

THE ROCKPORT PLANT - ANALYSIS OF TEMPORARY FAST TURBINE VALVING TESTS

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Abstract - A series of temporary fast turbine valving (TFTV) field tests were carried out on AEP's Rockport 1300 MW unit. These four tests at various unit loading levels verified the functional aspects of the TFTV scheme -- including the control and intercept valve stroke characteristics and the rapid reduction of mechanical driving power. This paper presents an analysis of the mechanical and electrical performance characteristics of the Rockport unit during TFTV. A comparison of the test measurements with digital simulation results is also provided to validate the simulation tools and models commonly used in system dynamics studies.

1. INTRODUCTION

1.1 Description of the Rockport Plant

The Rockport Plant, located in southern Indiana, will consist of two 1300 MW coal-fired generating units when completed in 1988, and will be connected to the AEP System via two 765 kV lines -- the Rockport-Jefferson line (111 miles) and the Rockport-Sullivan (Breed) line (97 miles). With only two 765 kV outlets integrating Rockport into the AEP System, a number of techniques new to the AEP System are being employed to assure reliable operation. The most notable of these are single-phase switching (SPS)[1,2] and temporary fast turbine valving (TFTV)[3]. The combined application of single-phase switching and temporary fast turbine valving provides the additional transient stability margin required to operate the Rockport Plant at 2600 MW on a firm basis.

Unit No. 1 was first synchronized with the system in September 1984. After a three-month test period, the unit was declared commercial on December 10, 1984. During this period, the TFTV field tests described in this paper were carried out.

1.2 Application of TFTV at Rockport

The application of TFTV at Rockport was based on extensive studies of transient stability performance under various plant/system operating conditions. A companion paper describes the Rockport Plant transient stability performance and other system planning considerations in greater detail[4].

As an aid in understanding the Rockport TFTV scheme, the steam flow path for a typical coal-fired

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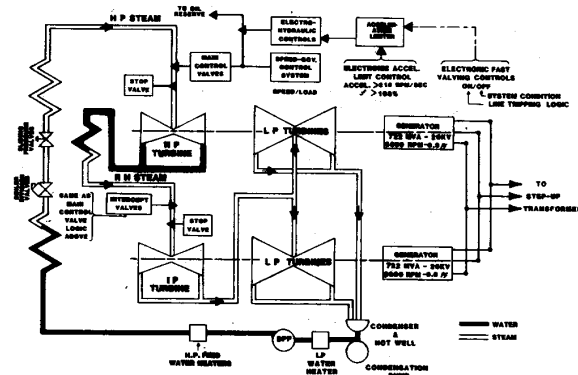


Fig. 1. Steam flow path and speed control for a typical BBC 1300 MW unit.

1300 MW, Brown Boveri, cross-compound turbo-generator, such as those installed at Rockport is illustrated in Figure 1. The steam from the boiler enters the high pressure (HP) turbine through the main control and stop valves. From the HP turbine the steam is exhausted to the reheater and enters the intermediate pressure (IP) turbine through the reheat intercept and stop valves. The exhaust steam from the IP turbine enters the low pressure (LP) turbine sections of each shaft before returning to the condenser for recycling.

In the temporary fast turbine valving scheme adopted for Rockport, both the control and intercept valves are used, with automatic reopening of the valves to the pre-fault position within 8 seconds after fast valving initiation. This quick reopening of the valves minimizes mechanical stresses and the possibility of lifting safety valves or causing turbine fatigue.

In developing the valving characteristic for implementation at Rockport, system performance (transient stability) and equipment (boiler and turbine) considerations were carefully balanced to achieve optimum benefits. Based on these considerations, it was determined that: 1) control and intercept valves should close as rapidly as possible; 2) control valves should close to 40% of full stroke and intercept valves to 26% of full stroke; 3) time delay before reopening valves should be minimized (0.3 seconds or less); and 4) accumulators are not required for the control valves to reopen to 100% position in approximately 6 seconds and intercept valves in approximately 7.5 seconds.

At Rockport, TFTV is initiated by a special electrical relay logic package designed by AEP. In developing the specifications for this package, speed and selectivity were important considerations. Speed was achieved by using the 765 kV line relays for the initiation signal. Selectivity was achieved by

including two pre-initiation signals to: 1) define system status (operator initiated) and 2) determine unit loading (from unit metering).

1.3 Overview of TFTV Field Test Program

In order to provide a complete functional check of the fast valving scheme implemented on Rockport Unit No. 1 and to verify the analytical studies, a series of four tests were carried out on November 3, 1984. These tests were performed with the unit on line and operating at various loading levels. All four fast valving operations were successful.

The primary objectives in carrying out the fast turbine valving tests were: 1) to verify the functional characteristics of the control and intercept valves in relation to their design characteristics; 2) to verify and examine the mechanical and electrical response characteristics of the boiler, turbine-generator and transmission system; and 3) to obtain sufficient information about the turbine-generator and its control system response for the development of enhanced computer simulation models.

The tests involved four separate temporary valving operations. Three of these tests were carried out with the unit operating in the sliding pressure mode at loading levels of 60%, 80% and 100%. The fourth test was performed at a unit loading of 80% with the unit operating in the constant pressure mode.

In the constant pressure mode, turbine power is controlled by throttling the control valves, which results in reduced efficiency at partial load. In sliding pressure operation of AEP's 1300 MW units, the control valves are fully open over the entire load range while the sliding pressure valves (See Figure 1) are throttled to adjust load. The net result is improved turbine efficiency at partial load[5]. Sliding pressure operation is the preferred mode of operating the Rockport unit.

The following table identifies the specific tests and the sequence in which the tests were carried out:

Table I: TFTV Field Test Sequence

| Field Test | Approximate Gross Output (MW) | Operating Mode |
|------------|-------------------------------|-------------------|
| FV1 | 820 | Sliding Pressure |
| FV3 | 1125 | Sliding Pressure |
| FV8 | 1125 | Constant Pressure |
| FV5 | 1350 | Sliding Pressure |

The four tests were run in this sequence of increasing load level in order to assure that the unit was not unnecessarily stressed as the severity of the TFTV duty was increased.

Since no electrical faults were applied during these tests, all four tests were performed by manually initiating the fast turbine valving signal in the control room. Prior to initial operation of Unit No. 2, additional TFTV tests are planned for Unit No. 1 in conjunction with 765 kV fault initiation and single-phase switching.

1.4 Instrumentation and Data Recorded

During the tests, an extensive amount of data was recorded using visicorders, tape decks, the plant digital computer and analog recorders, plant control room charts, and manual logs. Table II lists key data recorded on the two tape decks. Other important quantities captured on various high-speed recording devices included position of the other three control and three intercept valves. In addition, a number of pressure and temperature data points were captured using the plant's computerized data acquisition system. In order to synchronize all recorded data, a timing/event signal was generated with the initiation of fast valving and simultaneously recorded on the tape decks and all recording devices.

Table II: Data Recorded on Tape Decks

HP and RH Units

Gross Power
Field Current
Field Voltage
Speed (Front & Rear Std.)
Terminal Voltage (3 phases)
Stator Current (3 phases)
Control Valve A (one of four)
Intercept Valve 1 (one of four)

Other

765 kV Bus Voltage (one phase)
138 kV Reserve Bus Voltage (one phase)
FV Initiate Signal

1.5 Objectives of TFTV Field Test Data Analysis

The main objectives of the data analysis reported in this paper can be described as follows:

- (1) To evaluate the electrical and mechanical response characteristics of the Rockport HP and RH machines during TFTV.
- (2) To compare the electromechanical response of the Rockport unit observed during the TFTV field tests with simulation results in order to validate the simulation tools and enhance the simulation models of the turbine-generator and its control system.

Section 2 of the paper presents an analysis of the control and intercept valve characteristics during TFTV. The calculated mechanical power response of the Rockport unit during the four tests is presented in Section 3. Section 4 describes the electromechanical performance of the Rockport unit and its comparison with the digital simulation results. The electrical response of the unit and the power system to TFTV is presented in Section 5. Conclusions and a discussion of future work appear in Section 6. Although all four TFTV tests were carefully analyzed, the paper focuses on the results of test FV5. Being a full load test, FV5 produced the most meaningful and interesting results. Selected results of tests FV3 and FV8 are also presented to compare machine dynamic performance during sliding and constant pressure operation.

2. CONTROL AND INTERCEPT VALVE CHARACTERISTICS

During TFTV the steam flow of Rockport Unit No. 1 is controlled by the four main control and four intercept valves. The stroke characteristics (valve position vs. time) of these valves were recorded using magnetic tape decks and visicorders.

2.1 Summary of Field Test Measurements

Table III presents key valve stroke parameters derived from the test data. These parameter values represent average characteristics of the four control (CV) and four intercept (IV) valves during each test.

Table III: Valve Characteristics from Field Tests

| Field Test | Initial Time Delay* (ms) | | Closing Rate (%/s) | | Minimum Position (%) | | Opening Rate** (%/s) | |
|------------|--------------------------|-----|--------------------|-----|----------------------|----|----------------------|----|
| | CV | IV | CV | IV | CV | IV | CV | IV |
| FV1 | 88 | 108 | 568 | 216 | 32 | 22 | 16 | 8 |
| FV3 | 88 | 108 | 586 | 211 | 34 | 23 | 15 | 8 |
| FV8 | 75 | 113 | N/A | 211 | 38 | 23 | 11 | 8 |
| FV5 | 85 | 114 | 489 | 211 | 34 | 24 | 15 | 8 |

*Time between TFTV initiate signal and beginning of valve closure.

**The valves started to reopen almost immediately after reaching minimum position.

The control valve motion during the three sliding pressure tests (FV1, FV3 and FV5) was reasonably consistent. The initial time delays experienced by the control valves were almost equal and the closing rates were similar, although the rate was lower during the full load test FV5. During these three tests, the control valve minimum positions were within 2% of one another and these valves reopened at practically identical rates. Similarly, the behavior of the intercept valves was quite consistent during all four field tests.

Table III also indicates that during the constant pressure test FV8, the control valve movements were different from those during the sliding pressure tests. Further details of the control valve characteristics during test FV8 are provided in Section 2.2.

The average valve stroke parameters from Table III are included in Figure 2, which illustrates the valve stroke timing sequence during a typical TFTV operation following a system fault. The 765-kV line relay pick-up time is expected to be approximately 1 cycle (15 ms). The 30 ms fast valving relay pick-up time was determined from pre-test checkouts. Consequently, a signal to initiate valve closure would be received at the acceleration limiter approximately 45 ms after a critical fault condition occurs.

2.2 Comparison with Design Characteristics

By measuring the valve strokes during the field tests, the actual performance of the control and intercept valves can be compared with the manufacturer-provided design characteristics.

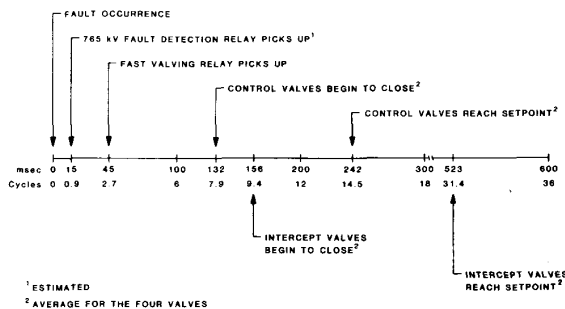


Fig. 2. Expected timing sequence during a typical TFTV operation at Rockport.

Figure 3 illustrates the motion of Control Valve "A" during test FV5. This motion is representative of the performance of all four control valves during the three sliding pressure tests. Also shown in Figure 3 is the design characteristic.

Following initiation of the TFTV signal, measured values indicate approximately twice the delay before initial valve motion as compared to the design parameter. On the other hand, the average closing rate of the control valves was faster than anticipated from design curves (548 vs. 375%/s).

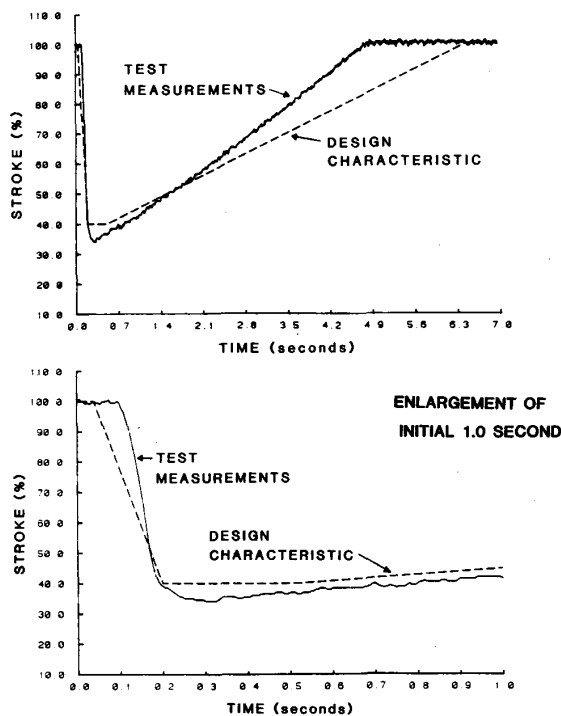


Fig. 3. Typical control valve motion during sliding pressure TFTV tests: Comparison between test FV5 and design characteristic.

The control valves overshoot their minimum design set point of 40%. As indicated in Table I, the average minimum position of the control valves was in the range of 32 to 38% during the four tests.

Although the initial delay in the actual response of the control valves was greater than anticipated, the valves reached their minimum expected set points of 40% at the pre-determined instant due to the faster closing rate. The opening rate of the control valves was faster than expected (15 vs. 10%/s), reaching their fully open positions approximately 1 1/2 seconds earlier than anticipated.

A similar analysis was carried out to examine the intercept valves' performance. Figure 4 provides a comparison of the actual intercept valve motions with the design curves. The delay in the actual initial valve motion was almost 1 1/2 times that indicated by the manufacturer. Also, the average minimum set point attained by the four intercept valves was in the range of 22-24% during the four field tests as compared to the design value of 26%. Finally, the opening rate of the intercept valves was slightly lower than anticipated (8 vs. 10%/s).

Despite these differences, the performance of the control valves during the three sliding pressure tests and of the intercept valves during all four tests was judged to be satisfactory relative to the requirements for successful fast valving.

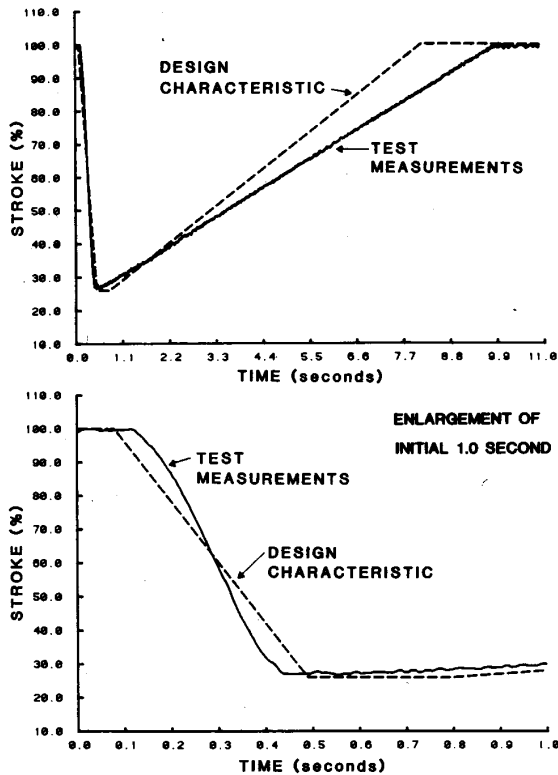


Fig. 4. Typical intercept valve motion during all TFTV tests: Comparison between test FV5 and design characteristic.

During the constant pressure test FV8, it was observed that the control valves' performance was sluggish as compared to their operation during the sliding pressure tests. As illustrated by Figure 5, the original positions of the control valves were approximately 60% during this test. After an initial delay of 75 ms, these valves reached their minimum position of 38% in approximately 0.4 s -- an average closing rate of 55%/s, significantly less than the anticipated 375%/s closing rate. During reopening, the control valves overshoot significantly, reaching as high as the 80% position. Even after 16 seconds following TFTV, the control valves did not return to their original 60% position, but rather drifted around the 65% position.

Further analytical studies will be carried out to investigate the impact on transient stability margins of such sluggish response. If the impact is significant, refinement of control valve response during constant pressure operation may be required. Meanwhile, since the Rockport units will normally be operating in the sliding pressure mode, this deviation is not critical.

3. DETERMINATION OF MECHANICAL POWER RESPONSE CHARACTERISTICS

The movements of the control and intercept valves during TFTV described above produce variations in the mechanical driving power of the Rockport unit. Although the turbine-generator mechanical power is one of the important parameters to monitor, it cannot be measured directly. Therefore, an analytical technique -- based on substituting the measured electrical power and speed deviation values into the "swing equation" -- was developed.

In its most useful form, the swing equation can be written as:

$$2H \frac{d\omega}{dt} = \frac{P_m - P_e}{\omega}$$

where: H = machine inertia constant (kW-s/kVA),
 ω = velocity of the rotational system (p.u.)
 P_m = input mechanical power (p.u.), and
 P_e = output electrical power (p.u.)

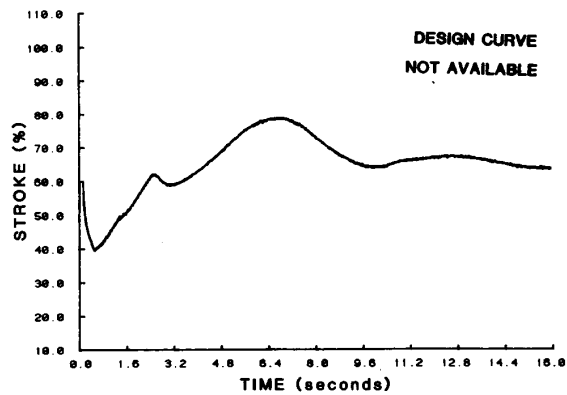


Fig. 5. Control valve motion during constant pressure TFTV test FV8.

With a knowledge of the time varying electrical power output and ensuing speed deviations, changes in the mechanical driving power can be calculated. Using this approach, the input mechanical power during the period of temporary fast turbine valving was derived.

Figure 6 illustrates the calculated time variations in mechanical power of the HP and RH machines as derived from TFTV test FV5. For comparison, the design characteristic supplied by the manufacturer is also shown in this figure. It can be observed that the reduction in the mechanical power of both the HP and RH machines during TFTV is more pronounced than that predicted by the manufacturer. This is because the control and intercept valves attained lower minimum positions than anticipated. From the transient stability performance viewpoint, this larger reduction in mechanical power is beneficial.

In order to examine the impact of unit operation mode (i.e., sliding pressure vs. constant pressure) on the mechanical power response of the Rockport unit, shaft mechanical power responses were derived for tests FV3 and FV8. The results (not shown here) indicate that for the HP machine, the per unit reduction in the mechanical power due to TFTV is larger when the unit is operating in the sliding pressure mode than in the constant pressure mode.

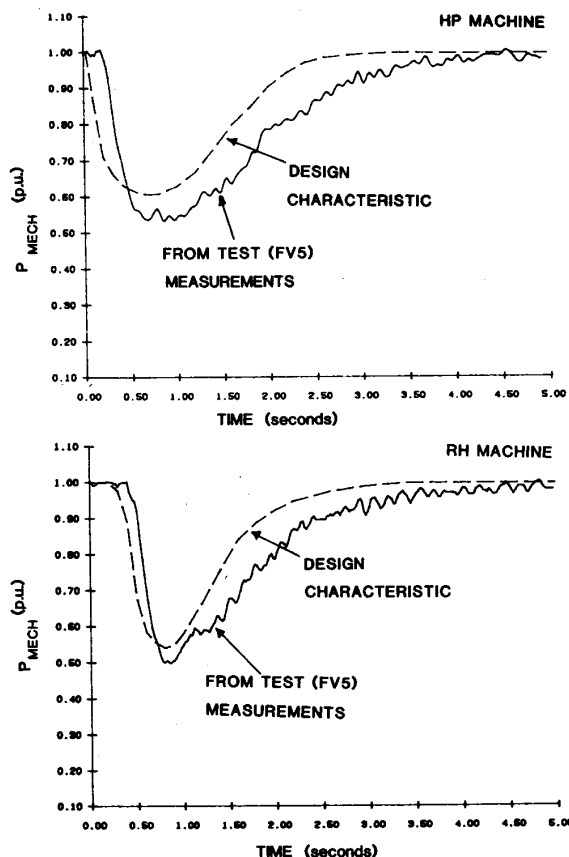


Fig. 6. Shaft mechanical power during test FV5: Comparison between test results and design characteristic.

This is because the control valve movement due to TFTV is larger during the sliding pressure mode (from 100% to 34% for test FV3) than that during the constant pressure mode (from 60% to 38% for test FV8). The results also indicate that the mechanical power response of the RH unit is not affected significantly by the mode of operation. This is because the response of the RH unit is influenced by the intercept valve motion characteristic, which is the same for both sliding pressure and constant pressure operation.

4. ELECTROMECHANICAL PERFORMANCE

The variations in the mechanical driving power of the Rockport unit described above influence its electromechanical performance during TFTV. In this section, the electromechanical performance of the Rockport unit observed during the TFTV field tests is summarized. A comparison of these measured data with the digital simulation results is then made in order to validate the simulation tools commonly used for planning studies.

4.1 Electromechanical Performance of Rockport Unit Observed During TFTV Field Tests

Table IV presents the maximum speed deviations, angular displacements and electric power excursions experienced by the Rockport unit during the four TFTV field tests. Both the HP and RH machines experienced relatively moderate deviations in rotor speed, angle and electrical power output. However, the RH machine experienced slightly larger deviations than the HP machine, because the per unit reduction in the mechanical driving power of the RH machine was larger than that of the HP machine.

Table IV: Electromechanical Performance of Rockport Unit During TFTV

| Field Test | Maximum Speed Deviation (%) | | Max Angle Change* | | Excursion of P_e (MW) | | | | | | | |
|-----------------------------------|-----------------------------|-------|-------------------|-------|-------------------------|------|-----|-------|-----|-----|-----|-----|
| | HP | RH | HP | RH | HP | | RH | | | | | |
| | | | | | Start | Max | Min | Start | Max | Min | | |
| <u>Field Test Measurements</u> | | | | | | | | | | | | |
| FV1 | -0.27 | +0.12 | -0.40 | +0.16 | -14° | -17° | 417 | 417 | 167 | 400 | 413 | 22 |
| FV3 | -0.33 | +0.16 | -0.50 | +0.23 | -15° | -19° | 567 | 567 | 251 | 555 | 595 | 58 |
| FV8 | -0.20 | +0.17 | -0.45 | +0.20 | -12° | -18° | 568 | 568 | 295 | 568 | 578 | 98 |
| FV5 | -0.39 | +0.19 | -0.62 | +0.30 | -16° | -20° | 688 | 688 | 281 | 672 | 722 | 102 |
| <u>Digital Simulation Results</u> | | | | | | | | | | | | |
| FV1 | -0.26 | +0.11 | -0.45 | +0.18 | -15° | -18° | 414 | 414 | 136 | 405 | 433 | 9 |
| FV3 | -0.33 | +0.17 | -0.56 | +0.25 | -19° | -21° | 567 | 567 | 206 | 555 | 596 | 49 |
| FV8 | -0.33 | +0.24 | -0.53 | +0.24 | -21° | -22° | 574 | 574 | 182 | 562 | 576 | 83 |
| FV5 | -0.39 | +0.21 | -0.63 | +0.31 | -20° | -23° | 687 | 687 | 261 | 673 | 726 | 96 |

*With respect to Rockport 765 kV bus.

A comparison of the results of tests FV3 and FV8 provides an indication of the difference in the electromechanical performance of the unit due to the operating mode. The HP unit experienced significantly less deviation in speed, rotor angle and electrical power when operating in the constant pressure mode (FV8) than in the sliding pressure mode (FV3), as a consequence of the difference in the control valve motion during these two operating modes. The larger movement of the control valves in the sliding pressure mode resulted in larger percentage reduction in mechanical power of the HP unit, which eventually translated into a larger reduction in the electrical power output of the HP machine.

4.2 Comparison of Digital Simulation Results with Test Measurements

A primary objective of the field test program was to evaluate the accuracy of various control system simulation models by comparing digital simulation results with the field test measurements. For this purpose, the actual field test conditions were duplicated using the AEP transient stability program. For the four transient stability simulations corresponding to the four TFTV tests, load flow cases were prepared to reflect the network conditions that existed just prior to each of the tests.

In order to simulate the effect of TFTV, a detailed turbine system representation, illustrated in Figure 7, was used. This representation evolved from a basic turbine model provided by the manufacturer. To this basic model, the average control and intercept valve characteristics (Table III) and stroke vs. flow nonlinearities were added. This computer model is a functional representation of the speed governor and TFTV scheme and correlates the per unit values of steam flow and shaft mechanical power.

Table IV compares the digital simulation results with the test measurements for the speed deviation, rotor angle and the electrical power swings of the Rockport unit. In general, the simulation results correlate very well with the measured values for the three sliding pressure tests -- FV1, FV3 and FV5. Minor discrepancies between the simulation results and the measured values can be attributed to limitations in the simulation program and the lack of precise data for representing various system components.

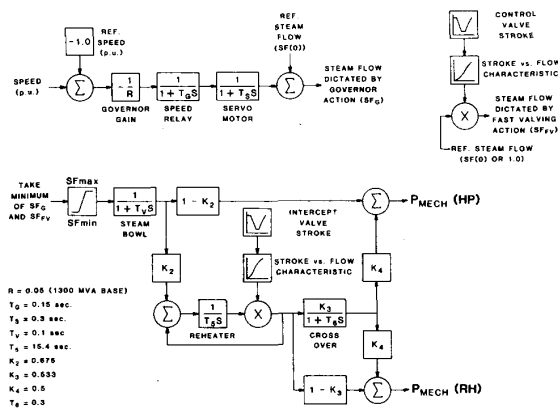


Fig. 7. Rockport turbine system computer model.

It is also apparent from Table IV that for the HP unit, the simulation results do not match with the measured values for the constant pressure test FV8. This indicates that the fast valving model used in the transient stability simulation program is not as accurate in predicting the electromechanical performance of the unit during the constant pressure mode as during the sliding pressure mode. Additional efforts to improve this accuracy are under way. Even with this minor deficiency in the fast valving model, this simulation tool is adequate for most purposes considering that the Rockport units are expected to operate normally in the sliding pressure mode.

Additional insight regarding the difference between the simulated results and measured data can be obtained by comparing waveforms of several electrical and mechanical parameters. Figure 8 compares the simulated electrical power outputs and those measured during field test FV5 for the HP and RH units. The simulated results indicate slightly larger excursions in the power swings and speed deviations (not shown in the figure) as compared with the field measurements. Although, the computer simulations include the net positive sequence damping generated by the interconnected network, any additional damping produced by the rotating elements is ignored. Therefore, the simulated swings are expected to be slightly larger than those measured during the field tests.

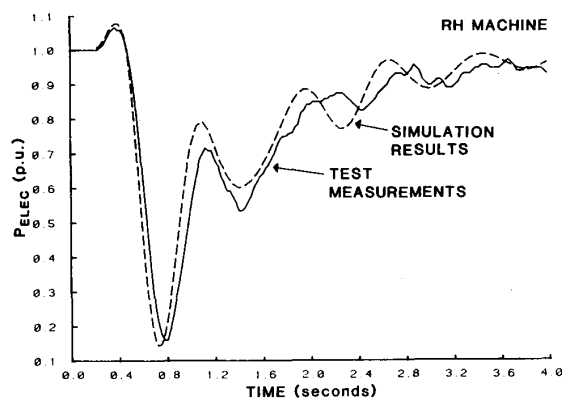
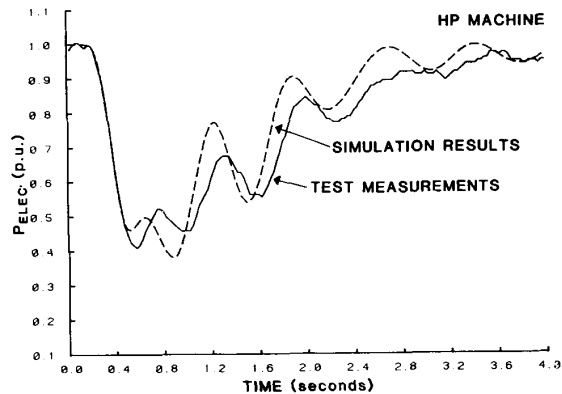


Fig. 8. Electrical power output: Comparison of digital simulation results with test measurements (test FV5).

Figure 9 compares the simulated shaft mechanical power and that derived from the field test measurements. The simulated results show good correlation with the derived mechanical power response.

Figure 10 shows a comparison between the simulated (positive sequence) machine terminal voltage and the positive sequence terminal voltage calculated from test data. The simulated results indicate a prolonged period of increased voltage due to TFTV. In addition, the voltage rise is of a slightly larger magnitude than the measured value. These differences can be attributed primarily to the inability of the excitation system model to faithfully reproduce the actual response of the Rockport exciter. Additional simulation studies are in progress to improve the excitation system model.

In general, the simulation studies have clearly demonstrated the validity of the various system component models. On the basis of comparisons with the field test measurements, it is expected that improved representation of the excitation system and the explicit representation of turbine-generator damping coefficients can provide even more realistic estimates of the expected system performance.

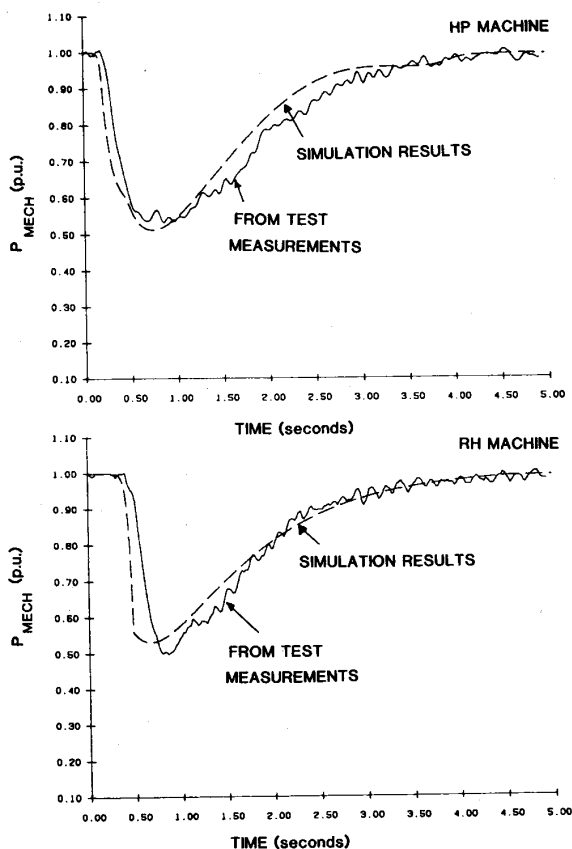


Fig. 9. Shaft mechanical power: Comparison of digital simulation results with calculated response from test measurements (test FV5).

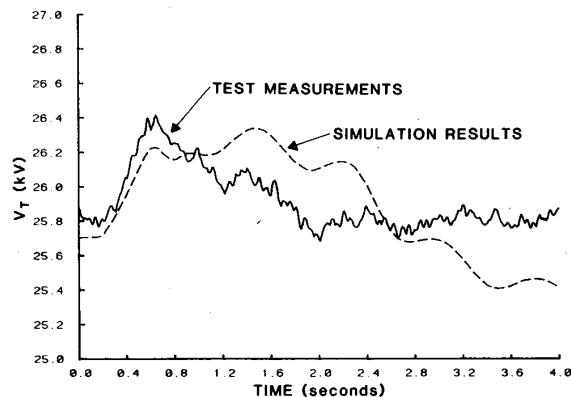


Fig. 10. Rockport 26-kV terminal voltage: Comparison of simulation results with calculations from test measurements (test FV5).

5. EVALUATION OF ELECTRICAL SYSTEM RESPONSE

The change in the unit mechanical driving power during TFTV results in variations in the electrical power output and other electrical parameters at Rockport. The response of the unit and the power system to TFTV was evaluated by examining the variations of several key electrical parameters, including: Rockport machine terminal voltage, stator currents, field quantities, real power output and reactive power output.

The three-phase 26-kV terminal voltage, HP machine stator current and RH machine stator current monitored during the field tests were converted to positive-sequence phasor quantities at discrete time steps using the Discrete Fourier Transform Method — developed by A. G. Phadke, et. al. for relaying applications[6]. The positive-sequence phasor quantities were then utilized in analyzing the electrical system response.

5.1 Reactive Power Response of Rockport Unit No. 1

One of the most interesting aspects of the electrical system response pertains to the reactive power output of the Rockport unit following TFTV. Figure 11 illustrates the reactive power responses — calculated from the positive-sequence machine terminal voltage and stator currents — of the HP and RH machines during test FV5.

Prior to TFTV, both the HP and RH units were operating in the underexcited mode. Moreover, the reactive loading on both units was balanced, with each unit absorbing approximately 30 MVAR. However, the transient reactive power outputs of the HP and RH machines following the application of TFTV were significantly different and acted in phase opposition to each other. While the reactive power output of the RH machine exhibits a tendency to oscillate about its pre-disturbance value, the HP machine reactive output tends to drive the machine further into the underexcited range before settling to its pre-disturbance level.

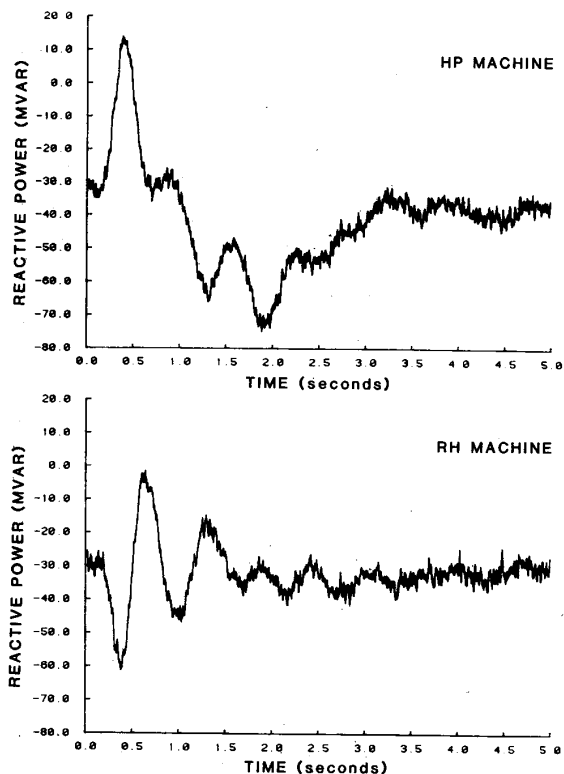


Fig. 11. Calculated reactive power output of Rockport HP and RH machines during TFTV (test FV5).

This machine behavior contradicts a common assumption that the responses of two similar units, operating in a cross-compound environment, would be similar following a disturbance. The tendency for the HP and RH machines to oscillate against each other suggests a possibility that the two units may develop even larger reactive power oscillations following a disturbance more severe than the TFTV action alone, e.g. TFTV initiated by an electrical fault.

This difference in reactive power responses of the two machines may be attributed to two primary factors. The delay in valving between the control and intercept valves results in different shaft mechanical power responses for the two units. Consequently, the electrical power outputs of these units would be dissimilar. Also, each unit has its own excitation system, with current droop and cross-current compensation. A difference in the responses of these two excitation systems can lead to dissimilar reactive power outputs.

An analytical study is under way to examine the causes of this behavior. This study will attempt to reproduce through digital simulations the excitation system responses of the HP and RH machines observed during the field tests. Preliminary simulations, using standard IEEE excitation system models and manufacturer-provided data, have not duplicated this unbalanced reactive power response. Therefore, more detailed excitation system models -- perhaps with explicit representation of cross-current compensation -- will be used in future simulations. Moreover, the

manufacturer-provided data will be verified by: (i) analyzing data obtained from exciter field tests carried out at Rockport in 1984, and (ii) calculating the exciter parameters based on the actual potentiometer settings.

5.2 Effect of TFTV on Local Frequency

The impact of TFTV on the local frequency was evaluated by examining the positive sequence voltage phasor at the Rockport 26-kV bus. Figure 12 illustrates the calculated system frequency during test FV5. The momentary drop in generation of approximately 1000 MW at Rockport resulted in a decrease in frequency to 59.9 Hz. This is due to the load/generation imbalance and the inherent inability of the automatic generation control system to follow rapid changes in generation levels. Although the frequency decreased momentarily, it recovered quickly, as generator output was automatically restored to its pre-test level.

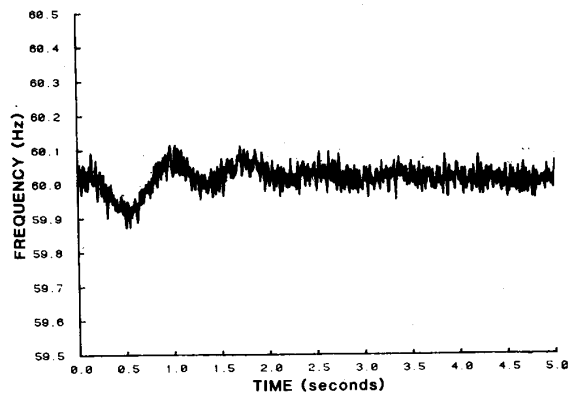


Fig. 12. Calculated frequency at Rockport 26-kV bus (test FV5).

6. SUMMARY

6.1 Conclusions

Valve Stroke Characteristics

In general, during the three sliding pressure field tests, the performance of the control and intercept valves equalled or exceeded expectations. During the constant pressure test FV8, however, the control valves were sluggish while closing and reopening and they overshoot significantly (relative to their initial position) during reopening. Despite the differences between the measured responses and the design characteristics, the performance of these valves was judged to be satisfactory for successful fast valving.

Shaft Mechanical Power Characteristics

It is possible to calculate shaft mechanical power response based on measured speed deviation and gross electrical power output of the generator. The calculated mechanical power indicates that the reduction in the mechanical power during the tests was more pronounced than that predicted by the manufacturer. This is because the control and intercept valves attained lower minimum positions than anticipated.

Electrical Performance Characteristics

Due to the generation-load imbalance caused by TFTV, the system frequency decreased momentarily and then recovered quickly during each field test. The reactive power responses of the HP and RH machines were found to be significantly different because of: (i) dissimilar mechanical power responses of the two units due to unequal valve motion of the control and intercept valves, and (ii) the difference in the excitation system response of the two units.

Comparison of Measurements with Simulation Results

In general, the simulation results were well correlated with the field test measurements. Minor differences were due to the limitations of the digital computer program and lack of precise data for representing various power system components. The magnitudes of the simulated speed deviations and electrical power swings were slightly larger than those recorded during the TFTV field tests due to the simplified representation of the rotational system damping in the simulations.

The simulated terminal voltage exhibited a slightly larger magnitude and more prolonged period of voltage rise than that observed during the field tests. This difference can be attributed to the lack of detailed excitation system representation.

6.2 Future Work

As part of the future Rockport TFTV/SPS field test program, two additional tests are being considered -- one involving a 765 kV fault without TFTV and the other involving a fault accompanied by TFTV. These test results should permit us to evaluate the turbine mechanical damping and the electrical system damping components independently.

Finally, the TFTV tests demonstrated unbalanced reactive power responses of the Rockport HP and RH units, including a tendency for the two units to oscillate against one another. This result indicates a need to carry out an additional study to examine the cause(s) of the unbalanced reactive power response and to determine corrective measures. More detailed simulation models of the excitation system will be required. These models should closely resemble the physical layout of the unit's excitation system to aid in formulating practical recommendations for voltage regulator settings.

Discussion

A. Keyhani (The Ohio State University, Columbus, OH): I would like to congratulate the authors for establishing a fast-valving test procedure, performing in-depth analysis of their results and proposing new control concepts. First, I would like to comment on the significance of the author's control concepts. Then I will discuss the importance of fast-valving test results in developing models for simulation study of the new control concepts.

I believe the control concepts presented in this paper have a significant impact on power system operation and planning. In system operation, when a phase-to-ground fault occurs, the fast-valving combined with single-phase switching followed by single-phase reclosing will provide an additional transient stability margin. At the same time, it will reduce the torsional impact on the turbine-generator shaft. If these control concepts are implemented in operations, then the system can be planned with fewer lines, resulting in a substantial reduction of capital expenditure and cost. This is indeed the most economical method of obtaining improved system dynamic performance. The authors should be commended for proposing these new control concepts.

7. ACKNOWLEDGEMENTS

The authors acknowledge the efforts of the Rockport Testing Task Force, which coordinated the development and execution of the various field tests, including the TFTV tests, and others called upon to support the task force's activities. The efforts of the Rockport Plant staff, who assisted in conducting the tests is greatly appreciated. Particular thanks are extended to the Canton Electrical Laboratory personnel, who collected and pre-processed the test data, and others who assisted in developing the computerized data analysis system. Finally, the assistance of our colleagues in the Bulk Transmission Planning Division in the preparation of this paper is gratefully acknowledged.

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The fast-valving test data provide valuable information for generator model development and validation. The data can be used to develop a model that would reflect the effect of normal operating flux levels on the parameters. This is an important aspect of the fast-valving test, since it facilitates the estimation of saturated machine parameters from operating data. An accurate generator model and its exciter would permit the simulation study of the new control concepts.

I would appreciate the authors' comments on the following questions:

- 1) Are the authors considering the use of sliding pressure mode of operation in automatic generation control [1]?
- 2) Were the Rockport units controlled by an automatic voltage regulator [AVR] during fast-valving tests?
- 3) What was the generator representation in the transient stability program? Were the saturation effects included?

Reference

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Manuscript received February 13, 1987.

W. R. Brownlee (Consulting Engineer, Birmingham, AL): The application of fast turbine valving to units such as those at Rockport is a major undertaking, and the authors are commended for their exhaustive studies, well-planned tests, and documentation for the benefit of others. Since exchanges of ideas are beneficial, certain comparisons will be made leading to questions for the authors.

The earliest field tests of fast turbine valving known to this discussor were made in the Spring of 1970, with preliminary publication in *Electrical World* [1]. The authors' Fig. 9 (reheat) shows a reduction in mechanical input to about 60 percent of normal after about 52 cycles, and the manufacturer's projection was slightly faster. This compares with a reduction of the Southern units to about 60 percent of normal after 24 cycles as shown by the manufacturer's projection [2]. However, the field tests showed that the intercept valves completely closed in about 15 cycles, compared with the manufacturer's projection of 65 cycles. It is evident that the reduction in mechanical input to 60 percent of normal occurred in much less than 24 cycles. While electric output lags and oscillates and is therefore less accurate, it is noted that in the authors' Fig. 8, power decreases to 60 percent of normal in about 40 cycles, compared with about 26 cycles for the Southern units. Figure 1 shows the rapid reduction in electric power output, and a comparison of field-tested and calculated performance [1]. After a delay of one second, the intercept valve opens rapidly to restore full output at about 2.5 s. This minimizes the need for intercept safety valve operation.

The Southern units closed the intercept valves completely, instead of partially, and at about twice the speed of the Rockport Units. These are 2000-lb. units and 3500-lb. units. The three units of one manufacturer performed very well, but one of the 3500-lb. units of another US manufacturer required complete replacement of the intercept valve mechanisms to assure acceptable operation. To what extent did the authors explore the possibility of improved valve mechanisms, complete closing, and doubling the speed of the intercept valves? If such changes were feasible for these foreign units, would it have been necessary to close also the main control valves, rather than relying on the intercept valves as most others do?

The use of protective line relays to initiate fast valving may save one or two cycles of time, but it requires connections such that all units in a plant could be valved at once, with no fault, by a single unsteady screw driver. Have the authors studied the effect of such an event?

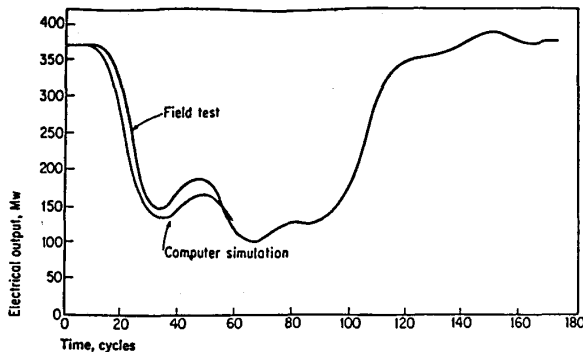


Fig. 1. Fast intercept valve testing produces sharp drop in electric output.

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J. Fish and R. Shinn (Tennessee Valley Authority, Chattanooga, TN): The authors are to be congratulated for this informative analysis of fast valving tests at the Rockport Plant. Papers like this are valuable in developing improved simulation techniques.

The results of our Cumberland Steam Plant fast valving tests were presented to the American Power Conference in April 1981. The Cumberland units have turbine-generators nearly identical to those at Rockport, but operate in the constant pressure mode only and employ a sustained fast valving scheme. In spite of these differences one might expect

the fast valving to respond similarly in the early portion of the event. However, our test results indicated considerably less delay before the valves began to move (CV-16 ms, IV-80 ms) with relatively slow closing rates (CV-280 %/s, IV-190 %/s). This resulted in earlier reduction in total mechanical driving power than was designed but with a slower decline. Computer simulations verified that this reduction was adequate for our requirements at the plant.

The authors concluded that despite the differences between the measured and designed characteristics of the valve strokes, the performance was judged to be satisfactory. They further noted that because the valves attained lower minimum positions, the reduction in mechanical driving power was more pronounced than designed and concluded that this is beneficial to transient stability. Although the closing rate of all valves was greater than the design values, the increased initial delay before valve movement is of serious concern and can be detrimental to unit stability. Figure 9 indicates that the simulated P-mech does not account for the initial delay observed in the measured data. We feel that the authors should consider this point in developing the control system model. Why is the measured value of P-mech not used directly in simulations rather than the turbine system computer model? We agree that the determination of P-mech from the data available during tests is an interesting exercise. To us, the critical part of the evaluation is the determination of speed deviation. Could the authors supply more details of how this was accomplished?

From Fig. 8, it can be observed that the period of the oscillations differ between measured and simulated values. Have the authors considered using values of machine reactances other than those supplied by the manufacturer to achieve a better match?

We are looking forward to seeing the results of the staged fault tests.

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M. C. Patel (Allegheny Power Service Corporation, Greensburg, PA): We would like to thank the authors for sharing their fast valving test results for the Rockport plant with the electric power industry. We also compliment them for the excellent presentation of their analysis in the paper. We have the following questions:

1) The paper does not mention the boiler's response during the fast valving tests. Boiler transients are usually one of the concerns in application of fast valving. Would the authors summarize and comment on the boiler transients during their fast valving tests?

2) The mechanical power response to fast valving shows a considerable difference between the response derived by the authors from the test data and the response predicted from the design characteristics. In the authors' experience, is the difference significant enough to warrant deriving all such data from tests for sufficiently accurate prediction of dynamic performance of the plant?

Manuscript received February 23, 1987.

John H. Doudna (Nebraska Public Power District, Columbus, NE): The authors' work on the development and testing of a remedial action scheme based upon steam turbine fast valving is of great interest to this discussor since I have been directly involved with fast valving applications on my own system. The authors' contributions in the application and testing of fast valving are a valuable asset to the utility industry as a whole. The concept of fast valving is often met with much opposition and skepticism. This opposition and skepticism has suppressed the use of fast valving by many utilities, and it has kept fast valving from being universally accepted as the practical, cost effective, and powerful system stabilization technique which it actually is. The authors' published work on the application of fast valving is important in helping to overcome the opposition and skepticism toward the application of fast valving techniques.

Despite the importance of the authors' work, I must criticize their use of the term "temporary" in conjunction with "fast valving." Although not a standard definition, fast valving of the type employed by the authors is most commonly referred to in technical literature as "momentary fast valving." The use of commonly accepted terms is important in preventing the proliferation of terms which may confuse those who are not intimately familiar with the subject of fast valving. I would greatly appreciate the authors' and other readers' comments on this matter.

In reading the paper, I did not find any explicit statements which specified the type of steam supply system used at Rockport. The authors' steam flow path diagram tends to portray the use of a supercritical, once-through boiler rather than a drum-type boiler. Is this observation correct? Did the application of fast valving have any bearing on the type of steam supply system selected for Rockport? Do the authors have any opinion as to which type of steam supply system is best suited for use with fast valving

applications?

Concerning the valve stroke characteristics of the control and intercept valves, the authors indicate that they measured the valve stroke characteristics of all four control valves and all four intercept valves, but information is only provided for one control and one intercept valve. Did all control valves and all intercept valves operate in an essentially identical manner? Among the four valves of each type, did any valve exhibit valve stroke characteristics which were unique and if so, what aspect of the valve stroke characteristic was unique (i.e., closing rate, reopening rate, time delay, etc.)? Was the uniformity of the valve stroke characteristics, or lack thereof, among each type of valve expected by the authors? Based on the discussor's experience with four intercept valves on a 600-MW, tandem compound, BBC steam turbine-generator, each of the four intercept valves, and especially one, exhibited a different valve stroke characteristic, and the valve stroke characteristics varied from test-to-test for different unit loading conditions. The only difference between the authors' application and the discussor's application is that the fast valving application on the discussor's system drives the intercept valves to the fully closed position before allowing them to reopen. Would the authors expect to see nonuniform intercept valve stroke characteristics for Rockport had they elected to drive the intercept valves to the fully closed position during fast valving operations?

As presented by the authors, analytical simulations of fast valving operations at Rockport predicted somewhat different mechanical power characteristics than those determined from test measurements. The analytical simulations predicted a faster power reduction than actually occurred, and the analytical simulations predicted that power recovery would also be faster. It is interesting to note that the discussor noted similar findings when comparing his analytical simulations with his actual fast valving test measurements. The similarity of results between the two work efforts would tend to indicate that the analytical models which exist today and rely on manufacturers' data do not accurately predict the performance of a steam turbine-generator during a fast valving operation. Thus, utilities investigating fast valving applications should not rely too heavily on such analytical simulation results of fast valving performance. It also reinforces the need to perform actual fast valving tests to verify such performance and to use this test information to refine the analytical simulation models so that they properly reflect the situations which exist.

As revealed through the authors' measurements of the fast valving tests, the delay time for the control and intercept valves following fast valving initiation was much longer than expected. As the authors state, this longer delay time was somewhat offset by a valve closure rate which was faster than expected. However, it would appear that the unit response rate during fast valving operations could be significantly enhanced if the valve delay time could be reduced to the originally expected values. As a matter of interest, were valve delay times, closing rates, and reopening rates specified as a requirement in the contract with the turbine-generator manufacturer? Have the authors identified the reason that the valve delay times measured were 1 1/2 to 2 times their expected value? Do the authors intend to investigate this matter further and pursue making modifications which might reduce the delay time? Although a comparison is difficult to make due to the different sizes of the authors' and the discussor's units which employ fast valving, the discussor measured delay times for his intercept valves which range from 40 ms to 76 ms with the average delay time being approximately 60 ms. Assuming that the valve springs, hydraulic dump valves, and other valve components as sized in proportion to the size of the valve, it would seem that the authors' intercept valves should exhibit delay times similar to those measured by the discussor on smaller valves made by the same manufacturer. Do the authors have any comment on this matter?

Manuscript received February 27, 1987.

R. H. Park (Fast Load Control Inc., Providence, RI): Like its companion paper, 87 WM 069-8, the paper under discussion represents a most useful treatment of its subject matter.

Besides presenting a very interesting set of test data, the paper carefully looks into the equally interesting question of the extent to which time-varying values of generated and turbine driving power predicted from test values of valve positions versus time, differ from test data on generated power, and changes in turbine driving power computed from test data.

In this, per Fig. 9 it can be seen that values labeled from test measurements, representing data computed from measured values of generated power and turbine speed change, consistently lag behind simulation results, which comprise predictions of turbine driving power based on test data covering changes in valve positions.

A principal way that a lag of the type in question can arise in the course of calculations involving changes in valve position is by the inadequate taking into account of steam volumes that are located downstream of control valves

yet ahead of a high pressure turbine's last stage, or downstream of intercept valves, yet ahead of the last stages of the reheat and low pressure turbines.

Reference to Fig. 7 will establish that it does not fail to include consideration of the volumes of the high pressure turbine steam bowl and the crossover, yet there appears to be no evidence of provision to take account of the volume of the reheat turbine's steam bowl.

Again, if not already done, it could be that there would be warrant to increase the value of T_1 as a way to allow for the effect of the steam volumes located between the high pressure turbine steam bowl and that turbine's last stage, while the same could apply to the reheat turbine. Also, would it possibly be worth doing, if not already done, to add the volume of the exhaust hood of the reheat turbine and the steam inlet volumes of the low pressure turbines to the volume of the crossover.

Another point that warrants taking note of is the fact that, per Fig. 9, and as noted in the writer's discussion of the companion paper, the driving power of the high pressure turbine dropped from 100 to around 60 percent of the full load combined driving powers of the high and one low pressure turbine, before, as of about 0.45 s following the start of test, the reheat turbine's intercept valves started to close.

What is notable here is that, based on the values of K_2 and K_3 , it would appear that the full load driving power rating of the high pressure turbine is .32 pu, or thereabout, of total rated turbine driving power, and that of each low pressure turbine .18, for a total of .50 as the rated per unit driving power of the high pressure turbine and one low pressure turbine taken together.

In view of reheat pressure decreasing only slowly, it would appear that, when the reheat turbine's intercept valves had not yet begun to close, reduction in the driving power of the low pressure turbines would have been inconsequential, which, in turn, would imply that, at that point in time the driving power of the high pressure turbine had dropped by around $.4 \times .5$ or .20 pu.

This drop, however, which amounts to .20/.32 or five-eighths of the high pressure turbine's rated driving power, took place at a stage when that turbine's control valves had closed only two-thirds as far as the full extent of closure that had been planned on.

This high ratio of decrease of high pressure turbine driving power to extent of fast control valve closure, which, it may be noted, also showed up in TVA's Cumberland tests (carried out with full steam pressure maintained), points to the fact that at any rate in the case of the high pressure turbine, where, in fast valving, turbine exhaust pressure during the key portions of the valving cycle would have held at near prefast valving value, and where entrained steam effects on steam flow behavior are on the high side during at least portions of that cycle, driving power will no longer behave as though it is substantially linearly related to steam flow.

This, in turn, brings with it the implication that turbine driving power as a function of valve position can differ considerably under sufficiently rapidly changing conditions, from what is commonly taken to apply when rapid change is absent, a proposition that a review of the Rockport test data doubtless could be made use of to more fully explore.

As bearing on speed of valve closure, it is worth noting the fact that whereas the Rockport delay between the instant of energization of the turbine's controls, and the start of control valve movement was 87 ms, in TVA's Cumberland tests this delay time averaged only around 40 ms, a value which, in due course, it may be determined can be yet further reduced.

Manuscript received February 27, 1987.

B. M. Pasternack and N. B. Bhatt: The authors would like to thank each of the discussors for their comments and questions which add considerably to the value of the paper. We will attempt to address each of the specific questions raised by the discussors after commenting on the significant broader issues.

First, we want to emphasize that the decision to apply fast valving at Rockport involved more than a simple cost comparison between transmission additions and supplementary controls. While we were able to avoid a substantial capital expenditure associated with construction of a third, long 765-kV transmission outlet by using a low-cost fast valving approach, other issues including equipment and system reliability were also carefully evaluated. Only after being convinced that the Rockport fast valving scheme would operate in a secure and reliable manner and that boiler and turbine-generator stresses due to fast valving would be minimal did we commit to the application of fast valving at Rockport. Had the economics not been so overpowering—as, for example, if a relatively short, third outlet could have been developed—the decision would have been different.

Because there was no standard terminology to describe various fast valving schemes when we first began investigating application of this technique in the early 1970's, we coined the term "temporary" fast turbine

valving for the Rockport scheme. Although "momentary" ultimately became a more commonly used description for the type of fast valving being applied at Rockport, we decided to retain the original terminology to avoid unnecessary confusion within our own company. We would welcome standardized industry definitions for broad classifications of fast valving.

As correctly observed by Mr. Doudna, the two 1300-MW Rockport units use supercritical once-through boilers. The application of fast valving did not have any bearing on the type of steam supply system selected since these two units are the fifth and sixth in AEP's series of 1300-MW units originally designed in the late 1960's. Regarding the question of whether the once-through or the drum-type steam supply system is better suited for fast valving, we feel that because of its inherent ability to absorb transients more readily, a drum-type system is perhaps better suited for fast valving. However, as evidenced by the variety of applications throughout the United States, fast valving can be implemented on units having either steam supply system. The important point is that the boiler and turbine thermal responses during fast valving should be carefully analyzed and necessary hardware modifications should be made to assure acceptable thermal response and compatibility with the fast valving controls.

To specify the fast valving requirements for the Rockport units, we provided to the turbine-generator manufacturer a bandwidth of mechanical driving power, which—based on our computer simulation studies—would ensure post-disturbance stability of the plant. The manufacturer then determined the achievable valve stroke characteristics that would produce the desired mechanical driving power. Since closing only intercept valves could not provide sufficient mechanical power reduction, we decided to close both sets of valves (control and intercept). Closing both sets of valves also distributes the impact of fast valving more evenly across the boiler and turbine and results in less stress than closing only the intercept valves.

Several discussers raised the issue of longer initial delays experienced by the control valves than predicted by the manufacturer (85 ms vs. 35 ms). We will be investigating the reasons for this delay. It should be pointed out, however, that because of the faster closing rate of the control valves, the resultant mechanical power response was quite acceptable from the stability performance viewpoint. The need to bring the actual delay times closer to the predicted values will be evaluated considering all these factors.

In response to Mr. Doudna's questions regarding valve stroke characteristics, all four control valves exhibited similar stroke characteristics during the field tests, as did three of the four intercept valves. The fourth intercept valve, however, consistently overshot its minimum setpoint (15 percent vs. an expected 26 percent) during all four tests because of calibration problems. Since control oil pressure vs. stroke characteristics of each of the four control valves and each of the four intercept valves are identical, we expected uniformity among the stroke characteristics of each set of valves. We would expect this uniformity even if we were to drive the intercept valves to the fully closed position.

The thermal responses of the boiler and turbine were carefully monitored during the tests since pressure transients are important considerations in fast valving, as pointed out by Mr. Patel. A main steam pressure rise of 5.6 percent (200 p.s.i.) was recorded during the full load test (FV5) about 0.8 s after fast valving initiation, with a return to original pressure in 6 s. A rise of 10 percent in the reheat steam pressure was the largest percentage increase observed during test FV5, although this pressure rise exceeded 5 percent for less than 1 s. The magnitudes of pressure transients observed during the tests do not encroach on the settings of the power relief, main steam, or reheat safety valves.

The mechanical power responses—derived from test measurements and simulated using a detailed turbine system model—generated many interesting comments. We agree with Messrs. Fish and Shinn that the speed

deviation determination is a critical part in the derivation of mechanical power from the test data. The speeds of the HP and RH machines were measured in rpm using magnetic speed pickup devices. The pulses from a manufacturer-installed transducer "reading" the toothed wheel on the shaft were recorded on a magnetic tape. These pulses were then processed through a digital-to-analog converter. The result was a dc level representing steady-state speed prior to fast valving. When the speed increased or decreased, the spread between pulses either decreased or increased, resulting in an increase or decrease of the dc voltage level, respectively.

The mechanical driving power characteristics provided by the turbine-generator manufacturer was based on the expected valve stroke characteristics for a full load condition and constant pressure unit operating mode. Because we planned to conduct field tests at various unit output levels (100 percent, 80 percent, and 60 percent) and under sliding pressure operating mode, we developed a detailed turbine system model (shown in Fig. 7 of the paper) to predict and evaluate the expected electromechanical performance of the unit using the AEP transient stability program.

Following the field tests, we used the same turbine system model to simulate the mechanical power (P_{mech}) response during the tests based on the measured valve stroke characteristics. A comparison of the simulated P_{mech} response with the response derived from the test measurements using the swing equation (Fig. 9 of the paper) indicates excellent correlation. This exercise provided validation of the turbine control system model for future simulation studies of the Rockport plant.

Mr. Park provided interesting insights into the turbine-generator modeling aspects with respect to the time constants. We intend to pursue some of his suggestions. Mr. Park's analysis of the relative mechanical power reductions of the HP and RH machines during the field tests was also very interesting.

The analysis of the field test data—especially the comparison of design and measured valve stroke and P_{mech} characteristics—has reinforced our decision to perform field tests to verify the functional characteristics of the fast valving scheme and the resultant electromechanical power response of the unit.

We concur with Dr. Keyhani that it is feasible to estimate generator parameters from the fast valving test data. The fast valving would cause redistribution of the air-gap flux, which consequently would result in variations in the operational impedances of the generator. We intend to calculate more accurate generator parameters from test data collected during the fast valving tests as well as other tests made on the Rockport unit. While we have not used various values of machine reactances to match the periods of oscillations of the measured and simulated values of electrical power (Fig. 8 of the paper), as suggested by Messrs. Fish and Shinn, there is a good possibility that more accurate generator parameters derived from test data will result in closer correspondence between the periods of oscillation.

In response to several specific questions asked by Dr. Keyhani: 1) the Rockport unit normally operates under automatic generator control whether in the sliding pressure or constant pressure mode, although it is not used as a peaking unit; 2) both Rockport generators were under automatic voltage regulator control during the tests; and 3) detailed machine representation with saturation effects was used for the simulation studies.

Mr. Brownlee raised a question regarding a false fast valving operation. In our planning studies, we did consider the impact of such operating errors. We believe that even if such a false operation occurs, it would not have a significant impact on the plant or the system, as evidenced by the fast valving tests, which produced a fast valving excursion even though there was no system requirement to do so.

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