MECHANICAL ENGINEERING | RESEARCH ARTICLE

Performance and emission study of low HCNG fuel blend in SI engine with fixed ignition timing

Vivek Pandey1*, Suresh Guluwadi2 and Gezahegn Habtam Tafesse2

Abstract: Natural gas (NG) has many advantages of clean fuels and can replace gasoline in spark ignition (SI) engines. However, it comes with disadvantages such as low energy density. Hydrogen blended NG can improve the fuel characteristics. Ignition timing also plays an important role in engine performance, especially at lean limits. Experiments were conducted for performance and emissions with hydrogen-compressed natural gas (HCNG) blends in a spark ignition (SI) engine with fixed ignition timing. HCNG blends with 0–15% hydrogen were tested with maximum brake torque (MBT) spark timing 30° before top dead center (BTDC) ignition. Significant reduction of carbon monoxide (CO), carbon dioxide (CO2) and hydrocarbons (HC) along with improved break thermal efficiency was observed with higher hydrogen fraction. There was no significant benefit in indicated thermal efficiency or NOx reduction by using MBT ignition. Hence, the expensive retrofitting of using variable spark timing apparatus can be avoided for on-road vehicles.

Subjects: Mechanical Engineering; Renewable Energy; Energy & Fuels; Combustion; Renewable Energy; Energy efficiency; Fossil and nuclear energy

Keywords: Alternative fuels; HCNG; emissions; hydrocarbons; nitrogen oxides; ignition timing

ABOUT THE AUTHOR
Dr. Vivek Pandey is involved in research activities that encompass hydrogen as a fuel for the future including hydrogen storage. He is also working on the effect of nanoparticles such as cerium oxide, titanium oxide, and carbon nanotubes (CNT) and their effects as additives on the combustion performance of low carbon footprint fuels including but not limited to biodiesels. He is also involved in augmenting the performance of solar energy devices through spectral-splitting technique. Another aspect of his research is on Computational Fluid Dynamics (CFD) of liquid-liquid two-phase flows in microchannels, and studying heat transfer characteristics. Dr. Vivek Pandey has also published research in Inverse Heat Transfer problems in engineering. The research presented in the paper deals with hydrogen as a clean burning fuel for on-road vehicles and its effects on the engine performance.

PUBLIC INTEREST STATEMENT
The investigation pertains to engines, such as in cars and other on-road vehicles, and the use of such engines with fuels such as hydrogen and natural gas. The use of hydrogen and natural gas is becoming an imperative because of decline in traditional, non-renewable, fossil fuels such as petrol and diesel. There is an added concern related to the emissions from the use of such fuels due to the fact that they are toxic for the environment and also to humans. When on road vehicles that use petrol or diesel are to be replaced with hydrogen and/or natural gas, engine modifications may be required. The results bring out the facts that expensive engine modifications are not needed when substituting hydrogen/natural gas with petrol/diesel in vehicles that are already in use or that would be produced later.
1. Introduction

It has become a widely accepted fact that fossil fuels have entered an interminable decline phase, and would be phased out from the present-day transport system in the near future. The use of natural gas (NG) as compressed natural gas (CNG) is preferred over gasoline or diesel in internal combustion (IC) engines due to various reasons. Compared to gasoline, CNG has lower carbon to hydrogen ratio; consequently, there is cleaner combustion with lower carbon emissions. Also, CNG has better anti-knock properties, is widely available, is highly compatible with IC engines and has low operational cost. Furthermore, it can be produced from both renewable and non-renewable resources (Sagar & Agarwal, 2018). However, there are several disadvantages with CNG such as lower energy density, comparatively smaller flame speed and narrower flammability limits (Niculae et al., 2020). Increase of CNG fraction in fuel blend lowers the volumetric efficiency and the flame speed, thereby reducing engine power, maximum cycle pressure and rate of heat release (Amirante et al., 2017; Singh et al., 2019). Therefore, more recent researches point to direct injection of natural gas, for example, Liu et al. (2021) extended the lean burn range of SI engine by using gasoline port injection combined with natural gas direct injection and showed stable lean combustion with 40% excess air. The coefficient of variance of indicated mean effective pressure (COVIMEP) was below 1.5. However, even with the new strategy employed, there was a 5–10% decrease in indicated mean effective pressure with CNG-gasoline blends.

The slower flame speed of natural gas when compared to hydrogen increases the ignition lag and also causes slower combustion; therefore coefficient of variance of indicated mean effective pressure (COVIMEP) is higher with CNG percentage. In order to overcome the disadvantages with CNG, hydrogen addition to CNG is considered. Hydrogen can be produced from renewable resources and has high energy density. It has wide flammability range and can be used for lean combustion in IC engines at wide open throttle (WOT) without significant variations in IMEP. This is because hydrogen has a laminar flame speed about eight times greater than natural gas (Hu et al., 2009; Huang et al., 2006; Ilbasa et al., 2006) This reduces the combustion duration and accounts for less cyclic variations, especially at extended lean operation. A higher efficiency due to hydrogen addition occurs due to an almost constant pressure cycle operation approaching the air-standard cycle. Another significant advantage is that of zero carbon emissions with hydrogen.

However, a major drawback is the energy intensive and expensive storage of hydrogen gas in compressed form. Furthermore, engines fuelled with pure hydrogen have higher peak combustion temperature and therefore give off higher thermal NOx at comparable operating conditions (Gong et al., 2019; Kosmadakis et al., 2016). In order to overcome the mutually exclusive limitations of CNG and hydrogen, a combination of these called as Hythane or HCNG is being increasingly researched. A smaller hydrogen percentage (usually up to 30%) is blended with CNG to increase the combustion limits and to improve the combustion and emission characteristics. Hereafter, the percentage of hydrogen in the Hydrogen-CNG blend is mentioned as xHCNG, where x is the volume percentage of hydrogen.

Investigations on HCNG fuelled spark ignition (SI) and compression ignition (CI) engines are available in the literature. The research related to HCNG fuelled SI engines are presented, due to more relevance to this article. Kosmadakis et al. (2021) evaluated the cyclic variation using coefficient of variance for indicated mean effective pressure (COVIMEP) in SI engines fuelled with up to 50HCNG blends. Smolenskaya and Smolenskii (2019) evaluated exhaust gas toxicity from a single-cylinder and a multi-cylinder spark ignition (SI) engine from addition of hydrogen in small quantities (5–15 %), to gasoline and CNG. The emissions were evaluated at engine idling. Prasad and Agarwal (2021) showed that replacing spark ignition with laser ignition in 40HCNG mixture extended the lean burning map with significantly reduced regulated emissions. Lee et al. (2014) studied emissions from a multi-cylinder, SI engine, fuelled with 30HCNG mixture. A 25% reduction
in fuel consumption as compared to pure CNG was noted, along with reduction in HC and NOx. Lower NOx for HCNG was due to less fuel consumed for stable idling condition, due to enhanced lean-burning limit for hydrogen. High specific fuel consumption at idle and lean condition for conventional gasoline fuelled SI engines warrants the need for investigating idle mode. Ji and Wang (2013) investigated a four-cylinder SI engine fuelled with hydrogen. Hydrogen was port injected and the injection duration was electronically controlled for varying \( \lambda \). They found an 18\% decrease in hydrogen fuel consumption compared to gasoline, with a 53\% increase in \( \lambda \) at idle and lean operation. Combustion stability was satisfactory with the maximum COV\(_{\text{IMEP}}\) at 1.1\%, which is far lower than the 10\% limit for lean combustion (Ma & Wang, 2008).

1.1. Port injection SI engines

Port injection (PI) and direct injection (DI) are the two broad categories of fuel injection in IC engines. Due to low density and viscosity as compared to gasoline, gaseous fuels require a redesign of the injector for the case of DI. DI allows a limited range of air-fuel ratios (AFR) because of limitations in injector design attributed to vastly different properties of CNG and higher injection pressure compared to PI systems (Erfan et al., 2015; Song et al., 2017). PI permits wider AFR and also reduces the cost of retrofitting a gasoline vehicle for CNG/HCNG, or redesign of the injector which obviates expensive manufacturing and design (Baratta et al., 2021). PI also accounts for highly homogeneous mixture. Moreover, DI systems are found to have lower brake thermal efficiency (BTE) at high loads, and higher hydrocarbon emissions at all load conditions. The reason is attributed to inferior charge homogenization due to DI (Moon, 2018). Therefore, PI is the dominant injection system in today’s CNG vehicle-market.

With increasing fraction of hydrogen in HCNG, ignition timing may be retarded with respect to maximum break torque (MBT) timing; this reduces NOx but also reduces BTE. Some investigators make changes to ignition timing for maximum brake torque, but that comes with the penalty of high NOx (Moreno et al., 2012). Therefore, it is important to assess the effects of fuel composition with fixed ignition timings, on SI engine performance and emission characteristics.

Some of the research relating to fixed ignition timing in SI engines is presented and it is shown that there are relatively fewer investigations with fixed ignition timing for port injection HCNG-fuelled SI vehicle engines.

1.2. Previous work on HCNG fuelled SI engines with fixed ignition timings

Akansu et al. (2007) experimented with 0–30HCNG mixtures in a four-stroke, four-cylinder SI engine at constant load, 2000 RPM and ignition timing of 30\(^\circ\) before top dead center (BTDC). They used excess air ratios (\( \lambda \)) from 0.8 to 1.6. They used port injection with a gas mixer. NOx and BTE were found to increase, whereas CO, CO\(_2\) and total hydrocarbons (THC) decreased, with an increase in H\(_2\) percentage. Ma et al. (2008) conducted experiments with 0–50HCNG in SI for assessing engine and emissions performance at idle. CH\(_4\) ppm in exhaust was found to decrease with increase in H\(_2\) percentage, at 20\(^\circ\) BTDC ignition. CO and CH\(_4\) decreased with increase in excess air ratio. H\(_2\) addition decreased the cyclic variability in terms of COV\(_{\text{IMEP}}\). Also, the lean burn stability at idle was superior with H\(_2\) percentage increase. However, they did not assess performance effects at fixed injection timing. Zhao et al. (2013) assessed a lean-burn SI engine performance with 55HCNG, at 1200 RPM and 16\(^\circ\) crank angle (CA). They studied two compression ratios (CR); 10:1, 12:1. They found that higher CR may serve to shorten the flame development period and has an acceptable COV\(_{\text{IMEP}}\) when \( \lambda \) is less than 1.8, but for higher values of \( \lambda \), it showed no benefit for lean operation. Significant difference in NOx for the two CR's was observed only for \( \lambda \leq 1.3 \).
Ma et al. (2007) investigated the effect of 0–50HCNG on lean burning characteristics of an SI engine at 30° BTDC ignition and MBT. Some relevant results for 30HCNG are summarized. For excess air ratio of 1.8, thermal efficiency was 37.5% and 37% for 30° BTDC ignition and MBT, respectively. Similarly, for excess air ratio of 2.0, thermal efficiency was 33% and 32.5% for 30° BTDC ignition and MBT, respectively. For \( \lambda \) less than 1.8, there was no statistically significant difference in indicated thermal efficiency. In addition, the maximum difference in engine efficiency while comparing 30° BTDC and MBT was seen for 30HCNG, the difference decreased with lower \( H_2 \) percentage. Therefore, it may be safely concluded that up to 30HCNG, a change in spark timing for attaining maximum brake torque is not needed. Fixed spark timing is sufficient for most ranges of operation. Furthermore, the extension of the lean burn limit to \( \lambda = 2 \) was demonstrated with 30HCNG. An improvement in engine thermal efficiency and NOx reduction potential with increasing \( H_2 \) percentage was demonstrated. As a result, we can use constant spark timing for HCNG fuelled SI engines along with exhaust gas recirculation (EGR) for reducing NOx. Wang et al. (2008) studied cyclic variations of peak cylinder pressure in a port injection SI engine fuelled with up to 40HCNG mixtures. For 40HCNG, they observed an extension of the lean limit to \( \lambda = 2.0 \), with \( \text{COV}_{\text{IMEP}} \) under 10%. The lean limit was found to increase with \( H_2 \) percentage. They hypothesized that increase in \( H_2 \) percentage would weaken the effect of turbulence induced fluctuations, thereby decreasing cyclic variations. This was due to the fact that the laminar and turbulent flame speeds of hydrogen-air mixtures have the same order of magnitude compared to an NG-air fuel mixture, whose turbulent flame speed is an order of magnitude higher. The heat release rate for laminar and turbulent states for large percentage hydrogen would be similar.

As evident from the previous research, investigations with port fuelled SI engines at fixed injection timing are relatively few in number, compared to the vast body of literature available for HCNG fuelled engines. Therefore, the aim of this article is to gain information on engine performance with fixed injection timing. Another motivation is to study the effects of low hydrogen additions. This paper is an experimental investigation of an SI engine test rig at fixed ignition timing of 30° BTDC. The engine was fuelled by 5HCNG and 15HCNG blends in order to evaluate the effects of hydrogen addition on exhaust emissions and engine efficiency. A dedicated pressure regulator was employed for HCNG to control the flow rate of the gas. Hydrogen and CNG were channeled through a mixer prior to port injection. A flame trap was installed prior to entering the inlet manifold. Constant ignition timing was maintained for all tests.
2. Experimental apparatus

Experimental investigation was carried out on a Maruti Suzuki Omni van engine (Figure 1), which was a spark ignited, vertical, three-cylinder and water-cooled engine. The engine details are shown in Table 1. The schematic of the test rig is shown in Figure 2. Table 2 shows relevant properties of NG and Hydrogen.

The engine was coupled to a hydraulic dynamometer, and a load cell was provided with torque arm of the dynamometer to measure the engine torque. The test rig was equipped with J-type thermocouples at the intake manifold, in the cooling water system and in the exhaust pipe. The HCNG mixture used in the test was premixed outside the intake manifold. The consumption of HCNG mixture by the engine was measured using a calibrated gas flow meter.

2.1. Description of exhaust gas analyzer

The Gas Analyzer used in the investigation was a standard INDUS PEA-205 5-Gas Analyzer, in order to monitor CO, CO$_2$, THC, O$_2$ and NOx emission. CO, CO$_2$ and THC were measured by Non-Dispersive
Infrared Ray method while O₂ and NOx were measured by Electro-chemical sensors with an absolute accuracy of ±0.02%.

3. Experimental procedure
Initially the engine was run on gasoline for about half an hour for engine warm up. Subsequently, experiments were conducted with pure CNG (0HCNG), 5HCNG, 10HCNG and 15HCNG blends to investigate and compare engine performance and exhaust emissions. The tests were conducted at constant speed, constant ignition timing with varying loads for both blends. Tests were also conducted to see the effect of varying the excess air ratio (λ) on indicated thermal efficiency and NOx emissions. The tests were repeated thrice for every load in order to reduce the error. The output torque was measured by a dynamometer; hence, the brake thermal efficiency for the engine could be calculated. The exhaust emissions, namely CO, CO₂, THC, NOx and also oxygen content in the engine exhaust were measured. A flame trap was installed in order to prevent back fire due to hydrogen in the fuel.

4. Results and discussions

4.1. Efficiency and load characteristics
Figure 3 shows the variation of Brake Thermal Efficiency (BTE) with Load. The trend shows an almost uniform increase with respect to load for all fuel compositions. At zero load, all fuel compositions indicate the same BTE but at full load there is a 50% increase of BTE for 15HCNG compared to 0HCNG mixture. This is due to increase in hydrogen concentration, higher mixture flame velocity and hence faster mixture burn rate and shorter combustion duration, along with

| Property                          | Hydrogen | Natural gas |
|-----------------------------------|----------|-------------|
| Density at (kg/m³)                | 0.082    | 0.754       |
| Stoichiometric fuel air ratio (by volume) | 0.420    | 0.106       |
| Volumetric lower heating value (MJ/m³) | 10.22    | 32.97       |
| Laminar flame speed (m/s)         | 2.9      | 0.38        |
| Minimum ignition energy (mJ)      | 0.02     | 0.28        |
| Lean flammability limit equivalence ratio | 0.1      | 0.5         |
| Quenching distance (mm)           | 0.6      | 1.9         |
a higher combustion cycle peak pressure and temperature. Higher peak temperature would account for higher thermal efficiency from first law of thermodynamics. In addition, the shorter combustion duration would account for lower heat transfer loss. It is also observed that even a small amount of hydrogen (5%) provides a substantial advantage to BTE as compared to pure CNG. However, with similar increments of hydrogen, the advantage is comparatively less.

Figure 4 shows the variation of specific fuel consumption (SFC) with load. All mixture compositions exhibit a characteristic “hockey-stick” trend. SFC at no load is considerably reduced by up to 80%, for 15HCNG compared to 0HCNG. It is evident that increase in hydrogen percentage in the HCNG blend reduces the fuel consumption. The reason can be attributed to heating value (MJ/kg) of hydrogen, which is 37 times that of NG. Another observation would be that there is on average, a 60% decline in SFC between 5% load to 20% load, and thereafter there is a more gradual decline. The fall in SFC is therefore more pronounced for low loads. However, even at high load, 50% reduction in SFC can be noticed.

4.2. Exhaust gas analysis
Carbon monoxide (CO) and total hydrocarbon (THC) emissions in the exhaust are indicative of incomplete combustion. Figure 5 shows the variation of CO as percentage of air with load. The graphs show a linearly decreasing trend with load. The 15HCNG mixture has lowest CO emission. There is a substantial decrease in CO emission as the hydrogen percentage increases in the HCNG
blend. For the 15HCNG mixture, there is an almost flat curve, indicating little change in CO emission with load. The other mixtures have a more pronounced downward CO emission trend with load. In general, hydrogen addition increases the combustion temperature due to faster heat release rate and higher heating value in MJ/kg. Higher temperature assists the completion of combustion reaction of CO₂ formation.

Figure 6 shows the decreasing trend of carbon dioxide (CO₂) percentage with load. There is a significant decrease of CO₂ emissions as the hydrogen percentage increases in the HCNG blend. Up to 50% reduction in CO₂ is observed between 0HCNG and 15HCNG at 25% load.

Figure 7 shows the variation of THC (ppm) with load. THC emissions decrease with increase in load and also with increasing hydrogen percentage in fuel. The curves show a trend similar to CO emissions (Figure 6). The chemical pathway of oxidation of CO and THC are similar. Therefore, at higher loads, higher cylinder temperatures are attained which aid in the complete oxidation of THC and CO. A further observation is a marked decrease (of up to 50%) in THC emissions with an increase in hydrogen percentage. This can be attributed to the contribution of hydrogen which accounts for quicker and more complete combustion. This again is due to the higher flame speed of hydrogen, and also due to higher peak cylinder temperatures on account of shorter burn duration due to hydrogen content. Also, since hydrogen is a zero carbon fuel, increase in hydrogen
fraction brings down the overall carbon to hydrogen ratio of the mixture, which is a contributing factor to the reduction in THC and CO emissions.

Figure 8 shows the variation of excess Oxygen ($O_2$) with load, in the engine exhaust. It is seen that the $O_2$ content in exhaust increases with increase in hydrogen fraction. On the whole, there is no significant change in exhaust $O_2$ with increasing load.

Figure 9 shows the variation of NOx ppm with load. The formation of NOx is mostly due to presence of nitrogen in air and to some extent in fuel. There is no fuel bound nitrogen in NG and H$_2$. Furthermore, major fraction of nitrogen-oxides are formed at high flame temperature and are highly correlated to the adiabatic flame temperature. Major contributor to NOx is thermal NOx, which is formed within high temperature zone of the diffusion flame. The trend is similar for all mixtures. NOx content in the exhaust clearly increases with load. In addition, NOx increases with increasing hydrogen fraction due to higher peak combustion temperatures, which again is due to quicker combustion of hydrogen due to its higher flame speed.

There are various ways for NOx reduction, two of which are exhaust gas recirculation (EGR), and varying the ignition timing so that the peak cylinder temperature may be lowered. Although the second NOx reduction strategy is employed by some researchers, it may not be more effective compared to EGR since it may be detrimental to engine performance. So, there is essentially a trade-off between engine performance and NOx, when going for variable ignition. This
disadvantage may be overcome by the use of hydrogen and there is a need to understand how hydrogen addition affects engine performance and NOx at different spark timings.

Variable spark timing is commonly implemented for the attaining maximum brake torque (MBT), and is also called as MBT ignition timing or MBT. MBT is found by a spark sweep on the engine, and some calculations. Its implementation is more commonly an open-loop process. The closed loop implementation is more involved and expensive since it would entail engine modifications which would be an issue for “on-road” vehicles. The open-loop implementation is also time consuming. Both methods depend on the fuel type and engine characteristics. Therefore it would be helpful to understand whether to implement variable ignition for maximum torque or for low NOx. The following section elucidates the results of MBT spark timing and fixed ignition timing (30° BTDC) on engine performance and NOx emissions.

4.3. Thermal efficiency and NOx variation with excess air ratio
The results of indicated thermal efficiency and NOx variation with excess air ratio (\(\lambda\)) are reported for MBT and 30° BTDC ignition timing. The limits of lean-combustion become apparent for variation
with excess air ratio. Figures 10 and 11 show indicated thermal efficiency (ITE) with \( \lambda \) for different hydrogen percentage in fuel. Figure 10 is for MBT ignition and Figure 11 is for 30° BTDC ignition. The black dashed line indicates 32.5 % efficiency. It is apparent from the two figures that beyond a certain value of \( \lambda \), ITE drops below the acceptable 32.5% value. We see the excess air ratios for each fuel composition at the 32.5% line. Excess air ratio at this value sets a limit on stable or efficient engine operation or engine lean burn limits. Above this value, we do not observe a significant variation for the two ignition timings. For 0HCNG, 2% increase in \( \lambda \) is seen for MBT timing, compared to ignition at 30° BTDC. Similarly, for 15HCNG, 1.6% increase in \( \lambda \) is seen for MBT timing, compared to ignition at 30° BTDC. Therefore, excess air ratio, that determine lean combustion limits are not much affected by change in spark timing.

Figures 12 and 13 show bar plots of brake specific NOx for 30° BTDC spark timing and MBT timing, respectively. Trends for all the fuel compositions are similar. NOx peaks at near \( \lambda = 1.2 \), which is 20% lean mixture, and thereafter declines to around one for \( \lambda = 1.8 \) and more. This is common observation for both the plots. Beyond excess air ratio of 1.6, NOx declines sufficiently so that it is not affected by change in spark timing. Therefore, change in spark timing does not give any
significant advantage to push the lean combustion limits. However, NOx values are higher for 30° BTDC spark timing, when excess air ratio is less than 1.6.

5. Conclusions
Experimental investigation on an SI engine fuelled by HCNG blends was carried out to investigate the effects of small H\textsubscript{2} fractions (0–15\%) and fixed spark ignition timings on performance and emission characteristics. By observing the exhaust performance graphs, we notice that 15HCNG blend accounts for the least amount of CO\textsubscript{2} emission, which is a major Green House Gas (GHG), while SHCN and pure CNG blend accounts for higher emissions. It can also be noted that 15HCNG blend results in the lowest emissions for CO and HC. Therefore, the use of up to 15HCNG can provide a relatively clean exhaust with higher thermal efficiency. NOx ppm in exhaust increases with load in case of 15HCNG compared to SHCN and pure CNG.

It is shown that HCNG mixture has a stable, lean combustion with excess air ratio (λ) up to 1.9 compared to 1.75 for pure CNG. NOx emissions with increasing hydrogen fraction were higher whereas hydrocarbons were less, compared to pure CNG; the difference in emissions between the two fuels diminished with increasing λ.

Indicated thermal efficiency is not significantly affected by change in spark timing and is well within an acceptable limit of 32.5\% for quite lean mixtures also. However, NOx ratings are somewhat affected by variation in spark timing, below excess air ratio of 1.6. Lower NOx for MBT spark timing conditions compared to constant spark timing of 30° is observed. However, even this becomes insignificant at lean combustion limits. EGR can be employed for NOx reduction for λ less than 1.6. For vehicles on road, it is more suitable to keep the spark timing unchanged and adopt EGR, retrofitting an SI engine vehicle for variable spark timing is a costly and unnecessary affair.

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Author details
Vivek Pandey\textsuperscript{1}
E-mail: vivek.fw@gmail.com
ORCID ID: http://orcid.org/0000-0001-9570-7751
Suresh Guluwadi\textsuperscript{2}
E-mail: suresh.guluwadi@gmail.com
Gezahegn Habtamu Tafesse\textsuperscript{1}
E-mail: vivek.pandey@astu.edu.et
\textsuperscript{1}Department of Thermal and Aerospace Engineering, SoMCME, Adama Science and Technology University, Adama, Ethiopia.
\textsuperscript{2}Department of Thermal and Aerospace Engineering, School of Mechanical, Chemical and Materials Engineering, Adama Science and Technology University, Adama, Ethiopia.

Disclosure statement
The authors of this work declare that they do not have any competing interest, also known as a “conflict of interest”, arising from our or our employer having any financial, commercial, legal, or professional relationship with other organizations, or with the people working with them, that could influence the research in this article.

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