Thermodynamic and Aerodynamic Meanline Analysis of Wet Compression in a Centrifugal Compressor

Jeong-Seek Kang*, Bong-Jun Cha, Soo-Seok Yang
Korea Aerospace Research Institute, Aeropropulsion Department,
45 Eoeun-Dong, Yuseong-Gu, Daejeon 305-333, Korea

Wet compression means the injection of water droplets into the compressor of gas turbines. This method decreases the compression work and increases the turbine output by decreasing the compressor exit temperature through the evaporation of water droplets inside the compressor. Researches on wet compression, up to now, have been focused on the thermodynamic analysis of wet compression where the decrease in exit flow temperature and compression work is demonstrated. This paper provides thermodynamic and aerodynamic analysis on wet compression in a centrifugal compressor for a microturbine. The meanline dry compression performance analysis of centrifugal compressor is coupled with the thermodynamic equation of wet compression to get the meanline performance of wet compression. The most influencing parameter in the analysis is the evaporative rate of water droplets. It is found that the impeller exit flow temperature and compression work decreases as the evaporative rate increases. And the exit flow angle decreases as the evaporative rate increases.

Key Words : Wet Compression, Centrifugal Compressor, Meanline Analysis

Nomenclature

C_θ : Tangential velocity component (m/s)
C_m : Meridional velocity component (m/s)
dw/dT : Evaporative rate
h : Enthalpy
L : Latent heat
$\dot{m}$: Mass flow rate (kg/s)
n : Polytropic exponent of dry air compression process
p : Pressure
R : Gas constant
T : Temperature
U : Blade speed (m/s)
w : Steam to air ratio
W : Work
X : Water/air mass flow ratio

Greeks

α : Flow angle (degree)
γ : Specific heat ratio
ρ : Density

Subscripts

1 : Compressor inlet
2 : Impeller exit
a : Air
dry : Dry air condition
o : Total condition
vap : Vapor
w : Water
wet : Wet condition

1. Introduction

The energy consumed by compressor in gas turbine is equivalent to 30～60% of energy produced by turbine and, so, the research on reducing compression work is important in increasing the output of gas turbine. And gas turbine output drops by 0.4～0.9% for every 1℃ rise in ambient tem-

* Corresponding Author,
E-mail : jskang@kari.re.kr
TEL : +82-42-860-2237; **FAX** : +82-42-860-2626
Korea Aerospace Research Institute, Aeropropulsion Department, 45 Eoeun-Dong, Yuseong-Gu, Daejeon 305-333, Korea. (Manuscript **Received** August 25, 2005; **Revised** June 16, 2006)

perature (Chaker et al., 2004a ; 2004b ; Bhargava et al., 2005a ; 2005b) which could cause significant problems to related facilities and power plants especially when electric demands are high during hot days. One method to solve this problem is to use inlet fogging system which injects small water droplets into the inlet duct of compressor. Then the water droplets evaporate in the inlet duct and cool the entering air down which boosts power output of gas turbine. About 700 gas turbines in the world are equipped with this inlet fogging system (Chaker et al., 2004a).

Another method to reduce the compression work and to boost gas turbine output is to inject small water droplets into the compressor. This method which is called wet compression or fog intercooling (Zheng et al., 2003 ; Chaker et al., 2004a ; Deneve et al., 2005) decreases compression work and increases the efficiency of gas turbines by decreasing the compressor exit temperature through the evaporation of water droplets inside the compressor. The air temperature in the compressor is higher than inlet air and the amount of vapor which can be contained in the air increases as the temperature increases. Therefore water droplets can be injected much more in wet compression than in inlet fogging.

Researches on wet compression, up to now, have established the thermodynamic and experimental foundation of wet compression where the decrease in exit temperature and compression work is demonstrated (Zheng et al., 2003 ; Deneve et al., 2005) and the increase in turbine output is demonstrated (Utamura et al., 1999 ; Bhargava et al., 2000 ; Meher-Homji et al., 2000b ; Chaker et al., 2004a ; 2004b ; Deneve et al., 2005 ; Meher-Homji et al., 2005). However, not only thermodynamic effects but also the aerodynamic effects of wet compression are important. Because the thermodynamic analysis of wet compression deals only with thermodynamic process and results of wet compression, it does not consider the geometrical parameters of compressor. It does not consider the diameter of impeller, blade angle, rotational speed, etc. So it does not consider flow angle change. When injected into the compressor, the water droplets evaporate inside the compressor and temperature, pressure, volume flow rate and flow angle become different from those of dry compression. Therefore, in multi-stage axial compressors where the stage matching is important and in centrifugal compressor with vaned diffuser where the impeller-vaned diffuser matching is important, not only the thermodynamic analysis but also the aerodynamic analysis on the effect of wet compression to the compressor performance is necessary. It is apparent that the matching of centrifugal compressor in wet compression is easier than axial compressor, because only the matching of impeller and diffuser is the main concern. The motive of this study began here : to provide the thermodynamic and aerodynamic analysis on wet compression in a centrifugal compressor.

This paper provides thermodynamic and aerodynamic analysis on wet compression in a centrifugal compressor for a microturbine. The meanline performance analysis of centrifugal compressor is coupled with the thermodynamic equation of wet compression to get the meanline performance of wet compression.

2. Analysis Methodology

2.1 Centrifugal compressor

Centrifugal compressor used in this study is the core compressor of a microturbine developed by Korea Aerospace Research Institute (KARI), and its specifications are shown in Table 1. The microturbine is composed of an 1-stage centrifugal compressor and a radial turbine with a power output of 67 kW.

2.2 Analysis methodology

As mentioned above, this paper coupled the

Table 1 Specifications of centrifugal compressor

Diameter of impeller (mm)	153.2
RPM	60740
Pressure ratio	4.2
Isentropic efficiency (%)	81
Number of impeller/splitter blades	13/13
Type of diffuser	channel diffuser
Number of diffuser vanes	24

thermodynamic process of wet compression and the aerodynamic meanline analysis of centrifugal compressor to obtain the aerodynamic performance of wet compression. The flowchart of analysis methodology is shown in Fig. 1. The first step is meanline performance analysis of dry compression. After that, the thermodynamic equation of wet compression is solved to calculate the thermodynamic effects of wet compression and, then, this output is used to calculate the modified aerodynamic and thermodynamic conditions at impeller exit. This result of modified flow conditions at impeller exit can be used to calculate modified performance of diffuser and volute, which is not included in this study. Because this method uses the results of dry compression analysis, it does not depend on the meanline analysis method of dry compression which is a merit of this analysis methodology. The meanline performance analysis of this study was performed by in-house code based on the method of Japikse (1996) which is a well known and proven method.

From the result of dry air compressor analysis, the polytropic exponent of dry air compression process (n) was defined as following.

$$\frac{p_{o2}}{p_{o1}}=\left(\frac{T_{o2}}{T_{o1}}\right)^{\frac{n}{n-1}} \tag{1}$$

Basic thermodynamic equations of wet compression was adopted from Zheng et al. (2003).

$$\frac{dp}{p}=\left(\frac{\gamma}{\gamma-1}+\frac{L}{R}\frac{dw}{dT}-\frac{1}{\gamma-1}\frac{n-\gamma}{n-1}\right)\frac{dT}{T} \tag{2}$$

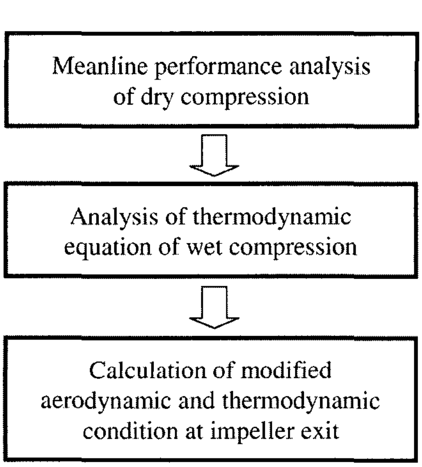

Fig. 1 Flowchart of analysis methodology

Where p is pressure, T is temperature, w is steam to air ratio, L is latent heat, γ is specific heat ratio, n is polytropic exponent of dry air compression process, and dw/dT is evaporative rate. dw/dT is the most influencing parameter in Eq. (2). w is determined from the amount of water droplets, droplet size distribution function, temperature distribution inside the impeller. It increases in the impeller and the dominant parameter from the psychrometric chart is temperature. So, Zheng et al. (2003) assumed it to be constant and solved the Eq. (2).

Assume dw/dT, L, γ, n in Eq. (2) be constant, then

$$\frac{\gamma}{\gamma-1}+\frac{L}{R}\frac{dw}{dT}-\frac{1}{\gamma-1}\frac{n-\gamma}{n-1}=\sigma \tag{3}$$

where σ is constant. Then, from Eq. (1)

$$\frac{p_{o2,wet}}{p_{o1}}=\left(\frac{T_{o2,wet}}{T_{o1}}\right) \tag{4}$$

For particle sizes that are relatively small (less than 15~20 microns) CFD studies have shown that the flow will tend to follow the air stream (Chaker et al., 2004a). And recent studies report that the size of the water droplet can be reduced to about 3 microns (Jeffs, 2004). This means that the water droplets evaporate continuously inside the impeller along the air stream and that the pressure loss due to water addition is negligible. Because the pressure ratio of wet compression is almost same as that of dry compression (Zheng et al, 2003), the following is assumed.

$$p_{o2,wet}=p_{o2,dry} \tag{5}$$

Then $T_{o2,wet}$ was calculated from Eq. (4). The equation of the conservation of energy can be described as following.

$$W_{wet}=(\dot{m}_a h_{2a}+\dot{m}_{vap}h_{vap}+\dot{m}_{2w}h_{2w})-(\dot{m}_a h_{1a}+\dot{m}_{1w}h_{1w}) \tag{6}$$

$$\dot{m}_{1w}=\dot{m}_{2w}+\dot{m}_{vap} \tag{7}$$

Where, subscript 1 is compressor inlet, 2 is compressor exit, a is air, w is water and vap is vapor. The enthalpy and other parameters of water and vapor are called from PROPATH subroutine to get accurate data. The compression work (W_{wet} (kW)) is obtained in Eqs. (6) and (7). Then,

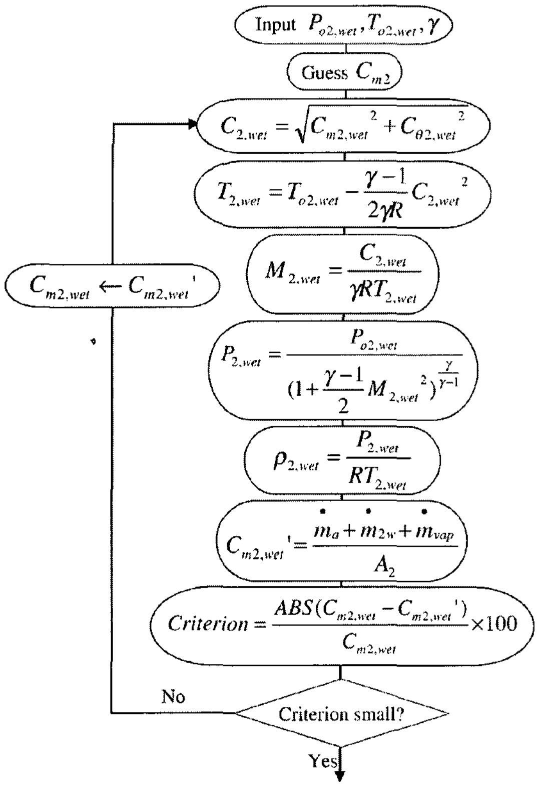

Fig. 2 Flowchart for the calculation of modified impeller exit flow condition

from the Euler-Turbomachinery equation

$$\Delta h_{wet} = \frac{W_{wet}}{\dot{m}_a + \dot{m}_{1w}} = U_2 C_{\theta 2,wet} \tag{8}$$

where the tangential velocity at impeller exit ($C_{\theta 2,wet}$) is obtained. Then the impeller exit flow parameters including velocity, pressure, static temperature, and other parameters are calculated as the flowchart shown in Fig. 2.

3. Analysis Result

3.1 Analysis at design condition

Wet compression process was analyzed at the design condition of a centrifugal compressor shown in Table 1. It was assumed that the size of the water droplets is so small that all the water droplets injected into the compressor might evaporate in the impeller. This is a realistic assumption based on experimental data (Jeffs, 2004). Then the mass flow rate of water is proportional to the evaporative rate. The water flow rate commonly used in the wet compression is about 0.5~2% of air mass flow rate (Bhargava et al., 2000). In this study, water/air mass flow ratio of 0~2.5% are assumed to be injected into the compressor.

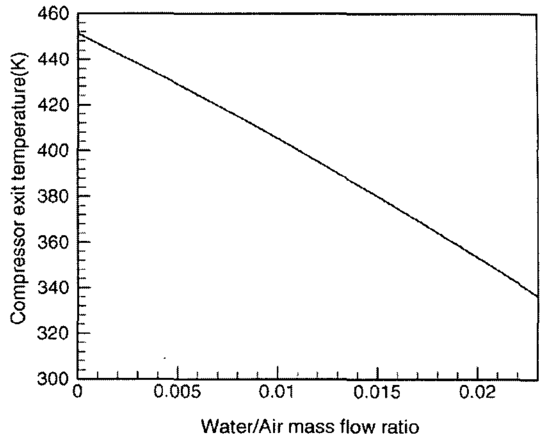

Fig. 3 Compressor exit temperature at design condition in wet compression

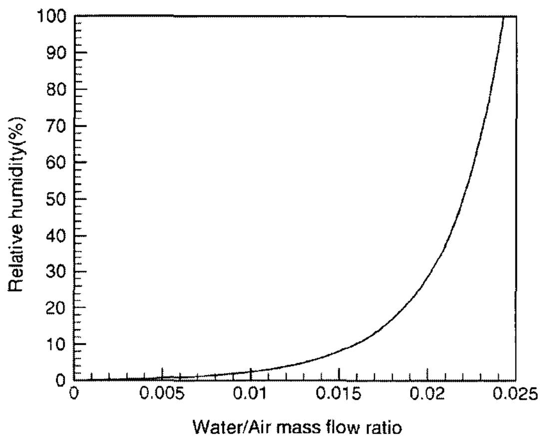

Fig. 4 Relative humidity of exit flow at design condition in wet compression

Figure 3 shows the temperature at compressor exit with respect to water/air mass flow ratio. Exit temperature in dry compression was 451 K. As water flow rate increases, the water droplets evaporates more and more in the impeller and the exit flow temperature decreases. Figure 4 shows relative humidity at the impeller exit. The relative humidity of atmosphere is assumed to be 0. As the water/air mass flow ratio increases, relative humidity increases and it reaches 100% at the water/

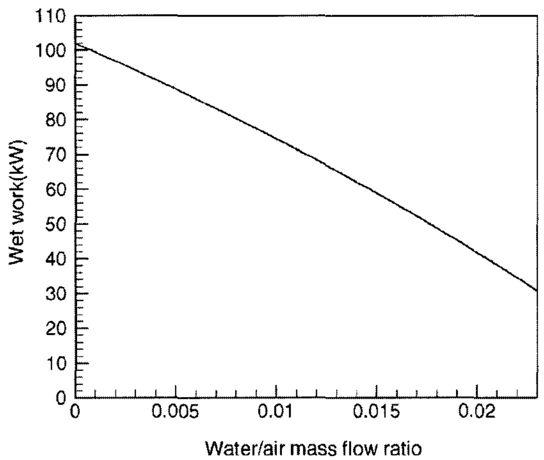

Fig. 5 Compressor power consumption (kW) at design condition in wet compression

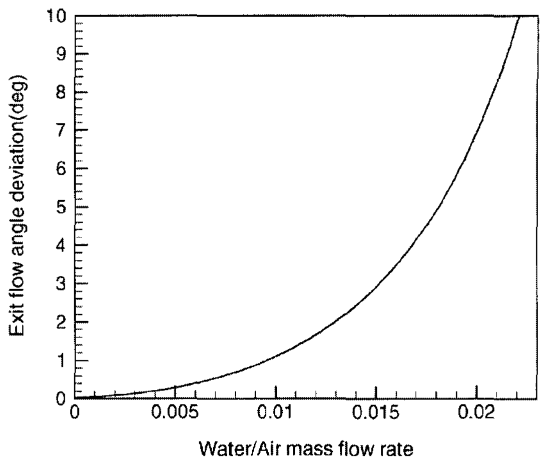

Fig. 6 Deviation of impeller exit flow angle (degree at design condition through wet compression process)

air mass flow ratio of 2.43%. Figure 5 shows the compression work with respect to the water/air mass flow ratio. As the water/air mass flow ratio increases the compression work decreases. The trend of compression work was found to be similar to that of the exit flow temperature.

In addition to the thermodynamic interests in wet compression, the most interesting aerodynamic phenomenon in wet compression is the flow angle change and the matching of impeller and vaned diffuser. If the impeller exit flow angle changed a lot, then the efficiency of compressor might deteriorate and flow separation could occur, resulting in the rotating stall and the change of surge limit line which could spoil the operation of a gas turbine. Therefore, it is necessary to obtain accurate information on the exit flow angle change due to wet compression. The change (deviation) of exit flow angle ($\Delta\alpha_2$) is defined in Eq. (9) and shown in Fig. 6.

$$\Delta\alpha_2 = \alpha_{2,dry} - \alpha_{2,wet} \tag{9}$$

where flow angle defined from radial direction. The change (deviation) of exit flow angle in Eq. (9) means the difference of flow angle between dry compression and wet compression. As the water/air mass flow ratio increases, the flow angle decreases and, hence, the change of exit flow angle ($\Delta\alpha_2$) increases. This is the result of decreased tangential velocity component at impeller exit shown

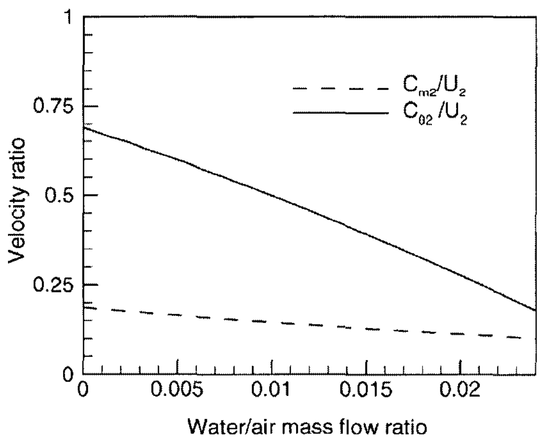

Fig. 7 Velocity ratio at impeller exit with water/air mass flow ratio

in Fig. 7. The decrease rate of tangential velocity is larger than that of meridional velocity. So the flow angle decreases as the water/air mass flow ratio increases. When the water/air mass flow ratio is 1%, the exit flow angle decreased by 1.1 degree and when the water/air mass flow ratio is 2%, it decreased by 7 degrees.

To make the data shown in Fig. 8 more useful, the following simple curve fitting was adopted.

$$\Delta\alpha_2 = 43473000X^4 \tag{10}$$

where X is the water/air mass flow ratio. Eq. (10) is the result of a specific centrifugal compressor

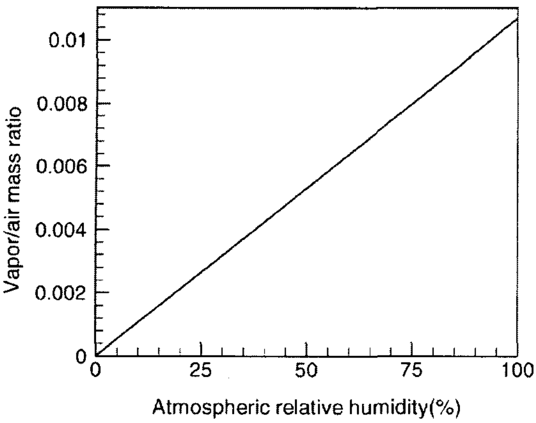

Fig. 8 Vapor and air mass ratio versus atmospheric relative humidity (atmospheric condition : 15℃, 101.3 kPa)

shown in Table 1. However, this result presents insights to the deviation angle of centrifugal compressors whose pressure ratio are around 4.2. Many compressors for microturbine belong to this case. For higher pressure ratio compressors the results are shown in the followings.

The analysis is the results under the assumption that the atmospheric relative humidity is 0% as mentioned above. The vapor mass contained in the air varies with relative humidity, temperature and pressure. The vapor mass contained in the air is shown in figure 8 where atmospheric condition is 15℃, 1 atm. The relative humidity of atmosphere limits the maximum water flow rate which can be evaporated in the impeller. The maximum water flow rate which can evaporate in the impeller is the maximum water/air mass flow ratio (0.0243) shown in Fig. 4 minus vapor/air mass ratio shown in Fig. 8.

3.2 Analysis at higher pressure ratio

Wet compression process was analyzed with evaporate rate and rpm on the condition that the inlet flow coefficient of compressor is same with the design condition of a centrifugal compressor shown in Table 1. As the rpm increases, the pressure ratio of the compressor increases as shown in the ordinate of Figs. 9~11.

The exit flow temperature with respect to the water/air mass flow ratio and pressure ratio is

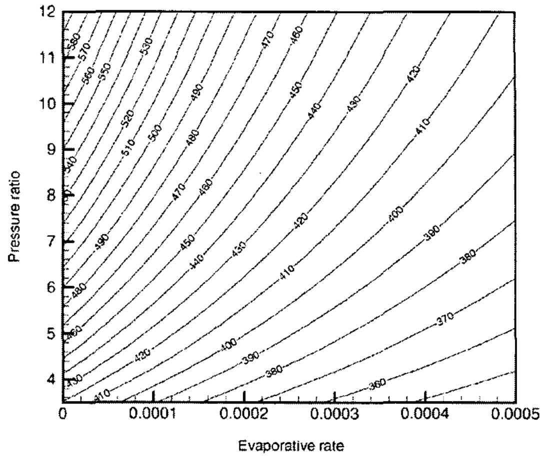

Fig. 9 Compressor exit temperature (K) with evaporative rate and pressure ratio in wet compression

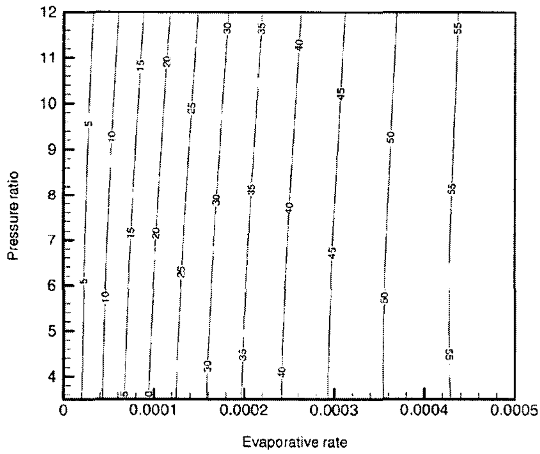

Fig. 10 Decreased compressor power consumption (%) with evaporative rate and pressure ratio in wet compression

shown in Fig. 9. As the water/air mass flow rate increases, the temperature decreases and as the pressure ratio increases, the amount of decrease in temperature with respect to the water/air mass flow ratio increases. The decreased compression work (%) with respect to the water/air mass flow ratio and pressure ratio is shown in Fig. 10. The pressure ratio has little effect on the percentage decreased work and as the water/air mass flow ratio increases the compression work decreases. The change of impeller exit flow angle is shown in

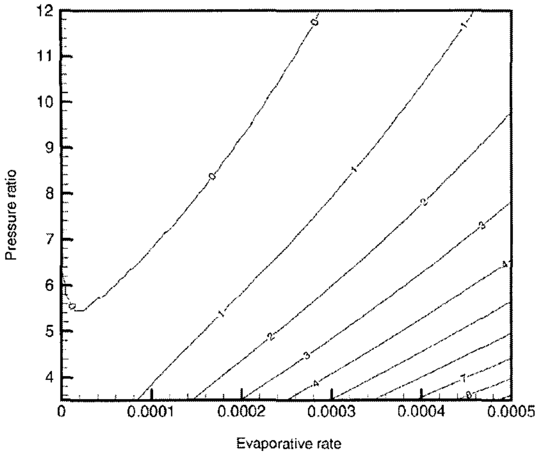

Fig. 11 Decreased impeller exit flow angle (degree) with evaporative rate and pressure ratio in wet compression

Fig. 11. As the water/air mass flow ratio increases, the flow angle decreases and, so, the change of exit flow angle ($\Delta\alpha_2$) increases. As the pressure ratio increases, the change of exit flow angle ($\Delta\alpha_2$) decreases and finally becomes 0.

3.3 Comment on the evaporative rate

To apply the result of this study in a real compressor, one thing should be considered — the evaporative rate. The analysis employed in this study assumed that the diameter of water droplets is small enough for all the water droplets injected into the inlet of compressor to evaporate inside the impeller. In a real centrifugal compressor for a microturbine, however, the mean passage length in the impeller is about 200 mm, and the passing time is on the order of 0.001 second which means that the passing time is so short and water droplets of very small diameter is needed to get the full profits of wet compression in centrifugal compressor. There are many reports on the researches to decrease the droplets size (Meher-Homji et al., 2000a ; Bhargava et al., 2002 ; 2005a ; Chaker et al, 2004a ; 2004b ; Chaker, 2005 ; Savic et al., 2005). And recently new system injects small water droplets of about 3 microns size and all of them evaporated in the first three stages of the GE frame 6 compressor (Jeffs, 2004). More studies are expected to decrease the size of the droplets in the near future. Therefore the assumption of perfect evaporation in the impeller is a realistic assumption based on experimental data.

4. Summary

This paper provided thermodynamic and aerodynamic analysis on wet compression in a centrifugal compressor for a microturbine. The meanline performance analysis of centrifugal compressor is coupled with the thermodynamic equation of wet compression to get the meanline performance of wet compression. One of the most interesting motives of this work is to investigate the flow angle deviation and the matching of impeller and vaned diffuser.

The most influencing parameter in the analysis is the evaporative rate of water droplets. The impeller exit flow temperature and compression work were found to decrease as the evaporative rate increases. And the exit flow angle decreases as the evaporative rate increases. When the water/air mass flow ratio is 1%, the exit flow angle decreased by 1.1 degree and when the water/air mass flow ratio is 2%, it decreased by 7 degrees.

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