frac{1}{\frac{1800}{60}} \times \frac{8^\circ}{360^\circ}$$

$$= .000740 \text{ Sec.}$$

Where;

α = number of degrees of engine crank travel during sampling.

n = revolutions per minute (rpm)

θ = sampling time in seconds

Volume of sample per engine cycle is found from the following relation:

$$\dot{V} = V \times \frac{60}{n} = .242 \times \frac{60}{900} \tag{5}$$

$$= .01614 \text{ in.}^3$$

The flow area required is found by:

$$A = \frac{\dot{V}}{\bar{V}_a(12)} = \frac{.01614}{342.5(12)} \tag{6}$$

$$= .0000392 \text{ in.}^2$$

These calculations show that the flow area was not a determining factor in the design of the sampling valve and that it would be possible to concentrate on the smallest lift and flow area for which it would be mechanically feasible to design. The extremely small amount of time which the valve would be in operation (0.00740 sec.) shows that every effort must be made to reduce the wieght of the moving parts and hence reduce the inertia forces.

A relationship between lift, cam angle, cam diameter, and engine cycle exists which must be investigated to establish the most satisfactory design. From the investigation of the equation for the force of inertia expressed in terms of weight, time, and distance,

$$F = \frac{2SW}{gt^2}, \tag{7}$$

it can be seen that the weight of the moving parts of the valve and the lift of the valve bears a direct relationship to the inertia force and the time varies with the inverse square law. From the above relationship it can be seen that the operating time of the valve presents the most critical limitations on the valve design and performance.

The lift of the valve was kept to within .01 to .02 inches to reduce the impact upon closing and hence the possibility of damaging the seating properties of the valve and also to reduce the possibility of valve surge after the time of closing, thus giving a sample which would not be representative of the point of sampling. The cam which actuates the valve operates at one-half engine speed. The angle of the cam lobe acting against the valve must be only one-half of the number of degrees of sampling, which would then be 4 degrees. The diameter of the cam should be kept as small as possible to reduce the tangential velocity between cam and valve upon contact as this will produce less wear of the parts due to friction.

The proper angle of the valve seat was investigated. Decreasing the angle of seat toward a flat seat gives the following results:

(1) Poor seating properties
(2) More bearing surface
(3) Less tendency of valve to stick when surface becomes dirty
(4) Less lift for a given flow area
(5) Thin edge which gives the valve a tendency to burn
(6) Faster conduction of heat away from seating surface because of increased area
(7) Increased weight of valve

By increasing the angle of the valve seat the opposite effect is obtained in each of the above cases. A 30° angle of the valve seat was selected as this gave the best combination of the above items.

The design of the valve and valve seat is given the next consideration. V.L. Maleev, gives the following equation for the flow area to equal the port opening;

$$D = \frac{h \cos\alpha}{.25}$$

$$= \frac{.02 \times .867}{.25}$$

$= 0.0694$ in.

Where;

D = diameter of valve port in inches

h = lift of valve in inches

α = angle of valve seat in degrees.[2]

The flow area is found by;

$$A = \frac{\pi}{4} D^2 = 0.7854 \times 0.0694^2 \tag{9}$$

$$= 0.003780 \text{ in.}^2$$

The width of valve seat is determined by Maleev's equation which states that:[3]

$$b = 1.45D$$

$$b = 1.45\ (0.0694) = 0.1005 \text{ inches} \tag{10}$$

The projected area (A) is determined by using the projected width, b = 1.45D, and then solving for the area from the relationship;

$$A = (2b + D)^2 \frac{\pi}{4} \tag{11}$$

$$A = \left((1.45 \times 0.0694 \times 2) + 0.0694\right)^2 \frac{\pi}{4}$$

$$= 0.0575 \text{ in.}^2$$

Next, the diameter of the valve stem was determined. Because of the rigid support offered by the valve guide and the relative short length of stem required (approximately 5 inches), the

2. Ibid., pp.325

3. Ibid., pp. 326

strength calculation can be for a short column under direct compression. Assuming that the maximum force against the valve stem is approximately 200 pounds the area of the stem can be determined by the equation based on direct compression of member:

$$A = \frac{F}{S} \tag{13}$$

$$A = \frac{200}{12,000}$$

$$A = .01667 \text{ in.}^2$$

Where:

F = force acting against valve in pounds

S = working stress, for the material under consideration which is high carbon steel, given in pounds per square inch.

The diameter is determined by;

$$D = \sqrt{\frac{.01677}{\frac{\pi}{4}}}$$

$$= .1462"$$

The above valve stem diameter is that based on direct compression. For a member to be under direct compression the ratio of length to diameter must not exceed $\frac{L}{D} = 8:1$ for steel, as given by V.L. Maleev.[4] The unsupported length of the valve stem extending from its guide will be assumed to be 2 inches. By using the above relationship the diameter of the valve stem can be determined as follows:

$$\frac{L}{D} = 8:1 \tag{14}$$

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4. V.L. Maleev, Machine Design, pp. 17.

$$D = \frac{2}{8} = .25 \text{ inches}$$

Thus the valve stem diameter is limited by the $\frac{L}{D}$ ratio.

The wieghts of the moving parts were determined next.

Approximate weight of valve:

$$W_v = \frac{\pi}{4} D^2 L d \tag{15}$$

$$W = 0.7854 \times 0.25^2 \times 5 \times 0.285$$

$$= 0.0700 \text{ pounds.}$$

Where;

d = density of steel

D = diameter of valve stem in inches

L = length of valve stem in inches.

The weight of the moving coils of the spring must be estimated and later checked against the weight of the spring used in the design. The following assumptions are made for the spring dimensions at this time:

(1) Mean coil diameter 3/4 of an inch

(2) Wire diameter 0.125 inches

(3) Number of coils will be 8.

The method of determining the part of the total weight of the spring which is moving is given by V.L. Maleev as follows:

> "However, only one end of the spring has a full movement; the movement gradually decreases toward the other end, which is still. Therefore it can be assumed that only one-half of the spring weight participates in the movement."[5]

The total weight of the spring is found from the following relation:

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5. V.L. Maleev, Internal Combustion Engines, pp. 462.

$$W_s = \frac{\pi^2}{4} D^2 D_o \; Nd \tag{16}$$

$$W_s = \frac{\pi^2}{4} \times .125^2 \times .75 \times 8 \times .285$$

$$= .0662 \text{ pounds}$$

Where;

D = diameter of wire in inches

D_o = mean diameter of coil in inches

N = number of coils

d = density of steel

The moving weight of the spring will then be $\frac{W_s}{2} = \frac{.0662}{2}$ = .0331 pounds. The total weight of the moving parts is equal to the weight of the valve plus the weight of the moving coils of the spring and is found to be:

$$W_t = W_v + \frac{W_s}{2}$$

$$= .0700 + .0331$$

$$= .1031 \text{ pounds}$$

The following table shows the effect of the various forces acting upon the valve during its opening and closing action. The sign convention applied is as follows: (+) aids the valve motion; (−) resists the valve motion. The action of the valve is divided into four equal time and distance increments. The ideal motion of a reciprocating valve is that it be accelerated and decelerated during the opening and the closing of the valve.

TABLE I

	Type of motion	Sign Conv.	Inertia of valve	Friction	Spring force	Press. above Atmos.	Inertia of spring
Opening of valve	1. Acc.	(+) desirable	–	–	–	–	–
	2. Dec.	(–) desirable	+	–	–	–	+
Closing of valve	3. Acc.	(+) desirable	–	–	+	+	–
	4. Dec.	(–) desirable	+	–	+	+	+

From the examination of the above table it can be seen that the most severe conditions occur at (1) and (4) but the cam will be in contact with the valve stem and control the motion. This leaves condition (3) as the one to be considered. The gas pressure will vary throughout the engine cycle; therefor the spring is the only force which can be depended upon to keep the valve and cam in contact at all times.

The inertia force at condition (3) will be determined because this will be the force which must be exerted by the spring to insure contact between the cam and valve. The inertia force at the beginning of the valve closure is found by equation (7) as follows:

$$F = \frac{2SW}{gt^2}$$

$$F = \frac{2 \times .02 \times .1031}{32.2 \times 12 \times 2 \times \left(\frac{.000740}{4}\right)^2}$$

$$F = 156 \text{ pounds}$$

Where;

W = weight of moving parts in pounds

g = acceleration of gravity in ft./sec.^{2}

S = total lift of valve in inches

t = total operative time of valve in seconds.

This force computed is the minimum force necessary to insure contact between the valve and cam. Consequently this force should be materially increased to allow for friction and the possibility of operating the valve above the design speed of 1800 rpm.

Desirable spring force = 1.25F = 1.25 (156)

= 195 pounds

The selection of the spring to be used in the sampling valve will depend on the amount of force which the spring will produce under most operating conditions. A spring should be selected which can be pre-loaded considerably to produce the necessary spring force for a particular sampling condition. The best combination of wire diameter, mean coil diameter, and number of active coils was determined by use of a valve spring chart.[6]

Selection of valve spring:

(1) Spring force = 195#

(2) Mean coil diameter = 21/32"

(3) Wire diameter = .177"

(4) Allowable deflection per coil = .04"

(5) Number of active coils = 7

(6) Total deflection = .28"

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6. Products Engineering, Design Work Sheets , pp. 132-133

This spring checks close enough to the assumed spring used in estimating the weight of the reciprocating parts that no change in the calculation of the inertia force need be made.

From these preliminary design determinations the working drawings were made. The availability of some necessary parts, which would otherwise have to be made, altered the over-all design of the sampling valve to some extent. These items were such things as; The phasing-gear-train, fuel injector pump bushing to act as a valve guide, and the adaption of the sampling valve to the engine to be tested.

OPERATING CHARACTERISTICS OF SAMPLING VALVE

The sampling valve was next checked for its operating characteristics so that the functioning of the valve could be established for its use in engine testing. The weight of the valve stem, spring, and spring keeper was determined. The total weight was found to be .1378# the spring selected had 10 coils and its weight was .0349#. Since only 1/2 of the coils would be acted upon by the inertia force the total moving weight of the parts was established as;

$$.1378 - \frac{.0349}{2} = .1204\#$$

The actual lift on the valve was established by using a depth gage and measuring the difference in position of the valve head from closed position to full open by turning the cam lobe across the valve stem with the sampling device assembled. This lift was found to be .015"

With the sampling valve assembled in place in the engine and the drive from engine to the sampling valve attached, the plate was removed from the back side of the sampling device thus exposing the cam and the valve stem. By turning the phase changing wheel and observing the beginning of contact and end of contact between the cam lobe and the valve stem and counting the number of notches on the phasing wheel passing a fixed point on the housing the number of degrees of sampling of the valve can be established. The number of notches passing a fixed point on the housing was found to be 6. The angular relationship between the phasing wheel and the engine crank travel was determined in order to establish the phase wheel position for the various events of the engine cycle. The phasing gear has 53 teeth and the phasing wheel in contact with it has 20 teeth, two revolutions of the phasing gear are required to rotate the cam through one revolution. From the gear ratio stated it can be established that 5.3 revolutions of the phasing wheel are required to rotate the cam through one revolution. The phasing wheel has 80 evenly spaced marks around its periphery. The number of degrees of crank travel per mark on the phasing wheel can be established as follows:

$$\text{degrees/mark} = \frac{720^\circ}{5.3 \times 80} = 1.7^\circ$$

Now the sampling time of the valve can be determined as $1.7\ (6) = 10.2^\circ$ of crank travel of the engine or expressed in seconds

$$t = \frac{10.2^\circ \times 60\ \text{sec}}{360^\circ \times 1800\ \text{rpm}} \tag{4}$$

$$= 0.000945\ \text{seconds}$$

With the operating condition of the valve established the inertia force can be computed as:

$$F = \frac{2SW}{gt^2} \tag{7}$$

$$= \frac{2 \times 0.015 \times 0.1204}{32.2 \times 12 \times 2\left(\frac{0.000945}{4}\right)^2}$$

$$= 84.0\ \text{pounds}$$

It can be seen that the inertia force has been reduced from the one computed in the original design calculations.

The spring force should be increased to allow for friction to about 1.25 x 84 = 105 pounds. Because of the light construction of some of the gears used in the phasing-gear-train a spring force of considerably less than the one computed above should be used.

A lighter spring can be used and the required sampling time of approximately 10° can be maintained by changing the design of the cam so that it accelerates the valve to full open and that the spring accelerates the valve to its seat. This type of valve will strike the seat with considerable force but with the small lift of the valve this type of design should work satisfactorily.

The inertia force computed for the above consideration is found to be:

$$F = \frac{2SW}{gt^2} = \frac{2 \times .015 \times .1204}{32.2 \times 12 \left(\frac{.000945}{2}\right)^2} \qquad (7)$$

$$= 42\#$$

Because of friction the force exerted by the spring will be increased to 1.25(42) = 52.5#.

The spring selected to be used in the instrument has the following specifications:

(1) Spring force = 58#

(2) Mean coil diameter = 11/16"

(3) Wire diameter = .12"

(4) Allowable deflection per coil = .06"

(5) Number of active coils = 10

(6) Total deflection = .6"

DESCRIPTION OF HIGH SPEED ENGINE TESTS WITH SAMPLING VALVE

The purpose of these tests was to first prove or disprove this particular design of sampling valve by performing a series of tests which would show conclusively the adaptability of such an instrument to both routine and investigative types of tests and secondly to be able to recommend design features which will improve the functioning of the valve from the standpoint of durability, adaptability, and shorter sampling periods.

A 3 1/2" x 4", 4 stroke cycle, single cylinder, air cooled Wisconsin engine was selected to be tested. The engine is directly connected to a D.C. generator which can be loaded by connecting the lead wires to movable electrodes in a brine tank. The housing of the generator is attached to the outer shell of the armature bearings and this makes a cradled generator dynamometer. The torque exerted by the generator housing is transferred to a platform scale by means of a lever arm welded to the housing and resting on a pedestal placed on the platform scales. This engine was a speed governed engine and it was desirable to disconnect the governor and attach a screw type throttle adjustment. This gave an exact throttle position which could be relocated at any time and made it possible to operate the engine at any desired speed.

The reasons for selecting this particular engine for testing were as follows: Ease of adapting a direct gear drive from the engine crankshaft to the sampling valve; no water jacket to seal off where the sampling valve was adapted to the engine; and the engine was in excellent mechanical condition.

The pressure gages and mercury manometer, Orsat gas analyzer, brine tank, speed indicator, and fuel measuring tank were all arranged so that the instruments could be read and adjustments of engine operating conditions could be made by the person performing the tests.

X Information pertaining to the functioning of the engine was derived from the data taken during a series of tests. The data obtained from the sampling valve was pressure changes with respect to engine crank angle and Orsat analysis during combustion and exhaust. From the Orsat analysis the functioning of the carburetor as to air-fuel ratio under varying loads was determined. Combustion analysis could only be carried out completely by use of complete gas analyzing set up similar to those found in fuel research laboratories. With the ordinary industrial gas analyzer only the percentages of CO, CO_2, and O_2 can be determined from the exhaust gases of an engine. The formation of water vapor and CH_4 was estimated from empirical relationships established for the general conditions of combustion in a gasoline engine. The pressure data recorded for successive crank positions of the engine gave information pertaining to the timing and functioning of the valves, duration of combustion, time of ignition, a means of determining the mean indicated pressure and hence the indicated horsepower of the engine, a study of the pumping losses during the suction and exhaust strokes, and a study of combustion pressure with reference to detonation and rich or lean gasoline-air mixtures.

The most conclusive proof as to the functioning of the sampling valve is obtained from the plotting of pressure volume diagrams from the pressure time data taken during a test and observing the shape of the cards, computing the value of n for compression and expansion and comparing with known values, and checking the calculated indicated horsepower against that determined by direct measurement. In the case of the test set-up being used the D.C. generator could be used as a motor to drive the engine at the test speeds and the brake out-put of the cradled unit would be equal to the friction horsepower of the engine (neglecting the small bearing friction present in the D.C. unit).

The data obtained with the sampling valve were the combustion and exhaust gas analysis by means of the Orsat analyzer and phase-to-phase pressures throughout the cycle. To insure accurate results the Orsat analyzer was completely cleaned and all tubes and rubber connections replaced and the CO_2, O_2, and CO cells were replenished with fresh chemicals. The exhaust gases are passed through the analyzer so that the gases are observed in the respective pipettes in the following order; CO_2, O_2, and CO. The pipettes contain solutions of 40 per cent caustic potash which absorbs CO_2, an alkaline solution of pyrogallic acid which absorbs the O_2, and an ammonical solution of cuprous chloride surrounded by some metallic copper to keep the solution energized, is used to absorb the CO.[1] The order of removal, as specified above, must be followed because each of the succeeding solutions will absorb any portion of the CO_2 and O_2 which remains in the sample. Even by following the correct procedure the sample should be flushed several times through each pipette before taking it to the next pipette so the correct percentage of each constituent is obtained from the analysis.

Before starting the tests the engine was started and loaded to the desired load and speed. When equilibrium of operating conditions was reached data were taken at 10° of crank travel increments throughout the engine cycle of 720°. Immediately upon obtaining a gas sample sufficient to analyze, the valve leading to the analyzer was closed, the one leading to the pressure gage was opened, and the pressure was allowed to build up to a maximum at the point of sampling while the Orsat was being used to obtain the percentages of the constituents in the combustion gases. This procedure made it possible to record the pressure and gas analysis under exactly the same operating condition, and at exactly the same position of crank travel of the engine, while maintaining a minimum continuous operating time of the sampling valve.

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1. Carroll M. Leonard, and Vladinir L. Maleev, <u>Heat Power Fundamentals</u>., pp. 350-352.

CALCULATIONS

The calculations are divided into two parts, that dealing with gas analysis, and that pertaining to pressures. The gas analysis calculations will be presented first.

A complete treatment of gas analysis for gasoline engines is presented in two NACA Technical Reports which were included in a Naval Reserve Officers Training course.[1]

The fuel to air ratio and completeness of combustion may be closely approximated by means of a common Orsat analysis of combustion gases. Unburned hydrocarbons and water or free carbon cannot be determined by it. In the case of gasoline-burning engines, determinations of unburned H_2 and CH_4 have been determined by a Bureau of Mines Orsat and it has been found that the H_2 is approximately 50% by volume of the CO and that there is always about .22% CH_4 in the dry exhaust gases. From these empirical relationships the air fuel ratio and completeness of combustion may be approximated.

The best approach to the analysis is by use of the mol system and, assuming that the total volume under consideration is 100 mols, then the % of each constituent is the % by volume of the total. From the Orsat analysis there will be determined the % of CO_2, O_2, and CO. By approximation there will be .22% CH_4 and .5(CO) will be the percent of H_2. The following computations are used to determine the air fuel ratio and completeness of

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[1] <u>NACA Technical Reports</u>, Nos. 476 and 616.

combustion:

1. The total air supplied in pounds = $\frac{100}{77} N_2 \times 28$
2. Carbon in fuel in pounds = 12 ($CO_2 + CO + CH_4$)
3. Hydrogen in fuel = (4 x $CH_4 + 2H_2 + 2H_2$ burned to H_2O)

   To find $\#H_2$ in fuel burned to H_2O:

   (a) The total $\#O_2$ in the air supplied to produce 100 mols of exhaust = 23% (total air supplied)

   (b) $\#O_2$ found in 100 mols of exhaust gases = 32 $(CO_2 + O_2 + \frac{CO}{2})$

   (c) $\#O_2$ to form H_2O = total $\#O_2$ to produce 100 mols of exhaust gases - $\#O_2$ found in 100 mols of exhaust gases

   (d) $\#H_2$ burned H_2O = $\frac{\#O_2 \text{ to form } H_2O}{8}$

4. Total fuel burned to form 100 mols of exhaust gas = total carbon in fuel + total H_2 in fuel
5. The air-fuel ratio = $\frac{\text{Total weight of air supplied}}{\text{Total weight of fuel}}$
6. Composition of fuel = $C_n H_m$

   (a) Mol weight of C_n in fuel = $\frac{\text{Total weight of carbon}}{12}$

   (b) Mol weight of H_2 in fuel = $\frac{\text{Total weight of } H_2 \text{ in fuel}}{2}$

   (c) Composition of fuel = $C_n H_m$ from parts (a) and (b) above

7. Completeness of combustion:

   (a) Total heat supplied to form 100 mols of exhaust gases = Total weight of fuel times the heating value of fuel. For gasoline it is 21,000 B.t.u./ pound.

(b) Heat lost in unburned materials in the exhaust gases is the percent of CO, CH_4, and H_2 multiplied by their respective heating values per mol. This is found to be 121,188 (CO), 121, 365 (H_2) and 380,981 (CH_4)

(c) Loss due to incomplete combustion = $\frac{\text{total heat lost}}{\text{total heat available}}$

The recording of the average pressure built up during each successive phase position of the sampling valve is recorded against degrees of crank travel of the engine and carried out at 6^{o} increments throughout the 720^{o} of crank travel per engine cycle. These pressure increments must be converted from crank angle or time increments to equivalent piston displacement increments. The construction of this pressure-volume diagram from the pressure-time diagram is done either graphically or analytically. The graphical solution offers sufficient accuracy and is a much more rapid method of converting from pressure-time coordinates to pressure-volume ones.[2]

The area of the pressure-volume diagram divided by the length of the displacement coordinate and multiplied by the scale factor of the pressure coordinate gives the mean effective pressure and by substituting this in the equation for the indicated horsepower, $IHP = \frac{PLAN}{33,000,}$ the power developed within the engine cylinder can be determined and hence the friction horsepower and mechanical

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2. *Elements of Diesel Engineering*, O.L. Adams, Sr. pp. 235-236.

efficiency. The friction horsepower represents the loss in power due to friction of the engine's working parts and is expressed as the difference between the brake horsepower and the indicated horsepower. The mechanical efficiency is simply the ratio of the brake horsepower to the indicated horsepower expressed as a percentage.

Upon constructing the pressure-volume diagram for some particular load on the engine, a study of the polytropic compression and expansion lines can be made by plotting on log paper these points from the expansion and compression lines. The slope of the line drawn through the average of these points equals the value of the exponent n in the polytropic equation PV^n equals a constant.

DISCUSSION AND RECOMMENDATIONS

The proof of the value of any piece of special test equipment such as the sampling valve depends upon the accuracy and consistency of the test results obtained from it. The most important function of the sampling valve is that it neither leaks when in closed position,nor leaks through the labyrinth seal above the gas outlet while a sample is being extracted. Leakage around the valve seat occured after a short period of operation necessitating the disassembly of the valve at frequent intervals and the relapping of the valve and seat to again obtain a gas tight seal. The first attempt to obtain cyclic pressure sampling was unsatisfactory because of the leakage encountered. This necessitated modifying the valve seat to a square shoulder instead of the original 30° seat and to change the poppet valve angle to 45°. In effect this gave a much larger force between valve and seat because of the reduced contact area for the same compressive force exerted by the spring.

X During the tests that were run, gas sampling for analysis was carried out only for the exhaust gases because the standard industrial Orsat which gives only the per cent of CO, CO_2, and O_2 was available for analyzing the products of combustion. Information pertaining to combustion phenomena could be attained only by use of a gas analyzer similar to the Bureau of Mines Orsat. An analyzer of this type was not available at this time therefore only the air-fuel ratios and combustion efficiencies were obtained.

The pressure sampling was carried out satisfactorily, however. The change described in valve seat design made the materials that were used for the valve and valve seat unsatisfactory. Because the valve functions as an exhaust valve during the power and exhaust strokes when sampling, it should be Stellite faced on its wearing surface and the valve seat should be made of cast iron which has good wear-resistant properties.

At the times the device was disassembled for relapping the valve, the gas seal formed by the close fit between the valve guide and valve was inspected and found to be very clean with a thin film of oil through its length. It was found that the cooling jacket allowed sufficient cooling of the valve and valve guide because at no time did the valve freeze in its guide. There was no evidence of oil leaking between the valve and valve guide into the stream of sampling gas because this would have left an obvious deposit of carbon around the point of leakage. The angular relationship between the phasing wheel and the degrees of engine crank travel made the point of sampling during the engine cycle a little difficult to establish, and a possible change in design so that one revolution of the phasing wheel corresponded to one revolution of the engine would simplify the phasing procedure. It would also be possible to link a reciprocating linkage to the phasing wheel, and hence an ordinary engine indicator could be attached to the phasing wheel so that the angular motion of the engine crank could be translated to a reciprocating motion corresponding to the motion of the engine piston. The gas sampling line could be connected to the indicator piston so that the stylus travel would correspond to the changing cylinder pressures. With this arrangement a pressure volume diagram would be built up directly on a point-by-point basis. If pressure time cards were desired, an indicator with a rotating drum could be connected directly to the phasing wheel shaft by means of a flexible drive cable and the pressure connection would be the same as in the first case.

Excessive wear was eliminated in the phasing gear train and valve actuating mechanism by having the parts running submerged in oil. The bulk of the sampling valve actuating mechanism and phasing gear train necessitated a very rigid bracing to insure that the assembly vibrated as a unit with the engine. This feature makes adaptability to other engines rather difficult, therefore possibly separating the valve and valve-actuating mechanism from the phasing gear train would have a decided advantage.

In the detailed study of combustion phenomena, a very short sampling period must be attained. This can be done by increasing the diameter of the cam, this also gives a smoother cam action against the valve. By increasing the diameter of the cam the rubbing velocity between the cam and valve would be increased, but with proper lubrication and selection of materials no excessive wear would result.

The most important feature that would insure a longer life sampling valve would be to incorporate into its design an adjustment on the lift of the valve so that it could be moved out of contact with the cam which operates it when sampling is not desired. This feature would have to be light in weight because it would become part of the reciprocating mass of the valve and, as was shown in the design calculation, this has an important effect on the operating speed of the valve.

The foregoing discussion has been to show the weak points of the present design and improvements that could be made which would lead to the construction of an improved sampling valve.

CONCLUSION

The exhaust gas analysis gives information pertaining to the air-fuel ratio and completeness of combustion. An examination of the curve, air-fuel ratio versus load (page 12a), shows that the carburetor was functioning properly because the air-fuel ratio curve follows that which is desirable for a spark ignition engine. The maximum economy air-fuel ratio for theoretical air and complete combustion is 15.2; this ratio should be 15.07 at 3/4 load on the engine tested. The air-fuel ratio for maximum power (12.7) should be approached for full load conditions of operation; for the engine tested it was found to be 12.78. The combustion efficiency follows the same trend as the air-fuel ratio curve (page 13a). The specific fuel consumption curve is inverted from the air-fuel ratio curve, but has the same general shape (page 13a). This will occur when the engine is being operated at the proper air-fuel ratio; in other words too large an air-fuel ratio would result in an increased specific fuel consumption.

It is interesting to note that the molecular formula for the fuel being used ranges between $C_8 H_{17.58}$ and $C_8 H_{17.8}$ as found by calculation (page 2a). The range for the premium grade gasoline used is within the range found above. The fact that the molecular formula computed is fairly constant shows that the combustion analysis determined by the Orsat was accurate.

The friction horsepower of the engine was found to be 2.95 at the test speed of 1800 rpm. This value was determined by direct measurement of the output of the D.C. generator directly connected to the engine when using a source of D.C. current to drive it as a motor. The friction horsepower thus determined can for all practical purposes be assumed to be constant throughout the load range at the test speed of 1800 rpm. The true indicated horsepower can be determined for each load on the engine by taking the sum of the friction horsepower, by direct measurement, and the brake horsepower. By comparing these values of indicated horsepower with those determined by calculations using pressure sampling data, it was found that at no load and 1/4 load the calculated indicated horsepowers were respectively 2.73 and 4.29 as against 2.95 and 4.35 by direct measurement.

The indicated horsepower as determined by calculation is slightly lower than that determined by direct measurement. This can be attributed to the fact that a sampling period of approximately 10^{o} is too long to accurately measure the phase-to-phase pressure during combustion and the early part of the expansion stroke. These pressures will appear lower than they actually are. As a result the area of the pressure-volume diagram will represent a lesser indicated horsepower than is actually developed within the engine cylinder. Sufficient data were not obtained at the higher load to make it possible to compute the indicated horsepower for the 1/2, 3/4, and full load conditions.

The values of the exponent n for compression and expansion at each load condition were plotted on log paper. No data were obtained for plotting the exponent n for expansion at the higher loads due to valve leakage and pressure fluctuation on the gage, but values of 1.4 to 1.38 were obtained for no load and 1/4 load respectively. The values of n for compression are fairly consistant for all loads. The values ranged from 1.32 at no load to 1.23 at full load. V.L. Maleev states in his book that the value of n decreases with an increase in temperature.[1] The value of n for compression decreases with increase of load because the operating temperature of the engine increases. It is noticed that at the end of the compression stroke a curve drawn through the points plotted would give a line whose slope would be less than the average value. This is explained by V.L. Maleev as follows: Since the heat exchange during the compression stroke is not uniform, the exponent n_c is not constant. During the latter part of the stroke the heat produced during compression cannot be readily dissipated through the decreasing cylinder wall area in contact with the gases; thus the temperature is raised which consequently lowers the value of n_c. The values of n_c vary in different engines from 1.28 to 1.38.[2]

During the expansion stroke the heat exchange is not constant. Therefore the value of n_e will vary throughout the stroke. The early part of the expansion stroke gives a value of n_e greater than the average value plotted because of the high rate of heat transfer through the engine's cylinder walls. Later in the stroke the value of n_e shows a decrease below the average value. This is caused by after-burning which retards the temperature decline. During the latter part of the stroke n_e gradually in-

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1. V.L. Maleev, Internal Combustion Engines., First Edition pp. 23.

2. Ibid., pp. 54.

creases. V.L. Maleev's discussion of the change of the exponent n_e during the expansion stroke verifies the above results.[3]

It is believed that the sampling time of the valve included too many degrees of rotation of the engine crank. This condition made the measurement of the phase-to-phase pressures during combustion and the early part of the expansion stroke in error as the pressure gage indication was difficult to read. During compression and expansion strokes with light loads on the engine the pressure fluctuation in the gage was somewhat reduced, making possible more accurate reading of the pressure gage.

The engine tested was not mounted firmly enough to the floor and as a result vibration of the unit was particularly severe. It is believed that this was a contributing factor to the short life of the driving mechanism which operated the valve. This wearing of the parts made the drive shaft within the phasing gear housing flexible and instead of the cam being rigid and forcing the valve open when its lobe came in contact with the valve stem, it flexed upward away from the valve stem when contact was made, and reduced the lift of the valve. The small lift resulting destroyed the scuffing and seating properties of the valve upon closing and allowed a deposit of carbon and lead to form when sampling during combustion. Resulting leakage made it impossible to record high pressure during combustion and the early part of the expansion stroke, because the pressure which was built up in the pressure gage by sampling through successive cycles would slowly drop because of gas leakage.

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[3] Ibid., pp. 56.

During the time that the driving mechanism described above was rigid and the valve seated properly, the data taken were reliable as previously shown. It can be concluded that with the recommendations previously suggested and their application to the revision of the present design of the sampling valve, a reliable and long-lived device can be constructed.

APPENDIX A

Test Data and Calculations

Engine tested: Wisconsin, single cylinder, air cooled, 4 stroke, cycle, Model AGH 3 1/2" x 4" engine; compression ratio 4.6:1; maximum bhp 8.4, continuous bhp 6.7, at 2200 rpm.

Test conditions: Zero to 6.41 bhp at 1800 rpm in 4 increments; fuel used was ethyl gasoline.

Method of loading engine: Cradled D.C. generator dynameter directly coupled to the engine; generator loaded by use of a brine tank; 14" lever arm extending from generator housing to pedestal on scales gives load to balance by scales.

Combustion Data and Calculations

TABLE I

ORSAT ANALYSIS % BY VOLUME

Load	# Load on scales	bhp	N_2	CO_2	CO	O_2	H_2	CH_4
0		0	76.88	7.6	10.0	.3	5.0	.22
1/4		1.605	81.23	10.5	4.3	1.6	2.15	.22
1/2		3.205	81.73	10.7	3.6	1.9	1.8	.22
3/4	12	4.81	82.48	10.9	2.8	2.2	1.4	.22
1	16	6.41	79.73	9.4	6.3	1.2	3.15	.22

TABLE II

AIR FUEL RATIO

(Calculation based on 100 mols of exhaust gases)

Load	#Air supplied	#Carbon in fuel	#O_2 in air supplied
0	2795	214	643
1/4	2950	180.3	679
1/2	2970	174.5	684
3/4	2995	167	680
1	2900	191.2	667

TABLE II (continued)

#O_2 in exhaust gases	#O_2 to form H_2O	#H_2 to form H_2O	#H_2 in fuel	#Fuel	Air Fuel Ratio
643	230	28.75	39.63	253.6	11.02
679	223	27.9	33.08	213.4	13.85
461	223	27.9	32.38	206.81	14.35
465	215	26.9	30.58	197.58	15.07
440	226	28.25	35.33	226.73	12.78

TABLE III

MOLECULAR FORMULA FOR FUEL

Load	Mols of carbon per 100 mols of exhaust gases	Mols of H_2 per 100 mols of exhaust gases	Formula $C_n\ H_m$
0	17.82	39.63	$C_8\ H_{17.75}$
1/4	15.02	33.08	$C_8\ H_{17.6}$
1/2	14.52	32.38	$C_8\ H_{17.8}$
3/4	13.92	30.58	$C_8\ H_{17.58}$
1	15.92	35.33	$C_8\ H_{17.7}$

TABLE IV

COMBUSTION EFFICIENCY

Based on 100 mols of exhaust gases

Load	Heat loss, B.t.u. in CO CO (121,188)	Heat loss, B.t.u. in H_2 H_2 (121,365)	Heat loss, B.t.u. in CH_4 CH_4 (380,981)
0	1,211,880	607,000	83,800
1/4	524,000	261,000	83,800
1/2	439,000	218,500	83,800
3/4	341,500	170,000	83,800
1	767,000	382,000	83,800

TABLE IV (continued)

Total heat loss, B.t.u.	Total heat available, B.t.u. #fuel (20,750)	Combustion loss per cent	Combustion efficiency per cent
1,902,800	5,260,000	36.2	63.8
868,800	4,430,000	19.6	80.4
741,300	4,300,000	17.23	82.77
595,300	4,100,000	14.51	85.49
1,232,800	4,710,000	26.2	73.8

Pressure Data and Calculations

TABLE V

COMPRESSION PRESSURES

PSIA

Degrees of crank travel from b.d.c.	No load	1/4 load	1/2 load	3/4 load	full load
0	16	16	17	17	17
6	16	16	17	17	17
12	16	16	17	17	18
18	17	17	18	18	18
24	17	17	18	18	19
30	18	18	19	19	19
36	18	18	19	19	20
42	18	18	20	19	20
48	19	19	20	20	21
54	19	19	20	21	21
60	20	20	20	22	22
66	21	21	22	23	23
72	22	23	23	24	25
78	23	24	25	26	26
84	25	25	27	28	28
90	27	27	29	29	30
96	29	29	31	32	32
102	31	32	33	34	36
108	35	35	36	38	38
114	37	37	38	39	40
120	41	42	43	45	47

TABLE V (continued)

Degrees of crank travel from b.d.c.	No load	1/4 load	1/2 load	3/4 load	full load
126	45	46	48	51	51
132	49	50	54	56	56
138	53	57	58	61	62
144	58	62	63	65	67
150	62	64	66	69	71
156	64	65	69	72	74
162	64	66	70	73	75
168	65	67	71	74	76
174	--	--	--	--	--
180	--	--	--	--	--

TABLE VI

COMBUSTION PRESSURES

PSIA

Degrees of crank travel from t.d.c.	No load	1/4 load	1/2 load	3/4 load	full load
0	---	---	---	---	---
6	---	---	---	---	---
12	---	---	---	---	---
18	---	---	---	---	---
24	200	220	---	---	---
30	180	200	---	---	---
36	140	160	---	---	---
42	115	135	---	---	---
48	90	115	---	---	---
54	80	105	100	---	---
60	68	93	90	---	---
66	64	80	85	---	---
72	60	70	80	---	---
78	56	65	74	---	---
84	47	58	67	---	---
90	44	52	64	---	---
96	41	49	59	---	---
102	39	47	54	---	---
108	37	45	50	---	---
114	35	43	48	---	---
120	33	40	44	---	---

TABLE VI (continued)

Degrees of crank travel from t.d.c.	No load	1/4 load	1/2 load	3/4 load	full load
126	32	38	41	---	---
132	31	37	39	---	---
138	30	36	37	---	---
144	28	33	35	---	---
150	25	31	34	---	---
156	20	28	32	---	---
162	18	24	26	---	---
168	17	18	20	---	---
174	16	18	18	---	---
180	16	18	18	---	---

TABLE VII

Crank Angle Versus Piston Displacement and Cylinder Volume

$$X = r\ (1 - \cos\alpha - \frac{r \sin^2\alpha}{2l})$$

Where;

X = distance from t.d.c. in inches

l = length of connecting rod which is 9 inches

r = radius of crank which is 4 inches

$$X_2 = \frac{V_1 - V_2}{r-1} = \frac{3.5^2 \times .7854 \times 4}{4.6-1} = \frac{384}{3.6} = 10.67 \text{ in.}^3$$

X_2 = clearance volume in.^{3}

$V_1 - V_2$ = piston displacement in.^{3}

r = compression ratio

Degrees from t.d.c.	% of stroke	Distance	Piston displacement in.^{3}	Cylinder volume in.^{3}
0	0	0	0	10.67
6	.54	.0216	.208	10.878
12	1.46	.0584	.562	11.232
18	3.14	.1255	1.208	11.878
24	5.46	.218	2.10	12.77
30	8.1	.324	3.12	13.79
36	11.6	.466	4.48	15.15
42	15.4	.617	5.94	16.61
48	19.7	.787	7.57	18.24
54	24.3	.974	9.36	20.03
60	29.2	1.168	11.22	21.89
66	36.8	1.40	14.12	24.79
72	39.7	1.584	15.23	25.90
78	45.0	1.79	17.22	27.89

TABLE VII (continued)

Degrees from t.d.c.	% of stroke	Distance	Piston displacement in.^{3}	Cylinder volume in.^{3}
84	50.3	2.01	19.35	30.02
90	55.6	2.22	21.4	32.07
96	60.7	2.425	23.35	34.02
102	65.7	2.625	25.25	35.92
108	70.5	2.82	27.10	37.77
114	74.9	2.99	28.8	39.47
120	79.2	3.16	30.4	41.07
126	82.92	3.31	31.8	43.47
132	86.4	3.45	33.2	43.87
138	89.6	3.58	34.4	45.07
144	91.7	3.64	35.0	45.67
150	94.7	3.78	36.4	47.07
156	96.4	3.85	37.0	47.67
162	98.0	3.91	37.6	48.27
168	99.0	3.95	38.1	48.77
174	99.6	3.99	38.4	49.07
180	100.	4.00	38.45	49.12

TABLE VIII

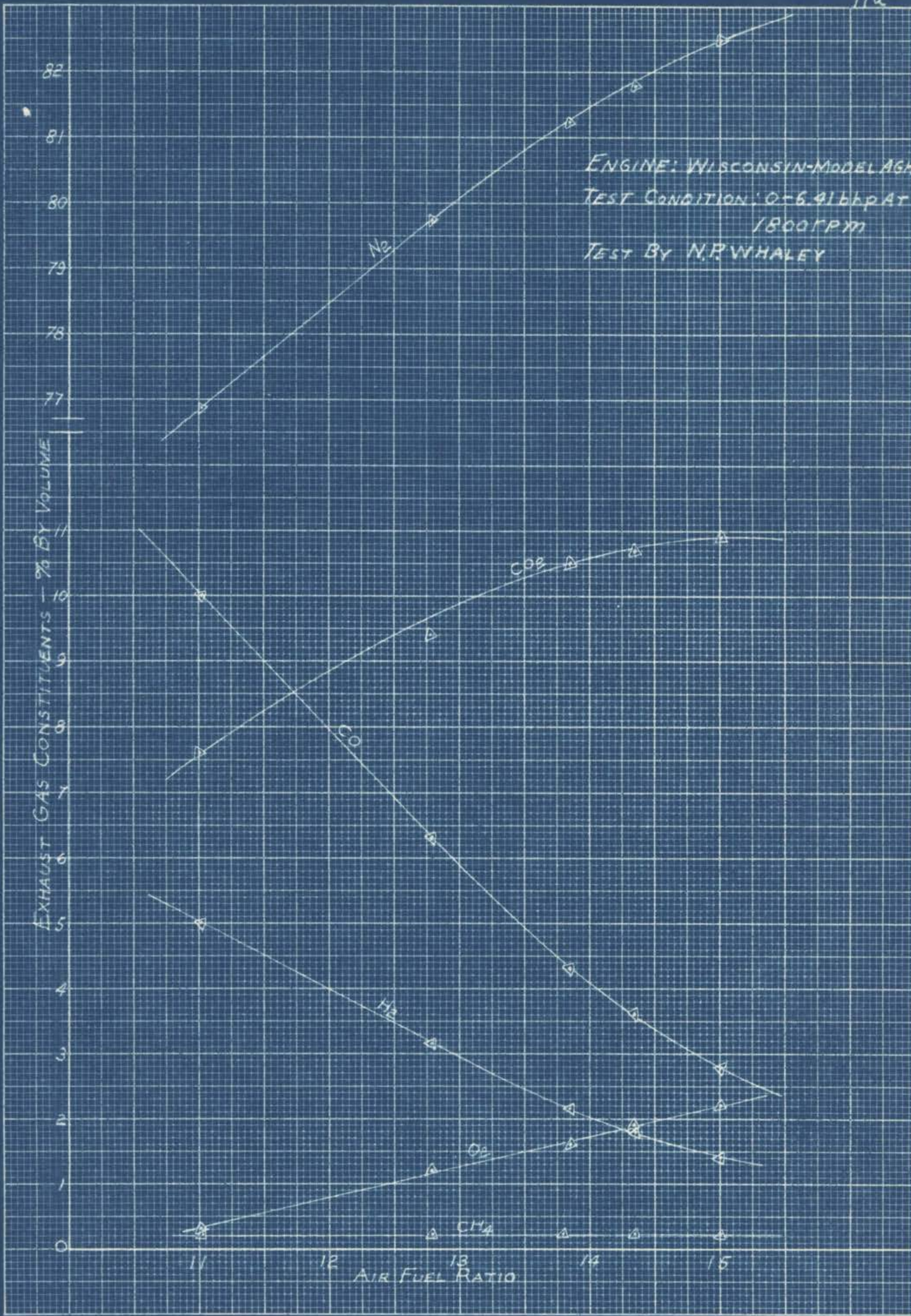
INDICATOR CARD DETERMINATIONS

Load conditions	Mean effective pressure	Indicated horsepower	Brake horsepower	Friction horsepower	Mechanical efficiency
0	31.25	2.73	0	2.73	0
1/4	49.2	4.29	1.605	2.685	37.5
1/2	----	----	3.205	----	----
3/4	----	----	4.81	----	----
1	----	----	6.41	----	----

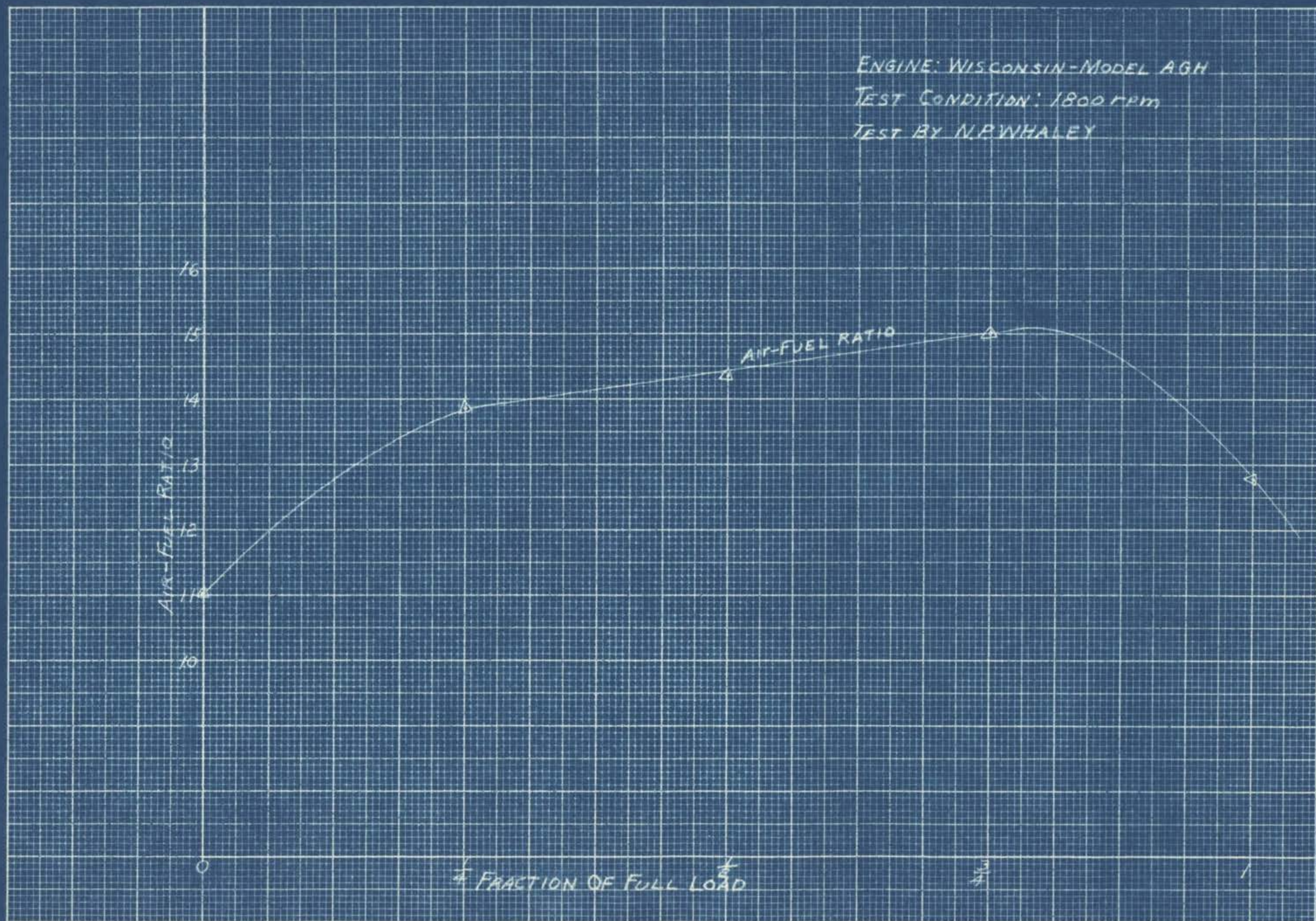
TABLE IX

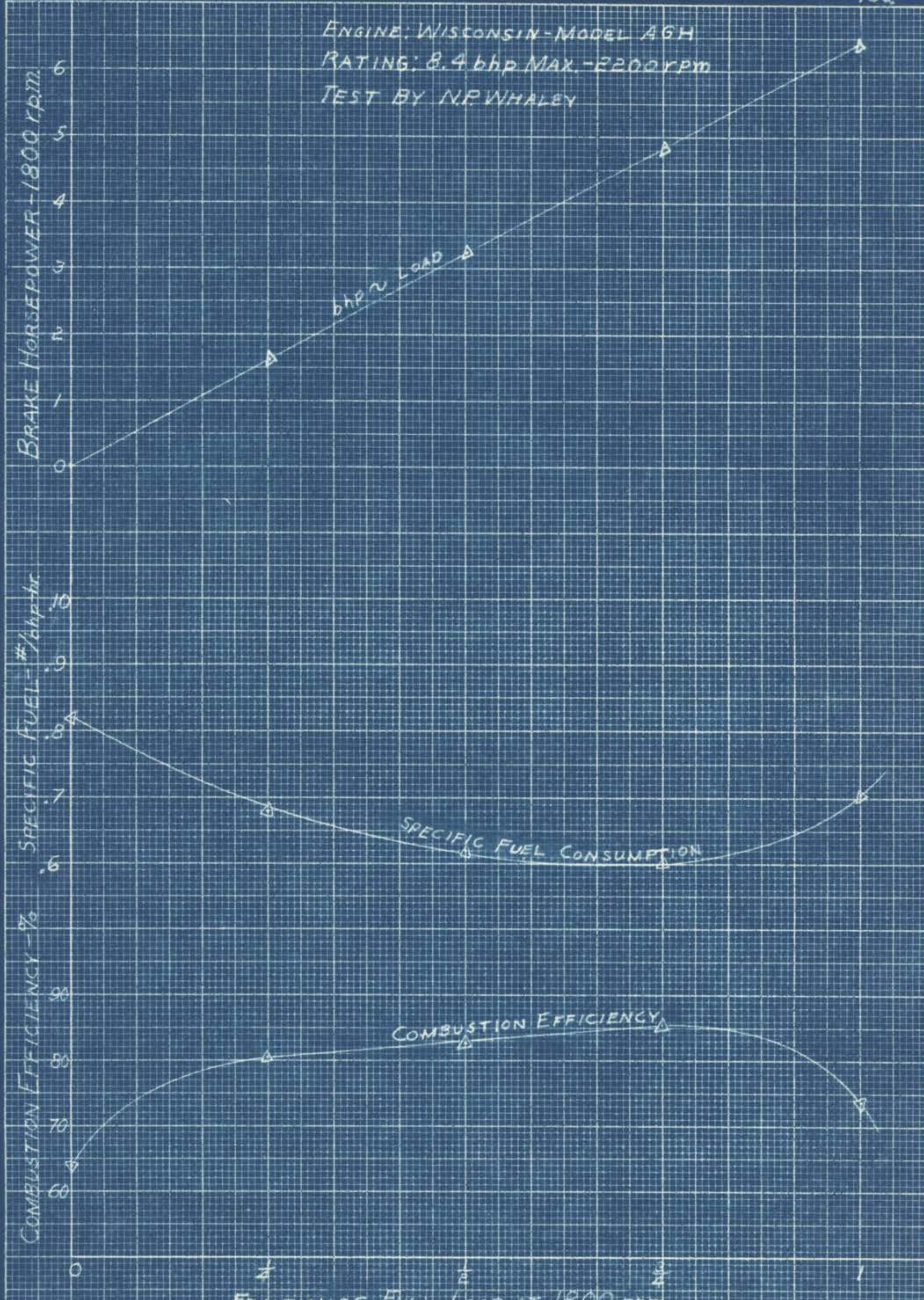
DETERMINATION OF n_c AND n_e FOR THE POLYTROPIC CURVES

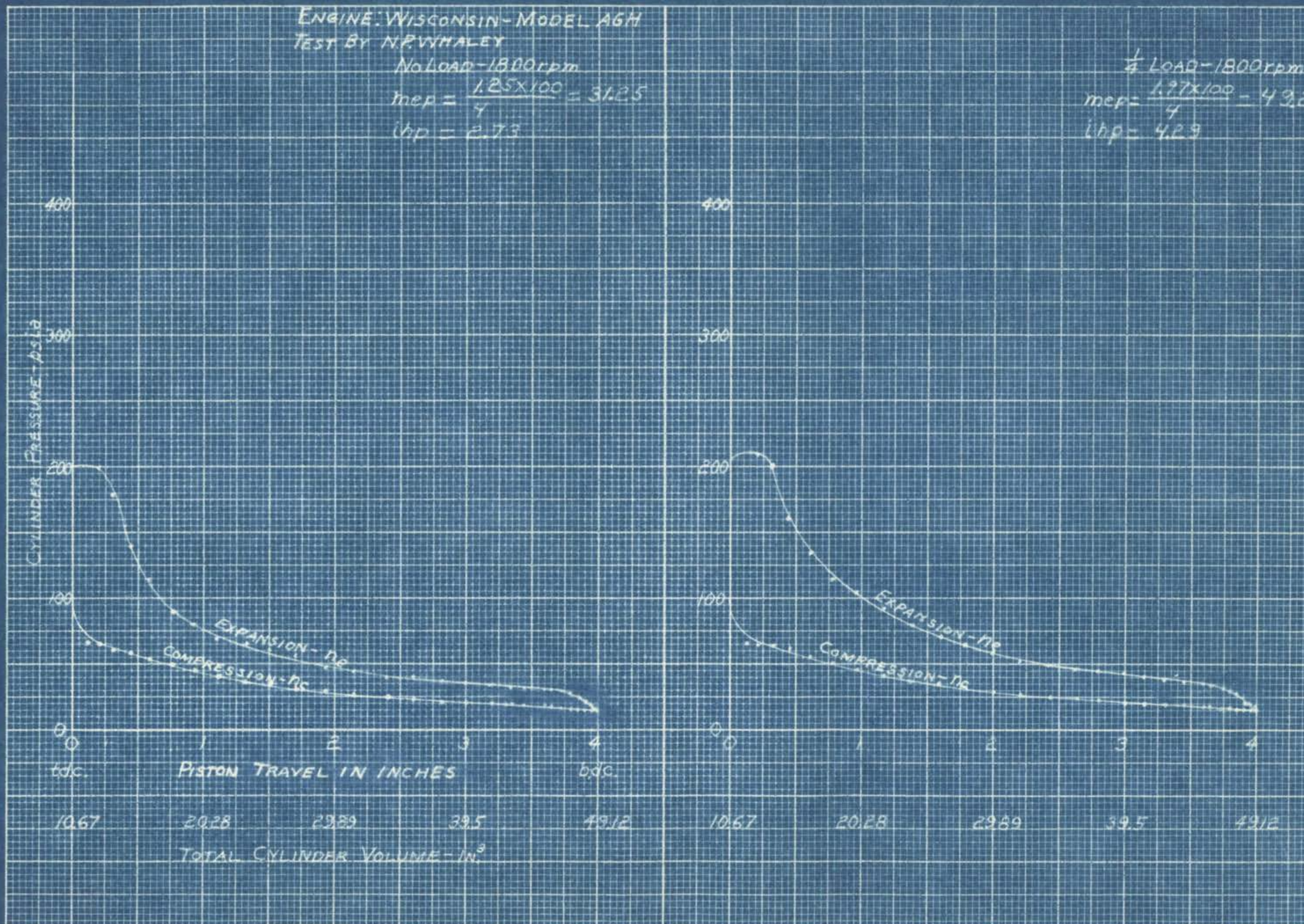
Load condition	Compression n_c	Expansion n_e
0	1.32	1.4
1/4	1.29	1.38
1/2	1.26	1.39
3/4	1.24	----
1	1.23	----

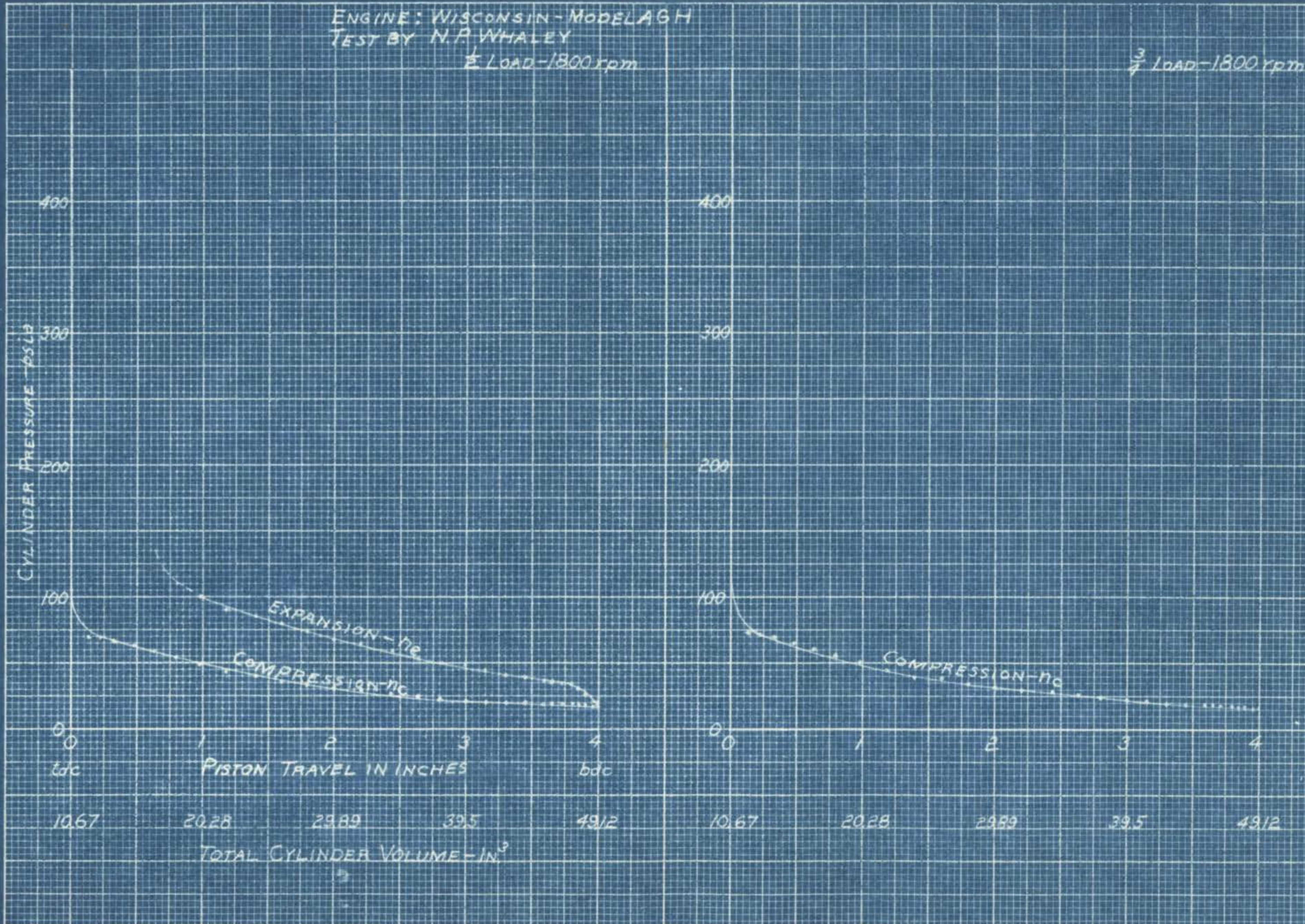


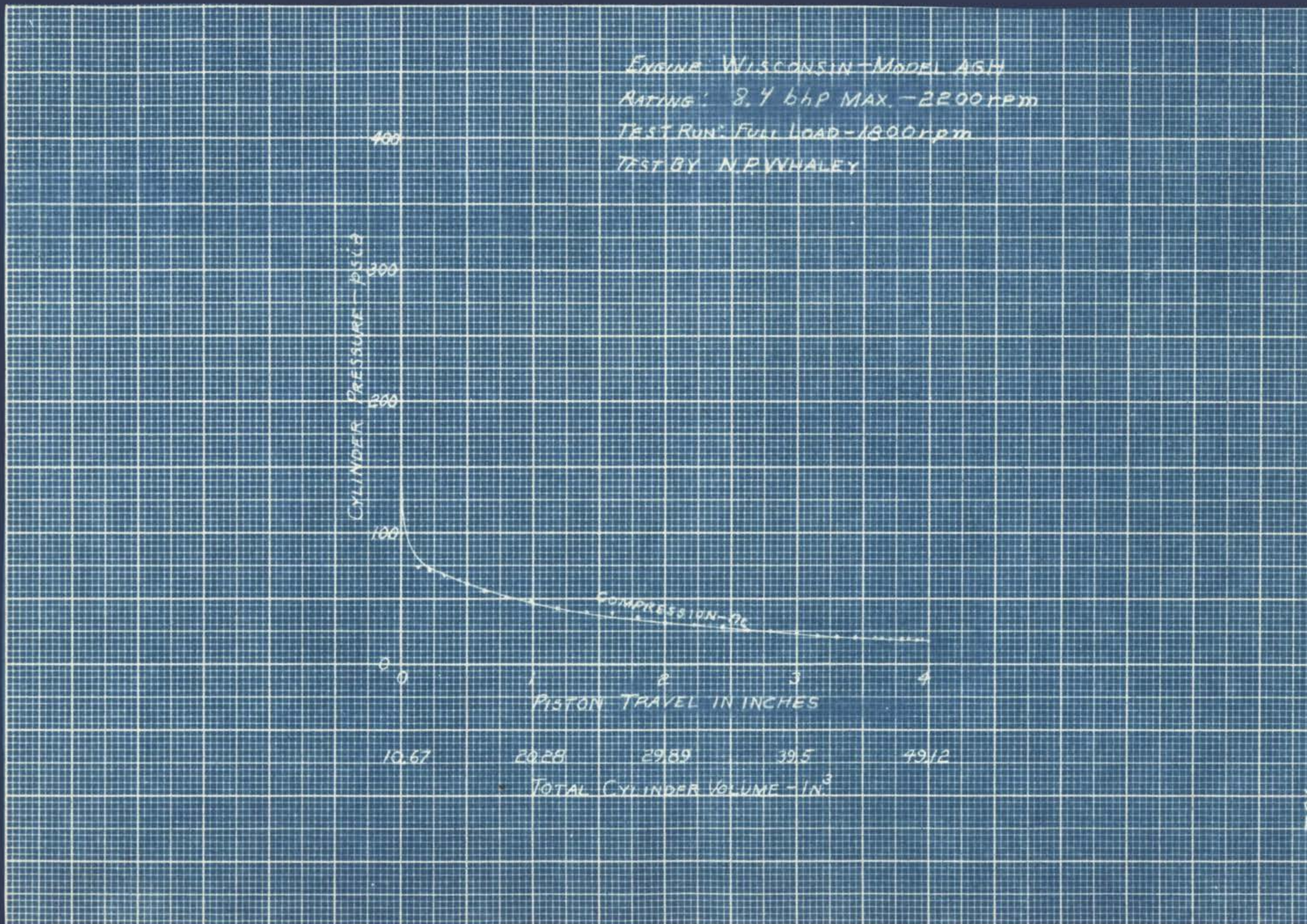
Engine: Wisconsin-Model AGH
Test Condition: 0-6.41 bhp At 1800 rpm
Test By N.P. Whaley
Exhaust Gas Constituents – % By Volume
Air Fuel Ratio
N2
CO2
CO
H2
O2
CH4
82
81
80
79
78
77
11
10
9
8
7
6
5
4
3
2
1
0
11
12
13
14
15

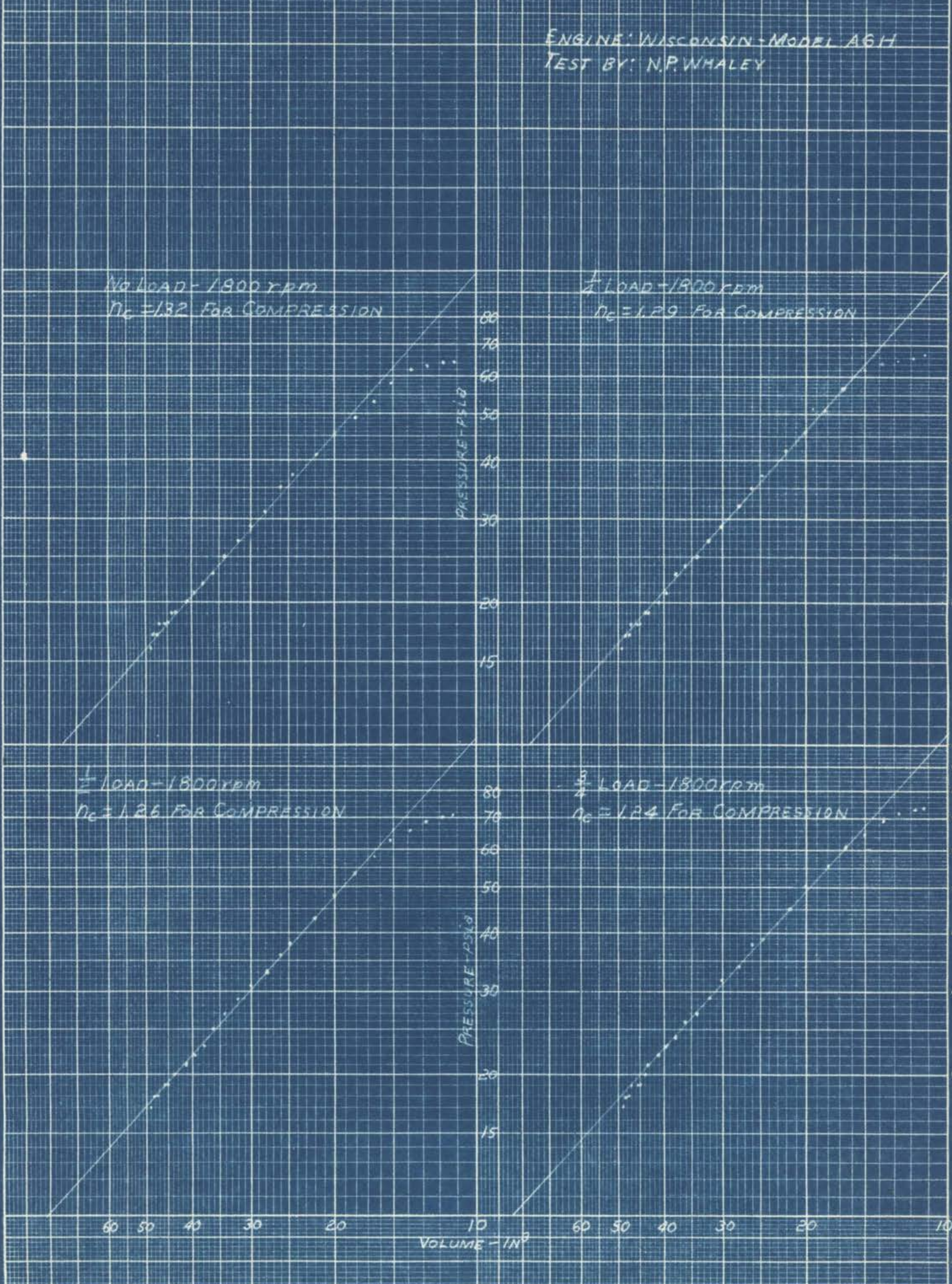

ENGINE: WISCONSIN-MODEL AGH
TEST CONDITION: 1800 rpm
TEST BY N.P. WHALEY
AIR-FUEL RATIO
16
15
14
13
12
11
10
0
1/4
1/2
3/4
1
FRACTION OF FULL LOAD

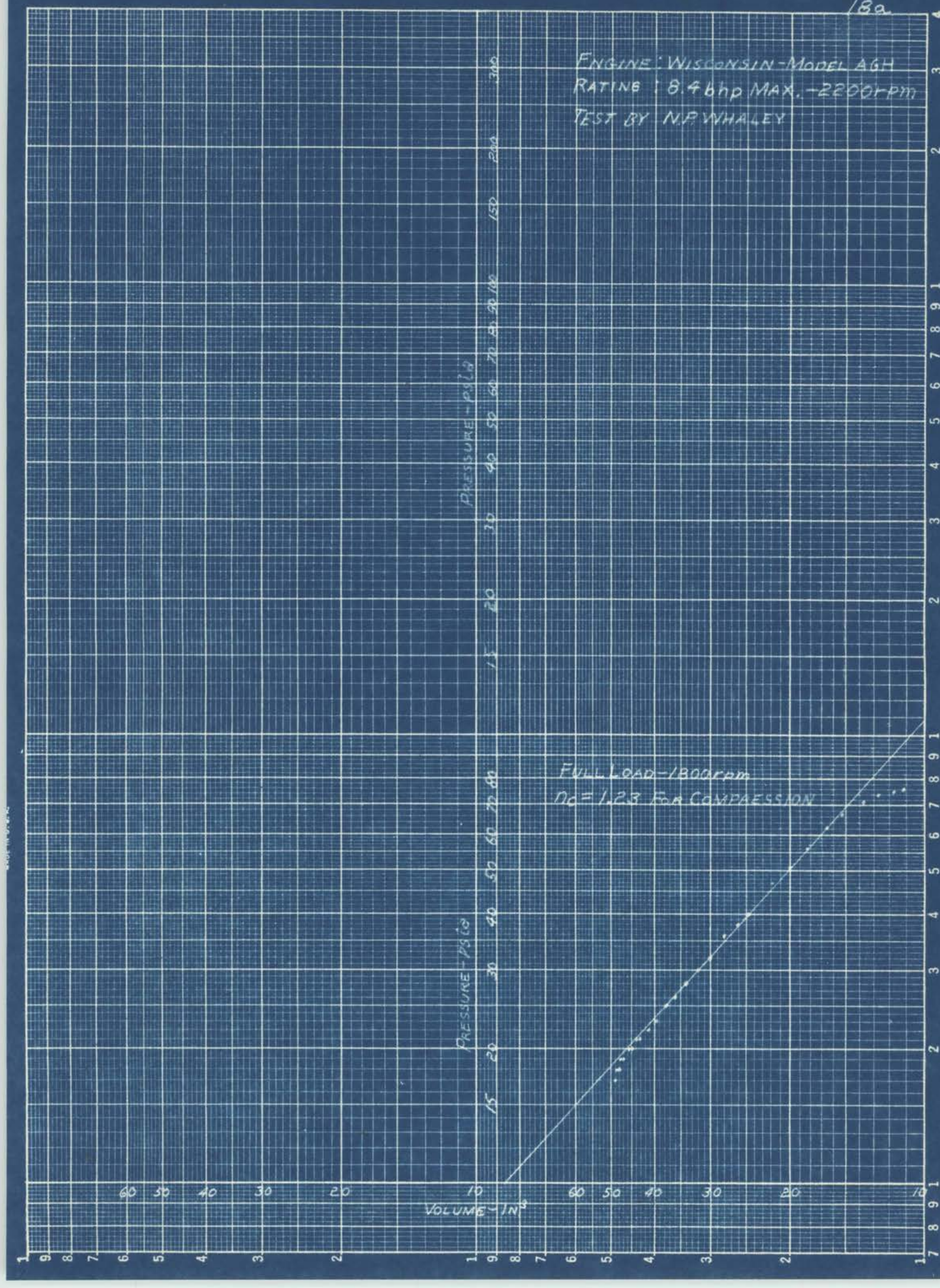

ENGINE: WISCONSIN-MODEL AGH
RATING: 8.4 bhp MAX.-2200 rpm
TEST BY N.P. WHALEY
BRAKE HORSEPOWER-1800 r.p.m.
bhp ~ LOAD
SPECIFIC FUEL-#/bhp-hr.
SPECIFIC FUEL CONSUMPTION
COMBUSTION EFFICIENCY-%
COMBUSTION EFFICIENCY
FRACTION OF FULL LOAD AT 1800 rpm.
0
1/4
1/2
3/4
1

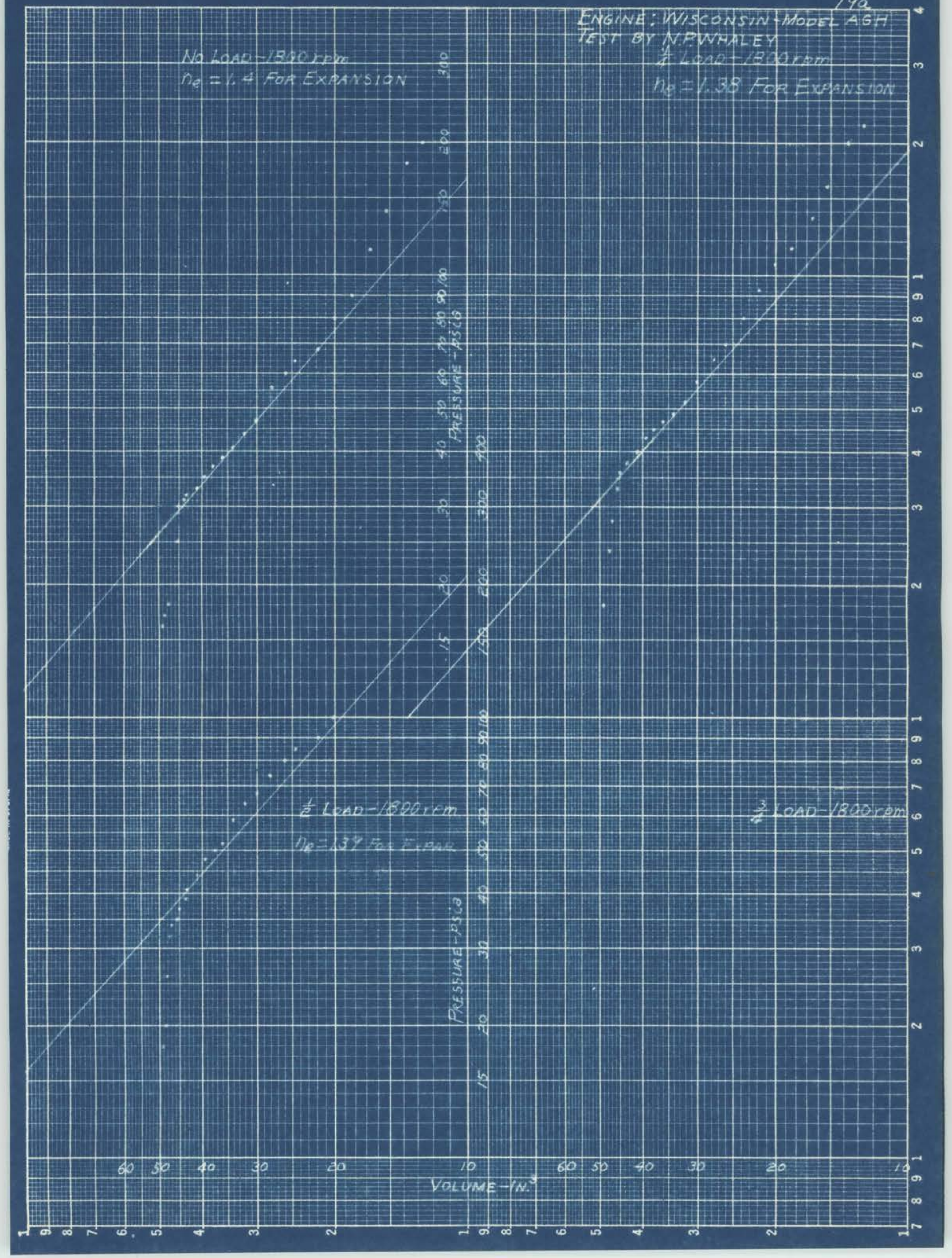

ENGINE: WISCONSIN - MODEL AGH
TEST BY N.P. WHALEY
No LOAD - 1800 rpm
mep = 1.25×100 / 4 = 31.25
ihp = 2.73
¼ LOAD - 1800 rpm
mep = 1.97×100 / 4 = 49.2
ihp = 4.29
CYLINDER PRESSURE - psia
400
300
200
100
0
EXPANSION - ne
COMPRESSION - nc
0
1
2
3
4
tdc.
bdc.
PISTON TRAVEL IN INCHES
10.67
20.28
29.89
39.5
49.12
TOTAL CYLINDER VOLUME - IN.³


ENGINE: WISCONSIN - MODEL AGH
TEST BY N.P. WHALEY
½ LOAD-1800 rpm
¾ LOAD-1800 rpm
CYLINDER PRESSURE - psia
400
300
200
100
0
EXPANSION - ne
COMPRESSION - nc
0
1
2
3
4
tdc
bdc
PISTON TRAVEL IN INCHES
10.67
20.28
29.89
39.5
49.12
TOTAL CYLINDER VOLUME - IN³

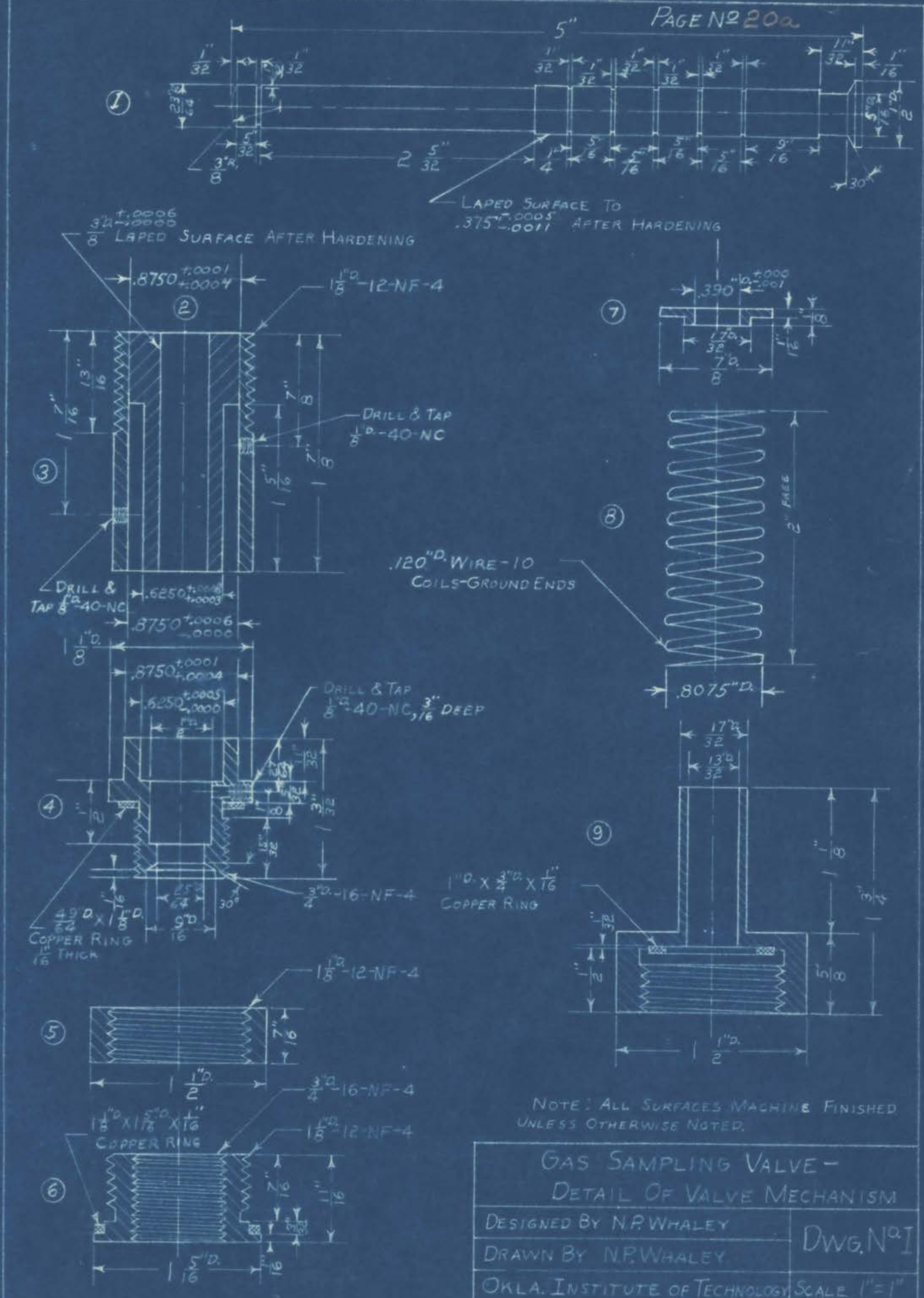

ENGINE WISCONSIN-MODEL AGH
RATING: 8.4 bhp MAX.-2200 rpm
TEST RUN: FULL LOAD-1800 rpm
TEST BY N.P. WHALEY
CYLINDER PRESSURE-psia
400
300
200
100
0
COMPRESSION-PC
0
1
2
3
4
PISTON TRAVEL IN INCHES
10.67
20.28
29.89
39.5
49.12
TOTAL CYLINDER VOLUME-IN^3


ENGINE: WISCONSIN-MODEL A6H
TEST BY: N.P. WHALEY
NO LOAD-1800 rpm
n_c = 1.32 FOR COMPRESSION
$\frac{1}{4}$ LOAD-1800 rpm
n_c = 1.29 FOR COMPRESSION
$\frac{1}{2}$ LOAD-1800 rpm
n_c = 1.26 FOR COMPRESSION
$\frac{3}{4}$ LOAD-1800 rpm
n_c = 1.24 FOR COMPRESSION
PRESSURE - PSIG
VOLUME - IN^3

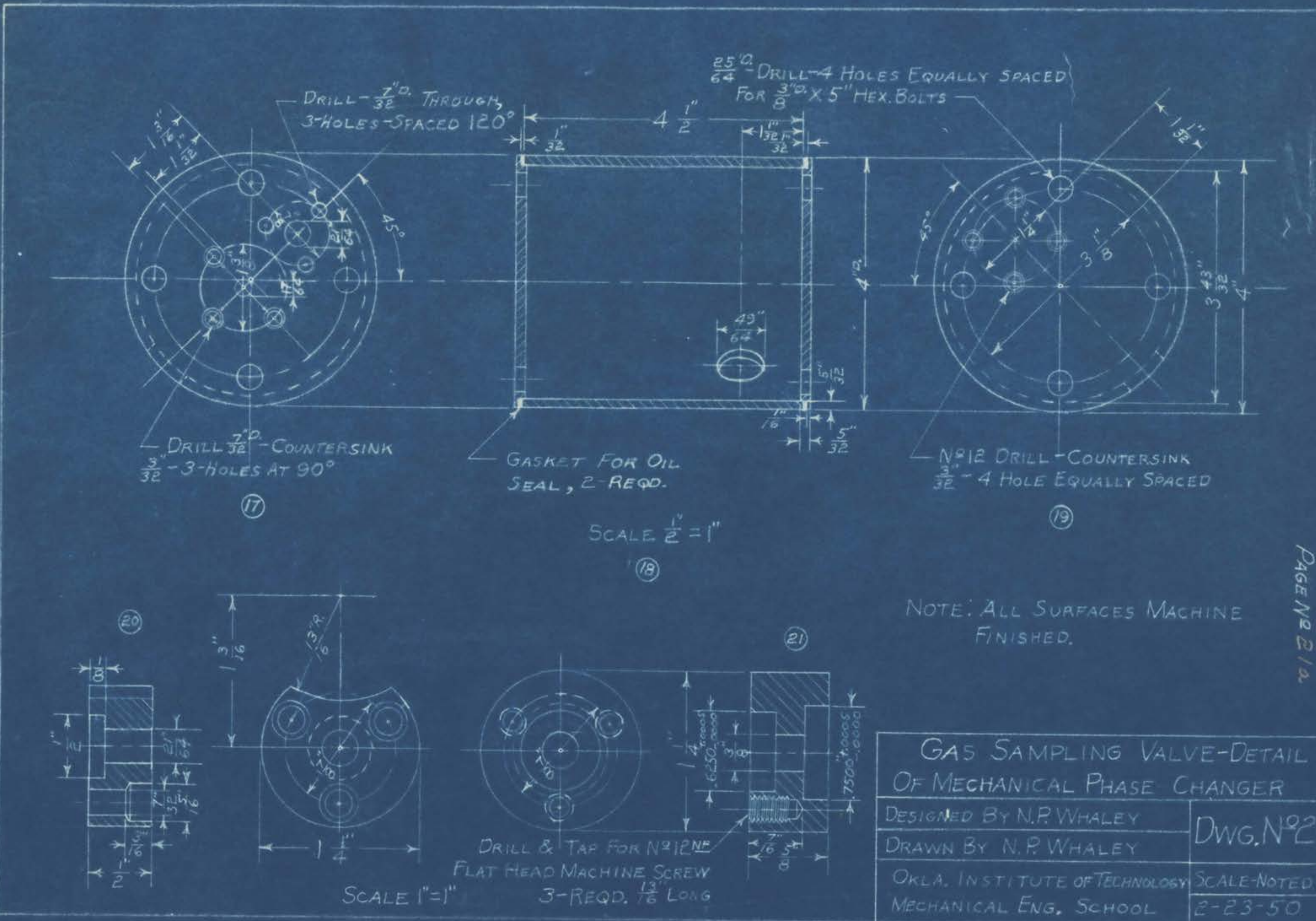

18a
ENGINE: WISCONSIN-MODEL AGH
RATING: 8.4 bhp MAX. - 2200 rpm
TEST BY N.P. WHALEY
FULL LOAD - 1800 rpm
n_c = 1.23 FOR COMPRESSION
PRESSURE - PSIA
PRESSURE - PSIG
VOLUME - IN³

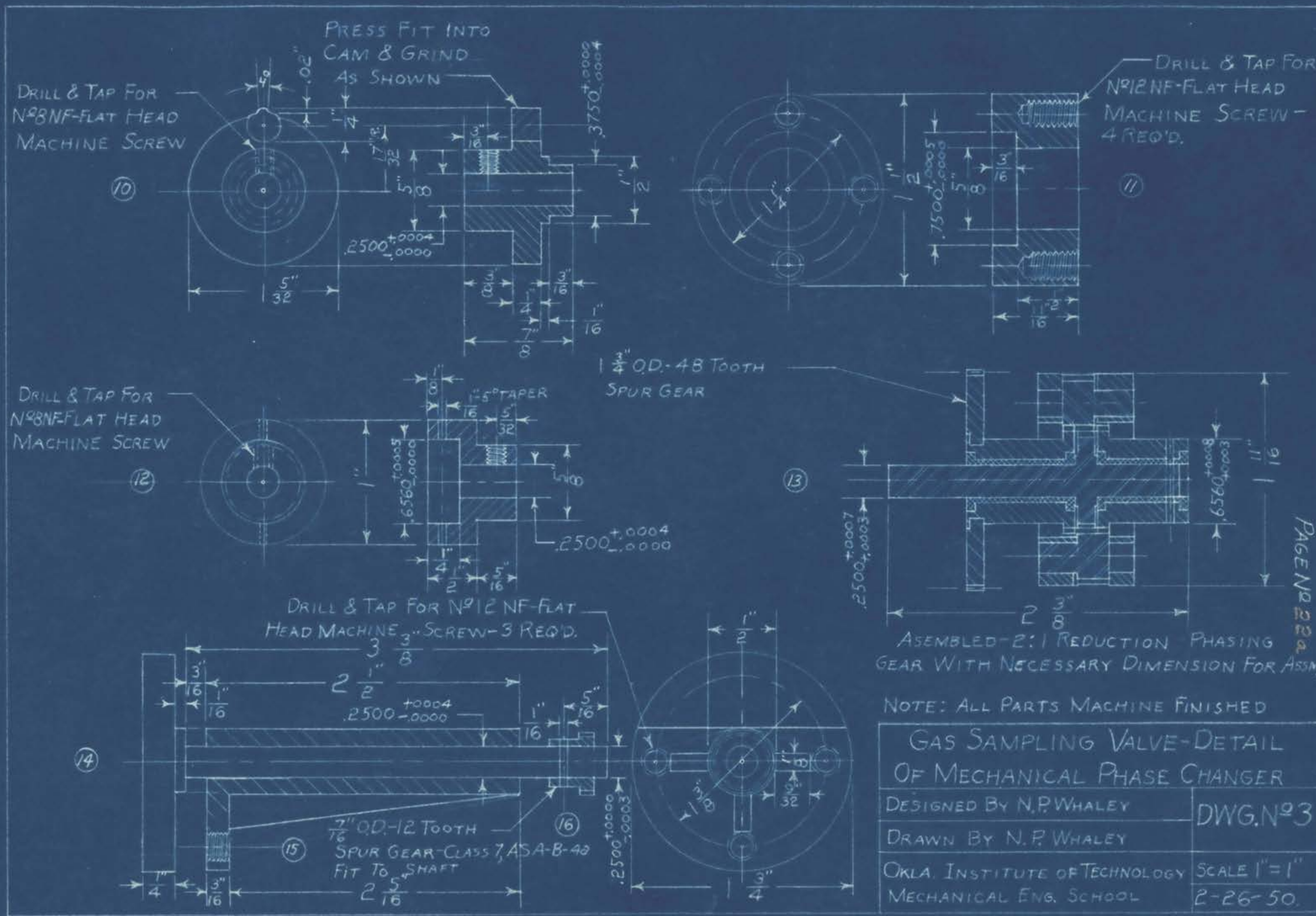


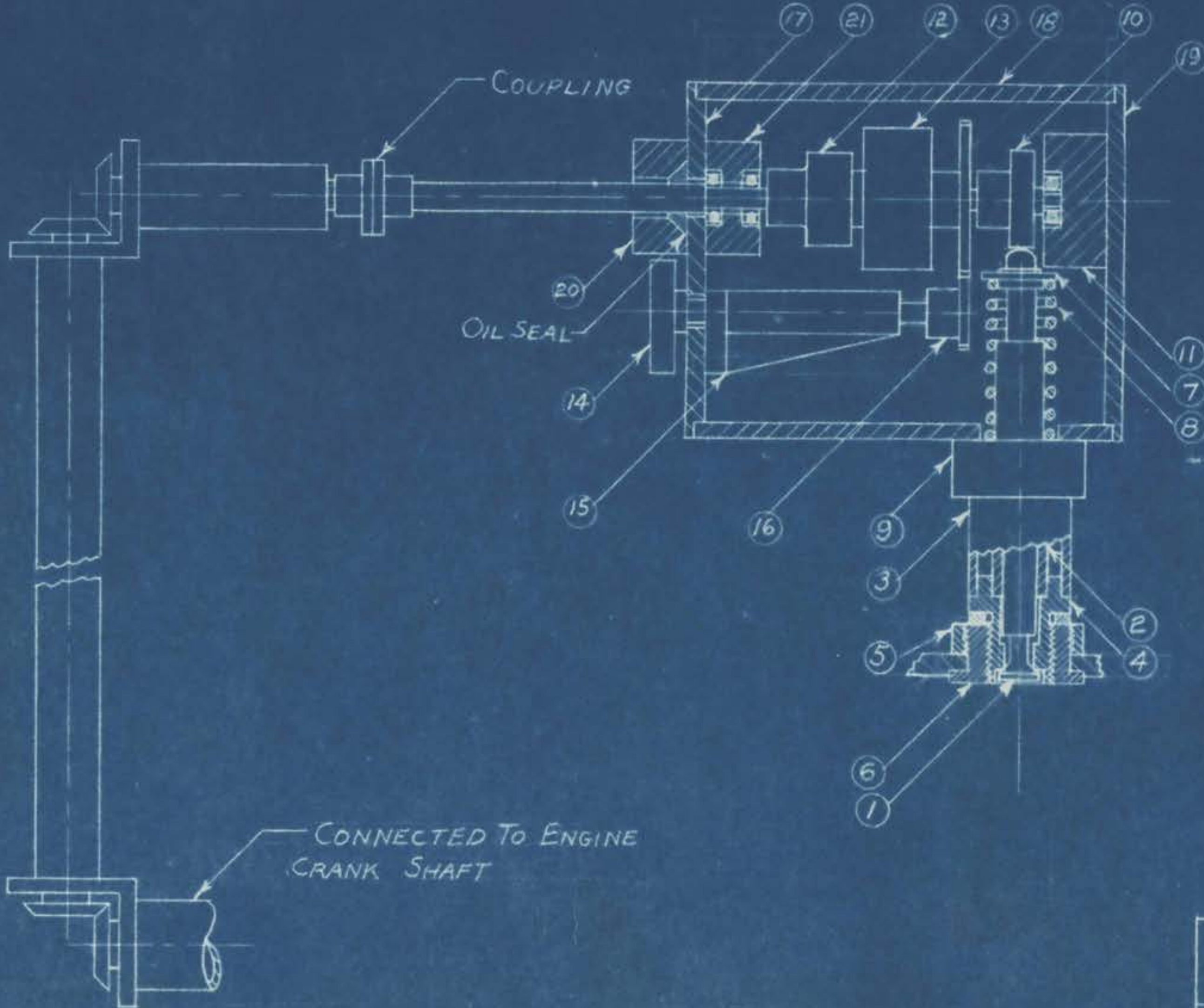
19a
ENGINE: WISCONSIN-MODEL AGH
TEST BY N.P. WHALEY
NO LOAD-1800 rpm
n_e = 1.4 FOR EXPANSION
¼ LOAD-1800 rpm
n_e = 1.38 FOR EXPANSION
½ LOAD-1800 rpm
n_e = 1.37 FOR EXPAN.
¾ LOAD-1800 rpm
PRESSURE-PSIG
VOLUME-IN.³



LAPED SURFACE TO .375 +.0005 -.0011 AFTER HARDENING
3/8 D +.0006 -.0000 LAPED SURFACE AFTER HARDENING
1 1/8 D - 12-NF-4
DRILL & TAP 1/8 D - 40-NC
DRILL & TAP 1/8 D - 40-NC
DRILL & TAP 1/8 D - 40-NC, 3/16 DEEP
3/4 D - 16-NF-4
49/64 D X 1 1/8 D COPPER RING 1/16 THICK
1 D X 3/4 D X 1/16 COPPER RING
.120 D WIRE - 10 COILS-GROUND ENDS
2" FREE
.8075" D.
1 1/8 D X 1 5/16 D X 1/16 COPPER RING
NOTE: ALL SURFACES MACHINE FINISHED UNLESS OTHERWISE NOTED.
GAS SAMPLING VALVE - DETAIL OF VALVE MECHANISM
DESIGNED BY N.P. WHALEY
DRAWN BY N.P. WHALEY
OKLA. INSTITUTE OF TECHNOLOGY
SCHOOL OF MECHANICAL ENG.
DWG. Nº I
SCALE 1" = 1"
2-20-50


DRILL - 7/32" D. THROUGH, 3-HOLES-SPACED 120°
DRILL 7/32" D. - COUNTERSINK 3/32" - 3-HOLES AT 90°
(17)
25/64" D. - DRILL - 4 HOLES EQUALLY SPACED FOR 3/8" D. X 5" HEX. BOLTS
4 1/2"
GASKET FOR OIL SEAL, 2-REQD.
SCALE 1/2" = 1"
(18)
№ 12 DRILL - COUNTERSINK 3/32" - 4 HOLE EQUALLY SPACED
(19)
NOTE: ALL SURFACES MACHINE FINISHED.
(20)
(21)
DRILL & TAP FOR № 12 NF FLAT HEAD MACHINE SCREW 3-REQD. 13/16" LONG
SCALE 1" = 1"
GAS SAMPLING VALVE-DETAIL OF MECHANICAL PHASE CHANGER
DESIGNED BY N.P. WHALEY
DRAWN BY N.P. WHALEY
DWG. № 2
OKLA. INSTITUTE OF TECHNOLOGY MECHANICAL ENG. SCHOOL
SCALE-NOTED
2-23-50


PAGE Nº 22a
PRESS FIT INTO CAM & GRIND AS SHOWN
DRILL & TAP FOR Nº8 NF-FLAT HEAD MACHINE SCREW
.3750 +.0000 -.0004
.2500 +.0004 -.0000
⑩
DRILL & TAP FOR Nº12 NF-FLAT HEAD MACHINE SCREW - 4 REQ'D.
.7500 +.0005 -.0000
⑪
1 3/4" O.D.-48 TOOTH SPUR GEAR
DRILL & TAP FOR Nº8 NF-FLAT HEAD MACHINE SCREW
1° 5° TAPER
.6560 +.0005 -.0000
.2500 +.0004 -.0000
⑫
.6560 +.0008 -.0003
.2500 +.0007 -.0003
⑬
ASEMBLED-2:1 REDUCTION PHASING GEAR WITH NECESSARY DIMENSION FOR ASSM.
DRILL & TAP FOR Nº12 NF-FLAT HEAD MACHINE SCREW - 3 REQ'D.
.2500 +.0004 -.0000
7/16" O.D.-12 TOOTH SPUR GEAR-CLASS 7, ASA-B-42 FIT TO SHAFT
.2500 +.0000 -.0003
⑭
⑮
⑯
NOTE: ALL PARTS MACHINE FINISHED
GAS SAMPLING VALVE-DETAIL OF MECHANICAL PHASE CHANGER
DESIGNED BY N.P. WHALEY
DRAWN BY N.P. WHALEY
OKLA. INSTITUTE OF TECHNOLOGY MECHANICAL ENG. SCHOOL
DWG. Nº 3
SCALE 1"=1"
2-26-50.



GAS SAMPLING VALVE - DETAIL OF ASSEMBLY	
DESIGNED BY N.P. WHALEY	DWG. Nº 4
DRAWN BY N. P. WHALEY	
OKLA. INSTITUTE OF TECHNOLOGY	SCALE $\frac{1}{2}'' = 1''$
MECHANICAL ENG. SCHOOL	4-29-50

APPENDIX D

PHOTOGRAPH

A. Sampling Valve

B. Driving Mechanism

C. Orsat Gas Analizer

D. Mercury Manometer

E. Scales

F. Low Pressure Gage (100 psi)

G. High Pressure Gage (600 psi)

H. Valve

I. Pressure Snubber

J. Leveling Bottle

K. Stop Watch

L. Throttle

M. Tachometer

N. Brine Tank

O. Fuel Measuring Tank

P. Electrodes

Q. Phasing Wheel

APPENDIX E

A Selected Bibliography

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10. Products Engineering. Design Work Sheets. New York and London: McGraw-Hill Publishing Company, Inc. 1933.

APPENDIX F

List of Abbreviations

B.t.u.	British thermal unit
b.d.c	bottom dead center
t.d.c	top dead center
psi	pounds per square inch
psia	pounds per square inch absolute
rpm	revolutions per minute
$^{\circ}$	degrees (angular measurement)
ft	feet
"	inches
/	per
#	pounds
in.^{2}	square inches
in.^{3}	cubic inches
cc	cubic centimeters
ft.^{2}	square feet
ft.^{3}	cubic feet
C	Carbon
CH_4	Methane
CO	Carbon Monoxide
CO_2	Carbon
H_2	Hydrogen
H_2O	water
N_2	Nitrogen
O_2	Oxygen

Typist: Mrs. Newton Philip Whaley,Jr.