

# Development of High-Performance, High-Speed, Compact Centrifugal Compressor Stage

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*In recent years, the multi-stage centrifugal compressors for chemical plants have been required to be more compact and to be highly efficient. Mitsubishi Heavy Industries, Ltd. (MHI) have been developing up-to-date stage (including impeller) configurations with a high flow coefficient, high revolution speed (high mach number) and high efficiency for Mitsubishi Centrifugal Compressor (MAC). This paper describes this development process.*

## 1. Introduction

Many of the centrifugal compressors used in chemical plants employ an one-shaft multi-stage type configuration in which the main shaft is supported by bearings that are lined up with the impellers. Each stage consists of an impeller which compresses gas and a diffuser channel which leads gas to the impeller in the next stage. The gas is compressed by each stage and its volume flow is reduced. The channel width of the impeller becomes narrower as the gas goes to the downstream stages, and this means that the shaft is lined up with many stages of different volume flows.

The Mitsubishi Centrifugal Compressor (MAC) holds a standard stage in these stages for each flow coefficient and renews the stages year by year with the aim of achieving a more compact, higher-speed and highly efficient compressor. This paper outlines the development process of this compressor.

## 2. Components of the centrifugal compressor stage

The components of the centrifugal compressor stage and

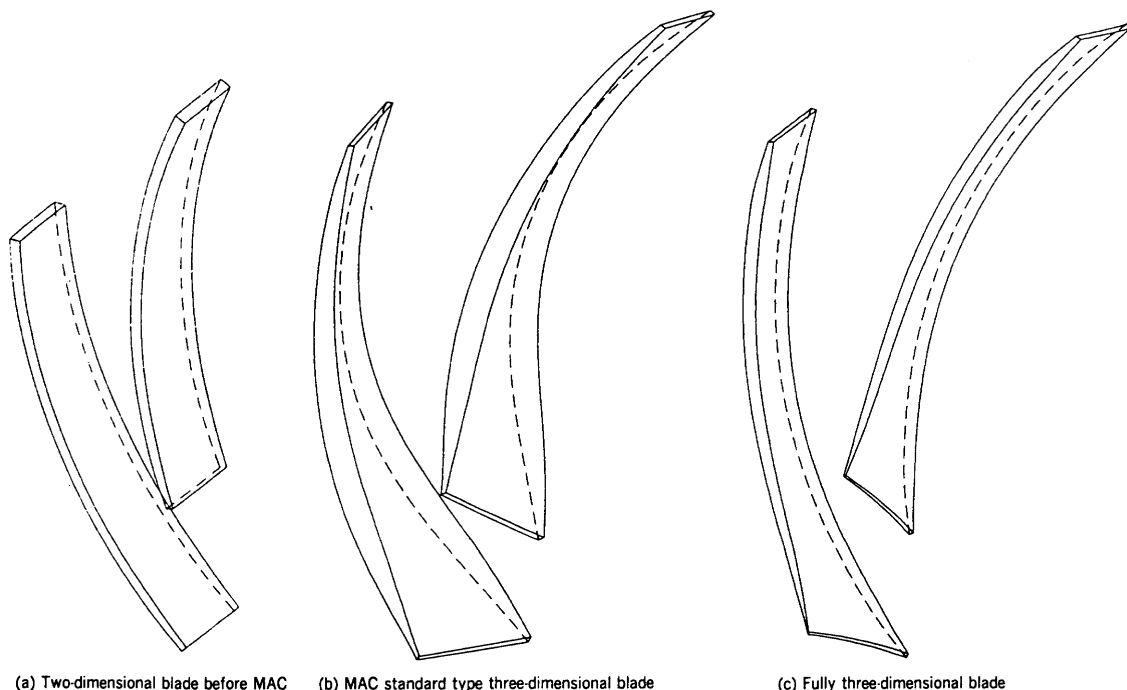
their characteristics are described below.

### (1) Impeller

The impeller is the most important part among the stage components, and is very important in raising pressure by efficiently giving energy to gas. As shown in Fig. 1, the impeller blade configuration has been modified for higher performance. Fig. 1(a) shows two-dimensional blades before development of the MAC. In many examples, part of a cylinder or cone has been used for the blade configuration. Fig. 1(b) shows the standard stage of the MAC, which has three-dimensional blades consisting of straight line elements. Recently, fully three-dimensional blades as shown in Fig. 1(c) have been developed using Computational Fluid Dynamics (CFD) and advance production technology, and the performance is being improved.

### (2) Diffuser

The diffuser is a channel which raises static pressure by changing dynamic pressure at the impeller exit to static pressure. Rapid deceleration causes separation of the boundary layer and a reverse-flow of gas. Therefore,



**Fig. 1 Configuration of centrifugal impeller blade**

Changes made in the configuration of the centrifugal impeller blade are shown.

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attention is paid to the design. Further, there are generally two types of diffusers: those without vanes and those with vanes. Either type is used depending on the use of the compressor.

### (3) Return channel

The return channel is a channel which leads the flow at the diffuser exit to the impeller inlet of the next stage. It consists of a bend, which changes the flow in the outer direction at the diffuser exit to a flow in the inner direction and return vanes which change the flow of gas to an axial flow at the impeller inlet of the next stage removing the inflow swirling flow.

The return channel is one type of diffuser, and reduction of loss is an important theme for improving performance.

### (4) Labyrinth seal

The labyrinth seal prevents gas leakage from the clearance between the rotating and static parts at the fore and aft sides of the impeller. There are direct type, step type and abradable type seals. The materials used in the seal consists of metal such as aluminium for the seal fin and nickel graphite spray, polymers, and the like which are easy-machinable materials for abradable type seals.

In impeller of low specific speed, the circulating loss leaking from the labyrinth forms a large portion of the total loss. Consequently, the low leakage, abradable type labyrinth seal is effective in improving performances.

## 3. Development of compressor stage

### 3.1 Method of development

#### (1) Aerodynamic design

The aerodynamic design of the standard stage is performed by the following procedure.

- The outlined one-dimensional value, that is, the radii of the impeller inlet and exit, channel width, blade angle, etc. are determined using a performance prediction program.
- Then, three-dimensional configuration of the channel is created by determining the channel wall surface and the blade angle distribution connecting the values of the inlet and exit.
- The three-dimensional configuration created is compared with the conventional one. The configuration is evaluated using fluid velocity distribution through inviscid flow analysis of the singular point method or the Denton code and using viscous flow analysis of the Dawes code base.
- The existing performance verification data is also considered in the evaluation.
- The one-dimensional value and three-dimensional distribution are modified, and the final configuration can be obtained by repeating this procedure.
- Accuracy in the viscous flow analysis is an important factor in this design work. Thus the accuracy of the analysis is also verified.

Fig. 2 shows the calculation results of the representative internal flow based on viscous flow analysis. A comparison is made of the analytical values and experimental data for improvement of the efficiency of the new and conventional impellers at the representative stage design flow point.

The flow inside the centrifugal impeller is characterized by very complicated flows, which include secondary flow on the blade surface and wall surface as well as a thick boundary layer which develops on the shroud suction surface. In spite of such complicated flow, the relative difference of efficiency in the analytical values shows the same trends in an increase and decrease of efficiency compared with the relative difference of efficiency in the experimental data.

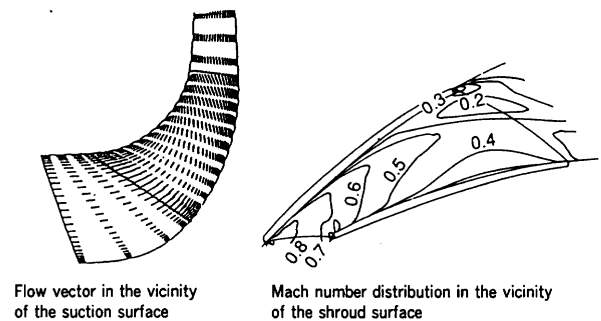
Fig. 3 shows a comparison between the viscous flow analysis results and experimental data obtained at the return channel exit. It can be seen that total pressure at the exit tends to decrease on the shroud side and the exit angle tends to approach zero before becoming negative in the vicinity of the wall surface, although the exit angle is positive in the center of the channel.

#### (2) Strength design

When the three-dimensional configuration of the impeller is created, structural strength is also evaluated in addition to carrying out the above flow analysis evaluation. The configuration is modified so that local stress as well as average stress due to centrifugal force are less than the material allowable stress. Further, vibration strength is evaluated and measures are taken in order to prevent resonance and reduce vibrating stress.

### 3.2 Development process

The development process up to now has been aimed of realizing a compact and highly efficient centrifugal compressor as introduced below. For details regarding the development



Example of calculation of internal flow

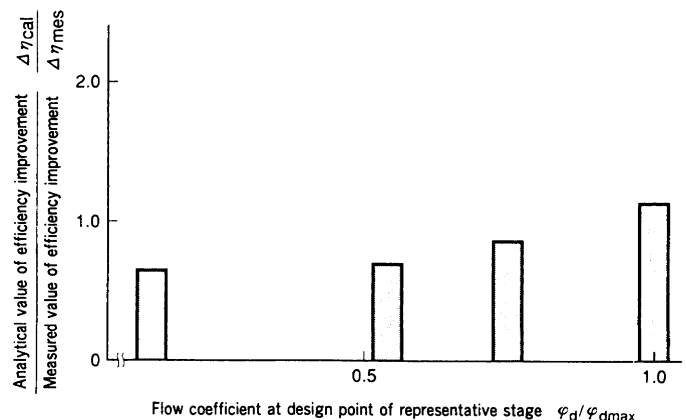


Fig. 2 Verification results of viscous flow analysis on improvement of impeller efficiency

Comparison of analytical values and experimental data for improving efficiency between new and conventional impellers at the representative stage design flow point is shown.

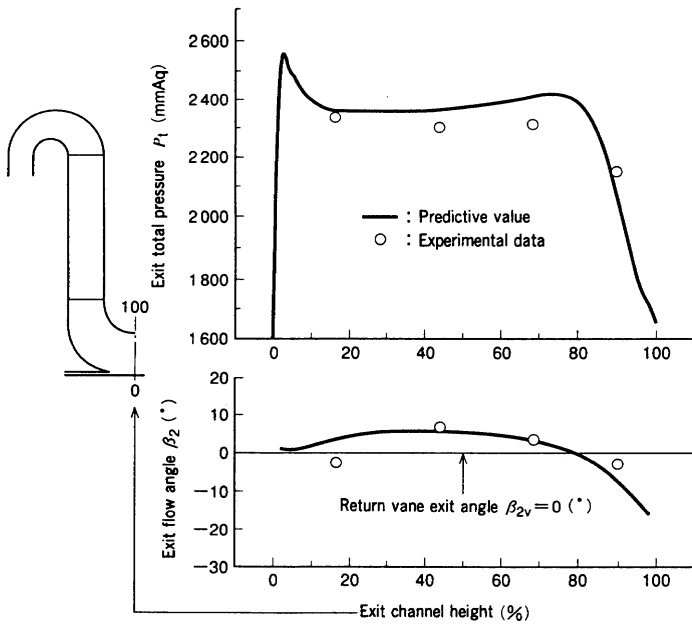


Fig. 3 Verification results of viscous flow analysis at return channel exit

Comparison between calculated values and experimental data in the viscous flow analysis at the return channel exit is shown.

of the prototype compressor refer to references (1) and (2). Further details regarding high-speed and high-mach number technology are presented in reference (3).

#### (1) Large flow coefficient stage

When the rotational speed  $U_2$  is considered as being constant in cases where a larger flow coefficient  $\varphi_d = Q_{o1} / \frac{\pi}{4} D_2^2 U_2$  is selected, the outer diameter of the impeller,  $D_2$ , becomes smaller and the compressor becomes compact even with the same volume flow  $Q_{o1}$ . However, when the flow coefficient  $\varphi_d$  is made larger, the blade height of the impeller becomes larger, and the three-dimensionality of the channel is increased and the internal flow becomes complicated.

The large flow coefficient stage has been developed by controlling increases of the fluid velocity at the channel inlet and optimizing loading distribution so that the internal flow is not distorted by secondary flow or other factors. Verification tests of the above developed configuration using the single stage test machine, described in Section 4, confirmed that polytropic efficiency was increased by 1 to 3% compared with the conventional one.

#### (2) High-mach number stage

The pressure ratio of the impeller is increased and the number of stages is reduced by selecting a higher rotational speed  $U_2$ , whereas the mach number of the relative inflow at the impeller inlet becomes larger by the increase of  $U_2$ .

When the mach number of the flow inside the impeller becomes larger, compressibility of the flow is increased and the deceleration ratio at the impeller inlet and exit becomes smaller resulting in a decrease in efficiency and an increase in dangerous stall.

In the design of the high-mach number stage, the deceleration ratio at the impeller inlet and exit was optimized, and a configuration capable of reducing the

maximum mach number inside the impeller as far as possible was employed. Verification tests of the high-mach number stage was conducted using a single stage test machine. It was confirmed that efficiency was improved by roughly 1 to 2% in the high-mach number zone 1.1 to 1.3 times as much as the conventional mach number.

#### (3) Higher-mach number and lower stress stage

When the rotational speed of the impeller is not so high although the machine mach number  $M_{o1}$  ( $M_{o1} = U_2 / a_{o1}$ ,  $a_{o1}$ : acoustic velocity in the inlet stagnation condition) is high in the gas of low acoustic velocity, the high-mach number stage described in the preceding paragraph can be applied. However, when the machine mach number is raised in the case of gas having a high acoustic velocity, centrifugal stress increases and may exceed the allowable limit stress of the material.

Therefore, the stage in which performance is not reduced up to a high-mach number and in which centrifugal stress is also reduced has been developed to deal with high rotational speed. The configuration of the low stress structure and enduring high-mach number has been determined by repeating the aerodynamic design and structural analysis.

As a result, as shown in Fig. 4, stress is decreased by 20% of that of the conventional type stage and efficiency is improved by roughly 1 to 3% in the high-mach number and high-rotational speed zone of 1.2 times as much as the conventional rotational speed.

#### (4) Development of highly efficient stage

Efficiency in the large flow coefficient zone and high-mach number zone has been improved for compact compressors. However, research and development is being

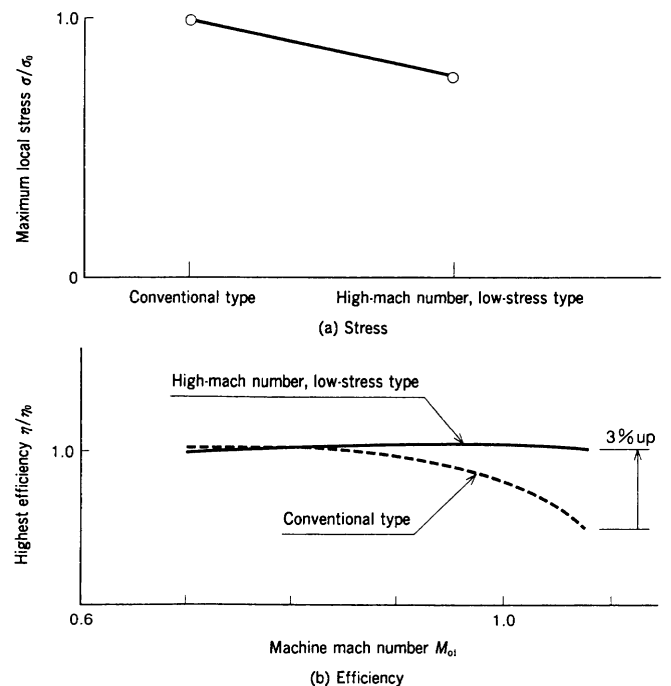


Fig. 4 Example of comparison of efficiency and centrifugal stress between high-mach number, low-stress stage and conventional stage

Comparison of efficiency test results and the maximum stress calculated using NASTRAN between high-mach number, low-stress stage and conventional stage are shown.

pursued with the aim of achieving even greater efficiency in up-to-date stages. A brief overview of some of these efforts is introduced below.

The impeller of the centrifugal compressor in general employs a blade surface constructed with straight line elements connected between the blade coordinate on the hub wall surface and the blade coordinate on the shroud wall surface. In this development, a free curved surface blade constructed with point groups is employed.

Further, supercomputers, EWS, and the like can be used at a lower cost and far the shorter time compared with the previous situation. The accuracy of the analysis code is also being improved. Therefore, evaluation by CFD analysis can be repeated many times in the design review stage.

Fig. 5 shows the verification results obtained using a single stage test machine. It was confirmed that efficiency at the design flow coefficient point was improved by 3.5%, and the operation range was extended compared with conventional impeller.

#### 4. Performance and strength verification

##### 4.1 Single stage performance test

The performance of the stage configuration including the impeller developed was verified using a single stage test compressor, a schematic of which is shown in Fig. 6. In this

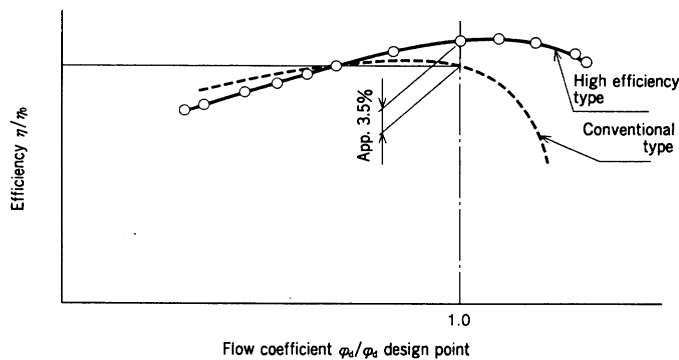


Fig. 5 Example of test results for improving efficiency of high efficiency stage  
Efficiency curves of high efficiency stage and test results of improvement in efficiency of the conventional stage are shown.

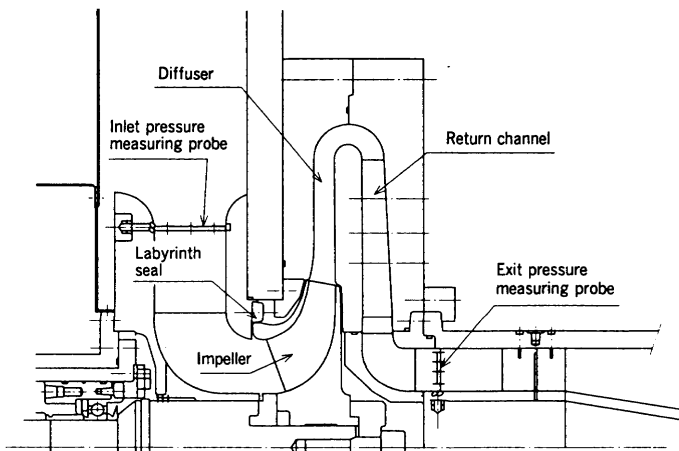


Fig. 6 Single stage test compressor  
Test equipment for performance and flow measurement in the standard stage is shown.

equipment, the single stage configuration of the multi-stage centrifugal compressor is completely reproduced and constructed with a suction channel, impeller, diffuser and return channel.

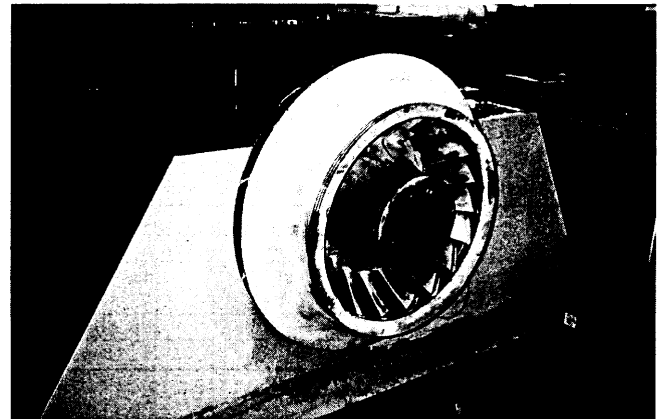
The impeller is placed on the overhanging shaft and driven by a thyristor motor via a step-up gear. Input to the impeller is measured by a torquemeter, while pressure rise is measured with a total pressure probe at the inlet and exit. Flow is measured using an orifice flow meter located inside the discharge piping. As a result of this measurement the efficiency, the flow coefficient and pressure coefficient are calculated, and the stage performance curve as shown in Fig. 5 can be obtained at the various machine mach number.

The test machine is constructed so as to be able to measure static pressure, total pressure and flow traverse on the way of the stage channel. As a result, not only the overall performance of the stage but also performance by the various channel elements, such as the impeller, diffuser, etc. can be separated.

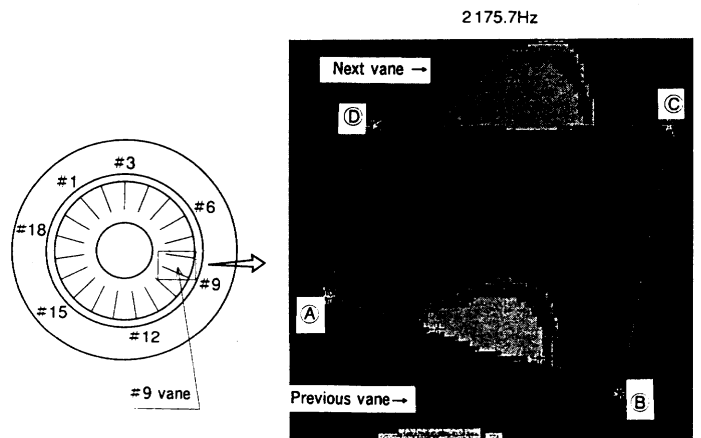
Further, verification tests of performance can be conducted in the structure of the installation of the suction casing channel before the impeller or installation of the scroll after the diffuser as well as in the construction shown in Fig. 6. Low-speed flow test equipment by which simple and detailed flow tests can be performed is also used in the development of the static channel such as the return channel.

##### 4.2 Spin test and vibration test

The strength of the impeller that has been developed is



(a) Test impeller



(b) Measurement results of vibration velocity distribution of the blade

##### Fig. 7 Example of vibration test of impeller

Measurement results of vibration velocity distribution of the impeller using the Laser Doppler vibration meter is shown.

verified through spin tests and vibration tests. The spin test is conducted at a rotational speed equivalent to 115% of the design rotational speed, and any deformation is confirmed to be within allowable limits.

Fig. 7 shows an example of the vibration test, which is the measurement results of the vibrational speed distribution of the impeller using a Laser Doppler vibration meter. The impeller vibrates at varying frequencies in the state shown in Fig. 7 (a). Fig. 7 (b) shows the measurement results of the vibrational speed distribution at a certain frequency in the resonance state. The vibration modes of the vanes are clearly found.

## 5. Conclusion

This paper has presented a brief overview of the develop-

ment process, representative examples and the verification test equipment for the Mitsubishi centrifugal compressor stage. Key points for maintaining and upgrading the quality of the real machine is research and development into improving stage performance and reliability. The authors intend to continue such development in the future.

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