INTRODUCTION

Large amounts of low-calorific fuel gas (LCFG) are generated as byproducts in industrial production processes such as coal mining, steel production, petrochemicals, coal/biomass gasification, and waste landfills.\(^1,2\) In addition to low-calorific value, such gases are typically characterized by fluctuating compositions, high water content, and corrosivity. The difficulty of utilizing such gases with conventional combustion technology means that they are often discharged directly to the environment,\(^3,4\) resulting in a lot of energy wastage and environmental pollution.\(^5\)
Therefore, novel technologies, such as regenerative thermal oxidation (RTO),\textsuperscript{6} exhaust gas recirculation (EGR),\textsuperscript{7,8} and porous media combustion,\textsuperscript{9–11} have been developed to handle LCFGs. However, in both EGR and RTO, large volume regenerators are required to preheat the fresh air, and the corrosivity of exhaust gases is also a major problem for the former. In view of this, porous media combustion has recently attracted attention owing to its wider flammability limit, smaller equipment volume, and lower CO and NO\textsubscript{x} emissions.\textsuperscript{12–14} Further, its advantageous lower flammability limit (LFL) makes LCFG combustion possible.\textsuperscript{15,16}

To achieve the clean and efficient utilization of LCFG, researchers have investigated the effects of porous burner structure on the LFL,\textsuperscript{17,20} as well as on flame stabilization,\textsuperscript{21–23} super-adiabatic effects,\textsuperscript{18,20,23–25} and emissions.\textsuperscript{26,27} Some studies featured porous burner designs with a periodic reciprocal flow.\textsuperscript{28–31} In such burners, an ultra LFL could be further extended for combusting LCFG, and the resulting flames usually stabilized in a specific zone due to periodic reciprocal flow.\textsuperscript{29–31} Some studies considered not only different types of porous media (e.g., packed particles, fiber, or foam), but also their material properties (e.g., thermal conductivity, specific heat, or absorption coefficient).\textsuperscript{16–20,32,33} Other studies focused on the multi-section design of porous media in the burner.\textsuperscript{34,35}

Early investigations into such a process were carried out by Kotani et al.\textsuperscript{17,18} in a burner filled with three sections of alumina tube bundles. They found that the combustion could be maintained at a lean equivalence ratio (ratio of theoretical air to actual air) of \(\phi = 0.2\). Subsequently, Koester et al.\textsuperscript{19} designed a two-layer porous burner with a gas combustion heater capable of both heating fresh gas and initiating the combustion. They reported that the LFL could be extended down to \(\phi = 0.1\) with an ignition temperature on the order of 1000°C. Zhdanok et al.\textsuperscript{20} later operated a similarly designed burner filled with 5.6 mm alumina spheres, and found that the critical equivalence ratio of lean combustion could be extended down to 0.153. This critical value was further corroborated by Shi et al.\textsuperscript{21} by means of numerical simulation. Mital et al.\textsuperscript{22} conducted experiments on the stability limit of a burner composed of a flame supporter and a diffuser layer. Their results showed that, depending on the firing rate, the lift-off limit lay within \(\phi = 0.6–0.7\), and the stable LFL within \(\phi = 0.5–0.6\). Zheng et al.\textsuperscript{37} later reported that the LCFG LFL could be further extended to an equivalence ratio of 0.08 in an SiC foam burner with an inlet velocity of 0.3 m/s.

The aforementioned studies have unequivocally shown the feasibility of utilizing LCFG via porous media combustion. However, most of these studies used simulated LCFG mixtures obtained by diluting a high calorific value gas (usually methane) with air. The air here acts not only as the diluent, but also as combustion components. The simulated LCFG mixtures, therefore, contain a large amount of oxygen. In addition to the LCFG with oxygen, there are also large amounts of LFGC without oxygen (LFGCWO) in industrial production. Apart from the flammable hydrocarbons, the principal components of LFGCWO mixtures such as blast furnace gas, landfill gas, and biomass gas are N\textsubscript{2} and/or CO\textsubscript{2}. The presence of inert gases will inevitably affect the ignition, flame position, combustion temperature, and emissions of LCFG, which results in different combustion behavior compared to that of LCFG, mixtures with oxygen.

Recently, the combustion of LFGCWO mixtures has attracted significant research interest.\textsuperscript{3,38–42} Al-Hamamre et al.\textsuperscript{4} first simulated the landfill gas and medical waste pyrolysis gas by changing the concentrations of CH\textsubscript{4} in N\textsubscript{2}. They reported that the blow-off of the burner occurred at 26% CH\textsubscript{4} at a power of 5 kW, and its blow-off was much higher for landfill gas compared to medical waste pyrolysis gas. Further, CO emissions increased upon reaching blow-off, whereas NO\textsubscript{x} emissions decreased with reductions in the CH\textsubscript{4} concentration or the firing rate. Francisco et al.\textsuperscript{38} simulated a range of LFGCWO mixtures composed of CH\textsubscript{4}, H\textsubscript{2}, CO\textsubscript{2}, CO, and N\textsubscript{2} by changing the lower heating value (LHV) from 5 MJ/Nm\textsuperscript{3} to 44 MJ/Nm\textsuperscript{3}. They studied the effect of mixture composition on flame stability and emissions and found that the range of flame stability noticeably increased with increasing H\textsubscript{2} concentration, and that the maximum emissions of CO and NO\textsubscript{x} were 0.75 g/kg and 0.09 g/kg, respectively. Subsequently, Gao et al.\textsuperscript{39} investigated the combustion of biogas simulated by diluting CH\textsubscript{4} with 25%–40% CO\textsubscript{2}, finding that the flame stability range of the biogas was wider than that of pure CH\textsubscript{4}, and that its flammable limits decreased with increasing CO\textsubscript{2} concentration. Similarly, Keramiotis et al.\textsuperscript{40} experimentally studied the combustion of a typical biogas mixture with 60% CH\textsubscript{4} and 40% CO\textsubscript{2}. They reported that the presence of CO\textsubscript{2} in biogas results in a ~50% higher CO, and a 50%–60% lower NO\textsubscript{x} by in comparison with pure CH\textsubscript{4}. Voss et al.\textsuperscript{41} studied the combustion of a H\textsubscript{2} and CO mixture diluted with CO\textsubscript{2} and N\textsubscript{2} in different volumetric mixing ratios in a porous burner. They analyzed the effect of inert components on the flame thickness and temperature profile along the axial center line and found that the presence of the inert porous medium promoted a submerged flame with thermal flame thickness higher than that in free flame mode. Al-attab et al.\textsuperscript{42} experimentally investigated the combustion of online LFGCWO produced by a biomass gasifier in a two-layer packed burner, finding that the equivalence ratio of ultra-lean stable combustion ranged from 0.33–0.71 for 5 MJ/m\textsuperscript{3} producer gas, and that the CO and NO\textsubscript{x}
emissions were reduced to 6 ppm and 230 ppm, respectively. Huang et al.\textsuperscript{43} numerically investigated the effect of the heat transfer performance of the porous medium in a two-layer porous burner on the combustion of three typical LCFGWO mixtures. They reported that the convective coefficient of the upstream layer had a greater influence on the temperature compared to the downstream layer, and the CO emissions were sensitive to the material properties upstream. Song et al.\textsuperscript{44} studied the combustion characteristics of the LCFGWO by diluting the LPG with N\textsubscript{2}, finding that the LFL of their burner could be extended to 1.4 MJ/m\textsuperscript{3}. Colorado et al.\textsuperscript{45} experimentally investigated the surface-stabilized combustion of the porous burner on a wide range of fuel compositions for blends of natural gas (NG) with CO\textsubscript{2} and H\textsubscript{2}. Their results showed that the amounts of both N\textsubscript{2}O and NH\textsubscript{3} present were small during steady state operation but increased significantly during the ignition process near the lean stability limit.

Most of the aforementioned studies focused on the effects of fuel composition on the flame stability and emissions from LCFGWO combustion. They usually investigated the LFL of LCFGWO by analyzing the dilution ratio, resulting in convenient to judge the LFL, but seldom from the aspects of the LHV. Furthermore, despite of their efforts on improving the structural arrangement of porous medium, so far few studies have been reported on the design of the porous medium length required in the burner. Meanwhile, for LCFG combustion in porous burner, as the porous medium located upstream has a preheating effects, its length has a significant influence on the preheated temperature of the fresh fuel gas, which in turns affects state of super-adiabatic combustion. Thus, the length of porous medium is important parameter needed to be designed and analyzed due to the flame propagation. Moreover, achieving the efficient combustion of LCFGWO mixtures in porous burners remains a significant challenge.\textsuperscript{15}

Against this backdrop, the present study aims to investigate the clean combustion of low-calorific NG/N\textsubscript{2} mixtures in a ceramic foam burner, and to develop a method to predict the porous medium length required in the burner. To this end, the influence of key operating parameters such as LHV, superficial velocity, and excess air ratio on the process was determined through analysis of the combustion characteristics. In addition to the super-adiabatic characteristics of the process (represented in terms of a super-adiabatic ratio), the clean combustion of low-calorific NG/N\textsubscript{2} was evaluated in terms of the stability of temperature profiles and the reduction of emissions. This allowed an optimal excess air ratio to be obtained, based on which the method of predicting the porous medium length required to burn the low-calorific NG/N\textsubscript{2} was developed. A mathematic correlation for designing the burner length was obtained by considering the effects of LHV and flame propagation velocity. These results are expected to contribute to the current scientific understanding of ultra-NG/N\textsubscript{2} combustion in porous burners.

2 | EXPERIMENTAL SECTION

2.1 | Experimental setup

Figure 1 shows a schematic of the experimental setup, consisting of a simple porous burner with a unidirectional flow. The four main parts of the setup included a gas/fuel supply and control system, a two-layer porous burner, a data acquisition system, and a flue gas analyzer. The air was supplied via a compressor and stored in an air tank for flow stability. The dilution gas was N\textsubscript{2} with 99.99% purity,
which was supplied from a high-pressure tank. Liquefied NG supplied from the Chunxiao gas field was stored in a high-pressure tank for use as the fuel gas. Its composition and physical properties are shown in Table 1. The gas flows were regulated using independent mass flow controllers (Sevenstar) with a precision of 0.01 L/min for NG and 0.1 L/min for N₂ and air. To obtain the simulated LCFGWO, the NG and N₂ were premixed in the primary mixer, then mixed with the air in the secondary mixer before entering the porous burner. Both primary and secondary gas mixers were filled with petal-like 18 mm alumina balls to ensure thorough mixing (Figure 2A).

The porous burner consisted of a corundum tube with an internal diameter \( D = 60 \) mm and a length of 500 mm. The outer wall of the tube was covered with a 40 mm thick alumina silicate insulation layer, and the inside was filled with a two-layer SiC ceramic foam of porosity \( \varepsilon = 0.85 \). The upstream layer, which was 50 mm thick and placed at the inlet of the burner, consisted of ceramic foam with 40 pores per inch (PPI) (Figure 2B). The downstream layer consisted of 15 PPI ceramic foam (Figure 2C) packed uniformly in the remaining part of the corundum tube. The uniform layout of the 15 PPI ceramic foam reduces the effect of the structural discontinuity of ceramic foam on combustion flame propagation. The data acquisition system was used to monitor the combustion temperature, with the temperature testing points distributed at intervals of 50 mm along the flow direction. The combustion temperatures were monitored using S-type thermocouples with a measurement range of 0–1573 K and a precision of 0.5% connected to a data logger (Agilent 34970A, with a precision of 1 ms), and recorded at 5 s intervals. A bypass was fitted on the outlet pipe of the burner for extracting the flue gas. The concentrations of CO and NOx were measured using a Horiba PG350 pre-calibrated flue gas analyzer with a precision of 1% at 10 s intervals.

### 2.2 Experimental procedure

After starting the electronic pulsing igniter at the burner outlet, NG was supplied from the high-pressure cylinder and mixed with air in the secondary mixer. The mixture fuel gas with a high equivalence ratio of \( \varphi = 0.8 \) was then ignited. Under \( \varphi = 0.8 \), surface combustion at the porous burner outlet will be converted into submerge combustion as the flame gradually propagates upstream. Thus, the position of the combustion flame can be determined by testing its maximum temperature. After about 45 min, the combustion flame propagates upstream to the layer interface (labeled as T1 in Figure 1). It should be mention that the transient period taking from the burner outlet to the layer interface upstream mainly depends on the flame propagation velocity. To shorten it, we can adjust the equivalence ratio near to 1, and/or decrease the superficial velocity. The flame propagation velocity then increases, that is, the transient period will be shortened correspondingly.

When combustion flame propagates upstream to the interface layer(T1), the N₂ flow was then switched on and adjusted according to the designated dilution ratio, \( \beta \) (defined as \( \beta = \frac{V_{N2}}{V_{NG}} \)), before being mixed with NG in the primary mixer. Here \( V_{N2} \) and \( V_{NG} \) are the volumetric flow-rates of N₂ and NG under standard conditions, respectively. Simultaneously, the air flowrate was also adjusted to the desired excess air ratio \( \phi \) (defined as \( \phi = \frac{m_{fu}/m_{air}}{m_{fu}/m_{air}} = \frac{V_{air,actu}}{V_{air,stoic}} \)), where \( m_{fu} \) and \( m_{air} \) are the mass flow rates of the NG/ N₂ and air, respectively, and \( V_{air,actu} \) and \( V_{air,stoic} \) are the volumetric flow rates of the actual and theoretical air (required for complete combustion per unit mass of fuel), respectively. For comparison, the correspondence of \( \alpha \) with \( \phi \) is shown in Table 2.

The NG/N₂ gas was then mixed with air in the secondary mixer before entering the porous burner. Similarly, NG/N₂ mixtures of different LHV can be obtained by adjusting \( \beta \), as shown in Table 3. The combustion temperature and emissions data were recorded once all temperatures were stable to within 3 K during a 5-min period.

The flow of N₂ was only switched off once the combustion flame had propagated downstream to the burner outlet. The value of \( \varphi \) was then readjusted back to 0.8, resulting in high calorific value combustion and causing the combustion flame to gradually propagate upstream again. Thus combustion heat was regenerated in porous media. As the combustion flame propagated to point T1, the N₂ flow was reopened and adjusted accordingly to achieve the next desired value of \( \beta \). In this way,
combustion of the different NG/N₂ mixtures was achieved in the unidirectional-flow porous burner.

2.3 | Parameter definitions

The global combustion reaction of the NG/N₂ mixtures is expressed as follows:

\[
\text{C}_n\text{H}_m + \beta \text{N}_2 + \left[ \left( n + \frac{m}{4} \right) + \left( \alpha - 1 \right) \right] (\text{O}_2 + 3.76\text{N}_2) \rightarrow \text{nCO}_2 + \frac{m}{2}\text{H}_2\text{O} + \left[ \beta + 3.76 \left( n + \frac{m}{4} \right) \right] \text{N}_2 + (\alpha - 1) (\text{O}_2 + 3.76\text{N}_2)
\]

(1)

In addition, the LHV corresponds to the β value of the NG/N₂ mixtures⁴⁶:

\[
\text{LHV} = \frac{Q_{\text{NG}}}{m_{\text{fu}}\rho_{\text{fu}}} = \frac{Q_{\text{NG}}}{(V_{\text{NG}}\rho_{\text{NG}} + \beta V_{\text{NG}}\rho_{\text{N2}}) \rho_{\text{fu}}}
\]

(2)

where \(\rho_{\text{NG}}, \rho_{\text{N2}},\) and \(\rho_{\text{fu}}\) are the densities of NG, N₂, and LCFG, respectively. Thus, the adiabatic combustion temperature, \(T_{\text{ad}}\), can be calculated as follows⁴⁶:

\[
\text{react} \sum N_i h_i = \text{prod} \sum N_i \left[ h_i^0 + c_{p,i} (T_{\text{ad}} - 298) \right]
\]

(3)

where \(h_i, h_i^0,\) and \(c_{p,i}\) are the enthalpy of formation, the standard enthalpy of formation, and the specific heat at constant pressure, respectively. The total premixed LCFG flow in the porous burner is represented by the superficial velocity,⁴⁷:

\[
u = \frac{V_{\text{total}}}{A} = \frac{4 \left( V_{\text{NG}} + V_{\text{N2}} + V_{\text{air,actu}} \right)}{\pi D^2}
\]

(4)

where \(V_{\text{total}}\) is the total volumetric flow rate under standard condition (Nm³/s), and \(A\) is the empty cross-sectional area of the porous burner (m²).

The oxygen concentration in the combustion flue gas is different for values of \(\alpha\). To facilitate a uniform comparison, the concentrations of CO and NOx emissions were converted to the values at 6% concentration of O₂ in the flue gas as follows:

\[
c = c' \times \frac{\alpha'}{\alpha_6}
\]

(5)

where \(c\) and \(c'\) are the converted and measured emissions concentrations, respectively. \(\alpha'\) and \(\alpha_6\) are the measured and converted (at O₂ 6%concentration) excess air ratios, respectively.


3 | RESULTS AND DISCUSSION

3.1 | Stable flame propagation

The temperature variation of each testing point in the burner with time is shown in Figure 3 (\(u = 0.3\) m/s, \(\alpha = 1.3\), and LHV = 3.36 MJ/Nm\(^3\)). The results show that the temperature of T1 is initially at its peak before decreasing as time increases, whereas the temperatures of the other testing points start lower and rise up to their peak values before coming back down. The temperature peak gradually emerges from T1 to T9 in sequence, which indicates that the flame propagates downstream from the two-layer interface to the exit. The reason for this is that the combustion flame is anchored right at the interface upstream when the combustion of NG/N\(_2\) starts. Thus, the peak temperature of T1 corresponds to the combustion flame temperature. Then, as the combustion flame gradually propagates downstream, the temperatures of the other testing points reach their peak values in turn. It is worth noting that the time intervals of the temperature peaks between two adjacent points are identical (~300 s). This indicates that the flame propagation velocity is constant (~0.167 mm/s), which is consistent with previous studies on LCFG mixtures containing oxygen.\(^{20,23,34,37}\) The results confirm that stable flame propagation was obtained for NG/N\(_2\) combustion under specified conditions.

After 45 min, the combustion flame propagated to T9 located near the outlet, where its peak temperature was much lower than the other points due to the combustion occurring outside the porous medium. To continue the combustion process, the N\(_2\) flow can be switched off and the equivalence ratio adjusted back to 0.8, as described in section 2.2. This causes the combustion flame to propagate upstream again, and the combustion heat to be regenerated in porous medium, which preheats the NG/N\(_2\) mixture. The cycle is then repeated at a different \(\beta\) of the NG/N\(_2\) mixture.

It is also worth noting that the temperature peaks at each testing point shown in Figure 3 are different. This is due to the randomness and complexity of the ceramic foam structure, as a result of which the flame propagation is a non-integral translation process, which is always accompanied by flame inclination, diversion, and break-up.\(^{20,22,24}\) Furthermore, the propagation flame does not necessarily pass through the exact locations of each thermocouple, which are fixed on the two-layer interface between the ceramic foam and internal wall of the burner. Thus, although the temperature peak of each testing point is different, the close proximity of these points to the propagating flame means that these peaks can be treated as the maximum combustion temperatures.

3.2 | Influence of LHV on temperature profiles

To understand the influence of LHV on the combustion temperature, the axial temperature profiles for NG/N\(_2\) combustion at different LHV are shown in Figure 4 (\(u = 0.3\) m/s, \(\alpha = 1.3\), \(t = 600\) s). The temperature peak at each LHV can be regarded as the maximum combustion temperature, and its position can, therefore, be considered as the initial position of the combustion flame. The maximum combustion temperature appears at T1 for LHV = 6.16 MJ/Nm\(^3\), which indicates that the flame is directly anchored at the two-layer interface upstream due to the discontinuity of pore diameter of SiC ceramic foam. When the LHV decreases to 3.36 MJ/Nm\(^3\), the combustion flame begins to propagate downstream. A new relative stability profile of combustion temperature is formed again at ~600 s, and its temperature peak appears at an axial position of \(x = 100\) mm, indicating that this is the location of the initial flame position. Subsequently, the newly formed flame begins to stably propagate downstream, which indicates that at lower LHV, the flame can escape from the interface of pore diameter discontinuity and propagate downstream, although this interface does help to stabilize the flame. As the LHV decreased further, the initial position of combustion flame gradually propagated downstream, appearing close to the outlet of the porous burner (\(x = 400\) mm) at LHV =1.03 MJ/ Nm\(^3\).

Based on the results in Figure 4, it may be inferred that if the LHV decreases further, the initial flame position will propagate to a position outside of the porous medium and blow-out will occur. The gradual propagation downstream of the initial flame position with decreasing LHV mainly...
arises from the ignition energy required by the NG/N₂. It is well known that the lower the LHV of the NG/N₂ mixture, the higher the ignition energy required. Hence, as the LHV decreases, the fresh premixed NG/N₂ needs to flow further and absorb more heat from the porous medium downstream to attain its ignition energy, resulting in a downstream shift of the initial flame position.

Next, the maximum combustion temperatures at different LHV (Figure 4) were compared with the corresponding adiabatic combustion temperatures calculated from combustion theory in Figure 5. The results show that the adiabatic combustion temperature was strongly positively correlated with LHV, while the experimentally measured maximum combustion temperatures varied only slightly with LHV. For LHV >3.36 MJ/Nm³, the adiabatic combustion temperature is higher than the corresponding experimental values, which indicates that sub-adiabatic combustion was occurring under these conditions. On the other hand, for LHV <3.36 MJ/Nm³, the adiabatic combustion temperature is lower than the corresponding experimental values, indicating super-adiabatic combustion. Therefore, 3.36 MJ/Nm³ is the critical LHV for this system. Furthermore, the super-enthalpy state is characterized by the super-adiabatic ratio (which is defined as the ratio between the maximum observed combustion temperature and its adiabatic counterpart). Hence, these results indicate that the super-adiabatic state gradually increased with decreasing LHV in the porous burner.

3.3 Influence of velocity on temperature profiles

Figure 6 shows the influence of superficial velocity on the axial temperature profiles of the porous burner (LHV = 1.76 MJ/Nm³, α = 1.3, t = 600 s). It is evident that the temperatures upstream of the flame (peak) position decrease with increasing superficial velocity, while those downstream of the flame position gradually increase. This indicates that the axial temperature profile gradually shifts downstream with increasing superficial velocity as it follows the flame propagation path. Based on the combustion theory, an increase in the fuel gas superficial velocity (the thermal power) should result in a rise in the maximum combustion temperature. However, for u = 0.4 m/s, the experimentally measured peak was smaller than that for u = 0.3 m/s. This is mainly caused by the effect of the flame propagation. For u = 0.3 m/s, the temperature of the testing peak is significantly higher than those of testing points on either side, demonstrating that the propagation flame passes through its peak.
position at exactly \( t = 600 \) s. Hence, its peak temperature can be considered as the maximum combustion temperature. For \( u = 0.4 \) m/s, on the other hand, the temperature difference between the testing point (\( x = 200 \) mm) and the peak (\( x = 250 \) mm) is relatively small. This indicates that at \( t = 600 \) s, the flame has already left the position at \( x = 200 \) mm but has not yet propagated to \( x = 250 \) mm. In other words, as the combustion flame is located in between these two points, its temperature is relatively higher than that at \( x = 200 \) mm, and the temperature at \( x = 250 \) mm is close to (but not exactly) the maximum combustion temperature. Then, the temperature of the testing peak at \( u = 0.4 \) m/s might be slightly smaller than that at \( u = 0.3 \) m/s. Furthermore, comparison with Figure 4 shows that the influence of superficial velocity on the combustion flame is much smaller than that of LHV.

### 3.4 Influence of LHV on emissions

Clean combustion of the NG/N\(_2\) mixture is evaluated not only on the basis of a stable combustion temperature, but also on the resulting pollutant emissions. Figure 7 shows the average emission of CO per unit of thermal power as a function of LHV at different values of \( u \) (\( \alpha = 1.3 \)). It is evident that for LHV \( > 1.42 \) MJ/Nm\(^3\), the CO level is ultra-low (<50 mg/kWh), and that the amount of emissions in this range (~30 mg/kWh) appears to be independent of the LHV. However, for LHV \(<1.42 \) MJ/Nm\(^3\), the CO emission increases rapidly, rising to over 624 mg/kWh for \( u = 0.2 \) m/s as the LHV decreases to its lean flammability value of 1.03 MJ/Nm\(^3\). This is mainly due to the initial flame position propagating downstream to the outlet of the burner (Figure 4). Although a relatively stable combustion can be maintained under lean flammability conditions, the maximum combustion temperature is significantly reduced as there is no porous media downstream absorbing the heat from the flue gas. If the LHV decreases further, the initial position of the combustion flame will propagate downstream outside of the porous media, resulting greatly decreasing its maximum combustion temperature. This will cause the CO emissions to increase sharply to a high level of >2000 mg/kWh, even further extinguish the combustion. On the other hand, it is clear that \( u \) has very little effect on CO emissions.

Figure 8 shows the NO\(_x\) emissions under the same conditions as in Figure 7. Clearly, the NO\(_x\) emissions per unit of flue gas volume for this burner are ultra-low (<30 mg/ Nm\(^3\)) in Figure 8A. Further, the NO\(_x\) emissions increase nonlinearly with increasing LHV. This is mainly because the NO\(_x\) produced by NG/N\(_2\) combustion is classified as thermal NO\(_x\), but not as prompt or fuel NO\(_x\). Based on the Zeldovich theory,\(^{46}\) the relationship between NO\(_x\) production and combustion temperature is exponential. Hence, as the combustion flame temperature increases with increasing LHV, the thermal NO\(_x\) increases.

Interestingly, when NO\(_x\) emissions are converted from unit of mg/Nm\(^3\) to mg/kWh, the function of NO\(_x\) emission with the LHV differs significantly (Figure 8B). The NO\(_x\) emissions increase quickly with increasing LHV first, then decrease slightly. The reasons are as follows. The NO\(_x\) emissions increase with the increasing combustion temperature as LHV increases, whereas the heat release also increases. For lower LHV, as the release rate of combustion heat is lower than the increase rate of NO\(_x\), the NO\(_x\) per unit of thermal power increases quickly with the increase in LHV. However, the release rate of combustion heat increases with increasing LHV, combined with the low NO\(_x\) effect of porous media combustion, the release rate of combustion heat will be larger than the increase rate of NO\(_x\) as the LHV increases to a critical threshold (about 1.42 MJ/Nm\(^3\) in present work). Therefore, the NO\(_x\) per unit of thermal power reaches to its peak, then slightly decreases with further increasing LHV. In addition, increasing \( u \) (thermal power) leads to the increase of NO\(_x\) emission.

### 3.5 Influence of excess air ratio on emissions

Figure 9 shows the average CO emissions as a function of \( \alpha \) at different \( u \) values (LHV = 3.36 MJ/Nm\(^3\)). For the sake of comparison, the CO concentrations were converted to the case with 6% \( \text{O}_2 \) in flue gas. The results show that the CO concentration was generally at a very low level of <200 mg/kWh, decreasing sharply as \( \alpha \) increased to 1.3, then basically stabilizing to a value of <40 mg/kWh. This is mainly due to the following reasons. There are large percent of non-combustible components in LCFGWO. For smaller \( \alpha \) (<1.3), the combustible components of LCFGWO are not fully mixed with oxygen due to the presence of the non-combustible components. Oxygen deficiency exists in local combustion zone, resulting in high CO emissions. As \( \alpha \) increases, the mixing effect between them can be improved, and combustion is more complete. The CO emissions then decreases correspondingly. However, when \( \alpha \) increases to a critical threshold (1.3), all combustible components can be fully mixed with oxygen. Under this condition, complete combustion is carried out, and CO emission decreases to its minimum. Therefore, increase of \( \alpha \) has further little effect anymore. In addition, at \( u = 0.5 \) m/s, the combustion flame propagates downstream outside of the porous medium for \( \alpha > 1.5 \), causing blow-out to occur.
Figure 10 shows the NO\textsubscript{x} emissions under the same conditions as in Figure 9. The results show that, just as for CO, the emissions level of NO\textsubscript{x} was also ultra-low, initially decreasing as $\alpha$ increases to 1.5 before slightly increasing in the range from $\alpha = 1.5$–2.0. For the thermal NO\textsubscript{x}, its formation rate is strongly influenced not only by the maximum combustion temperature, but also by the concentrations of N\textsubscript{2} and O\textsubscript{2} (being directly proportional to the former and proportional to the square root of the latter). For $\alpha < 1.5$, the formation of NO\textsubscript{x} is significantly affected by the maximum combustion temperature, and low concentrations of O\textsubscript{2} and N\textsubscript{2} in the combustion zone mean that their influence is negligible. As $\alpha$ increases, the maximum combustion temperature (and hence the NO\textsubscript{x} concentration) decreases. For $\alpha > 1.5$, the NO\textsubscript{x} formation is mainly affected by O\textsubscript{2} and N\textsubscript{2} due to their increased concentrations. Based on the

**FIGURE 7** Influence of LHV and inlet velocity on CO emissions in the porous burner. (A) NO\textsubscript{x} emissions per unit of flue gas volume (mg/Nm\textsuperscript{3}). (B) NO\textsubscript{x} emissions per unit of thermal power (mg/kWh)

**FIGURE 8** Influence of LHV and inlet velocity on NO\textsubscript{x} emissions in the porous burner

**FIGURE 9** Influence of the excess air ratio and inlet velocity on CO emissions in the porous burner

**FIGURE 10** Influence of excess air ratio and inlet velocity on NO\textsubscript{x} emissions in the porous burner
effect of \( \alpha \) on CO and NO\(_x\) emissions, it may be inferred that its optimal range for NG/N\(_2\) combustion is from 1.3–1.5. These results also show that the NO\(_x\) concentration is positively correlated with \( u \).

### 3.6 Estimation of porous medium length

The analysis in Section 3.5 showed that the optimal \( \alpha \) value is 1.3. According to the analysis in Section 3.3, the initial position of combustion flame is mainly determined by the LHV and not by \( u \). Further, the distance between the burner inlet and the initial position of the combustion flame can be considered as the minimum length of the porous medium required for NG/N\(_2\) combustion. Figure 11 shows this minimum length as a function of LHV (extracted from the data in Figure 5).

Based on the trend observed in Figure 11, the relationship between the minimum length, \( L_{\text{min}} \), and LHV was fitted using a simple exponential function, as follows:

\[
L_{\text{min}} = L_0 + A \exp(k \times \text{LHV})
\]  

(6)

where the constant term \( L_0 = 63.9 \) mm, pre-exponential factor \( A = 1121.1 \), and exponential factor \( k = -1.181 \). It is well known that the combustion flame does not stabilize at a specific position while burning the LCFG mixture in porous burner, usually resulting in a stable flame propagation downstream (as revealed by the analysis in Figure 3).\(^{21-24}\)

Therefore, it is necessary to increase the length of porous medium to maintain combustion mode. Thus, considering the flame propagation, a design length of porous medium, \( L \), can be further expressed in terms of the propagation velocity, \( w \), as follows:

\[
L = L_{\text{min}} + wt
\]  

(7)

For ultra-lean (or LCFG) combustion, the value of \( w \) can be calculated as follows:\(^{21,23}\):

\[
\omega \approx \frac{u \varepsilon}{(1 - \varepsilon)} \left( C_{\text{pg}} \rho_g \frac{\Delta T_{\text{ad}}}{\Delta T_{\text{s,i}}} \right) \left( 1 - \frac{\Delta T_{\text{ad}}}{\Delta T_{\text{s,i}}} \right)
\]  

(8)

where \( t \) is the combustion time, \( \varepsilon \) is the porosity of the porous medium, \( C_p \) is the specific heat, \( \Delta T_{\text{ad}} \) is the temperature rise due to adiabatic combustion, \( \Delta T_{\text{s,i}} \) is the temperature rise in the combustion wave, and the subscripts \( s \) and \( g \) denote the gas and solid phases, respectively.

It is worth noting that, Equation (7) may still have its adaptability for other excess air ratios although it is obtained in terms of \( \alpha =1.3 \). The reason is as follows. Despite of the different theoretical air volume required for LCFGWO with different compositions, whereas \( \alpha \) is usually selected from 1.10 to 1.50 in practical combustion. The difference between the selected value and 1.3 is relatively small. It can be treated as a smaller amount. Furthermore, the variation of \( \alpha \) can be reflected by the superficial velocity \( u \), which is already considered in Equation (8) of the flame propagation velocity.

On the other hand, the longer the porous medium, more combustion heat will be accumulated inside, which strengthen the preheating ability of the burner, resulting in combustion of lower NG/N\(_2\) ratio mixtures. Generally, due the production cost, the design criterion for the burner involves shorting the length of porous medium as much as possible while ensuring the stable LCFG combustion. Thus, according to Equation (8), a reasonable length of the porous medium can be designed by taking the time-dependent term \((wt)\) into consideration. That is, the stable propagation time of combustion flame determines the length of the porous burner required.

### 4 CONCLUSIONS

In this study, the combustion of low-calorific NG/N\(_2\) mixtures was experimentally investigated in a porous burner. The effects of varying the key operating parameters of the burner were analyzed with respect to the combustion characteristics occurring in porous medium. The length of porous medium was further estimated by considering the LHV and flame propagation velocity at the optimal excess air ratio. The results are helpful to the clean combustion of LCFGWO mixtures such as landfill gas, biomass.
gasification gas, and blast furnace gas. The main conclusions are as follows:

1) The initial position of combustion flame gradually moves downstream as the LHV decreases to its lower limit of 1.03 MJ/Nm$^3$. For LHV <3.36 MJ/Nm$^3$, the combustion is super-adiabatic, and the super-adiabatic ratio is negatively correlated with the LHV.

2) Both NO$_x$ and CO emissions are ultra-low. NO$_x$ emission per unit of flue gas volume was positively correlated with LHV, while it per unit of thermal power increases with increasing LHV first, then decrease slightly. CO emission decreases rapidly at lower LHV (<1.42 MJ/Nm$^3$), before becoming independent of LHV.

3) The optimal excess air ratio for clean combustion of the low-calorific NG/N$_2$ was determined to range from 1.3 to 1.5. The concentration of CO decreases sharply as the excess air ratio increases to 1.3, before becoming independent of the excess air ratio at all inlet velocities, while that of NO$_x$ shows a minimum value at an excess air ratio of 1.5.

4) The minimum length of the porous medium required to burn low-calorific NG/N$_2$ is negatively correlated with LHV, and can be estimated by fitting the exponential function $L_{\text{min}} = L_0 + A \times \exp(k \times LHV)$ to the experimental data. The design length of the porous medium (porous burner) can be estimated by considering flame propagation speed, according to equation $L = L_{\text{min}} + wt$.

ACKNOWLEDGEMENTS
This work was supported by the Zhejiang Provincial Natural Science Foundation of China (Grant No. LY15E060007) and Innovation Ability Promotion of Science & Technology SMEs of Shandong Province (Grant No. 2021TSGC1339).

CONFLICT OF INTEREST
The authors declare that they have no conflict of interest.

ORCID
Guanqing Wang https://orcid.org/0000-0003-3483-7311

REFERENCES
1. Chomiak J, Longwell JP, Sarofim AF. Combustion of low calorific value gases: problems and prospects. Prog Energy Combust Sci. 1989;15(2):109-129.
2. Göransson K, Söderlind U, Jie H, Zhang WN. Review of syngas production via biomass DFBGs. Renew Sustain Energy Rev. 2011;15(1):482-492.
3. Ptasinski KJ, Prins MJ, Pierik A. Exergetic evaluation of biomass gasification. Energy. 2007;32(4):568-574.
4. Al-Hamamre Z, Diezinger S, Talukdar P, Issendorff VF, Trimis D. Combustion of low calorific gases from landfills and waste pyrolysis using porous medium burner technology. Process Saf Environ Prot. 2006;84(4):297-308.
5. Abanades S, Abbaspour H, Ahmadi A, et al. A conceptual review of sustainable electrical power generation from biogas. Energy Sci Eng. 2021;10(2):630-655.
6. Chang DS, Cheng KP, Wang RW. Developing low temperature recovery technology of waste heat in automobile factory. Energy Sci Eng. 2018;6(5):460-474.
7. Jafarmadar S, Zehni A. Numerical investigation of the effects of dwell time duration in a two-stage injection scheme on exergy terms in an IDI diesel engine by three-dimensional modeling. Energy Sci Eng. 2013;2(1):1-12.
8. Zhang L, Wang HU, Zhong X, et al. Study on the influence mechanism of mixture stratification on GCI combustion and the compound injection strategy under high load operation. Energy Sci Eng. 2021;9(12):2434-2448.
9. Trimis D, Drust F. Combustion in a porous medium. Advances and applications. Combust Sci Technol. 1996;121(1):153-168.
10. Bedoya C, Zarzalis N, Habreurther P. Experimental and theoretical study of combustion under elevated pressure within porous inert media. Energy Technol. 2017;5(8):1124-1133.
11. Habib MA, Tahir F, Nemitallah MA, Ahmed WH, Bagd HM. Experimental and numerical analysis of oxy-fuel combustion in a porous plate reactor. Int J Energy Res. 2015;39(9):1229-1240.
12. Howell JR, Hall MJ, Elzey JL. Combustion of hydrocarbon fuels within porous inert media. Prog Energy Combust Sci. 1996;22(2):121-145.
13. Dehaj MS, Ebrahimi R, Shams M, Farzaneh M. Experimental analysis of natural gas combustion in a porous burner. Exp Thermal Fluid Sci. 2017;50(1):134-143.
14. Nemitallah MA, Mansir IB, Habib MA. Experimental and numerical study of oxy-methane flames in a porous-plate reactor mimicking membrane reactor operation. Int J Energy Res. 2019;43(13):7040-7057.
15. Wood S, Harris AT. Porous burner for lean-burn applications. Prog Energy Combust Sci. 2008;34(5):667-684.
16. Ling B, Ling ZQ, Kuang M, et al. Fuel-lean VOCs combustion in a porous burner stacked with alumina balls: a case for ethylene combustion. Int J Energy Res. 2019;43(2):970-982.
17. Kotani Y, Takeno T. An experimental study on stability and combustion characteristics of an excess enthalpy flame. Proc Combust Inst. 1982;19(1):1503-1509.
18. Kotani Y, Behbahani HF, Takeno T. An excess enthalpy flame combustor for extended flow ranges. Proc Combust Inst. 1984;20(1):2025-2033.
19. Koester GE, Kennedy LA, Subramaniam VV. Low temperature wave enhanced combustion in porous systems. In ASME/JSME Therm Eng Conf., 55-60. Department of Mechanical Engineering.
20. Zhdanok SA, Kennedy LA, Koester G. Superadiabatic combustion of methane air mixtures under filtration in a packed bed. Combust Flame. 1995;100(2):221-231.
21. Bubnovich VI, Zhdanok SA, Dobrego KV. Analytical study of the combustion waves propagation under filtration of methane–air mixture in a packed bed. Int J Heat Mass Transf. 2006;49(15–16):2578-2586.
22. Dobrego KV, Zhdanok SA, Zaruba AI. Experimental and analytical investigation of the gas filtration combustion inclination instability. Int J Heat Mass Transf. 2001;44(11):2127-2136.

23. Wang GQ, Tang PB, Li Y, Xu JR, Durst F. Flame front stability of low calorific fuel gas combustion with preheated air in a porous burner. Energy. 2019;170:1279-1288.

24. Saveliev AV, Kennedy LA, Fridman AA, Puri IK. Structures of multiple combustion waves formed under filtration of lean hydrogen-air mixtures in a packed bed. Proc Combust Inst. 1996;26(2):3369-3375.

25. Aldushin AP, Rumanov JE, Matkowsky BJ. Maximal energy accumulation in a superadiabatic filtration combustion wave. Combust Flame. 1999;118(1–2):76-90.

26. Keramiotis C, Stelzner B, Trimis D, Founti M. Porous burners for low emission combustion: an experimental investigation. Energy. 2012;45(1):213-219.

27. Mathis WM, Ellzey JL. Flame stabilization, operating range and emissions for a methane/air porous burner. Combust Sci Technol. 2003;175(5):825-839.

28. Comarin F, Saveliev AV, Fridman AA, Kennedy LA. A reciprocal flow filtration combustor with embedded heat exchangers: numerical study. Int J Heat Mass Transf. 2003;46(6):949-961.

29. Hoffmann JG, Echigo R, Yoshida H, Tada S. Experimental study on combustion in porous media with a reciprocating flow system. Combust Flame. 1997;111(1):32-46.

30. Jugjai S, Somjetlertcharoen A. Multimode heat transfer in cyclic flow reversal combustion in a porous medium. Int J Energy Res. 1999;23(3):183-203.

31. Dobrego KV, Gnesdilov NN, Lee SH, Choi HK. Lean combustibility limit of methane in reciprocal flow filtration combustion reactor. Int J Heat Mass Transf. 2008;51(9–10):2190-2198.

32. Barra AJ, Diepvens G, Ellzey JL, Henneke MR. Numerical study of the effects of material properties on flame stabilization in a porous burner. Combust Flame. 2003;134(4):367-379.

33. Gao HB, Qu ZG, Feng XB, Tao WQ. Combustion of methane/air mixtures in a two-layer porous burner: a comparison of alumina foams, beads, and honeycombs. Exp Thermal Fluid Sci. 2014;52:215-220.

34. Shi JR, Xie MZ, Liu H, Li G, Zhou L. Numerical simulation and theoretical analysis of premixed low-velocity filtration combustion. Int J Heat Mass Transf. 2008;51(7–8):1818-1829.

35. Hsu PF, Evans WD, Howell JR. Experimental and numerical study of premixed combustion within Nonhomogeneous Porous Ceramics. Combust Sci Technol. 1993;90(1):149-172.

36. Mital R, Gore JP, Viskanta R. A study of the structure of submerged reaction zone in porous ceramic radiant burners. Combust Flame. 1997;111(4):175-184.

37. Zheng CH, Cheng LM, Li T, Luo ZY, Cen KF. Filtration combustion characteristics of low calorific gas in SiC foams. Fuel. 2010;89(9):2331-2337.

38. Francisco RW, Rua F, Costa M, Catapan RC, Oliveira AAM. On the combustion of hydrogen-rich gaseous fuels with low calorific value in a porous burner. Energy Fuels. 2009;24:880-887.

39. Gao HB, Qu ZG, Tao WQ, He YL, Zhou J. Experimental study of biogas combustion in a two-layer packed bed burner. Energy Fuels. 2011;25(7):2887-2895.

40. Keramiotis C, Founti MA. An experimental investigation of stability and operation of a biogas fueled porous burner. Fuel. 2013;103(1):278-284.

41. Voss S, Mendes MAA, Pereira JMC, Ray S, Pereira JCF, Trimis D. Investigation on the thermal flame thickness for lean premixed combustion of low calorific H2/CO mixtures within porous inert media. Proc Combust Inst. 2013;34:3335-3342.

42. Al-attab KA, Ho JC, Zainal ZA. Experimental investigation of submerged flame in packed bed porous media burner fueled by low heating value producer gas. Exp Thermal Fluid Sci. 2015;62(4):1-8.

43. Huang R, Cheng LM, Qiu KZ, Zheng CH, Luo ZY. Low-calorific gas combustion in a two-layer porous burner. Energy Fuels. 2016;30(2):1364-1374.

44. Song FQ, Wen Z, Dong ZY, Wang EY, Liu XL. Ultra-low calorific gas combustion in a gradually-varied porous burner with annular heat recirculation. Energy. 2017;119:213-219.

45. Colorado A, McDonell V. Surface stabilized combustion technology: An experimental evaluation of the extent of its fuel flexibility and pollutant emissions using low and high calorific value fuels. Appl Therm Eng. 2018;136(5):206-218.

46. Glassman I, Yetter RA. Combustion, 4th ed. Elsevier Inc.; 2008.

47. Wu WY. Fluid Mechanics. Peaking University Publish House; 1982 (In Chinese).