The Iraqi Journal For Mechanical And Material Engineering, Vol.11, No.3, 2011

PERFORMANCE IMPROVEMENT OF A TURBO SHAFT ENGINE USING WATER INJECTION

Dr.Eng. Talib Z.Farge Assist. Prof. **Dr.Ismail I.Shibib** Lecturer **Msc.Eng. Mohammed A. Fayad** Assist. Lecturer **Msc.Eng. Majid A. Alwan** Assist. Lecturer

Fuel and energy research center
University of Technology
Iraq-Baghdad

ABSTRUCT

A theoretical study to calculate the performance of the turbo-shaft engine (TV3-117 MT), used to power the helicopter (MI-17). It is clear that the engine works less efficiently in a hot ambient especially during takeoff because the engine at takeoff needs maximum power. The engine performance is also calculated after water injection at compressor entrance. A new mixture (water and air), the dew point temperature has also been calculated at compressor entrance and in every stage to ensure that no condensation of water vapor is take place on the blades of the compressor. To satisfy this aim the amount of water injection should not exceed (0.8%) of air mass flow rate depending on the permissible payload of the engine.

The performance with water injection was calculated at different amounts of water injection (0.2 %, 0.5 %, 0.7 %, and 0.8 % of air mass flow rate with constant temperature (10 C^0)).The results showed that an increase in the amount of water injection leads to decrease in the mixture temperature and to an increase in the specific heat at constant pressure.

The percentage improvement in the performance for the specific power, thermal efficiency and specific fuel consumption at (Ta=55 C^0) and $\dot{m}$ =0.8% of air mass flow rate at the entrance of the compressor were (14.7, 12.6 and 11.3) respectively .

الخلاصة

تضمنت هذه الدراسة حساب أداء المحرك التوربيني المحوري (TV-117) و الذي يعمل على بعض الطائرات السمتية و من ضمنها الطائرة السمتية (MI-17) . حيث لوحظ ان عمل المحرك يتأثر تأثيرا كبيرا في الأجواء الحارة و خاصة عند الاقلاع لان المحرك يحتاج الى اقصى قدرة. تم حساب أداء المحرك بعد أجراء عملية الحقن المائي عند مدخل الضاغطة. لقد تم حساب مواصفات الخليط الجديد (ماء وهواء) ودرجة حرارة الندى عند مدخل الضاغطة و جميع مراحلها و ذالك لضمان عدم حدوث إي عملية تكثيف للماء المحقون على أسطح ريش الضاغطة . و على هذا الأساس تم تحديد كمية الماء المحقون على أن لا تتجاوز (0.8 %) من كتلة الهواء الجاري، و تحديد هذه الكمية على ضوء سماحة الحمل الإضافي للطائرة السمتية (MI-17).

تضمن البحث حساب أداء المحرك بعد عملية الحقن المائي و لكتل مختلفة من الماء المحقون(0.2% ,0.5% ,0.7% , 0.8% من كتلة الهواء الداخل) و لدرجات حرارة الهواء مختلفة (45 ,50 ,60 م)حيث بينت النتائج إن زيادة كتلة الماء المحقون تؤدي إلى نقصان في درجة حرارة الخليط و زيادة في المحتوى الحرارة عند ثبوت الضغط ، و كذالك زيادة نسبة التحسين في الأداء عند درجات الحرارة العالية للمحيط.

ان نسبة التحسن في الاداء للقدرة الناتجة والكفاءة الحرارية ومعدل صرف الوقود عند درجة حرارة ($Ta= 55\ C^0$) وكمية للماء المحقون 0.8% من كمية الهواء الداخل عند مدخل الضاغطة هي (14.7و 12.6 و 11.3) على التوالي.

Keywords, Performance, gas turbine, water injection, dew point

1. INTRODUCTION

Methods of increasing the thrust of turbojet and turboshaft engines for short periods of time have become of increasing importance in improving the effectiveness of this type of engine power plant. Thrust augmentation methods have found their principal application in improving the take off and climb characteristics of turboshaft aircraft as well as improving the combat and high speed performance of many military aircraft. Clinton etc. [1950] investigated the performance of a turbojet utilizing thrust augmentation by water injection at the compressor inlet was evaluated over a range of flight Mach numbers and altitudes and for a range of water injection rates.

Reece and Hensly [1952] evaluated the theoretical performance with inlet water injection of an axial flow compressor operating as a component of a gas turbine engine for a normal compressor pressure ratios of (4, 8, and 16) continuous saturation throughout the compression process. A theoretical analysis based on the assumptions of choked turbine nozzles and a compression efficiency dependent on the amount of evaporative cooling was used to evaluate the performance of an axial flow compressor operating in a gas turbine engine with inlet water injection and compressor-outlet saturation. Fortin and Bardon [1983] studied and evaluated a more practicable configuration in which the alcohol was injected between stages of multistage compressor, so that due to the high air temperatures, evaporation was complete before the mixture enters subsequent stages. The study showed that the percentage improvement in a cycle thermal efficiency is higher between the first and second stages and reduce toward the last stages.

Ansari [2003] studied the modification of the order to obtain higher efficiency for gas turbine cycle. The experimental results showed that as the mass spray water increased the fuel consumption decreased. Van Liere and Laagland [1998] described technology for over spray injection of gas turbines, called swirl flash technology. It uses hot pressurized water, to obtain tiny droplets, which behave like aerosols. When injected in a compressor the evapouration rate is extremely high and blade impact is avoided. The cooler and humidified compressed air provide also better cooling conditions for the hot gas components, thus increasing life time and reducing maintenance costs.

2. THEORY

The water injection in inlet of the compressor of gas turbine and mixing with air are related with gas-vapor mixture flow. It is therefore necessary to study the new parameters of this mixture because it is very necessary in the performance calculation of engine after injection.

2.1 Calculation of Mixture Temperature (t_m)

The enthalpy of air h_a can be calculated from the equation:

$$h = h_a + W_a h_g \tag{1}$$

$$h = Cp_a t_a + W_a h_g \tag{2}$$

$$Cp_a t_a = 1.007 t_a - 0.026 \tag{3}$$

The above equation (3) was obtained from paper No. 564 American International Measuring Specification office [Ashrae, 1997]. The enthalpy of evaporation was calculated (h_g)

From the table values as;

$$h_g = 2501 + 1.84 t \tag{4}$$

Substitution equations (2) and (3) into equation (1) gives:

$$h_a = (1.007 t_a - 0.026) + W_a (2501 + 1.84 t_a) \tag{5}$$

The enthalpy of a mixture (h_m) of perfect gases equals the sum of individual partial enthalpies of the components:

$$H_m = H_a + H_w \tag{6}$$

$$\dot{m}_m h_m = \dot{m}_a h_a + \dot{m}_w h_w \tag{7}$$

$$\dot{m}_m = \dot{m}_a + \dot{m}_w \tag{8}$$

From equation (7)

$$h_m = \frac{m_a^{\bullet} h_a + m_w^{\bullet} h_w}{m_m^{\bullet}} \tag{9}$$

The humidity of mixture (W_m) can be found from the conservation of mass flow rate equation of water in a humidifying section and can be expressed as: [Khaled Al Judi, (1986)]

$$m_m^{\bullet} W_m = m_a^{\bullet} W_a + m_w^{\bullet}$$

$$W_m = \frac{m_a^{\bullet} W_a + m_w^{\bullet}}{m_m^{\bullet}} \tag{10}$$

The temperature of mixture can be calculated in the similar method of equation (5) as:

$$h_m = (1.007 t_m - o.026) + W_m (2501 + 1.84 t_m) \tag{11}$$

2.2 Calculation of specific heat mixture C_{pm}

The energy balance of mixture is, [Yunus, A. Cengel and Michael,1997].

$$\dot{m}_m \quad C_{pm} T_m = \dot{m}_a C_{pa} + \dot{m}_w C_{pw} T_w \tag{12}$$

$$C_{pm} = \frac{\dot{m}_a C_{pa} T_a + \dot{m}_w C_{pw} T_w}{\dot{m}_m T_m} \tag{13}$$

And the specific heat ratio of mixture is:

$$\gamma_m = \frac{C_{p_m}}{C_{p_m} - R} \tag{14}$$

2.3 Calculation of dew point temperature (t_d):

The dew point temperature (t_d) for moist air with humidity ratio (w) and Pressure (P) was defined earlier as the solution t_d (P,w).for perfect, it is reduced to ,[Ashrae,1997].

<u>Partial Pressure P_p</u>

Humidity ratio

$$W = \frac{\dot{m}_w}{\dot{m}_a} \tag{15}$$

$$P_{pp} = \frac{P * W}{0.622 + W} \tag{16}$$

The dew point temperature can be calculated from stream tables or directly by the following equation,[Hand Book,1997].

$$t_d = 6.54 + 14.526 * \ln(P_p) + 0.7389 * (\ln(P_p))^2 + 0.09486 * (\ln(P_p)^3) + 0.4569 * (P_p)^{0.1984} \tag{17}$$

2.4 Dew point through stages

The dew point in front and through the stages of the compressor must be found to ensure that dew point of temperature does not exceed the temperature does not exceed the temperature of mixture. This is required for calculation the total temperature, total pressure, and partial pressure for every stage.

An axial-flow compressor stage consist of a rotor followed by a stator fig.(1) shows the (t-s) diagram for compressor stage,[Fortin, 1983].

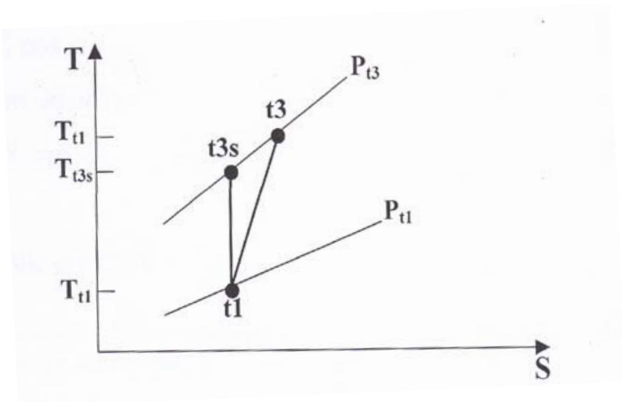


Fig. (1) state for definition of compressor stage.

The velocity diagram as shown in fig.(2).

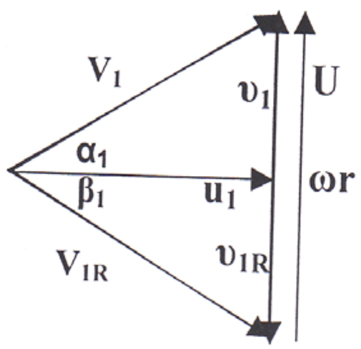


Fig. (2) velocity diagram.

The static temperature at inlet of first stage of the compressor is:

$$T_1 = \frac{T_{t1}}{1 + [(\gamma - 1)/2]M_1^{\ 2}} \tag{18}$$

The speed of sound:

$$a_1 = \sqrt{\gamma R T_1} \tag{19}$$

The absolute velocity of air :

$$V_1 = M_1 a_1 \tag{20}$$

The axial velocity:

$$u_1 = V_1 \cos\alpha_1 \tag{21}$$

The flow velocity:

$$\upsilon_1 = V_1 \sin\alpha_1 \tag{22}$$

The static pressure:

$$P_1 = \frac{P_{t1}}{[1 + [(\gamma - 1)/2]M_1^{\ 2}]^{\gamma(\gamma-1)}} \tag{23}$$

The mass flow parameter:

$$MFP(M_1) = \frac{MFP(M_1) * \sqrt{R/g_c}}{(\sqrt{R/g_c})_{S.A}} \tag{24}$$

The flow annulus area:

$$A_1 = \frac{m\sqrt{T_{t1}}}{P_{t1}(\cos\alpha_1)MFP(M)} \tag{25}$$

The relative flow velocity:

$$\upsilon_{1R} = \omega r - \upsilon_1 \tag{26}$$

The cascade flow angle

$$\beta = \tan^{-1}\frac{\upsilon_{1R}}{u_1} \tag{27}$$

The relative absolute velocity:

$$V_{1r} = \sqrt{u^2 + \upsilon_{1R}{}^2} \tag{28}$$

The relative Mach number:

$$M_{1R} = \frac{V_{1R}}{a_1} \tag{29}$$

The relative total temperature:

$$T_{t1R} = T_1 * (1 + \frac{\gamma - 1}{2} M_{1R}{}^2) \tag{30}$$

The relative total pressure:

$$P_{t1R} = P_1 * (\frac{T_{t1R}}{T1})^{\frac{\gamma}{(\gamma-1)}} \tag{31}$$

The total temperature of flow at the end of stage one:

$$T_{t3} = T_{t2} = T_{t1} * (\frac{P_{t2}}{P_{t1}})^{\frac{(\gamma-1)}{\gamma}} \tag{32}$$

The same equations from (eq. 18) to eq. (32) are used to calculate the properties of flow in all next stages of the compressor.

Assumption:

- The water injection is homogeneous and completely evaporated water injection at inlet of compressor.
- The air is to be assumed as a real gas after water injection.
- The total compressor ratio is constant.
- The inlet turbine temperature is constant.
- The efficiencies of all stages of engine are constant.

3. CYCLE ANALYSIS OF TURBO SHAFT ENGINE:

Parametric cycle analysis is used to determine how the engine performance (specific thrust and fuel consumption) varies with changes in flight conditions (e.g. Mach number) design limits (e.g. main burner exit temperature). Component performance (e.g. turbine efficiency), and design choices (e.g. compressor pressure ratio).

Starting with an equation of an uninstalled engine thrust, we rewrite this equation in terms of the total pressure and total temperature ratios: the ambient pressure (P_0), temperature (T_0), and speed of sound (a) and the flight Mach number (M_0).
For analysis, consider the turbo shaft engine as shown in Fig. (3).

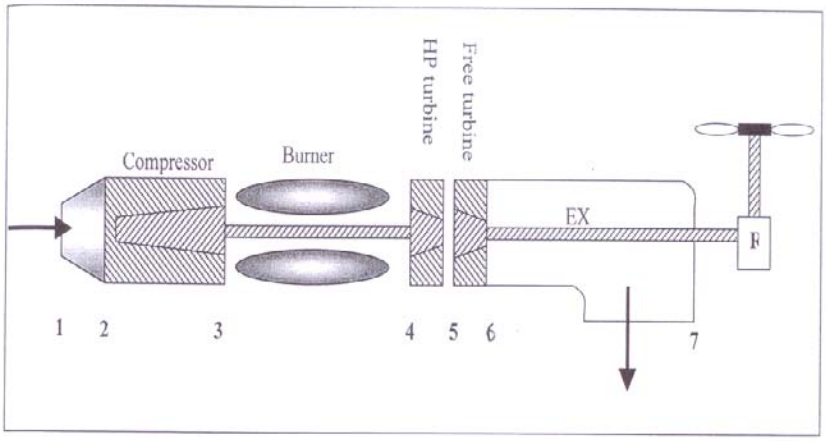


Fig. (3) Station numbering of turbo shaft engine.
1-2 inlet , 2-3 compressor , 3-4 combustion chamber , 4-5 low pressure turbine , 5-6 power turbine , 6-7 exhaust

The thermal energy added in the main combustor is given by:

$$\dot{Q}_{in} = \dot{m}_o CP(T_{t4} - T_{t3}) \qquad (33)$$

Where

$$\tau_\lambda = \frac{Cp_t T_{t4}}{Cp_c T_o} \qquad (34)$$

$$\tau_t = \frac{T_{t6}}{T_{t4}} \qquad (35)$$

The net output shaft power is the output of the turbine less the input power of the compressor and is given by:

$$\dot{W}_{out} = \dot{m}_o Cp[T_{t4} - T_{t6} - (T_{t3} - T_{t2})] \qquad (36)$$

$$\dot{W}_{out} = \dot{m}_o Cp[\tau_\lambda (1 - \tau_t) - \tau_r (\tau_c - 1)] \qquad (37)$$

Where

$$\tau_r = \frac{T_{to}}{T_o} \quad , \quad \tau_c = \frac{T_{t3}}{T_{t2}}$$

From equation (33) and (36) we can write the thermal efficiency of the gas turbine engine as follows:

$$\eta_T = \frac{\dot{W}_{out}}{\dot{Q}_{in}} = \frac{m_o CpT_o[\tau_\lambda(1-\tau_t) - \tau_r(\tau_c - 1)}{m_o CpT_o \tau_\lambda (1-\tau_t)} \tag{38}$$

$$\eta_T = 1 - \frac{\tau_r(\tau_c - 1)}{\tau_\lambda(1-\tau_t)} \tag{39}$$

Also for the isentropic expansion processes (0) to (t_3) we note that

$$\left(\frac{P_{t3}}{P_6}\right)^{\frac{\gamma-1}{\gamma}} = \tau_r\, \tau_c \tag{40}$$

Also for the isentropic expansion processes (t_4) to (t_5) and (t_6) to (7), we can write:

$$\left(\frac{P_{t4}}{P_6}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_{t4}}{P_{t6}}\right)^{\frac{\gamma-1}{\gamma}} * \left(\frac{P_{t6}}{P_6}\right)^{\frac{\gamma-1}{\gamma}} = \frac{1}{\tau_t}\frac{T_{t6}}{T_6} \tag{41}$$

For simplicity:

$$X = \frac{T_{t6}}{T_6} \tag{42}$$

Thus the ideal cycle ($P_{t4}= P_{t3}$) and ($P_6=P_0$)

Combining these ideal assumptions with the above definition and equations (40) and (41) gives:

$$\tau_t = \frac{X}{\tau_r \tau_c} \tag{43}$$

The maximum thermal efficiency of this cycle occurs when (x=1) which requires an infinite exit area .A more realistic value when (x) would be about 1.20 (M_0 of about 0.3). Not that for a given (T_{t4}) and (T_0), the compressor temperature ratio (τ_c) can be any value between unity and its maximum value is given by:

$$\tau_{c\,\max.} = \frac{\tau_\lambda \tau_t}{\tau_r}$$

$$\tau_{c\max.} = \frac{\sqrt{X\tau_{\lambda}}}{\tau_r} \tag{44}$$

For the turboshaft class of engine the shaft work is considered rather than the thrust. For the output shaft power, we define its work output coefficient:

$$C_{shaft} = \frac{\dot{W}_{shaft} / \dot{m}_o}{h_o} \tag{45}$$

Using equation (36) for the shaft power, we see that this coefficient becomes:

$$C_{shaft} = \tau_{\lambda}(1-\tau_t) - \tau_r(\tau_c - 1) \tag{50}$$

The turbine temperature ratio (τ_t) can be substituted in the above equation using equation (43)

$$C_{shaft} = \tau_{\lambda}(1 - \frac{X}{\tau_r \tau_c}) - \tau_r(\tau_c - 1) \tag{51}$$

The value of compressor ratio (π_c) is given by equation (32). It corresponds to that given the maximum of equation (34).
From the energy balance of the ideal burner [eq. (33)], have the fuel /air ratio.

$$f = \frac{CpT_o\tau_{\lambda}}{h_{pr}}(1 - \frac{X}{\tau_r \tau_c}) \tag{52}$$

The specific fuel consumption of a turbo shaft engine is expressed in terms of power specific fuel consumption (Sp). It is defined as the fuel weight flow rate per unit of shaft output, or

$$Sp = \frac{\dot{m} f}{\dot{W}_{shaft}} = \frac{f}{\dot{W}_{shaft} / \dot{m}_o}$$

$$Sp = \frac{\tau_{\lambda}}{C_{shaft} h_{pr}}(1 - \frac{X}{\tau_r \tau_c}) \tag{53}$$

Compressors are to a high degree of approximation adiabatic. The overall efficiency used to measure a compressor performance is the isentropic efficiency (η_c) defined by:

$$\eta_c = \frac{ideal\ work\ of\ compression\ for\ given\ \pi_c}{actual\ work\ of\ compression\ for\ given\ \pi_c} = 100\% \tag{54}$$

For calorically perfect gas, we can write:

$$\eta_c = \frac{W_{ci}}{W_c} = \frac{Cp(T_{t3i} - T_{t2})}{Cp(T_{t3} - T_{t2})} = \frac{\tau_{ci} - 1}{\tau_c - 1} \tag{55}$$

Here (τ_{ci}) is the ideal compressor temperature ratio, which is related to the compressor pressure ratio (π_c) by the isentropic relationship:

$$\tau_{ci} = \pi^{\frac{\gamma-1}{\gamma}}{}_{ci} = \pi_c^{\frac{\gamma-1}{\gamma}} \tag{56}$$

$$\eta_c = \frac{\pi^{\frac{(\gamma-1)}{\gamma}}{}_c - 1}{\tau_c - 1} \tag{57}$$

The isentropic efficiency of the turbine by:

$$\eta_t = \frac{1 - \tau_t}{1 - \pi_t^{\gamma-1/\gamma}} \tag{58}$$

4. RESULTS AND DISCUSSION

Fig.(1) shows the relation between the ambient temperature and mixture flow temperature when water was injected at the inlet at the compressor at different quantity of water injected. It is evident from this figure the temperatures of mixture slightly increase with the increase of the ambient temperature. Also the figure shows that the quantities of mixture flow temperature degrease when there is an increase in the percentage of water added to the air due to evaporation of the water droplets, resulting in the extraction of heat from the air. The effect of this is equivalent to a drop in the compressor inlet temperature and this behavior is reasonable.

Fig.(2) indicates the relation between the ambient temperature and the temperature of the mixture flow at different water temperature added with higher percentage of water injected (0.8%). It can be seen that there is a small effect of deferent water temperature at quantities of temperature of mixture because of the small quantity of water injected relative to air flow. The temperature of water used in this work is ($10C^0$).

Fig.(3) illustrates a relation between the ambient temperature and specific heat of mixture (C_{pm}).it can be seen that the specific heat of mixture is very little decreases when the ambient temperature increases because of the increase in the amount of the mixture temperature more than the ambient temperature. From the same figure it can be seen that the quantity of specific heat of mixture increases when the amount of the additional water increases due to the increase of the total mass flow, and specific heat of water is more than specific heat of air.

Fig.(4) illustrates the relation between the ambient temperature and the specific heat ratio is constant when the ambient temperature increases. Very little change is observed at different amounts of additional water.

Figs.(5-7) show the relation between the quantity of water injection and the temperature at operating condition (mixture temperature)and the dew point temperature of different amounts of water injection. It can be seen that the temperature of mixture decreases with the increase of the quantity of water added due to water evaporation which absorbs heat from the air mixing causing a temperature decrease compared with the entering air temperature. It is also evident that the dew point temperature increases when the injected water quantity increases due to the increase in the ratio of air to water causing increase in partial pressure.

Figs.(8-10) represent the relation between the mixture temperature and the dew point temperature across the stages of the compressor at $60C^0$ temperatures and different amounts of water injected. It can be seen that both of them increase with the increase number of stages, but the mixture temperature increase more than the dew point temperature or (the difference between the mixture temperature and the dew point temperature increase with increase of the number of stages). This may be due to the increase in the total pressure across the compressor. This shows that the condensation of water on the blades of the compressor will not happen at any stage of the compressor. The behavior of these curves is reasonable because the operating curve and dew point curve were diverged through the stages of the compressor.

Figures (11-14) Shows the performance of the (TV3-117) turboshaft engine at design condition with water injection. The figures show an improve in the performance more with increase of the amount of water injection. This is due to the lowering of the mixture temperature at the inlet temperature.

Figures(15-18) Show the percentage improvement in performance by using water injection at different ambient temperatures. The value of percentage of performance improvement was obtained from the following equation.

$$(\%)\,improvement = \frac{with\ \ water\,inj. - without\ water\,inj.}{without\ water\,inj.}$$

When fixing an ambient temperature the performance improvement percentage will increase with the increase of the amount of water injection. Similarly when an ambient temperature increase the improvement in performance percentage will be better at high quantity of water injection, i.e at $Ta=60\ C^0$. to get more improvement in performance percentage a greater amount of water injection than allowable is needed and cannot pass the allowable of an amount of water injection due to:

1. The temperature of mixture may become equal to the temperature of dew point and the water here will not evaporate (condensation). This will cause the condensation of water vapor on compressor which may lead to failure.
2. Action to the permissible to the permissible payload of the helicopter during the take off and flight duration.

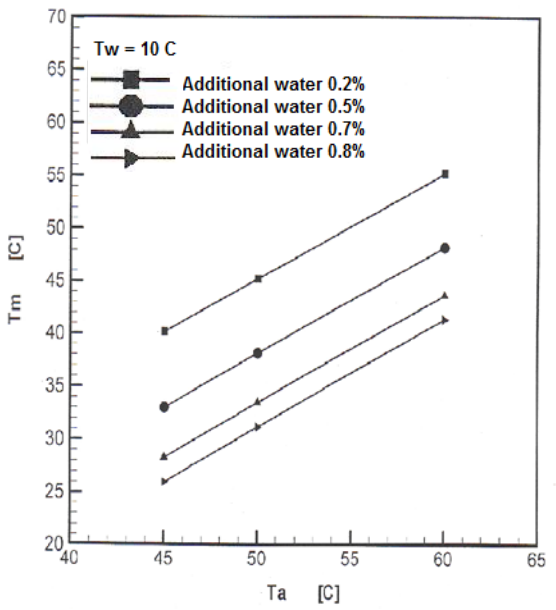


Fig.1. The relation between ambient temperature and mixture temperature.

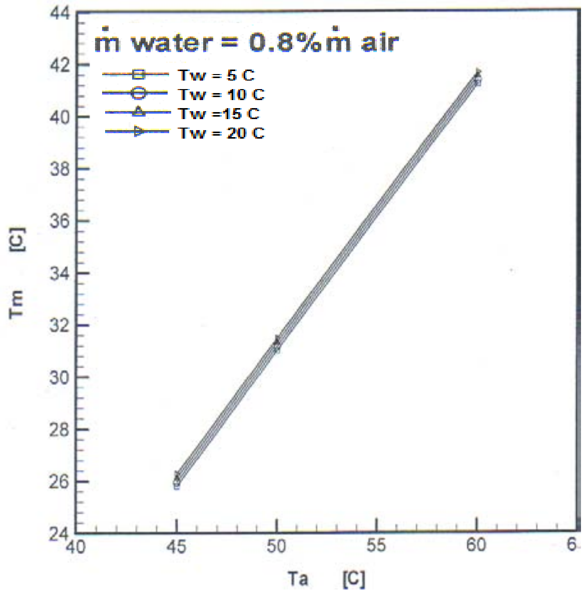


Fig.2.The relation between ambient temperature and temperature of mixture at different water temperature.

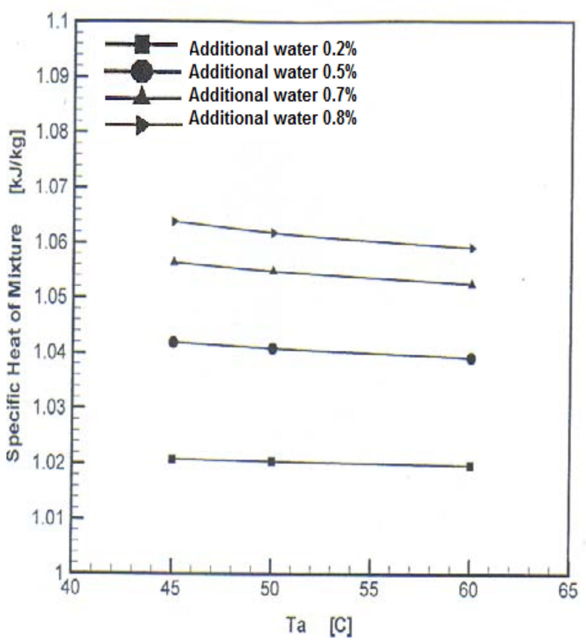


Fig.3. The relation between ambient temperature and specific heat of mixture.

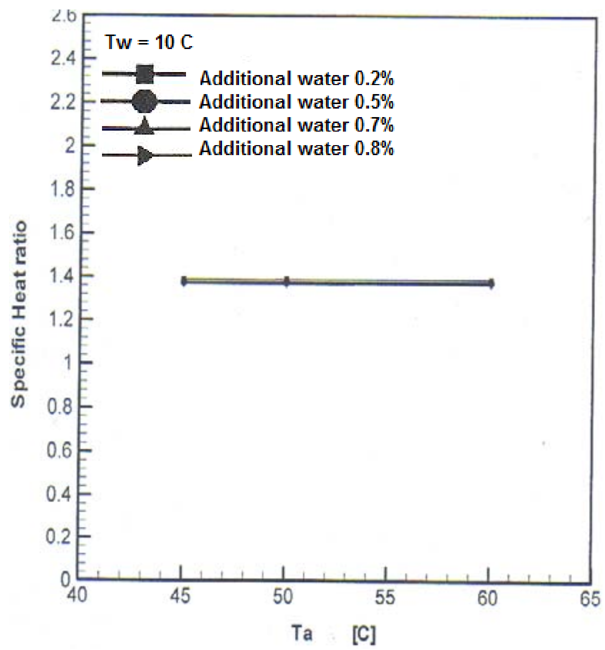


Fig.4. The relation between ambient temperature and specific heat ratio.

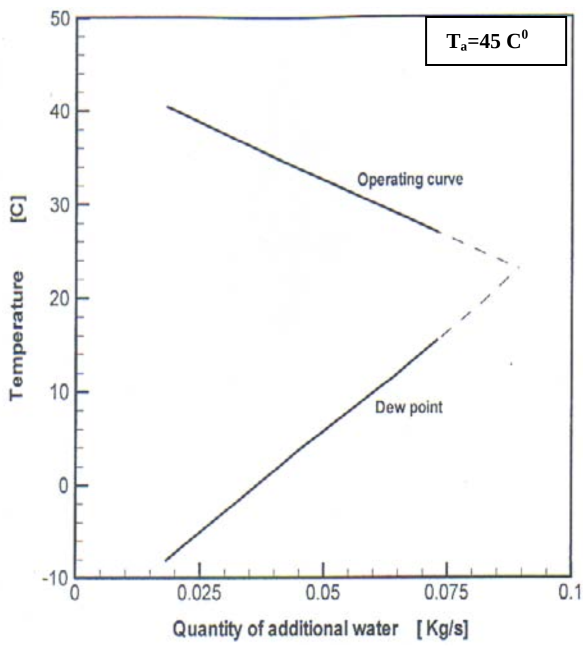


Fig.5. The relation between temperature and quantity of additional water at T_a =45C and intersection point.

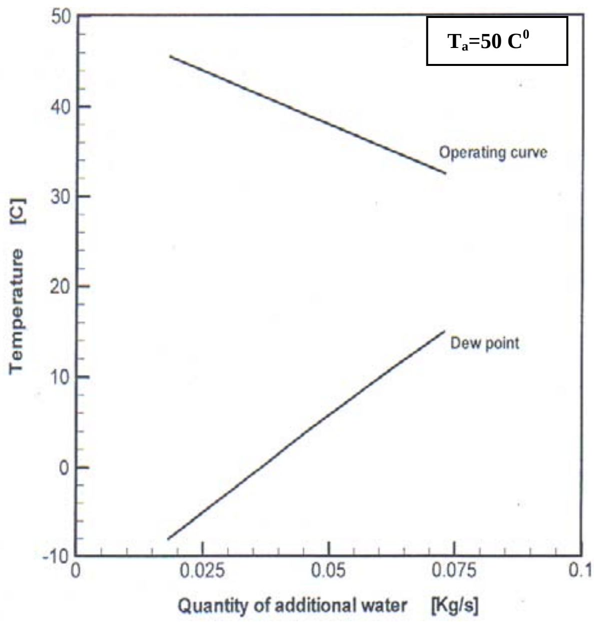


Fig..6. The relation between temperature and quantity of additional water at T_a =50C .

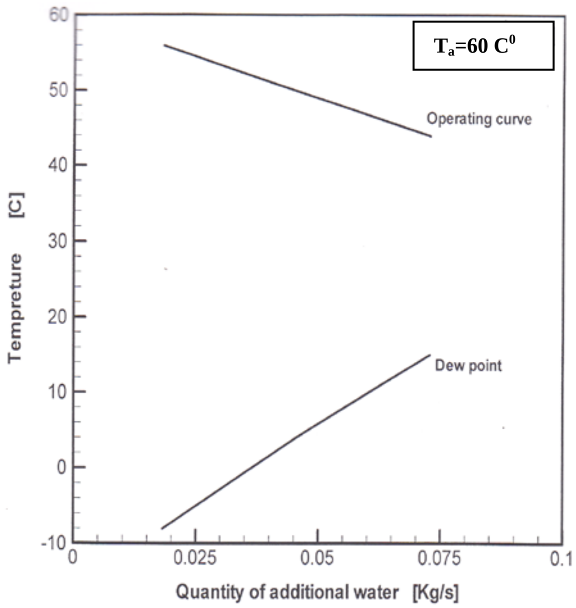


Fig.7. The relation between temperature and quantity of additional water at T_a =60C .

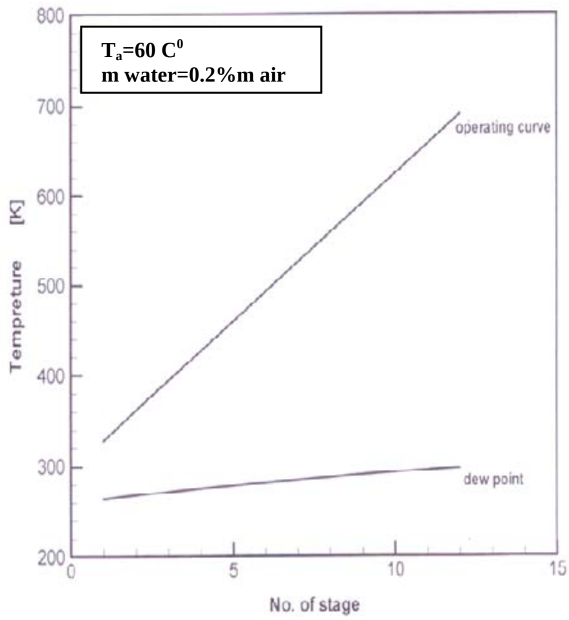


Fig.8. The relation between temperature and number of compressor stages at T_a =60C and m water=0.2% m air .

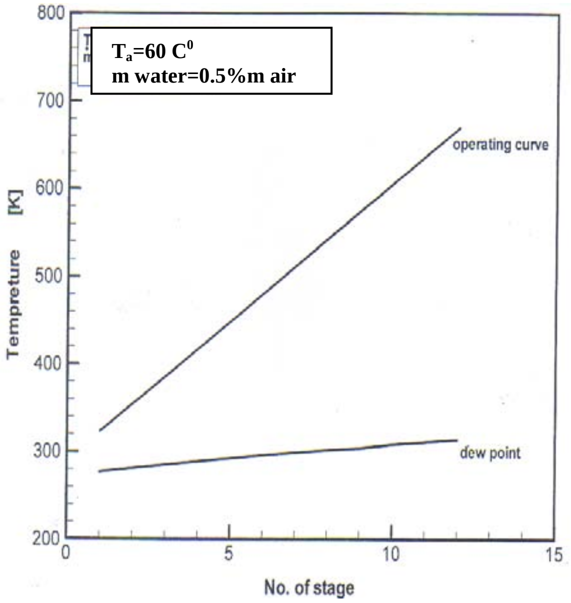


Fig.9. The relation between temperature and number of compressor stages at T_a =60C and m water=0.5% m air .

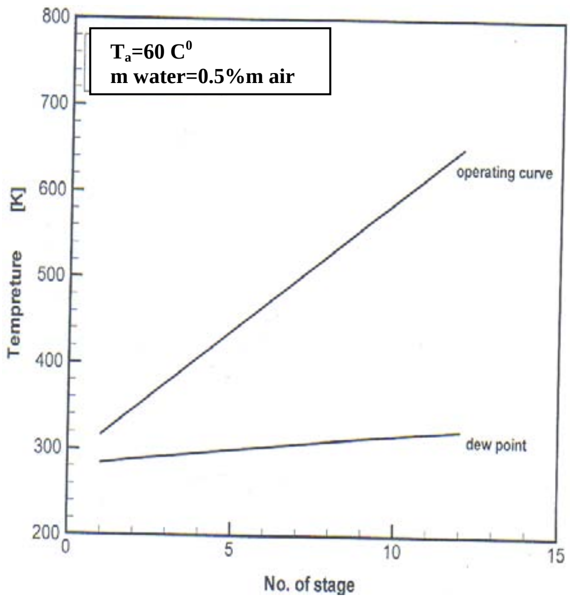


Fig.10. The relation between temperature and number of compressor stages at T_a =60C and m water=0.8% m air .

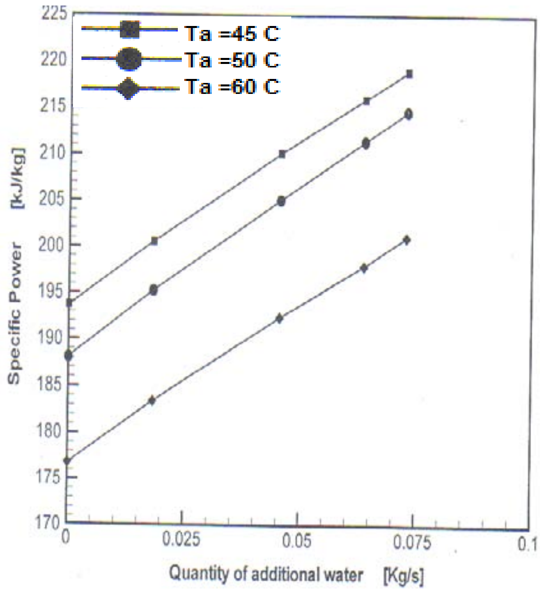


Fig.11. Turboshaft performance versus Quantity of additional water: specific power at different ambient temperature.

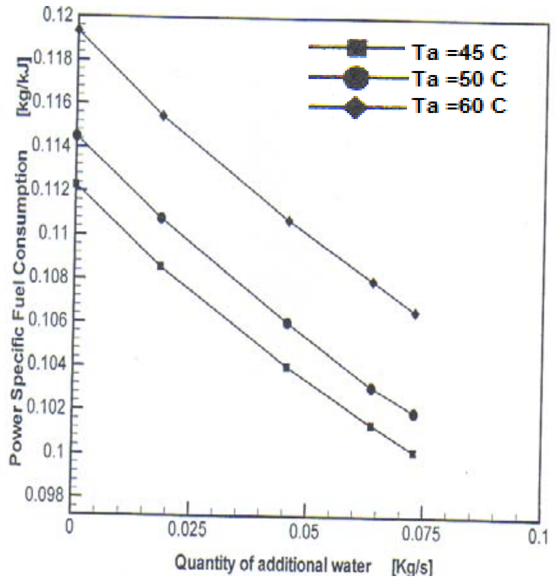


Fig.12. The relation between power specific fuel consumption and quantity of additional water at different ambient temperature .

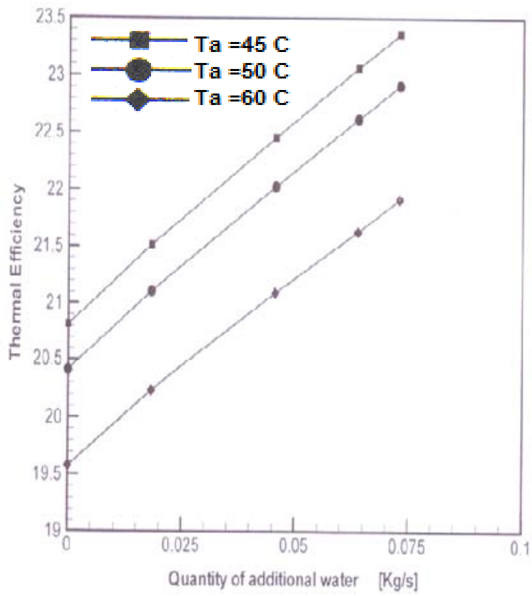


Fig.13. The relation between thermal efficiency and quantity of additional water at different ambient temperature .

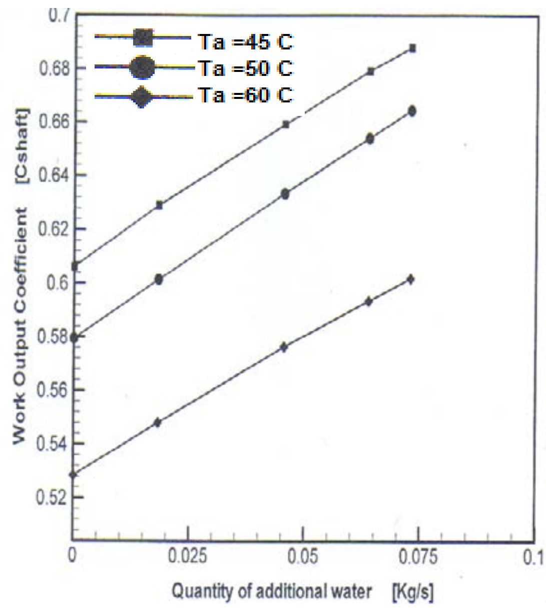


Fig.14. The relation between work output coefficient (cshaft) and quantity of additional water at different ambient temperature .

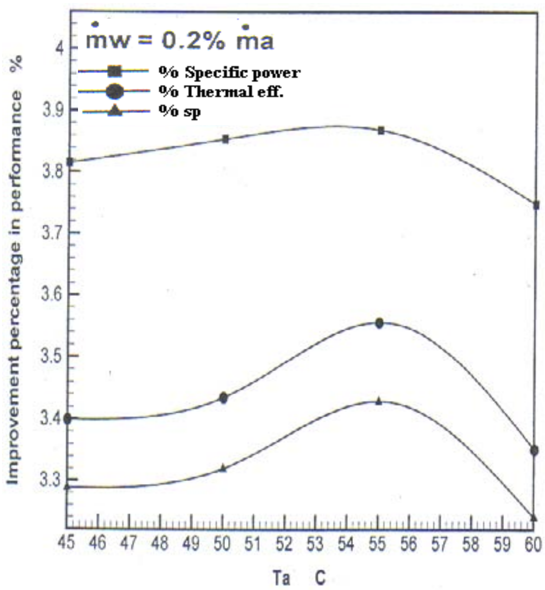


Fig.15. The improvement percentage in performance with water injection at mw=0.2%ma .

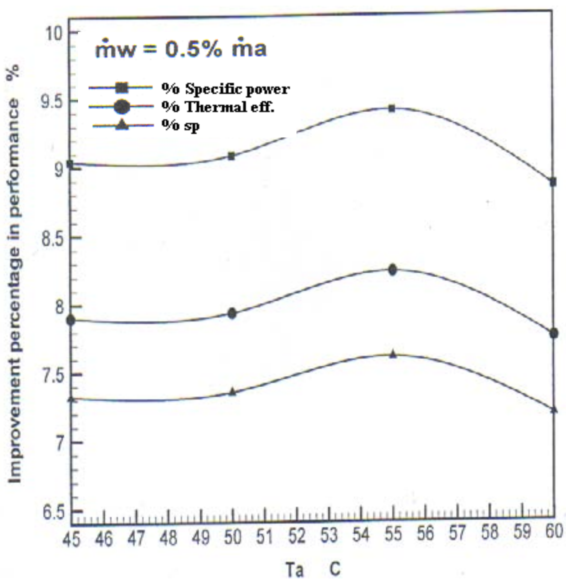
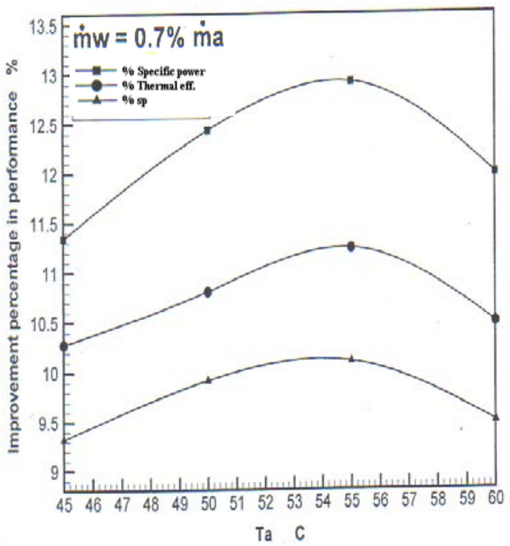


Fig.16. The improvement percentage in performance with water injection at mw=0.5% ma .



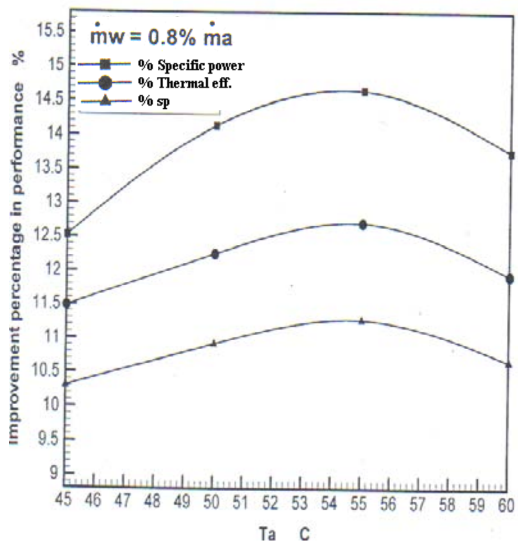


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Fig.18. The improvement percentage in performance with water injection at mw=0.8% ma .

5. CONCLSIONS:

The performance of the turboshaft engine with and without water injections studied. The calculation of new mixture property is done by water injection at the entry of the compressor. The following conclusions are:

1. The temperatures of the mixture decrease by the increase of the water injection the dew point.
2. There is a very little effect for changing water temperature in the mixture flow.
3. The specific heat of mixture flow increase by the increase of the amount of the water injected.
4. The dew point temperature is directly proportional with the quantity of the amount of water injection. The range of dew point temperature is far away from the operating temperature to give agreement results.
5. The behavior of dew point temperature and operating curve per stage of compressor is reasonable everywhere.
6. The percentage performance improvement increase at high temperature with the increase of the amount of water injection, Where at Ta=55C^{o} the specific power, thermal efficiency and specific fuel consumption improved at 14.7%, 12.6% and 11.3% respectively.

REFERENCE

ANSARI, M.R. "Gas turbine cycle efficiency improvement by spary water" mechanical engineering department, faculty of engineering, tarbiat modarres university, Tehran, iran,(2003).

Ashrae ,Hand Book. Fundamentals. , New York(1997).

Clinton, E. Wilcox and ARTHUR, M. Trout, "Analysis of Thrust Augmentation of Turbojet Engine" NACA Report-1006, 1950.

Fortin, J.A.C Compressor in a Gas – Turbine Operation" NACA Technical. And BARDON, M.F. "Gas Turbine Compressor Interstage Cooling Using Methanol" ASEM journal of Power, VOL.105, Oct. (1983) PP. 859-864.

Khaled Ahmed Al Judi, Principle Engineering of Air Condition ,Eng. College, Bassra Uneversity,1986.

REECE, V.HENSLY, "Theoretical Per-Formance of an Axial –Flow Note-2673, 1952.

Turboshaft engine (power all Russian medium helicopters)KLIMOV CORPORATION 11, Kamntemirovskaya str.,saint- petersburg , Russia, 194100.

(TV3-117MT) Series (3) MAINTENANC MANUAL,December ,25,1980.

Van J. Liere and laagand. G.H.M. "Retrofit of gas turbine by swirl flash " Kema Amhem, the Netherlands,1998.

Yunus, A. Cengel and Michael,A.Boles,"Thermodynamic:an Engineering APPROCH" Seconed edition Mcgraw- hill,inc.NEW YORK(1997).

NOMENCLATURE

Symbols	*Definition*	*Units*
A	Area	m^2
a	Speed of Sound	m/sec
C_c	Work output coefficient	-
C_{pa}	Specific heat of air at constant pressure	kJ/kg.K
C_{pm}	Specific heat of mixture at constant pressure	kJ/kg.K
C_v	Specific heat of at constant volume	kJ/kg.K
d	Diameter	m
e	Polytrophic efficiency	%
F	Thrust	N
f	Fuel / air ratio	–
g	Acceleration of gravity	m/ sec^2
H	Enthalpy	kJ
h	Enthalpy per unit mass	kJ/ kg
h_a	Enthalpy of air	kJ/kg

h_g	Enthalpy of evaporation	kJ/kg
h_m	Enthalpy of mixture	kJ/kg
h_{pR}	Low heating value of fuel	kJ/ kg
M	Mach number	-
m	Mass	kg
$\dot{m}$	Mass flow rate	kg/sec
$\dot{m}_a$	Mass flow rate of air	kg/sec
$\dot{m}_m$	Mass flow rate of mixture	kg/sec
$\dot{m}_w$	Mass flow rate of water	kg/sec
MFP	Mass flow parameter	-
N	Rotational speed	r.p.m
n	Number of stages	-
P	Pressure	N/m^2
P_t	Total pressure	N/m^2
Q	Heat	kJ
$\dot{Q}$	Rate of heat	kJ/sec
R	General gas constant	kJ/kg.K
Sp	Power Specific Fuel Consumption	kg/kJ
T	Static temperature	K
T_t	Total temperature	K
t	Temperature	C^0
U	Blade velocity	m/sec
u	Axial velocity	m/sec
ν	Absolute velocity	m/sec
V	Volume	m^3
υ	Flow velocity	m/sec
W	Moisture content	kg_w/kg_a
$\dot{W}/\dot{m}$	Specific power	kJ/kg

GREEK SYMBOLS

Symbols	***Definition***	***Units***
α	Rotor flow angle	Deg.
β	Cascade flow angle	Deg.
γ	Ratio of specific heat (C_p/C_v)	-

η	Efficiency	%
ρ	Density	kg/ m^3
T	Temperature ratio	
τ_λ	Enthalpy ratio	-
ω	Angular speed	r.p.m