Innovative VE SRB type Compressor Development compatible with R454B refrigerant

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ABSTRACT

In order to reduce greenhouse gas emissions, we have rapidly responded to refrigerant regulations in the North American market and aimed for miniaturization and weight reduction of twin rotary compressors. We addressed the challenges of reduced suction flow paths in thin CYLINDER, CRANKSHAFT run-out in a wide range, and refrigerant backflow in 1-pass SUCTION PIPE by introducing technologies such as circle-oval JOINT PIPE, asymmetric BALANCE WEIGHT, and Whistle structure. As a result, we have developed a lineup of 'SRB type compressors' compatible with R454B, achieving a weight reduction rate of 13.8% in compressors for the North American market while maintaining the same efficiency (energy-saving performance) as our conventional SNB type compatible with R410 refrigerant. Moving forward, we will standardize this specification and expand globally, including our existing bases in Japan, China, and Thailand, as well as new bases in India and USA, contributing to the realization of carbon neutrality.

Keywords: Compressor, R454B, Energy saving

1. INTRODUCTION

Our corporation defines the resolution of societal challenges through business activities as a fundamental mission. Within the air conditioning sector, our objective is to contribute to the establishment of a sustainable society by pursuing carbon neutrality through the reduction of greenhouse gas emissions. From an environmental policy perspective, the Montreal Protocol, adopted in 1987 to address ozone depletion and global warming, has positioned the phased elimination of chlorofluorocarbons (CFCs) and the phasedown of hydrofluorocarbons (HFCs) as international priorities. Following the Kigali Amendment to the Montreal Protocol in 2016, the United States enacted the American Innovation and Manufacturing (AIM) Act in December 2020 and subsequently ratified the Kigali Amendment in September 2022, thereby providing a regulatory framework for refrigerants. Consequently, beginning in 2025, refrigerants with a global warming potential (GWP) exceeding 700 will be prohibited for use in air conditioners and heat pumps (Fig. 1). This regulatory development renders the continued utilization of R410A, a refrigerant widely applied in the North American market, not feasible and necessitates a transition toward lower GWP alternatives.

In response, our company, in pursuit of carbon neutrality, developed a new refrigerating machine oil with excellent stability, lubricity, and low-temperature fluidity, thereby enabling compatibility with the R454B refrigerant (GWP 466). Through this advancement, we aimed to expand the product lineup of SRB type twin rotary compressors compatible with R454B, a lower-GWP alternative to R410A refrigerant (GWP 1924). Reducing greenhouse gas emissions over the product life cycle requires not only improving the energy efficiency of the products themselves but also reducing material

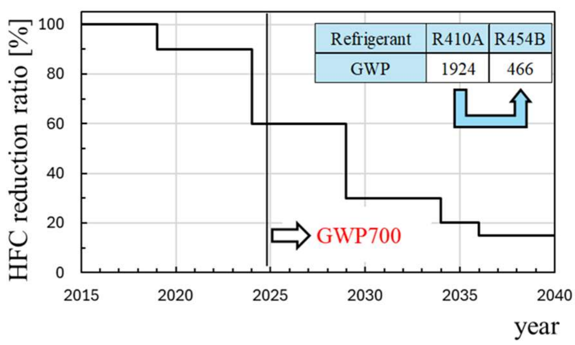


Fig.1 HFC reduction rate

usage, minimizing processing and assembly burdens, and shortening production times to lower simultaneously targeted greenhouse gas emission reductions through downsizing and weight reduction.

As a result, in the North American market, we realized the development of an R454B compatible SRB type compressor lineup (Fig.2) that achieves a 13.8% weight reduction compared with our conventional R410A compatible SNB type model, while maintaining

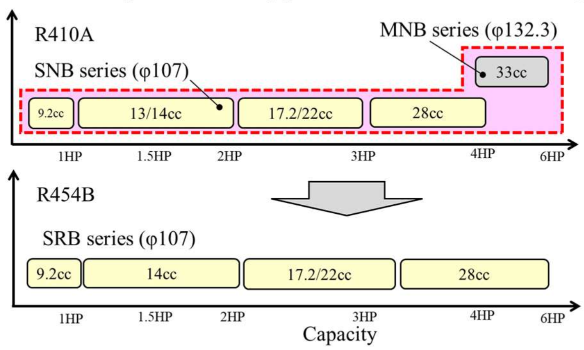


Fig.2 Compressor Lineup of S series

equivalent efficiency (energy-saving performance).
The subsequent sections describe the methodologies and technical considerations that enabled this achievement.

2. TECHNOLOGIES FOR IMPLEMENTATION

For downsizing and weight reduction of compressors, effective approaches include reducing the volume of individual components or decreasing the total number of components. When decomposed into major functional elements, the compressor can be classified into three parts: the compression mechanism, the motor, and the shell (including the suction muffler) (Fig.3). Accordingly, downsizing and weight reduction efforts were implemented for each of these three elements. Specifically, for volume reduction, cylinder height reduction (thin cylinder) and expansion of the permissible rotational speed range (wide range) were implemented, while for component count reduction, a 1 pass suction pipe configuration was introduced. On the other hand, since reducing the motor height leads to a decline in motor performance, efficiency improvements were pursued through the development of items in the compression mechanism. By allocating these efficiency gains to motor height reduction, compressor downsizing was successfully achieved.

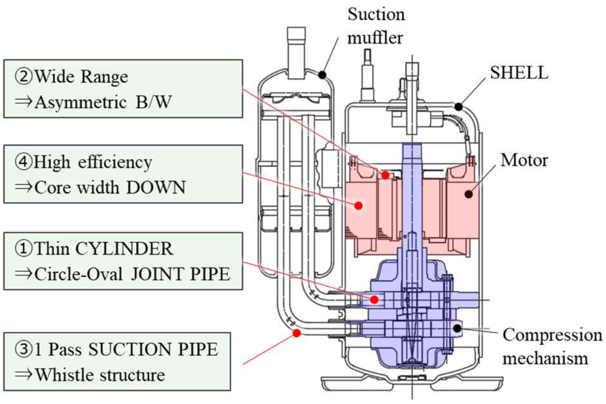


Fig.3 Structure of compressor and development elements

2.1 CHALLENGES AND SOLUTIONS IN THIN CYLINDER

The cylinder of a rotary compressor is a principal component forming the compression chamber and constitutes one of the largest parts in terms of volume. However, attempts to reduce the thickness of the cylinder present several challenges, such as increased assembly deformation due to reduced rigidity and a reduction in the suction flow passage resulting from dimensional constraints on the suction hole. To mitigate assembly deformation, our company applied a production technique known as heat caulking, which reduces cylinder strain during assembly. With respect to the issue of suction passage reduction, the conventional circular joint pipe [Fig. 4(a)] limited the effective flow area. To address this problem, we expanded the cylinder suction hole in the radial direction and introduced an oval joint pipe [Fig. 4(b)], thereby enlarging the effective cross-sectional area of the passage and overcoming the efficiency loss caused by reduced flow area. However, as shown in Fig. 4(b), making the cylinder, shell, joint pipe, and suction pipe all adopt an oval shape presented the challenge of increased processing load (longer processing time). To address this, we introduced a circle-oval joint pipe [Fig. 4(c)], with a circular rear end and an oval front end. This allowed the use of a circular suction hole in the shell and a circular suction pipe, both of which offer favorable formability, thereby achieving both high compressor efficiency and reduced processing load.

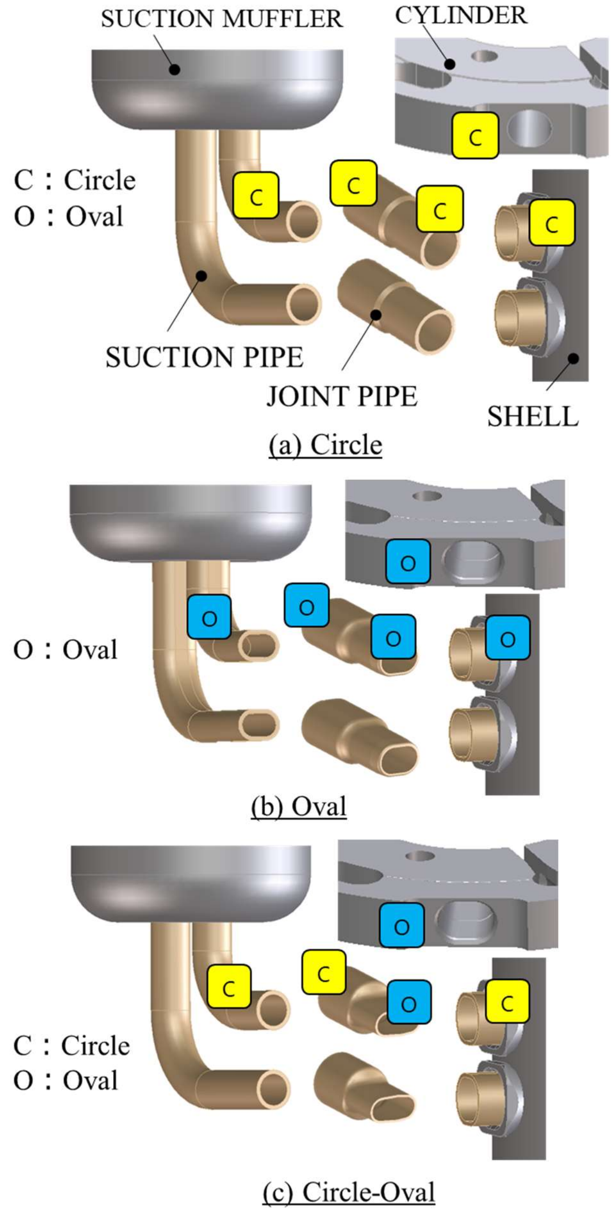


Fig.4 Circle-Oval structure

Furthermore, the compression chamber of a rotary compressor consists of the cylinder, rolling piston, and vane. A major portion of the refrigerant leakage from the high pressure region to the low pressure region occurs through the radial clearance between the inner diameter of the cylinder and the outer diameter of the rolling piston (Fig.5). The leakage area at the side surface of the rolling piston can be expressed as [cylinder width] × [radial clearance between the cylinder inner diameter and the rolling piston outer diameter]. Accordingly, reducing the cylinder thickness is effective in minimizing leakage loss, thereby contributing to higher efficiency. Based on these efforts, while improving efficiency, we achieved a 20.6% volume reduction in our 9.2 cc model by reducing the cylinder height from 13.6 mm to 10.8 mm.

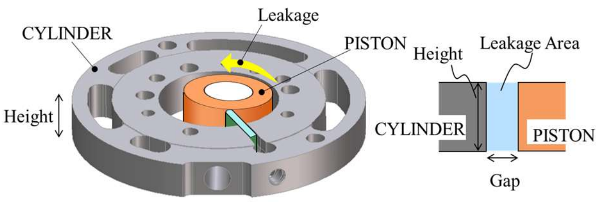


Fig.5 ROLLING PISTON side leakage

2.2 CHALLENGES AND SOLUTIONS IN WIDE RANGE

In rotary compressors, the shaft run-out under centrifugal force during high speed operation, leading to shaft wear and increased vibration; thus, an upper limit on rotation speed is generally imposed. Expanding this speed limit to allow a wide range enables a single compressor to cover a broader capacity range. Consequently, compared with current models, a smaller stroke volume can be selected, thereby allowing downsizing through the use of more compact and lightweight compressors. This approach not only contributes to size and weight reduction but also decreases the number of required product variants, which in turn reduces setup requirements during manufacturing and lowers power consumption. Furthermore, by adopting a smaller stroke volume, the compressor can operate at higher rotation speed to deliver the same capacity. Operation in this higher speed region suppresses refrigerant leakage from the compression chamber, thereby improving compressor efficiency. To achieve these benefits, we developed a one-dimensional CAE (1D-CAE) of the compressor to suppress shaft run-out during high speed operation (Fig.6).

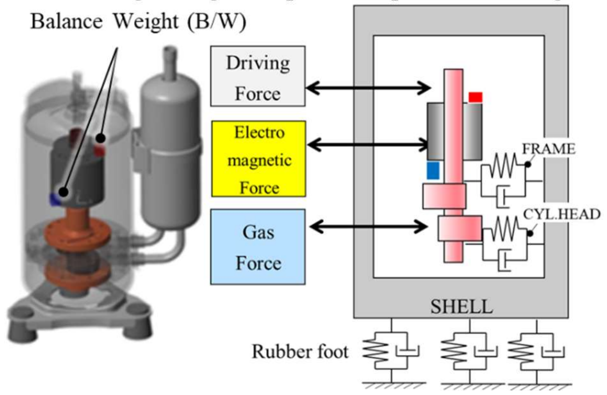


Fig.6 1D-CAE conceptual model

The compressor is subjected to complex phenomena involving forces across multiple fields, such as gas compression loads, motor magnetic attraction, and centrifugal forces, and therefore requires complicated analyses to resolve vibration transmission between components. However, the present 1D-CAE enables rapid analysis by utilizing Multiphysics models and Multibody dynamics models, and has established an optimal design method for balance weights, including "asymmetric balance weights" with differences in upper and lower mass, which traditionally require significant time for design and evaluation. As represented by our 14 cc model in Fig. 6, the shaft run-out increases up to 200% at 160 rps compared with the baseline at 130 rps, the maximum rotation speed of the conventional model. However, by introducing the optimally designed asymmetric balance weight(B/W) derived from the 1D-CAE analysis, the shaft run-out at 160 rps was suppressed by 37%. Furthermore, this 1D-CAE enables continuous refinement of analytical accuracy through feedback from experimental results, thereby improving its predictive performance throughout the development process. As a result, significant downsizing was achieved, including the replacement of the M type 33 cc model (shell outer diameter φ132.3) with the smaller S type 28cc model (shell outer diameter φ107).

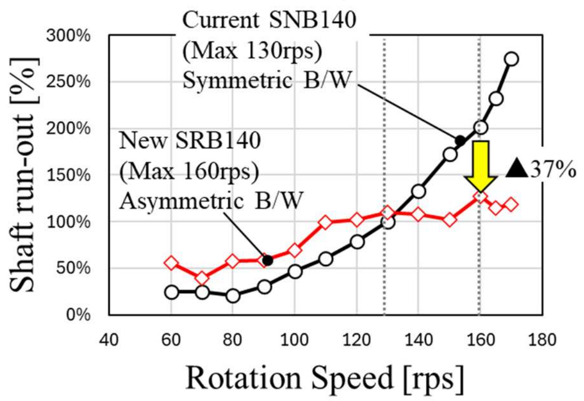
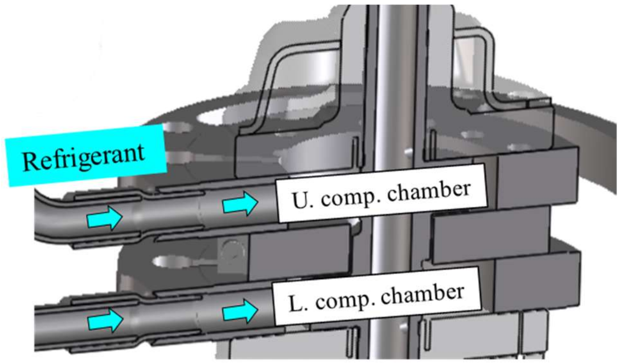


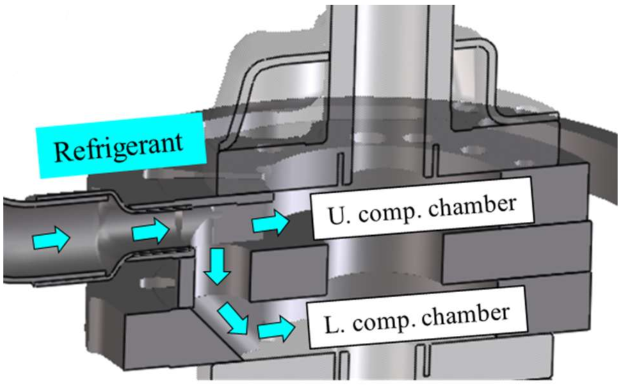
Fig.7 Shaft run-out

2.3 CHALLENGES AND SOLUTIONS IN 1 PASS SUCTION PIPE

Since a twin rotary compressor is equipped with two compression chambers, a 2 pass suction pipe configuration with two suction pipes, as shown in Fig. 8(a), is generally used. This suction pipe typically has an elbow structure and is made of copper, considering brazeability to the compressor shell body. However, due to the recent rise in copper prices, a 1 pass suction pipe configuration, as shown in Fig. 8(b), which enables both a reduction in the number of components and a decrease in copper usage, was adopted. In contrast to the 2 pass suction pipe, where each compression chamber is connected in series with a dedicated suction pipe, the 1 pass suction pipe structure allows the refrigerant flowing from a single suction pipe to branch in parallel inside the mechanism and then flows into each compression chamber. Under this configuration, backflow refrigerant



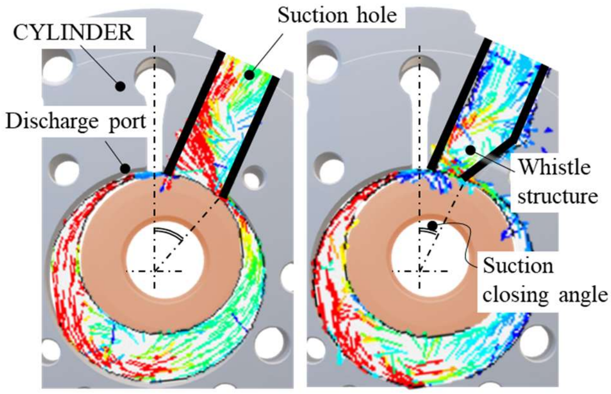
(a) 2Pass structure



(b) 1Pass structure

Fig.8 2pass and 1pass structures

generated when the rolling piston side leakage occurs during compression, or when the discharge valve opens and high pressure refrigerant flows from the discharge port to the suction port, can enter the opposite compression chamber (e.g., backflow from the upper cylinder flowing into the lower cylinder). Such backflow increases suction pressure loss and results in lower efficiency compared with the 2 pass suction pipe. To suppress this backflow during the opening of the discharge valve, a “whistle structure” (Fig.9) was introduced, in which the side of the suction hole opposite to the vane slot was narrowed toward the inner side of the suction hole, thereby reducing the suction closing angle. This structure reduces the suction pressure loss caused by backflow, which is particularly pronounced in the 1 pass structure, and thus enables a highly efficient 1 pass configuration. The circle-oval joint pipe must be capable of being inserted into the suction hole of the shell, as shown in Fig. 4(c). To permit insertion into the shell, the circle section of the circle-oval joint pipe must be larger than the oval section. Therefore, compared with the 2 pass suction pipe configuration, the 1 pass suction pipe configuration, which allows easier expansion of the suction pipe diameter, is well suited for use with the circle-oval joint pipe.



(a) Current (b)Whistle structure

Fig.9 Flow velocity distribution in the compression chamber immediately after the Discharge valve opening of Current and Whistle structure

3. CONCLUTION

In order to reduce greenhouse gas emissions, we have rapidly responded to refrigerant regulations in the North American market and aimed for miniaturization and weight reduction of twin rotary compressors. We addressed the challenges of reduced suction flow paths in thin cylinder, crankshaft run-out in a wide range, and refrigerant backflow in 1-pass suction pipe by introducing technologies such as circle-oval joint pipe, asymmetric balance weight, and whistle structure. As a result, we have developed a lineup of 'SRB type compressors' compatible with R454B, achieving a weight reduction rate of 13.8% in compressors for the North American market while maintaining the same efficiency (energy-saving performance) as our conventional SNB type compatible with R410 refrigerant. Moving forward, we will standardize this specification and expand globally, including our existing bases in Japan, China, and Thailand, as well as new bases in India and USA, contributing to the realization of carbon neutrality.

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革新的な VE を実現した新冷媒 R454B 対応 SRB 形圧縮機の開発
Innovative VE SRB type Compressor Development compatible with R454B refrigerant

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Key Word: Compressor, R454B, Energy saving

１．はじめに

当社は事業を通じた社会課題を使命としており、空調事業においてはカーボンニュートラル実現を目指して温室効果ガス排出量削減による「持続可能な社会の構築」に貢献することを目指している．昨今の環境情勢としてはオゾン層破壊および地球温暖化課題に対して 1987 年に採択されたモントリオール議定書をはじめ，フロン類の段階的廃止や HFC の段階的削減が世界的な重要課題として認識されるようになった. 2016 年のモントリオール議定書（キガリ改正）制定を受け，米国は 2020 年 12 月に AIM 法を制定，2022 年 9 月にモントリオール議定書（キガリ改正）に批准するなど急速に冷媒規制の基盤を固め，目下 2025 年からエアコンおよびヒートポンプに使用される冷媒は GWP700 未満に制限されることとなった（Fig.1）．これにより北米市場においてエアコン用として一般的な R410A 冷媒の継続使用が困難となり更に低 GWP な冷媒への移行必要性が高まっている．

そこで当社はカーボンニュートラル実現に向けて、安定性・摺動性・低温流動性に優れた冷凍機油を新たに開発することで R454B 冷媒（GWP466）への対応を可能とし、R410A 冷媒（GWP1924）に対して低 GWP である R454B 冷媒に対応したツインロータリ圧縮機（SRB 型）のラインアップ展開を目指した。また，温室効果ガス排出量削減には製品のライフサイクルを踏まえると製品自体の省エネルギ化だけでなく素材使用量削減や製品加工・組立時の負荷削減や時間削減による消費電力削減など様々な実現手段が存在するが，R454B 冷媒対応圧縮機開発では，小型軽量化による温室効果ガス排出量削減を同時に

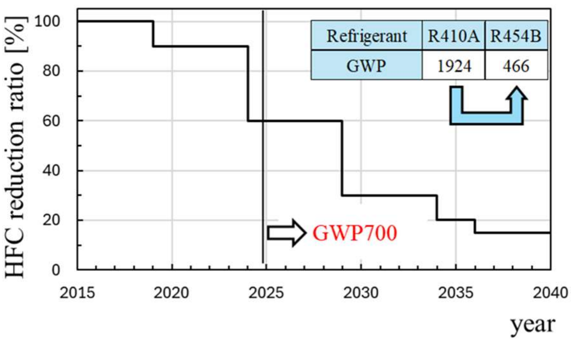


Fig.1 HFC reduction rate

目指した．

結果として北米市場において当社従来 R410 冷媒対応 SNB 型に対し効率（省エネルギ性）同等を維持しつつ13.8%の重量削減を実現する R454B 対応「SRB 形圧縮機」ラインアップ（Fig.2）展開を実現したため，本稿ではその実現手段について報告する．

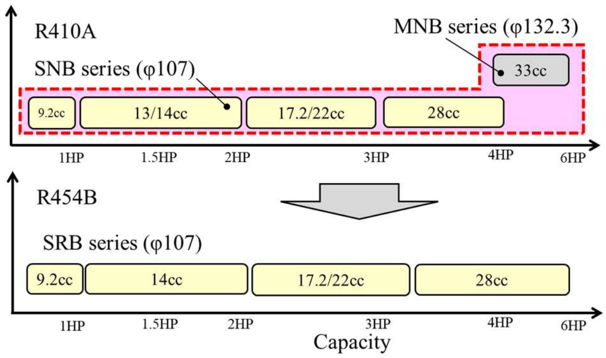


Fig.2 Compressor Lineup of S series

２．実現のための技術

圧縮機小型軽量化には部品の体積削減もしくは部品数削減が有効であり，圧縮機を大きな要素で分解すると圧縮メカ機構・モータ・シェル（吸入マフラー含む）の 3 要素に分けられる（Fig.3）ことから 3 要素それぞれにおいて小型軽量化を図った．具体的には，体積削減に対してはシリンダ高さ縮小（薄肉シリンダ化）と使用可能回転数の上限拡大（ワイドレンジ化）を実施し，部品数削減に対して 1 パス吸入管仕様の導入を実施した．一方でモータ高さを縮小するとモータ性能が低下してしまうことから圧縮メカ機構のアイテム開発にて高効率化を図り，効率向上分をモータ高さ縮小に充てることで圧縮機ダウンサイジングを実現した．

２．１　薄肉シリンダの課題と解決手段

ロータリ圧縮機のシリンダは圧縮室を構成する主要部品であり体積の大きい部品である．シリンダの薄型化には剛性低下に伴う組立歪の増加や吸入穴寸法制約による吸入流路縮小が発生す

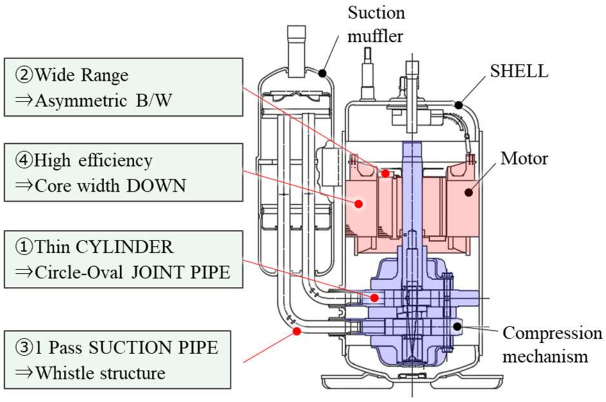
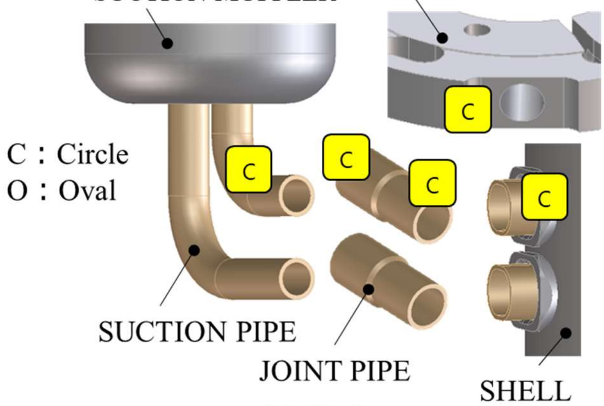
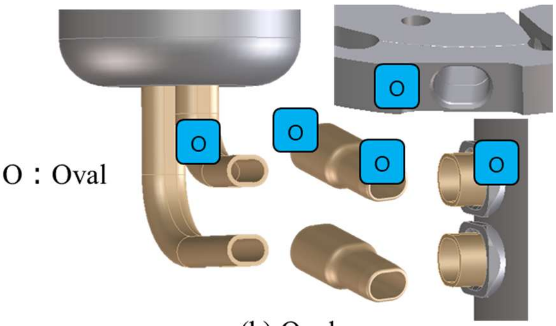


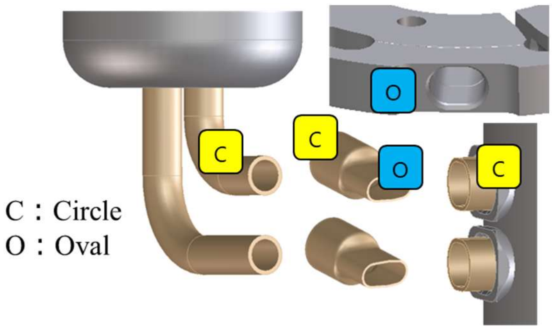
Fig.3 Structure of compressor and development elements

る課題が存在する．そのため当社はシリンダ歪を低減する生産技術である「熱かしめ固定」工法を活用することで組立歪を緩和した．吸入流路縮小については従来の「円連結管」Fig.4(a)に対し，当社はシリンダ吸入穴を径方向に広げることで流路有効面積を拡大する「扁平連結管」Fig.4(b)により流路縮小による効率低下の課題を解決してきた．しかしながら Fig.4(b)のようにシリンダ，シェル，連結管，吸入管全てを扁平形状にすることは加工負荷（加工時間）が上昇してしまう課題が存在した．そこで連結管の末端を丸形状，先端を扁平とした「丸-扁平連結管」Fig.4(c)を導入することで，成形性の良い丸形状のシェル吸入穴と丸形状の吸入管の構成で仕様を成立し圧縮機の高効率と低加工負荷を両立させた．





(b) Oval



(c) Circle-Oval

Fig.4 Circle-Oval structure

さらに，ロータリ圧縮機における圧縮室はシリンダ，ローリングピストン，ベーンで構成されており，高圧空間から低圧空間への冷媒漏れの多くはシリンダ内径とローリングピストン外径間の半径隙間から発生する（Fig.5）．このローリングピストン側面漏れ面積は[シリンダ巾]×[シリンダ内径とローリングピストン外径の距離（隙間）]で求められることからシリンダの薄型化が漏れ損失低減に有効であり高効率化に繋がる．これらから，効率改善しつつ当社 9.2cc モデルを代表するとシリンダ高さ 13.6mm 仕様の 10.8mm 化により体積 20.6%削減を実現した．

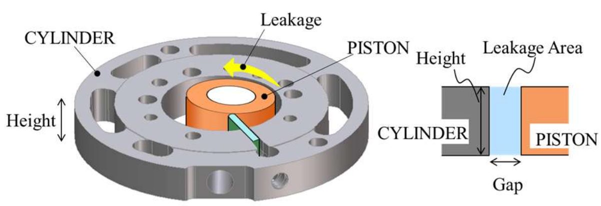


Fig.5 ROLLING PISTON side leakage

２．２　ワイドレンジの課題と解決手段

ロータリ圧縮機は高速運転時に遠心力によって軸が振れまわり，軸摩耗や振動増加に繋がることから回転数上限を設定している．そこで回転数上限を拡大しワイドレンジに使うことを可能とすれば 1 機種で賄う能力レンジが拡大し，現行搭載機種に対して 1 サイズ小さな行程容積で小型軽量の圧縮機を選定できるようになりダウンサイジングが可能となる．これらは小型軽量化効果だけでなく，機種ラインアップ仕様数削減に繋がり製造時の段取り削減による省消費電力化や，1 サイズ小さな行程容積にすることで同じ能力を出すための回転数を増加できることから圧縮室からの漏れ率が低い高速側での運転を可能とし圧縮機の高効率化が見込める．そこで，当社は高速運転時の軸振れまわりを抑制する為に圧縮機モデルの 1D-CAE を開発した（Fig.6）。

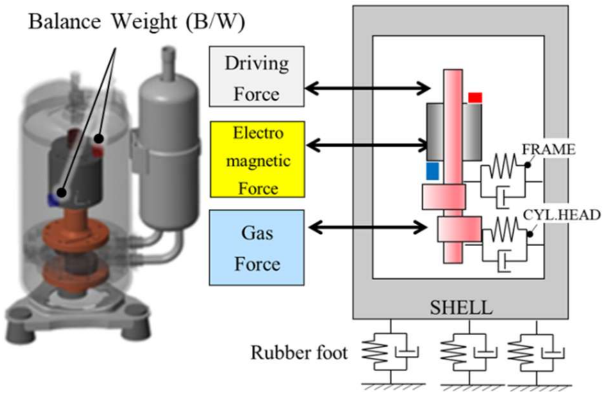


Fig.6 1D-CAE conceptual model

圧縮機はガス圧縮荷重，モータ磁気吸引力，遠心力など複合分野に跨る力を受ける複雑な現象であり部品間振動伝達を解く複雑な解析が必要とされるが，本 1D-CAE はマルチフィジックスモデルやマルチボディダイナミクスモデルを活用することで素早い解析を可能とし，設計・評価に時間を要する上下重量差のある「非対称バランスウェイト」を含めたバランスウェイトの最適設計手法を確立した．これにより当社 14cc モデルを代表すると Fig.7 のように，従来モデルの最大回転

数である 130rps 時の軸振れまわり量を基準にすると 160rps 時には 200%まで振れまわりが増大してしまうが，1D-CAE を活用した最適設計によって導出された非対称バランスウェイト搭載によって 37%の 160rps 時軸振れまわり量抑制を実現した．この結果，当社 M 形（シェル外径φ132.3）33cc モデルから S 形（シェル外径φ107）28cc モデルへの置き換えなどシェル径を含めた大幅なダウンサイジングを実現した．更に本 1D-CAE は実機結果のフィードバックによる解析値合わせ込みを可能としており，開発を通して常に精度が向上する特徴を持っている．

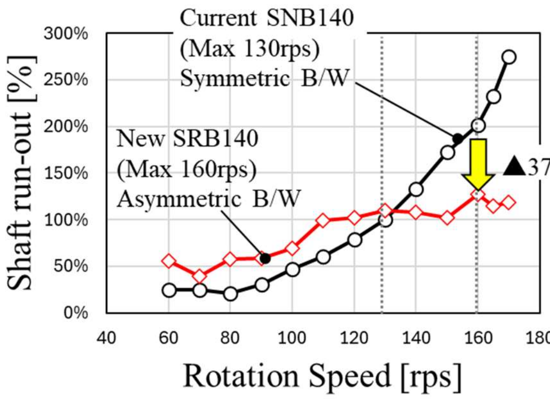


Fig.7 Shaft run-out

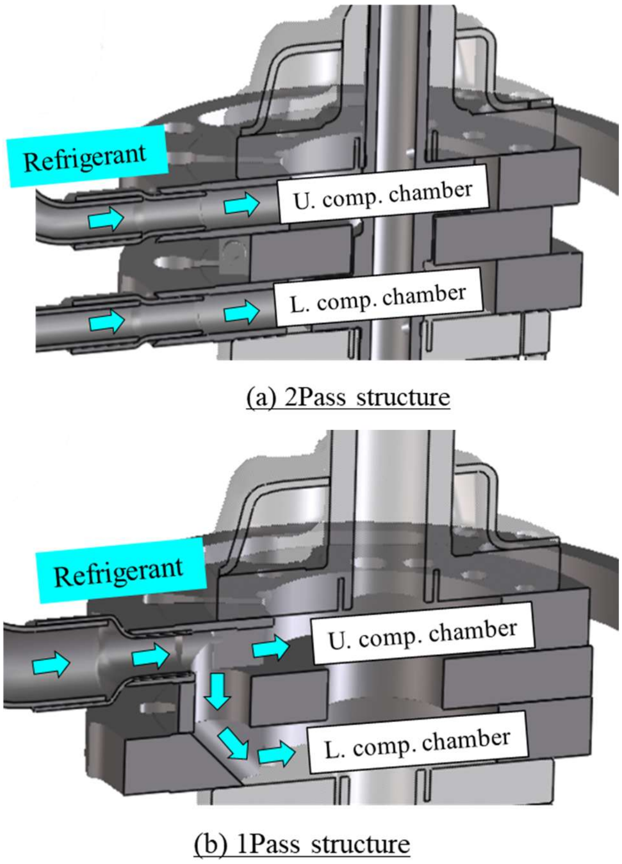


Fig.8 2pass and 1pass structures

２．３　１パス吸入管の課題と解決手段

ツインロータリ圧縮機は圧縮室を 2 つ備えることから吸入管を 2 本備えた 2 パス吸入管構造 Fig.8(a)が一般的である．また，この吸入管はエルボ構造をもち，圧縮機シェル本体へのロウ付け性も考慮し銅製にすることが一般的な部品だが，近年の銅価格高騰もあることから部品数削減かつ銅使用量削減が可能な 1 パス吸入管構造 Fig.8(b)を採用した．

しかし圧縮室それぞれに吸入管が直列で接続される 2 パス吸入管に対し，1 本の吸入管から流れ込んだ冷媒がメカ内部で並列に分岐してから各圧縮室に流れ込む 1 パス吸入管構造は圧縮時のローリングピストン側面漏れや吐出弁が開放し吐出穴と吸入穴が連通する際の高圧冷媒が吸入穴に向かう逆流冷媒が対の圧縮室に流入する（例：上シリンダからの逆流冷媒が下シリンダに流入する）．この逆流により吸入圧力損失が大きくなり 2 パス吸入管に対して低効率となる．そこで，吐出弁開放時の逆流を抑制する為に吸入穴の反ベーン溝側を吸入穴内側に絞り，吸入閉じ角を縮小させた「ホイッスル構造」（Fig.9）を導入した．これにより 1 パス構造で顕著となる逆流による吸入圧力損失を低減し高効率な 1 パス構造を実現した．

なお，丸-扁平連結管は Fig.4(c)のようにシェルの吸入穴部に連結管の丸管側が挿入可能である必要があり，シェルへの連結管挿入を可能とするには丸-扁平連結管は扁平部よりも丸管部が大きいことが要求される．このことから 2 パス吸入管仕様に対し吸入管径を拡大しやすい 1 パス吸入管仕様は丸-扁平連結管と相性が良い．

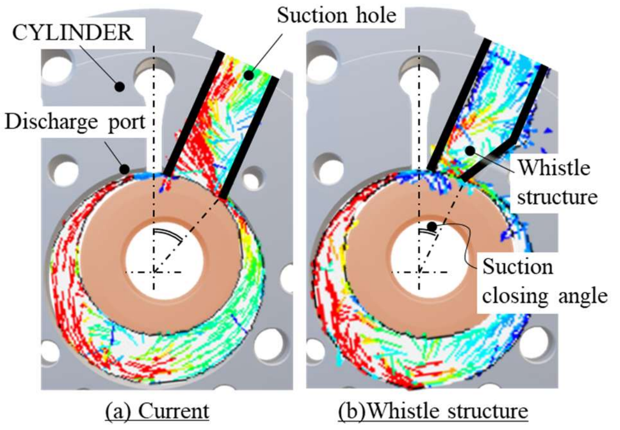


Fig.9 Flow velocity distribution in the compression chamber immediately after the Discharge valve opening of Current and Whistle structure

３．おわりに

温室温効果ガス排出量削減に向けて，ツインロータリ圧縮機の北米市場における急速な冷媒規制対応および小型軽量化を図った．薄肉シリンダにおける吸入流路縮小，ワイドレンジにおける軸振れまわり，1 パス吸入管における逆流冷媒の課題をそれぞれ丸-扁平連結管，非対称バランスウェイト，ホイッスル構造の技術導入によって解決し，当社従来 R410 冷媒対応 SNB 型に対し効率（省エネルギ性）同等を維持しつつ北米市場における圧縮機重量削減率 13.8%を実現可能な R454B 対応「SRB 形圧縮機」ラインアップ展開を実現した．今後は本仕様を標準仕様とし，当社既存拠点（日本・中国・タイ）に加え新拠点（インド・米国）を含めグローバルに展開し，カーボンニュートラル実現を目指し貢献していく．

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