



Honda R&D

Technical Review

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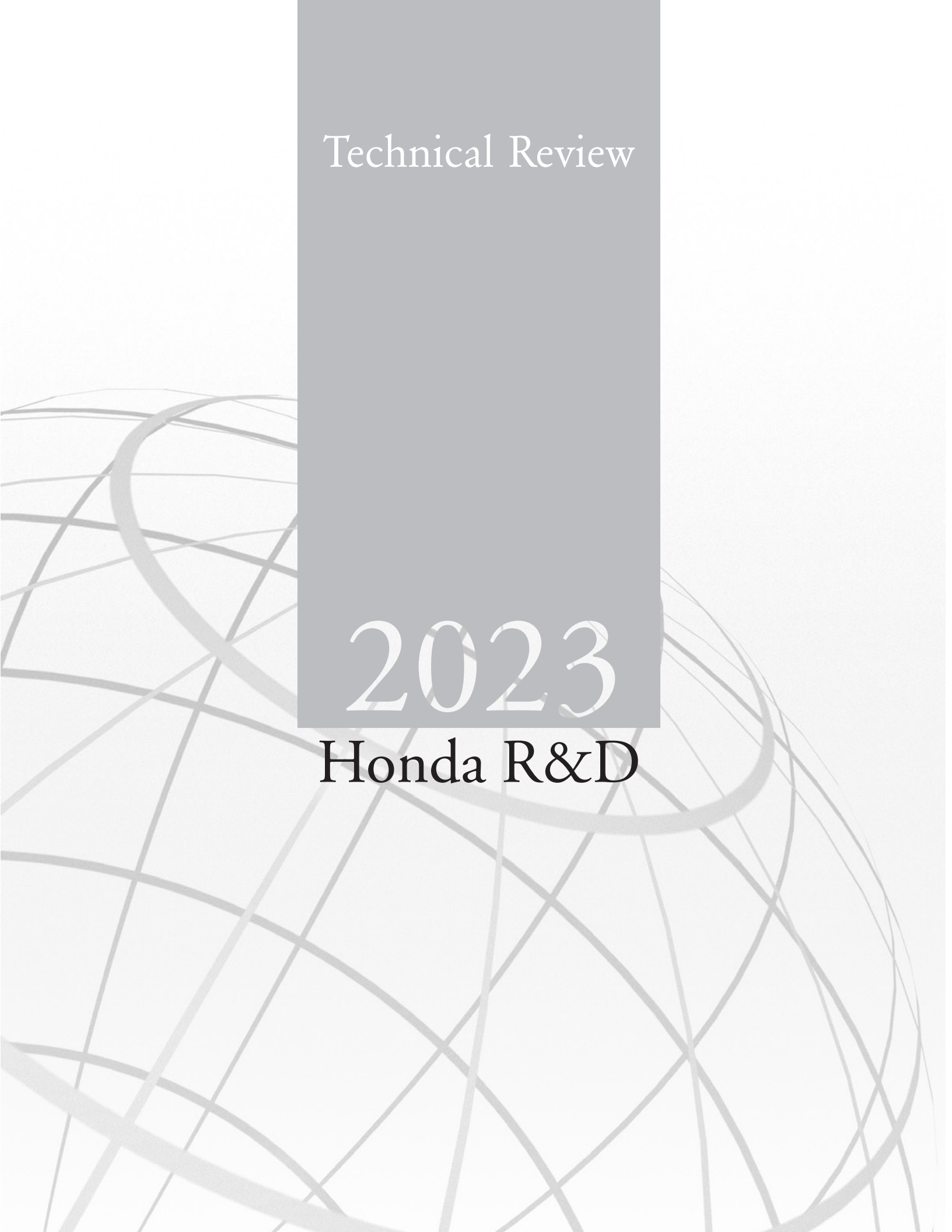


Comment on the Cover

For this cover, I wanted to give an impression of dramatic progress. As a motif for dramatic progress at Honda, there is the Nike of Samothrace. This graphic was created using lithographic printing to highlight out-of-sync printing, uneven printing, gradations, and so on, for your enjoyment. These effects were achieved by superimposing the hand-drawn image of Nike and the photographic image of Nike in two colors. I hope that this cover will convey the feeling of enthusiastic determination in going back to the state of mind one has at the beginning and starting work anew.



Design Center
Kosuke YONETA

The background features a light gray wireframe sphere on the left side, composed of intersecting curved lines. A solid gray vertical bar runs down the center of the page, serving as a backdrop for the title text.

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The Blossoming of Leonardo da Vinci, Engineer

Hidehiro Ikegami*

1. Introduction

“I am well aware that, because I am not a scholar, some presumptuous people will think that they can belittle me as an uneducated person. Bunch of ignoramuses themselves!”
—From the Codex Atlanticus¹

In this statement by Leonardo da Vinci in his Codex Atlanticus, “uneducated person” or *omo senza lettere* (*uomo senza lettere* in modern Italian) is a reference to his failure to understand Latin, which was the common language of the intellectual class of the time. Although many writers, including Dante and Petrarch, composed poetry and prose in the vernacular language (Italian), to qualify as a true man of culture, one had to write in Latin. The rather defensive quote from Leonardo above reveals his inferiority complex vis-a-vis the Latin language, which tormented him throughout his life.

Leonardo descended from a line of notaries, which was considered an intellectual class at the time, but was born out of wedlock (a bastard child) because his parents did not marry, probably because of their different class statuses. Ironically, the notaries’ union at the time did not admit illegitimate children, so it was clear from an early age that he would not inherit the family business. As a result, Leonardo grew up with an emotionally distant father, without access to Latin, the abacus, and other elements of a primary education that the sons of the wealthy at the time received.

Three times in Leonardo’s manuscripts, he makes note of the books currently in his possession. The first such mention is in the Codex Trivulzianus, where he lists five book titles: Donato, Lapidario, Plinjo, Abacho, and Morgante². The first of these, Donato, refers to a Latin grammar book by Donatus. It is interesting that Pliny’s *Natural History* and Luigi Pulci’s epic poem *Morgante* are accompanied by a grammar book and a book on calculation with the *abaco* (a kind of abacus). It is likely that Leonardo, who was in his mid-30s, taught himself out of necessity.

As this shows, his image as an “all-powerful genius” who can do anything without difficulty is not in line with reality. This paper will consider how Leonardo the engineer blossomed from such an inauspicious starting point.

2. Leonardo’s learning

Looking at Leonardo’s next two mentions of the books currently in his possession, you can see his collection expanding at an accelerating rate. The Codex Atlanticus lists 40 volumes, and the final mention in the Codex Madrid lists 116 volumes. The latter dates to around 1503–05, when Leonardo was already over 50 years old. Some classics such as Ovid, Pliny, Aristotle, and Augustine are listed, but most of the titles were written in the vernacular by contemporary writers such as Palmieri, Burchiello, and Alberti³. The scarcity of Latin texts can be attributed to Leonardo’s difficulties with the language mentioned above; even his editions of *Aesop’s Fables* and similar works were translations into the vernacular.

The list of books was apparently part of a list of belongings Leonardo left behind on his travels, so it is possible that there were other books that he brought with him. In any case, it can be seen that Leonardo’s collection of books expanded dramatically over the course of about 20 years, despite the fact that today’s printing technology did not exist, and books were basically copied by hand. Borrowing a manuscript and copying it yourself took a lot of time, and buying a completed manuscript was very expensive. Leonardo’s final cited inventory of 116 volumes represents a staggering number of books for an individual around the age of 50 to possess at the time.

While it is clear that he is a voracious reader, another of Leonardo’s qualities is his humility. Hearing that a famous bridge built in the time of ancient Rome still stands in Verona, he actually goes to the city and sketches it. In his manuscripts, there are several entries stating, in effect, “So-and-so is knowledgeable about this, so I must go ask him.” One example is a note to ask someone named “Gian di Paris” about “coloring methods using the secco technique (*il modo de colorire a secco*)”⁴. This was the French court painter Jean Perréal, also known as Jean De Paris, who actually came to Milan while in the service of Louis XII. This reminds us that Leonardo, who had just painted *The Last Supper* without the use of the fresco method, was always on the lookout for the best techniques. Despite having trained in Florence, the center of the art world, he humbly shows his willingness to learn from French painters,

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¹ Leonardo da Vinci, *il Codice Atlantico*, Biblioteca Ambrosiana, Milano, f. 327v.

² Leonardo da Vinci, *Ms. Trivulziano M 39*, Castello Sforzesco, Biblioteca Trivulziana, Milano, f. 2r.

³ Leonardo da Vinci, *il Codice Madrid*, Biblioteca Nacional de Madrid, Madrid, II, f. 2v.

⁴ Leonardo da Vinci, *il Codice Atlantico*, cit., f. 669r.

driven by a desire to learn more about oil techniques from the north.

In this way, he learned a great deal from his contemporaries, and indeed, there were apparently many learned people who had direct contact with him in Milan and Florence. Among them were the geographer Toscanelli; the navigator Amerigo Vespucci (namesake of America); Argyropoulos, a Greek lecturer in Florence; the architects Luca Fancelli and Giovanni Antonio Amadeo; the artillery engineer Giannino; the physician and mathematician Giovanni Marliani; university professor Fazio Cardano; and the intellectual Baldassare Castiglione. On the other hand, not all scholars and intellectuals are worthy of the name, and Leonardo wrote unfavorably about astrologers such as Ambrogio da Varese, an important figure in the Milan court.

3. Apprenticeship in a workshop

Once it is clear that young Leonardo will not take over the family business, his father, Ser Piero, sends him to apprentice in the workshop of an acquaintance. Andrea di Michele de' Cioni, aka Andrea del Verrocchio, born around 1435, was about 17 years older than Leonardo. He began his career as a goldsmith and established a workshop that was one of the largest in Florence at the time, rivaled only by the workshop of the Pollaiuolo brothers, who were about the same age. While Leonardo was at the workshop, Perugino (Pietro Vannucci), who would eventually become Raphael's mentor, was also there, and for a time Botticelli also came to the workshop to collaborate with the artists there. In short, Verrocchio was not only an excellent artist but also a good educator,



Fig. 1 Agostino Veneziano (Agostino de' Musi), "The Academy of Baccio Bandinelli," 1531, Metropolitan Museum of Art, New York.

and it was Leonardo's great fortune to become his pupil.

However, what we imagine today as an "artist's workshop" did not exist before the High Renaissance. To begin with, there was not yet a profession called "artist." Of course, there were people who painted pictures, carved statues, built houses, or practiced crafts such as goldsmithing and carpet weaving. However, their occupation was no different from that of bakers and leather goods manufacturers at the time, in the sense that creativity and individuality were rarely required.

An apprenticeship in a workshop takes years. At first, you do only small, simple drawings, and the rest of the day is mostly spent cleaning and running errands. Next, you learn how to make materials such as pigments, plaster, and stucco. The master would make several preliminary sketches and small models for each piece he completed, and the apprentices would use these as textbooks and learn by copying them. Unlike a modern art school, there were no instructors giving careful instruction; almost all of the education was based on the dutiful imitation of the professional's work.

After a few years, the apprentice is gradually given the responsibility of underpainting his master's works. By the time they had chosen their own subject matter and practiced drawing slightly larger sketches, a clear pecking order of who was better among the apprentices in the workshop was established, and their areas of expertise were solidified. At that point you would be entrusted with parts of the master's work as an assistant. For example, if you were good at musical instruments, you would be entrusted with drawing musical instruments. Similar to the roles of today's manga artists and their assistants, it was common for the master to draw the main characters and for the assistants to draw the

scenery. Verrocchio's workshop produced many excellent pupils, partly because he divided the work between his pupils and collaborators appropriately.

A little after Leonardo's time, an engraving was made depicting the sculptor Baccio Bandinelli (who was also a rival of Michelangelo) giving art lessons to his disciples in his workshop, which he called the *accademia* (engraving by Agostino Veneziano, Fig. 1). The word *accademia* refers to the Renaissance attempt to revive Plato's school in ancient Greece (Academia Platonica), the prototype for today's academies. In fact, Bandinelli started a private *accademia* in Rome in the 1530s and then in Florence in the latter half of the 1540s. The print depicts a group of young people gathered in a candlelit room, sketching around Bandinelli,



Fig. 2 *Cartelle of the Accademia Vinciana*, based on the design of Leonardo da Vinci, a noted Milanese printmaker, three types, 1497–1500, copperplate engraving, Milan, Ambrosiana Art Gallery.

who is holding a small statue of Venus. Small statues and vases in various poses are arranged on the shelves in the room, indicating that opportunities for this kind of practice were provided daily. Perhaps a similar kind of practical education was provided at the Verrocchio workshop.

Leonardo was another who sought to revive the academy. In his late 40s, he created at least six different original engravings known as the “cartelle of the Accademia Vinciana” (Fig. 2). Perhaps he was designing an emblem for his own school? The *cartelle* feature complex string-like patterns drawn with a single stroke, and one can imagine that he worked on them with considerable enthusiasm. However, perhaps due to his unsociable personality, in the end he never opened the school. Alternatively, he may have considered the *Accademia Vinciana* to be his small workshop in Milan, where he had five or six apprentices.

4. A direct mentor in the field of engineering

During his apprenticeship, Leonardo had several significant interactions that influenced his subsequent engineering career. The Cupola of the Cathedral of Santa Maria del Fiore (Fig. 3), a symbol of Florence, is a monumental masterpiece by the early Renaissance architect Filippo Brunelleschi. However, when Brunelleschi passed away, the final task of placing the bronze ball atop the tower remained unfinished. The Verrocchio workshop was contracted to undertake this unprecedented project, which required lifting a heavy object weighing more than two tons to a height of about 100 meters. Brunelleschi himself had already devised a crane for installing a lantern (lighting tower) at the top of the tower, and several people had copied the blueprints. Leonardo was one of them (Fig. 4).

The commission was received around 1468, and the Verrocchio workshop completed it in about three years. Leonardo, who was in the workshop at the time, was around 18 years old. This was shortly before he qualified as a master at the age of 20, and it is assumed that he was in a position of being his master’s right-hand man. Considering Leonardo’s remarkable activities in the engineering field later on, there is no doubt that this project helped him cultivate significant engineering skills.

Among his predecessors in the field of engineering, Siena-born Francesco di Giorgio Martini is a particularly large presence. Francesco himself was active in many countries as a military engineer, as well as an architect, painter, sculptor, and writer. In that respect, it can be said that he was quite Leonardo-like in his versatile talents. When Leonardo arrived at the Milanese court, Francesco, 13 years his senior, who

had previously worked in Urbino and Naples, was already widely known.

The two hit it off and went to visit the ancient equestrian statue of “Regisore” in Pavia together. Leonardo also obtained and studied Francesco’s manuscript draft, which now resides in the Biblioteca Laurenziana in Florence. To date, this is the only book definitively known to have been a part of the personal library of Leonardo, who annotated its margins in his distinct mirror writing (Fig. 5). Leonardo’s style of showing a figure and adding commentary to it, demonstrates how much he owed to his predecessor. Leonardo learned much of his engineering from him, and it is now clear that the famous self-propelled cars and other vehicles



Fig. 3 Cupola of the Cathedral of Santa Maria del Fiore, Florence

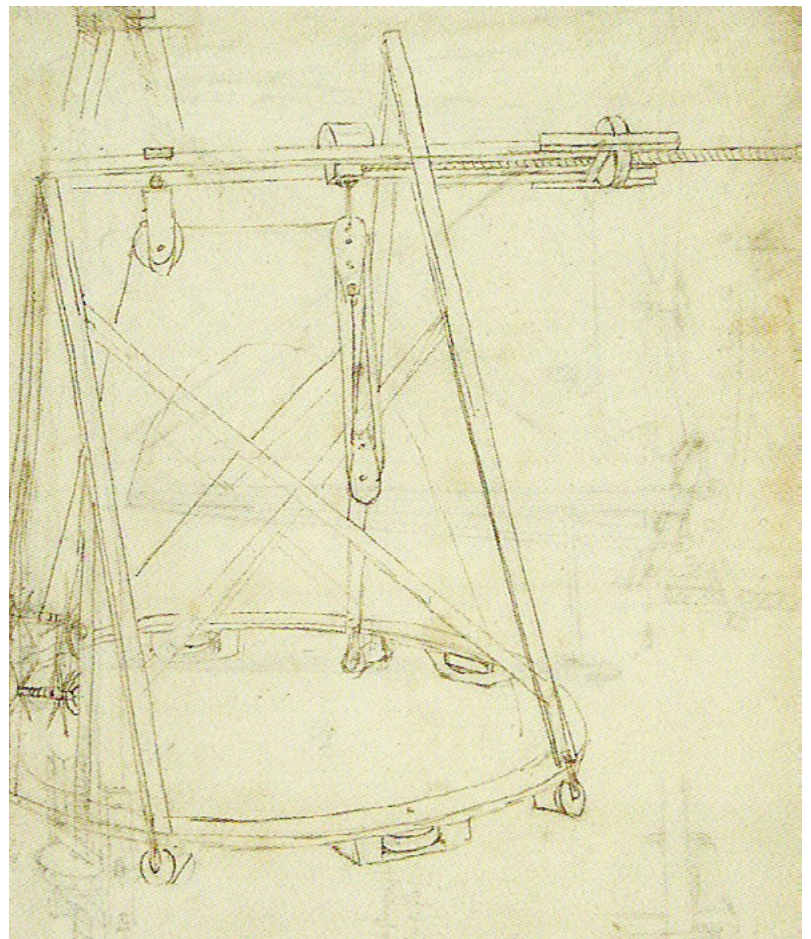


Fig. 4 Leonardo da Vinci, sketch of a crane, circa 1478–80, Codex Atlanticus, f. 808r.

that were once considered Leonardo's inventions owed most of their design to Francesco.

Leonardo was also active in Milan in the field of architecture. Fortunately, Milan was also home to the iconic Renaissance architect Donato Bramante, who must have been an invaluable mentor to Leonardo, who until then had had little opportunity to learn about architecture. The results soon became apparent. In 1487, about five years after Leonardo moved to Milan, a competition was held for the proposed renovation of the ceiling of the crossing tower on the Milan Cathedral. Of the five submitted designs, the judge, Bramante, selected Leonardo's, and there followed three recorded payments to Leonardo for this task.

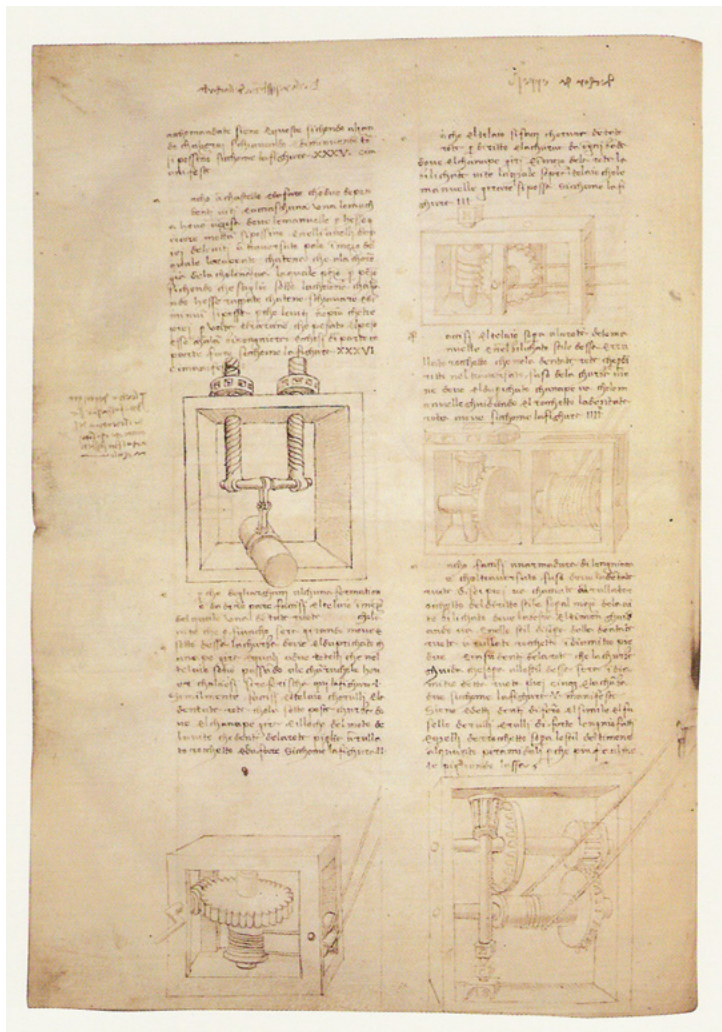


Fig. 5 Francesco di Giorgio Martini, *Treatise on Architecture* (formerly in the Leonardo Collection), Florence, Biblioteca Laurenziana, f. 44v.

5. Engineering development

Leonardo's greatest innovation in industrial design renderings is his "exploded diagram." This technique combines a diagram showing the overall appearance with a diagram that shows the disassembled parts and how they fit together (Fig. 6). In the device shown, when the operator moves the lever on the side of the device back and forth, the two rotating discs rotate in opposite directions and the bar rotates to raise the load. When gears are switched, each disc rotates in the opposite direction from before, and as a result, the load is lowered. In other words, it is a device that converts horizontal back-and-forth motion into vertical up-and-down motion⁵. The left side of the figure shows the overall view, and the right side of the figure shows an exploded view, with each part drawn at a slight distance from the others. The beauty of this method is that it not only reveals the shapes and roles of hidden and invisible parts, but also easily conveys the size, shape, and arrangement of all the parts used.

The introduction of exploded diagrams of mechanical parts was probably influenced by Leonardo's unique concept of "mechanical elements" (*elementi macchinali*). Leonardo used many parts such as springs and gears, but before him, there was no concept of functional classification or standardization in the engineering world, which is what prompted him to introduce the concept of "mechanical elements." For example, in the case of springs, he made illustrated lists of springs of various types⁶. In fact, he probably prepared stocks of pre-made parts and tools in advance, organized by function and specification, so that he

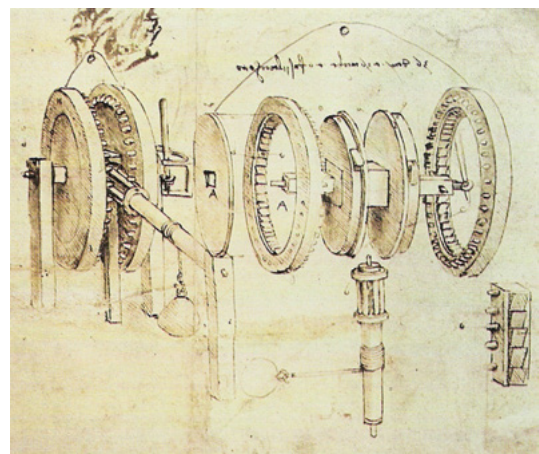


Fig. 6 Leonardo da Vinci, Converter from reciprocating motion to rotary motion, Codex Atlanticus, f. 30v.

⁵ This is one of Leonardo's engineering inventions that may have actually been built, as it could be made with the techniques and materials of the time. For a description of the device's operation, see the following: Domenico Lini, Kazuhiro Susowake, Shigetake Nagao, et al. (dir.), Catalog of the exhibition "Shirarezaru kagaku gijutsusha Leonardo da Vinci" NHK Kinki Media Plan, 1998. Alternatively: Domenico Lorenza et al., "Da Vinci tensai no shigoto" translated by Takako Matsui, Futami Shobo, 2007.

⁶ For Leonardo's unique concept of mechanical elements, see: Shingo Tabata, "Leonardo and Engineering," in Hidehiro Ikegami (ed.), *The World of Leonardo da Vinci*, Tokyo Do Publishing, 2007. In addition: Marco Cianchi, *Leonardo da Vinci's Machines*, Firenze (publication year unknown); *La mente di Leonardo*, a cura di Paolo Galluzzi, Firenze 2006.

could simply select them by desired function when devising a new apparatus.

It is not easy for those of us living in modern times to imagine the hardships that Leonardo and other engineers had to face in an era when they had to make most of their own tools and parts. Before inventing or building a machine, Leonardo first tried to obtain the necessary craft tools in an efficient manner (Fig. 7). In the device in the figure, the weight causes the shaft to rotate, lifting a hammer that periodically swings down, etching dents into the iron plate. The platform itself is also designed so that it inches forward as the shaft rotates, automatically shifting the position where the hammer hits.

Leonardo was keen to pursue such automation to reduce overall workload. That is why his papers so frequently mention ideas that harness not only water flow, which had been used since antiquity, but also the potential energy of a falling weight, as in the above-mentioned machine, or the reaction force of a large, stretched spring. An amusing example is a device that attempts to take advantage of the updraft generated by the heat of roasting meat on a spit to turn the spit mechanically, which is hard work when done manually (Fig. 8). Leonardo installed a fan on top of the narrowed chimney-like structure, with the idea that the rotation of the fan by the rising air current would automatically turn the roasting spit. While admittedly a bit ridiculous, this device could be called a primitive prototype of a steam engine.

I have already mentioned that not all of Leonardo's engineering ideas sprang from his own head. Of course, obsolete scientific ideas were ubiquitous. The "impetus" theory to explain the law of inertia is an example. In order to create a "perpetual motion machine" that can continue

to run indefinitely without power, Leonardo concocts a mechanism in which the rotation of a wheel causes bars with weights attached at the end to swing down, and the resulting reaction keeps the wheel spinning (Fig. 9). As we all know, a perpetual motion machine is impossible, and

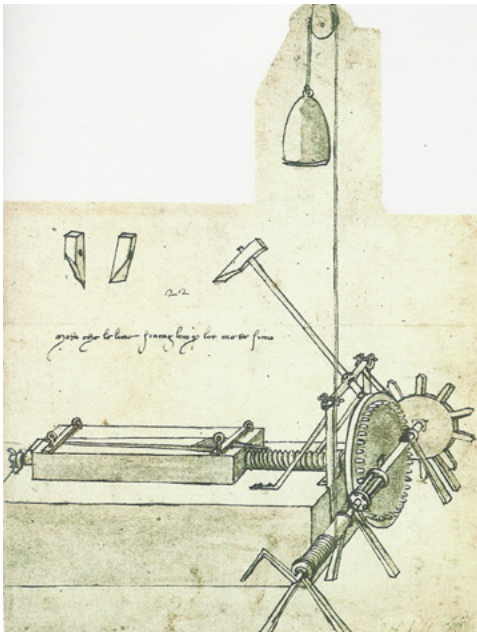


Fig. 7 Leonardo da Vinci, Automatic file-cutting machine, Codex Atlanticus, f. 24r.

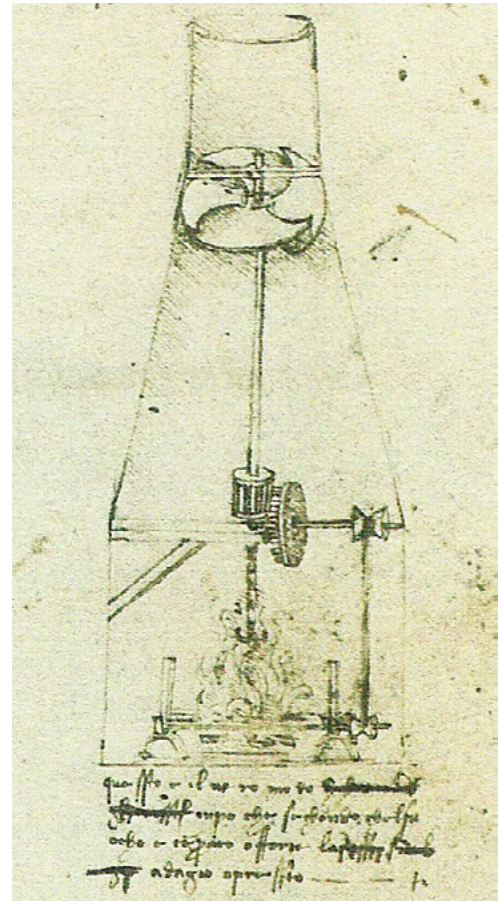


Fig. 8 Leonardo da Vinci, Automatic roasting mechanism, Codex Atlanticus, f. 21r.

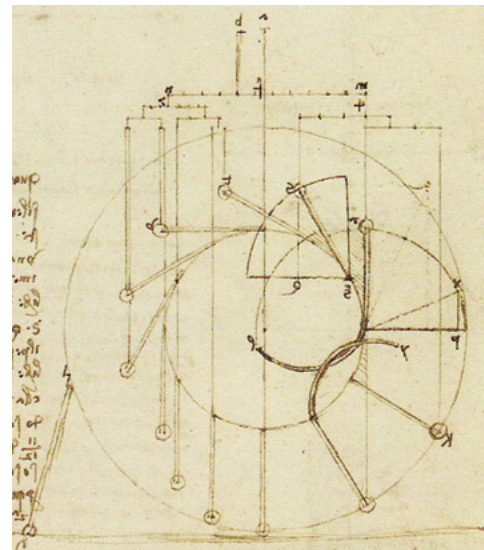


Fig. 9 Leonardo da Vinci, Perpetual motion machine, Codex Madrid I, f. 145r.

6. Conclusion

Leonardo soon realizes this and repudiates the alchemists who claim that perpetual motion is possible.

But Leonardo, being Leonardo, was not content to just assert “it can’t be done”—he also questioned why a perpetual motion machine was impossible. He verifies that the explanation was energy loss due to friction (Fig. 10) by putting weights on objects of various shapes and materials and observing the differences in the degree of friction.

Going one step further, he tests various ways to minimize the generation of frictional heat (Fig. 11). In this experiment, to reduce the energy lost in rotation, he uses a recess containing three balls as an axle bearing. This is clearly a prototype of modern thrust bearings.

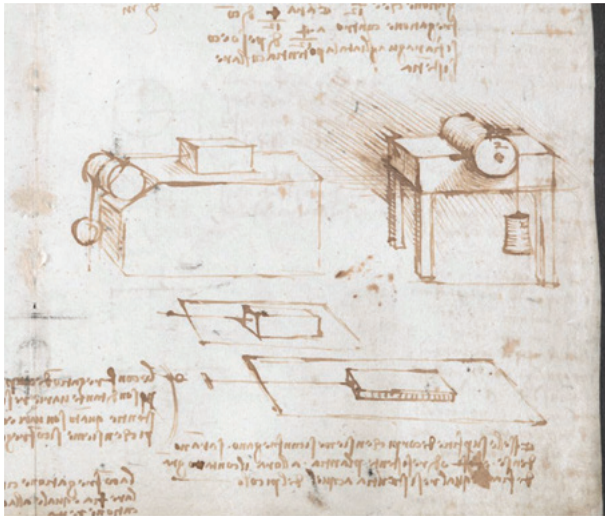


Fig. 10 Leonardo da Vinci, Friction experiment, Codex Arundel, f. 41r.

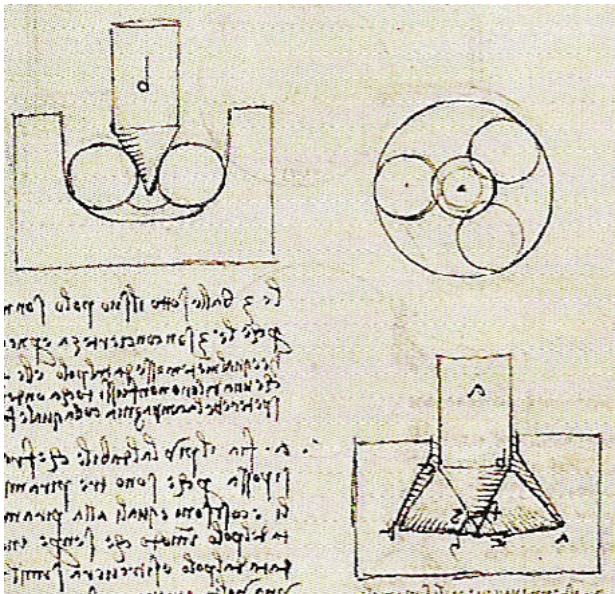


Fig. 11 Leonardo da Vinci, Thrust bearing experiment, Codex Madrid I, f. 101v.

Despite having written down a number of ideas connected with the modern world, Leonardo’s influence on the later history of engineering was not as great as it could have been. Above all, his biggest mistake was never publishing his ideas in a book. It is not that he did not intend to do so. In the past, some even went so far as to claim that Leonardo’s ideas were too radical to publish, a theory that also accounted for his mirror writing. In fact, however, he had a desire to publish and a clear vision for doing so. The Codex Leicester, for example, is clearly a draft intended for publication. There is also a list of chapters to be published under the title “Book on Water” documenting his long-standing observations on water flow. He also left behind a similar chapter-by-chapter structure for an Eastern travelogue, although this would have been completely fictional.

Had his publication plan come to fruition, he would have had a much greater impact on the development of science. However, Leonardo passed away without ever being published. Francesco Melzi, who was entrusted with the legacy of his mentor’s thoughts at a young age, throughout the remainder of his long life managed to organize only Leonardo’s notes on paintings and ended up compiling them into a single volume, *Treatise on Painting*.

After that, as is well known, a significant portion of Leonardo’s vast accumulation of thoughts was lost. Of course, as Pedretti points out⁷, some of Leonardo’s ideas, even if now lost, may have left some influence on future generations of engineers. Surely there were engineers, either contemporaries or later generations, who saw some of the devices he created.

In fact, it has been reported by Rosheim that the exact same “telescopic spring structure” found in a manuscript by Leonardo appears in the writings of Giovanni Alfonso Borelli in the 17th century⁸. This is one example of a contribution to the history of science by Leonardo’s engineering, which was thought to have had little influence on later generations.

■ Author ■



Hidehiro Ikegami

⁷ Carlo Pedretti, “Leonardo da Vinci: Art and Inventions [Mechanics],” translated by Kumiko Tanaka, Yasuyuki Ogura, and Manabu Morita, Toyo Shorin, 2008, pp. 102–113.

⁸ Mark Elling Rosheim, “L’automa programmabile di Leonardo,” in: *Lettura Vinciana*, XL, 2000.

Styling Design of 2022 Dax 125

Yuichi YOKOYAMA¹Takamasa SUMASU²Worawit CHAWALITNIMITKUL³

ABSTRACT

Upon an initiative from Honda's Thai Office, the Dax 125 has been developed as an addition to the "Redesigned Classic Models" such as C125, Monkey 125, and CT125. Since the combination of its small-diameter wheels and horizontally extended frame body resembled the long-bodied, small-sized dog, dachshund, the original DAX was named as DAX Honda in Japan. When designing the style of this model, the T-shaped pressed steel frame having a long beam extending from the steering head pipe to the seat was deemed as the key element to emphasize its uniqueness. We imagined various types of use such as "going out for a ride like taking a lighthearted walk wearing sneakers" and "enjoying a ride with family members on weekends." With that in mind, we chose "family & leisure sneakers" as the key words for the development of the product. In this project, rather than producing a clay model, we have used VR, which allows virtual viewing of the model from 3D CAD data, to acquire an image picture of the entire vehicle from an early period of styling design. Although it was a challenge to maintain the original product image in the new product having a larger engine displacement and higher performance, with the efforts of project members in Thailand and Japan focused on the common goal, the beautiful new model featuring the gentle and modern taste has been realized.

1. Introduction

Recently in Japanese motorcycle market, sales are growing in the small model segment having an engine under 125 cc.

Among such models, the "Redesigned Classic Models" such as Honda's C125, Monkey 125, and CT125 are especially popular among experienced users and so-called return users who choose to ride again after a long absence from the saddle (Fig. 1).

Meanwhile in Thailand where this new model is to be sold, the motorcycle market is growing more toward the hobby use by affluent users from the traditional use of

motorcycles for daily transportation. The customization of '60-'70 Super Cub is starting to become popular among such hobby users whose information via SNS provides impacts to worldwide audience. Honda's "Redesigned Classic Models" have been increasing their presence as an instrument to demonstrate their personality (Fig. 2).

In the said circumstances, upon the proposal from the office of Honda R&D in Thailand, the design concept model of Dax 125 was produced. The Dax 125 is planned as a product to add to the redesigned classical model line up using the original DAX, which was sold from 1969 until 2003, as a design motif.



Fig. 1 "Redesigned Classic Models" of Honda

¹ Design Center

² Honda Motor Co., Ltd.

³ Honda R&D Southeast Asia Co., Ltd.

2. History of DAX

The original DAX was a global model initially launched in 1969 and first sold in America as the ST50/70, then in Japan and Europe. Since the combination of the small-diameter wheels and horizontally extended frame body resembled the long-bodied, small-sized dog, dachshund, the original DAX was named as DAX Honda in Japan.

A general motorcycle design in the days the original DAX was planned was such that had mechanical parts such as the fuel tank, the seat, and the engine, and they were individually emphasized as a design feature. With the mechanical parts hidden and contained in the pressed steel frame, the original DAX realized a motorcycle that was user-friendlier and enjoyable for a wide variety of customers. The less mechanical and easily approachable styling design was well received by various types of customers and its production was continued until 2003 (Fig. 3).



Fig. 2 Snapshot of motorcycle dealer in Thailand



Fig. 3 History of DAX

3. Dax 125 Design Concept Model

3.1. Proposal from Thai Office

The design concept model was first proposed from the office of Honda R&D in Thailand.

What triggered the proposal was the prototype model having a 125 cc engine and a pressed steel frame that a Japanese R&D engineer, who was a DAX enthusiast, was trying to develop in advance. Although DAX models were previously not sold in Thailand, they were well known by the motorcycle manias called “enthusiasts” as well as the trend-setters. They perceived a high value in the one and only character attained from the T-shaped pressed steel frame of the original DAX.

3.2. Studies of the Concept Model

We considered that the key element to impress the uniqueness of the original DAX was the T-shaped pressed steel frame having a long beam extending from the steering head pipe to the seat. That in mind, we chose “horizontal rigid line” as the styling theme of Dax 125 (Fig. 4). These words are aimed to emphasize the T-shaped pressed steel frame symbolic in the original DAX. In the process of idea sketch, various images such as the one close to the original DAX to another having a high-tech image were drawn (Fig. 5). Among the various idea sketches, the key sketch



Fig. 4 Styling concept



Fig. 5 Idea sketches

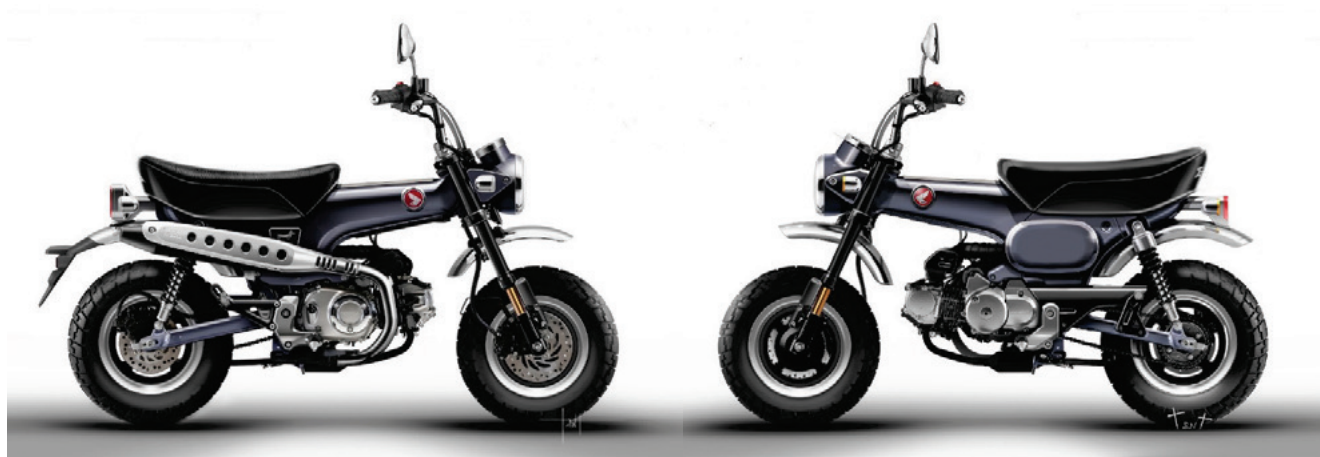


Fig. 6 Key sketches

was chosen because the original DAX image and the high-tech image were well blended in it (Fig. 6). The styling image model was produced by the office of Honda R&D in Thailand (Fig. 7).

3.3. Impressions of the Concept Model in Japan

Produced by Thai Office, the concept model was introduced to the product development team in Japan. As a motorcycle with a pressed steel frame has not been developed recently, some of the evaluation members were skeptical about the outcomes based only on the simulation. Although there were a number of hurdles to clear before bringing to a reality, the concept model had a charm that outbalanced the negatives, impressed the division upper managements, resulting in the finalization of mass-production model development. It was also decided to develop the new model in Japan because it involved a development of a pressed steel frame.



Fig. 7 Concept model produced by office of Honda R&D in Thailand

4. Development of Mass-production Model in Japan

In the redesigning of this model, the key point is how to restore the image of the iconic T-shaped pressed steel frame, which is an item that draws attention and creates user-friendly atmosphere by containing mechanical parts in it. It is an important item for realization of the one-and-only cute styling design.

4.1. Model Concept

In an attempt to realize a global model appropriate in countries in addition to Thailand and Japan, we reviewed the model concept.

We imagined various types of use including the one for a family fun on weekends while being suitable for a casual use like going out for a walk in sneakers, just like the original DAX. Thus, we have established the key words “family & leisure sneakers” (Fig. 8).

4.2. Main Target Market

Although the proposal came from the Thai Office, in addition to the limited market of hobby users in Thailand we set Japan as the main target where use of the product as a daily commuter is presumable. In the Japanese market, the



Fig. 8 Key words

redesigned classical models, i.e., C125, Monkey 125, and CT125 are popular among experienced users and so-called return users. We considered that the new model should have features to increase user options of product selection while avoiding rivalry with these three models. That in mind, we have chosen the long seat that allows carrying of a passenger in the standard trim and the four-speed transmission equipped with an automatic centrifugal clutch that does not require clutch operations. These features make the new model stand out from the existing ones (Fig. 9).

4.3. Styling Design

The conditions that should be strictly adhered to were established to maintain the styling image of the concept model and the original DAX which served as a design motif. In the said process, an icon was produced to clarify the design requirements (Fig. 10). With the icon produced, the balance and the key points of the unique styling became clearer than before. Accordingly, the T-shaped pressed steel frame, the high-mounted muffler, and the round lighting instruments (headlamp, taillamp, turn indicators) were set forth as the requirements.

Among those, the T-shaped pressed steel frame was considered as the most important style identity to emphasize the long horizontal beam that corresponds to the long body of the dog.

4.4. Studies of Styling Design by 3D CAD

In the conventional styling design of motorcycles, it was a common practice to evaluate the style using a 3D clay model after finalizing the 2D sketch. In this project, however, we elected to evaluate the design using the 3D CAD data rather than producing a clay model. Speaking specifically of the frame, the design was directly conducted on the 3D CAD without any reliance on the commonly-practiced reviewing of the design using 2D sketches.

When the pressed frame itself plays a key role of the styling design as in this model, its external shape itself directly affects the strength. In the development of a motorcycle using a pipe frame, the strength and/or the

stiffness can be adjusted by modifying the pipe diameter, gusset plate thickness and/or size. In this new model having a pressed steel frame, however, such a method cannot be taken. That in mind, we frequently exchanged the 3D CAD data between the styling design team and the function design team to achieve the attractive appearance while satisfying the functional requirements. The styling design by this method enables applications of outcomes from the analyses on the manufacturability, stiffness, strength, and crashworthiness that change on daily basis in the development process and affect with each other, as well as allowing adequate sharing of information with the functional design team. As adjustments and analyses on a surface height that should be increased or reduced had to be repeated several tens of times at an increment as small as 1 mm, this method was quite efficient.

4.5. T-shaped Pressed Steel Frame Design

The following styling design criteria were set forth to achieve the design target.

(1) Dense packaging of parts in the frame for a slim

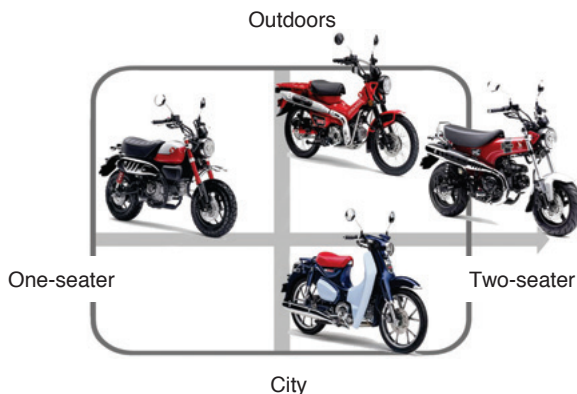


Fig. 9 Positioning map of various models

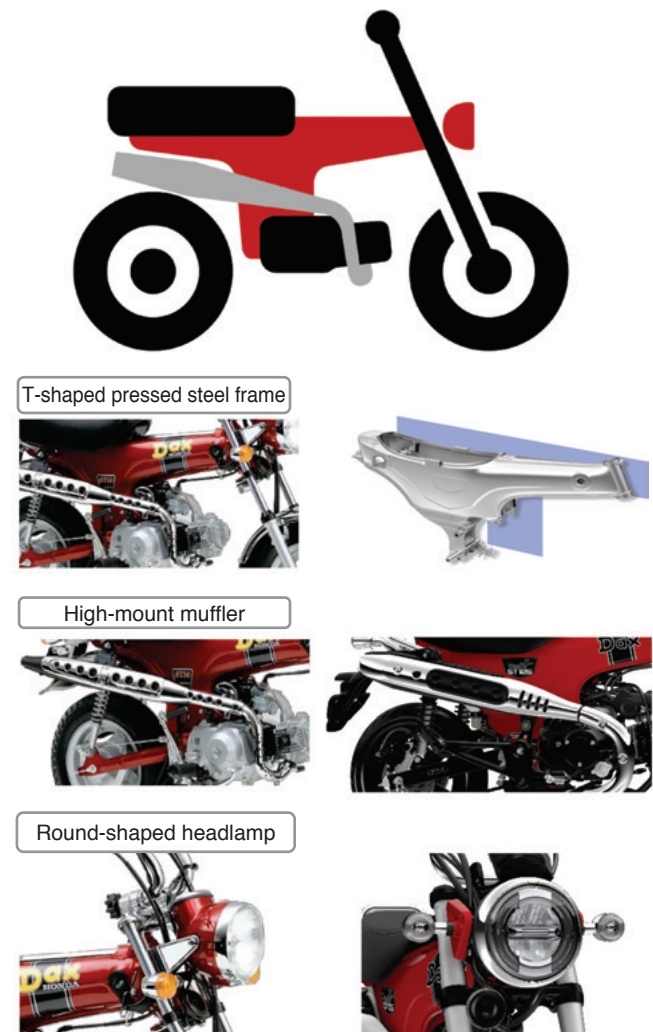


Fig. 10 Icon indicating balance of styling

appearance while maintaining an appropriate clearance
 (2) Gentle-looking surface attainable only from the pressed steel product

(1) Dense packaging of parts in the frame for a slim appearance while maintaining an appropriate clearance

A feature of the original DAX frame is such that the long and slim T-shape is clearly recognizable. In this new model, a larger number of parts than in the original DAX have to be contained in the frame due to an increase of engine displacement and ancillary devices necessary in a modern motorcycle. When compared to the original DAX having an engine displacement between 50 cc and 70 cc, the new model has to contain additional parts such as the ABS modulator, the canister, and the 3-axis IMU to detect body inclination while dealing with the increase of fuel tank capacity, air box volume, and battery size.

In the conventional method, various function development teams perform a parts layout of their own in advance. In such a method, an extra allowance has to be given to the clearance, causing an increase of the frame width, which does not allow restoration of the slim and long body of the original DAX character.

An increase of seat width resulting from an increase of the frame width hinders rider's easy reach of feet to the ground. Similarly, the high-mounted muffler, which is a key item of styling design, sticks out more toward the outside, requiring the passenger to extend his or her right leg to outside. A slim frame is important for the Dax 125 because

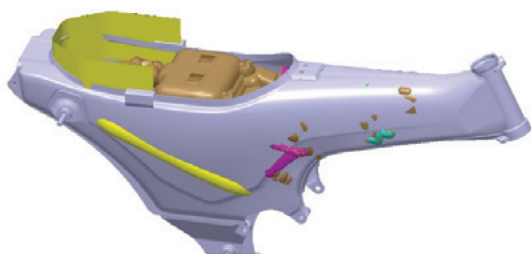


Fig. 11 Functional parts sticking out in early frame data

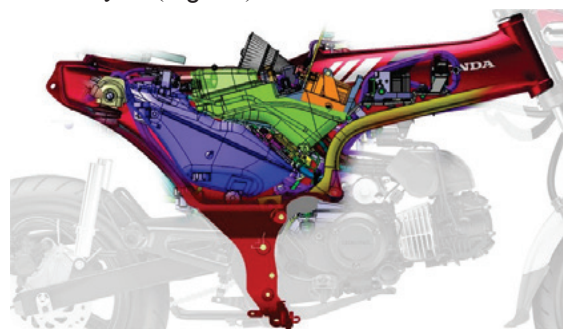


Fig. 12 Air box, fuel tank, and battery densely packaged in the pressed steel frame



Fig. 13 Pressed steel frame having an optimum volume

one of the goals is passenger comfort.

That in mind, the styling design team prepared 3D CAD data of the target frame configuration at the start of this project and provided the data to the frame body design team. The data formed a basis of component layout for various function development teams.

Among the functional part to be contained in the frame, bulky items are the air box and the fuel tank. The fuel tank having a 3.8 L capacity for convenience in daily use sticks out from the 3D CAD data of the frame provided by the styling design team (Fig. 11).

A lack of space also occurred in various other parts. To deal with these issues, the major components were located and relocated like a puzzle, and the layout was modified over and over. Along with an extensive change of layout, the styling design team also reviewed the style to come up with an optimum design satisfactory from the standpoints of function, law compliance, and style.

After several layout modifications and reviewing of the style, the layout is finalized in such a manner that the air box, the fuel tank, and the battery are densely packaged in the frame, and the electrical parts such as the ECU are stored in the side covers (Fig. 12).

The sharing of layout criteria within the teams in the early period of development contributed to the prompt addressing of issues, ultimately allowing realization of the pressed steel frame having an optimum volume as a 125 cc class motorcycle (Fig. 13).

- (2) Gentle-looking surface attainable only from the pressed steel product

In addition to the T-shape in the side view, we pursued a timeless and unique appearance only attainable from the pressed steel material in an attempt to make the product lovable and amiable for a long time for the users. Even now when models having cosmetic parts made of plastic are in the mainstream, there still are quite a few users who are attracted by the solidness and the structural charm of steel material.

Although the Dax 125 is an industrial product for mass production, we conducted studies to make the surface appear like a craft object. Here, the term “surface appearance” is used to refer to the tension and change of surface often used in the art-related industry which deals with three-dimensional objects.

In a craft object having a gentle appearance, the craftsman elaborates to create a user-friendly shape (Fig. 14). In the Dax 125, we have pursued such a surface appearance that the user can admire for a long time, and that is smooth and comfortable to touch while maintaining the requirements as a motor vehicle such as stiffness and strength (Fig. 15).

The press work of steel sheet is such a manufacturing method in that the flat steel sheet is placed between the upper and lower dies and pressed to the designated shape. As the necessary stiffness, strength, etc. are sometimes unachievable if the steel sheet is excessively stretched locally, a concave-folded portion has to be large in size. Another feature is that the entire surface needs to be bulged sufficiently to maintain necessary stiffness and strength.

Taking into account the said characteristics of the



Fig. 14 Craft object having gentleness



Fig. 15 Surface appearance that is smooth and comfortable to touch

pressed steel sheet, the frame is configured with an accent in the imaginal line running in the fore-and-aft direction, thus emphasizing the T-shape and expressing entire vehicle body's stretchiness (Fig. 16). Moreover, the modern and tight, yet gentle style has been realized by incorporating small, mechanical protrusions in the gently-curved and organic cross section.

In an attempt to grasp the final volume of entire vehicle, the shape was virtually reviewed from the 3D CAD data from the early period of styling design (Fig. 17). The shape was analyzed and modified repeatedly as many as to approximately several tens of times. Yet, it is not easy to produce a surface appearance that is smooth and comfortable to touch if the shape is reviewed only in the screen, even using the very fine images from the 3D CAD. Hence, we frequently cut out a wax model and reviewed the actual contour by touching and feeling with our hands (Fig. 18).



Fig. 16 Expression of entire vehicle body's stretchiness

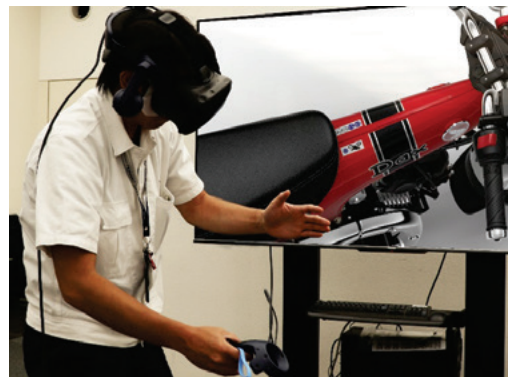


Fig. 17 Virtual evaluation using VR



Fig. 18 Wax model to touch and feel the shape

Thus completed is the T-shaped pressed steel frame featuring the dense packaging of parts in the frame while maintaining an optimum clearance as well as achieving the gentle appearance.

4.6. Welding

In the Dax 125 in that the pressed steel frame serves as a cosmetic part as it is, the appearance of welding beads plays an important role.

The beam, which stretches from the steering head pipe to the seat and is visible when the rider looks down from a seated position, is very critical from the standpoint of cosmetic finish. As the said part is composed of three pressed steel sheets, i.e., the right and left and bottom plates, welded together, the welding beads are visible from all angles.

In the Dax 125, therefore, a number of cosmetic reference samples of welding beads were produced to maintain the cosmetic quality of welding beads (Fig. 19).

To enhance cosmetic appearance of welding beads, the parameters such as the welding speed, the type and the amount of shield gas, the jig, and the torch angle were reviewed and specific conditions were established. Furthermore, the addition of finishing processes such as the spatter removal with the wire buff constantly helps ensure quality appearance of welding beads (Fig. 20).

In an attempt to help ensure necessary stiffness and strength in the steering head pipe area taking into account aesthetics, the data was analyzed in advance, thus realizing the simple yet beautiful finish (Fig. 21).

4.7. Styling Design of High-mounted Muffler

The high-mounted muffler is one of the necessities of DAX and is a critical item to produce such atmosphere in that the user feels it's good for casual rides and he/she may be able to go anywhere. The high-mounted muffler is often employed to increase ground clearance for off-road riding, and makes it easy to create light and agile, "go anywhere" image. The following criteria were set forth when designing the high-mounted muffler.

- (1) Exhaust pipe cover as beautiful as a brass instrument (Fig. 22)
- (2) Slim-looking muffler tucked along the frame

- (1) Exhaust pipe cover as beautiful as a brass instrument

As an exhaust pipe of mass-production motorcycle is machine-bent, its shape often looks mechanical because it is composed of straight lines and a bend having a constant radius. In this new model, we chose to realize an appearance like a brass instrument by creating an elegant and smooth line by hiding the exhaust pipe with the exhaust pipe cover made of chrome-plated pressed steel sheet while maintaining a sufficient space from the hot exhaust pipe.

With the pipe cover designed to match with the muffler cover, the high-mounted muffler having integrity and

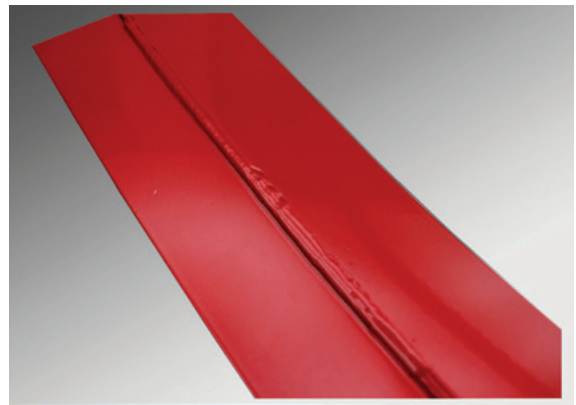


Fig. 19 Sample of welding bead appearance

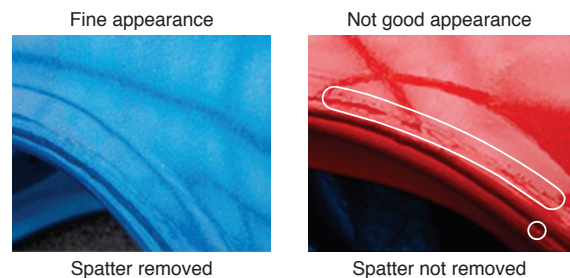


Fig. 20 Verification of spatter removal



Fig. 21 Simple yet beautiful steering head pipe area without gusset



Fig. 22 Elegant bending of brass instrument

beauty from the chrome-plating has been realized (Fig. 23). This design is aimed at being admired for a long time just like the pressed steel frame.

(2) Slim-looking muffler tucked along the frame

To keep the high-mounted muffler from sticking out sideways, the frame side of muffler outer shell is recessed to allow a space for the rear shock. Thus, the design having an integrity with the frame body has been realized. The use of black plastic protector on the muffler cover is a consideration to make the muffler look slim (Fig. 24).

4.8. Coloring

With respect to the coloring, two color options have been chosen. They are the heritage red that reminds of the original DAX of the old days with a target at return users, and the blue gray, which is a modern earth-color

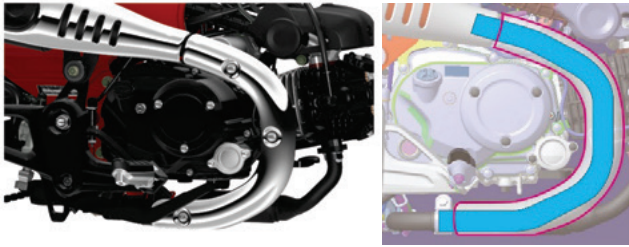


Fig. 23 Exhaust pipe cover as beautiful as a brass instrument

that young users having no knowledge of the original DAX also favor.

When the colors of past DAX models are compared, red and blue are seen frequently as the standard color of DAX (Fig. 25). Especially the red is employed in the first-generation DAX as well as in the revival version sold in 1996 in Japan. The red is also the corporate color of Honda. Accordingly, the red is selected. Among various kinds of red, the Pearl Nebula Red having a luxury image is selected.

In the meantime, young users tend to favor a quiet body color that makes the rider play the main role rather than the motorcycle. That in mind, we chose the less-colorful, Pearl Cadet Grey as a color targeted at young users.

The two colors are aimed at a wide variety of users from



Fig. 24 Slim-looking muffler tucked along the frame



Fig. 25 Colors of past DAX models

the experienced ones who know the original DAX up to the young ones (Fig. 26).

The vertical stripes on the pressed steel frame resembling a collar of dachshund are the standard graphic of the original DAX. The design having a luxury image that also satisfies adults has been achieved by the use of simple colors while maintaining the original image.

The original dachshund mark has been redesigned to make it modern while keeping the playful mind of the old days. Along with the increase of power from the engine displacement of 125 cc, the new mark is designed to a café-racer style to emphasize an uplifting image by the oval tires, the wind-swept ear, and the wearing of goggles. At the same time, the bold lines are used to highlight the character and to enhance recognizability (Fig. 27).



Fig. 26 Colors of new model: Pearl Nebula Red (left) and Pearl Cadet Grey (right)

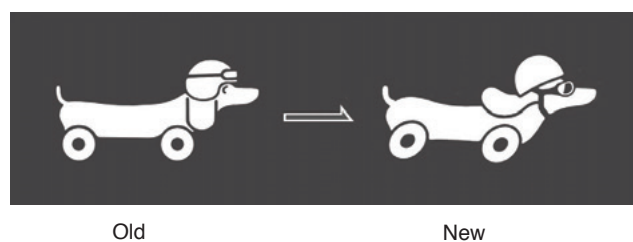


Fig. 27 Redesigned modern dachshund mark

5. Conclusion

The development of new Dax 125 is a redesigning of the historical motorcycle with a challenge in how to maintain the original image in the new model having a larger displacement engine and higher performance. Although the hurdle was much higher than we thought to package the devices in the simple and compact design to meet the latest law regulations, the goal has been achieved by employing the new styling design method using 3D CAD.

Although the roles were shared like the concept model produced in Thailand and the advance technical studies of frame and the mass-production product development conducted in Japan, each party had a strong fondness of the

Dax 125, resulting in the focusing of all their efforts into the common goal of product development. As a result, we have achieved a beautiful new model having a modern image in addition to the identity of the original DAX.

We would like to continue development toward a further evolution while monitoring how the users in the worldwide market receive the new model.

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Turbocharged and Naturally Aspirated Gasoline Direct Injection V6 Engines with Common Architecture based on New Valve Train System

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ABSTRACT

Two gasoline direct injection V6 engines were developed using a new valve train architecture with a common engine geometry: a turbocharged engine aimed at enhancing power performance, and a naturally aspirated engine aimed at enhancing environmental performance. Variable Timing Control for intake and exhaust was adopted to improve boost pressure responsivity for the turbocharged engine and thermal efficiency for the naturally aspirated engine. To maintain an engine size comparable to that of the previous engine, a new compact valve train was developed, combining a valve lifter-based Variable Cylinder Management system with built-in cylinder deactivation switching mechanism, hydraulic lash adjusters, and roller rocker arms. With these changes, these systems were equipped at the same cylinder head size as the previous SOHC engine. Furthermore, the new valve train allowed for a common engine geometry between the turbocharged and naturally aspirated engines, including cylinder head structure and rotational axis positions of the crank shaft and cam shafts, to help ensure commonality for the timing belt drive system and other systems. A single twin-scroll turbocharger was adopted for the turbocharged engine, achieving a maximum power of 265 kW and maximum torque of 480 Nm. The naturally aspirated engine features a fuel injection system with fuel pressure increased to 30 MPa, and a vehicle with the engine achieved an LEV III SULEV30 rating, producing PM of 1 mg/mile.

1. Introduction

In the North American market, there is a need for powertrains with driving force margin for SUVs and pickup trucks. In 1997, Honda developed a 3.0-L naturally aspirated (NA) SOHC VTEC V6 engine for increased power output. Then in 2003, to enhance environmental performance, Honda developed its Variable Cylinder Management (VCM) system⁽¹⁾, a rocker arm (R/A) based valve train with cylinder deactivation for switching between 6-cylinder and 3-cylinder operation. Next, looking to further enhance environmental performance in 2008, the company developed the VCM-II system for multi-stage switching between 6-cylinder, 4-cylinder, and 3-cylinder operation, pairing this VCM-II with an increased displacement engine for increased power in a modified NA 3.5-L V6 engine⁽²⁾. In 2013, a 3.5-L NA direct injection V6 engine⁽³⁾ was developed, combining VTEC, which switches valve lift to enhance power characteristics, with the VCM system for enhanced environmental performance.

This V6 engine is also used as the main powertrain for the Acura brand, which sells luxury vehicles mainly for

the North American market. In recent years, to increase the product appeal of its flagship high-performance models, Acura has seen demand for large power enhancements in the existing 3.5-L NA direct injection V6 engine. Forced induction with a turbocharger (T/C) is an effective means to increase power, although it comes with reduced launch response due to lag in boost pressure response in the low-rpm range. Boost pressure response can be improved with a twin-scrolling turbocharger to reduce exhaust interference and optimized control of valve overlap with Variable Timing Control (VTC) for intake and exhaust valves⁽⁴⁾.

Meanwhile, the Honda brand has demand in the North American market for a high displacement NA engine with a driving force margin for launching and smooth power characteristics from low to high rpm for SUVs and other vehicles used for hauling and towing. Also, demand for reduced environmental impact has increased in recent years across the automotive industry as a whole⁽⁵⁾. Thus, emissions performance at LEV III SULEV30 standards was targeted for the developed NA engine, which reduces nonmethane organic gas (NMOG) and NO_x to roughly a quarter of the LEV III ULEV125 standard achieved by the

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previous engine. Emissions performance can be enhanced by increasing fuel pressure in the fuel injection system and reducing fuel adhesion in the cylinder. Also, thermal efficiency can be improved with VTC for intake and exhaust to reduce pumping loss and optimized internal EGR volume.

Thus, it is desirable to adopt intake/exhaust VTC for the increased power and environmental performance. However, doing so requires a change in valve train architecture from the SOHC configuration of the previous engine to a DOHC configuration. Converting to DOHC and using the rocker arm-based VCM system means larger cylinder heads to support the valve train, making it non-trivial to maintain the same engine size as the previous engine. Therefore, a compact valve train was developed, combining a valve lifter-based VCM system adopting a built-in cylinder deactivation switching mechanism with hydraulic lash adjusters (HLAs) and roller rocker arms to help prevent an increase in engine size. Furthermore, this new valve train allowed for a common engine geometry between the turbocharged and NA engines, including cylinder head structure and rotational axis positions of the crank shaft and cam shafts. It also ensures commonality for the timing belt drive system.

This paper describes the newly developed valve train, along with the development aims and technologies applied for the two direct injection V6 gasoline engines using this valve train as a common structure (Fig. 1): the turbocharged model for enhanced power performance, and the NA model for enhanced environmental performance.

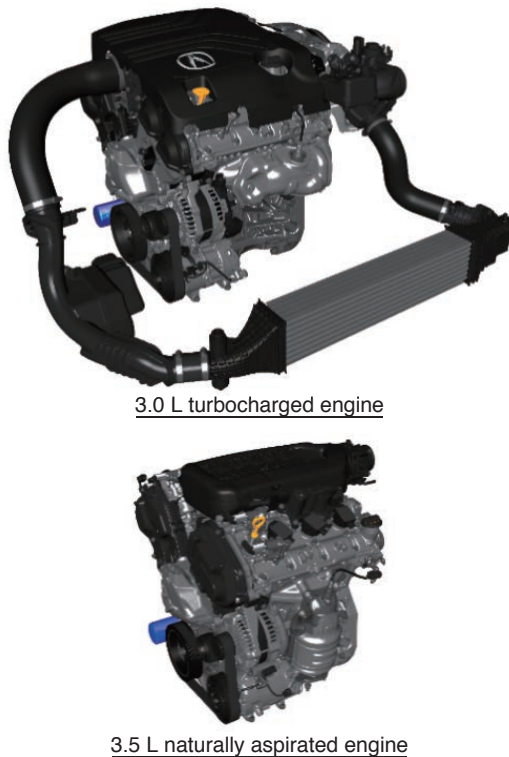


Fig. 1 Overview of newly developed engines

2. Development Targets

The development targets for the turbocharged and NA engines with the new valve train in common are as follows:

- (1) New valve train
 - Adopts intake/exhaust VTC and VCM system
 - Cylinder head size equivalent to the previous engine
 - Common engine geometry (valve train architecture, cylinder head structure, main rotational axis positions)
- (2) Turbocharged engine
 - Maximum power 265 kW, maximum torque 480 Nm (4.0–4.5-L NA engine equivalent)
 - Improved launch response
 - Maximum thermal efficiency of 36.5% or more
- (3) NA engine
 - Achieves LEV III SULEV30 category for emissions performance
 - Maximum thermal efficiency of 37.5% or more

3. New Valve Train

Figure 2 shows a comparison of valve train architectures for (a) the valve lifter-based VCM system for

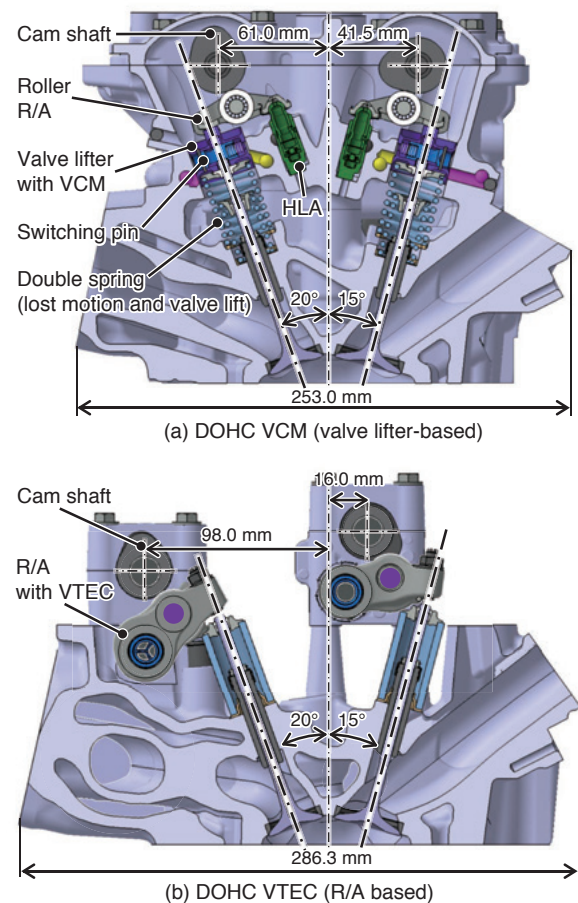


Fig. 2 Comparison of valve train architectures between previous and newly developed models

the developed DOHC architecture and (b) the rocker arm-based VTEC system as used in DOHC I4 engines.

The VCM system features cylinder deactivation switching pins built into the valve lifters and uses two valve springs with a retainer for each spring to consolidate the VCM system on the valve shaft together with the lost motion assembly. Also, using HLA and roller rocker arms allows the cam shafts to be positioned closer to the cylinder center than the valve shafts compared to the rocker arm-based DOHC VTEC architecture in Fig. 2(b). This prevents the adoption of a VCM system from enlarging the cylinder head size, while also achieving an equivalent valve angle.

The valve lifter-based VCM system has been newly developed for automobiles based on the Hyper VTEC valve control system with a valve deactivation mechanism for motorcycles⁽⁶⁾. Components for the system are shown in Fig. 3, and valve lifter motion during the cylinder deactivation switching operation is given in Fig. 4. Switching for cylinder deactivation is hydraulically controlled from oil galleries on the cylinder head with a switching pin built into the valve lifter. During valve lift activation, hydraulic pressure from the activation oil gallery pushes the switching pin against the stopper pin, and the valve is lifted via the switching pin. During valve lift deactivation, hydraulic pressure from the deactivation oil gallery slides the switching pin, aligning a through hole in the switching pin with the valve shaft. When the valve lifter is lifted, the valve stem end slides in the through hole and stops the valve lift. A slide retainer and outer spring that slide around the valve stem in the axial direction allow valve lifter motion to track with the cam profile.

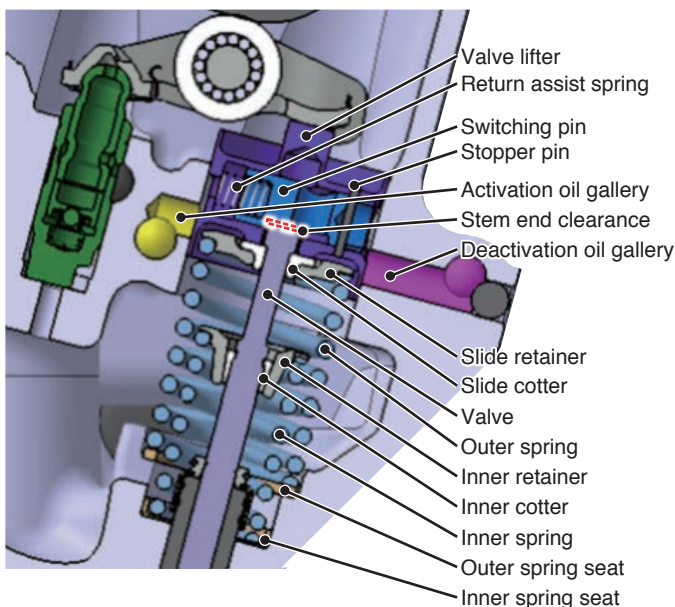


Fig. 3 Component of valve lifter-based VCM

The new valve train is the first Honda cylinder deactivation valve train to be combined with HLAs. By adopting HLAs, the clearances between rocker arm, cam lobe, and valve lifter as generated by tolerance variations in components can automatically be reduced to zero. This reduces cam profile and valve lift offsets, as well as variations in combustion performance due to reduced variations in the intake air flow rate, producing a maintenance-free valve train. In the VCM system with HLAs, the clearance required for activation of the cylinder deactivation switching pin was set between the switching pin and valve stem end to achieve reliable activation.

Figure 5 shows an overlay comparison of the cylinder head cross sections of the previous SOHC engine with VTEC-VCM system and the newly developed DOHC engine with VCM system. A cylinder head size equivalent to that of the previous engine was achieved while adopting the DOHC structure and VCM system. Also, by using an aluminum alloy cam cover with integrated cam journal, the cylinder head height including the cover was reduced by 30 mm from the previous engine.

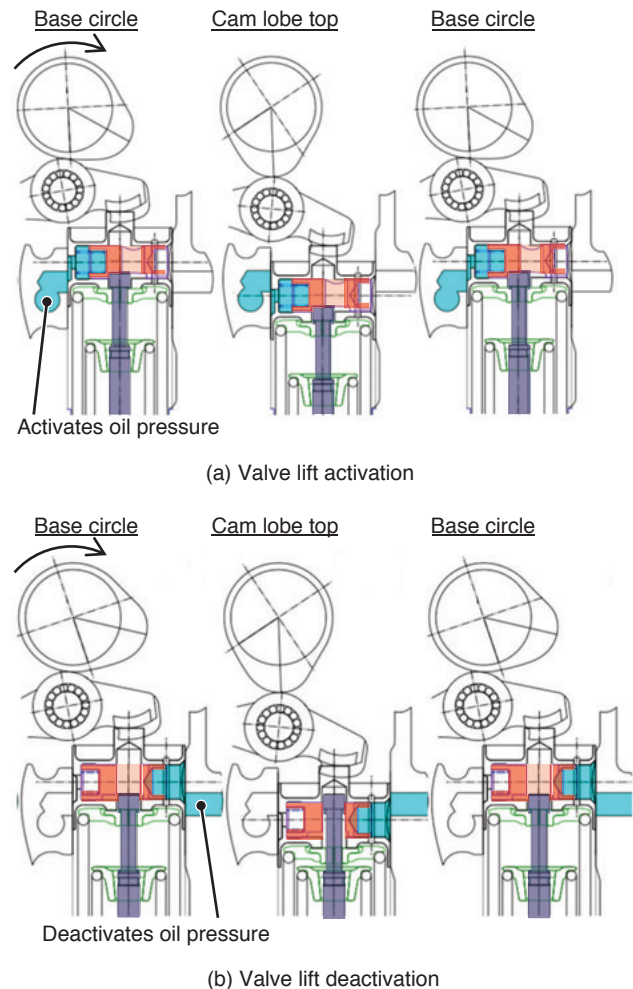


Fig. 4 Valve lifter motion

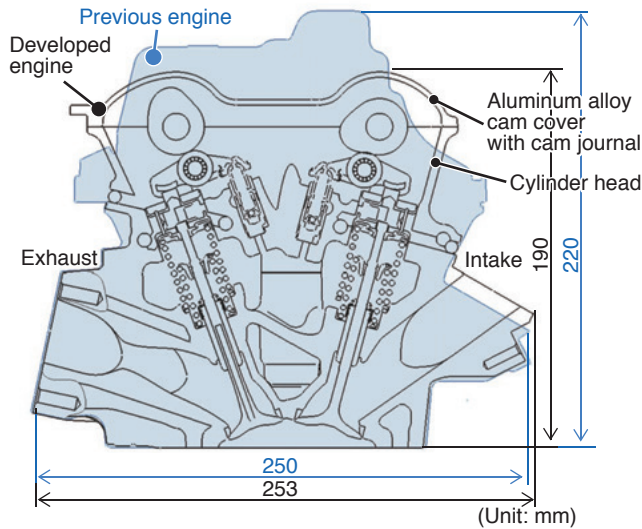


Fig. 5 Comparison of cylinder head appearances

4. Common Engine Geometry

In the developed engine, the new valve train allowed for a common engine geometry to be shared between the turbocharged and NA engines, including cylinder head structure and rotational axis positions of the crank shaft and cam shafts. Furthermore, the bore pitch, block height, bank angle, and other measurements were carried over from the previous engine. This allowed for an engine of the same size as the previous engine while adopting the DOHC architecture and intake/exhaust VTC.

With the common engine geometry, the same timing belt drive system could be shared for both turbocharged and NA engines. The timing belt system layout is shown in Fig. 6. In general, V engines employ timing chain

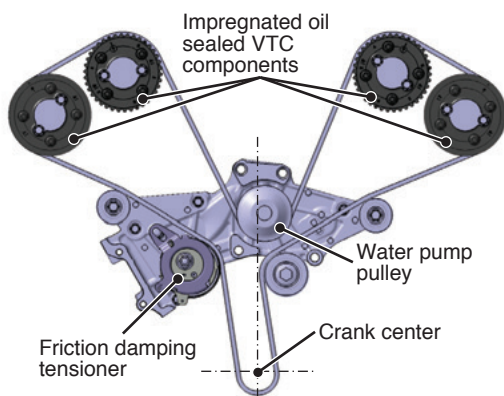


Fig. 6 Serpentine timing belt system

drive systems, which require multiple chains in a crank shaft driven system as each bank is offset. This enlarges the engine size in the crank shaft stroke direction. As with the previous engine, a serpentine layout was adopted to drive the crank shaft, cam shaft, and water pump with a single timing belt, preventing an increase in engine size.

The developed engines employ a DOHC structure and intake/exhaust VTC, which increases the amount of work driven by the valve train. Thus, to help ensure the strength of the timing belt, a friction damping tensioner was used in the auto-tensioning mechanism to reduce resonating tension.

The intake/exhaust VTC is integrated with the driven pulleys and driven by the timing belt. The structure of the driven pulleys with integrated VTC is shown in Fig. 7. As the pulleys are made of powder-sintered material, it is necessary to prevent oil seepage into the VTC from spoiling the timing belt. Therefore, O-rings were placed on the VTC component mating surfaces, and the powder-sintered pulleys were impregnated in resin to seal the oil and help prevent belt spoiling.

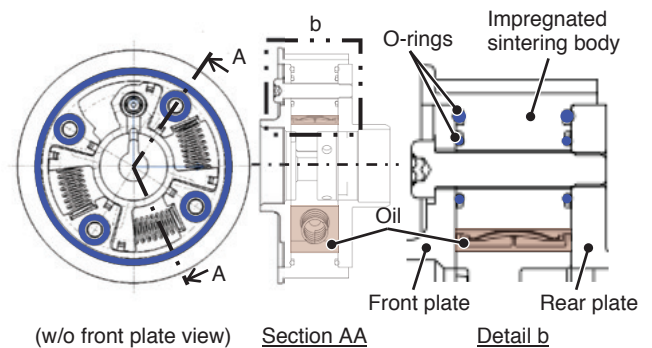


Fig. 7 Impregnation to seal oil for VTC

5. Turbocharged and NA Engine Specifications

Table 1 shows the main specifications of the turbocharged and NA engines with the new valve train in common as compared with the previous engine. By adopting a common structure between the turbocharged and NA engine models, the two new engines can be made with only minor changes, such as installing the turbocharger or increasing fuel injection system pressure. By adopting technologies suitable for each engine specification, the maximum power, maximum torque, and launch response were enhanced on the turbocharged engine, and emissions performance was increased on the NA engine.

Table 1 Comparison of engine specifications

	Previous NA engine	Developed NA engine	Developed T/C engine
Cylinder configuration	V6 cylinder	V6 cylinder	V6 cylinder
Displacement [cc]	3471	3471	2997
Compression ratio	11.5	11.5	9.8
Bore × Stroke [mm]	Φ89 × 93	Φ89 × 93	Φ86 × 86
Bore pitch [mm]	98	98	98
Block height [mm]	235	235	235
Bank angle [deg]	60	60	60
Valve train configuration	SOHC – 4-valve / cylinder	DOHC – 4-valve / cylinder	DOHC – 4-valve / cylinder
Variable valve control device	Intake/Exhaust VTEC and VCM	Intake/Exhaust VTC and VCM	Intake/Exhaust VTC and VCM
Variable cylinder management	6 – 3 cylinder switching	6 – 3 cylinder switching	6 – 3 cylinder switching
Cam drive system	Belt drive / Hydraulic tensioner	Belt drive / Friction damping tensioner	Belt drive / Friction damping tensioner
Aspiration system	Natural aspiration	Natural aspiration	Twin-scroll turbocharged aspiration
Fuel injection system	Direct injection	Direct injection	Direct injection
Maximum fuel pressure [MPa]	20	30	20
Fuel grade	RON91	RON91	RON95
Emission category	LEV III ULEV125	LEV III SULEV30	LEV III ULEV70
Maximum thermal efficiency [%]	36.5	37.5	36.5
Maximum power [kW/rpm]	209 / 6000	213 / 6100	265 / 5500
Maximum torque [Nm/rpm]	355 / 4700	355 / 5000	480 / 1400 - 5000

6. The Developed Turbocharged Engine

For the newly developed turbocharged engine, a single turbocharger was adopted to realize a compact layout, and the same engine length as the previous engine was achieved by mounting the turbocharger above the transmission (Fig. 8).

Using a single turbocharger to increase power means a larger turbocharger, making boost pressure response in the low-rpm range an issue. Thus, a twin-scroll turbocharger with lower exhaust interference was adopted to improve boost pressure response. Figure 9 shows a comparison of boost pressure response between twin-scroll and mono-scroll turbochargers. When changing load at a fixed engine speed of 1500 rpm, the twin-scroll turbocharger reached 80% of maximum torque 0.5 seconds faster than the mono-scroll turbocharger.

Combustion technology was carried over from the concept for the 2.0-L I4 turbo engine⁽⁴⁾, with thermal efficiency improved by suppressing knocking through rapid combustion with enhanced in-cylinder turbulent kinetic energy. Figure 10 shows the intake port, piston crown shape, and in-cylinder flow as calculated by CFD. The upright edge at the port bottom guides the mainstream of intake air to the upper surface of the intake valve and strengthens tumble flow, and the crown shape of a large-radius shallow dish enhances the retention of tumble flow.

An overview of the intake/exhaust VTC control strategy is shown in Fig. 11. In the low-rpm, high-load region, which dictates launch response, valve overlap was increased, and boost scavenging effects were used to increase torque (a). At low rpms and low to mid load, thermal efficiency was improved by optimizing internal

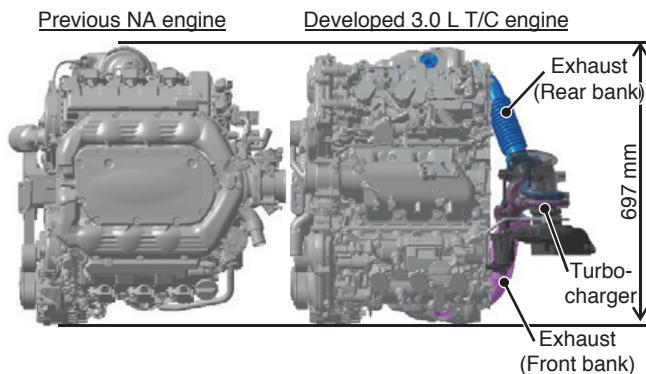


Fig. 8 Comparison of engine width between previous NA and developed T/C engines

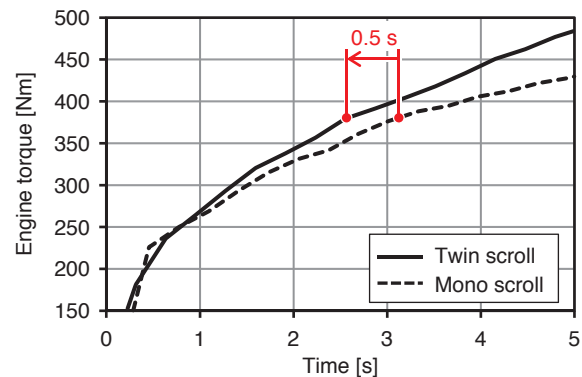


Fig. 9 Engine torque response at 1500 rpm

EGR volume, and fuel economy was improved with cylinder deactivation to reduce pumping loss (b, c). At mid to high rpms, the range was optimized to reduce valve overlap, and thermal efficiency was improved by enhancing exhaust efficiency to reduce knocking (d). In the engine start-up and idle range, valve overlap was reduced to reduce internal EGR volume and stabilize combustion (e).

Figure 12 shows the engine power and torque characteristics. Maximum power is 265 kW, and maximum torque is 480 Nm from 1400 rpm to 5000 rpm. Next, Fig. 13 shows a thermal efficiency map of the engine. The engine achieved a maximum thermal efficiency of 36.5%, and the area with a thermal efficiency of 34% or more was expanded while still enhancing power and torque.

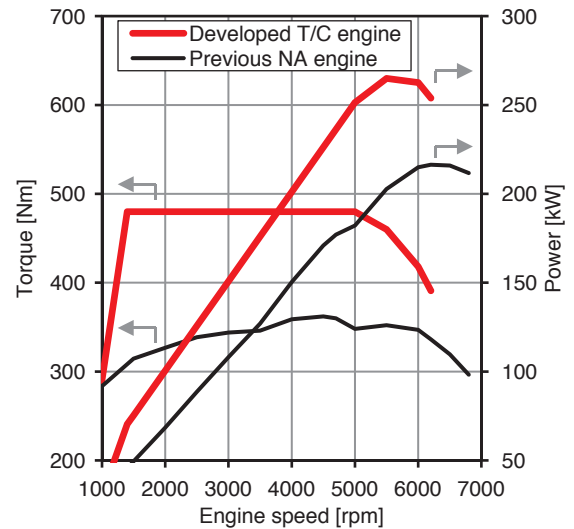
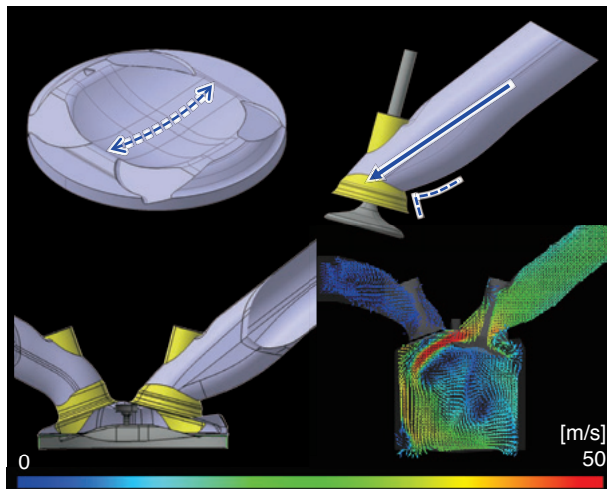


Fig. 12 Engine performance



(At 1500 rpm, wide open throttle (WOT), crank angle 470 deg)

Fig. 10 Intake-cylinder flow of T/C engine

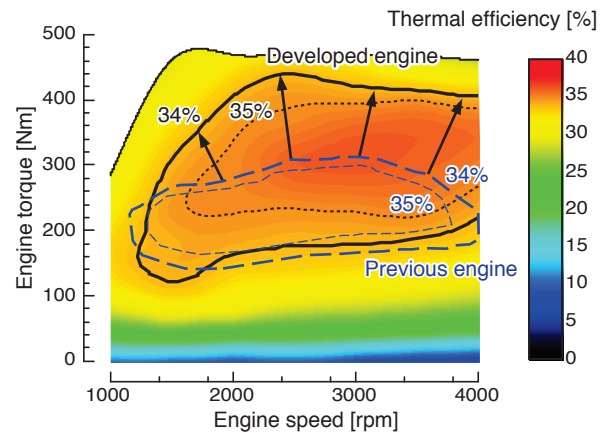


Fig. 13 Thermal efficiency map of developed T/C engine

7. The Developed NA Engine

For the developed NA engine, substantial reductions in unburned gas emissions and soot emissions were needed to achieve the LEV III SULEV30 regulation standards. Thus, the maximum fuel pressure for direct injection variable fuel pressure control was increased from 20 MPa in the previous engine to 30 MPa, and spray was atomized to reduce fuel adhesion in the cylinder. Also, the injector nozzle hole size was reduced while increasing fuel pressure to help ensure the same injection volume as the previous model, reducing the spray tip penetration (Fig. 14). Penetration was further reduced with multistage injection to reduce the continuous injection period (T_i).

Figure 15 shows the region in which multistage injection is applied. To help ensure stable combustion

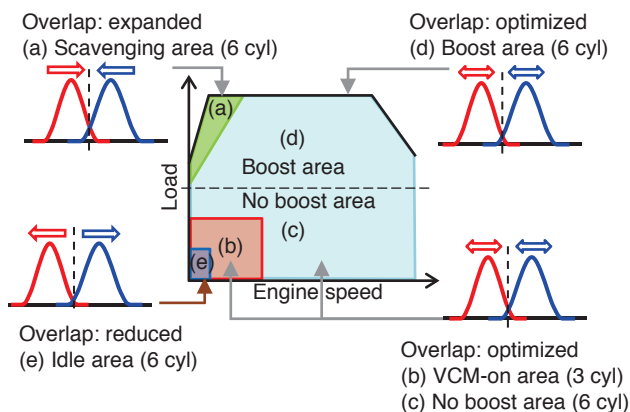


Fig. 11 Intake/Exhaust VTC control strategy

performance, the multistage injection conditions are changed for each driving condition. With these technologies, a vehicle with the NA engine achieved LEV III SULEV30 and PM of 1 mg/mile, enhancing emissions performance.

To improve thermal efficiency, the same as in the turbocharged engine, intake ports for increased tumble flow and a piston crown shape that maintains tumble flow were adopted to modify combustion by enhancing in-cylinder flow. Figure 16 shows the intake port, piston crown shape, and in-cylinder flow as calculated by CFD. The intake air flow rate was prevented from decreasing by using a larger sectional shape than the turbocharged engine, while enhancing flow with upright edges at the port bottom. In addition, the crown radius was designed to balance a high compression ratio with tumble retention.

An overview of the intake/exhaust VTC control

strategy is shown in Fig. 17. Valve overlap was optimized for all driving ranges. At low to mid rpms (a), thermal efficiency was increased by reducing pumping loss and optimizing internal EGR volume. By further combining with cylinder deactivation (b), thermal efficiency was increased for a wide range of driving conditions. At high rpms, the aim was to enhance knocking suppression through higher exhaust efficiency (c), and to increase power by optimizing valve overlap reductions.

Through these technologies, a maximum power of 213 kW and a maximum torque of 355 Nm were achieved, which are both at least as good as the previous engine. Finally, Fig. 18 shows the thermal efficiency map. Maximum thermal efficiency improved from 36.5% with the previous engine to 37.5%, and the area with a thermal efficiency of 35% or more was expanded.

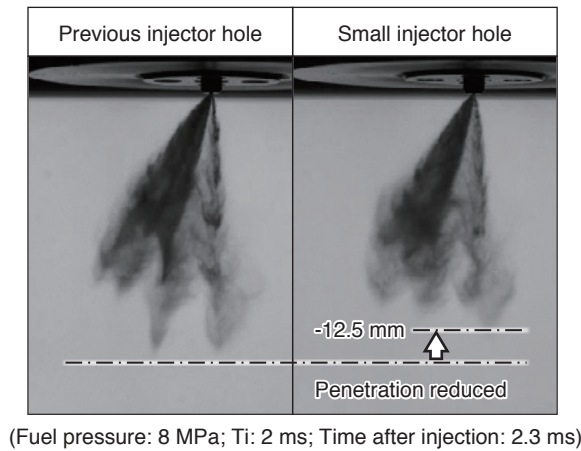


Fig. 14 Effect of small injector hole diameter

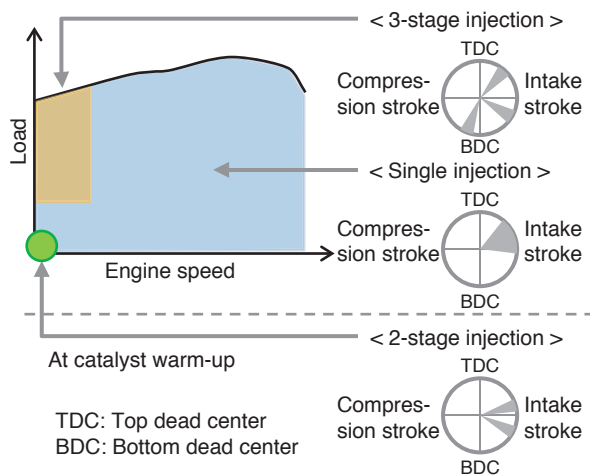


Fig. 15 Fuel injection operation

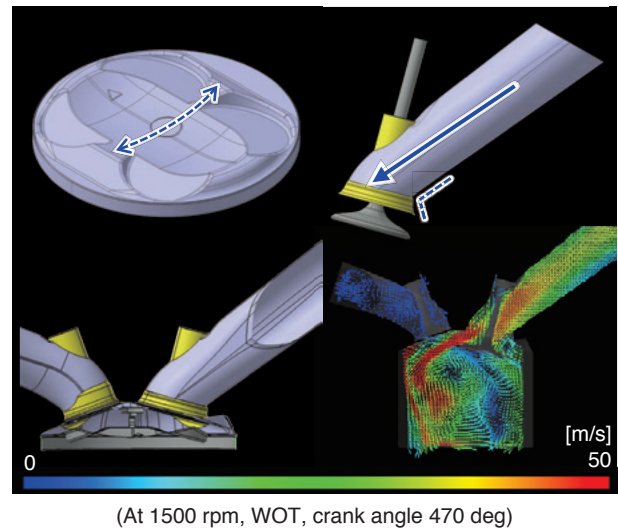


Fig. 16 In-cylinder flow of NA engine

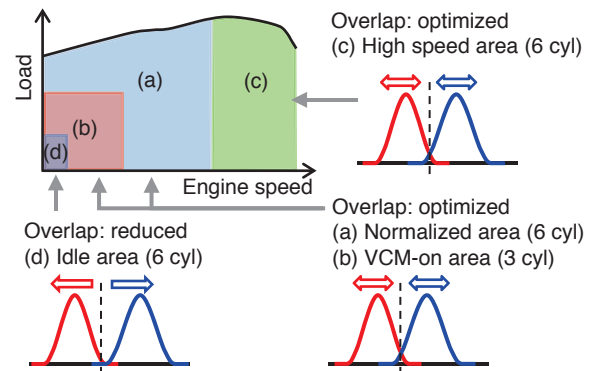


Fig. 17 Intake/Exhaust VTC control strategy

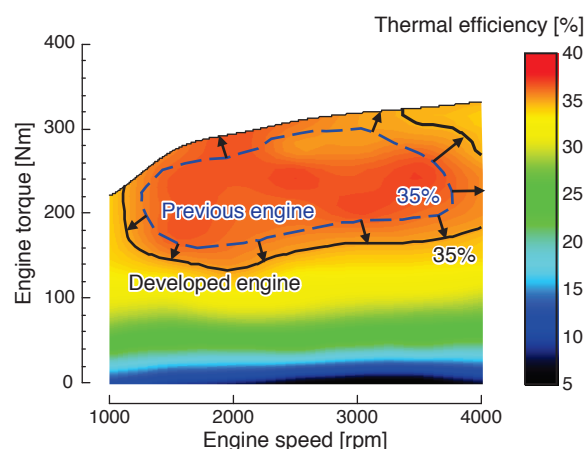


Fig. 18 Thermal efficiency map of developed NA engine

8. Conclusion

Turbocharged and NA direct injection V6 gasoline engines were developed with a new common valve train architecture, achieving the following results:

- (1) The new valve train has cylinder deactivation switching mechanisms built into the valve lifters, achieving a compact VCM system.
The engines combined this valve train with HLAs and roller rocker arms, also adopting DOHC and intake/exhaust VTC, and achieving a cylinder head size comparable to that of the previous engine.
Adopting HLAs both reduced variations in combustion performance due to reduced variations in the intake air flow rate and produced a maintenance-free valve train.
The engine geometry was made common between the turbocharged and NA engines, ensuring commonality for the timing belt drive system and other systems.
- (2) The turbocharged engine achieved a maximum power of 265 kW and maximum torque of 480 Nm, increasing power by 26.7% over the previous engine and torque by 35.2%. It achieved a maximum thermal efficiency of 36.5%.
- (3) The NA engine achieved a maximum power of 213 kW and maximum torque of 355 Nm. In terms of environmental performance, the maximum thermal efficiency improved from the previous engine's 36.5% to 37.5%. Also, a vehicle with the engine achieved LEV III SULEV30 and PM of 1 mg/mile.

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Technology for High-speed Manufacturing of High-pressure Hydrogen Tank by 14-axis Simultaneous Control Filament Winding Machine

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ABSTRACT

In the interest of reducing the cost of high-pressure hydrogen tanks, a 14-axis simultaneous control filament winding system was developed that shortens the filament winding process time and stabilizes tank pressure strength.

To accelerate the process, a mechanism capable of high-speed helical tumble winding was adopted, speeding up the process to approximately five times the conventional carbon fiber feeding speed. With the increased speed, a reservoir was implemented to keep the machine's band path distance constant at all times, allowing for reduced band tension fluctuations in winding. In addition, feed rollers capable of widening tow prepreg for feeding at a stable bandwidth were developed, along with a control method for the rollers, to help enhance carbon fiber reinforced plastic laminate quality. Upon studying the cost reduction effects of the increased speed, the machine was configured for dual head filament winding.

As a result, process time was reduced by approximately 80% compared with the conventional process, and the effectiveness of the machine was confirmed to reduce variance in pressure resistance for the high-pressure hydrogen tanks.

1. Introduction

The spread of fuel cell vehicles (FCVs) using hydrogen as fuel is expected to help in achieving a carbon-free society. An important component of FCVs is their high-pressure hydrogen tanks. For FCVs to spread in the future, it will be important to build manufacturing technologies that reduce costs with stable production quality in order to help guarantee the safety of high-pressure hydrogen tanks.

Vehicle high-pressure hydrogen tanks are dominated by 70-MPa type 4 vessels⁽¹⁾. As seen in the sectional view in Fig. 1, type 4 tanks comprise a plastic liner to seal in the enclosed hydrogen gas, a boss for device connection, and a multiple layer laminate of carbon fiber reinforced plastic (CFRP) to reinforce the outer surface of the liner. The CFRP layer is molded through filament winding (FW), a process that involves winding and then thermosetting a band of carbon fibers impregnated with an epoxy resin onto the resin liner. Given the high-pressure resistance and durability required of high-pressure hydrogen tanks, the liner is wound with dozens of layers of CFRP. The filament

winding process takes time, which greatly affects the cost of hydrogen tanks⁽²⁾.

In order to help enhance productivity, Honda has jointly developed a new high-speed multi-axis filament winding machine with high degrees of freedom together with Roth Composite Machinery. This machine is capable of world-

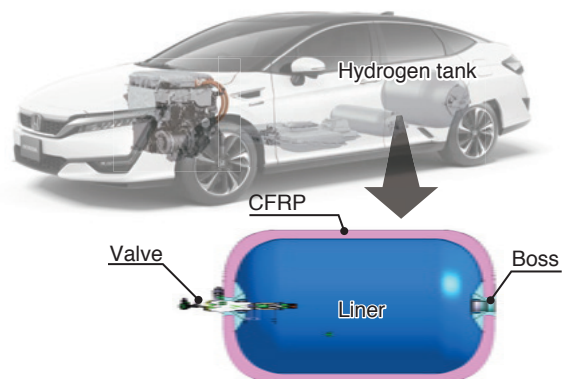


Fig. 1 Type 4 high-pressure hydrogen tank

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class filament winding speeds, reducing winding time by approximately 80% from the conventional process for filament winding machines. In addition, the process achieves a stable CFRP laminate quality, which can be an issue for high-speed filament winding. Therefore, the technology will be introduced here.

2. Development Goals for High-speed Filament Winding Machine

Figure 2 shows the methods used for winding the carbon fiber bands in filament winding for high-pressure hydrogen tanks. Hoop winding is used to wrap the cylinder to help ensure strength around the tank circumference, and helical winding is used to reinforce the tank dome. Helical winding is classified into high- and low-helical winding according to the winding angle. A currently popular filament winding machine is shown in Fig. 3. In low-helical winding, the band feeding speed fluctuates as the delivery head needs to reciprocate in the Y-axis direction of the filament winding machine in the figure. The band winding tension fluctuates due to acceleration and deceleration of the band, and large changes in tension will impact tank pressure strength. Thus, the inability to speed up low-helical winding is one issue in the filament winding process. As shown in Fig. 4, low-helical winding takes up approximately three-quarters of the total molding time.

Another issue in the filament winding process is with wet filament winding, in which the carbon fiber band is impregnated with epoxy resin before winding. Using this

method, the resin can be scattered when winding at high speed, or resin flow can occur in the CFRP layers when spinning the tank⁽³⁾. By contrast, high-speed winding is possible with dry filament winding, which uses tow prepreg, a tow (carbon fiber bundle) that comes preimpregnated with high-viscosity epoxy resin. However, to date, dry filament winding has presented issues in maintaining uniform tow prepreg bandwidth and thickness during tank winding, making it a challenge to achieve a uniform CFRP laminate quality. Where this has resulted in decreased pressure strength, measures such as winding more CFRP layers were necessary.

Thus, speeding up the filament winding process for high-pressure hydrogen tanks requires the following two technologies:

- (1) Faster low-helical winding with more stable tensioning
- (2) High-speed tow prepreg delivery and high-quality lamination

In response, the machine answers these two technical needs with the following three new technological developments:

- (1) A 14-axis simultaneous control high-speed filament winding machine capable of high-speed low-helical winding
- (2) Simultaneous multiple-filament winding technology for multiple tow prepreps, compatible with a new control high-speed filament winding machine
- (3) Bandwidth stabilization technology for tow prepreps

Each development will be described in order in the sections to follow.

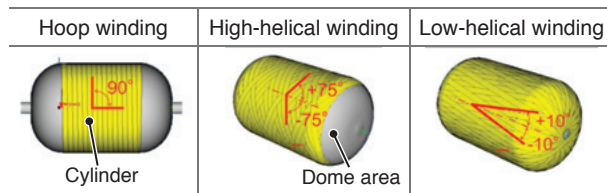


Fig. 2 Various filament winding methods

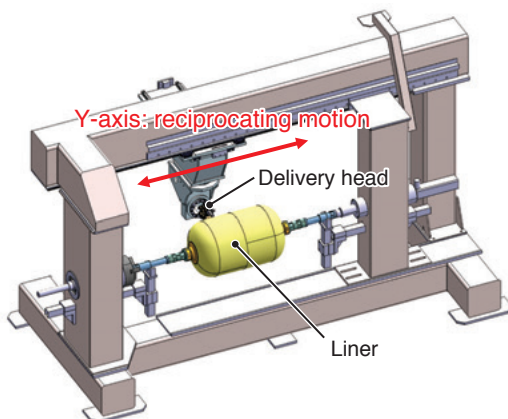


Fig. 3 Conventional FW machine

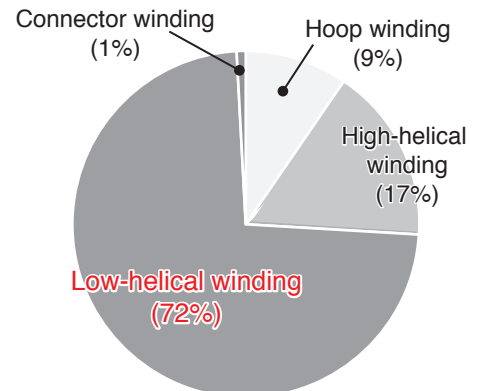


Fig. 4 Proportions of total FW time by winding method

3. Development of 14-axis Simultaneous Control Filament Winding Machine

3.1. Development of High-speed Helical Tumble Winding

3.1.1. Machine mechanism enabling tumble winding

The filament winding machine jointly developed with Roth Composite Machinery is shown in Fig. 5. The

machine has a high degree of freedom thanks to the 14-axis system outlined in Table 1. The machine has two creel stands for delivery from bobbin-shaped tow preregs, with tow prepreg fed from the stands along two band paths. Each band path is controlled by the system across 6 axes (Y-axis, Z-axis, U-axis, V-axis, W-axis, and I-axis).

The liner mounted to the filament winding machine adds a tumble axis (C-axis) to the rotation axis (X-axis) on conventional machines, allowing it to rotate on two axes. As shown in Fig. 6, low-helical winding allows for band winding while tumbled, with the band winding speed kept constant by maintaining a constant C-axis speed. However, even at a constant band winding speed, operating the delivery head will make the band path distance fluctuate, in turn fluctuating the band feed speed in the band path, as shown in Fig. 7. Fluctuations in the band feed speed may cause tension fluctuations, which impact tank pressure strength. Therefore, in order to keep tension constant throughout the filament winding process, the band path distance needs to be kept constant.

Table 1 Functions of each axis of FW machine

Axis	Function	Number of axes
C-axis	Mandrel tumble	1
X-axis	Mandrel drive	1
Y-axis	Horizontal carriage	2
Z-axis	Horizontal cross carriage	2
U-axis	Rotating pay-out eye	2
V-axis	Horizontal swivel axis	2
W-axis	Vertical carriage	2
I-axis	Reservoir	2

In the newly developed filament winding machine, reservoirs (I-axis) have been installed in between the tank winder and creel stands to maintain a constant band path distance (Fig. 8). This machine mechanism reduces band tension fluctuation, as shown in Fig. 9, allowing revolutionary high-speed helical tumble winding.

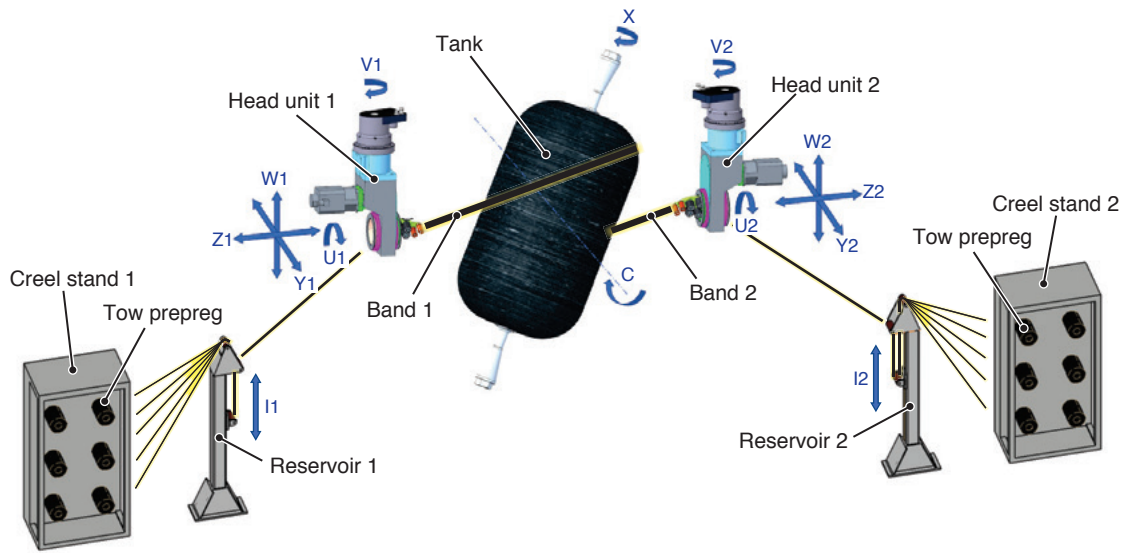


Fig. 5 14-axis simultaneous control FW machine

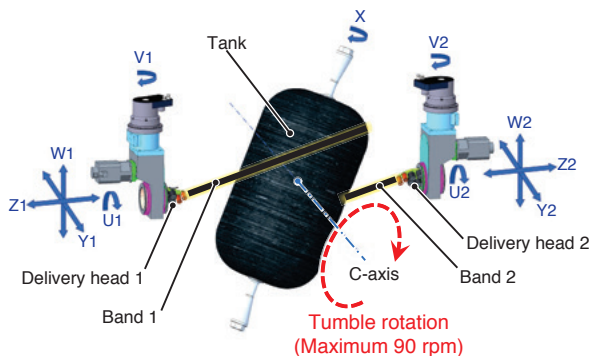


Fig. 6 High-speed helical tumble winding

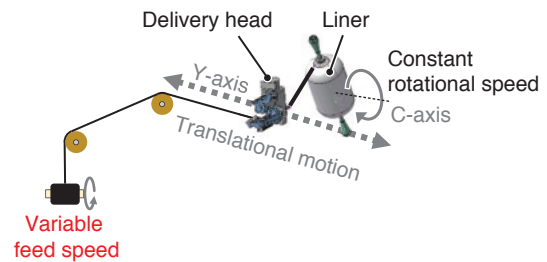


Fig. 7 Velocity change due to change in band path distance

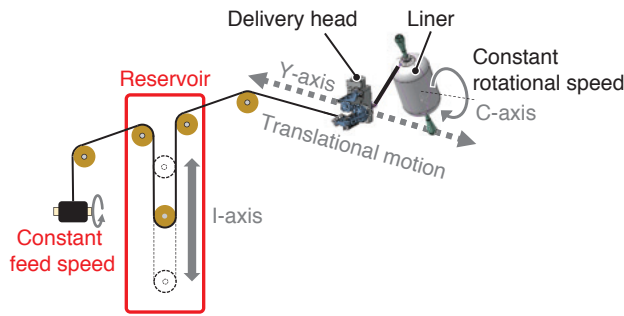


Fig. 8 Reservoir installed in FW machine

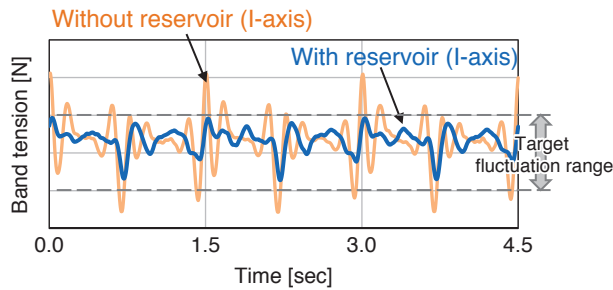


Fig. 9 Effect of reservoir on band tension stability

3.1.2. Reservoir control (I-axis) and higher winding speed

Helical winding in the machine is accelerated by high-speed rotation along the tumble axis (C-axis). The machine needs to feed the band along the band path at speeds corresponding to the liner rotational speed under high-speed tumbled rotation. The control method using the reservoir (I-axis) described above is important to achieving high-speed delivery at a stable tension. In the feedback control initially assumed, maintaining steady control was not easy. The band path distance fluctuated greatly at high speeds, and there was no control designed to respond to those fluctuations. Thus, an additional control was implemented to employ feed-forward control. A Honda proprietary computational algorithm estimates band path fluctuation to high precision, and I-axis motion is configured to correspond with the estimated fluctuation. With this, the fluctuations in tension were kept to no greater than the fluctuations for helical winding on a conventional filament winding machine for high-speed helical tumble winding at rotation speeds of up to 90 rpm. Band feeding speeds for helical winding on the developed filament winding machine are approximately five times conventional band feeding speeds.

3.2. Multi-filament Winding

3.2.1. Examination of number of delivery heads

To further reduce the time required for the filament winding process, a multi-filament process was examined. Simultaneous feed of bands from multiple locations can

shorten filament winding time. However, multi-filament winding creates constraints on the CFRP laminate design.

Multi-filament winding impacts CFRP laminate design as follows. In CFRP laminate design, the filament winding parameters considered are the number of cycles and coverage rate. The number of cycles is how many cycles the band is wound around the liner to form one layer. The coverage rate is the ratio of the total surface area of actual band as wrapped relative to the total surface area of the tank cylinder. If there are gaps between bands and the entire cylinder is not covered, the coverage rate will be less than 100%. Coverage rates of less than 100% decrease tank pressure strength. Therefore, in CFRP laminate design, the number of cycles needed to keep the coverage rate of each layer at 100% or higher is determined. In the interests of reducing CFRP weight and shortening filament winding process time, preferably, the minimum number of cycles to achieve a 100% coverage rate will be selected. As the number of cycles for multi-filament winding is multiplied by the number of band feeds, this limits the number of cycles that can be selected in the laminate design and can be an issue that the minimum number of cycles to achieve a 100% coverage rate cannot be selected. Thus, multi-filament winding can lead to increased CFRP weight.

With that, CFRP laminate designs were performed with between one and four band delivery heads, calculating the CFRP weight and process time for each case. The results are given in Fig. 10. In the calculations, the number of cycles needed to achieve a 100% coverage rate was determined for each number of delivery heads. With the CFRP weight of a single head as the baseline, CFRP weight increased by 2% with two heads and by 10% with four heads.

Figure 10 also shows the relationship between the number of delivery heads and time required for the filament winding process. The time required for filament winding was estimated for the CFRP weight at each number of delivery heads as calculated above, then calculating the rate of reduction in process time compared with a single head. It was found that process time could be reduced by 44% using two heads, and by 65% using four heads.

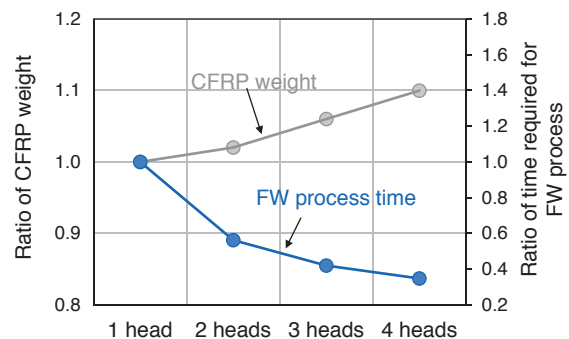


Fig. 10 Effect of the number of delivery heads on CFRP weight and FW process time

Based on the estimates of CFRP laminate weight and filament winding process time from using multi-filament winding, the impacts on hydrogen tank cost were examined. Figure 11 shows the cumulative estimated CFRP material costs and filament winding manufacturing costs per hydrogen tank for each number of delivery heads. At three or more delivery heads, the cost reductions from shortening filament winding time are outweighed by cost increases due to increased CFRP weight, meaning that cost is lowest at two heads. Therefore, the newly developed filament winding machine was configured for dual head filament winding.

3.2.2. Dual head filament winding machine enabled for tumbling

To perform high-speed helical tumble winding from dual bands, the two band delivery heads 1 and 2 need to be arranged on opposing sides and operated in point symmetry with the tank in the center, as shown in Fig. 6. This arrangement is effective in reducing head interference and band crossing during filament winding. With tumble winding speed five times faster than conventional speeds and the dual heads, the combination allows for high-speed helical winding at approximately 10 times the speed of conventional filament winding machines, and high-speed helical tumble winding is still stable even with the dual heads.

3.3. High-quality Lamination of Tow Prepregs

3.3.1. Tow prepreg widener mechanism

In the development of a 14-axis simultaneous control filament winding machine to enable high-speed filament winding, it is also important that the tow prepregs used for high-speed feeding are laminated at a high quality. High-quality lamination requires that the tow prepreg be widened before winding. However, the filament winding process in which the tow prepreg is used applies tension to the tow prepreg, making the filament bundle shrink in width. Comprising high-viscosity epoxy resin, bundles will keep their shape once shrunk, and thus widening a tow prepreg once shrunk is no simple task. For this reason, a tow prepreg

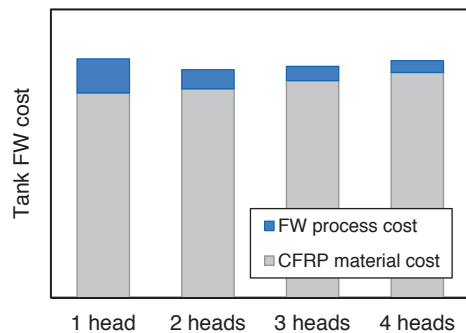


Fig. 11 CFRP material cost and FW process cost by number of delivery heads

widener mechanism with the following characteristics was installed in the band feed port of the filament winding machine⁽⁴⁾:

- (1) Heaters inserted in the rollers to heat the roller surface
- (2) Roller surfaces given multiple convex ridges at equal intervals in the axial direction
- (3) Multiple rollers equipped in the feed direction

Each of these characteristics is shown in Fig. 12.

First, in order to stabilize and widen the tow prepreg, the roller surfaces are heated to heat the tow prepreg as it contacts the rollers. This reduces the viscosity of the epoxy resin, lowering the holding strength of the filament. Multiple rollers are installed to help ensure more contact time with the rollers even with high-speed feeding and heat the tow prepreg to the required temperature. Next, surface pressure is applied to the entire tow prepreg by the axially convex roller surface to widen the prepreg. With the narrow interval between convex ridges on the rollers, the convex ridges are able to apply continuous surface pressure to the filament bundle before it shrinks in width, thereby suppressing band shrinkage in the width direction.

Using the widener mechanism as described here widened tow prepregs by more than 50% compared with their width before passing through the widener mechanism.

3.3.2. Stable control for tow prepreg bandwidth

In order to wind the expanded tow prepreg around the liner at a stable bandwidth without it shrinking, the position and attitude of the band feed port are controlled. If the band is twisted at a distance from the feed port to the liner winding point, the band will shrink. Thus, it is necessary to match the sectional attitude of the band at the liner winding point with the sectional posture at the feed port. Especially in helical winding, the band sectional attitude will change dynamically on the curved surface of the dome. To accommodate the dynamic changes of the curved surface, the band feed port requires two degrees of freedom for translational motion and three degrees of freedom for rotational motion. The band feed for this machine has a mechanism that allows motion in the above five degrees

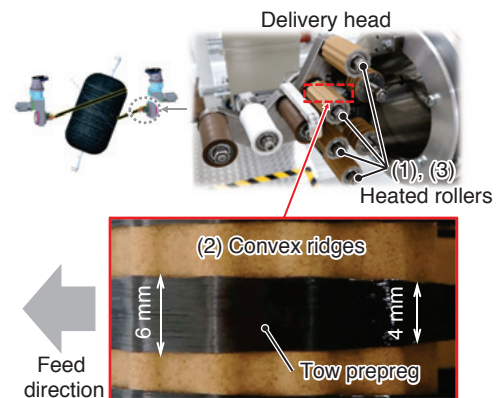


Fig. 12 Tow prepreg widener mechanism

of freedom. By controlling the position and attitude of the band feed port through the five degrees of freedom while winding the tow prepreg, the high-quality dome shown in Fig. 13 was successfully laminated at a stable bandwidth.

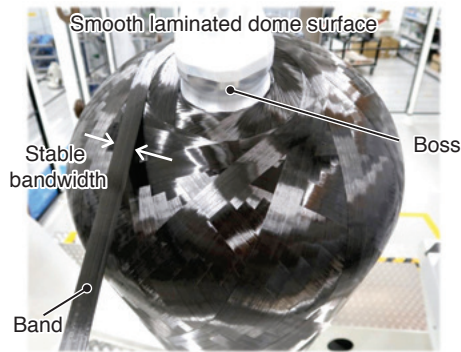


Fig. 13 Lamination quality at dome

4. Reductions in Filament Winding Process Time

Due to the high-speed helical tumble winding and dual heads, the time required for the filament winding process to manufacture a high-pressure hydrogen tank with the 14-axis simultaneous control filament winding machine was reduced by approximately 80% compared with a conventional filament winding machine. Table 2 shows the size, volume, and CFRP weight of filament-wound tanks and filament winding time with the new machine. Tank A is a high-pressure hydrogen tank shorter in length and larger in outer diameter, assuming installation behind the rear seatback of the FCV. Tank B is greater than 1 m in length with a smaller outer diameter, assuming applications for installation in the center tunnel of FCVs or for commercial FCVs. For both tank sizes, the CFRP filament winding speed was 1.69 kg/min.

5. Verification Results for Tank Pressure Strength

Table 3 gives a comparison of burst pressure stabilities between two 120-L tanks: the first manufactured by a conventional filament winding machine using wet filament winding, and the second manufactured by the developed high-speed filament winding machine using dry filament winding with tow prepreps. Stability was evaluated using the coefficient of variation (standard deviation/average) as an index. The same lot of carbon fiber was used to manufacture the two tanks.

By using filament winding tension-fluctuation control technology, a tow prepreg widener mechanism, and feeds with stable bandwidth, pressure strength (burst pressure) variation was reduced to one-fifth compared with conventional manufacturing methods.

Table 2 Results for high-speed FW production

	Tank A	Tank B
Tank length [mm]	915	1120
Tank diameter [mm]	530	435
Tank volume [L]	120	102
CFRP weight [kg]	48.9	42.2
FW process time [min]	29	25
Average FW speed [CFRP kg/min]	1.69 (0.26)	1.69 (0.31)

() : Conventional Wet-FW speed

Table 3 Results of burst pressure stability of 120 L tanks

	Conventional FW (Wet-FW)	High-speed FW (Dry-FW)
Coefficient of variation of burst pressure (standard deviation / average)	0.059	0.012

6. Conclusions

In this paper, we described a 14-axis simultaneous control filament winding machine enabling high-speed helical tumble winding with dual delivery heads in the interest of achieving high-speed filament winding manufacturing and high-quality lamination for high-pressure hydrogen tanks.

- (1) We introduced the structure of a newly developed 14-axis simultaneous control filament winding machine capable of high-speed helical tumble winding with dual delivery heads.
- (2) High-speed filament winding tension-stabilization technology, tow prepreg widening technology, and band feed port control technology were applied to the machine for stable lamination.
- (3) The machine reduced the time required for the filament winding process by approximately 80%, achieving world-class filament winding speeds.
- (4) This development also confirmed that variances in tank pressure strength due to manufacturing impacts could be reduced to one-fifth the conventional values.

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Modeling of Efficient Lifetime Prediction for Lithium-ion Battery Employing Orthogonal Array

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ABSTRACT

Estimation of lithium-ion battery durability by modeling that simulates performance degradation is an important technology during the initial phase of development of electrified products. For this study, a low-cost, highly efficient technique for creating a model was investigated by applying the orthogonal array used in quality engineering to the process of creating a performance degradation prediction model. As a result, it was found that for the storage durability test, it was productive to apply an experimental design that uses an L16 orthogonal array, whereas for the cycle durability test, it was productive to apply an experimental design that uses an L27 orthogonal array. It was also found that by fitting to the test results, when the storage degradation coefficient is modeled by expressing it with the state of charge and temperature in a quadratic linear equation together with an interaction term, and the cycle degradation coefficient is modeled by expressing it with the state of charge, window of operating state of charge, temperature, charge current, and discharge current in a first-order linear equation, then it was possible to simulate the results of accelerated durability testing highly accurately with a deviation of 1.3%. The method of modeling discussed in this paper reduced the amount of testing necessary in the conventional, factorial experiment method by 58% while realizing an equivalent level of accuracy, confirming that enhancement of experiment efficiency can be expected.

1. Introduction

Since lithium-ion batteries (LiBs) have high energy density and power density, they are coming into wide use not only in compact electronic equipment, but also in the large-scale mobility of motorcycles, automobiles, and aircraft⁽¹⁾⁻⁽⁴⁾. Durability in LiBs for electric vehicles such as hybrid electric vehicles (HEVs), battery electric vehicles (BEVs), and so on, has been sought over a period of 10 years or more, and various regions and patterns of use also have to be taken into consideration. It is necessary, therefore, to confirm by means of prediction using a model simulating performance degradation (hereinafter, degradation prediction model) whether the durability satisfies development objectives in the initial phase of development^{(5), (6)}. The rate of LiB degradation depends on the temperature and state of charge (SOC) of a battery in storage, the current while charging, the current while discharging, the magnitude of changes in the SOC, and other such factors. Consequently, the rate of degradation of a battery changes according to the environment where the electric vehicle is used and how it is used. In general, when a user drives frequently in a region with high temperatures

and repeatedly accelerates and brakes suddenly, then the battery in that user's electric vehicle will undergo degradation sooner than one in the vehicle of a user in a region with low temperatures who does not drive frequently and who accelerates and brakes gently. A degradation prediction model needs to have functions that work out the rate of degradation to match the various ways in which such electric vehicles are used and calculate cumulatively the amount of degradation per unit of time⁽⁷⁾. At Honda up to now, a semi-physical model has been created that very accurately calculates performance degradation behavior by predicting the internal state of a battery^{(8), (9)}. Going forward, there will be a need for efficient creation of highly accurate prediction models to meet the further increasing need for electric vehicle development. In order to efficiently create prediction models, it is necessary to set conditions for testing whether they satisfy development objects and to investigate optimal methods for creating models.

Figure 1 shows a simplified view of the flow of the simulation of degradation behavior in the repetition of park-drive-charge modes implemented by Honda. The LiB performance is expressed by its capacity retention, which diminishes as the battery is used. The rates of degradation

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corresponding to the individual park-drive-charge modes of use are calculated from a degradation coefficient map that is created in advance by fitting with the training data from durability tests. The calculated rate of degradation and the time period during the mode ($\Delta time$) are multiplied together to calculate the amount of degradation during each mode. These are added together to predict the amount of degradation. In order to predict degradation with high accuracy, it is necessary to account for a large number of conditions in a way that will include usage by users and to conduct long-term tests. However, durability tests of on-board batteries involve a charge-discharge machine for charging and discharging the batteries, a thermostatic chamber, an evaluation device for reading electrical properties, the expense of personnel to conduct the durability tests, and so on. As these are needed for extended periods of time, there is a large cost burden. In this study, therefore, an experimental design based on orthogonal arrays was applied in creating an LiB model, the number of tests was reduced, and a highly accurate method of modeling was realized.

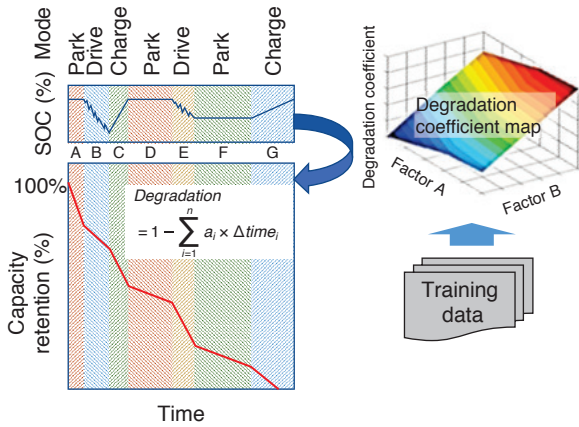


Fig. 1 Concept of calculating performance degradation

2. Current Issues and Proposals

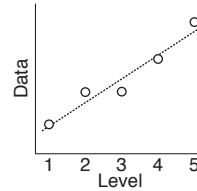
Up to now, Honda has been conducting durability tests with 100 or more load conditions over a period of half a year or more in order to create degradation prediction models for on-board LiBs. The tests have been conducted with a focus on battery load using two types of tests: storage mode (parked) and cycle mode (driving and charging). The test conditions have been set according to fractional factorial experiment to five levels or more for each degradation factor. Figure 2 shows an example. For each factor (from A to E), five levels are set. In this method, factors other than the focal factors are fixed (the gray cells in Fig. 2(b)), and only the focal factors are changed one level at a time, thereby obtaining test data on factorial effect

that can be analyzed. The advantage is that the effect due to the change of a single factor can be obtained directly so that the readability of the data is high and the data can readily be understood intuitively. For example, plotting the test data items 1 to 5 from Fig. 2(b) in the factorial effect diagram in Fig. 2(c) makes it possible to readily grasp that there is a steady upward trend. The disadvantage, on the other hand, is that the various test levels are concentrated on a certain level, and as a result there is a concern that imbalance between levels may occur when viewing the data as a whole. The imbalance of levels tends to diminish the robustness of the prediction model. In addition, when deriving the factorial effect, one data item is extracted per level in order to grasp the dependency of each factor, and thus there is a susceptibility to being affected by variation and noise. In order to address this issue, the fixed levels were changed and two to three sets of fractional factorial experiments were conducted. This resulted in the issue, however, that the number of durability test conditions became large, ranging from 65 to 100 conditions, and the testing cost became high.

In this study, these issues were addressed by applying an orthogonal design. Figure 3 shows basic experimental designs created with two levels of each factor. Three

	Level				
	1	2	3	4	5
Factor	A	A1	A2	A3	A4
	B	B1	B2	B3	B4
	C	C1	C2	C3	C4
	D	D1	D2	D3	D4
	E	E1	E2	E3	E4

(a) Level / Factor matrix



(c) Factorial plot

Experiment No.	Factor				
	A	B	C	D	E
1	A1	B1	C1	D1	E1
2	A2	B1	C1	D1	E1
3	A3	B1	C1	D1	E1
4	A4	B1	C1	D1	E1
5	A5	B1	C1	D1	E1
6	A1	B2	C1	D1	E1
7	A1	B3	C1	D1	E1
8	A1	B4	C1	D1	E1
9	A1	B5	C1	D1	E1
*	A3	B3	C3	D3	E3
*	A3	B3	C3	D3	E4
*	A3	B3	C3	D3	E5

Note: Grey cell is unchanged from cell above it

(b) Test design

Fig. 2 Conventional testing method

Experiment No.	Factor				
	A	B	C	D	E
1	1	1	1	1	1
2	2	1	1	1	1
3	1	2	1	1	1
4	2	2	1	1	1
5	1	1	2	1	1
6	2	1	2	1	1
29	1	1	2	2	2
30	2	1	2	2	2
31	1	2	2	2	2
32	2	2	2	2	2

Note: Colored cells show level 2

(a) Full factorial experiment

Experiment No.	Factor				
	A	B	C	D	E
1	1	1	1	1	1
2	2	1	1	1	1
3	1	2	1	1	1
4	1	1	2	1	1
5	1	1	1	2	1
6	1	1	1	1	2

(b) Fractional factorial experiment

Experiment No.	Factor				
	A	B	C	D	E
1	1	1	1	1	1
2	1	1	1	2	2
3	1	2	2	1	1
4	1	2	2	2	2
5	2	1	2	1	2
6	2	1	2	2	1
7	2	2	1	1	2
8	2	2	1	2	1

(c) Orthogonal array design

Fig. 3 Variations of basic designs of experiments

designs are shown. The full factorial experiment tests the combinations of each factor at all the levels. The fractional factorial experiment tests a portion of the factorial effect. The orthogonal array design tests the factorial effect on the basis of an orthogonal array.

In an orthogonal array design, each level is included in the test conditions the same four times in each factor, and the combinations of factors are orthogonal. Because of this, the factor characteristics of each level can be calculated as the average of the characteristic values of the four tests. The fact that the data can be averaged and handled is a link with factorial effect analysis, which has little variation. It also becomes possible to have combinations that are balanced for each level, which increases the model's robustness. In addition, it is possible to greatly reduce the number of test conditions compared to the full factorial experiment.

3. Process Development

In obtaining efficient model training data, it is desirable to acquire highly reliable data without reducing model prediction accuracy. Here, model prediction accuracy means the degree of divergence between training data and other test data used for verification. Highly reliable data means data that is less influenced by noise. In order to obtain data for creating an LiB degradation prediction model using the orthogonal array described above, data acquisition was carried out with attention to the following matters.

3.1. Orthogonal Array Size Optimization

An orthogonal array is denoted by the letter L combined with a number indicating the number of experiments, as in L8 and L16. There are many orders of orthogonal array, such as L8, L9, L16, L18, L25, L27, L32, L36, and so on. When the size of an orthogonal array is decided, then the number of levels and the number of factors that can be evaluated are automatically limited. For example, if an orthogonal array is an L9, then it will be possible to evaluate at most four factors on three levels. When there is visible interaction between factors, then the number of factors that can be evaluated is further reduced. Out of these kinds of constraints, it is necessary to take into consideration the factors that should be evaluated and the number of levels that should be evaluated when selecting an orthogonal array. There are two main types of LiB modes of use: storage mode and cycle mode. The factors that determine the magnitude of their respective loads are, for storage mode, the two factors of temperature and SOC, whereas for cycle mode, the load is determined by the five factors of temperature, SOC, Δ SOC, charge current (I-charge), and discharge current (I-discharge) (Fig. 4)⁽⁹⁾. In deciding the number of levels, it is necessary to take into consideration the linearity of the characteristic values for each factor. If the factorial effect is linear, then it is possible to approximate a straight line from the test results for two levels. If it is non-linear, however, it will be necessary to set

at least three levels and to model the results by the response surface method. For this study, the optimal orthogonal array was configured from the analysis of on-board LiB durability test results.

3.2. Optimization of the Number of Levels

When setting the levels, it is necessary to take into consideration two things: the feasibility of the test and the level extrapolation risk. In experimental design based on an orthogonal array, the occurrence of incomplete testing and missing data undermines the reliability of the data as a whole. Therefore it is necessary to set test levels that are realizable. As tests that are set with conditions at the usage limit value for multiple factors may sometimes turn out to be not possible to execute, these were avoided. However, when the range of levels set is conspicuously narrower than the usage range, it is necessary to calculate the degradation coefficient from outside the range of levels set by extrapolation, and this can become a cause of increased prediction deviation. Levels were therefore set to the largest possible range while keeping the balance among factors and staying within the range of possible test execution.

3.3. Model Formula Development

For the model formula, it is necessary to select the model formula that is best capable of reflecting the training data and that is capable of making accurate predictions with respect to verification data that is separate from the training data. In the formulas formulated as candidates for the model, the number of terms and the order were controlled and multiple models with differing degrees of complexity were prepared.

3.4. Model Formula Selection

In selecting the model formula, it is necessary to make the determination from the coefficient of determination R^2 for the training data and the simulation deviation with respect to verification data. A model formula that is complex, with an order and coefficients that are large, will fit well with the training data, but a state of overfitting may

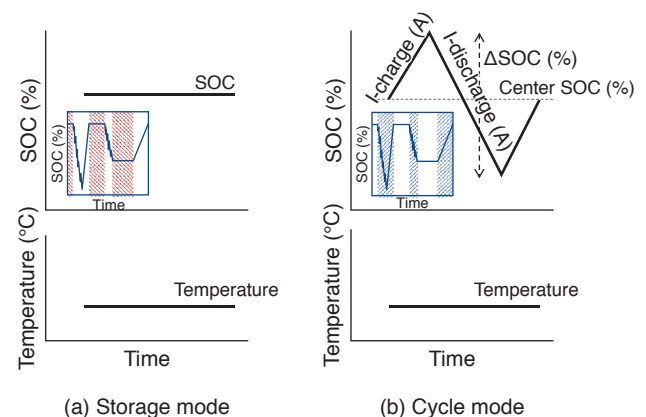


Fig. 4 Factors of storage and cycle modes

sometimes occur when accuracy is reduced with respect to conditions other than the training data, such as verification conditions. For this reason, multiple model formulas were planned based on the LiB degradation characteristics and with differing degrees of complexity. From among those, the model formula with the highest accuracy was selected using the coefficient of determination and simulation deviation as indices.

4. Process Investigation Results

Durability tests were conducted under many conditions and the test results were analyzed in order to investigate methods suited to the creation of LiB degradation prediction models. The tests used LiBs of the 5-Ah class for hybrid use as samples. Durability tests were conducted over a six-month period by factorial experiment with 95 conditions in all for both storage durability tests and cycle durability tests. The rate of degradation under each condition was calculated using a general model formula, Eq. (1)⁽¹⁰⁾, that expresses LiB durability behavior, and the parameters of the model formula were identified by the least squares method.

$$\text{Capacity retention} = a \times \text{Time}^{0.5} + 1 \quad (1)$$

$$\begin{aligned} \text{Degradation coefficient} &= a_{11} \times \text{SOC}^2 + a_1 \times \text{SOC} \\ &+ a_{22} \times \text{Temperature}^2 \\ &+ a_2 \times \text{Temperature} \\ &+ a_{12} \times \text{SOC} \times \text{Temperature} \\ &+ a_0 \end{aligned} \quad (2)$$

4.1. Orthogonal Array Selection

Figure 5(a) and (b) show factorial effect diagrams that have the degradation coefficient a calculated from the results of each test plotted against each factor. Figure 5(c) shows the response surface obtained on the basis of a quadratic model formula, Eq. (2). The storage durability tests indicated the non-linearity of the SOC dependency and temperature dependency, which meant that it would be necessary to select an orthogonal array allowing selection of three levels or more. Also, the fact that the slope of the SOC dependency changed according to the temperature suggested that there was an interaction between the SOC and the temperature. These results pointed to an orthogonal array having three levels or more for two factors and allowing evaluation of interaction, and as such it was found to be desirable to use a three-level L9 orthogonal array and an L16 that transforms a two-level orthogonal array into four levels.

Figure 6 shows the factorial effect analysis results from the cycle durability tests. The dependencies show a steady upward trend with respect to five factors, and non-linear behavior was observed in three of those five factors,

namely temperature, ΔSOC , and I-charge. Also, with regard to the four factorial effects other than SOC, slopes that differed according to the SOC were observed when factorial effects under differing SOC were evaluated. This can be considered to occur because there are interactions in

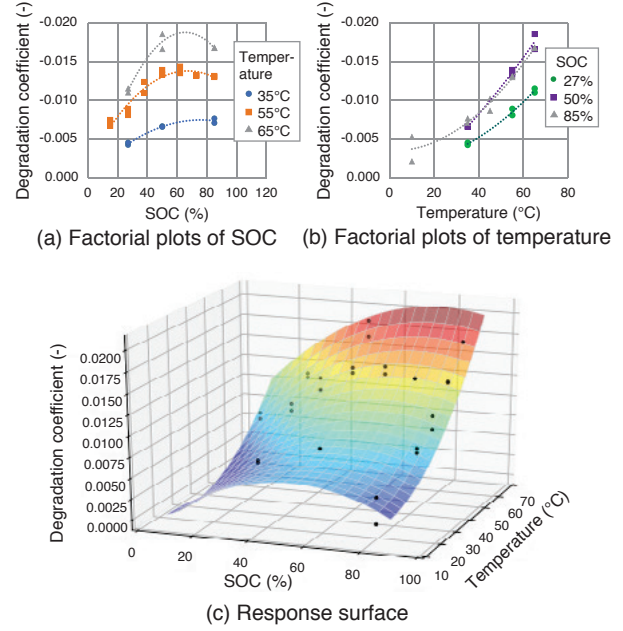
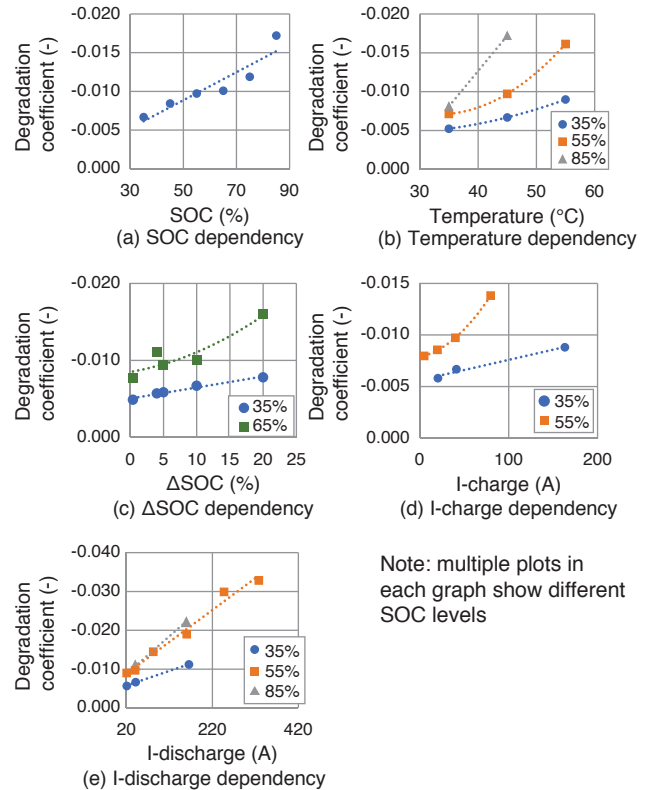


Fig. 5 Factorial plots and response surface of two variables in storage mode



Note: multiple plots in each graph show different SOC levels

Fig. 6 Factorial plots of five factors in cycle patterns

each factorial effect. It was found from these results that an L27 that enables factor analysis of three levels or more and that enables evaluation of four types of interaction at most would be appropriate for the cycle durability test. The total number of tests necessary for the storage durability test (L16) and the cycle durability test (L27) comes out to the sum of 16 and 27, or 43 conditions.

4.2. Selection of Levels

When setting the levels for each factor, it is necessary to consider factors that are completely independent and factors that are not independent separately. Factors of SOC and temperature in the storage durability test are independent. In the cycle durability test, on the other hand, I-charge and I-discharge, and center SOC and Δ SOC, are not independent because the sums of the values of each pair are constrained. Figure 7 shows those constraints. The range of values that can be taken by I-discharge and I-charge is at or below the upper limit on current value in the battery specifications, and it is necessary to conduct tests at or below the effective current value (A-rms limit) at which cell heat generation does not become an issue [Fig. 7(a)]. When the amount of heat generated in the cell is large, battery temperature cannot be readily controlled. In order to conduct tests safely, levels were also set so that the range of the sum of the center SOC and Δ SOC would be such that it did not cause the over-discharge phenomenon and lithium plating phenomenon even when overvoltage is taken into account [Fig. 7(b)]. For conditions that have factor values outside the level setting range, it becomes necessary to determine the degradation coefficient by extrapolation, and thus levels were selected so that the highest level values could be chosen with all the variables kept in balance.

4.3. Planning the Model Formula

Multiple model formulas were created while adjusting their complexity by the order of the factorial effect and the number of terms in the model formula. With regard to the order, first-order linear and quadratic linear versions were prepared. With regard to the number of terms, interaction terms were added and reduction of the number of terms was carried out by analysis of variance. Six types of model formula were planned, as shown in Table 1. With regard to

Table 1 List of model formulas

No.	Model	Formula	Number of coefficients
(i)	Linear	$\sum_{i=1}^5 a_i \times X_i + a_0$	6
(ii)	Linear + interaction	$\sum_{i=1}^5 a_i \times X_i + \sum_{i=2}^5 a_{1,i} \times X_1 \times X_i + a_0$	10
(iii)	Quadratic	$\sum_{i=1}^5 a_{i,i} \times X_i^2 + \sum_{i=1}^5 a_i \times X_i + a_0$	11
(iv)	Quadratic + interaction	$\sum_{i=2}^5 a_{i,i} \times X_i^2 + \sum_{i=1}^5 a_i \times X_i + \sum_{i=2}^5 a_{1,i} \times X_1 \times X_i + a_0$	21
(v)	Linear + interaction (ANOVA)	$\sum_{i=1}^5 a_i \times X_i + a_{1,3} \times X_1 \times X_3 + a_0$	7
(vi)	Quadratic + interaction (ANOVA)	$\sum_{i=2}^5 a_{i,i} \times X_i^2 + \sum_{i=1}^5 a_i \times X_i + a_{1,3} \times X_1 \times X_3 + a_0$	12

X_1 : SOC; X_2 : Temperature; X_3 : Δ SOC; X_4 : I-charge; X_5 : I-discharge

Table 2 Analysis of variance of L27 cycle mode test

Factor	Degree of freedom	Sum of squares	Mean square	F-value	p-value	Statistical significance
Center SOC	2	7.5E-05	3.7E-05	4.7E+01	0.002	*
Temperature	2	1.2E-04	5.8E-05	7.3E+01	0.001	*
Δ SOC	2	2.1E-04	1.1E-04	1.3E+02	0.000	*
I-charge	2	1.1E-05	5.5E-06	7.0E+00	0.049	*
I-discharge	2	5.0E-05	2.5E-05	3.2E+01	0.004	*
Center SOC x I-charge	4	1.0E-06	2.5E-07	3.2E-01	0.853	
Center SOC x Δ SOC	4	1.8E-05	4.5E-06	5.6E+00	0.061	*
Center SOC x Temperature	4	2.8E-06	7.0E-07	8.9E-01	0.544	
Errors	4	3.2E-06	7.9E-07	-	-	
Total	26	4.9E-04	2.2E-05	-	-	

*Lower than the alpha level of 0.10

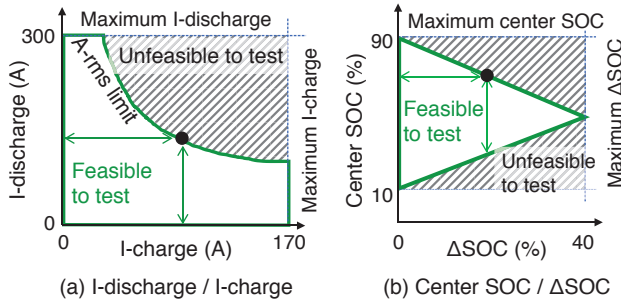


Fig. 7 Trade-off factors in cycle mode

the degradation factor, linear Models (i)(ii)(v) with factorial effects that are in first-order linear form were formulated. Those to which simple linear form did not apply were formulated as quadratic linear Models (iii)(iv)(vi), and analysis-of-variance (ANOVA) Models (v)(vi) made up only of statistically significant terms were formulated. In order to select the statistically significant terms, the results of tests conducted using the orthogonal array were analyzed by analysis of variance, as shown in Table 2, and the statistically significant terms (those factors in Table 2 marked with asterisks) were identified by the F-test. The F-test is a method for determining whether or not there is a significant difference in the variance between two factors, that is, the variation. In the analysis of variance, significant factors can be identified by conducting F-tests of the variance from each factorial effect and of the variance from the deviation factors. Since factors determined not to be significant disrupt the accuracy of the model, they are excluded as much as possible. In this study, six factors out of the five principal factors and four interactions were determined to be statistically significant.

Table 3 Simulation results of six models

	Model	Number of coefficients	Coefficient of determination (R^2) vs training data	RMSE (%) vs test data		
				Simple pattern	Driving pattern	Average
(i)	Linear	6	0.92	2.2%	1.1%	1.6%
(ii)	Linear + interaction	10	0.95	6.4%	1.0%	3.7%
(iii)	Quadratic	11	0.93	3.3%	2.1%	2.7%
(iv)	Quadratic + interaction	15	0.97	6.7%	2.8%	4.7%
(v)	Linear + interaction (ANOVA)	7	0.94	3.0%	1.6%	2.3%
(vi)	Quadratic + interaction (ANOVA)	12	0.98	7.4%	1.4%	4.4%

4.4. Model Formula Selection

The model formulas were selected from among the six types of model formula in Table 1 by means of the coefficient of determination R^2 of the training data and the simulation deviation (root mean square error: RMSE) from the verification data. Verification test results different from the training data were prepared for use in selecting the model formulas, and those formulas with higher prediction accuracy were selected. It is necessary to prepare multiple test data as much as possible and to select the model formulas comprehensively. In this study, simple patterns that combine multiple training test conditions and complex actual driving patterns were set as test conditions, and the data from durability tests conducted over the same period (four months) as the training tests were used as data for verification. The coefficient of determination of the training data for each model and the deviation from the test data used for verification are shown in Table 3. Of these, Model (i) was successfully fitted to the training data with a coefficient of determination of 0.9 or higher, and its prediction has the lowest deviation from the test data used for verification. Therefore it was determined that the linear model of Model (i) was good. The reason that the linear Model (i) had the best accuracy this time is thought to be that it was a simple model formula that had fewer coefficients than the other Models (ii) through (vi) and thus was more robust in its predictions of patterns that differed from the training data. As data that supports this reasoning, the results from the analysis of variance (Table 2) described above show that the terms included in the linear models (SOC, temperature, Δ SOC, I-charge, I-discharge) have low p-values in all eight terms. Here the p-value indicates the probability from 0 to 1 that there will be significance in the variance values based on the variance ratio of each factor and the deviation factors and the degree of freedom. The closer the value is to 0, the greater the variance by comparison to the deviation factors, which means it is significant. The Model (i) formula is considered to have enhanced prediction accuracy because the formula is made up of statistically significant terms.

5. Degradation Prediction Results

Cycle mode data was obtained using the L27 orthogonal array with 27 tests, and Fig. 8 shows the results from prediction of accelerated durability testing at the high temperature of 45°C using the optimized model. The accelerated durability testing simulated the waveform applied to the battery cells during actual driving. The result was an RMSE of 1.3% for the deviation between the prediction results and the measured values.

Meanwhile, durability tests were also conducted not using the orthogonal array but rather the conventional method of fractional factorial experiment that was conducted with 65 conditions. The model was created with the number of conditions used for modeling changed from 18 conditions to 65 conditions, and Fig. 9 shows the accuracy achieved in prediction of accelerated durability testing as plotted against the number of conditions. It is apparent that in the factorial experiment, the deviation decreased as the number of conditions used in modeling increased, indicating that prediction accuracy has been enhanced. The prediction deviation when 65 conditions were used was 1.4%, whereas the prediction deviation with the model created using the orthogonal array with

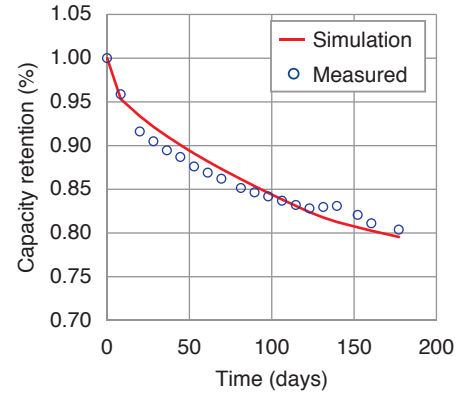


Fig. 8 Simulation result of driving pattern

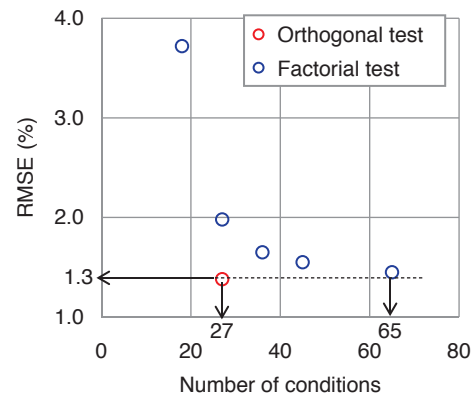


Fig. 9 RMSE of various number of conditions

27 tests was 1.3%. Therefore this confirmed that both approaches yield largely the same values. The model using the orthogonal design had prediction accuracy equivalent to the conventional method while reducing the number of conditions by 58%.

6. Conclusion

In creating an LiB performance degradation prediction model, the process of acquiring training data based on an orthogonal array and the method of modeling were investigated. The results yielded the findings below.

- (1) The orthogonal arrays suited to acquiring the training data necessary in creating an LiB degradation model are L16 (storage durability test) and L27 (cycle durability test).
- (2) For the battery degradation model formulas used in this study, the quadratic linear model with interaction is optimal for storage degradation and the first-order linear model is optimal for cycle degradation.
- (3) Implementing test design using the orthogonal array yielded prediction results that maintained the same 1.3% accuracy while reducing the number of conditions by 58%.

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Development of Multi-disciplinary Optimization Design Assist System for Gas Turbine Rotor Blades

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ABSTRACT

An optimization design assist system was developed for gas turbine rotor blade design that integrates an optimization algorithm into a workflow automating blade shape creation and aerodynamic and structural analysis. In recent years, thinner blades and higher loads have been used to enhance aerodynamic performance. In view of the impact of increased blade shape design sensitivity on structural mechanics, the system was designed to enhance structural mechanics evaluation functions. This paper presents an overview of the system and example cases of its application to the design of a centrifugal compressor rotor blade and a radial turbine rotor blade. By introducing the system, engineers were able to consider a wide variety of design elements in aerodynamic and structural design, as well as to propose suitable solutions that simultaneously satisfy both aerodynamic and structural design goals and to present quantitative design prospects.

1. Introduction

Gas turbine engines are designed to have a finite life. Particularly, rotor blade components, which see use in environments under large centrifugal forces, pressure, and thermal stresses, require part inspection and replacement at operating times predetermined by the designers. Extending part life can reduce part replacement costs and increase marketability.

As fuel economy performance also contributes to increase marketability, the designers provide the maximum stress and weight of the components, the specific vibration mode frequencies of the rotor blade, and other necessary structural constraints, and pursue longer component life and higher aerodynamic performance.

Given the nature of gas turbine engines, which start with the compression of intake air and then extract energy from the combustion gas using a turbine, the compressor and turbine rotor blades exert a major impact on engine fuel economy performance. However, sufficient structural strength needs to be ensured under the harsh operating environment.

In the design of rotor blades to date, aerodynamic design and structural design have been performed in successive order. If targets are not met, both are repeated in succession until satisfied.

In recent years, however, there has been a trend toward thinner rotor blades and higher loads to increase

aerodynamic performance^{(1), (2)}, increasing the impact of blade shape changes on the structural strength compared to conventional design. Thus, more care is required in the detailed design to help ensure the prerequisite structural strength. For these reasons, a front-loading design method that considers both aerodynamics and structural strength at the initial design stage is needed.

We have therefore created a workflow with the aim of designing gas turbine rotor blades that automates the process from blade shape definition through to analysis and, in combination with an optimization algorithm, developed a design assist system that considers aerodynamics and structural mechanics concurrently.

This paper first gives an overview of the system, and then explains the structural analysis module in detail. In many cases, general turbomachinery optimization calculations only consider aerodynamic performance, or only examine relatively simple evaluation metrics for the structural design, such as aerodynamic performance and disk center stress⁽³⁾⁻⁽⁶⁾. In this system, the structural evaluation function was enhanced to first satisfy the necessary structural constraints and then to increase product competitiveness by increasing aerodynamic performance and service life. Finally, example cases are presented of the system being applied in design of centrifugal compressors and radial turbine rotor blades which are widely used in turbomachinery.

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2. Composition of the Multi-disciplinary Design Assist System

The overall system is as shown in Fig. 1. The system searches for blades by simultaneously analyzing aerodynamics and structural mechanics and evaluating the results with the optimization algorithm.

The optimization software used in this system is HScriptOpt, developed by the Honda Research Institute Europe based on next-generation optimization algorithms.

2.1. Optimization Calculation Flow

As shown in Fig. 1, this system consists of four processes, starting with A. Blade definition, then passing through two analysis modules in B. Aero module and C. Structure module before ending with D. Evaluation.

At the start of the calculation, 11 different blades are generated in A. Blade definition, either at random with predefined ranges for the possible design variables or from concrete given initial values. We call this the first generation. This first generation is then subjected to aerodynamic analysis and structural analysis using B. Aero module and C. Structure module, respectively. During the optimization, both modules are run in parallel.

In D. Evaluation, the individual different designs are evaluated by defining an evaluation function called Fitness. An example of Fitness function is shown in Eq. (1). Here, the weight of the isentropic efficiency is taken as α , with the remaining weight given to low-cycle fatigue (LCF) life. Maximizing their sum is the target of the optimization problem. Note that there is no limit to the number of objective variables that can be incorporated into the Fitness function.

$$\text{Fitness} = \alpha \times (\text{isentropic efficiency}) + (1 - \alpha) \times (\text{LCF life}) \quad (1)$$

The key variables that are often set in the system as objective variables and constraints are summarized in Table 1. The variables have been grouped from the viewpoints of aerodynamics and structural mechanics.

System optimization calculations repeatedly generate individual cases for subsequent generations based on Fitness and evaluate the results. The Covariance Matrix

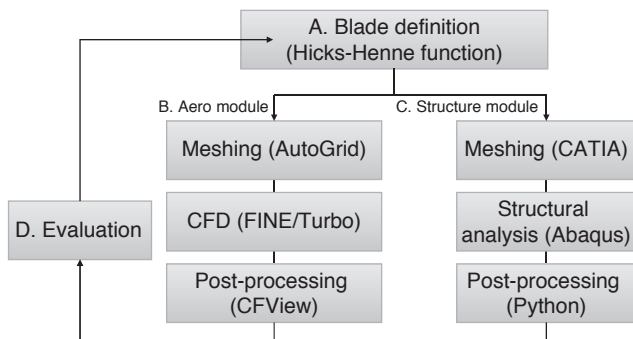


Fig. 1 Optimization workflow

Table 1 Objective variables

	Objectives
Aero performance	Efficiency
	Mach number
	Flow rate
	Pressure ratio
	Expansion ratio
Mechanics	Stress
	LCF life
	Modal frequency
	Vibration modes number
	Mass

Adaptation Evolution Strategy (CMA-ES)⁽⁷⁾ was used as the optimization algorithm for verifying the effectiveness of the system implemented in Chapter 4.

Each process was automated using Python. Each process from A to C is described below.

2.2. A. Blade definition

If too many input variables are used in the optimization calculations, the calculation results will take a long time to converge. To avoid this, the input variables used in defining the optimization problem need to be reduced while increasing design variation.

Blades are generally defined by a free curve using a Bézier curve⁽⁸⁾, but adding more nodes to increase the degrees of freedom will increase the number of input variables. Thus, blades for this program are defined using a Hicks-Henne function⁽⁹⁾, which can represent complex curved surfaces with fewer input variables.

Blades can be defined with any number of sections. A sample 5-section blade as defined with this system is shown in Fig. 2. Adding Hicks-Henne functions to a

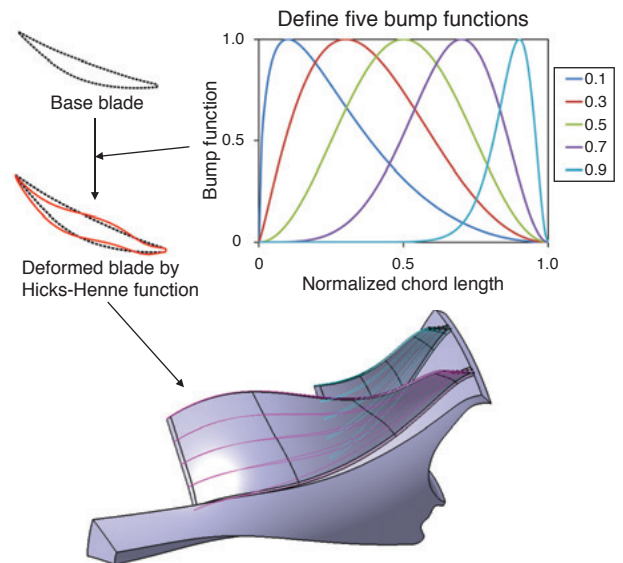


Fig. 2 Blade definition

baseline shape generates a curve by superimposing waves and defining the blade shape by changing the height of the waves at each position with an input variable. By using Hicks-Henne functions, blades can be defined to a similar precision as when using a normal Bézier curve but with about half of the input variables per cross-section.

2.3. B. Aero module

The blade shapes defined in A are subjected to aerodynamic meshing and aerodynamic analysis, and the results are post-processed. An example mesh is given in Fig. 3. Meshes are generated using Cadence Autogrid 14.2 meshing software, and aerodynamic analysis is performed using Cadence Fine/Turbo 14.2.

2.4. C. Structure module

For the structural analysis model, meshes as shown in Fig. 4 are generated for the blades defined in A. During mesh generation, inputting different mesh definition data for each part such as the blade surface, blade root fillet, and disk allows the model to maintain mesh quality and produce highly precise analysis results.

When mesh generation is complete, a structural analysis input file including the mesh data and boundary conditions is generated. The process sequence is automated using

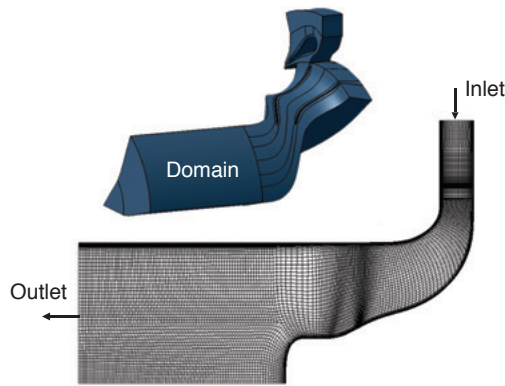


Fig. 3 CFD mesh



Fig. 4 Structural analysis mesh

CATIA Script⁽¹⁰⁾, a CATIA-specific macro language. When handling component weights and shape measurement results during optimization calculations, data can be extracted here. The structural analysis was performed using Dassault Systèmes Abaqus/standard 2018. As structural analysis results in CATIA can be extracted individually for each mesh definition part, the analysis results can be imposed with fine constraints.

The system's structural analysis includes multiple selectable functions, such as Stress Analysis and Modal Analysis. The necessary analyses can be selected in isolation to shorten analysis time for producing the analysis results.

3. Structural Mechanics Evaluation Items in Detail

The structural mechanics evaluation function for this system was enhanced during rotor blade design in order to maximize aerodynamic performance and service life.

The structural mechanics evaluation items break down from (i) to (v) as shown in Fig. 5. In (i) Temperature mapping, the temperature history is mapped, and in (ii) Hot-to-cold conversion, the blades are converted for reasons described below. Next, (iii) Stress analysis calculates blade surface stress and maximum stress at random locations; (iv) Modal analysis calculates resonance points and natural frequencies; and (v) LCF analysis calculates the LCF life based on the stress history during engine operation.

3.1. (i) Temperature mapping

This process presents temperature data for the structural analysis model in the form of the temperature profile during operation. The temperature profile is calculated from a thermal conductivity analysis based on gas temperatures as determined by CFD and component heat transfer rates.

In the system, the three-dimensional temperature profile, which is the result of thermal conductivity analysis for the initial blade shape, is mapped three-dimensionally to a

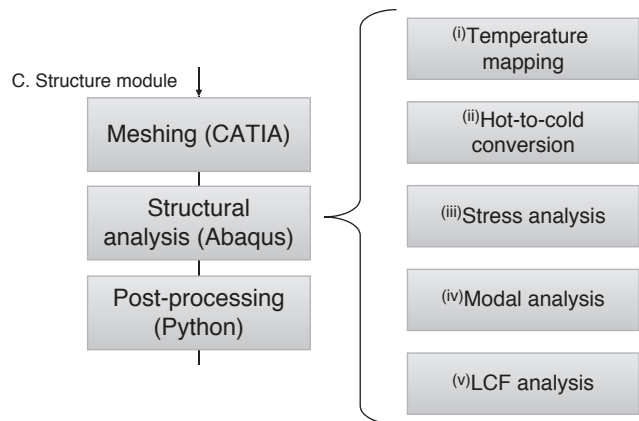


Fig. 5 Structure module

random blade. A mapped temperature profile is shown in Fig. 6.

The model is given the temperatures for required times, such as the temperature at the time to evaluate LCF life. Here, an in-house tool was used for thermal mapping. Generally, thermal stress greatly impacts structural design given that the absolute temperature values for the turbine are greater than those for the compressor and, consequently, large gradients arise.

3.2. (ii) Hot-to-cold conversion

In blade aerodynamic design, there are a number of typical conditions for determining engine performance, such as for takeoff and cruising at high altitude. This section will focus on the most important of these conditions. The rotational speed determined from these conditions is called the aerodynamic design rotational speed.

As the blades defined by this system are hot shapes, which are deformed by external forces at aerodynamic design rotational speeds, the blades need to be reverse deformed to their cold shapes at the zero-state rotational speed prior to structural analysis. This is a hot-to-cold conversion.

The difference between hot and cold shapes is illustrated in Fig. 7. In the front view, the hot shape has deflected to the left compared to the cold shape. Meanwhile, in the side view, the hot shape has a slightly higher blade height than the cold shape.

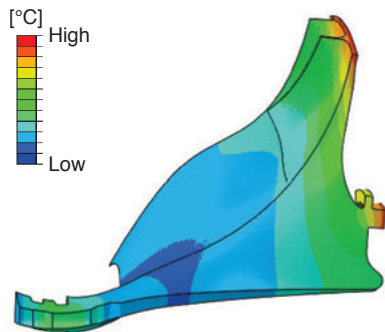


Fig. 6 Mapped temperature

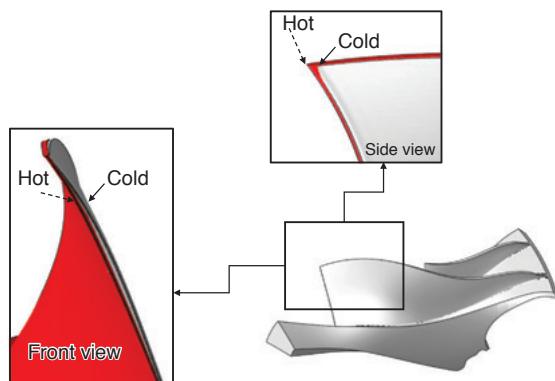


Fig. 7 Hot to cold

If the hot shape is used as the initial shape for structural mechanics calculation, the external forces will be duplicated in the shape calculations. Thus, hot blade shapes are converted to their cold shapes to evaluate structural mechanics precisely. Within this process, calculations are iterated using the finite element method (FEM), reverse calculating to determine the cold shape from the aerodynamic design shape.

3.3. (iii) Stress analysis

As rotor blades rotate at high speed, they generate centrifugal stress, pressure, and thermal stress. As shown in Fig. 8, stress peaks are observed in each of the fillet, disk bore, and blade region. Structural mechanics is to be evaluated to help ensure that no major plastic deformation occurs beyond the yield stress.

3.4. (iv) Modal analysis

A rotor blade may resonate with a blade natural frequency or generate large vibrational stress, when vibrations generated by the nozzle or stator blade located upstream act as the source of excitation. The Campbell diagram shown in Fig. 9 illustrates the number of excitation

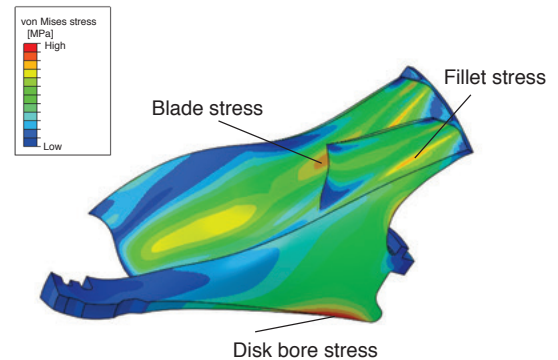


Fig. 8 Stress contour

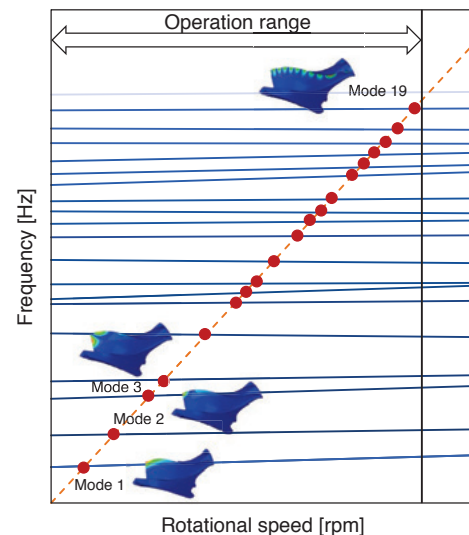


Fig. 9 Campbell diagram

force frequencies and resonance points with the blade natural frequencies (points where the diagonal dashed line and solid lines for each vibration mode intersect, denoted in the diagram by dots). In this example, 19 resonance points are found in the operation range.

If a design with resonance points in the operation range is unavoidable, verification is needed to help ensure that no resonance modes generate harmful vibration stress. When thinner blades are promoted in the interest of aerodynamic performance, higher order modes tend to increase, making it harder to measure vibrational stress and identify each vibrational mode. This issue becomes easier to address by reducing the number of vibration modes included in the operation range, which is therefore one target of the design process.

3.5. (v) LCF analysis

With rotor blades designed to have a finite life, it is necessary to ensure a certain minimum LCF life, and extending this LCF life can reduce part replacement costs. The LCF life of rotor blades is governed mainly by centrifugal and thermal stress time history. Given the difference in time constants between centrifugal and thermal stresses, this difference produces response delays that will differ depending on the location. Thus, assuming blades with multiple operating states, LCF life is calculated based on the time history at each location within the component by performing multiple centrifugal stress analyses under different operating conditions, as shown in Fig. 10.

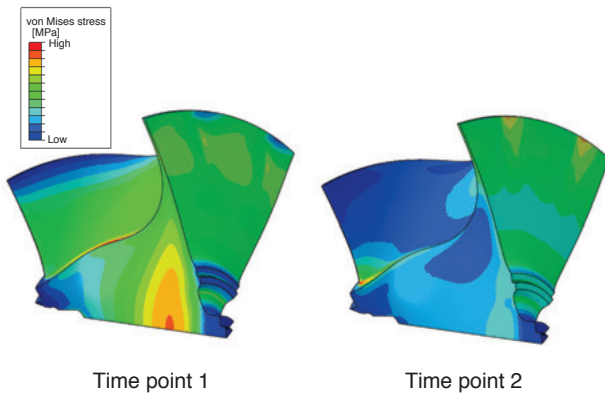


Fig. 10 Stress analysis result for LCF calculation

4. Effectiveness of System Application

Here, two cases of blade design will be considered to verify the effectiveness of applying the design assist system. The first case is an optimization calculation for a centrifugal compressor, evaluating the relationship between the number of blade vibration modes, disk bore stress, and isentropic efficiency. The second case is a calculation for a

radial turbine, evaluating the relationship between LCF life and isentropic efficiency considering temperature gradients.

4.1. Centrifugal Compressor Optimization Calculations

In considering how to reduce the number of rotor blade vibration modes within the operation range, the impact of the number of vibration modes on stress and isentropic efficiency was evaluated. The aim of these calculations was to maximize isentropic efficiency and minimize the number of vibration modes, with at least a 10% reduction in the number of vibration modes as the constraint. A part of the optimization calculation results is shown in Fig. 11. Here, the initial base line design is shown with a black square (■), and other individual cases with white circles (○). The vertical axis represents disk bore stress (disk bore stress positions shown in the figure), while the horizontal axis represents the number of vibration modes. Both horizontal and vertical axes were normalized to the base line value.

As seen in Fig. 11, disk bore stress increases as the number of natural vibration modes decreases. This relationship appears as a dotted line, illustrating the trade-off between disk bore stress and the number of vibration modes. This relationship results because blade rigidity increased due to the increased blade thickness, while disk bore stress increased due to the increased self-weight.

Next, the relationship between isentropic efficiency and number of vibration modes is shown in Fig. 12. Here, the vertical axis represents the isentropic efficiency, and the horizontal axis represents the number of natural vibration modes. The figure illustrates that isentropic efficiency decreases as the number of vibration modes decreases from the base line.

Presumably, this reduction is due to increased blade thickness to reduce the number of vibration modes, resulting in reduced isentropic efficiency. These results indicate that

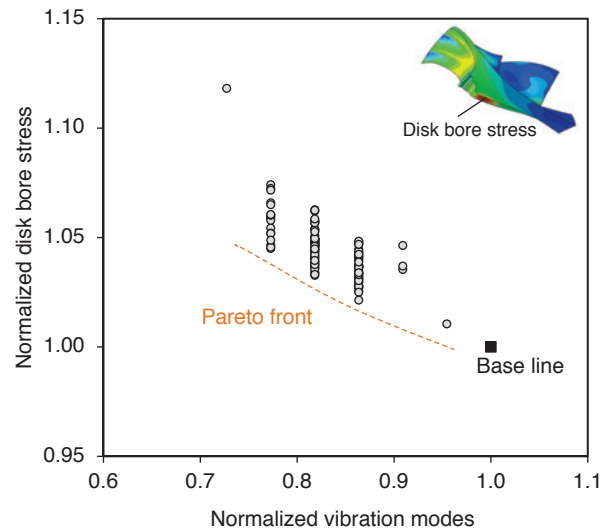


Fig. 11 Disk bore stress vs. vibration modes

a decrease in isentropic efficiency of approximately 0.7% will reduce the number of vibration modes by 10%.

4.2. Radial Turbine Optimization Calculations

The isentropic efficiency of the radial turbine and the LCF life of the blade root fillet were increased. Figure 13 shows the calculation results, along with the root fillet position of the blade for which LCF life is calculated. Here, centrifugal and thermal stresses were calculated for two points during operation to regulate LCF life, and LCF life was calculated.

The base line is shown as a black square (■), and other individual cases are shown as white circles (○). The horizontal axis represents the isentropic efficiency, and the vertical axis represents LCF life. Both horizontal and vertical axes were normalized to the base line value. The Pareto front of LCF life and isentropic efficiency is shown

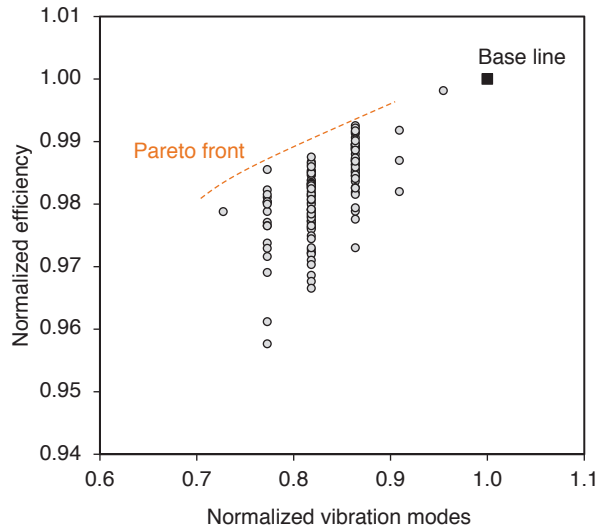


Fig. 12 Efficiency vs. vibration modes

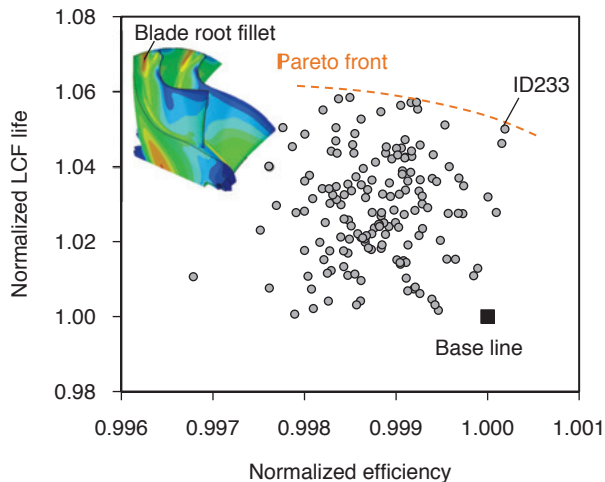


Fig. 13 Efficiency vs. LCF life

by the dotted line. The most efficient solution on the Pareto front is ID233. Focusing now on ID233, a comparison superimposing the blades of ID233 and the base line is shown in Fig. 14. While there are no major changes in blade thickness or airfoil between the two, the inclination angle of the blade tip is increased. As a result, while no major changes were observed in the flow field, reducing the blade root fillet stress by a few percentage points increased LCF life by 5%.

From these optimization calculation results for the radial turbine, in studying how to maximize isentropic efficiency and LCF life, it is predicted that LCF life can be increased by about 6% at the same isentropic efficiency. When further extending LCF life, it is expected that isentropic efficiency would inevitably be decreased. In addition, with the current design constraints left unchanged, it can be surmised that dramatically increasing isentropic efficiency while maintaining LCF life would be no trivial task.

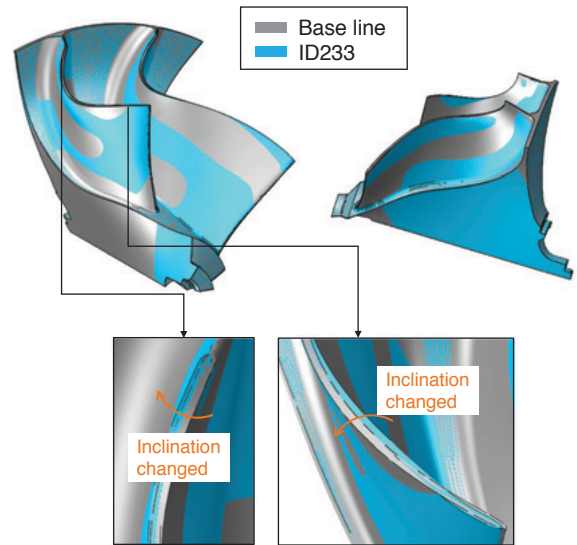


Fig. 14 Base line design vs. ID233 design

5. Conclusion

This paper introduced a design assist system for gas turbine rotor blades. In confirming system effectiveness, a case study was presented considering the two areas of aerodynamics and structural mechanics.

By utilizing this system, it is possible both to select suitable solutions for design variables, and to quantitatively demonstrate the impact on design elements with associated trade-offs, such as varying disk bore stress by b% and isentropic efficiency by c%, respectively, to reduce the number of vibration modes by a%.

The ability to propose suitable solutions that satisfy both aerodynamic and structural design goals while satisfying the required structural constraints and to present quantitative

design prospects, all at the initial blade design stage, should help in streamlining design and be effective in the development of gas turbines.

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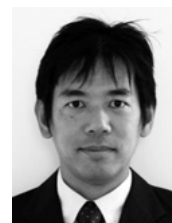
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Knocking Mitigation of Gasoline Engine by Reduction of Unburned Area Using Pre-chamber Jet Combustion

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ABSTRACT

Analysis was conducted of knocking mitigation when using passive pre-chamber jet combustion as compared to a gasoline engine with stoichiometric spark ignition combustion. Jets emitting from the pre-chamber enabled the propagation flame to rapidly reach the bore end, shortening the combustion duration. This reduced the unburned area of end-gas in the latter half of combustion and mitigated knocking. It was also shown that by creating more nozzles and thus increasing the number of jets, the unburned field is partitioned and made smaller, which suppresses auto-ignition and mitigates knocking. It was confirmed that setting specifications to maximize the knocking mitigation effect and thermal efficiency can raise the compression ratio by 1.8 and increase thermal efficiency by 1.1 pts.

1. Introduction

Countries around the world are stepping up their efforts to become carbon neutral. Under the scenario where an increase in temperature caused by climate change is kept below 1.5°C, it will be necessary to reduce emissions of the greenhouse gas CO₂ by at least 45% in 2030 as compared to 2010⁽¹⁾. To that end, increasing the thermal efficiency of internal combustion engines is an effective way of reducing CO₂ emissions.

In gasoline spark ignition (SI) combustion, the ignition timing needs to be retarded more than the minimum advanced for best torque (MBT) in order to help avoid knocking. This increases time loss and is an issue in the effort to increase thermal efficiency. The retarding of ignition timing leads to lower thermal efficiency in the normal load zone, and because it is necessary to suppress increases in exhaust gas temperature at the output point, it leads to lower output. For the sake of maintaining output, fuel enrichment has been used to date to protect the exhaust system, however this increases CO emissions. Because nations are adopting Real Driving Emissions (RDE) legislation, regulations on CO emissions are being strengthened. Moreover, RDE legislation going forward is likely to ban the use of fuel enrichment itself. Therefore, it will not be easy to maintain output as in the past. As the above indicates, knocking mitigation is key to increasing

thermal efficiency and output, lowering CO emissions, and conforming to RDE legislation.

As countermeasures, methods are being attempted that use pre-chamber jet combustion and, with the discharging jets, increase turbulence kinetic energy on the flame propagation side and shorten combustion duration to mitigate knocking⁽²⁾⁻⁽⁴⁾. Many case studies have been reported of using pre-chamber jet combustion in order to stabilize lean-burn SI combustion, which has a tendency to be sluggish⁽⁵⁾⁻⁽⁷⁾. As for gasoline stoichiometric SI combustion, there are case studies that analyzed the knocking mitigation effect when trying to increase the compression ratio⁽²⁾⁻⁽⁴⁾, but those studies do not have any detailed pre-chamber jet combustion status analysis. Because the jets of combustion gas ejected from the many nozzles exhibit complex combustion forms, there are many unknowns about the effect pre-chamber jet combustion has on knocking and its auto-ignition behavior inside the cylinder. To establish the specifications of pre-chamber jet combustion, it is necessary to elucidate these phenomena.

This research clarified phenomena relating to pre-chamber jet combustion knocking mitigation in gasoline stoichiometric SI combustion. It also used the knocking mitigation effect of pre-chamber jet combustion to study the effect of increasing the compression ratio to increase thermal efficiency.

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2. Principle of Pre-chamber Jet Combustion

This research used pre-chamber jet combustion with a passive pre-chamber⁽²⁾⁻⁽⁴⁾. A passive pre-chamber is a pre-chamber with no interior fuel injection nozzles. Rather, the air-fuel mixture is ingested from the main-chamber side. An illustration of the passive pre-chamber jet combustion chamber used in this study is provided in Fig. 1.

In the passive pre-chamber, the pressure differential between the main chamber and pre-chamber during engine compression stroke is used to bring the air-fuel mixture of the main chamber into the pre-chamber. A spark plug in the pre-chamber then ignites the air-fuel mixture and the subsequent rise in pre-chamber pressure causes hot burned gas and unburned gas to be ejected from the pre-chamber nozzle holes⁽⁴⁾. This increases the ignition surface area of the main chamber air-fuel mixture and the turbulence kinetic energy on the main chamber flame surface, enabling rapid combustion.

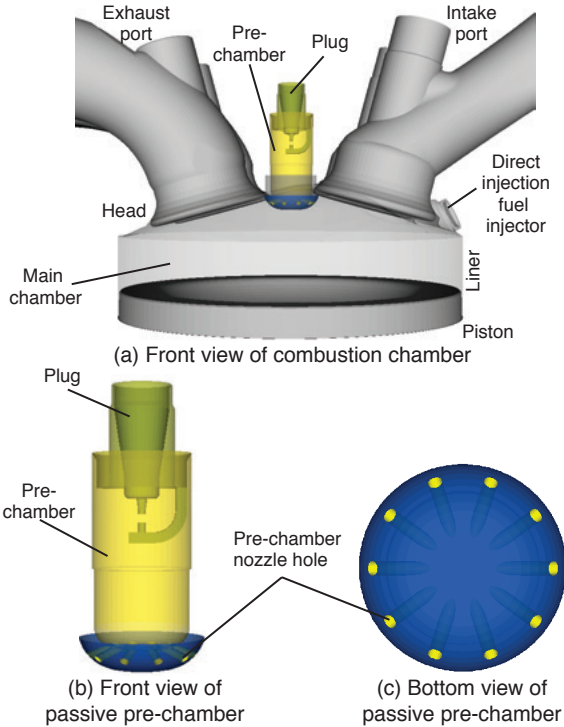


Fig. 1 Schematic diagram of combustion chamber and passive pre-chamber

3. Knocking Analysis Method

3.1. Test Engine and Evaluation Load

For the analysis, a pre-chamber was added to a single-cylinder engine that was developed based on mass-production 1.5 L direct injection turbo inline four-cylinder specifications. Specifications of the engine used for the analysis are provided in Table 1.

Table 1 Test engine specifications

Number of cylinders	1
Displacement volume [cm ³]	374.5
Bore [mm]	73
Stroke [mm]	89.4
Compression ratio	10.6 (base), 12.4
Valve system	DOHC 4-valve
Intake port tumble ratio	1.8
Fuel injection system	Direct injection (Injection pressure: 10 [MPa])
Fuel	Japanese regular gasoline (Research octane number: 91)
Ignition system	Direct ignition (Ignition energy: 60 [mJ])
Plug heat value	#8
Pre-chamber specifications	Hole number: 4, 6, 8, 10, 12
	Hole diameter [mm]: 0.3, 0.4, 0.5, 0.65, 0.8, 0.95
	Volume [cm ³]: 0.42, 0.52, 0.83, 1.12, 1.36, 1.92
Boost device	Super charger

In pre-chamber jet combustion, nozzle hole (hereinafter, hole) diameter and pre-chamber volume are involved in jet penetration (hereinafter, penetration)⁽⁴⁾ and it is believed that the number of holes is involved with the partition state of unburned area inside the cylinder. Analysis was therefore conducted as below using these as design variables for the pre-chamber specifications, as shown in Table 1.

The evaluation load point was set at an engine speed of 2,000 rpm and indicated mean effective pressure (IMEP) of 1,300 kPa (fuel injection of 30 mg/st per cycle per cylinder), conditions under which the time loss caused by the knocking limit becomes an issue.

3.2. Computational Technique

To analyze both the knocking mitigation effect of pre-chamber jet combustion and the phenomenon, testing was done on actual equipment, and phenomenon analysis was also conducted of the combustion by computational fluid dynamics (CFD). Combustion CFD was performed using Convergent Science's 3D-CFD software CONVERGE 3.0.14. An overview of the computational model is given in Table 2.

Here, the Extended Coherent Flame Model (ECFM)⁽⁸⁾ was used as the combustion model. In this model, flame propagation combustion is calculated using a flame surface density equation. Also included in this approach are the Tabulated Kinetics of Ignition (TKI) model, which uses various parameters to express auto-ignition in a table, as the auto-ignition model⁽⁹⁾ and the Imposed Stretch Spark Ignition model (ISSIM), which models the ignition circuit and ignition part combustion

Table 2 Calculation software and model

CFD software	CONVERGE v3.0.14
Turbulence model	RNG k- ζ -f
Spray model	Kelvin-Helmholtz and Rayleigh-Taylor
Combustion model	Extended Coherent Flame Model
Ignition model	Imposed Stretch Spark Ignition
Auto-ignition model	Tabulated Kinetics of Ignition
Fuel	Regular gasoline

gas, as the ignition model⁽⁹⁾. This model makes it easy to analyze combustion parameters of flame propagation combustion and can also isolate auto-ignition combustion for analysis. This makes it a good model for analyzing SI combustion, which consists of both flame propagation combustion and auto-ignition combustion. The reproducibility of conventional SI combustion and pre-chamber jet combustion with this model has been confirmed⁽⁹⁾.

Details of the combustion model are shown in Fig. 2. The propagation flame combustion rate (OMEGA_CFM) indicates the unburned fuel chemical reaction speed on the flame propagation reaction surface, while the auto-ignition combustion rate (OMEGA_TKI) indicates the unburned area's auto-ignition reaction speed. In this research, the effect that changing the various pre-chamber jet parameters has on flame propagation is evaluated by OMEGA_CFM and the effect it has on auto-ignition is evaluated by OMEGA_TKI.

3.3. Knocking Judgment Method

The conventional method of judging knocking is to calculate a Fourier transform of the pressure as measured

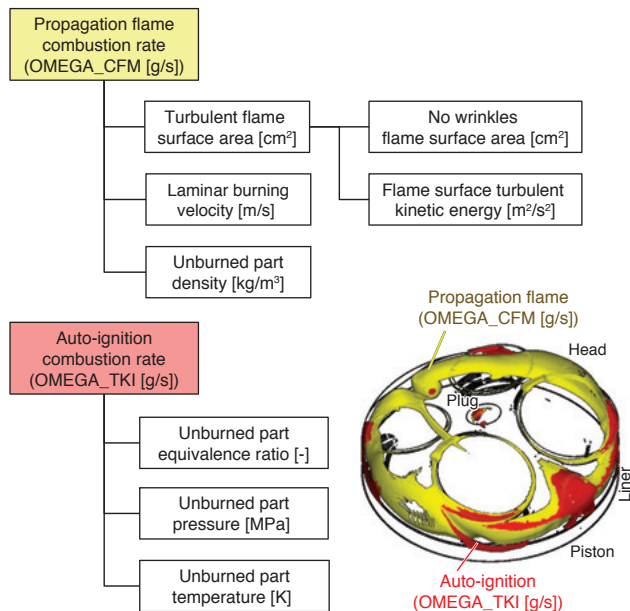


Fig. 2 Factor analysis of flame propagation combustion and auto-ignition combustion

inside the cylinder and then use band-pass filtering, after which knock intensity (KI), the difference between the maximum and minimum values of pressure amplitude, is calculated⁽¹⁰⁾. Then, after ignition timing characteristics are measured, the combustion timing at which KI suddenly changes is found and the knocking limit combustion timing is judged.

For analyzing pre-chamber jet combustion by CFD, this research devised a new knocking judgment method using CFD calculation results. Because knocking is so closely related to auto-ignition of unburned area of end-gas in the latter half of combustion⁽¹¹⁾, the research team believed it could judge knocking based on the auto-ignition combustion rate calculated by CFD. An overview is given in Fig. 3. Figure 3(a) shows combustion over time and how the auto-ignition combustion rate approaches the peak. This peak value is defined as the maximum auto-ignition combustion rate (OMEGA_TKI_max). The results for OMEGA_TKI_max at the time when the ignition timing shown in Fig. 3(b) is changed confirm the behavior such that the combustion center of gravity timing (Mass Fraction Burned 50%: MFB 50%) is advanced and OMEGA_TKI_max is increased. Therefore, it seems possible to estimate knocking limit MFB 50% by setting a fixed threshold for OMEGA_TKI_max. When actual equipment with the various combustion specifications used as parameters was compared to the CFD knocking limit in pre-chamber jet combustion, OMEGA_TKI_max 1.5 g/s was used as the knocking threshold for the

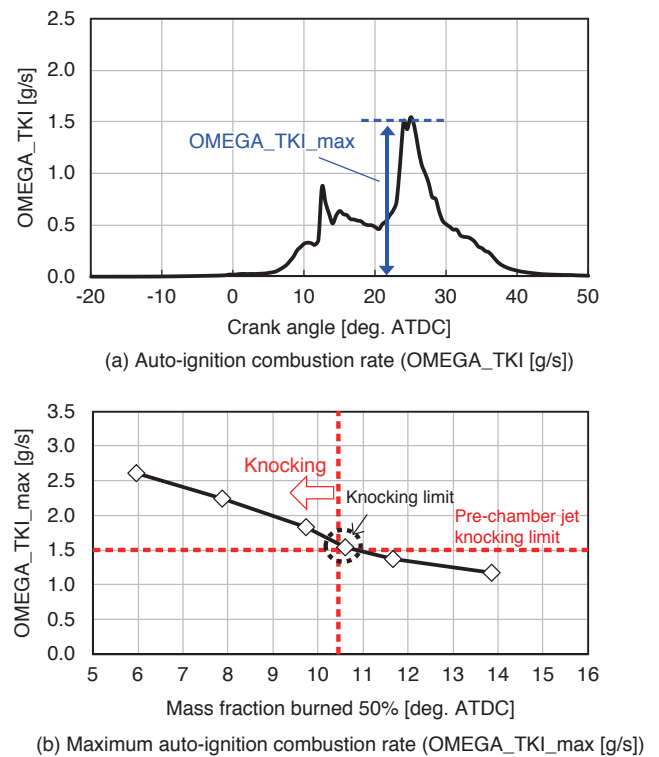


Fig. 3 Knocking judgment method by CFD auto-ignition observation in pre-chamber jet combustion

test engine's evaluation load, which confirmed that there is a correlation between the actual equipment and CFD at knocking limit MFB 50%.

The correlation between actual equipment and CFD at knocking limit MFB 50% in pre-chamber jet combustion when the pre-chamber hole diameter, volume, and number of holes were changed is shown in Fig. 4. Using CFD, this judgment method can estimate the actual equipment's knocking limit MFB 50% with a precision within 0.5 deg., confirming that a judgment method using the auto-ignition combustion rate is valid.

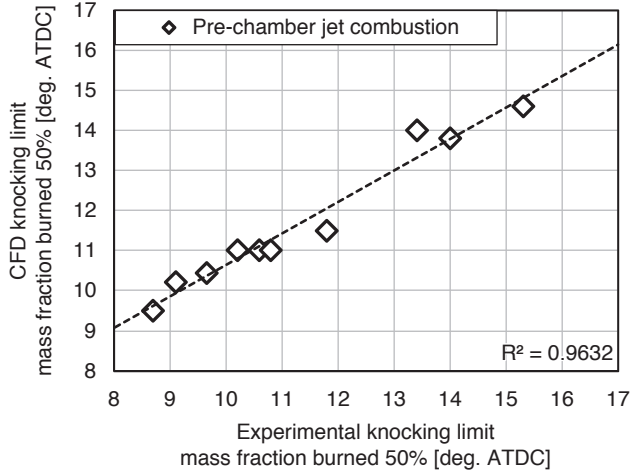


Fig. 4 Knocking limit between experiment and CFD on pre-chamber jet combustion

4. Analysis of Knocking Mitigation Effect

4.1. Pre-chamber Hole Diameter and Knocking

CFD was used to analyze the relationship between pre-chamber hole diameter and knocking. Here, the CFD auto-ignition combustion rate was used to judge knocking and comparisons were made under knocking limit conditions. This comparison makes it possible to observe the auto-ignition phenomenon when knocking is occurring so that the knocking mitigation effect can be quantitatively evaluated. Penetration, as used below, is defined as the distance reached from pre-chamber holes to flame propagation reaction surface, whereas end-gas auto-ignition volume is defined as the value in the range within 10 mm inside the liner wall. Pre-chamber specifications analyzed by CFD are given in Table 3.

The MFB 50% and combustion duration at the knocking limit are shown in Fig. 5. For 1 deg. of shortening of combustion duration, MFB 50% at the knocking limit advances approximately 1 deg. Combustion duration is shortest at hole diameter $\Phi 0.5$ mm, at which time the knocking mitigation effect is greatest. This presumably means that maximizing penetration at $\Phi 0.5$ mm maximizes turbulent flame surface area and minimizes combustion duration, as shown in Fig. 6. Moreover, at $\Phi 0.4$ mm and

Table 3 Pre-chamber jet specifications on CFD

	Pre-chamber jet combustion $\epsilon 10.6$					
Excess air ratio	Stoichiometric					
Pre-chamber hole specification	10 holes, 0.3 mm diameter	10 holes, 0.4 mm diameter	10 holes, 0.5 mm diameter	10 holes, 0.65 mm diameter	10 holes, 0.8 mm diameter	10 holes, 0.95 mm diameter
Pre-chamber volume [cm ³]	0.83	0.83	0.83	0.83	0.83	0.83

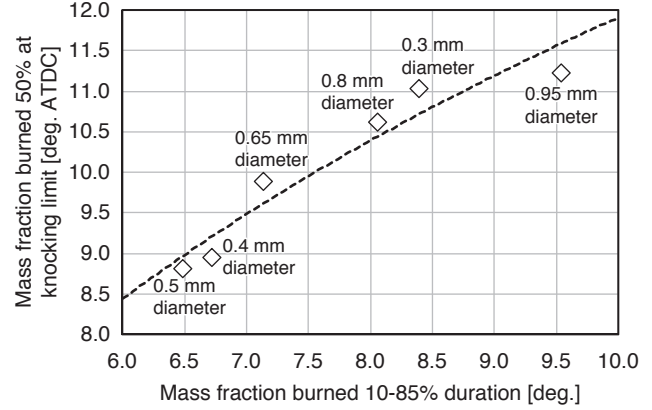
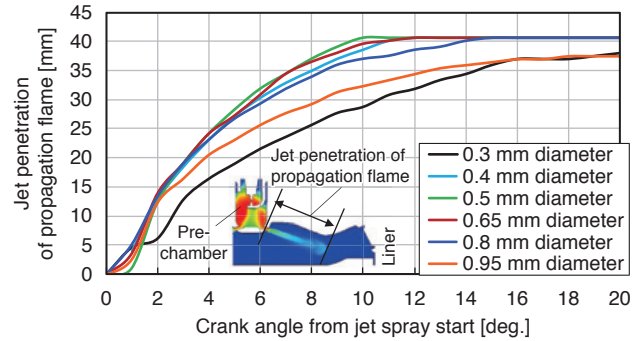
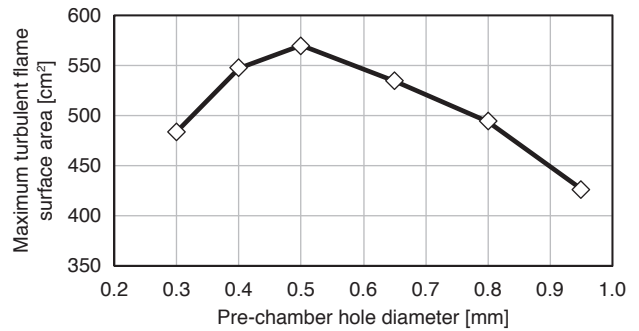


Fig. 5 Effect of pre-chamber hole diameter on knocking by CFD



(a) Relationship between pre-chamber hole diameter and jet penetration of propagation flame



(b) Relationship between pre-chamber hole diameter and maximum turbulent flame surface area

Fig. 6 Effect of pre-chamber hole diameter on jet penetration of propagation flame and maximum turbulent flame surface area by CFD

below, the jet flow volume from the apertures declined, whereas at $\Phi 0.65$ mm and greater, the jet flow velocity from the apertures also declined, presumably indicating that in each case penetration declined and combustion duration increased.

OMEGA_CFM and OMEGA_TKI are shown in Fig. 7. In specifications where knocking was mitigated, ignition timing could be advanced and combustion happened earlier. As Fig. 7(a) shows, the timing at which propagation combustion moved from the peak to the latter half was advanced to 8 deg. ATDC at hole diameter $\Phi 0.5$ mm, in contrast to the 11 deg. ATDC at hole diameter $\Phi 0.95$ mm. Moreover, as Fig. 7(b) shows, auto-ignition combustion started rising in the latter half of propagation combustion and was advanced to 11 deg. ATDC at hole diameter $\Phi 0.5$ mm in contrast to the 14 deg. ATDC at hole diameter $\Phi 0.95$ mm. The subsequent auto-ignition combustion peak was synchronous with the end of propagation combustion and was advanced to 22 deg. ATDC at hole diameter $\Phi 0.5$ mm in contrast to the 27 deg. ATDC at hole diameter $\Phi 0.95$ mm.

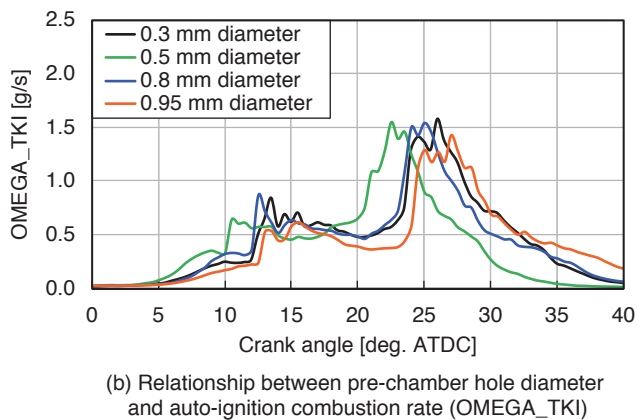
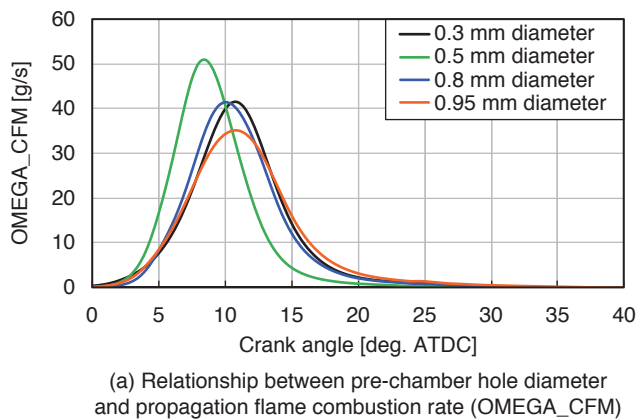


Fig. 7 Effect of pre-chamber hole diameter on relationship between OMEGA_CFM and OMEGA_TKI by CFD

Figure 8 shows an image of the flame while Fig. 9 shows the end-gas auto-ignition volume when OMEGA_TKI rises. Here, at hole diameter $\Phi 0.5$ mm, the time at which OMEGA_TKI rose was the most advanced and the end-gas auto-ignition volume was at its minimum.

These results indicate that extending penetration to produce rapid flame propagation reduces end-gas unburned air-fuel mixture and suppresses auto-ignition.

4.2. Pre-chamber Volume and Knocking

CFD was used to analyze the relationship between pre-chamber volume and knocking. The CFD auto-ignition combustion rate was used to judge knocking and comparisons were made under knocking limit conditions. Pre-chamber specifications analyzed by CFD are given in Fig. 10.

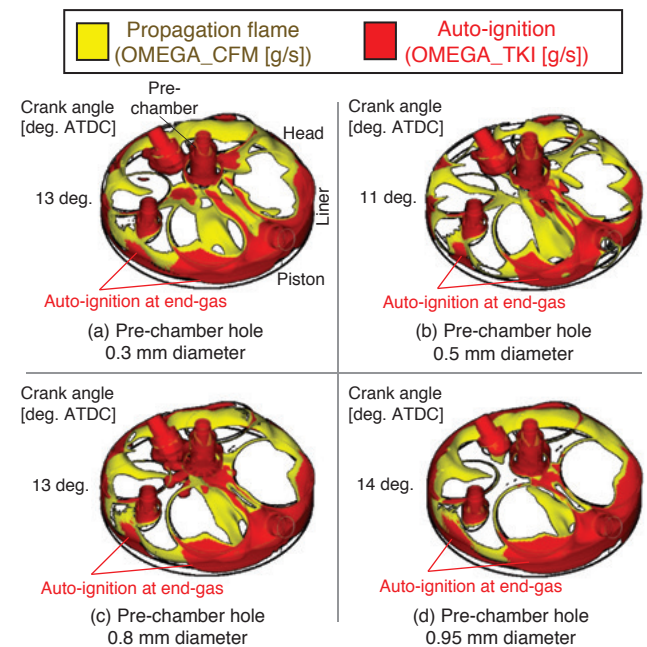


Fig. 8 CFD image analysis of pre-chamber hole diameter

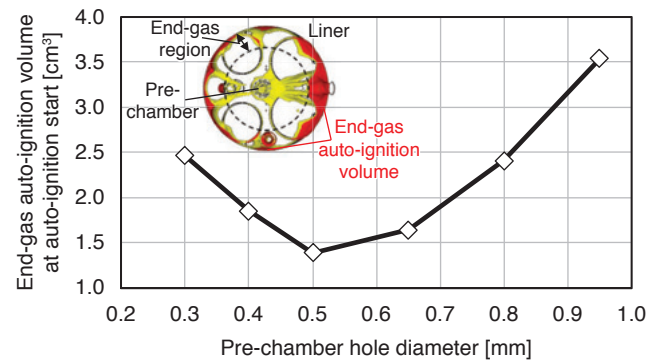


Fig. 9 Effect of pre-chamber hole diameter on end-gas auto-ignition volume by CFD

The combustion duration and MFB 50% at the knocking limit are shown in Fig. 11. For 1 deg. of shortening of combustion duration, MFB 50% at the knocking limit advances approximately 2 deg. Enlarging the pre-chamber volume shortened the combustion duration and increased the knocking mitigation effect. Penetration and maximum turbulent flame surface area are shown in Fig. 12. Enlarging the pre-chamber volume extended penetration and increased turbulent flame surface area. These appear to be factors in shortening combustion duration.

OMEGA_CFM and OMEGA_TKI are shown in Fig. 13. In the same manner as differences in hole diameter, in specifications where knocking was mitigated, ignition timing could be advanced and combustion happened earlier. As Fig. 13(a) shows, expanding pre-chamber volume from 0.52 cm³ to 1.92 cm³ made the timing at which propagation combustion moved from the peak to the latter half advance to 6 deg. ATDC from 11 deg. ATDC. Moreover, as Fig. 13(b) shows, auto-ignition combustion started rising in the latter half of propagation combustion and was advanced to

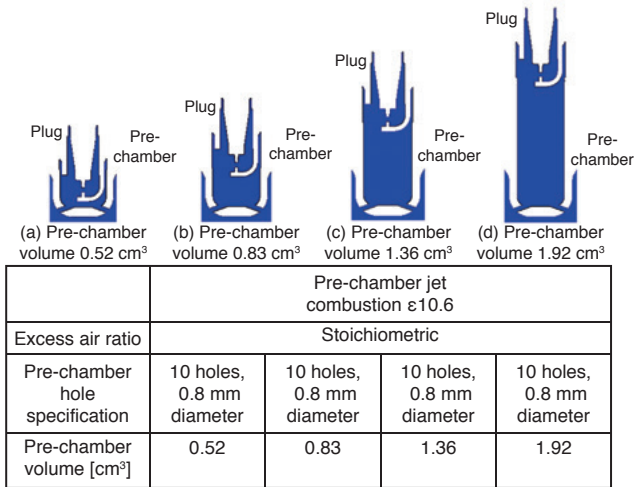


Fig. 10 Pre-chamber jet specifications on CFD

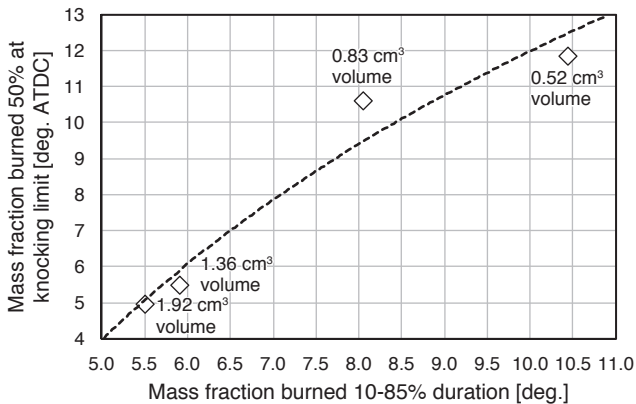
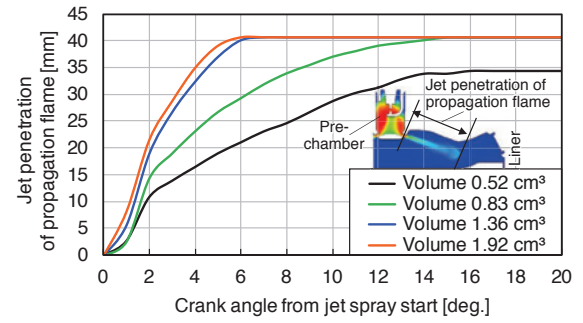
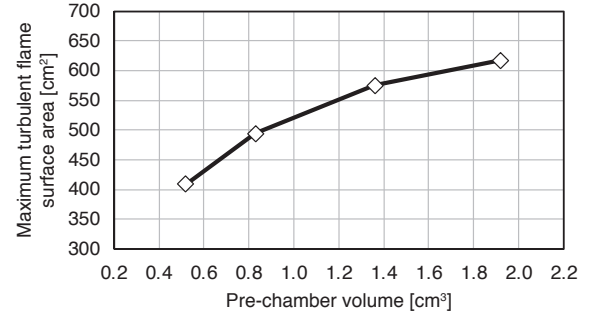


Fig. 11 Effect of pre-chamber volume for knocking by CFD

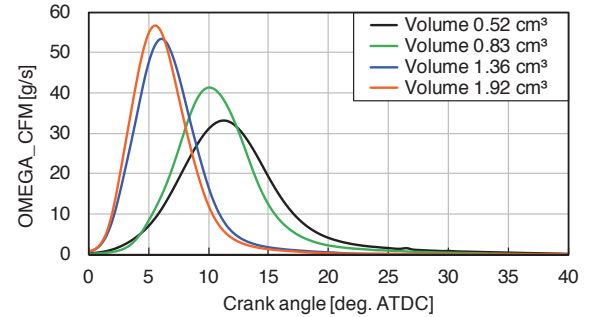


(a) Relationship between pre-chamber volume and jet penetration of propagation flame

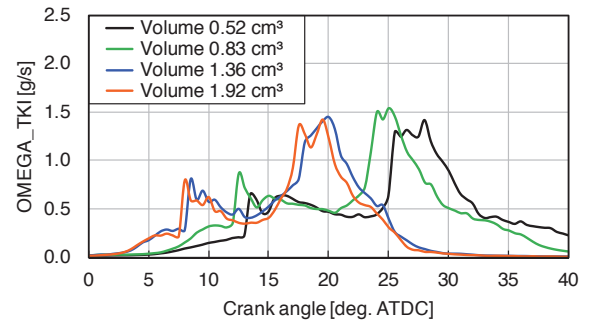


(b) Relationship between pre-chamber volume and maximum turbulent flame surface area

Fig. 12 Effect of pre-chamber volume on jet penetration of propagation flame and maximum turbulent flame surface area by CFD



(a) Relationship between pre-chamber volume and propagation flame combustion rate (OMEGA_CFM)



(b) Relationship between pre-chamber volume and auto-ignition combustion rate (OMEGA_TKI)

Fig. 13 Effect of pre-chamber volume on relationship between OMEGA_CFM and OMEGA_TKI by CFD

8 deg. ATDC from 14 deg. ATDC. The subsequent auto-ignition combustion peak was synchronous with the end of propagation combustion and was similarly advanced to 19 deg. ATDC from 27 deg. ATDC.

Figure 14 shows an image of the flame when OMEGA_TKI rises, while Fig. 15 shows the end-gas auto-ignition volume. It was confirmed that the propagation flame on the high-volume pre-chamber side where combustion duration is short reaches the bore end before auto-ignition of the end-gas and that the auto-ignition area shrinks.

The above indicates that enlarging pre-chamber volume to extend penetration produces rapid flame propagation, reduces end-gas unburned air-fuel mixture, and helps suppress auto-ignition.

However, there will presumably be an increase in pre-chamber heat loss as the surface area inside the pre-chamber grows larger and an increase in main-chamber heat loss as the jet contact surface area on the main-chamber wall increases. Therefore, the effect on heat loss was confirmed in CFD. As Fig. 16 shows, it was confirmed that for a 1 cm³ enlargement of the pre-chamber volume, pre-chamber heat loss increased by approximately 1 pt. and main-chamber heat loss increased by approximately 2 pts. In pre-chamber jet combustion, it is important both to reduce heat loss and shorten combustion duration to mitigate knocking. The results indicate that it is necessary to consider a balance of the two when deciding the volume.

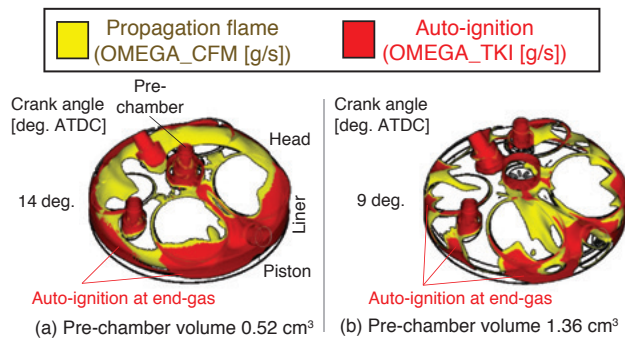


Fig. 14 CFD image analysis of pre-chamber volume

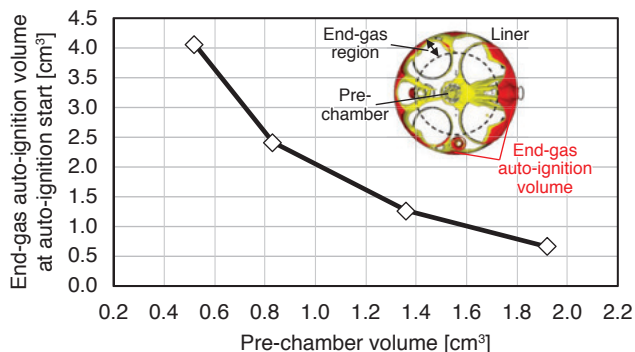


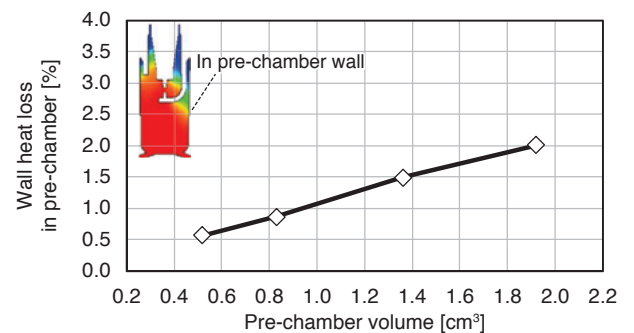
Fig. 15 Effect of pre-chamber volume on end-gas auto-ignition volume by CFD

Table 4 Pre-chamber jet specifications on CFD

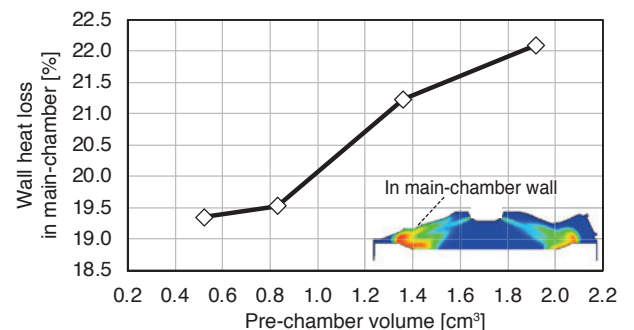
Excess air ratio	Pre-chamber jet combustion $\epsilon 10.6$				
	Stoichiometric				
Pre-chamber hole specification	4 holes, 1.27 mm diameter	6 holes, 1.05 mm diameter	8 holes, 0.9 mm diameter	10 holes, 0.8 mm diameter	12 holes, 0.73 mm diameter
Pre-chamber volume [cm³]	1.36	1.36	1.36	1.36	1.36

4.3. Pre-chamber Number of Holes and Knocking

In combustion with different numbers of holes in the pre-chamber, as the number of partitions of unburned area by jet will vary, it is anticipated that there will be variations in knocking behavior. Here, CFD was used to analyze the relationship between changes in number of holes in the pre-chamber and knocking behavior. Similarly to what was done before, the CFD auto-ignition combustion rate was used to judge knocking and comparisons were made under knocking limit conditions. To be able to evaluate pre-chamber specifications with different numbers of holes while providing the same amount of jet output, the number of holes and hole diameter were set so that the total surface area of the holes was constant. Table 4 gives the pre-chamber specifications analyzed by CFD, while Fig. 17 gives the combustion duration and MFB 50% at the knocking limit. As the number of holes increases, so does the knocking mitigation effect. Maximum knocking mitigation occurred with 10 holes. When there were more than 10 holes, however,



(a) Wall heat loss in pre-chamber by heat balance analysis



(b) Wall heat loss in main-chamber by heat balance analysis

Fig. 16 Effect of pre-chamber volume on heat loss by CFD

combustion speed declined slightly while knocking limit MFB 50% was largely retarded. Penetration and maximum turbulent flame surface area are shown in Fig. 18. From four to 10 holes, as the number of holes increases, the turbulent flame surface area tends to expand, and the combustion duration tends to shorten. We believe the reason for this is that, although the increase in the number of holes caused penetration itself to decline, the flame surface area expanded with the increasing number of jets.

OMEGA_CFM and OMEGA_TKI are shown in Fig. 19. With 10 holes, at which knocking was mitigated,

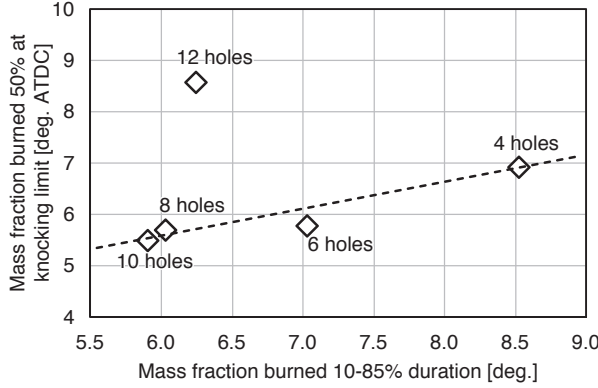
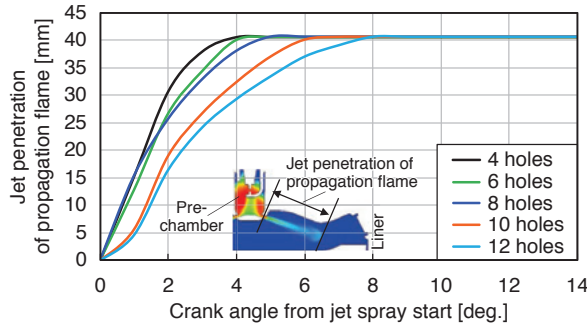
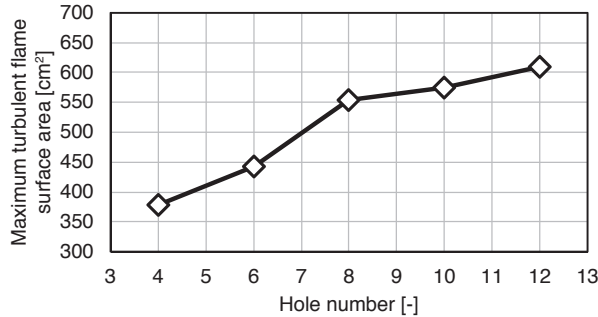


Fig. 17 Effect of pre-chamber hole number on knocking by CFD



(a) Relationship between pre-chamber hole number and jet penetration of propagation flame



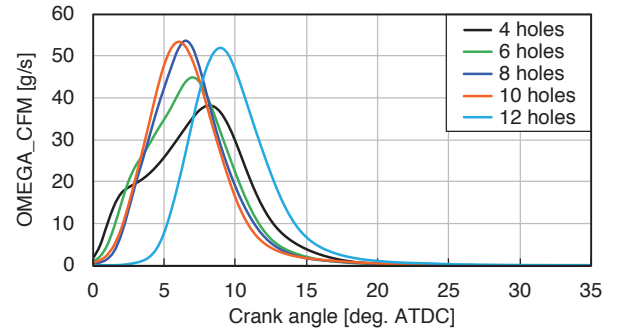
(b) Relationship between pre-chamber hole number and maximum turbulent flame surface area

Fig. 18 Effect of pre-chamber hole number on jet penetration of propagation flame and maximum turbulent flame surface area by CFD

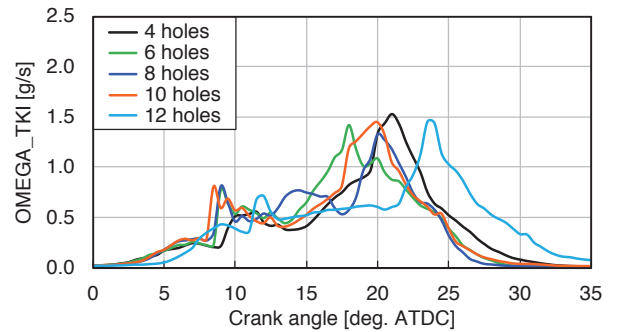
ignition timing could be advanced, and the combustion happened earliest with this number. As Fig. 19(a) shows, the timing at which propagation combustion moved from the peak to the latter half was advanced to 6 deg. ATDC with 10 holes in contrast to the 9 deg. ATDC with 12 holes. Moreover, as Fig. 19(b) shows, auto-ignition combustion began rising in the latter half of propagation combustion and was advanced to 9 deg. ATDC with 10 holes in contrast to the 24 deg. ATDC with 12 holes. The subsequent auto-ignition combustion peak was synchronous with the end of propagation combustion and was advanced to 20 deg. ATDC with 10 holes in contrast to the 24 deg. ATDC with 12 holes.

Here, observation of the flame image when OMEGA_TKI rises in Fig. 20 shows that as the number of holes increases, the increasing number of jets causes flame surface area to expand. Moreover, it was observed that the volume of each auto-ignition part tends to shrink when the unburned field is dispersed, and it was confirmed that the unburned field was most fragmented with 10 holes. With 12 holes, however, the unburned field was not completely dispersed because of interference between adjacent flames. As a result, knocking occurred and ignition timing was not advanced, which presumably retarded knocking limit MFB 50%.

These results indicate that having 10 holes suppresses auto-ignition and offers the best knocking mitigation owing to the effect of avoiding interference between adjacent



(a) Relationship between pre-chamber hole number and propagation flame combustion rate (OMEGA_CFM)



(b) Relationship between pre-chamber hole number and auto-ignition combustion rate (OMEGA_TKI)

Fig. 19 Effect of pre-chamber hole number on relationship between OMEGA_CFM and OMEGA_TKI by CFD

flames and dispersing the unburned field and making it smaller. This analysis found that selecting 10 holes, which is the jet interference limit in this engine, advanced the knocking limit MFB 50% by 1.5 deg. compared to four holes as a benchmark.

4.4. Compression Ratio and Knocking

Generally, an increased compression ratio raises theoretical efficiency but also increases end-gas temperature and pressure, which promotes auto-ignition and becomes a factor in knocking. Here, CFD was used to analyze the effect that compression ratio has on knocking in pre-chamber jet combustion.

Table 5 gives the specifications analyzed by CFD, while Fig. 21 shows the MFB 50% and combustion duration when ignition timing characteristics under each

Table 5 Engine specifications of conventional SI combustion and pre-chamber jet combustion on CFD

	Conventional SI combustion base ϵ 10.6	Pre-chamber jet combustion ϵ 10.6	Pre-chamber jet combustion ϵ 12.4
Excess air ratio	Stoichiometric	Stoichiometric	Stoichiometric
Pre-chamber hole specification	-	10 holes, 0.8 mm diameter	10 holes, 0.5 mm diameter
Pre-chamber volume [cm ³]	-	1.12	0.42

specification were measured. When the knocking mitigation effect under different compression ratios was confirmed, the knocking limit MFB 50% was 5 deg. advanced in pre-chamber jet combustion at compression ratio 10.6 as compared to conventional SI combustion with the same compression ratio. Moreover, even when the pre-chamber jet combustion compression ratio was raised to 12.4, the knocking limit MFB 50% was about the same as conventional SI combustion at compression ratio 10.6. This is presumably because the above-described shortening of combustion duration and the unburned area dispersion effect in pre-chamber jet combustion caused the end-gas's auto-ignition energy to decline.

Also, combustion duration of pre-chamber jet combustion with compression ratio 12.4 was approximately 40% shorter than that of conventional SI combustion with compression ratio 10.6.

5. Results of Tests with Actual Single-cylinder Engine

By using the knowledge gained from analysis up to this point through examination of the specifications of pre-chamber jet combustion, knocking and thermal efficiency of pre-chamber jet combustion in an actual single-cylinder engine were confirmed in order to verify the effectiveness of such combustion. The tested specifications are given in Fig. 22. Moreover, pre-chamber specifications were set based on the results described above.

Figure 23 shows the results of actual single-cylinder engine tests with conventional SI combustion and pre-chamber jet combustion when the ignition timing is varied. The evaluation load for the test was engine speed of 2,000 rpm and IMEP of 1,300 kPa. As Fig. 23(a) shows, combustion duration during pre-chamber jet combustion was approximately 40% shorter than for a conventional SI combustion engine, which is similar to the CFD study results. Moreover, in pre-chamber jet combustion with a compression ratio 1.8 pts. higher than conventional SI combustion, the knocking limit MFB 50% was about the same. These indicate that pre-chamber jet combustion replaced the amount of knocking mitigation with compression ratio, raising the compression ratio by 1.8. Since it is possible to raise the theoretical thermal efficiency with a higher compression ratio, the indicated thermal

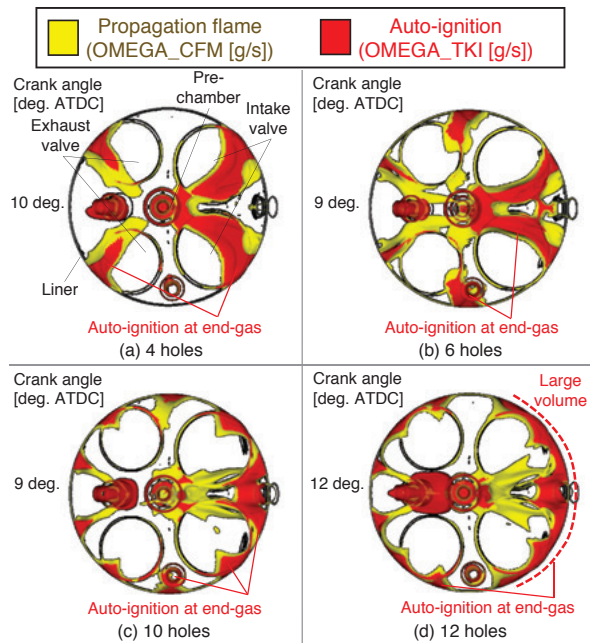


Fig. 20 CFD image analysis of pre-chamber hole number

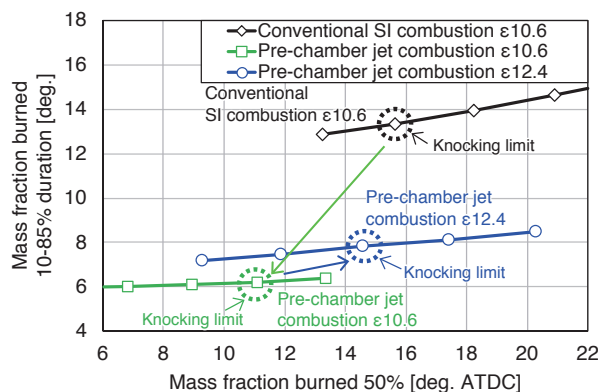


Fig. 21 Relationship between MFB 10-85% duration and MFB 50% in various compression ratios on CFD

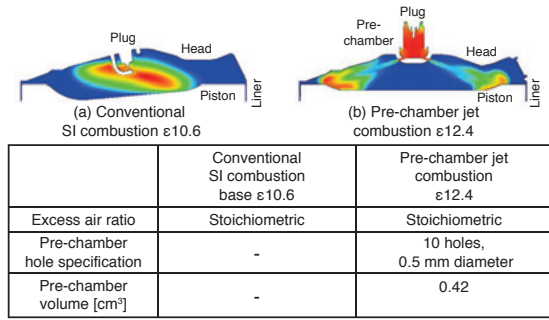
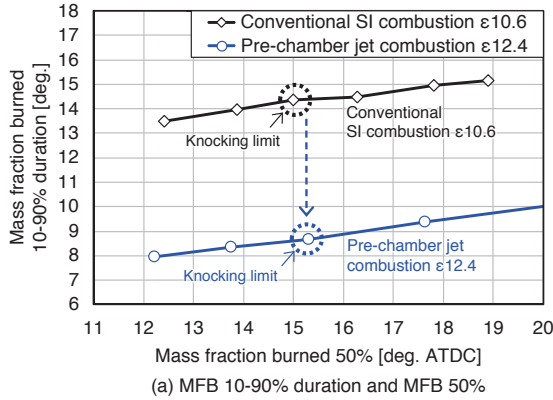
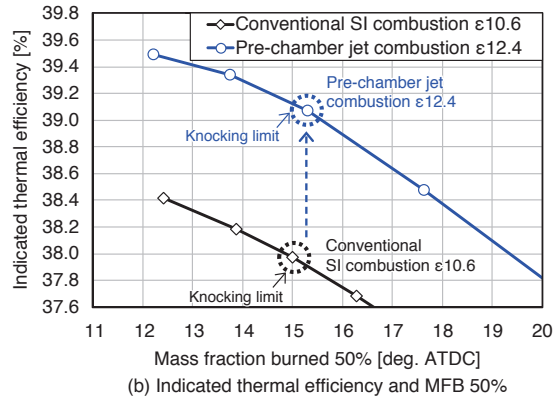


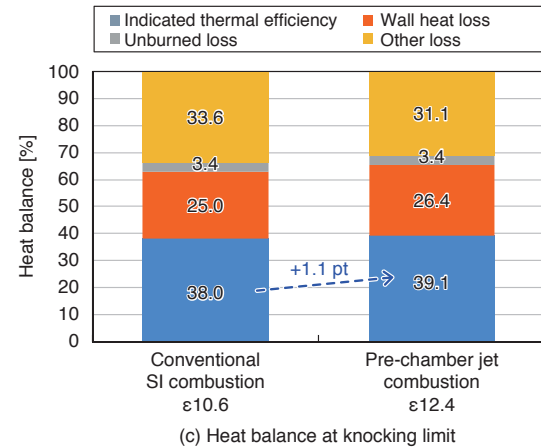
Fig. 22 Engine specifications of conventional SI combustion and pre-chamber jet combustion



(a) MFB 10-90% duration and MFB 50%



(b) Indicated thermal efficiency and MFB 50%



(c) Heat balance at knocking limit

Fig. 23 Experimental results of conventional SI combustion and pre-chamber jet combustion

efficiency of pre-chamber jet combustion with compression ratio of 12.4 was 1.1 pts. higher than conventional SI combustion with compression ratio of 10.6, even after deducting for the increase in heat loss resulting from rapid combustion, as shown in Figs. 23(b) and (c).

6. Conclusion

CFD and an actual engine were used to reveal the following about the knocking mitigation effect and thermal efficiency increasing effect of using pre-chamber jet combustion in comparison to a gasoline stoichiometric SI combustion engine.

- (1) Optimizing pre-chamber hole diameter and enlarging pre-chamber volume advanced the knocking limit MFB 50% from 1 deg. to 2 deg. for each 1 deg. combustion duration that was shortened. This is believed to be because the jets caused the propagation flame to reach the bore-end quickly, which reduced the unburned area of the end-gas and suppressed auto-ignition.
- (2) Selecting 10 holes, which is the jet interference limit in this engine, mitigated knocking by the greatest amount and advanced the knocking limit MFB 50% by 1.5 deg. compared to four holes as a benchmark. This is believed to be because increasing the number of pre-chamber holes and thereby increasing the number of jets partitions the unburned field and makes it smaller, suppressing auto-ignition.
- (3) Pre-chamber jet combustion in an actual single-cylinder engine produced a similar knocking mitigation effect as that found by CFD. By replacing the amount of knocking mitigation with compression ratio and raising the compression ratio by 1.8 pts., this study demonstrated that thermal efficiency increased under the evaluation load by 1.1 pts. compared to a conventional stoichiometric SI combustion engine.

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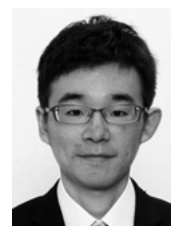
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A Model for Turbulent Flame Speed Based on Theory of Turbulent Flame Brush Thickness Dynamics

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ABSTRACT

In this work, a new theory is proposed for determining the instantaneous turbulent flame speed using a simplified but physics-based approximation of the dynamics of the transient turbulent flame brush thickness. A summary of the main points of this new model's derivation are reviewed, which can be thought of as an extension of Damköhler's steady state flame speed model to include the effect of the transient development of a turbulent flame. The resulting flame brush thickness dynamics model is compared against historical measurements available in the literature as well as against 3D realizations of resolved turbulent flame propagation using the instantaneous G-equation. This comparison shows the newly derived model can describe both the transient and steady state turbulent flame speed across a range of conditions with a singular modeling form and is therefore expected to find utility as turbulent flame speed model for combustion devices where representation of the developing turbulent flame is of crucial importance.

1. Motivation and Premise of the Present Analysis

Replacement of the chemical source terms in the energy equation with an alternative flame kinematic term is a convenient simplification that is commonly applied in turbulent combustion modeling. This has given rise to the concept of a turbulent flame speed and for decades researchers have proposed numerous models to characterize the propagation rate of the wrinkled turbulent flame.

The common approach for modeling the instantaneous turbulent flame speed, s_T , simplifies the problem by separating the steady state flame speed, $s_{T,\infty}$, from the transient response of the flame speed shown by the right-hand side (RHS) of Eq. (2).

$$\frac{s_{T,\infty}}{s_L} = 1 + \frac{\bar{u}'}{s_L} \quad (1)$$

$$\frac{s_T}{s_{T,\infty}} = 1 - \exp(-\tau/t) \quad (2)$$

There are several examples of steady state turbulent flame speed models available in the literature with a similar form to Eq. (1). A selection of these models⁽¹⁾⁻⁽¹⁵⁾ have been plotted in Fig. 1. Note that despite having the same input

conditions⁽¹⁶⁾, the range in predicted values of the turbulent flame speed ratio, s_T/s_L , at a given relative turbulence intensity, $\bar{u}'/s_L$, spans across an order of magnitude.

This variation in modeled turbulent flame speed ratios is

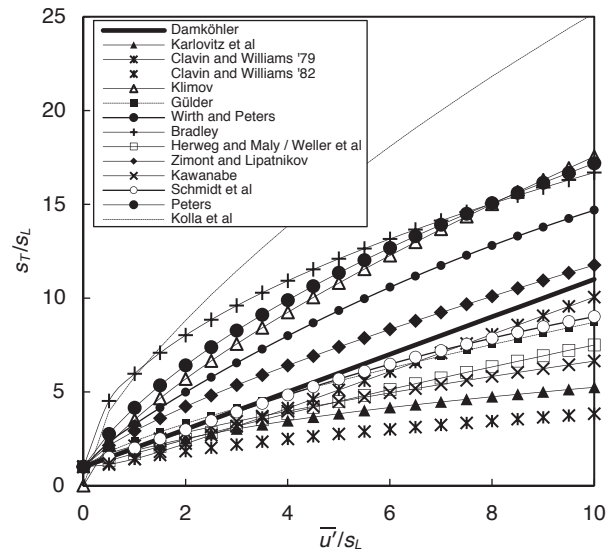


Fig. 1 Comparison of steady state flame speed ratios predicted by various models at typical engine conditions

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reflective of the range of the observed turbulent flame speed ratios in the literature. Turbulent flame speed measurements compiled by Abdel-Gayed⁽¹⁷⁾⁻⁽²³⁾ from all known turbulent flame speed measurements at the time are shown in Fig. 2. Note the similar span of values in measured turbulent flame speed ratios compared to the predicted values presented in Fig. 1. Indeed, many of these models referred to this same measurement database during their development.

Models of the turbulent flame speed shown in Fig. 2 tend to be empirical fits rather than phenomenological or physics-based models. This empiricism in the turbulent flame speed models will lead to necessary modeling calibration of the turbulent flame speeds during their application. Ultimately this limits the ability of many flame kinematics-based approaches to represent turbulent combustion processes rather than simply representing the details of a specific dataset used to train a particular model. As it is desirable to minimize this type of empiricism while increasing representation of the relevant turbulent flame propagation phenomena, the primary focus of this work was to develop a simple, physics-based model capable of capturing the macroscopic dynamic phenomena of turbulent flame propagation.

An alternative to the typical empirical fitting functions encountered for the steady and transient development of s_T , like Eqs. (1) and (2), might be to identify the underlying influential terms that lead to the dynamic response of the flame speed and then to allow the steady state condition to be developed naturally once dynamic equilibrium has been reached. A model of transient turbulent flame speed based on this principle of dynamic equilibrium does not appear to have been widely explored throughout the literature. This concept will be the main approach investigated here.

The remaining structure of this paper is as follows:

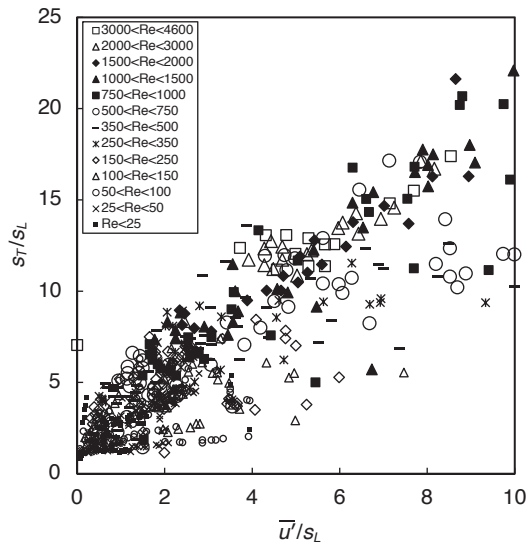


Fig. 2 Steady state turbulent flame speed measurements compiled from many sources by Abdel-Gayed

Chapters 2-4 discusses a detailed analysis of resolved 3D turbulent flame propagation using the G-equation, which was used to better understand the underlying mechanics of turbulent flame propagation. The results of this analysis lead to the development of a new Turbulent Flame Brush Thickness Dynamics (FBTD) model which is derived in Chapter 5. The FBTD model results are first compared against the measurement results of Abdel-Gayed in Chapter 6, and then against realized flame propagation of the instantaneous 3D G-equation through a synthetic frozen turbulent flow field in Chapter 7. Finally, a summary of the key findings and discussion of the results is given in Chapter 8. Readers should note that the present work is a summary of the main findings from a more comprehensive investigation by the author which is available elsewhere in the literature⁽¹⁶⁾.

2. The G-equation

A simple yet fundamental model for studying the details of a developing turbulent flame is the instantaneous G-equation of Williams⁽²⁴⁾,

$$\frac{\partial G}{\partial t} + \frac{\partial G}{\partial x_f} \vec{u} = s_L \left| \frac{\partial G}{\partial x_f} \right| \quad (3)$$

The G-equation is a member of the level set approach⁽²⁵⁾,⁽²⁶⁾ where a self-propagating interface interacts with an embedded velocity field. If the instantaneous velocity vector, $\vec{u}$, can be attributed to the fluctuating velocity of turbulence, and if the local normal propagation speed of the G_0 interface can be attributed to the laminar flame speed, s_L , then the resulting motion of the G level set at G_0 can be interpreted as an instantaneous realization of a turbulent flame, and so is suggested to be a suitable model for the purposes of the present analysis.

The instantaneous G-equation will be simulated for a set of conditions that are relevant for practical devices. The results will then be studied with the intent of ascertaining, and then modeling, the underlying influential behaviors most responsible for the unsteady and dynamic behavior of turbulent flame propagation.

The use of the G-equation for this purpose requires several assumptions to be made. The effect of turbulent motions on the flame are implicitly assumed to be entirely kinematic in nature, which is consistent with the flamelet regime of combustion⁽²⁷⁾. Equivalently, this assumption implies the chemistry is infinitely fast and therefore the flame surface is infinitely thin, both suitable approximations for typical premixed engine combustion. The use of William's form of the G-equation does not include stretch corrections to the laminar flame speed, and this assumption is less justifiable, as even weakly turbulent flames have regions of the flame surface which experience significant curvature effects⁽²⁸⁾. However, such impacts are presumed

to be highly local and short-lived phenomena, and so it has been decided to neglect these impacts in the aim of simplicity. Additionally, inclusion of the isothermal assumption makes the solution of the G-equation trivial, as the flame propagation becomes decoupled from the flow, and was already decoupled from the chemistry by the flamelet assumption. Research has indicated that the expansion ratio across the flame have only a very weak dependence on the turbulent flame speed⁽²⁹⁾, and so once again this assumption will be kept in the aim of simplicity.

The input laminar flame speed can be defined arbitrarily as it represents an external source term. However, a realistic instantaneous turbulence velocity field is required to continue the analysis, and this cannot be defined arbitrarily. In prior works the fluctuation velocity is frequently precomputed⁽³⁰⁾ using a grid structure or other means of inducing turbulent flow. In some examples a solenoidal fluctuating velocity is specified using a Gaussian random variable which conforms with a chosen energy spectrum⁽³¹⁾. In either case, coupling of the flow solver with an intense swirling flow is known to rapidly dissipate the fluctuating velocity unless a source of energy is used to sustain the fluctuations. To help prevent this, forcing functions have been used to artificially maintain the fluctuating velocity by reintroducing energy at a desired filter length⁽³²⁾, where typically this length is associated with the integral length scale of turbulence.

In this work a different approach will be used to mitigate the effects of turbulence energy dissipation. The flow field will be treated as frozen and non-interacting with the flame propagation, which when combined with the assumptions discussed above greatly reduces the numerical burden of the analysis while still providing insight into the fundamental transient behavior of a developing turbulent flame.

The details of this method will be discussed in the following chapter.

3. Synthetic Turbulent Flow Field

The flow field will be synthesized using the following scheme which was devised to generate a realistic turbulent flow field with a high degree of control during initiation without requiring a detailed solution of the associated conservation equations. First the desired length scale of turbulence, l_0 , is passed as an input which sets the overall scale of the problem to be solved. The size of the box-shaped computational domain is then specified according to the number of eddy lengths desired in each of the three Cartesian coordinates; N_i, N_j, N_k . The total volume of the computational domain, V_{cd} , can be estimated by,

$$V_{cd} = N_i N_j N_k l_0^3 \quad (4)$$

If an individual eddy can be assumed to have a local circular cross-sectional area, and if it spans the domain with a length $V_{cd}^{1/3}$, then the expected volume of an individual

eddy can be estimated as,

$$V_{eddy} = \frac{\pi}{4} l_0^2 V_{cd}^{1/3} \quad (5)$$

This leads to an estimate for the total number of eddies required to fill the computational domain to the desired span as,

$$N_{eddy} = V_{cd} / V_{eddy} = V_{cd}^{2/3} / \left(\frac{\pi l_0^2}{4} \right) \quad (6)$$

Substituting (4) into (6) provides the total number of eddies to be initialized in terms of the eddy lengths desired in the three Cartesian coordinates.

$$N_{eddy} = \frac{4}{\pi} (N_i N_j N_k)^{2/3} \quad (7)$$

To initialize the velocity field the location of each individual eddy within the computational domain should be anchored in space. These datum points are positioned within the subsections of the entire domain using randomly assigned coordinates, with the intention of randomly, but evenly, distributing the eddies. Each eddy's 3D centerline is specified in the form of a randomly determined 2nd order parametric polynomial of the form,

$$L = L_i \hat{i} + L_j \hat{j} + L_k \hat{k} \quad (8)$$

$$L_n(t) = \alpha_n t^2 + \beta_n t + \gamma_n \quad (9)$$

Where α, β, γ are randomly determined coefficients for each of the n dimensions (i, j , and k) of the centerline specified for each eddy along the parametric dimension t .

The 3D velocity field of an individual eddy is then determined by using the normal to the vortex center line by use of a proposed variant of the solenoidal 2D Lamb-Oseen vortex. The tangential velocity of the Lamb-Oseen vortex is given by,

$$u_\theta = \frac{\Gamma}{2\pi r} g(r, t) \quad (10)$$

Where the first term represents the tangential velocity profile of an irrotational vortex from 2D potential flow theory, and the function $g(r, t)$ represents the viscous core of a rotational vortex given by,

$$g(r, t) = 1 - \exp(-r^2/4\nu t) \quad (11)$$

The Lamb-Oseen vortex is intended to model a decaying line vortex over time due to viscosity, but it can be modified to allow the specification of a rotational vortex core of diameter, l_0 , by replacing the function $g(r, t)$ with the following,

$$\gamma(r, t) = 1 - \exp\left(-\frac{1}{2}\left(\frac{r\pi}{l_0}\right)^2\right) \quad (12)$$

The variable r is the radial-normal distance to the vortex centerline. Example plots of the modified Lamb-Oseen vortex described by Eqs. (10) and (12) are shown in the Fig. 3 below. It is assumed that the rotational vortex core which contains the vorticity and rotational inertial of the turbulent eddies is a likely candidate feature of the vortex to be associated with the integral length scale of turbulence as a convenience. As justification, it can be seen than the radial distance between the peak tangential velocities across the diameter of the eddy is found to be near the specified

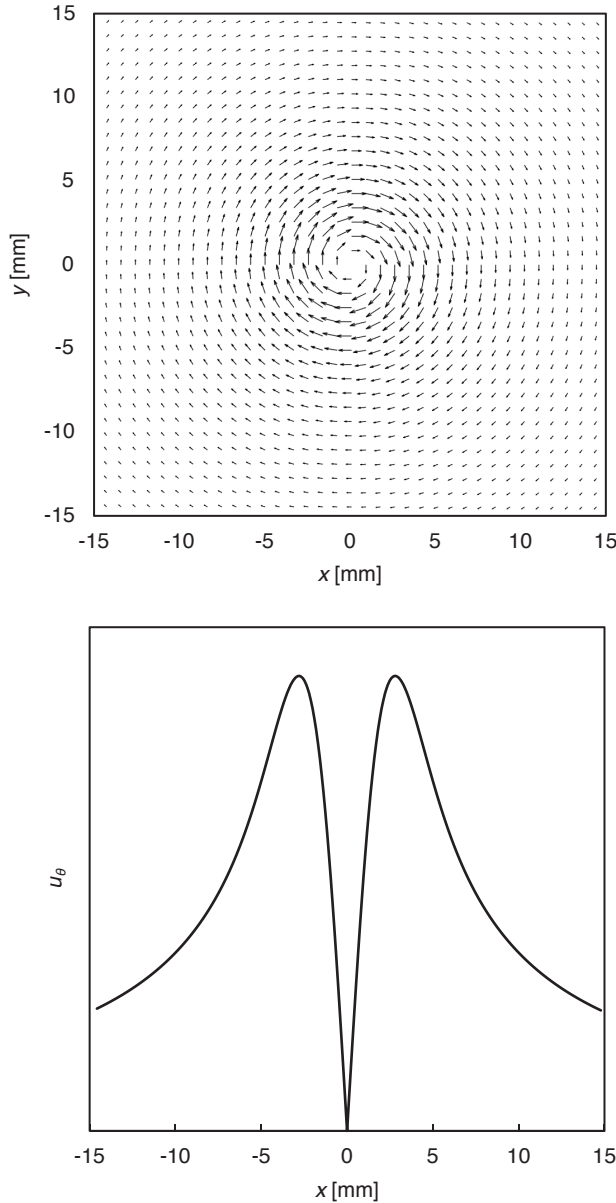


Fig. 3 Example 2D vortex velocity vectors and tangential centerline velocity from Eqs. (10)-(12)

integral length scale of 3 mm.

Next the radial distance r will be determined for every cell center in the computational domain. In a 2D profile this can simply be found as the radial distance within the plane to the specified vortex center point. In 3D however, the radial distance r of an arbitrary point is defined as the shortest distance from that point to the vortex centerline. This shortest distance can be found by solving the following optimization problem,

$$t_{min} \rightarrow \min(\text{dist}\{p, L(t)\}) \quad (13)$$

The minimum distance found with the equation above is associated with the optimal parametric value of $t_{min}(i, j, k)$ which is unique for each point $p(i, j, k)$ in the domain for every individual eddy. Point p has coordinates p_i, p_j, p_k , and point $L(t)$ has components L_i, L_j, L_k at the parametric line distance t , each defined in Euclidean space. Having determined these geometric points within the domain, the radial distance r can then be found by application of the Pythagorean theorem by,

$$r = \left[(L_i(t_{min}) - p_i)^2 + (L_j(t_{min}) - p_j)^2 + (L_k(t_{min}) - p_k)^2 \right]^{1/2} \quad (14)$$

The figure below shows the graphical relationship between the parametric curve, $L(t)$, the arbitrary point within the domain, p , the closest point on the curve L to point p shown at $L(t_{min}, p)$, and finally, the distance r between these two points.

The total velocity field is then found by superimposing the individual contribution from N_{eddy} number of eddies within the volume V_{cd} . In Fig. 5 below, the isosurfaces of velocity magnitude after the frozen turbulent flow field has been fully synthesized are shown. As a qualitative comparison, an image of DNS isotropic homogenous box

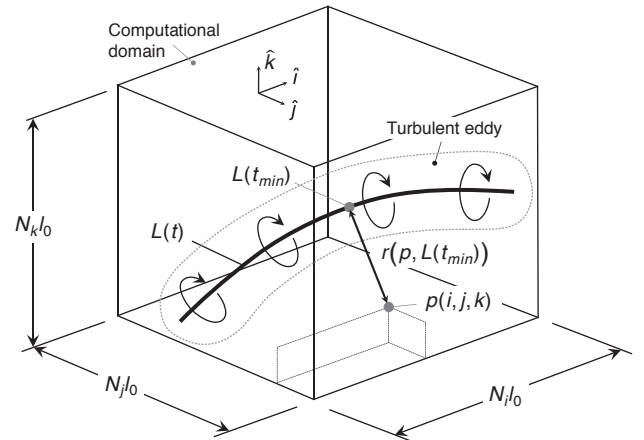


Fig. 4 Graphical representation of the distance r between points p and L at t_{min}

turbulence simulations of Chang et al⁽³³⁾ are presented in Fig. 6 using the Q-criterion, which after review suggests that the artificial flow field synthesized is reasonable.

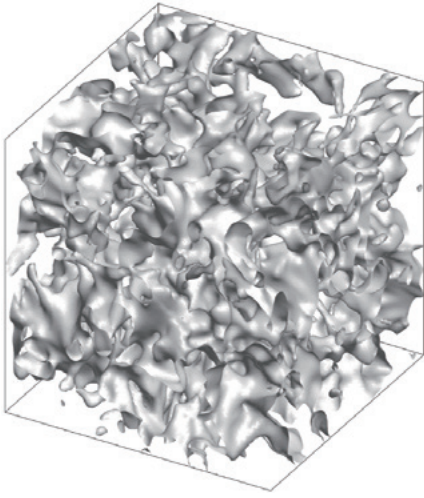


Fig. 5 Synthetic turbulent flow field initiation with 150 eddies inside the computational domain

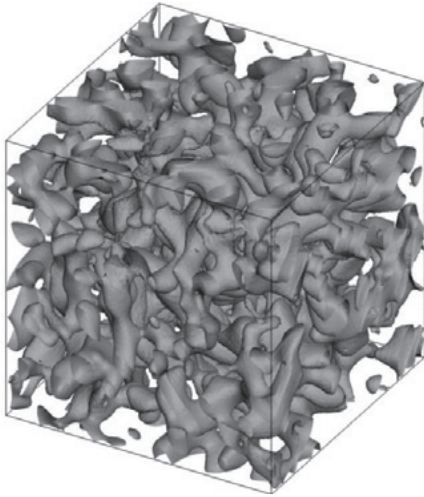


Fig. 6 DNS results of an isotropic turbulent flow by Chang et al

4. The G-equation Results

Following from the scheme described in Chapter 4, the box shaped computational domain was initiated using the following values for the relevant parameters: $l_0 = 2$ mm, $N_i = 20$, $N_j = 20$, $N_k = 40$, $N_{eddy} = 809$, $V_{cd} = 128,000$ mm³, #cells = $120 \times 120 \times 240$.

Shown in Fig. 7, the G field was initiated to give an initially planar flame surface located just inside the domain to avoid the possibility of eddy velocity advecting

a portion of the flame surface outside of the computational domain where it can no longer be tracked. The G_0 level set value was chosen arbitrarily as 0.5, and G upstream of this value was initialized to 0, while downstream values were initialized to 1. Cyclic boundary conditions were implemented to walls normal to the $\hat{i}$ and $\hat{j}$ directions and fixed values of G were applied to the walls normal to the $\hat{k}$ direction.

Figure 8 shows nine sequential timesteps of the G-equation solution for $\bar{u}'/s_L = 1$. The initially planar surface undergoes a significant transient period while the

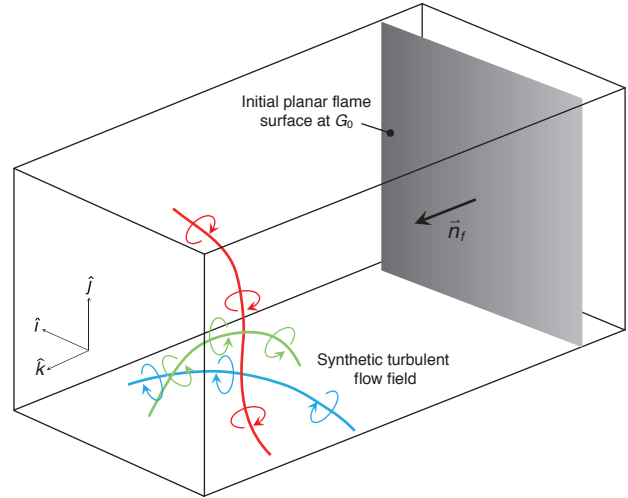


Fig. 7 Depiction of the initially planar flame surface at G_0 propagating into a synthetic frozen turbulent flow field

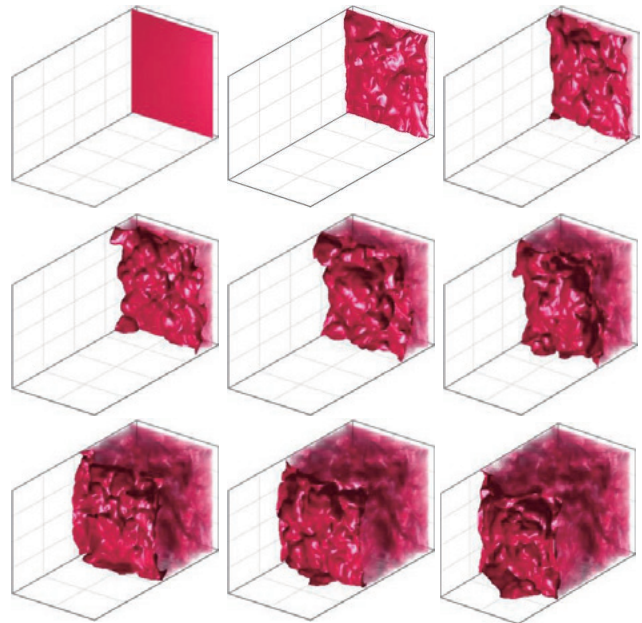


Fig. 8 Example G-equation results of the transient development of an initially planar flame unity turbulence intensity

turbulent flow begins to wrinkle and distort the surface before ultimately reaching a quasi-steady propagation rate.

After careful study of these representations of turbulent flame propagation like shown in Fig. 8, it became apparent to the author that the dynamics of the flame brush thickness might be usable as a simpler means of predicting the turbulent flame speed.

Traditionally, the mechanism for the increased consumption rate of turbulent flames has been associated with the increased wrinkled flame area, A_L . This is shown explicitly in modeling equalities like,

$$\frac{dm_b}{dt} = \rho_u A_L S_L = \rho_u A_T S_T \quad (15)$$

Which lead to,

$$s_T = \frac{A_L}{A_T} s_L \quad (16)$$

Where for a fixed laminar flame speed, Eq. (16) shows that increasing the wrinkled laminar flame surface area A_L will result in a proportionally faster apparent turbulent flame speed, s_T . These conclusions are not debated here. As depicted in Fig. 9, increasing turbulence intensity will increase the wrinkling of the flame, but additionally it can be seen that on average more of the flame is not

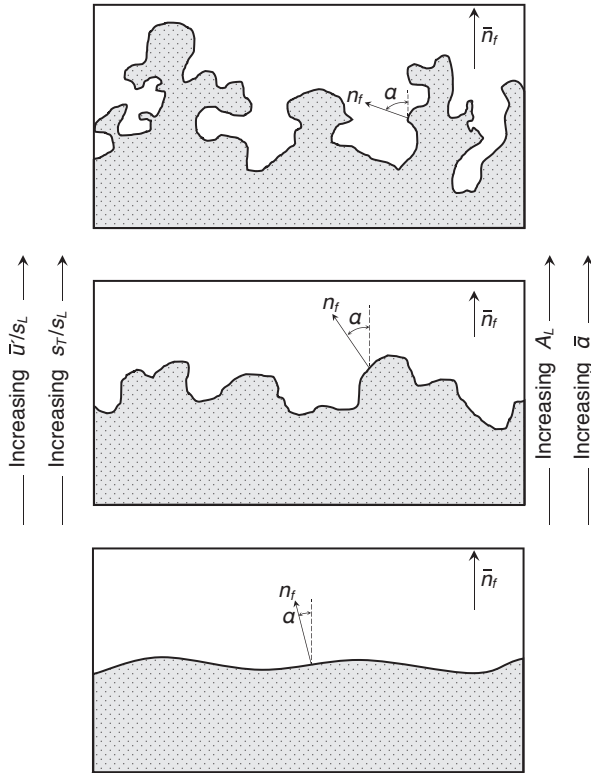


Fig. 9 Depiction of influence to various turbulent flame properties as turbulence intensity is increased

locally propagating along the mean vector, $\bar{n}_f$, but instead propagating locally along n_f at some angle α with respect to that mean propagation vector.

The statements above are not a new observation, and in fact many turbulent flame speed models already implicitly include this influence of a deviating flame angle with increasing turbulence intensity. For example, this is essentially the same concept that led to Damköhler's original turbulent flame speed model⁽¹⁾ shown here,

$$\frac{s_{T_{\infty}}}{s_L} = 1 + \frac{\bar{u}'}{s_L} \quad (17)$$

Where it was assumed the mechanism for enhanced flame speed was due to a proportional increase in the flame angle α and not strictly due to the increasing flame area itself. However, Damköhler's model only intended to predict the steady state turbulent flame speed and does not consider the dynamic relationship between these flame variables. These kinds of dynamic interactions will now be captured by introducing a new theory of turbulent flame brush thickness dynamics.

5. Turbulent Flame Brush Thickness Dynamics

An approximation for the mean flame angles $\bar{\alpha}$ and $\bar{\theta}$ can be found from the stencil in Fig. 10 below. Here the same characteristic triangle is used that Damköhler applied, but the characteristic lengths of the sides of the triangle have been assigned to the flame brush thickness and the integral length scale of turbulence, rather than the laminar flame area and the turbulent flame area.

By referencing the geometry in the figure below, it can be seen that an approximation for the mean flame angle, $\bar{\theta}$, can be found by,

$$\tan \bar{\theta} = l_{fb}/l_0 \quad (18)$$

For a very thin flame brush ($l_{fb} \rightarrow 0$) the angle $\bar{\theta}$ approaches zero radians, and for a very thick flame brush ($l_{fb} \rightarrow \infty$), the angle $\bar{\theta}$ approaches $\pi/2$ radians. The angle $\bar{\theta}$

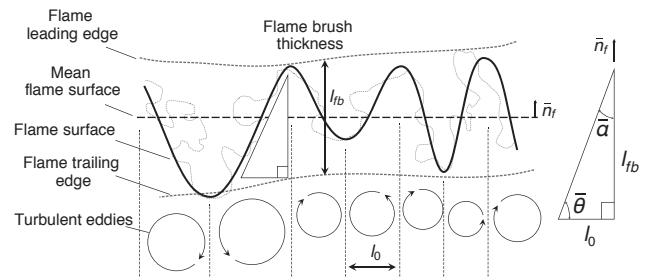


Fig. 10 Turbulent flame angle relationship by mean turbulence and turbulent flame geometrical length scales

can be thought of as the mean local deviation angle from the global mean propagation vector. Therefore, when considering a purely laminar flame that has no flame brush thickness, $\bar{\theta}$ is zero. This indicates that locally the flame is propagating in the mean normal vector everywhere, and that there is no deviation from the mean, as would be expected. While a flame brush approaching infinite thickness would yield $\bar{\theta}$ that is approaching $\pi/2$ radians. Indicating that an infinitely thickened flame brush is propagating perpendicularly to the mean on average.

5.1. Flame Brush Thickness Growth

Figure 11 shows a depiction of an initially zero thickness flame brush being thickened by a field of nearly homogeneous and isotropic turbulence across a span of finite time Δt . Note that while isotropic turbulence has no statistical directional preference, it is the aligned eddies that have the greatest contribution to the instantaneous flame brush thickness growth. If a turbulent eddy is assumed to be a vortex or swirling region of flow where a local axis of rotation can be defined, then it is the eddies whose rotation axis are aligned perpendicularly to the mean propagation vector, $\bar{n}_f$, which have the highest effective brush thickness growth. This has been shown explicitly in the 2D depiction of the flame brush growth process below of Fig. 11.

Here the common counter rotating vortex structure has been assumed. This implies that aligned eddies can both advance and retreat the front as the flame passes through the counter rotating vortex pair. Using this stencil an estimate for the discrete change in the flame brush thickness due only to the effect of turbulence, and ignoring all other factors, is given by,

$$\begin{aligned} \left. \frac{\Delta l_{fb}}{\Delta t} \right|_{growth} &= \Delta t \{ (s_L + u_{ch}) - (s_L - u_{ch}) \} \\ &= 2u_{ch}\Delta t \end{aligned} \quad (19)$$

Where u_{ch} is a characteristic turbulence velocity associated with the maximum and minimum instantaneous velocities normal to $\bar{n}_f$. Due to local rotational symmetry

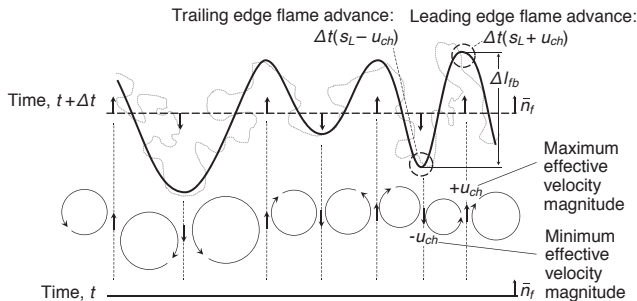


Fig. 11 Stencil of an initially planar flame at time t being thickened by a field of eddies after a discrete increment of time

of turbulent eddies, the maximum and minimum values are assumed to be equal in magnitude but opposite in sign.

The characteristic velocity u_{ch} is an inherently instantaneous quantity, but it can be estimated from the statistical $\overline{u'}$ by assuming that the rms fluctuation velocity is a Gaussian distributed random variable. Higher order statistics of small scale turbulence are known to be non-Gaussian⁽³⁴⁾, but the statistics of the large scale turbulence, especially near the core of the probability density function, are often assumed to be quite well approximated by a Gaussian distribution⁽³⁵⁾. As it is explicitly assumed that the energy containing eddies are the ones most responsible for thickening of the flame brush, a suitable estimate for the maximum instantaneous fluctuation velocity experienced by the flame over the interval Δt can be found from,

$$u'_{max} = u'_{min} \approx 3\bar{u}' \quad (20)$$

Where it has been assumed that six standard deviations cover the statistical range of the fluctuation velocity for both the maximum and minimum instantaneous fluctuation velocity.

Following along the approximate flame streamlines (the dotted vertical lines in Fig. 11), it is assumed that the instantaneous fluctuation velocity experienced by the flame varies between u'_{max} and u'_{min} and follows in a nearly sinusoidal path according to the wave length $2l_0$. This allows the average characteristic velocity experienced by the flame brush, u_{ch} , to be estimated by finding the average positive values of the sine wave over half of one cyclic period,

$$u_{ch} \approx u'_{max} \frac{1}{\pi} \int_0^\pi \sin\phi d\phi \quad (21)$$

The mean characteristic velocity of the turbulent fluctuations is then found to be,

$$u_{ch} \approx \frac{6}{\pi} \bar{u}' \quad (22)$$

Substituting Eq. (22) into (19) and taking the limit as $\Delta t \rightarrow 0$ we find the following differential modeling approximation for the flame brush thickness growth term as a function of the rms fluctuation velocity.

$$\begin{aligned} \lim_{\Delta t \rightarrow 0} \left. \frac{\Delta l_{fb}}{\Delta t} \right|_{growth} &= \left. \frac{dl_{fb}}{dt} \right|_{growth} \\ &= \frac{12}{\pi} \bar{u}' \end{aligned} \quad (23)$$

5.2. Flame Brush Thickness Decay

In the absence of other effects, a thickened flame brush will tend to thin over time due to self-annihilation caused by regions of the flame surface intersecting with one another. This effect is also commonly referred to as the kinematic

restoration^{(36), (37)} of the flame surface area. An estimate for the discrete quantity of the flame brush thickness that decayed over the time interval Δt can be determined from the stencil shown in Fig. 12. In this image a segment of the thickened flame brush from Fig. 11 has been depicted as if in isolation from the turbulence.

In the image below it can be seen that the trailing edge of a wrinkled self-normal propagating surface has an apparently faster rate of travel as compared to the leading edge. This is due to the self-annihilation mechanism which is a function of the mean local deviation angle, $\bar{\theta}$. An estimate for the discrete change in the flame brush thickness due only to the effect of self-annihilation and ignoring all other factors is therefore given by,

$$\begin{aligned} \left. \frac{\Delta l_{fb}}{\Delta t} \right|_{decay} &= \Delta t \{ (s_L) - (s_L / \cos \bar{\theta}) \} \\ &= s_L (1 - \sec \bar{\theta}) \Delta t \end{aligned} \quad (24)$$

As before with the brush thickness growth term, the time derivative for the brush thickness decay due to self-annihilation is found by rearranging Eq. (24) and taking the limit as $\Delta t \rightarrow 0$.

$$\begin{aligned} \lim_{\Delta t \rightarrow 0} \left. \frac{\Delta l_{fb}}{\Delta t} \right|_{decay} &= \left. \frac{dl_{fb}}{dt} \right|_{decay} \\ &= s_L (1 - \sec \bar{\theta}) \end{aligned} \quad (25)$$

5.3. Freely Propagating Turbulent Flames

A dynamic equation for describing a freely propagating turbulent flame brush is given below, where freely propagating in this context specifically excludes the thickness destruction due to flame-wall interaction. These effects will be considered later in Section 5.5. Here a

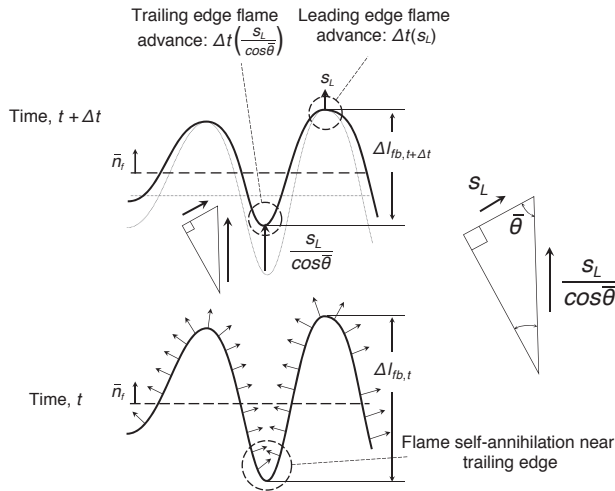


Fig. 12 Depiction of a thickened flame brush at time t thinning by self-annihilation after discrete time increment Δt

freely propagating turbulent flame brush is assumed to be influenced only by thickness growth due to turbulent stretching and thickness decay due to self-annihilation.

$$\left. \frac{dl_{fb}}{dt} \right|_f = \left. \frac{dl_{fb}}{dt} \right|_{growth} + \left. \frac{dl_{fb}}{dt} \right|_{decay} \quad (26)$$

Substituting in terms defined in Sections 5.1. and 5.2. yields the following,

$$\left. \frac{dl_{fb}}{dt} \right|_f = \frac{12}{\pi} \bar{u}' + s_L (1 - \sec \bar{\theta}) \quad (27)$$

Note that $\bar{\theta}$ is a nonlinear function of the flame brush thickness from the definition below,

$$\bar{\theta} = \tan^{-1}(l_{fb}/l_0) \quad (28)$$

It is seen that the derived equation for the freely propagating turbulent flame brush thickness dynamics, Eq. (27), is a first order nonlinear ordinary differential equation. It describes that an initially thickness-free flame brush ($l_{fb} = 0$) subjected to constant input conditions of rms fluctuation velocity, integral length scale of turbulence, and laminar flame speed will first rapidly thicken due to the initially zero-magnitude decay term on the RHS of Eq. (28), followed by a declining rate of change in brush thickness over time as the magnitude of the decay term grows due to the increasing influence of self-annihilation for a thickened flame. Finally, a steady state value of brush thickness will be reached once the growth and decay terms have equal magnitudes but opposite signs. This signifies that the process has reached the dynamic equilibrium condition discussed at the introduction.

An example of the characteristic dynamic response of the flame brush thickness approaching dynamic equilibrium is given in the Fig. 13 for the fixed values of $s_L = 1$ m/s, $\bar{u}' = 1$ m/s, and $l_0 = 1$ mm.

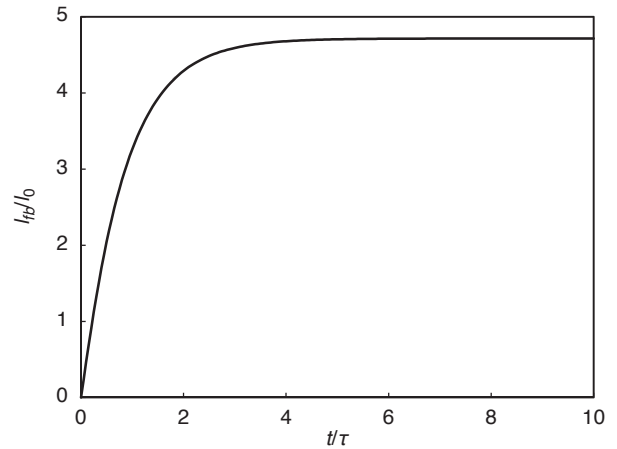


Fig. 13 Dynamic response of an initially thickness-free turbulent flame brush for constant conditions

5.4. Turbulent Flame Speed Interpretation

Although the derivations given in the preceding sections for the flame brush thickness dynamics equations have not specifically stated so, the information needed to interpret the transient motion of a rapidly propagating mean surface in the presence of turbulence from these equations has already been included. This rate of displacement rate of the flame brush thickness centerline is the same transient turbulent flame speed which was desired to be estimated with a simple physical model.

In Fig. 12 at time $t+\Delta t$, it is seen that the mean flame surface has advanced more rapidly than the leading edge of the flame when isolated from external effects. This is a consequence of the more rapid consumption of reactants near the trailing edge of a flame than near the leading edge of the flame, embodied by the decay term given in Eq. (27).

In the absence of turbulence, the mean flame surface is expected to propagate at the laminar flame speed relative to the unburned gas. In the presence of turbulence the flame brush thickens, leading to more rapid consumption and propagation near the trailing edge of the flame. A suitable approximation for the free turbulent propagation speed of the mean flame surface can therefore be given by,

$$s_T = s_L - \frac{1}{2} \frac{dl_{fb}}{dt} \Big|_{decay} \quad (29)$$

The first term on the RHS of the equation above represents the non-turbulent propagation, and the second term represents the apparent advancement of the front due to a reduction in the flame brush thickness. If the flame brush thickness is not decreasing or has a zero value, turbulent flame speed simply reduces to the laminar flame speed, as would be expected. Also note that if the flame brush thickness is strictly increasing, in the absence of other effects, the brush is still assumed to be advancing at the laminar flame speed as the increasing thickness is assumed to occur proportionally to the mean brush center line, both in the directions of leading and trailing edges equally. The leading coefficient on the second term represents that the mean flame surface is located at the flame brush centerline halfway between the leading and trailing edges of the flame, respectively. A discrete reduction in the brush thickness would therefore lead to an apparent advancement of the brush center by one half the brush thickness reduction magnitude.

Substituting in the definition for the brush thickness decay rate and rearranging gives a model for the instantaneous turbulent flame speed ratio s_T/s_L for a freely propagating flame,

$$\frac{s_T}{s_L} = \frac{1 + sec\bar{\theta}}{2} \quad (30)$$

The steady state turbulent flame speed can be found by setting the left-hand side of Eq. (27) to zero, substituting

terms into Eq. (30) and simplifying. This results in the FBTD-based equation below for the steady state turbulent flame speed,

$$\frac{s_{T,\infty}}{s_L} = 1 + \frac{6}{\pi} \frac{\bar{u}'}{s_L} \quad (31)$$

The equation above recovers the commonly proposed form of the steady state turbulent flame speed correlations embodied by Eq. (1). The historical model which most closely resembles the new FBTD model derived here is the same one proposed by Damköhler, and perhaps not unexpectedly, as many of the key interpretations and features of the stencils used in the model derivation were similarly assigned and defined.

5.5. Confined Flame Propagation

When the flame brush can freely interact with the turbulence, its dynamics are expected to be approximated by Eq. (27) and the freely propagating flame brush will ultimately reach a maximum sustainable thickness once the terms on the RHS reach equilibrium. Results for the steady state flame brush thickness for a freely propagating flame found from Eq. (28) over a range of relative turbulence intensities are shown in Fig. 14.

Other than the rapid increase near $\bar{u}'/s_L = 0$, the maximum sustainable brush thickness of a freely propagating flame is predicted to increase linearly with relative turbulence intensity according to the FBTD model. Note that even for the relatively weak turbulence conditions the flame brush is anticipated to be many times thicker than the integral length scale.

A simple conclusion that can be drawn is that a combustor needs to be of a sufficiently large geometrical scale relative to the turbulence length scale in order to contain the freely developing flame brush thickness, or the flame can no longer be truly freely propagating but rather is confined by the combustor. Representing such complicated flame-wall interaction in a simple model is unlikely to be universally valid. However, a coarse approximated criteria has been discovered from the study of the instantaneous G-equation solutions for highly confined flame development. The criteria are given below which classifies flames as those which are either freely developing or developing within confined combustor geometry.

$$\begin{aligned} \text{Freely Propagating Flame: } \frac{2l_{fb}}{d_0} &< 1 \\ \text{Confined Flame: } \frac{2l_{fb}}{d_0} &\geq 1 \end{aligned} \quad (32)$$

Where d_0 is the macro length scale of the geometry of the combustor which inhibits the free propagation of the flame. For rectangular shaped cross sections, it is found that the maximum transverse span tends to be the limitation. An approximate model for the confined flame propagation can

be made by modifying Eq. (27) as below,

$$\left. \frac{dl_{fb}}{dt} \right|_c = \zeta \frac{12}{\pi} \bar{u}' + s_L(1 - \sec \bar{\theta}) \quad (33)$$

Where ζ is the probability that the entire flame brush will be influenced by the fluctuating intensity of turbulence according to the following scheme:

Table 1 Confined flame propagation criteria

Condition:	Criteria:	ζ :
Free	$2l_{fb}/d_0 < 1$	1
Confined	$2l_{fb}/d_0 \geq 1$	$d_0/2l_{fb}$

From the values of ζ given above we can see that when a combustor's geometry is sufficiently large, Eq. (33) becomes equivalent to Eq. (27) for a freely propagating flame. While a very thick flame brush contained within an overly small combustor will experience a reduction in the magnitude of the flame brush thickness growth term due to interference with the walls of the combustor. In the table above, the interference with the combustor is considered in the transverse direction only. Normal direction interference is handled by an additional flame brush thickness destruction term which is discussed elsewhere⁽¹⁶⁾.

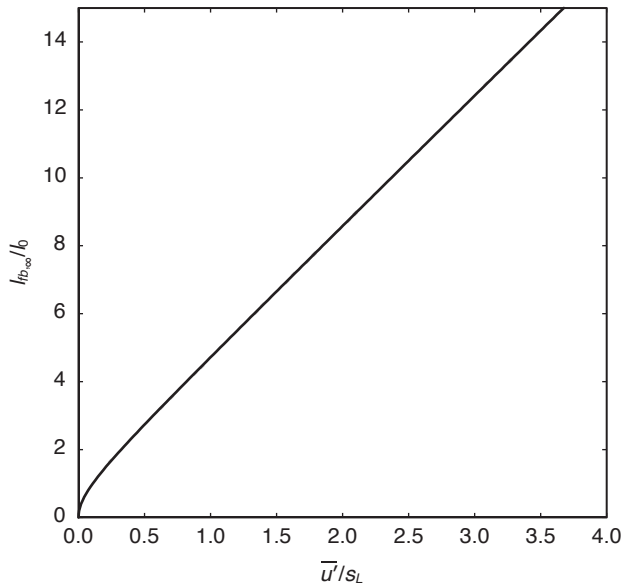


Fig. 14 Steady state flame brush thickness of a freely propagating flame versus relative fluctuation intensity

6. Comparison of FBTD vs Abdel-Gayed Database

The predicted values for a steady state, freely propagating, turbulent flame speed based on the model

derived in the previous chapter have been compared against the compiled database results of Abdel-Gayed in Fig. 15 and are shown as a solid line. We see there is a generally good agreement between results predicted from Eq. (27) and the measurements starting near $s_T/s_L = 1$ and $\bar{u}'/s_L = 0$ that increases linearly with turbulence intensity. Many of these data points were recorded from open flame burners, where the turbulence and the flame were able to interact freely without constraint by the combustor directly and is therefore expected to be well represented by Eqs. (27)-(29) for freely propagating turbulent flames. These data points have been noted in Fig. 15 by the present author as “Freely propagating flames”.

However, the compiled database results also contain a secondary nonlinear trend in the measurements which is not well represented by the equations for freely propagating flames. This so called “bending over” in the measured turbulent flame speed is commonly found in the literature. In the particular case of Fig. 15, most of these points exhibiting the nonlinear trend were collected by Petrov and Talantov⁽³⁸⁾ in the 1960s. Their experiment measured turbulent flame speeds of a stationary inverted flame cone that was stabilized by a burner within a 150 mm diameter pipe flow supplied by grid generated turbulence. They reported laminar flame speeds in the range of 0.2-0.6 m/s and Abdel-Gayed calculated the integral length scales in the range of 2.67-3.9 mm using correlations from Baines and Peterson⁽³⁹⁾.

By setting the left-hand side of Eq. (27) to zero and

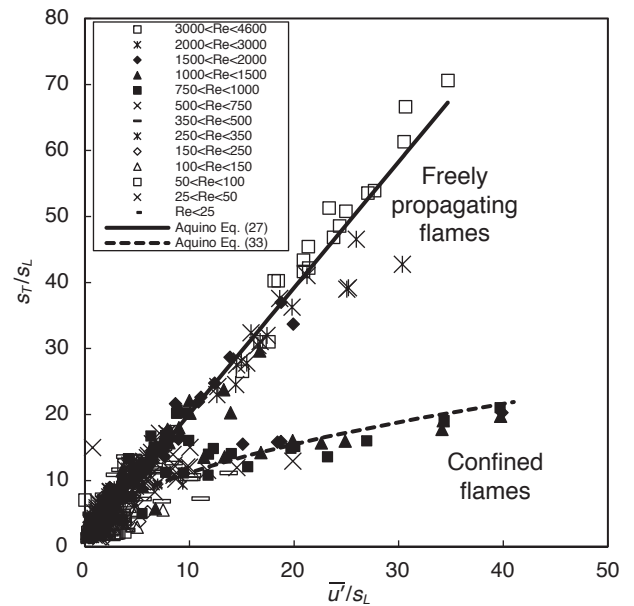


Fig. 15 Comparison of derived steady state turbulent flame speed model against turbulent flame speed measurements compiled by Abdel-Gayed

applying the given characteristic conditions of $\overline{u}/s_L = 50$, $s_L = 0.4$ m/s, and $l_0 = 3.29$ mm, we can solve for the maximum steady state flame brush thickness of a freely propagating flame expected to be found during Petrov and Talantov's experiments as approximately 632 mm. Using the criteria defined in Table 1 we find $2l_{fb}/d_0 \approx 8.5$, and therefore conclude that the flame speed measurements of Petrov and Talantov are highly confined by the combustor and are therefore not expected to be represented by Eqs. (27)-(29) for freely propagating flames. Considering that the inverted flame cone of Petrov and Talantov needs to fit within the combustor's pipe diameter of 150 mm, this analysis demonstrates that a freely propagating flame could not develop under such conditions without the geometry of the combustor interfering with the turbulent flame's propagation.

The dynamics of this type of confined flame brush thickness can be calculated from Eq. (33) using values of ζ between 1-0.118 determined by parameters found from the referenced works above. The confined turbulent flame speed ratios are then found by application of Eqs. (28) and (29). As expected from the prior discussion, the resulting confined turbulent flame speeds were found to be substantially lower than their freely propagating counterparts operating under the same conditions. These confined flame results have been added to Fig. 15 as a dashed line and the datapoints in the vicinity have been noted by the present author as "Confined flames". We can see that both the nonlinear trend versus turbulence fluctuation intensity and the magnitude of the turbulent flame speed ratio are well represented with respect to the measurements. Most of the low fidelity turbulent flame speed models have not attempted to account for these type of macro scale interference effects and therefore might incorrectly associate some aspects of the so-called "bending" phenomena to other mechanisms which are fundamentally unrelated.

7. Comparison of FBTD vs G-equation

Detailed measurements of transient flame development necessary for validating the FBTD model are not frequently present in the literature. Therefore, the dynamics of the FBTD model will be validated by comparison against calculated instantaneous G-equation solutions of resolved turbulent flame propagating for the same input physical conditions. The initialization and solving procedure used for the G-equation analysis here is the same as was discussed in Sections 4.1.-4.3.

Since the flame brush thickness is a macroscopic quantity, some method of spatially filtering the instantaneous solutions of the G-equation needs to be applied in order extract statistical information for the purposes of this comparison. In the case of simple planar flame propagation used here, a 2D sliding interrogation window technique⁽¹⁶⁾ can be implemented to extract the

relevant quantities. A depiction of this approach used to sample from the instantaneous G-equation results is shown in Fig. 16.

Processing and tabulating of the results followed the method proposed by Aquino⁽¹⁶⁾ where the three primary output quantities of interest were chosen to be the mean flame position x_0 , the flame brush thickness l_{fb} , and the flame speed ratio s_T/s_L . Several statistical quantities are reported for the G-equation results as a function of time as the detailed 3D results tend to exhibit a high degree of spatial and temporal variation. The corresponding mean values predicted by the derived FBTD model are then compared against these results in Fig. 17. Note that a more comprehensive comparison between the G-equation results and the proposed FBTD model is available by the author in the literature⁽¹⁶⁾, where the laminar flame speed, the magnitude of the fluctuation velocity, and the turbulence length scale were varied independently of one another in order to test the FBTD model's ability to represent the overall behavior of a developing turbulent flame for a much wider variety of conditions.

In the present work the comparison shown in Fig. 17 is limited to a single realization of turbulent flame propagation for the same conditions discussed at the beginning of Chapter 4. Despite the high degree of local fluctuations present in the G-equation results, the FBTD model was able to represent the mean motion of the flame surface, the dynamic response of the flame brush thickness, the resulting steady state flame brush thickness occurring at dynamic equilibrium, and finally allowed the interpretation of these behaviors into an apparent macroscopic turbulent flame speed of the mean flame surface. It was found that a similar level of agreement to the singular case presented here was also observed as other parameters were varied in general, and this suggests the simplifying assumptions used to derive the FBTD model in Chapter 5 are reasonable approximations for describing the overall dynamic behavior of a developing turbulent flame despite maintaining a relatively simple mathematical form.

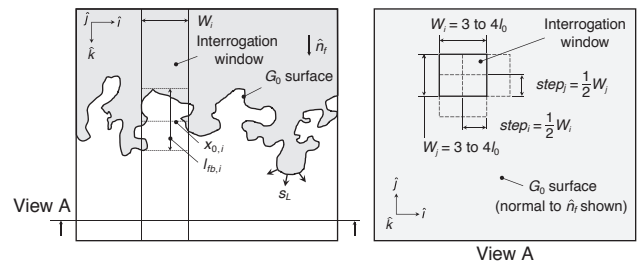


Fig. 16 Sliding interrogation window technique used for extracting statistical information from the instantaneous G-equation results

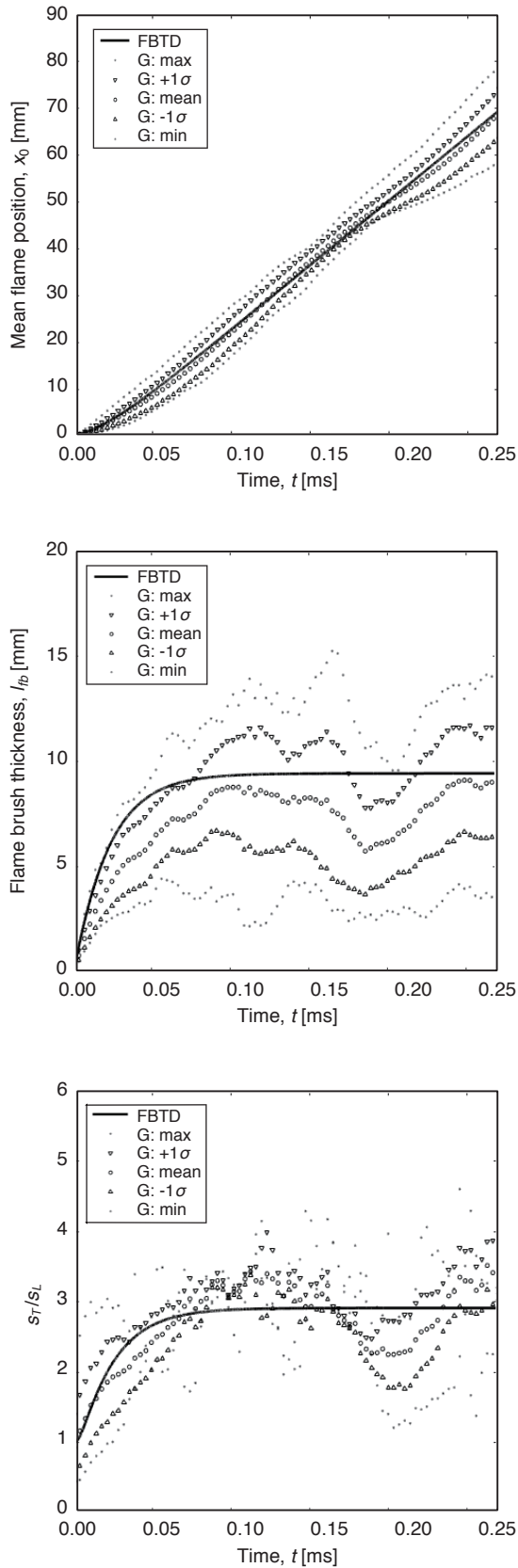


Fig. 17 Comparison of FBTD vs single realization of the G-equation for the same conditions used in Chapter 4

8. Discussion and Conclusions

The primary focus of this work was to develop a simple, physics-based model that could describe the dynamic phenomena of turbulent flame propagation. Analysis of detailed G-equation solutions lead to the concept of the dynamics of the turbulent flame brush thickness being usable as a model for the turbulent flame speed which was derived throughout Chapter 5.

Subsequent analysis found that the FBTD model gave good agreement with the dynamic behavior of the G-equation solution for many developing turbulent flames, where the details of one realization were presented in Fig. 17. Predicted steady state turbulent flame speeds were also found to compared favorably against the compiled measurement database of Abdel-Gayed across a wide range of turbulence intensity and experimental arrangements.

An interesting consequence of FBTD theory is the observation that large spatial dimensions relative to the turbulence length scales are necessary for a turbulent flame to develop to the peak turbulent flame speeds possible under given conditions. Analysis of the historical record indicates that the experimental apparatus used in combustion measurements are occasionally of insufficient dimension to prevent interference with the developing turbulent flame brush, and so although a sampled turbulence intensity may be quite high, the measured turbulent flame speed can be found to be significantly lower than those for otherwise similar operating conditions when either a larger combustor or smaller turbulence length scales are present.

Turbulent flame speed models do not typically contain criterion for the kind of spatial dimension requirement discussed above, and so many models present in the literature may incorrectly associate some aspects of the relationship between the turbulent flame speed and the turbulence intensity to other mechanisms which are fundamentally unrelated. This observation was applied in the analysis of Petrov and Talantov's measurements given in Chapter 6 and the significance of the described effect can be seen in Fig. 15.

8.1. The FBTD Model

$$\left. \frac{dl_{fb}}{dt} \right|_c = \zeta \frac{12}{\pi} \bar{u}' + s_L(1 - \sec \bar{\theta}) \quad (34)$$

$$\bar{\theta} = \tan^{-1}(l_{fb}/l_0) \quad (35)$$

$$\frac{s_T}{s_L} = \frac{1 + \sec \bar{\theta}}{2} \quad (36)$$

The equations listed above are collectively referred to as the FBTD model. Equation (34), along with the definition provided by Eq. (35), are intended to be used for predicting the transient turbulent flame brush thickness for a given

laminar flame speed, rms fluctuation velocity, and integral length scale of turbulence. Equation (36) then interprets the dynamics of the turbulent flame brush thickness to allow calculation of the apparent instantaneous turbulent flame speed.

It is anticipated that the FBTD model may find utility in the analysis of the dynamics of a developing turbulent flame present, like those present in many practical combustion devices such as the common internal combustion engine.

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Phillip AQUINO

Hermetically Sealed Cooling Structure for Small High-current Density Motors

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ABSTRACT

A motor cooling structure for electrification vehicle drive motors capable of absorbing the increases in heat generation from miniaturization with high current has been developed. The conventional cooling method involves dripping refrigerant onto exposed parts of the stator coil, relying on thermal conduction to indirectly cool the coil inside the stator slots where the refrigerant cannot reach. In this study, we have established a technology that directly cools the entire coil by hermetically sealing the stator to immerse the entire coil in automatic transmission fluid. Using a prototype motor, the technology has been demonstrated to have more than twice the cooling performance of the conventional cooling method. It was also confirmed that further enhancements in cooling performance can be expected by increasing refrigerant flow rate.

1. Introduction

In recent years, electrification vehicles such as battery electric vehicles (BEVs) and fuel cell vehicles (FCVs) are rapidly growing in popularity due to demands to reduce CO₂ emissions during vehicle travel. In addition to the existing basic performance requirements of high power, high torque, and high efficiency, the authors see compactness as another important performance requirement for the traction motors used in electrification vehicles. With these electrification vehicle motors using rare metals and copper, which are of concern for material supply risk⁽¹⁾, smaller motors would lead to reductions in materials used.

There are two main methods of reducing motor size. First, motor rotating speed can be increased⁽²⁾. As motor output is the product of torque and rotating speed and torque is proportional to motor size, the motor size can be reduced while maintaining the same motor output by increasing the rotating speed⁽²⁾. Taking this route requires strengthening the rotating bodies in the reduction gear and rotor, which generally requires use of high-strength materials containing rare metals in quantity⁽³⁾. Also, the reduction ratio on the reduction gear will be increased, resulting in increased friction loss and reduced overall efficiency of the drive unit⁽⁴⁾. The other method is to increase torque density and output density by applying high current to the motor. Given the increased heat generation per unit area, this method requires higher cooling performance than offered by

conventional cooling methods^{(5),(6)}.

Given the supply risks of rare metals and other materials, in this study, we propose a hermetically sealed cooling structure with more than twice the conventional cooling performance to achieve a smaller motor by increasing current density. Using the hermetically sealed cooling structure, we succeeded in increasing motor torque density and output density to approximately double their conventional values⁽⁷⁾.

Traction motors are generally cooled using one of two main methods: water cooling, in which refrigerant in the form of Long Life Coolant (LLC) flows through channels formed in the housing, and oil cooling, in which Automatic Transmission Fluid (ATF) as the refrigerant is dripped directly on exposed portions of the coil to cool the motor⁽⁵⁾. In water cooling systems, as the system uses thermal conduction to cool the motor through the stator core, cooling performance is high for the windings inside the slots but weaker at the coil ends outside the stator core⁽⁵⁾. Meanwhile, in oil cooling systems, the coil ends can be cooled directly, but the windings inside the slots where the refrigerant does not reach are cooled by thermal conduction in the conductor itself⁽⁵⁾. Both systems depend on thermal conduction to cool certain sites and struggle at handling the increase in heat generation per unit area needed when reducing motor size. Therefore, in this study, we have developed a method based on oil cooling to directly cool the entire coil, including the slot interior. This paper will open with an overview of the

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cooling technology application. Next, a proof of concept of the cooling effects on a test piece will be explained, followed by challenges with the hermetically sealed cooling structure and how to handle these challenges. Finally, results confirming the effects will be shown from test measurements with an actual prototype motor.

2. Cooling Technology Application

2.1. Target Cooling Performance

The goal of this study was to double existing cooling performance. The following represents the estimated impact that a twofold enhancement in cooling performance would have on motor performance.

First, so as to help ensure sufficient drive power for an electrification vehicle with a single motor, taking into account the ratio of reduction gears and balance of friction, the assumed motor for development was given a maximum torque of 280 Nm, maximum power of 150 kW, and maximum rotating speed of 15,000 rpm. If cooling performance can be doubled from existing levels, current density will also at least be doubled, resulting in torque density and output density of at least 1.7 times existing levels. With this, existing motors could be reduced in volume by 40%.

2.2. Cooling Method Concept

To enhance cooling performance, it is important to reduce thermal resistance between the heating element, or coil, and the refrigerant using heat transfer Eq. (1). According to Eq. (2), thermal resistance is determined by the heat transfer area and heat transfer coefficient. The heat transfer coefficient can be expressed by Eq. (3) using a Nusselt number, a dimensionless quantity indicating whether the heat transfer is convective or conductive. Nusselt numbers positively correlate with Reynolds numbers, which indicate the ratio of inertial force to viscous force. According to Eq. (4), the Reynolds number is proportional to the flow velocity of the fluid. Thus, the heat transfer coefficient can be enhanced by increasing flow velocity. Based on the above, the aim of the proposed structure will be to increase the heat transfer area and, by increasing flow velocity, to enhance the heat transfer coefficient.

$$Q = \frac{(T_{\text{coil}} - T_{\text{ATF}})}{R} \quad (1)$$

$$R = \frac{1}{Ah} \quad (2)$$

$$Nu = \frac{hL}{k} \quad (3)$$

$$Re = \frac{UL}{\nu} \quad (4)$$

Q : Amount of heat removed (W)

R : Thermal resistance (K/W)

T_{coil} : Temperature of coil (K)

T_{ATF} : Temperature of ATF (K)

A : Heat transfer area (m^2)

h : Heat transfer coefficient ($\text{W}/\text{m}^2\text{K}$)

Nu : Nusselt number

L : Characteristic length (m)

k : Thermal conductivity (W/mK)

Re : Reynolds number

U : Flow velocity (m/s)

ν : Dynamic viscosity (m^2/s)

Figure 1 shows a sectional view and overview of a conventional cooling structure⁽⁶⁾, and Fig. 2 shows a sectional view and overview of the proposed cooling system for development. In the conventional cooling structure, ATF refrigerant is dripped on the coil ends from directly above to cool the coil. In the conventional structure, thermal conduction from the coil ends, not the ATF, is used to transfer heat to the coil inside the slots, and the heat transfer area of the coil ends is limited to the surface area where ATF is applied. This limits the total amount of heat removed from the coil as a whole.

As shown in Fig. 2, in the hermetically sealed cooling structure in this study, the entire coil, including inside the slot as well as the coil ends, is immersed in ATF to help ensure a heat transfer area 2.2 times larger than the conventional heat transfer area. Furthermore, the ATF is pressure-fed along the axis of rotation from one coil

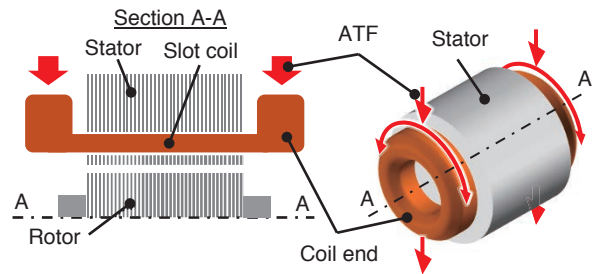


Fig. 1 Conventional motor cooling structure

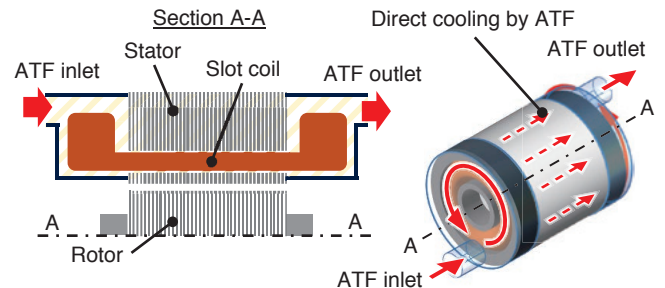


Fig. 2 Proposed cooling method

end through the inside of the slot and discharged out the opposite coil end, increasing the flow velocity of ATF in contact with the coil and enhancing the heat transfer coefficient.

3. Proof of Concept for Cooling Effect

3.1. Thermal Fluid Analysis

The results of a thermal fluid analysis simulating part of the cooling concept described above are given in Fig. 3. Thermal fluid analysis was performed through CFD simulations. The analysis model is a reproduction of a single slot in the motor stator. The ATF flows in from one coil end and through the slot interior to be discharged from the opposite coil end. For the boundary conditions, flow velocity at the inlet was constant, discharge at the outlet was by atmospheric release, and heat flux on the outer surfaces of the stator yoke and covers were set to zero. To simulate heat release from the coil, the coil wall surfaces were given a heat flux equivalent to twice the heat generated by the motor used in the conventional cooling system.

The CFD results show that the maximum temperature is below the heat resistance temperature of the coil.

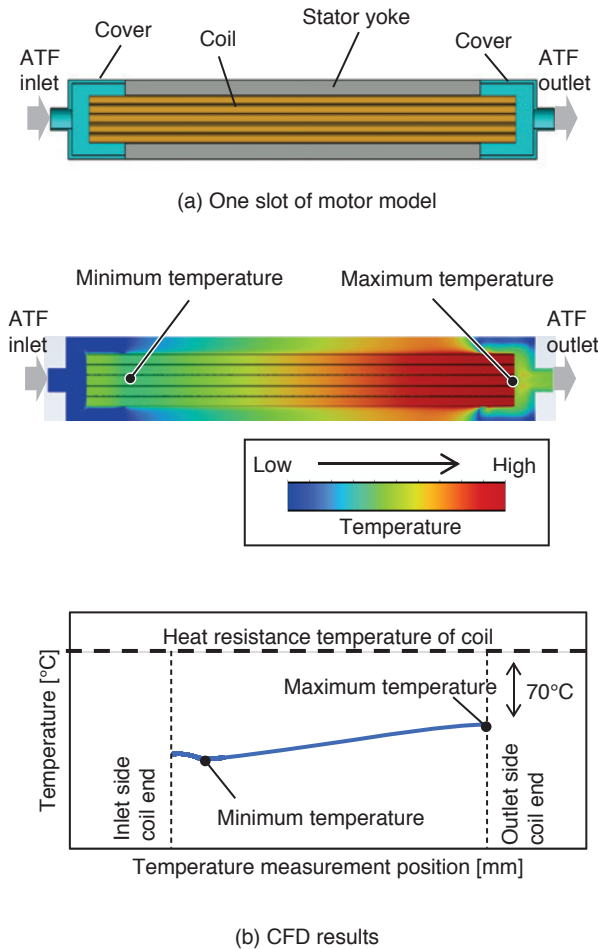


Fig. 3 Fluid analysis

Additionally, the maximum temperature for the coil was at the ATF discharging coil end, not inside the slot as in the conventional system. By using the hermetically sealed cooling structure to enable direct cooling of the entire coil with ATF, including the coil in the slot interior, it was confirmed that the maximum temperature of the coil is located at the coil end exposed from the stator.

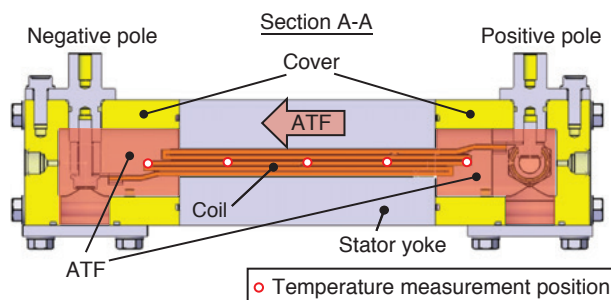
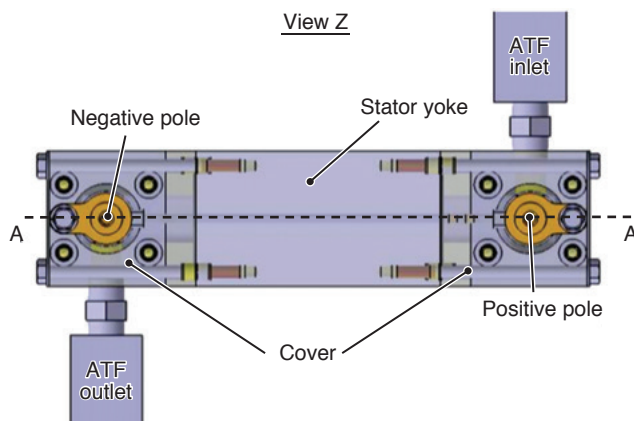
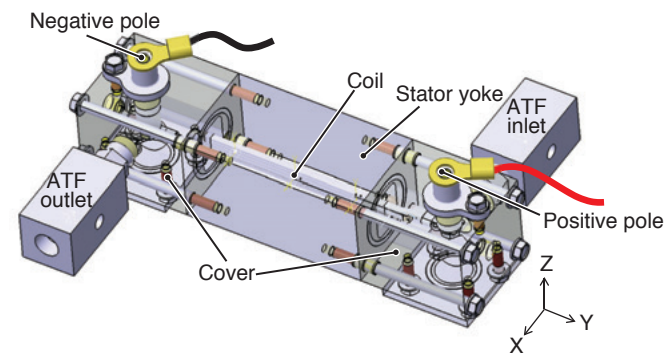
3.2. Verification Test with Test Piece

A test piece simulating a 1-slot section of the stator was fabricated based on the CFD model described above. The test piece is shown in Fig. 4. The test piece is equipped with a power supply terminal for energized heating of the coil. The test piece is built with the ability to simulate both a conventional cooling structure, dripping refrigerant on the coil ends, and the hermetically sealed cooling structure devised for this study. The conventional cooling structure was simulated by dripping refrigerant from the cover at both coil ends. With it, refrigerant could be discharged by partially releasing the cover. The hermetically sealed cooling structure was simulated by supplying refrigerant from the inlet, running it through the gaps in the coil within the slot, and discharging it out the outlet side. Inside the slot, the conventional cooling structure is composed of a coil, insulating paper, and varnish. By contrast, the hermetically sealed cooling structure is composed of the coil and insulating paper, using no varnish.

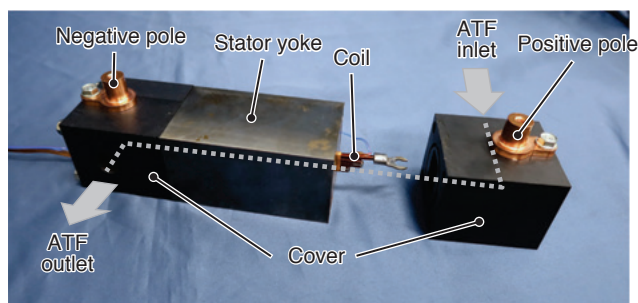
In the test system, coil temperature is measured when the coil is energized and heated to compare between the conventional cooling structure and hermetically sealed cooling structure. For the test conditions, the same temperature and flow rate for the refrigerant ATF and input power for the coil were used for both cooling structures.

Figure 5 shows the relationship between current density and coil temperature during coil heating for each cooling structure. Coil temperature was measured at equal intervals in the coil longitudinal direction using five thermocouples, plotting the maximum temperature of these measurements. Coil temperature was lower for the hermetically sealed cooling structure than in the conventional cooling structure. Even at twice the current density of the conventional structure, which is the input power with the motor reduced to 40% the conventional volumetric size, the maximum temperature is confirmed to be below the coil heat resistance temperature.

In conventional cooling structures, only the exposed parts of the coil is directly cooled by ATF. Thus, the coil ends are directly cooled, but the middle of the coil in the slots is relatively hard to cool. By contrast, as the hermetically sealed cooling structure can directly cool the entire coil with ATF, including the spaces within the slots, this greatly increases the heat transfer area and should increase the cooling effect. The hermetically sealed cooling structure was successfully reproduced according to the concept in testing with the test piece.



(a) CAD model



(b) Actual model

Fig. 4 Test piece

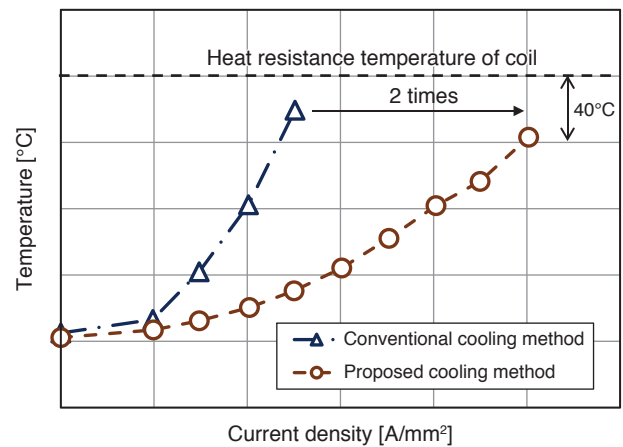


Fig. 5 Relation between current density and coil temperature

4. Challenges and Their Measures with Hermetically Sealed Cooling Structures

In the hermetically sealed cooling structure, the entire coil is immersed in ATF, which is circulated throughout the system. To apply the structure to an actual motor, upon insulating the system, the following three technical issues will need to be addressed:

- (1) Fix the coil and stator yoke in the slot
- (2) Create oil channels in the slots
- (3) Seal the coil ends with a cover to make them oil tight

First, figures will be used to explain technical issues (1) and (2) in detail. Figure 6 is a sectional drawing of a slot in a conventional motor. The coil and stator yoke are insulated from one another by wrapping insulating paper around the

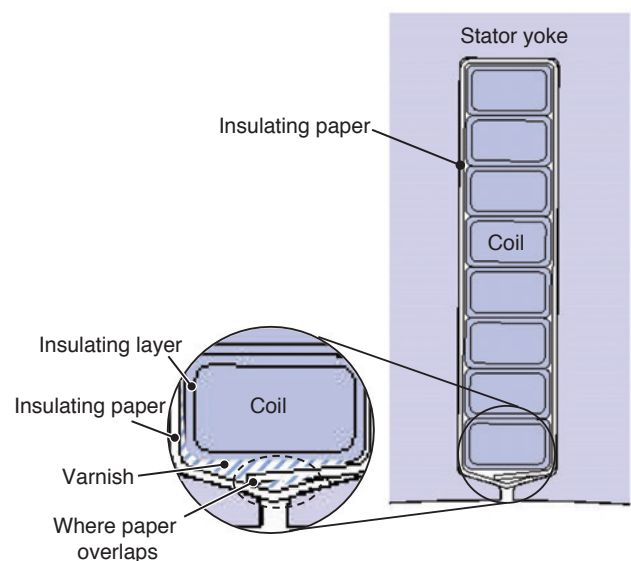


Fig. 6 Cross section of slot for conventional motor

coils inside the slots. When the proposed cooling system is applied to this structure, ATF will need to flow between the insulating paper and coils, meaning that ATF will leak through where the insulating paper overlaps. In conventional motors, the gaps between the slot coils, insulating paper, and stator yoke are impregnated in varnish to fix each part in place, thus blocking the space that will serve as the ATF flow path required for the proposed cooling method.

An overview of the structure devised to address issues (1) and (2) is shown in Fig. 7. A sheet with a foam adhesive layer is applied to the surface of newly developed insulating paper, modifying the seal where the insulating paper overlaps and sufficiently affixing the coil without using varnish. Coil fixing force and ATF channels are ensured in the slots by adopting a coil shape with straight sides for a wide contact area where facing the stator yoke and arc-shaped corners for large spaces on the sides between coils.

A detailed view of the internal slot configuration is shown in Fig. 8. The slot is composed of a coil with a

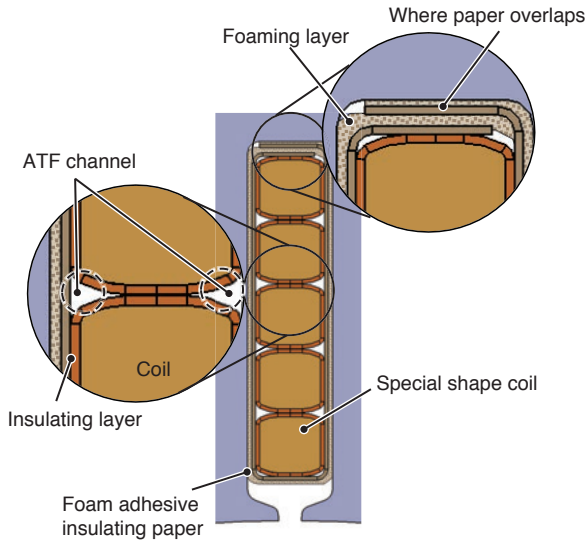


Fig. 7 Cross section of slot for proposed motor

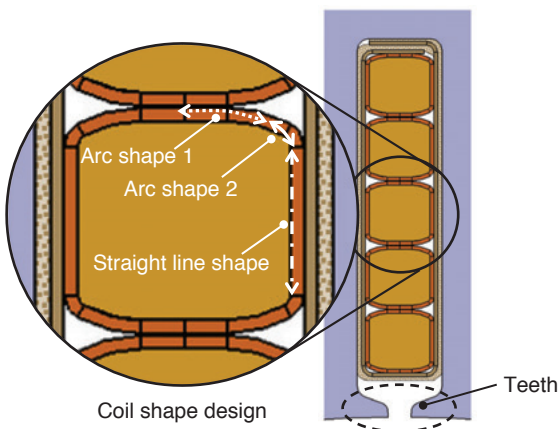


Fig. 8 Slot configuration

special sectional shape with multiple arcs forming its corners, surrounded by insulating paper with heat-attached foam adhesive. The coil corners are shaped with double arcs in the vicinity of the coils to create space for ATF channels between the conductors, and the straight sections are made longer on the sides contacting the insulating paper than the sides contacting an adjacent coil to provide a wide adhesional surface area with the insulating paper.

Finally, the following is an explanation of technical issue (3). In the conventional structure, refrigerant is dripped onto the coil ends, so no coil end seals are necessary. However, in forming the hermetically sealed cooling structure described in this paper, no refrigerant can be allowed to leak from the coil ends. The coil end sealing structure used for this motor is shown in Fig. 9. Along the inner diameter of the stator are multiple teeth, as shown in Fig. 8. To seal these teeth, a resin cuff support is positioned on the end face of the stator. The gaps between the cuff support and insulating paper are filled with a foam adhesive sheet to help ensure a good seal. The sealing structure was formed by applying an O-ring between the coil cover and cuff support.

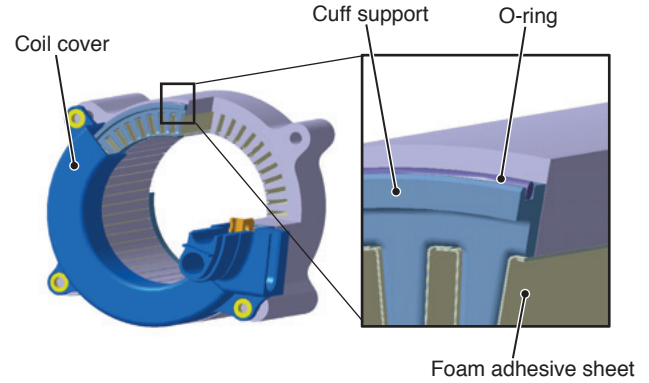


Fig. 9 Sealing structure (coil is not shown)

5. Demonstration Testing with the Hermetically Sealed Cooling Structure

5.1. Overview of the Prototype

Figure 10 is an external view of the prototype. A prototype motor using the hermetically sealed cooling structure was fabricated to confirm the pressure loss and cooling performance characteristics of the cooling structure.

5.2. Basic Specifications

Table 1 shows the basic specifications of the motor. The current density is double the conventional values to produce equivalent torque at 40% the conventional size. By reducing the area occupied by the coil, the specification allows for the appropriate number of turns to be set even with the smaller diameter.

Table 1 Motor specifications

Motor type	IPMSM
System voltage	350 V
Number of poles	8
Number of turns / slot	5
Current density	Twice conventional density
Coil type	Segment conductor



Fig. 10 Prototype motor

Figure 11 is a sectional view of the magnetic circuit applied for the motor. The motor stator has a smaller outer diameter than in conventional motors. As a result, the magnetic torque, which depends on the magnet flux, is limited. The motor adopts a multilayer arc magnet shape to maximize reluctance torque while ensuring the magnet amount.

5.3. Pressure Loss Characteristics

As the cooling structure is hermetically sealed, pressure is lost when pressure-feeding the ATF. Running the ATF into the slight spaces inside the slots requires an electric oil pump to pressure-feed the ATF. If pressure loss when feeding is high, a high-flow electric oil pump will be required, which would impact the overall drive unit efficiency.

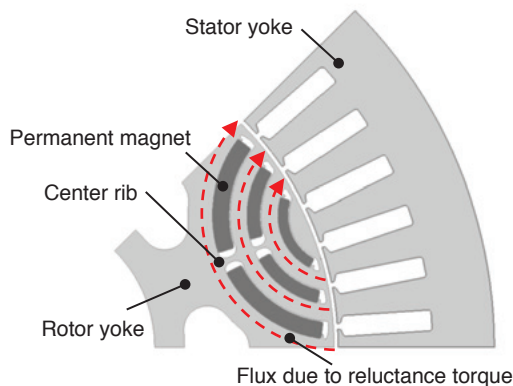


Fig. 11 Magnetic circuit model

Pressure loss in the cooling structure was measured to confirm the feasibility of establishing the cooling channels. Figure 12 shows the relationship between flow rate and pressure loss, and the rated output curve of the electric oil pump. The electric oil pump had an output of 40 W and volumetric efficiency of 50%, which is typical for automotive applications⁽⁸⁾. With the pressure loss at each target flow rate falling below the oil pump output curve, this confirms that the cooling structure can be fed with a typical automotive use electric oil pump.

5.4. Cooling Performance Characteristics

This section will compare the cooling performance of the prototype motor and a conventional motor. Given the many differences between the conventional cooling structure and hermetically sealed cooling structure, including motor structure, an evaluation method capable of comparing the cooling performance was used. For the test conditions, temperature and flow rate were set equal for the refrigerant ATF, and cooling amount was evaluated as copper loss of the power applied until the coil temperature reached equilibrium at a controlled temperature. Thus, greater copper loss means higher cooling performance. Figure 13 shows sample time series data for measurements of actual coil temperature and copper loss.

Figure 14 shows a graph of the relationships between cooling performance and refrigerant flow, comparing the proposed motor and conventional motor. The results confirmed that the proposed cooling method achieved more than twice the cooling performance of the conventional method. They also show that cooling amount increases relative to refrigerant flow differently than with the conventional cooling method. With the conventional method, the cooling amount tends to become saturated relative to refrigerant flow. Presumably, this may be due to refrigerant dripped on the coil ends being reflected and

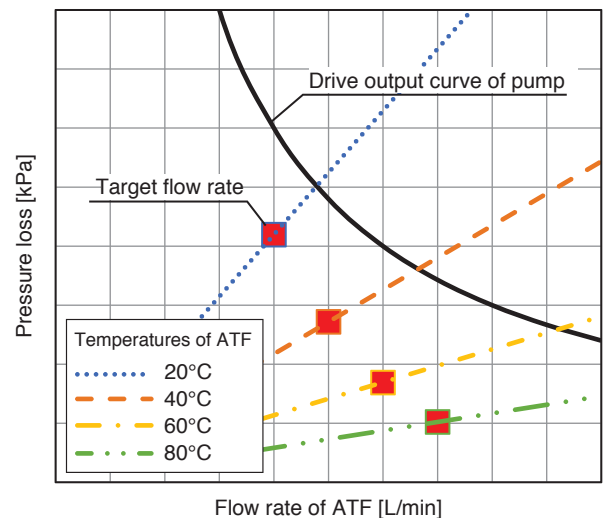


Fig. 12 Relation between flow rate and pressure loss

scattered at increased flow rates, meaning that the added refrigerant does not contribute to cooling. Meanwhile, in the proposed cooling structure, the cooling amount tends to increase linearly with refrigerant flow. As the structure is ATF immersed, an increase in refrigerant flow should increase flow velocity, which in turn should lead to an increase in heat transfer coefficient. In this test, the maximum cooling performance was confirmed to be quadruple that of the conventional motor.

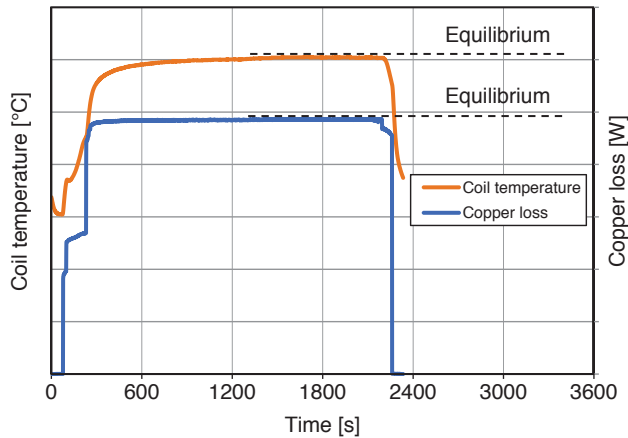


Fig. 13 Time series measurement data

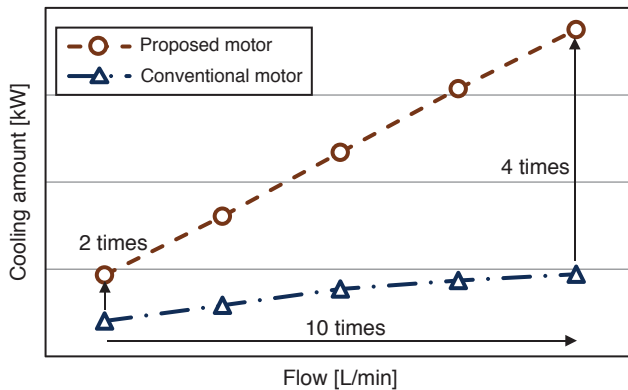


Fig. 14 Measurement of cooling capacity

6. Conclusions

A hermetically sealed cooling structure was developed to achieve a compact motor with high current density. A test piece and prototype motor applying this structure were fabricated to demonstrate the cooling performance.

- (1) More than twice the cooling performance of conventional cooling was achieved by immersing and circulating ATF throughout all windings to directly cool the coil, expanding the heat transfer area, and increasing the heat transfer coefficient.

- (2) The hermetically sealed cooling structure is highly sensitive to refrigerant flow, and cooling performance can be further enhanced by increasing refrigerant flow.

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Influence of Inverter Surge Voltage Waveform and Environmental Change on Partial Discharge Phenomena in Automobile Main Motor Magnet Wire

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ABSTRACT

Evaluation was conducted of the partial discharge inception voltage resulting from inverter surge voltage in the magnet wire, also known as winding wire, of the main motor used in an electric-powered vehicle.

Having clarified the influence of the waveform on partial discharge, the voltage waveform used for evaluation was configured to include waveform conditions under which partial discharge is most likely to occur during operation of the inverter in an electric-powered vehicle. With use of the inverter surge voltage, it was found that the partial discharge inception voltage rose 22% higher than the usual sine wave voltage. Envisioning the environment in which the main motor would be used, the influence of the environmental factors of ambient temperature, atmospheric pressure, and relative humidity on the partial discharge phenomena was also clarified. The use of inverter surge voltage for evaluation based on the above results yields the expectation that a rise in the partial discharge inception voltage can be expected. It also indicated that insulation design that takes the influence of the environmental factors into consideration can contribute to a reduction in the thickness of the insulation film.

1. Introduction

In light of the heightened awareness of environmental issues in the world in recent years, the electrification of automobiles has advanced rapidly. In Japan, the hybrid electric vehicle (HEV) began to rise in popularity as an electric-powered vehicle from the late 1990s. Because of demand for further reduction of environmental impact, popularity is presently shifting to the battery electric vehicle (BEV), which can use renewable energy. The performance of the main motor in this kind of electric-powered vehicle has an influence on the vehicle's cruising range, so raising motor efficiency to a high level is particularly desirable.

For that reason, many studies are underway⁽¹⁾ into methods for suppressing the motor losses from iron loss and copper loss, which are factors that diminish motor efficiency. It is possible to suppress copper loss in the main motor magnet wire by increasing the cross-sectional area of

the conductor, thereby lowering the resistance of the magnet wire. Recently there have been instances of approaches that reduce the relative permittivity of the magnet wire by introducing minute air bubbles into the insulation film on the wire, thereby making the film as thin as possible so as to increase the copper filling factor of the magnet wire slot^{(2),(3)}. However, making the insulation film on the motor magnet wire thinner than necessary reduces the insulating properties of the magnet wire, and partial discharge ends up occurring. The occurrence of partial discharge causes erosive wear of the insulation film surface, which ultimately lowers its insulation performance⁽³⁾. It is necessary, therefore, to accurately grasp the partial discharge inception voltage (PDIV) in the magnet wire specifications for that motor and design it so that partial discharge does not occur. Also, given the characteristics of the main motor magnet wire in an electric-powered vehicle, the magnet wire is used under the environmental conditions of the circumstances in

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which the vehicle is used. During high load operation, the temperature will rise, and in areas at high elevations, the atmospheric pressure will be lower. The influence that such environmental changes have on partial discharge also needs to be taken into consideration when studying ways to assure the insulation.

With the main motor of an electric-powered vehicle, the direct current voltage supplied from the batteries is converted into an intermittent pulse wave voltage by high-speed switching of the inverter devices, and that voltage is applied. As the voltage rise of that pulse wave voltage is steep, the surge voltage is superimposed on the pulse wave voltage from the inverter and is applied to the main motor input terminal as an inverter surge voltage. In PDIV evaluation of magnet wire, on the other hand, sine wave voltage in the commercial frequency range is most commonly used. It was reasoned for the present study that this is the case because methods of PDIV evaluation using inverter surge voltage have not become widespread and because sine wave voltage waveforms can easily be standardized and have general applicability, among other such reasons. In the phenomena of partial discharge due to inverter surge voltage, however, the course that arrives at discharge is complex by comparison with sine wave voltage, so that PDIV is known to be different⁽⁴⁾.

The insulation for main motor magnet wire is designed by first setting the PDIV for the magnet wire on its own and then taking into consideration the motor configuration, the environmental conditions, and other such PDIV variable factors. When PDIV evaluation of the magnet wire uses a sine wave voltage so that it diverges from the actual PDIV in the main motor, it becomes necessary to design a film thickness that is greater than needed in order to assure a safety factor. This had become an issue.

The aim of this research was to enable the design of an optimal thickness of insulation film by clarifying the PDIV characteristics of motor magnet wire resulting from inverter surge voltage, and enhancing main motor performance by reducing the thickness of the insulation film while assuring the necessary insulating performance.

2. Purpose

The purpose was defined as clarification of the following items (1) through (4), which have not previously been clarified sufficiently, for the study of insulation design for main motor magnet wire in electric-powered vehicles, and particularly the thick-film rectangular wire used for segment conductors⁽⁵⁾.

- (1) Influence of the shape of the inverter surge voltage waveform
- (2) Influence on PDIV due to the change of applied voltage from sine wave voltage to inverter surge voltage
- (3) Correlation between PDIV and the gas density estimated from temperature and atmospheric pressure
- (4) Correlation between relative humidity and PDIV

3. Evaluation Method

3.1. Sample Material

The material used as samples for PDIV evaluation was thick-film rectangular wire with insulation film of polyimide with relative permittivity ϵ_r of 3.0. A film thickness of 120 μm was chosen. It is a characteristic of polyimide that it absorbs and releases moisture according to the relative humidity of the atmosphere. This characteristic has an influence on partial discharge, so before measuring the PDIV, the sample material was placed in a constant-temperature, constant-humidity chamber adjusted to 25°C and 50% RH, which is close to the laboratory atmosphere environment, and given stabilization treatment to suppress change in the percentage of moisture content of the polyimide insulation film during testing. The atmosphere environment while discharge testing was being conducted was also set to the basic conditions of 25°C and 50% RH, except when verifying the influence of temperature and relative humidity.

3.2. PDIV Test Samples

For PDIV evaluation of magnet wire, a pair of magnet wires with film insulation that are connected together are held in place. One of the wires is grounded and the other wire has a gradually higher voltage applied to it. This causes a strong electric field to form in the space between the magnet wires. The test evaluates the value of the voltage applied when partial discharge occurs.

The partial discharge occurs according to Paschen's law⁽⁶⁾ in the small air gap between the insulation films of the two magnet wires. This is because, due to the difference between the relative permittivity of a solid body and a gas, the strength of the electric field is increased in the gaseous portion where the relative permittivity is lower. The space in which this partial discharge occurs is called the electric field air gap, and it is an important factor in studying the partial discharge phenomena.

For the research here, reproducibility of the shape of the electric field air gap was sought by adopting the new test sample shape shown on the right in Fig. 1. The new test sample shape is made up of a pair of magnet wires, one of which is in a straight line and the other of which has a fixed curvature. The magnet wires themselves are connected together in a slightly bent state and held in a holder so that they remain connected at a single point and a constant surface pressure. In the present research, these test samples are referred to as single R pair test samples, and these were used to evaluate partial discharge.

Figure 1 shows a conventional arrow pair test sample and a single R pair test sample to which voltage at or above the PDIV is applied to cause partial discharge. In the area where partial discharge occurs, a purple light is emitted in the space of the electric field due to the collision between electrons and gas molecules. The change in color therefore makes it possible to know the area where discharge occurs.

With the arrow pair test sample, partial discharge occurs across a wide range from the straight line portion where the two magnet wires are side by side to the fork portion where they branch away. When the area where partial discharge occurs extends across a wide range, the shape of the electric field air gap cannot readily be reproduced over multiple test samples. This causes variation in measurement. With the single R pair test samples, partial discharge occurs within the comparatively narrow range of the area between the straight line and the fixed curvature parts. This makes it possible to raise the reproducibility of the electric field air gap shape as well as to suppress measurement variations.

3.3. PDIV Evaluation Method

Figure 2 shows an overview of the PDIV evaluation device. The single R pair test sample is connected to the output of the inverter, and the partial discharge detector is connected in parallel with that. When partial discharge occurs in the single R pair test sample, the charge balance in condenser Ck breaks down so that charge transfer occurs and voltage is generated on both sides of detection impedance Z. The noise of the frequency component produced by the applied voltage is filtered out from that voltage to detect the partial discharge signal more clearly. For quantification of the discharge quantity, the detection voltage and discharge quantity calibration values that were acquired in advance are used to derive the quantity.

Figure 3 shows an example of measurement of the relationship between the applied voltage, discharge quantity, and time. The partial discharge signal appears at the time of $0.55 \mu\text{s}$, immediately before the applied voltage reaches the set peak value of 1.5 kV . At the time

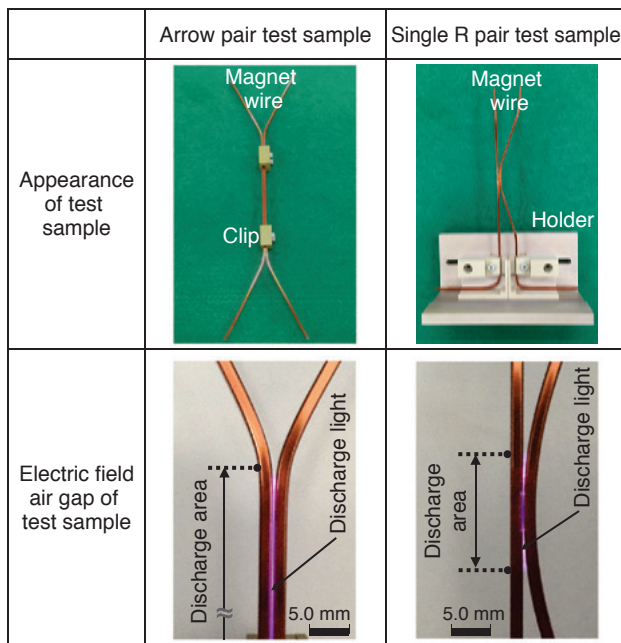


Fig. 1 Appearance of arrow pair test sample and single R pair test sample

of $0.1 \mu\text{s}$, which is the rise time of the applied voltage, a switching noise signal from the inverter device appears that the filter was not able to remove. However, this does not interfere with measurement because it is in a different temporal phase from the partial discharge. The criterion used in this research to determine when PDIV has occurred is that partial discharge with a discharge quantity of 100 pC or more per discharge has occurred at or above a certain probability. As there is no clear standard for setting the value of the discharge quantity⁽⁷⁾, it was decided to make it the same as the discharge quantity generally used for sine wave voltage PDIV measurement.

Figure 4 shows the process by which the initial

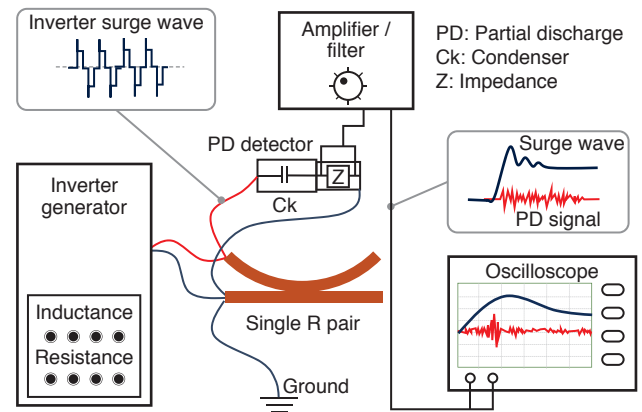


Fig. 2 Schematic diagram of PDIV measurement system

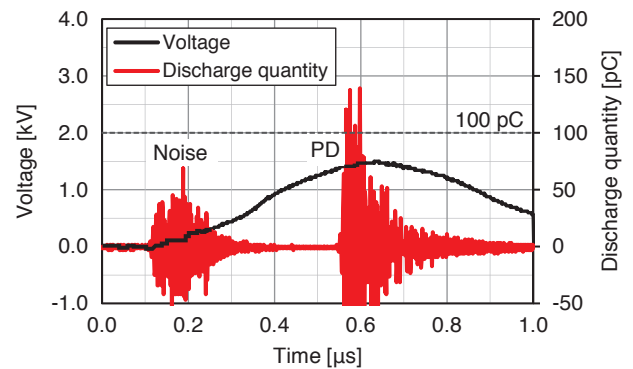


Fig. 3 Example of measurement of applied voltage and discharge quantity

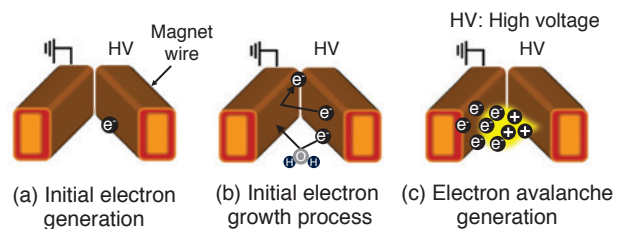


Fig. 4 Process of initial electron growth to electron avalanche

electrons generated in the electric field space between the magnet wires grow into the electron avalanche that is the initial stage of discharge.

- Initial electrons are generated from the electric field space and film.
- The initial electrons are accelerated by the electric field and collide with water molecules in the air and with the surface of the solid film, generating new electrons.
- The multiplied electrons grow into the electron avalanche formed from the electron swarm and positive ion cloud.

Although this is an extremely brief phenomenon, even in the partial discharge from the inverter surge voltage shown in Fig. 3, the same course is followed to arrive at discharge⁽⁴⁾. With the partial discharge phenomena caused by inverter surge voltage, the process of growth from the generation of initial electrons up to the electron avalanche is changed by the influencing factors of voltage waveform and environmental conditions. It is important, therefore, to clarify those influences.

4. Evaluation Results

4.1. PDIV Influence of Inverter Surge Voltage Waveform

As shown in the example in Fig. 5(a), the voltage waveform for driving the main motor appears as a square wave pulse waveform as a result of pulse width modulation (PWM) conversion. The inverter changes that

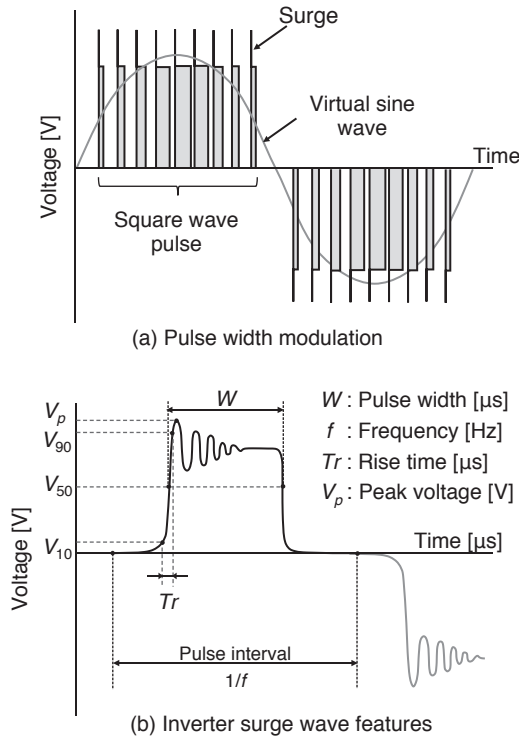


Fig. 5 Square wave pulse controlled by PWM and inverter surge wave features

Table 1 Test conditions for pulse width, frequency, and rise time

Scope Test	Pulse width W [μ s]	Frequency f [kHz]	Rise time Tr [μ s]
Condition 1	1-500	1.0	0.06
Condition 2	10	0.1-50	0.06
Condition 3	10	1.0	0.06-0.35

form according to the requirements for output to the main motor. In addition, this square wave pulse is affected by the characteristics of the inverter devices, by impedance mismatching between the power cable and the main motor, and other such structural influences so that it takes on the shape of an inverter surge waveform specific to that main motor. That pulse waveform is expressed by the feature values of pulse width W , frequency f , and pulse rise time Tr (the time taken to rise from 10% to 90% of the peak voltage value). The definitions of the respective feature values are shown in Fig. 5(b).

In order to verify the influence that the feature values have on the PDIV, while taking into account the conditions of the waveform output by the actual electric vehicle inverter and the range supported by the evaluation device, the pulse widths, frequencies, and pulse rise times shown in Table 1 were set as the parameters and the sensitivity of PDIV to the influence of each parameter was evaluated.

4.1.1. Pulse width W

Figure 6 shows that the measurement results for the relationship with PDIV of pulse width W varied from 1 to 500 μ s. In the pulse width region of 100 μ s and below, a longer pulse width W has an influence that lowers the PDIV.

4.1.2. Frequency f

Figure 7 shows that the measurement results for the relationship with PDIV of frequency f varied from 0.1 to 50 kHz. Higher frequency f has an influence that lowers the PDIV.

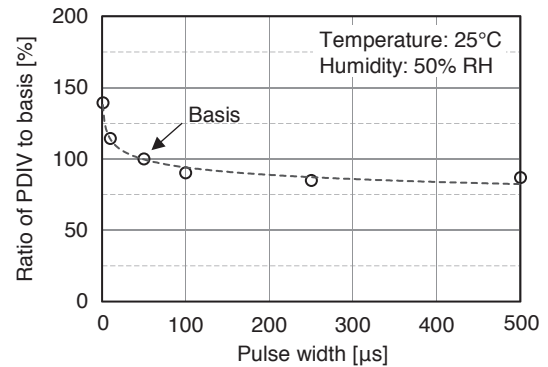


Fig. 6 Relationship between PDIV and pulse width as inverter surge voltage wave features

4.1.3. Pulse rise time T_r

Figure 8 shows that the measurement results for the relationship with PDIV of pulse rise time T_r varied from 0.06 to 0.35 μs . Longer pulse rise time T_r has an influence that lowers the PDIV.

4.2. PDIV Influence of Sine Wave Voltage and Inverter Surge Voltage Waveforms

In order to verify the influence of sine wave voltage and inverter surge voltage waveforms, the same magnet wires were used to conduct PDIV evaluation of the voltage waveform of both sine wave voltage and inverter surge voltage. When setting the waveform conditions for the inverter surge voltage, it was taken into consideration that the various feature values obtained in Section 4.1. have a tendency to lower the PDIV. Therefore, with a view to assuring the insulation, the voltage waveform that lowers the PDIV the most was set so as to include the voltage waveform output by the inverter in an actual electric-powered vehicle. The results are shown in Fig. 9, and this was taken as the basic condition for inverter surge voltage in this research.

Figure 10 shows the PDIV comparison results. This indicates that the PDIV with inverter surge voltage rises 22% higher than that with sine wave voltage. This indicates that, by including the inverter surge voltage in the design

of the insulation film, it will be possible to reduce the thickness of the insulation film by an amount corresponding to the amount by which the insulation film had to be thicker with the previous PDIV. A factor in this rise in the PDIV, as shown in Fig. 11, is that the sine wave voltage and inverter surge voltage differ in the duration that voltage is applied in the high-voltage region that can result in partial discharge. Focusing on the duration that voltage is applied in the high-voltage region, it is apparent that because the sine wave voltage time is given in millisecond units, the application of voltage is sustained until the initial electrons occurring in the electric field space grow into an electron avalanche. With the inverter surge voltage, on the other hand, as the time is given in nanosecond units, there are cases when the applied voltage ends up disappearing before the initial electrons have grown into an electron avalanche.

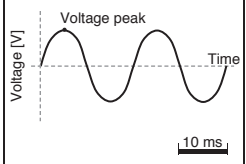
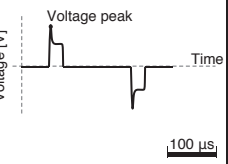
	Sine wave voltage	Inverter surge voltage
Waveform image		
Waveform specifications	Waveform: Sine Frequency: 60 [Hz]	Waveform: Bipolar pulse Pulse width: 50 [μs] Frequency: 10 [kHz]

Fig. 9 Basic conditions of sine wave voltage and inverter surge voltage

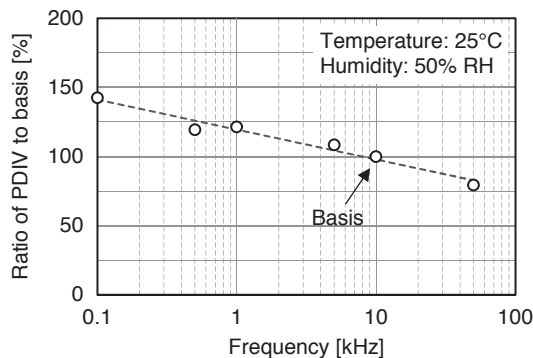


Fig. 7 Relationship between PDIV and frequency of pulse as inverter surge voltage features

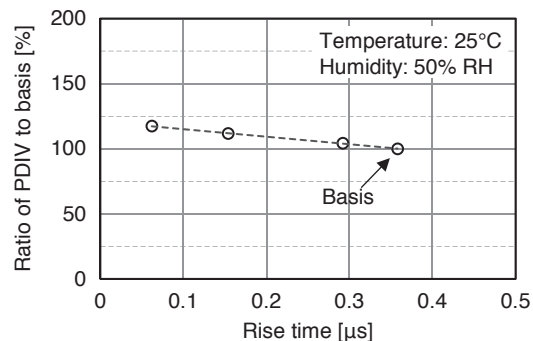


Fig. 8 Relationship between PDIV and rise time of pulse as inverter surge voltage features

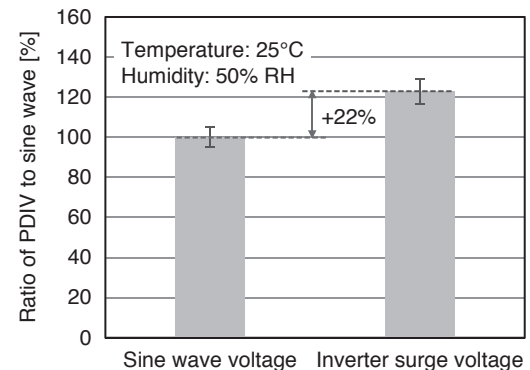


Fig. 10 Comparison of PDIV between sine wave voltage and inverter surge voltage

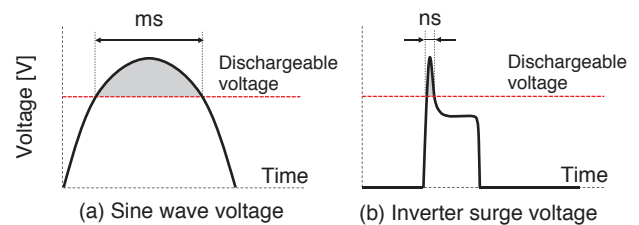


Fig. 11 Comparison between dischargeable times of sine wave voltage and inverter surge voltage

4.3. Influence of Environment on Inverter Surge Voltage

In the occurrence of partial discharge with inverter surge voltage, the probability of initial electrons occurring in the electric field space and the process of their growth into an electron avalanche are factors, and these factors are influenced by environmental conditions. It is important for the study of assuring the insulation, therefore, to ascertain the relationship between the PDIV and the environmental factors that have an influence on the partial discharge phenomena. For the present research, the ambient temperature, atmospheric pressure, and relative humidity were isolated as the environmental factors, and their relationship with the PDIV was verified.

4.3.1. PDIV influence of temperature and atmospheric pressure

In order to verify the influence of temperature and atmospheric pressure on the PDIV, the gas density was estimated from the temperature and atmospheric pressure and its relationship with the PDIV was sought.

The gas density was estimated using the state equation of Eq. (1) and converting it into Eq. (2). In an ideal gas, the temperature and atmospheric pressure are described by a state equation. The properties of a real gas also resemble those of the ideal gas, excluding high-density states at low temperatures and at high pressures. Here, therefore, the state equation was assumed to be valid and the study was pursued accordingly.

$$\text{State equation: } PV = nRT \quad (1)$$

$$\text{Gas density: } \frac{n}{V} = \frac{1}{R} \times \frac{P}{T} \quad (2)$$

P : Pressure

V : Volume

n : Number of moles

R : Gas constant

T : Temperature

For the measurement conditions, the operating environment of the actual motor and the range covered by the evaluation equipment were taken into consideration, the atmospheric pressure was taken as the ambient atmospheric pressure in a constant state with the temperature varied from 40°C to 170°C, the temperature was taken as a constant state at 25°C with the atmospheric pressure varied from 0.1 atm to 1.0 atm, and the relative humidity was kept in the range of 30% RH or lower. Figure 12 shows the measurement results for the relationship between gas density and PDIV.

It is apparent from these results that increases in gas density cause the PDIV to rise. Also, although some deviation in the PDIV value may be seen due to the difference in relative humidity conditions between the

measurements along the plotted line under atmospheric pressure control and the measurements along the plotted line under temperature control, it has become clear that the sensitivity of PDIV to the influence of changes in gas density remains the same.

The reason for this is that the main cause of the influence of changing gas density on the PDIV arises largely from changes in the number of gas particles that obstruct the activity of the initial electrons⁽⁸⁾. When the gas density decreases so that the free path for the electrons becomes longer, the electrons are accelerated sufficiently in the electric field space so that collisions between electrons and gas particles, and further collisions between the positive ions generated by collision and the solid film, begin to occur more frequently. This causes new electrons to be generated in the electric field space, making it more likely that discharge will occur and lowering the PDIV.

4.3.2. PDIV influence of relative humidity

In order to verify the influence of relative humidity on the PDIV, the relative humidity was varied from 10% to 95% RH and measurements were made to determine the relationship between it and the PDIV. Figure 13 shows the results. The measurement atmosphere had the temperature fixed at 40°C at ambient atmospheric pressure to enable control of the relative humidity over a wide range.

It is apparent from Fig. 13 that when the relative humidity is increased from the dry state of 10% RH, the PDIV tends to decrease. This is because as the relative

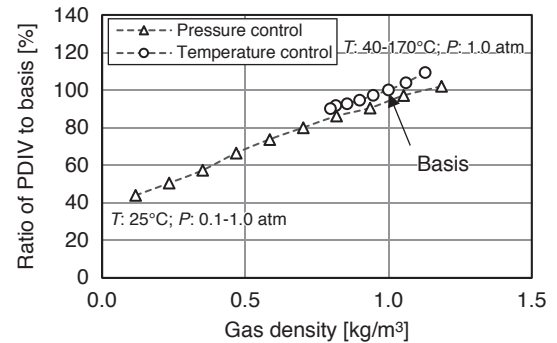


Fig. 12 Relationship between gas density and PDIV

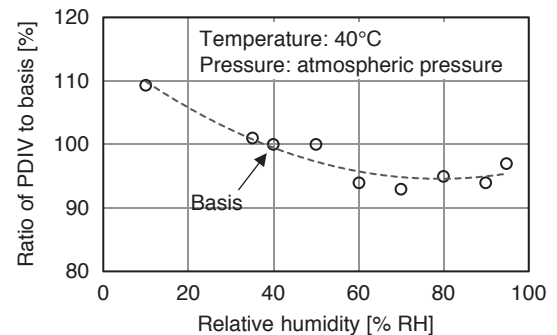


Fig. 13 Relationship between relative humidity and PDIV

humidity rises to a high level, the number of water molecules suspended in the electric field air gap increases, and those molecules ultimately become the source of oxygen negative ions and their hydrates that are sources of initial electrons⁽⁹⁾. However, where the PDIV tended to go lower as the relative humidity rose, in a high relative humidity environment at 60% RH or higher, such tendency to go lower was no longer seen.

Even though an environment is prepared in which discharge can occur more readily because of the rise in relative humidity, the phenomenon whereby the tendency for the PDIV to decrease disappears in a high relative humidity environment of 60% RH or higher suggests that some other influencing factor is at work.

5. Discussion of PDIV Behavior in a High Relative Humidity Environment

5.1. Differences from the General View of the Influence of Relative Humidity

There have been many reports regarding the relationship between relative humidity and PDIV in magnet wire with a film of polymeric insulating materials^{(5), (10)}, and the general view is that raising the relative humidity causes the PDIV to decrease. However, in the verification of inverter surge voltage conducted for this study, as shown in Fig. 13, the tendency for the PDIV to decrease was no longer visible in a high relative humidity environment of 60% RH or above. The cause of this is presumed to be that in a high relative humidity environment, a change of physical properties of some kind occurs in the surface of the insulation film that inhibits the generation of the initial electrons. Verification was therefore conducted of relative humidity, the discharge quantity, and the surface resistivity of the insulation film.

5.1.1. Relative humidity and discharge quantity

Figure 14 shows the relationship between relative humidity and the initial discharge quantity. In order to make it possible to evaluate even very small discharge quantities here, measurement included even discharges with very small discharge quantities that do not satisfy the

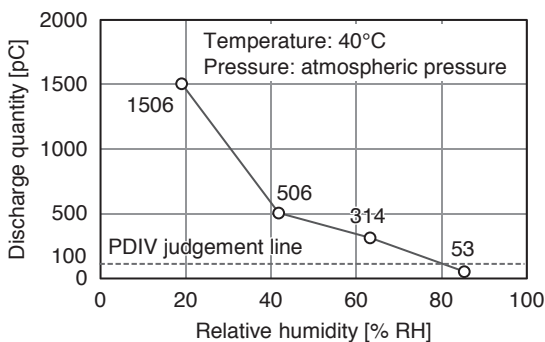


Fig. 14 Relationship between relative humidity and initial discharge quantity per one discharge signal

100-pC criterion for determining PDIV occurrence. The measurement atmosphere was 40°C at ambient atmospheric pressure, and the discharge quantity for the single initial discharge was derived from the mean value for initial discharge quantities in the first 10 sampling iterations. This showed that where the initial discharge quantity at the relative humidity of 19% RH was 1506 pC, the figure decreased to 53 pC at the relative humidity of 85% RH. It was apparent from this that even if partial discharge did occur in a high relative humidity environment, the discharge did not develop to the point of having an electron avalanche accompanied by a large discharge quantity as found when the relative humidity was low. It was found to develop only into a slight electron avalanche with a relatively small discharge quantity. It is presumed from this that when the discharge quantity for determining PDIV is set to 100 pC, there is a low possibility that partial discharge that satisfies the PDIV discharge quantity criterion will occur in a high relative humidity environment and that the tendency toward a lower PDIV is inhibited.

5.1.2. Surface resistivity and electric field strength simulation

The fact that when the relative humidity is increased, the initial discharge quantity decreases is clear, and the factors causing this were inferred.

In a high relative humidity environment, it is assumed that moisture adheres to the insulation film surface such that the surface resistivity is changed. The change in surface resistivity has an influence on the strength of the electric field in the electric field air gap. Simulation analysis was therefore conducted of the strength of the electric field formed in the electric field air gap when there was no film of moisture on the insulation film and when there was a moisture film 1.0 μm thick.

The analysis used the discharge area of the single R pair test sample as a model. The analysis conditions were set as insulation film with thickness of 120 μm , applied voltage of 1.5 kV, and two levels of conductivity of moisture of 5.0×10^{-6} S/m and 5.0 S/m on the assumption that the water would have different degrees of cleanliness. Figure 15 shows the simulation analysis results. It was found from the analysis results that the presence of a moisture film caused

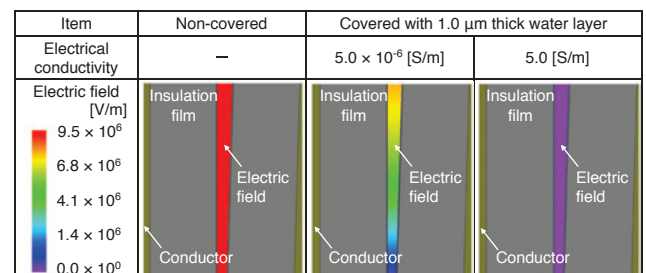


Fig. 15 Simulation of electric field strength at air gap with 1.0 μm thick water layer on insulation film

relaxation of the electric field that forms in the electric field air gap between magnet wires. It also became clear that this tendency was intensified when the conductivity of the moisture was higher. It can be inferred from this that when a moisture film forms on the insulation film so that electric field relaxation occurs in a high relative humidity environment, this is one of the factors in the decrease of the initial discharge quantity.

5.1.3. Relative humidity and surface resistivity measurement

As reduction in the surface resistivity of the insulation film was inferred by simulation to result in electric field relaxation in the electric field air gap, the relationship between relative humidity and surface resistivity was verified using actual polyimide film. Figure 16 shows the sample material and measurement method. A polyimide film 120 μm thick was used for evaluation and electrodes were fabricated by gold (Au) sputtering. The film was prepared using a drying furnace at 100°C so it would be in an absolute dry condition. The surface resistivity was then measured at a temperature of 40°C while varying the relative humidity from 30% to 95% RH.

Figure 17 shows the relationship between the relative humidity and the surface resistivity of the polyimide film. A tendency was observed for the surface resistivity of the polyimide film to decrease as the relative humidity

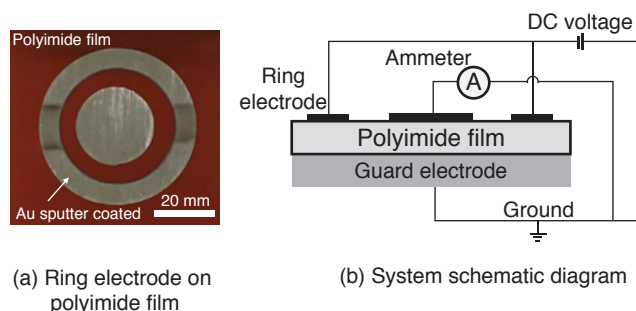


Fig. 16 Surface resistivity measurement method

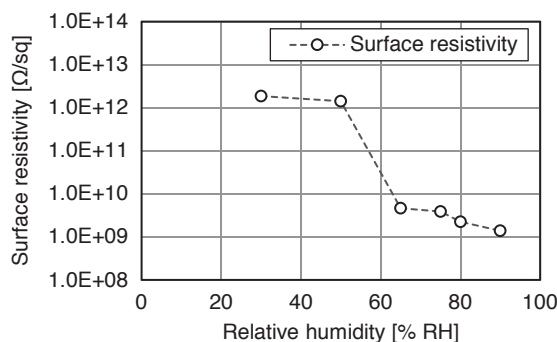


Fig. 17 Relationship between surface resistivity of polyimide film and relative humidity

increased. It also became clear that under these testing conditions, a sharp decrease occurred in the relative humidity range from 50% to 65% RH.

It can be inferred here that the rise in relative humidity results in water molecules that are in the atmosphere adhering to the surface of the polyimide film, ultimately forming multiple layers of moisture film. As the moisture film contains ions, it reduces the surface resistivity. It can be inferred from this that the decrease in surface resistivity in a high relative humidity environment due to adhesion of moisture causes electric field relaxation to occur so that the tendency of the PDIV to decrease is no longer visible.

6. Conclusion

This research successfully yielded the following findings regarding the PDIV characteristics of main motor magnet wire resulting from inverter surge voltage, a matter that had not previously been sufficiently clarified.

- (1) When the inverter surge voltage feature values of pulse width and pulse rise time increase, and frequency becomes higher, this has an influence that lowers the PDIV. The causal factor in this for each of the feature values is that when the time that such voltage is applied in the high-voltage range grows longer, this promotes development into an electron avalanche by the initial electrons that are generated in the electric field space.
- (2) In PDIV evaluation that assumes the inverter surge voltage applied to an actual main motor, there was a 22% increase over the PDIV using sine wave voltage. This indicated that a reduction in the thickness of the insulation film would be possible.
- (3) Converting temperature and atmospheric pressure into gas density allows it to be treated as a single influencing factor, and increased gas density raises the PDIV. Assuming the environment in which the main motor of an electric-powered vehicle is used, the influence on PDIV from atmospheric pressure is greater than that from temperature.
- (4) Raising the relative humidity from 10% to 60% RH resulted in the PDIV decreasing by 15%. At 60% RH or above, however, the tendency of the PDIV to decrease is no longer visible. One factor in this is that the increase in relative humidity results in a decrease in the surface resistivity of the insulation film so that electric field relaxation occurs in the electric field air gap.

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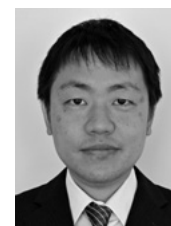
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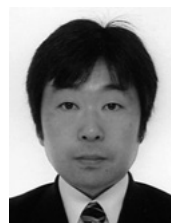
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Fuel Cell Stack Power Prediction Model Using Gaussian Process Regression Model

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ABSTRACT

A fuel cell stack power prediction model that takes into consideration the various stack control parameters is important in the optimization design of the controls for each item of auxiliary equipment in a system equivalent to that of an actual vehicle. However, creating a model for quantitative prediction of stack power requires large amounts of data concerning the materials and structure inside the fuel cell. Moreover, since the internal phenomena are complex, large-scale modeling is necessary. For this research, a design of experiment method known as the space filling technique was used to acquire data efficiently. With the acquired data as a basis, the use of Gaussian process regression made it possible to create a model capable of predicting stack performance as well as the temperature and pressure in the various parts of the stack in a short computation time. It was also made clear that this model could be used to calculate operating conditions that would maximize stack power, and verification by testing showed that it would be possible to obtain a power prediction model that could be used to investigate stack performance from a limited amount of test data.

1. Introduction

1.1. Background

The polymer electrolyte fuel cell (FC) converts chemical energy directly into electrical energy using hydrogen (H_2) as a fuel. This gives it higher energy efficiency than an internal combustion engine, while water is the only substance it emits during operation. It is, therefore, a power generating device with low environmental impact, and as such it is expected to have a wide range of uses, including in automobiles and trucks, as backup power sources for factories, and so on⁽¹⁾. The automobile industry, in particular, is pursuing development of FC automobiles with low environmental impact as a way of achieving electrification according to the laws and regulations of different countries. However, the penetration rate of FC automobiles is low for various reasons, including inadequate infrastructure development and the high cost of FC stacks⁽²⁾. Realizing lower-cost FC stacks will require achieving the necessary power and durability with limited catalyst loading and active area⁽³⁾. The development of materials and optimization of control to enhance power output are therefore considered to be necessary.

Current distribution and electrochemical reaction activity inside the FC stack change according to the gas, temperature, and water produced by the generation of electricity in the electric power-generating environment.

Since the stack performance changes as a result, multiple studies are being conducted on the reproduction of the stack interior environment by mathematical modeling for the purpose of predicting stack performance with respect to control^{(4),(5)}. When performing mathematical modeling of the FC stack, the various parameters influence each other, and the modeling becomes complicated as a result. This also requires understanding of the various theoretical formulas for the physical phenomena involved, the physical properties for the various materials, and so on⁽⁶⁾. Consequently, it becomes necessary to have accurate measurements and coefficient fitting in order to obtain multiple parameters every time a change is made to the materials or FC structure.

Regarding other modeling methods, with the development of artificial intelligence technology, increasing research is being done on modeling that uses machine learning⁽⁷⁾. The advantages of FC stack performance modeling by machine learning include, for example, the possibility of modeling the complex state of fluids and water inside the stack without performing a simulation.

Supervised learning is one type of machine learning. By appropriately labeling the input and output of feature values in measurement data, it learns the data and creates a model. Stack power prediction models that use artificial neural networks, which are one type of machine learning that is supervised learning, are highly accurate and capable

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Table 1 Definition of input and output parameters

	Anode				Cathode			Coolant			Electrical
Input parameter	Flow rate (An_Q)	Relative humidity (An_RH)	Pressure (An_P)	N ₂ concentration (An_N2)	Flow rate (Ca_Q)	Pressure (Ca_P)	Relative humidity (Ca_RH)	Flow rate (Co_Q)	Pressure (Co_P)	Temperature (Co_T)	Current density (CD)
Output parameter	Differential pressure (An_dP)				Differential pressure (Ca_dP)			Outlet pressure		Outlet temperature	Power

of quantitative prediction of performance^{(8), (9)}. However, creating a performance prediction model that uses artificial neural networks requires large quantities of measurement data. Furthermore, the examples in the references^{(8), (9)} all have about five explanatory variables, and since the distribution of explanatory variables is limited to about three levels for each variable, the possible range of accurate stack power prediction is also limited. For this research, therefore, measurement data was acquired efficiently by the design of experiment (DoE) method, and Gaussian process regression⁽¹⁰⁾ was used to create from that data a predictive model for FC stack power, differential pressure, and outlet temperature with respect to 11 variables that serve as input control parameters. In order to enhance the output of the created stack power prediction model, overall optimization of the control parameter variables was carried out and the enhancement of stack power was verified using an actual FC stack. This made clear that a stack power prediction model using Gaussian process regression with a small quantity of training data can be used to investigate control for the purpose of enhancing output.

1.2. Overview of Stack Modeling

Figure 1 shows an overview of the stack that was evaluated in the course of creating a Gaussian process regression model for this research.

First, the space filling technique⁽¹¹⁾, which is a DoE, was used to set the experiment conditions so that training data can be acquired efficiently for use in machine learning. A

device was also created with the capability to perform the experiment fully automatically as well as to preprocess the data. This is done simply by installing the FC stack on the device, and it creates the experiment conditions, performs the experiment, acquires the experiment data, and organizes the data. The experiment data acquired by the fully automated evaluation system was used to create the machine learning statistical model of the FC stack by Gaussian process regression. The input and output for the created model are shown in Table 1. The ETAS ASCMO tool from ETAS was used to formulate the experiment design and to create the Gaussian process regression model.

2. Stack Performance Evaluation

2.1. Setting the Experiment Conditions

When using DoE to set the experiment conditions, it is necessary to set the upper limits and the lower limits for each of the 11 control parameters. The upper and lower limits were set according to two perspectives, that of the range within which the stack components will not be damaged, and that of the upper and lower limits for control of the evaluation device. As one example, in cases when differential pressure is generated inside the stack between the anode and cathode, then from the perspective of stack protection, an upper limit was set for the differential pressure between the anode and cathode in order to protect components from differential pressure at or above a certain level⁽¹²⁾.

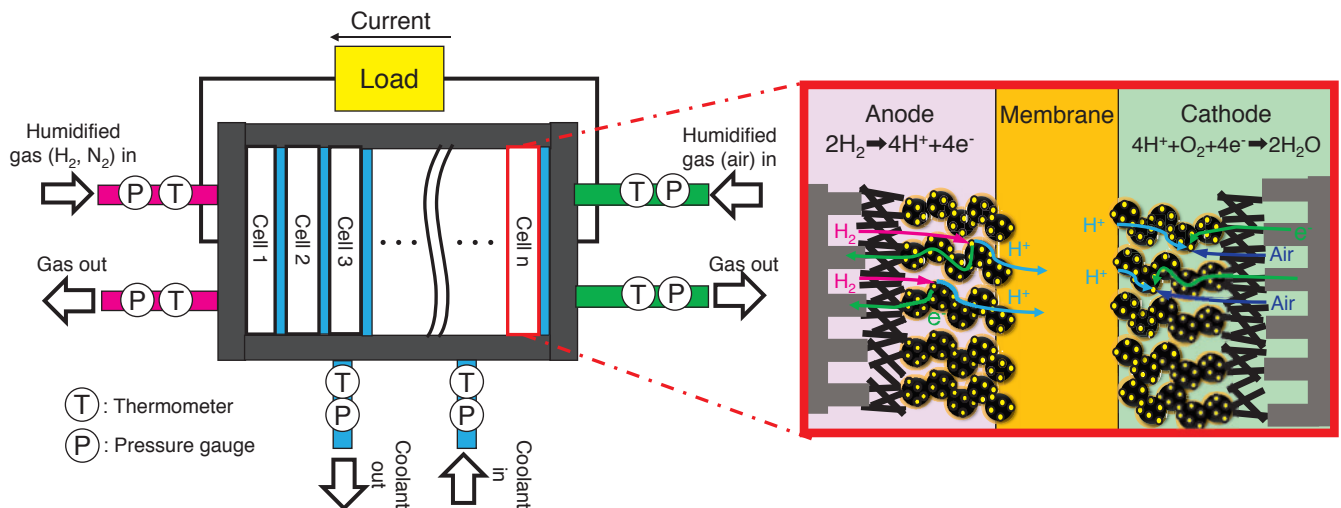


Fig. 1 Schematic image of stack

2.2. Experiment Procedure

A stack of nine cells in layers was fabricated to conduct the experiment. The experiment procedure was set as shown in Fig. 2 for conducting the evaluation under each of the measurement conditions based on DoE.

The anode and cathode were supplied with H_2 and nitrogen (N_2) and the temperatures of the stack and humidifier were adjusted. Next, air was introduced to the cathode and the backpressure of the anode and cathode was adjusted. If the open circuit voltage of the stack is held to the vicinity of 1 V at this time, the components will be progressively degraded. The current value was therefore set so that the voltage would be decreased slightly.

Generation would proceed at 1 A/sec until the current reached the set value. That current would be maintained for 30 minutes, and the average value for the last minute would be taken as the stack power performance under the target measurement conditions. After measurement, N_2 would again be introduced to the cathode, the stack voltage would be lowered, and discharged water would be treated under temperature adjustment and H_2 and N_2 supply conditions.

2.3. Fully Automated Evaluation System

A system capable of automated evaluation was created in order to conduct the experiment under the above conditions and procedure and to organize the data for use in creating the Gaussian process regression model. An overview is shown in Fig. 3.

The fully automated evaluation system is made up of three main parts, which are the experiment control system, the experiment device system, and the data analysis part. In the experiment control system, a device control file is created using 11 variables based on the DoE created using the ETAS ASCMO software made by ETAS, and a control profile converted for use in experiment device control is transmitted. In the experiment device system, control of the above-mentioned experiment procedure is implemented based on the control profile in which only the control parameter portion related to the 11 variables is different. The experiment data under each condition is measured by each of the measuring instruments and recorded by the logger. The measurement data is organized automatically in the data analysis part and then labeled according to Table 1

to process it into data for machine learning use. A Gaussian process regression model is then created using that created data set. ETAS INCA-FLOW software from ETAS was used as the management system to operate this device in a fully automated manner.

2.4. Gaussian Process Regression Model of the Stack

Gaussian process regression was used for modeling the stack. A feature of Gaussian process regression is that it is nonlinear and uses probability theory so that model predicted values have uncertainty and the probability of predicted values can be distinguished⁽¹⁰⁾.

The prediction equation based on Gaussian process regression used in modeling is shown in Eq. (1)⁽¹³⁾.

$$y(\vec{x}) = \sum_{i=1}^N C_i \cdot e^{-\frac{1}{2} \sum_{l=1}^D \frac{(x_{i,l} - x_l)^2}{r_l^2}} \quad (1)$$

For the Gaussian kernel function in the exponential function part, the generally used ARD squared exponential kernel was used. $\vec{x}$ is the input vector of the explanatory variable, N is the number of training data items, and C_i is the coefficient of the i th training data item, derived from

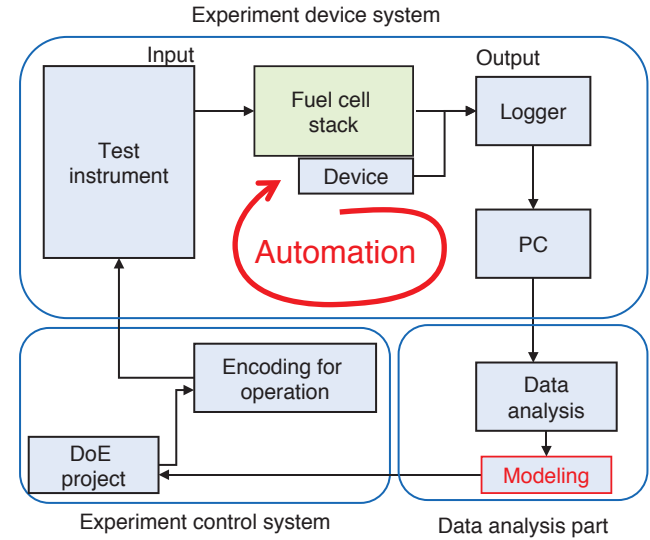


Fig. 3 Schematic image of automated test process

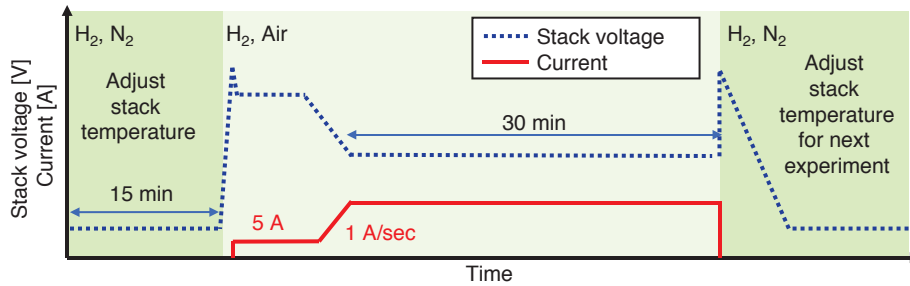


Fig. 2 Schematic image of experiment protocol

the hyperparameters. D is the number of dimensions, $X_{i,l}$ indicate the training data location, X_l is the input value, and r_l is the length scale. Of these, C_i and r_l are hyperparameters, and in ASCMO, these hyperparameters are determined by marginal likelihood optimization algorithms⁽¹⁴⁾.

Out of the experiment conditions set by the space filling technique, the Gaussian process regression model was created on the basis of the data set consisting of 169 conditions acquired by the fully automated evaluation system. Table 1 shows the definitions (labeling) of the parameters for modeling. The N_2 concentration here is the proportion of N_2 in the gas supplied to the anode inlet. The differential pressures of the anode and cathode are the difference in pressure at the stack inlet and outlet. In modeling the power, the stack output was converted from the total for nine cells to the equivalent for one cell. For the 11 defined input parameters, each output item was modeled by Gaussian process regression.

The accuracy of the created model was evaluated using the leave-one-out (LOO) method⁽¹⁵⁾. Model accuracy obtained by the LOO method can be evaluated by the determination coefficient R^2 . If R^2 is 0.9 or greater, it can be determined that a model capable of quantitative prediction has been obtained, and if R^2 is 0.6 or greater, it can be determined that a model capable of qualitative prediction has been obtained⁽¹⁵⁾.

3. Results and Discussion

3.1. Stack Model Evaluation

The anode differential pressure, cathode differential pressure, coolant outlet pressure, coolant outlet temperature, and stack power were modeled on the basis of the input conditions defined in Table 1. Figure 4 shows the results for evaluation of the created model by the LOO method. The modeling used 169 items of measurement data. An accuracy of $R^2 > 0.9$ is indicated for the prediction of stack power and the various outlet parameters. This indicates that a model has been obtained that is capable of quantitative

prediction of stack power in a steady state as well as cathode differential pressure, anode differential pressure, and coolant outlet temperature and pressure.

The contributions to stack power made by the 11 variables were evaluated based on the Gaussian process regression model. The parameter making the greatest contribution to power is current, and when that contribution is taken as 1, the relative contributions of the other parameters are shown in Fig. 5. It is apparent that the contribution of the anode control parameter is low and the contribution of the cathode control parameter is high. Since the FC overvoltage is more dependent on the oxygen reduction reaction in the cathode than on the hydrogen oxidation reaction in the anode⁽¹⁶⁾, it is apparent that the stack characteristics for the power prediction model in this research have also been selected appropriately.

Figure 6 shows the polarization curve for the stack measured under the various standard conditions and the average power values predicted by the power prediction

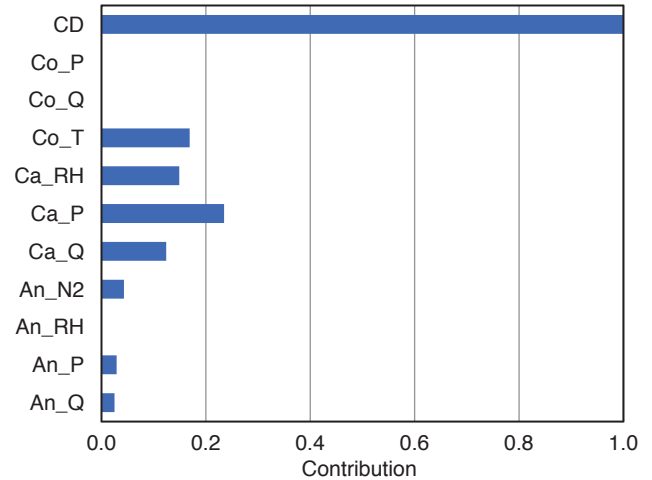


Fig. 5 Input relevance to stack power

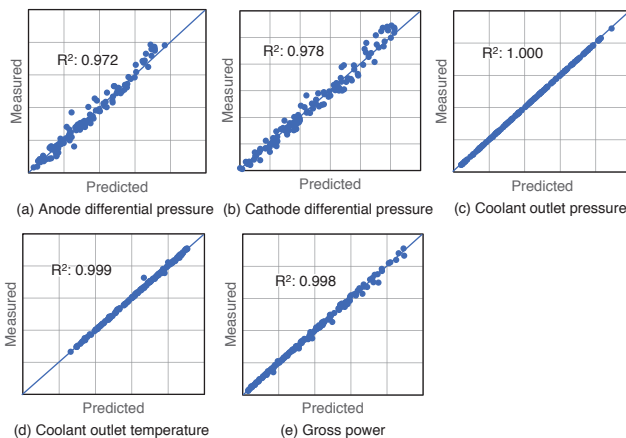


Fig. 4 Results of leave one out evaluation

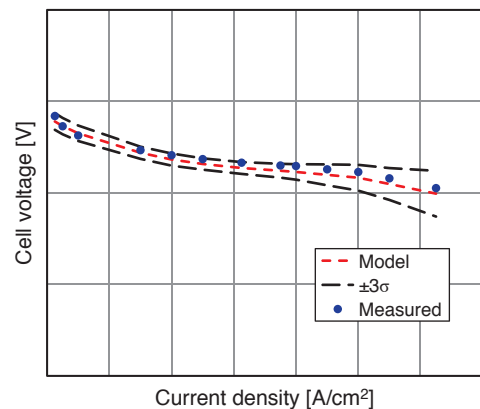


Fig. 6 Comparison of model prediction and measured data

model when the control parameters under the various standard conditions have been input. The model created by Gaussian process regression yields the predicted value averages and standard deviation $\sigma^{(10), (17)}$. Figure 6 also shows the results for model prediction average values within $\pm 3\sigma$. The polarization curve from actual measurement of the stack falls within the $\pm 3\sigma$ range, indicating that output under the various operating conditions can be predicted.

In Fig. 6, the σ spread is greater at high current density than at low current density. This is because the number of measurement data items on the high load side is approximately 12% of the total number of measurement data items and there is less learning data than for low loads.

Up to now, stack power had been predicted by the results of parametric study of experiment designs created using multiway layout⁽¹⁸⁾. Consequently, there was an issue in that the number of measurement points increased conspicuously when multiple variables changed simultaneously. Furthermore, in order to evaluate 11 variables in a multiway layout and create a highly accurate prediction model, it is necessary to conduct tens of thousands of experiments. The experiment design for this research used the space filling technique, which is a DoE method, to perform modeling by Gaussian process regression. This made clear that prediction of the various output items was possible with the number of evaluation points reduced to several hundred and while the 11 correlated variables underwent simultaneous change.

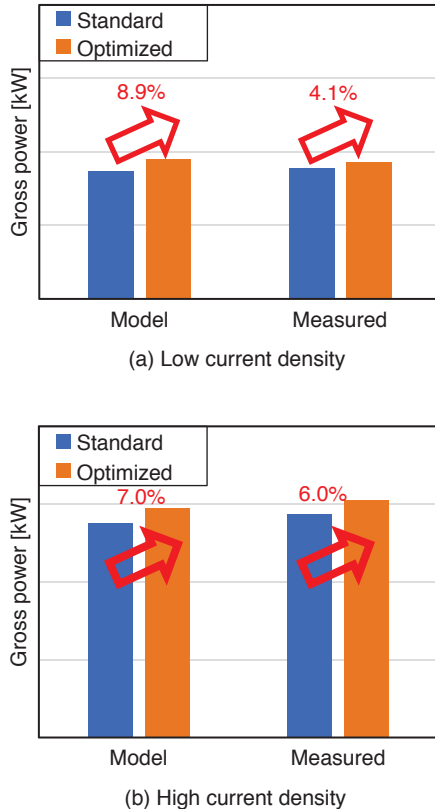


Fig. 7 Effect of optimized parameters on stack power

3.2. Output Enhancement by Examination of Control Parameters Using a Power Prediction Model

Verification was performed regarding whether or not it is possible to investigate the various control parameters using the stack power prediction model and enhance output. Verification involved overall optimization of the control parameters with the aim of maximizing both the low current density and high current density output of the stack.

Figure 7 shows average values for model prediction output and measured output under standard conditions and optimized conditions. The specifications of the measured stack were the same as those of the stack used in creating the model, but a different stack was used.

The average power prediction values calculated from the power prediction model can be expected to show power enhancement of 8.9% for low current density and 7.0% for high current density when the control parameters are changed from the standard conditions. As measured, the figures were 4.1% and 6.0%. Therefore it is possible to use the power prediction model created for this research to investigate the enhancement of stack power.

Figure 8 shows the ratio of the control parameters for both low current density and high current density as optimized overall to those parameters under standard operating conditions. Under the optimized conditions, the cathode pressure and cathode flow rate are set to higher levels than under the standard operating conditions, and the

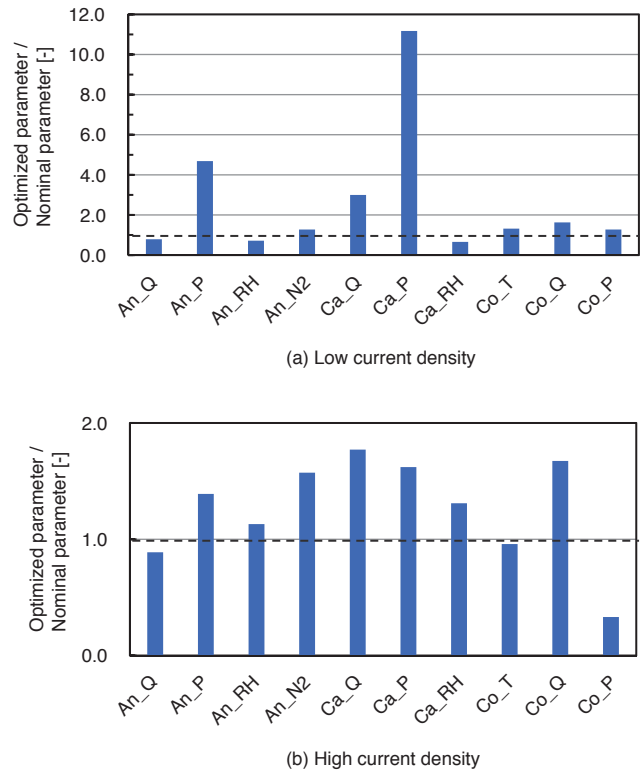


Fig. 8 Ratio of optimized parameter / nominal parameter

enhancement of power can be considered to occur because the cathode overvoltage is reduced. In the anode, on the other hand, the conditions call for flow to be reduced and N_2 concentration to be increased, and this can be considered to be because anode overvoltage makes almost no contribution to power performance. These adjustments of the control parameters are also supported by the results in Fig. 5 and by general FC characteristics⁽¹⁶⁾. Creating a more accurate power prediction model appears likely to necessitate increased density of data overall in the space formed from 11 dimensions.

The stack power prediction model created in this research will be used going forward to proceed with optimization of an FC system equivalent to that of an actual vehicle that also combines accessories and related items.

4. Conclusion

The results obtained in this research are shown as follows.

- (1) With the number of stack performance evaluation points reduced to the smallest amount by means of the space filling technique, the various output items were modeled using Gaussian process regression on the basis of measurement data acquired by an automated system for experimentation and data analysis.
- (2) Using the Gaussian process regression model, the control parameters were optimized for the purpose of maximizing stack power, and this made it possible to investigate the enhancement of stack power.

Going forward, the model described in this paper will be used to carry out optimization at the system level.

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Editorial Afterword

Thank you very much for reading this issue, Vol. 35, of the Honda R&D Technical Review. We put a half year or more of effort into producing every issue of this technical journal. In that process, authors persevere through several rounds of pointers and guidance they receive from reviewers and members of the manuscript committee, starting from their first manuscript, refining their papers until both sides are satisfied.

As seen from the perspective of the Technical Review editorial staff, authors have put together the findings of their own research in a single written work, and in writing it they must have constantly kept in mind what they most wanted to convey to their readers and how they should write in order to convey it accurately. In going through revision after revision of their papers, I suppose they felt very keenly how difficult it is to write so as to communicate to a third party. I expect no small number of them feel they have had their fill of writing papers.

However, that written work they underwent such pains to author becomes an element of evidence that can change the world. In addition, through this writing they can build networks with engineers and researchers inside and outside the company. So as we members of the Technical Review editorial staff see it from our perspective, the authors' work becomes an irreplaceable asset of great value. Over 10 years have already passed since I became involved in this editing work, and now I am again given the opportunity to publish the fruits of everyone's efforts in this issue of the historic Honda R&D Technical Review. As an editor, nothing could give me greater pleasure. In closing, I would like to express my wish, going forward, to continue contributing my own efforts, however slight they may be, to assisting with the Honda R&D Technical Review.

—Hisako MAJIMA—

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