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DESIGN, FABRICATION AND TEST OF A STIRLING ENGINE FOR AGRICULTURE

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ABSTRACT

A kinematic stirling engine with a domed heater was designed, fabricated and tested. In designing and fabrication of the engine various problems were confronted and solved. Among various parts of the engine, cooler and main seal needed sophisticated techniques to fabricate in order to prevent leakage of working gas from the parts and to ensure their proper functions in the engine.

The engine had a series of experiment at various working gas pressures, heater temperatures and engine speeds to evaluate its performance. Indicated and brake power outputs and indicated and brake thermal efficiencies were determined from the experimental data. The engine resulted a little inferior performance to that of the GPU-3 engine of which performance was well reported. Several recommendations were made to improve the performance of the engine during the evaluation of its performance.

Key Word : Stirling Engine, Design, Performance, Fabrication, Kinematic Engine

INTRODUCTION

Agricultural by-products and solar energy are prospective future alternative energy sources for agricultural works. They are better to be used on the spot of agricultural operation, however, they are not good sources to convert them to mechanical work for agriculture. External combustion engines could be used for the purpose but most of them are not favourable for agricultural operations because specific power output of the engine and thermal efficiency of the engine are relatively low compared to those of the internal combustion engines.

Stirling engine is an external combustion engine of multi-fuel usable and is known as possibly compact in size but high in thermal efficiency. Hence the engine has been expected as an engine useful for agriculture to fuel agricultural by-products.

Recently various technology for the engine has been developed by stirling engine researchers - Hoehn(1978), Isshiki et al(1983), Reader and Hooper(1983), West(1986) and Beale(1986) - and most of them had expected that the engine could be in use practically in the future because of its simplicity in structure, low noise and vibration, and less contamination of environment. However, techniques for designing and fabrication of the engine are not fully materiallized.

This study was carried out to design and fabricate a engine which is capable to use combustion heat of agricultural by-products. In the process of designing and fabrication of the engine, it was intended to itemize various problems to fabricate the engine at a local machine shop in Korea and to find out practical solutions for the problems. Finally the fabricated engine was tested at various operating conditions to evaluate performance of the engine.

DESIGNING AND FABRICATION

Among various types of the engine, beta type kinematic engine with rhombic drive mechnism was selected to fabricate in this study. Main reason for the selection was that the type of engine is fit to use combustion heat of agricultural by-products and is easy to scale up to an engine of MW power output with almost same technique to design and fabricate an 1 or 2 kW engine. And also the type of engine was considered as an engine of easy to design and to fabricate because various works on the type of engine were done by many researchers and a lot of informations on the engine are open in public.

Hydrogen was selected as working gas of the engine because performances of several engines charged with hydrogen were well reported and it was intended in this study to compare performance of the fabricated engine with those of engines of which performance was known.

Stroke volume of the engine was determined using the Beale number (Senft;1982) of 0.15 and the maximum operating pressure of working gas, the highest temperature of the heater and engine speed were assumed as 3 MPa, 650 ℃ and 2000 rpm, respectively.

Power piston diameter and stroke were determined from the stroke volume determined above and a piston diameter-stroke ratio of 2.4. Power piston was made of solid stainless steel bar with 3 grooves for piston rings. Displacer piston was made of hollow stainless steel cylinder(t=1.5 mm) and a heat baffle of steel plate was installed inside the displacer in order to cut off convection and radiation heat transfer from top to bottom of the displacer which was located between heater and cooler. Displacer had one groove for a piston ring. Material of all the piston rings was Teflon.

To seal the working gas spaces of the engine, several attempts were made. Most of the attempt were to seal reciprocating piston rod. Finally rotational sealing on an axle for power output was selected instead of the reciprocating rod sealing. A mechanical seal of fine grade was implemented on the crankcase of the engine which was made to stand high pressure, and working gas was charged inside the crankcase as well.

Structure of heater of the engine was one of important part to design the engine to use agricultural by-products as a fuel for the engine. Generally tubes of heater of a stirling engine are placed around the wall of combustion chamer or in the center of burner of the engine. But most of agricultural by-products are solid and not easy to burn because their sizes and shapes are not homogeneous, heater of thin tubes was not considered. Instead of this, a domed heater, which is simple and durable in its structure, was selected. SUS 304 was used as material for the heater.

Regenerater was made of very fine stainless steel wire mesh of #200 and was shaped by rolling the wire mesh fairly tight.

Cooler was made like a shell and tube type heat exchanger. 150 units of fine cooper tube(2.7 mm OD and 1.2 mm ID) were welded on a cylindrical cooper shell by brazing. Cooler was the hardest part to fabricate because the tubes were too thin and too many to weld mannually.

Cross sectional view of the fabricated engine is shown in Fig. 1.

MATERIALS AND METHODS

Materials

The engine was equiped with a working gas charging system, a heater heating system and a cooling water supplying system. Working gas charging system was consisted of a gas tank, a pressure regulator, a valve and a pressure gage. A propane gas burner was used to heat the engine for the engine test purpose and flame temperature of the burner was controlled by a gas flow control valve. Cooling water was supplied at a constant flow rate to the engine to keep constant cooling temperature.

A disc brake was mounted to the engine to load the engine and the load was controlled by variable braking forces to the engine.

Instrumentation on the engine was done to determine heater temperature, mean pressure of the working gas, heat rejected by the cooler and the indicated power and the brake power of the engine.

Heater temperature was determined by averaging 3 temperatures measured at evenly spaced 3 locations on the surface of the heater by K type thermocouples. Mean pressure of the working gas was determined by averaging experimental data for pressure variations in the compression space of the engine measured by a pressure transducer(Kyowa, PE-50KP). A dial type pressure gage was also installed to observe charging pressure of the working gas. Heat rejected by cooler was determined from the specific heat of water and experimental data of cooling water flow rate and inlet-outlet water temperatures.

In order to obtain the indicated power, position of power piston was measured along with the time varying gas pressure. A pressure transducer(HBM, P8A) was set to measure inside pressure of crankcase which was the basis to determine position of power piston. To determine the brake power, axle torque and rotational speed were measured. Axle torque was measured by strain gages and a slip ring and the speed was measured by a magnetic pick up type tachometer. The strain gages and the tachometer were installed on an axle connected between the engine and the disc brake.

The time varying experimental data were acquired through an A/D converter (12 bits resolution, 0.6 ms/data) and were collected in a 8 bits computer for further analyses.

Skematic diagram of the experimental setup is shown in Fig. 2.

Methods

Performance of the engine was tested at the mean gas pressures of 0.8, 1.3 and 2.1 MPa, and the heater temperatures of 400℃, 500℃ and 600℃. At an engine operating condition of a certain gas pressure and a heater temperature, the engine was tested at a minmum and a maximum operatable loads and at a mean load of the minimum and the maximum loads.

DISCUSSION OF RESULTS

Indicated Power Output (IPO)

IPOs of the engine were calculated from the obtained experimental data of working gas pressure, volume of working space and speed of engine using the well-known thermodynamic equation to determine work done by a cylinder -piston system during a quasi-equilibrium process.

IPOs of the engine are shown in Fig. 3. As shown, IPO and the maximum operable speed of the engine were increased as the working gas pressure and the heater temperature were increased. The maximum IPO of the engine was about 670 W within the experimental condition ranges noted above. The maximum IPO was obtained at 2.1 MPa and 600 ℃ of the working gas pressure and the heater temperature, respectively, and speed of the engine was 1442 rpm.

Brake Power Output (BPO)

BPOs of the engine were calculated from the experimental data of torque on the axle and speed of the engine. Fig. 4 shows the BPOs of the engine. BPOs were increased as the working gas pressure and the heater temperature were increased like IPO, however, the trend was fairly different from that of IPO. The difference was that BPO was influenced more by the heater temperature than the working gas pressure. The maximum BPO was marked as 184 W, only about 1/3 of the maximum IPO, at the same experimental conditions for the maximum IPO except the speed of the engine.

BPOs of the engine were decreased rapidly as speed of the engine was increased as shown in the figure, and the trend was come up sharply when the speed was higher than 850-1000 rpm. Hence the maximum BPO within the experimental condition ranges was obtained around the engine speed which was much lower than the maximum operable speed of engine at which the maximum IPO was observed. The maximum BPO was obtained at a speed of 841 rpm.

Indicated Thermal Efficiency (Eit)

Eits of the engine were determined by the classical definition of Eit with the data for IPO divided by an estimated total heat input(Qin) to the working gas of the engine. Qin was estimated by an addition of IPO and heat rejected by the cooler which was determined from the experimental data for the flow rate of cooling water and the temperature difference between the inlet and the outlet of cooling water.

Fig. 5 shows Eits of the engine at various operating conditions. Within the experimental conditions, Eits of the engine were in a range of 11 - 22 %, which were fairly lower level than Eit of a general heat engine. Main reason for such low level of Eit of the engine was that considerable amount of heat was not transferred to the working gas in the engine but lost to the cooler of the engine directly.

Eit of the engine was increased as the working pressure and speed of the engine was increased. However, Eit was decreased as the heater temperature was increased. The fact obeyed the thermodynamic rule for an ideal heat engine, Carnot cycle engine, and was interpreted as that the higher the heater temperature was, the higher rate of heat loss by conduction heat transfer from the heater to the cooler through outer shell of the engine occured. It was concluded that careful geometrical or thermal arrangement of the heater and the cooler to reduce the conduction heat loss is necessary to increase the thermal efficiency of the engine.

Performance of various stirling engines was put togrther by Martini(1982) and Crowley(1984). Among the engines, the GPU-3 engine developed by General Motors Research Corporation is similar to the engine of this study from viewpoint of structure and size of the engines, and performance of the engine was well reported.

Eits of the engine of this study were compared with those of the GPU-3 engine. Since Eit of a stirling engine depends on operating conditions of the engine and operating conditions of both engines were not equal, the comparison was performed after Eit of the GPU-3 engine at a certain operating condition was estimated by interpolation. The comparison resulted that Eits of the engine of this study were 1 - 2 % lower those of the GPU-3 engine at operating conditions for high IPO of the engine.

Brake Thermal Efficiency(Ebt)

Ebts of the engine were determined by the same manner as for Eit with the data for BPO and the estimated total heat input(Qin) described above. Fig. 6 shows Ebts of the engine. Ebts of the engine were increased as the working gas pressure and the heater temperature were increased. The same trend of Ebts is observed from the test data for the GPU-3 engine(Martini;1982).

The maximum Ebt of the engine was recorded as 7.5 % at 2.1 MPa, 600 ℃ and 841 rpm of the gas pressure, the heater temperature and the engine speed, respectively. The maximum Ebt is fairly low value as an Ebt of an engine, however, the value is close to Ebt of the GPU-3 engine which was 9 % when IPO of the engine was 999 W(Crowley;1984).

Mechanical Loss

Mechanical loss of the engine at the operating condition for the maximum Ebt was 264 W. The loss was incresed as the speed of the engine increased, and recorded 618 W at 1442 rpm of the speed with the same gas pressure and the heater temperature for the maximum Ebt.

Martini(1982) noted that the mechanical loss of the GPU-3 engine was about 400 W when BPO of the engine was within 400 - 600 W, and was about 500 W when BPO of the engine was 1100 - 1500 W. He noted that the least mechanical loss of an engine of the same type and size was expected about 400 W. Reader and Hooper(1983) estimated the mechanical loss for a sliding seal alone for a small kinematic stirling engine as 700 - 1000 W.

It was concluded from the results that mechanical loss of the engine was about the same as that of the GPU-3 engine, and the loss was a considerable portion of energy input to the engine. In order to improve performance of the engine, it was necessary to decrease the mechanical losses from the seal and the drive mechanism of the engine.

CONCLUSION

1. Fabrication of a kinematic stirling engine had two main difficulties; one was to make a seal for a reciprocating connecting rod and the other was to make a cooler which was made by welding numerous thin tubes on a cooler shell. The seal was successfully substituted by a rotational mechanical seal but more work was still needed to improve performance of the engine. The cooler needed a special welding tool to fabricate like a welder in vacuum.
2. The fabricated engine had a maximum brake power output of 184 W within the experimental conditions of the engine. Indicated thermal efficiencies of the engine were in a range of 11 - 22 % and the maximum brake thermal efficiency was 7.5 %. Overall performance of the engine was a little inferior to that of the GPU-3 engine.
3. Performance of the engine could be improved by careful arrangement of the heater and the cooler geometrically and thermally to reduce conduction heat loss from the heater to the cooler, and by development of a seal and a drive mechanism to reduce mechanical energy loss in the engine.

REFERENCES

1. Beale, W.T. 1986. Applications of simple stirling engines. Third International Stirling Engine Conference. Rome, Italy. p.49-
2. Crowley, J.L. 1984. Standardized efficiency terms: Comparison of some stirling engine performance figures. IECEC Proc., Paper No. 849064. ACS.
3. Hoehn, F.W. and A.R. McDougal. 1978. Design of a prototype stirling laboratory research engine. IECEC Proc., Paper No. 789271. SAE.
4. Isshiki, N., et al. 1983. Experimental studies on small stirling engines of different types. IECEC Proc., Paper No. 839137. AICE.
5. Martini, W.R. 1982. Stirling engine design manual, 2nd ed. USDOE, Office of Conservation and Solar Applications. p.1-153.
6. Reader, G.T. and C. Hooper. 1983. Stirling Engines. E. & F. N. Spon.
7. Senft, J.R. 1982. A simple derivation of the generalized Beale number. IECEC Proc., Paper No. 829273. IEEE.
8. Urieli, I. and D.M. Berchowitz. 1984. Stirling cycle engine analysis. Adam Hilger Ltd., Bristol.
9. West, C.D. 1986. Principles and applications of stirling engines. Van Nostrand Reinhold Co., N.Y.

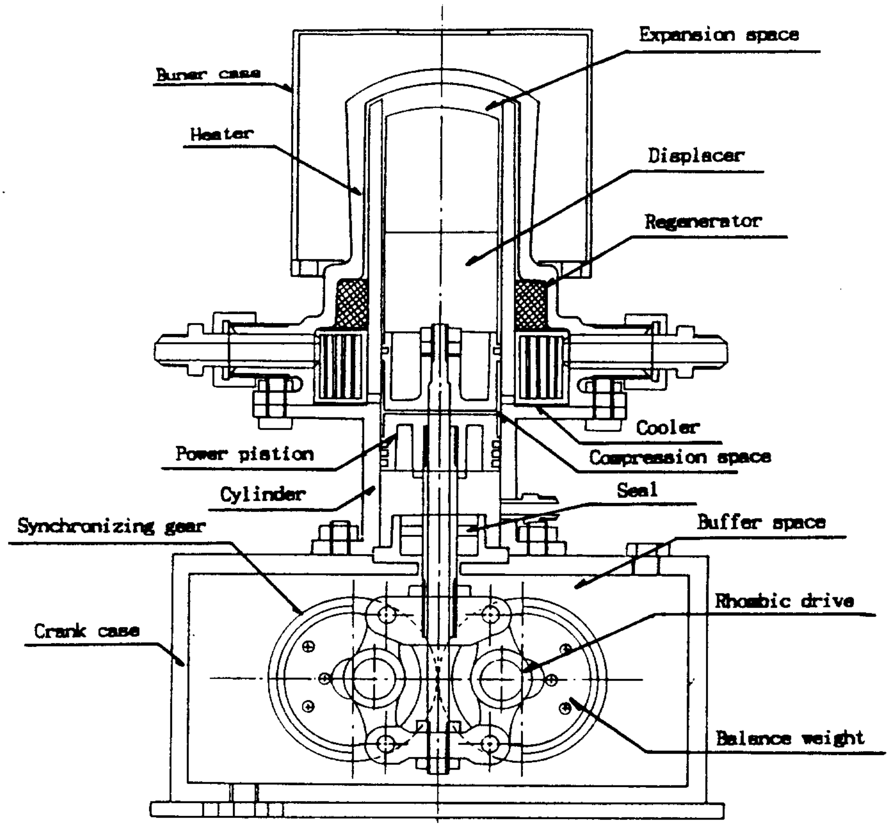

Fig. 1. Cross section of the designed engine

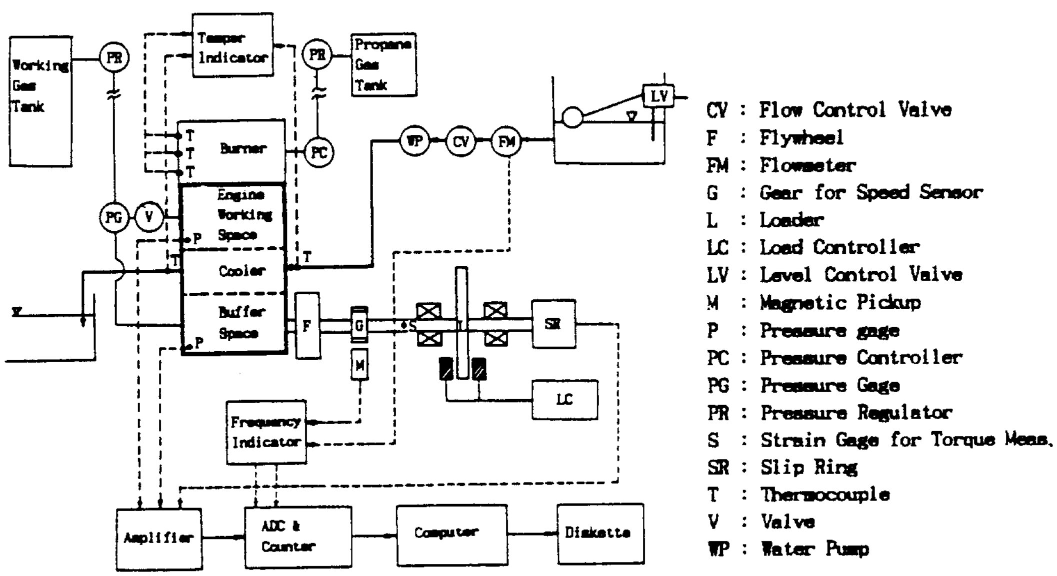

Fig. 2. Instrumenttation of the engine for the performance test

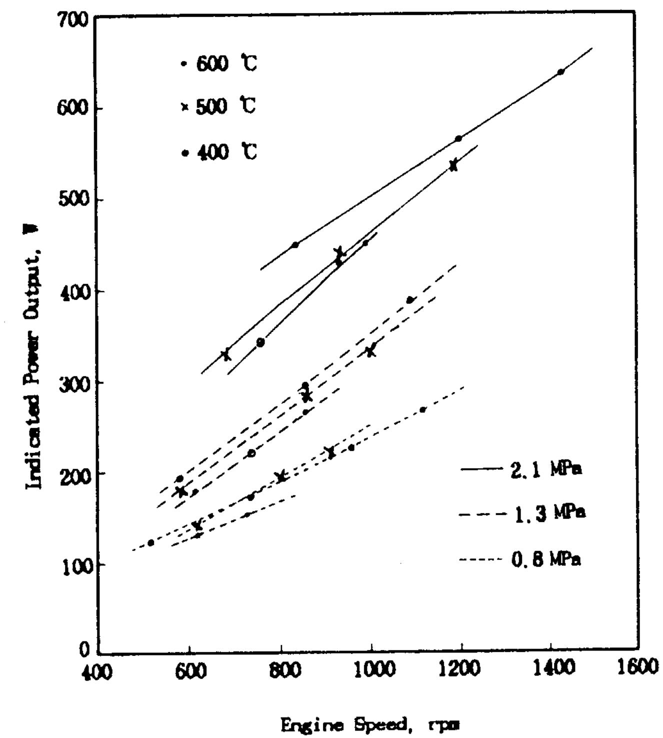

Fig. 3. Indicated power outputs of the engine

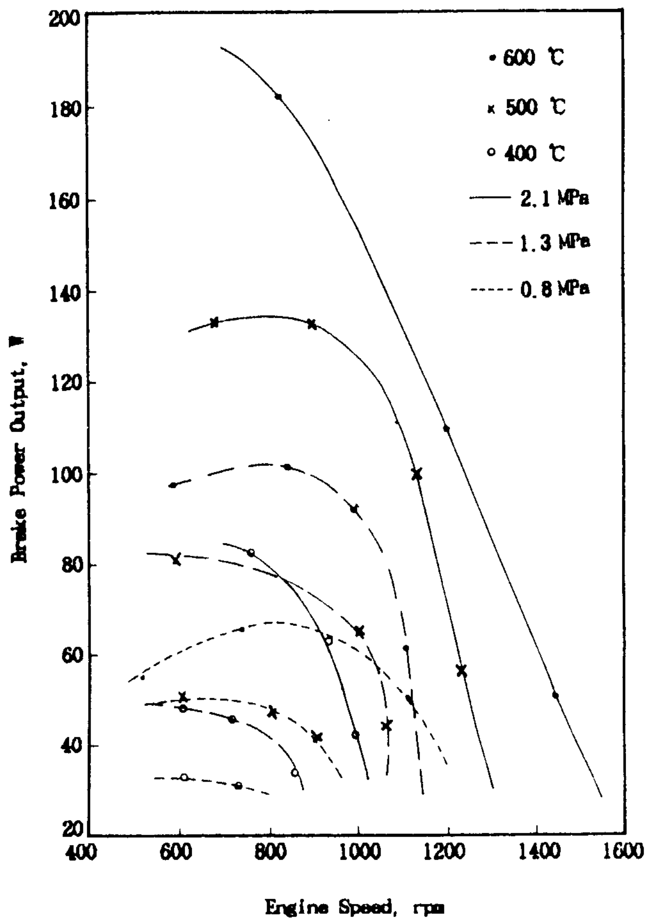

Fig. 4. Brake power outputs of the engine

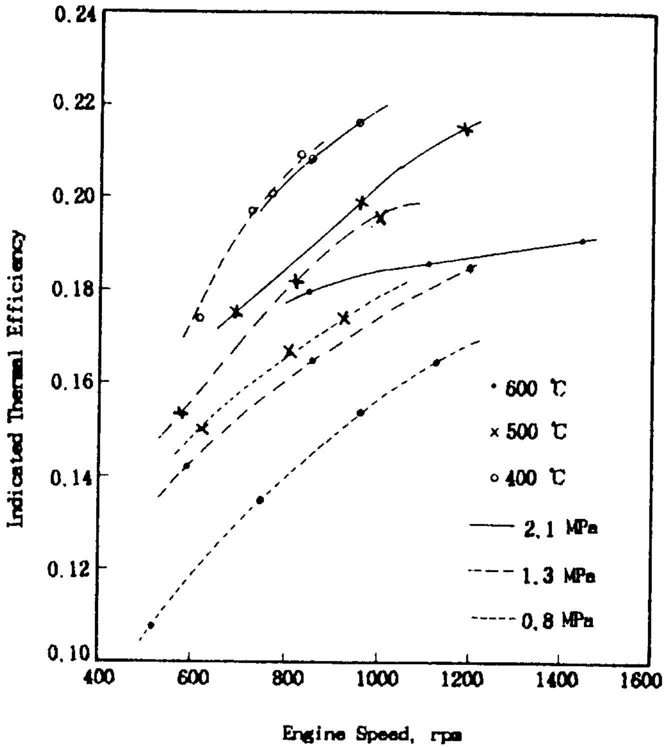

Fig. 5. Indicated thermal efficiencies of the engine

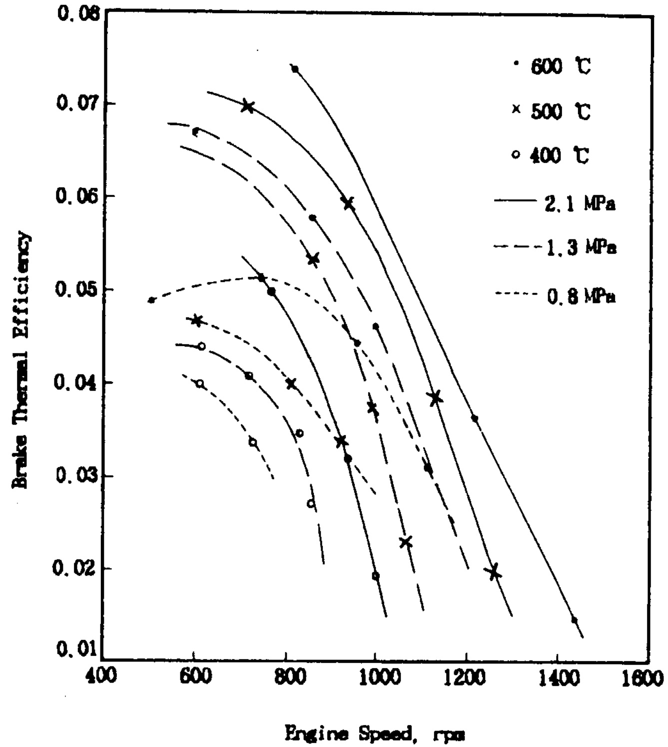

Fig. 6. Brake thermal efficiencies of the engine