Short overview on combustion systems scale-up with emphasis on NOx emissions of gas-fired furnaces

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Abstract
Historically, scale-up of technologies have been a fundamental driver for successful economies in enhancing performance of applications. In the context of combustion technologies, several scale-up approaches were developed with the aim of transferring relevant characteristics of combustion systems from laboratory- to industrial-scale system. A reasonable selection of scaling approaches is required to perform a proper scaling of a combustion system. In the current study, the development of scaling methods and their use for technical applications is presented. In order to provide a guideline for the examination of combustion systems to be scaled up, common and recently published scale-up methods are compared qualitatively, as well as the investigation on scale-up of combustion chambers, is addressed. Due to the expected further technical and environmental restrictions, the focus of the implementation is directed towards NOx emissions.

KEYWORDS
combustion, NOx emissions, scale-up, standardization

1 INTRODUCTION

The research and development of technological applications is time-consuming and resource-intensive. In order to keep development costs at the minimum necessary level, investigations of newly developed systems are initially carried out on a laboratory scale. Scaling methods are required to transfer these systems into industrial scale. In this ‘enlargement process’, the primary processes and the main properties of the system (stability, heat transfer rate, pollutants emissions) must be kept as similar as possible. In such a way, predictions of numerous properties and simplified standardization of design concepts in a scaled system are possible. Penner\(^1\) provided a definition for scale-up of a combustion system as the ability to design new combustion devices with predictable performance on the basis of test experience with old devices.

To elaborate on a historical development of scale-up of the technology, Figure 1 shows the progress of scale-up based on a time scale:

The publications listed in Figure are related with different fields, including production, energy and combustion technology. The time period under consideration follows the industrialization era and spans more than 150 years. In contrast, Figure 2 shows number of publications that can be found in the database ‘Scopus’ with keywords containing ‘scale-up’. Three of the considered keywords ‘scale-up and emission’, ‘scale-up and combustion’ and ‘scale-up and flame’ refer in varying degrees to combustion technology. Given that ‘scale-up’ is the most...
general of the considered keywords, it yields the largest number of publications.

The first research paper using ‘scale-up’ as keyword was published in 1933. The paper is concerned with the scale-up of the destructive hydrogenation process. The number of published articles increases to a greater or lesser extent for all considered keywords from about 2005. This can be connected with an increasing requirement of mass production, as well as with the desire to improve process developments. These requirements indicate a desire for simplified standardization of design concepts.

2 | NITROGEN OXIDES

Nitrogen oxides (NOx) are known for their negative effects on humans and the environment. NOx usually refers to the combination of NO and NO2 molecules. NO emissions can be formed during combustion processes by three main formation mechanisms: the extended Zeldovich-, Fenimore- and Fuel-NO mechanism. The NO mechanisms in non-premixed flames are summarized in numerous publications. Moreover, NO formation can occur via intermediate N2O-, NO2- and NNH-paths. For high temperature combustion processes of natural gas typically, only thermal NO is considered. In addition to the temperature, the concentration of O and OH, as well as the residence time, are of primary importance for thermal NO production.

The NO formed by the Fenimore mechanism is called prompt NO. Prompt NO becomes relevant in fuel rich, non-premixed flames and is formed by a complex formation mechanism via radicals: nitrogen, cyanides and amidoxides, which in turn, are formed by a reaction between CH radicals and bimolecular N compounds. In fuel NO, on the contrary, the N radical descends from the fuel. Fuel NO is formed at temperatures about 1100 K via a complex radical formation mechanism (see Figure 17.8. in Ref.24).

Correlations of NOx to the thermal input power \( \dot{Q} \) are of a particular interest, because \( \dot{Q} \) provides direct information about the size of the combustion system according to (1):

\[
\dot{Q} = H_u \cdot m_{\text{fuel}} = \frac{\pi}{4} \cdot H_u \cdot \rho_{\text{fuel}} \cdot U_{\text{fuel}} \cdot D_{\text{fuel}}^2
\]

where is \( H_u \) is the mass-related lower calorific value, \( m_{\text{fuel}} \) the mass flow of fuel, \( \rho_{\text{fuel}} \) the density of fuel, \( U_{\text{fuel}} \) the fuel exit velocity and \( D_{\text{fuel}} \) the fuel nozzle diameter.

3 | COMPARISON OF METHODS FOR BURNER SCALING

The most commonly used scaling approaches for industrial combustion systems are the constant velocity (CV) and constant residence time (CRT) approaches. By showing the scaling effect of the mixing process depending on the characteristic burner diameter \( D_0 \) and the characteristic inlet velocity \( U_p \), Hawthorne et al. can be considered as one of the initiators of the CV and CRT methods. An ideal scaling of all relevant processes, such
as the mixing process and chemical reaction kinetics, at varying thermal input power levels in the laboratory and industrial scale is not possible. Since the value of a dimensionless parameter can cover considerably more operating points than a value of a dimensional parameter, common scaling approaches are based on keeping dimensionless parameters constant.

With the CV approach, the characteristic inlet velocity $U_0$ and consequently the ratio $U_{01}/U_{02}$ between system 1 and system 2 is kept constant. The characteristic burner velocity and the characteristic burner diameter are usually related to the fuel lance so that applies $U_0 = U_{\text{fuel}}$ and $D_0 = D_{\text{fuel}}$. Hence, according to (1) it can be concluded that $D_{01}/D_{02} = (Q_1/Q_2)^{1/2}$. From this relationship, all geometric quantities are calculated. With the CRT approach, the convective time scale $D_0/U_0$ is kept constant, whilst the thermal input power $Q$ is varied. Both approaches are based on the scaling of the large macro-scale turbulent mixing process. Table 1 compares used scale-up approaches similar to CV and CRT.

In the context of acoustic stabilization of a non-premixed jet flame, Cole's scaling method was developed. The laboratory-scale burner is shown in Figure 2 in Ref. The acoustic driving frequency was set so that diminished pressure amplitude at high frequencies as well as momentum-dominated mixing and premature ignition at low frequencies are avoided. In addition, the characteristic mixing time should be kept as constant as possible in order to improve the emissions performance. Based on a self-defined scaling factor $S = \bar{V}_2/\bar{V}_1$, where $\bar{V}$ is the central air jet volumetric flow rate of the respective scale, the following scaling criteria resulted:

$$U_{\text{air},2}/U_{\text{air},1} = S^{1/2}$$

$$A_2/A_1 = S^{1/2}$$

$$d_2/d_1 = S^{1/4}$$

where $U_{\text{air}}$ is the central air jet velocity, $A_j$ is the jet area and $d_j$ is the jet diameter. Consequently, the approach is based on a proportional increase of the central air jet velocity and the jet area during scaling.

In order to maintain a high heat release rate constant and to achieve a high recirculation rate at a combustion process for moderate or intense low-oxygen dilution (MILD), Kumar et al. developed a scaling approach. In this scale-up method the characteristic burner diameter is increased in proportion to the cubic value of the thermal input power, that is $D_{01}/D_{02} = (Q_1/Q_2)^{1/3}$, for the purpose of keeping the heat release rate constant. All the major geometric quantities are determined from this relationship. In order to limit the pressure drop the empirically determined value of 100 m/s is set for the air injection velocity. However, the position and the injection velocity of secondary air are determined numerically to improve the recirculation rate and to burn the residual CO. The mild combustion regime was successful at the convective timescale of $D_0/U_0 < 80$ µs. The method of Kumar et al. has been proven successful for fuels with calorific values in the 4.5–45 MJ/kg range.

For a reverse-flow combustor, Pramanik and Ravikrishna developed two scale-up methods due to the disadvantages of the CV, CRT, approaches of Cole et al. and Kumar et al. (see Table 1). These are named as constant volume of the combustor to jet momentum ratio (CM) and constant volume of the combustor to jet kinetic energy ratio (CK). The combustor volume was related to inflow conditions in both approaches since for reverse-flow combustors the inflow conditions are crucial. To keep the recirculation rate constant, the following relationship was used in the derivation of these two approaches:

$$L_{\text{OCC}} \sim D_0$$

where $L_{\text{OCC}}$ is the characteristic length of the combustion chamber. For ease of use, Pramanik and Ravikrishna have equated $L_{\text{OCC}}$ with the characteristic diameter of the combustion chamber $D_{\text{OCC}}$.

In the 3.3–25 kW thermal input range considered, the near-nozzle mixing is independent of $D_{\text{OCC}}$ and is controlled by the oxidizer-to-fuel momentum flux ratio $J_{\text{ox}}/J_f \gg 1$. For larger thermal input ranges, on the other hand, the ratio is $J_{\text{ox}}/J_f \leq 1$, where the characteristic momentum flux is defined as follows: $J_0 = A_0 \cdot U_0^2$. Since in the CM approach the ratio of the characteristic volume of the combustion chamber and the incoming jet momentum is kept constant during scaling, the following basic formula can be obtained:

$$\frac{L_{\text{OCC}}^3}{U_{\text{OCC}}^2 \cdot D_{\text{OCC}}^2} = \text{const.}$$

In the CK approach, however, the formula for the characteristic kinetic energy $E_{\text{kin}}$ of the inlet flow used for turbine blades is applied:

$$E_{\text{kin}} = 0.5 \cdot A_0 \cdot \rho \cdot U_0^3$$

where $A_0$ is the characteristic blade area and $\rho$ is the density of the fluid. In the case of a burner, $A_0$ corresponds to the nozzle cross-section area. Using equation (7), the
| Method       | \( \frac{D_{02}}{D_{01}} \sim \) | \( \frac{U_{02}}{U_{01}} \sim \) | Heat release rate | \( \frac{\eta}{\eta_0} \sim \) | Re~ | Investigation on NOx?                                                                                     | Advantages ⊕ and disadvantages ⊝ referring to stated combustion systems, investigated experimentally and/or numerically, in the certain thermal input range                              |
|-------------|-----------------------------------|-----------------------------------|-------------------|-------------------------------|-----|-----------------------------------------------------------------------------------------------------------------|----------------------------------------------------------------------------------------------------------|
| CV          | \( \left( \frac{Q_2}{Q_1} \right)^{1/2} \) | Const.                            | \( \left( \frac{Q_2}{Q_1} \right)^{-1/2} \) | \( \frac{Q_2}{Q_1} \)         | 1/2 | Yes; for example\(^22,25,26,28,30,33,35,46,48-50,52-54\); Clear correlations exist;\(^26,53\) depends on operating conditions and previous investigations | ⊕ Methane/air, premixed, \( \phi = 0.5, \dot{Q} = \text{const.}; \) constant pressure drop and flame stability;\(^48\) |
| CRT         | \( \left( \frac{Q_2}{Q_1} \right)^{1/3} \) | \( \left( \frac{Q_2}{Q_1} \right)^{1/3} \) | Const. Const. | \( \left( \frac{Q_2}{Q_1} \right)^{2/3} \) | Yes; for example\(^22,25,26,28,30,33,35,46,48-50,52-54,35\); Clear correlations exist;\(^26,53\) depends on operating conditions and previous investigations | ⊕ Methane/air, premixed, \( \phi = 0.5, \dot{Q} = \text{const.}; \) Stable combustion performance;\(^8\) |
| CRT         | \( \left( \frac{Q_2}{Q_1} \right)^{1/3} \) | \( \left( \frac{Q_2}{Q_1} \right)^{1/3} \) | Const. Const. | \( \left( \frac{Q_2}{Q_1} \right)^{2/3} \) | Yes; for example\(^22,25,26,28,30,33,35,46,48-50,52-54,35\); Clear correlations exist;\(^26,53\) depends on operating conditions and previous investigations | ⊕ Methane/air, premixed, \( \phi = 0.5, \dot{Q} = \text{const.}; \) Stable combustion performance;\(^8\) |
| CRT         | \( \left( \frac{Q_2}{Q_1} \right)^{1/3} \) | \( \left( \frac{Q_2}{Q_1} \right)^{1/3} \) | Const. Const. | \( \left( \frac{Q_2}{Q_1} \right)^{2/3} \) | Yes; for example\(^22,25,26,28,30,33,35,46,48-50,52-54,35\); Clear correlations exist;\(^26,53\) depends on operating conditions and previous investigations | ⊕ Methane/air, premixed, \( \phi = 0.5, \dot{Q} = \text{const.}; \) Stable combustion performance;\(^8\) |

\( CV \) and \( CRT \) refer to stated combustion systems, investigated experimentally and/or numerically, in the certain thermal input range.
| Method               | \(\frac{\dot{m}_a}{\dot{m}_a} \sim\) | \(\frac{U_{a}}{U_{m}} \sim\) | Heat release rate | \(\frac{\dot{m}_b}{\dot{m}_a} \sim\) | Re~ | Investigation on NOx? | Advantages ⊕ and disadvantages ⊖ referring to stated combustion systems, investigated experimentally and/or numerically, in the certain thermal input range |
|----------------------|--------------------------------------|-------------------------------|-------------------|--------------------------------------|-----|------------------------|----------------------------------------------------------------------------------|
| Cole et al.\(^{34}\) | \(\left(\frac{\dot{Q}_2}{\dot{Q}_1}\right)^{1/4}\) | \(\left(\frac{\dot{Q}_2}{\dot{Q}_1}\right)^{1/2}\) | \(\left(\frac{\dot{Q}_1}{\dot{Q}_0}\right)^{1/4}\) | \(\left(\frac{\dot{Q}_1}{\dot{Q}_0}\right)^{-1/4}\) | \(\left(\frac{\dot{Q}_2}{\dot{Q}_1}\right)^{3/4}\) | Yes; measured 200 ms downstream of the burner, without mentioning the number of measurements: <100 ppm | Pyrolysis gases/air, non-premixed, acoustically excited, \(\phi = 0.53 - 0.9, \dot{Q} = 4.75 - 700\ kW:\) ⊕ Stable combustion control in case of fluctuations of certain input parameters ⊖ Reduced emissions of NOx and CO at lower input power ⊖ Reduced mixing quality throughout the system beyond a certain thermal input value |
| Kumar et al.\(^{28,35}\) | \(\left(\frac{\dot{Q}_2}{\dot{Q}_1}\right)^{1/3}\) | \(\text{1, with } U_{O_2} = 100\ m/s\) | Const. | \(\left(\frac{\dot{Q}_1}{\dot{Q}_0}\right)^{1/3}\) | \(\left(\frac{\dot{Q}_2}{\dot{Q}_1}\right)^{1/3}\) | Yes; some scattered global values without clear regularity: <26 ppm | LPG or producer gas/air (MILD), non-premixed, \(\phi = 1, \dot{Q} = 3 - 150\ kW:\) ⊕ Low O\(_2\) mass fraction and high temperature zone in most of the combustion chamber volume at higher thermal input ⊖ Low emissions (CO, NO) at scaling ⊖ Wide range of operating points (very different fuels) ⊖ Relatively inflexible with regard to configuration of the combustion system\(^{28,35}\) |
| CM\(^{35}\) | \(\left(\frac{\dot{Q}_2}{\dot{Q}_1}\right)^{2/3}\) | \(\left(\frac{\dot{Q}_2}{\dot{Q}_1}\right)^{1/3}\) | \(\left(\frac{\dot{Q}_1}{\dot{Q}_0}\right)^{-1/5}\) | \(\left(\frac{\dot{Q}_1}{\dot{Q}_0}\right)^{1/5}\) | \(\left(\frac{\dot{Q}_2}{\dot{Q}_1}\right)^{3/5}\) | No | Syngas/air (MILD), non-premixed, reverse-flow, \(\phi = 0.89, \dot{Q} = 3.3 - 25\ kW:\) ⊖ Pressure drop smaller than with CRT ⊖ Mixing time increases at a lower rate as compared to the CV approach |
| CK\(^{35}\) | \(\left(\frac{\dot{Q}_2}{\dot{Q}_1}\right)^{3/7}\) | \(\left(\frac{\dot{Q}_2}{\dot{Q}_1}\right)^{1/7}\) | \(\left(\frac{\dot{Q}_1}{\dot{Q}_0}\right)^{-2/7}\) | \(\left(\frac{\dot{Q}_1}{\dot{Q}_0}\right)^{2/7}\) | \(\left(\frac{\dot{Q}_2}{\dot{Q}_1}\right)^{4/7}\) | No |
relationship on which the CK approach is based on can be formulated as follows:

\[
\frac{I_{0CC}^3}{U_{0CC}^3 \cdot D_{0CC}^2} = \text{const.} \quad (8)
\]

All relevant geometric quantities are calculated from the resulting relationship \(D_{01}/D_{02} = (Q_1/Q_2)^{2/5}\) for the CK approach and from \(D_{01}/D_{02} = (Q_1/Q_2)^{0.7}\) for the CM approach, respectively (see Table 1). Since neither large thermal inputs nor NOx emissions have been investigated using CM and CK approaches, this area requires further research.

Table 1 summarizes NOx correlations for different scaling approaches. It should be noted that NOx values were investigated only for a few thermal power ranges. Thus, no reliable NOx predictions via the scaling approaches are achievable. The lack of NOx data related to Scale-up methods can be explained by the fact that usually the range of scales and operating conditions covered with a focus on NOx pollutant emissions has been very limited.\(^8,22\) One of the few such studies, where gaseous fuels and their NOx pollutant emissions have been examined extensively throughout a wide thermal input range is the Scaling 400 Study.\(^38-44\)

Since scale-up usually refers to laboratory- and industrial-scale systems, the question arises how these scales can be assigned to the thermal input power levels. According to\(^8,45\) the range of 0.5–30 kW for the non-premixed flame can be attributed to the laboratory scale and the range of 5–70 MW to the full industrial scale. In this respect, Figure 3A shows these lower and upper values of the thermal input power within laboratory and full industrial scales as a function of the characteristic burner diameter. The values of the characteristic burner diameter at base points in Figure 3A are obtained from the non-premixed combustor investigated by Sadakata and Hirose.\(^45\) Resulting from scaling with approaches listed in Table 1, the differences in the characteristic burner diameter across the laboratory and full industrial scale are shown. Figure 3B completes this overview illustrating the intermediate area between laboratory- and industrial-scale by scaling the design point \((\bar{Q}_{Lab,\text{max}}, \bar{D}_{Lab,\text{max}})\) from Figure 3A, where \(\bar{Q}_{Lab,\text{max}}\) corresponds to the maximum value of the thermal input power in the laboratory range and \(\bar{D}_{Lab,\text{max}}\) to the averaged characteristic diameter. The further design points \((\bar{Q}_j, D_{0j})\) in Figure 3B result from the scale-up to two, four and eight times of the thermal input power \(\bar{Q}_{Lab,\text{max}}\). It can be seen that the mean values of the characteristic diameters \(\bar{D}_{0j}\) for a certain thermal input power \(Q\) can be described by a parabolic curve. This is due to the stronger scattering of the \(D_0\) values at higher thermal input powers for different scale-up approaches.

Considering the characteristic velocity constant during scale-up by means of the CV method, the fuel volume rate and consequently the characteristic diameter increase strongly. On the contrary, the approach of Cole et al.\(^34\) yields the smallest value of the characteristic diameter. The bigger value of \(D_{0CV}\) compared to \(D_{0Cole}\) can also be derived to \(D_{0CV}/D_{0Cole} = Q_{2CV}^{1/4}/Q_1^{1/4} > 1\) from Table 1. In general, the greater the exponent of the ratio \(D_{01}/D_{02}\), the greater is the characteristic burner diameter \(D_{02}\) in the scaled system.

For more complex burner designs, the above-mentioned scale-up approaches are not always sufficient because scaling of certain burner components or certain process variables can lead to local falsifications.\(^26\) For instance, Niguse and Agrawal\(^46\) have modified a scaled-up

![Figure 3](image-url)
swirl number by means of two concentric co-swirlers, so that swirl number can be kept constant. Barroso et al. have used the $\pi$-Theorem to determine dimensionless ratios keeping CO emissions constant. Individual components, however, were scaled using both approaches CV and CRT.

4 | COMBUSTION CHAMBER SCALING

Weber and Mancini\(^8\) point out that the scale-up of combustion systems with a combustion chamber must be performed by retaining the chamber. This implies that the combustion chamber influences relevant combustion process properties, namely the aerodynamic, ignition, flame length and pollutant emissions. An approximation given by Weber and Mancini\(^8\) states the following: if the ratio between the characteristic diameter of combustion chamber $D_{0,CC}$ and $D_0$ is greater than the value of three, the influence of the combustion chamber wall on the near-burner zone can be neglected. The $D_{0,CC}/D_0$-ratio controls the aerodynamics of the combustion chamber until a certain combustion chamber length $L_{0,CC}$. That is, the influence of $L_{0,CC}$ on the combustion process does not exist above a certain value. According to Ariatabar et al.\(^48\) the combustion chamber length $L_{0,CC}$ can be estimated by means of the characteristic combustion chamber volume $V_{0,CC}$:

$$L_{0,CC} \sim V_{0,CC}^{1/3} \quad (9)$$

When determining NOx-production, the combustion chamber volume $V_{0,CC}$ must be taken into account due to the influence of the residence time. At full-load conditions and larger burner scales, the highest thermal NOx-production rate is in the near-burner region for a non-premixed combustion operating with natural gas fuel.\(^22,49\) Also in partially premixed flames operating with H\(_2\) as fuel, the near-burner zone is one of the largest contributors to the total NO.\(^26\) In certain constellations,\(^50\) prompt NO also makes a non-negligible contribution to the total NO in this region. If a sufficiently large increase in performance is carried out by the CRT approach, there occur evermore local regions in the combustion chamber with, to some extent, very different local residence times. This issue leads to increased NOx values. It represents a failure of the CRT approach and suggests that the influence of the combustion chamber should be taken into account during scaling.\(^22,26\)

An example of changes in combustion chamber volume according to Pramanik and Ravikrishna\(^35\) due to scaling are shown in Figure 4 for different scale-up approaches. The differences for scaled combustors are up to about 1300%. With the CV approach and consequently $U_0 = \text{const.}$, the increase in thermal input power $Q$ is defined according to (1) solely via the $D_{CV,0}^2$ - increase. With the three other scaling approaches CK, CM and CRT, the relationships $D_{0,2}/D_{0,1} = f(Q_{2}/Q_1)$ according to Table 1 yield a smaller value of $D_{0,2}$. From Table 1 it can be seen that $D_{CV,0}^2/D_{CK,0}^2 = Q_{2}^{1/14}/Q_1^{1/14} \approx 1.16, D_{CV,0}^2/D_{CM,0}^2 = Q_{2}^{1/10}/Q_1^{1/10} \approx 1.22$ and $D_{CV,0}^2/D_{CRT,0}^2 = Q_{2}^{1/6}/Q_1^{1/6} \approx 1.40$, so that the burner diameters decreases as $D_{CV}^2 > D_{CK}^2 > D_{CM}^2 > D_{CRT}^2$. If $D_{0,CC}/D_0 = \text{const.}$ is assumed for all cases about the regarded power input range, the same ratios for the combustion chamber diameters are applied. For the correlation between the combustion chamber lengths and the combustion chamber diameters, however, the expression (9) can be used. Kumar et al.\(^28\) has also investigated the change of combustion chamber dimensions in dependence of scaling approaches. His results correspond to ones shown in Figure 4.

Considering the variation of the aerodynamic and pollutant emissions during the scale-up, the increase in uniform heat dissipation and the thermal similarity in the combustion chamber. Thermal similarity can be achieved with suitable insulation and the use of cooling water pipes. However, the thermal insulation properties of the combustion chamber have to be improved, the higher the ratio between the surface and the volume of the combustion chamber is. A typical relationship between combustion chamber wall temperature and NOx emissions at a constant thermal input power is exponential.\(^51\) Regarding this, a suitable combustion chamber wall temperature has to be maintained. Moreover, the $D_{0,CC}/D_0$-ratio has to be kept approximately constant to maintain aerodynamic similarity.\(^22\)

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**Figure 4**: Relative changes in combustion chamber volume whilst scaling the base combustion system of 3.3–25 kW with approaches CV (constant velocity), CRT (constant residence time), CM (constant volume to jet momentum ratio) and CK (constant volume to jet kinetic energy ratio).\(^35\)
5 | CONCLUSIONS

A short overview of scale-up approaches in the context of combustion systems is provided. The focus is on gaseous fuels and their NOx pollutant emissions. After outlining the historical development of scale-up methods, the derivations for scaling approaches are delineated. The listed scale-up criteria are qualitatively compared by showing advantages and disadvantages of each criterion for different operating points $\varphi \in [0.4; 1]$ and $Q \in [0.4 \text{ kW}; 700 \text{ kW}]$. In addition, previous experiences on determined NOx correlations during scale-up have been summarized. The experience that has already been accumulated with the scale-up of burners and combustion chambers can be used to carry out the scale-up process more efficiently in the future.

In order to forecast the NOx pollutant emissions more effectively using the scale-up methods, more research is acquired in the application of different scale-up approaches over a wide range of scales and operating conditions for different combustion system configurations. In this respect, a further research for the scale-up methods of constant volume to jet momentum ratio (CM) and constant volume to jet kinetic energy ratio (CK) is advocated, in particular, for higher thermal inputs.

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