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PERFORMANCE ANALYSIS OF INTEGRATED GASIFICATION COMBINED CYCLE CONSIDERING TURBINE MODIFICATION

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ABSTRACT

In IGCC plants, the operating environment of the gas turbine changes from its designed condition due to its integration with the gasifier block, especially with the air separation unit. The theoretical IGCC power and efficiency enhances as the integration degree becomes lower. However, low integration degree designs would reduce the compressor surge margin and cause overheating of turbine metal considerably. The main reason for these problems is that turbine inlet gas flow increases considerably because the heating value of the syngas is much lower than that of natural gas and also additional air is supplied by the auxiliary compressor. The problems can be mitigated by modulating gas turbine operating parameters. However, the problems can better be overcome through modifications of gas turbine components. This study analyzed the modification of the turbine to accommodate the increased turbine flow. The entire IGCC plant was modeled and a full off-design operation of the gas turbine was simulated. The performances of the IGCC plant with and without the turbine modification were compared. The limitations of the compressor surge margin and the turbine blade temperature were applied in both cases. The turbine modification enables a larger net power output in the low integration degree regime. The net plant efficiency does not depend very much on the integration degree.

INTRODUCTION

Worldwide efforts are being focused on research and development of producing power from coal which has the largest reserve among fossil fuels, while minimizing carbon dioxide emission. Among others, the integrated gasification combined cycle (IGCC) has been considered to be the most environmentally friendly method of using coal. Furthermore, the possibility of adopting pre-combustion carbon dioxide capture has accelerated the research and development effort for the IGCC plant [1,2]. An IGCC Plant is composed of two major parts: the power block and the gasifier block. Even though the power block of an IGCC plant looks the same as those in conventional combined cycle plants, there exist several key differences because it is integrated with the gasifier block and the operating environment is different from the original condition. The heating value of the syngas, which consists mainly of carbon monoxide and hydrogen, is much lower than that of natural gas which gas turbines are usually designed with. Thus, much more fuel is supplied to the gas turbine. The gas

Nomenclature

A	[m^2]	Area
ASU	[-]	Air separation unit
C	[-]	Cooling constant
c_p	[kJ/kgK]	Specific heat
GT	[-]	Gas turbine
ID	[-]	Integration degree
$IGCC$	[-]	Integrated gasification combined cycle
HHV	[kJ/kg]	Higer heating value
$\dot{m}$	[kg/s]	Mass flow rate
P	[kPa]	Pressure
PR	[-]	Pressure ratio
R	[kJ/kgK]	Gas constant
SM	[-]	Surge margin
ST	[-]	Steam turbine
T	[K]	Temperature
$TRIT$	[K]	Turbine rotor inlet(firing) temperature
$\dot{W}$	[MW]	Power

Special characters

ϕ	[-]	Cooling effectiveness
γ	[-]	Specific heat ratio
η	[-]	Efficiency
κ	[-]	Constant

Subscripts

aux	Auxiliary
b	Blade
c	Coolant
d	Design point
g	Gas
in	Inlet
∞	Asymptotic

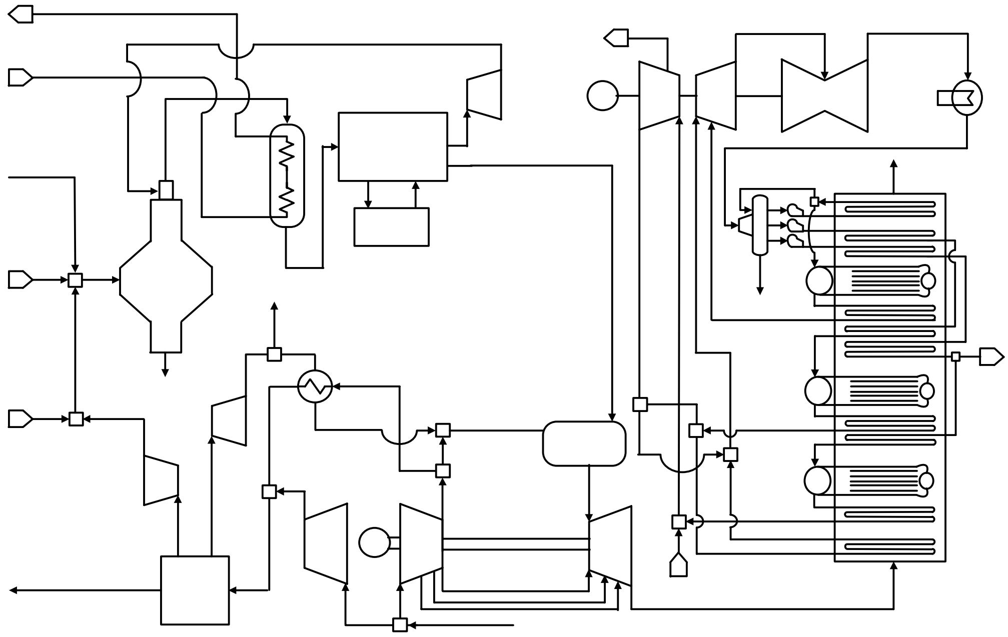

Figure 1 IGCC plant layout

turbine is integrated with the air separation unit (ASU) which produces oxidant for the gasification process. The air to ASU can be fed either by an independent air compressor or by the compressor of the gas turbine (or combination of both). Diverse integration methods are available in commercial plants [3,4]. The integration is the primary design factor of the IGCC plant, which affects much the plant performance [5]. Affected by such factors, the operating condition of the gas turbine in IGCC plants deviates from the original design condition. It has been predicted that the reduction of compressor surge margin and the rise of the turbine metal temperature can be critical issues depending on the GT-ASU integration [6]. The main reason for these problems is that turbine inlet gas flow increases considerably because the heating value of the syngas is much lower than that of natural gas and also additional air is supplied by the auxiliary compressor. As the integration degree becomes lower (air supply from the auxiliary compressor to the ASU increases), IGCC performance (power and efficiency) may increase but the two problems become more severe [6]. Modulation of gas turbine parameters such as the firing temperature and the diluting nitrogen flow may alleviate the two problems, but is accompanied by a reduction in the plant performance [6].

An active solution to avoid the surge margin reduction is to increase the swallowing capacity of the turbine by widen the turbine flow path [5,7]. However, mere modification may aggravate the overheating of the turbine. Accordingly, this study aims to investigate the influence of turbine modifications on IGCC performance considering the limitations on both the compressor surge and turbine blade temperature. The entire IGCC plant including a gasifier block, a gas turbine and a bottoming steam cycle is modelled. A full off-design analysis is given to the gas turbine using operating characteristics of the compressor and the turbine blade cooling model.

SYSTEM MODELING

An entire IGCC plant was modeled, as shown in Figure 1. The system includes the power block that consists of a gas turbine and a bottoming steam turbine cycle. Also included in the analysis are the gasification block and the ASU which interact with the gas turbine. Shell gasification process was simulated using HYSYS [8] based on a literature [9]. A sophisticated model was made to include all of the components in the literature. Since this study is focused on the performance of the power block, detailed descriptions on the gasifier block will not be given here. However, we confirmed that the predicted syngas properties are very close to those in the literature. Table 1 lists coal and syngas properties. The temperature and pressure of the syngas are 589K and 2193 kPa, and the purities of the oxygen and nitrogen from the ASU are 95% and 98.9%. The agreement of the simulated syngas properties with the reference data is very good. In order to simulate the off-design operation of the gas turbine, GateCycle [10] is used. A triple pressure bottoming steam cycle is also modelled by HYSYS. All thermodynamic properties and mass flows at every flow line between the gas turbine, modelled by GateCycle, and the remaining parts of the plant, modelled by

Table 1 Design specifications of the gas turbine

Coal (Wt. %, dry)		Syngas (Mole %)		
			Modelling	Reference[8]
N_2	7.75	N_2	4.03	4.32
H_2O	10.91	H_2O	0.3	0.3
CO	1.41	CO	64.28	62.7
CO_2	0.33	CO_2	1.06	2.06
CH_4	5.06	CH_4	Nearly 0	0.04
H_2	2.82	H_2	29.25	29.7
AR	71.72	AR	1.07	0.92
HHV (kJ/kg)	30531.1	HHV (kJ/kg)	12846	12689

Table 2 Design specifications of the gas turbine

Pressure ratio	16
Firing temperature (K)	1600
Exhaust temperature (K)	874
Exhaust gas flow (kg/s)	445
Net power output (MW)	171.6
Net thermal efficiency (%)	36.8

HYSYS, are deliberately matched to ensure the thermal and mass balances of the entire plant.

The ASU separates oxygen from the air. The oxygen is supplied to the gasifier to produce syngas. The air to the ASU can be supplied in various ways. The gas turbine compressor can supply the whole amount of the air required at the ASU. This is called 100% integration degree (ID). On the other hand, if the auxiliary compressor supplies the whole amount of air, the ID is 0%. If the two compressors share the ASU air supply, the ID ranges between 0 and 100%. Depending on the ID, the gas flow rate at the turbine varies greatly, which affects the operating condition of the gas turbine.

The design specifications of the gas turbine are shown in Table 2. They represent a state-of-the-art F-class gas turbine with the firing temperature (turbine rotor inlet temperature, TRIT) of 1600K. The change of the operating condition of the gas turbine fuelled by syngas instead of natural gas was simulated through a full off-design analysis. A compressor map was used to model the operating characteristics of the compressor. Figure 2 shows the full speed line of the compressor map. The design surge margin of the compressor was assumed to be 20%, which is a representative value in high pressure ratio axial compressors of industrial gas turbines. The surge margin is defined as follows.

$$SM = \frac{PR_{surge} - PR_{operation}}{PR_{operation}} \quad (1)$$

The off-design operation of the turbine is represented by the following chocking condition.

$$\frac{\dot{m}_{in}\sqrt{T_{in}}}{\kappa A_{in} P_{in}} = \text{constant,where } \kappa = \sqrt{\frac{\gamma}{R}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}} \quad (2)$$

Thus, the operating condition of the gas turbine is determined by a matching between the characteristics of the compressor and the turbine. Therefore, if the turbine inlet condition changes (e.g., increase of mass flow), the matching between the turbine and the compressor causes a change in the working pressure ratio of the compressor, thus affecting the surge margin. In the case of turbine modification, the turbine inlet flow area (vane throat area) of Eq. (2) is assumed to be widened (re-manufactured) to keep the turbine inlet pressure, thus the compressor pressure ratio.

A turbine cooling model was required to predict the variation of the turbine blade temperature. We focused on the change of the temperature of the stator vane of the first turbine stage. A

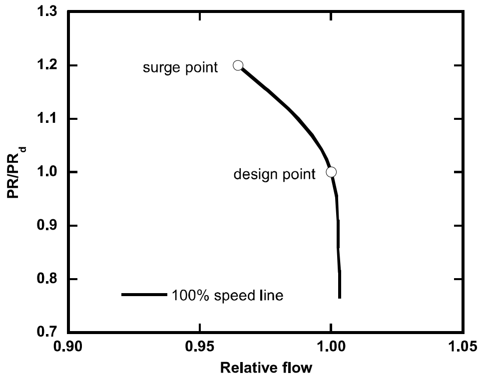

Figure 2 compressor map

simple thermodynamic model [11] was adopted. The cooling effectiveness is defined as follows.

$$\phi = \frac{T_g - T_b}{T_g - T_c} \quad (3)$$

At the design point, all of the gas, coolant and metal temperatures are given. Then, the cooling effectiveness is determined. The design temperatures of the inlet gas, the coolant and the blade metal of the first stage vane are 1670K, 688K and 1144K respectively. The major parameter governing the change in the cooling effectiveness at off-design conditions is the ratio between the thermal capacities (mass flow times specific heat) of the coolant and the hot gas. The following equation represents this relationship.

$$\frac{\dot{m}_c \cdot c_{p,c}}{\dot{m}_g \cdot c_{p,g}} = C\frac{\phi}{\phi_\infty - \phi} \quad (4)$$

Here, ϕ_∞ represents a cooling effectiveness corresponding to a very high thermal capacity ratio. 1.0 is given to ϕ_∞ in this study. C represents the technology level of the cooling scheme, and was decided to yield the thermal capacity ratio at the design point. The equation is an asymptotic curve between the thermal capacity ratio and the effectiveness [6], and was used to determine the blade metal temperature at any off-design condition. At the off-design condition, the coolant flow will also change due to the variation of operation condition. The following model [12] was adopted.

$$\dot{m}_c = \dot{m}_{c,d}\left(\frac{P_c}{P_{c,d}}\right)\left(\frac{T_{c,d}}{T_c}\right)^{0.5} \quad (5)$$

The steam bottoming cycle is a triple-pressure type as shown in Fig. 1 and Table 3 lists its major parameters. The thermal interaction between the steam and the raw gas was also considered. The net IGCC power output and efficiency are defined as follows. The auxiliary power is a sum of all of the power consumptions of additional components including the

Table 3 Major parameters of the bottoming cycle

HP/IP TIT (K)	838.5
LP TIT (K)	524
HP/IP/LP pressure (kPa)	13200 / 2420 / 500
Condener pressure (kPa)	5
HP/IP/LP pinch temperature (K)	35 / 35 / 10

auxiliary air compressor, various other compressors and pumps.

$$\dot{W}_{IGCC} = \dot{W}_{GT} + \dot{W}_{ST} - \dot{W}_{Aux} \tag{6}$$

$$\eta_{IGCC} = \frac{\dot{W}_{IGCC}}{\left(\dot{m} \cdot HHV\right)_{coal}} \tag{7}$$

ANALYSIS AND RESULTS

Two different approaches to solve the compressor surge problem and the overheating of the turbine blade were investigated. This first is to modulate a couple of operating parameters of the gas turbine such as the firing temperature and the dilution nitrogen flow without modifying any component. The second is to widen the turbine flow area to keep the design surge margin with simultaneous modulation of the firing temperature to keep the design blade temperature.

Solution by modulation of operating parameters

If none of gas turbine operating parameters were modulated, the compressor surge margin would decrease and the turbine blade temperature would rise substantially in comparison to the design condition [5,6]. The main cause is the increased turbine mass flow, which makes the compressor pressure ratio and the corresponding coolant temperature rise above the design values. According to the simulation of this study, the compressor surge margin and the first turbine stage vane temperature are maintained around design levels with 100% ID, but they deviate from the design values (the surge margin decreases and the blade temperature rises) as the ID decreases. At 0% ID, the surge margin becomes nearly zero and the blade temperature rises by more than 35K.

The previous study [6] examined the possibility of mitigating the two problems by modulating three parameters: firing temperature, diluting nitrogen flow and the turbine coolant flow. Here, modulations of the firing temperature and the nitrogen flow are considered. Figure 3 shows the operating strategy of the gas turbine, which can be summarized as follows.

- Regime A : The firing temperature is modulated to meet the design temperature of the first stage vane. The entire amount of the nitrogen separated from ASU is supplied to gas turbine combustor.
- Regime B : The surge margin is kept at the minimum allowable level (10%) by reducing the nitrogen flow. The first stage vane temperature is still maintained at the design value.

In the regime A, a reduction of the firing temperature leads to a decrease in the compressor discharge pressure (i.e. an increase of surge margin) as well as a decrease in the blade temperature. Without the firing temperature reduction, the blade temperature would rise above the design value and the surge margin reduction would be more serious. The trend would become more severe as the ID decreases because a reduction of the ID means an increase in the turbine inlet gas flow. Figure 4 shows the variations in turbine inlet flow and nitrogen flow. Below an ID (42%) where the surge margin reaches the minimum value (10%), the surge margin and the firing temperature are forced to be kept constant regardless of the integration degree. Without this restriction, the turbine inlet gas flow tends to increase. However, by reducing the nitrogen flow, the net turbine inlet flow can be kept constant to secure the target surge margin.

Figure 5 shows the IGCC power and efficiency. The results of non-modulated gas turbine operation are also shown for comparison. Without the modulations, the net IGCC power would become larger as the ID gets lower. The efficiency would have a similar trend even though its ID dependency is not very considerable. The firing temperature reduction causes power and efficiency drops in the regime A. In the active surge prevention regime (B), the gap between the non-modulated and

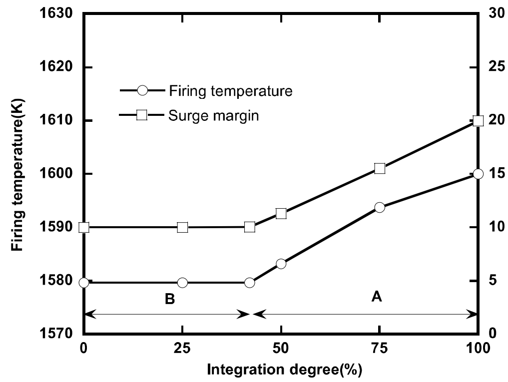

Figure 3 Operation strategy to meet the surge margin limitation and the blade temperature without component modification

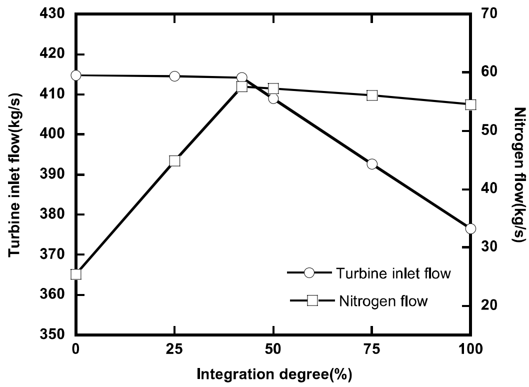

Figure 4 Variations in turbine inlet flow and nitrogen flow with integration degree (no component modification)

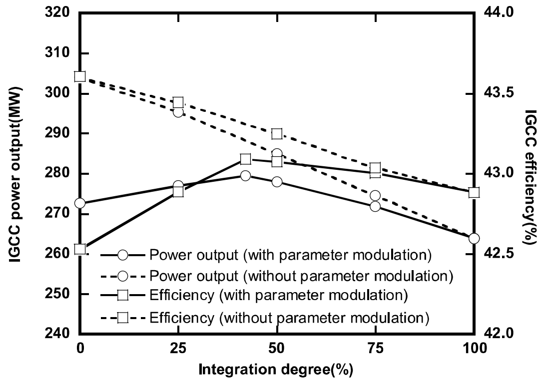

Figure 5 Variations in IGCC performance with integration degree (no component modification)

the modulated operations is considerable. Both the power output and efficiency peak at the ID where the surge margin restriction begins to be applied. Simultaneous application of both the surge margin (10%) and blade temperature restrictions causes more than 30 MW (11%) decline of the power output at 0% ID.

Solution by turbine modification

In the previous section, we observed that the two problems of the compressor surge margin reduction and the turbine metal overheating can mitigated by modulations of operating parameters. However, they are partial solutions because the compressor surge margin is still smaller than the design margin. In addition, the modulations incur serious reduction of net power output, especially in the low ID regime. A more active solution to the surge problem is to re-design or modify the turbine in order for the turbine to accommodate the increased mass flow. In this study, increase of the vane area (A in of Eq. (2)) of the first stage is considered. The target is to maintain the design value of the compressor pressure ratio. Simultaneously, the firing temperature is modulated to meet the design temperature of the first stage vane.

Figure 6 shows the compressor pressure ratio and the firing temperature. The pressure ratio is maintained constant at the design value as intended (see the dotted line for the case of no-modification for comparison). The firing temperature needs to be reduced by as much as 70K at 0% ID to maintain the design temperature of the first stage vane. The required firing temperature reduction is greater than that of the non-modified operation (compare the firing temperatures of Figures 3 and 6). It was estimated that the blade temperature would increase by 45K at 0% ID if the firing temperature were not modulated. Figure 7 shows the required increase of the vane area (percentage increase relative to the design condition). Figure 8 shows the resulting turbine inlet flow and the dilution nitrogen flow. The flows can be steadily increased up to 0% ID thanks to the increased vane area (see Figure 4 for comparison). At 0% ID, nearly 20% increase of the vane area is needed and the result is almost 11% more gas flow admission to the turbine (compare Figures 4 and 8).

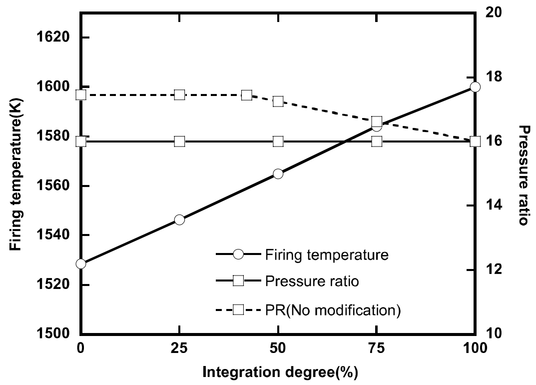

Figure 6 Firing temperature and pressure ratio setting in the case of turbine modification

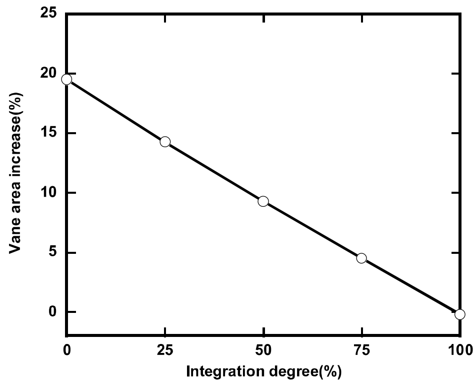

Figure 7 Relative increase in the vane area in the case of turbine modification

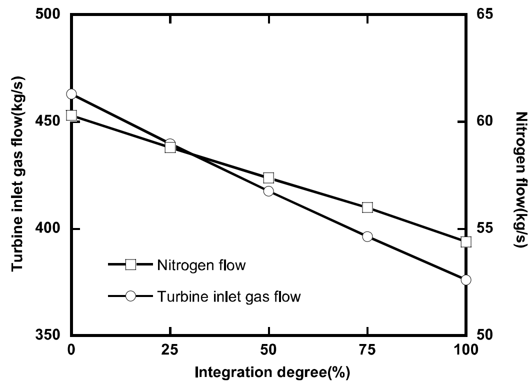

Figure 8 Variations in turbine inlet gas flow and nitrogen flow with integration degree in the case of turbine modification

Figure 9 shows the variations of gas turbine and steam turbine power outputs and the auxiliary power consumption. Results for the non-modified operation which correspond to Figure 4 are also shown for comparison. The turbine

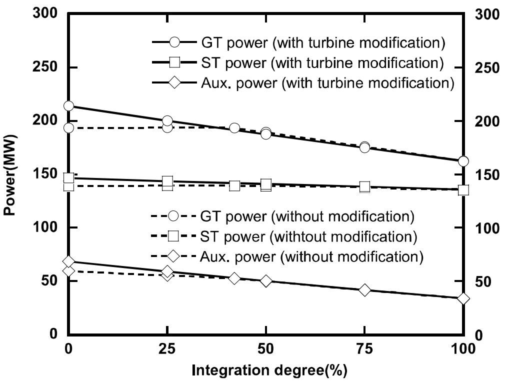

Figure 9 Variations in power outputs and auxiliary power consumptions with integration degree

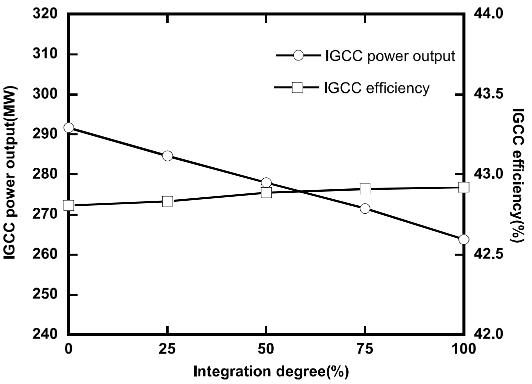

Figure 10 Variations in IGCC performance with integration degree in the case of turbine modification

modification does not yield much improvement in the gas turbine power outputs in the high ID regime (the regime A defined in the previous section) compared with the non-modified operation. This is because the negative effect of relatively lower firing temperature counteracts the positive effect of relatively larger turbine flow. However, in the low ID regime (the regime B defined in the previous section), the much greater turbine flow yields considerably larger power outputs. The increased gas flow provides larger steam turbine power capacity as well. The auxiliary power also increases as the ID decreases because of the greater air supply from the auxiliary air compressor. The resulting net IGCC power output and efficiency are shown in Figure 10. The net power output in the low ID regime is much larger compared with the non-modified operation of the previous section (see Figure 5 for comparison). At 0% ID, it reaches 292MW which is about 20MW larger than the non-modified case. The net thermal efficiency is effectively insensitive to the ID (only a very slight decrease with decreasing ID).

CONCLUSION

Performance of IGCC plants subject to operating limitations in terms of compressor surge margin and turbine blade temperature is investigated. Two different strategies to mitigate the two problems were compared: modulation of operating parameters and modifications of the turbine. Modulation of the firing temperature and the dilution nitrogen flow can solve the blade overheating problem and mitigate the surge problem. The net power output and efficiency increase as the integration degree decreases, but only up to a certain point. Then, they decrease sharply with decreasing integration degree. The turbine modification, i.e. increase of the vane area, to keep the design surge margin of the compressor was simulated. No sensible advantage is found in the high integration degree regime. However, in the low integration degree regime, the turbine modification provides much greater net system power output. Accordingly, the turbine modification is effective in designing an IGCC with a low integration degree.

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