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SECOND LAW ANALYSIS OF IDEAL DIESEL CYCLE

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ABSTRACT

The present paper is an application of the second law of thermodynamics to the air standard diesel cycle. The study investigates the effects of cycle parameters (compression ratio, dimensionless initial and maximum temperatures and initial pressure) on the cycle performance (the first law and second law efficiencies, dimensionless heat added, work done, and available and unavailable energies along the cycle).

The study shows that about half of the energy lost must under any case be given to the surroungings, while some portion of the other half may be regained as work.

INTRODUCTION AND REVIEW

The air standard diesel cycle (Fig.1) is a theoretical thermodynamic representation of the actual diesel cycle, and it is based on the following approximation: The working fluid throughout the cycle is taken to be solely air as a perfect gas (with ideal gas equation of state and constant specific heats). The standard diesel cycle (here-after will be called the cycle), among other standard cycles (e.g. Otto, dual, etc.), is well covered in numerous texts of thermodynamics. In most of these texts (e.g. [2,3,10 and 12]), the principle of conservation of energy (which is the first law of thermodynamics) has been the main tool in the analysis of the cycle. The analysis in these texts is concluded to the first law (thermal) cycle efficiency. When both the first and second laws of thermodynamics are used in the analysis, more parameters of cycle performance appear to be equally important as the first law efficiency. There have been many studies (e.g. [1,4,5,6,7,9 and 11]) of the concepts of second law of thermodynamics and specifically the concept of available energy. No literature was found in detailed second law application to diesel cycles.

The present investigation of the cycle incorporates the first and second

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laws of thermodynamics along with ideal gas equation of state. A dead state which is needed for the second law application is defined as the atmospheric temperature and pressure. The following variables are used as the cycle parameters: compression ratio, dimensionless initial and maximum temperatures and initial pressure. The performance parameters are: first and second laws efficiencies, dimensionless internal, available and unavailable energies and heat added. The effects of cycle parometers on cycle performance are discussed for different levels of cycle parameters. The histories of internal, available and unavailable energies along the cycle are presented for different levels of cycle parameters.

The advantage of this analysis is that the main parameters and their effects which govern the cycle performance are made more apparent. Although this study is highly qualitative, it is still an important tool in the advancement of engineering analysis and design of the actual diesel cycle.

ANALYSIS

Cycle Description

The air standard diesel cycle (a-b-c-d in Fig.1) is an idealized representation of actual diesel cycle. It is composed of: isentropic compression process a-b, followed by isobaric heat addition in process b-c. The expansion proceeds isentropically in process c-d and is followed by isochoric heat rejection in process d-a.

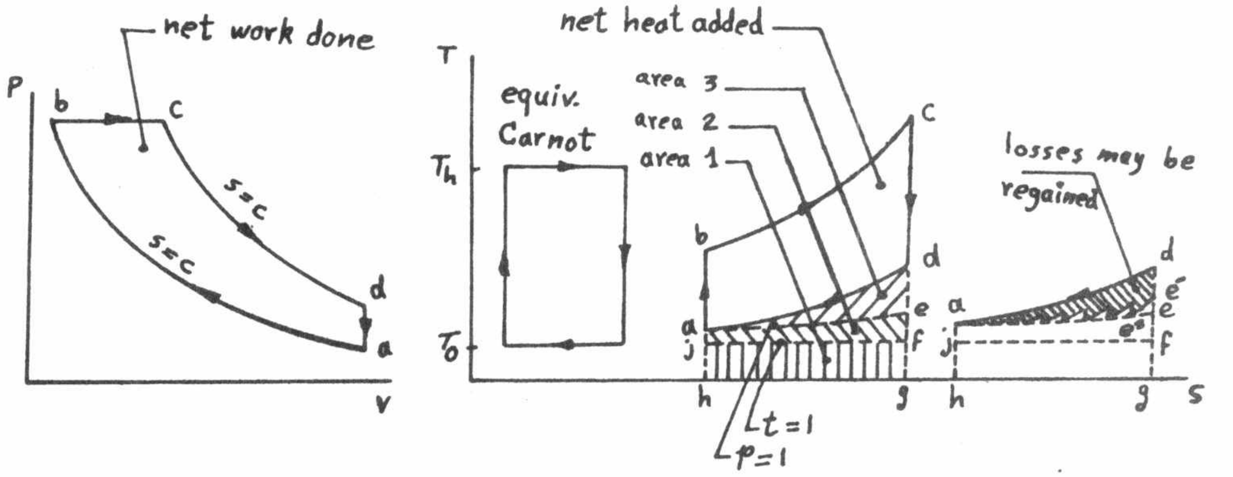

Fig.1 Standard diesel cycle with net heat added, net work done as well as equivalent Carnot cycle.

Governing Equations

The governing equations for cycle analysis are: perfect gas equation of state, the first and second laws of thermodynamics, constant specific heats, first Tds relation and isentropic process; namely:

$$Pv = RT \tag{1a}$$

$$\delta Q = \Delta U + \delta W \tag{1b}$$

$$\Delta A = \Delta U + P_o \Delta V - T_o \Delta S \tag{1c}$$

$$c_p = \text{const.}, \quad c_v = \text{const.} \tag{1d}$$

$$Tds = du + P\,dv \tag{1e}$$

$$P\,v^k = \text{const.} \tag{1f}$$

First Law Efficiency

The first law efficiency of the cycle η_I is defined as the ratio of cycle net work to heat added, i.e.:

$$\eta_I = W_{net}/Q_{add} \tag{2}$$

which can be reduced using Eqns.(1), and the relations $(T_b/T_a) = r^{k-1}$, $(T_c/T_b) = \alpha$ and $t_c = \alpha t_a\, r^{k-1}$ into:

$$\eta_I = 1 - r^{1-k^2} \frac{(rt_c/t_a)^k - (r^k)^k}{(rt_c/t_a) - (r^k)} \tag{3}$$

which relates η_I to k,r, t_a and t_c .

Second Law Efficiency

The second law efficiency η_{II} of the cycle is defined as the ratio of W_{net} to the available energy added A_{add}. i.e.:

$$\eta_{II} = W_{net} / A_{add} \tag{4}$$

Introducing Eqns.(1) into the above equation and rearranging, then:

$$\eta_{II} = \eta_I / (1 - T_o/T_h) = \eta_I / \eta_{eqcar} \tag{5}$$

where $T_h = (T_c - T_b) / \ell_n (T_c/T_b)$ is the logarithmic mean temperature difference (LMTD) of the heat addition process b-c, and η_{eqcar} is the efficiency of an equivalent Carnot cycle shown in Fig.1. The above equation shows that η_{II} is a function in k,r,t_a and t_c.

History of u, a and $\bar{a}$ Along The Cycle

Define dead state energy $U_o = m\, c_v\, T_o$, and dimensionless energies, heat, temperature, pressure and volume as:

$$u = \Delta U/U_o, \quad a = \Delta A/U_o, \quad q = Q/U_o \tag{6}$$

$$t = T/T_o, \quad p = P/P_o, \quad x = V_a/V$$

For process a-b, u can be written as: $u = (T_a/T_o)(T/T_o-1)$. Using the above dimensionless terms, then:

$$u = t_a(x^{k-1}-1), \text{ x varies from r to 1} \tag{7a}$$

represents the history of u along a-b. The available energy a along a-b is written as: $a = u - P_o V_a(x-1)/(U_o)$. Substituting ideal gas relations from Eqns.(1), then the history of a and $\bar{a}$ along process a-b reads:

$$a = u - (k-1)\, t_a(x-1)/(x.p_a), \text{ and } \bar{a} = u - a \tag{7b}$$

where u is given in Equ.(7a). Similar procedure is used to determine the histories of u, a and $\bar{a}$ along the remaining processes b-c-d-a. Results are given in Table 1. The table also shows values of w_{comp}, q_{add}, w_{exp} and q_{rej}.

DISCUSSION

The functional relationships of cycle performance parameters (namely η_I, η_{II}, w_{comp}, w_{exp}, q_{add}, q_{rej}, a and $\bar{a}$) to cycle design and operating parameters (namely r, k, t_a, p_a and t_c) are given in Equs.(3) to (7) and Table 1. Equations (3) and (5) show that p_a has no effect on both η_I and η_{II}.

Effect of Cycle Parameters on η_I and η_{II}

Figures 2 through 6 show the effect of cycle parameters r, k, t_a and t_c on η_I and η_{II}. The effect of r on η_I and η_{II} is shown in Figs.2 to 4 for different values of k, t_a and t_c. In Figs.2 and 3, both η_I and η_{II} increase with r, with η_I is always less than η_{II} by a factor of about %37 which corresponds to $t_c = 10$. High values of k or t_a result in high values of both η_I and η_{II}. Figure 4 shows that both η_I and η_{II} increase with r and η_{II} is always higher than η_I by a factor of about %37 at $t_c = 10$ regardless of the value of r. At $t_c = 8$, η_{II} is higher than η_I by a factor which increases with r. The figure shows that η_{eqcar} at $t_c = 8$ decreases with r, while it remains almost constant at $t_c = 10$. The effects of t_a and t_c on η_I and η_{II} are shown in Figs.5 and 6. Both efficiencies increase with t_a and decrease with t_c for all values of r, with η_{II} is always larger than η_I.

Effect of Cycle Parameters on q_{add}, a_{add} and w_{net}

Figure 7 shows the effect of r on q_{add}, a_{add} and w_{net} at two levels of parameters k, t_a and t_c. It is noted that q_{add} and a_{add} linearly decrease very slightly with r for all levels of k, t_a and t_c, while w_{net} always increases with r. High values of k or t_c and low values of t_a result in high values of q_{add}, a_{add} and w_{net}.

History of u, a and $\bar{a}$ Along The Cycle

Figures 8 through 12 show the history of energy terms u, a and $\bar{a}$ along the cycle for different levels of cycle parameters r, t_a, p_a and t_c as given in Table 1. The cycle progresses from BDC to TDC and back to BDC. The energy

6

terms are assumed to start with zero values at the BDC. With r=12, t_c=10 and ambient initial conditions, Figure 8 shows that about 60% of the total energy supplied to the cycle is lost in the heat rejection process at BDC. About 66% of this lost energy is available energy and the rest is unavailable energy. The figure also shows that the peak energy content in the cylinder (occurs about 50^o after TDC) is about 75% available and 25% unavailable energies. Figure 9 shows two cycles with r=12 and 16 respectively, while other parameters are kept the same for the two cycles. These cycles have almost same peak values of available and unavailable energies occuring at about 50^o and 40^o after TDC respectively. The cycle with r=16 has higher compression work, steeper heat addition and less available energy losses (about 37% of peak available energy is lost) than the cycle with r=12 (about 51% losses). Figure 10 shows similar effect of t_a on two cycles. Figure 11 shows the effect of t_c on two cycles. In that figure, it is shown that increasing t_c from 10 to 12 results in an increase in peak available energy content by about 27% and an increase in available energy losses by about 50%. The initial pressure has the least effect on history of a and $\bar{a}$ as shown in Fig.12. Decreasing p_a from 1.2 to 0.8 results in slightly less peak available energy and same amount of losses.

Cycle Losses

The energy losses of the cycle (during the heat rejection process), as shown in Fig.1 can be explained as follows: (1) Because the heat is introduced at a finite temperature (process b-c), only that part of it corresponding to the Carnot cycle efficieny (area above the constant temperature line T_o), can be converted into shaft work, while the remaining part must in any case be given to the surroundings. This contributes to the first part of the energy losses; area 1 . (2) A further portion of energy (area 2) is lost due to the fact that the temperature is not lowered back by a reversible process to the atmospheric temperature T_o. (3) The last portion (area 3) results from the fact that on account of design limitations, the expansion in the cylinder does not continue to the atmospheric pressure p_o.

The relative amounts of these areas can be investigated using Fig.1 as follows: Applying the following relationships:

$$Tds = du + Pdv, \quad Tds = dh - v\,dP \text{ and } \Delta S_{12} = c_v.\,\ell n\,(T_2/T_1) + R.\,\ell n\,(v_2/v_1)$$

to processes d-a and e-a, then for process d-a; (constant volume process): $Tds = du$ and $T_d/T_a = \exp(\Delta s/c_v)$, and for process e-a: $Tds = dh$ and $T_e/T_a = \exp(\Delta s/kc_v)$. Using the last results, the following integrals can be evaluated:

$$\text{area } 1 = \text{area j-f-g-h} = \int_j^f T\,ds = T_o\,\Delta s$$

$$\text{area } 1{+}2 = \text{area a-e-g-h} = \int_a^e T\,ds = c_p\,T_a\,[\exp(\Delta s/c_p)-1]$$

$$\text{area } 1{+}2{+}3 = \text{area a-d-g-h} = \int_a^d T\,ds = c_v\,T_a\,[\exp(\Delta s/c_v)-1]$$

The last equation is the total losses from the cycle, while area 1

represents the lost energy which must in all cases be given to the surroundings. Normally, for the actual diesel cycle, the exhaust temperature is approximately 3 to 4 times the initial temperature, i.e. T_d/T_a = 3 to 4. Assuming $T_a = T_o$ and $T_d/T_a = 3.5$, then the last three equations reduce to:
area 1 = 50% of total losses
area 2 = 30% of total losses
area 3 = 20% of total losses
The final point to be mentioned here is that, part of the losses can be recovered (equals to shaded area in Fig.13) by using an exhaust turbine in the heat rejection process (which will be driven by pressure drop d-e' and then the air is cooled and recompressed in stages (e',e",...) to P_o.

CONCLUSIONS

The following conclusions are based on the analysis discussed in the preceeding sections:

1. A far deeper and better understanding of the effect of cycle parameters on its performance has been achieved through the application of the second law of thermodynamics to the cycle analysis.

2. The study shows that the controlly parameters of the cycle k, r, t_a and t_c greatly affect the cycle performance parameters η_I, η_{II}, u, a, $\bar{a}$, q_{dd} and w_{net}. The initial pressure p_a has no effect on either η_I or η_{II} and has a slight effect on the other performance parameters.

3. The analysis shows that both η_I and η_{II} increase with r and η_{II} is always larger than η_I by a factor = $\eta_{eqcar} = 1 - T_o/T_h$ which is the efficiency of an equivalent Carnot cycle working between temperature limits T_o and T_h = LMTD of the heat addition process b-c of the cycles.

4. High values of r,k or t_a and low values of t_c result in improving cycle performance (i.e. high work output and less heat added and available energy added).

5. With same initial conditions and maximum temperature = 10, increasing r from 12 to 16 results in decreasing available energy losses by about 14% .

6. With r = 12, t_a = 1 and t_c = 12, about half of the peak available energy content of the cycle is lost in the heat rejection process.

7. Decreasing t_a from 1.2 to 0.8 while keeping r = 12, t_c = 10 and p_a = 1 results in a net decrease of available energy losses of about 12%.

8. Peak available energy content of the cycle increases with the maximum cycle temperature t_c .

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NOMENCLATURE

Symbol	Meaning	Dimensionless form
A	available energy	$a = A/U_o$
$\bar{A}$	unavailable energy	$\bar{a} = \bar{A}/U_o$
BDC	bottom dead center	
c_p	const. press. sp. heat	
c_v	const. vol. sp. heat	
k	specific heat ratio c_p/c_v	
LMTD	log. mean temp. difference	
m	mass of cylinder content	
P	pressure	$p = P/P_o$
Q	heat transfer	$q = Q/U_o$
R	gas constant	
r	compression ratio = V_a/V_b	
S	entropy	
s	specific entropy	
T	temperature	$t = T/T_o$
TDC	top dead center	
U	internal energy	$u = U/U_o$
V	volume	
v	specific volume	
W	work done	$w = W/U_o$
x,y,z	parameters in Table 1	
η	efficiency	

Differential Operators

Δf	total change of a point function f
δf	total change of a path function f
df	differential change of a function f
$\ln(f)$	natural logarithm of f

Subscripts

0	atmospheric (dead state) condition.
I	first law
II	second law
add	amount added
car	Carnot cycle
comp	compression
eqcar	equivalent Carnot cycle
exp	expansion
h	high temp. of Carnot cycle
i	at state "i" of the cycle, i = a,b,c and d
net	net amount

Table 1. History of u, a, and $\bar{a}$ along the cycle.

Process	u	a	$\bar{a}$	heat and work	remarks
a-b	$t_a(x^{k-1}-1)$	$u-t_a(k-1)(x-1)/(x.p_a)$	u - a	$w_{comp} = -u$	$x = 1 \rightarrow r$
b-c	$t_c(y-1)/\alpha$	$u-t_c(k-1)(1-y)/(r^k.\alpha.p_a)-k\ \ell n\ y$	u - a	$q_{add} = u$	$y = 1 \rightarrow \alpha$
c-d	$t_c(z^{1-k} - 1)$	$u-t_c(k-1)(1-z)/(r^k.p_a)$	u - a	$w_{exp} = u$	$z = 1 \rightarrow r/\alpha$
d-a	$t_a\ \alpha^k\ (y^{-k} - 1)$	$u + k\ \ell n(y)$	u - a	$q_{rej} = -u$	$y = 1 \rightarrow \alpha$

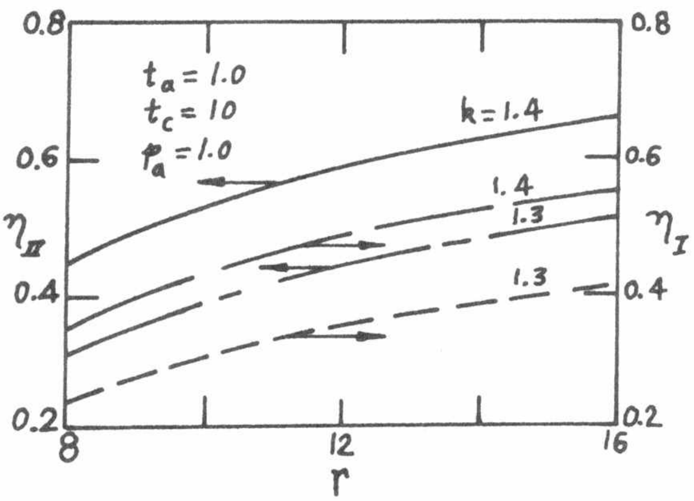

Fig.2. η_I and η_{II} versus r for different values of K.

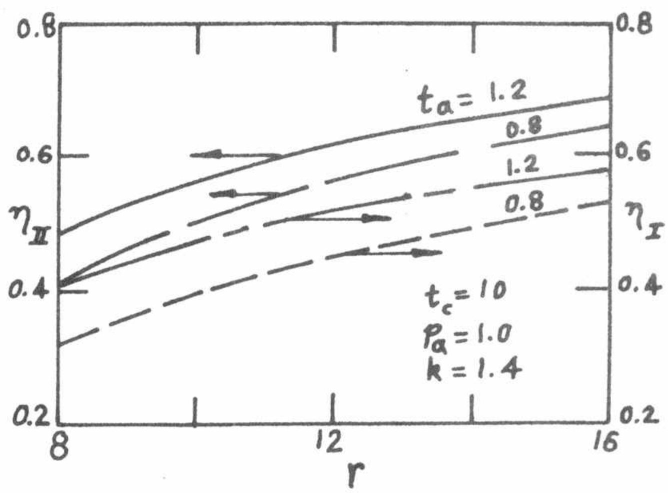

Fig.3. η_I and η_{II} versus r for different values of t_a.

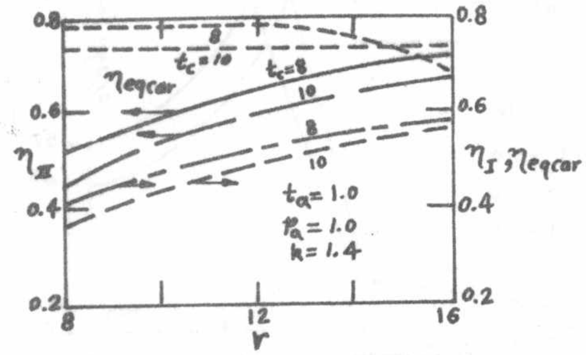

Fig.4. Effect of r on η_I, η_{II} and η_{eqcar} for different t_c.

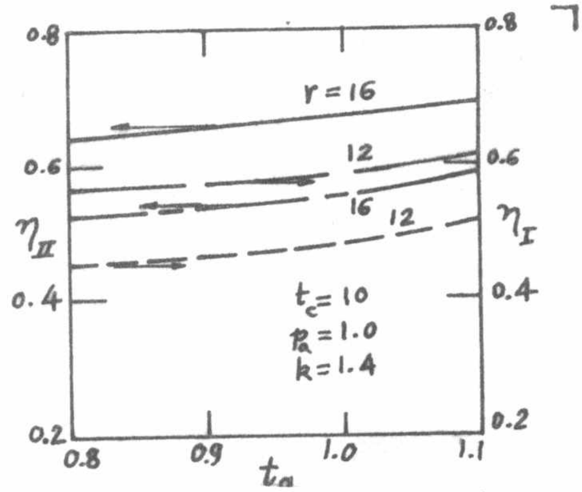

Fig.5. η_I and η_{II} versus t_a.

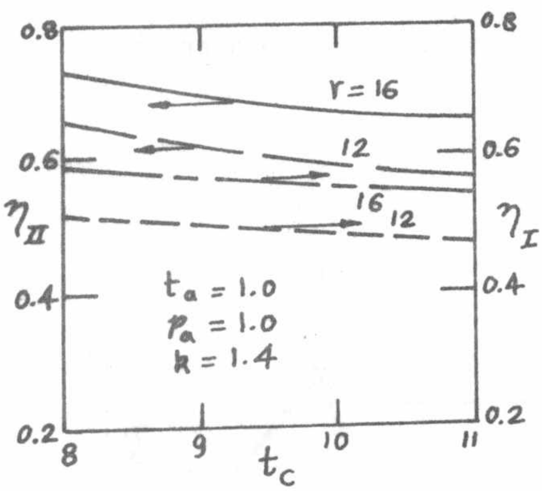

Fig.6. η_I and η_{II} versus t_c.

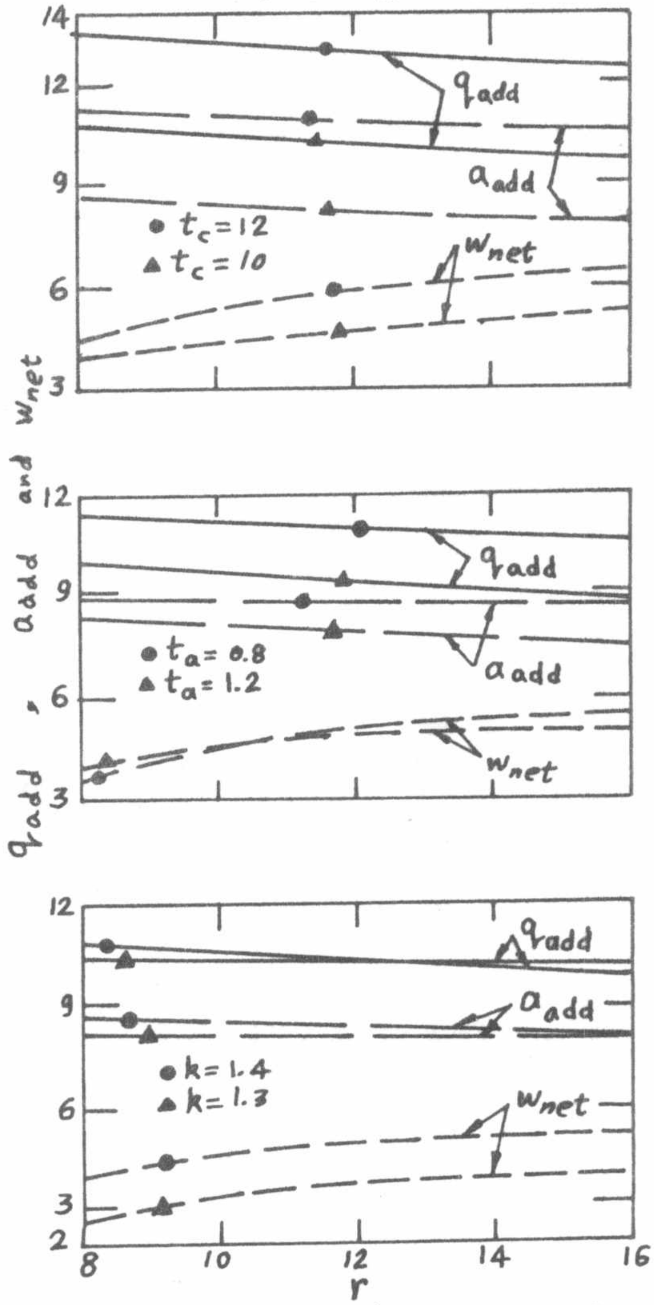

Fig.7. q_{add}, a_{add} and w_{net} versus r .

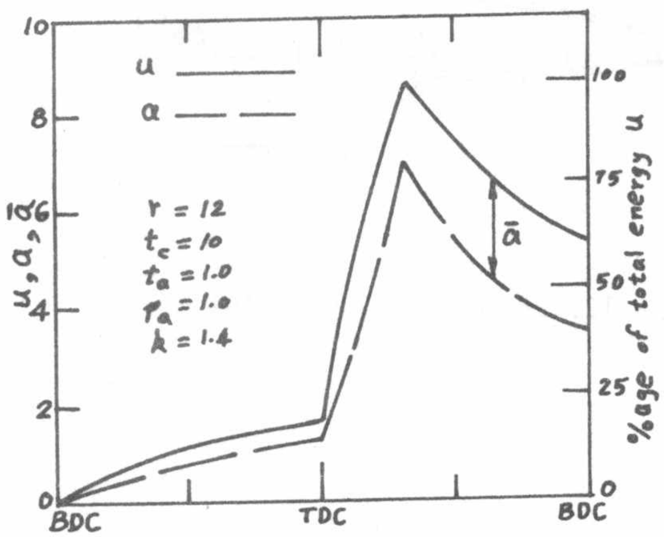

Fig.8. History of u, a and $\bar{a}$ along the cycle.

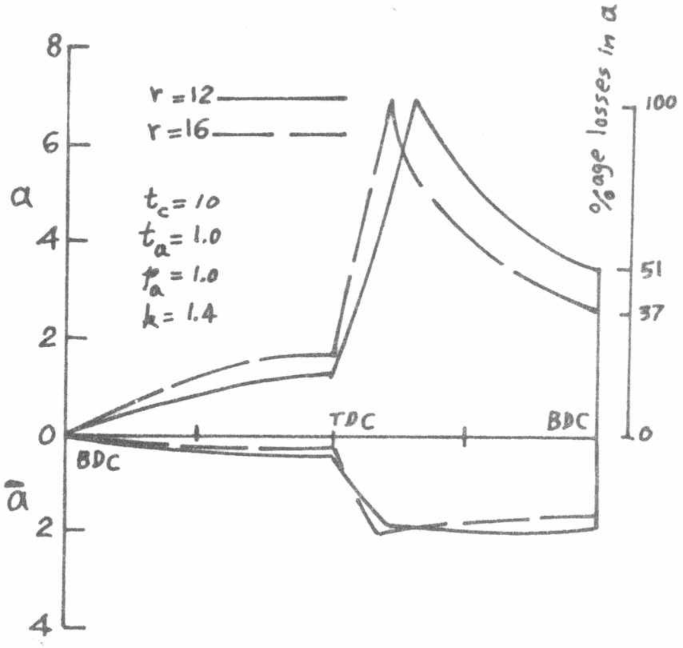

Fig.9. Effect of r on a and ā .

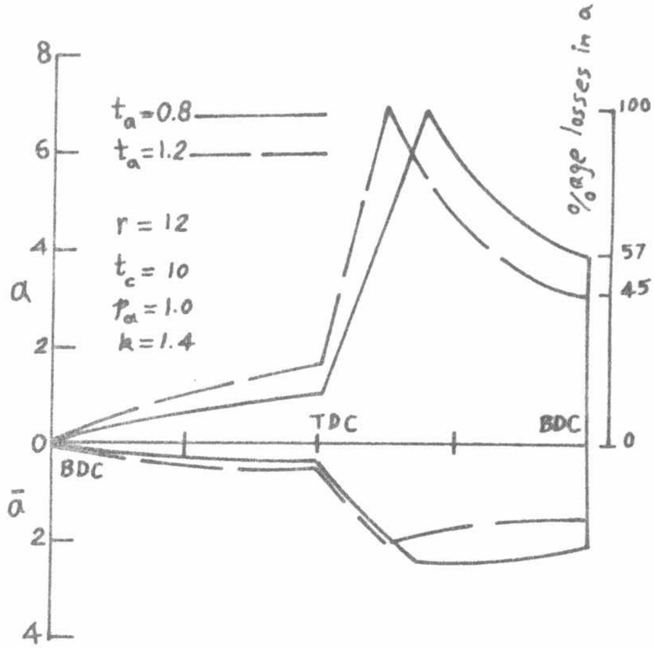

Fig.10. Effect of t_a on a and ā .

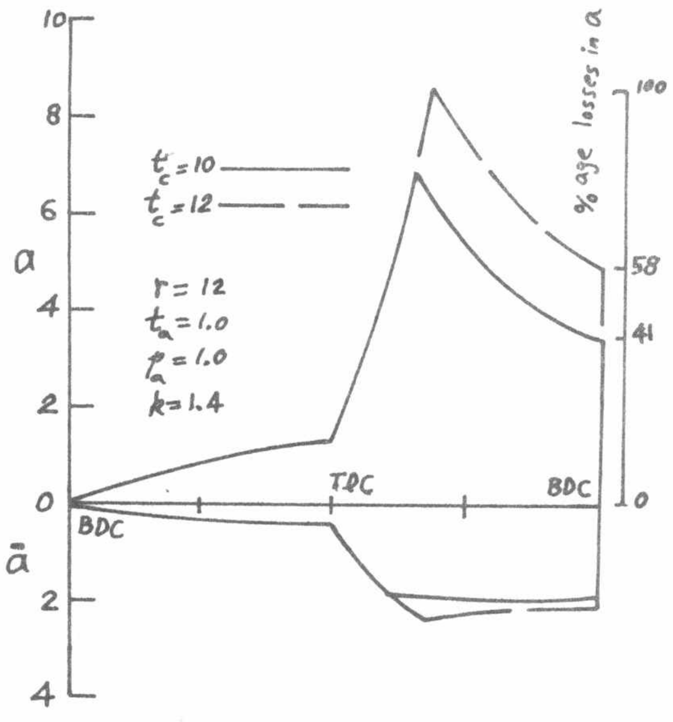

Fig.11. Effect of t_c on a and ā .

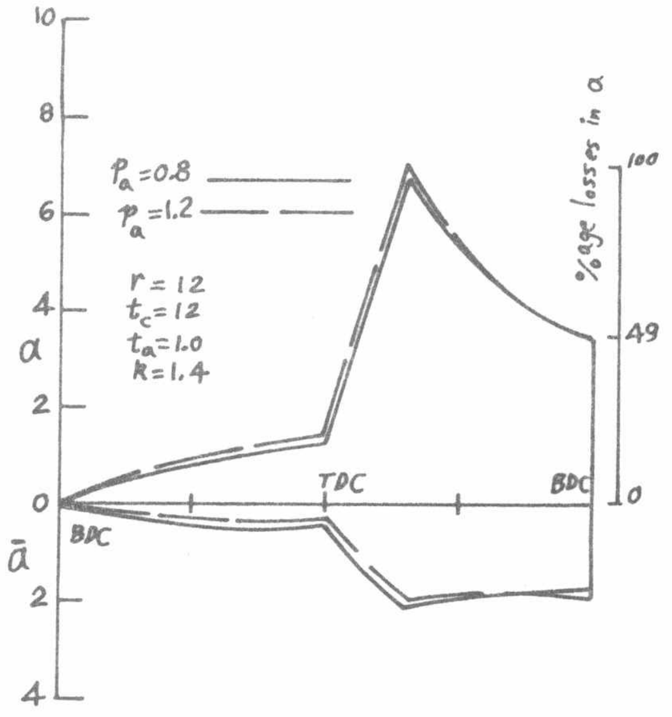

Fig.12. Effect of p_a on a and ā .