ABSTRACT

In order to reduce the production of green house gases, the combustion of biomass has been gaining importance in electricity generation. Especially the direct combustion of biomass in gas turbines of a few MW output would offer a very attractive option because of low investment costs and high operational flexibility. Therefore, since 1991 the Institute of Thermal Turbomachines and Powerplants at the Vienna University of Technology has been working on realising a wood particle fired gas turbine with direct combustion. With reference to earlier studies (c.f. Hamrick (1991), Fredriksson and Kallner (1993)), it had been concluded that the design and the operating characteristics of the fuel feed system would strongly influence the combustion and so would be a very important part of the whole facility.

Following an overview of the planned gas turbine test facility including the combustion chamber and the recently installed pneumatic fuel feed system, the paper will deal with three basic requirements of fuel feeding in the case of a directly fired gas turbine: feeding against back pressure, continuous fuel flow rate and a low conveying air ratio, which is the ratio of fuel conveying air to total combustion air of the combustion chamber. While the first two requirements, i.e. feeding against back pressure and continuous feeding, are briefly considered, the minimisation of the conveying air ratio is discussed in detail. For instance, important parameters affecting the conveying air ratio are fuel moisture, combustion air ratio and, in particular, velocity. Following theoretical estimation of the conveying air ratio, results of fuel feeding measurements are presented and conclusions drawn with respect to system integration.

1. INTRODUCTION

Due to low public acceptance of nuclear power generation and the production of green house gases by conventional power generation with coal, oil or gas, during the last years the interest in biomass utilization has increased. Wood as a primary energy source plays an important role because it is a very widespread and locally available natural resource. However, due to its lower density and a lower heating value in comparison to fossil fuels it does not make sense economically and ecologically to transport wood over great distances to central powerplants of large power output. Therefore, the ideal use of wood as an energy source is electricity and heat generation (cogeneration) by smaller decentralized power facilities of only a few MW of thermal power. Today, power generation from wood and other biomass is frequently done by conventional or fluidized bed steam generators and steam turbines (i.e., plants with indirect firing). However, steam powerplants are expensive and therefore can only be operated economically in the high power range. On the other hand, gas turbines with direct firing would be a very attractive option because of low investment costs.

Beside this general consideration, the lumber industry, which is a very important branch of the economy in many countries, produces large amounts of wood waste, e.g., saw dust and wood powder. However, presently only a small fraction of this biomass is used to the power generation. Due to low public acceptance of nuclear power generation and the production of green house gases by conventional power generation with coal, oil or gas, during the last years the interest in biomass utilization has increased. Wood as a primary energy source plays an important role because it is a very widespread and locally available natural resource. However, due to its lower density and a lower heating value in comparison to fossil fuels it does not make sense economically and ecologically to transport wood over great distances to central powerplants of large power output. Therefore, the ideal use of wood as an energy source is electricity and heat generation (cogeneration) by smaller decentralized power facilities of only a few MW of thermal power. Today, power generation from wood and other biomass is frequently done by conventional or fluidized bed steam generators and steam turbines (i.e., plants with indirect firing). However, steam powerplants are expensive and therefore can only be operated economically in the high power range. On the other hand, gas turbines with direct firing would be a very attractive option because of low investment costs.

Presented at the International Gas Turbine & Aeroengine Congress & Exhibition
Indianapolis, Indiana — June 7—June 10, 1999
the corresponding amount of saw dust. This means that the lumber industry today is an example for the potential application of gas turbines fired directly with wood. Therefore, a research project at the Vienna University of Technology is focused on the development of a wood particle fired gas turbine. The realization of this project is divided into different phases. In a first step a combustion chamber design was developed. Afterwards the pneumatic fuel feed system was designed and manufactured. This is now tested under special consideration of stable operation of the combustion chamber over a wide range of power settings. Later, the combustion chamber will be tested including operation together with the fuel feed system. Finally the combustion chamber will be connected to a small gas turbine of the Institute. During this phase, the risk to the gas turbine of fouling and possibly corrosion will also be studied.

The actual feeding system mainly consists of mechanical conveyors metering and conditioning the fuel mass flow and of pneumatic feeding components such as the injector and the conveying pipe to the combustion chamber. Following a brief consideration of the conveying chamber design it will be shown that a reduction of the conveying air flow rate supports and improves the combustion chamber operation. Therefore, a low conveying air ratio is a main objective of research reported in this paper.

2. DIRECTLY WOOD PARTICLE FIRED GAS TURBINE

To offer a very simple and economical option of wood particle fired gas turbines, the principal idea is to use a conventional gas turbine, leave its compressor and turbine unchanged and replace the existing gas or oil fired combustion chamber by a combustion chamber designed for wood particle firing. While gas or oil are fuels with almost constant properties, wood is a fuel with strongly varying moisture content. With regard to increasing fuel moisture, there is the following benefit in the case of direct firing: While the efficiency and the power which is transmitted at the coupling of a steam turbine or an indirectly fired gas turbine drop with increasing fuel moisture, the power of a directly fired gas turbine rises due to the increasing flow rate through the turbine.

Figure 2.1 shows a schematic diagram of the wood particle fired gas turbine. For initial combustion tests, only the combustion chamber is tested. A separate compressor supplies the combustion air and a throttle is used to maintain the combustion chamber pressure. Later, assuming stable combustion and that minimization of the ash content of the combustion gas will have been reached by a cyclone dust separator, the gas turbine will be added to the system. The entire system including the fuel feed system is designed to meet a maximum thermal power of about 800kW and a maximum operation pressure of the combustion chamber of 2.8bar. These figures directly correspond to the gas turbine which will be used in the test facility.

The turbine inlet temperature of the gas turbine used in these tests is limited to 800°C according to manufacturer specifications. Since the combustion temperature is much higher, the gas temperature downstream of the combustion chamber is reduced by mixing the combustion gas with cooler compressor air. While modern gas turbines with conventional fuels run at turbine inlet temperatures of about 1300°C, directly wood particle fired turbines must be limited to 1100°C to avoid fuel slag. A consequence is that the thermal efficiency of the directly wood particle fired gas turbine is correspondingly lower. But this fact is not so significant because small engines due to less advanced cooling techniques are limited to turbine inlet temperatures of approximately 1000°C.

**Combustion Chamber Concept.** An advantage of this temperature limitation is the reduction of thermal nitrogen oxide. In order not to exceed a combustion temperature of 1100°C the combustion chamber is divided into two stages. In the first stage (the primary combustion chamber), the wood powder is burned with an understoichiometric combustion air ratio of about 0.7 (0.5≤λ≤0.9). In the second stage (the secondary combustion chamber), the combustible gas coming out of the first stage is overstoichiometrically afterburned (λ=1.6).

Figure 2.2 shows the principal outline of the combustion chamber corresponding to Sengschmied (1995). Primary and secondary combustion chamber are designed as swirl chambers where
the combustion air is blown in tangentially. The second stage follows directly the primary chamber. Both stages are separated by a partition with a collar like in some cyclone designs. Moreover, with reference to Sengschmied, who estimated the fuel particle distribution inside the primary combustion chamber with regard to stable combustion, the wood particle inlet and the auxiliary gas burner are located at the bottom. Starting the gas turbine, the gas burner is used to preheat the insulating liner up to a minimum temperature of 600°C. The incoming wood particles will then be directly ignited.

A quantity characterizing operation of the combustion chamber is the combustion intensity \( q_{cc} \) defined as

\[
q_{cc} = \frac{P_{th}}{V_{cc} \rho_{cc}} \quad \text{with} \quad P_{th} = m_{fuel} \cdot H_{fuel} \tag{1}
\]

The unit of the combustion intensity \( q_{cc} \) is \([s^{-1}]\). It is a measure of the residence time of the combustion gas inside the combustion chamber. In case of the wood particle fired swirl combustion chamber, \( q_{cc} \) should be between a minimum of 12 \( s^{-1} \) and a maximum of 20 \( s^{-1} \). In general, the volume of the combustion chamber \( V_{cc} \) is a constant. Assuming also constant operating pressure \( P_{cc} \) and constant lower heating value of the fuel \( H_{fuel} \), the needed fuel mass flow \( m_{fuel} \) is directly proportional to \( q_{cc} \). Therefore, the selection of \( q_{cc} \) has an important influence on fuel feeding.

3. REQUIREMENTS OF FUEL FEEDING

There is one main requirement with respect to fuel feeding of a directly fired gas turbine: Fuel feeding against back pressure. Two other important requirements follow from the present combustion chamber design: In contrast to other combustion chamber concepts for solid fuels, the present combustion chamber has a very small volume and so the time lag between a change in fuel feeding and gas turbine power output is short. This improves control characteristics of the system. On the other hand, fuel mass flow fluctuations have a greater effect on gas turbine operation. E.g., according to manufacturer specifications for turbines in the range of 4 to 8 MW of electric power, changes in the gas heating value greater than ±10% during operation at rated output have to be avoided. In case of turbines of a lower power class, somewhat higher deviations are acceptable and vice versa. Therefore, one goal of fuel feeding must be a steady fuel flow rate. On the other hand, as noted in chapter 2.1, the fuel is blown in at the bottom of the primary combustion chamber and consequently the fuel inlet flow does not support the swirl flow in the first stage. Therefore, with respect to fuel feeding a further very essential task must be to minimize the conveying air flow rate. The loading \( \mu \), i.e., the ratio of the fuel mass flow rate \( m_{fuel} \) and the conveying air mass flow rate \( m_{conv.\_air} \), should be as high as possible. However, there is a quantity restricting the maximum loading: According to the TRD414 (1993) the conveying air inlet velocity of a wood particle fired furnace must be higher than 18m/s in order to prevent flash back.

3.1 Basics of the Conveying Air Ratio

The conveying air ratio \( k_1 \) is introduced as the fraction of the conveying air flow rate \( m_{conv.\_air} \) to the total combustion air flow rate of the first combustion stage \( m_{prim.\_comb.\_air} \):

\[
k_1 = \frac{m_{conv.\_air}}{m_{prim.\_comb.\_air}} \tag{3}
\]

As seen before, the conveying air ratio \( k_1 \) should be as low as possible and therefore, it can be regarded as a measure of fuel feeding quality. To get a stable swirl flow, the conveying air ratio should be lower than approximately 33%. However, at the present time it is impossible to give an exact limit of the conveying air ratio because no combustion chamber tests have been done so far. In the future, owing to combustion chamber operation, it will be possible to exactly specify the allowed range of conveying air ratio. The total combustion air flow rate of the primary combustion chamber is given by the product of the combustion air ratio \( \lambda_1 \), the stoichiometric air to fuel ratio \( 1_{\text{min}} \) and the fuel flow rate \( m_{fuel} \):

\[
m_{prim.\_comb.\_air} = \lambda_1 \cdot 1_{\text{min}} \cdot m_{fuel} \tag{4}
\]

Using equation (2), the conveying air ratio can then be written as

\[
k_1 = \frac{1}{\lambda_1 \cdot 1_{\text{min}} \cdot \mu} \tag{5}
\]

3.2 Parameters Affecting the Conveying Air Ratio

According to equation (5), the conveying air ratio depends on the following parameters: the combustion air ratio \( \lambda_1 \), the loading \( \mu \) and the fuel moisture \( u \).

Fuel Moisture

![Fig. 3.1 Conveying Air Ratio at a Combustion Air Ratio of 0.7 Dependent on Loading and Fuel Moisture](http://fluidsengineering.asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1999/78590/V002T01A013/4217995/v002t01a013-99-gt-353.pdf)
In contrast to the combustion air ratio and the loading, the fuel moisture is a parameter which is usually determined by conditions outside of the system. Figure 3.1 shows the influence of the fuel moisture on the conveying air ratio. The figure applies to an average combustion air ratio of 0.7. It can be noticed that high moisture requires high loading to decrease conveying air ratio. The normal range of moisture of saw dust is up to 45% (Wagner (1979)). Moisture contents of 100% or higher only occur in case of freshly cut lumber. However, by applying a drying process the fuel moisture can be reduced. This means, if loadings higher than 2 can be realized, the conveying air ratio will be lower than 20% or even much less.

**Combustion Air Ratio.** Another parameter is the combustion air ratio as presented in figure 3.2 which applies to a constant fuel moisture of 10%. This value also corresponds to the material used for the present tests. Intending low NOx emission, the combustion air ratio is allowed to vary between 0.5 and 0.9. But this can cause significant variations of the conveying air ratio. For example, at a loading of 2, the maximum difference of the conveying air ratio for the above range of combustion air ratios amounts to almost 8 percentage points. As a conclusion, if the combustion chamber is operated at low combustion air ratio, the conveying air ratio will be much higher or otherwise the loading will have to be increased.

**Pipe Diameter and Fuel Inlet.** Although conveying techniques enable feeding at high loadings, with reference to TRD (1993), a minimum air velocity of 18m/s at the combustion chamber inlet has to be met and so it is a quantity limiting the loading. Due to this fact, the pipe diameter which determines the air inlet velocity, strongly influences the conveying air ratio. The air inlet velocity \( v_{\text{inlet}} \) can be written as a function of the state of the air (temperature \( T \), pressure \( p \)), the thermal power requirement of the combustion chamber \( P_{\text{th}} \), the loading \( \mu \) of the pneumatic conveying process and the cross sectional area \( A_{\text{pipe}} \) of the conveying pipe at the combustion chamber inlet. Then, according to equation (1), pressure and thermal power can be substituted by the combustion intensity \( q_{\text{cc}} \) so that the conveying air inlet velocity is calculated by:

\[
v_{\text{inlet}} = \frac{q_{\text{cc}} V_{\text{cc}}}{A_{\text{pipe}} H_{\text{fuel}}}
\]

Figure 3.3 shows the inlet velocity as a function of loading and pipe diameter. The combustion intensity is kept constant at 12s\(^{-1}\) at a combustion chamber volume of 0.15m\(^3\) and the fuel moisture is kept at 10%. With these data and with respect to the minimum allowed inlet velocity of 18m/s, which is also drawn into figure 3.3, the attainable loadings will be the lowest. Now, independent of feeding techniques, the maximum realizable loading follows from this limiting velocity and from the pipe diameter. E.g., a maximum loading of 1.05 results from a pipe diameter of 25mm at a conveying air temperature of 25°C. The loading rises by an increase either of the combustion intensity or of the fuel moisture. Finally, the minimum reachable conveying air ratio can now be obtained from figures 3.1 and 3.2 under consideration of the highest loading read from figure 3.3.

But there are two possibilities to raise the loading and hereby to reduce the conveying air ratio: the first one by reducing the pipe diameter or just the pipe cross section at the combustion chamber inlet, and the second one by heating the conveying air (see fig. 3.3). On the other hand, due to the risk of ignition, it should be watched that the air temperature does not exceed 90°C.
4. FUEL FEED SYSTEM

Figure 4.1 is a schematic diagram of the experimental fuel feed system. In order to carry out feeding tests without combustion, the feeding is done to a specially designed cyclone which replaces the combustion chamber. Downstream of the cyclone, a constant pressure valve is used to simulate the operating pressure of the combustion chamber. The fuel flow rate is related to the weight change of the receiving bin which is positioned on a scale working on electromagnetic force compensation. At the beginning of operation, the bin is filled into the sending bin. After filling, the bin is closed and pressurized together with screw feeder and vibrator housing. The pressure corresponds approximately to the operating pressure of the combustion chamber. In this way the back pressure between combustion chamber (cyclone) and fuel storage is overcome. Later, for long time operation, it is possible to add a lock hopper at the inlet flange of the storage bin. A stirring device inside the storage bin supports the fuel flow into the screw feeder where the fuel flow rate is metered by the rotational speed of the screw shaft. At the screw outlet, the fuel is discharged onto the vibrator. In contrast to conventional conveying applications, the vibrator is not used for conveying but to produce a uniform flow. From the vibrator the powder drops into the funnel of an injector. In the mixing chamber of the injector, the fuel is entrained into the conveying pipe by a propulsion jet (c.f. section 4.1). The conveying pipe ends at the entrance of the combustion chamber.

It should be mentioned, that the mixture of wood powder and air inside of the fuel conveying components (such as the injector, the conveying pipe and the vibrator housing) is within the explosion limits. Therefore, with respect to component and system design the question of explosion protection has to be carefully dealt with (c.f. Joppich and Hartelbacher (1998)). Hence, various isolating and venting valves connected to an automatic control system including detectors and sensors have been installed. So, if any controlled variable goes out of range, the system will switch over to a safety operation state or even will shut down immediately.

Overview of Injector Feeding. Figure 4.2 shows a diagram of an injector feeder which consists of a driving nozzle, a hopper, a mixing tube and a diffuser. The fuel falling into the hopper is entrained and accelerated by the driving jet entering through the nozzle. The jet spreads by momentum exchange between the driving jet, its surroundings and the fuel. As the gas flow is decelerated the static pressure rises along the injector axis. In conventional feeding applications at ambient pressure the pressure gain is used to overcome the pressure drop of the conveying pipe. The pressure gain mainly depends on the following parameters: the entrance velocity of the driving jet, the diameter of the mixing tube and the mass flow rates of sucked-in air (secondary air) \(m_{\text{air}}\) and of the fuel \(m_{\text{fuel}}\) which corresponds to the loading \(\mu\). However, in the case of fuel feeding, the hopper pressure \(p_{\text{H}}\) can be adjusted, and therefore reaching a high pressure gain is not as important as in other feeding applications.

With reference to section 3.1, a low conveying air ratio requires high loading. Therefore, the variable parameters of the injector have to be primarily considered with regard to high loading. Similar to the pressure gain, the attainable loading strongly depends on the entrance velocity of the driving jet, the secondary air mass flow rate as well as on the mixing tube diameter.

The entrance velocity of the driving jet depends on the total temperature and the pressure ratio across the nozzle. Injectors are frequently classified by operating modes in low, medium and high pressure injectors. While low pressure injectors are working on subsonic driving jet flows, medium pressure injectors have supersonic jets close to the critical state. By comparison, high pressure injectors have supersonic jets with much higher pressure ratios and, therefore, are used only for special applications. To reach high loadings, medium pressure injectors are used for fuel feeding (Muschelknautz (1994)). So, assuming for example air as driving medium with a total temperature of about 300K, the entrance velocity of the driving jet could be in the range of about 320m/s to 380m/s. In chapter 5, the
influences of the secondary air mass flow rate and of the mixing tube diameter on the loading \( \mu \) and in relation to the pressure gain are studied. To provide driving jet velocities within the above range, the test adjustments with respect to nozzle diameter and nozzle total pressure \( p_0 \) are as given in table 4.1.

Table 4.1 Test Adjustments of the Nozzle as a Function of the Primary Air Ratio at 2bar Back Pressure and a Total Conveying Air Mass Flow Rate of 48.38 kg/h

<table>
<thead>
<tr>
<th>primary air ratio [%]</th>
<th>100</th>
<th>80</th>
<th>60</th>
<th>40</th>
</tr>
</thead>
<tbody>
<tr>
<td>secondary air ratio [%]</td>
<td>0</td>
<td>20</td>
<td>40</td>
<td>60</td>
</tr>
<tr>
<td>nozzle diameter [mm]</td>
<td>4.0</td>
<td>3.6</td>
<td>3.3</td>
<td>2.6</td>
</tr>
<tr>
<td>nozzle total pressure ( p_0 ) [bar]</td>
<td>5.1</td>
<td>5.0</td>
<td>4.5</td>
<td>4.8</td>
</tr>
</tbody>
</table>

5. RESULTS OF FUEL FEEDING TESTS

Wood Powder Used in the Feeding Tests. The results are obtained by using a wood powder that is a commercial fuel in Sweden. The mean particle diameter is about 0.36\( \text{mm} \) and the dust content (<0.063\( \text{mm} \)) is approximately 7.5\%. A detailed particle distribution is shown in figure 5.1. The material is mainly from pine and fir with 10% birch bark added. The bulk density is about 285\( \text{kg/m}^3 \) and the moisture is between 5\% and 10\%. In the following investigations a moisture of 10\% is assumed.

5.1 Steady Fuel Flow Rate

An earlier study by Fredriksson and Kallner (1993) showed that screw feeders alone do not produce the necessary steadiness of the fuel flow rate. Moreover, in the present tests, it was observed that the frequency of fluctuations caused by screw feeding corresponds directly to the rotational speed of the screw shaft. The fluctuations of the fuel flow rate decrease, when the screw diameter is reduced and the rotational speed is correspondingly increased. But there are two reasons to refrain from such a measure: A too small screw inlet obstructs the fuel flow from the sending bin into the screw. In addition, the maximum rotational speed must be limited to less than 1m/s to avoid the risk of ignition due to heat generation by high friction (VDI 2263 (1992)).

The solution is a vibrating conveyor placed after the screw feeder. Figure 5.2 shows the improvement in steadiness of the fuel flow rate when the vibrator is used in comparison with screw feeding alone. The standard deviations of feeding fluctuations can be reduced to values of less than 11\% and even 5\% in the operating range between 100kg/h and 300kg/h. It is essential that the vibrator operates at its optimum amplitude. A too low amplitude conveys too little material so that wood powder is accumulated between screw and vibrator. In contrast to this, a too high amplitude conveys too fast and so transmits the fluctuations of the screw as can also be seen in figure 5.2.

Fig. 5.1 Particle Size Distribution of the Used Wood Powder

5.2 Feeding Against Back Pressure

In case of gas turbines of 1MW to 2MW electrical power, the operating pressure of the combustion chamber is about 5bar to 6bar. Regarding the test facility, the operating pressure will be a lower, i.e. as high as 2.8bar. First tests of feeding against back pressure were successfully conducted at a simulated combustion pressure of up to 2.5bar. However, to determine the effects of the parameters mentioned in chapter 4, the pressure in the cyclone is kept constant at 2bar in the following investigations.

5.3 Low Conveying Air Ratio

In section 3.2, the parameters affecting the conveying air ratio were discussed: combustion air ratio, fuel moisture and loading of the
conveying process. Now, the loading, which is the only parameter affecting the conveying air ratio that is determined by feeding techniques, and its enhancement are considered. This is of interest because reaching high loadings is the prerequisite for the reduction of the conveying air ratio.

The presentation of the pressure gain as a function of the loading in figures 5.3 to 5.5 corresponds to the characteristic lines of injectors. In general, the characteristic lines of an injector are approximately straight lines. Comparing figure 5.3 and figure 5.4 it is apparent that an increase of the secondary air ratio significantly improves the maximum loading (loading at plugging). While at 0% secondary air flow loadings are very low, loadings even higher than 5 are possible at 40% primary and 60% secondary air flow. Figure 5.5 shows the pressure gain of the injector with a mixing tube diameter of 25mm at different primary air ratios.

Table 5.1 Combustion Intensities $q_{cc}$, their Required Loadings $\mu$ and the Resulting Conveying Air Ratios $k_i$ at 10% Fuel Moisture and a Mean Combustion Air Ratio of $\lambda=0.7$

<table>
<thead>
<tr>
<th>$q_{cc}$ [s$^{-1}$]</th>
<th>12</th>
<th>16</th>
<th>20</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\mu$ [%]</td>
<td>1.59</td>
<td>2.12</td>
<td>2.64</td>
</tr>
<tr>
<td>$k_i$ [%]</td>
<td>16.20</td>
<td>12.15</td>
<td>9.72</td>
</tr>
</tbody>
</table>

Table 5.2 Combustion Intensities $q_{cc}$, their Required Loadings $\mu$ and the Resulting Conveying Air Ratios $k_i$ at 40% Fuel Moisture and a Mean Combustion Air Ratio of $\lambda=0.7$

<table>
<thead>
<tr>
<th>$q_{cc}$ [s$^{-1}$]</th>
<th>12</th>
<th>16</th>
<th>20</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\mu$ [%]</td>
<td>2.1</td>
<td>2.8</td>
<td>3.5</td>
</tr>
<tr>
<td>$k_i$ [%]</td>
<td>15.62</td>
<td>11.71</td>
<td>9.37</td>
</tr>
</tbody>
</table>

After consideration of the conveying air ratio, which has first priority in fuel feeding, the pressure gain of the injector is briefly reviewed: Regarding figures 5.3 to 5.5, low secondary air ratios enable high pressure gain. As a conclusion, the secondary air ratio has to be chosen high enough to meet the required loading, but on the other hand, should not be exceeded because the resulting pressure gain is becoming lower again. As seen in figure 5.4, the diameter of the mixing tube also strongly influences the pressure gain. At a loading lower than 3.5 a smaller diameter is favourable. However, exceeding a loading of 3.5, larger diameters lead to higher pressure gains.
combustion intensities between 12 s\(^{-1}\) and 20 s\(^{-1}\) as well as a mean combustion air ratio of 0.7, the conveying air ratio does not exceed 16.2%. In case of upsaling to systems of higher power output, it may be expected that even lower conveying air ratios can be realized because the ratio of the conveying pipe diameters of an upscaled facility and the test facility is lower than the power ratio. This is because in case of too small pipe diameters, the friction forces increase significantly and thus aggravate pneumatic conveying. The results of this experimental project are a good base for further development and testing of the combustor chamber and its fuel system.

ACKNOWLEDGEMENT

The financial support by the Austrian Science Fund and the Jubilee Fund of the City of Vienna is gratefully acknowledged.

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CONCLUSION

With respect to a directly wood particle fired gas turbine, the present investigations show that the recently installed pneumatic fuel feed system meets the necessary requirements which are feeding against back pressure, steady and stable fuel flow rate and, moreover, feeding at low conveying air ratios. With regard to fuel moistures between 10% and 40% and combustion chamber operation at