

A Simulation Model for the Determination of the Air Cell Size of DI Diesel Engines as One of the Means to Control Soot Emission

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ABSTRACT

The soot emission from Diesel engines has become one of the social problems. According to our studies, it is effective, for the soot emission control, to promote the oxidation of the growing carbonaceous particulates at the ending period of the fuel injection. A combustion chamber with an air cell may be one of the feasible combustion systems for this purpose. The authors establish a simulation model of the air jet behavior for a DI Diesel engine with an air cell. The sizes of the air cell and its flow passage may bring considerable effects on the engine operation and performance as well as on the soot emission. The part of the results of the calculation of this model is very similar with the work of Dr. Kamimoto and others. In addition to this part, the authors consider the effect of the temperature for the blowing air jet from the air cell in this paper.

INTRODUCTION

The carbonaceous particulates or soot emission from Diesel engines has become one of the serious environmental problems in the highly developed countries and also even in the developing countries. In addition, through the several times of the oil crisis impacts, people pay more attention to the fuel economy of the oil consuming equipments, such as steam boilers, industrial furnaces, automobiles, trucks, etc. To decrease the specific fuel consumption, the Diesel engine designers are fond of the direct injection type Diesel engines than that of the indirect injection type Diesel engines.

If we can mix fuel with air quickly and let fuel has sufficient oxygen to oxide in the combustion chamber of the DI Diesel engines, it is possible to reduce soot emission from the engines. There may be many ways to obtain the quick mixing of fuel and air, but it is recognized that the effective designs of the intake air passage and combustion chambers are significantly important. Good intake air passage enables the incoming air through the passage to get better turbulence or high vortex flow in the cylinder. And

good chamber design by the suitable combination of the combustion chamber and piston crown shapes may maintain appropriate air turbulence during the compression and expansion periods. To solve soot emission problems thoroughly, it is necessary, first, to understand the soot formation process. Many researches and studies were made on this phenomena [1], [2],[3],[4],[5],[6], the results of these studies pointed out that the soot formation is one of the inherent characteristics for the compression ignition engines, but this does not mean that the emission of smoke and carbonaceous particulates is inevitable. If we are able to burn or oxide these suspended particulates before expelling them from the cylinder, the soot may not be exhausted from the cylinder. Aoyagi et al [7] indicated that the soot formation starts simultaneously with the commencement of the diffusion combustion, and that the soot concentration increases rapidly during the injection period, reaching a maximum amount around the end of injection. The soot is then oxidized at a rapid rate, but after the flame temperature decreases the oxidation hardly proceeds further more and the soot is exhausted at a certain concentration level. According to Chang's investigation [6], the oxidation of the growing particulates in the combustion chamber before the exhaust process is one of the effective means to reduce the soot emission from the engine. On the other hand, Dr. Kamimoto et al [8] developed a DI Diesel engine with an air cell as one of the combustion systems for reducing the soot emission from Diesel engines. But they did not present the mathematical deduction in their paper. In this paper, the authors attempt to establish a mathematical model of a DI Diesel engine with an air cell and use this model to analyze the behavior of air jet from the air cell in the combustion chamber during the combustion period in order to promote the oxidation of the carbonaceous particulates as one of the means to control soot emission. Because the authors believe that this is one of the possible ways to minimize soot emission to some extent. And from this

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analysis, we hope to obtain some data to determine the optimum sizes of the air cell and its flow passage to the main chamber. As indicated in Kamimoto's paper, the pressure differential between the air cell and the main combustion chamber with the injection timing and the momentum of the air jet from the air cell are the important factors to control the extent of the particulate oxidation. In addition to these factors, the authors consider the effect of the temperature for the blowing air jet from the air cell in this paper. It is, generally, recognized that the temperature is too low to ignite the combustible substances, therefore, under certain temperature, the oxidation of the carbonaceous particulates proceed further more, then the soot emission will be held to a certain level.

ASSUMPTIONS

Before establishing the simulation model, the following assumptions are made:

1. The working medium is air, and it is assumed to be an ideal gas.
2. The combustion takes place in the main chamber only.
3. There is no heat loss, i.e., the compression and expansion processes are adiabatic.
4. The air excess ratio is taken as 1.5.
5. Using the fuel with a cetane number of 57, and the pattern of the heat release rate curve as shown in Fig. 1.
6. The timing for exhaust valve close is set at 20° ATDC, and for intake valve close at 38° ABDC; at this instant assuming that the cylinder pressure is atmospheric and the temperature is at the room temperature of 25° C.
7. Engine size is 102 x 106 mm, connecting rod length is 190 mm, and compression ratio is 16.5 without having the air cell.
8. The fuel injection timing is set at 15° BTDC and assuming 5° ignition lag in crank angle.
9. The coefficient of discharge, C_d , is taken as 0.8. [8]
10. The flow through the flow passage of the air cell is a steady flow.

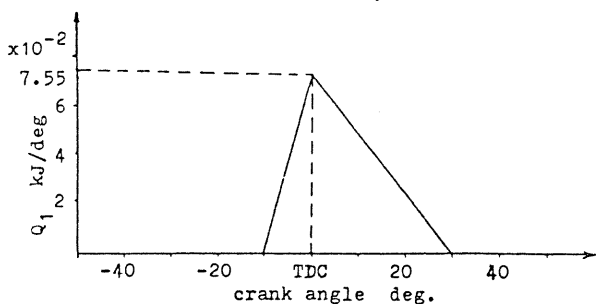


Fig. 1 The pattern of heat release rate

ESTABLISHING A SIMULATION MODEL

The schematic diagram of an engine with an air cell is shown in Fig. 2. In

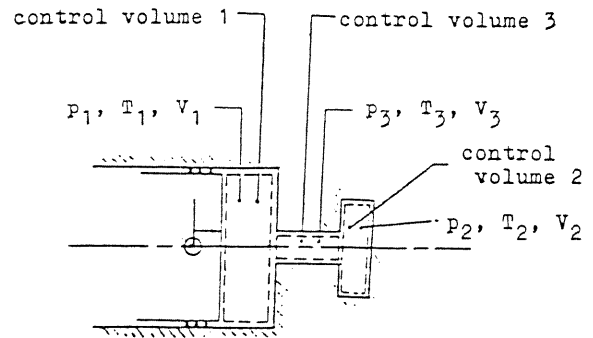


Fig. 2 The engine with an air cell

the operation, air is pressed into the air cell during the compression stroke and is then expelled into the main combustion chamber as an air jet during the expansion stroke when the air pressure in the air cell is higher than the gas pressure in the main chamber. The air jet strikes the fuel particles that are injected into the combustion chamber by the injection nozzle, agitates the fine fuel particles, and promotes the oxidation of the carbonaceous particulates formed and grown during the combustion period, especially near the ending period of the fuel injection.

In Fig. 3,

$$r \sin \theta = L \sin \phi \quad (1)$$

Here, for the convenience of computer calculation, we take $\theta = 0$ at BDC. And

let $r/L = \bar{R}$,
for the displacement of piston, s ,

$$s = r(1 - \cos \theta) + L(B - 1) \quad (2)$$

$$\text{where } B = \sqrt{1 - \bar{R}^2 \sin^2 \theta}$$

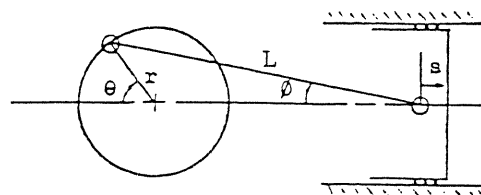


Fig. 3 The crank mechanism

$$\text{and } V_1 = V_0 - A \cdot s$$

$$= V_0 - A[r(1 - \cos \theta) + L(B - 1)] \quad (3)$$

then

$$\frac{dV_1}{dt} = -\omega A r (\sin \theta - \frac{1}{2} \frac{\bar{R} \sin 2\theta}{B}) \quad (4)$$

$$\text{where } \omega = \frac{d\theta}{dt}$$

The combustion chamber with the air cell is divided into three parts, namely, the main combustion chamber, the air cell and the flow passage (or orifice) between the main chamber and the air cell, and designate them as, control volume 1, control volume 2 and control volume 3,

respectively.

From the equations of energy balance, mass balance and state among these three control volumes, the following relations are obtained.

A. When $p_1 > p_2$,

$$\frac{dT_1}{d\theta} = \frac{RT_1\dot{Q}_1 - C_d A_f R^2 p_1 T_1^2 \cdot E + \omega R A r p_1 T_1 (\sin \theta - \frac{1}{2} \frac{\bar{R} \sin 2\theta}{B})}{\omega c_v p_1 \{V_0 - A[r(1 - \cos \theta) + L(B-1)]\}} \quad (5)$$

$$\text{where } E = \left\{ \frac{2k}{(k-1)RT_1} [(p_{i2}/p_1)^{2/k} - (p_{i2}/p_1)^{(k+1)/k}] \right\}^{\frac{1}{2}}$$

$$\frac{dp_1}{d\theta} = \frac{\omega A r p_1 (\sin \theta - \frac{1}{2} \frac{\bar{R} \sin 2\theta}{B}) - C_d A_f R p_1 T_1 E + \frac{p_1}{T_1} \{V_0 - A[r(1 - \cos \theta) + L(B-1)]\} \frac{dT_1}{d\theta} \omega}{\omega \{V_0 - A[r(1 - \cos \theta) + L(B-1)]\}} \quad (6)$$

$$\frac{dT_2}{d\theta} = \frac{RT_2\dot{Q}_2 + C_d A_f R p_1 T_2 E \{c_v (T_{i2} - T_2) + RT_{i2} + \frac{k}{k-1} RT_1 [1 - (p_{i2}/p_1)^{(k-1)/k}]\}}{\omega c_v p_2 V_2} \quad (7)$$

$$\frac{dp_2}{d\theta} = \frac{p_2}{T_2} \cdot \frac{dT_2}{d\theta} + \frac{C_d A_f R p_1 T_2 E}{\omega V_2} \quad (8)$$

$$T_{i2} = T_1 (p_{i2}/p_1)^{(k-1)/k}$$

When $p_2/p_1 \geq [2/(k+1)]^{k/(k-1)}$, take $p_{i2} = p_2$,

and $p_2/p_1 < [2/(k+1)]^{k/(k-1)}$, take $p_{i2} = [2/(k+1)]^{k/(k-1)} p_1$

B. When $p_1 < p_2$,

$$\frac{dT_1}{d\theta} = \frac{RT_1\dot{Q}_1 + C_d A_f R p_2 T_1 E' \{c_v (T_{i1} - T_1) + RT_{i1} + \frac{k}{k-1} RT_2 [1 - (p_{i1}/p_2)^{(k-1)/k}]\}}{\omega c_v p_1 \{V_0 - A[r(1 - \cos \theta) + L(B-1)]\}} + \frac{\omega R A r p_1 T_1 (\sin \theta - \frac{1}{2} \frac{\bar{R} \sin 2\theta}{B})}{\omega c_v p_1 \{V_0 - A[r(1 - \cos \theta) + L(B-1)]\}} \quad (9)$$

$$\frac{dp_1}{d\theta} = \frac{\omega A r p_1 (\sin \theta - \frac{1}{2} \frac{\bar{R} \sin 2\theta}{B}) + C_d A_f R p_2 T_1 E' + \frac{p_1}{T_1} \{V_0 - A[r(1 - \cos \theta) + L(B-1)]\} \frac{dT_1}{d\theta} \omega}{\omega \{V_0 - A[r(1 - \cos \theta) + L(B-1)]\}} \quad (10)$$

$$\frac{dT_2}{d\theta} = \frac{RT_2\dot{Q}_2 - C_d A_f R^2 p_2 T_2^2 \cdot E'}{\omega c_v p_2 V_2} \quad (11)$$

$$\frac{dp_2}{d\theta} = \frac{p_2}{T_2} \cdot \frac{dT_2}{d\theta} - \frac{C_d A_f R p_2 T_2 E'}{\omega V_2} \quad (12)$$

$$\text{where } E' = \left\{ \frac{2k}{(k-1)RT_2} [(p_{i1}/p_2)^{2/k} - (p_{i1}/p_2)^{(k+1)/k}] \right\}^{\frac{1}{2}}$$

$$T_{i1} = T_2 (p_{i1}/p_2)^{(k-1)/k}$$

When $p_1/p_2 \geq [2/(k+1)]^{k/(k-1)}$, take $p_{i1} = p_1$,

and $p_1/p_2 < [2/(k+1)]^{k/(k-1)}$, take $p_{i1} = [2/(k+1)]^{k/(k-1)} p_2$

RESULTS

Using Runge-Kutta method, the above set of non-linear differential equations is solved for T_1 , p_1 , T_2 and p_2 . The effects of the air cell volume and the flow passage diameter are shown in Fig.4. θ_1 represents the crank angle when the air in the air cell begins to flow into the combustion chamber, and θ_2 represents the crank angle for maximum M° . Here, $M^\circ = \dot{M}/\omega = C_d A_f \Delta p / \omega$. Larger diameter of the flow passage or the smaller volume of the air cell causes the earlier θ_1 and θ_2 . The compression ratio of a Diesel engine has to be maintained to some degree, and, generally, low compression ratio is not suitable for a Diesel engine. For the compression ratio above 12, θ_1 ranges between 9° and 22° , θ_2 ranges between 30° and 42° for the air cell volume of 4--20 cc when M° is maximum. The change of the main combustion chamber temperature is very similar to the change of M° in the range of the air cell volume mentioned above, and the temperature at M°_{\max} are in the neighbor of 2500°K . The air temperature in the air cell T_2 is nearly constant and in the range of 1300 - 1800°K . The fuel injection timing seems to have little effect on V_2 and D , as shown in Fig. 5. From these results, we may choose 12 cc for the appropriate air cell volume and 3 mm as for the flow passage diameter. With the air cell, the main combustion chamber temperature, T_1 , increases slightly, because less working medium in the main chamber receives a given quantity of energy input, and air in the air cell keeps low temperature. In Fig. 6, we can find that the temperature difference between the main chamber and the air cell exceeds 1000°K . This may cause low flame temperature and have some possibility to reduce NO_x emission. Fig.7 shows the changes of the main chamber pressure, the air cell pressure and M° with the crank angles. During the compression stroke, air in the air cell increases steadily, and, then, increases suddenly after the commencement of combustion. Passing the point where two pressure curves intercourse, air in the air cell starts to inject into the main chamber.

The effect of heating the air cell is shown in Fig. 8. It results in the increase of T_2 and has little effect on θ_1 and θ_2 , but the decrease in M°_{\max} is slightly larger. Therefore, the reduction of the soot emission may be difficult in this case. On the contrary, if we cool the air cell, as the cooling proceeds, the

air temperature in the air cell decreases, and θ_1 and θ_2 increases slightly, but M°_{\max} increases to the maximum point, and then falls down slowly, as shown in Fig.9. The increase in M°_{\max} may promote the oxidation of the carbonaceous particulates and assist the reduction of the soot emission.

CONCLUSION

In this paper, the authors have established a simulation mathematical model for a DI Diesel engine having an air cell. Using this model, we can find the appropriate volume of the air cell and the suitable diameter of its flow passage for a given size of a DI Diesel engine, and reduce the soot emission from this engine. And, by applying the simulation calculation, we may save much time and much money for the experimentation. Here, we choose 12 cc and 3 mm as the air cell volume and the flow passage diameter, respectively, for a DI Diesel engine with an engine size of 102×106 mm. Referring to the results of Dr. Kamimoto and others, for M°_{\max} we have obtained, it may reduce the soot emission from the engine.

NOMENCLATURE

A	= cross sectional area of the cylinder, m^2
A_f	= cross sectional area of the flow passage between the main chamber and the air cell, m^2
C_d	= coefficient of discharge at the flow passage,
c_v	= constant volume specific heat, $\text{J/kg}\cdot\text{K}$
D	= diameter of the flow passage, m
L	= length of connecting rod, m
\dot{M}	= momentum per unit time, $\text{N}\cdot\text{s}/\text{s}$
M°	= momentum per crank angle, $\text{N}\cdot\text{s}/\text{deg}$
p	= pressure, N/m^2
Q°	= heat added or rejected, kJ/deg
r	= radius of crank circle, m
\bar{R}	= r/L (ratio of crank circle radius to connecting rod length)
s	= displacement of piston, m
t	= time, sec
T	= temperature, $^\circ\text{K}$
V_0	= combustion chamber volume when piston is at BDC, m^3
V	= volume, m^3
θ	= crank angle, deg
k	= c_p/c_v (specific heat ratio)
θ	= angle of connecting rod with cylinder axis, deg
ω	= angular velocity, rad/sec

Subscripts:

i	= inlet,
e	= exit,
1	= property at control volume 1,
2	= property at control volume 2,

3 = property at control volume 3.

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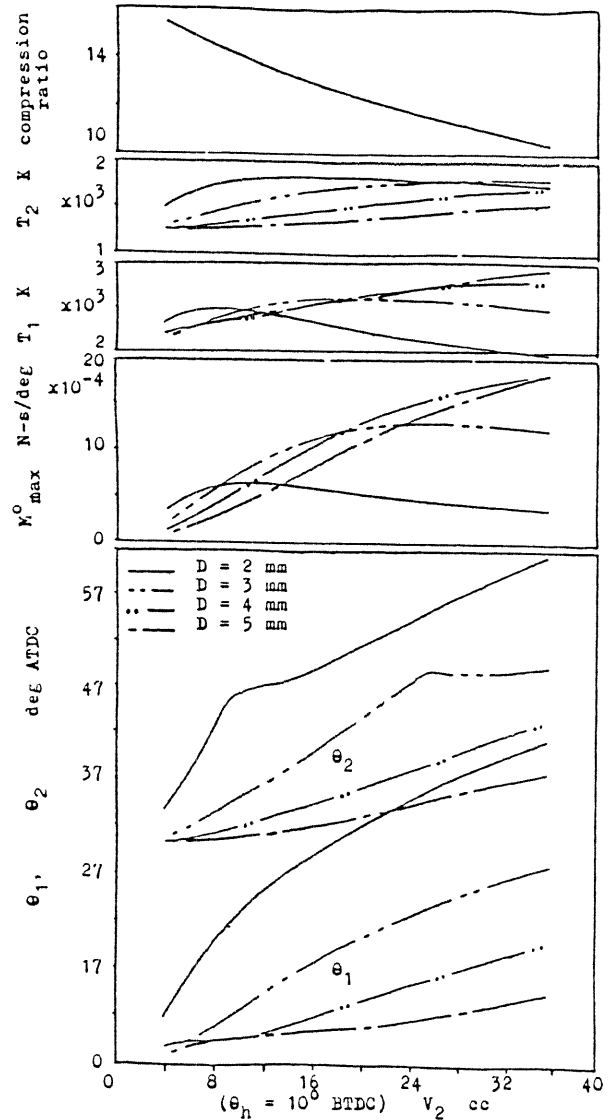


Fig. 4 The effects of the air cell volume and the flow passage diameter

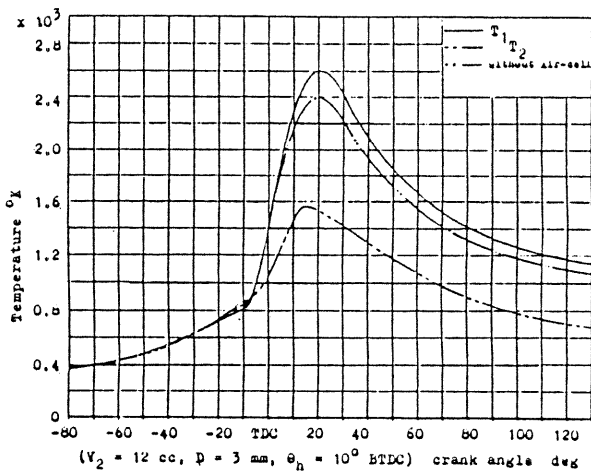


Fig. 6 The temperature changes

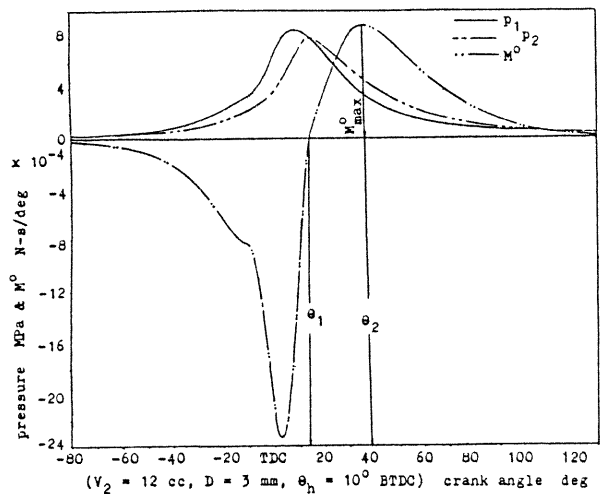


Fig. 7 The changes of p_1 , p_2 and M^0

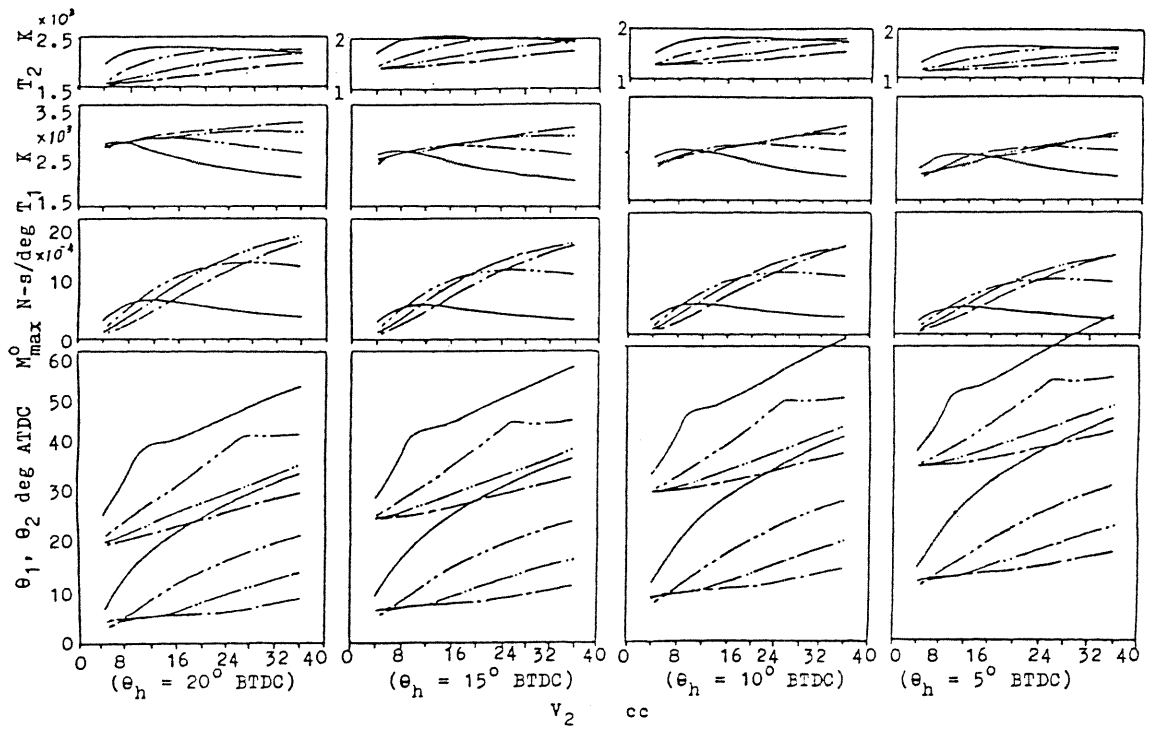


Fig. 5 The effect of fuel injection timing

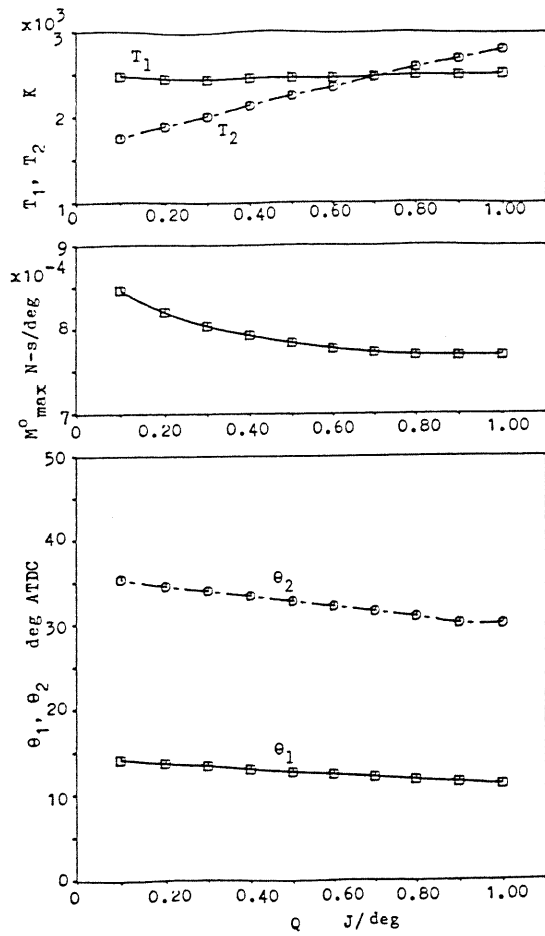


Fig. 8 The effect of heating

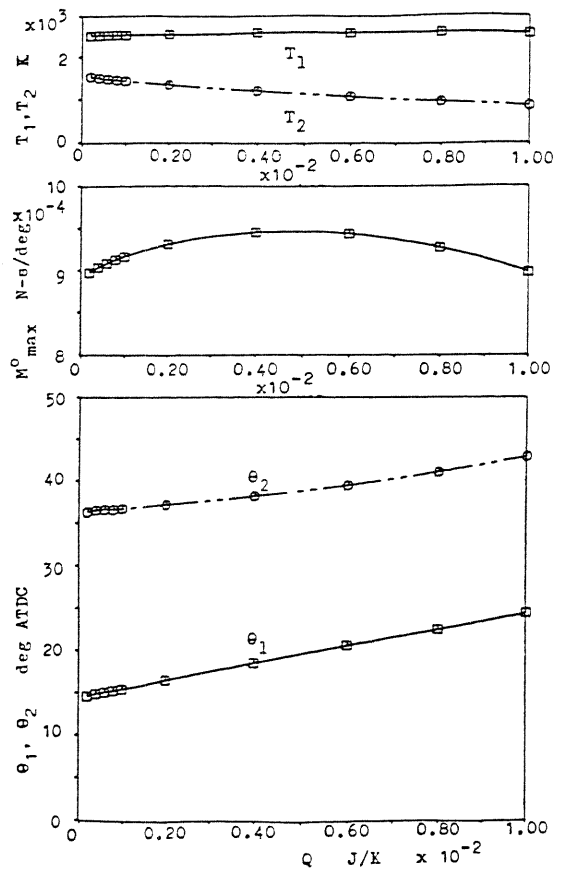


Fig. 9 The effect of cooling