DYNAMIC SIMULATION OF CENTRIFUGAL COMPRESSOR SYSTEMS

by

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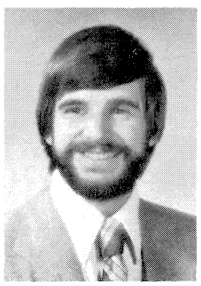

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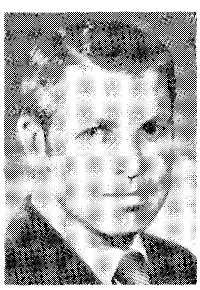

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ABSTRACT

The capacity, pressure, speed and horsepower of centrifugal compressors have been trending higher and higher in the past few years. Not only are machine designers faced with increasing aerodynamic and mechanical problems, but application engineers now have to cope with greater complexity in designing safe, economic control systems.

This paper discusses the application of dynamic simulation to centrifugal compressor control system design. Three complex compressor systems which have been designed with the aid of simulation are presented. Two of the compressors are refrigeration units installed in a liquified natural gas plant. The third is a high pressure gas injection compressor. Each of the units consumes power in the range of 25,000 to 40,000 BHP.

The simulation studies resulted in design recommendations concerning the number and location of recycles required, sizing of recycle control valves, and setpoint, gain, and reset settings for control system instrumentation. The studies also led to a better understanding of how the compressors would interact with processes. Start-up procedures were tested and amended where required.

The procedures followed in developing the simulation models are outlined. The structure of the mathematical models and the method of solution of the models are discussed.

INTRODUCTION

Centrifugal compressors have been assigned increasingly tougher process roles in recent years. Not only have capacity, horsepower and speed increased, but, in some processes, the centrifugal compressor has become a central component. In these processes, profitability can rest directly upon the efficiency and operability of the plant's compressors.

The compressor designer and the application engineer, who is responsible for the design of the entire process, are faced with the problem of designing machines which are often larger and more complex than ever built. Somehow they must also provide reasonable assurances that the machine will operate as intended once it is installed. This process of satisfying one's self (and the plant's owner) that the compressor system design will perform satisfactorily can be very subjective. Traditional methods of analysis cannot provide accurate predictions of how a compressor and a process will interact. So system design usually has been based upon the designer's experiences and opinions.

A tool which has been extremely valuable in evaluating the performance of a compressor system prior to installation is dynamic simulation. Dynamic simulation involves writing a set of differential and algebraic equations, which describe the time-dependent behavior of a compressor and a process, and implementing this mathematical model on a computer. The computer representation of the compressor system is then used to predict how the real system will behave, both during steady-state operation and during the transition between steady-states.

A general discussion of the applications of simulation to compressor system design is below. Following are discussions of three compressor systems which have been designed with aid of simulation. Included are an outline of the systems, the objectives of each study and the major benefits obtained. The final section of this paper addresses the procedures of development and implementation of simulation models for compressor systems.

USE OF SIMULATION IN COMPRESSOR SYSTEM DESIGN

In the past the design of centrifugal compressor control systems has been based upon rules of thumb, calculations based upon steady-state conditions, and the designer's personal experience. The systems were installed and tuned in the field. If difficulties were encountered, modifications were made and tested until satisfactory performance was achieved. Often, if performance still was not satisfactory, the system was operated manually.

This approach cannot be tolerated for many of the centrifugal compressor systems being installed today. The potential losses in terms of capital costs of field modifications, lost production due to excessive downtime, and increased production costs due to inefficient operation are too great to allow the degree of field experimentation done in the past.

For even a moderately complex compressor such as that shown in Figure 1, it is impractical if not impossible to do manual calculations or use intuition to predict the dynamic performance of the system.

The system shown consists of a two-casing centrifugal compressor with individual recycles around each casing. Each recycle is controlled by a flow-ΔP anti-surge system. The compressor is driven by a steam turbine whose speed is varied to maintain a constant pressure in the low pressure suction drum. The performance characteristics of each casing are shown in Figure 2.

In this example, the high pressure casing has a greater rise to surge and stable range than the low pressure casing. Suppose now that an occurrence downstream reduced the flow demand on the compressor. The HP casing would be first to sense the upset. The pressure at its discharge would rise and the flow rate would decrease causing the casing's anti-surge controls to open its recycle valve. The decrease in flow demand and the recycle flow would then cause the LP casing's discharge pressure to rise. The questions that now arise are: Will the LP casing with its low rise to surge and stability go into surge? Should the set point of the LP anti-surge controller be set at a value further from surge? Or, should the speed with which the HP casing's controls react be slowed, and if this is done, is the HP casing's protection being sacrificed to protect the LP casing? Or, finally, do any problems exist at all?

For some systems, answers to these questions might be obtained using intuition and manual calculation. If one then considers how the anti-surge and suction pressure controls

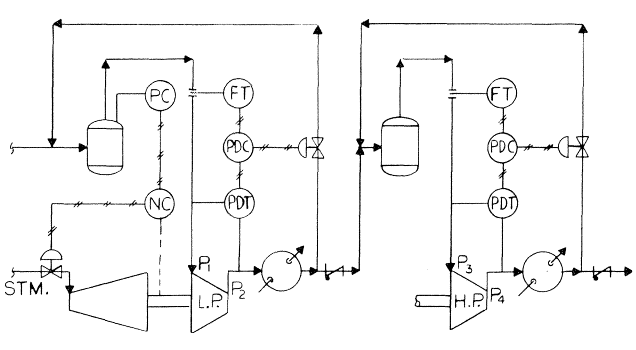

Figure 1. Typical Centrifugal Compressor System.

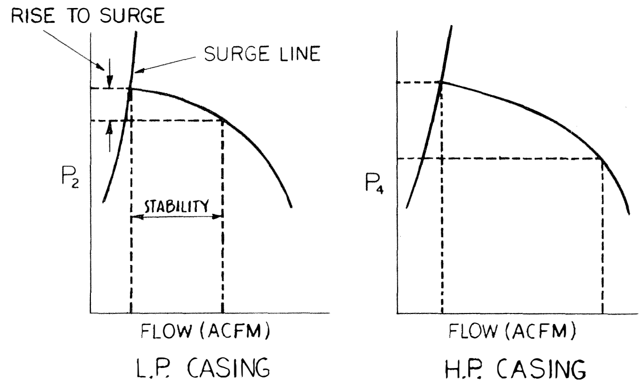

Figure 2. Centrifugal Compressor Characteristics.

might interact or considers molecular weight variations, the situation becomes even more complex and probably insoluble. It is not hard to understand why much of the system design has been a matter of building what one thought would be a workable system, and then modifying it in the field if necessary.

The system in Figure 1 is the final design of an actual compressor system. This system was not simulated and was originally designed with a single recycle as shown in Figure 3. The recycle valve was manipulated by the anti-surge controller requiring the greater recycle flow. Intuition told the designers that there should have been two individual recycle loops, but the capital cost of a single loop was lower, and it couldn't be proved that the single loop would not work. So the system was installed with a single recycle.

When the system was started up, there were some conditions at higher molecular weights where the controls could not prevent surge. After considerable analysis, it was determined that one casing was operating in stonewall while the other was operating in surge. The speed controls interacted with the recycle controls so there was no recycle valve position which resulted in sufficient flow to satisfy the process while keeping both casings out of surge.

The controls were adjusted and modified but they could not be made to work. In the end, the system was changed to a two recycle configuration which operated successfully.

Had this system been simulated during the design pahse of the project, this process of trial, analysis and modification in the field, and the expense of modification and loss of revenue could have been avoided.

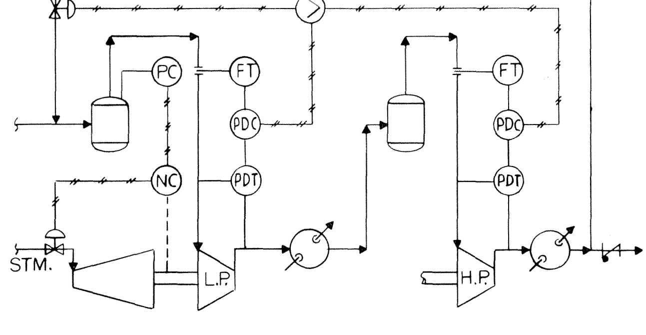

Figure 3. Initial Design of Compressor System Shown in Figure 1.

Simulation allows the engineer to pre-test a design both statically and dynamically. The procedure of trial and analysis, modification and another trial and analysis is done on a computer. A proven final design results and the risk of expensive field modifications or delays is greatly reduced.

Some of the specific benefits of simulation are:

- The operability of a proposed system can be evaluated, and any necessary modifications made before construction.
- A wide range of alternative control schemes can be tested. The system can then be designed with more efficient controls rather than with excessively conservative schemes.
- Start-up procedures can be tested. In some systems, the driver may have a torque limit. If the initial inventory is too high, this limit could be exceeded. In multi-recycle valve systems, the sequence and timing of the valve closures can be critical to keeping the compressor out of surge. By using simulation, these limits can be determined.
- The sizing and stroke times for recycle or other control valves can be checked for all operating conditions.
- Initial controller proportional band and reset settings can be determined. The scaling of transmitters and computing instruments can be checked.
- The overall arrangement of exchangers, check valves, vents, etc., can be evaluated to maximize controllability and safety.
- System performance during emergency trip or the effects of equipment failure or operator error can be evaluated.
- Acceptance criteria for ASME performance loop tests can be developed.

CASE HISTORIES

Three centrifugal compressor systems which have been designed with the aid of dynamic simulation are presented below. Two of the systems provide refrigeration for a liquefied natural gas (LNG) plant. The third system is a high-pressure gas injection compressor. All of the simulation studies were performed on a large capacity hybrid computer.

PROPANE REFRIGERATION SYSTEM

This system, schematically shown in Figure 4, provides secondary refrigeration to the LNG process shown in Figure 5 [1,2]. Propane refrigerant is used to cool the natural gas feed and to condense the primary multi-component refrigerant (MCR™)[1]. Refrigeration is provided at three pressure (temperature) levels. The compressor is a single casing machine with two side-streams. Design power consumption of the steam turbine driven compressor is more than 35,000 BHP at a speed of about 3000 RPM.

The propane from the compressor discharge is cooled, then condensed against water. An inventory of liquid propane is held in an accumulator. The liquid propane flows from the accumulator to shell and tube heat exchangers at three propane pressure levels. The propane evaporated in these exchangers then flows back to the compressor.

[1]MCR is the trade-mark for Air Products and Chemicals Incorporated's patented multi-component refrigerant natural gas liquefaction process.

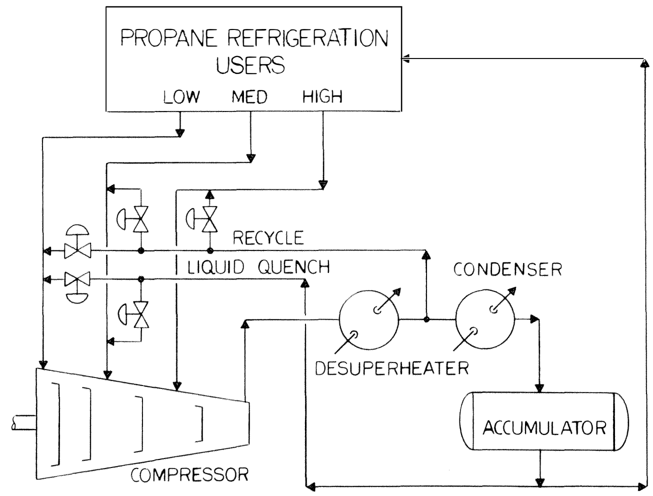

Figure 4. Propane Refrigeration Compressor System.

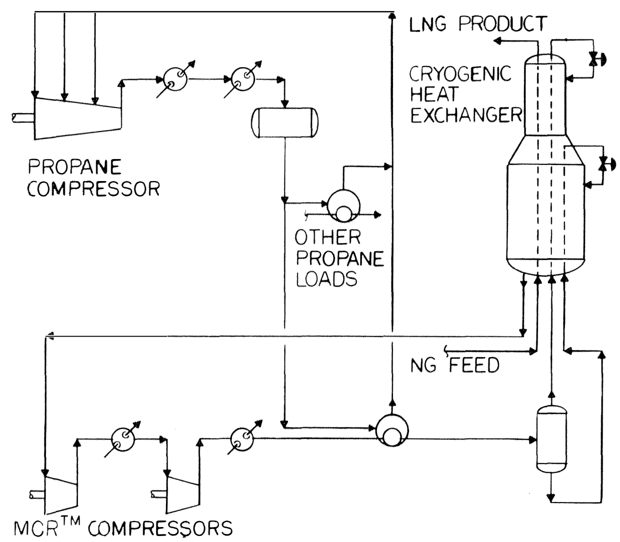

Figure 5. Propane Pre-Cooled Multi-Component Refrigerant (MCR™) Natural Gas Liquefaction Process. (Patented Cycle Courtesy of Air Products and Chemicals Inc).

Compressor surge protection is provided by recycle of warm gas from the desuperheater exit to the suction drum of each section. Side-stream temperatures are maintained during recycle operation by injection of liquid propane from the accumulator into the first and second section suction drums.

This compressor system combines a compressor in which the dynamic response of one section is directly affected by the responses of the other sections with a process which has very large volumes and a large liquid inventory. The system responds slowly to control actions because of the large system capacitance. There was concern that process upsets or equipment failures might impose conditions in which surge could not be prevented. This concern was supported by control system difficulties encountered during a simulation study of a similar system [3,4]. Therefore, a simulation study was proposed and approved.

The study was performed in two phases during the detailed engineering of the unit. The initial phase resulted in an evaluation of the overall performance of the compressor and its control system when responding to process upsets and equipment failures. The first section recycle valve was found to be too small and a larger size was tested and found to be adequate. A potential problem with an interlock to open all recycle valves simultaneously upon a plant shutdown was identified, but could not be investigated further due to limitations in the mathematical model and computer implementation. It was also found that all three sections of the compressor entered surge almost immediately after a turbine trip. This also could not be investigated further because of the simulation program limitations.

A second phase study was undertaken to investigate the problems found by the first phase study and also to determine what shaft torques could be expected during start-up of the compressor and to evaluate the sensitivity of the control system to the rise-to-surge characteristics of each section of the compressor.

The objectives of the second phase study required a simulation model capable of accurately representing the system performance over an operating range from zero speed to full speed, full load operation. The mathematical model was revised to allow for this operating range and reprogrammed to overcome the computational problems experienced during the first phase. The resulting simulation model satisfied all performance requirements.

The principle results of the study were:

- The satisfactory performance of the anti-surge control system was demonstrated prior to installation. Initial controller set points, gain and reset settings were determined.
- Operating limitations were identified. For example, it was found that the system could not be operated at 100 percent speed, full recycle (no load) and no liquid quench flow without exceeding the temperature limit of the desuperheater.
- The interlock circuits to open the recycle valves upon plant shutdown were removed.
- Many control system alternatives were tested in an attempt to prevent surge during trip. None of the alternatives successfully prevented surge in the first or second sections. It was found that a faster stroke time for the third section recycle valve could prevent surge in the third section only.
- The sensitivity of the anti-surge control system to the rise-to-surge of each section was evaluated. Satisfactory anti-surge control was found for values equal to the manufacturer's guarantee.
- After the ASME test loop curves were available, the simulation was retested and design recommendations concerning acceptable rise-to-surge were confirmed.
- The compressor start-up procedure was tested. It was found that the torques experienced during start-up would not be a problem. A need to introduce liquid quench before reaching 100 percent speed was demonstrated. The system was also found to be sensitive to the sequence of recycle valve closure during start-up.
- The simulation led to an excellent understanding of the overall static and dynamic behavior of the system.

MULTI-COMPONENT REFRIGERATION SYSTEM (MCR™)

The system circulates the primary refrigerant for the LNG process in Figure 5. The compressor is schematically shown in Figure 6. The refrigerant is compressed in a two-casing compressor with intercooling between casings. Each casing has individual steam turbine drivers and absorbs over 35,000 BHP at speeds of approximately 4,600 RPM. Compressor antisurge protection is by flow - ΔP control using individual casing recycles.

The compressed refrigerant is cooled by water and then condensed against liquid propane. The condensed stream then flows into a cryogenic heat exchanger where the natural gas feed is condensed into LNG by vaporization of the refrigerant. The vaporized refrigerant is then returned to the compressor suction.

In this simulation, it was proposed that the compressor system performance and the interaction of the compressor controls with the controls for the cryogenic heat exchanger be studied. Some of the specific objectives of the study were:

- Evaluate the performance of the compressor anti-surge controls.
- Confirm the sizing of the recycle control valves and the scaling of the surge control system computing elements.
- Test the compressor start-up procedure and determine the maximum expected shaft torques during start-up.
- Determine maximum cooling water temperatures during transients and during operation in total recycle.
- Determine if relief valves located at the discharge cooler would open during operation in total recycle.
- Determine if the compressor and cryogenic heat exchanger controls would interact.

All of the intended objectives were achieved. It was found that the discharge relief valves would open during the transient between zero recycle and total recycle upon a sudden load loss; but, after some venting, the valves would reseat. The loss of inventory under these conditions was predicted. It was also found that the cryogenic heat exchanger controls had to be tuned for slow response in order to minimize compressor/exchanger interaction. It was also found that when production was reduced enough to cause compressor recycle, the cryogenic exchanger controls were ineffective, and manual exchanger control was then necessary.

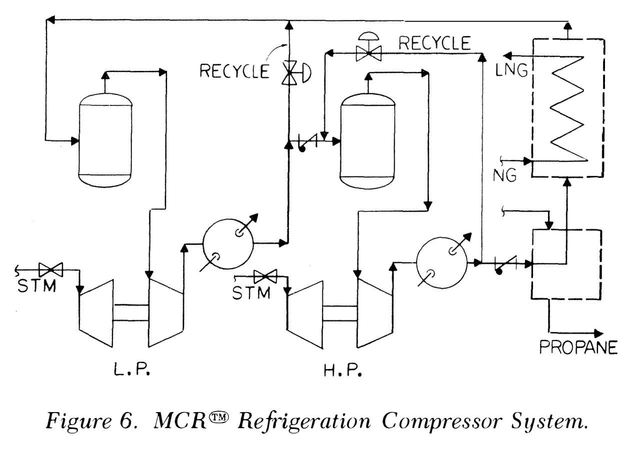

Figure 6. MCR™ Refrigeration Compressor System.

GAS INJECTION SYSTEM

The gas injection system shown in Figure 7 is part of a gas field development in which natural gas is drawn from various wells, natural gasoline and other condensates stripped off, and the resulting "dry" gas piped to a LNG plant or injected back into the field.

This compressor consists of two casings with four sections of compression, each with an inter-stage air-cooler and individual recycles. The compressor is operated at about 10,000 RPM and driven by a gas turbine with an ISO rating of 30,000 BHP. The suction pressure is approximately 1000 psig and the discharge pressure is about 7000 psig. The injection flow rate is controlled by a pressure controller which maintains a constant suction pressure by manipulating speed, then opening the first section recycle valve if speed control alone is insufficient. If enough gas is available, the compressor will operate at maximum speed.

When simulation of this system was being considered, the questions to be answered were:

- What degree of dynamic interaction among each section's anti-surge controls would occur?
- Were proposed process bypasses around the entire compressor needed? Would the presence of the bypasses result in unacceptable interactions with other controls?
- Were the control valves properly sized and were stroke times fast enough?
- Could the compressor operate well over wide flow ranges?

The simulation resulted in answers to all of these questions. A significant interaction among the suction pressure controller, speed and the first section recycle valve was identified. This problem was quickly resolved by the use of the simulation, but had the system been installed as initially intended, it is possible that several trips or at least severe surges would have occurred before the problem would have been diagnosed and corrected in the field.

The simulation also identified a potential for overspeed upon trip due to the location of the inventory vents located on each section's discharge piping which opened simultaneously with trip. These vents would quickly depressurize each section's discharge and the compressor would act as a turbine, thus tending to accelerate the rotor.

The simulation report recommended that additional testing be done to analyze the need for mounting the inventory vents at the discharge. It was felt that suction mounting would be acceptable, provided that surge would not occur during rundown after trip. Additional tests were made and a satisfactory design was obtained with suction mounting.

Also, as a result of the experience gained from testing the system, several other questions regarding timing of certain actions, the method of bringing the compressor on line during start-up and the need for the inter-stage check valve arose. Simulation tests were made to answer all of these questions.

The impellers in each casing have back-to-back orientations, i.e., the impellers in one section have an orientation opposite to that of the other section. This arrangement imposes limits on the relative ΔP's across each section. To assure reliable operation, a study of compressor thrust bearing loading was made for such conditions as start-up, trip, sudden recycle, venting, etc. Review of the simulation data showed that in most cases, the ΔP limits were not exceeded. For the cases in which the ΔP limits were exceeded, control system modifications which would prevent transgression of the limits were recommended and implemented.

The dynamic simulation played a significant role in the design of these compressor systems. The specifications of critical equipment were confirmed, or where an inadequacy was found, the simulation was used as a design tool to determine what the proper design should be. In some cases, proposed design changes were demonstrated to be unacceptable.

PERFORMANCE OF SIMULATION STUDIES

In the previous sections of this paper, compressor system simulation was discussed in terms of the application of simulation models to system design. This portion of the paper is focused upon the development of those simulation models. Discussed are the major steps involved in performing a simulation study and an overview of mathematical models and computer implementation.

SIMULATION PROCEDURES

Figure 8 illustrates the flow of a typical simulation study.

The initial and most crucial step in the study is the definition of the system. This step involves not only the identification of the physical boundaries of the system, but also the definition

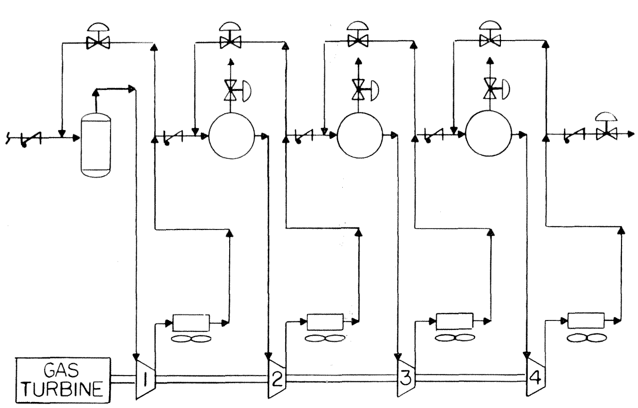

Figure 7. Gas Injection Compressor.

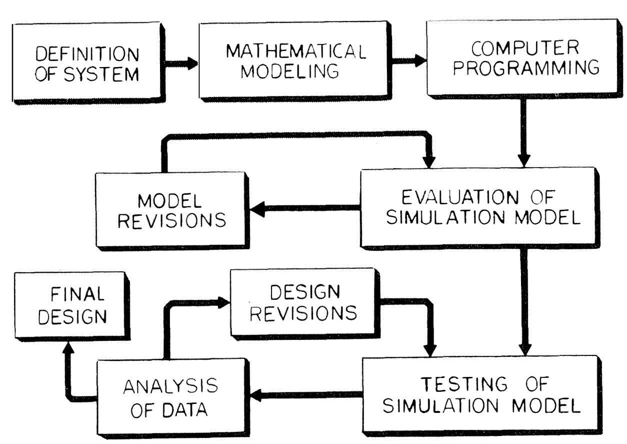

Figure 8. Procedure for Performance of Dynamic Simulations.

of the variables and types of behavior which the mathematical model must represent.

The model of a compressor system can be very complex, in which dynamic variations in pressures, temperatures, and molecular weights are represented. Complex models are not always necessary. For example, if it is known that temperatures will remain essentially constant, the mathematical model can be much simpler.

The more rigorous a model is, the larger (in terms of number of equations) it becomes, and is, therefore, more expensive to program and run on a computer. Because cost and complexity are related, it is important that a model be as simple as possible, yet be detailed enough to meet the objectives of a study. Therefore, it is is extremely important to define what information is expected from the simulation before modelling and programming begin. If this is not done, the result may be a model which cannot adequately describe the system.

After the boundaries of the system and the performance criteria for the simulation study have been established, the mathematical model is written. The mathematical model is then programmed on a computer. The basic forms of mathematical models and the types of computers used in simulation are discussed later.

Once the simulation model has been programmed and the majority of programming errors eliminated, a period of model evaluation follows. This involves making a series of computer runs to observe the dynamic behavior of the model and to determine if the behavior is realistic. If the model is not yet adequate, revisions are made and the model is retested until all parties agree that the model is accurately representing the system.

The system performance tests are made after the model evaluation phase is completed and all program modifications have been made. This testing consists of a series of preplanned upsets. The simulation data is analyzed and, if system performance is not satisfactory, design revisions are made and tested until an acceptable design is obtained.

In practice, some design deficiencies are often identified during the model evaluation phase. The primary purpose of the evaluation phase is to identify and correct modelling and programming errors, but also the majority of test runs occur during the evaluation phase so design problems do appear. Often over 100 runs (many invalid due to programming errors) may be made during model evaluations and only 20 or 30 runs made during performance testing.

MATHEMATICAL MODELS

A simulation model represents the steady-state and dynamic behavior of a system as a set of simultaneous algebraic and differential equations. Figure 9 illustrates how the mathematical model for a typical compressor study is organized.

The model for a compressor system can be considered to consists of three separate modules between which information is passed. These modules are:

- The process module
- The compressor module
- The control system module.

PROCESS MODULE

The process module consists of algebraic and differential equations describing the accumulation of mass and energy at various places in the system such as vessels or piping junctions (nodes), the transfer of mass and energy between the nodes, and thermodynamic characteristics such as enthalpies or phase

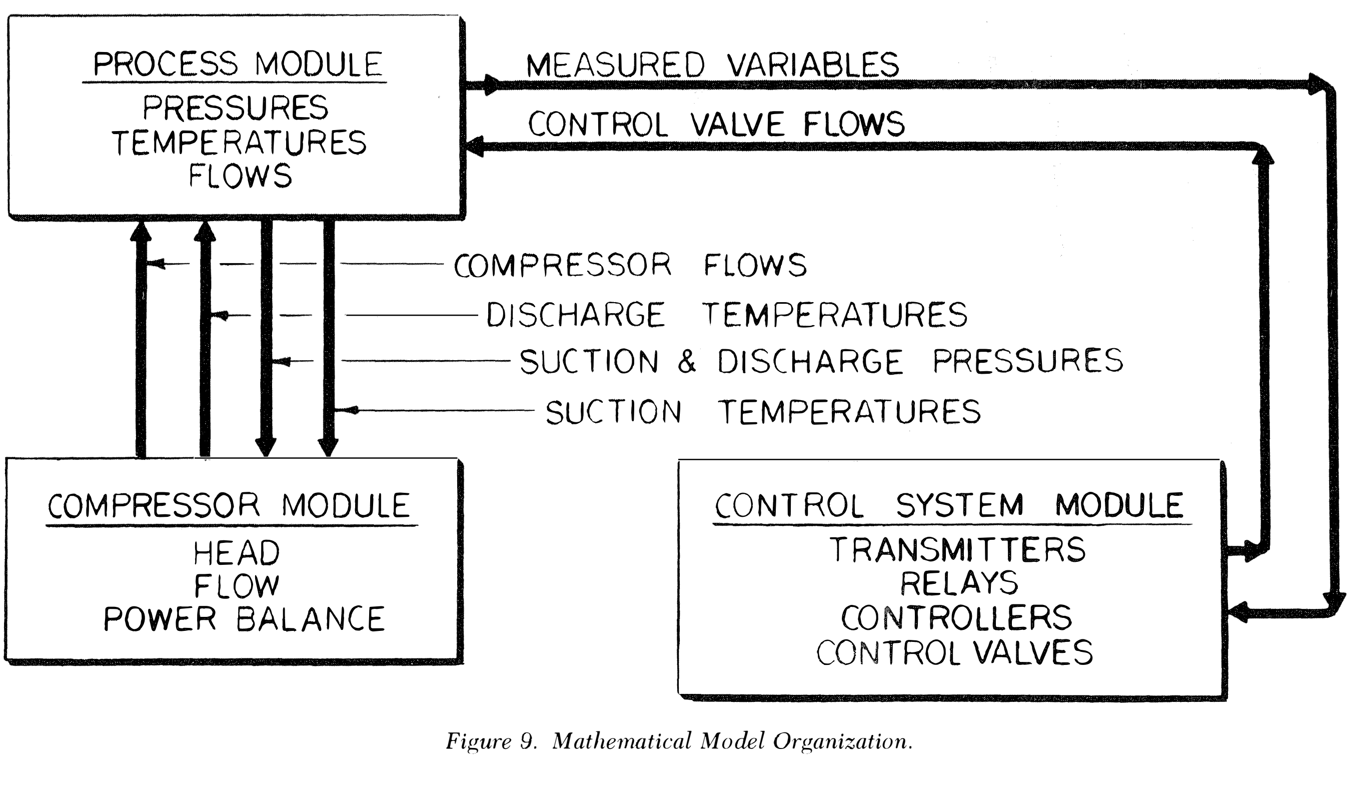

Figure 9. Mathematical Model Organization.

equilibria. The process model can be relatively simple such as the model for the injection system in which variations in temperature or molecular weight were not significant so only mass balances needed to be considered; or may be highly complex, such as the refrigeration system models in which heat transfer, phase equilibria and temperature variations had to be accounted for in addition to mass balances.

COMPRESSOR MODULE

The compressor module consists of the empirical head/flow characteristics of each section as provided by the manufacturer, thermodynamic compression equations, and a torque balance between the compressor and its driver. Figure 10 illustrates the information flow in a typical compressor model.

The compressor module receives suction pressure and temperature, and discharge pressure from the process module. The required compressor head is calculated by:

$$H = \frac{Z \times R \times T_s \times n \times \eta}{\overline{MW} \times (n-1)} \times \left[\left(\frac{P_D}{P_S}\right)^{\frac{n-1}{n\eta}} - 1\right] \quad (1)$$

Where:

- H = Polytropic head, ft - lb_f/ lb_m
- $\overline{MW}$ = Molecular weight, lbm/ lb-mole
- P_D = Discharge pressure, psia
- P_S = Suction pressure, psia
- R = Gas constant, 1,545 (ft - lb_f)/(lb - mole)-(°R)
- T_s = Suction temperature, °R
- Z = Average compressibility, dimensionless
- n = Polytropic exponent, dimensionless
- η = Polytropic efficiency, dimensionless

The volumetric flow is then determined from the manufacturer's characteristic curves as a function of speed and head:

$$Q = f\,(H, N) \quad (2)$$

Where:

- Q = Inlet volumetric flow, ACFM
- N = Compressor speed, RPM

The compressor curves can be represented in several ways. For a constant speed compressor, a single head-flow curve is used. This curve can be stored in the computer as a table of flow and head values [3], or fit to a polynomial function.

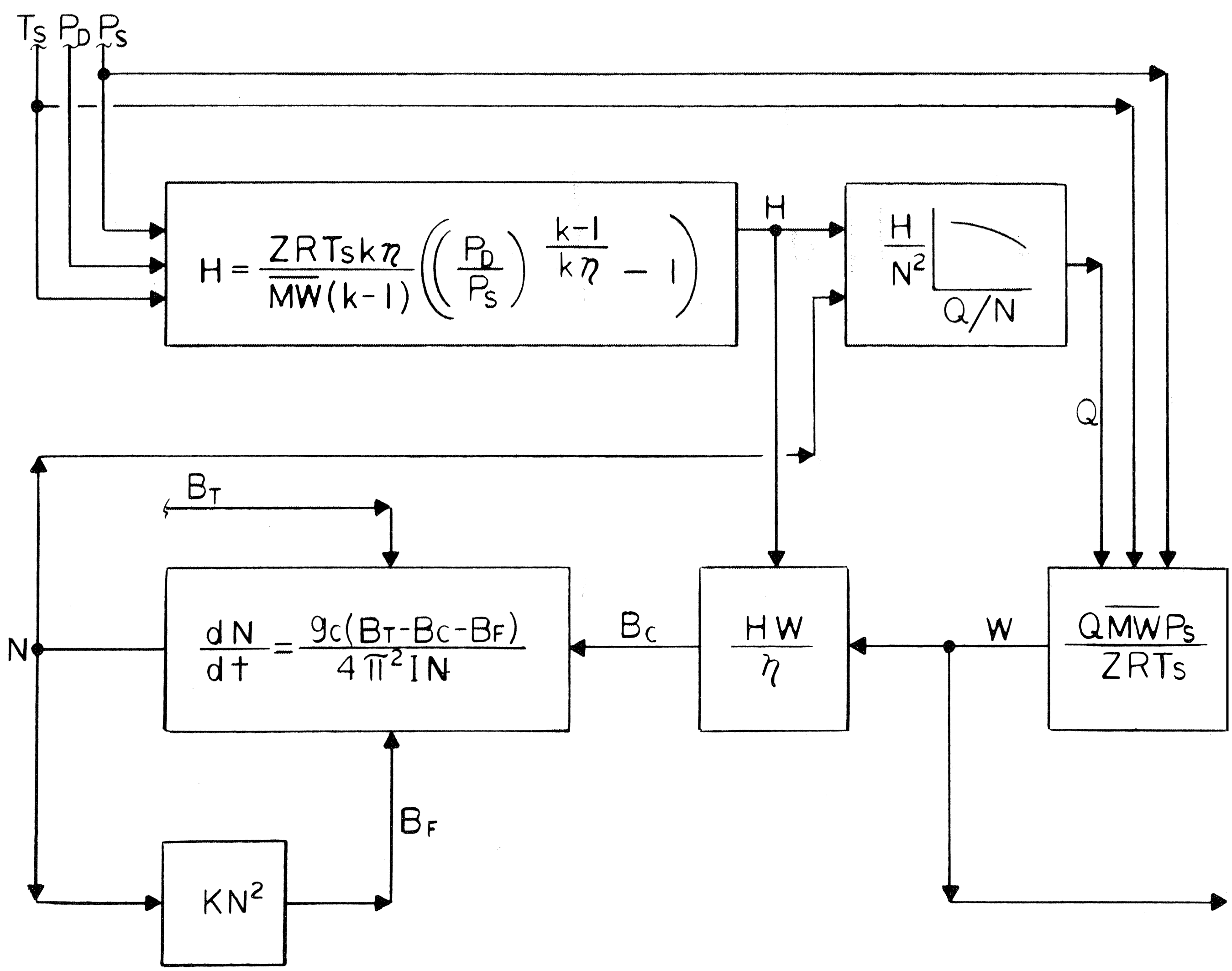

Figure 10. Compressor Model Block Diagram.

For a variable speed compressor, a series of curves, each for a given speed, can be stored in tabular form [3], or a single speed curve can be used and variation in speed accounted for by the use of fan laws [5].

The head, flow, speed, efficiency performance map of a compressor's performance has also been fitted to polynomial equations using head and flow coefficients [3].

The mass flow through the compressor is calculated from the suction conditions, volumetric flow and a gas law equation of state:

$$W = \frac{Q \times \overline{MW} \times P_s}{Z \times R/144 \times T_s} \tag{3}$$

Where:

W = mass flow, lb_m/min

Z = compressibility factor, dimensionless

For variable speed machines, the compressor speed is calculated by a differential torque balance

$$\frac{dN}{dt} = \frac{3600 \times g_c \times (B_t - B_c - B_f)}{4 \times \pi^2 \times I \times N} \tag{4}$$

Where:

B_t = Power supplied by driver, ft - lb_f/min

B_c = Power consumed by compressor, ft - lb_f/min

B_f = Power lost by friction, ft - lb_f/min

I = Axial moment of inertia, lb_m - ft^2

g_c = Constant 32.17 lb_m - $ft/(lb_f$ - $sec^2)$

The power supplied by the driver can come from a simplified governor model. The governor model typically is a proportional plus integral controller whose set point and input are speed and output is power [3]. If necessary, a more detailed driver model can be used [5].

The power consumed by the compressor is calculated from head, mass flow and efficiency

$$B_c = \frac{H \times W}{\eta} \tag{5}$$

The frictional loss is calculated by assuming the losses are proportional to speed squared:

$$B_f = K \times N^2 \tag{6}$$

Where K is a constant.

The shaft torque can be calculated from either supplied or consumed power

$$\tau_c = \frac{B_c + B_f}{N} \tag{7a}$$

Where τ_c = torque imposed on shaft by compressor, ft - lb_f

$$\tau_c = \frac{B_t}{N} \tag{7b}$$

Where τ_t = torque imposed on shaft by driver, ft - lb_f

Note that τ_c and τ_t have opposite signs and that $|\tau_c| = |\tau_t|$ only in the steady state.

If $|\tau_t| > |\tau_c|$ the compressor accelerates. If $|\tau_t| < |\tau_c|$ the compressor decelerates.

The compressor model may be a simple representation of a constant speed machine or a very complex one in which variation in speed, efficiency and polytropic coefficients are accounted for. The degree of sophistication required is a function of both the type of system under study and the objectives of the study. For example, a model to study start-up would have to be more complex than a model used to study constant speed operation.

CONTROL SYSTEM MODULE

The control system module represents the behavior of the process control instrumentation and final control elements. Controllers, transmitters, computing relays and control valves are usually modelled mathematically. In hybrid or analog computer simulations, the controllers used sometimes are actual industrial electronic controllers rather than mathematical models [3,4]. Control valve dynamics, essentially the valve stroking period, are modelled, as are measurement lags when they are significant.

COMPUTER IMPLEMENTATION

Because of the extremely large amounts of computation involved, dynamic simulation did not become practical until high speed computers were developed. Most early simulation was performed using analog computers [6]. As digital computer speeds increased, digital solutions of simulation models became practical and several simulation oriented languages were developed (i.e., CSMP and MIMIC). A third type of computer, the hybrid computer, couples a digital computer and an analog computer together. An interface module containing analog to digital and digital to analog converters and control logic enables the two computers to communicate.

A discussion of the relative merits of analog, digital and hybrid computers is presented by Smith & Cadman [7]. The use of digital computers and CSMP in performing dynamic simulations is discussed by reference [8].

The simulations discussed in this paper were performed on a hybrid computer. This computer system is shown in Figure 11.

The computer consists of three large analog computers, the Pacer 681's, tied to one digital mini-computer, the Pacer 100. The analog computers each have 120 operational amplifiers, or enough capacity to solve a mathematical model consisting of around 30 first order differential equations. The digital computer has 32,000 words of memory with access to disc mass storage and input/output devices of card reader, lineprinter, and a CRT teletype.

In addition to the hybrid equipment, there are 20 industrial electronic controllers and an older generation TR-48 analog computer. The TR-48 is used primarily as an interface to the controllers because the controllers accept and output 1-5V signals while the analog computers operate on 0-10V.

The electronic controllers provide a very powerful point of interaction with the computer programs. In effect, the behavior of a compressor being simulated can be monitored in the laboratory on these instruments in precisely the same manner as it would be from a control panel in the field. It is this type of interaction between the simulation and the engineers designing a compressor system that has made hybrid simulation much more attractive than digital simulation for this application.

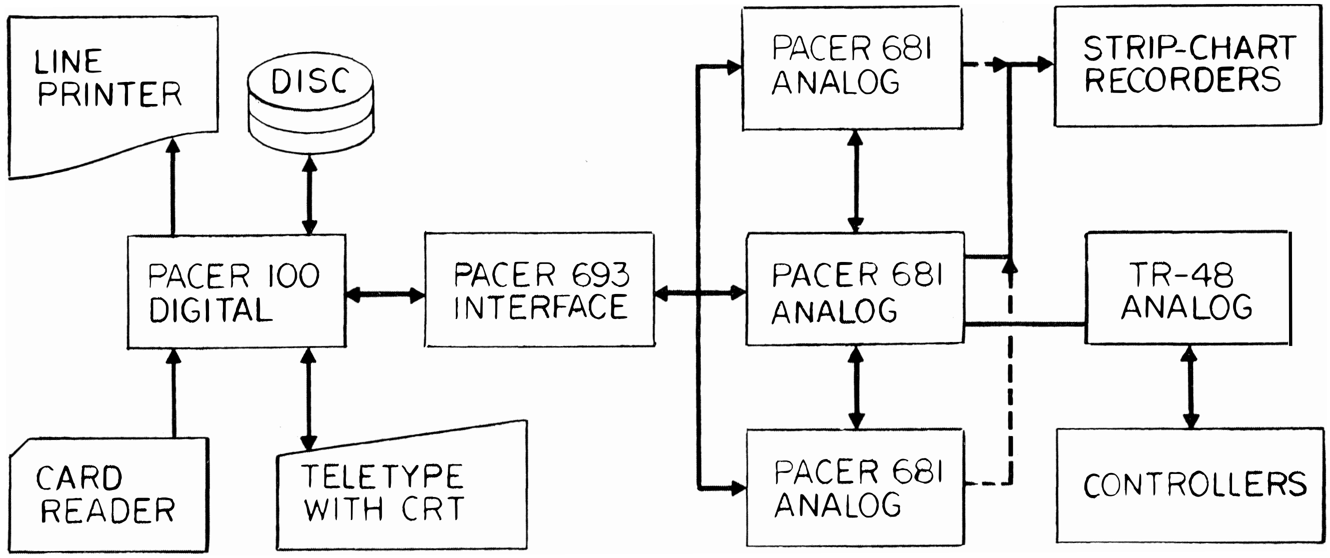

Figure 11. Hybrid Computer System.

The choice of a computer depends on the size of the mathematical model, the number of runs to be made and the degree of engineer/solution interaction required. Small problems, or problems for which only a few runs are to be made probably are more economically done by digital solution. For large problems with many runs required, or problems which require a high degree of engineer/solution interaction hybrid computation is often more suitable.

CONCLUSIONS

Dynamic simulation has been found to be a highly effective design tool for use in the design of complex centrifugal compressor systems. This tool has been used to test the operability of control systems, evaluate operating procedures, check equipment arrangement and provide acceptance criteria for ASME performance test compressor characteristics. The hybrid computer implementation of the simulation models was very effective in promoting engineer/simulation model interaction. This resulted in the use of the simulation models as an integral part of the design process.

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