STRATIFIED CHARGE SCAVENGING OF A TWO-STROKE ENGINE AT PART THROTTLE

by

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B.A.Sc., University of British Columbia, 1960

A THESIS SUBMITTED IN PARTIAL FULFILMENT OF
THE REQUIREMENTS FOR THE DEGREE OF

MASTER OF APPLIED SCIENCE

in the Department

of

Mechanical Engineering

We accept this thesis as conforming to the
required standard

THE UNIVERSITY OF BRITISH COLUMBIA

September, 1962

Department of Mechanical Engineering,
The University of British Columbia,
Vancouver 8, B.C., Canada.

September, 1962.

ABSTRACT

The high speed 2-stroke cycle engine designed for high power-to-weight ratio is relatively inefficient at part throttle. It would be advantageous to incorporate a simple method of allowing extra air to enter the cylinder prior to the fresh mixture, thus stratifying the charge and increasing the proportion of the air-fuel mixture retained in the cylinder at part loads while not deleteriously affecting the maximum power at full throttle. A series of tests on an engine fitted with a reed valve connecting the atmosphere to the passageway leading to intake ports, were carried out with varying amounts of extra air; the results showed that power, speed, thermal efficiency, and fuel trapping efficiency gave increases at nearly all settings, but with a large excess of extra air, the air-fuel ratio through the carburetor had to be decreased to maintain stable operation.

ACKNOWLEDGEMENT

The experimental work described in this report was carried out in the Mechanical Engineering Laboratory at the University of British Columbia and the calculations performed at the University Computing Centre. The use of these facilities is gratefully acknowledged.

To all the people who made this project possible, the author would like to express his thanks. Particular thanks are due to the following:

Professor W.O. Richmond for his valuable guidance and assistance during all phases of the project and for the use of the facilities of the Mechanical Engineering Department;

Professor J. Young for his direction during the initial stages of this project;

Power Machinery Limited of Vancouver, and Mr. Jack Stainsby in particular for their co-operation and for supplying the engine and spare parts, and fabricating special parts;

Consolidated Mining and Smelting Company of Trail for making this project financially possible through the Cominco Fellowship and Grant.

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Chapter I. INTRODUCTION

Scavenging, the process of replacing the exhaust gases in the cylinder with a fresh charge, is inefficient in the 2-stroke engine because some exhaust gases are not displaced, thereby reducing the quantity of fresh charge available for combustion. Some mixing between the exhaust gases and fresh charge always takes place so that some fresh charge escapes with the exhaust. If this fresh charge contains gasoline vapours, raw fuel flows through the engine unburned, and unutilized.

Throttling, the process of restricting the entry of the fresh charge into the engine, reduces the brake horse power. At the start of compression, the volume of fresh charge retained in the cylinder together with the volume of exhaust gas remaining, constitute a constant cylinder volume. Thus throttling, by restricting the fresh charge entry, indirectly increases the proportion of the exhaust gases in the combustion mixture, resulting in reduced efficiency, reliability, and durability.

The effect of the disadvantages of the two processes may be reduced by admitting air into the cylinder prior to the fresh air-fuel charge, that is, by stratifying the charge. This stratified charge scavenges the exhaust gases and reduces the loss of fuel-air mixture through the exhaust ports.

In the system under investigation, the stratification of the charge is accomplished by admitting air through an auxilliary valve into the passageways between the crankcase and ports as close to the ports as possible as shown in figure 1.

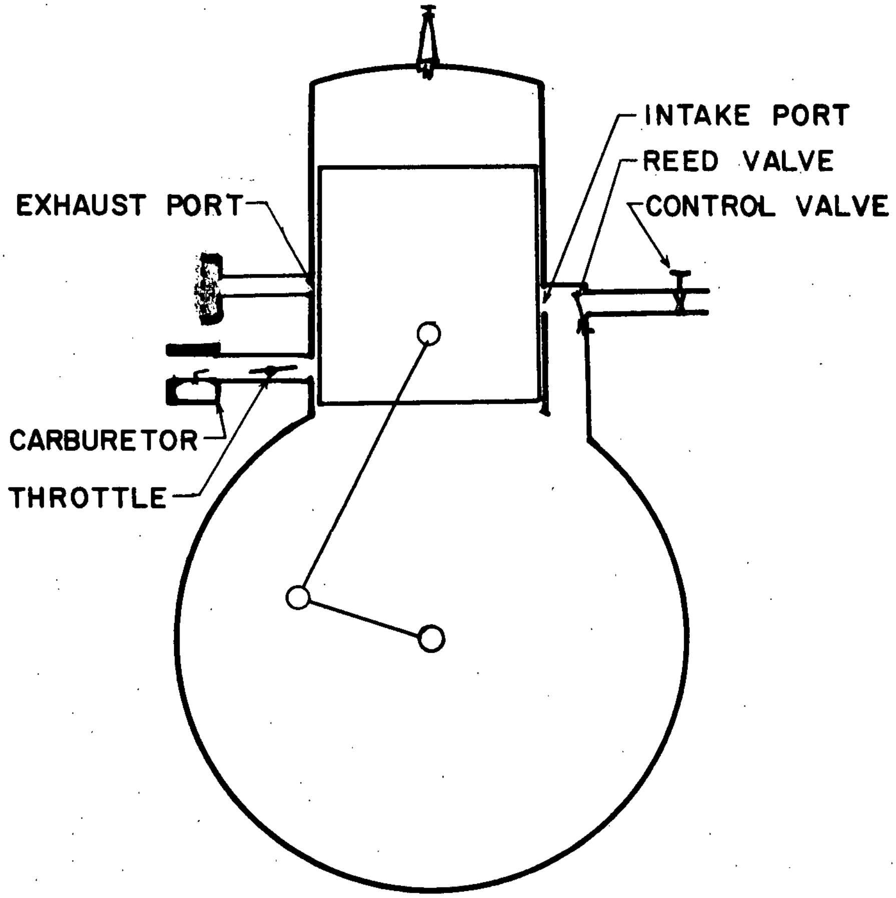

Figure 1. Schematic view of a 2-cycle engine with a reed valve positioned.

The upward movement of the piston creates a partial vacuum in the crankcase and passageway drawing a mixture of air and fuel through the carburetor into the crankcase, and a separate quantity of air through the auxilliary scavenging air valve into the passageway. Downward movement of the piston precompresses the charge. As the air is ahead of the air-fuel mixture, it is the first to enter the cylinder when the intake port is uncovered; in so doing it expels most of the exhaust gases leaving an atmosphere of air in the cylinder. The fuel-air mixture from the main crankcase follows the entrance of the air, scavenging the air-exhaust mixture, leaving an atmosphere of air and air-fuel mixture. The scavenging air serves both to scavenge the exhaust almost completely and to supply an excess of oxygen for more efficient combustion.

The chemical equation for combustion gives a stoichiometric ratio of 15 lb of air per lb of gasoline in order to supply the amount of oxygen required to burn the fuel completely. For complete combustion in an actual engine, there must be an excess of air and thorough mixing of fuel and air. There is a limit however to the air-fuel ratio for satisfactory combustion. Maximum efficiencies are usually obtained when the air-fuel ratio is somewhere between 15 and 16. Specific fuel comsumption increases as the air-fuel ratio increases above 16 due to slower burning and/or incomplete combustion. Cyclic irregularity also increases because in some cycles the combustion velocity has decreased to such an extent that combustion is not completed by the time the exhaust ports open. The increase in cyclic irregularity eventually produces missing, limiting the upper air-fuel ratio of homogeneous mixtures to about 18.

In the other direction, greater power is obtained by supplying more fuel than can be burned with the air available. The hydrogen in the fuel has a greater affinity for oxygen than has the carbon so that in a shortage of oxygen, the hydrogen reacts with more oxygen reducing the amount available for the reaction with carbon. During the reaction of hydrogen with oxygen to form water a greater amount of energy is released than in the reaction of carbon with oxygen to form carbon monoxide or carbon dioxide. Thus more energy is released in the combustion of a mixture with excess fuel. Also the probability of all the oxygen combining with the hydrocarbons in the fuel is increased if the hydrocarbons in the fuel are increased due to a lower air fuel ratio. The engine will run cooler at low air-fuel ratios. This is due to an increase in the specific heat of the gases.

To increase the upper critical air-fuel ratio two conditions must be met:

(1) the mixture must remain ignitable, and

(2) a sufficiently rapid rate of combustion must be maintained.

These requirements necessitate an increase in the ignition energy, achieved in practice either by means of a precombustion chamber, or by a stratified charge. In both of these practices a small, easily ignited portion of the charge initiates and sustains combustion of the main charge.

If a prechamber is employed with homogeneous mixtures, its walls are maintained at a high temperature to impart the required energy to the combustible mixture. The burning gases issuing from the prechamber into the main combustion chamber act as a torch to ignite the main charge. Conditions approaching those of pre-ignition would be desirous to assist rapid and complete combustion.

For satisfactory combustion to occur in a stratified charge with or without a prechamber, a volume of the correct or slightly rich air-fuel ratio is directed to surround the spark plug. The expansion of the burning gases and temperature increase of this burning mixture raises the pressure and temperature of the remaining lean charge to its ignition point. To facilitate rapid and complete combustion the cylinder head and prechamber must be designed to ensure turbulence and mixing of the burning gases with the main charge.

Engine knock is greatly reduced as the air-fuel ratio is increased. Detonation has been explained as being due to the rapid combustion of the fuel-air mixture in the unburned portion preceding the flame. A flame front radiating outwardly from the point of ignition, compresses and heats the yet unburned gases in advance of it to such a degree that self-ignition takes place. Flame propagation is slower in leaner mixtures so that in a high air-fuel ratio mixture, the temperature and pressure of the unburned portion of the cylinder volume does not reach self-ignition conditions before the piston has expanded the gases and reduced the temperature and pressure. As the energy requirements for ignition increases with the air-fuel ratio, leaner mixtures require higher temperatures before self-ignition will take place.

The completeness of combustion is one of the factors which affects the specific heat of a working substance. The value of C_v which determines k, depends on the temperature of the mixture as well as on the air-fuel ratio. Table I gives the value of C_v at several different air temperatures and air-fuel mixture ratios.

C_v	CONDITIONS
0.1715	Air @ 540°R
0.185	Air @ 1200°R
0.235	Air @ 5000°R
0.282	Fuel and air during lean mixture combustion
0.296	Fuel and air during correct mixture combustion
0.347	Fuel and air during rich mixture combustion

Table I. Specific Heat (C_v) at Constant Volume. (1) *

The apparent increase in C_v with decreasing air-fuel ratios or increasing temperature is due to dissociation, the mechanism which tends to reverse the initial chemical reaction of combustion or to form new intermediate compounds such as nitric oxides. Because the chemical reaction partially reverses or does not go to completion, not all of the potential chemical energy of the fuel is released in the process of burning. Because the value of k varies with the completeness of combustion it is possible to predict a change in the thermal efficiency as the air-fuel ratio is varied by considering the ideal Otto cycle efficiency equation. It may be stated as:

* Numbers in parenthesis refer to Bibliography at end of report.

$$e = 1 - \left(\frac{1}{r}\right)^{k-1}$$

where r = compression ratio, and

k = ratio of specific heats = $\frac{C_p}{C_v}$

As an example the average value of k is 1.3 for a correct carburetted mixture and equal to 1.4 for pure air. Using these values of k and a compression ratio of 10, the ideal Otto cycle efficiencies are 50% for correct air-fuel mixtures and 60% for pure air. Further calculations of the possible theoretical gains relative to the efficiency with a correct air-fuel ratio give a gain of 8% for an air-fuel ratio increase from 15 to 20, and a gain of 12% for an air-fuel ratio increase from 15 to 27 (2). These calculations predict that if an engine could be run satisfactorily on a lean air-fuel mixture higher efficiencies would be the result.

To reduce the power produced by an engine at a constant speed, the quantity of heat liberated in the cylinder per cycle must be reduced. In the ordinary carburetted 2-stroke cycle engines, throttling of the intake charge produces the desired result by restricting the quantity of fuel and air mixture entering the crankcase. The consequential lowering of the precompression pressure results in less charge entering the cylinder during intake so its partial pressure is reduced. For the total pressure in the cylinder to remain approximately constant at the start of compression, the partial pressure of the exhaust gases must increase which is brought about by more of the exhaust gases remaining in the cylinder. The residue gases dilute the fresh mixture.

The exhaust gases prevent rapid and complete mixing of the combustible gases by isolating the oxygen and fuel. With less oxygen available for combustion and the overall air-fuel ratio increased, combustion is incomplete. The lack of oxygen is partially compensated for by the high temperature exhaust gases increasing the energy of ignition and velocity of the flame front in the exhaust-fresh charge mixture. Nevertheless to maintain smooth and reliable firing at part loads, the overall mixture strength should be kept above stiochiometric by enriching the fresh charge.

With the stratified charge scavenging system, the reduction of the heat liberated and consequently the power output, is accomplished by admitting more air through the scavenging air valve instead of restricting the air-fuel mixture. The reduction in partial pressure of the air-fuel mixture is achieved by increasing the partial pressure of the air entering through the scavenging air valve so that the exhaust partial pressure remains approximately constant. The result is that as the power requirements drop, more air is admitted through the scavenging air valve thereby reducing the amount of fresh air-fuel mixture entering through the carburetor while retaining approximately the same amount of residual gases. At part load the stratified charge scavenging system will result in supplying the fuel with an excess amount of air instead of an excess of exhaust gases as is the case in the conventional engine.

With conventional scavenging, three hypothetical relationships exist between the scavenging efficiency defined as the ratio of the mass of charge retained to the ideal mass that the piston displaces, trapping efficiency defined as the ratio of mass of charge retained to the mass supplied, and scavenging ratio defined as the ratio of mass supplied to the ideal mass that the piston displaces. These relationships are shown in figure 2. Although these representations are hypothetical, they do permit an approximate analysis of a scavenging system to be undertaken.

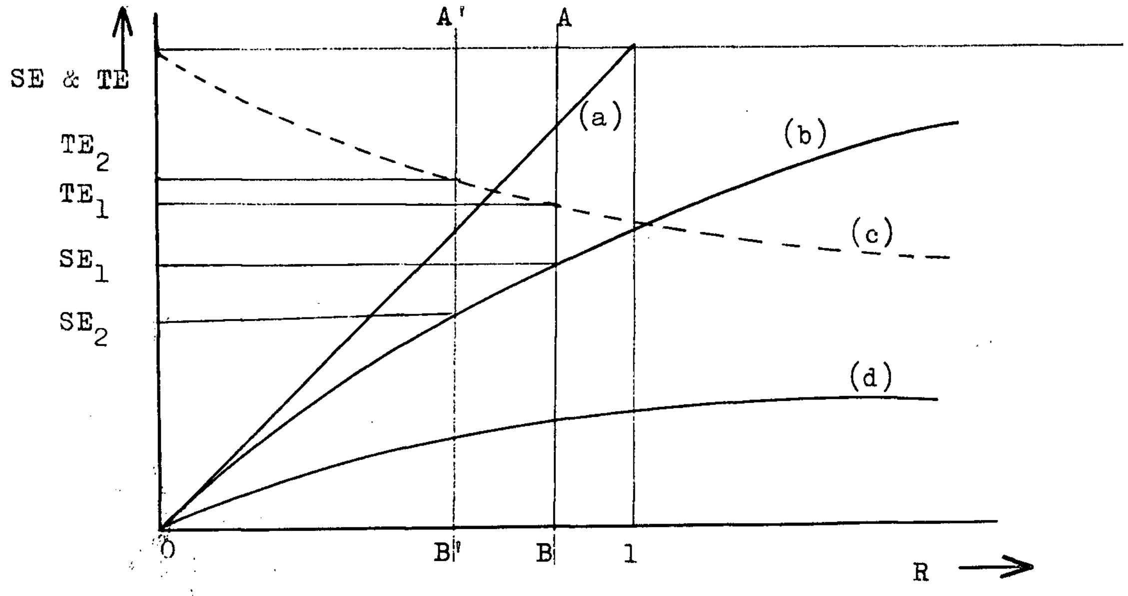

Figure 2. Theoretical relationships between scavenging efficiency (SE) trapping efficiency (TE) and scavenging ratio (R) according to Taylor (3): (a) SE with perfect scavenging (b) SE with perfect mixing (c) TE with perfect mixing, and (d) SE with complete short circuiting.

In the case of perfect scavenging the fresh charge is assumed to replace the exhaust gases completely without mixing or short circuiting. This means that for a given volume of fresh charge entering the intake ports, an equal volume of undiluted exhaust gases leave by way of the exhaust ports. Under this assumption, the scavenging efficiency is equal to the scavenging ratio at all points and the trapping efficiency is equal to unity.

The second and most important representation, because it approximates the actual conditions best, assumes that the volume of fresh charge mixes completely with the residual gases as soon as it enters the cylinder and that an equal volume of the resultant mixture leaves the cylinder. To get a simple relationship, it is necessary to assume that the residual exhaust gases are at the same temperature and have the same molecular weight as the fresh charge, and that the piston remains at bottom dead center during the scavenging process. The resultant equations, derived by Taylor (3) are:

$$SE = 1 - e^{-R}$$

$$TE = \frac{1 - e^{-R}}{R}$$

where SE = scavenging efficiency,
TE = trapping efficiency,
R = scavenging ratio.

As more mixture enters the cylinder at constant speed so that the scavenging ratio increases, the amount retained will increase and the proportion of amount retained to amount supplied will decrease.

The third possibility is that the fresh charge flows through the cylinder in a separate stream without mixing with the residual gases or pushing them out. This process is called short circuiting and results in very little fresh charge retention.

In an actual engine the above three processes occur together. In the scavenging process some of the exhaust is pushed out without mixing, some of the fresh charge mixes with the exhaust and both flow out, and some of the fresh charge flows out without mixing.

Studies conducted by Taylor at Massachusetts Institute of Technology (ref.3) reveal that the actual curve of the scavenging ratio versus scavenging efficiency has the same general shape as the curve for perfect mixing. This fact, together with some high-speed motion picture of the scavenging process in a large 2-cycle engine made by Boyer et al. (4) indicate that there is much mixing and little "piston" action in the actual scavenging process. The actual curves of the scavenging efficiency versus scavenging ratio as determined experimentally by Taylor, generally lie below the curves for perfect mixing indicating considerable short circuiting.

Because the actual curve has the same general shape as the curve for perfect mixing, and because for the purposes of this investigation the variation of the scavenging efficiency and trapping efficiency are more important than their magnitude, the following analysis assumes that perfect mixing takes place in the scavenging process.

The process of throttling the fresh charge in a conventional scavenging system increases the restriction to the air flow which results in a smaller quantity of mixture entering the cylinder. The reduction in the mass decreases the scavenging ratio which corresponds to the line A-B in figure 2 moving to the left, say to A'-B'. The result is a decrease in the scavenging efficiency and an increase in the trapping efficiency as the figure also shows. An increase in the trapping efficiency means that a greater proportion of the charge remains in the cylinder and a smaller proportion escapes with the exhaust gases out the exhaust ports.

A new set of equations are necessary to describe the efficiencies resulting from scavenging the exhaust gases with a stratified charge. The relationships as derived by Taylor may still be used to approximate the scavenging and trapping efficiencies of the total charge of air entering through the scavenging air valve and the air-fuel mixture entering through the carburetor, but new definitions were necessary to estimate the retained proportion of the carburetted air-fuel mixture. The ideal equation for fuel scavenging efficiency, defined as the ratio of mass of fuel-air mixture inspired through carburetor to ideal mass that the piston displaces, and the fuel trapping efficiency, defined as the ratio of the mass of carburetted fuel-air mixture retained to the mass supplied, are derived in appendix III, employing similar assumptions to those used by Taylor in his derivation of the charge scavenging and trapping efficiencies. The resultant relationships assuming perfect mixing are:

$$FSE = 1 - e^{-R(1-SAR)}$$

$$FTE = \frac{1 - e^{-R(1-SAR)}}{(1-SAR)R}$$

where FSE = fuel scavenging efficiency,
FTE = fuel trapping efficiency,
SAR = scavenging air ratio,
R = scavenging ratio.

These efficiency equations are represented in graphical form in figure 3.

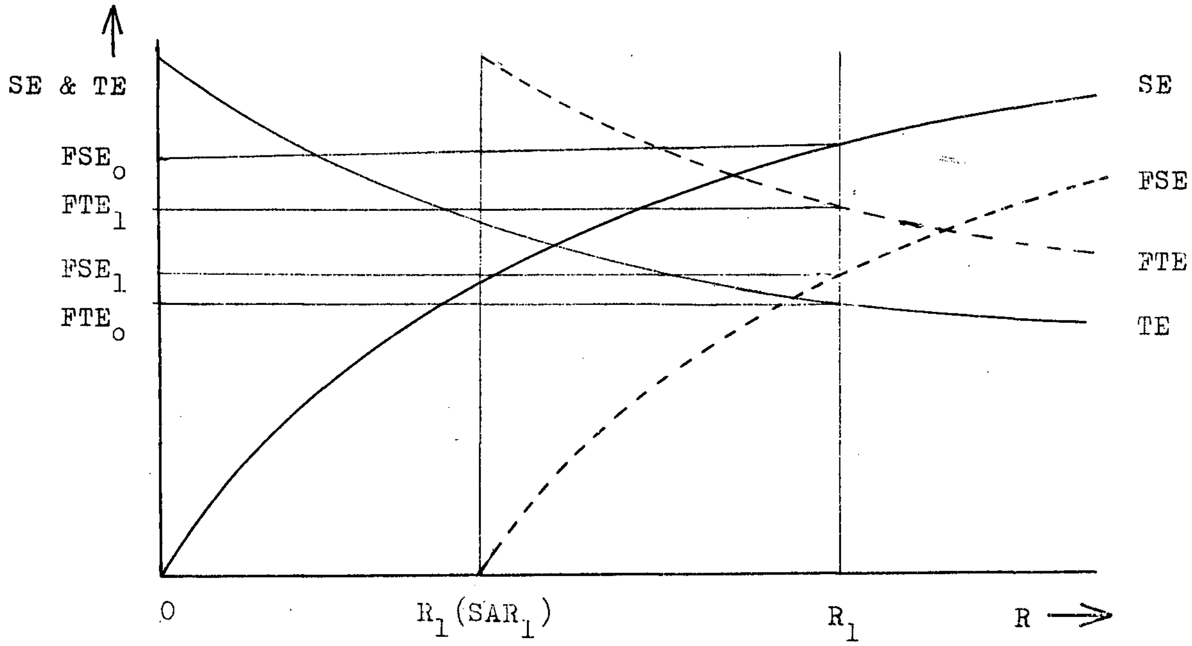

Figure 3. Theoretical relationships between scavenging efficiency, trapping efficiency, and scavenging ratio for stratified charge operation assuming perfect mixing.

When the scavenging air valve is closed so that the charge is homogenous, the fuel trapping efficiency is equal to the charge trapping efficiency and the fuel scavenging efficiency is equal to the charge scavenging efficiency and the scavenging air ratio is equal to zero as shown by the full lines in figure 3. When the scavenging air valve is opened to allow a small quantity of scavenging air to enter cylinder prior to the air-fuel mixture, (with the engine running at full throttle), the scavenging air ratio is no longer zero so that the fuel trapping efficiency and fuel scavenging efficiency are different from the charge trapping and scavenging efficiency respectively. These efficiency equations are given in graphical form by the dotted lines in figure 3.

The shape of the curves are independent of the scavenging air ratio as the only change that occurs is that the origins of the fuel trapping efficiency and fuel scavenging efficiency curves are displaced to the right by an amount equal to the value of $(R)_X(SAR)$. The charge efficiency relationships are not altered so they can still be represented by the solid lines in figure 3. Total air consumption increases only to a small extent because the amount of the air-fuel mixture entering the carburetor reduces with an increase in the amount of air entering through the valve. Because the total air consumption change is small the value of the scavenging ratio remains approximately constant. If, for a particular speed, the value of the scavenging ratio is R_1 the fuel scavenging efficiency will drop from FSE_0 to FSE_1, and the fuel trapping efficiency will increase from FTE_0 to FTE_1, as the scavenging air valve is opened to give the scavenging air ratio a value of SAR_1. The fuel scavenging efficiency decrease means that the amount of air-fuel mixture retained to ideal mass displaced decreases but this decrease is the result of a smaller quantity of air-fuel mixture being supplied. The air-fuel mixture is reduced by an amount approximately equal to $(R)(SAR)$. The increase in the fuel trapping efficiency indicates that more of the air-fuel mixture being supplied by the carburetor is retained in the cylinder.

When the scavenging air valve is opened with the engine running at part throttle, additional air enters the crankcase accompanied by a reduction in the amount of air flowing through the carburetor. The result of the reduced restrictions to air flow due to the additional path made available for the air through the scavenging air valve, is that the total air consumption increases. The speed also increases due to combustion efficiency increase and the reduction in the crankcase vacuum. The effect of the air consumption increase is to increase the scavenging ratio which is partially offset by the speed increase so the resultant scavenging ratio is R_1, figure 4. But as scavenging air ratio is now a positive number the

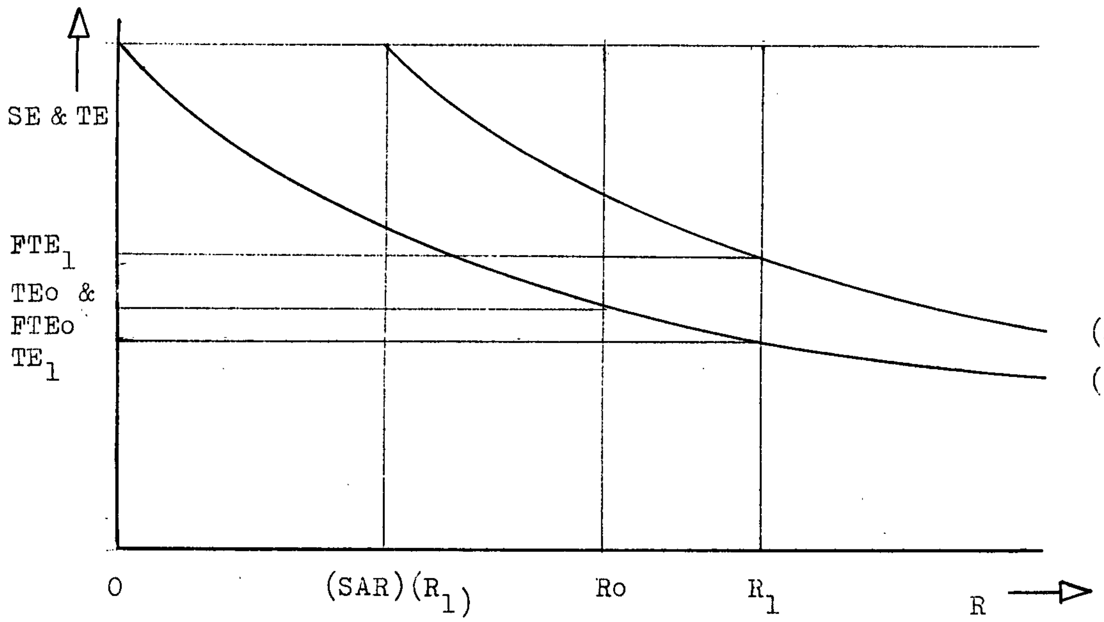

Figure 4. Theoretical relationship between trapping efficiency and scavenging ratio for stratified charge operation at part throttle.

(a) fuel trapping efficiency with scavenging air valve open, (b) fuel trapping efficiency with scavenging air valve closed, or trapping efficiency with scavenging air valve open or closed.

origin of the fuel trapping efficiency will be to the right. The overall result is that the fuel trapping efficiency will increase from FTE_0 to FTE_1 and the charge trapping efficiency will decrease from TE_0 to RE_1. By stratifying the charge for purposes of exhaust scavenging the proportion of the total air retained to total amount supplied is reduced and the proportion of the air-fuel mixture retained to amount supplied is increased. The increase in the amount of fresh air in the exhaust and a decrease in the amount of fuel wasted in scavenging, will increase the thermal efficiency of the engine as well as decrease the exhaust temperatures for lower specific fuel consumption and a more durable engine.

In this investigation the stratified charge scavenging system was tested on a powerful, high-speed lightweight engine, piston ported throughout. The requirement of simplicity ruled out any modifications requiring fuel injection or elaborate precombustion chambers but did not negate reed valves. Even then in future designs it should be possible to simplify the system even more by controlling the scavenging air entrance with the piston as is done at the present time with the control of the air-fuel mixture. It is also a simple matter to revert back to ordinary scavenging with no deterent effects on performance at maximum power if conditions warrant it.

The intake ports must deflect the rich mixture to the spark plug so that at top dead center the spark will cause ignition of the volume surrounding the plug, which in turn will ignite the remaining charge. The degree to which the overall fuel air ratio may be reduced with this system while maintaining satisfactory combustion, would depend on how well the stratification could be controlled by the intake port deflections, and on the scavenging air temperature.

Chapter II. PREVIOUS WORK

Various methods of burning a stratified charge in a 4-stroke engine have been investigated throughout the world. One method considered at the Institut Francais du Petrole and reported by Baudry (2) is to inject a small amount of fuel, roughly corresponding to the idle requirements of the engine, into a precombustion chamber and to aspirate a lean mixture through the carburetor. This procedure involves a number of difficulties. The presence of a prechamber always entails a reduction of thermal efficiency and the requirement of fuel injection pumps and nozzles greatly increases expenses as well as involving all the drawbacks of fuel injection.

Experimental results have been published by Nilov in Russia (5) whereby aspiration of a carburetted mixture has been substituted for fuel injection as shown in Figure 5. During the induction period, the engine

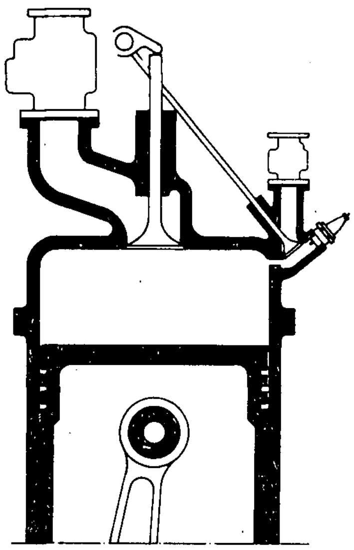

Figure 5. Russian "Jet" Ignition engine (5)

inspires a lean mixture through the main carburetor and a rich mixture through the prechamber inlet valve. The rich mixture is ignited by the spark plug located in the prechamber and the lean mixture is ignited by the torch effect of the prechamber. This system has the disadvantage that the thermal losses produced by the prechamber lowers the peak engine output.

It is possible to accomplish stratification without using a prechamber by direct injection of the fuel into the vicinity of the spark plug. However cost and complexity are increased as stratification by injection is practical only when used in conjunction with a special combustion chamber as revealed by experiments at the Citroen works in France, (U.S.A. patent no. 2,929,250), and by Texaco in the United States (6). Barber and his associates at Texaco eliminated engine knock by means of stratification. Their system is characterized as one involving injection near top dead center and a non-homogeneous charge.

A system for heterogeneous mixture formation in the cylinder without a prechamber was developed at the Institut Francais du Petrole by Baudry (2). The carburetted mixture was divided into two separate streams of completely different fuel-air mixtures as shown in Figure 6. The rich mixture is delivered into the vicinity of the spark plug by a supplementary

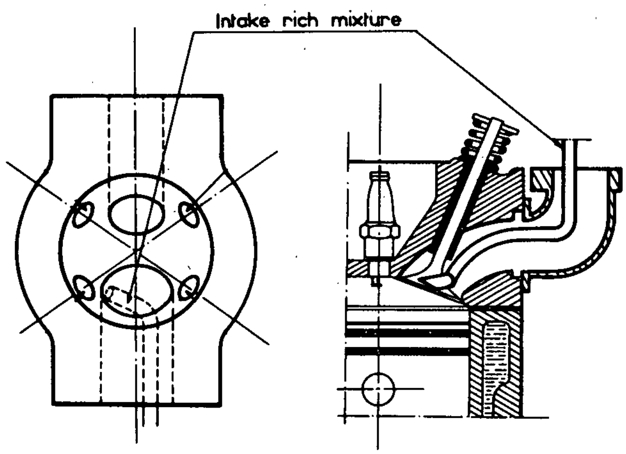

Figure 6. I.F.P. engine modified for heterogeneous carburetion.

carburetor through a heater. The standard intake pipe supplies either pure air or a lean mixture to the remaining cylinder volume as determined by power requirements.

In the United States, Conta and Durbetaki at the University of Rochester employed the Broderson method to achieve stratification (7). This is characterized by injection of fuel near bottom dead center into a throat between the prechamber and the main combustion chamber as shown in figure 7.

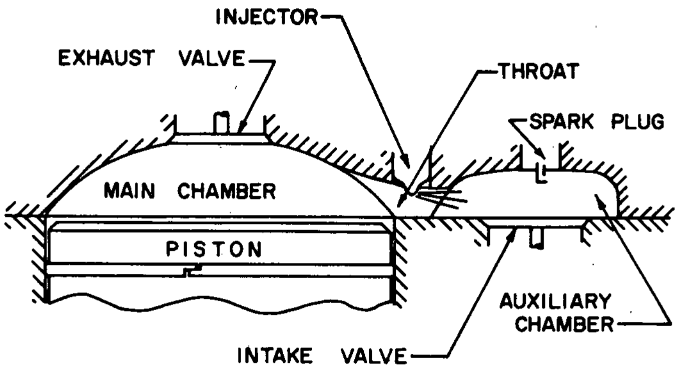

Figure 7. Combustion chamber for Broderson method of stratification

Proper stratification of the fuel is insured by simultaneously controlling the quantity of fuel injected and the time injection begins. Early injection when the intake valve is still open and air is flowing into the main chamber draws fuel into the main chamber; injection after the valve is closed and piston is compressing air into the auxilliary chamber restricts fuel to the prechamber.

Harry Ricardo of London, England, ran a 2-cycle engine on a stratified charge system in 1905, (8). The system he used required a prechamber into which the rich mixture was inspired through a separate intake valve and carburetor. But he was unable to obtain more than one-half or two-thirds of the maximum power obtainable from a similar engine burning a homogeneous mixture.

Later Ricardo experimented with two separate fuel injections through one injector to fulfill the need for very thorough mixing of the fuel and air before ignition. The injector was placed vertically in the head of the cylinder and the injection was in the form of a hollow cone directed to meet the air as it entered through the inlet ports in the sleeve valve as shown in figure 8. Additional fuel was injected after the end of normal

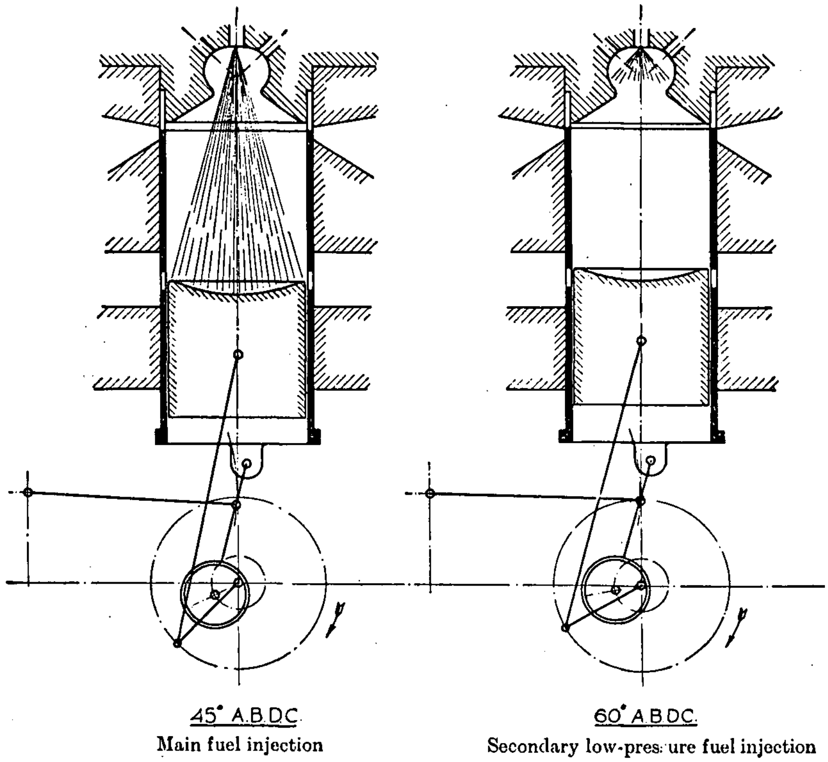

Figure 8. Ricardo system of stratification in a 2-stroke engine

injection period, in order to provide, locally, a mixture sufficiently rich to be ignited by the spark plugs even though the mixture in the main cylinder was too lean for ignition from a spark, though not too lean for ignition by a flame issuing from a prechamber. However, in this system a gap existed between 15 and 40% full torque where ignition was irregular and uncertain. Below 15% of full-load torque, combustion took place in the prechamber alone; from 35-40% full torque and upwards combustion took place in the prechamber and main chamber. But between 15% and 40% the mixture in the main combustion chamber was too weak to be ignited from the prechamber.

In 1911 a patent was granted by the U.S. Patent Office to William Stephenson, (No. 1,012, 288); the patent was to increase the efficiency and speed of the then slow turning 2-stroke engine by allowing air to enter the pre-compression chamber independently of the mixed charge through a spring loaded poppet valve as shown in Figure 9.

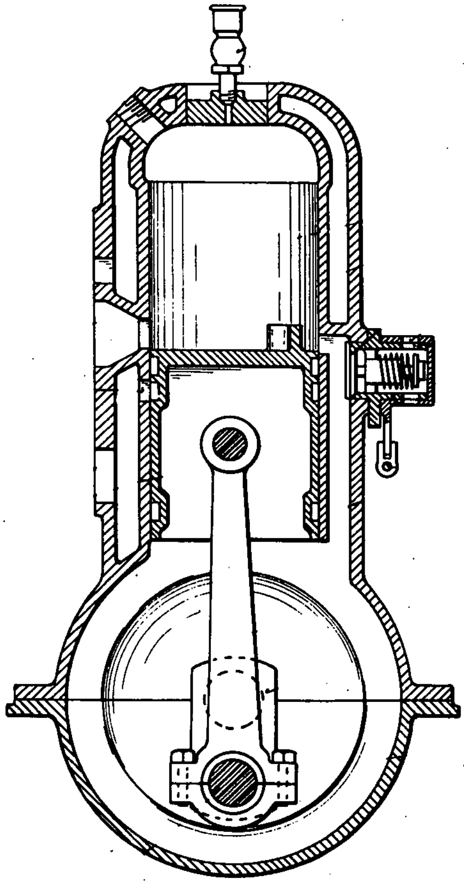

Figure 9. Stephenson method of stratification in a 2-stroke engine

At the University of British Columbia in 1958 Peter Koch installed a reed valve to admit extra air into the crankcase of two outboard motors. A limited number of tests indicated an increase in power and a decrease in the specific fuel comsumption. Apparently the normally rich mixture was being diluted by the air in the crankcase giving a combustible mixture which approaches the stoichiometric ratio, thus increasing the power and decreasing the specific fuel consumption. The present principle of using a reed valve to allow extra air to enter the cylinder prior to the main charge is similar to the principle Koch employed in his outboard motors.

The method of stratification under investigation in this report is similar to Stephenson's method but a reed valve is used in place of a poppet valve. Stephenson's method was for a slow speed engine whereas the present system is for high-speed, low specific weight engines where the scavenging ratio is below one as the time for scavenging is very short. His engine had valves whereas the engine presently used is piston-ported throughout.

Ricardo's system reduced the maximum power whereas the present system has no deterrent effects at full power and throttle. Fuel injection is completely impractical for the size of engine being considered.

No experimental results on the effects that stratifying the charge has on the scavenging and throttling of a two-cycle engine have been located, although W. Seaver made an analytical investigation of stratified charging of an internal combustion engine for his Master's degree thesis at Yale School of Engineering (9).

Chapter III. DETAILS OF TEST ARRANGEMENT

The engine used for this test was a model 270 Canadian chain saw engine made in Vancouver by Power Machinery Ltd. It had the following specifications:

Bore - 2 5/16 in.

Stroke - 1 3/8 in.

Capacity - 5.8 cubic in.

Compression ratio - 10.

Fuel-oil mixture - 16:1

Carburetor - Tillotson diaphragm

Ports - piston ported throughout

Recommended carburetor setting - main jet 3/4 - 1 turn open

The outside walls of the passageway were built up with weld material as shown in figures 10 and 11. A hole was drilled and tapped to take a

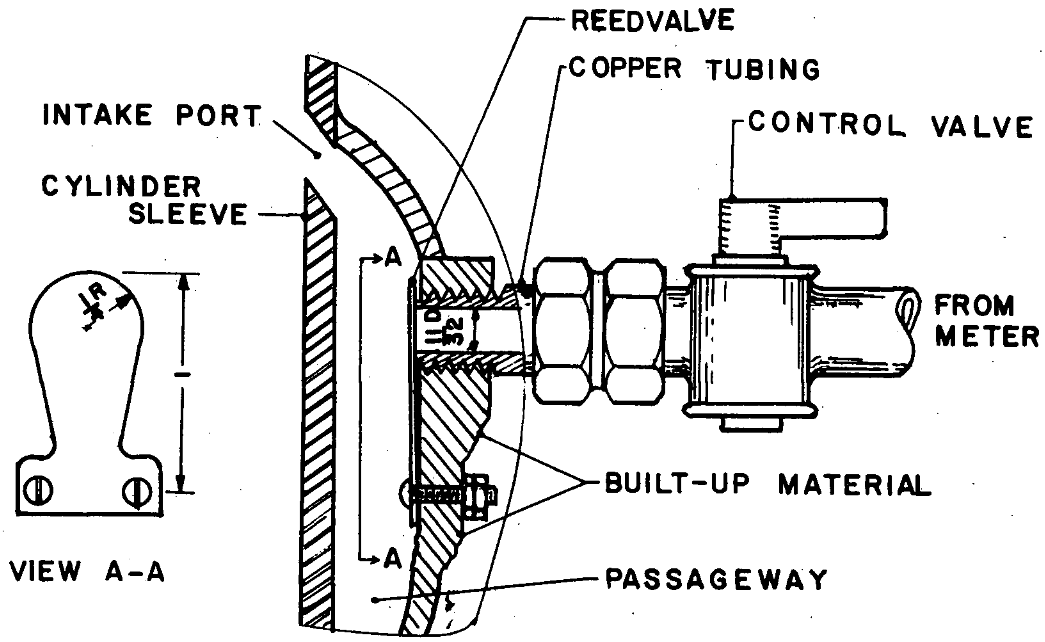

Figure 10. A cross-section of the scavenging air valve

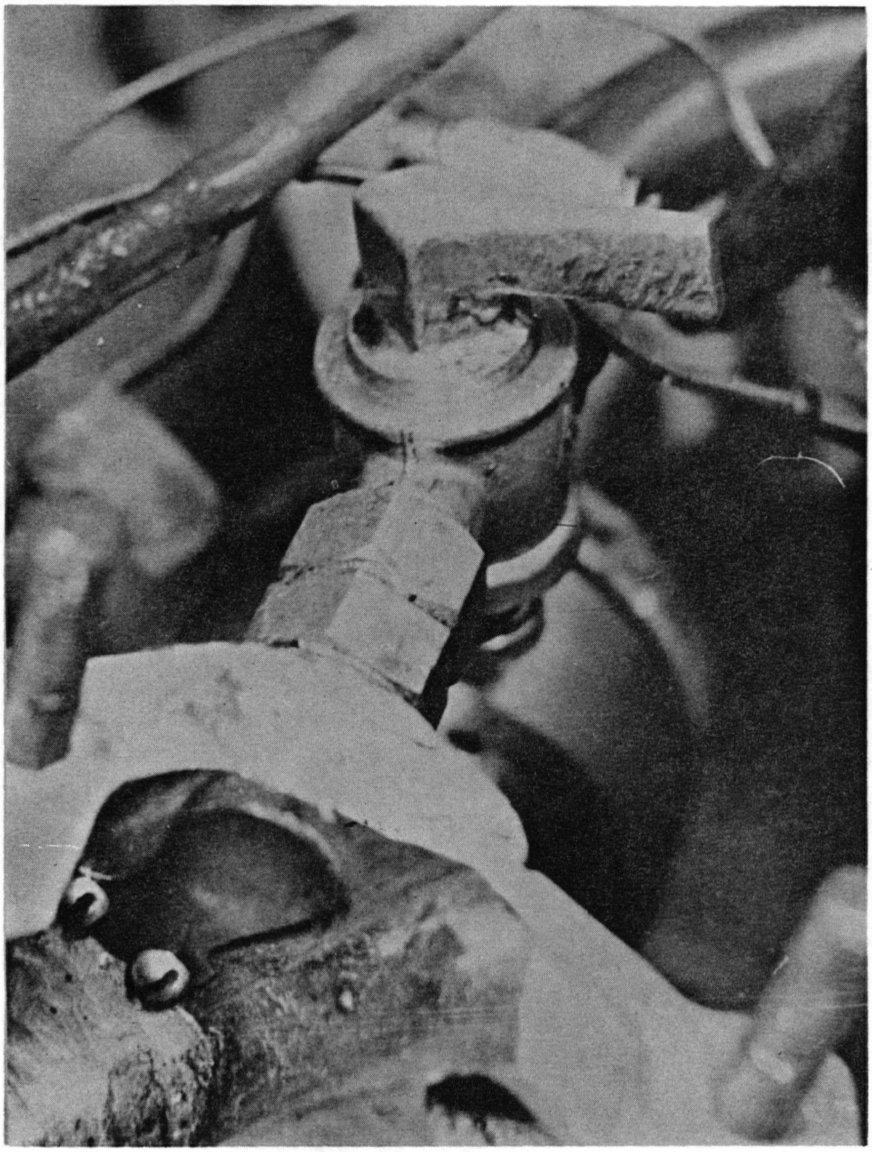

Figure 11. A photograph of the scavenging air valve arrangement

threaded copper tube. The tube projected into the passageway and served as a seat for the high carbon steel reed valve. The valve itself was fastened by two screws in the passageway machined flat to permit the valve to seat properly. On the other end of the copper tube an Imperial cock valve was fastened; the hole of the valve had been opened up to allow as much air as possible to enter when the valve was full open.

The engine was mounted on a small engine test bed in the Mechanical Engineering Laboratory, and coupled to a Froude hydraulic dynamometer, absorbing a maximum of 30 hp at 10,000 rpm. A photograph of the setup is shown in figure 12. The engine was bolted to an adjustable mount but separated from it by a wooden plate to absorb some of the engine vibration.

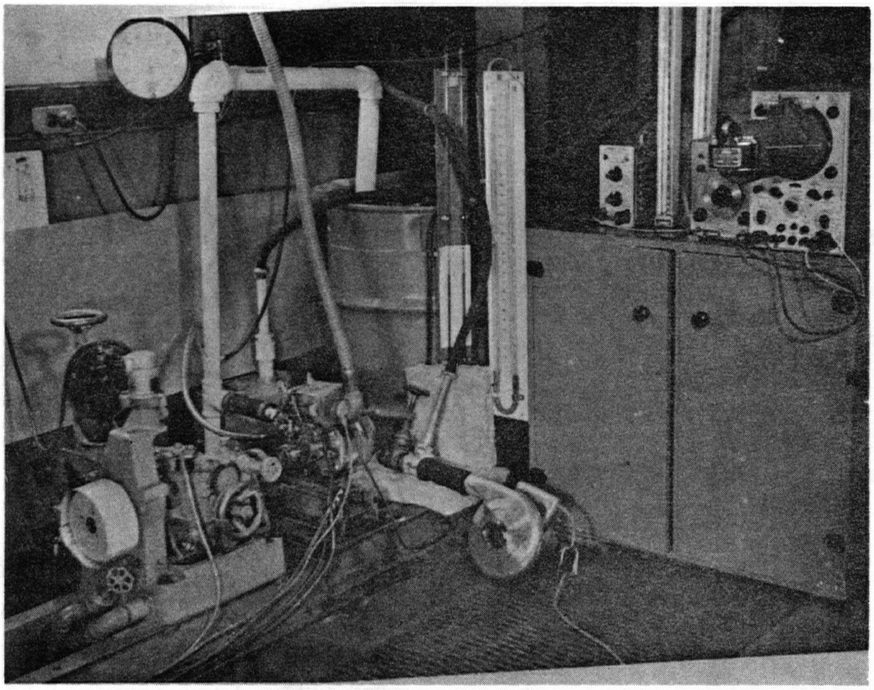

Figure 12. General view of engine on test bed

The engine was instrumented as shown in figure 13, so that the following quantities could be determined:

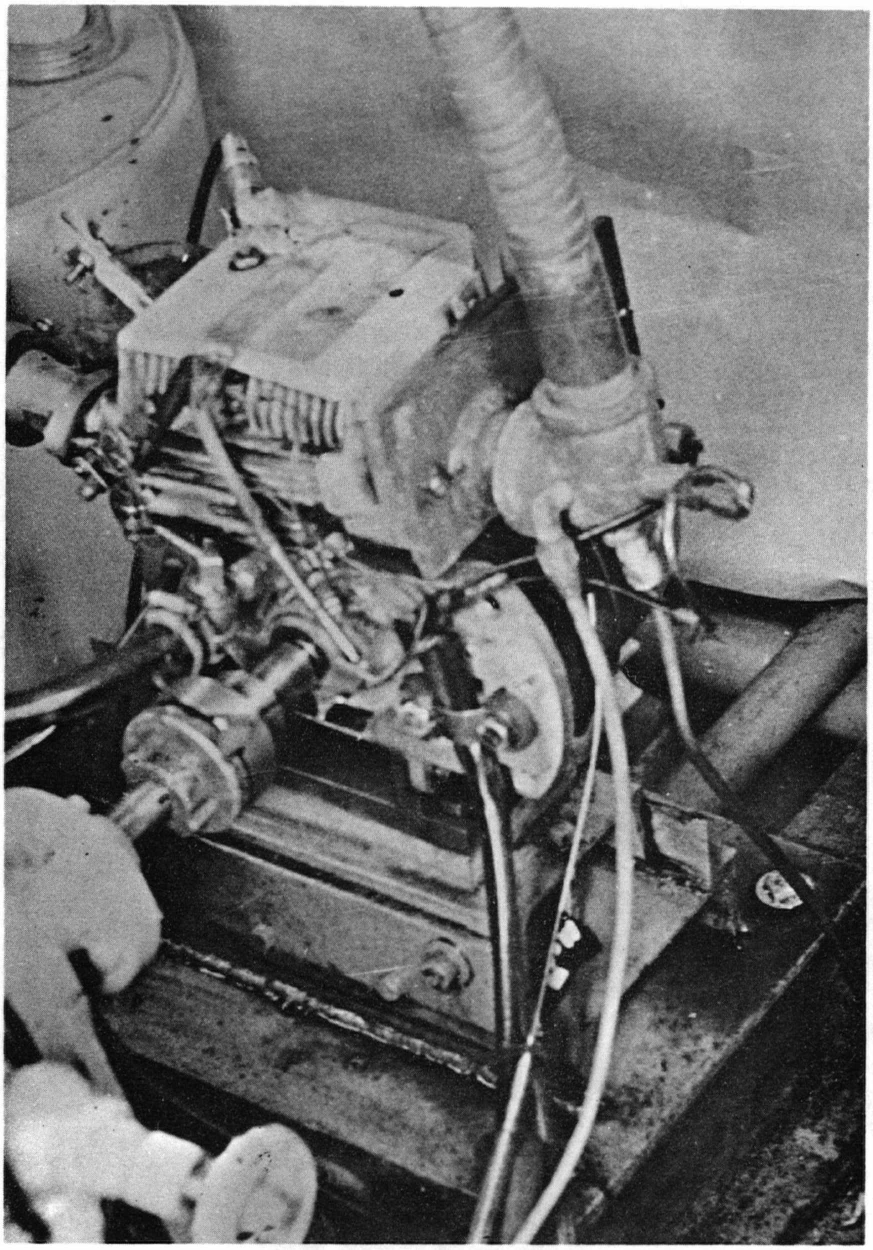

Figure 13. View of instrumented engine

(1) air consumption,

(2) fuel consumption,

(3) speed and power,

(4) engine temperatures

(5) exhaust gas constituents,

(6) combustion pressures.

From these quantities the operating characteristics of the engine were determined.

(1) Air Consumption

The air inspired through the carburetor was drawn from a 45 gallon drum fitted with bypass valves and several different size nozzles. The nozzle used was machined according to VDI standard dimensions (inside diameter .4000 inches), with the standard nozzle coefficients being read from published graphs (10). The average reading of two water manometers connected across the nozzle was taken as the nozzle pressure drop. The pressure in the straight pipe leading to the nozzle (inside diameter .58 inches), was read on another water manometer. A bi-metallic thermometer projecting into the surge tank was read for the air temperature.

The pressure drop across the nozzle varies with the square of the mass flow. Thus if the pressure upstream of the nozzle is at atmospheric pressure, the pressure downstream will be below the barometric reading and will vary as the mass flow varies. A high flow rate will result in a large pressure drop across the nozzle so that the difference between the carburetor intake pressure which is downstream of the nozzle and the barometric pressure will be great. This condition is analogous to operating the engine with the choke partly closed. The effect of varying the air flow would be similar to varying the choke setting.

A vacuum cleaner blower was used to boost the nozzle upstream pressure to above atmospheric pressure so that the pressure downstream of the nozzle could be maintained at atmospheric conditions. The quantity of air flowing through the nozzle was controlled with by-pass valves so that the desired downstream pressure could be maintained. The nozzle and blower were employed only when the nozzle pressure drop reading was required; at other times the auxilliary valves leading to the surge tank were opened and the air allowed to by-pass the nozzle.

The quantity of air flowing through the scavenging air valve was measured by a gas flow meter. Saturated conditions were assumed to prevail downstream of the meter.

(2) Fuel Consumption

The fuel consumption was measured by weighing the fuel in the tank from which fuel was being drawn, or by drawing the fuel from a graduated pipette. The former method gave the weight consumption over the complete test whereas the latter method gave the volume of fuel consumed for only a short time, usually one or two minutes.

The fuel weighing method was generally used and consisted of weighing the fuel and the tank at the start of a test, and whenever readings were required. The fuel tank was connected to the fuel lines leading to engine with flexible tubing which had a negligible effect on the weight readings. The tank was positioned on a balancing scale calibrated in grams. As the fuel was consumed, the weight of fuel in the tank decreased

A pipette calibrated in cubic centimeters could be filled with fuel and then allowed to drain to the engine as it required the fuel. The number of cubic centimeters of fuel used per minute could thus be determined. All fuel flowed through a Dwyer variable area flowmeter to indicate instantaneous fuel consumption. This reading was of value in determining flow fluctuations.

(3) Speed and Power

A Berkely electronic counter with a maximum counting ability of 6000 counts per minute was adapted to count the number of revolutions. The counter is shown in figure 14. The tripping switch was connected to the

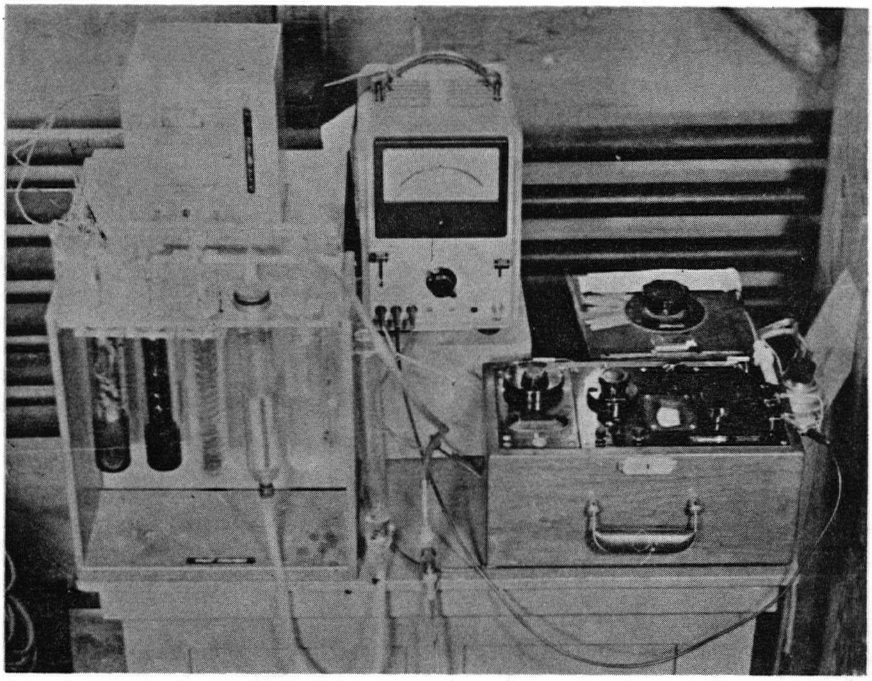

Figure 14. Readout instruments

tachometer drive shaft which has a drive reduction of 10,000 to 1818. Two copper set screws served double purposes in positioning a bakelite wheel on the drive shaft and acting as a conductor for the tripping circuit. A flat spring made contact with the bakelite wheel, closing the circuit whenever the spring touched a set screw. The electronic counter registered the number of times the circuit was closed.

The tachometer on the dynamometer was used to check the variation in speed and indicate a nominal operating speed.

The torque produced by the engine was measured by the Froude hydraulic dynamometer.

(4) Engine temperatures

Two copper-constantin "megopak" thermocouples indicated the temperature of the mixture in each of the passageways leading to the ports. As these unshielded thermocouples were located between hot walls, the readings were probably a few degrees too high due to radiation, but as the variation in temperature was more important than the correct temperature they were assumed to give readings of sufficient accuracy. Two copper-constantin thermocouples were installed in the cooling air stream downstream of the engine, another was silver-soldered to a washer placed under a cylinder head nut, and a iron-constantin thermocouple was silver-soldered to a copper ring placed under the spark plug. A copper-constantin and iron-constantin thermocouple indicated carburetor intake temperatures and valve intake temperatures respectively.

Two iron-constantin thermocouples were fitted into steel probes and screwed into the exhaust pipe and exhaust muffler. The thermocouple in the pipe was in the direct path of exhaust issuing from the exhaust ports so was completely surrounded by hot exhaust gases. The thermocouple in the muffler was not in a direct exhaust flow path so was in somewhat stagnant exhaust gases.

The reference junctions of all thermocouples were individually placed in an ice-bath with the cold junctions placed in glass tubes filled with kerosene. A vacuum tube voltmeter was connected across the thermocouples via a rotary switch. This instrument was used to read the potentials for the lower temperatures as well as indicate when the exhaust temperature had become steady. A potentiometer was connected in parallel with the vacuum tube voltmeter to read the higher temperatures more accurately. These instruments are shown in figure 14.

(5) Exhaust gas constituents

A Republic continuous CO_2 recorder drew in a continuous sample of the exhaust and analysed it. The results from this instrument showed a continuous variation in the CO_2 content of the exhaust.

An orsat analyser, shown in figure 14, indicated the percentage CO_2, O_2 and CO at one particular time and not an average over the run.

(6) Combustion pressures

The pressure in the cylinder was displayed on an oscilloscope by means of a Kistler transducer system. Any number of cycles from one to eleven on one scope were possible by adjusting the sweep synchronizing control through which the triggering signal passed.

The cylinder pressure acted on a P2-14 SLM quartz pickup which changed the pressures into an electrostatic charge, through a hole in a specially fabricated spark plug. The piezo-calibrator in turn amplified the electrostatic charge and produced a voltage signal which was proportional to the pressure. The voltage was displayed on the cathode ray oscilloscope. The calibrator allowed various ranges of pressure to be displayed, not only as the primary trace, but also as a means of calibrating the pressure signal on the scope.

The x axis was on a time base triggered by an electro-magnetic signal initiatedby the dynamometer shaft. The triggering signal first of all passed through a sweep-synchronizing control which regulated the number of pulses the oscilloscope received so that a constant number of cycles would be displayed on the scope, independently of speed. The number of cycles desired was set on the sweep-synchronizing control. The apparatus is shown in figure 12.

Five or six of the eleven cycle displays were photographed side by side on a single frame, with a calibration pressure superimposed, usually 400 psi, as shown in figure 15. The peak cylinder pressures were measured by

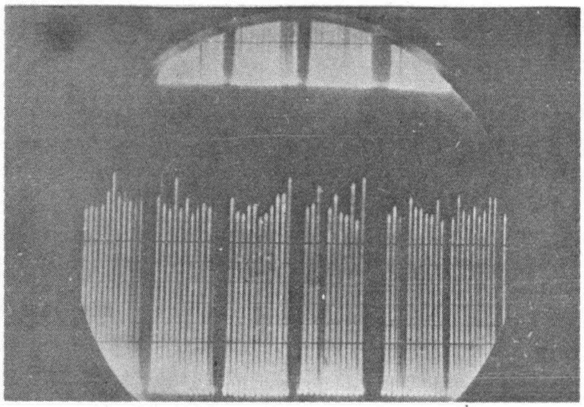

Figure 15. Sample photograph of peak combustion pressures

projecting the frame on to graph paper with an enlarger. From the values of the peak combustion pressures the average peak combustion pressure and average variation were determined.

A vertical water manometer, connected to the exhaust pipe, measured the average exhaust pressure over the whole cycle.

Chapter IV. DETAILS OF TEST PROCEDURE

The procedure generally followed in making a test is described in the following paragraphs; variations to this are described with the results. Before the test was started and while the engine was still warming up with the scavenging air valve closed, the blower was switched on and the valves adjusted to deliver the amount of air required by the engine and yet maintain the carburetor intake pressure the same as when the surge tank and nozzle are bypassed. After the adjustments were completed, the air was allowed to bypass the surge tank, nozzle, and blower through an additional valve, and the blower was stopped.

When the exhaust temperature had become steady, the test was commenced by starting the revolution counter and counter stopwatch simultaneously, starting the fuel flow stopwatch when the fuel weighing scale was balanced, and starting the scavenging air flow stopwatch at an even flowmeter reading. The blower was then again started and the nozzle pressure drop, the nozzle intake pressure, and carburetor intake pressure noted when all the air was being drawn in through the nozzle. When the readings were completed the air was diverted and blower stopped. A complete set of all pressure and temperature readings was then taken. After the readings were completed, about 10 minutes after the test was started, the revolution counter and time were noted simultaneously, the fuel weighing scale was adjusted to read slightly underweight so when it balanced the time and scale reading noted, and the air flow meter read on an even minute. The pressure drop across the nozzle and dynamometer reading were again noted, the two exhaust temperatures read on the potentiometer, and the passageway temperature read on the voltmeter.

Twenty-five or thirty minutes after the test was commenced, the procedure for a complete set of readings was repeated. The scavenging air valve was then fully opened and the whole test procedure repeated. The full open scavenging air valve position was followed by several tests with intermediate openings.

Results of both sets of readings were reduced on the IBM 1620 computer but the 10 minute reading results were used only as a check on the results of the second set of readings. The nozzle pressure drop and dynamometer readings varied during a test as did the exhaust temperature and passageway temperature so these important readings were averaged over the duration of the test and used in the second set of calculations.

Chapter V. DISCUSSION AND RESULTS

Chapter V. DISCUSSION AND RESULTS

The main objective of the tests was to compare the normal operating characteristics of the engine on the conventional scavenging system with the operation on the stratified charge scavenging system. For these tests the only adjustment made was that the scavenging air valve was changed to various positions.

The second objective was to see what effect throttling has on the performance of an engine with stratified charge scavenging. For one set of these tests, the only adjustment made was that the throttle was changed to various positions; for the other set of tests, the throttle was adjusted and then the dynamometer setting changed to bring the speed back to its original value.

The results of the tests on the engine operating on the stratified charge scavenging system are included on the pages that follow. An analysis of the results follows the set of graphs for the particular test. Each test is identified by the throttle position, jet adjustment, nominal speed, and the sheet number on which the data was recorded during the actual run. The sheet number is cross-referenced with the tables of observed results transcribed from the data sheets and included in appendix IV, and the tables of calculated results transcribed from computor results and included in appendix V.

The first few tests, sheets 14, 15, 17, 19, 20 and 21, were with a constant throttle setting, constant dynamometer setting and wide open choke. The engine was run with a valve position varying from full closed to full open. Tests recorded on sheets 22 and 23 were similar to the above except the choke was adjusted when necessary to give stable engine operation. Sheet 24 shows the results of tests with the valve full open, constant dynamometer setting, and a throttle position varying from idle to full open. The choke was adjusted as required. The results on sheet 25 were taken from runs with speed constant, the valve full open, the throttle varied and the dynamometer adjusted to obtain the desired speed. A comparison of the results of the conventional scavenging and the stratified charge scavenging is made from all the results plotting them on a bhp base instead of the usual scavenging air ratio base.

Figures 16-20 with TP 50%, Jet 20/16, NS 4700, from sheet 20

The fuel weighing method was used for fuel consumption and gave satisfactory results. The surge tank was kept connected during the run but the auxilliary nozzles were opened when the pressure drop was not being read. The vacuum in the surge tank remained at about .04 inches of water.

Analysis of figure 16 reveals that the speed, torque and power did not increase until the scavenging air ratio was greater than 5%. Nevertheless the specific fuel consumption decreased up to a ratio of 10% when it increased. From these curves it appears that the best scavenging air ratio would be 10%.

The reason for the power not increasing as would have been indicated by a speed increase, may be explained with the aid of figure 19. The power increase due to a decrease in the specific fuel consumption was offset by a reduction in the fuel consumption due to a decrease in the quantity of air flowing through the carburetor. With a further increase in the scavenging air ratio, the fuel consumption increased as the air-fuel ratio through carburetor decreased, but as the overall air-fuel ratio was increasing, the thermal efficiency also increased so that the net result was a power increase. With a further increase in the scavenging air ratio, the air-fuel ratio in the carburetor still decreased, but the air flow through carburetor increased, so that the overall air-fuel ratio decreased, reducing the thermal efficiency.

Comparing the fuel scavenging efficiency, figure 17 with the specific fuel consumption, figure 16, the curves seem to be of the same general form. Because of the similarity, the decrease in the specific fuel consumption seems to be the result of a larger proportion of fuel remaining in the cylinder.

The approximately linear relationship between the air-fuel ratio and the weight of air flow through carburetor breaks down above a ratio of 10 which means that the fuel flow fluctuated independently of the air flow through the carburetor, possible due to fuel control valve sticking, or to changes in air or fuel densities, or to fuel flow restrictions. It should be mentioned that the first test was with a scavenging air ratio of 11.5% and the last test, two hours later, was with a ratio of 10.0%.

Comparing the average combustion pressure, figure 20, fuel trapping efficiency, figure 17, and the overall air fuel ratio, figure 19, it appears that the peak combustion pressure varies with the amount of fuel-air mixture trapped. The exhaust temperature, figure 18, and the exhaust pressure drop, figure 20, vary inversely with the average combustion pressure. These relationships indicate that when the combustion pressure is high, more useful energy is removed from the cylinder. This increase in combustion pressure is probably the result of a more rapid rate of burning so that the combustion is completed earlier. With the peak combustion pressure occuring shortly after top dead centre, it has a longer time to act on the piston so more energy is removed, lowering the exhaust temperature. The exhaust pressure varies with the total air consumption as well as with the combustion pressure. Combustion would also be more complete if rapid combustion occured so that the amount of after-burning in the exhaust would reduce.

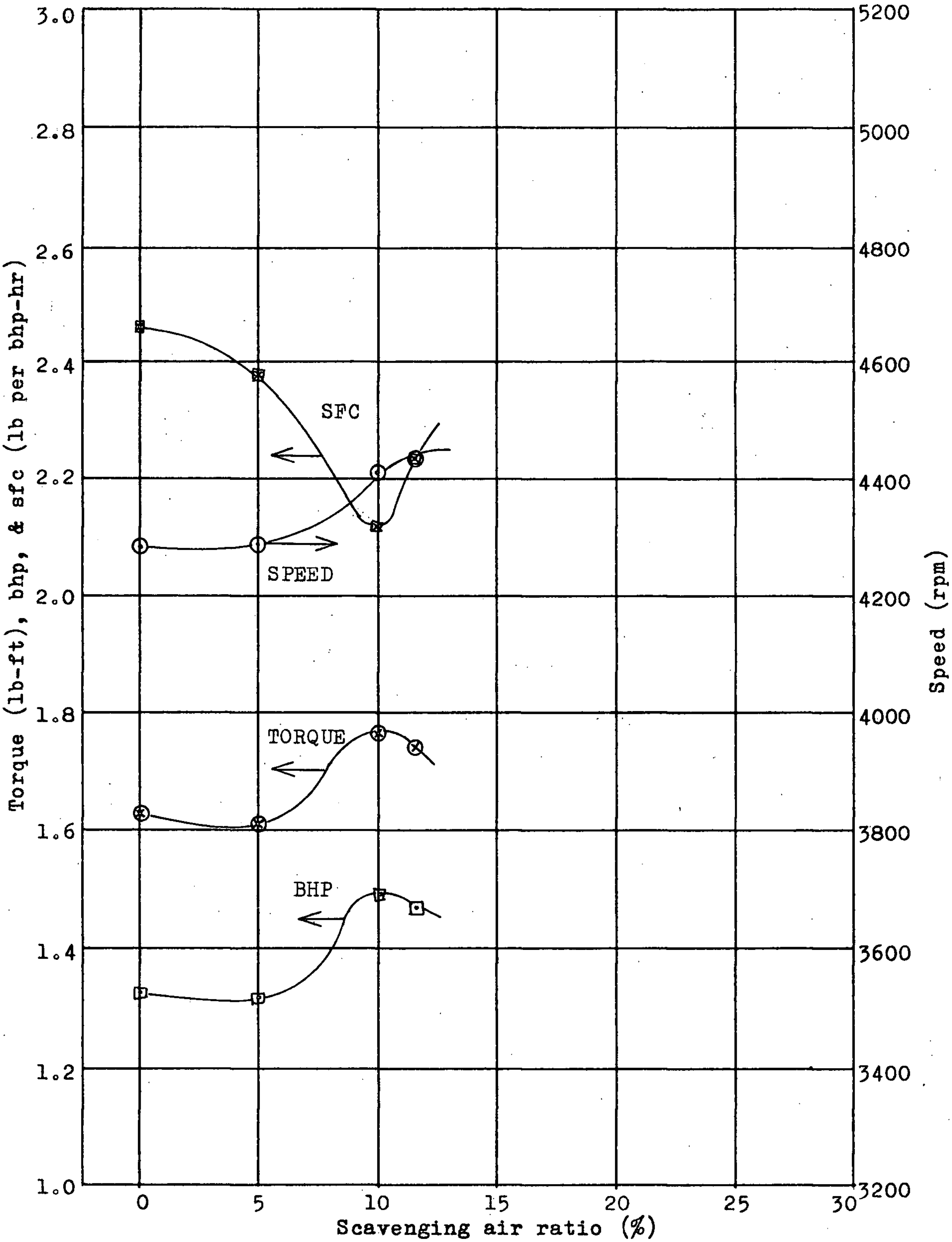

Figure 16. Performance curves with TP 50%, Jet 20/16, NS 4700, from sheet 20.

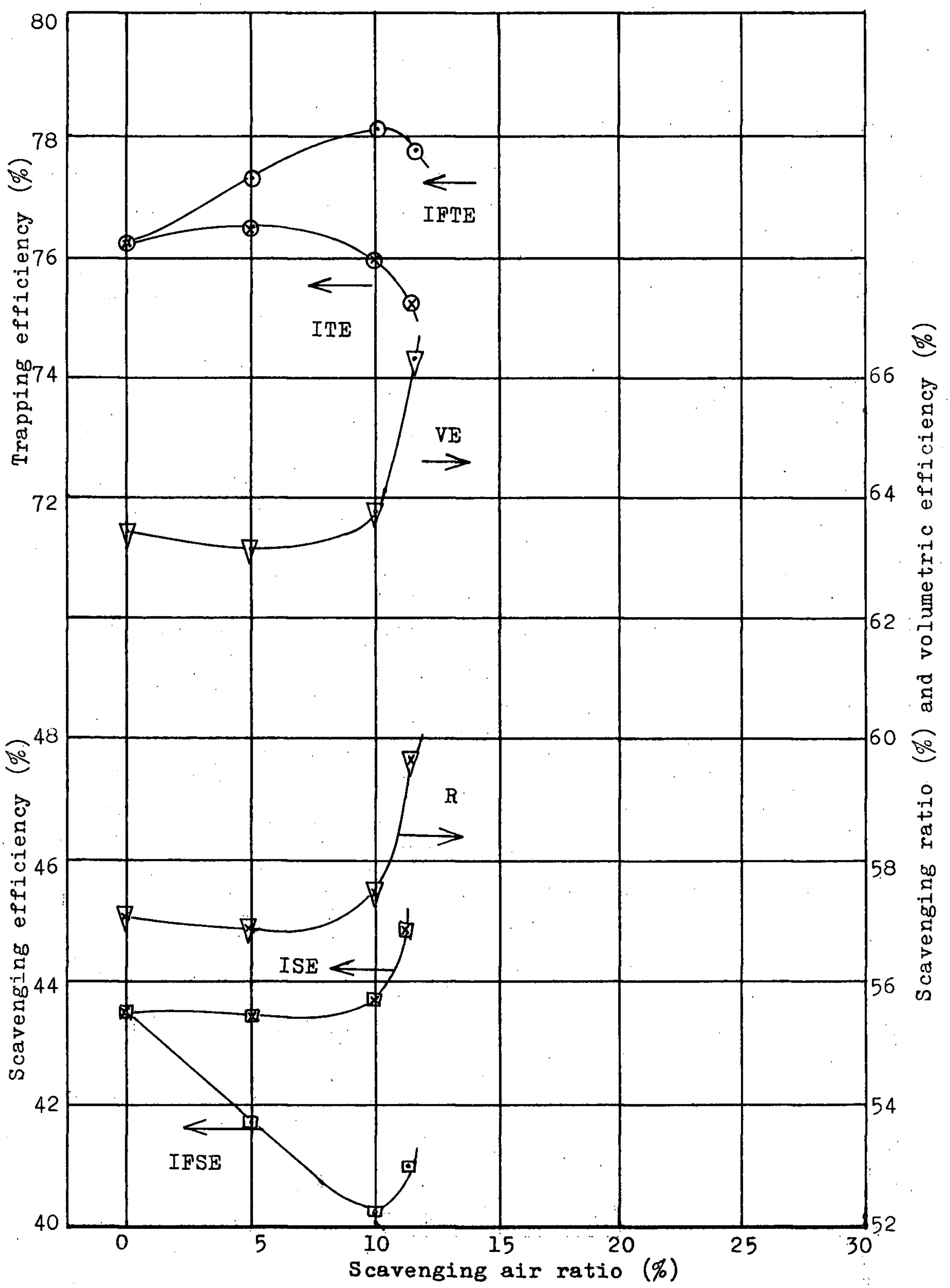

Figure 17. Efficiency curves with TP 50%, Jet 20/16, NS 4700, from sheet 20.

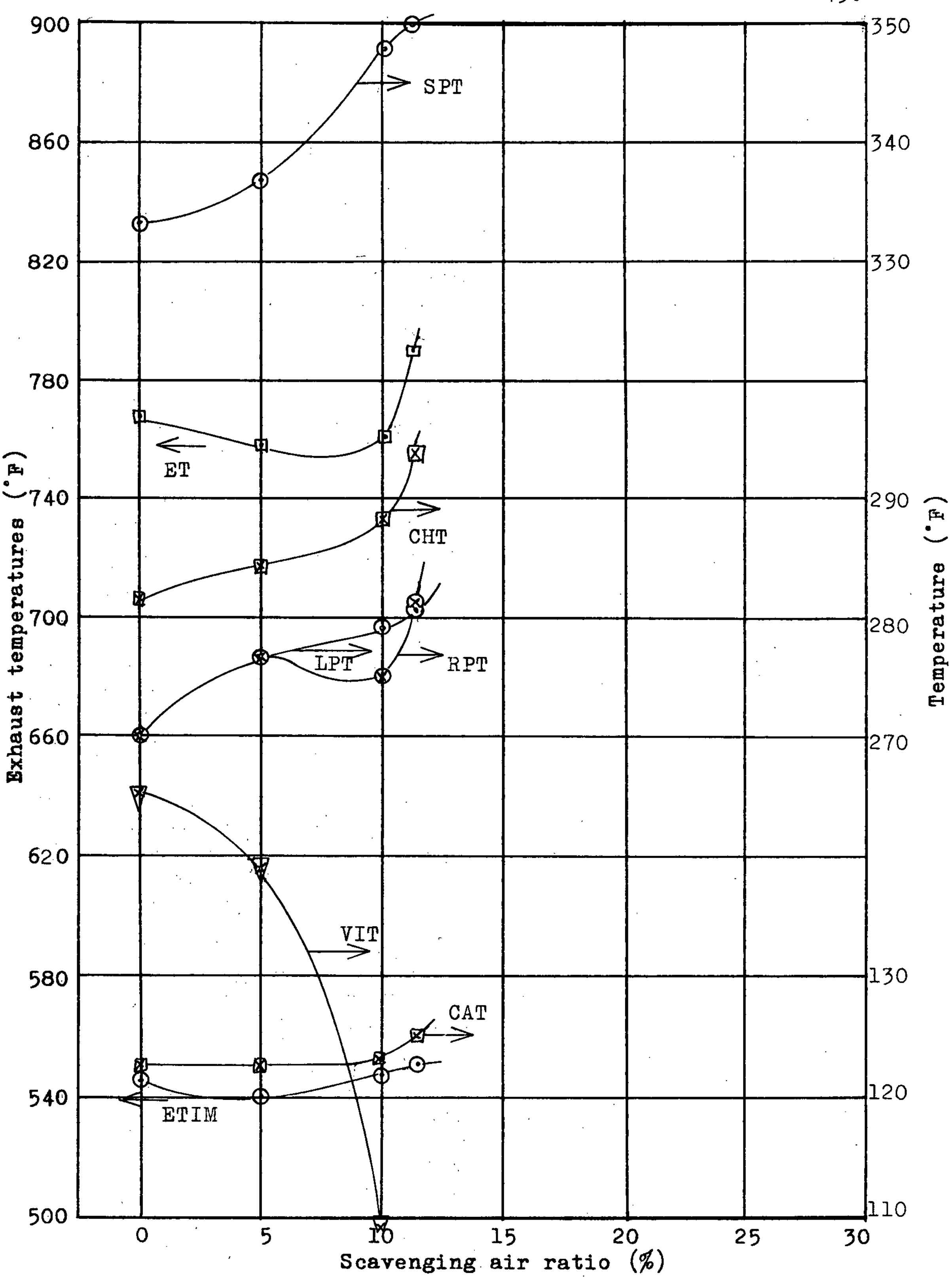

Figure 18. Temperature curves with TP 50%, Jet 20/16, NS 4700, from sheet 20.

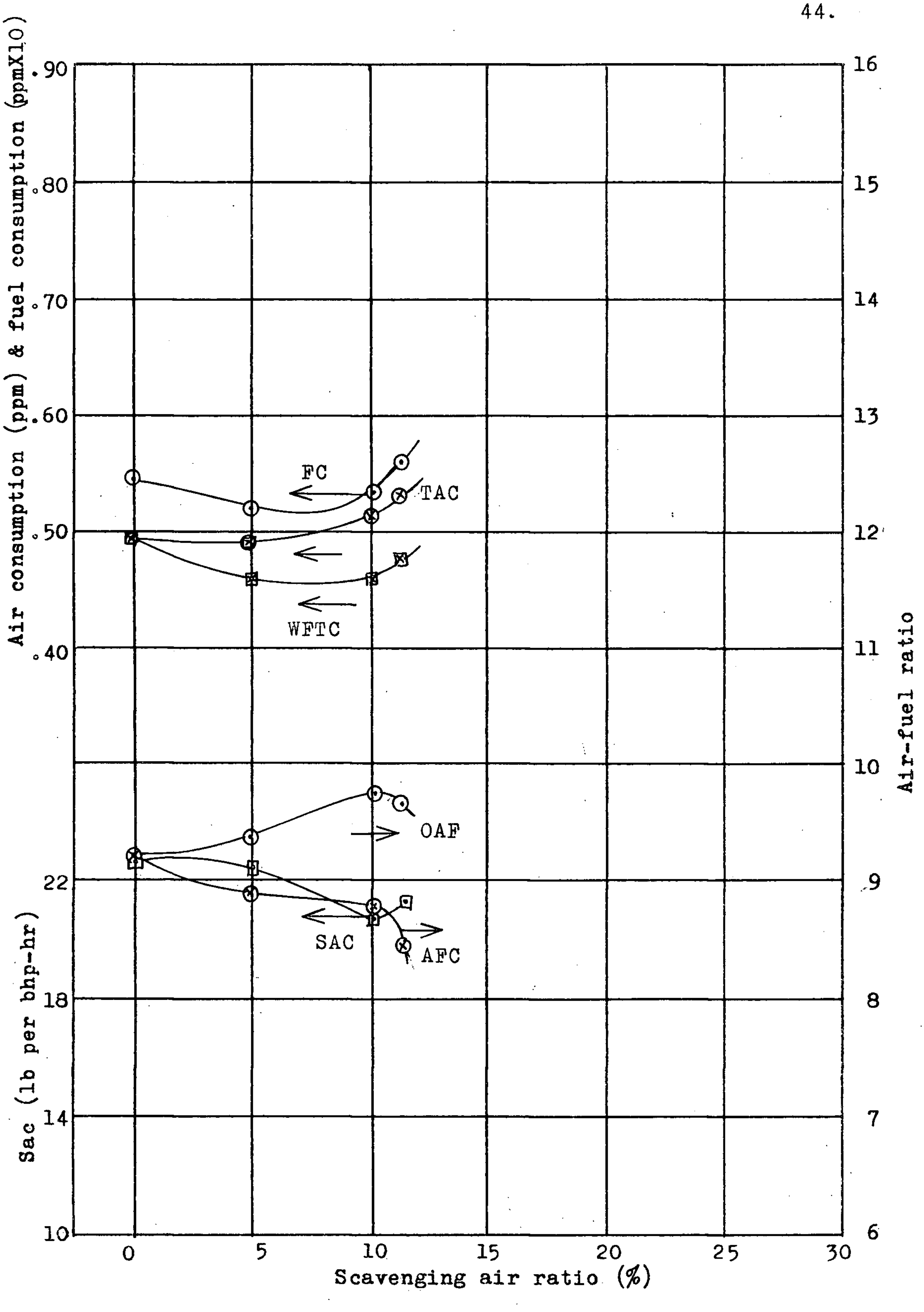

Figure 19. Air & fuel consumption curves with TP 50%, Jet 20/16, NS 4700, from sheet 20.

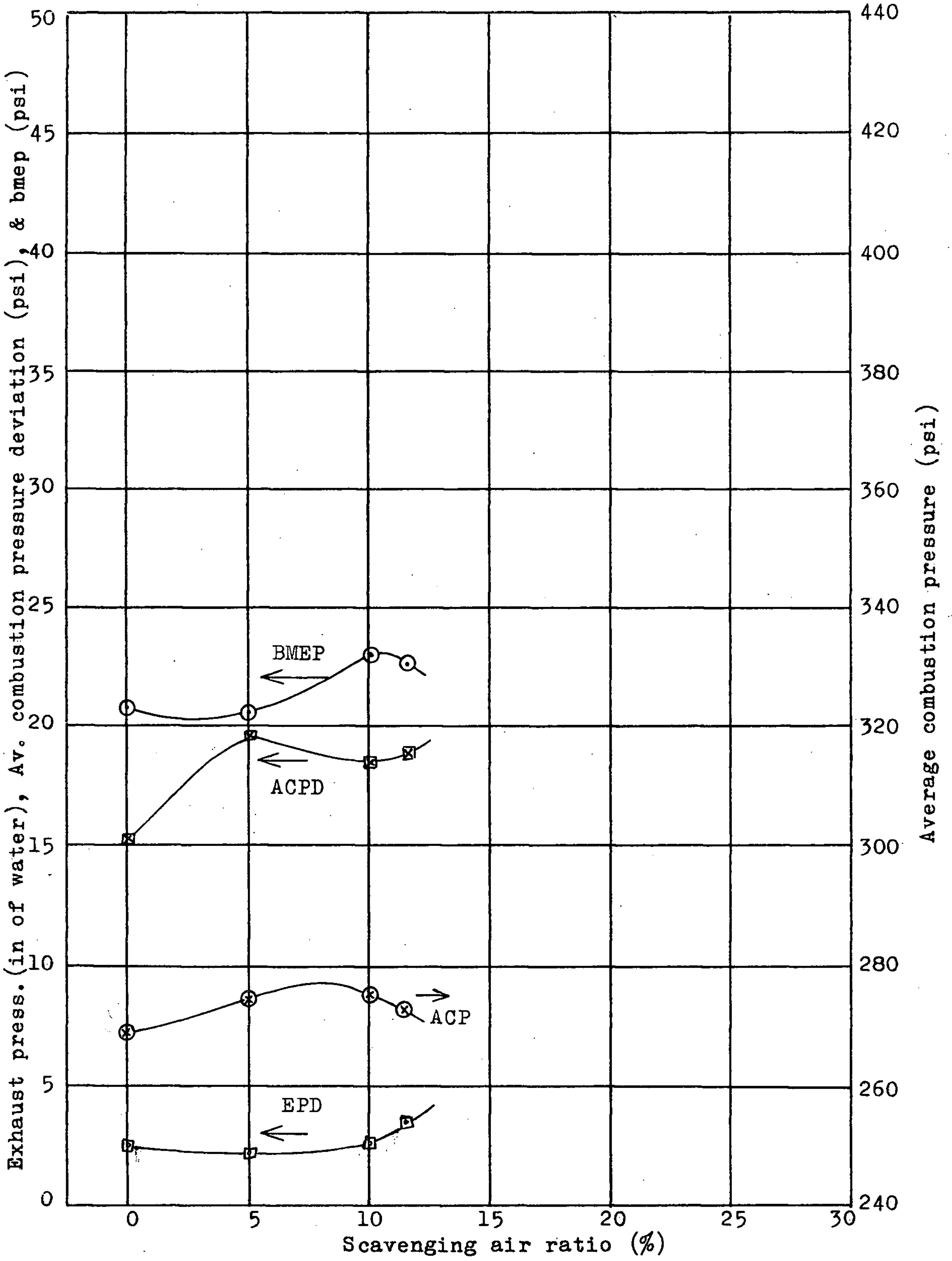

Figure 20. Pressure curves with TP 50%, Jet 20/16, NS 4700 from sheet 20.

(2) Figures 21-26 with TP 50%, Jet 16/16, NS 4500, from sheet 17

This test was with the throttle half open, the main jet one complete turn open, and with a nominal speed of 4500 rpm. During the tests, the engine suddenly slowed down to about two-thirds speed and then gradually crept back up to operating speed again. This phenomenon occured twice with scavenging air valve closed, once when the valve was 70% open, and once when it was 100% open, but not at all when the valve was 40% open. The sudden drop in speed seemed inexplicable but it may have been caused by carburetor malfunction or slugs of oil in the gas. Fuel consumption was measured by drawing the fuel from the pipette for one minute during each test. Carburetor air was drawn from surge tank through auxilliary nozzles except when the pressure drop across the nozzle was being measured. The carburetor intake pressure was thus below atmospheric pressure. The cylinder head temperatures could not be read as the thermocouple to the nut had broken off. Generally the temperatures were quite constant during a test.

The specific fuel consumption versus scavenging air ratio is not a straight line as the other main operating characteristics seem to be, as indicated on figure 21. As the fuel consumption drops off at scavenging air ratio of 3.5%, figure 25, one could suspect that the fuel consumption measurement over the one minute interval was not representative of the average consumption over the whole time. But the exhaust temperature dropped and the left port temperature increased probably due to the valve intake temperature increase so that, due to rapid combustion in a greater amount of air, more energy was removed before the exhaust was discharged. This could explain why the fuel consumption decreased yet the power continued to increase. The total air consumption, figure 25, increased beyond proportion at this scavenging air ratio so that the air-fuel ratio curve also was not a straight line.

As the brake mean effective pressure and the average peak combustion pressure are proportional, figure 26, the increased effective pressure was the result of higher pressures acting for longer periods as the combustion was completed sooner. With higher overall pressures, the overall temperature should also be higher; this is born out by the spark plug temperature in figure 24. Speed and power increase, figure 21, is probably the result of more fuel being retained and more efficient combustion in a greater proportion of air. The valve intake temperature was high, figure 24, so when the scavenging air valve was opened and the hot scavenging air entered the crankcase, the passageway temperature increased. As more air flowed past the hot scavenging valve where it picked up its heat, its temperature reduced, thus decreasing the passageway temperature until the increasing temperature of the passageway walls and crankcase counteracted the valve intake temperature decrease.

The hot scavenging air has a high ignition energy so it is not very surprising to notice that the brake thermal efficiency was high, figure 23, the exhaust temperature low, figure 24, and the exhaust pressure drop low, figure 26, when the valve intake temperature was high at a scavenging air ratio of 3.5%. The exhaust temperature and pressure are low because more energy is removed from the cylinder when combustion is faster, leaving a smaller amount in the exhaust gases, as well as reducing the amount of afterburning in the exhaust pipe. The exhaust pressure also depends on the quantity of air being discharged, which is confirmed by comparing these quantities on figure 25 and 26. Nevertheless the average peak cylinder pressure deviation increased rapidly indicating greater cyclic irregularity.

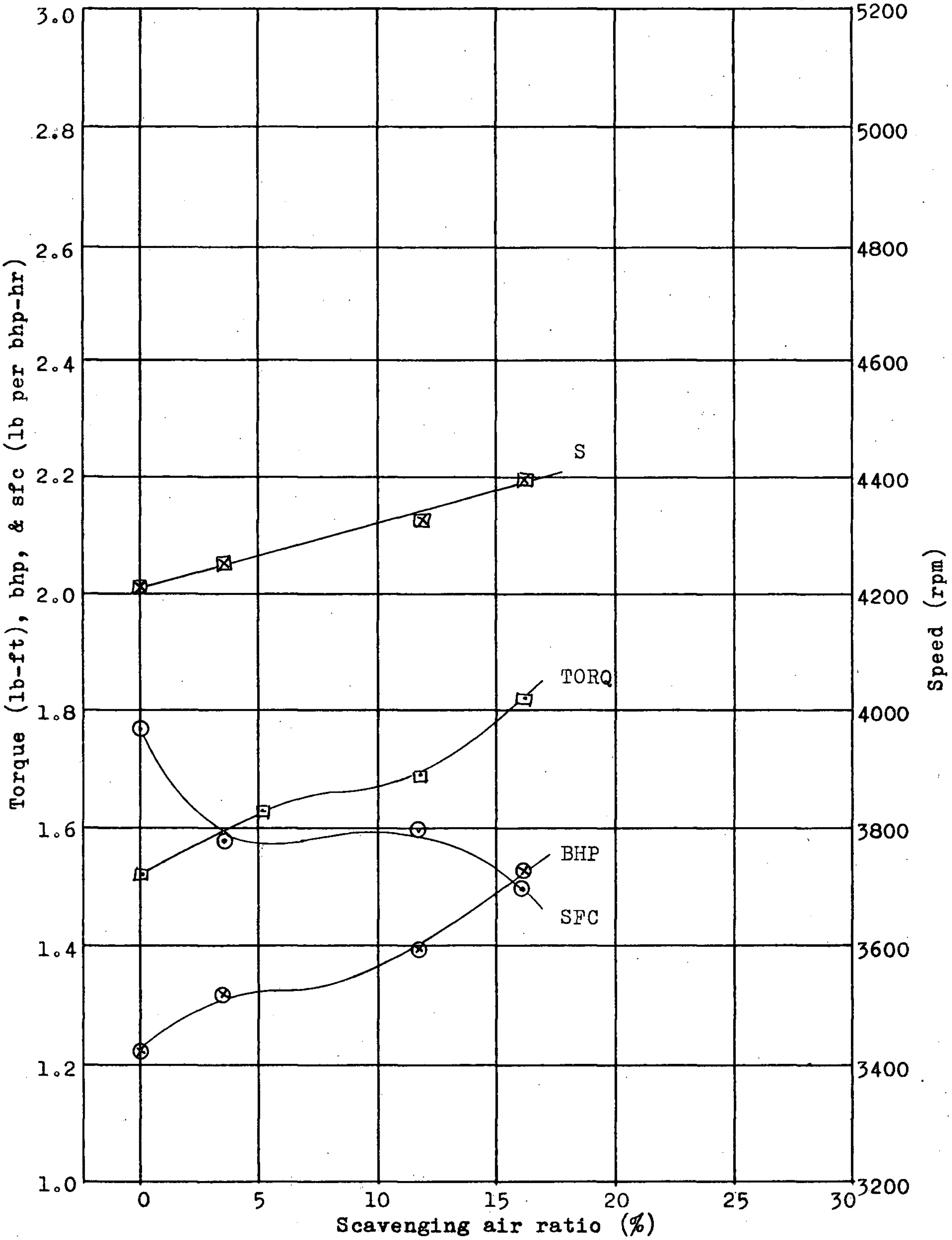

Figure 21. Performance curves with TP 50%, Jet 16/16, NS 4500, from sheet 17.

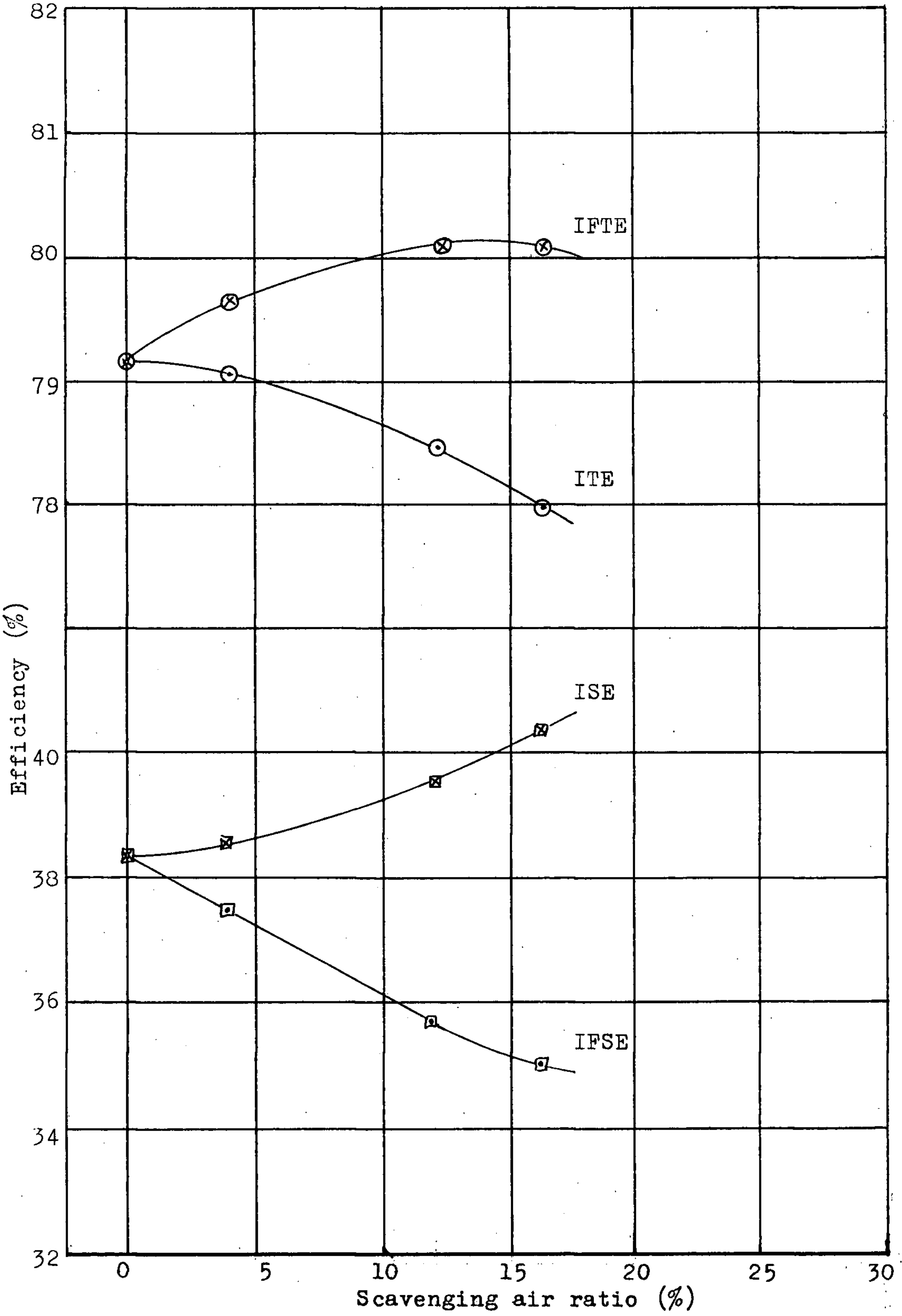

Figure 22. Efficiency curves with TP 50%, Jet 16/16, NS 4500, from sheet 17.

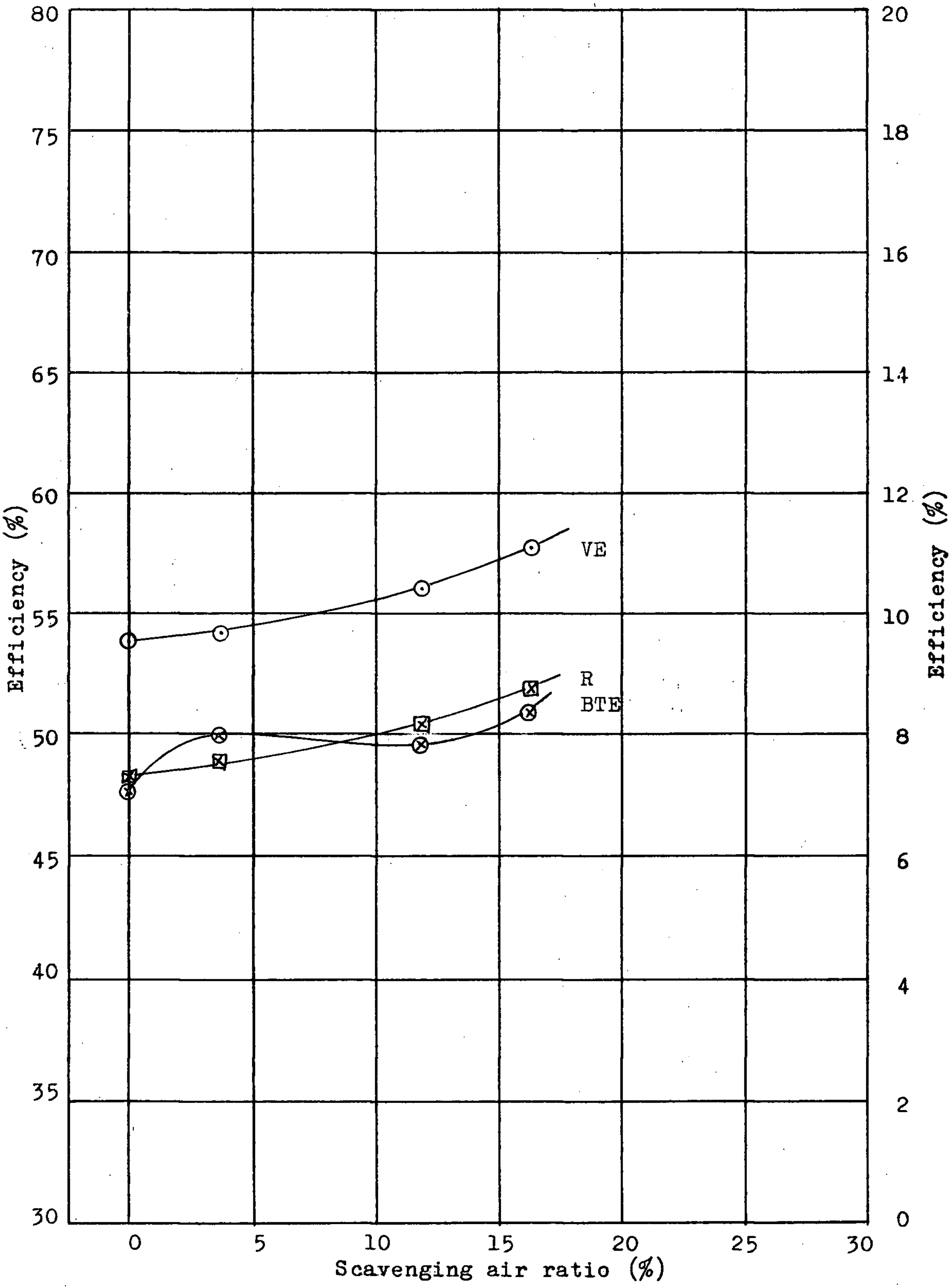

Figure 23. Efficiency curves with TP 50%, Jet 16/16, NS 4500, from sheet 17.

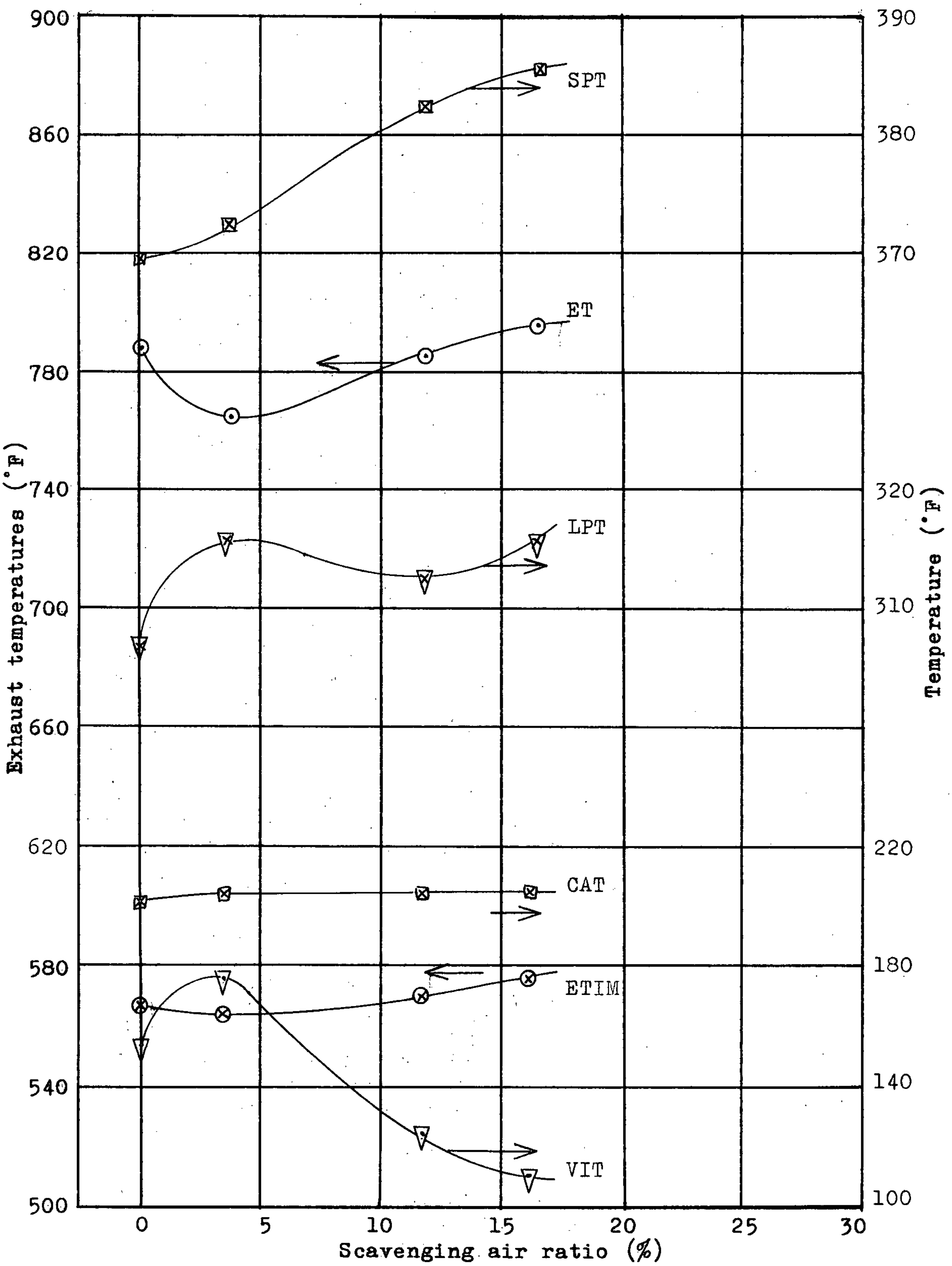

Figure 24. Temperature curves with TP 50%, Jet 16/16, NS 4500, from sheet 17.

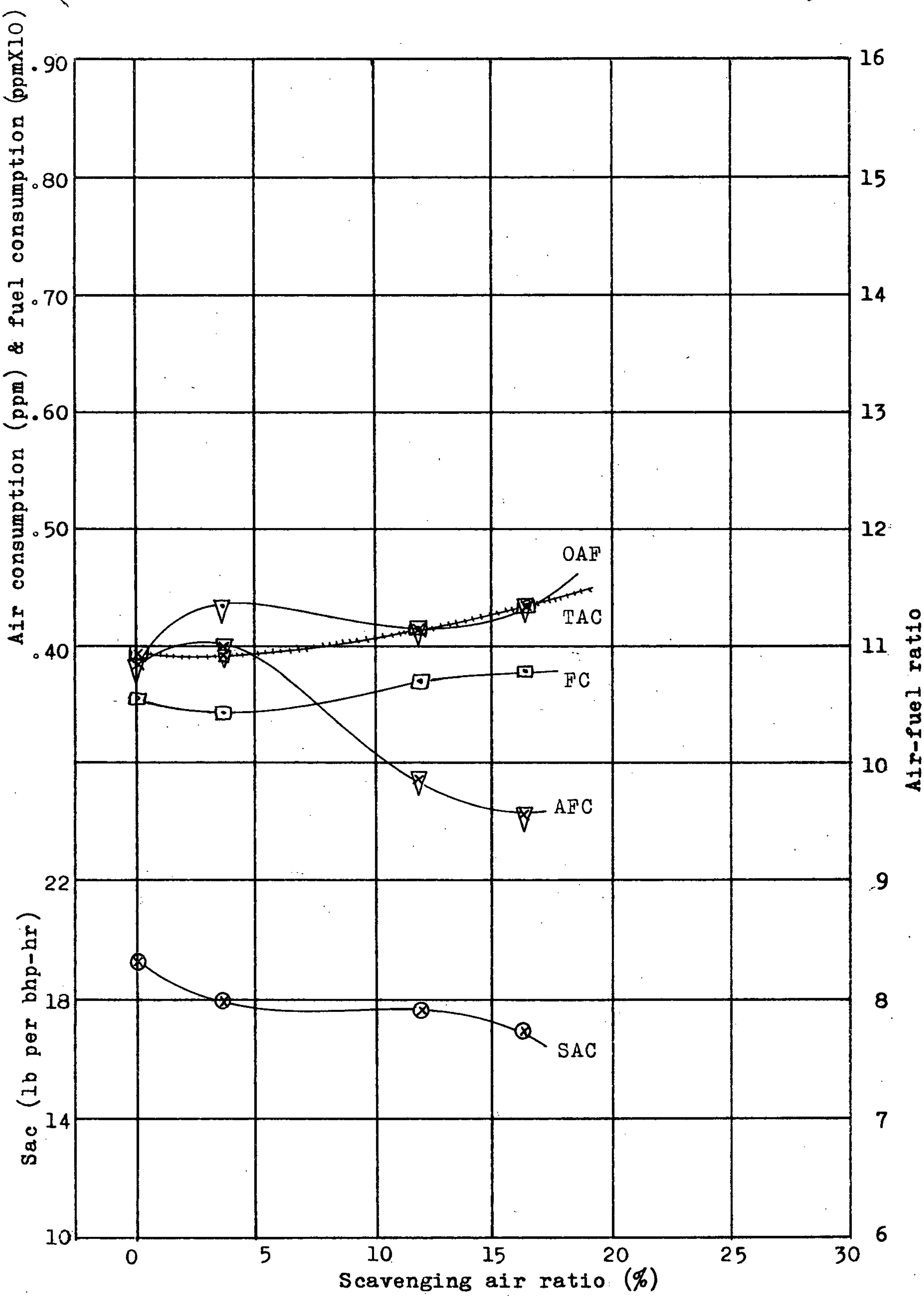

Figure 25. Air & fuel consumption curves with TP 50%, Jet 16/16, NS 4500, from sheet 17.

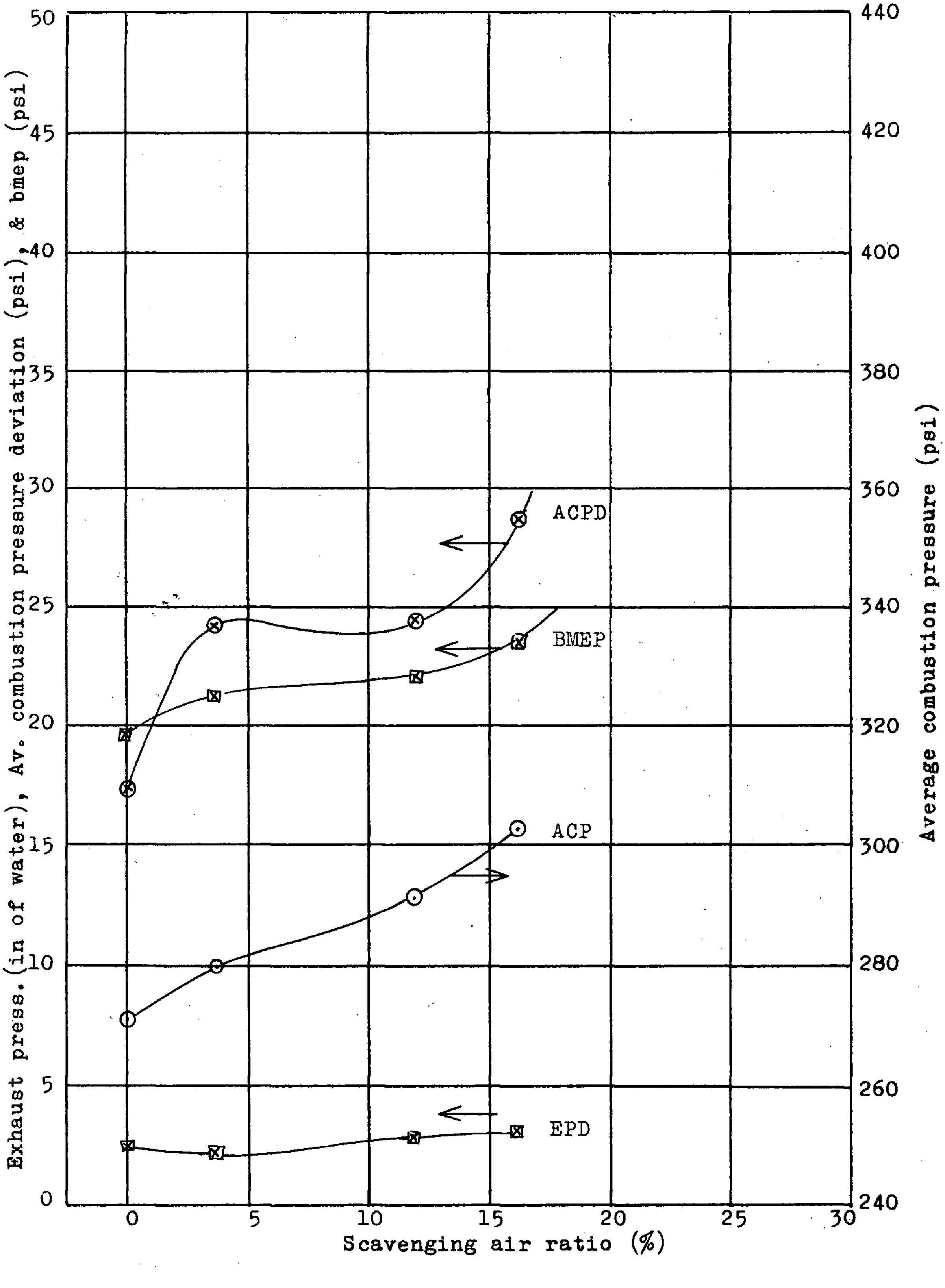

Figure 26. Pressure curves with TP 50%, Jet 16/16, NS 4500, from sheet 17.

(3) Figure 27-32 with TP 50%, Jet 14/16, NS 4900, from sheet 15

The test was conducted at 50% full open throttle, jet open 7/8 full turn, and a nominal speed of 4900. When not being measured, the air entered the surge tank through auxilliary nozzles so the pressure in the surge tank varied between .04 and .06 in. of water. The maximum amount of air that could be admitted through the full open valve was only 5% of total air consumption.

Power, speed, and brake mean effective pressure, figures 27 and 32, dropped at first as air was admitted through the savenging air valve. This is probably due to the peak combustion pressure decreasing at first, and then increasing. The probability of this being the cause is quite high as the pressure in the cylinder did actually drop and then rise again, figure 32. The spark plug temperature did similarly, figure 30, suggesting that combustion was slower at a scavenging air ratio of 2.5% and then becoming faster. The higher exhaust temperature indicated that more energy was lost through the exhaust gases. The reason why the exhaust pressure varied inversely with the exhaust temperature instead of directly, as would have been expected with an increase in the exhaust energy, is that the total air consumption decreased and then increased, the curve being similar in form to the exhaust pressure curve.

From the increase in the fuel trapping efficiency, figure 28, one would expect that the brake thermal efficiency would increase, but this is not the case; the efficiency remains constant up to 2.5% and then drops. But in figure 31, the air-fuel ratio is shown to decrease continually, so that, even though more of the mixture remains in the cylinder, combustion is slower due to a shortage of oxygen.

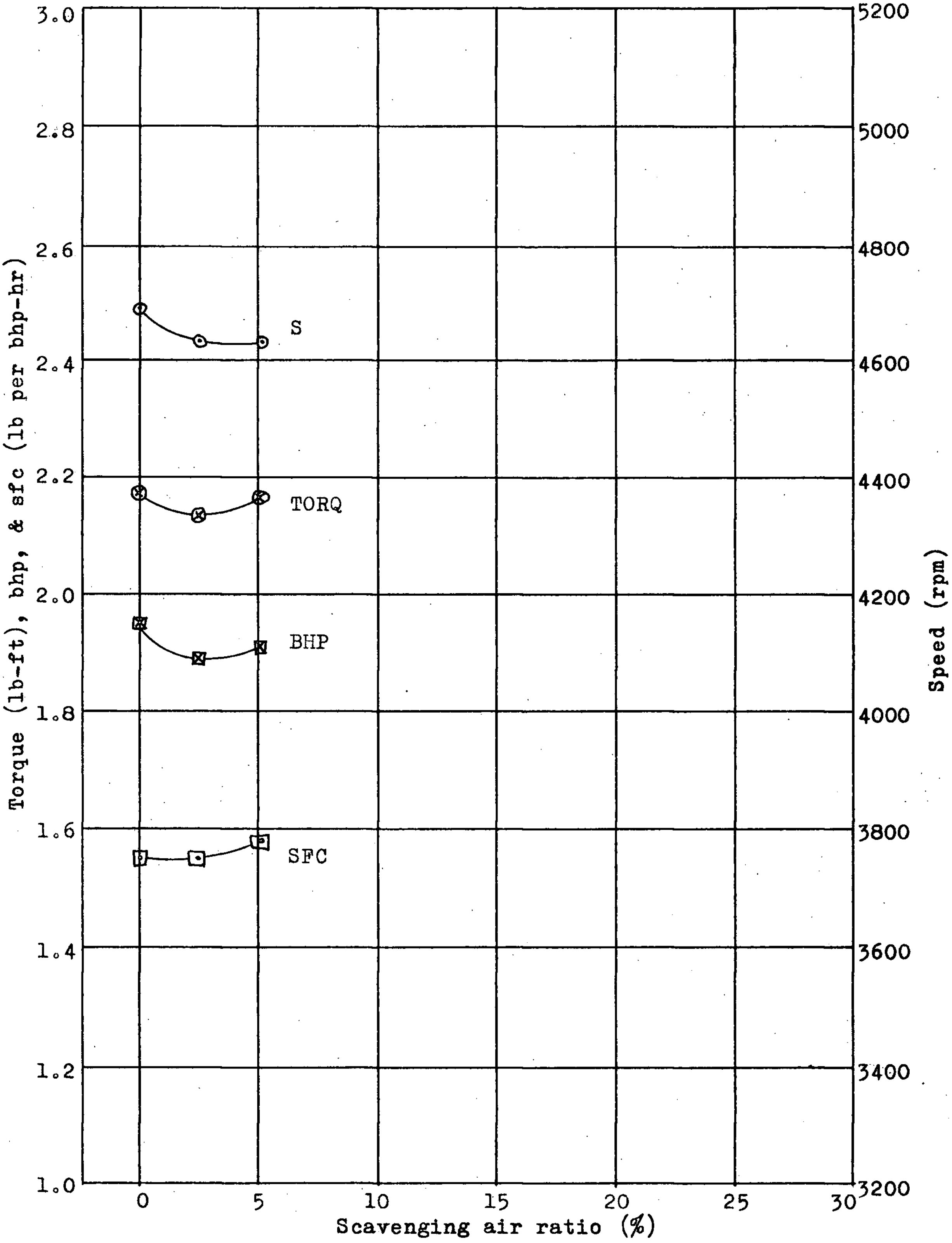

Figure 27. Performance curves with TP 50%, Jet 14/16, NS 4900, from sheet 15.

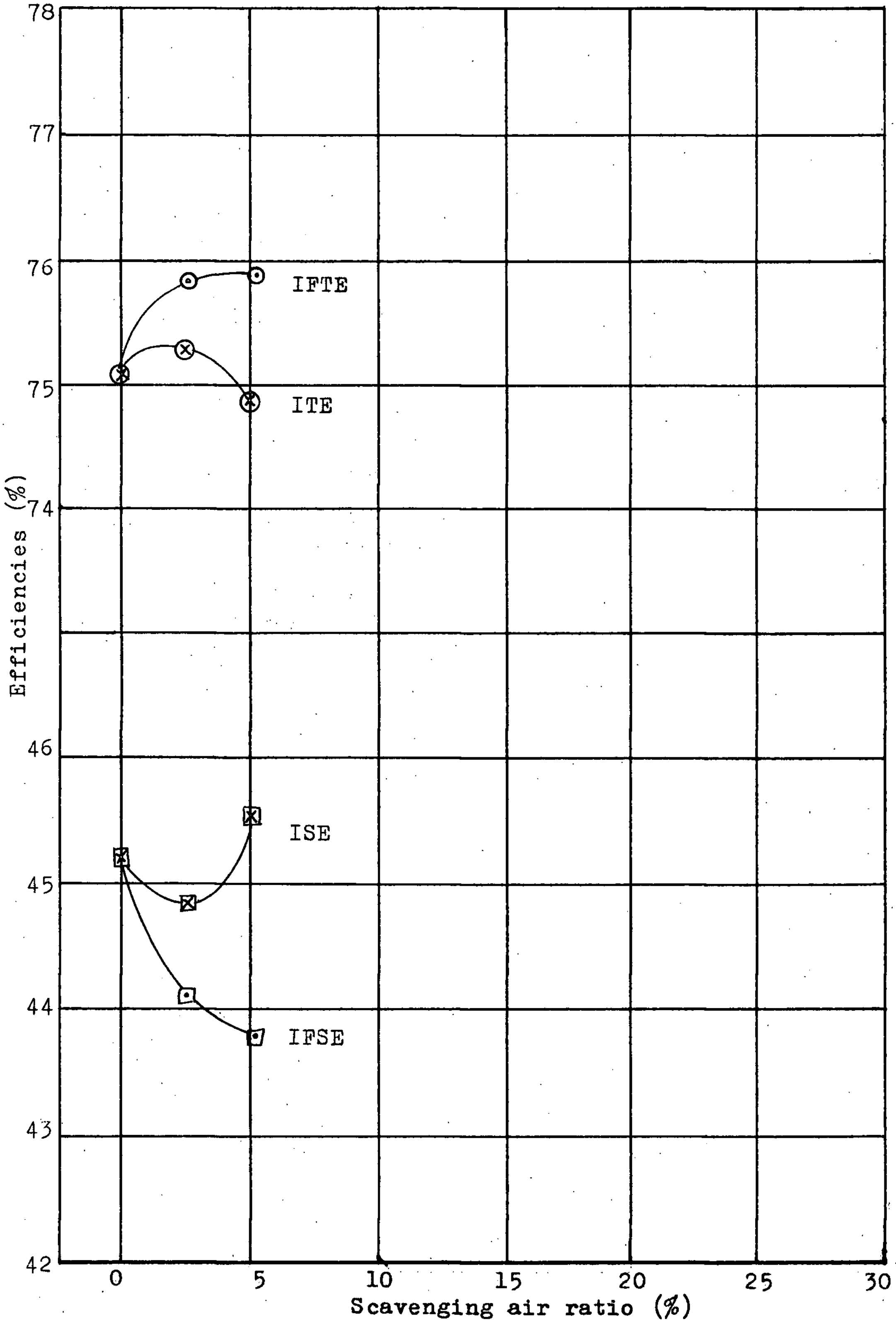

Figure 28. Efficiency curves with TP 50%, Jet 14/16, NS 4900, from sheet 15.

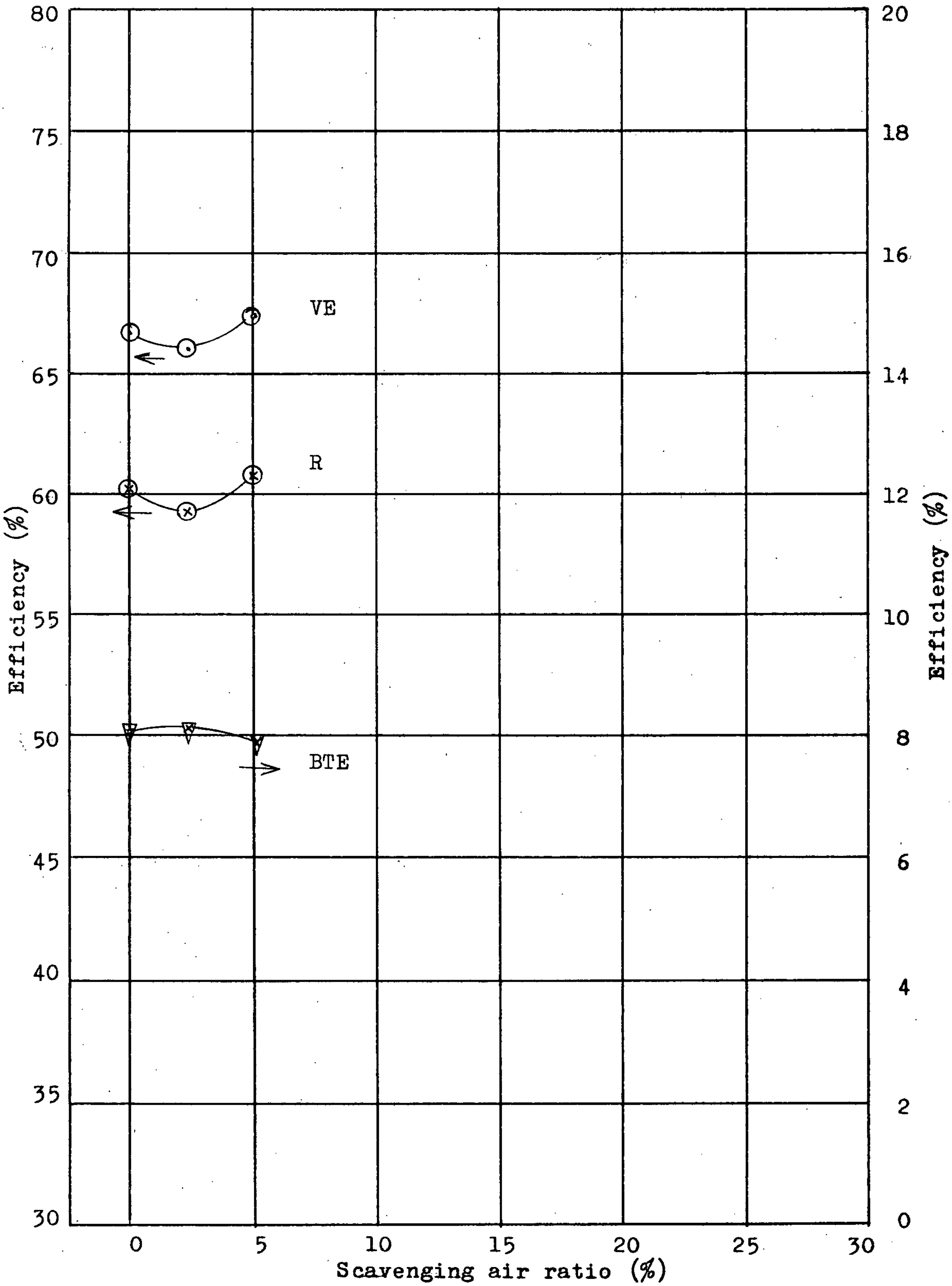

Figure 29. Efficiency curves with TP 50%, Jet 14/16, NS 4900, from sheet 15.

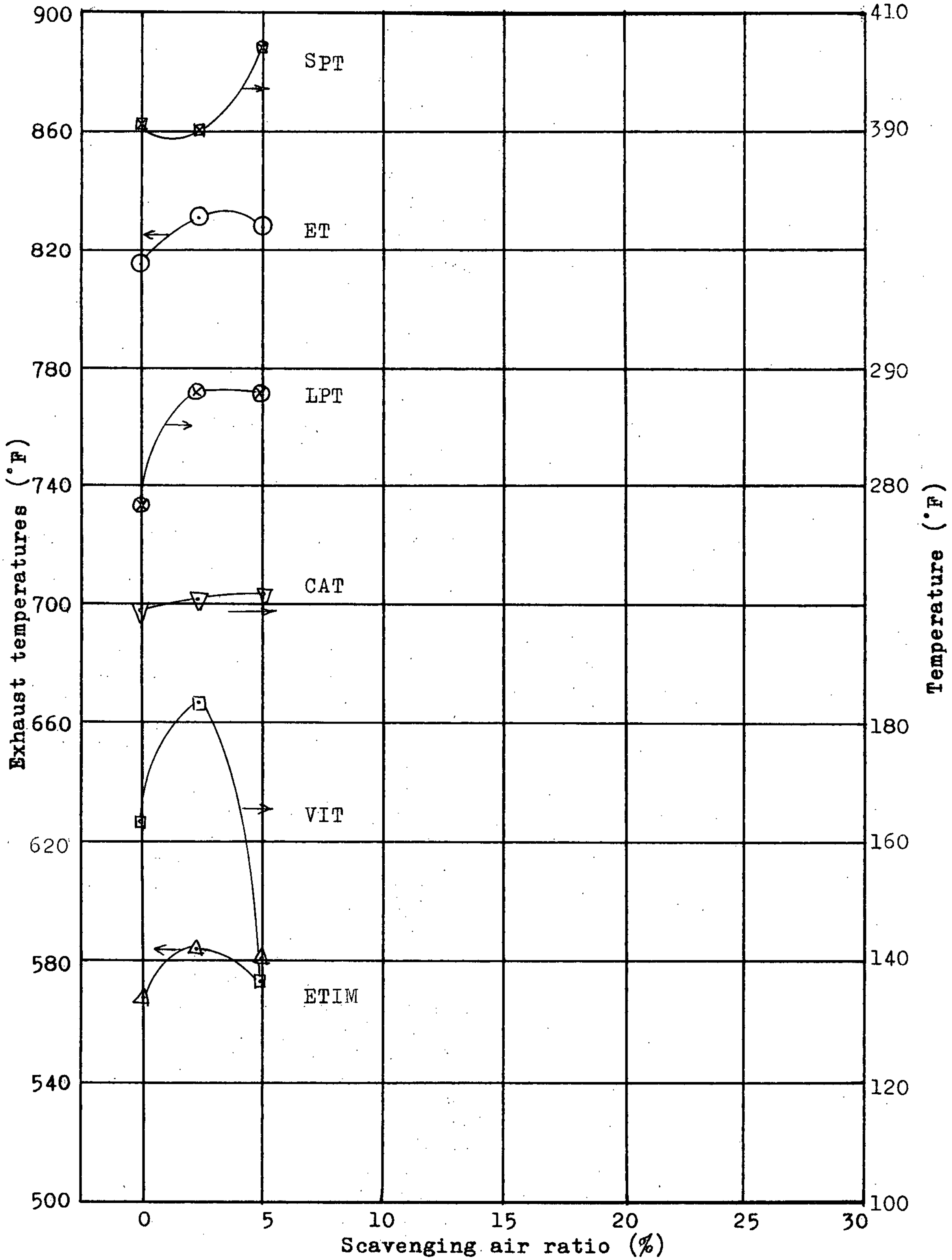

Figure 30. Temperature curves with TP 50%, Jet 14/16, NS 4900, from sheet 15.

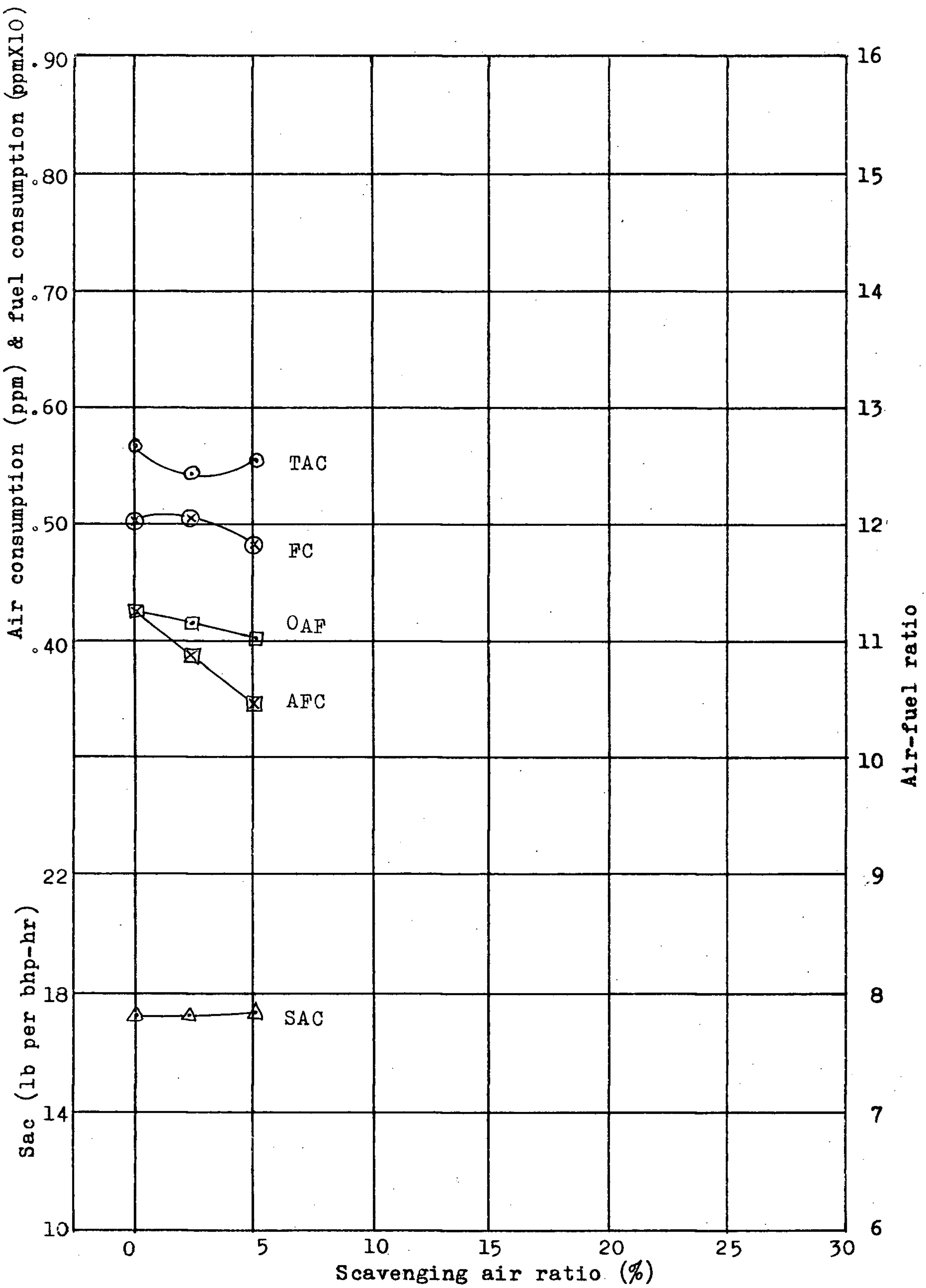

Figure 31. Air and fuel consumption curves with TP 50%, Jet 14/16, NS 4900, from sheet 15.

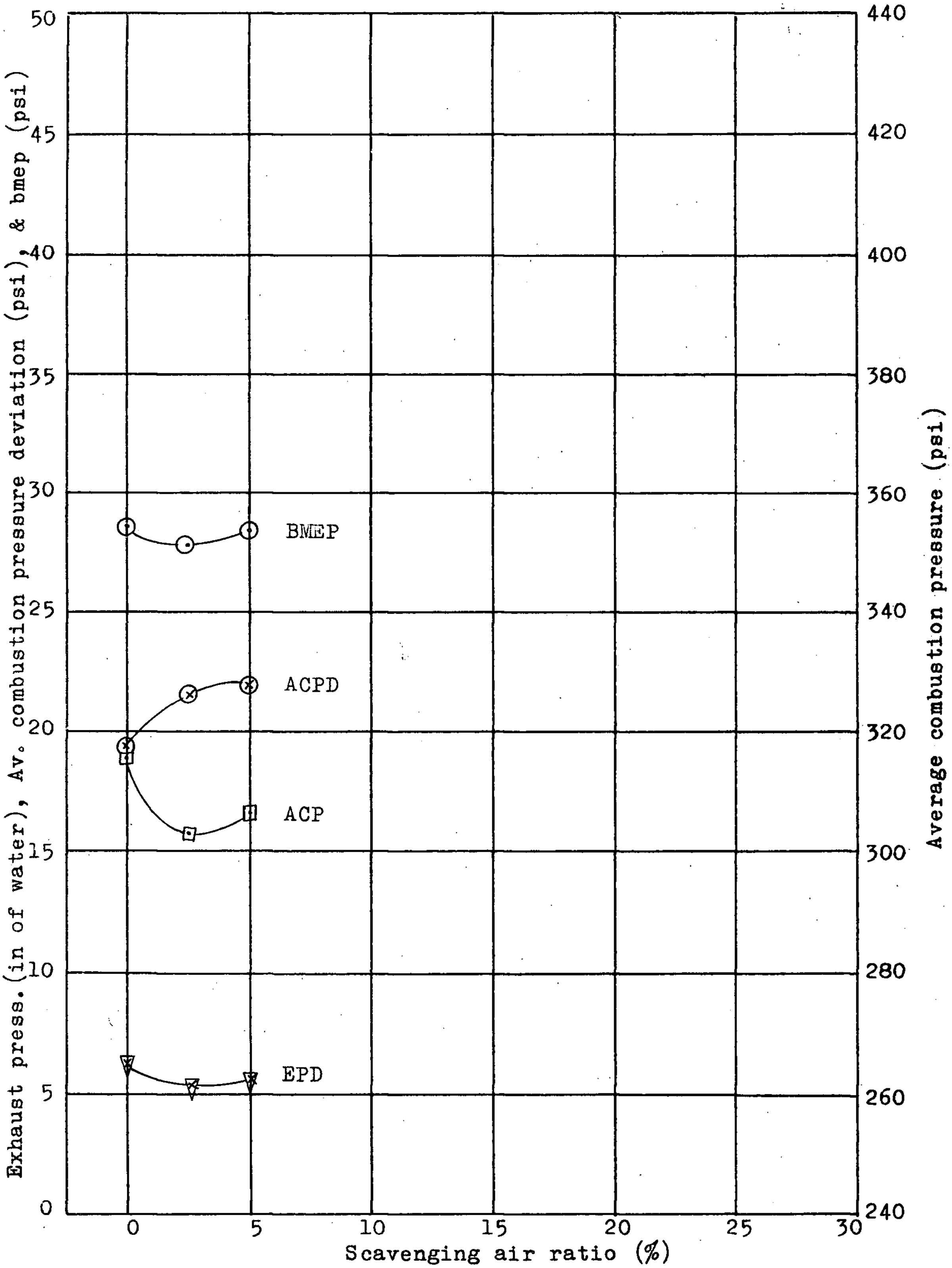

Figure 32. Pressure curves with TP 50%, Jet 14/16, NS 4900, from sheet 15.

(4) Figures 33-38 with TP 60%, Jet F/A, NS 5000, from sheet 21

This test at a throttle position 60% open and nominal speed 5000 rpm, was with a new carburetor adjusted at the factory. But as the instrumented engine had greater restrictions to the carburetor air flow than an uninstrumented one at the factory would have, the fuel flow in carburetor would not be the same in both cases. Nevertheless the engine ran relatively smooth during the test but it did not seem to be running under optimum conditions: when choked or when the carburetor intake pressure was increased, the engine would speed up.

The resultant operating characteristics were far from normal, e.g. specific fuel consumption was 4.0 lbs per bhp-hr. The air-fuel ratio in the carburetor dropped but the overall air-fuel ratio increased slightly, as shown in figure 37.

The air-flow restrictions were reduced so that the volumetric efficiency increased. As the ratio of the exhaust temperature to the total air consumption is nearly constant, the amount of heat lost by the exhaust gases depends on the total air consumption and independent of the combustion variation. Thus the only major gain in efficiency is due to an increase in the trapping efficiency.

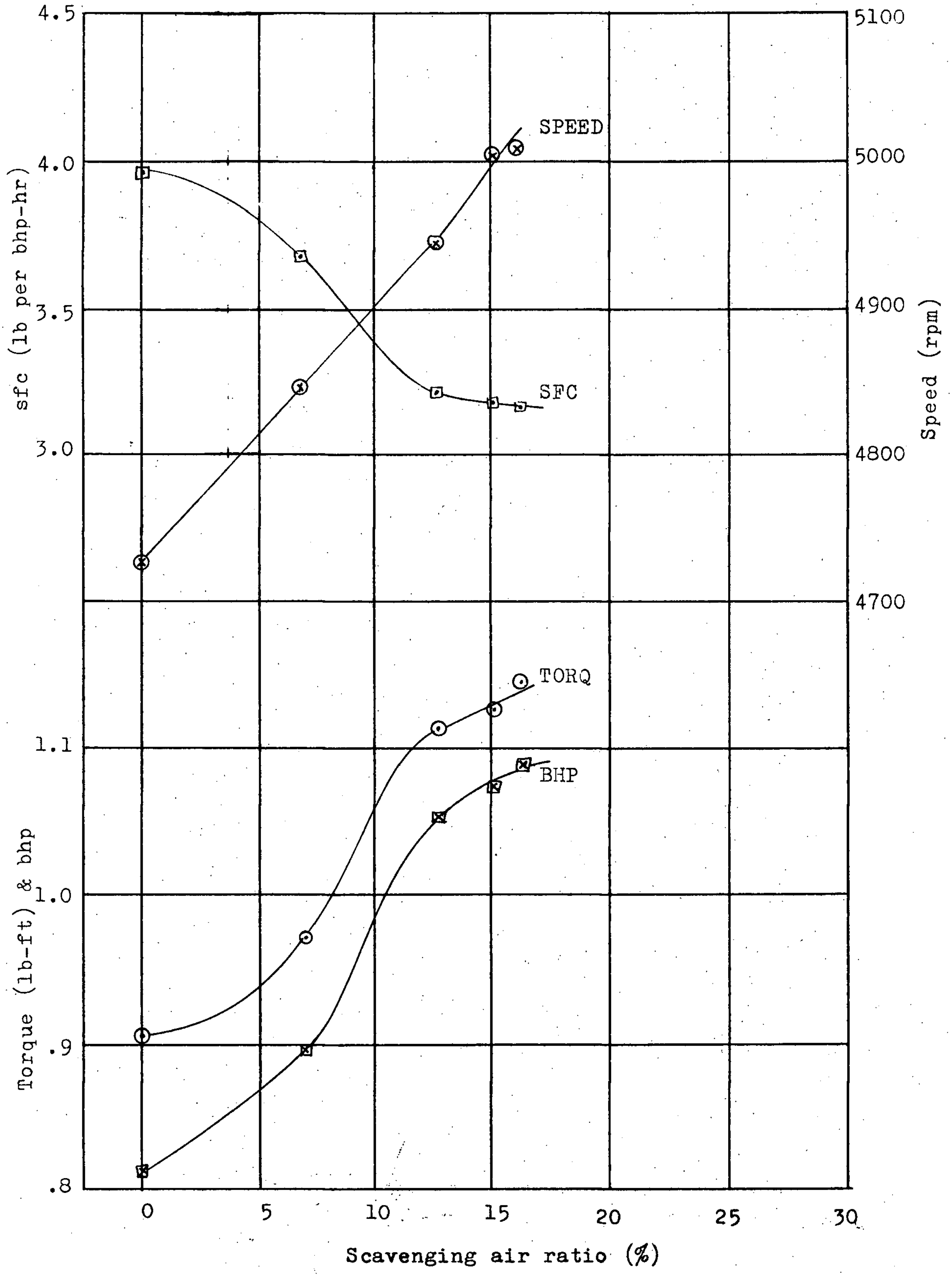

Figure 33. Performance curves with TP 60%, Jet FA, NS 5000 from sheet 21.

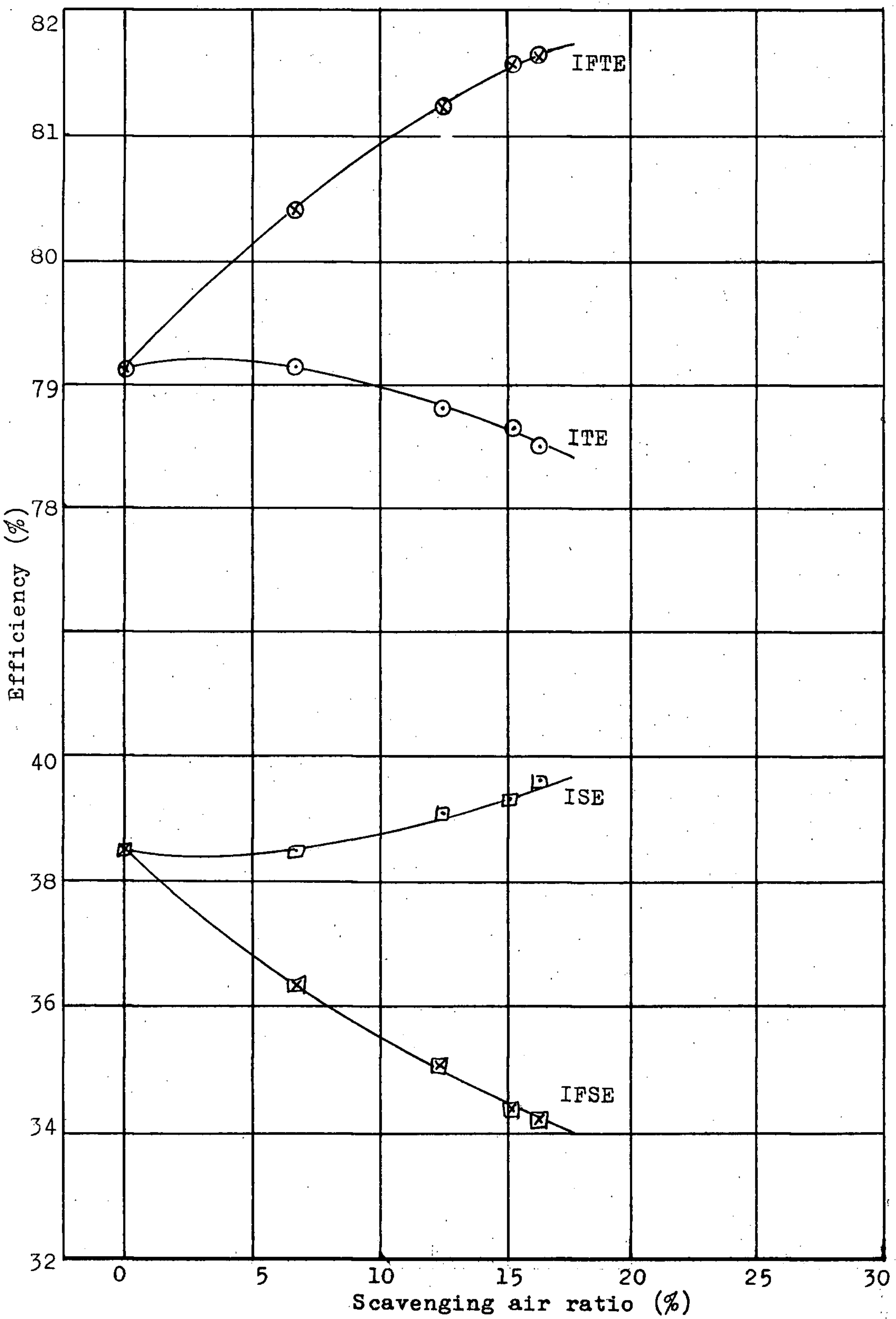

Figure 34. Efficiency curves with TP 60%, Jet FA, NS 5000, from sheet 21.

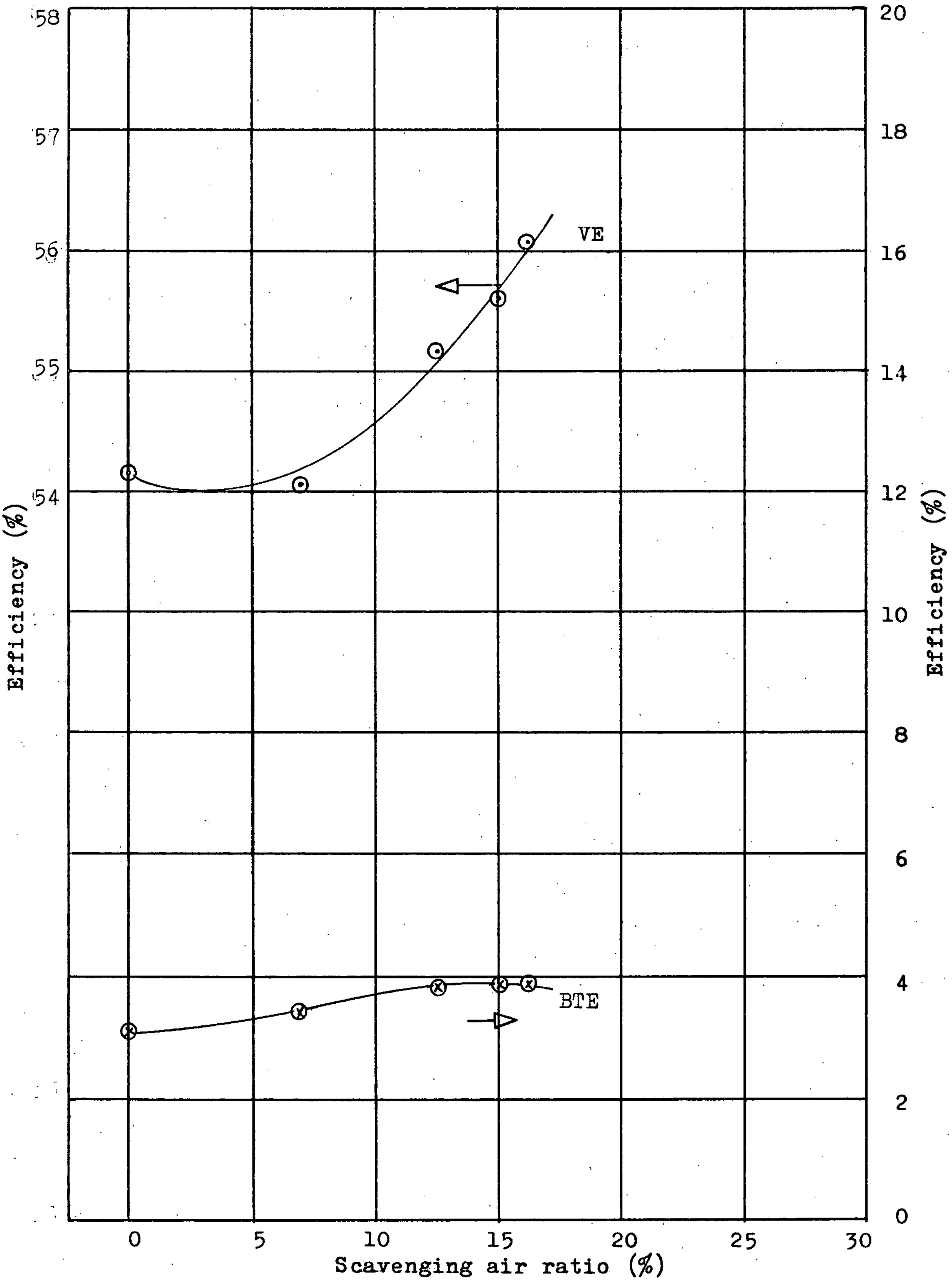

Figure 35. Efficiency curves with TP 60%, Jet FA, NS 5000, from sheet 21.

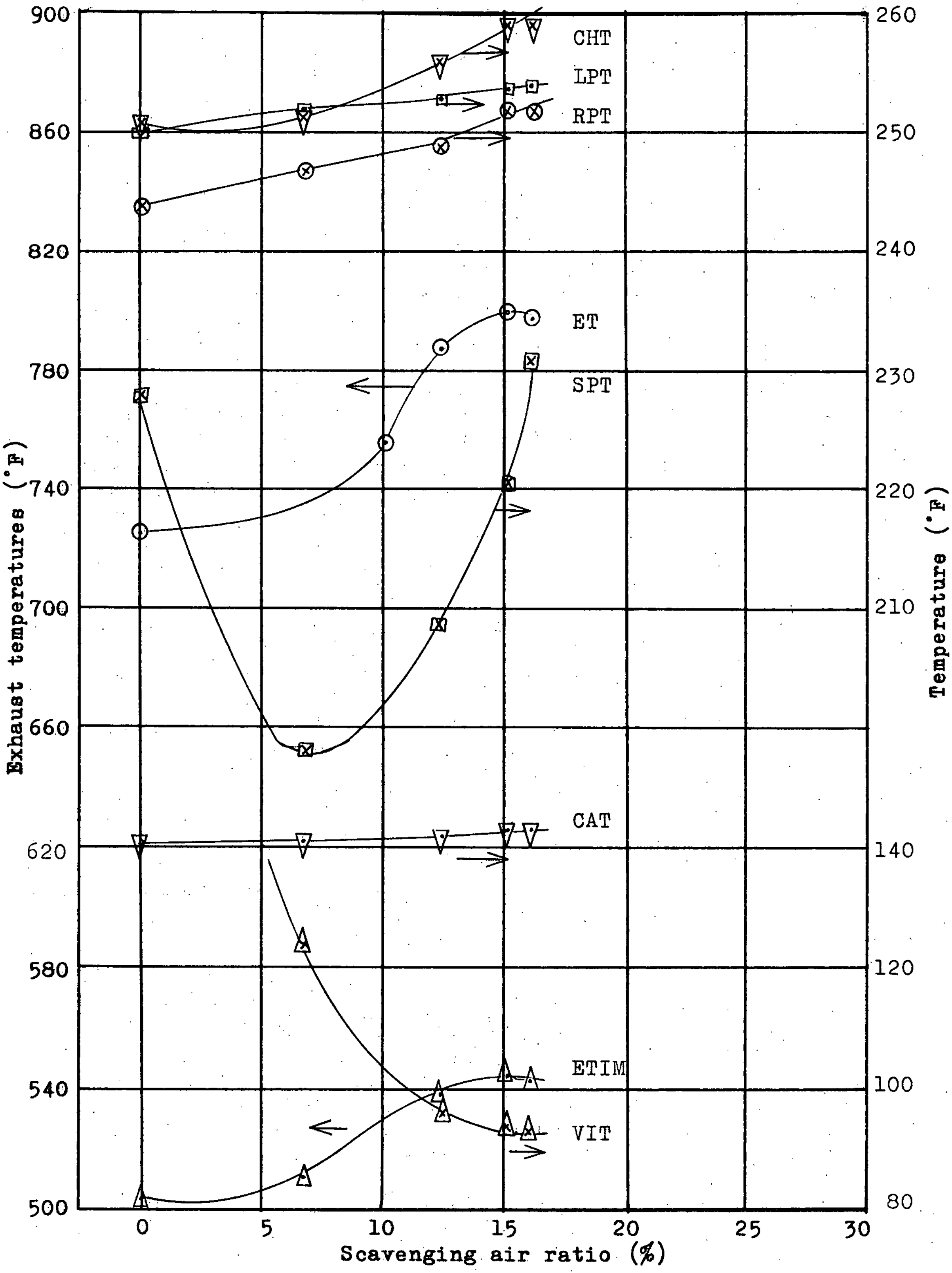

Figure 36. Temperature curves with TP 60%, Jet FA, NS 5000, from sheet 21.

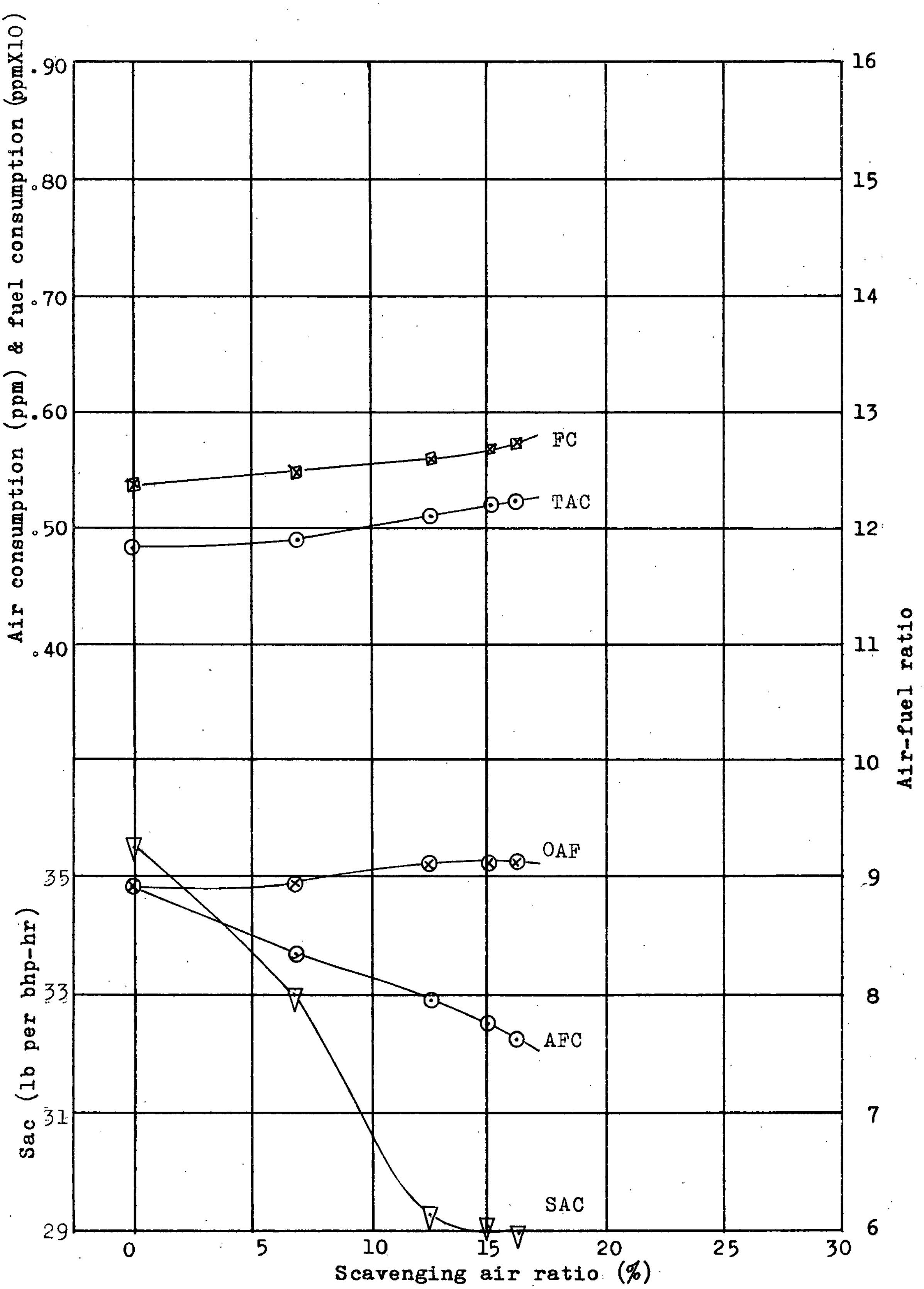

Figure 37. Air & fuel consumption curves with TP 60%, Jet FA, NS 5000, from sheet 21.

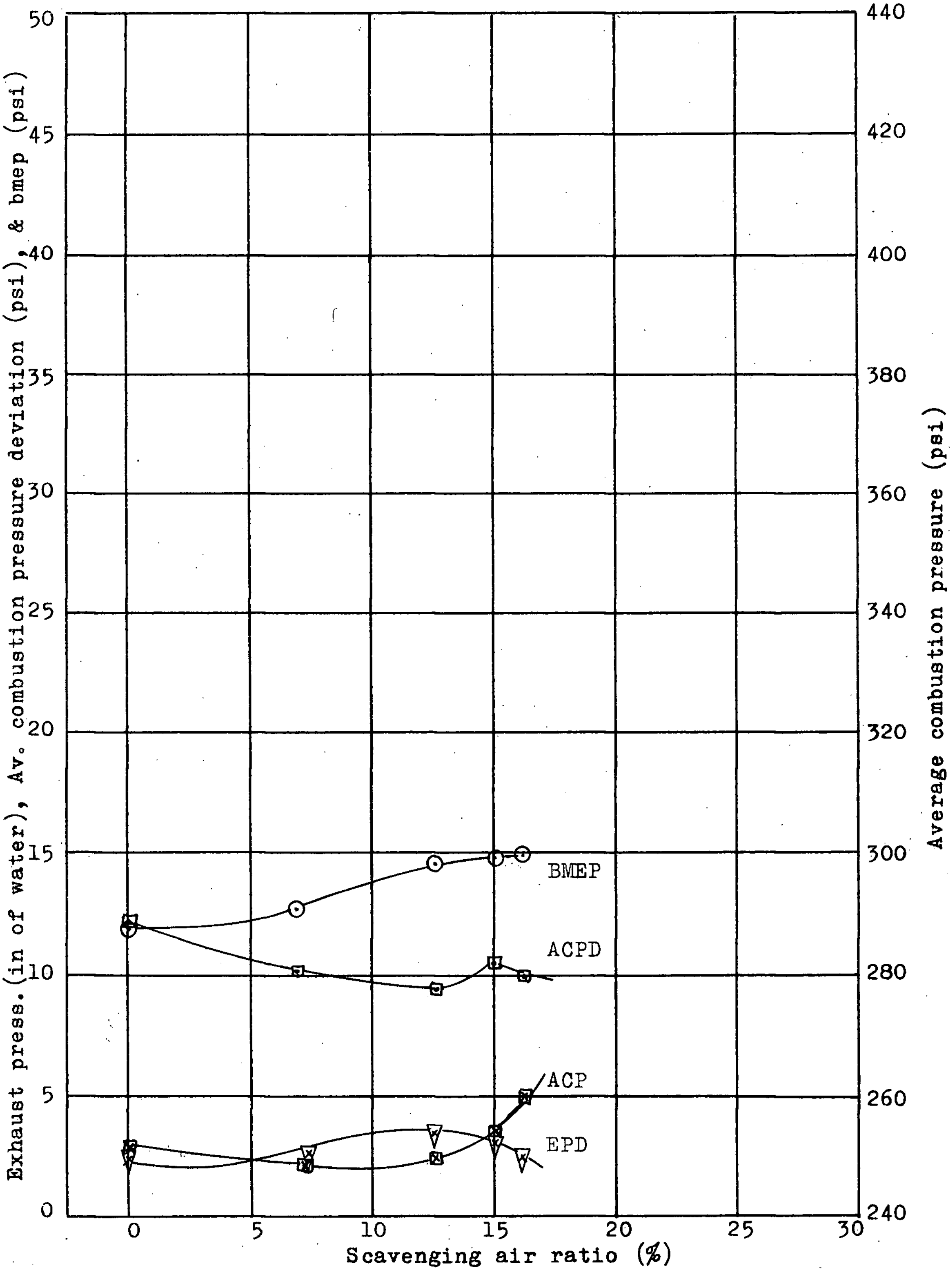

Figure 38. Pressure curves with TP 60%, Jet FA, NS 5000, from sheet 21.

(5) Figures 39-43 with TP 85%, Jet 17/16, NS 5300 from sheet 19

The jet was set at 1 1/16 turns open at the start of this test to give the maximum power with the throttle 85% open and a nominal speed of 5300 rpm. When the scavenging air valve was opened the fresh charge backed out through the valve and meter instead of air being drawn in. The result is a decrease in the specific fuel consumption even though approximately 2% of the fuel vapour escaped through the valve unburned; the vapour could be seen issuing from the meter. The reversing of the air flow could have been the result of poor reed valve sealing with the pressure difference between the inside of the valve and the outside of the valve not as great as when the throttle is closed further which stopped the leakage and resulted in normal entrance of the scavenging air. The force to keep the valve on its seat against the vibrating forces would not be sufficient to seal the valve, thus allowing the charge to leak out on pre-compression.

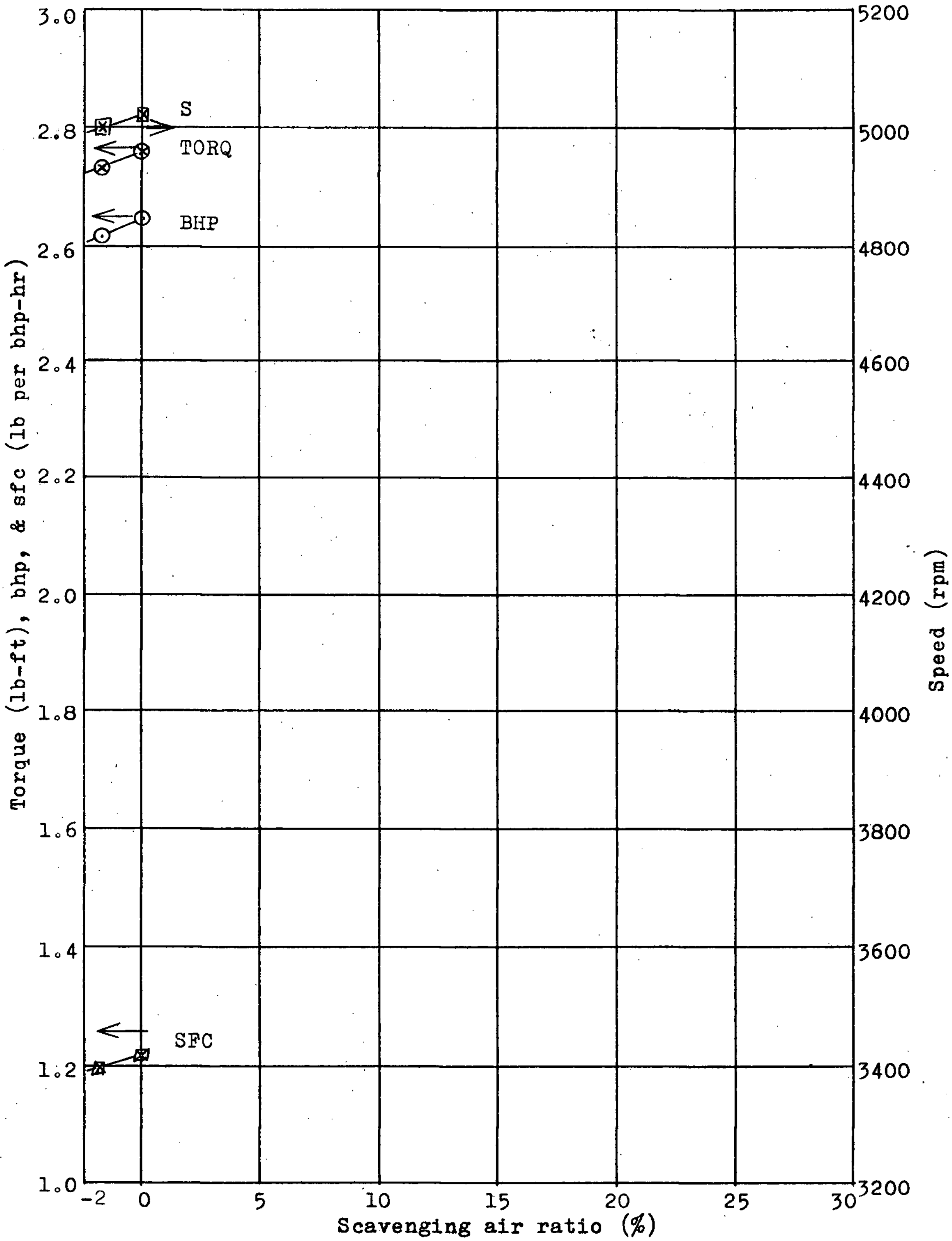

Figure 39. Performance curves with TP 85%, Jet 17/16, NS 5300, from sheet 19.

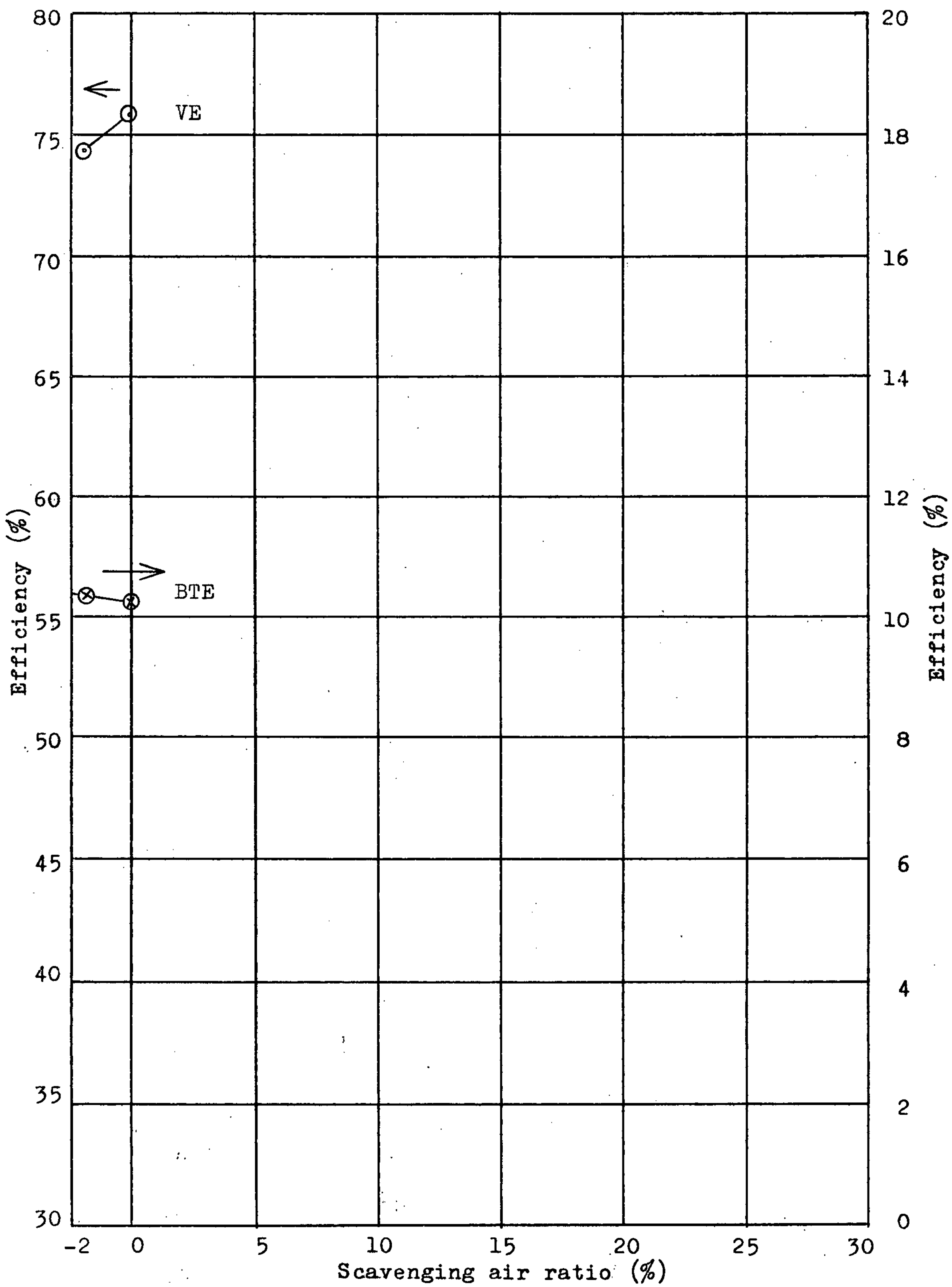

Figure 40. Efficiency curves with TP 85%, Jet 17/16, NS 5300, from sheet 19.

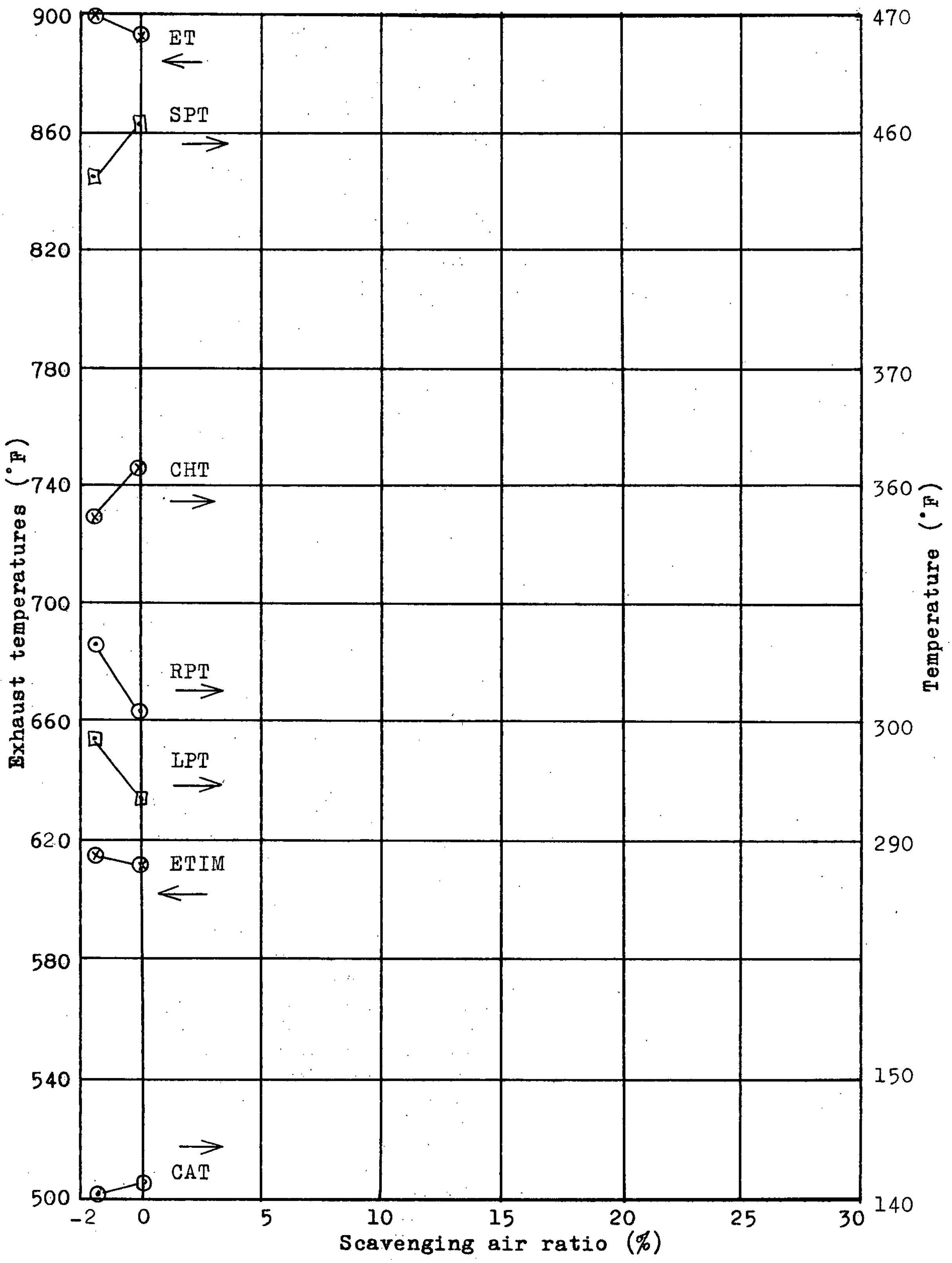

Figure 41. Temperature curves with TP 85%, Jet 17/16, NS 5300, from sheet 19.

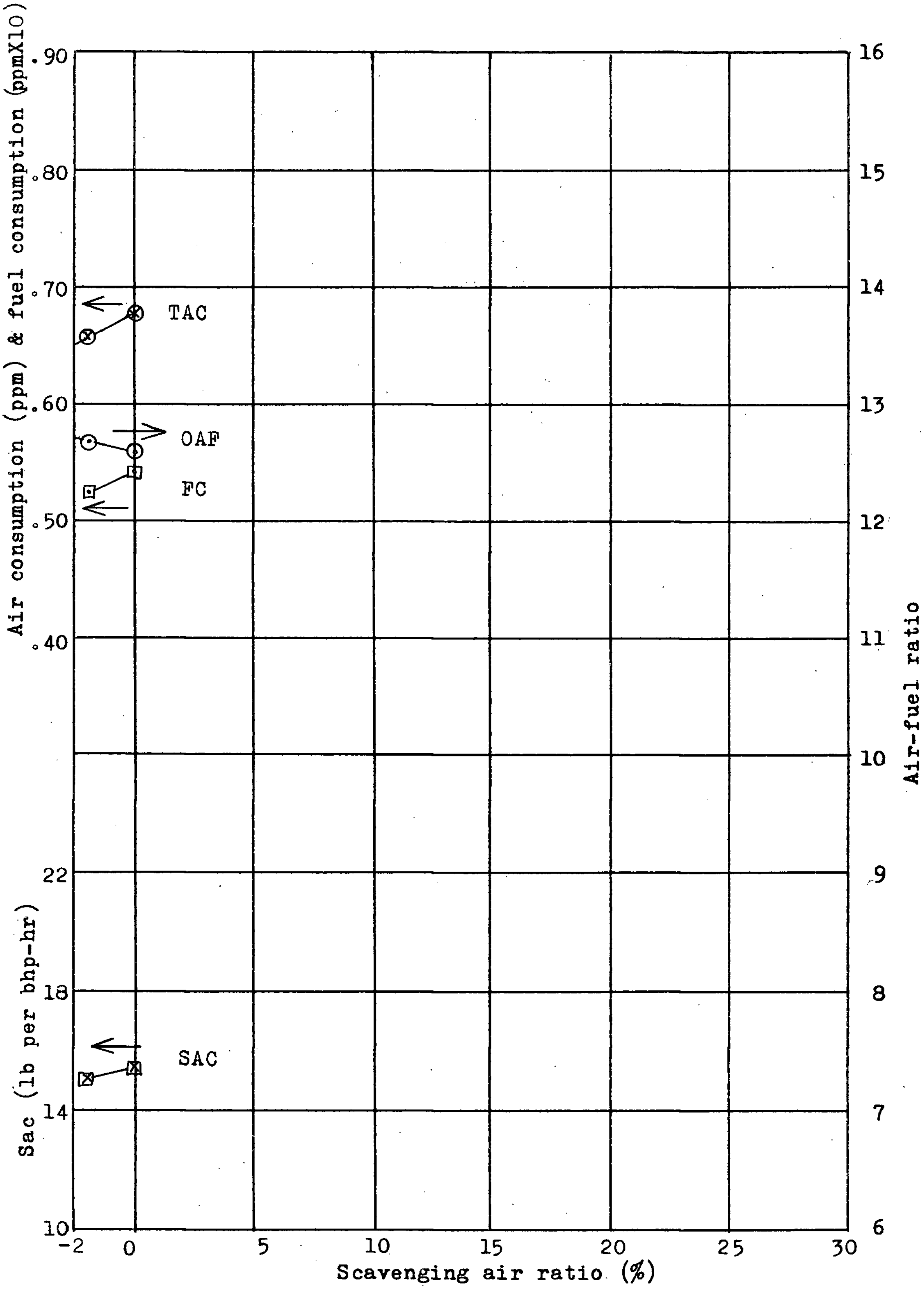

Figure 42. Air & fuel consumption curves with TP 85%, Jet 17/16, NS 5300, from sheet 19.

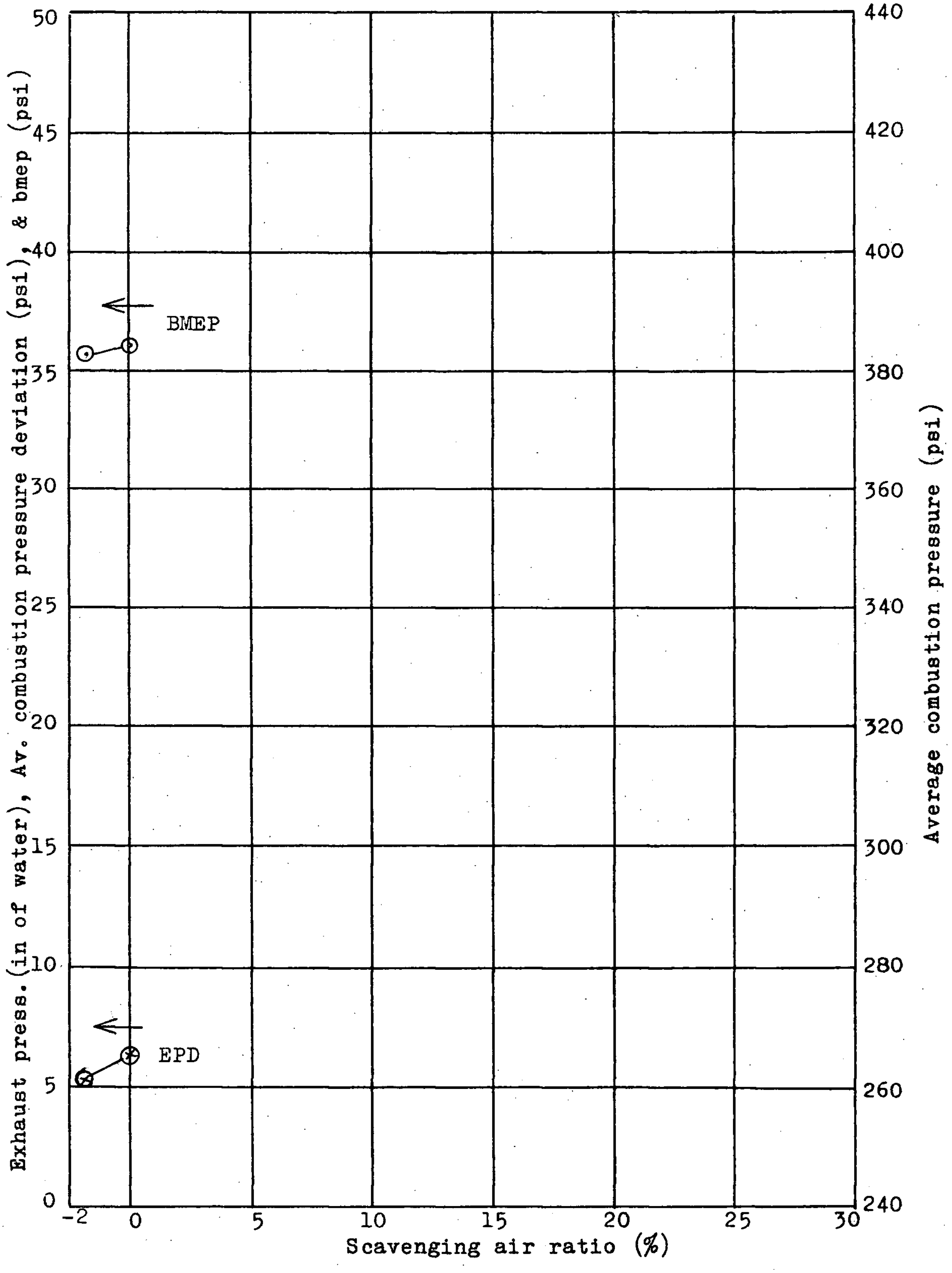

Figure 43. Pressure curves with TP 85%, Jet 17/16, NS 5300, from sheet 19.

Figures 44-48, with TP 100%, Jet 14/16, NS 2700, from sheet 14

Fuel measurements for these tests were taken over two minute intervals with the graduated pipette. As the air flow through carburetor was small, Reynolds number for the nozzle was below the lower limit of the graphs from which the nozzle flow coefficients were read. It was necessary to extrapolate the curve for these coefficients.

At the full open position of the valve the power and specific fuel consumption was lower than it was at the closed valve position, figure 44. The low specific fuel consumption was probably due to the higher overall air-fuel ratio and the higher fuel trapping efficiency, figure 45 and 47. Even though the overall air-fuel ratio was above the stoichiometric ratio the actual air-fuel ratio in the engine cylinder was lower as some air, but little fuel, escaped out the exhaust ports.

The reduced restrictions to air flow and the decrease in speed resulted in the specific air consumption increasing even though the total air consumption remained approximately constant. The spark plug and cooling air temperatures dropped indicating a reduced combustion temperatures, figure 46. The constant exhaust temperature indicates that the energy lost in the exhaust remained approximately constant.

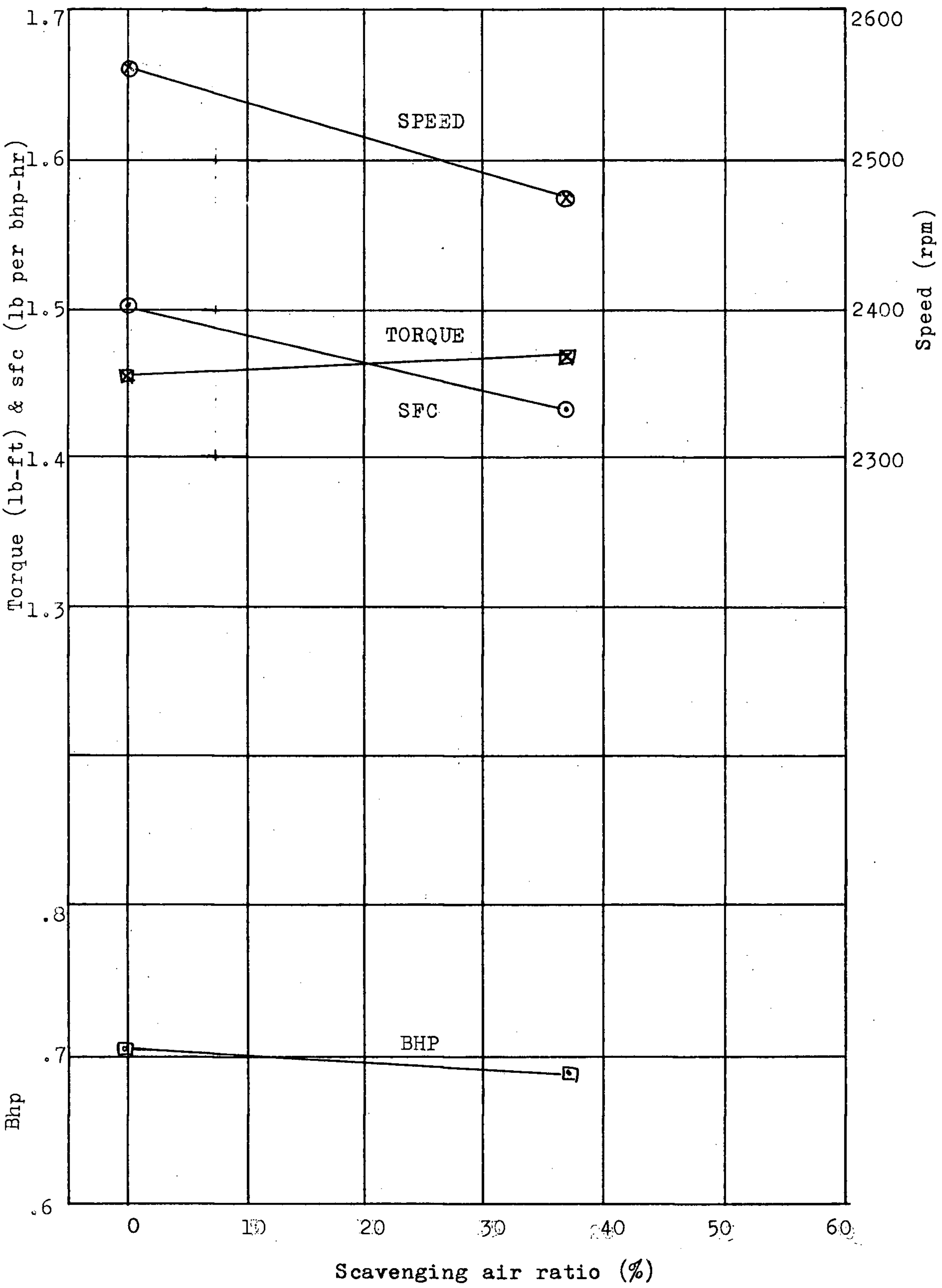

Figure 44. Performance curves with TP 100%, Jet 14/16, NS 2700, from sheet 14.

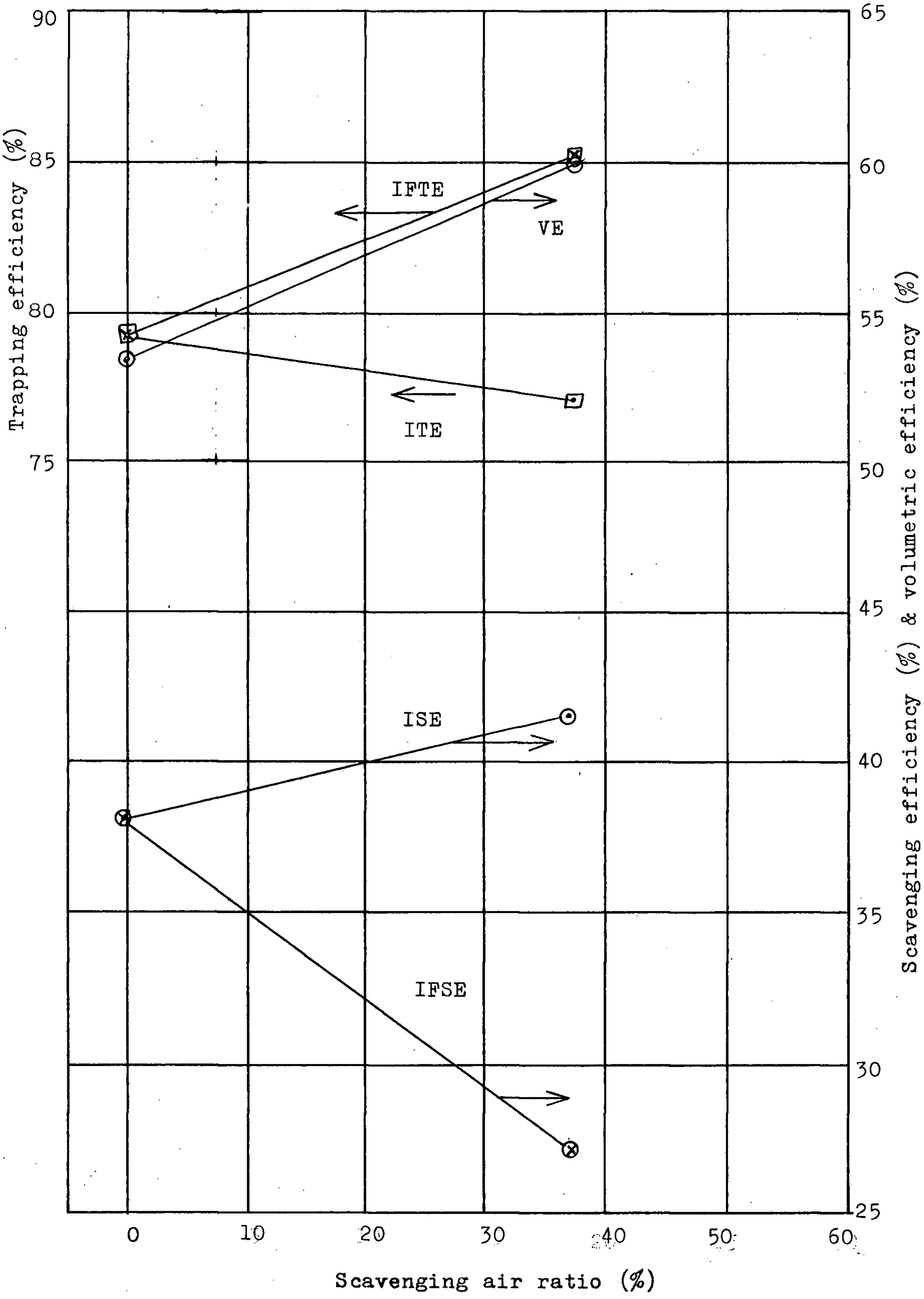

Figure 45. Efficiency curves with TP 100%, Jet 14/16, NS 4700, from sheet 14.

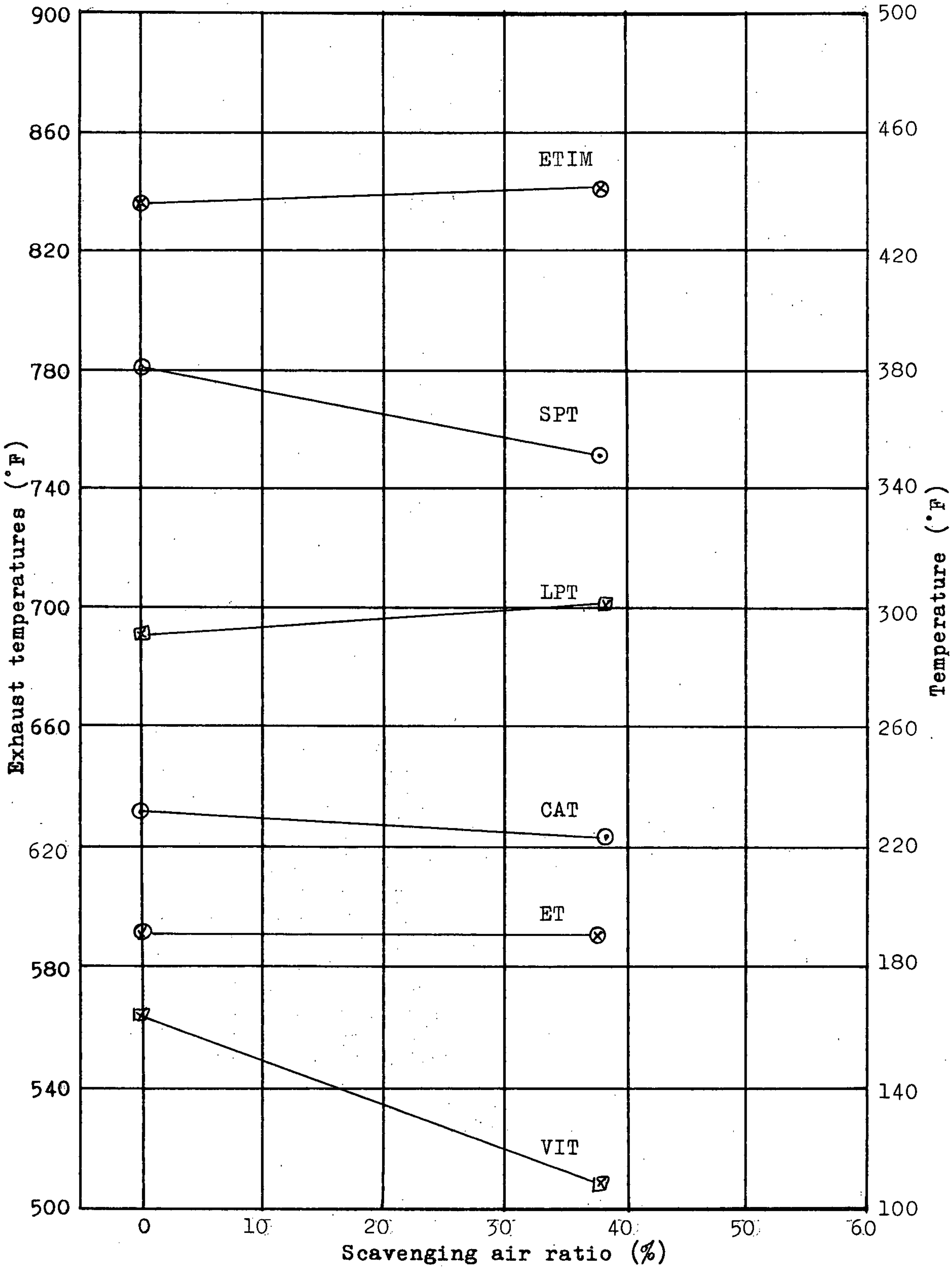

Figure 46. Temperature curves with TP 100%, Jet 14/16, NS 2700 from sheet 14.

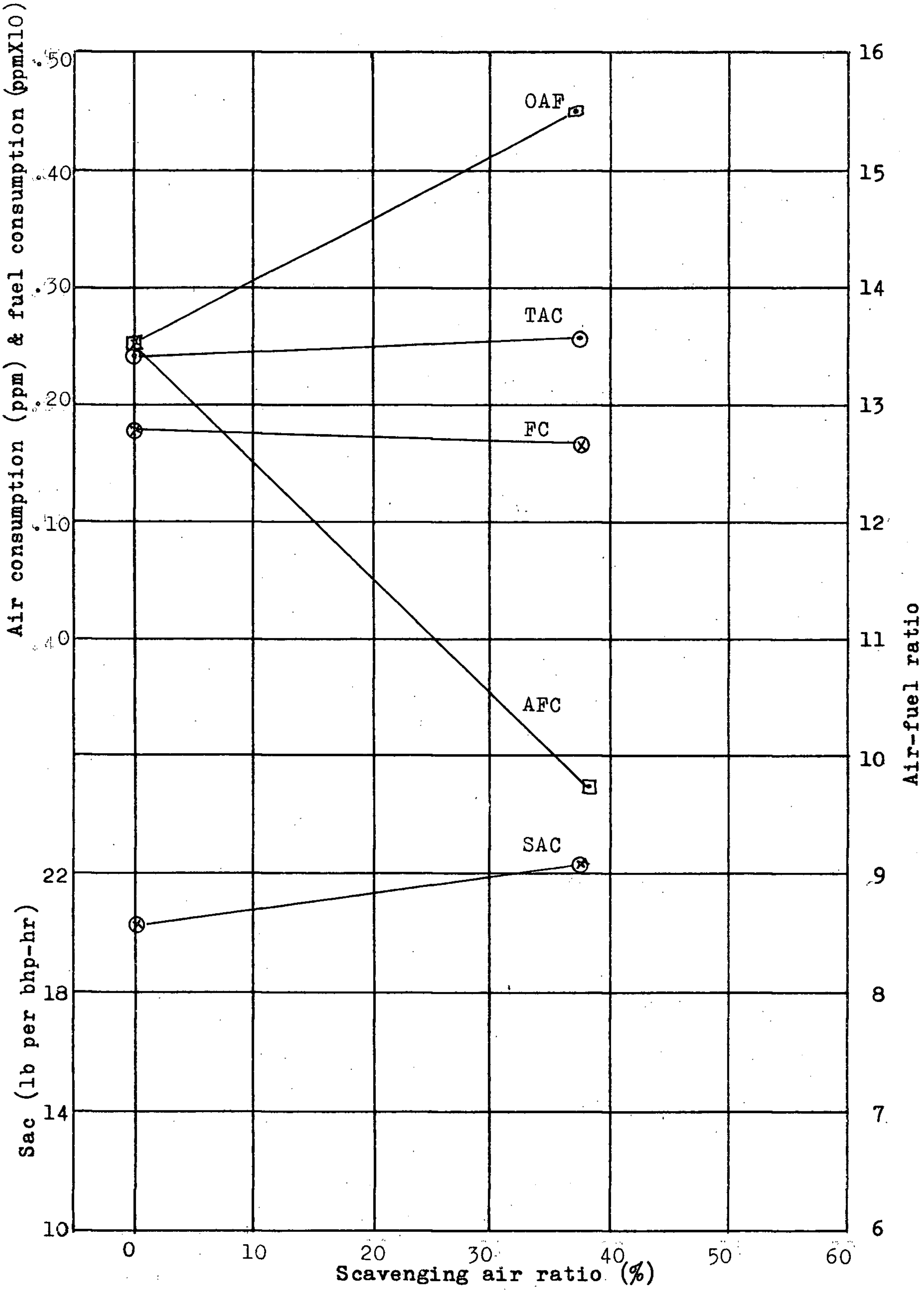

Figure 47. Air and fuel consumption curves with TP 100%, Jet 14/16, NS 2700, from sheet 14.

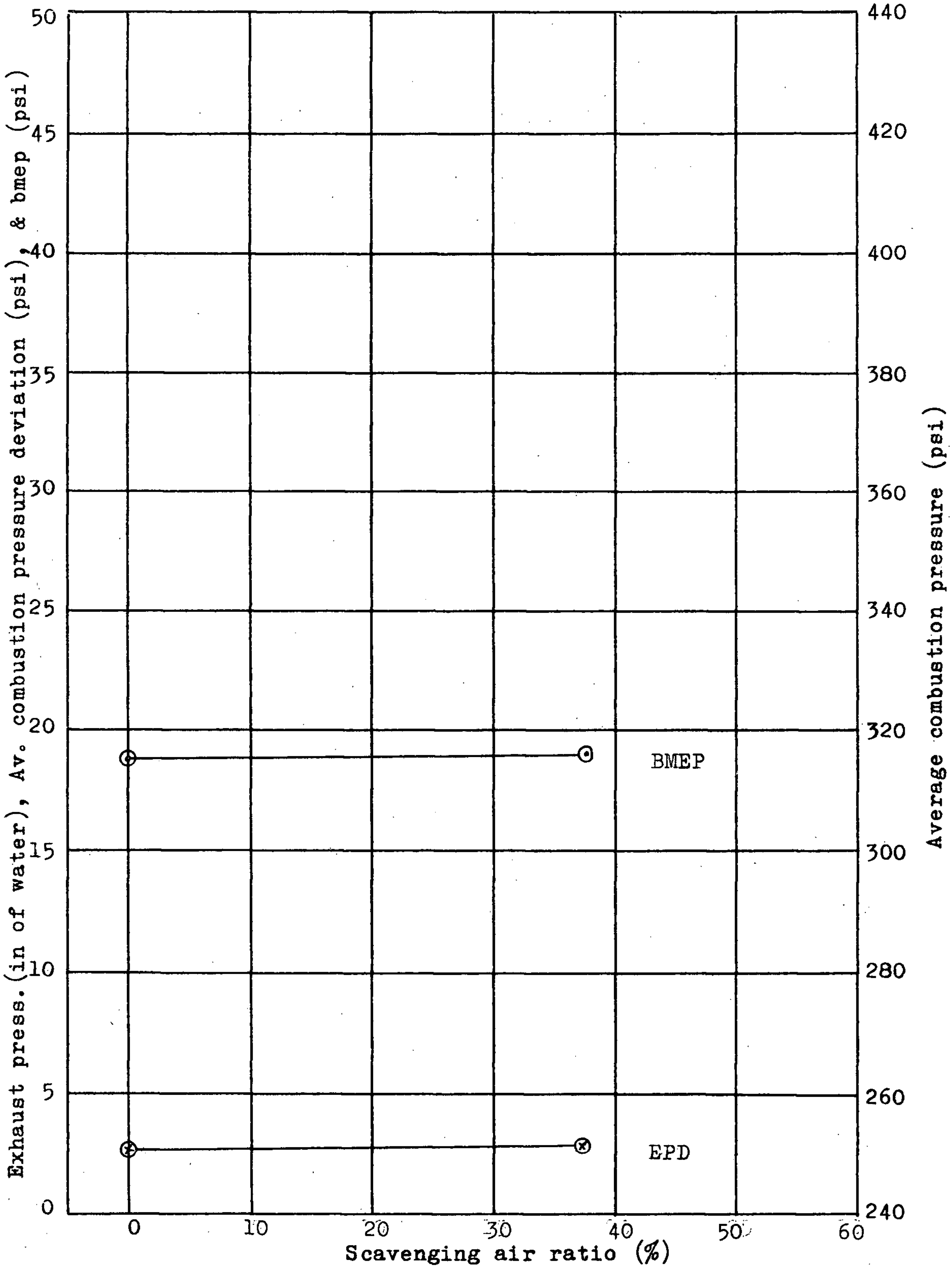

Figure 48. Pressure curves with TP 100%, Jet 14/16, NS 2700, from sheet 14.

Figure 49-54 with TP 70%, Jet F.A., NS 4300, from sheet 22.

Tests were conducted with the above settings, first with the scavenging air valve closed, then 55% open, and finally with it wide open; the jet adjustment on the new carburetor remained as it had been received from the factory. During the second test, the engine speed suddenly dropped approximately 500 rpm and then picked up again; this fluctuation occured 4 times during the test. The speed dropped approximately 1500 rpm twice just after the valve was opened 100%, but remained steady during the test.

From figure 49 it appears that the power dropped slightly for low scavenging air ratios and then increased for higher ratios while the fuel consumption dropped more rapidly at first and then increased again, figure 53. Thus the specific fuel consumption decreased rapidly and then increased again as the scavenging air ratio increased, figure 49. The fuel trapping efficiency, figure 50, increased continuously so that less fuel was wasted. As the charge trapping efficiency decreased, more of the air was lost in the exhaust which cooled the exhaust. Figure 52 shows that the exhaust temperature remained nearly constant even though the overall combustion temperature had increased as indicated by the spark plug and cylinder head temperatures. The valve inlet air temperature affected the right passageway temperature.

With less restriction to air flow, the total air consumption and volumetric efficiency increased although the first increase in the weight of air flowing through valve was compensated for by the corresponding reduction in the weight of air flowing through carburetor, figure 53.

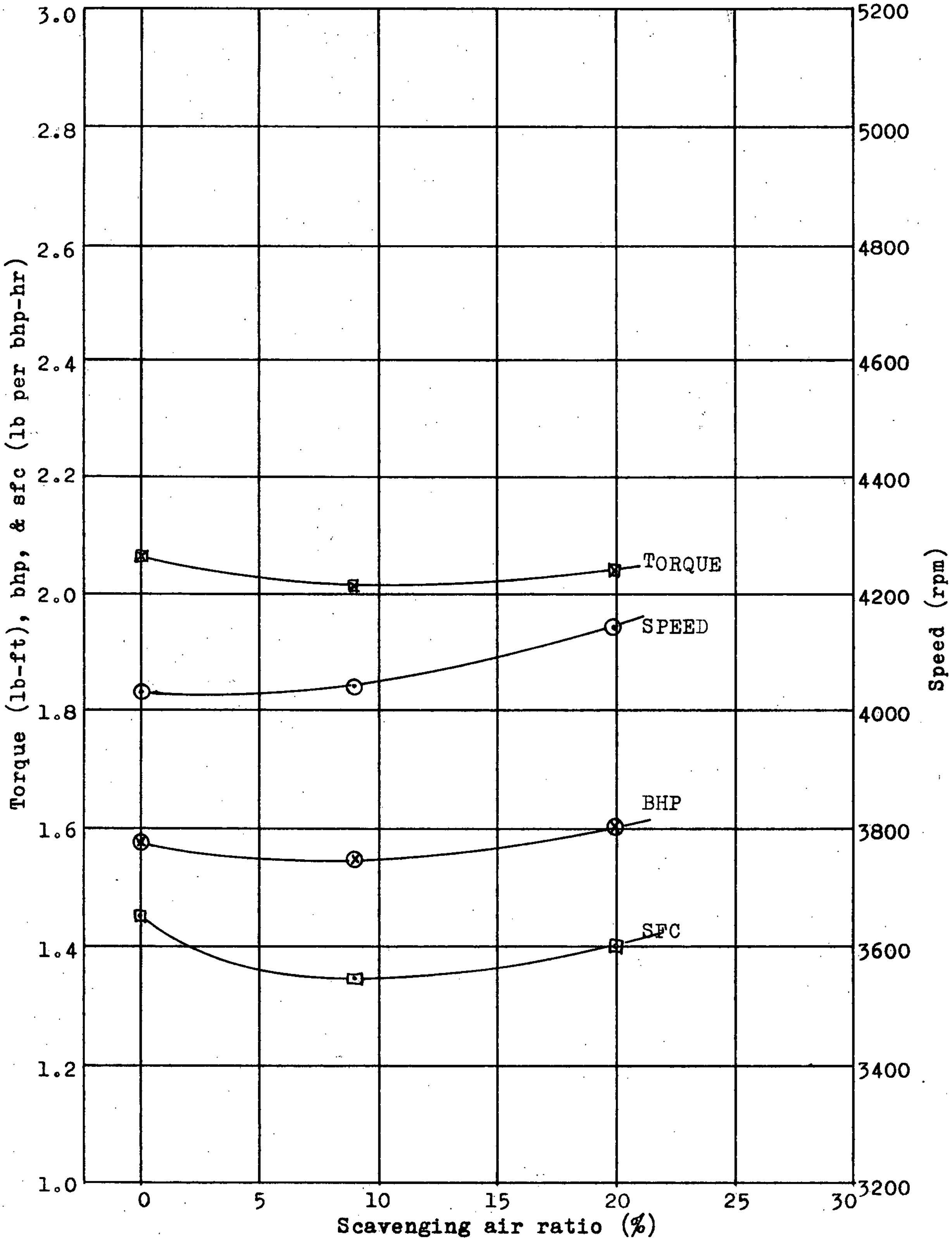

Figure 49. Performance curves with TP 70%, Jet FA, NS 4300, from sheet 22.

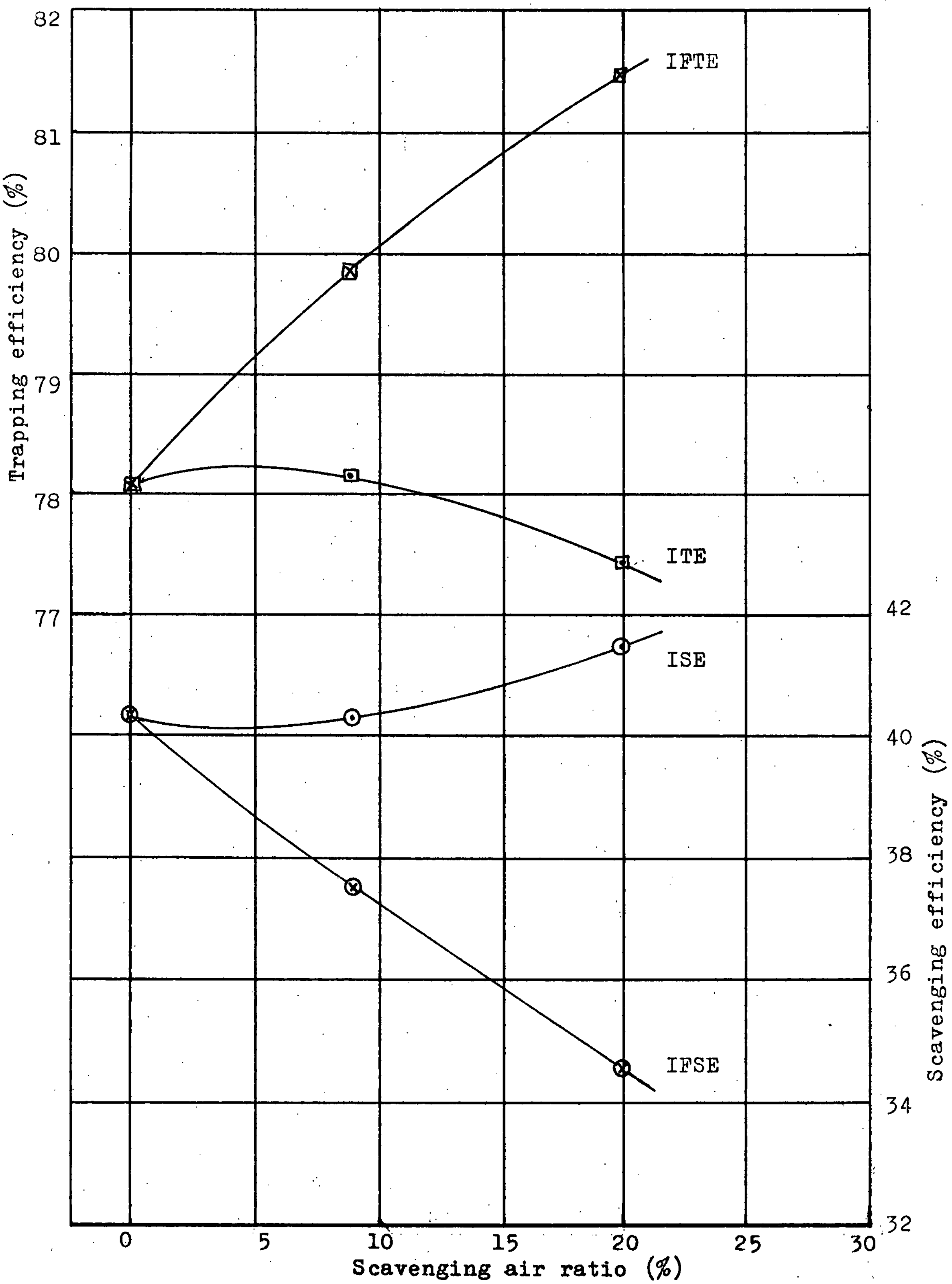

Figure 50. Efficiency curves with TP 70%, Jet FA, NS 4300, from sheet 22.

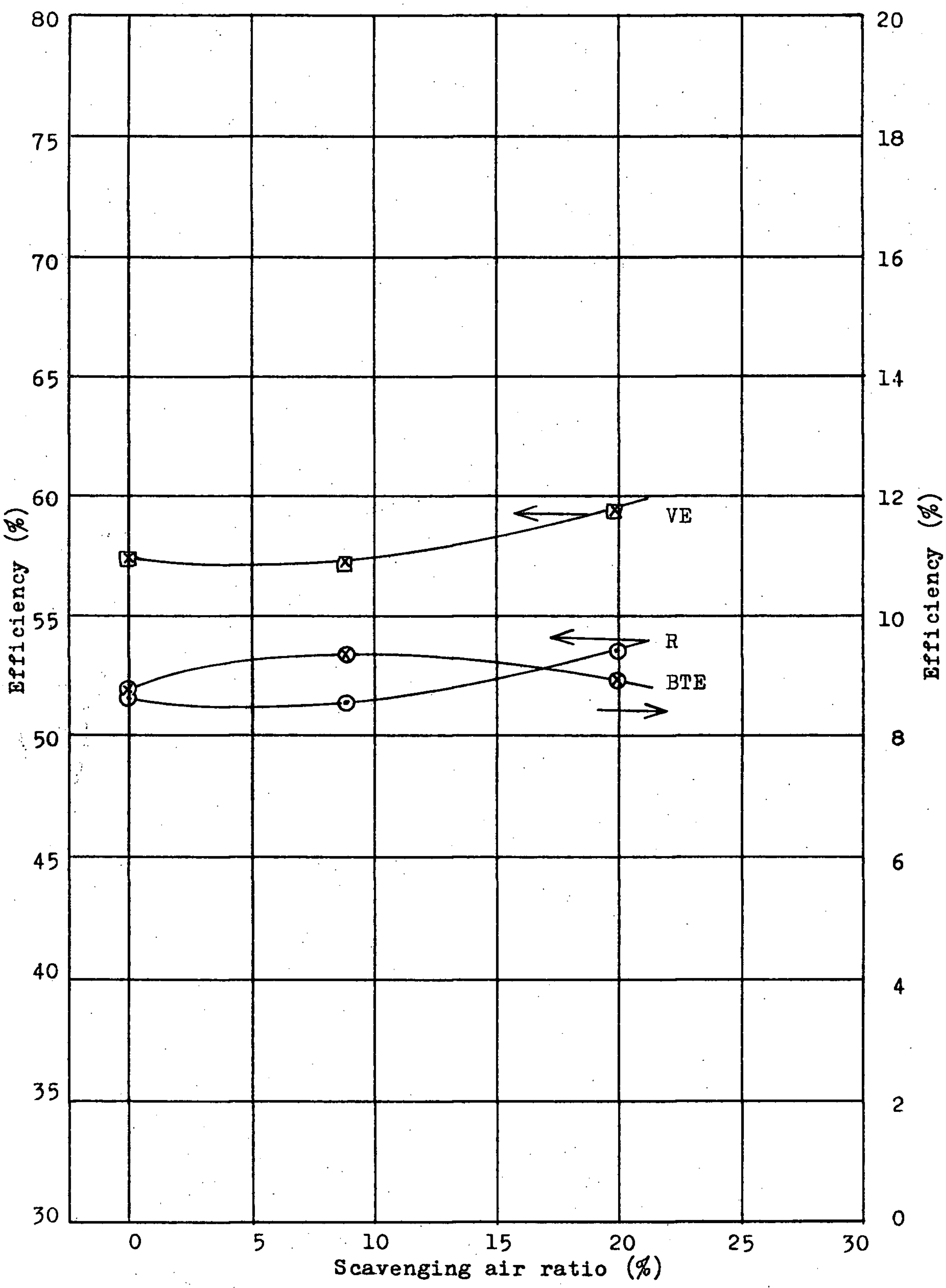

Figure 51. Efficiency curves with TP 70%, Jet FA, NS 4300, from sheet 22.

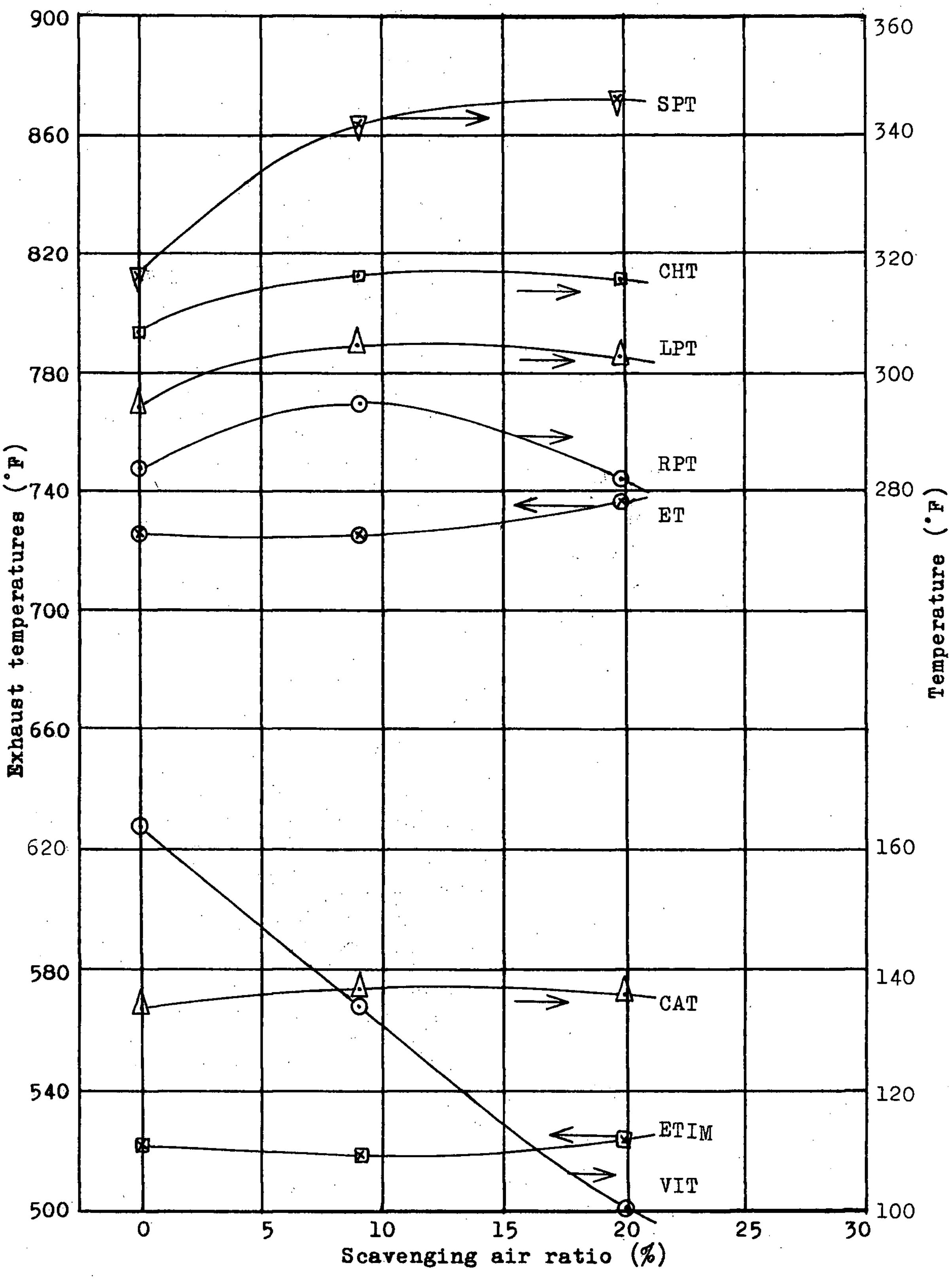

Figure 52. Temperature curves with TP 70%, Jet FA, NS 4300, from sheet 22.

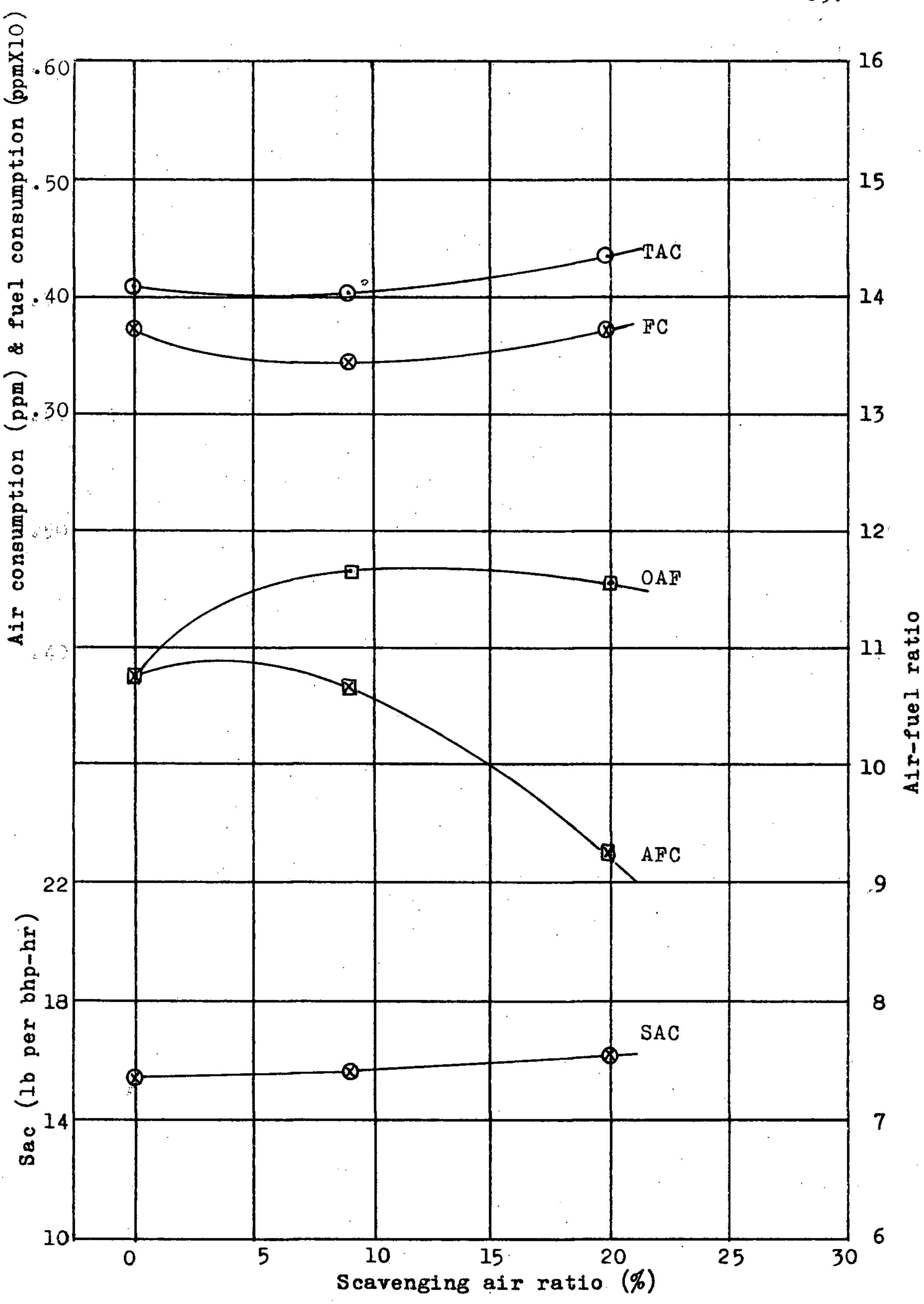

Figure 53. Air and fuel consumption curves with TP 70%, Jet FA, NS 4300, from sheet 22.

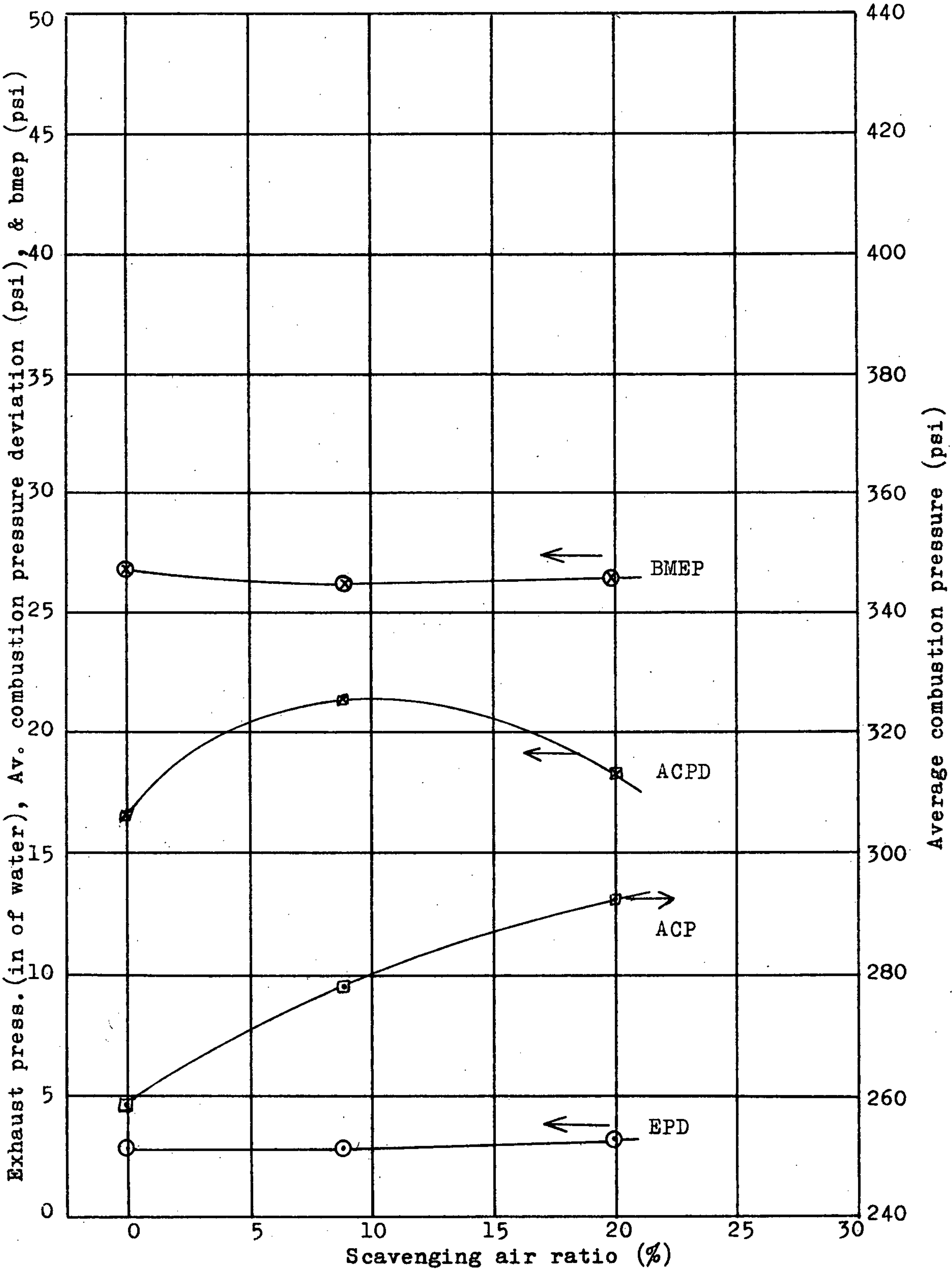

Figure 54. Pressure curves with TP 70%, Jet FA, NS 4300, from sheet 22.

Figure 55-62 with TP 30%, Jet F.A., from sheet 23

With the scavenging air valve wide open resulting in a scavenging air ratio of 54%, the engine would run smoothly for 30-40 seconds and then suddenly drop down from 4900 rpm to 3000 rpm, gradually pick up speed, and then remain constant for the 30-40 seconds before repeating the cycle. The engine seemed to be running out of fuel. Thus the tests at the valve setting were limited in length to 30 seconds. Several readings were taken when the engine was running steady and then the readings were averaged. For the next test, the only adjustment made was to choke the engine so that the speed would not drop more than several hundred rpm; this adjustment increased the scavenging air ratio to 56%. During the last test with the scavenging air ratio 48.4%, speed fluctuated between 4900 rpm and 4600 rpm and the temperature fluctuated. During the test, the engine slowed down by 1000 rpm once. The engine exhibited severe vibrations between 3800-4000 rpm so that no tests were run with speeds in this range.

Figure 61 shows that the fuel consumption remained approximately constant as the scavenging air ratio increased but made a sudden jump when the critical vibration speed was exceeded. Nevertheless the specific fuel consumption decreased approximately linearly, figure 55. The lowest specific fuel consumption obtained was .91 lb per bhp-hr., but to maintain steady speed operation, choking was necessary which increased the specific fuel consumption to 1.06 lb per bhp-hr. Choking reduced the power and speed and the air fuel ratios, figure 60. Figure 56 shows that the ideal fuel trapping efficiency increased continually with the scavenging air ratio, while the ideal trapping efficiency remained approximately constant until the scavenging air ratio exceeded 30%. The decrease in the specific fuel consumption is the result of an increase in the fuel trapping efficiency and an increase in the air-fuel ratio made possible by stratifying the charge.

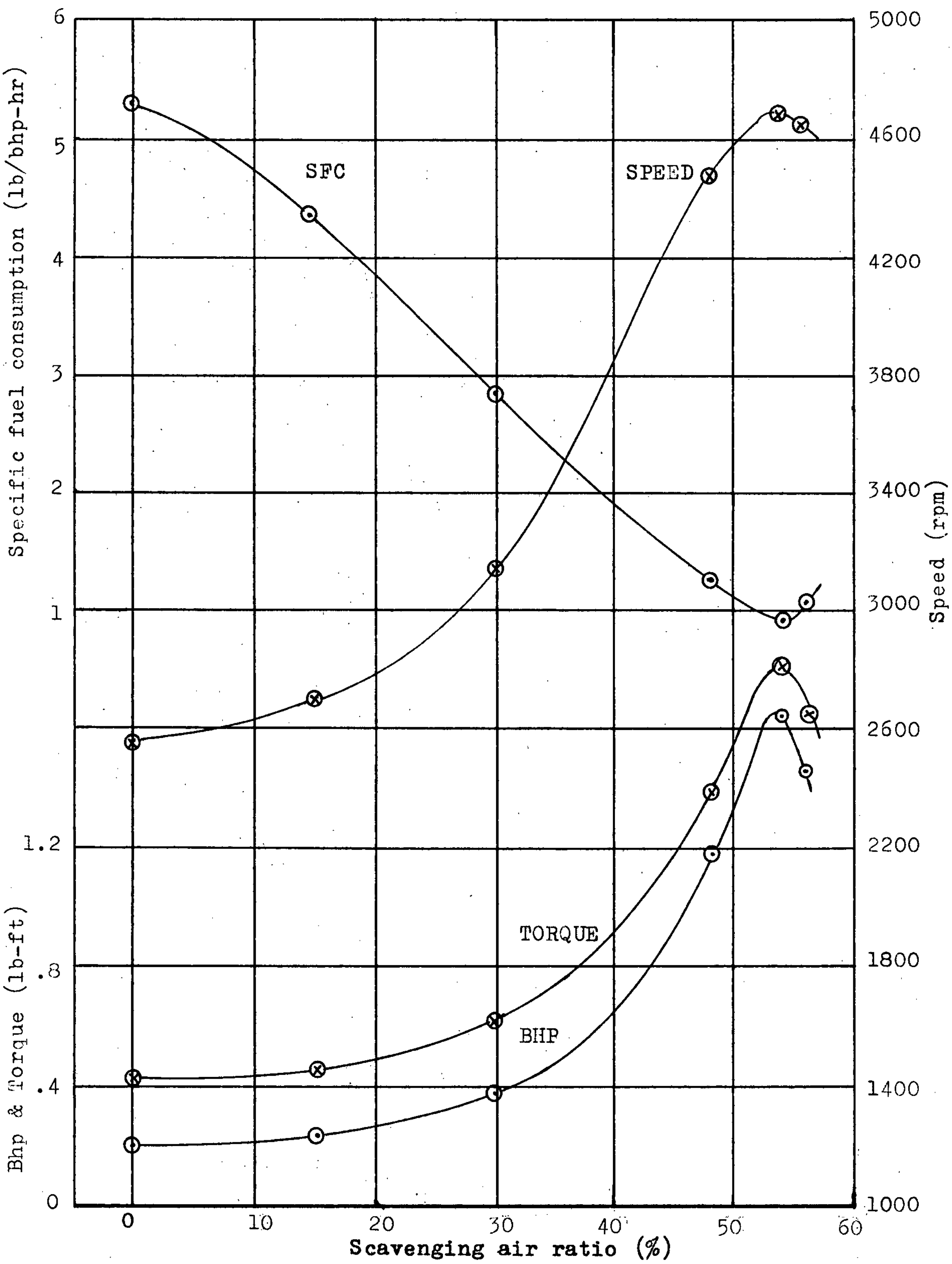

Figure 55. Performance curves with TP 30%, Jet FA, from sheet 23.

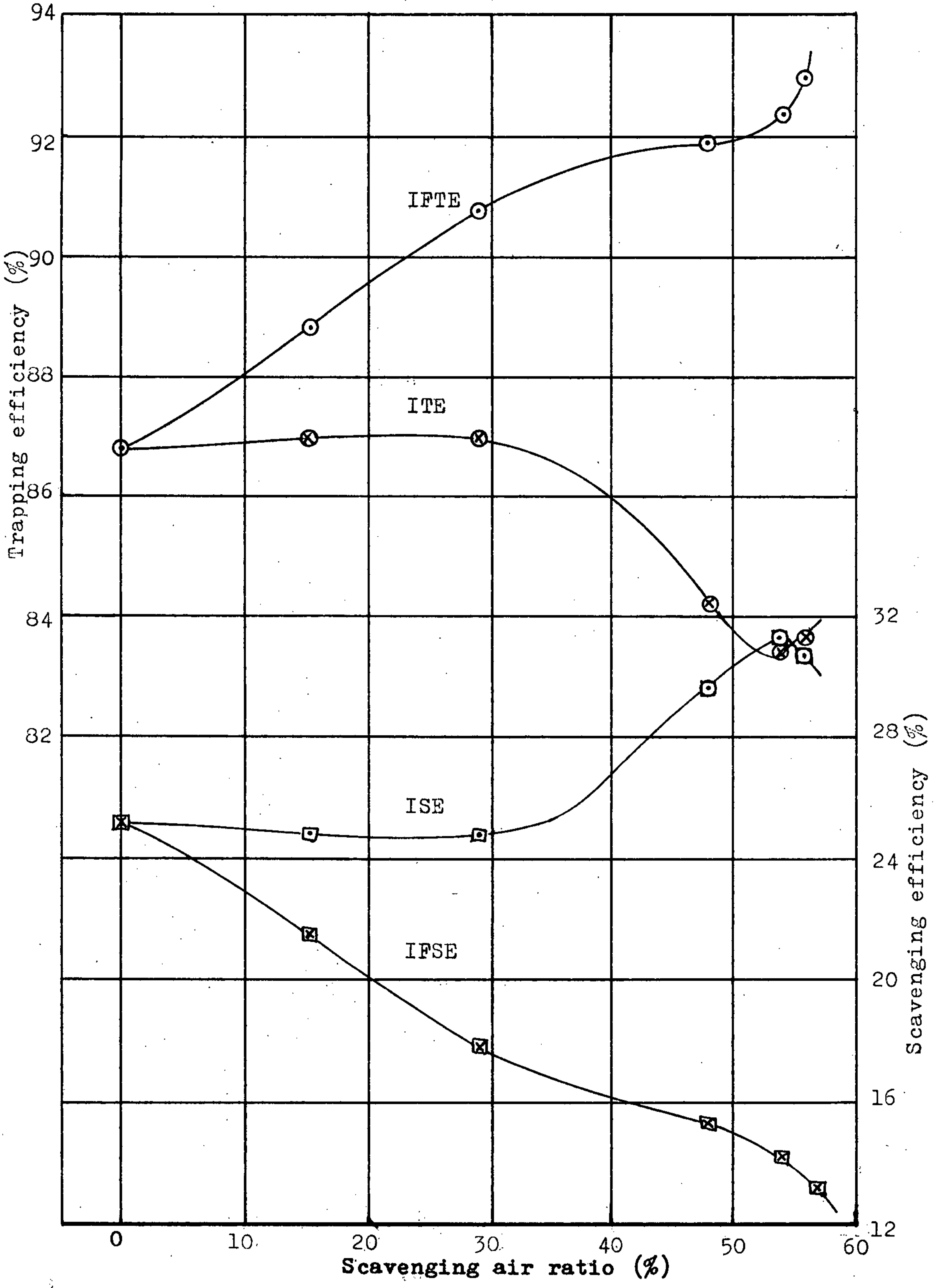

Figure 56. Efficiency curves with TP 30%, Jet FA, from sheet 23.

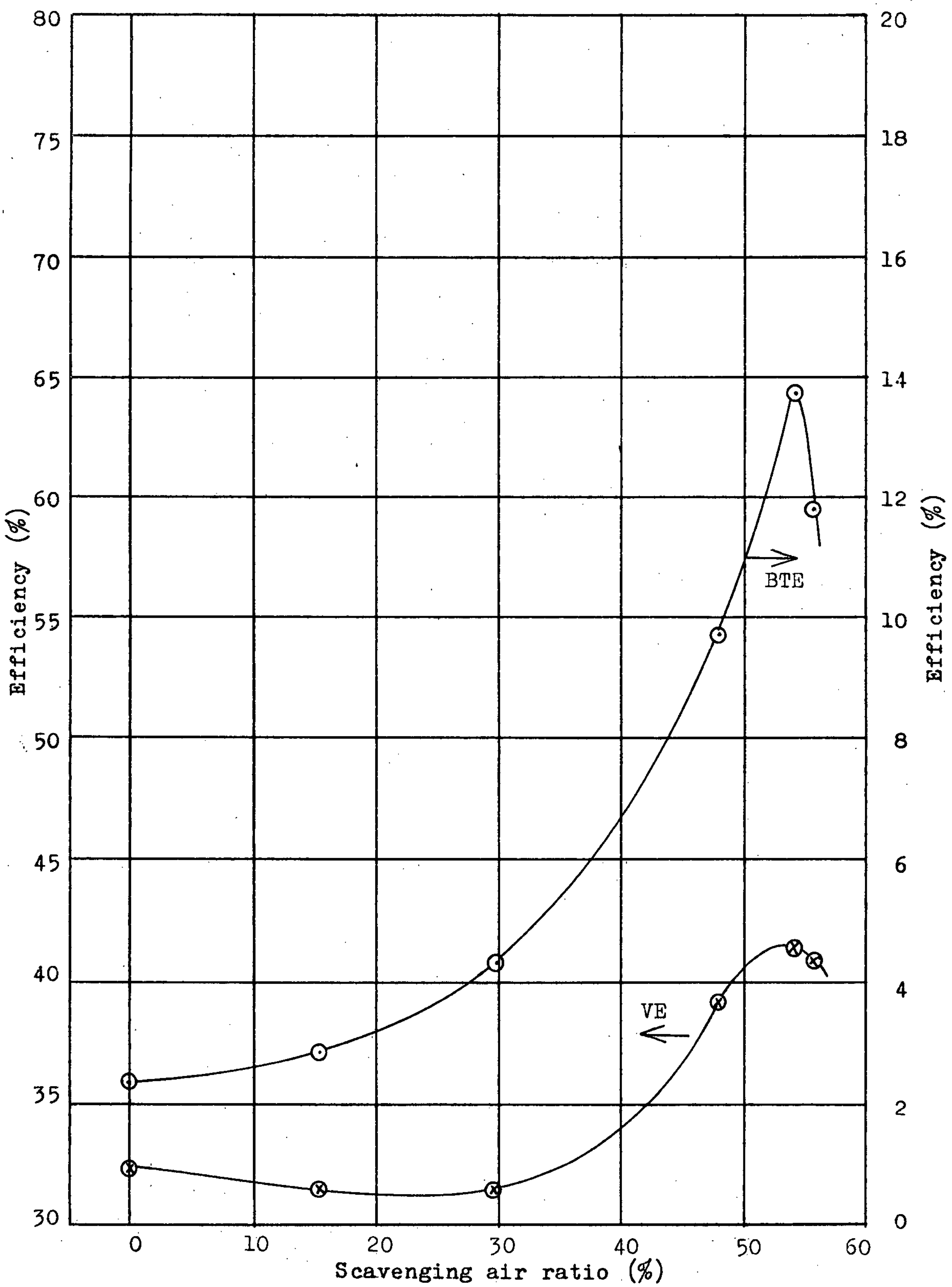

Figure 57. Efficiency curves with TP 30%, Jet FA, from sheet 23.

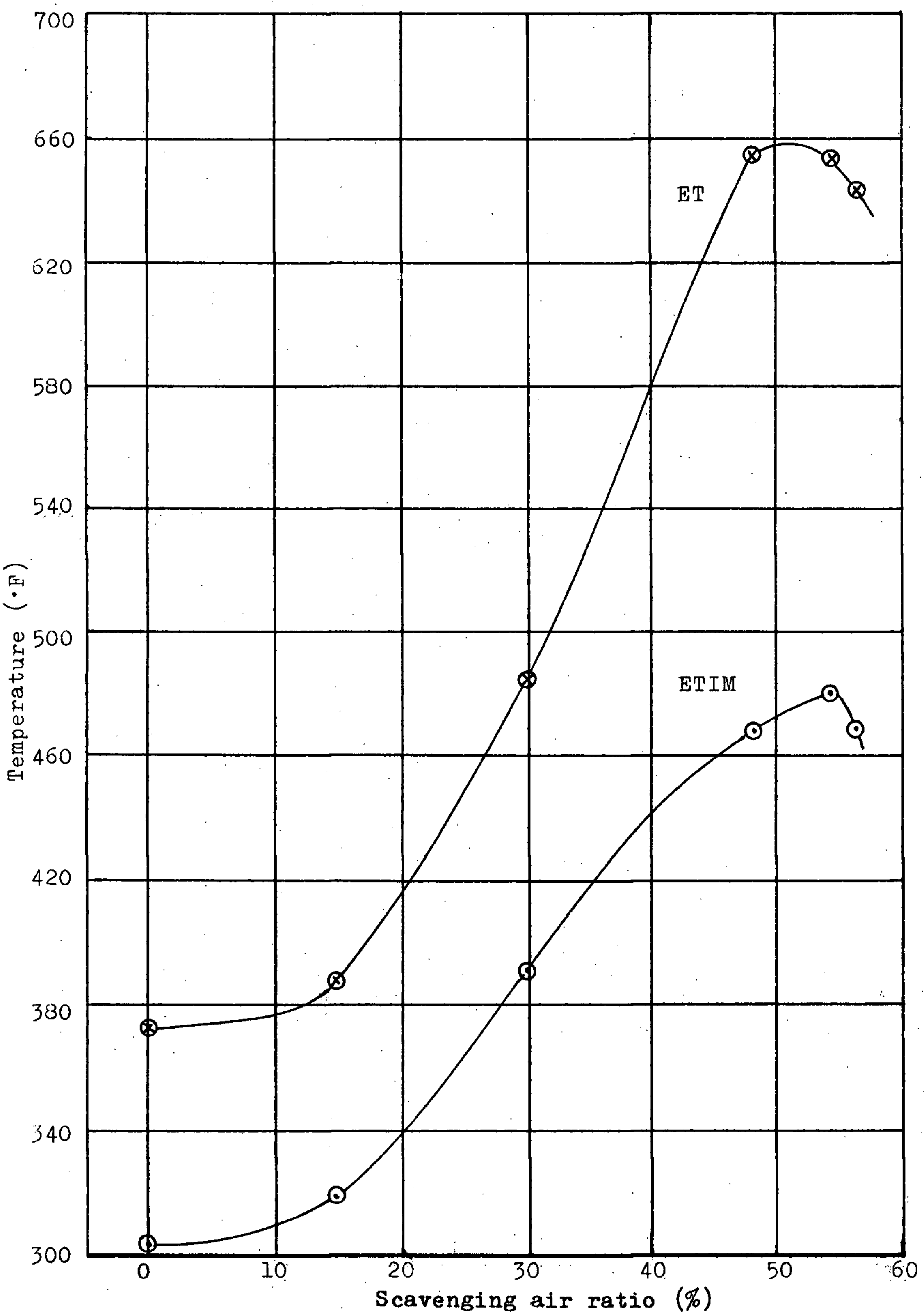

Figure 58. Exhaust temperature curves with TP 30%, Jet FA, from sheet 23.

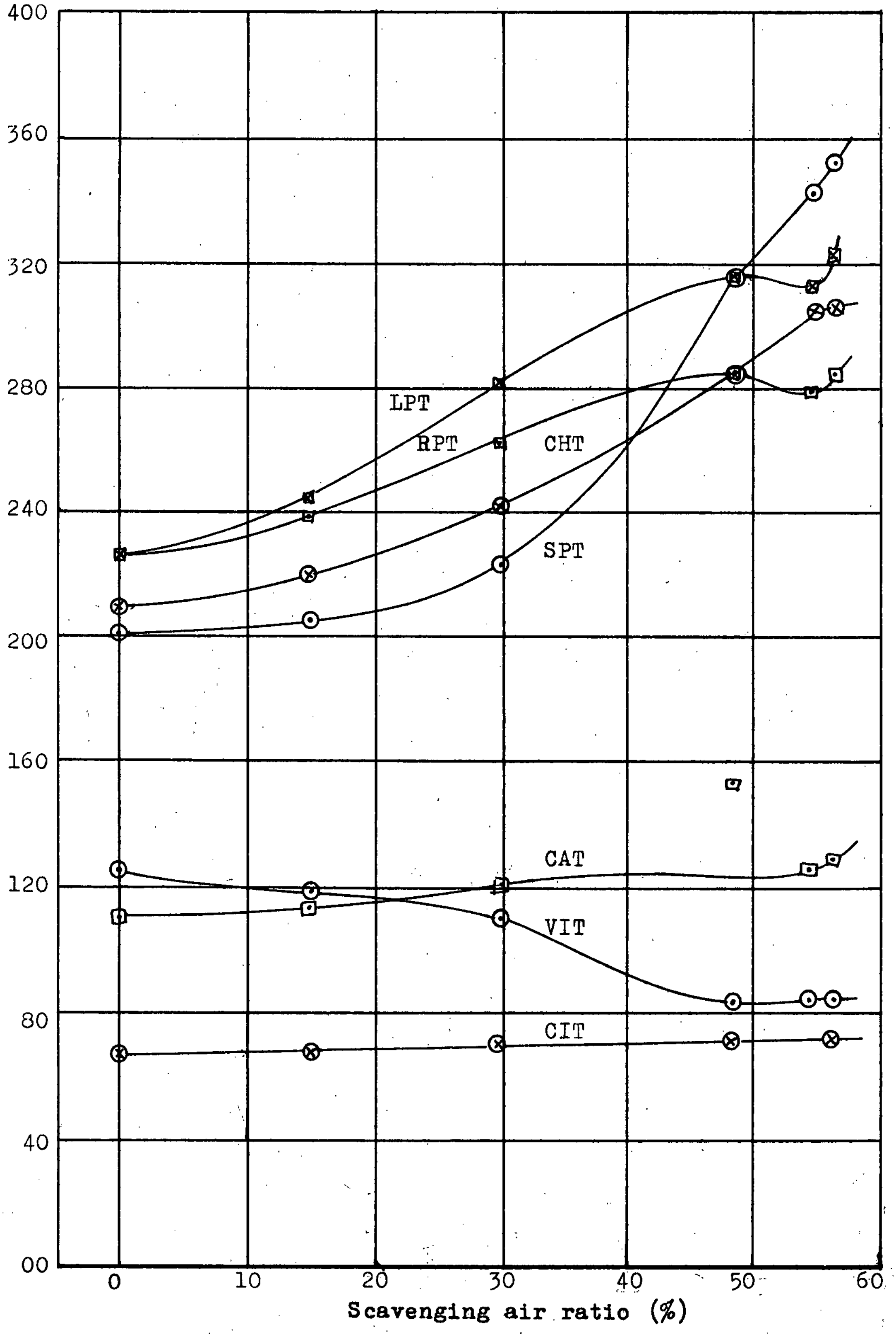

Figure 59. Temperature curves with TP 30%, Jet FA, from sheet 23.

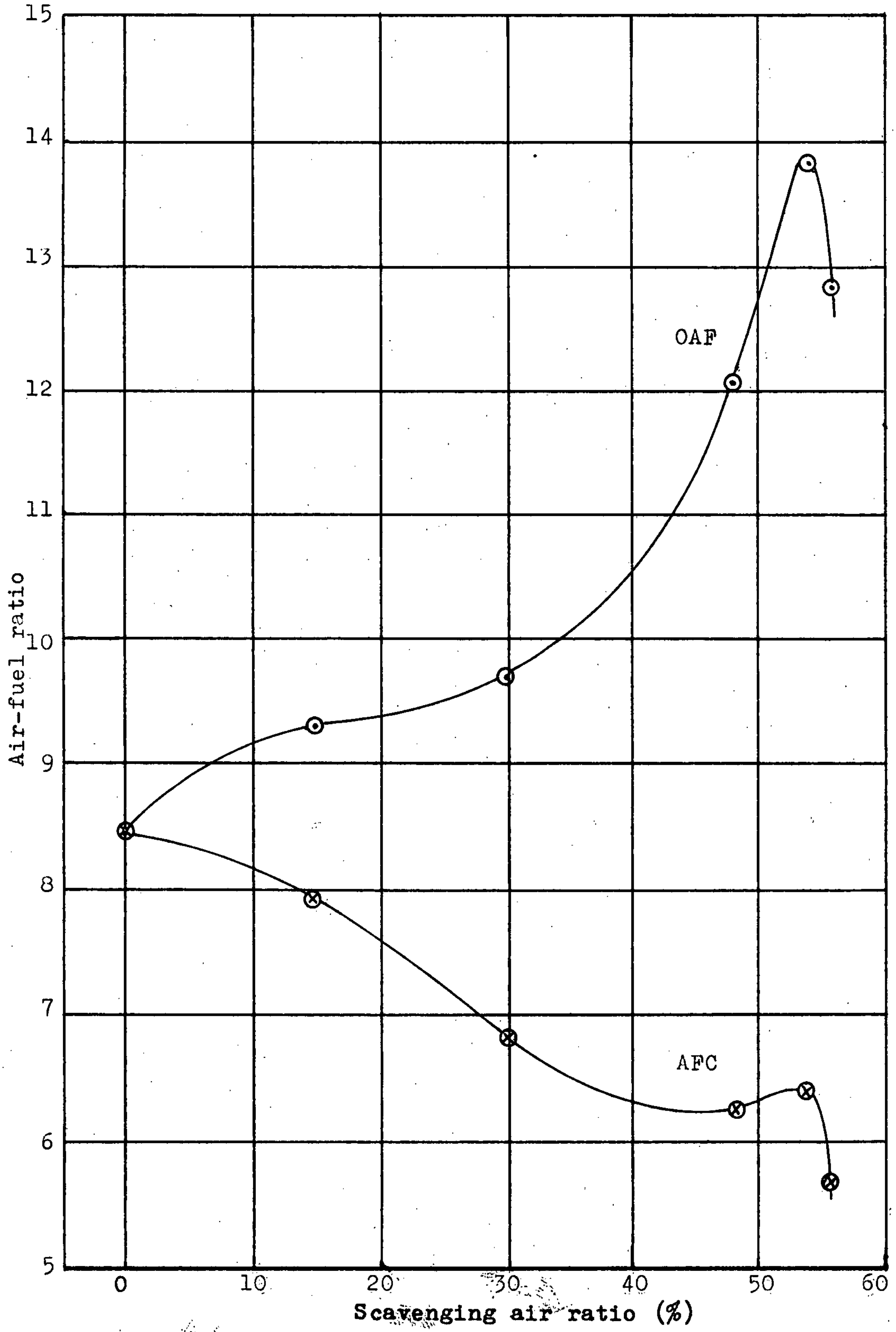

Figure 60. Air-fuel ratio curves with TP 30%, Jet FA, from sheet 23.

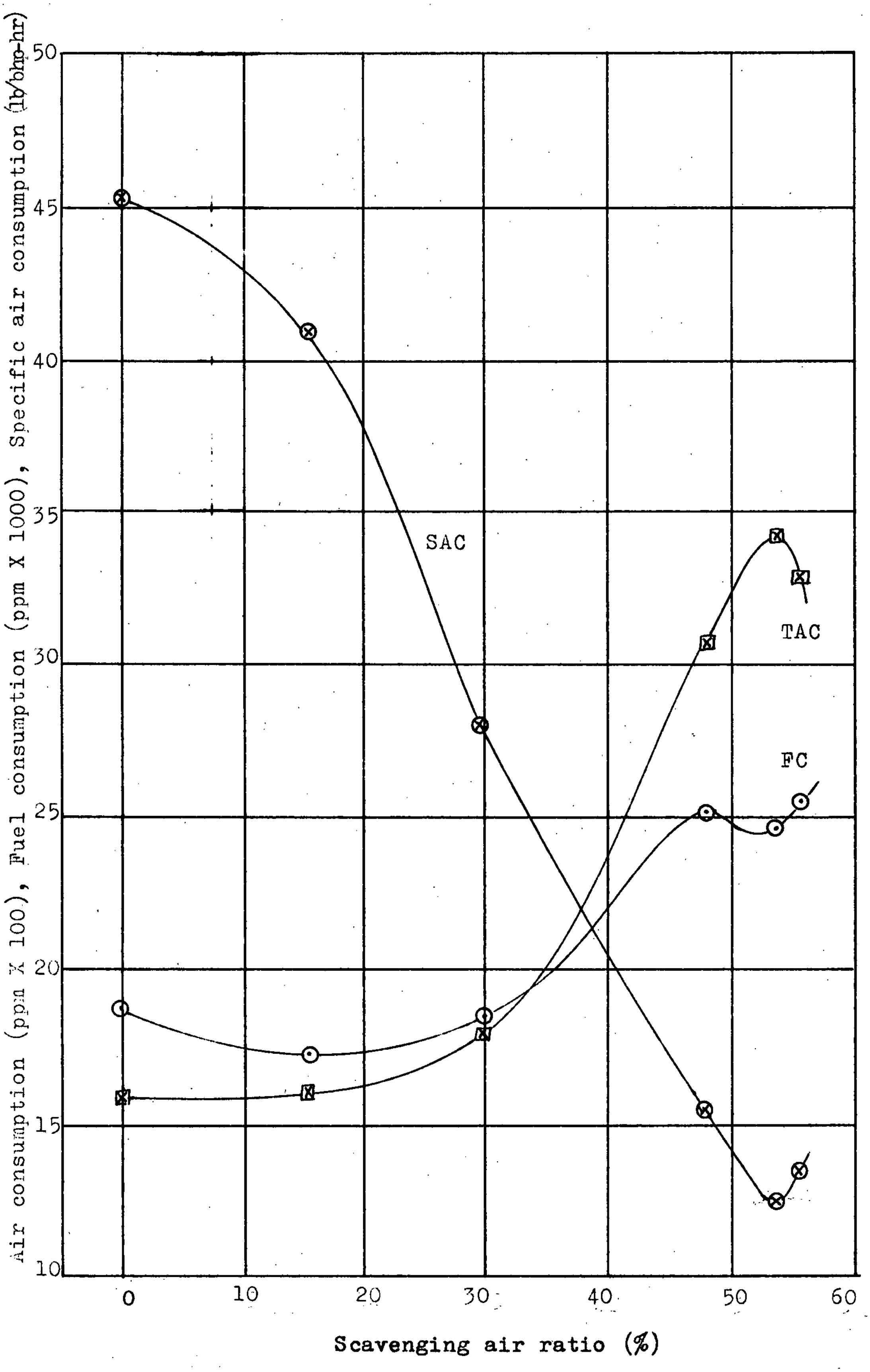

Figure 61. Air and fuel consumption curves with TP 30%, Jet FA, from sheet 23.

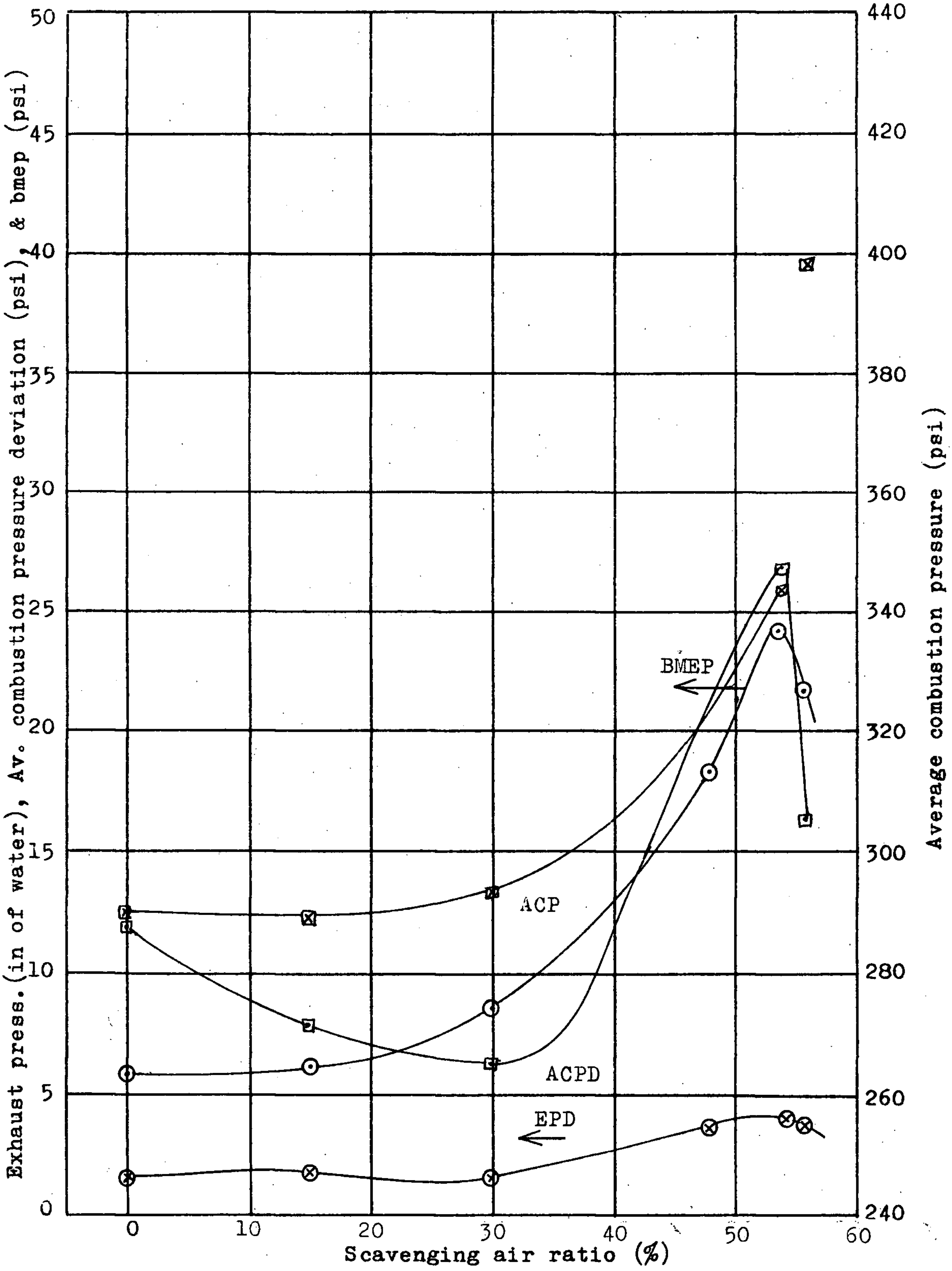

Figure 62. Pressure curves with TP 30%, Jet FA, from sheet 23.

Figures 63-70 with VPI00%, Jet F.A., from sheet 24

The purpose of this series of tests was to determine the performance of the engine at a constant scavenging air valve position with varying throttle positions which would change the scavenging air ratio. The first test was with the valve closed and the rest of the tests were with the valve wide open. The choke was adjusted as required to give steady speed operation. The inaccurate choke adjustment and fluctuations in air-fuel ratio in carburetor will account for the variation in the air-fuel ratios and fuel consumption, figures 68 and 69.

After the tests with the scavenging air valve open were completed and the valve closed, the speed could not be brought up to the initial value so that the test was conducted at the maximum speed obtainable. The results are plotted as isolated points at a scavenging air ratio equal to zero. The other series of isolated points at a scavenging air ratio equal to 3.4 % resulted from a test with conditions identical to those which gave results at scavenging air ratio equal to 8.1% except that the choke was fully opened.

The fuel trapping efficiency and charge trapping efficiency both increased so that more air and more fuel were trapped in the cylinder. As the quantity of air consumed was reduced, the volumetric efficiency decreased. Nevertheless the specific air consumption and specific fuel consumption remained nearly constant up to a scavenging air ratio of 50%, figures 68 and 69.

The exhaust temperature dropped continuously while the cylinder head and spark plug temperatures gradually increased up to 50% scavenging air ratio, figure 66 and 67, although these latter temperatures fluctuated with the overall air-fuel ratio, figure 68.

A peak in the specific fuel consumption at a scavenging air ratio of 12% was probably due to excessive choking. If a carburetor was designed to supply the required mixture strength at all throttle settings, the specific fuel consumption fluctuations would not be as great.

The shape of the average combustion pressure curve, figure 70, is analogous to the shape of the overall air-fuel ratio curve, figure 68, and the brake thermal efficiency curve, figure 65. Although the average combustion pressure deviation varies considerably, it likewise follows the general trend of the overall air-fuel ratio, although it does not drop off at the maximum scavenging air ratio.

The exhaust pressure drop varies with the total air consumption although it does increase at maximum scavenging air ratio. The increase at this ratio may be due to slow burning as indicated by the low combustion pressures and the drop in the spark plug and cylinder head temperatures. If the peak combustion temperature and pressure occured much after top dead centre, the exhaust temperature would increase as less energy is removed from the cylinder as power. However the trapping efficiency levelled off whereas the fuel trapping efficiency did not, figure 64, so that a larger proportion of the cool scavenging air went out the ports with the exhaust gases. The cool air in the exhaust negated some of the exhaust temperature increase due to delayed combustion.

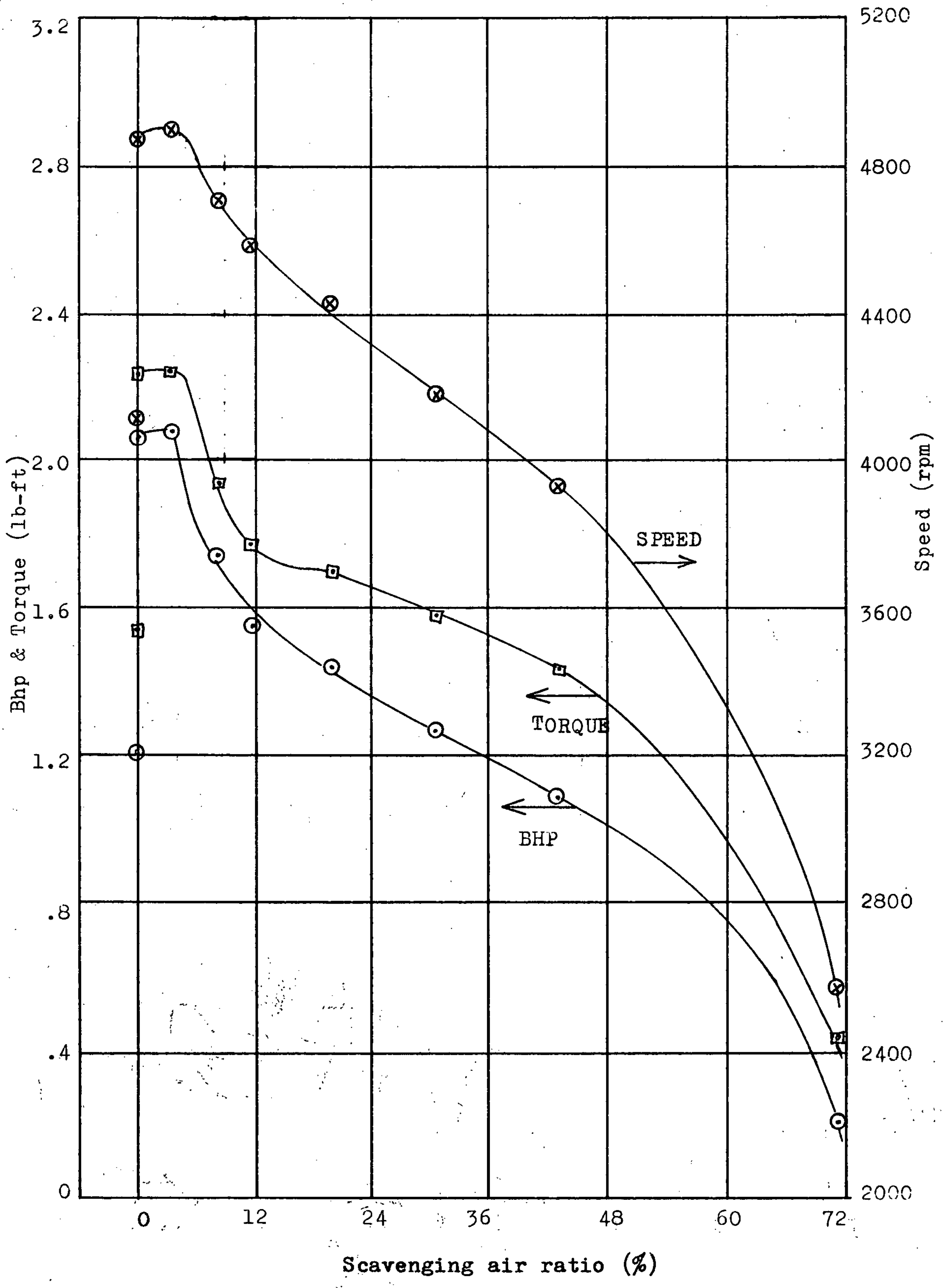

Figure 63. Performance curves with VP 100%, Jet FA, from sheet 24.

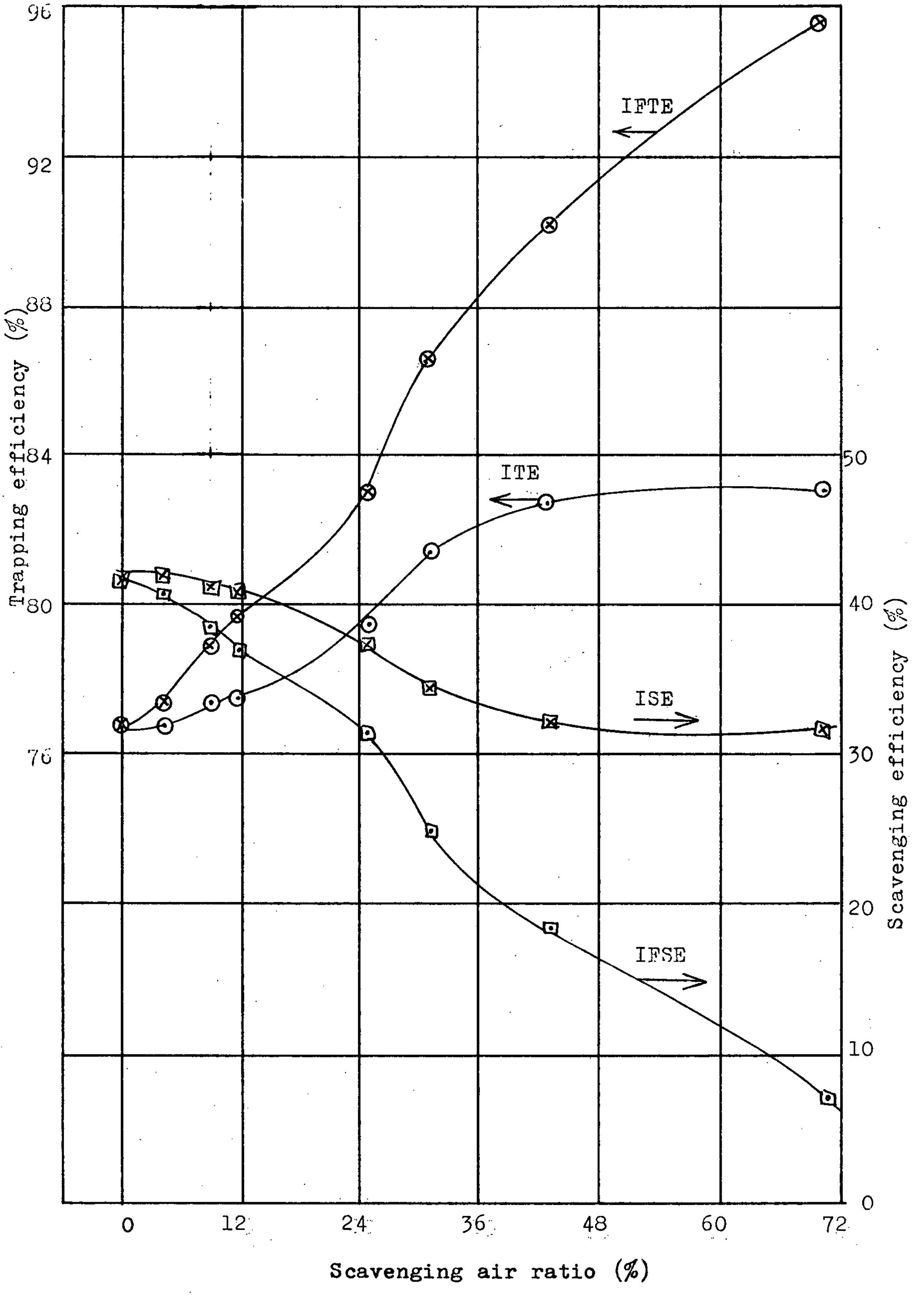

Figure 64. Efficiency curves with VP 100%, Jet FA, from sheet 24.

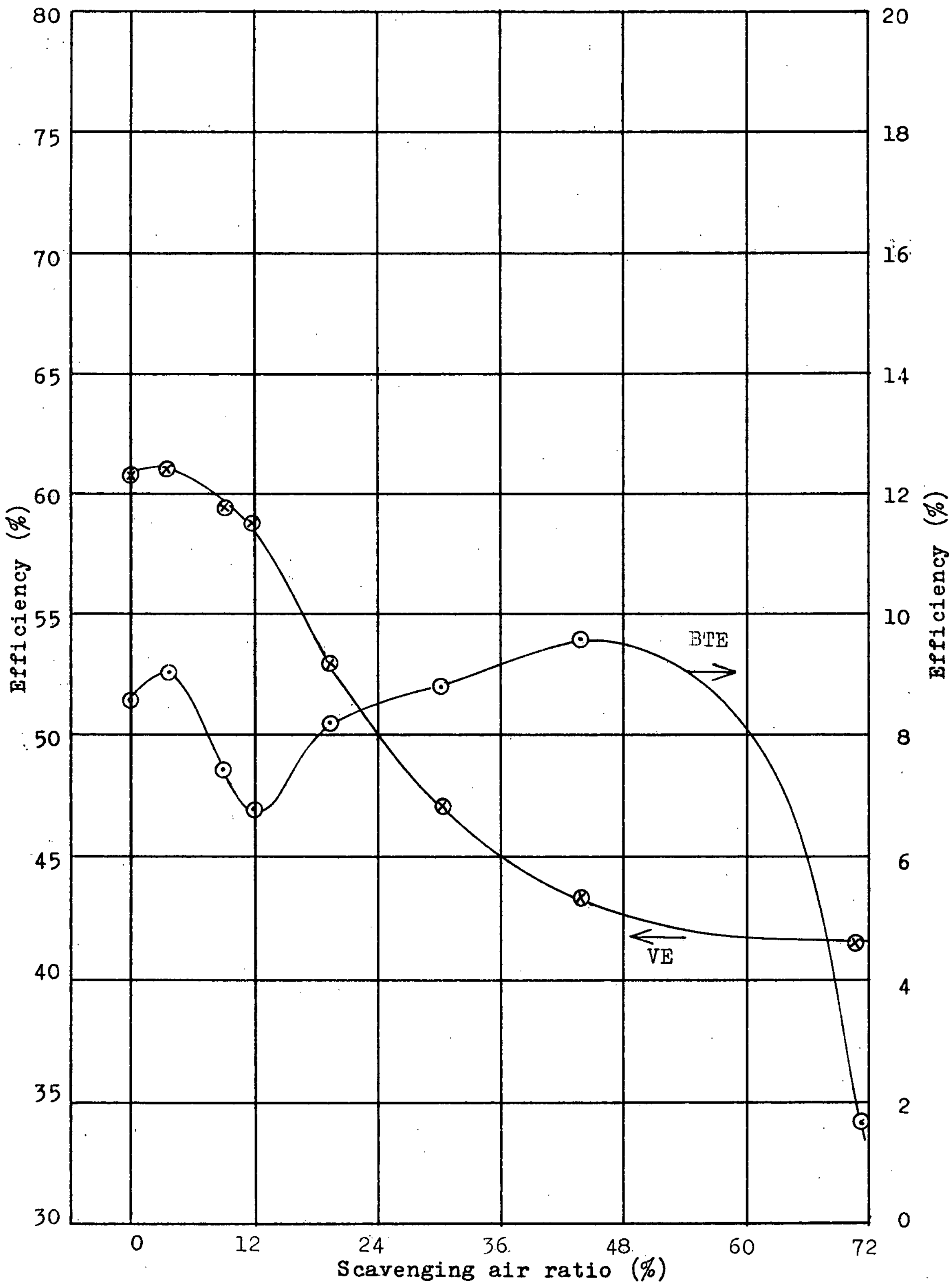

Figure 65. Efficiency curves with VP 100%, Jet FA, from sheet 24.

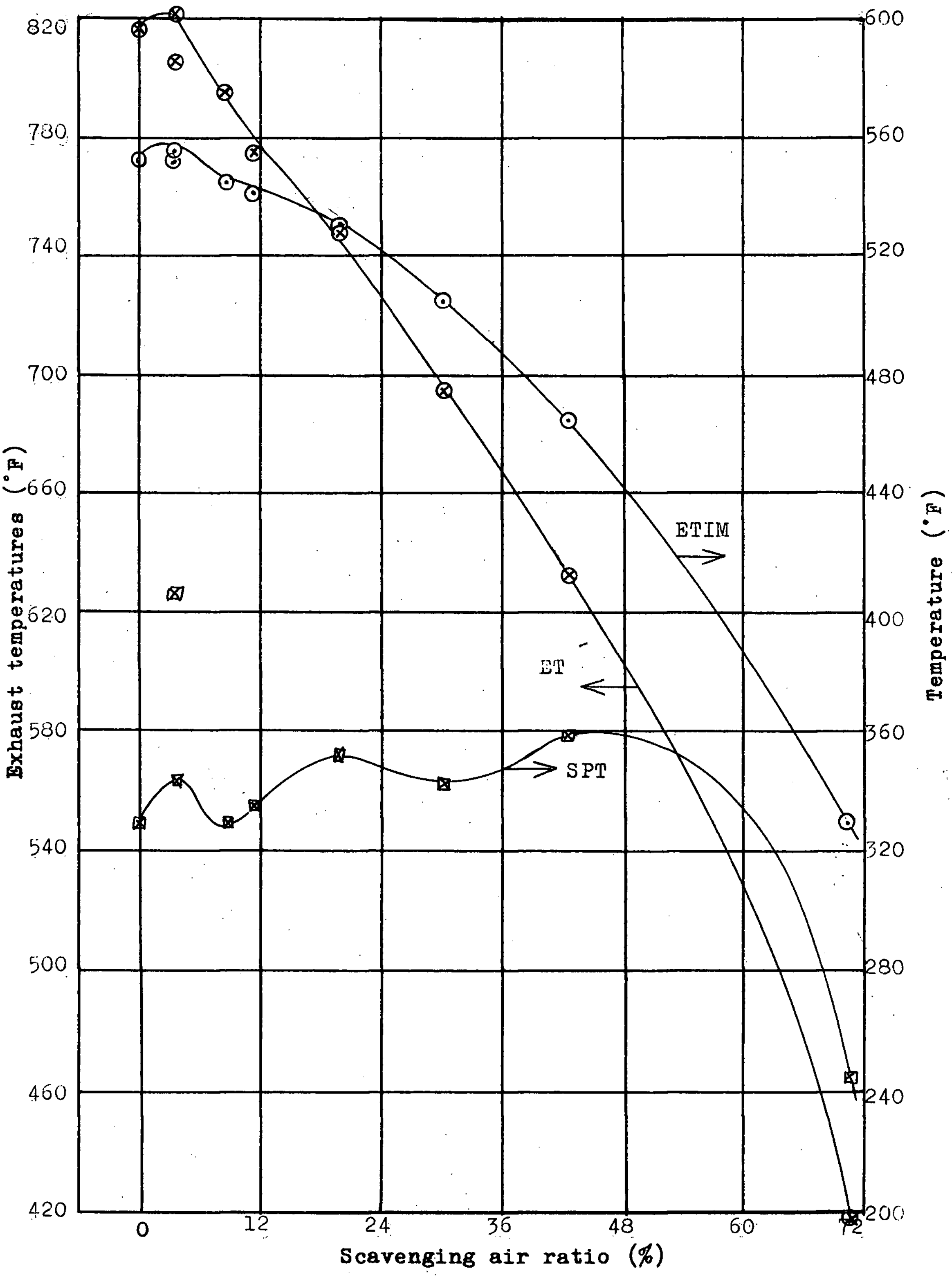

Figure 66. Temperature curves with VP 100%, Jet FA, from sheet 24.

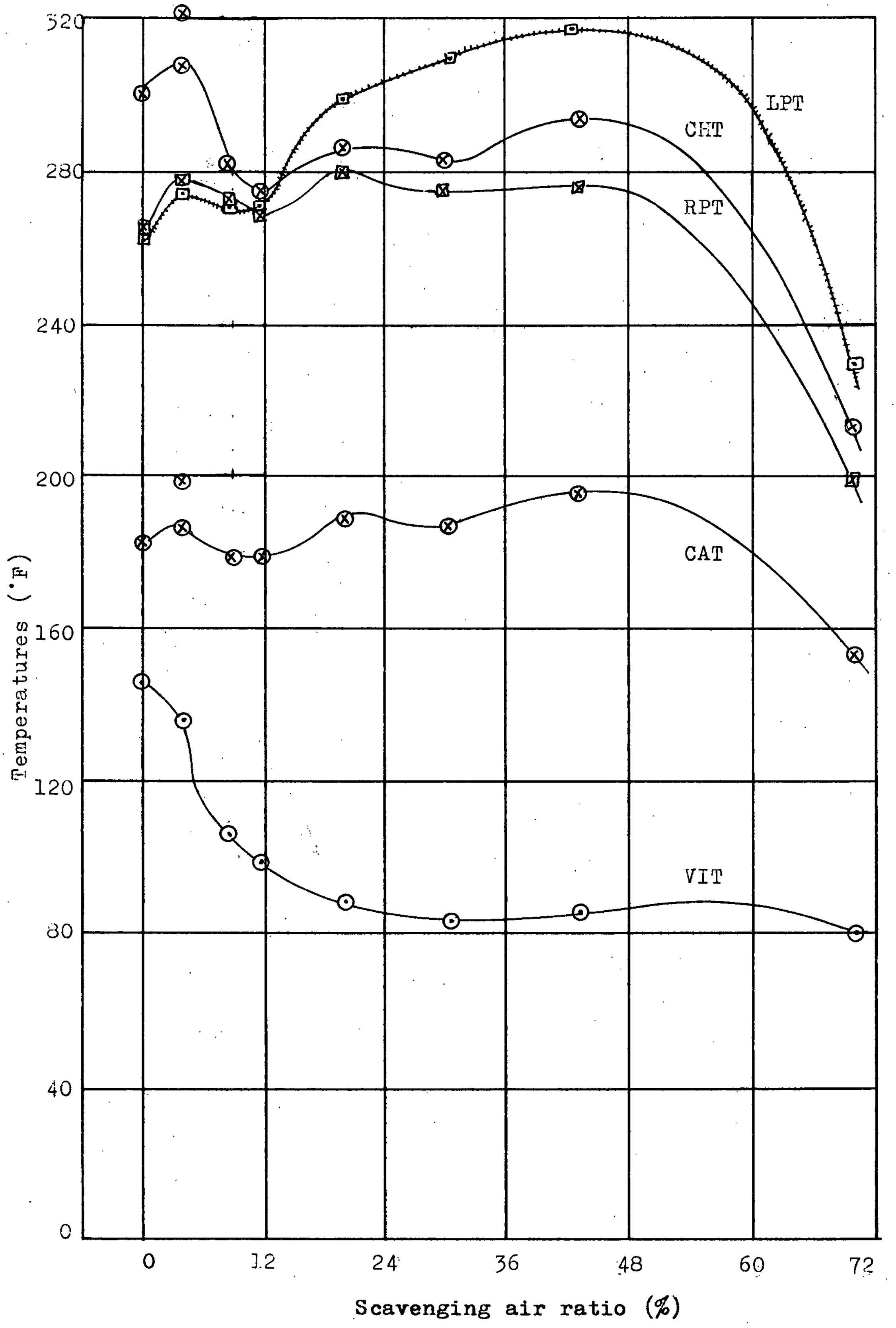

Figure 67. Temperature curves with VP 100%, Jet FA, from sheet 24.

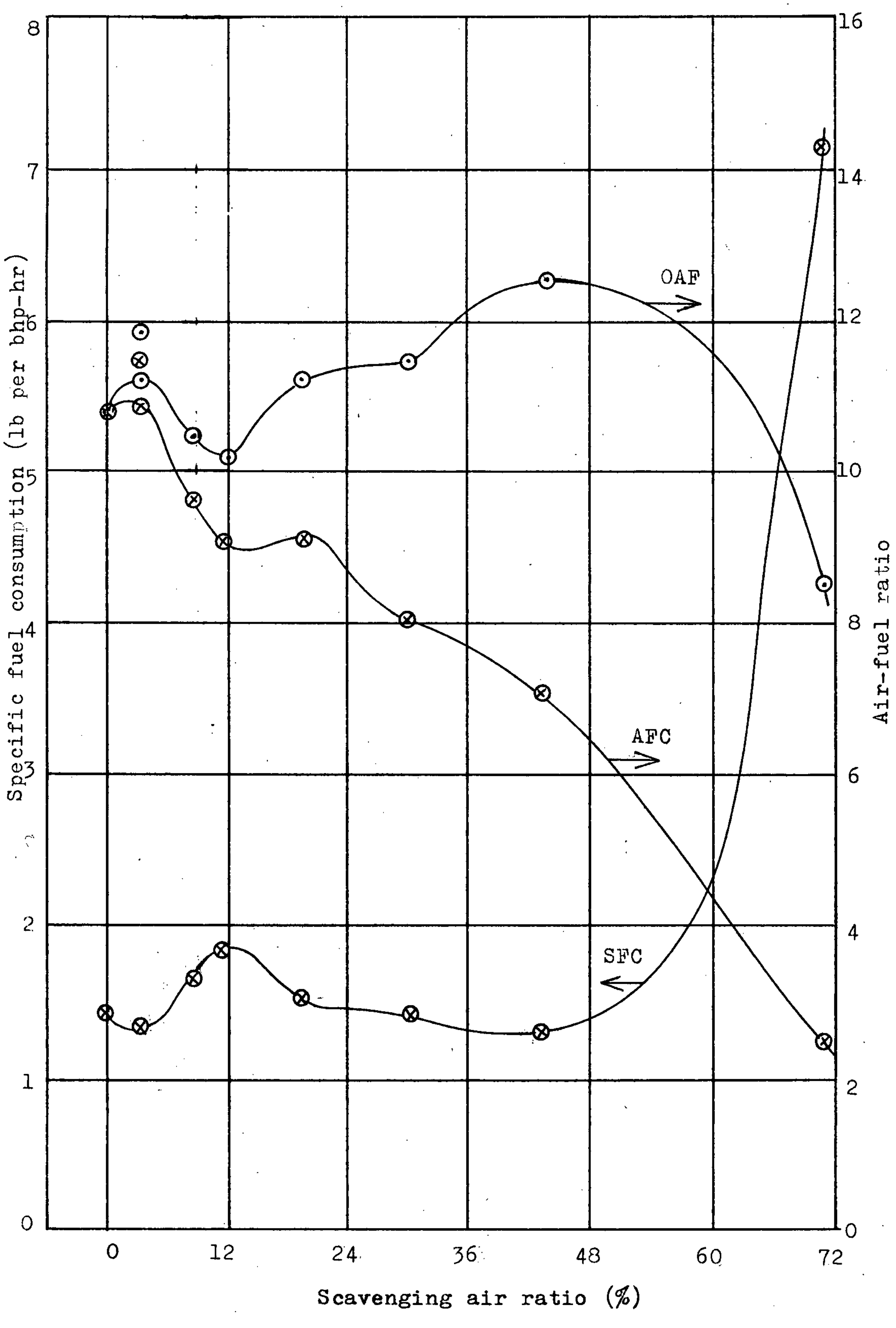

Figure 68. Air & fuel consumption curves with VP 100%, Jet FA, from sheet 24.

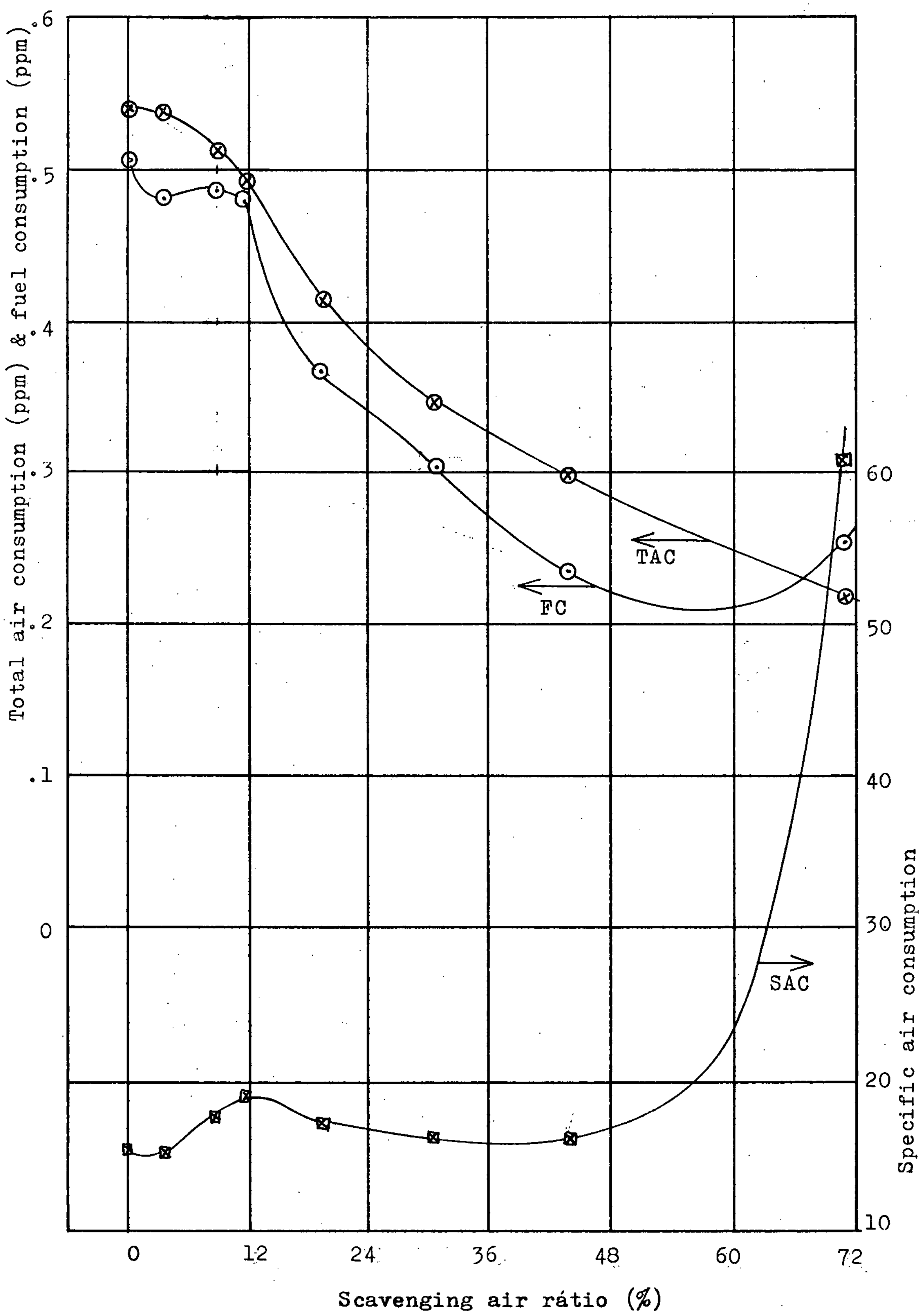

Figure 69. Air & fuel consumption with VP100%, Jet FA, from sheet 24.

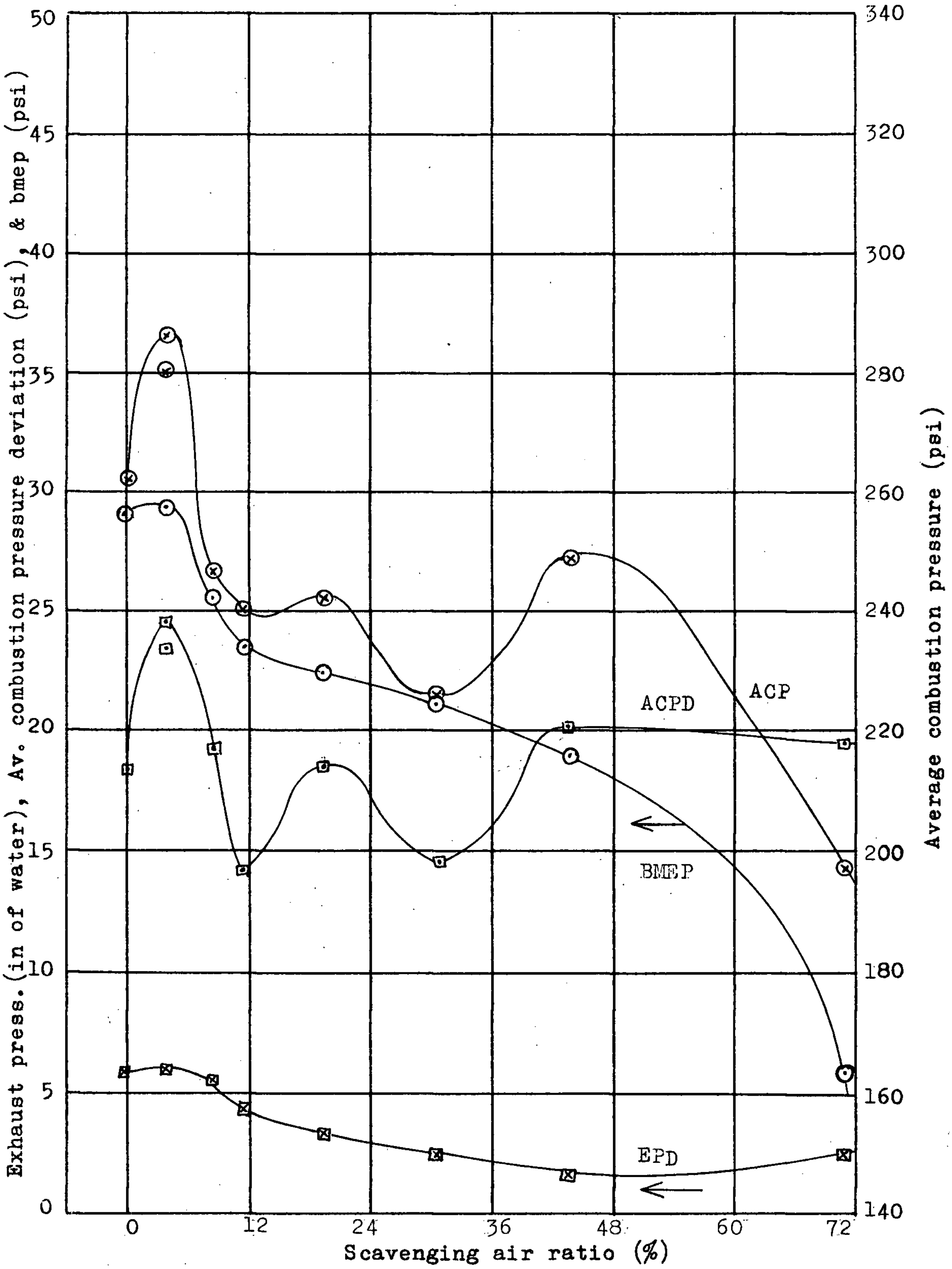

Figure 70. Pressure curves with VP 100%, Jet FA, from sheet 24.

Figures 71-78 with VP 0% or 100%, Jet F.A., NS 4500, from sheet 25

The purpose of this series of tests was to compare the performance of the engine at constant speed on the basis of power produced. The scavenging air valve was kept wide open and the throttle varied to obtain several different torque readings with the stratified charge scavenging system; the valve was then closed and several more tests taken to obtain the torque readings with conventional scavenging.

The first few tests were performed with the valve full open and the throttle varied from full open to a minimum setting which still allowed the engine speed to be kept at 4500 rpm. The choke was adjusted as required to maintain smooth operation. Engine exhaust produced excessive smoke during most of the tests. Over the duration of each of the tests, the engine slowed down 50 - 250 rpm.

When the valve was closed after the tests with the open valve were finished, the speed could not be brought up to the desired value of 4500 rpm, so that a test was conducted at maximum speed obtainable. The performance at this setting is plotted as the points at .92 bhp. These points will not be considered representative of normal engine performance, as the exhaust ports were found to be almost completely closed with carbon when the engine was stopped. The carbon was removed and several days later the tests were continued with the valve closed, at several throttle positions. As in all the tests, the speed was brought up to 4500 rpm by adjusting the dynamometer after the throttle had been set.

Figure 71 indicates a definite reduction in the specific fuel consumption with the stratified charge scavenging system. But the temperatures, figure 75 and 76, indicate that the engine ran hotter when the scavenging was done with a stratified charge. The exhaust temperature as well as the spark plug and cylinder head temperatures were all generally higher. The reason for this may be explained with reference to the air-fuel ratio, figure 77. As the low air-fuel ratio approaches the stiochiometric ratio, as a result of stratifying the charge, combustion is more efficient as explained in chapter 1. The result is an increase in the combustion temperatures. The specific fuel consumption, figure 78, is higher as extra air is admitted to cylinder which also generally increases the volumetric efficiency, figure 74. The fuel trapping efficiency is increased, figure 72, indicating that more of the fuel remains in the cylinder. Due to the few points for conventional scavenging the shape of the curve may be closer to the trapping efficiency with stratified charge scavenging than the curve that is drawn would indicate. Similar observations may be made about the scavenging efficiencies as shown in figure 73.

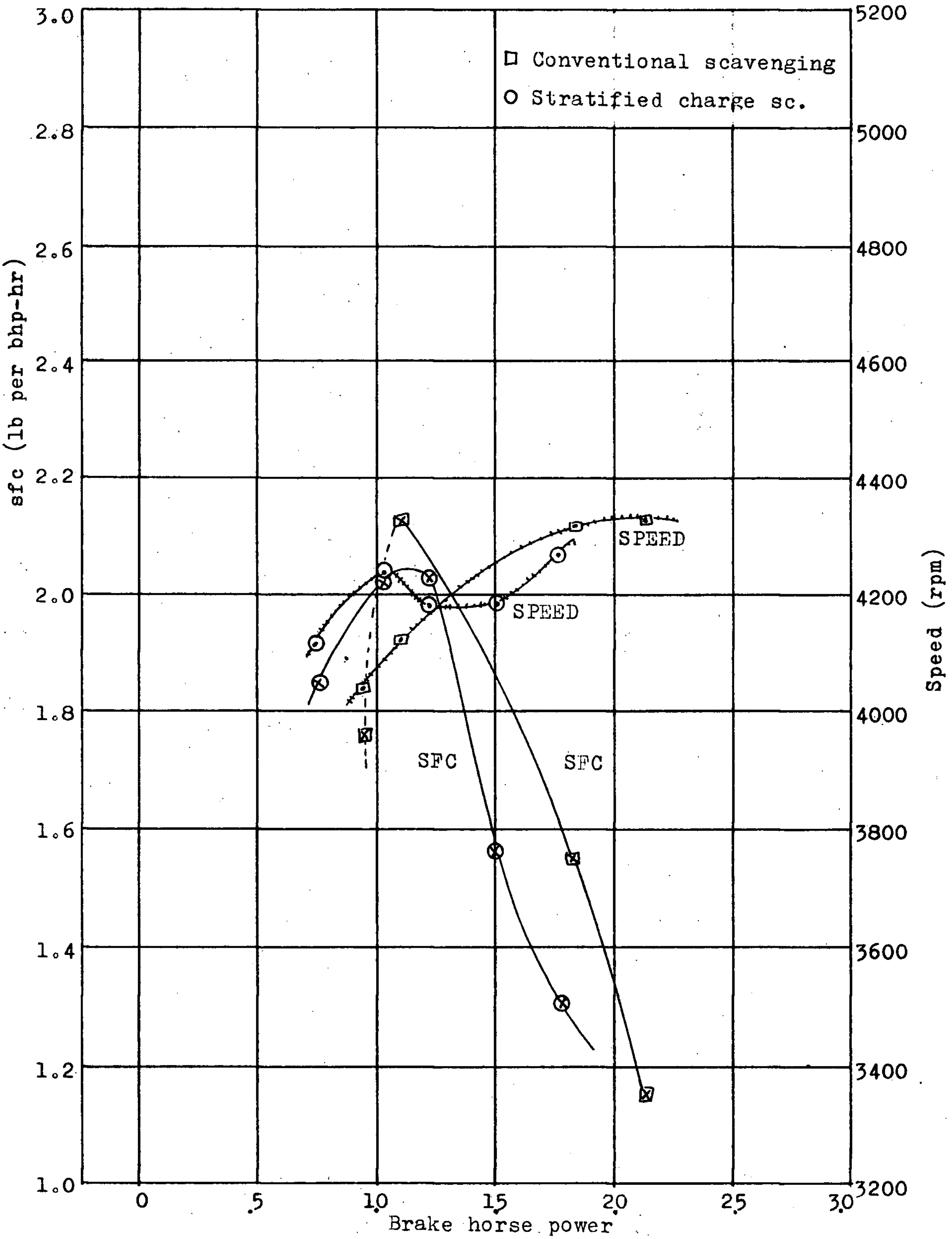

Figure 71. Performance curves with VP 100%, Jet FA, NS 4500, from sheet 25.

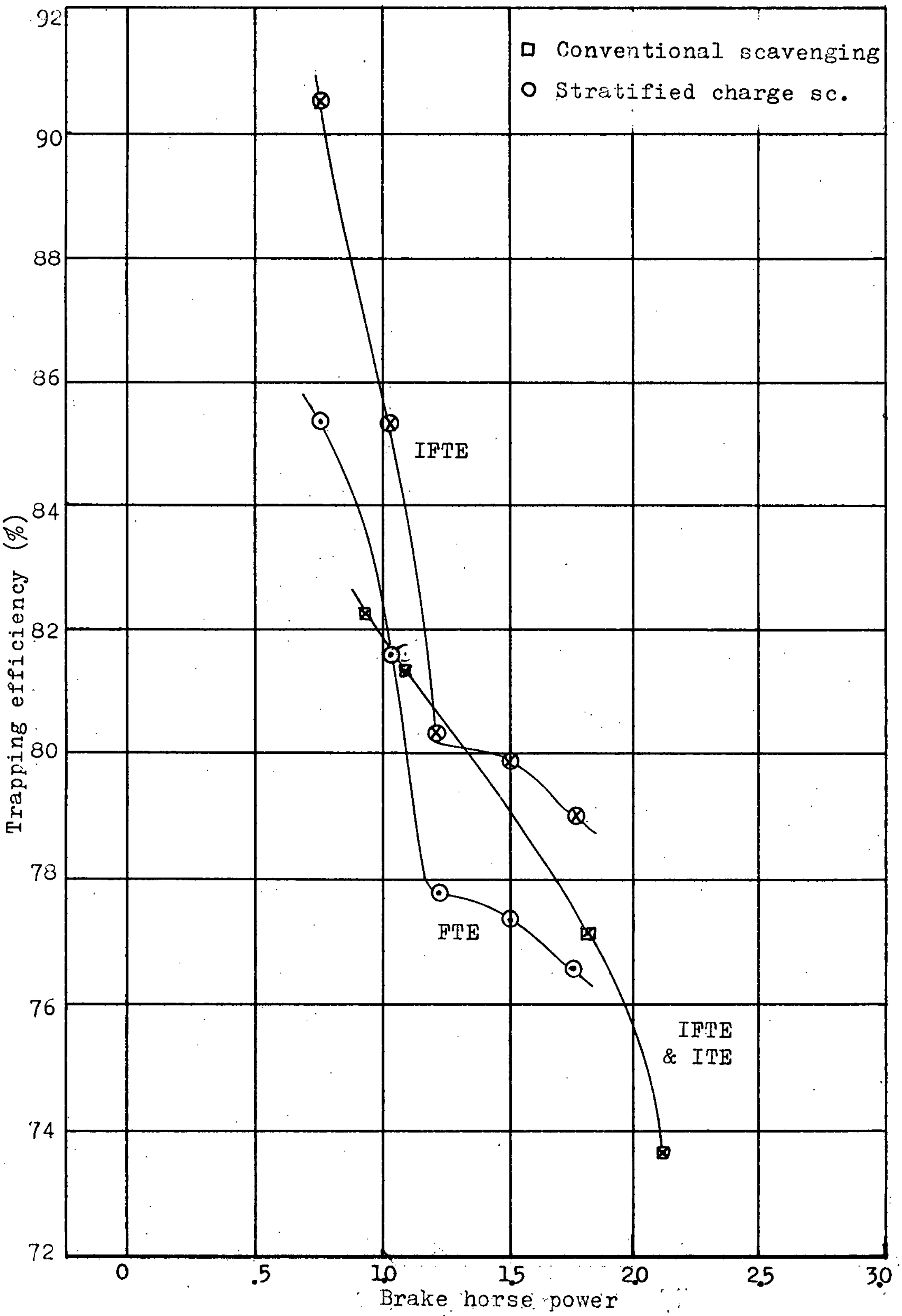

Figure 72. Trapping efficiency curves with VP 100%, NS 4500, from sheet 25.

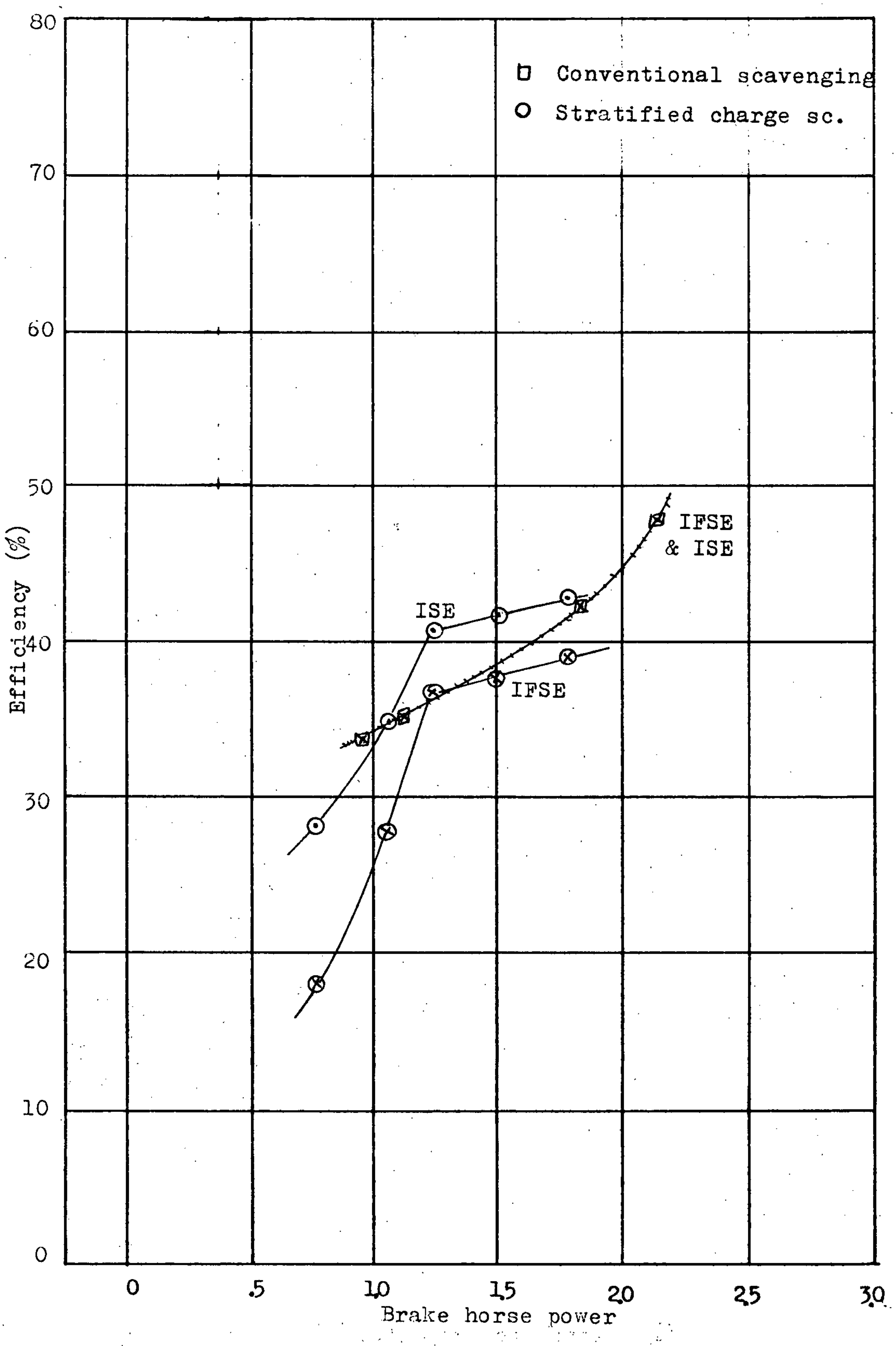

Figure 73. Scavenging efficiency curves with VP 100%, Jet FA, NS 4500, from sheet 25.

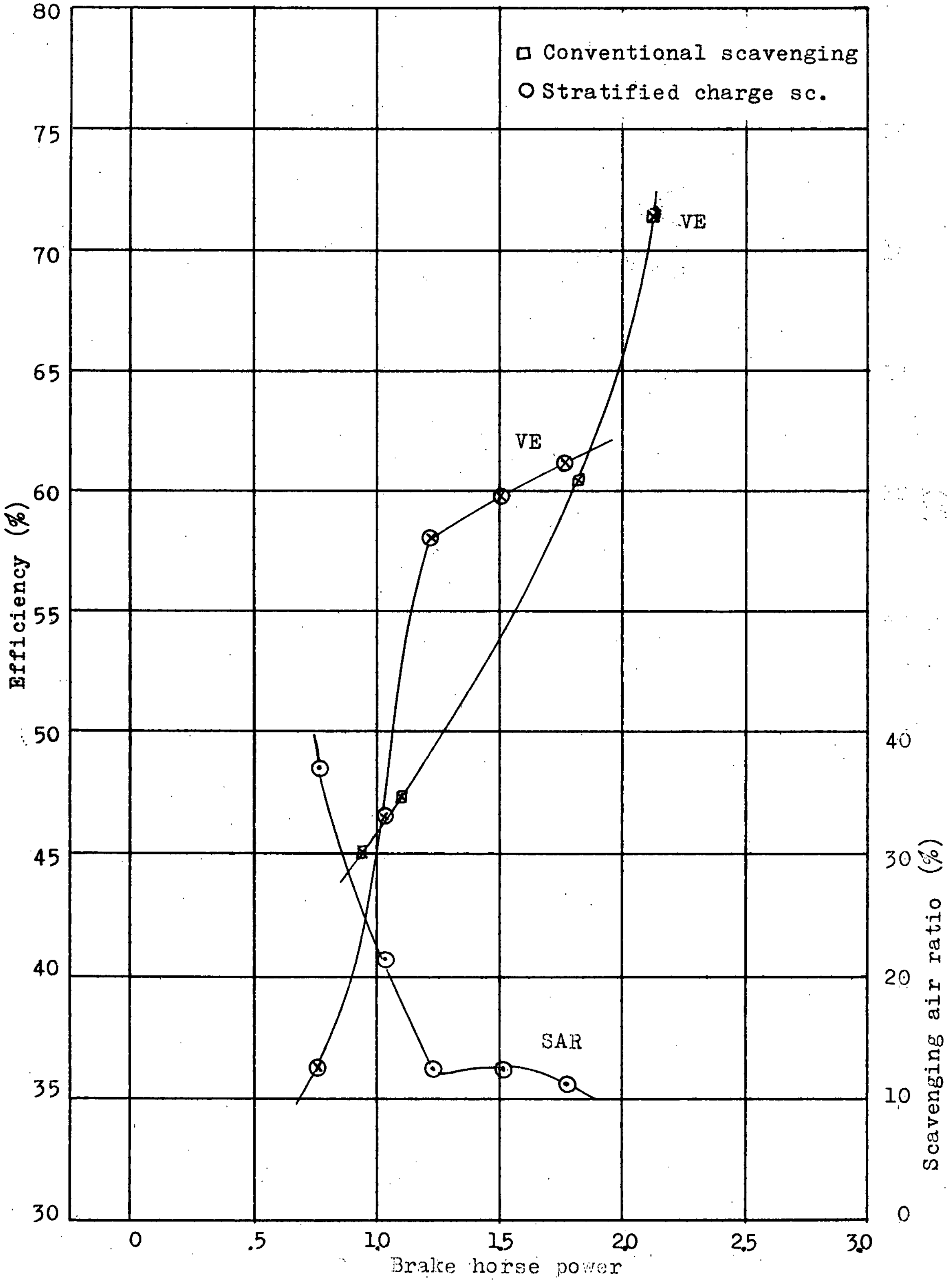

Figure 74. Efficiency and air ratio curves with VP 100%, Jet FA, NS 4500, from sheet 25.

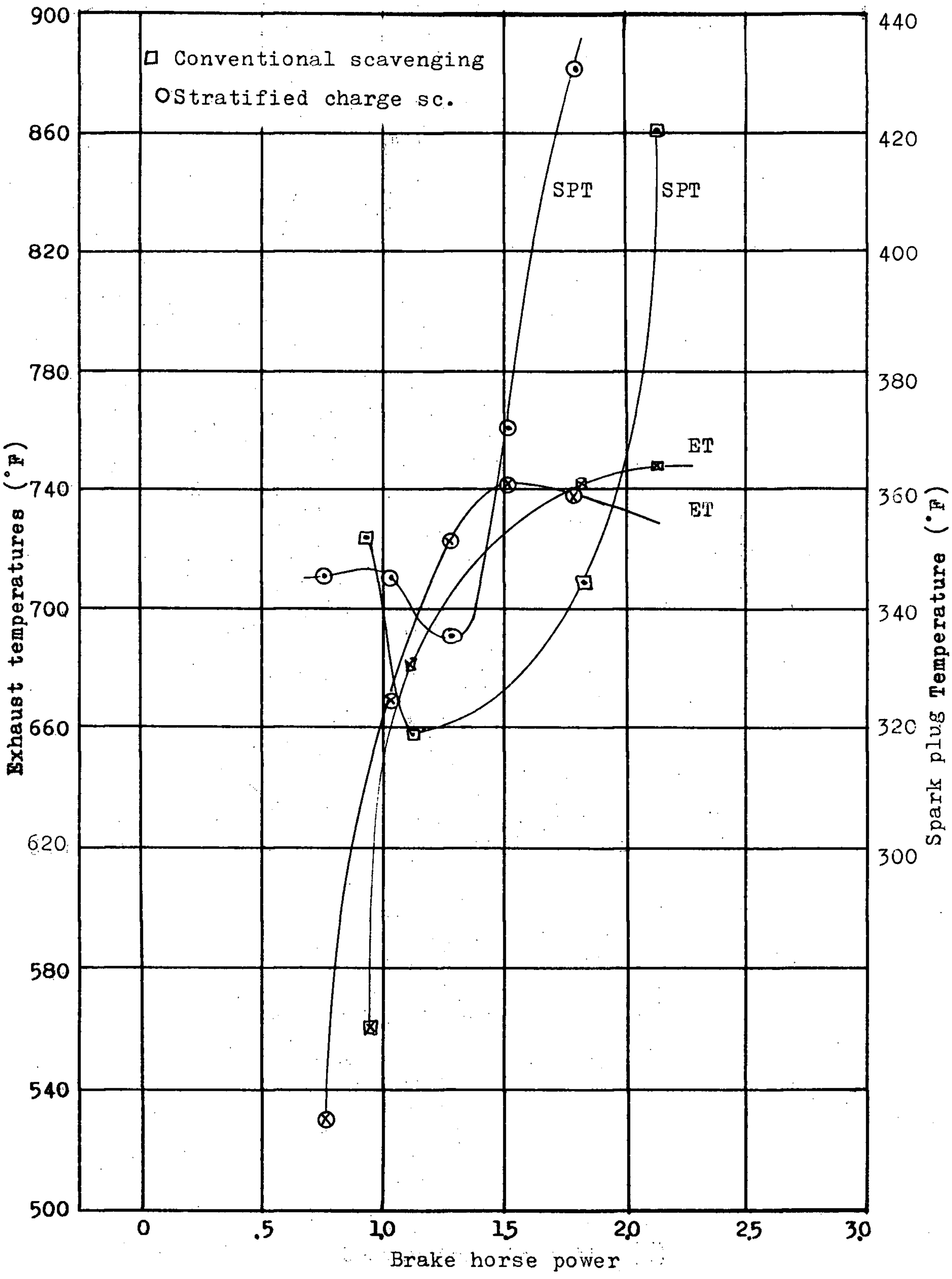

Figure 75. Temperature curves with VP 100%, Jet FA, NS 4500, from sheet 25.

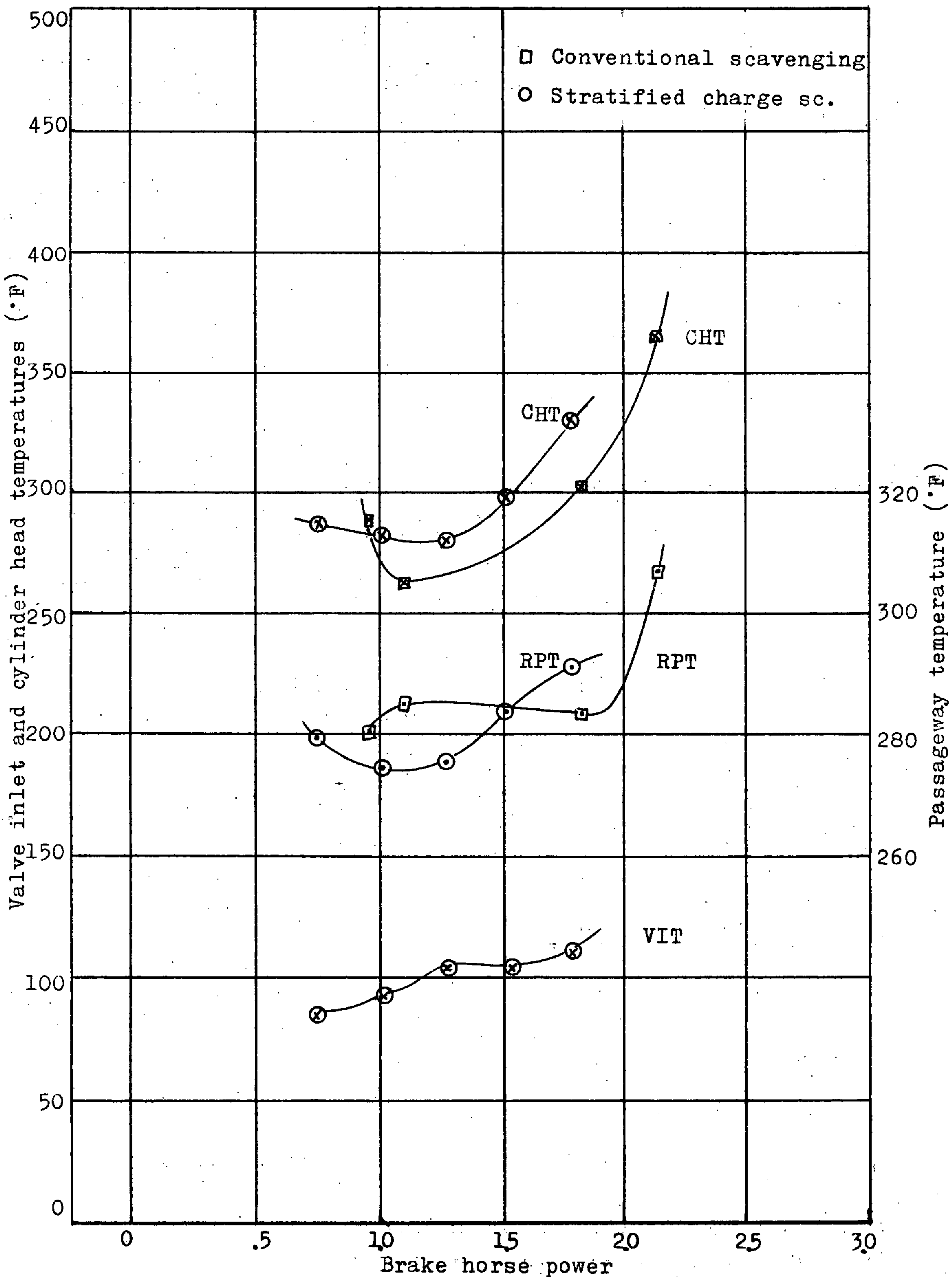

Figure 76. Temperature curves with VP 100%, Jet FA, NS 4500, from sheet 25.

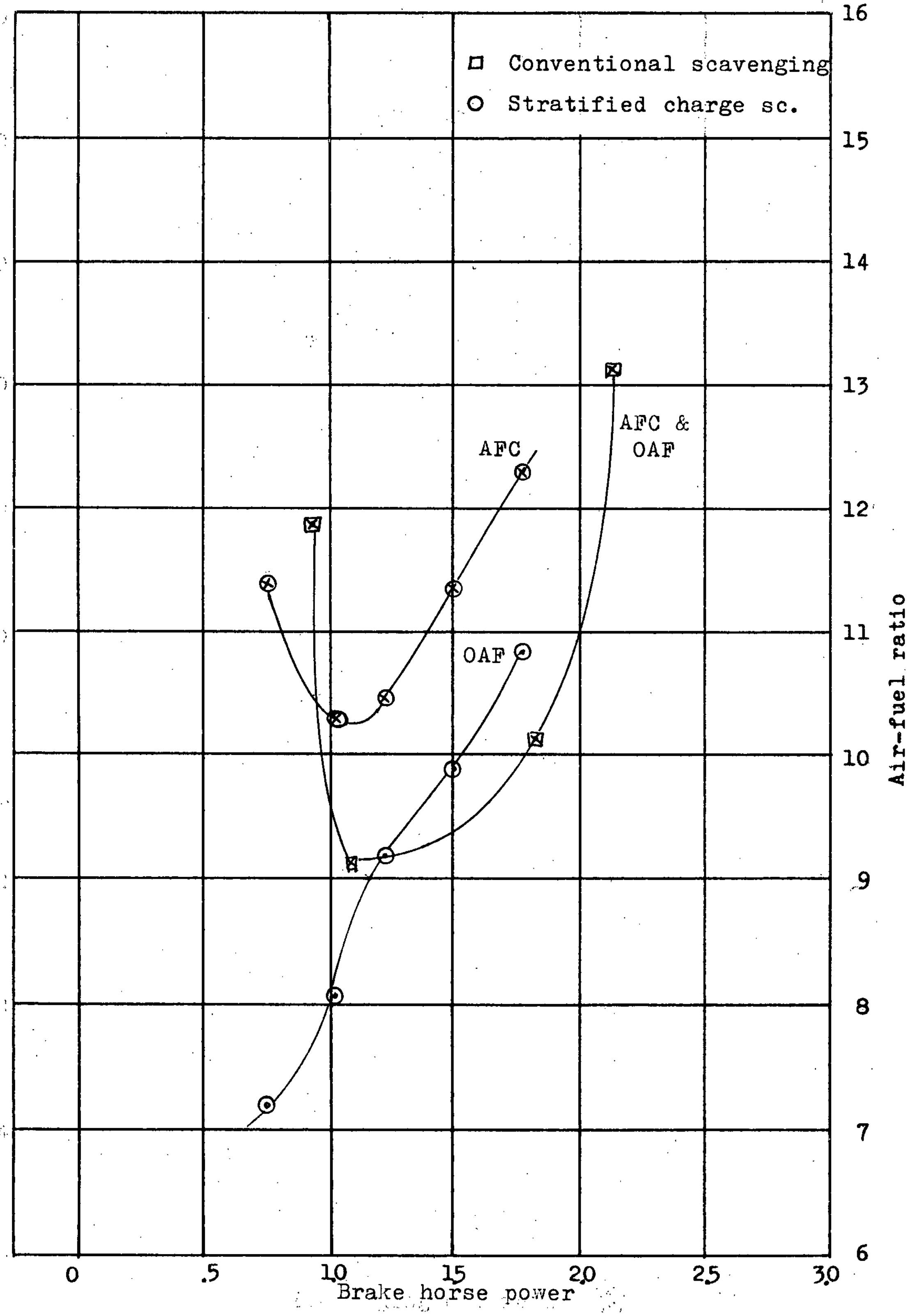

Figure 77. Air-fuel ratio curves with VP 100%, Jet FA, NS 4500, from sheet 25.

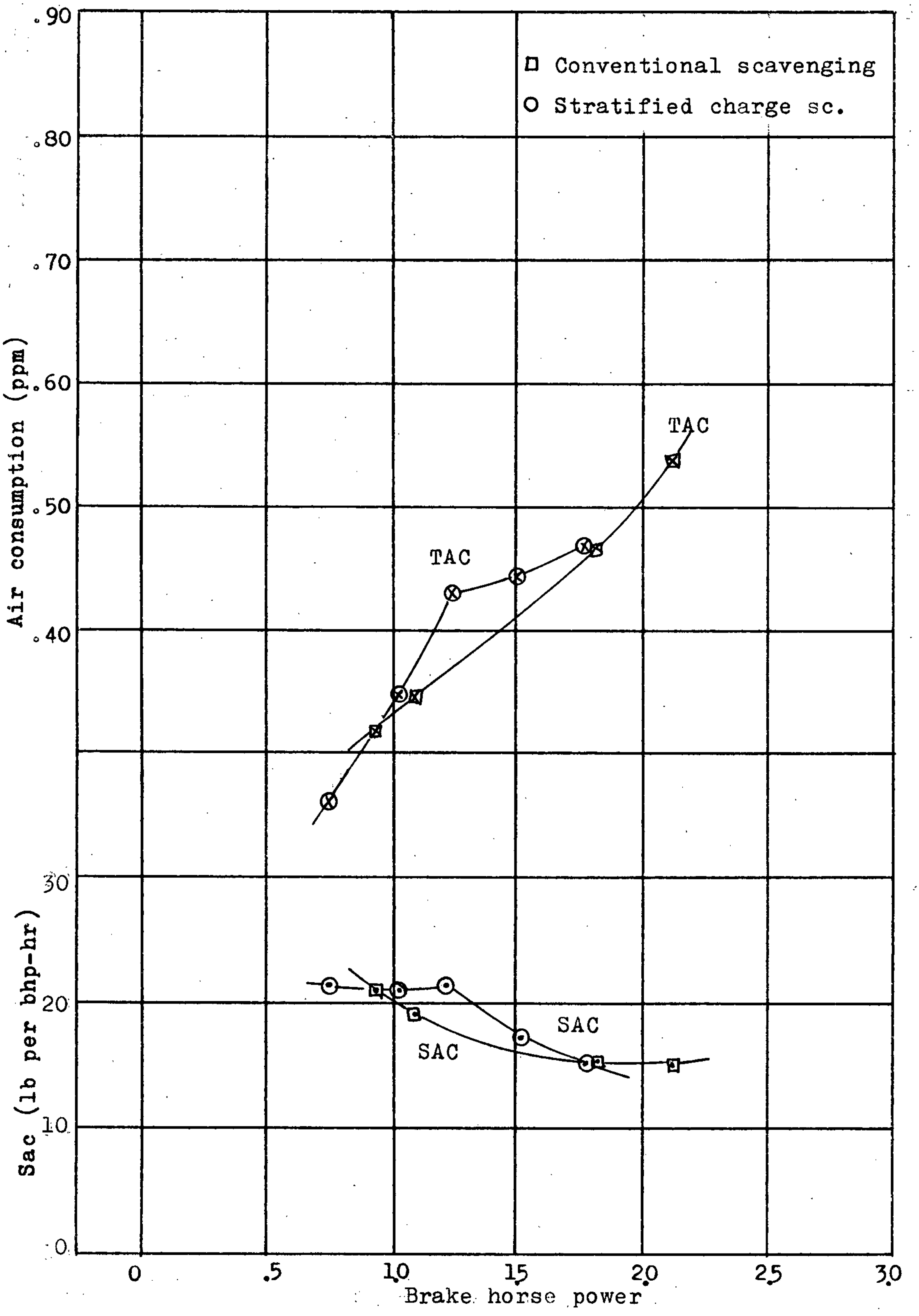

Figure 78. Air consumption curves with VP 100%, Jet FA, NS 4500, from sheet 25.

Figures 79-85 Speed and specific fuel consumption curves

The specific fuel consumption and speed for each series of tests were plotted on a brake horse power base to enable a comparison to be made at a particular brake horse power. In most cases interpolation of the curves was necessary. The specific fuel consumption was generally lower when the engine was running on the stratified charge scavenging system, although in several cases the consumption was the same for both the new and the conventional system and in one case the consumption was higher. The unreliable interpolation of the line from only two points may account for the exception in the latter case. There is very little difference between the speeds for the two scavenging methods as the power absorbed by the dynamometer varies with the speed of the engine.

Figure 85 may be taken as representing the results as these curves were plotted from the results obtained from tests performed with the throttle position varying from full open to 23% open. The curve indicates a very definite decrease in the specific fuel consumption especially at the lower end of the horse power scale.

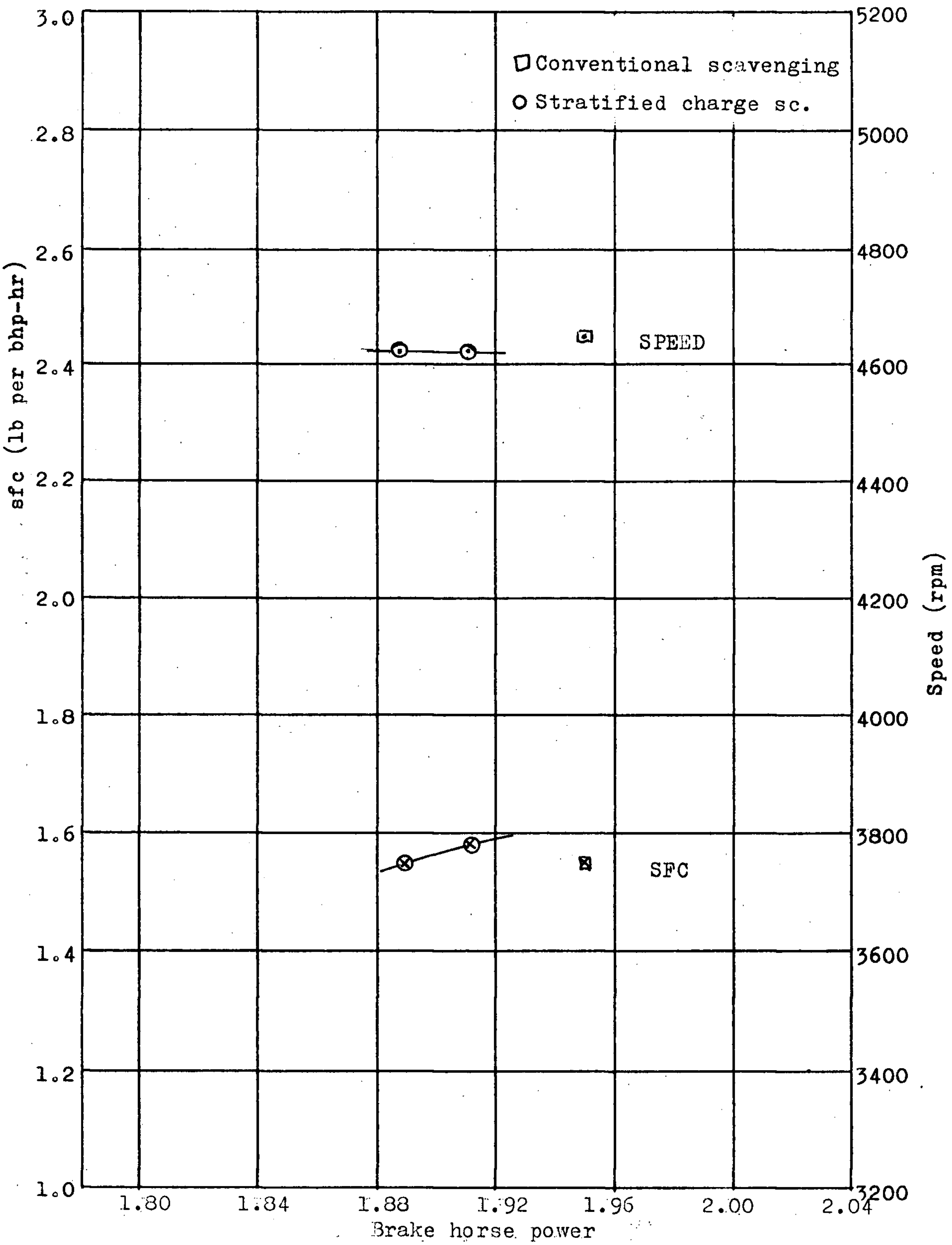

Figure 79. Speed & sfc curves with TP 50%, NS 4900, from sheet 15.

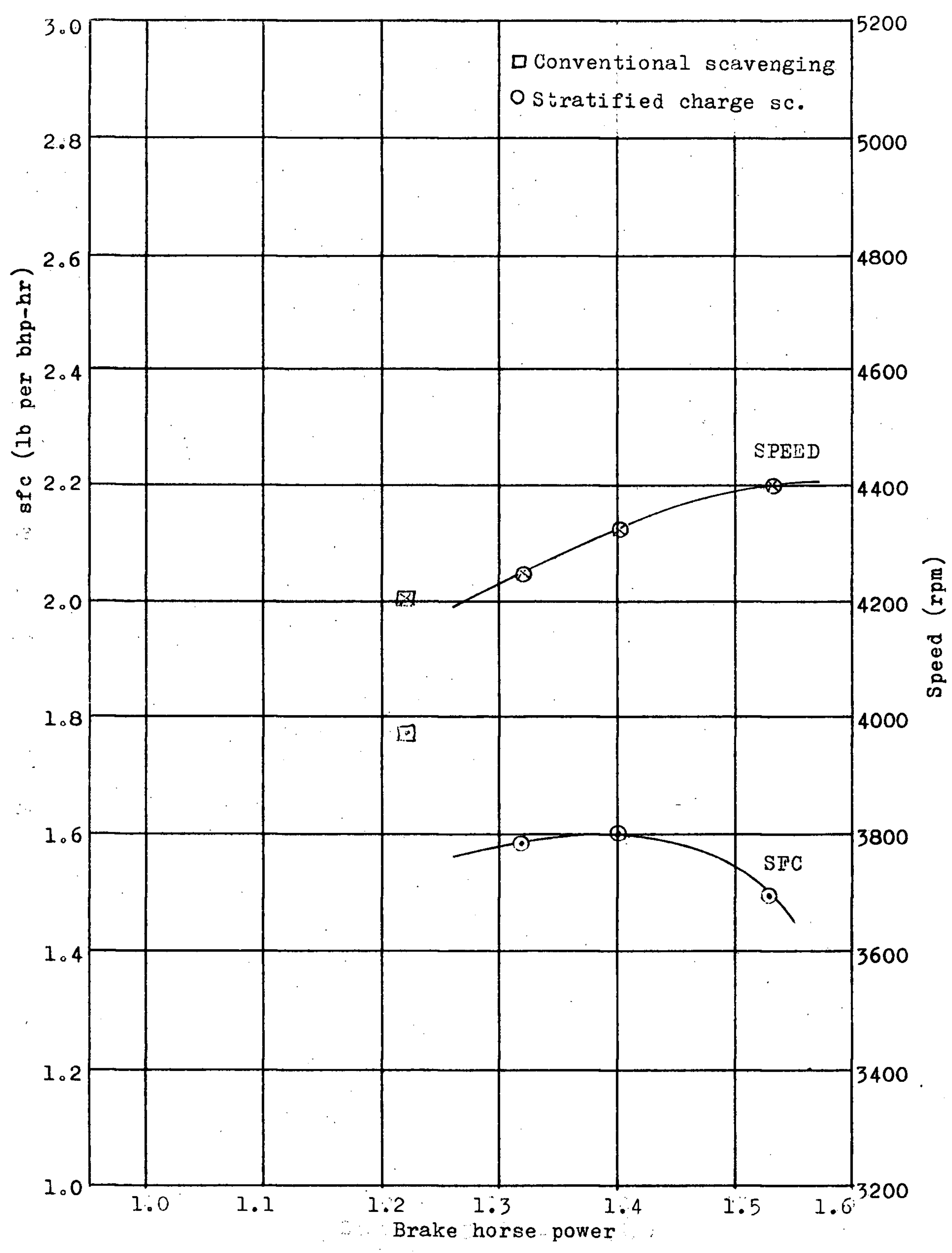

Figure 80. Speed and sfc curves with TP 50%, NS 4500, from sheet 17.

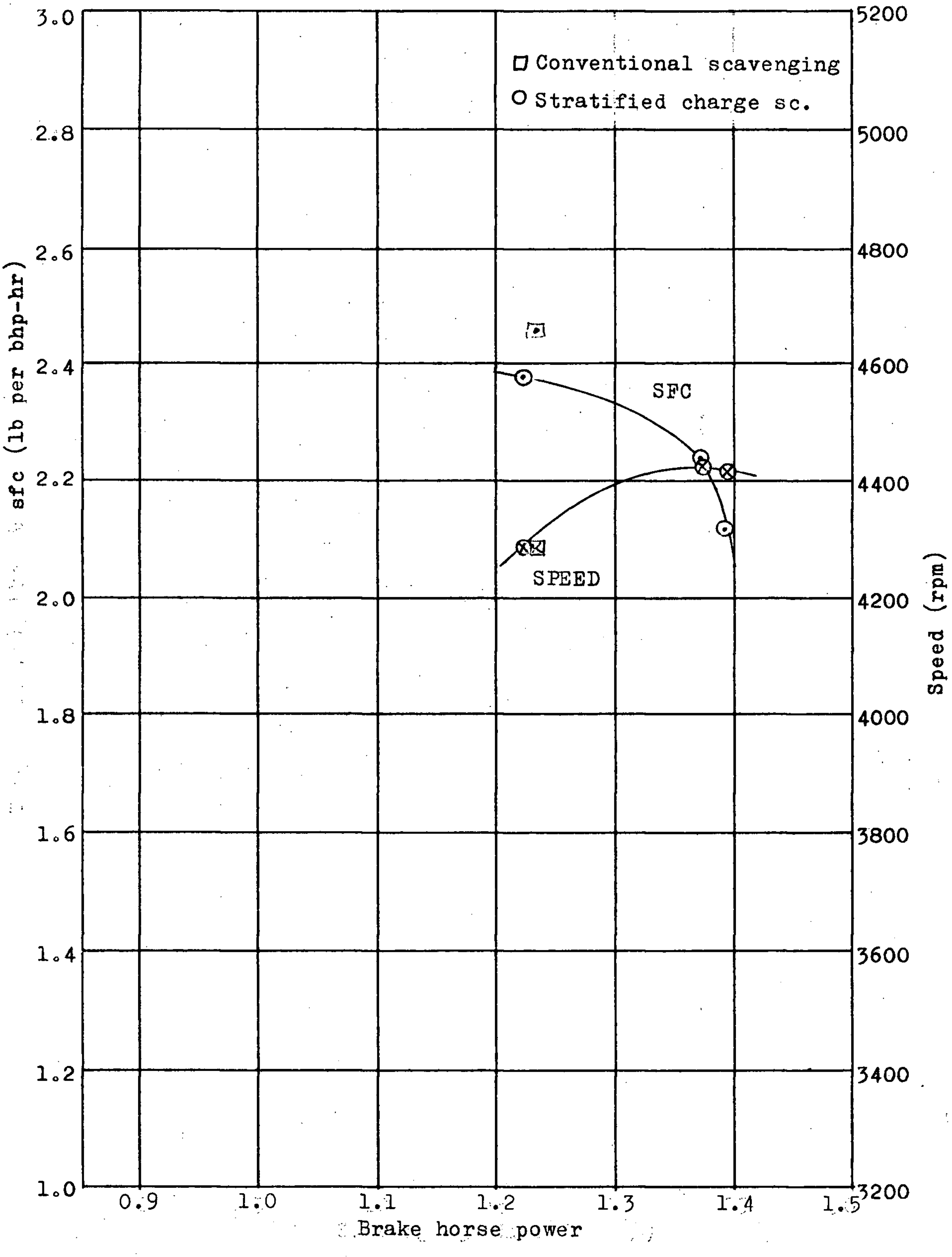

Figure 81. Speed and sfc curves with TP 50%, NS 4700, from sheet 20.

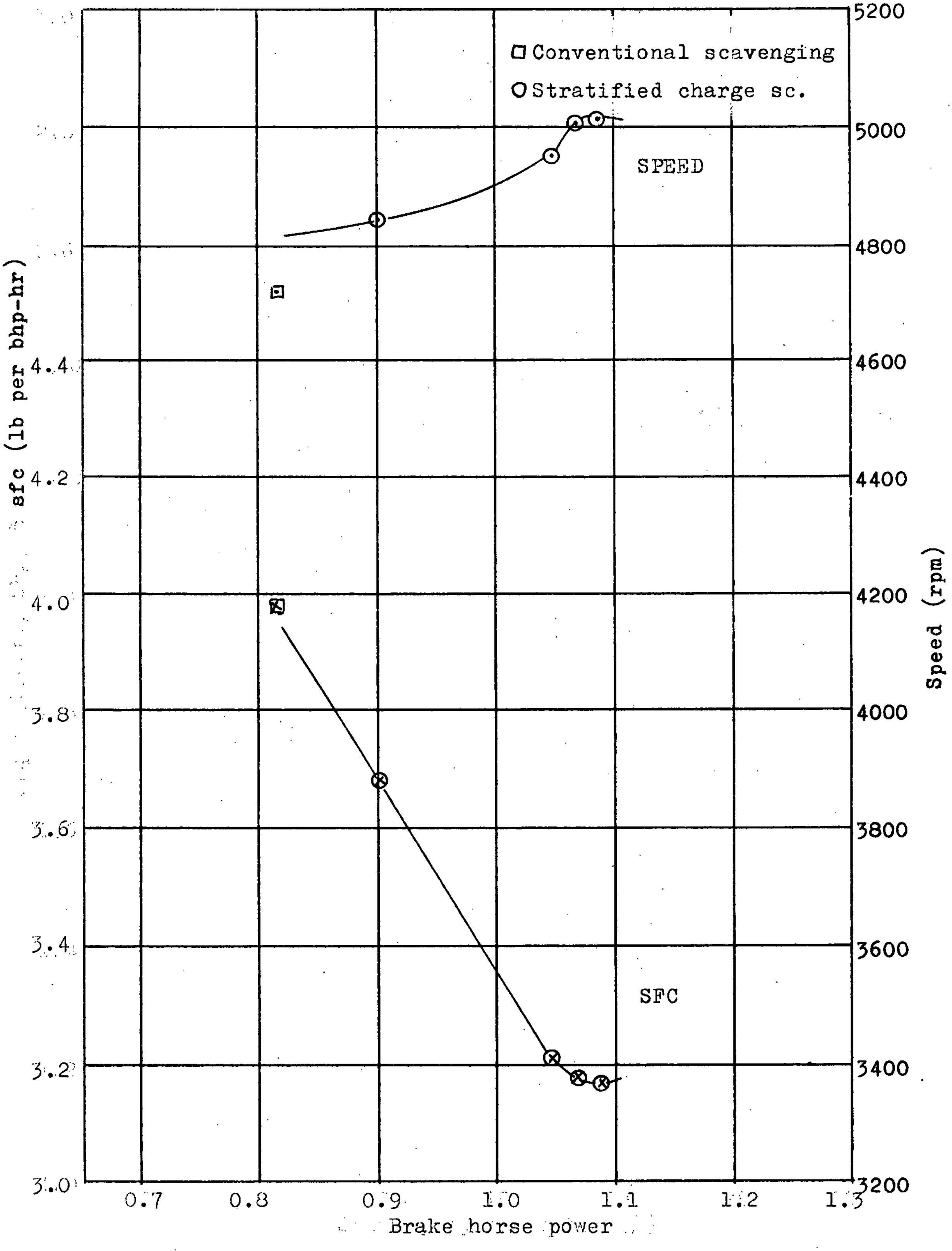

Figure 82. Speed & sfc curves with TP 60%, NS 5000, from sheet 21.

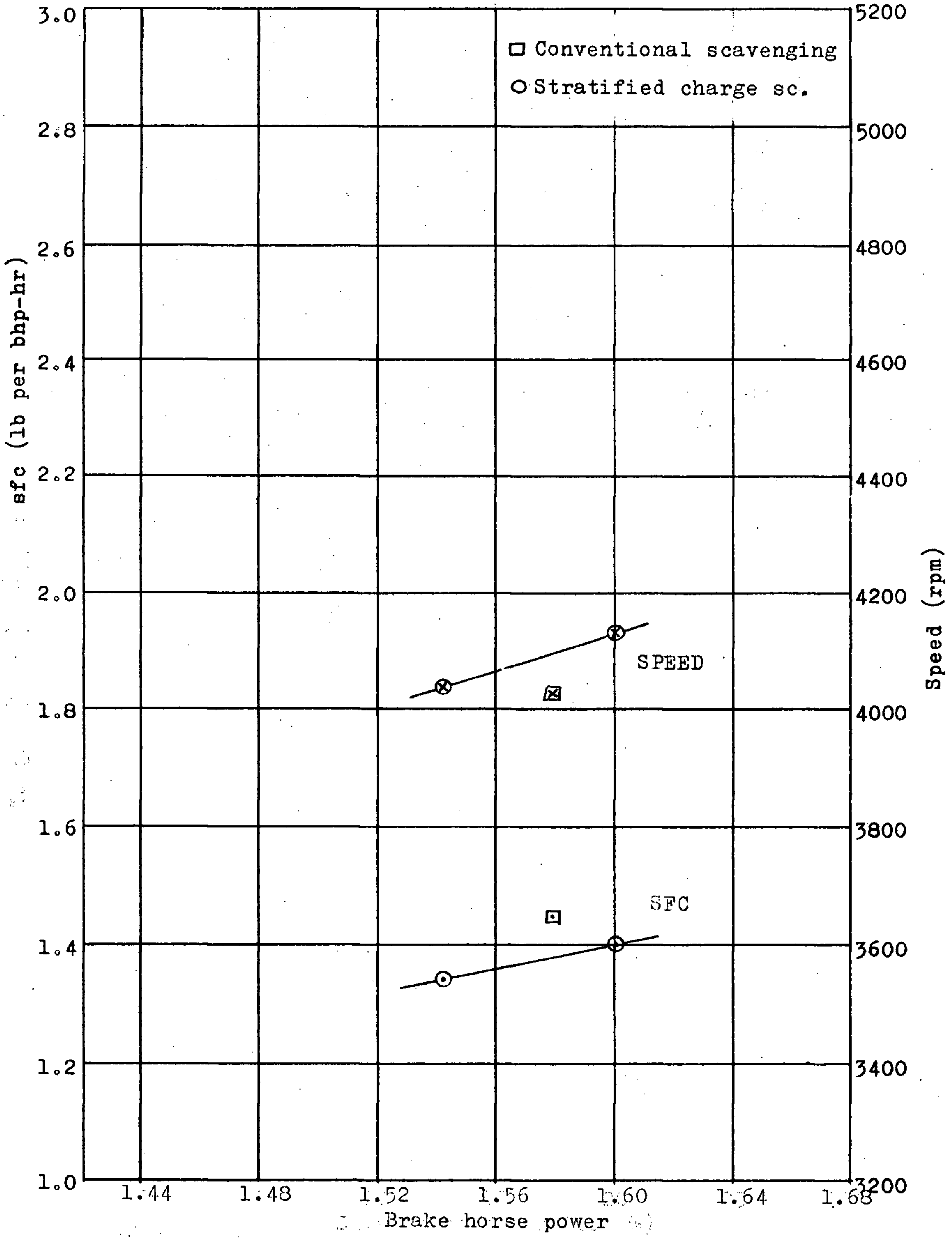

Figure 83. Speed & sfc curves with TP 70%, NS 4300, from sheet 22.

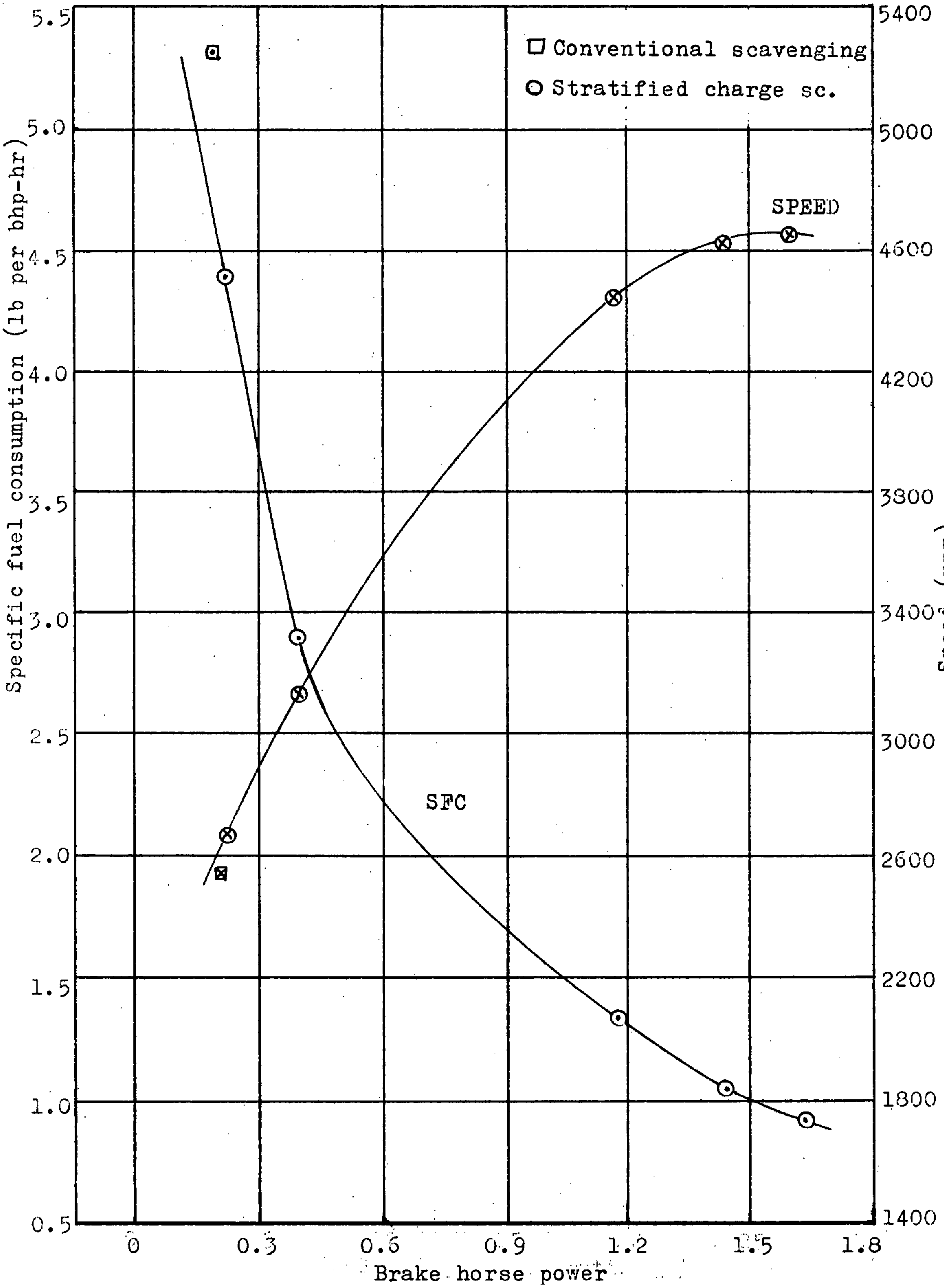

Figure 84. Speed & sfc curves with TP 30%, from sheet 23.

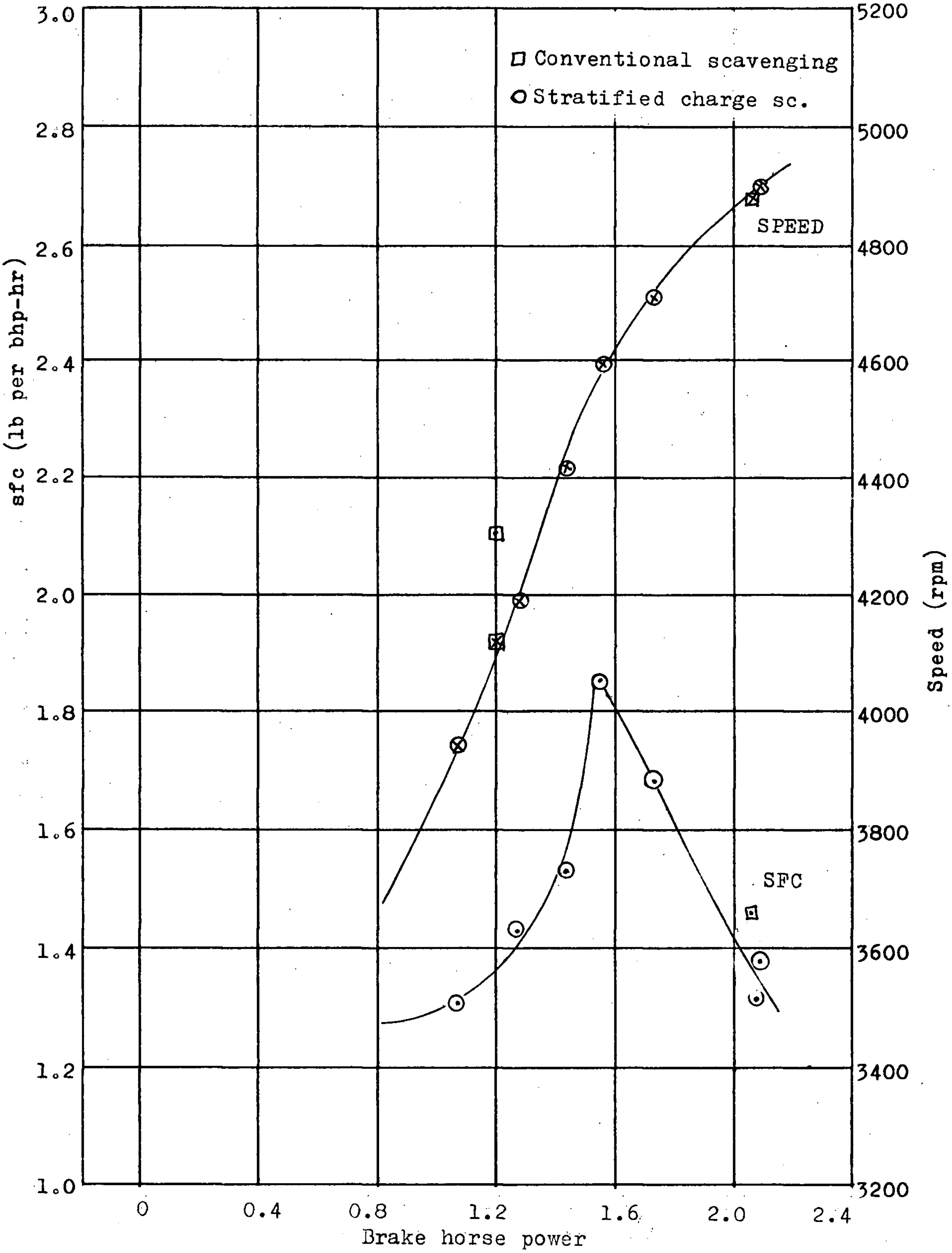

Figure 85. Speed & sfc curves with VP 100%, from sheet 24.

Figures 86-92 Experimental curves of exhaust and spark plug temperatures

The exhaust and spark plug temperatures were plotted on a brake horse and power base, to enable a comparison to be made on the basis of a particular brake horse power.

The exhaust temperatures are generally lower for stratified charge although in three tests it was slightly higher. The spark plug temperature was sometimes higher, and sometimes lower.

Figure 92 can be taken as representing the tests generally as the points for this curve were drawn from tests conducted at a constant valve position for purposes of determining the performance of the engine as the incoming charge was throttled. The curves show that the exhaust temperature is lower and the spark plug temperature is higher for the stratified charge scavenging system. The higher spark plug temperature indicates more efficient combustion as the overall combustion temperature is higher. The lower exhaust temperature indicates that more energy is removed from the exhaust gases before being discarded. The lower temperature may also be due to an increase in the quantity of cool air present in the exhaust pipe which had escaped from the cylinder through the exhaust ports.

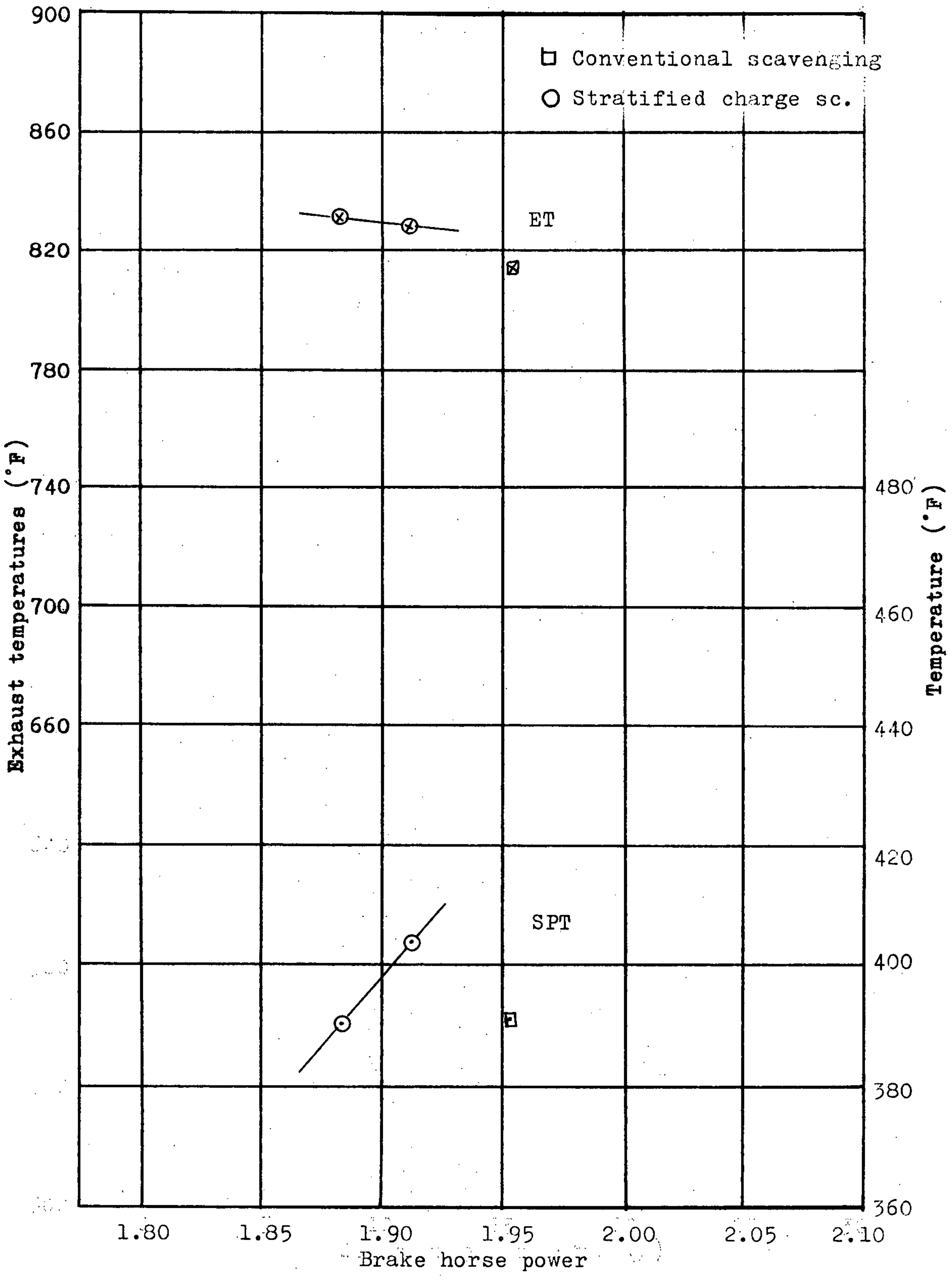

Figure 86. Spark plug & exhaust temperature curves with TP 50%, NS 4900, from sheet 15.

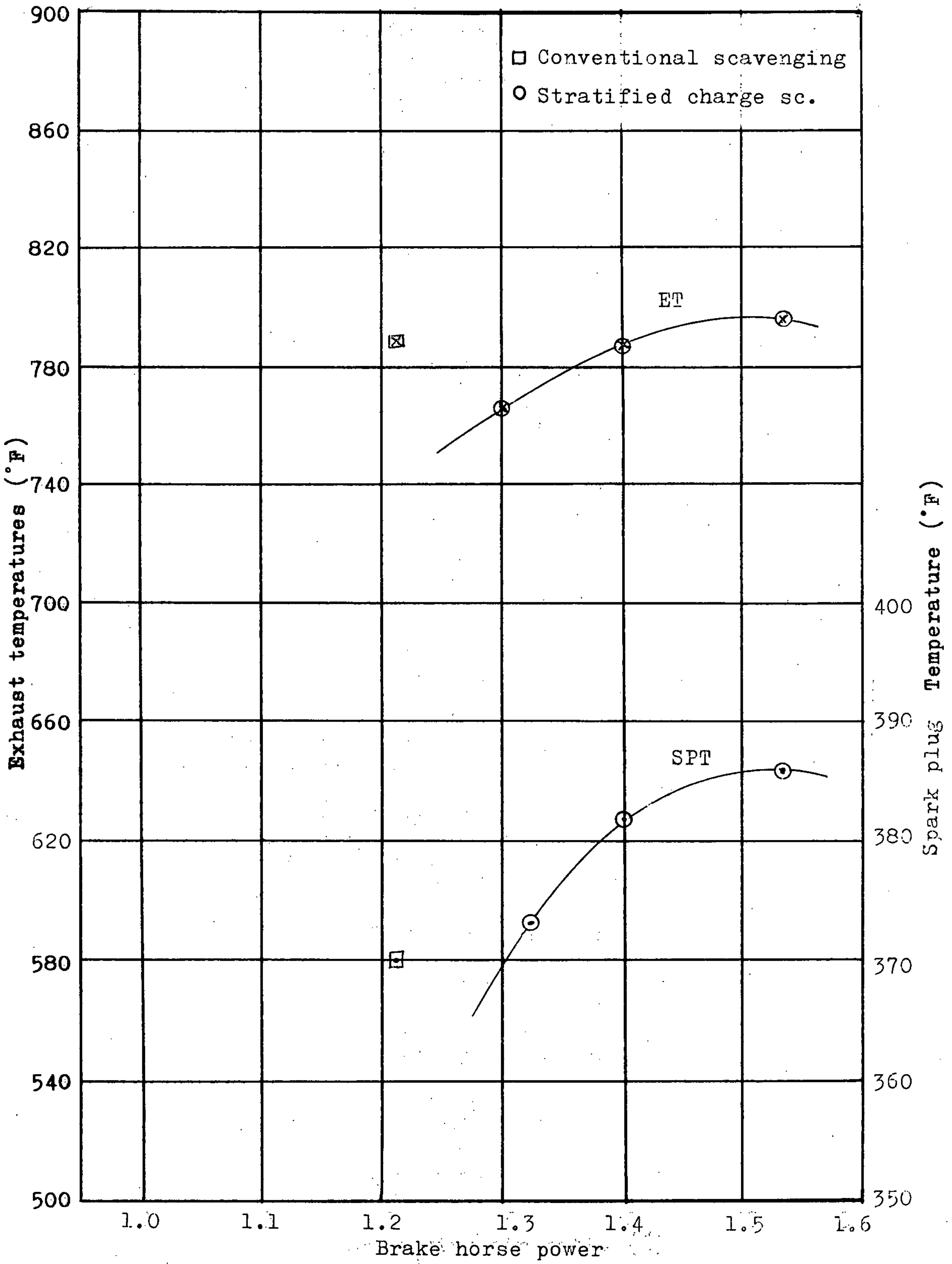

Figure 87. Exhaust and spark plug temperature curves with TP 50%, NS 4500, from sheet 17.

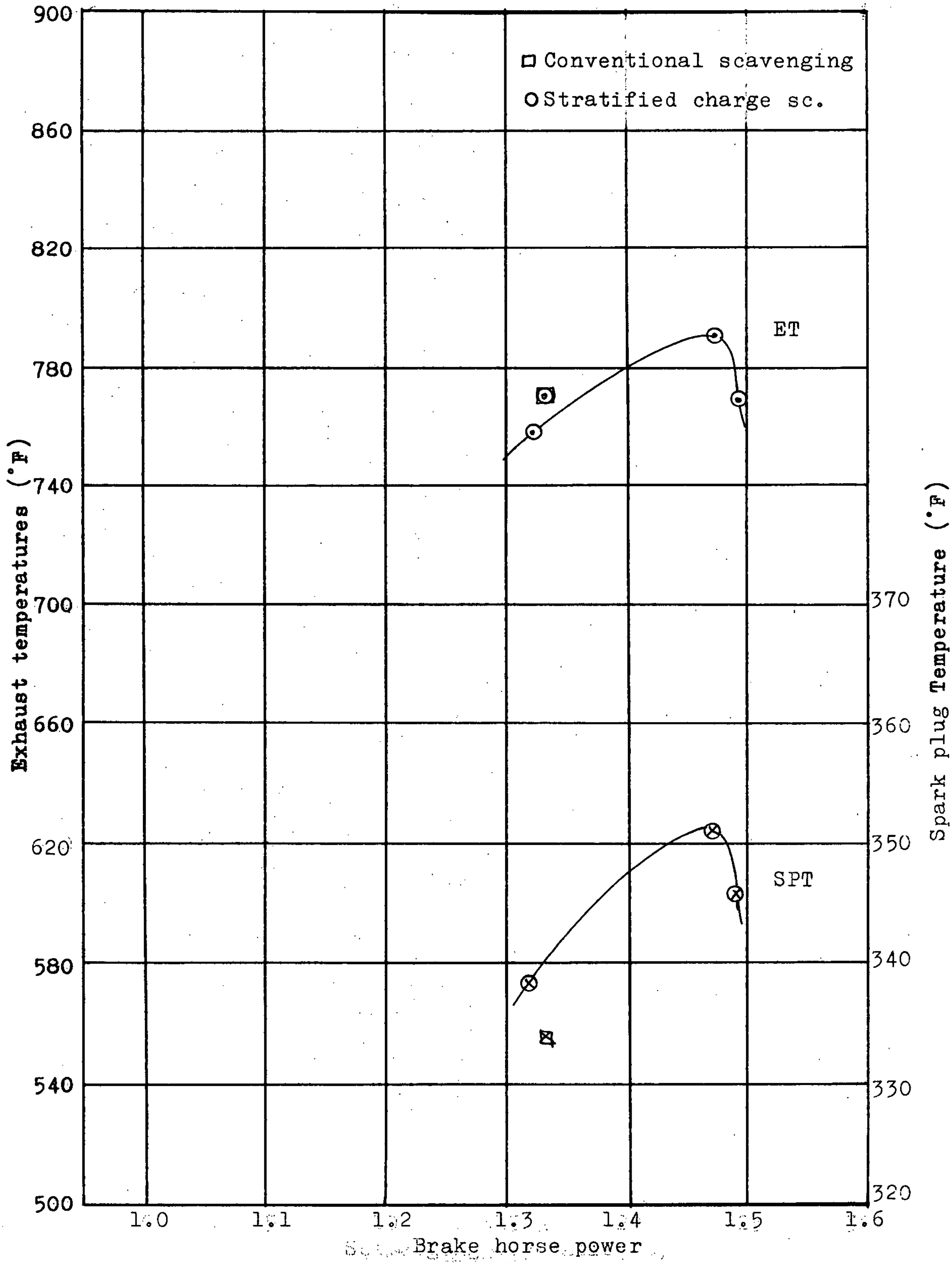

Figure 88. Exhaust & spark plug temperature curves with TP 50%, NS 4700, from sheet 20.

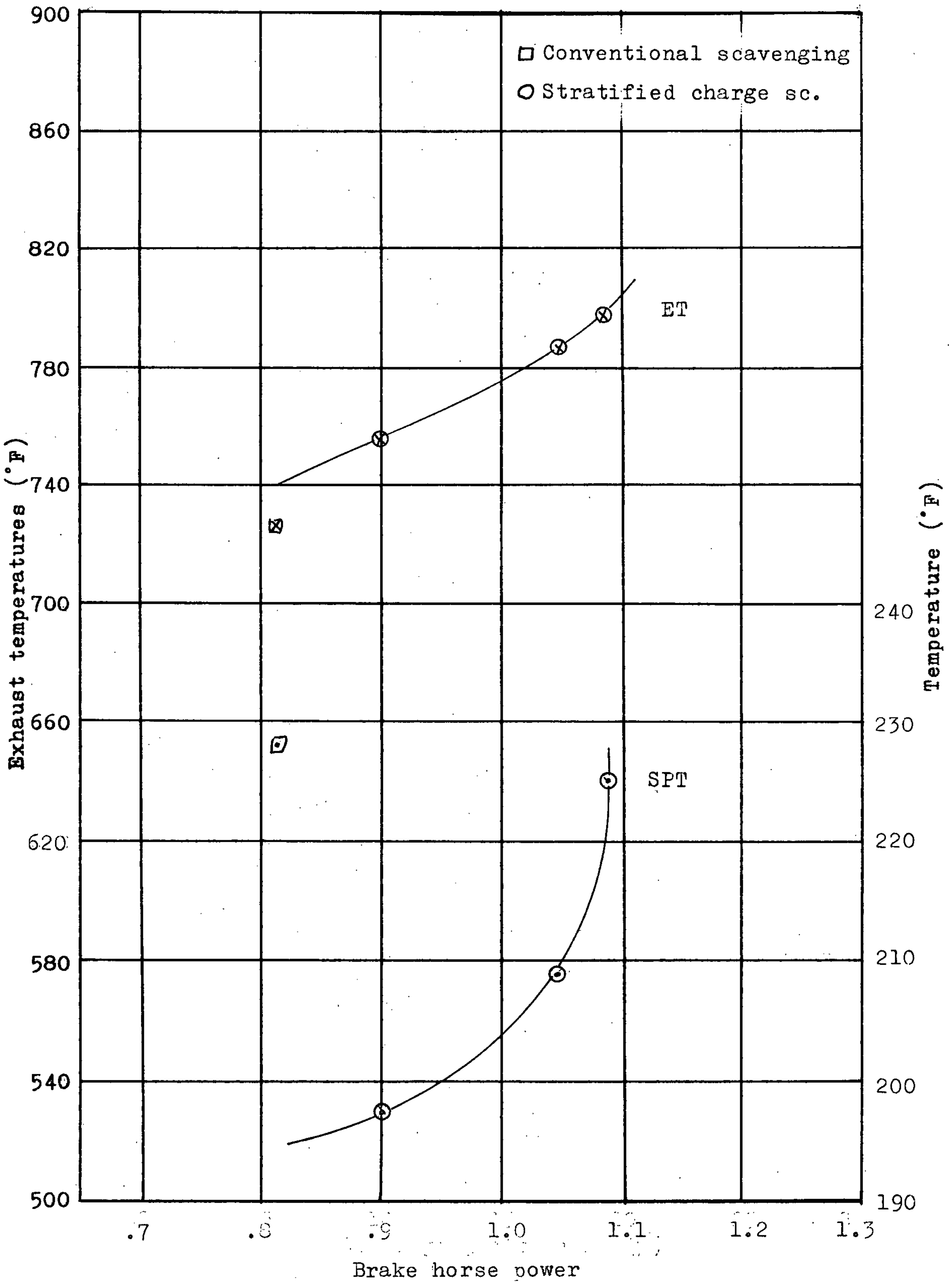

Figure 89. Exhaust and spark plug temperature curves with TP 60%, NS 5000, from sheet 21.

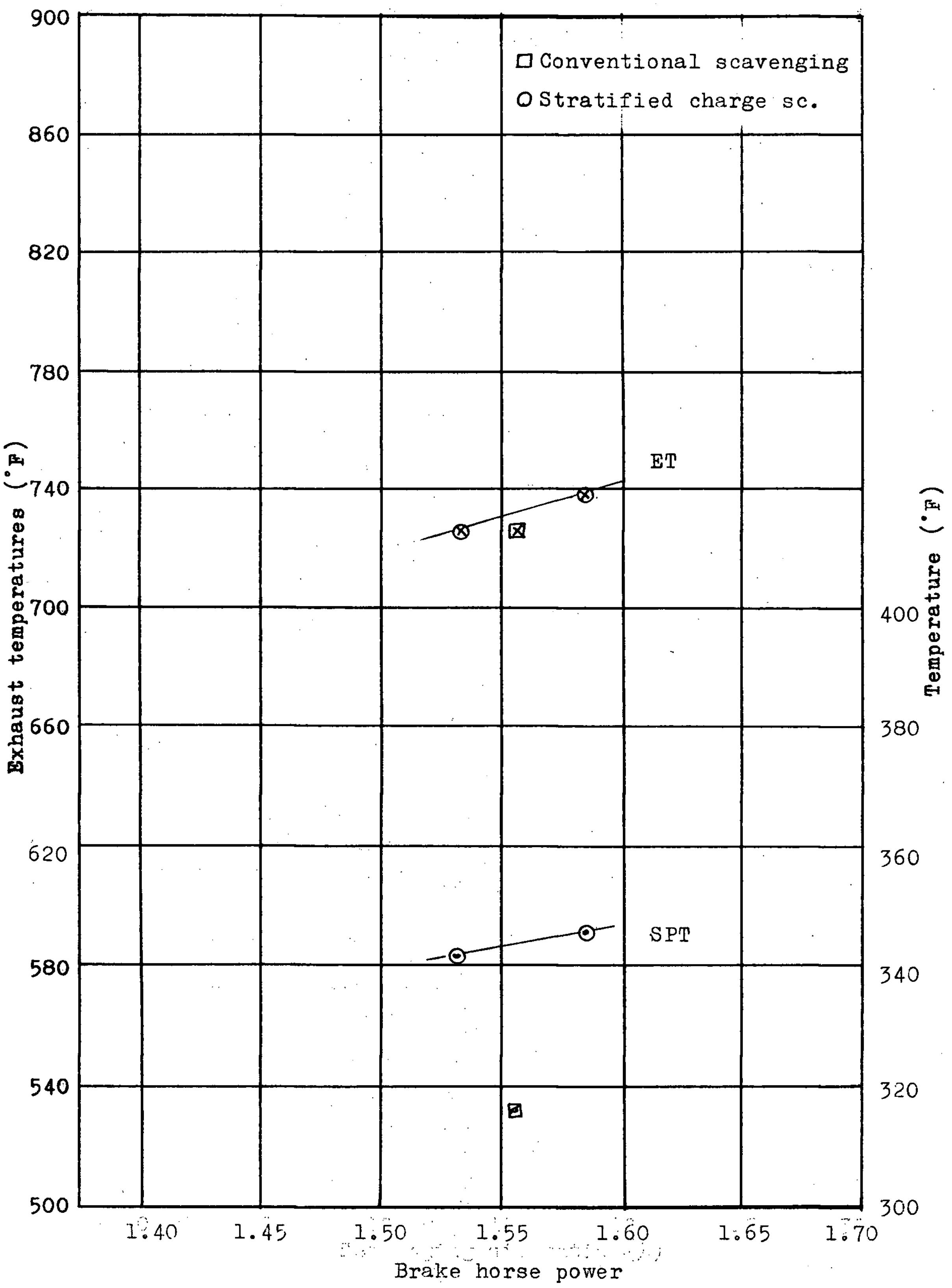

Figure 90. Exhaust & spark plug temperature curves with TP 70%, NS 4300, from sheet 22.

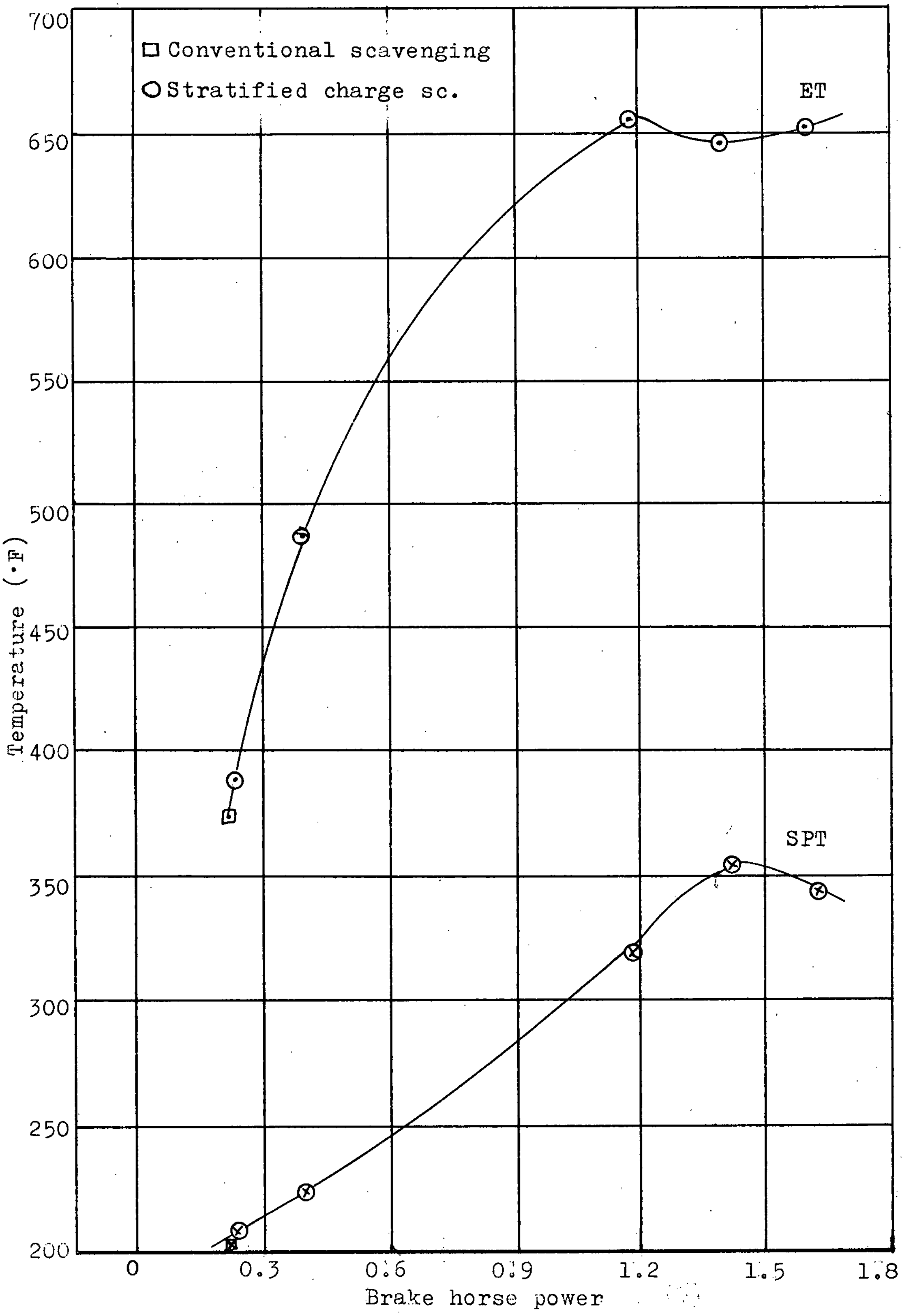

Figure 91. Exhaust & spark plug temperature curves with TP 30%, from sheet 23.

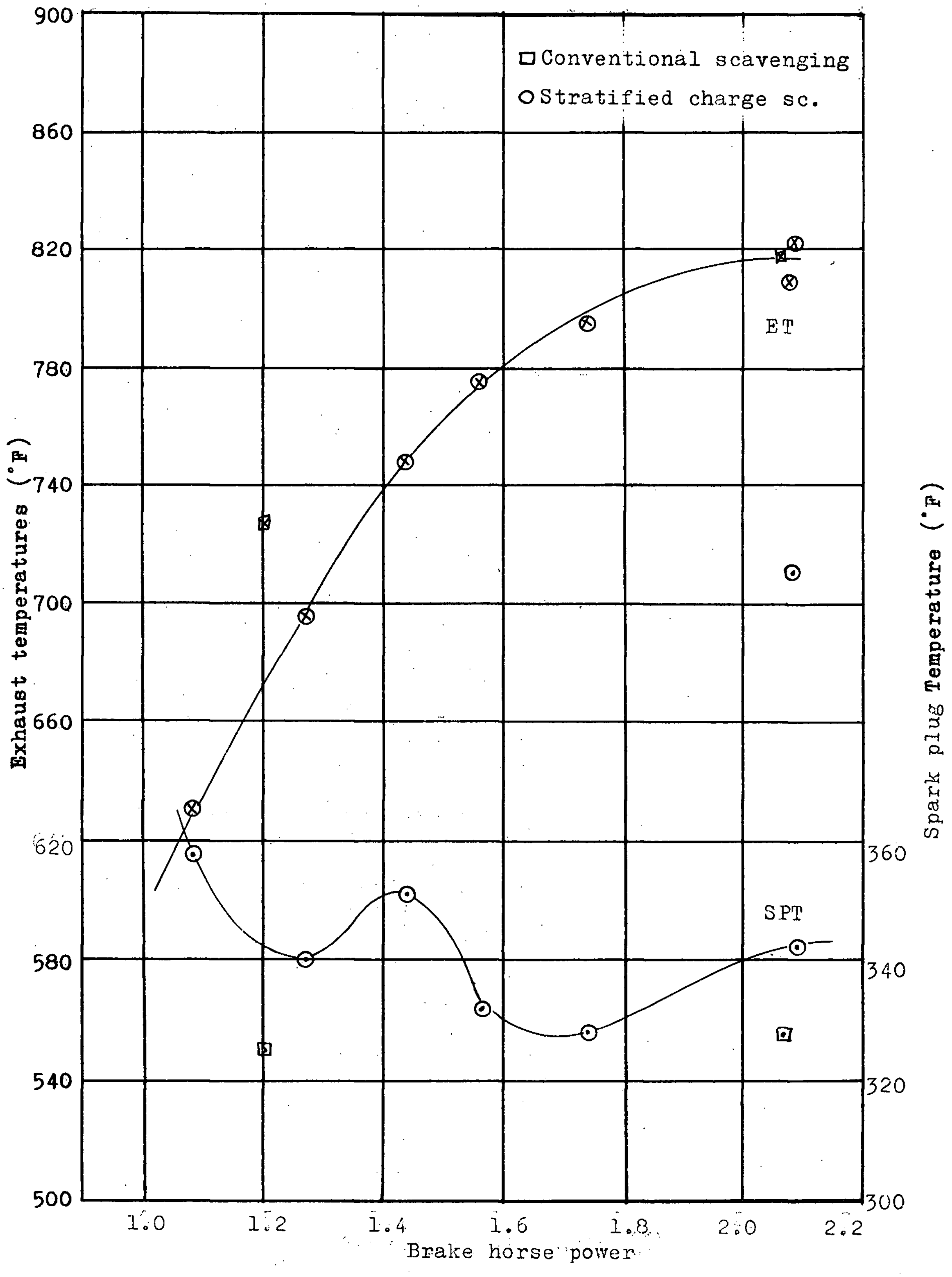

Figure 92. Temperature curves with VP 100%, from sheet 24.

Chapter VI. SUMMARY AND CONCLUSIONS

Chapter VI. SUMMARY AND CONCLUSIONS

From the results of the tests described, it is obvious that there are great fluctuations in the operating characteristics of the engine. For a small high-speed 2-stroke cycle engine operating at part throttle, this is to be expected. The conclusions which can be drawn from experiments on such an engine can only be generalizations and may not hold true in all cases.

The greatest problem to consistent results seemed to be the variation in the air-fuel ratio through the carburetor. The continuous CO_2 recorder showed fluctuations of about $\pm$ 1-2% CO_2 during all of the tests. Average CO_2 content was usually around 6%. As the results of the tests show, the air-fuel ratio through the carburetor was not always consistent even when the intake was not choked. A contributing factor to this may have been the variation in the momentum of the air mass entering the carburetor from a straight pipe.

In the tests minimum specific fuel consumption was usually obtained at maximum overall air-fuel ratio as the ratio was generally below the stoichiometric ratio at part throttle. In the stratified charge scavenging system, the additional air entering through the scavenging air valve increases the overall air-fuel ratio. As the air-fuel mixture enters the cylinder after the scavenging air, more of the mixture is trapped in the cylinder. These two factors increase the brake thermal efficiency. As the charge trapping efficiency decreases, more air escapes with the exhaust gases so that the exhaust is cooler and as the fuel trapping efficiency increases, less fuel is available for afterburning in the exhaust pipe. The higher overall air-fuel ratio results in more efficient combustion raising the overall combustion temperature and negating the cooling effect of the excess air in the exhaust gases.

It is quite satisfactory to keep the valve at a constant position and vary the throttle setting to change the power produced, as is done in the conventional throttling system. But as the throttle is closed, the scavenging air flow increases, so that it is necessary to limit the scavenging air ratio to obtain efficient operation at low power requirements. On the other hand the scavenging air valve is closed when maximum power is required with no deterrent effects to peak engine output.

Generally the stratified charge scavenging system shows definite reductions in the specific fuel consumption, especially at low power requirements. The specific fuel consumption decreased 10.6% at 2.1 hp and 34.8% at 1.2 hp during one test. The normal decrease was between 5 and 15%. The temperatures are not as consistent although the exhaust temperatures are generally lower, and the cylinder head and spark plug temperatures generally higher. Nevertheless much work is still required to adapt the principle of stratified charge scavenging to a practical, trouble-free, consistent system.

The following list contains some of the work still to be done on similar engines to obtain more detailed information about the effects of stratifying the charge for more efficient combustion at part throttle.

(1) comparison of the performance with a scavenging air valve in each of the two passageways instead of only one;

(2) comparison of the performance at various valve intake temperatures by preheating the air entering the valve by means of a heat exchanger from the exhaust gases or by an external heat source;

(3) compare the actual scavenging and trapping efficiency with the theoretical. This would require a method of determining the amount of air and fuel which has escaped into the exhaust pipe;

(4) method of determining the degree of mixing between the scavenging air and the carburetted air-fuel mixture, and the degree of mixing of the exhaust gases with the fresh charge. This may require studies in models;

(5) more tests with the present system with a carburetor that allows closer control of the air-fuel ratio.

Appendix I. GLOSSARY

GLOSSARY

(1) Scavenging air valve, (SAV): valve arrangement of reed valve and cock control valve that allows additional air to enter passageway to ports;

(2) Scavenging air: air that enters through the scavenging air valve;

(3) (Carburetted) air-fuel mixture: mixture of fuel and air that enters crankcase through the carburetor;

(4) Charge: Fresh gases entering the engine or cylinder and made up of scavenging air and carburetted air-fuel mixture;

(5) Scavenging air ratio, (SAR): defined as the mass ratio of scavenging air supplied to total charge supplied;

(6) *Scavenging efficiency, (SE): defined as the mass ratio of charge retained to the ideal mass retained;

(7) Scavenging ratio, (R): defined as the ratio of mass of charge supplied to ideal mass;

(8) *Trapping efficiency, (TE): defined as mass ratio of charge retained to charge supplied;

(9) *Fuel scavenging efficiency, (FSE): defined as the ratio of mass of carburetted air-fuel mixture retained to ideal mass that the cylinder should have retained;

(10) Fuel scavenging ratio, (FR): defined as the mass ratio of carburetted air-fuel mixture supplied to ideal mass that the cylinder should retain;

(11) *Fuel trapping efficiency, (FTE): defined as mass ratio of carburetted air-fuel mixture retained to carburetted air-fuel mixture supplied.

*NOTE: For ideal efficiencies, perfect mixing is assumed.

Appendix II. SYMBOLS, ABBREVIATIONS AND UNITS

SYMBOLS, ABBREVIATIONS AND UNITS

The following is a list of symbols and abbreviations used throughout the text. A symbol not listed is defined on introduction. The units refer to the values as specified in the tables of data and results in Appendix IV and Appendix V.

ACP	- average of peak combustion pressures (psi);
ACPD	- average deviation of peak combustion pressures from ACP (psi);
AF	- overall air-fuel ratio;
AFC	- air-fuel ratio of mixture inspired through carburetor;
BARO	- barometric pressure (in. of Hg);
BHP	- brake horse power;
BMEP	- brake mean effective pressure (psi);
BTE	- brake thermal efficiency (%);
CAT	- cooling air temperature ($^{\circ}$F);
CHT	- cylinder head temperature ($^{\circ}$F);
CIT	- carburetor intake temperature ($^{\circ}$F);
CIPD	- carburetor intake pressure drop (in. of water);
COUNT	- number of counts of revolution counter in time TS;
DBT	- dry bulb temperature ($^{\circ}$F);
DYNA	- dynamometer torque reading
ET	- exhaust temperature in pipe (F);
ETIM	- exhaust temperature in muffler (F);
EPD	- exhaust pressure drop (in. of water);
FC	- fuel consumption (ppm);
FR	- fuel scavenging ratio;
FSE	- fuel scavenging efficiency (%);
FTE	- fuel trapping efficiency (%);
IFSE	- ideal fuel scavenging efficiency (%);
IFTE	- ideal fuel trapping efficiency (%);

ISE - ideal charge scavenging efficiency (%);

ITE - ideal charge trapping efficiency (%);

JET - 16 times fraction of one turn that jet is open;

MIT - flowmeter inlet temperature (^{o}F);

NS - nominal speed from tachometer (rpm);

OAF - overall air-fuel ratio;

IOD - nozzle (orifice) inlet density (lb per cubic ft.);

OIT - nozzle (orifice) inlet temperature (^{o}F);

OIPR - nozzle (orifice) intake pressure (rise) above BARO (in. Hg);

OPD - nozzle (orifice) pressure drop (in. of water);

PT - passageway air temperature (^{o}F); (RPT - right and LPT - left);

QM - volume reading of flowmeter in time TM (cubic ft.);

R - scavenging ratio;

RT - room temperature (^{o}F);

REYN - Reynolds number;

S - speed (rpm);

SAC - specific air consumption (lb per bhp-hr.);

SAR - scavenging air ratio;

SFC - specific fuel consumption (lb per bhp-hr.);

SPT - spark plug temperature (^{o}F);

SP GR - specific gravity;

TAC - total mass air consumption (ppm);

TF - time for WF lb fuel flow (min);

TM - time for QM cubic ft air flow through meter (min);

TORQ - torque (lb-ft.);

TP - throttle position - ratio to full open (%);

TRIAL - reference to data sheets taken during the tests;

TS - time for COUNT counts on rev. counter (min.);

VDI - Verein Deutscher Ingenieure - nozzle standards;

VE - volumetric efficiency (%);

VIPD - valve intake pressure drop (in. of Hg);

VIT - valve intake air temperature (^{o}F);

VP - valve position - ratio to full open (%);

WBT - wet bulb temperature (^{o}F);

WF - fuel consumed in time TF (Kilogram or cubic centimeters);

WFTC - weight air flow through carburetor (ppm);

WFTV - weight air flow through scavenging air valve (ppm);

Appendix III. EFFICIENCY DERIVATIONS WITH PERFECT MIXING

EFFICIENCY DERIVATIONS WITH PERFECT MIXING

Derivation of the relationships between scavenging efficiency, trapping efficiency, and scavenging ratio with perfect mixing requires the following assumptions:

(1) Scavenging air and the carburetted air-fuel mixture do not mix prior to cylinder entry;

(2) the scavenging air mixes completely with the exhaust gases as soon as the air enters, and then the carburetted air-fuel mixture mixes completely with the resultant air-exhaust mixture as soon as the fuel-air mixture enters the cylinder;

(3) the exhaust, air, and air-fuel mixture are at the temperature of the mixture in the passageway;

(4) the exhaust and air-fuel mixture have the same molecular weight as air;

(5) the pressure is assumed constant and equal to the average exhaust pressure.

<u>For an homongeneous charge</u>

Let:

y = volumetric fraction of charge in the cylinder at any instant to total volume of cylinder,

v = volume of mixture that has flowed into cylinder,

V = cylinder volume.

The volumetric increase in the mixture is equal to the volume of fresh charge which has flowed in, minus the volume of the fresh charge which has escaped. In mathematical terms this relationship may be expressed as:

$$Vdy = dv - ydv$$

$$dy = \frac{(1-y)dv}{V}$$

Integration yields:

$$y = 1 - e^{-\frac{u}{V}} \qquad (1)$$

By definitions:

$$R = \frac{TAC}{(PID)(S)(V)} \tag{2}$$

$$FR = \frac{WFTC}{(PID)(S)(V)} \tag{3}$$

$$TE = \frac{SE}{R} \tag{4}$$

$$FTE = \frac{FSE}{R} \tag{5}$$

$$SAR = \frac{WFTV}{TAC} \tag{6}$$

For an homogeneous mixture:

$$WFTC = TAC$$

$$SAR = 0$$

$$y = SE = FSE$$

$$\frac{u}{V} = R = FR = \frac{TAC}{(PID)(S)(V)}$$

Substituting these values into equations 1, 4 and 5 gives:

$$y = SE = 1 - e^{-R} \tag{7}$$

$$FSE = 1 - e^{-R} \tag{8}$$

$$TE = \frac{1 - e^{-R}}{R} \tag{9}$$

$$FTE = \frac{1 - e^{-R}}{R} \tag{10}$$

For a stratified charge - symmetrical admission into cylinder

The idealized equations 7 and 9 still hold for the total amount of scavenging air and carburetted air-fuel mixture (charge) supplied. To determine the equations for scavenging efficiency and trapping efficiency of only the carburetted air-fuel mixture v and y must be defined as follows:

y = volumetric fraction of the carburetted air-fuel mixture in the cylinder to total cylinder volume,

v = volume of the carburetted air-fuel mixture that has flowed in up to time "t".

With these definitions

$$y = FSE$$

$$\frac{v}{V} = FR$$

Equation 1 becomes

$$FSE = 1 - e^{-FR} \tag{11}$$

and $$FTE = \frac{FSR}{FR} = \frac{1 - e^{-FR}}{FR} \tag{12}$$

To determine the value of FR, consider

v = total volume of charge flowed in per cycle minus the volume of scavenging air flowed in during the same cycle

$$= \frac{TAC}{(PID)(S)} - \frac{WFTV}{(PID)(S)}$$

$$= \frac{1}{(PID)(S)} (TAC - WFTV)$$

Then

$$\frac{v}{V} = \frac{1}{(V)(PID)(S)} (TAC - WFTV) \tag{13}$$

By equation 6,

$$WFTV = (SAR)(TAC)$$

Substituting into equation 13 yields:

$$\frac{v}{V} = \frac{1}{(V)(PID)(S)} (TAC - (TAC)(SAR))$$

$$= \frac{TAC}{(V)(PID)(S)} \left(1 - SAR \right)$$

Using equation 2,

$$\frac{u}{V} = R (1 - SAR) \tag{14}$$

Substituting into equation 11 and 12

$$FSE = 1 - e^{-R(1-SAR)} \tag{15}$$

$$FTE = \frac{1 - e^{-R(1-SAR)}}{R(1-SAR)} \tag{16}$$

For stratified charge - non symmetrical charge admission

If all of the scavenging air enters one scavenging air valve and the assumption is made that this air enters the cylinder through only one port (in a symmetrical 2-ported engine), say the right one, then equations 15 and 16 mustbbe modified although equations 7 and 9 are still valid.

Consider that the charge entering the left port is separated from the charge entering the right port, that is, divide the cylinder into two equal volumes and analyse them separately. The equations for perfect mixing for the left port with no scavenging air, will be the same as for ordinary scavenging, that is, equations 15 and 16 cancel down to equations 8 and 10, as SAR is zero.

For left volume:

$$(FR) = R = \frac{TAC}{(S)(PID)V}$$

$$(FSL)_L = 1 - e^{-R}$$

$$(FTE)_L = \frac{1 - e^{-R}}{R}$$

The efficiencies for the right volume containing all the scavenging air, can be expressed by modified forms of equation 15 and 16. For right volume:

$$(SAR)_R = \frac{WFTV}{(WFTV + WFTC)\frac{1}{2}}$$

$$= \frac{2(WFTV)}{TAC}$$

$$= 2(SAR) \tag{17}$$

$$(FR)_R = \frac{\left(\frac{TAC}{2} - WFTV\right)}{(S)(PID)(V)\,\frac{1}{2}}$$

But $WFTV = TAC(SAR)$

Therefore

$$(FR)_R = \left(\frac{TAC}{2} - TAC(SAR)\right) \frac{2}{(S)(PID)(V)}$$

$$= \frac{2(TAC)}{(S)(PID)(V)} \left(\tfrac{1}{2} - SAR\right)$$

$$= R(1 - 2\,(SAR)) \tag{18}$$

Equation 1 checks with equation 15 if the SAR in equation 15 is taken as $(SAR)_R$.

Substituting equation 18 into equation 11 and 12

$$(FSE)_R = 1 - e^{-R(1 - 2\,(SAR))}$$

$$(FTE)_R = \frac{1 - e^{-R(1 - 2\,(SAR))}}{R(1 - 2\,(SAR))}$$

For the overall effect of both volumes of the cylinder, the efficiencies must be averaged

$$FSE = \frac{(FSE)_L + (FSE)_R}{2}$$

$$FSE = 1 - \tfrac{1}{2}\left[e^{-R(1 - 2\,(SAR))} + e^{-R}\right] \tag{19}$$

$$FTE = \frac{(FTE)_L + (FTE)_R}{2}$$

$$FTE = \frac{1 - e^{-R(1 - 2\,(SAR))}}{2(R)(1 - 2\,(SAR))} + \frac{1 - e^{-R}}{2(R)} \tag{20}$$

Charge efficiency equations 7 and 9 still hold:

$$SE = 1 - e^{-R} \tag{7}$$

$$TE = \frac{1 - e^{-R}}{R} \tag{9}$$

Where R and SAR are as previously defined, i.e.,

$$R = \frac{TAC}{(PID)(S)(V)} \qquad (2)$$

$$SAR = \frac{WFTV}{TAC} \qquad (6)$$

Equations 19, 20, 7, 9, 2, and 6 are the ones used for approximating the scavenging efficiency and trapping efficiency of the 2-stroke engine stratified charge scavenging system.

Appendix IV. TABLES OF OBSERVED RESULTS

TABLE II. OBSERVED EXPERIMENTAL DATA

TRIAL	JET	NS	TP	BARO	VP	ACP	ACPD	COUNT	TS	DYNA	WF	TF	OPD	CIPD
15-2	14	4900	50	29.67	0	316.0	18.90	59756	35	1.64	31.3cc	1.00	4.46	.06
5					100	307.0	21.97	55600	33	1.65	31.4cc	1.00	3.87	.04
8					50	303.2	21.57	60648	36	1.63	30.4cc	1.00	3.94	.04
17-2	16	4500	50	29.94	0	271.3	17.38	53599	35	1.16	22.5cc	1.00	2.15	.02
5					70	292.1	24.54	48810	31	1.29	23.3cc	1.00	1.88	.02
7					100	303.0	28.78	57809	31	1.37	23.8cc	1.00	1.83	.02
11					40	279.7	24.30	51067	33	1.24	21.5cc	1.00	2.02	.02
19-2	17	5300	82	29.90	0	-	-	54860	30	2.11	.760	31.00	6.24	.07
5					100	-	-	58259	32	2.09	.740	31.05	6.05	.06
20-2	20	4500	50	29.94	100	275.2	18.77	70928	44	1.33	1.240	49.64	3.00	.04
5					0	268.0	15.21	62458	40	1.23	.980	39.52	3.37	.03
7					50	277.6	19.87	32800	21	1.22	.420	17.70	2.98	.03
11					70	278.2	18.17	48171	30	1.35	.710	29.70	2.88	.02
21-2	F.A.	5000	60	30.02	0	251.2	12.14	56714	33	.69	.800	32.62	3.18	.04
4					100	260.2	9.98	29150	16	.86	.390	14.92	2.68	.03
5					100	254.8	10.61	76450	42	.86	.980	37.90	2.68	,03

TABLE II. (Con't)

TRIAL	JET	NS	TP	BARO	VP	ACP	ACPD	COUNT	TS	DYNA	WF	TF	OPD	CIPD
21-8	F.A.	5000	60	30.02	70	249.8	9.65	53953	30	.84	.669	26.22	2.74	.03
-11					50	248.9	10.20	54667	31	.74	.64	25.66	2.88	.03
22-1	F.A.	4300	70	29.84	0	258.0	16.30	26340	18	1.57	.30	17.43	2.31	.01
-5					55	279.2	21.46	51350	35	1.53	.54	34.45	1.89	.01
-8					100	292.6	18.12	72150	48	1.55	.73	42.88	1.69	.01
23-2	F.A.	2800	30	29.81	0	189.7	11.98	33482	36	.33	.23	27.01	.36	.00
-5					40	190.2	7.96	24513	25	.35	.16	20.38	.27	.00
-7					55	193.7	6.42	24060	21	.49	.10	11.87	.23	.00
-9					100	298.2	16.26	1706	1	1.39	.10	8.87	.36	.00
-12					100	244.2	27.10	57269	34	1.25	.35	30.10	.30	.00
-15					80	-	-	50499	31	1.06	.19	16.45	.36	.00
24-2	F.A.	5200	100	29.80	0	263.3	18.26	53229	30	1.70	.58	25.38	4.04	.02
-5			100		100	287.4	24.72	57019	32	1.71	.58	26.50	3.75	.02
-8			70		100	246.6	19.12	53110	31	1.48	.62	28.03	3.05	.02
-11			70		100	280.1	23.42	53512	30	1.70	.57	27.71	3.71	.02
-14			60		100	239.0	14.31	53399	32	1.36	.64	29.30	2.63	.02

TABLE II. (Con't)

TRIAL	JET	NS	TP	BARO	VP	ACP	ACPD	COUNT	TS	DYNA	WF	TF	OPD	CIPD
24-17	F.A.	5200	50	29.80	100	241.6	18.62	40194	25	1.30	.40	24.00	1.57	.01
-20			40			225.0	14.70	31986	21	1.21	.31	22.60	.83	.01
-23			30	29.87		248.8	20.18	48720	34	1.10	.33	30.77	.40	.01
-26			23			197.3	19.31	27295	29	.33	.28	24.12	.06	.01
-29			50		0	-	-	29990	20	1.17	.39	20.30	2.46	.01
25-2	F.A.	4500	100	29.86	100	-	-	52772	34	1.65	.57	33.01	2.36	.02
-5			70			-	-	35020	23	1.43	.36	20.25	2.09	.02
-8			65			-	-	44116	29	1.17	.46	24.35	2.03	.02
-11			50			-	-	38517	25	.96	.33	21.15	1.08	.01
-14			40			-	-	41950	28	.73	.25	23.71	.40	.00
-17			100		0	-	-	20540	14	.92	.15	12.15	1.47	.01
14-8	14	4700	100	29.73	0	-	-	32550	40	1.12	22.1cc	2.00	.88	.00
-12			100		100	-	-	30630	34	1.11	20.6cc	2.00	.38	.00
25-20	F.A.	4500	100	29.80	0	-	-	42483	27	1.98	.43	22.95	4.02	.01
-23			60		0	-	-	40851	26	1.65	.54	25.80	3.02	.01
-26			50		0	-	-	25464	17	1.04	.32	18.52	1.70	.01

TABLE III. OBSERVED EXPERIMENTAL DATA

TRIAL	QM	TM	OIPR	VIPD	EPD	CAT	CIT	CHT	SPT	VIT	ET	ETIM	RPT	LPT	OIT	MIT	DBT	WBT
15-2	0	0	4.46	0	6.2	199	84	121	391	163	815	568	278	-	82	76	75	64
5	11.93	30	3.87	1.3	5.5	202	83	125	404	136	829	581	288	-	82	76		
8	6.99	38	4.00	.9	5.4	201	83	125	390	183	831	584	288	-	82	76		
17-2	0	0	2.15	.0	2.5	202	75	108	370	156	788	568	307	-	74	69	68	60
5	16.49	25	1.88	.8	2.6	205	76	113	383	127	787	571	313	-	74	69		
7	33.66	31	1.83	1.2	3.1	206	76	113	386	112	798	578	316	-	74	69		
11	5.99	32	2.02	.2	2.3	205	77	108	373	179	766	565	316	-	75	70		
19-2	0	0	6.25	0	6.6	142	79	362	461	160	896	614	294	301	78	74	71	60
5	-4.34	27	6.07	.9	5.4	141	80	358	457	230	900	617	299	307	78	74		
20-2	35.22	43	3.1	.10	3.3	126	72	294	351	100	791	553	282	281	72	68	66	58
5	0	0	3.6	0	2.5	124	72	282	334	146	770	545	270	270	72	68		
7	5.26	16	3.1	.03	2.3	124	72	284	337	139	759	541	277	276	72	68		
11	20.69	30	2.9	.06	2.5	124	72	288	347	108	770	545	275	279	72	68		
21-2	0	0	3.3	0	2.4	140	73	252	228	149	726	505	249	244	72	67	68	59
4	15.93	14	2.7	.14	2.5	143	74	259	232	93	796	543	254	252	73	67		
5	35.97	34	2.7	.14	2.9	143	74	259	226	94	798	546	254	252	72	67		

TABLE III. (Con't)

TRIAL	QM	TM	OIPR	VIPD	EPD	CAT	CIT	CHT	SPT	VIT	ET	ETIM	RPT	LPT	OIT	MIT	DBT	WBT
21-8	20.03	23	2.74	.12	3.5	142	73	257	209	97	789	539	253	251	72	67	68	59
-11	13.08	29	2.95	.07	3.0	141	72	251	198	124	757	512	251	247	72	67		
22-1	.00	0	2.40	.00	2.9	134	75	307	316	163	727	522	295	284	72	71	69	62
-5	14.27	30	1.91	.05	2.9	137	75	316	342	134	726	519	305	295	74	71		
-8	44.70	38	1.72	.15	3.2	136	75	316	346	100	738	523	303	282	74	71		
23-2	.00	0	.36	.00	1.5	111	70	210	201	127	372	303	227	228	69	68	68	60
-5	7.67	24	.29	.04	1.6	114	70	220	206	120	388	319	245	240	69	68		
-7	15.91	22	.26	.07	1.4	122	72	242	223	111	485	391	282	264	71	68		
-9	2.49	1	.37	.47	3.9	126	73	305	343	85	654	480	314	280	71	68		
-12	79.55	32	.30	.46	3.5	130	74	306	353	86	643	468	326	286	72	68		
-15	60.18	30	.37	.31	3.4	153	73	283	317	84	656	468	316	286	72	69		
24-2	.00	00	4.15	.00	5.7	182	68	300	329	145	817	553	262	265	68	66	65	59
-5	7.14	29	3.81	.04	5.9	186	68	307	343	135	821	556	274	277	70	67		
-8	15.61	28	3.12	.02	5.3	178	68	282	329	105	796	546	270	271	70	67		
-11	7.38	30	3.84	.04	5.9	198	69	322	406	133	807	553	275	282	71	67		
-14	24.66	33	2.69	.09	4.4	179	68	275	333	98	776	542	271	269	71	67		

TABLE III. (Con't)

TRIAL	QM	TM	OIPR	VIPD	EPD	CAT	CIT	CHT	SPT	VIT	ET	ETIM	RPT	LPT	OIT	MIT	DBT	WBT
24-17	28.96	27	1.59	.14	3.1	188	67	286	352	89	749	531	296	280	71	67	69	65
-20	35.46	25	.86	.19	2.3	186	66	283	341	83	696	505	309	275	70	67		
-23	61.09	35	.41	.26	1.7	195	71	294	359	85	632	465	317	276	70	67		
-26	53.86	26	.07	.34	2.4	151	72	212	243	80	418	329	229	199	71	67		
-29	.00	00	2.48	.00	3.1	182	74	268	325	135	724	529	269	263	72	69		
25-2	22.46	31	2.41	.09	3.4	164	72	333	430	110	738	541	294	291	72	68	69	61
-5	17.13	22	2.20	.09	3.1	152	72	299	370	103	742	540	294	283	73	69		
-8	20.25	27	2.09	.09	3.1	143	73	280	335	103	722	532	283	275	74	70		
-11	25.71	25	1.10	.12	2.1	143	71	283	345	92	670	-	296	274	74	71		
-14	33.20	25	.41	.17	1.1	146	70	286	346	85	530	-	300	279	74	71		
-17	.00	00	1.50	.00	1.1	148	69	288	352	131	563	-	286	280	74	73		
14-8	.00	00	.92	.00	2.6	232	85		382	167	591	439	292	-	80	75	74	64
-12	45.16	34	.40	.00	2.9	226	87	134	354	111	591	444	303	-	79	75		
25-20	0	0	4.30	.01	4.8	186	76	366	420	154	747	531	300	307	74	71	69	62
-23	0	0	3.18	.01	3.7	158	77	302	344	146	742	510	284	283	74	71		
-26	0	0	1.90	.01	2.5	146	77	266	319	139	680	474	288	285	74	71		

Appendix V. TABLES OF CALCULATED RESULTS

TABLE IV. REDUCED EXPERIMENTAL RESULTS

TRIAL	BHP	SFC	SAR	TAC	IFTE	BMEP	TORQ	VE	BTE	ISE	SAC	ITE	AFC	OAF	FC
15-2	1.95	1.55	.0	.566	75.1	28.5	2.18	66.9	8.10	45.2	17.4	75.1	11.2	11.2	.0503
5	1.91	1.58	5.1	.556	75.9	28.3	2.17	67.5	7.93	45.5	17.5	74.9	10.5	11.0	.0504
8	1.89	1.55	2.4	.544	75.8	27.9	2.14	66.1	8.10	44.8	17.3	75.3	10.9	11.1	.0488
17-2	1.22	1.77	.0	.395	79.2	19.9	1.52	53.9	7.06	38.4	19.4	79.2	10.9	10.9	.0361
5	1.40	1.60	11.7	.418	79.8	22.1	1.69	56.0	7.82	39.6	17.9	78.5	9.9	11.2	.0373
7	1.53	1.50	16.0	.437	81.1	23.8	1.82	57.7	8.38	40.5	17.1	78.0	9.6	11.4	.0382
11	1.32	1.58	3.5	.396	79.7	21.3	1.63	54.2	7.97	38.6	18.0	79.1	11.0	11.4	.0346
19-2	2.65	1.22	.0	.679	72.4	36.2	2.77	76.0	10.27	49.6	15.4	72.4	12.6	12.6	.0540
5	2.62	1.20	-1.8	.657	----	35.8	2.74	74.6	10.42	48.9	15.1	----	12.7	----	.0525
20-2	1.47	2.24	11.5	.530	77.9	22.8	1.75	66.3	5.60	44.9	21.6	75.3	8.5	9.6	.0550
5	1.33	2.46	.0	.498	76.2	21.3	1.63	63.5	5.09	43.5	22.5	76.2	9.1	9.1	.0546
7	1.32	2.38	5.0	.491	77.4	21.1	1.61	63.2	5.28	43.4	22.3	76.2	8.92	9.4	.0523
11	1.49	2.12	10.1	.511	78.1	23.1	1.77	63.9	5.92	43.7	20.6	76.0	8.7	9.7	.0527
21-2	.82	3.98	.0	.484	79.1	11.8	.90	54.1	3.15	38.6	35.6	79.1	8.9	8.9	.0540
4	1.09	3.17	16.2	.527	81.7	14.9	1.14	56.1	3.95	39.6	29.0	78.5	7.6	9.1	.0576
5	1.08	3.18	15.1	.523	81.6	14.7	1.13	55.6	3.95	39.5	29.1	78.7	7.8	9.2	.0570

TABLE IV. (Con't)

TRIAL	BHP	SFC	SAR	TAC	IFTE	BMEP	TORQ	VE	BTE	ISE	SAC	ITE	AFC	OAF	FC
21-8	1.05	3.21	12.7	.514	81.2	14.6	1.12	55.1	3.91	39.1	29.3	78.8	8.0	9.1	.0561
-11	.90	3.68	6.8	.494	80.4	12.7	.97	54.0	3.41	38.5	33.0	79.2	8.4	9.0	.0549
22-1	1.58	1.44	.0	.410	78.1	26.9	2.06	57.4	8.71	40.3	15.5	78.1	10.8	10.8	.0379
-5	1.54	1.34	8.7	.404	79.8	26.2	2.00	57.2	9.35	40.3	15.7	78.1	10.7	11.7	.0345
-8	1.60	1.40	20.0	.435	77.4	26.6	2.04	59.5	8.93	41.5	16.3	77.4	9.3	11.6	.0375
23-2	.21	5.34	.0	.159	86.8	5.7	.43	32.3	2.34	25.2	45.2	86.8	8.5	8.5	.0187
-5	.24	4.40	14.7	.161	88.8	6.0	.46	31.6	2.85	24.8	41.0	87.0	7.9	9.3	.0173
-7	.39	2.89	29.8	.180	90.7	8.4	.64	31.6	4.35	24.8	28.0	87.0	6.8	9.7	.0185
-9	1.63	.91	53.8	.344	92.4	23.8	1.83	41.6	13.73	31.2	12.6	83.4	6.4	13.8	.0248
-12	1.45	1.06	56.1	.329	93.0	21.4	1.64	40.9	11.82	30.8	13.6	83.7	5.6	12.8	.0256
-15	1.19	1.29	48.4	.307	91.8	18.2	1.39	39.2	9.76	29.7	15.5	84.3	6.2	12.0	.0254
24-2	2.07	1.46	.0	.547	77.0	29.1	2.23	60.7	8.61	42.1	15.8	77.0	10.9	10.9	.0503
-5	2.09	1.38	3.4	.545	77.6	29.3	2.24	61.1	9.10	42.3	15.6	76.9	10.9	11.3	.0482
-8	1.74	1.68	8.1	.514	79.0	25.4	1.94	59.6	7.48	41.5	17.7	77.4	9.7	10.5	.0487
-11	2.08	1.30	3.4	.541	77.7	29.1	2.23	60.8	9.62	42.1	15.6	77.0	11.5	11.9	.0453
-14	1.56	1.85	11.3	.493	79.8	23.3	2.23	58.9	6.78	41.1	19.0	77.6	9.1	10.2	.0481

TABLE IV. (Con't)

TRIAL	BHP	SFC	SAR	TAC	IFTE	BMEP	TORQ	VE	BTE	ISE	SAC	ITE	AFC	OAF	FC
24-17	1.44	1.53	19.2	.416	83.1	22.3	1.71	53.0	8.18	37.9	17.4	79.5	9.1	11.3	.0367
-20	1.27	1.43	30.3	.349	86.8	20.7	1.59	47.2	8.77	34.6	16.5	81.5	8.0	11.5	.0302
-23	1.08	1.31	43.7	.298	90.2	18.9	1.44	43.4	9.59	32.3	16.5	82.8	7.1	12.6	.0236
-26	.21	7.19	70.7	.218	95.7	5.6	.43	42.7	1.74	31.9	61.2	83.0	2.5	8.5	.0255
-29	1.21	2.11	.0	.423	78.6	20.1	1.54	56.0	5.96	39.6	21.0	78.5	10.0	10.0	.0423
25-2	1.76	1.30	11.5	.468	79.0	28.3	2.17	62.0	9.68	42.7	15.9	76.6	10.9	12.3	.0380
-5	1.50	1.57	12.9	.447	79.9	24.5	1.87	59.9	7.99	41.7	17.9	77.3	9.9	11.4	.0391
-8	1.22	2.04	12.7	.438	80.3	20.1	1.54	58.1	6.15	40.7	21.5	77.8	9.2	10.5	.0416
-11	1.02	2.03	21.5	.353	85.3	16.5	1.26	46.8	6.18	34.4	20.8	81.6	8.1	10.3	.0343
-14	.75	1.85	37.0	.265	90.5	12.5	.96	36.4	6.77	28.0	21.2	85.5	7.2	11.4	.0232
-17	.93	1.76	.0	.325	82.2	15.8	1.21	45.1	7.13	33.4	21.0	82.2	11.9	11.9	.0272
14-8	.71	1.50	.0	.240	79.4	19.0	1.46	53.4	8.34	38.1	20.3	79.4	13.5	13.5	.0177
-12	.69	1.43	37.4	.259	85.3	19.2	1.47	60.1	8.75	41.8	22.4	77.2	9.75	15.5	.0166
25-20	2.14	1.16	0	.543	73.7	33.9	2.60	71.7	10.85	47.6	15.2	73.7	13.1	13.1	.0413
-23	1.78	1.55	0	.469	77.1	28.3	2.17	60.6	8.08	42.0	15.8	77.1	10.2	10.2	.0461
-26	1.07	2.13	0	.349	81.3	17.8	1.36	47.7	5.88	34.9	19.6	81.3	9.2	9.2	.0380

TABLE V. REDUCED EXPERIMENTAL RESULTS

TRIAL	CHOKE	R	IFSE	SPEED
15-2	0	.602	45.2	4695
-5	0	.608	43.8	4633
-8	0	.595	44.1	4633
17-2	0	.485	38.4	4211
-5	0	.504	35.8	4330
-7	0	.519	35.1	4400
-11	0	.488	37.6	4256
19-2	0	.684	49.6	5029
-5	0	.671	-	5007
20-2	0	.597	41.0	4433
-5	0	.571	43.5	4294
-7	0	.569	41.8	4295
-11	0	.572	40.3	4416
21-2	0	.487	38.6	4726
-4	0	.504	34.2	5010
-5	0	.500	34.4	5006

TABLE V. (Con't)

TRIAL	CHOKE	R	IFSE	SPEED
21-8	0	.496	35.0	4946
-11	0	.486	36.4	4849
22-1	0	.517	40.3	4024
-5	0	.515	37.5	4035
-8	0	.536	34.5	4134
23-2	0	.291	25.2	2557
-5	0	.285	21.5	2696
-7	0	.285	17.8	3151
-9	0	.374	14.2	4691
-12	30/70	.368	13.1	4632
-15	30/70	.353	15.4	4480
24-2	48/70	.546	42.1	4879
-5	48/70	.550	41.2	4900
-8	48/70	.537	38.9	4711
-11	45/70	.547	41.0	4905
-14	0	.530	37.4	4589

TABLE V. (Con't)

TRIAL	CHOKE	R	IFSE	SPEED
24-17	5/70	.477	31.7	4421
-20	27/70	.425	25.0	4189
-23	20/70	.390	18.6	3940
-26	13/70	.385	7.3	2588
-29	0	.504	39.6	4124
25-2	20/70	.558	38.8	4268
-5	16/70	.539	37.3	4187
-8	0	.523	36.5	4183
-11	10/70	.421	27.8	4237
-14	18/70	.328	18.1	4120
-17	18/70	.406	33.4	4035
14-8	0	.480	38.1	2559
-12	0	.541	27.3	2477
25-20	17/70	.646	47.6	4327
-23	0	.545	42.2	4321
-26	12/70	.429	34.9	4119

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