Fuel Consumption Minimization in Compressor station Operation Under Non-isothermal Condition

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ABSTRACT

Mathematical modeling is one of the most cost-effective tools that can be used to aid in design, operation, and optimization studies. The systems under consideration actually operate in an unsteady nature, and although much effort has been and continues to be spent on unsteady mathematical models, many over-simplifications are introduced that bring into question the simulation results.

One of the primary concerns in the operation of a compressor station is minimization of fuel consumption while maintaining the desired throughput of natural gas. In practice, the station operator tries to achieve this by shutting down units or controlling individual unit speeds based on experience. This is generally a trial-and-error process without any guarantee of optimality. In this paper we present a robust structured solution process for tackling this problem using simulation-based optimization.

INTRODUCTION

Natural gas enters the pipeline from a supply source, and then is transported to one or more delivery points. One of the most important collections of components in this system is the compressor station located about every 60 miles along the pipeline. The compressor station overcomes the gas pressure drop in the pipe. Consequently, detailed mathematical modeling of compressor stations is critical for optimizing and understanding the ability of the gas pipeline system to deliver natural gas to the end-user.

Several investigators tried to simulate unsteady conditions for pipeline systems and some of them focused in compressor station modeling. *Bryant [1]* modeled compressor station control, which had some advantages such as the ability to set individual unit swing priority, the ability to try and meet multiple setpoints and the ability for units to automatically come on-line and off-line. The model used automatic linepack tuning instead of automatic pipeline roughness tuning.

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Turner and Simonson [2,3] developed a computer program for compressor stations that is added to SIROGAS, which is program for solving a pipeline network for steady state and transient mode.

Stanley and Bohannan [4] discussed the application of dynamic simulation to centrifugal compressor control system design. 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. This paper solves equations in ordinary form without considering pipe equations into compressor station.

Botros et al. [5,6] and Botros [7] presented a dynamic compressor station simulation that consisted of nonlinear partial differential equations describing the pipe flow together with nonlinear algebraic equations describing the quasi-steady flow through various valves, constrictions, and compressors. This model included mathematical descriptions of the control system, which consists of mixed algebraic and ordinary differential equations with some controller limits.

Schultz [8] derived the real-gas equations of polytropic analysis and to show their application centrifugal compressor testing and design. *Odom [9]* reviewed the theory of centrifugal compressor performance, and also presented a set of polynomial equations for centrifugal compressor maps, which used constant coefficient for these equations for different compressor.

Letniowski [10] presented an overview of the design process for a compressor station model that is part of a network model.

Jenicek and Kralik [11] developed optimized control of a generalized compressor station. The model described algorithm for optimizing the operation of the compressor station with fixed configuration.

Carter [12] presented a hybrid mixed-integer-nonlinear programming method, which is capable of efficiently computing exact solutions to the restricted class of compressor models, and attempted to place station optimization in context with regard to simulation.

Boyd et al. [13] considered the fuel cost minimization problem in the steady-state gas pipeline networks by using mathematical model over compressor station.

Carter [14] developed a nonsequential Dynamic Programming (DP) algorithm to handle looped networks when the mass flow rate variables are fixed. The main advantages of DP are that a global optimum is guaranteed to be found and that nonlinearity can be easily handled.

Botros [15] presented a numerical study of gas recycling during surge control, and furnished a basic understanding of the thermodynamic point of view and showed the variation of gas pressure, temperature and flow.

Wu et al [16] presented two-model relaxation, one in the compressor domain and another in the fuel cost function, and derive a lower bounding scheme. The empirical evidence has been presented that showed the effectiveness of the lower bounding scheme.

Siregar et al [17] developed a mathematical model, which in turn solved analytically, and numerically for optimum pipeline diameter and routing.

Cobos-Zaleta and Rios-Mercado [18] used a MINLP model for the problem of minimizing the fuel consumption in a pipeline network. A computational experience was presented by evaluating an outer approximation with equality relaxation and augmented penalty method.

Edgar et al [19] presented a computer simulation to optimize the design of a gas transmission network, which considered the number of compressor stations, the diameter and length of pipeline segments, and the operating conditions of each compressor station. Two solution methods were used.

Osiadacz [20] described the algorithm for optimal control of a gas network based upon hierarchical control and decomposition of the network. This work is concerned with the minimization of operating costs for high-pressure gas networks under transient conditions.

The work presented in this paper is an important advance over current methods in the accurate simulation of transient non-isothermal behavior in natural gas pipelines, and extends the knowledge found in the literature by demonstrating the impact of varying boundary conditions on compressor station components. In addition, it also shows how this type of detailed simulation can be used for optimizing the operation of a compressor station to minimize fuel consumption while maintaining desired throughput.

NOMENCLATURE

A Cross- sectional area of pipe (m^2)
b_1 - b_6 Coefficients for centrifugal compressor map (-)
C_p Specific heat at constant pressure (J/kg. K)
D Pipe diameter (m)
f Friction factor (-)
g Gravitational acceleration (m/s^2)
h Specific enthalpy (J/kg)
$Head$ Isentropic head (kJ/kg)
LHV Low heating value (kJ/kg)
$\dot{m}$ Mass flow rate (kg/s)
N Number of node (-)
N_r Speed (rpm)
P Pressure of the gas (Pa)
Q Capacity (m^3/hr)
R Specific gas constant (kJ/kg)
t Time (s)
T Temperature (K)
v Velocity of the gas directed along the axis of the pipe (m/s)
V_w Isentropic wave speed (m/s)
W Frictional force per unit length of pipe and per unit time (N/m)
x Distance along the pipe (m)
Z Compressibility factor (-)
η Efficiency (-)
γ_g Specific gravity (-)
θ Angle of inclination of pipe to the horizontal (radian)
σ Isentropic exponent (-)
ρ Density of the gas (kg/m^3)
Ω Heat flow (J/ms)

Subscripts

ac Actual condition
d Discharge
dr Driver
f Fuel
is Isentropic
$mech$ Mechanical
s Suction
Sc Standard condition

GOVERNING EQUATION

The first step to develop this solution process is to devise an analysis scheme that provides the simulation support required by the optimization. The non-isothermal flow of natural gas in a compressor station is governed by the time-dependent continuity, momentum, and energy equations, and an equation of state for homogeneous, geometrically one-dimensional pipe flow. The compressors within the compressor station are modeled using centrifugal compressor map-based polynomial equations.

Pipe Equations

Issa and Spalding [21], Deen and Reintsema [22], Thorley *and Tiley [23], and Price et al. [24]* developed the basic equations for one-dimensional, unsteady, compressible flow that include the effects of wall friction and heat transfer:

Continuity Equation

$$\frac{\partial \rho}{\partial t}+\frac{\partial}{\partial x}(\rho v)=0 \tag{1}$$

Momentum Equation

$$\rho\frac{\partial v}{\partial t}+\rho v\frac{\partial v}{\partial x}+\frac{\partial P}{\partial x}=-\frac{w}{A}-\rho g sin\theta \tag{2}$$

Where:

$$w=\frac{f\rho v|v|}{8}\pi D$$

Conservation of Energy

$$\rho\frac{\partial h}{\partial t}+\rho v\frac{\partial h}{\partial x}-\frac{\partial P}{\partial t}-v\frac{\partial P}{\partial x}=\frac{\Omega+wv}{A} \tag{3}$$

Ω is heat flow into the pipe per unit length of pipe and per unit time.

Equation of State

$$\frac{P}{\rho}=ZRT \tag{4}$$

To obtain h in terms of P, Z, and T, *Zemansky [25]* described the thermodynamic identity:

$$dh=CpdT+\left\{\frac{T}{\rho}\left(\frac{\partial \rho}{\partial T}\right)_P+1\right\}\frac{dP}{\rho} \tag{5}$$

The resulting set of equations is:

$$\left(\frac{\partial P}{\partial t}\right)+v\left(\frac{\partial P}{\partial x}\right)+\rho V_w^2\left(\frac{\partial v}{\partial x}\right)=\frac{V_w^2}{CpT}\left[1+\frac{T}{Z}\left(\frac{\partial Z}{\partial T}\right)_P\right]\frac{\Omega+wv}{A} \tag{6}$$

$$\left(\frac{\partial v}{\partial t}\right)+v\left(\frac{\partial v}{\partial x}\right)+\frac{1}{\rho}\left(\frac{\partial P}{\partial x}\right)=-\frac{w}{A\rho}-g sin\theta \tag{7}$$

$$\left(\frac{\partial T}{\partial t}\right)+v\left(\frac{\partial T}{\partial x}\right)+\frac{V_w^2}{Cp}\left[1+\frac{T}{Z}\left(\frac{\partial Z}{\partial T}\right)_P\right]\left(\frac{\partial v}{\partial x}\right)$$

$$=\frac{V_w^2}{CpP}\left[1-\frac{P}{Z}\left(\frac{\partial Z}{\partial P}\right)_T\right]\frac{\Omega+wv}{A} \tag{8}$$

The parameter V_w is:

$$V_w=\sqrt{\frac{ZRT}{\left\{1-\frac{P}{Z}\left(\frac{\partial Z}{\partial P}\right)_T-\frac{P}{\rho CpT}\left[1+\frac{T}{Z}\left(\frac{\partial Z}{\partial T}\right)_P\right]^2\right\}}} \tag{9}$$

Centrifugal Compressor Equations

To simulate the compressor station, the following equations are used for centrifugal compressor. Compressor head will obtain by:

$$Head=0.28704\frac{T_sZ_s}{\sigma\gamma_g}\left(\left(\frac{P_d}{P_s}\right)^{\sigma}-1\right) \tag{10}$$

and the relationship between the flow rate for standard condition and actual mass flow rate is:

$$Q_{sc}=\frac{\dot{m}_{ac}R}{97.67\times10^{-3}} \tag{11}$$

and the power that needs for compressor is:

$$Power=\frac{Head\times\dot{m}_{ac}}{\eta_{is}\eta_{mech}} \tag{12}$$

For the purpose of inputting centrifugal compressor characteristics into a pipeline simulation model, it is suggested that the entire head versus capacity map be digitalized and stored as a table. However, a simplified but still accurate representation of the head versus capacity curve can be obtained through the use of the non-dimensional characteristics. Figure 1. shows a sample compressor map. Three non-dimensional parameters are necessary to describe a compressor map, $\frac{Head}{N_r^2}$, $\frac{Q_{ac}}{N_r}, \eta_{is}$. Using standard polynomial curve-fit procedures for each centrifugal compressor [9] , the relationship between these parameters could be found by:

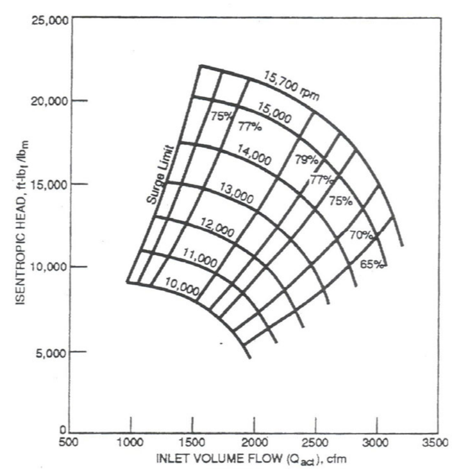


Fig.1 Compressor Map [9]

$$\frac{Head}{N_r^{\,2}} = b_1 + b_2\left(\frac{Q_{ac}}{N_r}\right) + b_3\left(\frac{Q_{ac}}{N_r}\right)^2 \qquad (13)$$

and:

$$\eta_{is} = b_4 + b_5\left(\frac{Q_{ac}}{N_r}\right) + b_6\left(\frac{Q_{ac}}{N_r}\right)^2 \qquad (14)$$

Where:

$b_1, b_2, b_3, b_4, b_5, b_6$ are coefficients that make Eqs. 13 and 14 fully characterize the specific centrifugal compressor map. With the coefficients for Eqs. 13 and 14 stored in the computer, knowing the isentropic head and inlet volumetric flow will allow computation of compressor speed and isentropic efficiency.
The fuel consumption for the compressor driver is currently obtained by:

$$\dot{m}_f = \frac{Power}{LHV \times \eta_{dr}} \qquad (15)$$

The gas discharge temperature is obtained by:

$$T_d = T_s + \frac{T_s}{\eta_{is}/100}\left[\left(\frac{P_d}{P_s}\right)^{\sigma} - 1\right] \qquad (16)$$

and the mass balance for suction and discharge of compressor is:

$$\dot{m}_{ac,s} = \dot{m}_{ac,d} + \dot{m}_f \qquad (17)$$

The fully implicit method consists of transforming Eqs. 6, 7, and 8 from partial differential equations to algebraic equations by using finite difference approximations for the partial derivatives. These equations are nonlinear and the Newton-Raphson method is applied to solve these equations for the compressible, non-isothermal transient flows through a pipe. Quasi-steady flow can be assumed at each time step of the numerical solution for the centrifugal and reciprocating compressor equations.

Formulation of the Optimization Problem

In order to optimize the operation of the network, we first formulate the problem at hand in the format of a standard nonlinear programming problem (NLP). This standard form is as follows:

Find the values of the design variables $[b_1, b_2, \ldots, b_r]^T$ to:
Minimize an objective function f(b)
Subject to the constraints: $h_j(b) = 0$, j = 1,…, m
and $g_j(b) \le 0$, j = m+1,…,n

The formulation of the network operation problem in the standard NLP form must be done carefully, making sure that the NLP formulation captures all the relevant aspects of the associated network problem.

Let the number of compressor stations in the pipeline network be N and let the number of compressors in station j be NC_j. Let n_{ik} be the speed of compressor k in station i. Further, let $nmin_{ik}$ and $nmax_{ik}$ represent the allowable minimum and maximum speeds of compressor k in station i. Let the fuel consumption rate of station i be m_{fi}. Finally, let the mass flow rate at station i be m_i and and let the specified minimum allowable mass flow rate at station i be $mmin_i$.
Then, the set of design variables is defined by

$\{n_{ik}\}$, i = 1,…,N; k = 1, …NC_i

while the objective function is given by

$f = \Sigma(m_{fi})$, i = 1,…,N

and the constraints are

$nmin_{ik} \le n_{ik} \le nmax_{ik}$, i = 1,…,N; k = 1, …$NC_I$

$mmin \le m_i$, I = 1,…,N.

Solution of the Optimization Problem

Once the network operation problem has been formulated as an optimization problem as outlined above, it can be solved using any of a variety of available methods. In this work, we used the sequential unconstrained minimization technique (SUMT) with an exterior penalty function. A directed grid search method was used for the unconstrained minimization that is required by the SUMT approach.

RESULT AND DISCUSSION

In this section we consider two parts, first simulation and second optimization.

Simulation

The purpose of this initial simulation is to show the behavior of parameter before reaching to steady sate condition. Figure 2. shows the schematic of a compressor station that explains the boundary conditions and geometry of the compressor station. Note that the number of compressors could be different. To know the dynamic response of the compressor station's parameters, an example is provided.

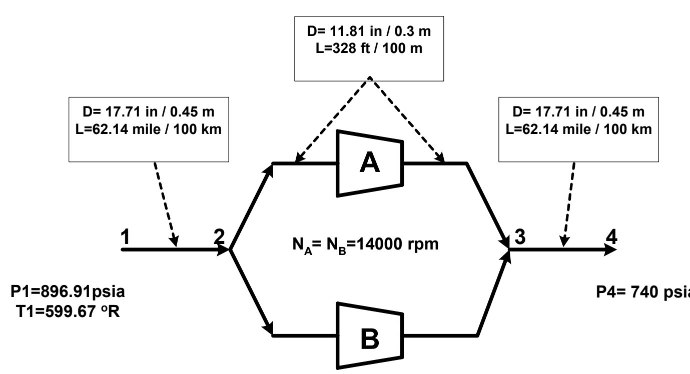
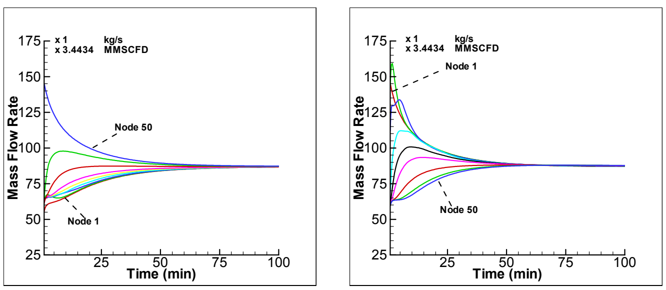


Figure 2 – Schematic of compressor station

The compressor station is located between two long pipes (100 km/ 62.14 mile).
The constant pressure boundary conditions are considered for

the head of the inlet pipe, which enters to the compressor station, and end of the outlet pipe, which exits from the compressor station, as shown in Figure 2. Another boundary condition for this simulation is constant speed (14000 rpm) for each compressor. Some of the results are described as follow.

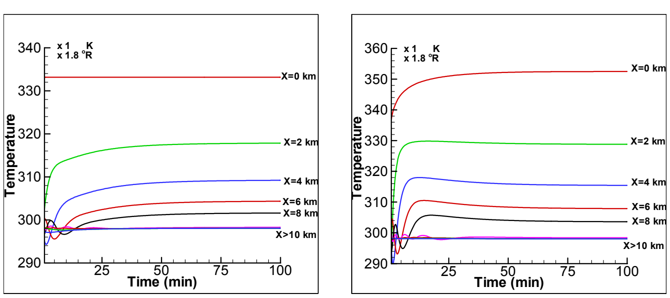
Figure 3 shows the variation of mass flow rate of the inlet pipe to compressor station for different node. As shown in this figure, after a while the mass flow rates at all of the nodes converge to the same value because of conservation of mass. And we can consider the behavior of flow as steady- state. Same manner can apply for outlet pipe from compressor station as shown shows the fuel consumption in compressor station with respect to time, which is calculated by using the difference between the mass flow rate at inlet and outlet of compressor station.



(a) (b)

Figure 3 - Mass flow rate for inlet pipe to (a) and outlet from compressor station (b)

Figure 4 illustrates the temperature distribution within the inlet pipe to, and the outlet pipe from, the compressor station. Because of conservation of energy and heat transfer between the pipe and environment, the values of temperature at each node are different as shown in these figures. But the interesting thing is that the temperature of gas after about 10 km of pipe length will be constant (surrounding temperature). Control of exhaust temperature at outlet of compressor is one of the most important goals of operation, which can control it using the non- isothermal model to reach the suitable value.



(a) (b)

Figure 4 - Temperature distribution for inlet pipe to (a) and outlet from compressor station (b)

Figure 5 shows the fuel consumption in compressor station with respect to time, which is calculated by using the difference between the mass flow rate at inlet and outlet of compressor station

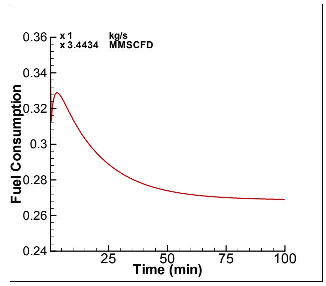


Figure 5- Fuel consumption at compressor station

Figure 6 represents the behavior of the centrifugal compressor parameters during this condition. Because the compressor map is design for steady-state condition, then with constant compressor speed as a boundary condition, the working point at compressor map will be changed till reaching to steady state condition (point B). This point is exactly for 14000 rpm.

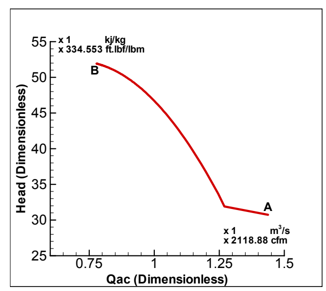


Figure 6. Compressor map for this simulation

Optimization

The system considered here is a single compressor station with ten identical compressors as shown in Figure 7. The compressor speed limits for this case are given in Table 1, and the goal of the optimization is to minimize the total fuel consumption while maintaining a station throughput of 600 kg/s (2066.0566 MMSCFD).

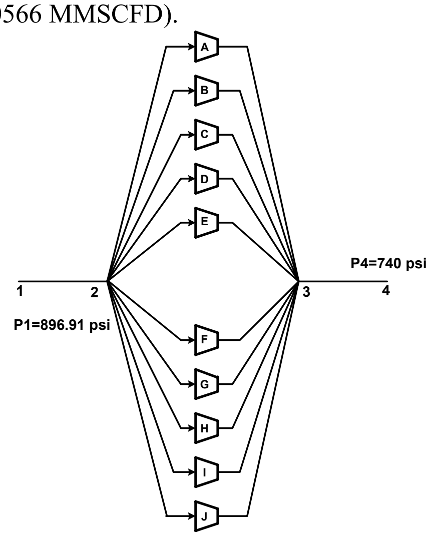


Figure 7 Compressor station with ten identical compressors

In this case, we also consider the possibility that the optimum operating condition for this station may require the shutdown of one or more units. Accordingly, we run the optimization separately using eight, nine, and ten compressors as shown in Table 2. By comparing the optima thus obtained, we can see that the best solution is to operate all ten compressors, with one compressor running at 12,397 rpm and the remaining nine compressors running at 12,400 rpm. Generally, it is believed that running fewer units in a compressor station is a way to improve the efficiency of the station. This example shows that this heuristic is not necessarily true, and by using numerical optimization we can find solutions that are much more fuel efficient. It can also be seen from Table 2 that at the ten compressor optimum, the efficiency of each unit actually from 76.11% to the 74.71%, - 74.93% range, i.e., at the optimum we may be operating at lower efficiency, but the total fuel consumption is reduced. It is also seen from Table 2 that in this case, the outlet temperature dropped from 623.3 degrees to 599.3 degrees.

CONCLUSIONS

This study used a fully implicit finite difference method to analyze transient and non-isothermal flow within a pipe and a quasi-steady flow assume at each time step of the numerical solution for centrifugal and reciprocating compressor equations to simulate compressor station in non-isothermal condition. The numerical results show that:

1. The effect of treating the gas in a non-isothermal manner is very necessary for pipeline flow calculation accuracies, and is extremely necessary for rapid transient processes.
2. By using a computer simulation, the dynamic response of compressor could be found by changing boundary condition with respect to time.
3. Foundation on which to build more complete compressor station equipments such as scrubbers, coolers, etc.
4. Numerical optimization is an effective tool for optimizing compressor speeds, and can yield significant reductions in fuel consumption. This, in turn, will increase throughout and reduce emissions.
5. Determination of the optimal number of compressors to shutdown in a compressor station and selection of optimal speeds for the remaining compressors can be done simultaneously using the methods developed herein.

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Table 1- The input data for optimization

	N_{rA}	N_{rB}	N_{rC}	N_{rD}	N_{rE}	N_{rF}	N_{rG}	N_{rH}	N_{rI}	N_{rJ}
Initial Value	14500	14500	14500	14500	14500	14500	14500	14500	14500	14500
Max. Value	15000	15000	15000	15000	15000	15000	15000	15000	15000	15000
Min. Value	10000	10000	10000	10000	10000	10000	10000	10000	10000	10000
Mass flow rate (kg/s- MMSCFD)		600 – 2066.05								

Table 2- Final result for speed and fuel consumption

	10 Compressors on-line		9 Compressors on-line		8 Compressors on-line	
	Initial	Final	Initial	Final	Initial	Final
N_{rA}	14500	12397	14500	13125	14500	14188
N_{rB}	14500	12400	14500	13144	14500	14200
N_{rC}	14500	12400	14500	13150	14500	14200
N_{rD}	14500	12450	14500	13200	14500	14206
N_{rE}	14500	12450	14500	13200	14500	14200
N_{rF}	14500	12450	14500	13200	14500	14250
N_{rG}	14500	12450	14500	13200	14500	14250
N_{rH}	14500	12450	14500	13200	14500	14250
N_{rI}	14500	12450	14500	13200	0	0
N_{rJ}	14500	12450	0	0	0	0
Fuel Consumption (kg/s- MMSCFD)	1.78 – 6.11	1.14 – 3.94	1.58 – 5.44	1.19 – 4.12	1.34 – 4.61	1.26 – 4.35
Mass flow rate (kg/s- MMSCFD)	662.51 – 2281.31	600.00 – 2066.06	639.36 – 2201.58	600.00 – 2066.06	608.10 – 2093.97	600.00 – 2066.08
η_{isA}	76.11	74.93	72.85	72.04	68.43	68.29
η_{isB}	76.11	74.92	72.85	71.96	68.43	68.25
η_{isC}	76.11	74.92	72.85	71.94	68.43	68.25
η_{isD}	76.11	74.71	72.85	71.75	68.43	68.23
η_{isE}	76.11	74.71	72.85	71.75	68.43	68.25
η_{isF}	76.11	74.71	72.85	71.75	68.43	68.09
η_{isG}	76.11	74.71	72.85	71.75	68.43	68.09
η_{isH}	76.11	74.71	72.85	71.75	68.43	68.09
η_{isI}	76.11	74.71	72.85	71.75	0	0
η_{isJ}	76.11	74.71	0	0	0	0
T_3 (^oR)	623.3	599.3	617.2	602.2	609.3	606.2

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