A Proposed Sequential Turbocharging System for Constant Speed Diesel Engines

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ABSTRACT

This paper discusses the possibility of improving the part load performance of diesel electric turbocharged engines operating at constant speed conditions. A sequential turbocharged system is proposed, where the compressors are connected in series. The study focused on two turbocharged diesel-electric generating sets existing at Ameria Petroleum Refining Company in Alexandria, Egypt. The results showed that, at part load, both the maximum pressure and temperature were increased, and the brake specific fuel consumption was reduced considerably (by about 10%).

1. INTRODUCTION

When a turbocharged diesel engine operates at part loads, it suffers from lack of air flow associated with low supercharged pressure and temperature. This leads to an incomplete combustion , and will ultimately worsen the engine performance. Many investigators have tried to improve the turbocharged diesel engine performance at part loads using various approaches.

In 1982, early trials were made by Flaxington [1], Wallace and Ziarati [2] to improve the turbocharged diesel engine performance at low speed operating conditions by using variable geometry turbine. At low speed range, the smaller nozzle area generates greater pressure drop across the turbine. The brake specific fuel consumption is improved by over 10%
Holzhausen and Alfano [3] used a new hydraluic boost compressor system to improve the acceleration and the transient response of turbocharged diesel engine. An improvement of 15.8% in brake specific fuel consumption has been made.

Before 1984, the hyperbar system was introduced by the German diesel manufacturer MTU [4], to solve the problems of the turbocharged diesel engines at start-up conditions as well as part load operation where the air mass flow and its supercharging pressure is low. The main disadvantage of this system is the requirement of an external power source to start up the engine. The cut-out system was also developed by MTU [4], in which some cylinders are to be cut out by cutting fuel delivery to them at part loads. The cylinders to be cut out can be selected according to the harmonic vibration characteristics.A new series of sequential turbocharging systems was introduced by MTU [4] to improve the part load operation of turbocharged diesel engines. This method requires at least two turbochargers. All turbochargers are working active during full load operation. Some of them are deleted at part load conditions, so that the active turbochargers always operate at optimum efficiency and provide higher charge-air pressure. In 1986, Borila [5] developed another model of the sequential turbocharging systems consisting of two unequal size turbochargers with pulse convertor connected in parallel arrangement, demonstrated in Figure (1). The two turbochargers are operating at rated power in parallel while the small one is switched - off at part load operation in order to improve the brake mean effective pressure, and consequently the brake specific fuel consumption.

In 1989, Meier and Czerwinski [6] had paid attention to the medium speed engines, for the marine and stationary applications. They recommended the use of independent exhaust gas-driven power turbine, that can be shut off at part load, or blowing air from the compressor outlet to the turbine inlet through a controlled bypass system.
Two-stage pressure boosting systems were also tried for large diesel engines. The scavenging pressure required for the pressure boosting of diesel engines can be reached in one or in several stages; [7] and [8]. The use of a power

turbine was also tried in large diesel engines; [9], [10]. The power turbine is opened in the range from 50% to 100% load, and is closed in loads below 50%. Figure (2) sketches the power turbine method.

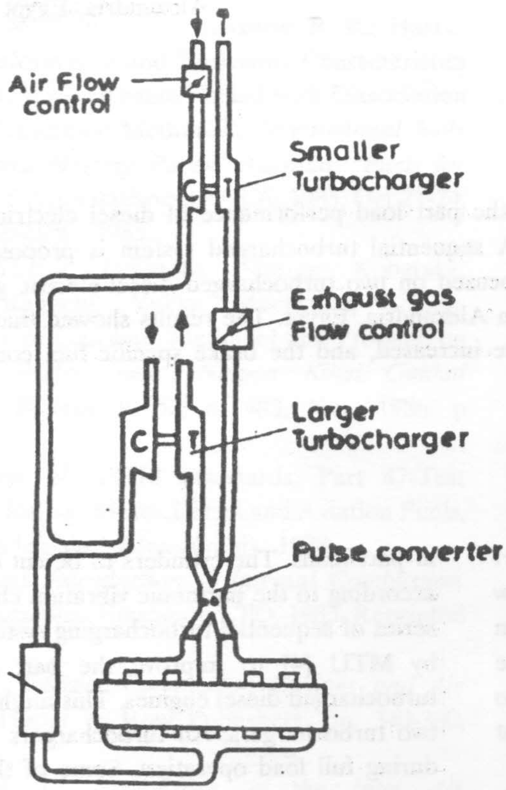

Figure 1. A sequential turbocharging system with turbochargers of unequal size and a pulse convertor.

2. PRESENT WORK

The aim of the present work is to study the improvement of performance of constant speed diesel engines at part load. Two engines were considered, both are installed at Ameria Petroluim Refining company in Alexandria, Egypt. One engine is Blackstone V12, rated at 2148 horse power and 1500 rpm. The other engine is Fuji 8-inline rated at 1800 horse power and 750 rpm.

The addition of an extra turbocharger to the engine, for operation at part loads, was studied keeping the already existing turbocharger unchanged. The idea is to operate the second turbocharger at part load to maintain the same inlet pressure to the engine at both part load and full load operating conditions. The suggested sequential turbocharging system is different in that the two compressors are connected in series and the second compressor is driven by an air turbine. A computer model is constructed to accomodate the proposed sequential turbocharging system. To test the model, the performance data, for the convetional system, obtained from the computer implementation of the model are compared with that of the actual engines performance. The comparison showed that the model and computer program are quite satisfactory.

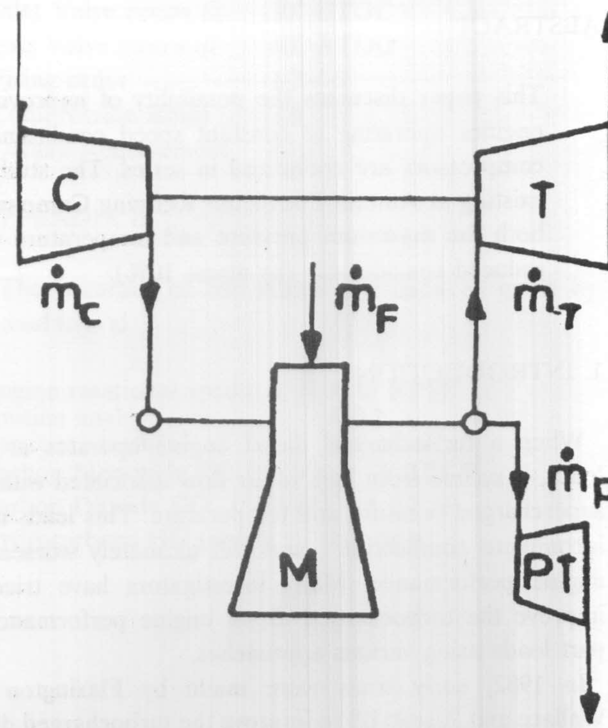

Figure 2. A turbocharger with power turbine.

The results of the present investigation show that the maximum cylinder pressures obtained, using the proposed sequential method, at 50% and 75% loads are near that at full load with a single turbocharger. The maximum cylinder temperature has also higher values at part loads. The temperature of the exhaust gases, delivered to the turbocharger turbine has also increased. The brake specific fuel consumption was reduced by about 10% for the two engines at part loads.

3. THE TURBOCHARGING SYSTEM

Figure (3) shows both the conventional and the proposed sequential turbocharging systems. The conventional turbocharger comprises one turbocharger (T1-C1) and aftercooler for operation at both full and part loads. This conventional system is already installed for the Blackstone and Fuji diesel engines at Amria Petroleum Refining Company.

The proposed sequential turbocharging system comprises an additional turbocharger (T2-C2), where the two compressors act in series in order to maintain the same inlet pressure to the engine at both full and part load operation. In case of part load operation, the aftercooler is by-passed to keep the inlet air temperature near its value at full load operation. The automatic valve (V1) will by-pass part of the exhaust gases according to load variation to maintain the engine inlet pressure about 1.8 bar at all operating range. The automatic valve (V2) will direct the air flow into the new turbocharging system at part load operation. The piping system, following valve (V2), will divide the air mass flow rate between the air turbine (T2) and the second compressor (C2) according to the overall mass conservation of the system.

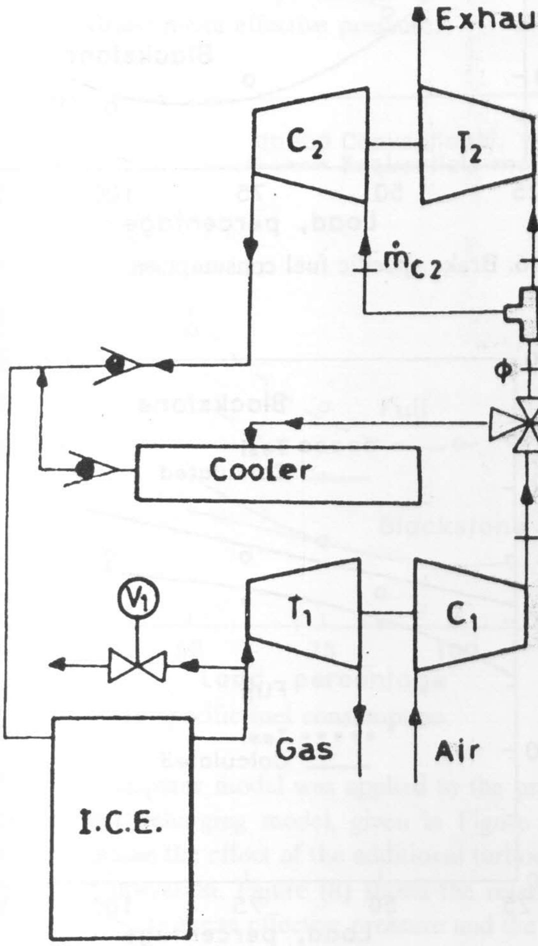

Figure 3. The proposed sequential turbocharging system.

4. DIESEL CYCLE ANALYSIS USING THE PROPOSED SYSTEM

Figure (4) shows the cycle analysis of the sequential turbocharged diesel engine working at part loads. The air at atmospheric conditions, point (a), is compressed in the first compressor adiabatically to point (b). The turbocharger used was VTR 251 type with pressure ratio of 2.4 at 75% load and 1.8 at 50% load. The second compressor was used at part loads to raise the manifold pressure at point (1) to the value 2.95 bar for both 75% and 50% loads. The specific heats for different mole fractions of the working gas mixture are calculated as reported in the appendix.

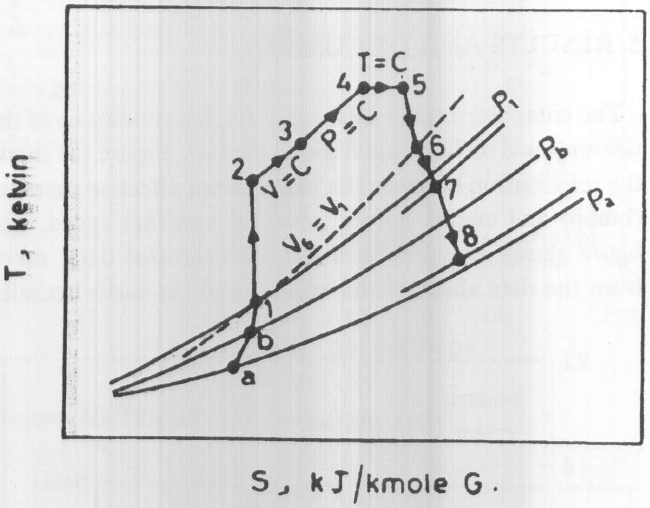

Figure 4. Cycle analysis of the diesel engine using the proposed turbocharging system.

The diesel engine cycle analysis comprises the isentropic compression from point (1) to point (2) according to the engine effective compression ratio (e.g. 12.7 for Blackstone engine). The general combustion equation, with λ being the excess air factor, is

$$C_nH_m + \lambda(n+\frac{m}{4})(O_2+3.76N_2) \rightarrow nCO_2 + \frac{m}{2}H_2O + (\lambda-1)(n+\frac{m}{4})O_2 + 3.76\lambda(n+\frac{m}{4})N_2 \quad (1)$$

from which the lower calorific value of the diesel engine fuel used ($C_{12}H_{26}$) is obtained in kilo joule per kilo mole of product gases as follows:

$$L.C.V. = \frac{7259911.2}{6.5 + 88.06\,\lambda} \qquad (2)$$

The heat added in the conventional diesel engine was assumed to be 40% at constant volume, 40% at constant pressure and 20% at constant temperature. The cooling losses (calculated from the data of the actual engine test sheet) were found to be 19.49% at full load, 21.65% at 75% load and 27% at 50% load. Then an adiabatic expansion from point (5) to point (6) followed by exhaust blowdown from point (6) to point (7). Finally, the expansion in the turbine from point (7) to point (8). It is to be noticed that the turbine inlet pressure (P_7) is equal to or lower than (P_1), while the turbine exit pressure (P_8) is equal to or greater than (P_a).

5. RESULTS AND DISCUSSION

The computer model was tested for the conditions of the conventional turbocharged diesel engines. Figure (5) shows the relationship between the brake mean effective pressure (bmep) and engine output power at constant speed. The figure shows an agreement between the test data, taken from the data sheet of the engine, and the model results.

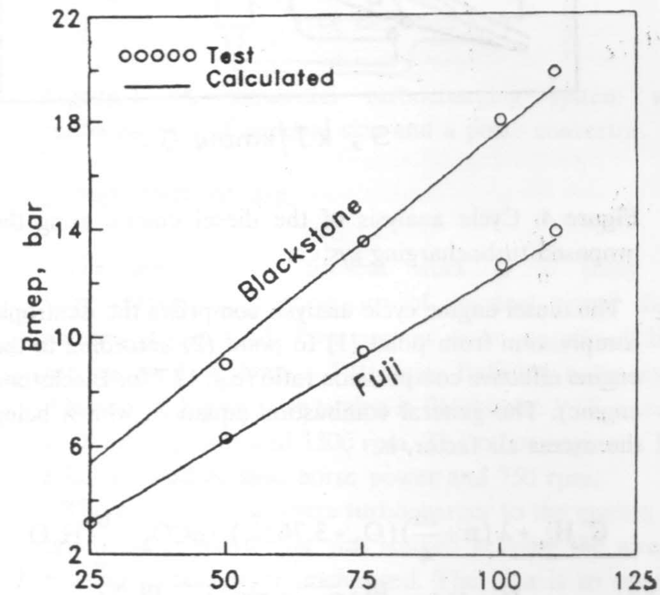

Figure 5. Brake mean effective pressure.

Figure (6) shows that the brake specific fuel consumption (b.s.f.c.) is decreasing rapidly in the part load range, while increases within the overload range. Figure (7) represents the relation between the gas temperature at turbine inlet for both model and data sheet results, and the engine output power. The results shown in Figures (5), (6), and (7) show a good agreement between the model results and the actual engine test runs.

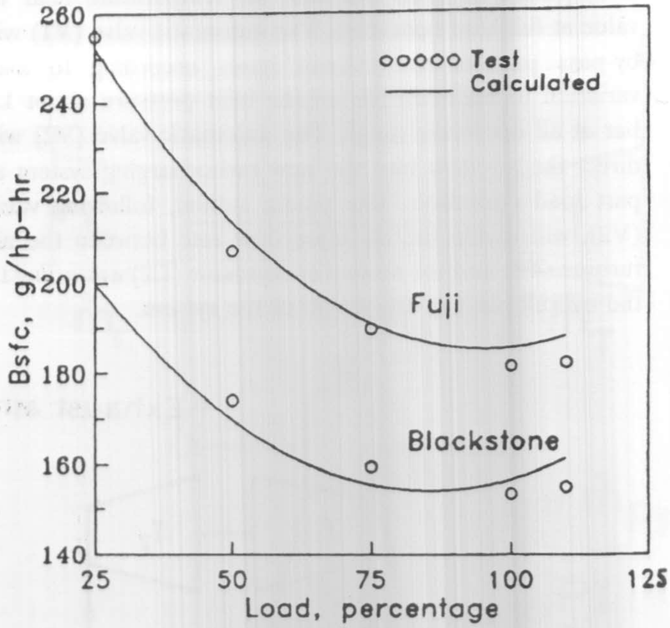

Figure 6. Brake specific fuel consumption.

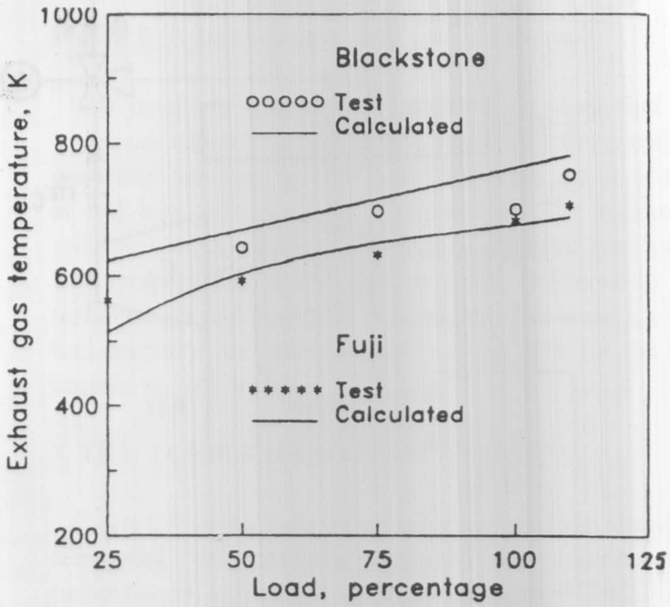

Figure 7. Temperature at turbine inlet.

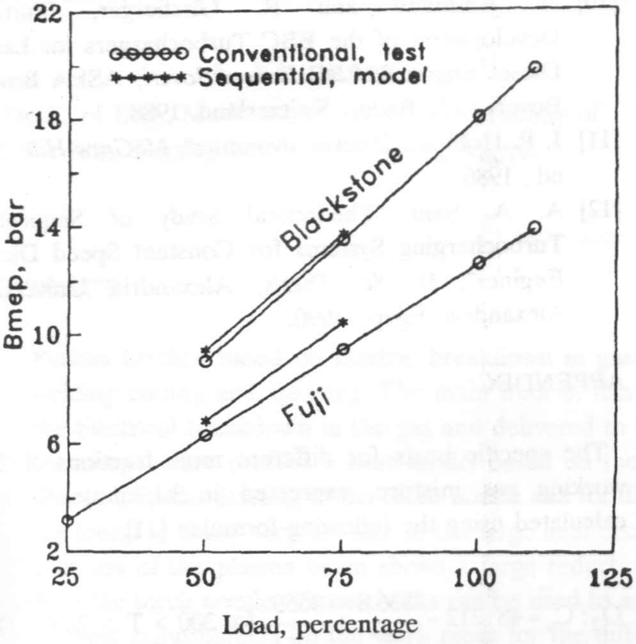

Figure 8. Brake mean effective pressure.

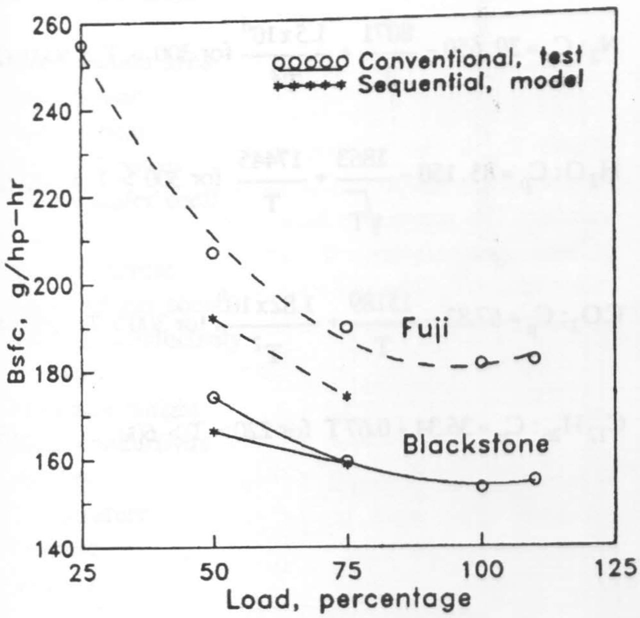

Figure 9. Brake specific fuel consumption.

Then the computer model was applied to the proposed sequential turbocharging model, given in Figure (4), in order to examine the effect of the additional turbocharger at part load operation. Figure (8) shows the relationship between the brake mean effective pressure and the output power. There is a slight increase in bmep at part loads. However, a remarkable reduction in the brake specific fuel consumption can be achieved when the proposed sequential turbocharging system is used, Figure (9). The maximum cylinder pressure at part loads increases due to the sequential turbocharging to a value near that of the full load in the conventional system as can be seen from Figure (10). Due to the increase in the maximum cycle pressure, the exhaust gas temperature and the turbine inlet temperature also increase as shown in Figure (11).

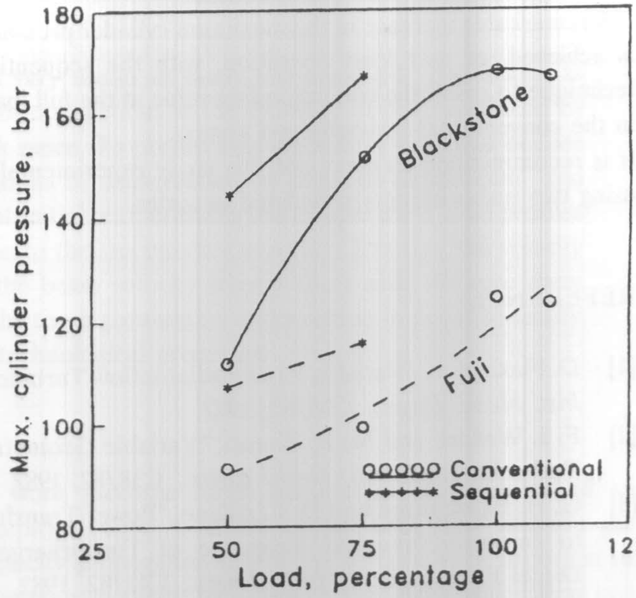

Figure 10. Maximum cylinder pressure.

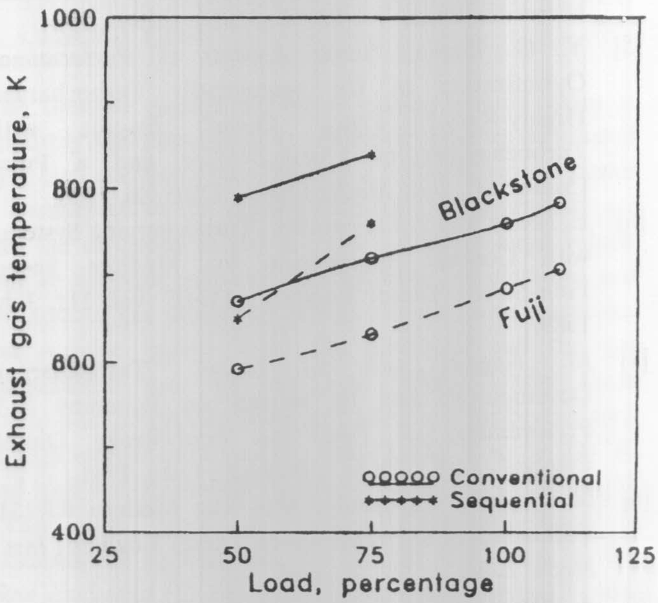

Figure 11. Temperature of exhaust gases at turbine inlet.

6. CONCLUDING REMARKS

The proposed sequential turbocharging technique can be used for the conventional turbocharged diesel engines to improve the part load operating conditions.

The proposed sequential turbocharging technique achieves a remarkable reduction in specific fuel consumption at part loads, particularly in the range of 50% to 75% of the full load operation.

A remarkable increase in the maximum cylinder pressure is achieved, at part load operation, with the sequential technique to reach the corresponding value at the full load in the conventional turbocharged system.

It is recommended to carry out this study experimentally using two turbochargers connected in series.

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APPENDIX

The specific heats for different mole fractions of the working gas mixture, expressed in kJ/kmole K, are calculated using the following formulae [11]:

$$O_2: C_p = 48.212 - \frac{536.8}{\sqrt{T}} + \frac{3559}{T} \quad \text{for } 300 > T > 2800 \qquad (1)$$

$$N_2: C_p = 39.650 - \frac{8071}{T} + \frac{1.5\times10^6}{T^2} \quad \text{for } 300 > T > 5000 \qquad (2)$$

$$H_2O: C_p = 83.150 - \frac{1863}{\sqrt{T}} + \frac{17445}{T} \quad \text{for } 300 > T > 3000 \qquad (3)$$

$$CO_2: C_p = 67.83 - \frac{15189}{T} + \frac{1.82\times10^6}{T^2} \quad \text{for } 300 > T > 2800 \qquad (4)$$

$$C_{12}H_{26}: C_p = 36.34 + 0.67T \quad \text{for } 220 > T > 600 \qquad (5)$$