Mixture Temperature-Controlled combustion: a revolutionary concept for ultra-low NOX emission

Viktor Józsa

Budapest University of Technology and Economics, Faculty of Mechanical Engineering, Department of Energy Engineering, 1111 Budapest, Műegyetem rkp. 3., Hungary

Abstract

Mixture Temperature-Controlled (MTC) combustion is a novel concept, offering extremely low NOX emission without a compromise. Unlike flameless combustion or exhaust gas recirculation techniques, the oxidizer can be ambient air, offering robust realization in potential applications. The essence of the concept is the central injection of cold air to delay the ignition of the fuel-air mixture. Hence, the flame root is not localized near the fuel nozzle, facilitating distributed combustion, which ultimately leads to a reduced NOX emission. The distributed flame also means low flame luminosity. The MTC combustion mode can be maintained easier under lean conditions, which is highly favorable in gas turbine applications. Compared to V-shaped flames, more than 50% reduction in NOX can be realized without an increase in CO emission. During the experiments, no unstable flame behavior was observed up to an equivalence ratio of 0.57. The central cold air jet, which also serves as the atomizing medium of the airblast atomizer, leads to a low swirl number. MTC combustion also offers reduced noise, and the spectrum contains geometry-related components principally. Hence, it is hypothesized that this concept has a lower tendency to thermoacoustic instabilities than V-shaped flames.

Keywords: MTC combustion; NOx; CO; swirl; lean; pollutant emission
1 Introduction

Gas turbine technology for aviation and power generation is almost a century old. Initially, flame stabilization, control, and advanced materials were in the focus of development [1]. The increasing concern about the environmental impact, especially at high altitudes, lead to systematic development to eliminate soot and mitigate NO\textsubscript{X} emission by the millennium [2]. Earlier combustors used non-premixed flames since they are highly stable [3], however, their pollutant emission was excessive [4].

The time scale of NO formation significantly exceeds the flow time scale [5], hence, rich burn-quick quench-lean burn (RQL) combustors were introduced [6]. The rich flame root in RQL ensures stable combustion while the quick dilution makes combustion lean, leading to a notably reduced NO\textsubscript{X} emission. This concept is still under development in aero engines by leading jet engine manufacturers [7], however, lean premixed prevaporized (LPP) burners offer a further reduction in NO\textsubscript{X} emission. Nevertheless, these burners have a tendency for serious combustion instabilities [8], which is the major drawback for their general introduction in aviation. To solve this issue, a pilot flame [9] or advanced, online control algorithms can be used [10]. Catalytic combustion, which theoretically provides a fully homogeneous temperature field, was never introduced to gas turbines due to incomplete combustion [11]. Otherwise, it offers low NO\textsubscript{X} and CO emission.

The trivial way to achieve zero NO\textsubscript{X} emission is oxyfuel combustion, i.e., there is no N\textsubscript{2} in the oxidizer as it is pure O\textsubscript{2}. This technology offers increased cycle efficiency as the flue gas temperature can be increased up to the capabilities of the structural materials and the cooling system without worrying about excessive NO\textsubscript{X} formation. However, unresolved technological challenges make oxyfuel combustion economically unfeasible [12]. No practical application can withstand pure O\textsubscript{2}-fuel combustion, hence, flue gas recirculation helps in lowering the flame temperature. Overall, the competitiveness of oxyfuel combustion with carbon capture
and storage facility will increase with CO$_2$ emission prices. Even though solar and wind power are on a steep incline, gas turbines are excellent technological complements in balancing their fluctuating power generation [13].

A recent concept under development is flameless combustion [14], which bears the potential to implement in gas turbines [15]. To further reduce the thermal NO$_X$ emission present in LPP burners, exhaust gas recirculation (EGR) is applied to decrease flame temperature [15]. Hence, the heat release is occurring in a large volume, and pressure fluctuations are inferior to those of an LPP burner [16]. The implementation of EGR is not trivial for either boiler [17] or gas turbine. Nevertheless, the technology of EGR is successfully used in solid fuel combustion, e.g., in fluidized bed systems [18] and grate boilers [19]. A more widespread application is internal combustion engines, including both compression ignition [20] and spark ignition [21] variants. The beneficial effect of distributed combustion on NO$_X$ emission has been shown by Karyeyen et al. [22] by diluting the mixture by both CO$_2$ and N$_2$. The former diluent gas contributed to an increased CO emission, while the latter one reduced it until 18% O$_2$. Further combustion air dilution kept this pollutant at the same level.

By evaluating the temperature field, decreasing the peaks and the reduction of high-temperature zones is in the focus of the development of existing combustion chambers. NO$_X$ emission mitigation can be achieved by both fuel staging [23] and air staging [24,25]. The latter option includes both primary air [26] and secondary air [27] control. These techniques are all aiming to provide a more homogeneous mixture. Nevertheless, creating the perfect mixture is hindered by the realization of fuel inlet, air inlet, cooling, and other design considerations, which are necessary for reliable operation. The homogeneous mixture is also critical in hypersonic vehicles [28], and internal combustion engines [29] to meet the continuously stringent pollutant emission standards, incorporating CO$_2$ emission of cars [30].
The above concepts and their principles point to that homogeneous mixture combustion in a large volume would be the most beneficial solution for the mitigation of NO\textsubscript{X} emission. It also maintains a proper temperature pattern for turbine blades, allowing the reduction of the share of cooling air, ultimately leading to higher efficiency [1]. A possible solution without the difficulties of EGR implementation or oxygen enrichment is the Mixture Temperature-Controlled (MTC) combustion concept, which is introduced in the present paper. The fundamentals, background, and technical details are summarized in Section 2. Section 3 discusses flame characteristics of the maintained volumetric combustion, pollutant emission analysis, including the comparison with LPP swirl combustion, which is the most advanced solution with numerous successful practical applications. Lastly, the acoustic characteristics are evaluated, showing that MTC combustion has a relatively low tendency to thermoacoustic instabilities.

2 Materials and methods

The present section begins with the discussion of the MTC combustion concept to highlight the practical requirements of designing a burner around it. Secondly, the experimental setup is introduced, also including some operational experiences. The last subsection details the atomization characteristics. Due to the novelty of the concept, this section includes a more in-depth explanation than usual.

2.1 The Mixture Temperature-Controlled combustion concept

Steady-operating turbulent burners highly benefit from swirl vanes to ensure a homogeneous fuel-air mixture, hence, it is also a core part of MTC combustion. Similar to other swirl burners, hot combustion air flows through the swirl vanes. The principal phenomenon to exploit is providing a relatively cool stream at the center to avoid the increased heat release of
lean premixed burners at the flame root [31–33], which is the principal source of their NOX emission. The MTC burner solves this issue by having a central plain-jet airblast atomizer. Consequently, flashback or fuel nozzle coking cannot occur in MTC combustion due to the cold central air flow into which the fuel is injected. Besides generating a fine spray, the nozzle generates a high-speed cold air stream that surrounds the fuel stream through an adiabatic expansion as:

\[
T_2 = T_1 \cdot \left(\frac{p_2}{p_1}\right)^{\frac{y-1}{y}},
\]

where \( T \) is the absolute gas temperature, \( p \) is the static pressure, and \( y \) is the specific heat ratio. Subscripts 1 and 2 represent the pre and post-expansion points. The schematic of the mentioned setup, which enables MTC combustion mode, is shown in Fig. 1.

![Fig. 1. Burner layout of the MTC combustion mode.](image)

The turbulent flow induces notable temperature fluctuations in the mixture, decreasing the heat release rate [34]. The regime diagram and the theoretical background, which describes distributed combustion, was discussed by Im et al. [35]. Also, the mixture temperature correlates logarithmically with the ignition delay [36]. Both phenomena contribute to distributed combustion, facilitated by the MTC burner design. Increasing \( p_1 \) in Eq. (1) seems
favorable, however, there is a reasonable limitation set by the application. For instance, the spray cone angle, hence, the spreading of the spray is decreasing with the increase of $p_1$ [37], even though intense turbulence facilitates spray spreading more [38]. The spray characteristics are detailed in Subsection 2.3, and the effect of expansion at the nozzle on the average mixing tube temperature is discussed in Subsection 3.1. A video showing MTC combustion is available as supplementary material of ref. [39]. This earlier work also demonstrated that MTC combustion mode is sensitive to the fuel properties, nevertheless, the same burner allowed smooth operation with both diesel oil and 100% biodiesel. Probably, MTC burners can be designed for a wide range of fuels, since fuel flexibility is increasingly important [40]. Nevertheless, it has to be noted that a single design is suspected of working flawlessly in a narrower range of fuel properties. Flame luminosity in MTC combustion mode is very low due to the highly homogeneous mixture, similar to air dilution with inert gas [41]. Hence, the effect of radiative heat transfer has a significantly lower impact on droplet evaporation than that on fuel sprays of internal combustion engines [42].

2.2 Experimental setup

The experimental setup is shown in Fig. 2. The fuel was diesel oil, according to the EN590:2017 standard, which was delivered from a pressurized tank. Its flow rate was measured by an Omega FPD3202 flow meter with 1.5% calibrated measurement uncertainty. The thermal power was uniformly 13.3 kW in all the cases. During volumetric combustion, the flame size was approximately $150 \times 150 \times 150$ mm, meaning a $4 \text{ MW/m}^3$ volumetric heat release rate.
The overall equivalence ratio was varied in the range of $\phi = 0.57 - 0.86$ in four steps, corresponding to 3, 5, 7, and 9% $O_2$ concentration in the flue gas. Stable flame for an extended operation was only possible up to 9% $O_2$; lean flame blowout was reached in a minute at 10% $O_2$, while 11% $O_2$ lead to an immediate blowout. This result is in line with the theoretical lean flammability limit of hydrocarbon fuels, which is $\phi \approx 0.5$ [43]. Besides $O_2$, CO and NO concentrations were also measured by a Testo 350 flue gas analyzer. The corresponding uncertainties are shown in Table 1, considering that the dry pollutant emissions in gas turbine applications are corrected to 15% $O_2$ [44]. All the discussed emission data in Subsection 3.2 were also subjected to this correction.
It was shown in Eq. (1) that the atomizing gauge pressure, $p_g$, is a governing parameter of the MTC combustion mode. In the present setup, it was varied from 0.3 bar to 0.9 bar in five equidistant steps. Since atomizing air also contributes to $\phi$, the combustion air flow rate was hence decreased when $p_g$ was increased. Even though the combustion conditions were controlled by the $O_2$ concentration in the flue gas, the atomizing air flow rate was required to calculate the spray characteristics, which is discussed in Subsection 2.3. A pre-calibrated Omega FMA1842A flow meter with 1 liter/min uncertainty was used, which meant 2.3–4.2% relative uncertainty. The electric preheater provided a constant 200 °C combustion air temperature, based on previous experiences [39] since MTC combustion was observed only up to 250 °C in the case of diesel oil. All the thermometers in the cold lines were B-class Pt100 resistance thermometers with 0.4 °C accuracy, while K-type thermocouples were used along the path of the hot combustion air with 2.2 °C accuracy.

The annular swirl vanes were designed for axial combustion air flow. Initially, 60° vanes were used to have a high swirl number due to the notable contribution of the axial thrust of the atomizing jet. However, the flame could not be stabilized in this case. Hence, a 45° swirl vane was used in all subsequent measurements, resulting in a geometric swirl number, $S = 0.787$ [45]. The overall swirl number, considering the atomizing jet, is presented in Subsection 3.1.

The acoustic signal recording was performed by a GRAS 146AE microphone with a DT 9837B data acquisition card at 20 kHz for 30 s at each setup. To keep the sensor cool, an in-house designed water-cooled socket was used with a Helmholtz resonator, which was tuned to 20 kHz eigenfrequency. Consequently, the 5% positive bias due to the amplification of the resonator in the spectral domain occurs at 4.3 kHz. This is acceptable in combustion since the

| Gas / $\phi$ (1) | 0.86 | 0.76 | 0.67 | 0.57 |
|------------------|------|------|------|------|
| NOx [ppm]        | 0.69 | 0.75 | 0.86 | 1.0  |
| CO [ppm]         | 1.00 | 1.13 | 1.29 | 1.50 |
| $O_2$ [V/V%]     | 0.067| 0.075| 0.086| 0.1  |
spectral range of interest is located below this frequency [46]. A similar configuration was used by Noiray and Denisov [47] in the case of a turbulent swirl burner.

### 2.3 Atomization characteristics

Estimation of the atomization characteristics was based on the fuel flow rate at 13.3 kW thermal power and the physical properties of the fuel, shown in Table 1. The below calculations were based on a similar atomizer configuration [38].

**Table 2. Relevant properties of the diesel oil.**

| Property                           | Value |
|-----------------------------------|-------|
| Lower heating value [MJ/kg]       | 43    |
| Stoichiometric air requirement [kg/kg] | 14.4  |
| Fuel mass flow rate [kg/h]        | 1.11  |
| Density [kg/m³]                   | 820   |
| Surface tension [mN/m]            | 25.6  |
| Kinematic viscosity [mm²/s]       | 2.53  |

Since the high-speed free jet also acts as a cold air stream to delay ignition, enabling MTC combustion, both the air-to-fuel mass flow ratio, AFR, and the momentum flux ratio, MFR, are higher than usual in airblast atomization. These non-dimensional quantities are defined by Eqs. (2) and (3).

\[
AFR = \frac{\dot{m}_A}{\dot{m}_F}, \tag{2}
\]

\[
MFR = \frac{\rho_A \cdot w_A^2}{(\rho_F \cdot w_F^2)}, \tag{3}
\]

where \( \dot{m} \) is the mass flow rate, \( \rho \) is the density, and \( w \) is the flow velocity. Subscripts \( A \) and \( F \) refer to air and fuel, respectively. Since fuel evaporates from the droplet surface, the most representative droplet diameter of the generated spray in combustion is the surface-to-volume, or the Sauter Mean Diameter, SMD. This measure of airblast atomization was found to correlate with both the Weber number, We, and the Ohnesorge number, Oh [48]. The latter one is the
ratio of We and Reynolds number, Re, to eliminate the flow velocity. They are calculated by Eqs. (4)–(6) as:

$$Re_A = \rho_A \cdot d_0 \cdot w_A / \mu_A,$$

(4)

$$We_A = \rho_A \cdot d_0 \cdot w_A^2 / \sigma,$$

(5)

$$Oh = \sqrt{We_F / Re_F} = \mu_F / (\sigma \cdot d_0 \cdot \rho_F)^{0.5},$$

(6)

where $d_0 = 1.2$ mm is the initial diameter of the liquid jet, $\mu$ is the dynamic viscosity, and $\sigma$ is the surface tension. The SMD of the spray can be estimated by Eq. (7), based on [38]:

$$SMD / d_0 = (0.477 \cdot We_A^{-0.5} + 0.35 \cdot Oh) \cdot (1 + 1/AFR).$$

(7)

The above-detailed variables for all $p_g$ are presented in Table 3, except Oh. It was 0.013 at all conditions, as it is calculated from the physical properties of the diesel oil and the initial liquid jet diameter.

| $p_g$ [bar] | $AFR$ [1] | $MFR$ [1] | $Re_A$ [1] | $We_A$ [1] | $SMD$ [μm] |
|-------------|------------|------------|------------|------------|------------|
| 0.3         | 2.05       | 830        | 12825      | 3550       | 22.1       |
| 0.45        | 3.53       | 3410       | 20331      | 11.0       |
| 0.6         | 4.34       | 29039      | 14578      | 12.2       |
| 0.75        | 5.19       | 35374      | 20331      | 11.0       |
| 0.9         | 5.19       | 4755       | 23100      | 14.1       |

Based on the Re and We values, the jet breakup mechanism is atomization [49], and the droplet breakup mode is shear breakup [50].
3 Results and discussion

This section details the swirl number and the average inlet temperature, followed by a map, highlighting the characteristic flame shapes at various setups. Images of selected setups are also presented in the first subsection. Then NO\textsubscript{X} and CO emissions are shown in Subsection 3.2, quantifying the difference between the presented flames. Subsection 3.3 discusses the spectral analysis of the acoustic signal, focusing on the comparison of the observed flame shapes.

3.1 Flame characteristics

The MTC combustion mode features a weak swirl ($S < 0.6$), even though the swirl vane would otherwise generate a strong swirl, leading to a V-shaped flame. The axial momentum of the atomizing jet significantly reduces $S$, while the increasing preheated combustion air flow rate for lower equivalence ratio counteracts it, shown in Fig. 3a. Also, the overall combustion air temperature, $T_{OCA}$, is affected, presented in Fig. 3b. The increasing $p_g$ results in lower temperature after the expansion at the atomizer nozzle, according to Eq. (1), which is counteracted by the combustion air flow rate. Hence, the trend is similar to that of Fig. 3a. Figure 3c shows the percentage of the atomizing air mass flow rate and the total air flow rate, $r_A$. This is the inverse of Figs. 3a and 3b, as the high atomizing air flow rate results in low $S$ and $T_{OCA}$. 
There were three stable flame shapes distinguished during the combustion tests. They were the straight flame, V-shaped flame, and distributed combustion, which corresponds to
the well-set conditions of MTC combustion mode. Also, a transitory behavior was observed between various stable flame shapes, as shown in Fig. 4.

| $p_g$ [bar] | $\Phi$ [°C] |
|------------|-------------|
| 0.9        | 0.57 d d d s |
| 0.75       | 0.67 d d d s |
| 0.6        | 0.76 d d d s |
| 0.45       | 0.86 d t t s |
| 0.3        |              |

Fig. 4. Flame shapes at each measurement point. Light blue (d): MTC, distributed combustion; brown (v): V-shaped flame; orange (s): straight flame; light green (t): transitory flame between the upper and lower neighboring flame shapes.

The inhomogeneous flow field, i.e., the swirling hot air outside and the cold central axial air flow characterizes the operation of this burner setup since V-shaped flames would not be possible otherwise at such a low $S$. Distributed combustion was only possible with leaner mixtures and from $p_g = 0.45$ bar, concluding that the higher the temperature difference between the central region and the annular swirling flow, the more favorable the conditions are for this operation. This is enabled by the phenomenon that two fluids with notably different viscosity do not mix, making MTC combustion possible. Distributed combustion was not observed at the highest equivalence ratio, hence, having the highest share of the atomizing air and the lowest average temperature does not automatically lead to more favorable conditions for distributed combustion. This result indicates that extensive further research is required to understand the criteria of MTC combustion.

Figure 5 shows six images of various stable flame shapes. The top row corresponds to straight flames, presenting the effect on atomizing pressure on the flame structure. Increasing $p_g$ results in decreasing $SMD$, hence, droplet evaporation and mixing with the combustion air occurs faster. Consequently, the flares are disappearing, the flame luminosity is decreasing, and the fuel-air mixture becomes more homogeneous.
The bottom row of Fig. 5 contains a V-shaped flame and two flames in the MTC combustion mode. The former one has a significantly higher luminosity, however, this is far lower than that of the straight flames. In the case of distributed combustion, the presented images are brighter than the other ones captured subsequently, similar to the effect of combustion air dilution [41]. This result ultimately points to the advantage of the MTC combustion mode: the flame occupies a large volume, hence, the heat release can be more homogeneous and less intense. Also, it overcomes the disadvantage of the V-shaped flame: the
heat release in the flame root occurs in a small volume, unavoidably leading inhomogeneous temperature field. Based on the observations, MTC combustion mode qualitatively approaches flameless combustion since it also features delayed ignition and lower volumetric heat release rate [14]. However, the combustion air does not have to be diluted with either an inert or recirculated flue gas, making this novel combustion concept attractive for gas turbine applications. Even more so, as the leaner the mixture, the distributed combustion is easier to achieve and maintain.

All operating points were investigated for at least one minute, while the average was three minutes. The flame of distributed combustion was relatively well-localized, and no blowout or notable acoustic fluctuations were occurred up to \( \phi = 0.57 \). Consequently, it can be stated that MTC combustion mode matches the blowout stability of all the other flame shapes observed presently.

### 3.2 Pollutant emissions

The ultimate measure of a new combustion concept from the viewpoint of regulations is the offered reduction in pollutant emissions. Nevertheless, flame stability, operational flexibility, availability, and potentials in burner tuning for increased efficiency are all critical in industrial technologies. Among the pollutants, NO\(_X\) is of greatest concern since it can be avoided in the case of perfect combustion, which exists only theoretically.

Figure 6a shows the NO\(_X\) emission at all conditions, corrected to 15% O\(_2\) in all the cases. There are two general trends. The first one is that straight flame is characterized by high NO\(_X\) concentration since the released heat is concentrated to a small volume. The other one is the decreasing concentration with decreasing \( \phi \) and increasing \( p_g \), as both dilution and lower overall combustion air temperature decrease the adiabatic flame temperature. In the case of transitory flames, their NO\(_X\) emission is similar to that of the flame shape with higher emission.
Both V-shaped flames and distributed combustion are characterized by low NO\textsubscript{X} compared to the straight flame, however, the latter one features a 53\% reduction on average compared to the former one. More precisely, the NO\textsubscript{X} emission of the V-shaped flame was 12.4 ppm at \(\phi=0.67\), while the average of that of distributed combustion was 4.7 ppm at the same equivalence ratio. At \(\phi=0.57\), these values were 9 ppm and 5 ppm, respectively. Nevertheless, further reduction is possible with inert gas dilution [22], since the adiabatic flame temperature will also decrease.

Figure 6b shows the CO emission, which can be considered marginal at all conditions, compared to current pollutant emission regulations worldwide. Since there is no correlation
between the two emissions, it can be concluded that all three flame shapes are appropriate for complete combustion. The qualitatively outstanding points are local features, hence, there is no obvious fundamental reason for them. Concluding from the pollutant emission data, MTC combustion is a highly favorable concept since it provides a further significant decrease in NO\textsubscript{X} emission compared to the widely used V-shaped flames, while the CO emission remains low.

3.3 Acoustic characteristics

The acoustic spectrum of various setups, which were discussed earlier in Fig. 5, is shown in Fig. 7. The Fourier transformation was performed with a 4096 sample window, using Hamming weighting. The instantaneous results are shown in Fig. 7a, indicating a high variation of the sound pressure level, SPL. It is caused by the temporal fluctuation of the temperature field, affecting the speed of sound, hence, the characteristic frequencies. Consequently, averaging of the 4096 sample windows with a 50% overlap was performed over the 30 s signal, shown in Fig. 7b. There was zero (Z) spectral weighting used. The recorded signal of the selected conditions was free from temporal bursts, hence, spectral bias. Consequently, these results are respective to a smooth operation.

The straight flame was characterized by the highest overall SPL, OASPL, 123.5 dB. That of the background noise, i.e., without combustion, was 108.4 dB, which was originated from the shearing flow of the atomizing free jet and the external cooling jets of the glass windows. The overall noise of the V-shaped flame was 119.5 dB, notably lower than that of the straight flame. The MTC combustion mode resulted in the lowest OASPL values, 114.0 and 113.0 dB at $\phi = 0.76$ and 0.67, respectively. Consequently, this combustion mode offers a significant reduction in the overall acoustic load.
The averaging used in Fig. 7b also helps in identifying the characteristic peaks, summarized in Table 4 with their hypothesized source, assuming rectangular duct and cylinder geometries [51]. This is possible due to the fact that combustion, as a phenomenon, has no characteristic frequency; the peaks are the result of the interaction of the chemical reactions with the flow field and the combustion chamber, CC, geometry. The highest SPL peak is suitable for determining the average speed of sound since it corresponds to the quarter-wave inside the combustion chamber, i.e., the wavelength is $0.5 \times 4 = 2$ m. The highest SPL in the hot cases, H, was uniformly located at 234.4 Hz, implying similar temperature field average, originated from the identical thermal power. Nevertheless, this result is not evident; both the
different heat release patterns and the flame luminosity differences would suggest at least flame shape-dependent frequency peaks, which is observed at higher frequencies. The average speed of sound was hence $a_H = 468.8 \text{ m/s}$. In the case of the background noise (noted as the cold case, C), the peak frequency was 190.4 Hz, meaning $a_C = 388.8 \text{ m/s}$. Even though $a_H$ and $a_C$ values alone seem low in the field of combustion, it should be noted that the central cold jet inside the mixing tube, MT, significantly affects the temperature field, so the propagation of the sound waves.

| Frequency [Hz] | Case | Wave | Source                  |
|----------------|------|------|-------------------------|
| 190.4          | C    | 1/4  | CC length: 0.5 m        |
| 234.4          | H    | 1/4  | CC length: 0.5 m        |
| 566.4          | C    | 3/4  | CC length: 0.5 m        |
| 683.6          | C    | 1/4  | MT length: 0.116 m      |
| 835.0          | H    | 1/2  | CC width: 0.3 m         |
| 922.9          | H    | 5/4  | CC length: 0.5 m        |
| 1001           | H    | 1/4  | MT length: 0.116 m      |
| 1425           | H    | 1    | CC width: 0.3 m         |
| 3877           | H    | 2    | CC width: 0.3 m         |

Local frequency peaks apart of the listed ones in Table 4 are most probably originated from various flow structures. Revealing these sources requires computational fluid dynamic simulations, which is a potential direction for future research. The time-averaged SPL of MTC combustion in Fig. 7b contains primarily well-localized peaks related to the combustion chamber and burner geometry. Hence, it can be concluded that thermoacoustic instabilities are less likely to endanger stable operation near the lean blowout limitation than that in the case of V-shaped flames. This favorable characteristic is similar to that of flameless combustion [15].

4 Conclusions

The present paper introduced a novel combustion concept, Mixture Temperature-Controlled (MTC) combustion. It enables about -50% NOX emission compared to the widespread V-shaped flames of lean premixed swirl combustion, while ambient air can be used
as the oxidizer. Currently, MTC combustion was presented for diesel oil, however, it is hypothesized that other liquid and gaseous fuels may also work. Since this is the introductory paper of MTC combustion, numerous further investigations, e.g., laser measurements and numerical simulation, are required to understand this novel concept in detail. Since MTC combustion was demonstrated in a turbulent swirl burner at 13.3 kW thermal power without a notable practical issue, this concept bears the potential of the next advancement in low emission industrial and gas turbine burners. The key findings were the following.

- The essence of the MTC combustion concept is the cold central air inlet, delaying ignition. Consequently, combustion occurs downstream of the burner, occupying a large volume in the combustion chamber. Hence, this burner configuration is free from flashback and fuel nozzle coking by design.

- The distributed heat release leads to low NOₓ emission, while CO emission remains low. Hence, the NOₓ advantage is not a result of a compromise.

- Flame luminosity is significantly lower than that of V-shaped and straight flames. Hence, optical sensing and control of the process bear further challenges.

- The overall sound pressure level of MTC combustion was 6 dB lower than that of a V-shaped flame, meaning a notably reduced acoustic load on both the device and affected personnel.

- The time-averaged averaged acoustic spectrum contains well-localized peaks, which are related to the eigenmodes of the combustion system geometry. Consequently, its tendency to thermoacoustic instabilities is hypothesized to be significantly lower than that of V-shaped flames and similar to that of flameless combustion.
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Conflict of interest

The author declares that there is no conflict of interest.

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