THE USE OF COMPUTATIONAL FLUID DYNAMICS AND SCALE MODEL COMPONENT TESTING FOR A LARGE FCC PROTOTYPE AIR COMPRESSOR

by

Michael Kotlyar

Senior Staff Rotating Equipment Engineer

ARCO Indonesia

Houston, Texas

Reid Engstrom

Principal Mechanical Engineer

ARCO Products Company

Carson, California

and

Marco Giachi

Senior Engineer

General Electric/Nuovo Pignone

Florence, Italy

Michael Kotlyar is a Senior Staff Rotating Equipment Engineer on the ARCO Indonesia Tangguh LNG Project, in Houston, Texas. He is involved with specification and selection of large industrial type gas turbines and compressors and development of the plant reliability model. Prior to this assignment, he was a Senior Machinery Engineer at the ARCO Los Angeles Refinery, responsible for engineering, maintenance and reliability of rotating and reciprocating equipment. Previous positions include Senior Member of Technical Staff for Parsons Corporation, involved with equipment selection, testing, installation, and commissioning for the North Slope Development Program, and Design Engineer for Rotoflow Corporation, involved in design of high speed cryogenic turboexpanders and compressors.

Mr. Kotlyar has an M.S. degree (Mechanical Engineering, 1968) from the Moscow Automechanical Institute. He is a member of ASME, has coauthored four technical papers, and is the inventor named in two U.S. patents.

Reid W. Engstrom is a Principle Mechanical Engineer at ARCO Products Company, in Carson, California. He has more than 18 years experience in the industry and is presently involved in management of rotating equipment projects, rotating equipment preferred supplier agreements, and maintenance/reliability of existing refinery machinery.

Mr. Engstrom has a B.S.M.E. from the University of Wisconsin and is a registered Professional Engineer in the State of California. He presently serves as ARCO's representative on the API Mechanical Subcommittee.

Marco Giachi is Senior Engineer responsible for Centrifugal Compressors in the Research and Design Group, General Electric/Nuovo Pignone, in Florence, Italy. His professional background is mainly concentrated in experimental aerothermodynamics. He joined General Electric/NuovoPignone in 1994 after experience as an aerodynamicist and wind tunnel test engineer in the aeronautical and automotive industries. His activity at GE/NP has been primarily dedicated to centrifugal compressor design.

Dr. Giachi received his degree (Aeronautical Engineering, 1985) from Pisa University and a post graduate diploma (Fluid Dynamics, 1987) from Von Karman Institute in Brussels.

ABSTRACT

The purpose of this paper is to describe the use of computational fluid dynamics (CFD) and scale model testing techniques that were used to accurately predict and confirm the performance of a large centrifugal air compressor. Compressor performance was later verified by a field performance test.

INTRODUCTION

A fluid cat cracking unit (FCCU), at a large California refinery had been seriously limited by air compressor performance for about 10 years. Two previous attempts to rerate "the main air blower" had failed. Other FCCU debottlenecking projects were successful, so costly oxygen injection was used to support the feed rate to the unit. A project was initiated to replace the four-stage, centrifugal "main air blower" with one of higher capacity and better efficiency. Goals of the project were to:

- Install a machine that would increase the capacity by approximately 30 percent.
- Install a machine that would continuously operate for a minimum of five years between servicing or overhauls.

- Install a machine that would fit the existing compressor "footprint" to minimize structural and piping modifications.
- Reuse the existing 20,000 hp steam turbine driver. The new compressor would have to be significantly more efficient in order to achieve a 30 percent rate increase without increasing the power of the driver.

Application discussions were initiated with several compressor manufacturers. A new three-stage prototype design with three-dimensional impellers was chosen after review of preliminary CFD studies that confirmed the feasibility of the project. The physical size of the chosen design is noteworthy. The diameter of each impeller is approximately six feet and the shaft diameter is approximately two feet.

Schedule constraints precluded a full-scale ASME PTC 10 Class 3 test. Furthermore, full-scale testing would require the erection of a large, costly structure to support the test driver (steam turbine) and the new compressor. A decision was therefore made to use scale model testing of all flow path components, followed by field testing, to verify the performance of the compressor.

Three 1:4.625 scale models were manufactured and tested using specially dedicated testing facilities. Each scale model reproduced a stage of the compressor, consisting of an impeller and its upstream and downstream stationary components. The inlet of the first stage impeller, the upstream suction line, and the compressor inlet volute were also reproduced in scale and tested, resulting in a "complete flow path" series of tests from the atmospheric inlet to the compressor's discharge flange. Comparisons were made between CFD predictions and actual test results.

Field performance tests were later conducted to verify the compressor performance in the field and to confirm the validity of the CFD predictions and the scale model test results.

BACKGROUND

The fluid catalytic cracking unit (FCCU) is the main gasoline producer in modern "conversion type" refineries, where heavy gas oils are "cracked" into lighter hydrocarbon chains. This reaction coats the catalyst with a hydrocarbon film that must be burned-off to make the process continuous. The burning occurs in a large vessel called a regenerator, and requires huge quantities of air at low pressure. The compressor that supplies the air to the regenerator is commonly called "the main air blower" because of the high flow and low head requirements.

The original "main air blower" (i.e., the "old" blower) was installed when this FCCU was built in 1953. The design came from the steel industry, which used large "blowers" to supply air to blast furnaces. It was a four-stage, foot mounted, centrifugal compressor with a cast iron case and riveted steel impellers. A 10,000 hp condensing steam turbine was direct connected to provide power at approximately 2500 rpm. The "old blower" was designed to supply approximately 108,000 acfm at 22 psig discharge pressure, with a polytropic efficiency of approximately 75 percent.

In 1968, the turbine was rerated to approximately 13,000 hp at 2700 rpm. The higher speed resulted in a capacity of approximately 132,500 acfm at 22 psig discharge (a flow increase of about 23 percent). There were no modifications made to the compressor.

Further production increases were desired by the mid-1980s. A study was performed to determine if the capacity of the blower could be increased again. This resulted in a project to rerate the steam turbine to approximately 18,000 hp at 2700 rpm, and to rerate the compressor for a flow of approximately 145,000 acfm at 29 psig discharge. The compressor rerate consisted of four new impellers—diaphragm design was not changed. The turbine power increase was successful, but the compressor rerate was not. Extensive field tests were made and actual turbine power was determined to be approximately 20,000 hp. With a speed increase to 2800 rpm, the compressor would supply about 141,000 acfm at a discharge pressure of 28 psig. The measured polytropic efficiency was 72 percent.

One final attempt to rerate the compressor was made in 1992 by redesigning and replacing the diaphragms. This attempt was also not successful—there was no flow improvement. Individual stage tests indicated the machine was "choking" in the interstage return bends and in the discharge volute. The turbine governor was adjusted for a speed increase to 2900 rpm. This resulted in a capacity of approximately 147,000 acfm at 28 psig discharge pressure. At this point, the rest of the FCCU had been extensively debottlenecked, and costly oxygen injection was needed to support plant feed rates. The measured efficiency of the compressor was only 68 percent at this point. It was now clear that rerates had taken this machine as far as it was going to go, and further increases would have to come from different equipment.

PROJECT SCOPE AND DEVELOPMENT

The 1998 FCC turnaround was the next opportunity to improve the air supply situation. The only realistic option was to install a new compressor in place of the old one. This was a tall order—the requirement was to produce 30 percent more flow with no reduction in head and with no power or speed increase from the driver. The economics were such that the machine had to fit into the same "footprint" as the old, in order to avoid costly structural and piping modifications.

In mid-1996, a study was initiated to take a fresh look at the idea of a new compressor. The biggest constraint was the existing footprint. As was mentioned above, the old blower was a four-stage machine and the flow was choking in the stationary components (i.e., return bends and discharge volute). The natural inclination would be to build a longer machine in order to obtain greater spacing between stages and on the discharge end of the casing. This option was rejected because the equipment deck size and the opening for the blower were not suitable—a longer machine would be extremely expensive to install. The only space available was radial. The idea of using a larger diameter was therefore put forth. This approach could provide for more head per stage (the old blower was making approximately 10,000 ft per stage). One manufacturer proposed a three-stage compressor with impellers that would measure 1.85 m (approximately six feet) in diameter. After careful review of this proposal, it was concluded that this approach was feasible. As with many refinery projects, time was of the essence, with about 15 months available to build, test, ship, and install the new compressor.

The decisive factor that permitted the project to proceed were the manufacturer's computational fluid dynamic (CFD) studies and testing capabilities. Advances in fluid dynamics, modern computational techniques, and significant improvements in testing methodology led to the conclusion that a three-stage compressor could be designed and manufactured. It was understood that all three impellers would have to be of a three-dimensional design and would be the largest three-dimensional centrifugal impellers the manufacturer had ever made (probably the largest ever made by any company). Given the axial length restrictions, the design of the inlet and discharge volutes was an interesting and challenging task. The compressor would use a forged-steel fabricated casing to avoid the risk of cracking, which had been a problem with the old cast iron blower.

This is a machine with downward inlet and discharge nozzles. The inlet nozzle is 80 inch in diameter. To conduct a PTC 10 Class 3 test would require construction of a highly elevated and costly structure to support the blower and the steam turbine test driver. While not technically challenging, the time required to mount both machines and do the performance tests would be considerable. The project schedule dictated that other options be considered. Also, in order to minimize the risk at the end of the project, there was a great desire to conduct some advance testing. Past experience has taught the lesson that problems discovered during full-scale performance testing will have a grave impact on the project schedule. Considering the size of the forgings, and their delivery time, advanced testing was a prudent thing to do. The decision was made to conduct all performance testing using scale models and to limit full scale factory testing to a standard API mechanical run. This is the first time an entire compression system had been modeled and tested by this manufacturer. The goal of the scale-model testing program was to confirm compressor performance early in the project. That would allow changes in design, if necessary, to be implemented prior to the manufacturing of its critical components. This testing program included all three compressor stages, the discharge volute, and the compressor inlet volute (including a scaled-down model of the existing plant inlet piping).

MODEL TESTING

The new compressor is a three-stage machine characterized by large diameter impellers (1.850 m diameter; Figures 1 and 2). The three compressor stages were reproduced utilizing models 4.625 times smaller than the real size. The three models reproduced the stages inclusive of the impeller and the stator stationary parts downstream and upstream. Each stage is defined between an inlet section and an outlet section in such a way as to reproduce the entire gas path through the machine, as shown in Figure 3. Swirl conditions are reproduced on inlet to the second and third stages by an array of stator blades that simulate the effect of the stages upstream. The scale was selected in such a way as to have impellers of 400 mm diameter, as is standard for this manufacturer's rig testing of models. The models are made of aluminum and steels (Figure 4).

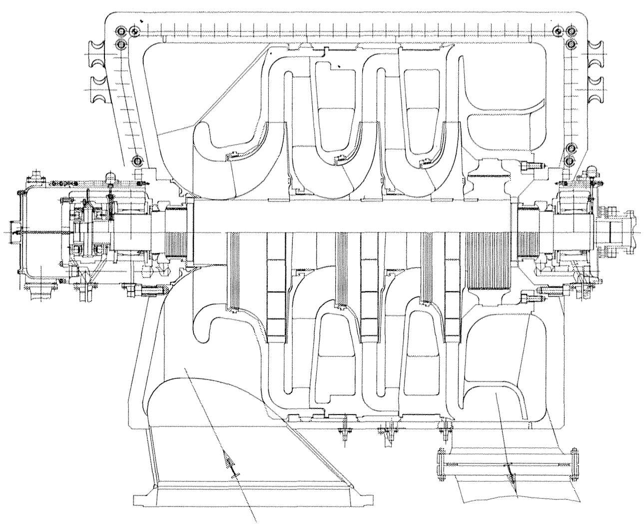


Figure 1. Meridian Section of Compressor.

Prior to testing the stages, a test was conducted on a model of the inlet (Figure 5) to verify the distortion conditions of the flow on the inlet to the first impeller. The presence of a radial inlet preceded by a 180 degree curve in the suction duct, and the project constraints imposed to the inlet plenum geometry (i.e., to fit the compressor into the existing plant footprint) may, in extreme cases, affect impeller performance. Consequently, it is important to verify that the first stage test conditions are effectively representative of real conditions. The duct model was built to the same scale as that of the stages to allow the component parts of the suction volute to be utilized also in tests on the first stage.

INSTRUMENTATION AND DATA PROCESSING

The three models were instrumented according to the manufacturer's standard (Benvenuti, 1978a, 1978b), which calls for rake-mounted "Kiel" probes for measuring total pressure (Figure 6), three-hole "Cobra" probes for measuring direction of velocity, static pressure taps on the walls for static pressure, and thermocouples for measuring total temperature.

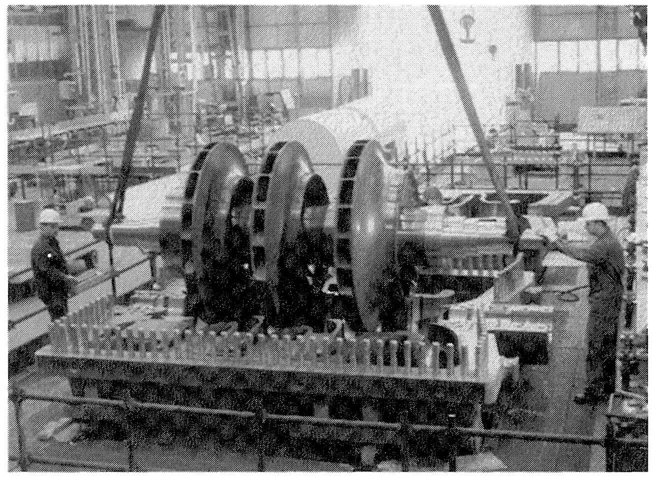
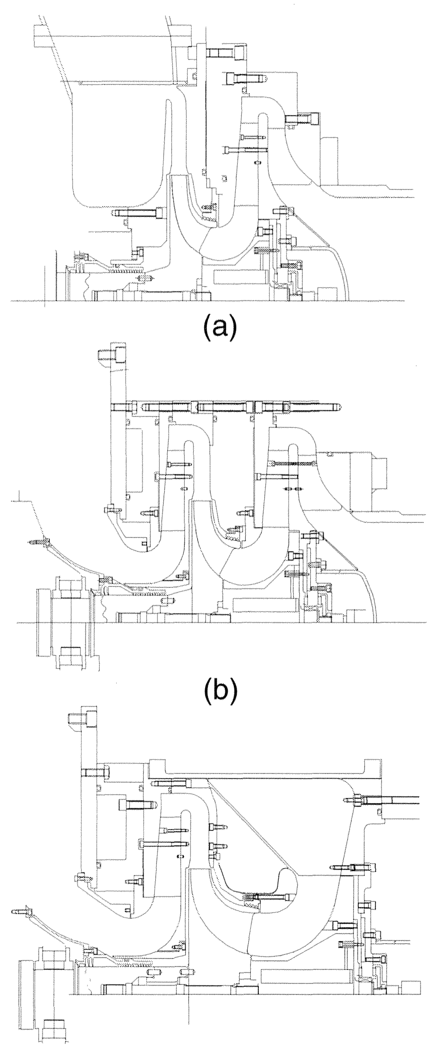


Figure 2. The Compressor During Installation of the Rotor in the Casing.

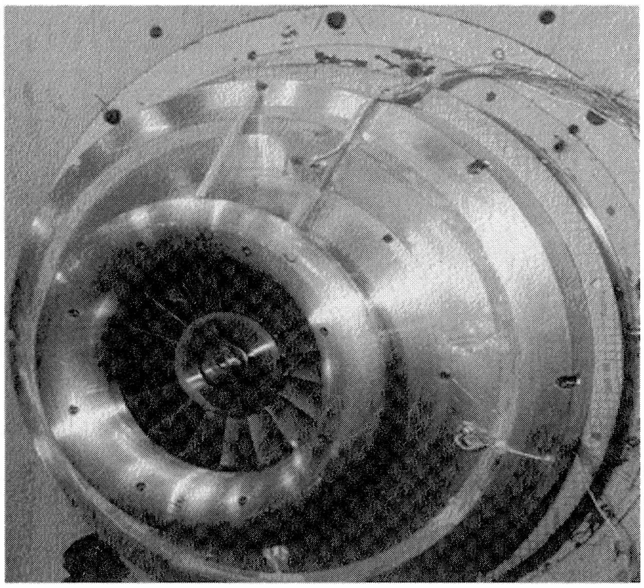


(a)

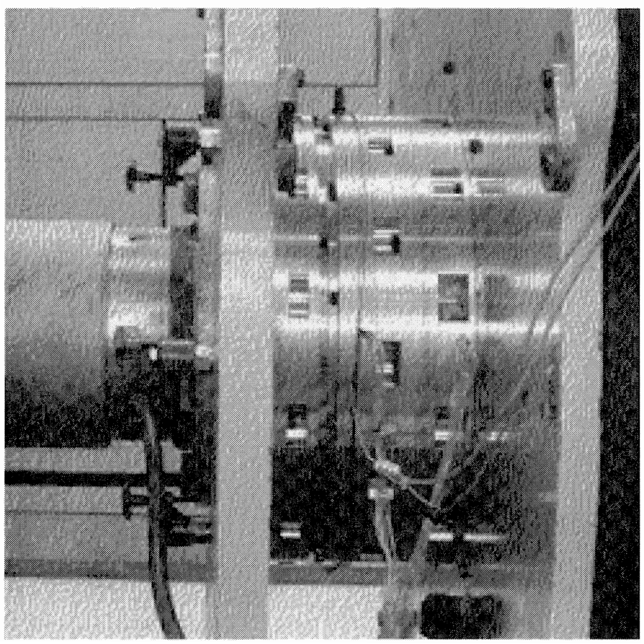
(b)

(c)

Figure 3. Meridian Sections of the Three Models.



(a)



(b)

Figure 4. Scale Models of First and Second/Third Stages.

DYNAMIC SIMILITUDE

The primary quantities for the compressor performance evaluation are: efficiency and energy input to the flow. These two quantities depend on the amount of volumetric flow that is handled by the machine, the rotational speed, and the properties of the gas. It is well known from dimensional analysis (Dixon, 1978) that all the above-mentioned quantities can be reduced into some dimensionless groups in such a way that one can predict the performance of a prototype from tests conducted on a scale model.

Measurements for calculating efficiency and nondimensional head coefficients consist of the total temperatures and pressures on the inlet to the stages and on the outlet from the stages. The measured temperature and pressure values along with that of flow rate are processed to furnish the performance values for the stage in nondimensional terms (polytropic efficiency and coefficients of work and of head) in relation to the flow coefficient and the tip Mach speed.

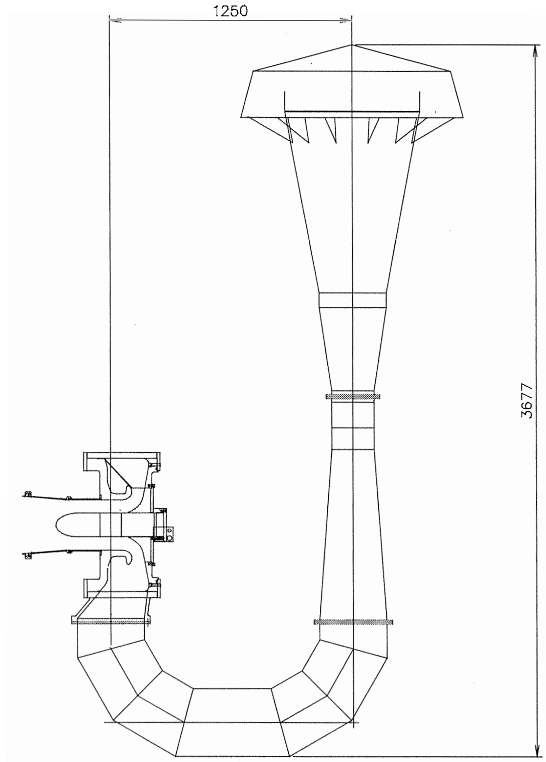


Figure 5. Model of Inlet.

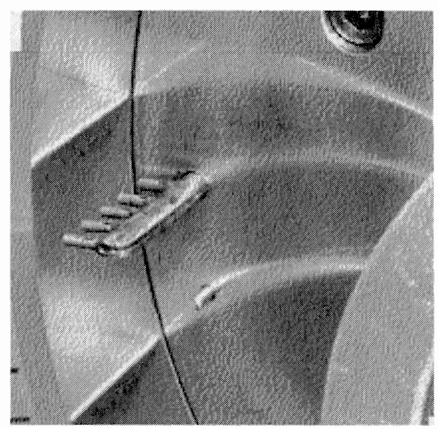


Figure 6. Rake of "Kiel" Probes for Measuring Total Pressure.

- Work coefficient $\tau = \tau\,(\varphi_1$, Mach, Reynolds, gas$)$
- Polytropic efficiency $\eta = \eta\,(\varphi_1$, Mach, Reynolds, gas$)$
- Head coefficient $\tau\eta = \tau\eta\,(\varphi_1$, Mu$)$

The coefficients are defined as indicated in Table 1, with their values in Table 2.

The tests were conducted in dynamic similitude with the real conditions regarding nondimensionalized flow rates and Mach number. Similitude in the ratio between volumes is automatically satisfied due to the fact that the test gas (air) is the same in real and in laboratory conditions. The conversion thermodynamics are exactly the same for real conditions and for those of the model in the laboratory. Only the Reynolds number, which is obviously affected by the scale, is lower in the model by a factor of 4.625. It is considered that this has no significant influence on the results, since both the values of this parameter (model and real conditions) are higher than the critical values. ASME standards (ASME, 1997) indicate a critical value of 2×10^5 for the Reynolds number. A comparison between real conditions and model conditions is given in Table 2.

Table 1. Definition of Quantities Utilized in Processing Test Data.

Quantity	Expression	Meaning of symbols
Coefficient of volume flow rate	$\varphi_1 = \frac{4 \cdot Q}{\pi \cdot D_2^2 \cdot U_2}$	Q = Volume flow rate with reference to conditions on inlet to stage D_2 = Impeller diameter (0.4 m for the models) U_2 = Impeller tip speed
Tip Mach speed	$Mu = \frac{U_2}{a}$	a = Speed of sound with reference to conditions on inlet to stage
Reynolds number	$Re = \frac{U_2 \cdot b_2 \cdot \rho}{\mu}$	b_2 = Width of impeller outlet ρ, μ = Density, viscosity of air at stage inlet conditions
Coefficient of work	$\tau = \frac{C_p \cdot (T_{out} - T_{in})}{U_2^2}$	C_p = Specific heat (1007 for air) T_{in} = Total temperature on inlet to stage T_{out} = Total temperature on outlet from stage
Polytropic efficiency	$\eta = \frac{k-1}{k} \cdot \frac{\log\left(\frac{P_{out}}{P_{in}}\right)}{\log\left(\frac{T_{out}}{T_{in}}\right)}$	k = Ratio of specific heat values (1.4 for air) P_{in} = Total pressure on inlet to stage P_{out} = Total pressure on outlet from stage

Table 2. Comparison Between Real Conditions and Model Conditions.

	Quantity	Real Conditions	Model
1st stage	Impeller diameter (m)	1.850	0.4
	RPM (1/min)	2900	12600
	Tip Mach number	0.775	0.775
	Flow coefficient	0.1171	0.1171
	Gas constant	1.4 (air)	1.4 (air)
	Reynolds number	2.4×10^6	5.1×10^5
2nd stage	Impeller diameter (m)	1.850	0.4
	RPM (1/min)	2900	11700
	Tip Mach number	0.72	0.72
	Flow coefficient	0.0892	0.0892
	Gas constant	1.4 (air)	1.4 (air)
	Reynolds number	2.1×10^6	4.5×10^5
3rd stage	Impeller diameter (m)	1.850	0.4
	RPM (1/min)	2900	10850
	Tip Mach number	0.668	0.668
	Flow coefficient	0.0705	0.0705
	Gas constant	1.4 (air)	1.4 (air)
	Reynolds number	1.7×10^6	3.7×10^5

Normally, the size of the model is selected to find a compromise between the achievable Reynolds number and the cost, together with the power required to run the scale model test. Furthermore, the model has to be big enough to make its geometry a good reproduction of the real one and to be unaffected by standard machining tolerances. For the present case, impeller diameters of 400 mm (scale of 4.625:1) were selected.

The tests were conducted using rigs 1 and 2 (Figure 7), which are normally used for tests on scale models. The two test rigs are practically the same in general structure, but differ in the power of the electric motor installed, which is 1000 kW on rig 1 and 500 kW on rig 2. In both cases, the impeller is driven by a variable-speed electric motor, while flow is controlled by a valve inserted in the system and measured by an orifice plate. Speed of rotation and flow can be controlled separately so that, for each test, the entire curve of the stage, from surging to choking, is explored.

RESULTS OF TESTS ON MODEL OF INLET SECTION

These tests were conducted to determine distortions in the flow on inlet to the first impeller. The measurements of total pressure, static pressure, and direction of flow were organized in the form of maps, shown in Figure 8, of distortion at the reference section (aerodynamic interface plane (AIP)).

The purpose of these tests was not that of evaluating performance, but rather of determining these conditions in order to compare them with those present on the inlet to the impeller in the first stage model. As a general comment, it can be observed that, in spite of the footprint constraints and the 180 degree bend, the first stage inlet flow conditions for this machine are not unlike those normally present in machines of this type (Michelassi and Giachi, 1997).

From the maps of total pressure and Mach number, the absence of local areas at low pressure and low speed can be noted,

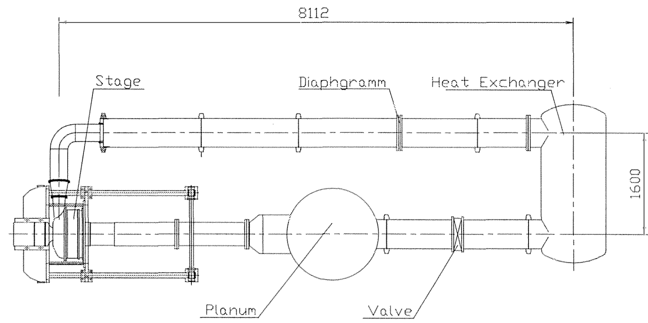


Figure 7. Schematic Layout of Test Rig 2.

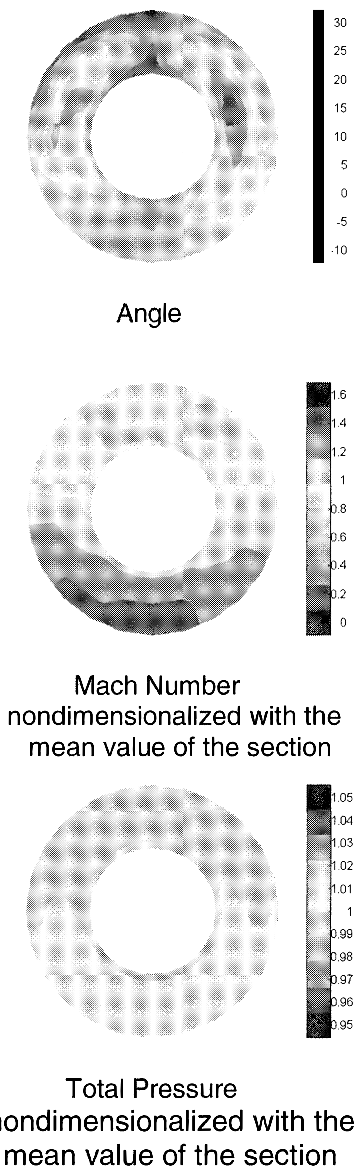


Figure 8. Maps of Distortion at the First Stage Impeller Eye.

indicating the absence of any areas of separated flow. The flow rate is higher in the lower half of the inlet section, due to the effect of the radial suction flange that is positioned in the lower part. The 180 degree curve immediately upstream of the suction flange produces no appreciable effects.

TEST RESULTS

The results of the tests are summarized in the curves of the nondimensional coefficients shown in Figure 9. The polytropic efficiencies of the chosen stages are in the range of 0.83 to 0.87. Considering the geometrical constraints that have been imposed on this design, these are judged to be exceptional, and represent a huge improvement over the measure efficiency of the existing plant compressor (i.e., the existing "main air blower"). The work coefficient is quite typical for impellers having a blade exit angle of 50 degrees.

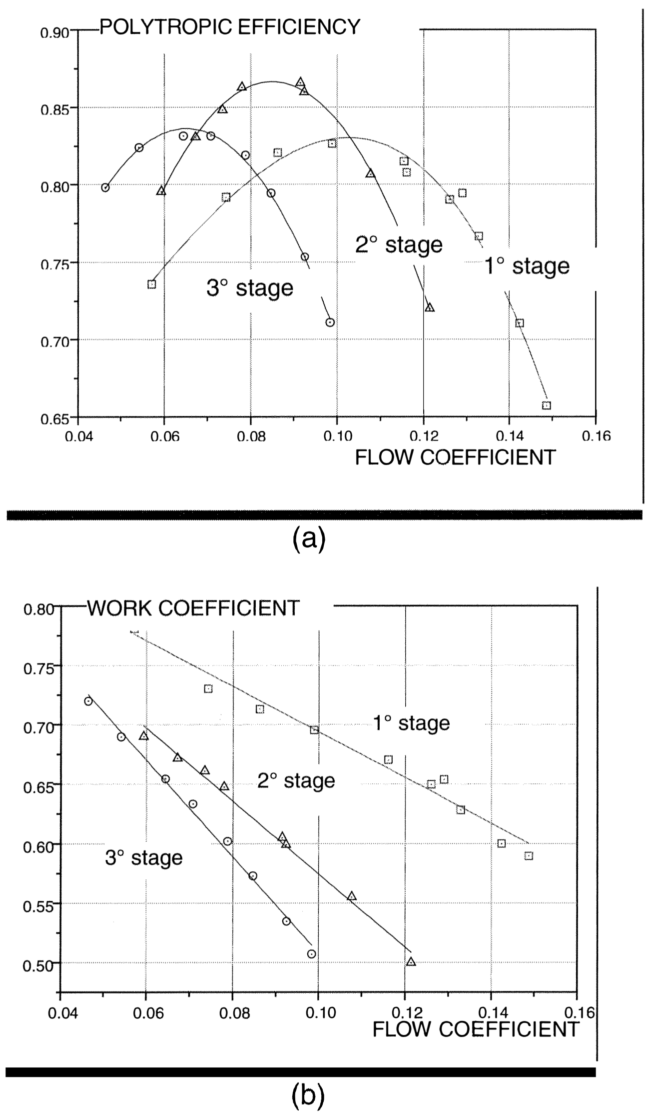


Figure 9. Thermodynamic Performance in Terms of Nondimensional Coefficients.

The performance curves of the stages have been repeated at Mach numbers lower and higher, respectively, than that of design (Figures 10 and 11). The availability of specific data relevant to different speeds has made it possible to calculate the entire operating map of the real machine at different speeds, based on performance actually measured at each rpm.

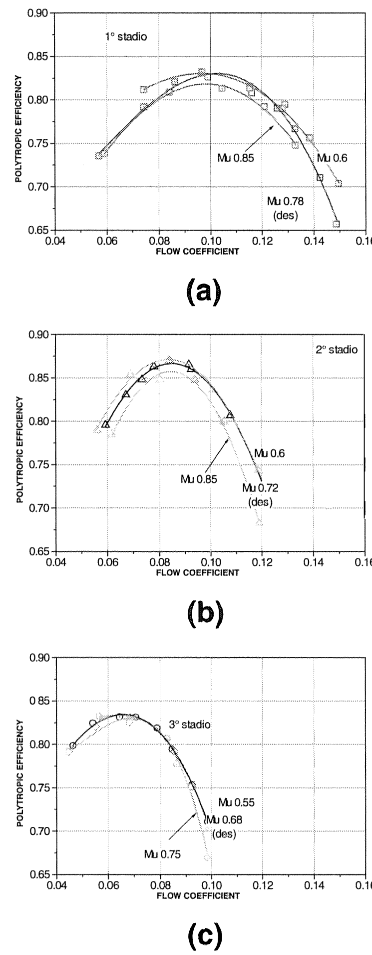


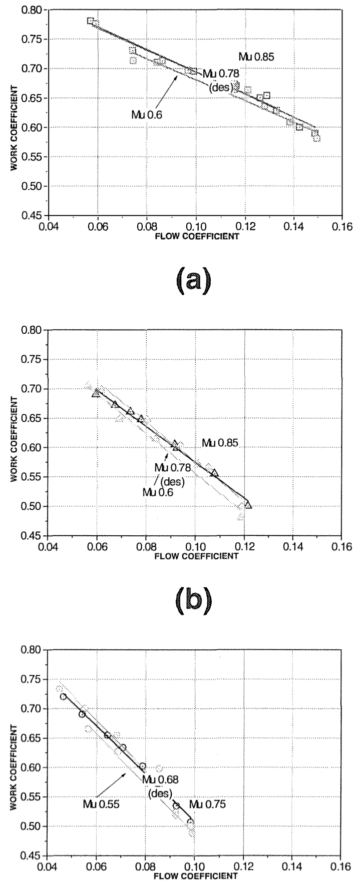
Figure 10. Efficiency of Stages at Mach Numbers Other than Design.

Efficiency appears lower at the higher Mach numbers for each stage (more evident for the first and second stages), while the coefficient of work is slightly higher at the higher Mach numbers (to a similar degree for each of the three stages). The differences are, however, very slight and performance appears substantially independent of the Mach number. This agrees with the fact that the impellers operate at quite low Mach numbers, and thus in fields that are certainly subsonic, where physical phenomena are substantially Mach-independent.

INSTALLATION AND FIELD TESTING

The "new" main air blower was delivered on schedule. The old blower was removed and the new one installed in early 1998 (Figure 12). The new blower fit like a glove—there was about one-half inch of radial "room to spare" between the sidewall of the casing and the deck beams. A field performance test was then conducted to verify contractual obligations and also to determine the correlation between CFD studies, the scale model tests, and the actual field performance.

Field performance was verified through a set of measurements taken directly onsite during startup of the machine. The measurements were made during the period of February to March 1998, and thus at a rather low ambient temperature, corresponding to design case D. The compressor was instrumented with pressure taps and thermocouples on the suction and discharge flanges, in accordance with ASME standards. Flow rate was measured utilizing a pitot tube inserted directly in the narrow section of the suction duct. Calibration measurements had been previously made to determine the calibration coefficient of the section and to be able



(c)

Figure 11. Coefficients of Work of Stages at Mach Numbers Other than Design.

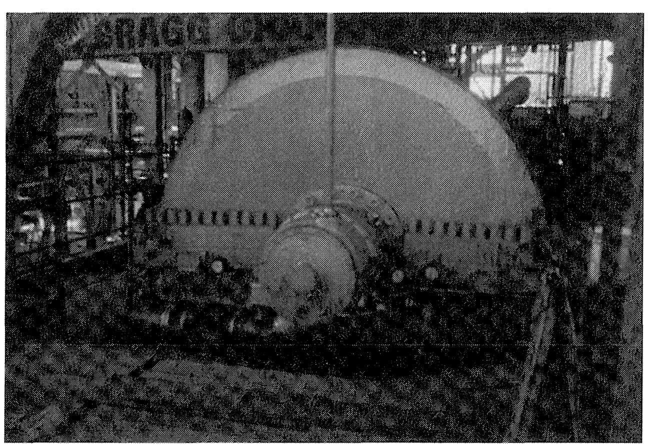


Figure 12. The Air Compressor Installed in the Refinery.

to correlate the speed measured at one point through the pitot tube with the mean effective speed (i.e., the flow rate) in the section.

Comparison between the performance values estimated on the basis of the data obtained from tests on the models and the performance values actually measured is shown in Figure 13.

The diagrams in Figure 13 show excellent agreement between the predicted values and those measured onsite. The differences between the values obtained from the field testing of the real machine and those predicted from the model testing are approximately ± 1.5 percent for efficiency and + 3 percent for

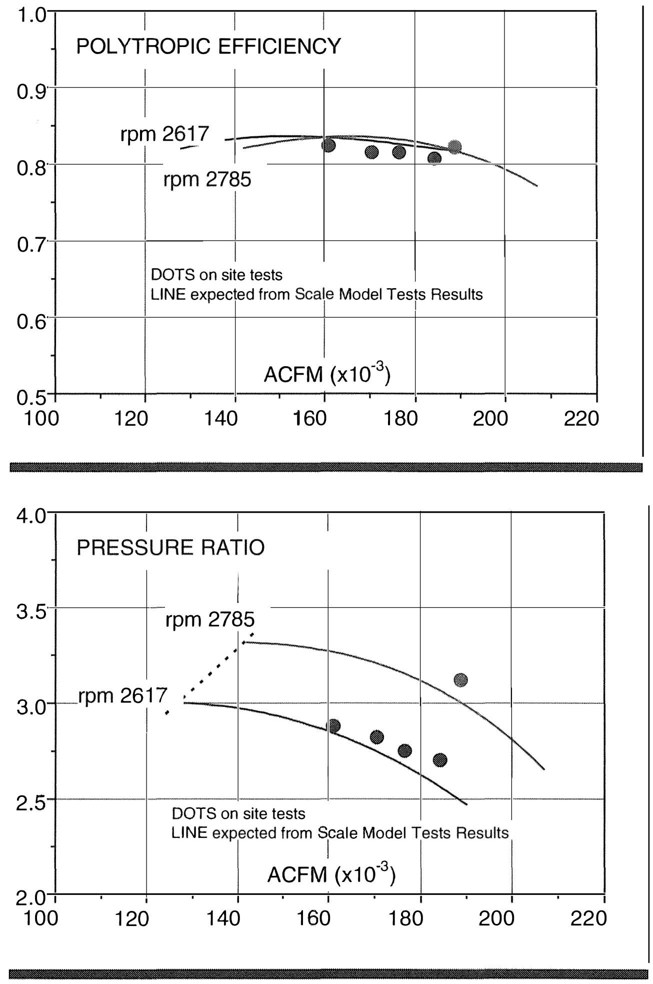


Figure 13. Comparison Between Performance Values Estimated on the Basis of Tests on Models and Performance Values Measured Onsite on the Real Machine.

compression ratio, depending on the operating point. The differences are of the same order as the precision of measurement system.

Field measurements were taken at two different speeds during the startup procedure. This would have been impossible during normal plant operation, where it would have to run at the speed imposed by the plant.

Field tests confirmed the success of the new blower. The FCCU is no longer constrained due to lack of air, and costly oxygen injection is no longer needed.

CONCLUSION

Perhaps the most obvious conclusion from the years of experience with this application is that rerates to old equipment can only go so far. At some point, equipment has to be replaced with the help of new and better technology. In addition, the following conclusions can be drawn from this project:

- Computational flow dynamics can be used to accurately predict prototype performance. The CFD studies were critical to the decision to proceed with the project. Subsequent model and field tests confirmed their accuracy and validity.

- The three models reproduce, separately, the whole machine, so that the entire gas path through the compressor is simulated. From the results of the tests on models, the predicted performance values have been calculated and compared with the real values measured on the machine onsite. The comparison has confirmed the correctness of this approach with differences of less than three percent in compression ratio and 1.5 percent in polytropic efficiency.

- The approach to verification of performance utilizing models has the further advantage of saving time. In the case described here, having test data available prior to assembly of the real compressor made it possible to shorten delivery time by several months, a determinant factor to the success of the project.

REFERENCES

Benvenuti, E., 1978a, "Aerodynamic Development of Stages for Industrial Centrifugal Compressor. Part 1: Testing Requirements and Equipment—Immediate Experimental Evidence," ASME Paper 78-GT-4.

Benvenuti, E., 1978b, "Aerodynamic Development of Stages for Industrial Centrifugal Compressor. Part 2: Test Data Analysis, Correlation and Use," ASME Paper 78-GT-5.

Dixon, S. L., 1978, *Fluid Mechanics*, *Thermodynamics of Turbomachinery*, Oxford, United Kingdom: Pergamon Press.

ASME PTC 10, 1997, "Compressors and Exhausters," American Society of Mechanical Engineers, New York, New York.

Michelassi, V. and Giachi, M., 1997, "Experimental and Numerical Analysis of Compressor Inlet Volute," ASME 97-GT-481.

ACKNOWLEDGEMENTS

The authors wish to thank ARCO and General Electric/Nuovo Pignone for permission to publish this article. The authors also wish to express their appreciation to the following individuals for their help and guidance in this project: John Wolflick, Steve Konig, Greg Crowell, Raul Morales, Ernie Leuenberger, Jack Schrock, Roy Carroll, Shawn Enarson, John Lemanski, Elio Ceccarelli, Franco Sarri, Alberto Viti, Paolo Bendinelli, Massimo Camatti, Antonio Bartolini, Angelo Dei, and Alfredo Paci.