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THERMODYNAMIC ANALYSIS OF BUSH ENGINE

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ABSTRACT

The Bush Engine used as the pressure generator, is thermodynamically analyzed by defining an ideal cycle. Heat exchanges in the hot space, cold space and regenerator were determined in terms of temperature and pressure limits of the cycle. The thermal efficiency of the engine was described on the basis of the work availability of the compressed air and is found to be equal to the Carnot cycle. The optimum tank pressure, which corresponds to the maximum cyclic availability, is stated with an equation. Introducing practical working conditions, numerical results are obtained and discussed. The engine is found to be practically valuable.

Key Words: Hot gas engine, thermal compressor, bush engine, Ericson engine

BUSH MOTORUNUN TERMODİNAMİK ANALİZİ

ÖZET

Basınç jeneratörü olarak kullanılan Bush Motoru ideal bir çevrim tanımlamak sureti ile termodinamik yönden analiz edilmiştir. Sıcak hacim soğuk hacim ve rejeneratördeki ısı alış verişi çevrimin basınç ve sıcaklık limitleri cinsinden belirlenmiştir. Sıkıştırılmış havanın iş yapma kabiliyeti (availability) esas alınarak motorun ısıl verimi tanımlanmış ve Carnot çevriminin verimine denk olduğu belirlenmiştir. Sıkıştırma basıncının belirli bir değerinde çevrimlik availability makximum değere ulaşmakta olup bu hale karşı gelen sıkıştırma basın-cı bir eşitlik ile ifade edilmiştir. Pratiğe uygun çalışma şartlarını kullanmak sureti ile sayısal değerler elde edilmiş ve motorun pratik değerinin olduğu kanaatine varılmıştır.

Anahtar Kelimeler: Sıcak hava motoru , ısıl kompresör, Bush motoru, Ericson motoru

1. INTRODUCTION

The Bush Engine is a special type of Hot Gas Engines, which compresses the air using thermal energy. The engine works with a cycle consisting of two isobaric and two polytropic processes. The processes are performed by means of the mechanical arrangement in Figure 1, where displacement of working gas is performed by a displacer and two check valves. From the thermodynamic point of view, the Bush Engine is in the class of Ericson Engines, Walker (1980).

Walker(1980) reported that an externally driven arrangement having gas operated valves had been patented by Bush as a pressure generator in 1949. The same arrangement was used as a power source in artificial heart studies undertaken by Buck (1968). In Massachusetts Institute of Technology, a development project, named VHGE, was carried out by J.L.Jr Smith. The practical results obtained by Smith was much lower than theoretically predicted. The difference was attributed to the inadequate heat transfer rate between the wall and gas in the cylinder.

The subject of In-Cylinder Heat Transfer has been extended by Smith and his students (Smith and Lee, 1978; Lee, 1983; Kornhauser,1992; Kornhauser and Smith,1994).

In the present study, the thermal efficiency, cyclic availability and pumping characteristics of the Bush Engine was examined as a preliminary study to a solar energy research.

The advantages of the Bush Engine lies in its structural simplicity and functional difference which works as a thermal compressor. In Hot Gas Engines with pistons, the following disadvantages arise.

1. The co-working surfaces of piston and cylinder are machined at a high quality of finish and fitted to each other with a minimal working clearance.

2. To prevent the leakage of compressed gas through the gap between the piston and cylinder, where no lubrication is made, PTFE rings which have a limited life period are used.

3. The mechanical friction between piston and cylinder surfaces dissipates a portion of cyclic work produced by the engine.

4. Pistons used in Hot Gas Engines are heavy elements that increase the total weight of the engine and creates vibration.

In the Bush Engine, the only element subject to reciprocating motion causing mechanical friction is the rod of displacer. Since the diameter of the displacer rod is very much smaller than that of a piston, disadvantages mentioned above are avoided. In addition, the compressed air is a source of energy as practical as electricity. Compressed air can be used by many kinds of energy consuming machines. Electricity and the other types of energy may not be stored but the compressed air is a stored energy itself. Therefore the Bush Engine may be more practical in solar energy applications.

Buck (1968) reported that the prediction of thermodynamic performance characteristics of the Bush Engine had been made by using a Schmidt type simplified analysis and a Nodal analysis. In available literature, the details of the analysis of Bush Engine has not been presented. In the present study, the analysis of the Bush Engine is presented.

2. ENGINE AND THERMODYNAMIC CYCLE

The Bush Engine consists of a cylinder, a displacer, a crank shaft, a flywheel, a connecting rod, two check valves and a tank (Figure 1). One end of the cylinder is kept at the hot source temperature while the other at the cold source temperature. The volumes in the hot and cold ends of the cylinder are named as the hot and cold spaces respectively. In a Bush Engine designed for air pumping, the gas pumped into the tank functions as working fluid as well.

With a cycle consisting of four thermodynamic processes, the engine compresses the air into the tank. The position of the displacer, at which the hot space becomes zero, may be assumed to be the beginning of the first process. At this position of displacer, all of the air in the cylinder is in cold space. The pressure and temperature of the air are equal to the ambient pressure and temperature. After the initiation of the first process, some of the air in the cold space passes to the hot space and the pressure begins to increase.

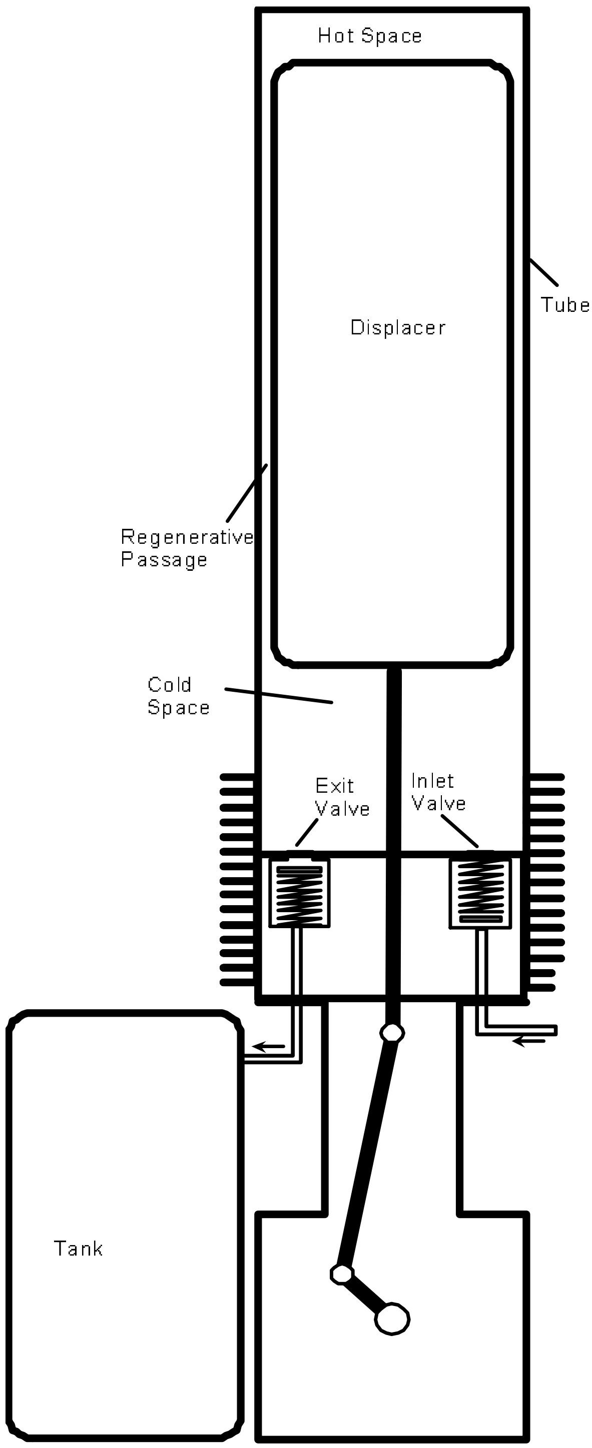

Figure 1. The Bush Engine

The variation of pressure with the volume of hot space is shown in Figure 2, where the first process of cycle is illustrated between the points 1 and 2. During this process, the total mass of air in the cylinder, both in cold and hot spaces, remains constant. The pressure continues to increase as the hot space increases and consequently reaches the tank pressure. When the cylinder pressure exceeds the tank pressure, the exit valve discloses and the first process terminates.

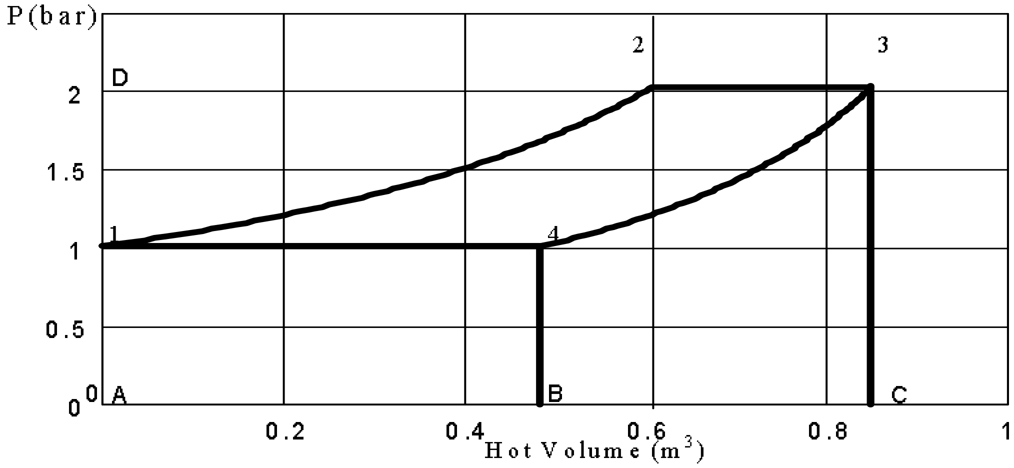

Figure 1. Variation of Pressure With Hot Space Volume.

The second process begins when the exit valve is disclosed, and terminates when the cold space is vanished. Over the second process the pressure remains constant. During the second process the mass of air in the cold space splits into two portions; a portion passes to hot space as the other to the tank. The second process is illustrated between the points 2 and 3 in Figure 2.

The third process begins when the displacer turns backward to the hot end of the cyl-inder. This process is illustrated between the points 3 and 4 in Figure 2. In this process, the hot space is reduced as the cold space expanded. The mass of air in the cylinder remains con-stant. Due to the displacement of the air from hot space to the cold space, the pressure begins to decline. As the displacer proceeds, the pressure in the tube drops to the ambient pressure and the inlet valve is disclosed. Disclosing of inlet valve is the end of third process.

In the fourth process, the expansion of cold space and reduction of hot space continues. This process is illustrated between the points 4 and 1 in Figure 2. The pressure remains constant at ambient pressure. All of the air in the hot space maves to the cold space. At the same time some ambient air enters into the cold space. When the displacer reaches to the end of its stroke, the fourth process and cycle terminates.

In the Bush Engine the rod of the displacer creates a cycle in the cylinder and generates some work. The work generated by the rod of the displacer is proportional to the volume of the rod. When the displacer is at the hot end of the cylinder, the pressures in and out the cylinder are in equilibrium. The slight displacement of the displacer causes the inner pressure to prevail the outer pressure. When the displacer moves towards the cold end of the cylinder, the rod of the displacer moves out of the cylinder and the gas in the cylinder expands generating some work. The positive work generation by inner pressure continues during the downward stroke of the displacer. In the upward stroke, the inner pressure generates some negative work. However, this work is less than the positive work generated during the downward stroke. Some of the work generated during the downward stroke is stored on a flywheel as the kinetic energy and again used in the upward stroke of the displacer. The cycle created by the displacer rod resembles the cycle seen in Figure 2. The only difference is the amount of variation of volume in the first and third processes.

3. THERMODYNAMIC ANALYSIS

The analysis aims to determine the heat withdrawn form the hot source, the heat rejected to the cold source and the efficiency of the machine as an energy generation system. In the analysis the following assumptions are made.

1) The hot and cold sources are surfaces surrounding the hot and cold spaces. The passage connecting the hot and cold spaces functions as a regenerator.

2) The air in different sections of the machine is thermally in equilibrium with the surrounding solid surfaces. For that reason, the heat exchange between the air and solid surfaces occurs at reversible conditions, that is, the variation of the total entropy of the isolated system is zero.

3) At every point in the machine, the instantaneous pressure is the same. When the air displaces between different sections of the machine no frictional loss occurs.

4) The thermal efficiency of the engine is described as the proportion of cyclic availability to the heat withdrawn from the hot source.

5) For the simplicity of analyses, the volumes of the displacer rod and connecting pas-sage are ignored.

The total volume of the machine is

$$V_C + V_H = V_T = \text{Constant} \quad [1]$$

The instantaneous pressure in the machine is described by

$$p = \frac{m_t\, R\, T_C\, T_H}{V_C\, T_H + V_H\, T_C} \quad [2]$$

Using Equation [1] and [2] the work generated in the hot space during the first process is stated as

$$W_{H12} = \frac{m_t R T_C T_H}{T_C - T_H} \ln \frac{p_1}{p_2} \quad [3]$$

Using the first law of thermodynamics, the cyclic rates of heat exchanged in hot space, cold space and regenerator are stated respectively as

$$Q_{H12} = W_{H12} - m_{H2}\, R\, T_H \quad [4]$$

$$Q_{C12} = W_{C12} + m_{H2}\, R\, T_C \quad [5]$$

$$Q_{R12} = m_{H2}\, C_p\, (T_H - T_C) \quad [6]$$

In the second process, the work generated in the hot space, the rates of heat exchange in the hot space, the cold space and the regenerator are stated respectively as

$$W_{H23} = (m_{H3} - m_{H2})\, R\, T_H \quad [7]$$

$$Q_{H23} = 0 \quad [8]$$

$$Q_{C23} = 0 \quad [9]$$

$$Q_{R23} = (m_{H3} - m_{H2})C_P(T_H - T_C) \quad [10]$$

In the third process, the work generated in the hot space, the rates of heat exchange in the hot space, the cold space and the regenerator are stated respectively as

$$W_{H34} = \frac{m_{H3} R T_C T_H}{T_C - T_H} \ln \frac{p_3}{p_4} \quad [11]$$

$$Q_{H34} = W_{H34} + (m_{H3} - m_{H4}) R\, T_H \quad [12]$$

$$Q_{C34} = W_{C34} - (m_{H3} - m_{H4}) R\, T_C \quad [13]$$

$$Q_{R34} = (m_{H3} - m_{H4}) C_P (T_C - T_H) \quad [14]$$

In the fourth process, the work generated in the hot space, the rates of heat exchanged in the hot space, the cold space and the regenerator are stated respectively as

$$W_{H41} = -m_{H4} R\, T_H \quad [15]$$

$$Q_{H41} = 0 \quad [16]$$

$$Q_{C41} = 0 \quad [17]$$

$$Q_{R41} = m_{H4} C_P (T_C - T_H) \quad [18]$$

Summation of Equations 6, 10, 14 and 18 indicates that, the total rate of heat exchange in the regenerator is zero. In Figure 2, the area D-2-1 represents the heat given by Equation 4. The area D-2-3-4-1 represents the heat given by Equation 12. It is seen that the net heat withdrawn from the hot source is represented by the area 1-2-3-4. Summation of Equations 4 and 12 also shows that the net heat withdrawn from the hot source is

$$Q_H = W_{H12} + W_{H34} \quad [19]$$

The work availability of the air compressed into the tank is

$$Q_H = W_{H12} + W_{H34} \quad [19]$$

The work availability of the air compressed into the tank is

$$W_{exp} = (m_t - m_{H3}) R\, T_C \ln \frac{p_2}{p_1} \quad [20]$$

By using Equations 3, 11, 19 and 20 in

$$\eta = \frac{W_{exp}}{Q_H} \quad [21]$$

the thermal efficiency of the machine is determined as

$$\eta = 1 - \frac{T_C}{T_H} \quad [22]$$

which is Carnot efficiency. Choosing m_{H3} from the General State Equation of Perfect Gases and substituting into Equation 20 yields

$$W_{exp} = m_t \left(1 - \frac{p_2}{p_1}\frac{T_C}{T_H}\right) R\, T_C \ln \frac{p_2}{p_1} \quad [23]$$

From the last equation it is seen that W_{exp} does not infinitely increase with p_2. Deriving the last equation with respect to p_2, and then setting equal to zero yields

$$1 + \ln \frac{p_2}{p_1} = \frac{T_H}{T_C}\frac{P_1}{P_2}$$

In the last equation Replacing p_1 with p_A and p_2 with p_0 results in

$$1 + \ln \frac{p_O}{p_A} = \frac{T_H}{T_C}\frac{P_A}{P_O} \quad [24]$$

which can be used to determine the optimum pressure to get the maximum cyclic availability.

The maximum tank pressure corresponding to any T_H / T_C is stated as

$$\frac{P_{max}}{p_A} = \frac{T_H}{T_C} \quad [25]$$

The ratio between the air mass pumped into the tank per cycle and the total air mass initially found in the cylinder is stated as,

$$\frac{m_{out}}{m_t} = \left(1 - \frac{p_O}{p_A}\frac{T_C}{T_H}\right) \quad [26]$$

The inclusion of the work in the analysis, generated by the displacer rod, requires the use of more complicated equations. To secure the simplicity of this analysis, the work generated by the displacer rod is excluded.

4. RESULTS AND DISCUSSION

Within the range of temperature ratio $2.0 \le T_H / T_C \le 3.5$, results obtained from Equations 24, 25 and 26 are plotted in Figure 3. For $T_H/T_C = 2$, taking $T_C = 300$ K and $p_A = 101314$ N/m^2, the results documented in Table 1 are obtained per kg air. In the first process of cycle in the hot space, we may expect a heat flow from the wall to the working gas. But as long as the air enters into the hot space at hot source temperature (T_H), the heat flow is adverse. In practice, since the regenerator efficiency is less than 100 %, the fluid enters into the hot space with a temperature lower than T_H and the adverse heat flow in the hot space becomes lower than those documented in the Table 2. The other adverse heat flow occurs in the third process in the cold volume.

The cyclic availability and thermal efficiency are found to be 8800 J and 50 % re-spectively. With respect to this cyclic availability, an engine having 1500 rev/min speed would generate about 250 W specific power per liter stroke volume. The temperature ratio chosen in this examination, may be obtained with an ordinary solar energy collector.

Table 1. Heat Balance for

Process	Q_C	Q_H	Q_R
1-2	-25400	-13775	137182
2-3	0	0	82159
3-4	7803	31372	-137182
4-1	0	0	-82159

Using a sophisticated solar energy collector consisting of concentrators and receivers, the hot source temperature may be improved above 1000 K (Bean and Diver, 1992; Andraka et al., 1992; Moreno et al., 1992; Shaltens et al., 1992). If the hot and cold source tempera-tures are assumed to be 1000 K and 300 K, the ratio of optimum and maximum pressure to ambient pressure becomes1.98 and 3.33 respectively. Since the air pumped into the tank is at ambient temperature, a second compression process can be applied under the same hot and cold source temperatures.

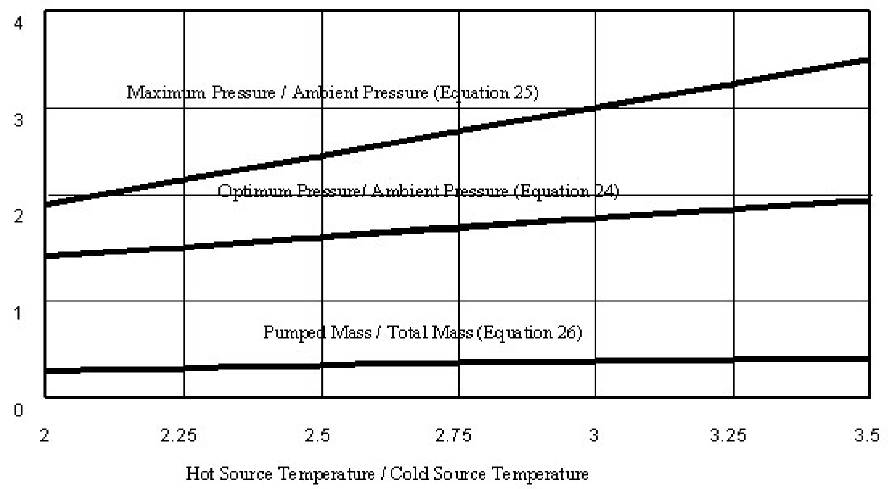

Figure 3. Variation of pressure mass ratio with temperature ratio

For 1000 K hot source and 300 K cold source temperatures the heat exchange rates in Table 2 are obtained. The cyclic availability and thermal efficiency becomes 23880 J and 70 % respectively. An engine having 1500 rev/min speed would generate about 700 W specific power per liter stroke volume. For a Hot Gas Engine, the specific power obtained in the above examinations are reasonable as long as the aim is develop of a Solar Energy Engine.

As seen in the Tables 1 and 2, the regenerative heat exchange is much greater than the heat transfer occurring in hot and cold spaces. From the practical point of view, this is a useful feature that the heat transfer surface in the regenerative channel can be enhanced by relatively easier methods.

Table 2. Heat Balance for

Process	Q_c	Q_H	Q_R
1-2	-47859	-36519	295470
2-3	0	0	122409
3-4	13746	70631	-295470
4-1	0	0	-122409

It is notable that, in hot and cold spaces, the heat exchange between the gas and the wall occurs only in compression and expansion processes. Kornhauser and Smith (1994) showed that in compression and expansion processes, a large enough heat transfer occurs between the wall and the gas in the cylinder. In the first process, the air, blowing into the hot space, impinges on the upper surface of the cylinder. The same phenomenon occurs in the third process in the cold space. In the impinging flow of a fluid, the convection heat transfer coefficient is large enough. These incidents provide a development chance for the Bush Engine to have a simple mechanic structure.

As known, all of the ideal cycles deviate from the practical cycles and cause mis estimations. In the cycle described in this study, it is assumed that the air in different sections of the machine is thermally in equilibrium with the surrounding solid surfaces. This is possible if only there is an infinitely large heat transfer coefficient between the air and the solid wall. In practice, the heat transfer coefficient has a finite value. Therefore, especially in hot and cold spaces, some difference may occur between temperatures of the gas and the solid surface.

5. CONCLUSION

By defining an ideal cycle, the Bush Engine was thermodynamically analyzed. The ideal thermal efficiency of the engine was found to be equal to the Carnot cycle. For a the Bush Engine working with ambient air between 1000 K hot source and 300 K cold source, the power output was predicted to be 700 W. For the same conditions the tank pressure was determined to be approximately 2 bars. The cyclic heat exchanges in the hot and cold spaces were found to be plausible from the heat transfer point of view. The engine was found to be more practical than Stirling Engines for solar energy applications.

C_P	Specific Heat at constant Pressure (J/kg K)
m	Mass (kg)
m_t	Initial mass in the cylinder
m_{out}	The mass pumped into the tank per cycle
p	Pressure (Pa)
p_A	Ambient Pressure
p_o	Optimum Pressure

p_{max}	Available Maximum Pressure (Pa)
Q	Heat (J)
T	Temperature (K)
V	Volume (m^3)
W	Work (J)
W_{exp}	Availability (J)
η	Thermal efficiency

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